PART 5 Machinery

CONTENTS

CHAPTER 1 General Requirements

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SECTION 1 General

1.1 Machinery to be constructed under survey

1.1.1 All significant parts of the machinery equipment are to be surveyed at the manufacturer's premises, to the Surveyor's satisfaction, in ships built under Special Survey. The Surveyor is to examine that all the components are appropriate for the intended purpose and duty. Some of such units are the following:

– Main engines and their associated gears, superchargers and flexible couplings.

– Boilers supplying steam for propulsion or for services essential for the safety or the operation of the ship at sea and other boilers having working pressures exceeding 3,4 bar and having heating surfaces greater than 4,65 m^2.

– All pumps associated to main propulsion, essential machinery and the safety of the ship, (boiler feed, cooling water circulating, condensate extraction, oil fuel and lubricating oil pumps, fire, bilge and ballast pumps)

– Auxiliary engines essential for safety or for the operation of the ship at sea.
– All heat exchangers associated to main propulsion and essential machinery

– Air compressors, air receivers and other pressure vessels necessary for the operation of main propulsion and essential machinery. Any other unfired pressure vessels for which plans are required to be submitted as detailed in Part 5, Chapter 7.

– Valves and other components intended to be installed in pressure piping systems having working pressures exceeding 6,8 bar.
– Steering machinery and control mechanisms.
– Athwart ship thrust units.
– Alarm and control equipment as detailed in Part 8.
– Electrical equipment and electrical propelling machinery as detailed in Part 6.

1.2 Classification survey

1.2.1 Workmanship and materials should be examined by the Surveyors from the beginning of work until the final test of the machinery under full load. After this, the Surveyors will submit a report and if this is found to be satisfactory by LHR a certificate will be granted, and an appropriate notation will be assigned in accordance with Part 1.

1.3 System of inspection of mass-produced machinery items

1.3.1 LHR will be prepared to adopt a survey procedure of mass-produced machinery items based on quality assurance concepts utilizing regular and systematic audits of the approved manufacturing and quality control processes and procedures as an alternative to the direct survey of individual items.

1.3.2 In order to obtain approval, the requirements of SECTION 7 are to be complied with.

1.4 Deviation from the Rules

1.4.1 Requests for departure from the requirements of the Rules, due to special circumstances, will be subject to special consideration.

SECTION 2 Documents for approval

2.1 Plans

2.1.1 Plans in triplicate, of all machinery items, as detailed in the Chapters giving the requirements for individual systems, are to be submitted for consideration, before the commencement of the work. The particulars of the machinery, including power ratings and design calculations, where applicable, necessary to verify the design, are also to be submitted. Any subsequent modifications or alterations to initial design, materials or manufacturing procedure are to be re-submitted for consideration. A plan showing the arrangement of the machinery is also to be submitted.

2.2 Materials

2.2.1 The requirements for the materials used in the construction of all machinery items are those described in Part 2 of the LHR's Rules. Materials for which provision is not made therein may be accepted, provided that they comply with an approved specification and such tests as may be considered necessary.

SECTION 3 Operating conditions

3.1 Ambient conditions (IACS UR M28 (1978), UR M40 (1981))

3.1.1 Ambient reference conditions

For the purpose of determining the power of main and auxiliary reciprocating internal combustion engines, the following ambient reference conditions apply for ships of unrestricted service:

–	Total barometric pressure	1000 mbar
–	Air temperature	+45°C
–	Relative humidity	60%
–	Sea water temperature	32°C (charge air coolant-inlet)

NOTE:

1. The engine manufacturer shall not be expected to provide simulated ambient reference conditions at a test bed.

3.1.2 Temperatures

The ambient conditions specified in Table 1.3.1 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances as to ensure proper operation.

Table 1.3.1: Temperatures

(a) Air

Installations, components	Location, arrangement	Temperature range (°C)
Machinery and electrical installations (1)	In enclosed spaces	0 to +45 (2)
	On machinery components, boilers, In spaces subject to higher and lower temperatures	According to specific local conditions
	On the open deck	-25 to +45 (2)

(b) Water

Coolant	Temperature (°C)
Seawater	+32 (2)
Charge air coolant inlet to charge air cooler	+32 (2) see 3.1.1

NOTES:

1. Electronic appliances are to be suitable for proper operation even with an air temperature of +55°C.

2. LHR may approve other temperatures in the case of ships not intended for unrestricted service.

3.2 Inclinations (IACS UR M46 Rev. 2 (2018))[1]

3.2.1 The ambient conditions specified under Table 1.3.2 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances to ensure proper operation.

3.2.2 Any proposal to deviate from the angles given in Table 1.3.2 will be specially considered by LHR, taking into account the type, size and service conditions of the ship.

Table 1.3.2: Inclinations

Installations, components	Angle of inclination [°] (2)			
	Athwartships		Fore-and-aft	
	static	dynamic	static	dynamic
Main and auxiliary machinery	15	22,5	5 (4)	7,5

[1] Note:

1. These requirements are to be uniformly implemented by LHR on ships contracted for construction on or after 1 January 2020.
2. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder.

Safety equipment, e.g. emergency power installations, emergency fire pumps and their devices Switch gear, electrical and electronic appliances (1) and remote-control systems	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 for the carriage of liquefied gases and of chemicals the emergency power supply must also remain operable with the ship flooded to a final athwartships inclination up to a maximum of 30°.
4. Where the length of the ship exceeds 100m, the fore-and-aft static angle of inclination may be taken as 500/L degrees where L = length of the ship, in metres, as defined in Part 3, Chapter 1, SECTIONS 3.2 and 3.4.

3.3 Vibrations

3.3.1 Machinery, equipment and hull structures are normally subjected to vibration stresses.

Design, construction and installation must in every case take account of these stresses.

3.4 Definitions

3.4.1 Units and formulae included in the Rules are shown in SI units.

3.4.2 Where the metric version of shaft power, i.e. (shp), appears in the Rules, 1 shp is equivalent to 75 kgf m/s or 0,735 kW.

3.4.3 Pressure gauges may be calibrated in bar, where: $1\ \text{bar} = 0{,}1\ \text{N/mm}^2 = 1{,}02\ \text{kgf/cm}^2$

3.5 Fuels

3.5.1 The flash point (closed cup test) of fuel oil for use in ships classed for unrestricted service is, in general, to be not less than 60°C.

3.5.2 Fuels with flash points lower than 60°C will be specially considered in accordance with Part 5, Chapter 10, SECTION 2.

3.6 Power ratings

3.6.1 In the Chapters where the dimensions of any particular component are determined from shaft power, P, in kW, and revolutions per minute, n, the values to be used are to be derived from the following:

– for main propelling machinery, the maximum shaft power and corresponding revolutions per minute giving the maximum torque for which the machinery is to be classed,
– for auxiliary machinery, the maximum continuous shaft power and corresponding revolutions per minute which will be used in service.

3.7 Starting arrangements of Internal Combustion Engines (IACS UR M61 Rev.1 (2022))

3.7.1 Mechanical starting

3.7.1.1 The arrangement for air starting is to be such that the necessary air for the first charge can be produced on board without external aid.

3.7.1.2 Where the main engine is arranged for starting by compressed air, two or more air compressors are to be fitted. At least one of the compressors is to be driven independent of the main propulsion unit and is to have the capacity not less than 50 % of the total required.

3.7.1.3 The total capacity of air compressors is to be sufficient to supply within one hour the quantity of air needed to satisfy 3.7.1.5 by charging the receivers from atmospheric pressure. The capacity is to be approximately equally divided between the number of compressors fitted, excluding an emergency compressor which may be installed to satisfy 3.7.1.1.

3.7.1.4 Where the main engine is arranged for starting by compressed air, at least two starting air receivers of about equal capacity are to be fitted which may be used independently.

3.7.1.5 The total capacity of air receivers is to be sufficient to provide, without their being replenished, not less than 12 consecutive starts alternating between Ahead and Astern of each main engine of the reversible type, and not less than six starts of each main non-reversible type engine connected to a controllable pitch propeller or other device enabling the start without opposite torque. When other consumers such as auxiliary engines starting systems, control systems, whistle, etc., are to be connected to starting air receivers, their air consumption is also to be taken into account.

Regardless of the above, for multi-engine installations the number of starts required for each engine may be reduced upon the agreement with LHR depending upon the arrangement of the engines and the transmission of their output to the propellers.

3.7.2 Electrical starting

3.7.2.1 Where the main engine is arranged for electric starting, two separate batteries are to be fitted. The arrangement is to be such that the batteries cannot be connected in parallel. Each battery is to be capable of starting the main engine when in cold and ready to start conditions. The combined capacity of the batteries is to be sufficient without recharging to provide within 30 minutes the number of starts of main engines are required above in case of air starting.

3.7.2.2 Electric starting arrangements for auxiliary engines are to have two separate batteries or may be supplied by separate circuits from the main engine batteries when such are provided. In the case of a single auxiliary engine only one battery may be required. The capacity of the batteries for starting the auxiliary engines is to be sufficient for at least three starts for each engine.

3.7.2.3 The starting batteries are to be used for starting and the engine’s own monitoring purposes only. Provisions are to be made to maintain continuously the stored energy at all times.

3.8 Power conditions for generator jets

3.8.1 Auxiliary engines coupled to electrical generators are to be capable under service conditions of developing continuously the power to drive the generators at full rated output (kW) and, in the case of oil engines and gas turbines, of developing for a short period (15 minutes) an overload power of not less than 10% (see Part 6, Chapter 4).

3.8.2 Engine builders are to satisfy the Surveyors by suitable tests that the above requirements are applied. With oil engines and gas turbines any fuel stop fitted is to be set to permit the short period overload power of not less than 10 per cent above full rated output (kW) being developed.

SECTION 4 Machinery room arrangements

4.1 Accessibility

4.1.1 Equipment are to be so arranged in the engine room that all the erection holes and inspection parts provided by the engine manufacturer for inspections and repairs are accessible.

4.2 Machinery fastenings

4.2.1 All fastenings are to be of robust construction. The machinery is to be securely fixed to the ship's structure to the Surveyor's satisfaction.

4.3 Flexible mountings

4.3.1. The Shipbuilder is to ensure that the vibration levels of flexible pipe connections, shaft couplings and mounts remain always within the limits specified by the component manufacturer. The vibration levels are to be checked in the following conditions:

- Condition of maximum dynamic inclinations to be expected during service.
- Start-stop operation.
- Operation on the natural frequencies of the system.

Due account is to be taken of any creep that may be inherent in the mount.

4.3.2. Anti-collision chocks are to be used to ensure that manufacturers limits are not exceeded. Suitable means are to be provided to accommodate the propeller thrust.

4.4 Fire protection

4.4.1. All surfaces of machinery where the surface temperature may exceed 220°C and where impingement of flammable liquids may occur are to be properly shielded and insulated.

4.5 Communications

4.5.1 The existence of two independent means of communication between the bridge and engine room control station from which the engines are normally controlled is compulsory. See also Part 8.

4.5.2 One of these means of communication shall visually indicate the order and the corresponding response, both at the control station of the engine room and on the bridge.

4.5.3 At least one mean of communication is required between the bridge and any other control position from which the propulsion machinery may be controlled. One mean of communication is also required between the bridge and the steering gear compartment.

4.5.4 Indication of the direction of rotation of the propeller shaft and its revolutions per minute, in both the wheelhouse and the engine room, is required.

SECTION 5 Astern power for main propulsion (IACS UR M25 Rev. 4 (2017))

5.1 General requirements

5.1.1 In order to maintain sufficient manoeuvrability and secure control of the ship in all normal circumstances, the main propulsion machinery is to be capable of reversing the direction of thrust so as to bring the ship to rest from the maximum service speed. The main propulsion machinery is to be capable of maintaining in free route astern at least 70% of the ahead revolutions (1).

Note (1): The ahead revolutions as mentioned above are understood as those corresponding to the maximum continuous ahead power for which the vessel is classed.

5.1.2 Where steam turbines are used for main propulsion, they are to be capable of maintaining in free route astern at least 70% of the ahead revolutions for a period of at least 15 minutes. The astern trial is to be limited to 30 minutes or in accordance with manufacturer's recommendation to avoid overheating of the turbine due to the effects of "windage" and friction.

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

5.1.4 Main propulsion systems are to undergo tests to demonstrate the astern response characteristics. The tests are to be carried out at least over the manoeuvering range of the propulsion system and from all control positions. A test plan is to be provided by the yard and accepted by the surveyor. If specific operational characteristics have been defined by the manufacturer these shall be included in the test plan.

5.1.5 The reversing characteristics of the propulsion plant, including the blade pitch control system of controllable pitch propellers, are to be demonstrated and recorded during trials.

Note (2):

1. These requirements are to be uniformly implemented be LHR on:
   (a) Ships contracted for construction on or after 1 July 2018.
   (b) Ships other than those specified in the preceding (a) on which astern testing is carried out in accordance with IACS UR Z18 on or after 1 July 2018.
2. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder.

SECTION 6 Trials

6.1 Tests

6.1.1 All tests of particular components and trials of machinery are to be carried out to the satisfaction of the Surveyors, according to the Chapters giving the individual requirements for each part.

6.2 Sea trials

6.2.1 For the sake of reliability, due attention is required to the duration of all sea trials. Sea trials for all types of installation are to be carried out under normal manoeuvering conditions, to verify the machinery under power. During the trials and within the operating speed range any generated vibration is not to exceed the manufacturers values.

6.2.2 Diesel engines for main propulsion are to undergo trials as specified in Part 5, Chapter 2, SECTION 9. In case of controllable pitch propellers, the free route astern trial is to be carried out with the propeller blades set in the full pitch astern position. Where emergency manual pitch setting facilities are provided, their operation is to be demonstrated to the satisfaction of the Surveyors.

6.2.3 Before the full power sea trials, in geared installations, the gear teeth are to be suitably coated to demonstrate the contact markings, and on conclusion of the sea trials all gears are to be opened up in order to permit the Surveyors to make an inspection of the teeth. The marking is to indicate freedom from hard bearing, particularly at the ends of the teeth, including both ends of each helix where applicable. The contact is to be not less than that required by Part 5, Chapter 4.

6.2.4 The stopping time, ship headings and distances recorded on trials, together with the results of trials to determine the ability of ships having multiple propellers to navigate and manoeuvre with one or more propellers inoperative, are to be available on board for the use of the master or designated personnel.

SECTION 7 Quality assurance scheme for machinery

7.1 General

7.1.1 The proposed certification scheme is applicable to mass produced items manufactured under controlled conditions and will be restricted to works where the use of quality control procedures is well established. LHR will have to be satisfied that the practices employed will ensure that the quality level of the finished products is equivalent to the standards which would be required from the use of traditional survey methods.

7.1.2 An extensive survey is to be made by the Surveyors of the actual operation of the quality control programme including workmanship. The Committee is to consider proposed alternative designs for compliance with LHR's Rules, or other relevant requirements.

7.1.3 The procedures and practices of manufacturers which have been granted approval will be kept under review.

7.1.4 Approval by another organization will normally not be accepted as sufficient evidence that a manufacture's arrangements comply with the LHR's requirements.

7.2 Quality systems requirements

7.2.1 A quality system in accordance with requirements of the ISO 9001 Standard may be considered as meeting the requirements of LHR. Other quality systems will be subject to special consideration.

7.3 Mass production of machinery equipment

7.3.1 Internal combustion engines having a bore not exceeding 300mm and turbo blowers produced in series may be subject to a special procedure of inspection in accordance with requirements of Part 5, Chapter 2, SECTION 9, 9.2. Similar procedures may be applied to other machinery items upon request.

CHAPTER 2 Piston Engines

CONTENTS

SECTION 1 General

1.1 Scope

1.1.1 The Rules of this section apply to internal combustion engines used as main propulsion units, auxiliary units and also to air compressors.

For the purpose of these Rules, internal combustion engines are diesel engines.

1.2 Documents for the approval of diesel engines

1.2.1 For each type of engine that is required to be approved refer to Section 13 of this Chapter.

SECTION 2 Rated power

2.1 General

2.1.1 Diesel engines are to be capable to deliver their rated power when running at rated speed as a continuous net brake power. Diesel engines are to be capable of continuous operation within power range 1 in Figure 2.2.1 and of short-period operation in power range 2. The extent of the power ranges is to be stated by the engine manufacturer.

2.1.2 For the purposes of this Chapter, continuous power means the net brake power which an engine is capable of delivering continuously, provided that the maintenance prescribed by the engine manufacturer is carried out, between the maintenance intervals stated by the engine manufacturer.

2.1.3 To verify that an engine is rated at its continuous power, it is to be demonstrated that the engine can run at an overload power corresponding to 110% of its rated power at corresponding speed for an uninterrupted period of 1 hour. Deviations from the overload power value require the agreement of LHR.

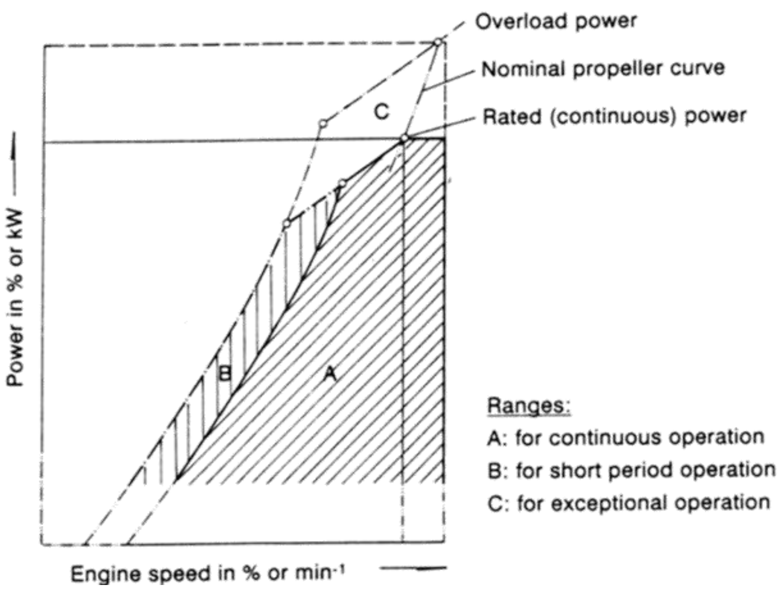
2.1.4 After running on the test bed, the fuel delivery system of main engines is normally to be so adjusted that overload power cannot be given in service.

2.1.5 Subject to the prescribed conditions, diesel engines driving electric generators must be capable of overload operation even after installation on board.

2.1.6 Subject to the approval of LHR, diesel engines for special vessels and special applications may be designed for a continuous power (fuel stop power) which cannot be exceeded.

2.1.7 For main engines, a power diagram (Figure 2.2.1) is to be prepared showing the power ranges within which the engine is able to operate continuously and for short periods under service conditions.

Figure 2.2.1: Example of a power diagram



SECTION 3 Calculation of Crankshafts for Internal Combustion Engines (IACS UR M53 Rev. 4 (2019))

3.1 General

3.1.1 Scope

These Rules for the design of crankshafts are to be applied to internal combustion engines for propulsion and auxiliary purposes, where the engines are capable of continuous operation at their rated power when running at rated speed.

Where a crankshaft design involves the use of surface treated fillets, or when fatigue parameter influences are tested, or when working stresses are measured, the relevant documents with calculations/analysis are to be submitted to the LHR in order to demonstrate equivalence to the Rules.

3.1.2 Field of application

These Rules apply only to solid-forged and semi-built crankshafts of forged or cast steel with one crank throw between main bearings.

3.2 Principles of calculation

The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas.

The calculation is also based on the assumption that the areas exposed to highest stresses are:

- fillet transitions between the crankpin and web as well as between the journal and web,
- outlets of crankpin oil bores.

When journal diameter is equal or larger than the crankpin one, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate documentation of fatigue safety may be required.

The calculation of crankshaft strength consists initially in determining the nominal alternating bending (see 3.3.1) and nominal alternating torsional stresses (see 3.3.2) which, multiplied by the appropriate stress concentration factors (see 3.4), result in an equivalent alternating stress (uni-axial stress) (see 3.8). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (see 3.9). This comparison will show whether or not the crankshaft concerned is dimensioned adequately (see 3.11)

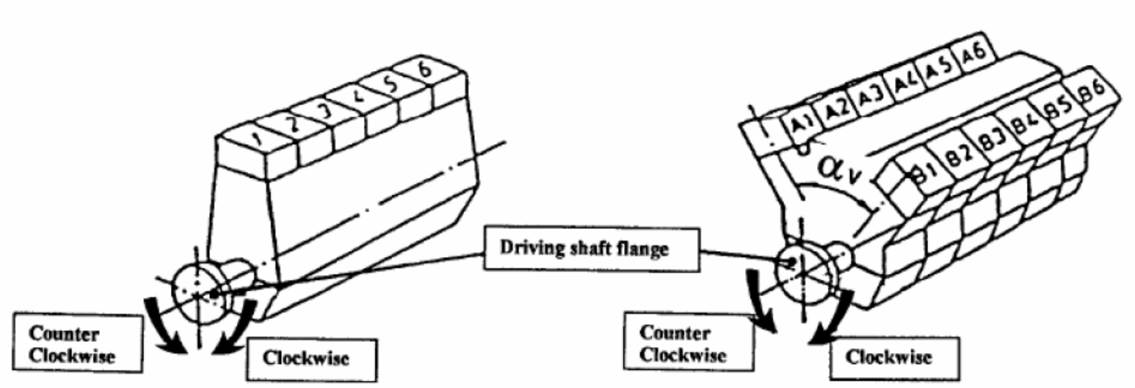
3.2.1 Drawings and particulars to be submitted

For the calculation of crankshafts, the documents and particulars listed below are to be submitted:

- crankshaft drawing (which must contain all data in respect of the geometrical configurations of the crankshaft),
- type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod),
- operating and combustion method (2-stroke or 4-stroke cycle/direct injection, precombustion chamber, etc.),
- number of cylinders,
- rated power [kW]
- rated engine speed [rpm]

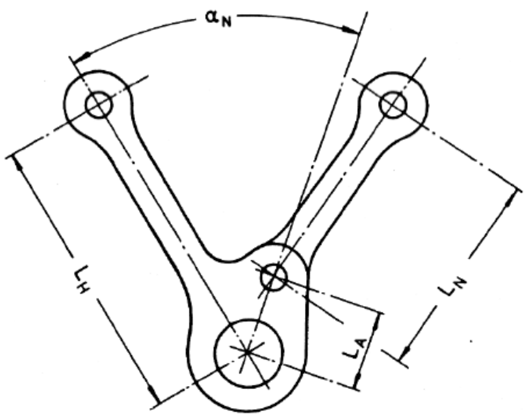
- direction of rotation (see Figure 2.3.1),
- firing order with the respective ignition intervals and, where necessary, V-angle α_v [°] (see Figure 2.3.1),

Figure 2.3.1: Designation of the cylinders



- cylinder diameter [mm]
- stroke [mm]
- maximum net cylinder pressure P_{max} [bar]
- charge air pressure, bar (before inlet valves or scavenge ports, whichever applies),
- connecting-rod length L_H [mm]
- all individual reciprocating masses acting on one crank [kg]
- digitalized gas pressure curve presented at equidistant intervals, bar versus Crank Angle (at least 5° CA),
- for engines with articulated type connecting-rod (see Figure 2.3.2),
  - distance to link point L_A [mm]
  - link angle a_N [degrees]
  - connecting-rod length L_N [mm]

Figure 2.3.2: Articulated-type connecting rod



- details of crankshaft material:
  - material designation (according to ISO, EN, DIN, AISI, etc.)
  - mechanical properties of material (minimum values obtained from longitudinal test specimens)
    - tensile strength [N/mm²]
    - yield strength [N/mm²]
    - percentage reduction in area at break,
    - percentage elongation A_5 [%]

  - impact energy – KV [J]
  - o type of forging (free form forged, continuous grain flow forged, drop-forged, etc.; with description of the forging process)
- every surface treatment affecting fillets or oil holes shall be subject to special consideration, and shall be specified so as to enable calculation according to Section 3.14
- particulars for alternating torsional stresses calculations, see 3.3.2.

Figure 2.3.3 Crankthrow for in line engine

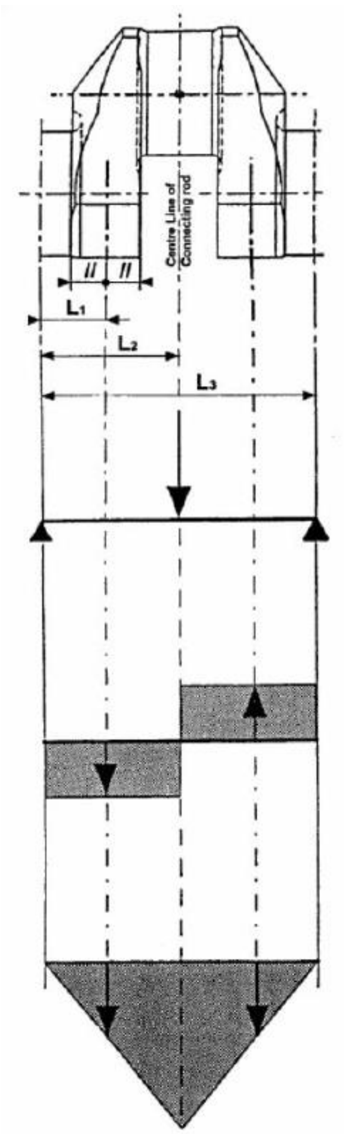
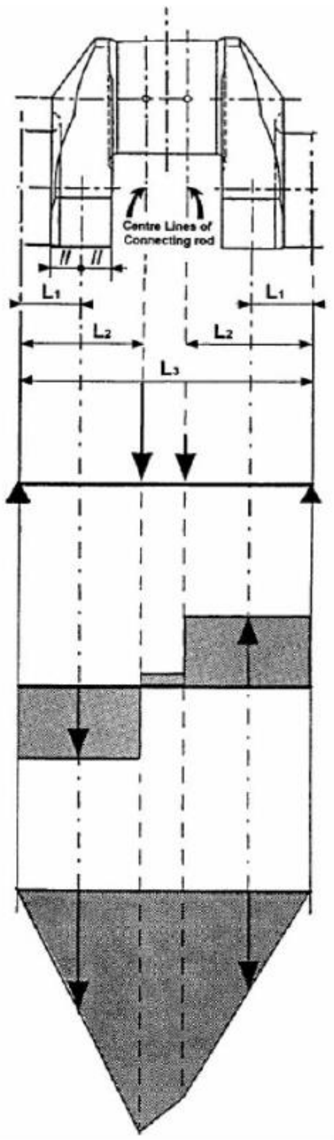


Figure 2.3.4 Crankthrow for Vee engine with2 adjacent connecting-rods



Connecting-rod acting component forces (F_R or F_T)

Radial shear force diagrams (Q_R)

Bending moment diagrams (M_{BR} or M_{BT})

L_1 = Distance between main journal centre line and crankweb centre (see also Fig 5 for crankshaft without overlap)

L_2 = Distance between main journal centre line and connecting-rod centre

L_3 = Distance between two adjacent main journal centre lines

3.3 Calculation of stresses

3.3.1 Calculation of alternating stresses due to bending moments and radial forces

(1) Assumptions:

The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearings midpoints (distance L_3, see Figure 2.3.3 and Figure 2.3.4).

The bending moments M_{BR}, M_{BT} are calculated in the relevant section 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 Figure 2.3.3).

For crankthrows with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see Figure 2.3.4).

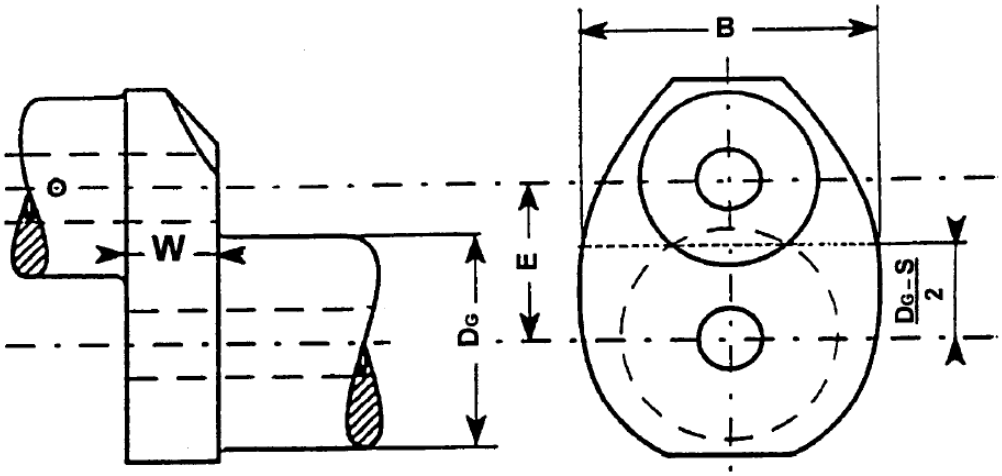
a) Bending moments and radial forces acting in web

The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of the solid web (distance L_1) and are derived from the radial component of the connecting-rod force.

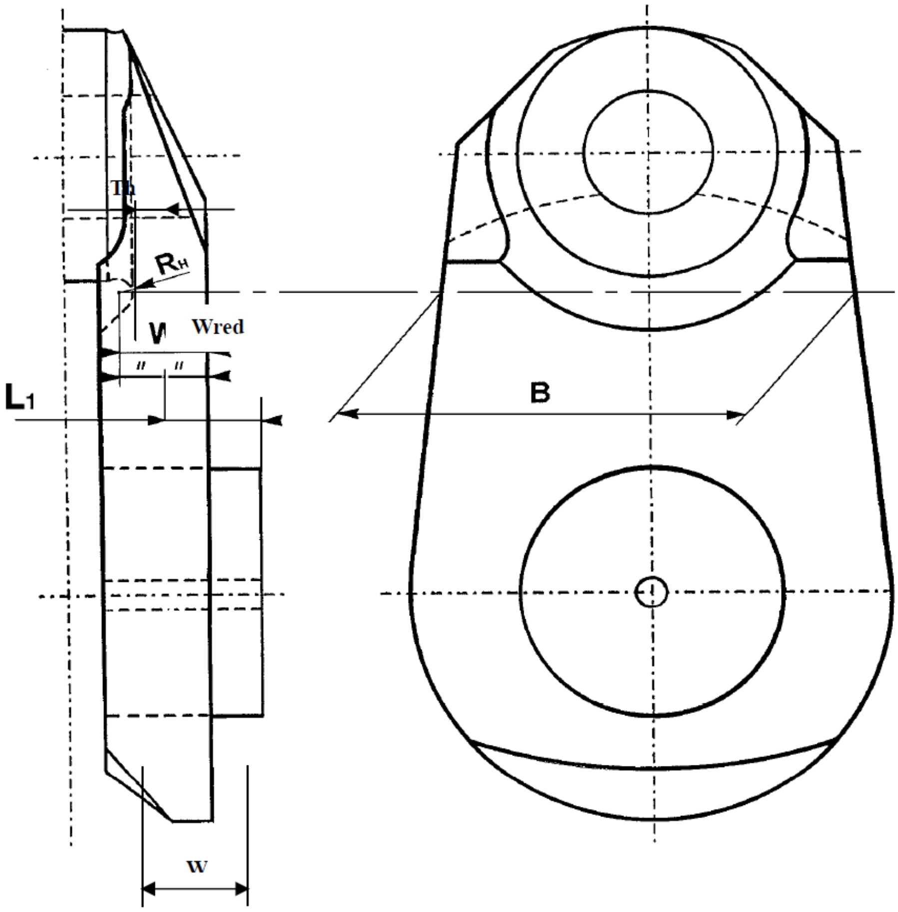
The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crank web. This reference section results from the web thickness W and the web width B (see fig. 5).

Mean stresses are neglected.

Figure 2.3.5: Reference area of crankweb cross section



Overlapped crankshaft

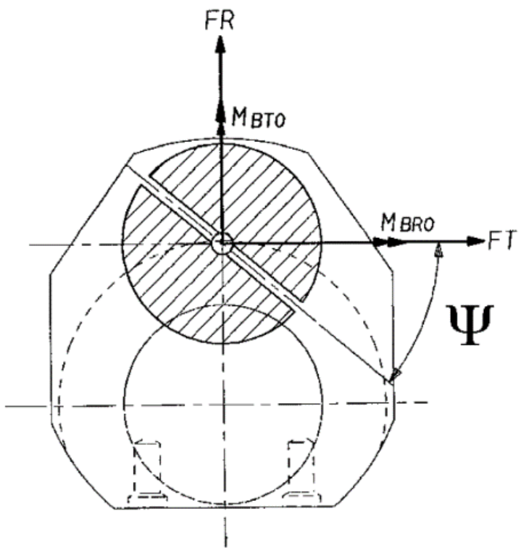


Crankshaft without overlap

b) Bending acting in outlet of crankpin oil bore

The two relevant bending moments are taken in the crankpin cross-section through the oil bore.

Figure 2.3.6: Crankpin section through the oil bore



M_{BRO} = the bending moment of the radial component of the connecting-rod force

M_{BTO} = the bending moment of the tangential component of the connecting-rod force

The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

(2) Calculation of nominal alternating bending and compressive stresses in web

The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position will be calculated over one working cycle.

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 3.3.1(1)(a) and 3.3.1(1)(b) - will then be calculated.

In case of V-type engines, the bending moments - progressively calculated from the gas and inertia forces - of the two cylinders acting on one crankthrow are superposed according to phase. Different designs (forked connecting rod, articulated-type connecting rod or adjacent connecting rods) shall be taken into account. Where there are cranks of different geometrical configurations in one crankshaft, the calculation is to cover all crank variants.

The decisive alternating values will then be calculated according to:

$$X_N = \pm \frac{1}{2}(X_{max} - X_{min})$$

where:

X_N = is considered as alternating force, moment or stress

X_{max} = is maximum value within one working cycle

X_{min} = is minimum value within one working cycle

a) Nominal alternating bending and compressive stresses in web cross section

The calculation of the nominal alternating bending and compressive stresses is as follows:

$$\sigma_{BFN} = \pm \frac{M_{BRFN}}{W_{eqw}} \cdot 10^3 \cdot K_e$$

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \cdot K_e$$

where:

σ_{BFN} = nominal alternating bending stress related to the web [N/mm²]

M_{BRFN} = alternating bending moment related to the centre of the web [Nm] (see Figure 2.3.3 and Figure 2.3.4)

$$M_{BRFN} = \pm \frac{1}{2}\left[M_{BRF_{max}} - M_{BRF_{min}}\right]$$

W_{eqw} = section modulus related to cross-section of web [mm³]

$$W_{eqw} = \frac{B \cdot W^2}{6}$$

K_e = empirical factor considering to some extent the influence of adjacent crank and bearing restraint, with:

= 0,8 for 2-stroke engines

= 1,0 for 4-stroke engines

σ_{QFN} = nominal alternating compressive stress due to radial force related to the web [N/mm²]

Q_{RFN} = alternating radial force related to the web [N] (see Figure 2.3.3 and Figure 2.3.4)

$$Q_{RFN} = \pm\frac{1}{2}\left[Q_{RF_{max}} - Q_{RF_{min}}\right]$$

F = area related to cross-section of web, mm²

$$F = B \cdot W$$

b) Nominal alternating bending stress in outlet of crankpin oil bore
The calculation of nominal alternating bending stress is as follows:

$$\sigma_{BON} = \pm\frac{M_{BON}}{W_e} \cdot 10^3$$

where:

σ_{BON}= nominal alternating bending stress related to the crank pin diameter, N/mm²

M_{BON}= alternating bending moment calculated at the outlet of crankpin oil bore

$$M_{BON} = \pm\frac{1}{2}\left[M_{BO_{max}} - M_{BO_{min}}\right]$$

with

$M_{BO} = M_{BTO} \cdot \cos\psi + M_{BRO} \cdot \sin\psi$, and

ψ = angular position, degrees

(see Figure 2.3.6)

W_e = section modulus related to cross-section of axially bored crankpin, mm³

$$W_e = \frac{\pi}{32} \cdot \left[\frac{D^4 - D_{BH}^4}{D}\right]$$

(3) Calculation of alternating bending stresses in fillets

The calculation of stresses is to be carried out for the crankpin fillet as well as for the journal fillet.

For the crankpin fillet:

$$\sigma_{BH} = \pm(\alpha_B \cdot \sigma_{BFN})$$

where:

σ_{BH} = alternating bending stress in crankpin fillet, N/mm²

α_B = stress concentration factor for bending in crankpin fillet (determination-see 3.4),

For the journal fillet (not applicable to semi-built crankshaft):

$$\sigma_{BG} = \pm(\beta_B \cdot \sigma_{BFN} + \beta_Q \cdot \sigma_{QFN})$$

where:

σ_{BG}= alternating bending stresses in journal fillet, N/mm²

β_B= stress concentration factor for bending in journal fillet (determination - see 3.4),

β_Q = stress concentration factor for compression due to radial force in journal fillet (determination - see 3.4).

(4) Calculation of alternating bending stresses in outlet of crankpin oil bore

$$\sigma_{BO} = \pm(\gamma_B \cdot \sigma_{BON})$$

where:

σ_{BO} = alternating bending stress in outlet of crankpin oil bore, N/mm²

γ_B = stress concentration factor for bending in crankpin oil bore (determination - see 3.4),

3.3.2 Calculation of alternating torsional stresses

(1) General:

The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in item 3.3.2(2). The manufacturer shall specify the maximum nominal alternating torsional stress.

(2) Calculation of nominal alternating torsional stresses:

The maximum and minimum torques are to be ascertained for every mass point of the complete dynamic system and for the entire speed range by means of a 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. Whilst doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring[2] in one of the cylinders). The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected. Where barred speed ranges are necessary, they shall be arranged so that satisfactory operation is possible despite their existence. There are to be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for normal firing conditions. The values received from such calculation are to be submitted to LHR. The nominal alternating torsional stress in every mass point, which is essential to the assessment, results from the following equation:

$$\tau_N = \pm \frac{M_{TN}}{W_P} \cdot 10^3$$

$$M_{TN} = \pm \frac{1}{2}\left(M_{T_{max}} - M_{T_{min}}\right)$$

$$W_P = \frac{\pi}{16} \cdot \left[\frac{D^4 - D_{BH}^4}{D}\right] \text{ or } W_P = \frac{\pi}{16} \cdot \left[\frac{D_G^4 - D_{BG}^4}{D_G}\right]$$

where:

τ_N	=	nominal alternating torsional stress referred to crankpin or journal, N/mm²,
M_{TN}	=	maximum alternating torque, Nm,
W_P	=	polar section modulus related to cross-section of axially bored crankpin or bored journal, mm³,
M_{Tmax}	=	maximum value of the torque, Nm
M_{Tmin}	=	minimum value of the torque, Nm.

For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in further calculations is the highest calculated value, according to above method, occurring at the most torsionally loaded mass point of the crankshaft system. Where barred speed ranges exist, the torsional stresses within these ranges are not to be considered for assessment calculations.

[2] NOTE: Misfiring is defined as cylinder condition when no combustion occurs but only compression cycle.

The approval of crankshaft will 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 by suitable calculation that the approved nominal alternating torsional stress is not exceeded. This calculation is to be submitted for assessment.

(3) Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore:

The calculation of stresses is to be carried out for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore.

For the crankpin fillet:

$$\tau_H = \pm(\alpha_T \cdot \tau_N)$$

where:

τ_H = alternating torsional stress in crankpin fillet, N/mm²

α_T = stress concentration factor for torsion in crankpin fillet (determination - see 3.4)

τ_N = nominal alternating torsional stress related to crankpin diameter, N/mm²

For the journal fillet (not applicable to semi-built crankshafts):

$$\tau_G = \pm(\beta_T \cdot \tau_N)$$

where:

τ_G = alternating torsional stress in journal fillet, N/mm²

β_T = stress concentration factor for torsion in journal fillet (determination - see 3.4)

τ_N = nominal alternating torsional stress related to journal diameter, N/mm²

For the outlet of crankpin oil bore:

$$\sigma_{TO} = \pm(\gamma_T \cdot \tau_N)$$

where:

σ_{TO} = alternating stress in outlet of crankpin oil bore due to torsion, N/mm²

γ_T = stress concentration factor for torsion in outlet of crankpin oil bore (determination - see 3.4)

τ_N = nominal alternating torsional stress related to crankpin diameter, N/mm²

3.4 Evaluation of stress concentration factors

3.4.1 General

The stress concentration factors are evaluated by means of the formulae according to items 3.4.2, 3.4.3 and 3.4.4 applicable to the fillets and crankpin oil bore of solid forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. It must be noticed that stress concentration factor formulae concerning the oil bore are only applicable to a radially drilled oil hole. All formulae are based on investigations of FVV (Forschungsvereinigung Verbrennungskraftmaschinen) for fillets and on investigations of ESDU (Engineering Science Data Unit) for oil holes.

Where the geometry of the crankshaft is outside the boundaries of the analytical stress concentration factors (SCF) the calculation method detailed in 3.12 may be undertaken.

All crank dimensions necessary for the calculation of stress concentration factors are shown in Figure 2.3.7.

The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (VON MISES) - occurring in the fillets under bending load acting in the central cross-section of a crank - to the nominal bending stress related to the web cross-section (see Figure 2.3.8).

The nominal stress has to be determined under the bending moment in the middle of the solid web.

The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum equivalent shear stress - occurring under torsional load in the fillets - to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see Figure 2.3.8).

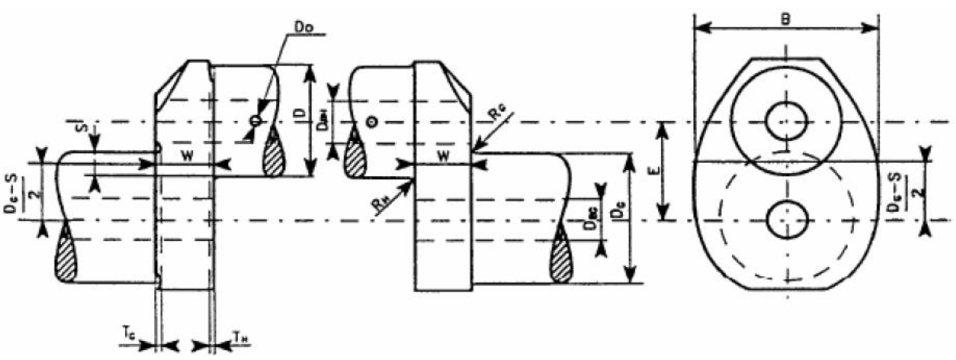
The stress concentration factor for compression (β_Q) in the journal fillet is defined as the ratio of the maximum equivalent stress (VON MISES) - occurring in the fillet due to the radial force - to the nominal compressive stress related to the web cross-section.

The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress – occurring at the outlet of the crankpin oil-hole under bending and torsional loads – to the corresponding nominal stress related to the axially bored crankpin cross section (see Figure 2.3.9).

When reliable measurements and/or calculations are available, which can allow direct assessment of stress concentration factors, the relevant documents and their analysis method have to be submitted to LHR in order to demonstrate their equivalence to present rules evaluation. This is always to be performed when dimensions are outside of any of the validity ranges for the empirical formulae presented in 3.4.2 to 3.4.4.

Paragraphs 3.12 and 3.15 of this Section describe how FE analyses can be used for the calculation of the stress concentration factors. Care should be taken to avoid mixing equivalent (VON MISES) stresses and principal stresses.

Figure 2.3.7: Crank dimensions



Actual dimensions:

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,

B (*) = web width, mm.

(*) In the case of 2 stroke semi-built crankshafts:

- when $T_H > R_H$, the web thickness must be considered as equal to:

$$W_{red} = W - (T_H - R_H), \qquad \text{(see Figure 2.3.5)}$$

- web width B must be taken in way of crankpin fillet radius centre according to Figure 2.3.5

The following related dimensions will be applied for the calculation of stress concentration factors:

Crankpin fillets	Journal fillets
$r = R_H/D$	$r = R_G/D$
$s = S/D$	
$w = W/D$ crankshafts with overlap	
W_{red}/D crankshafts without overlap	
$b = B/D$	
$d_o = D_o/D$	
$d_G = D_{BG}/D$	
$d_H = D_{BH}/D$	
$t_H = T_H/D$	
$t_G = T_G/D$	

Stress concentration factors are valid for the ranges of related dimensions for which the investigations have been carried out. These ranges are as follows:

$$s \le 0{,}5$$
$$0{,}2 \le w \le 0{,}8$$
$$1{,}1 \le b \le 2{,}2$$
$$0{,}03 \le r \le 0{,}13$$
$$0 \le d_G \le 0{,}8$$

$$0 \leq d_H \leq 0{,}8$$

$$0 \leq D_o \leq 0{,}2$$

Low range of s can be extended down to large negative values provided that:

- If calculated f(recess) < 1 then the factor f (recess) is not to be considered (f(recess) = 1),
- If s < - 0,5 then f(s,w) and f(r,s) are to be evaluated replacing actual value of s by – 0,5.

3.4.2 Crankpin fillet

The stress concentration factor for bending (α_B) is:

$$a_B = 2{,}6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(recess)$$

where:

f(s,w) = $-4{,}1883 + 29{,}2004 \cdot w - 77{,}5925 \cdot w^2 + 91{,}9454 \cdot w^3 - 40{,}0416 \cdot w^4 + (1-s) \cdot .(9{,}5440 - 58{,}3480 \cdot w + 159{,}3415 \cdot w^2 - 192{,}5846 \cdot w^3 + 85{,}2916 \cdot w^4) + (1-s)^2 \cdot (-3{,}8399 + 25{,}0444 \cdot w - 70{,}5571 \cdot w^2 + 87{,}0328 \cdot w^3 - 39{,}1832 \cdot w^4)$

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) = $1 + (t_H + t_G) \cdot (1{,}8 + 3{,}2 \cdot s)$

The stress concentration factor for torsion (α_T) is:

$$\alpha_T = 0{,}8 \cdot f(r,s) \cdot f(b) \cdot f(w)$$

where:

f (r,s) = $r^{(-0{,}322+0{,}1015 \cdot (1-s))}$

f (b) = $7{,}8955 - 10{,}654 \cdot b + 5{,}3482 \cdot b^2 - 0{,}857 \cdot b^3$,

f (w) = $w^{(-0{,}145)}$

3.4.3 Journal fillet (not applicable to semi-built crankshaft)

The stress concentration factor for bending (β_B) is:

$$\beta_B = 2{,}7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(recess)$$

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) = $1 + (t_H + t_G) \cdot (1{,}8 + 3{,}2 \cdot s)$

The stress concentration factor for compression (β_Q) due to the radial force is:

$$\beta_Q = 3{,}0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$$

where:

f_Q (s) = $0{,}4368 + 2{,}1630 \cdot (1 - s) - 1{,}5212 \cdot (1 - s)^2$

f_Q (w) = $\dfrac{w}{0{,}0637 + 0{,}9369 \cdot w}$

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) = $1 + (t_H + t_G) \cdot (1{,}8 + 3{,}2 \cdot s)$

The stress concentration factor for torsion (β_T) is:

$$\beta_T = \alpha_T$$

if the diameters and fillet radii of crankpin and journal are the same

If crankpin and journal diameters and/or radii are of different sizes

$$\beta_T = 0{,}8 \cdot f(r,s) \cdot f(w) \cdot f(b)$$

where:

f(r,s), f(b) and f(w) are to be determined in accordance with item 3.4.2 (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter:

$$r = \frac{R_G}{D_G}$$

3.4.4 Outlet of crankpin oil bore

The stress concentration factor for bending (γ_B) is:

$$\gamma_B = 3 - 5.88 \cdot d_O + 34.6 \cdot {d_O}^2$$

The stress concentration factor for torsion (γ_T) is:

$$\gamma_T = 4 - 6 \cdot d_O + 30 \cdot {d_O}^2$$

3.5 Additional bending stresses

3.5.1 In addition to the alternating bending stresses in fillets (see item 3.3.1(3)) 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 given by the following table:

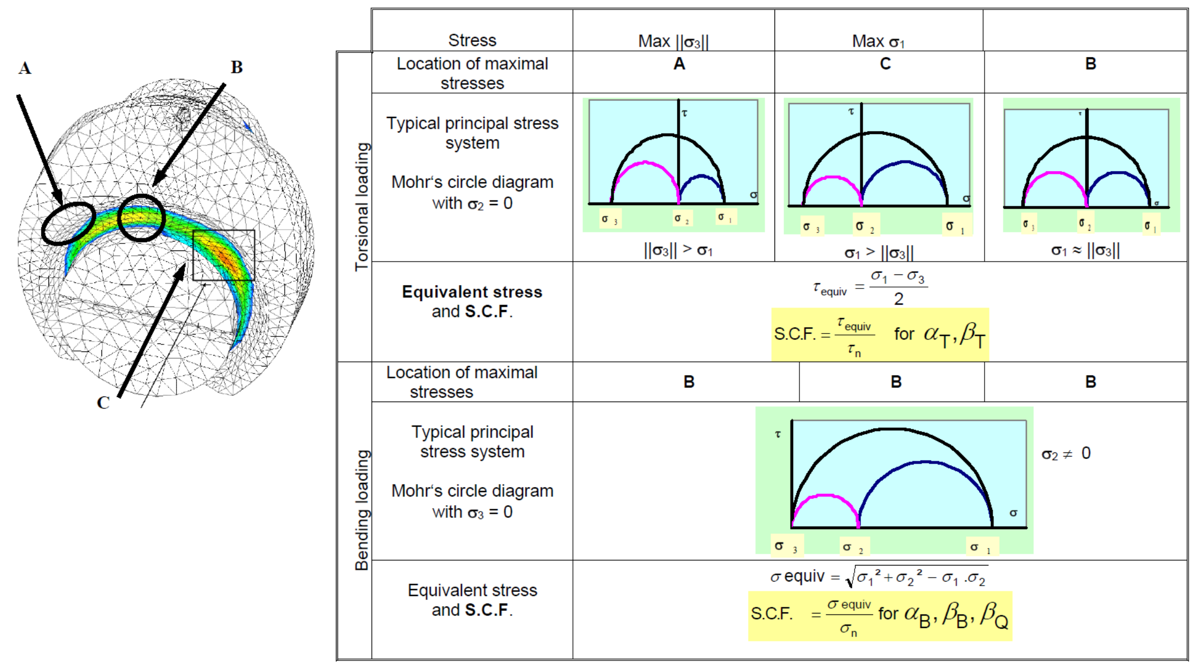
Type of engine	σ_{add} [N/mm²]
crosshead engines	±30 (3)
trunk piston engines	±10

3.6 Calculation of equivalent alternating stress

3.6.1 General

In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a Von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (see Figure 2.3.8). As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the Von Mises criterion.

Figure 2.3.8: Definition of Stress Concentration Factors in crankshaft fillets



At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see Figure 2.3.9).

---

3 NOTE:
The additional stress of ±30 N/mm² is composed of two components:
1. an additional stress of ±20 N/mm² resulting from axial vibration
2. an additional stress of ±10 N/mm² resulting from misalignment / bedplate deformation
It is recommended that a value of ±20 N/mm² be used for the axial vibration component for assessment purposes where axial vibration calculation results of the complete dynamic system (engine/shafting/gearing/propeller) are not available. Where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

The above two different ways of equivalent stress evaluation both lead to stresses which may be compared to the same fatigue strength value of crankshaft assessed according to Von Mises criterion.

3.6.2 Equivalent alternating stress

The equivalent alternating stress is calculated in accordance with the formulae given.

For the crankpin fillet:

$$\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3 \cdot \tau_H^2}$$

For the journal fillet:

$$\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3 \cdot \tau_G^2}$$

For the outlet of crankpin oil bore:

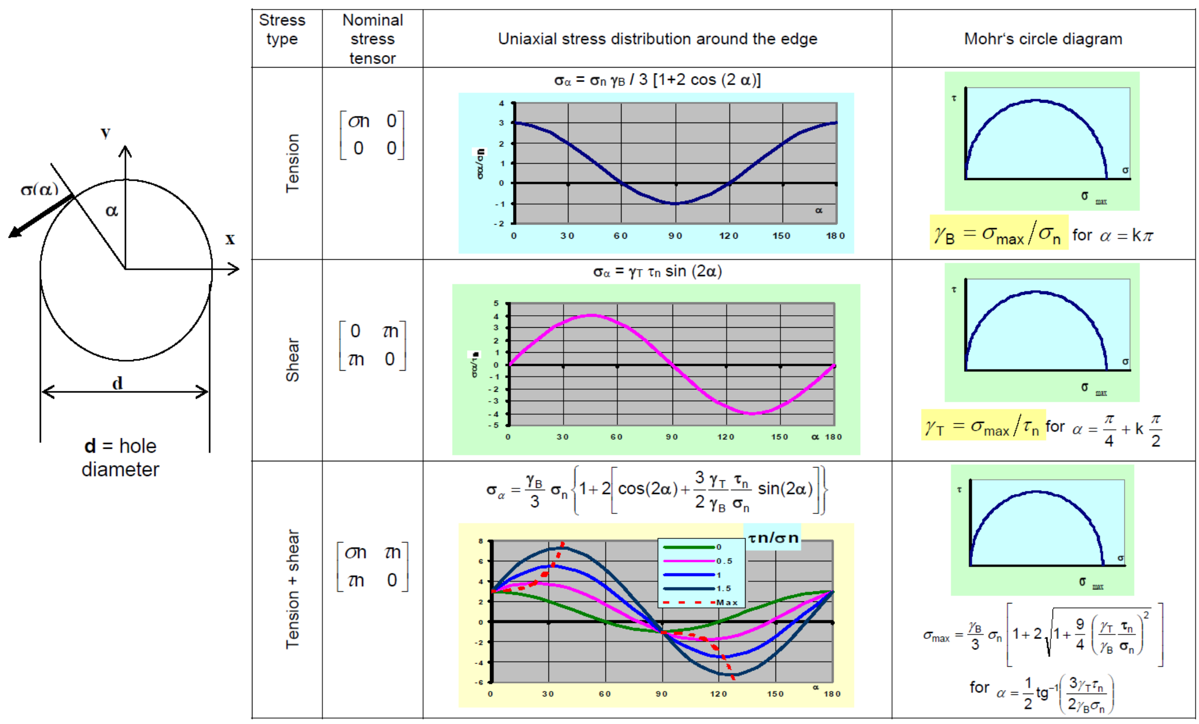
$$\sigma_V = \pm\frac{1}{3} \cdot \sigma_{BO}\left[1 + 2 \cdot \sqrt{1 + \frac{9}{4} \cdot \left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

where:

σ_V = equivalent alternating stress, N/mm².

For other parameters see 3.3.1(3), 3.3.2(3) and 3.7.

Figure 2.3.9: Stress Concentration Factors and stress distribution at the edge of oil drillings



3.7 Calculation of fatigue strength

3.7.1 The fatigue strength is to be understood as that value of equivalent alternating stress (Von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength may be evaluated by means of the following formulae:

Related to the crankpin diameter:

$$\sigma_{DW} = \pm K \cdot (0{,}42 \cdot \sigma_B + 39{,}3) \cdot \left(0{,}264 + 1{,}073 \cdot D^{-0{,}2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{R_X}}\right)$$

where:

R_X = R_H in the fillet area

R_X = D_o / 2 in the oil bore area

Related to the journal diameter:

$$\sigma_{DW} = \pm K \cdot (0{,}42 \cdot \sigma_B + 39{,}3) \cdot \left(0{,}264 + 1{,}073 \cdot D_G^{-0{,}2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{R_G}}\right)$$

where:

σ_{DW} = allowable fatigue strength of crankshaft, N/mm².

K = factor for different types of crankshafts without surface treatment. Values greater than 1 are only applicable to fatigue strength in fillet area.

= 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 area,

= 0,93 for cast steel crankshafts manufactured by companies using an approved by LHR cold rolling process,

σ_B = minimum tensile strength of crankshaft material, N/mm².

For other parameters see 3.4.3.

When a surface treatment process is applied, it must be approved by LHR. Guidance for calculation of surface treated fillets and oil bore outlets is presented in 3.12 of this SECTION.

These formulae are subject to the following conditions:

- surfaces of the fillet, the outlet of the oil bore and inside the oil bore (down to a minimum depth equal to 1.5 times the oil bore diameter) shall be smoothly finished,
- for calculation purposes R_H, R_G or R_X are to be taken as not less than 2 mm.

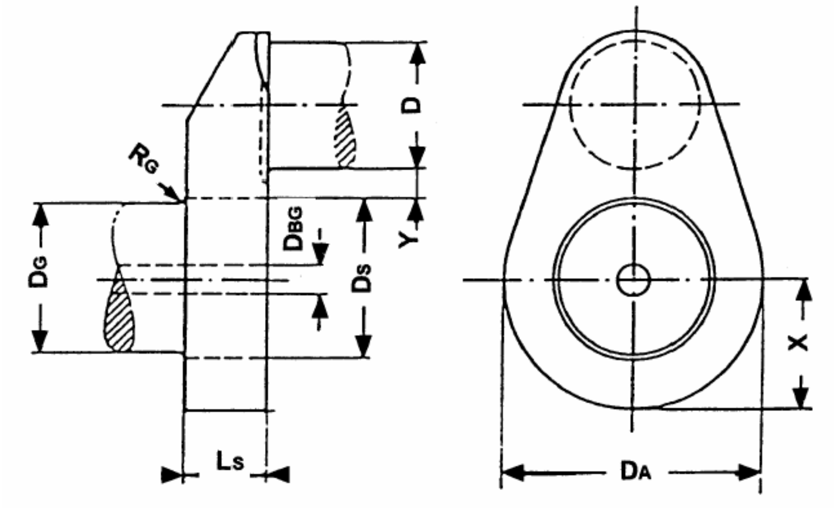
As an alternative the fatigue strength of the crankshaft can be determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow. For evaluation of test results see 3.14. In any case the experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment have to be submitted for approval to LHR (method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, confidence number, etc).

3.8 Calculation of shrink-fits of semi-built crankshaft

3.8.1 General

All crank dimensions necessary for the calculation of the shrink-fit are shown in Figure 2.3.10.

Figure 2.3.10: Crankthrow of semi-built crankshaft



where:

D_S	=	shrink diameter, mm,
D_G	=	journal diameter, mm,
D_{BG}	=	diameter of axial bore in journal, mm,
L_S	=	length of shrink-fit, mm,
R_G	=	fillet radius of journal, mm,
D_A	=	outside diameter of web, mm, or twice the minimum distance x between centre-line of journals and outer contour of web, whichever is less, mm,
y	=	distance between the adjacent generating lines of journal and pin, mm.
y	≥	$0{,}05 \cdot D_S$.

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.

Respecting the radius of the transition from the journal to the shrink diameter, the following should be complied with:

$$R_G \geq 0{,}015 \cdot D_G, \text{ and}$$

$$R_G \geq 0{,}5 \cdot (D_S - D_G),$$

where the greater value is to be considered.

The actual oversize Z of the shrink-fit must be within the limits Z_{min} and Z_{max} calculated in accordance with 3.8.3 and 3.8.4.

In the case where 3.8.2 condition cannot be fulfilled then 3.8.3 and 3.8.4 calculation methods of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such case Z_{min} and Z_{max} have to be established based on FEM calculations.

3.8.2 Maximum permissible hole in the journal pin

The maximum permissible hole diameter in the journal pin is calculated in accordance with the following formula:

$$D_{BG} = D_S \cdot \sqrt{1 - \frac{4000 \cdot S_R \cdot M_{max}}{\mu \cdot \pi \cdot D_S^2 \cdot L_S \cdot \sigma_{SP}}}$$

where:

S_R = safety factor against slipping, however a value not less than 2 is to be taken unless documented by experiments,

M_{max} = absolute maximum value of the torque M_{Tmax} in accordance with 3.3.2, Nm,

μ = coefficient for static friction, however a value not greater than 0,2 is to be taken unless documented by experiments,

σ_{SP} = minimum yield strength of material for journal pin, N/mm²

This condition serves to avoid plasticity in the hole of the journal pin.

3.8.3 Necessary minimum oversize of shrink-fit

The necessary minimum oversize is determined by the greater value calculated in accordance with the formulae below:

$$Z_{min} \geq \frac{\sigma_{sw} \cdot D_s}{E_m}$$

$$Z_{min} \geq \frac{4.10^3}{\pi \cdot \mu} \cdot \frac{S_R \cdot M_{max}}{E_m \cdot D_S \cdot L_S} \cdot \frac{1 - (Q_A^2 \cdot Q_S^2)}{(1 - Q_A^2) \cdot (1 - Q_S^2)}$$

where:

Z_{min} = minimum oversize, mm,

Q_A = web ratio, $Q_A = D_S / D_A$

Q_S = shaft ratio, $Q_S = D_{BG} / D_S$

E_m = Young's modulus, N/mm².

σ_{SW} = minimum yield strength of material for crank web, N/mm².

3.8.4 Maximum permissible oversize of shrink-fit

The maximum permissible oversize is calculated in accordance with the following formula:

$$Z_{max} \leq D_S \cdot \left(\frac{\sigma_{sw}}{E_m} + \frac{0,8}{1000}\right)$$

where:

Z_{max} = maximum oversize, mm.

This condition serves to restrict the shrinkage induced mean stress in the fillet.

3.9 Acceptability criteria

3.9.1 The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. This comparison has to be carried out for the crankpin fillet, the journal fillet and the outlet of crankpin oil bore and is based on the formula:

$$Q = \frac{\sigma_{DW}}{\sigma_V}$$

where:

Q = acceptability factor

Adequate dimensioning of the crankshaft is ensured if the smaller of all acceptability factors satisfies the criteria:

$$Q \geq 1{,}15$$

3.10 Guidance for Calculation of Stress Concentration Factors in the web fillet radii of crankshafts by utilizing Finite Element Method

3.10.1 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 according to the rules of this document 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 of this Section or the alternative methods, von Mises stresses shall 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 not covered by this document.

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.

3.10.2 Model requirements

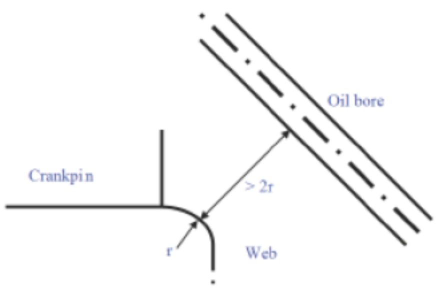
The basic recommendations and perceptions for building the FE-model are presented in 3.10.2(1). It is obligatory for the final FE-model to fulfill the requirement in 3.10.2(3).

(1) Element mesh recommendations

In order to fulfil the mesh quality criteria, it is advised to construct the FE model for the evaluation of Stress Concentration Factors according to the following recommendations:

- The model consists of one complete crank, from the main bearing centerline to the opposite side main bearing centerline
- Element types used in the vicinity of the fillets:
  - 10 node tetrahedral elements
  - 8 node hexahedral elements
  - 20 node hexahedral elements
- Mesh properties in fillet radii. The following applies to ±90 degrees in circumferential direction from the crank plane:
- Maximum element size a=r/4 through the entire fillet as well as in the circumferential direction. 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 r is the local fillet radius. (If 8 node hexahedral elements are used even smaller element size is required to meet the quality criteria)
- Recommended manner for element size in fillet depth direction
  - First layer thickness equal to element size of a
  - Second layer thickness equal to element to size of 2a
  - Third layer thickness equal to element to size of 3a
- Minimum 6 elements across web thickness
- Generally the rest of the crank should be suitable for numeric stability of the solver
- Counterweights only have to be modeled only when influencing the global stiffness of the crank significantly
- Modeling 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, see figure 2.3.11
- Drillings and holes for weight reduction have to be modeled
- Sub-modeling may be used as far as the software requirements are fulfilled

Figure 2.3.11: Oil bore proximity to fillet



(2) Material

This Section does not consider material properties such as Young's Modulus (E) and Poisson's ratio (ν). In FE analysis those material parameters are required, as strain is primarily calculated and stress is derived from strain using the Young's Modulus and Poisson's ratio. Reliable values for material parameters have to be used, either as quoted in literature or as measured on representative material samples.

For steel the following is advised: E= $2{,}05 \cdot 10^5$ MPa and ν=0,3.

(3) Element mesh quality criteria

If the actual element mesh does not fulfil any of the following criteria at the examined area for SCF evaluation, then a second calculation with a refined mesh is to be performed.

The quality of the mesh should be assured by checking the stress component normal to the surface of the fillet radius. Ideally, this stress should be zero. With principal stresses σ_1, σ_2 and σ_3 the following criterion is required:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0{,}03 \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

(4) Averaged/unaveraged stresses criterion

The criterion is based on observing the discontinuity of stress results over elements at the fillet for the calculation of SCF:

Unaveraged nodal stress results calculated from each element connected to a $node_i$ should differ less than by 5 % from the 100 % averaged nodal stress results at this $node_i$ at the examined location.

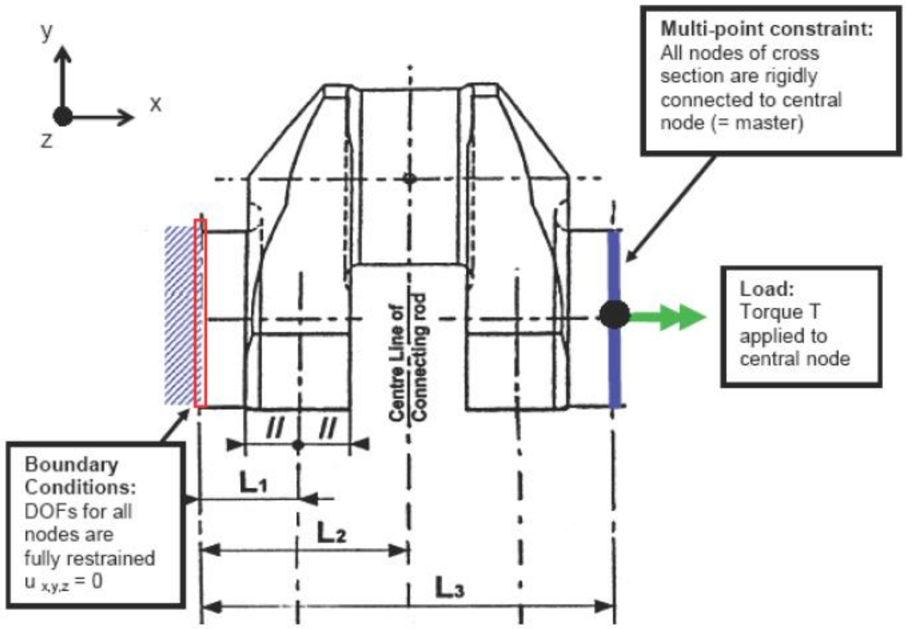
3.10.3 Load cases

To substitute the analytically determined SCF in this Section, the following load cases have to be calculated:

(1) Torsion

In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded pure torsion. In the model surface warp at the end faces is suppressed. Torque is applied to the central node located at the crankshaft axis. This node acts as the master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face. Boundary and load conditions are valid for both in-line and V-type engines.

Figure 2.3.12: Boundary and load conditions for the torsion load case



For all nodes in both the journal and crank pin fillet principal stresses are extracted and the equivalent torsional stress is calculated:

$$\tau_{\text{equiv}} = \max\left(\frac{|\sigma_1 - \sigma_2|}{2}, \frac{|\sigma_2 - \sigma_3|}{2}, \frac{|\sigma_1 - \sigma_3|}{2}\right)$$

The maximum value taken for the subsequent calculation of the SCF:

$$\alpha_{\text{T}} = \frac{\tau_{\text{equiv},\alpha}}{\tau_{\text{N}}}$$

$$\beta_{\text{T}} = \frac{\tau_{\text{equiv},\beta}}{\tau_{\text{N}}}$$

where τ_{N} is nominal torsional stress referred to the crankpin and respectively journal as per 3.3.2(2) of this SECTION with the torsional torque T:

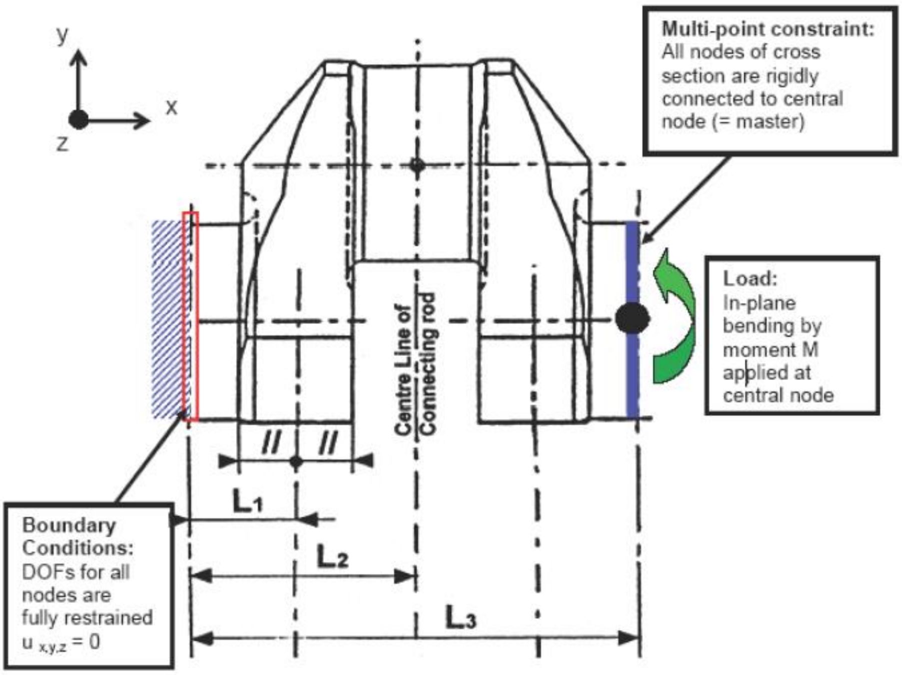
$$\tau_N = \frac{T}{W_P}$$

(2) Pure bending (4 point bending)

In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded in pure bending. In the model surface warp at the end faces is suppressed. The bending moment is applied to the central node located at the crankshaft axis. This node acts as the master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

Boundary and load conditions are valid for both in-line- and V- type engines.

Figure 2.3.13: Boundary and load conditions for the pure bending load case.



For all nodes in both the journal and pin fillet von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the SCF according to:

$$\alpha_B = \frac{\sigma_{\text{equiv},\alpha}}{\sigma_N}$$

$$\beta_B = \frac{\sigma_{\text{equiv},\beta}}{\sigma_N}$$

Nominal stress σ_N is calculated as per 3.3.1(2) a) with the bending moment M:

$$\sigma_N = \frac{M}{W_{eqw}}$$

(3) Bending with shear force (3-point bending)

This load case is calculated to determine the SCF for pure transverse force (radial force, β_Q) for the journal fillet.

In analogy to the testing apparatus used for the investigations made by FVV, the structure is loaded in 3-point bending. In the model, surface warp at the both end faces is suppressed. All nodes are connected rigidly to the centre node; boundary conditions are applied to the centre nodes. These nodes act as master nodes with 6 degrees of freedom.

The force is applied to the central node located at the pin centre-line of the connecting rod. This node is connected to all nodes of the pin cross sectional area. Warping of the sectional area is not suppressed.

Boundary and load conditions are valid for in-line and V-type engines. V-type engines can be modeled with one connecting rod force only. Using two connecting rod forces will make no significant change in the SCF.

The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The SCF in the journal fillet can be determined in two ways as shown in 3.10.4 and 3.10.5.

Figure 2.3.14: Boundary and load conditions for the 3-point bending load case of an inline engine

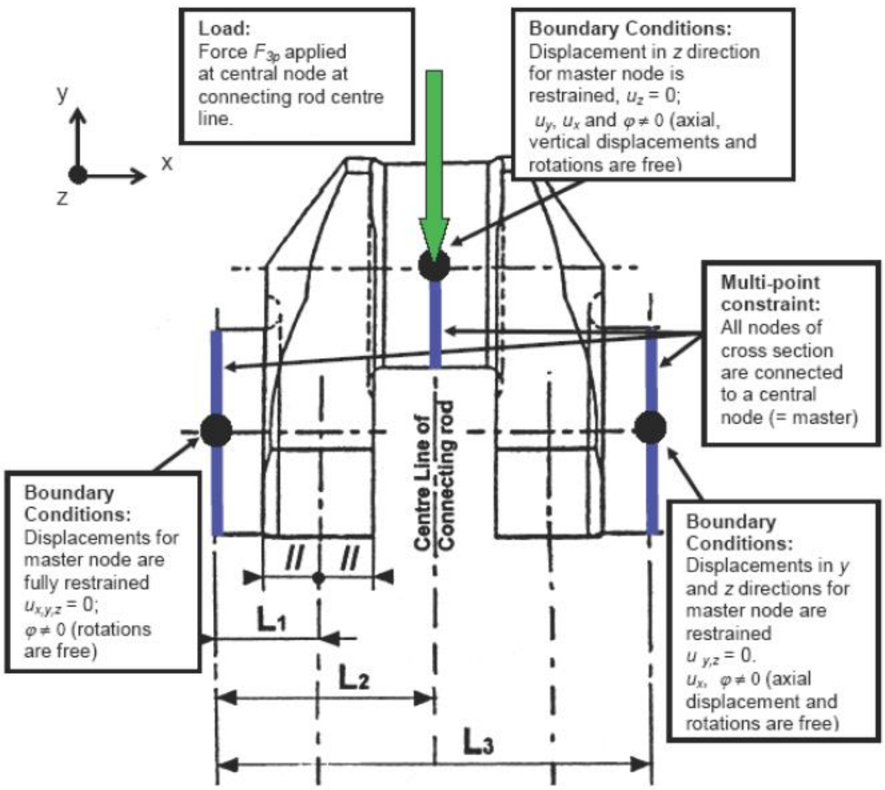
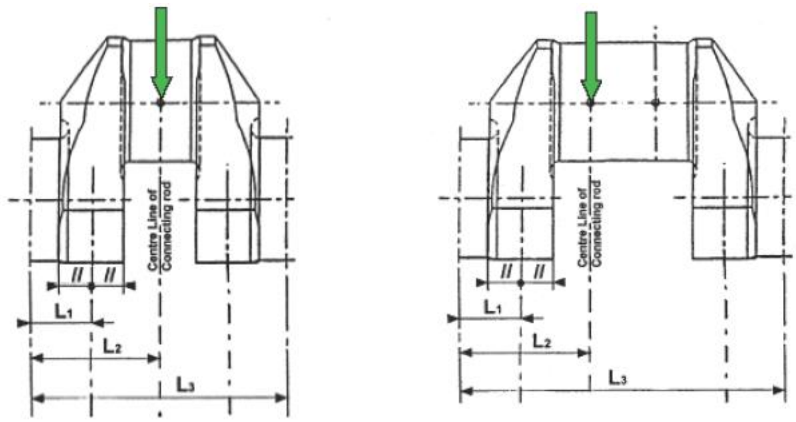


Figure 2.3.15: Load applications for in-line and V-type engines



3.10.4 Method 1

This method is analogue to the FVV investigation. The results from 3-point and 4-point bending are combined as follows:

$$\sigma_{3P} = \sigma_{N3P} \cdot \beta_B + \sigma_{Q3P} \cdot \beta_Q$$

where:

σ_{3P} = as found by the FE calculation

σ_{N3P} = Nominal bending stress in the web centre due to the force F_{3P} [N] applied to the centre-line of the actual connecting rod, see Figure 2.3.15

β_B = as determined in paragraph 3.10.3(2)

σ_{Q3P} = $Q_{3P}/(B \cdot W)$ where Q_{3P} is the radial (shear) force in the web due to the force F_{3P} [N] applied to the centreline of the actual connecting rod, see also Figures 2.3.3 and 2.3.4

3.10.5 Method 2

This method is not analogous to the FVV investigation. In a statically determined system with one crank throw supported by two bearings, the bending moment and radial (shear) force are proportional. Therefore the journal fillet SCF can be found directly by the 3-point bending FE calculation. The SCF is then calculated according to

$$\beta_{BQ} = \frac{\sigma_{3P}}{\sigma_{N3P}}$$

For symbols see 3.10.4.

When using this method the radial force and stress determination in this section becomes superfluous. The alternating bending stress in the journal fillet as per 3.3.1(3) is then evaluated:

$$\sigma_{BQ} = \pm\left|\beta_{BQ} \cdot \sigma_{BFN}\right|$$

Note that the use of this method does not apply to the crankpin fillet and that this SCF must not be used in connection with calculation methods other than those assuming a statically determined system as in this section.

3.11 Guidance for Evaluation of Fatigue Tests

3.11.1 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 document. Other statistical evaluation methods may also be applied.

3.11.1.1 Small specimen testing

For crankshafts without any fillet surface treatment, the fatigue strength can be determined by testing small specimens taken from a full-size crank throw. When other areas in the vicinity of the fillets are surface treated introducing residual stresses in the fillets, this approach cannot be applied.

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.

3.11.1.2 Full-size crank throw testing

For crankshafts with surface treatment the fatigue strength can only be determined through testing of full size crank throws. For cost reasons, this usually means a low number of crank throws. 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.

3.11.2 Evaluation of test results

3.11.2.1 Principles

Prior to fatigue testing the crankshaft must 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.

The test samples should 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 should be corrected with a confidence interval: a 90% confidence interval may be used both for the sample mean and the standard deviation.

The test results, when applied in this SECTION, shall be evaluated to represent the mean fatigue strength, with or without taking into consideration the 90% confidence interval as mentioned above. The standard deviation should 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.

If the evaluation aims to find a relationship between (static) mechanical properties and the fatigue strength, the relation must be based on the real (measured) mechanical properties, not on the specified minimum properties.

The calculation technique presented in 3.11.2.4 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.

3.11.2.2 Staircase method

In the original staircase method, the first specimen is subjected to a stress corresponding to the expected average fatigue strength. If the specimen survives 10^7 cycles, it is discarded and the next specimen is subjected to a stress that is one increment above the previous, i.e. a survivor is always followed by the next using a stress one increment above the previous. The increment should be selected to correspond to the expected level of the standard deviation.

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. With this approach, the sum of failures and run-outs is equal to the number of specimens. This original staircase method is only suitable when a high number of specimens are available. Through simulations it has been found that the use of about 25 specimens in a staircase test leads to a sufficient accuracy in the result.

3.11.2.3 Modified staircase method

When a limited number of specimens are available, it is advisable to apply the modified staircase method. Here the first specimen is subjected to a stress level that is most likely well below the average fatigue strength. When this specimen has survived 10^7 cycles, this same specimen is subjected to a stress level one increment above the previous. The increment should be selected to correspond to the expected level of the standard deviation. This is continued with the same specimen until failure. 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.

With this approach, the number of failures usually equals the number of specimens. The number of run-outs, counted as the highest level where 10^7 cycles were reached, also equals the number of specimens.

The acquired result of a modified staircase method should 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).

If the confidence calculation is desired or necessary, the minimum number of test specimens is 3.

3.11.2.4 Calculation of sample mean and standard deviation

A hypothetical example of tests for 5 crank throws is presented further in the subsequent text. 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

Furthermore, the method distinguishes between

Less frequent event is failures C=1

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, and vice versa.

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.

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. The sample mean can be calculated as follows:

$$\overline{S_\alpha} = S_{\alpha 0} + d\left(\frac{A}{F} - \frac{1}{2}\right) \qquad \text{when } C = 1$$

$$\overline{S_\alpha} = S_{\alpha 0} + d\left(\frac{A}{F} + \frac{1}{2}\right) \qquad \text{when } C = 2$$

The standard deviation can be found by

$$s = 1{,}62\ d\left(\frac{F \cdot B - A^2}{F^2} + 0{,}029\right)$$

where:

$S_{\alpha 0}$ = is the lowest stress level for the less frequent occurrence

d = is the stress increment

$F = \sum fi$

$A = \sum i \cdot fi$

$B = \sum i^2 \cdot fi$

i = is the stress level numbering

fi = is the number of samples at stress level i

The formula for the standard deviation is an approximation and can be used when

$$\frac{B \cdot F - A^2}{F^2} > 0{,}3 \quad \text{and} \quad 0{,}5\ s < d < 1{,}5\ s$$

If any of these two conditions are not fulfilled, a new staircase test should be considered or the standard deviation should be taken quite large in order to be on the safe side.

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.

3.11.2.5 Confidence interval for mean fatigue limit

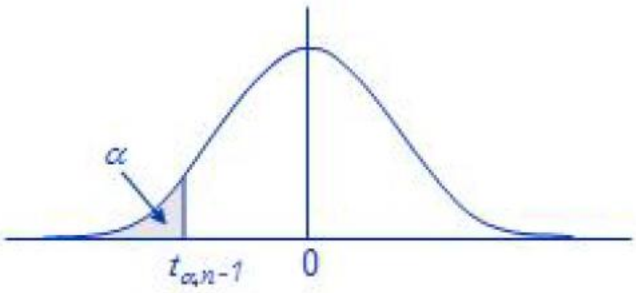
If the staircase fatigue test is repeated, the sample mean and the standard deviation will most likely be different from the previous test. Therefore, it is necessary to assure with a given confidence that the repeated test values will be above the chosen fatigue limit by using a confidence interval for the sample mean.

The confidence interval for the sample mean value with unknown variance is known to be distributed according to the t-distribution (also called student's t-distribution, see Figure 2.3.16) which is a distribution symmetric around the average.

If S_α 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-α)·100% confidence interval for the mean is:

$$P\left(S_\alpha - t_{\alpha,n-1}\frac{s}{\sqrt{n}} < S_{\alpha X\%}\right) = 1 - \alpha$$

Figure 2.3.16: Student's t-distribution



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 Figure 2.3.16 shows the t-value for (1 - α)·100% confidence interval for the sample mean.

The resulting confidence interval is symmetric around the empirical mean of the sample values, and the lower endpoint can be found as:

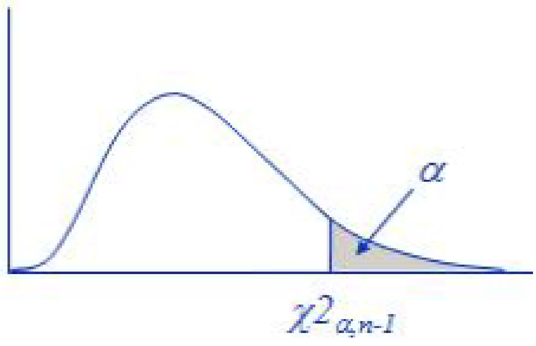
$$S_{\alpha X\%} = S_{\alpha} - t_{\alpha,n-1} \frac{s}{\sqrt{n}}$$

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.

3.11.2.6 Confidence interval for standard deviation

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, Figure 2.3.17.

Figure 2.3.17: Chi-square distribution



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. The figure shows the chi-square for (1-α)·100% confidence interval for the variance.

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 - α)·100% confidence interval for the variance is:

$$P\left(\frac{(n-1)\, s^2}{\sigma^2} < X^2{}_{\alpha,n-1}\right) = 1 - \alpha$$

A (1–α)·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 found by

$$s_{X\%} = \sqrt{\frac{n-1}{X^2{}_{\alpha,n-1}}} \cdot s$$

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.

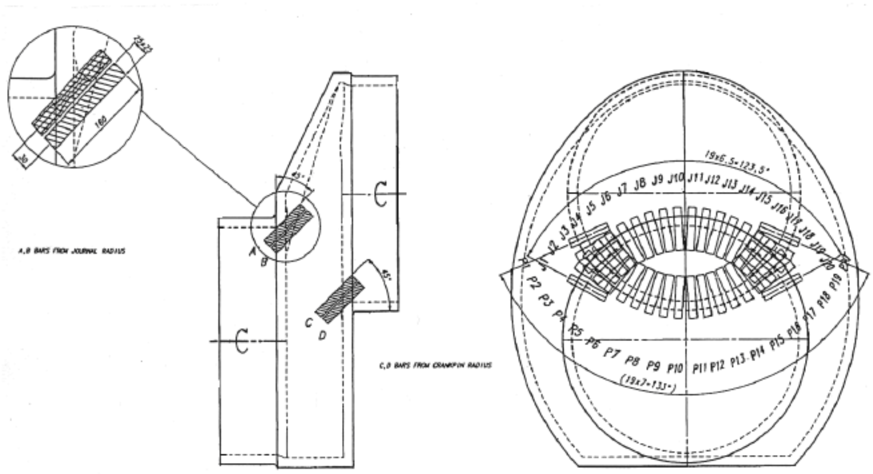
3.11.3 Small specimen testing

In this connection, a small specimen is considered to be one of the specimens taken from a crank throw. Since the specimens shall be representative for the fillet fatigue strength, they should be taken out close to the fillets, as shown in Figure 2.3.18.

It should 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.

The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

Figure 2.3.18: Specimen locations in a crank throw



3.11.3.1 Determination of bending fatigue strength

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.

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 shall be taken in 45 degrees angle as shown in Figure 2.3.18.

a. 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 Figure 2.3.18.
b. If the objective of the testing is to document the influence of continuous grain flow (cgf) forging, the specimens should be restricted to the vicinity of the crank plane.

3.11.3.2 Determination of torsional fatigue strength

a. If the specimens are subjected to torsional testing, the selection of samples should follow the same guidelines as for bending above. The stress gradient influence has to be considered in the evaluation.
b. If the specimens are tested in push-pull and no further information is available, the samples should be taken out at an angle of 45 degrees 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).

3.11.3.3 Other test positions

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.

When using test results from a prolonged shaft piece, it must be considered how well the grain flow in that shaft piece is representative for the crank fillets.

3.11.3.4 Correlation of test results

The fatigue strength achieved by specimen testing shall be converted to correspond to the full-size crankshaft fatigue strength with an appropriate method (size effect).

When using the bending fatigue properties from tests mentioned in this section, it should 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, e.g. by using no credit for cgf. This approach is applicable when using the Gough Pollard criterion. However, this approach is not recognized when using the von Mises or a multi-axial criterion such as Findley.

If the found ratio between bending and torsion fatigue differs significantly from $\sqrt{3}$, one should consider replacing the use of the von Mises criterion with the Gough Pollard criterion. Also, if critical plane criteria are used, it must 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.

Any addition of influence factors must 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.

3.11.4 Full size testing

3.11.4.1 Hydraulic pulsation

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.

Although the applied load should 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.

Furthermore, it is important that the test rig provides boundary conditions as defined in 3.10.3(1) to 3.10.3(3). The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

3.11.4.2 Resonance tester

A rig for bending fatigue normally works with an R-ratio of -1. Due to operation close to resonance, the energy consumption is moderate. Moreover, the frequency is usually relatively high, meaning that 10^7 cycles can be reached within some days. Figure 2.3.19 shows a layout of the testing arrangement.

The applied load should be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains.

Clamping around the journals must 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.

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.

A rig for torsion fatigue can also be arranged as shown in Figure 2.3.20. 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.

Figure 2.3.19: An example of testing arrangement of the resonance tester for bending loading

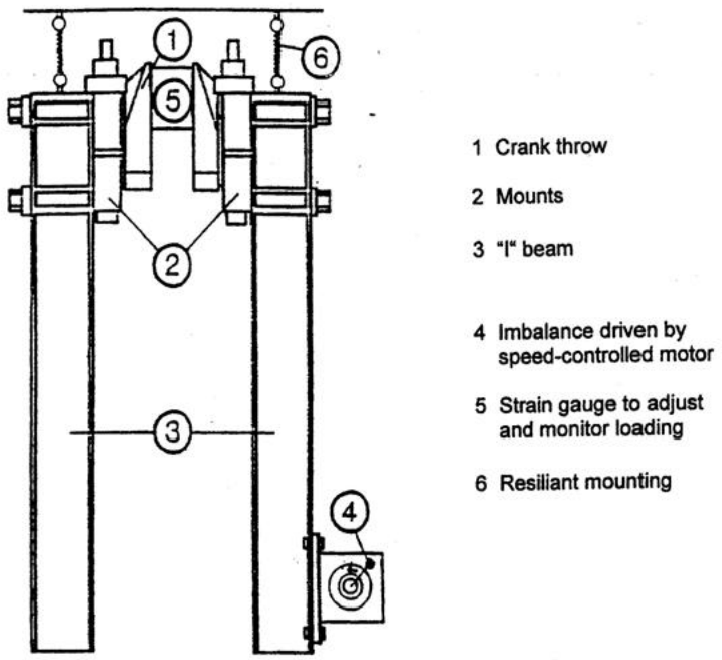
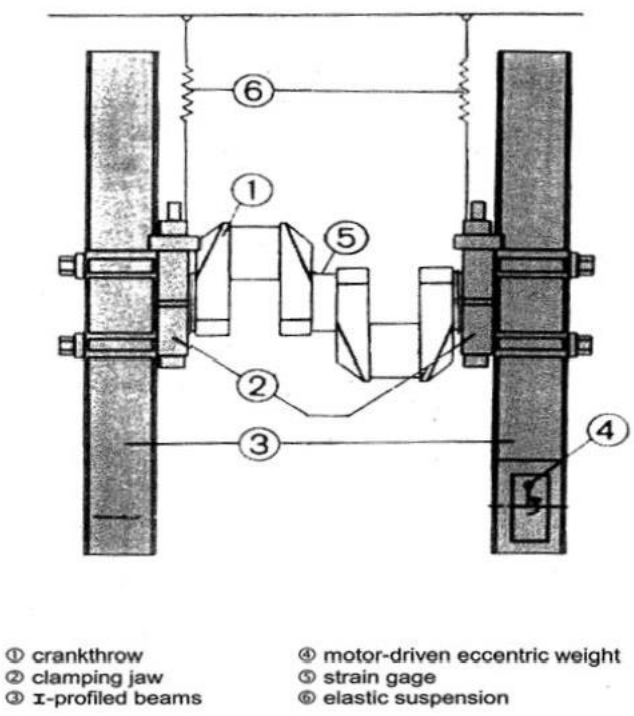


Figure 2.3.20: An example of testing arrangement of the resonance tester for torsion loading with double crank throw section



Since sideway movements can cause some bending stresses, the plain portions of the crankpins should also be provided with strain gauges arranged to measure any possible bending that could have an influence on the test results.

Similarly, to the bending case the applied load shall be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains as well.

3.11.4.3 Use of results and crankshaft acceptability

In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability, (see 3.11), the Gough-Pollard approach and the maximum principal equivalent stress formulation can be applied for the following cases:

Related to the crankpin diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BH}}{\sigma_{DWCT}}\right)^2 + \left(\frac{\tau_{BH}}{\tau_{DWCT}}\right)^2}\right)^{-1}$$

where:

σ_{DWCT} = fatigue strength by bending testing

τ_{DWCT} = fatigue strength by torsion testing

Related to crankpin oil bore:

$$Q = \frac{\sigma_{DWOT}}{\sigma_v} \; ; \qquad \sigma_v = \frac{1}{3}\,\sigma_{BO}\left[1 + 2\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

where:

σ_{DWOT} = fatigue strength by means of largest principal stress from torsion testing

Related to the journal diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BG}}{\sigma_{DWJT}}\right)^2 + \left(\frac{\tau_G}{\tau_{DWJT}}\right)^2}\right)^{-1}$$

where:

σ_{DWJT} = fatigue strength by bending testing

τ_{DWJT} = fatigue strength by torsion testing

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 according to the calculation where the surface treatment had not been taken into account.

3.11.5 Use of existing results for similar crankshafts

For fillets or oil bores without surface treatment, the fatigue properties found by testing may be used for similar crankshaft designs providing:

- Material:
  - Similar material type
  - Cleanliness on the same or better level
  - The same mechanical properties can be granted (size versus hardenability)

- Geometry:
  - Difference in the size effect of stress gradient is insignificant or it is considered
  - Principal stress direction is equivalent. See 3.11.3.

- Manufacturing:
  - Similar manufacturing process

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.

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.

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.

3.12 Guidance for Calculation of Surface Treated Fillets and Oil Bore Outlets

3.12.1 Introduction

This paragraph deals with surface treated fillets and oil bore outlets. The various treatments are explained and some empirical formulae are given for calculation purposes. Conservative empiricism has been applied intentionally, in order to be on the safe side from a calculation standpoint.

Please note that measurements or more specific knowledge should be used if available. However, in the case of a wide scatter (e.g. for residual stresses) the values should be chosen from the end of the range that would be on the safe side for calculation purposes.

3.12.2 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.

3.12.2.1 Surface treatment methods

Table 2.3.1 covers possible treatment methods and how they influence the properties that are decisive for the fatigue strength.

Table 2.3.1: 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

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 document. In addition, die quenching can be considered in the same way as induction hardening.

3.12.3 Calculation principles

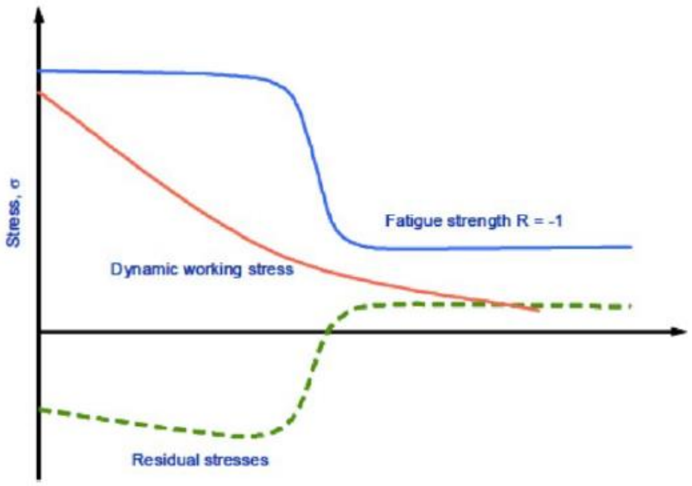
The basic principle is that the alternating working stresses shall be below the local fatigue strength (including the effect of surface treatment) wherein non-propagating cracks may occur, see also 3.12.6.1 for details. This is then divided by a certain safety factor. This applies through the entire 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.

Consideration of the local fatigue strength shall include the influence of the local hardness, residual stress and mean working stress. The influence of the 'giga-cycle effect', especially for initiation of subsurface cracks, should be covered by the choice of safety margin.

It is of vital importance that the extension of hardening/peening in an area with concentrated stresses be duly considered. 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.

Alternating and mean working stresses must 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. Figure 2.3.21 indicates this principle in the case of induction hardening. The base axis is either the depth (perpendicular to the surface) or along the fillet contour.

Figure 2.3.21: Stresses as functions of depth, general principles



The acceptability criterion should 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.

3.12.3.1 Evaluation of local fillet stresses

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 3.10. 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. 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 3.4 if within its validity range, and a relative stress gradient inversely proportional to the fillet radius. Bending and torsional stresses must be addressed separately. The combination of these is addressed by the acceptability criterion.

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. These functions $\alpha_{B-local}$ and $\alpha_{T-local}$ have different shapes due to the different stress gradients.

The SCFs α_B and α_T are valid at the surface. The local $\alpha_{B-local}$ and $\alpha_{T-local}$ drop with increasing depth. The relative stress gradients at the surface depend on the kind of stress raiser, but for crankpin fillets they can be simplified to $2/R_H$ in bending and $1/R_H$ in torsion. The journal fillets are handled analogously by using R_G and D_G. The nominal stresses are assumed to be linear from the surface to a midpoint in the web between the crankpin fillet and the journal fillet for bending and to the crankpin or journal centre for torsion.

The local SCFs are then functions of depth t according to Equation 3.1 as shown in Figure 2.3.22 for bending and respectively for torsion in Equation 3.2 and Figure 2.3.23.

$$\alpha_{B-local} = (\alpha_B - 1)e^{\frac{-2t}{R_H}} + 1 - \left(\frac{2t}{\sqrt{W^2 + S^2}}\right)^{\frac{0,6}{\sqrt{\alpha_B}}} \qquad (3.1)$$

$$\alpha_{T-local} = (\alpha_T - 1)e^{\frac{-t}{R_H}} + 1 - \left(\frac{2t}{D}\right)^{\frac{1}{\sqrt{\alpha_T}}} \qquad (3.2)$$

Figure 2.3.22: Bending SCF in the crankpin fillet as a function of depth. The corresponding SCF for the journal fillet can be found by replacing R_H with R_G

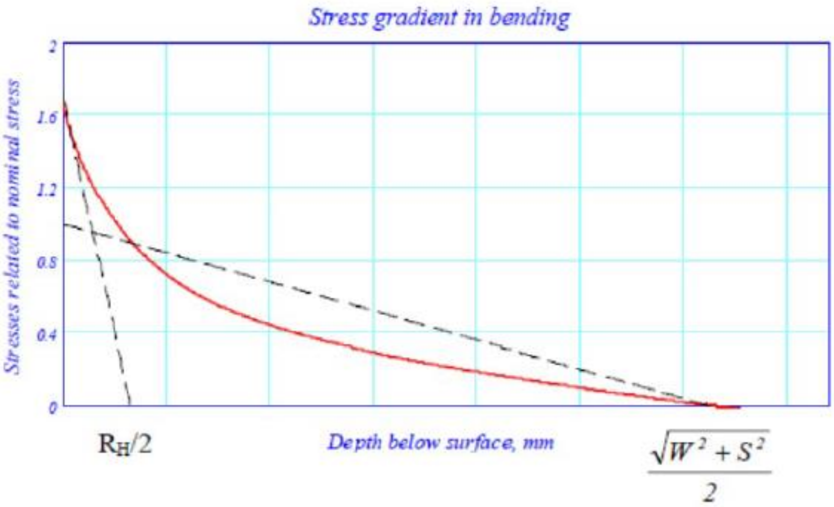


Figure 2.3.23: Torsional SCF in the crankpin fillet as a function of depth. The corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G

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 should be used to determine the actual stresses in the transition zone.

3.12.3.2 Evaluation of oil bore stresses

Stresses in the oil bores can be determined also by FEA. The element size should be less than 1/8 of the oil bore diameter D_o and the element mesh quality criteria should be followed as prescribed in

3.10. The fine element mesh should continue well beyond a radial depth corresponding to the hardening depth.

The loads to be applied in the FEA are the torque, see 3.10.3(1), and the bending moment, with four-point bending as in 3.10.3(2).

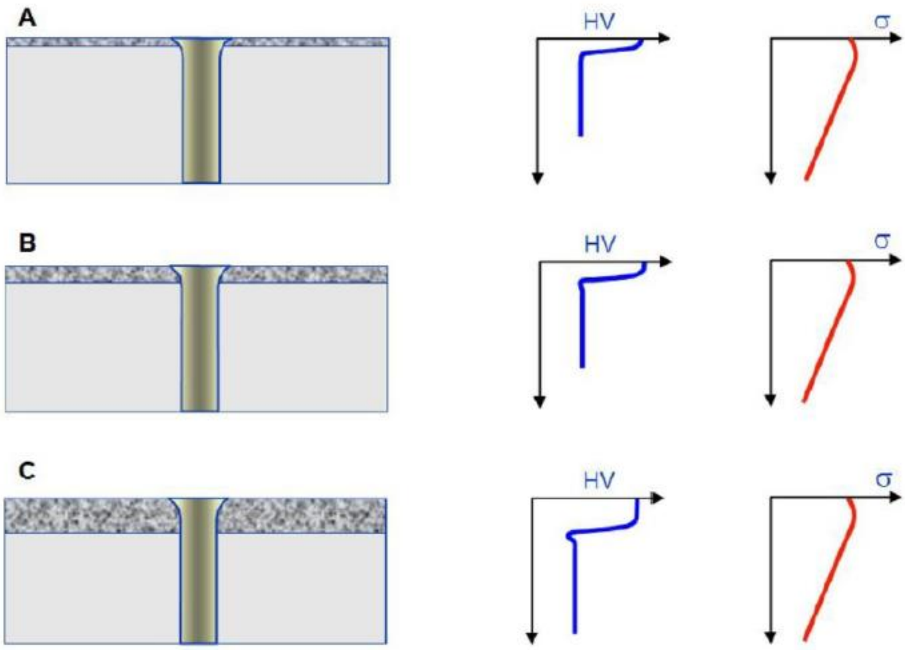
If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from 3.4 if within its applicability range. Bending and torsional stresses at the point of peak stresses are combined as in 3.6.

Figure 2.3.24 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.

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 Figure 2.3.24, for shallow (A) and intermediate (B) hardening, the transition point practically coincides with the point of maximal stresses. For deep 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.

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. These functions $\gamma_{B-local}$ and $\gamma_{T-local}$ have different shapes, because of the different stress gradients.

Figure 2.3.24: Stresses and hardness in induction hardened oil holes



The stress concentration factors γ_B and γ_T are valid at the surface. The local SCFs $\gamma_{B-local}$ and $\gamma_{T-local}$ drop with increasing depth. The relative stress gradients at the surface depend on the kind of stress raiser, but for crankpin oil bores they can be simplified to 4/D_o in bending and 2/D_o in torsion. The local SCFs are then functions of the depth t:

$$\gamma_{B-local} = (\gamma_B - 1) \cdot e^{\frac{-4t}{D_o}} + 1 \qquad (3.3)$$

$$\gamma_{T-local} = (\gamma_T - 1) \cdot e^{\frac{-2t}{D_O}} + 1 \qquad (3.4)$$

3.12.3.3 Acceptability criteria

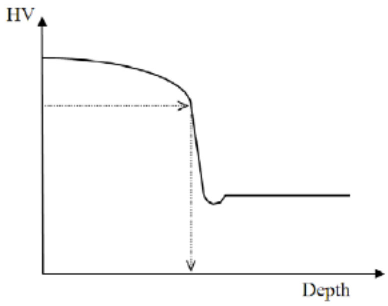
Acceptance of crankshafts is based on fatigue considerations; This section compares the equivalent alternating stress and the fatigue strength ratio to an acceptability factor of Q ≥ 1,15 for oil bore outlets, crankpin fillets and journal fillets. This shall be extended to cover also surface treated areas independent of whether surface or transition zone is examined.

3.12.4 Induction hardening

Generally, the hardness specification shall 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 HV0.5...HV5.

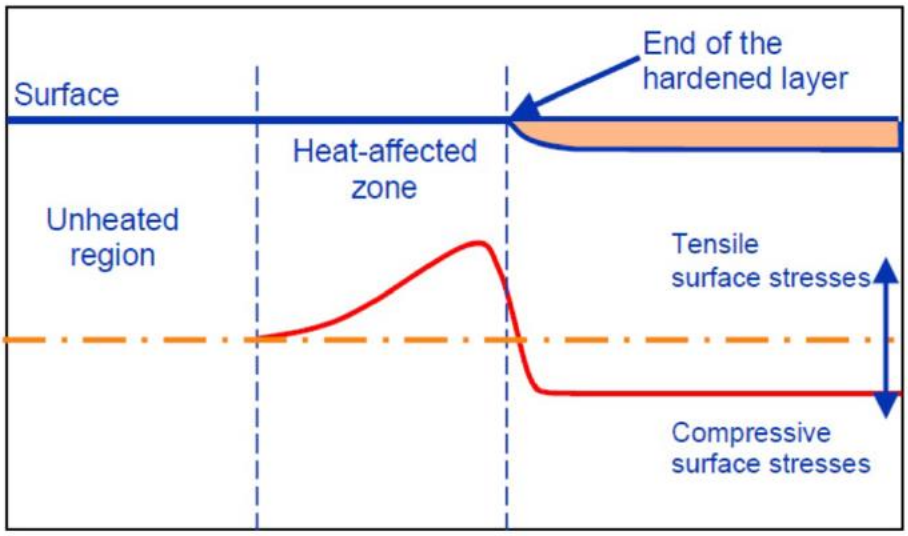
The induction hardening depth is defined as the depth where the hardness is 80% of the minimum specified surface hardness.

Figure 2.3.25: **Typical hardness as a function of depth. 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.**



In the case of crankpin or journal hardening only, the minimum distance to the fillet shall be specified due to the tensile stress at the heat-affected zone as shown in Figure 2.3.26.

Figure 2.3.26: Residual stresses along the surface of a pin and fillet



If the hardness-versus-depth profile and residual stresses are not known or specified, one may assume the following:

- The hardness profile consists of two layers (see Figure 2.3.25):
  - Constant hardness from the surface to the transition zone
  - Constant hardness from the transition zone to the core material
- Residual stresses in the hard zone of 200 MPa (compression)
- Transition-zone hardness as 90% of the core hardness unless the local hardness drop is avoided
- Transition-zone maximum residual stresses (von Mises) of 300 MPa tension

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.

3.12.4.1 Local fatigue strength

Induction-hardened crankshafts will suffer fatigue either at the surface or at the transition to the core. The fatigue strengths, for both the surface and the transition zone, can be determined by fatigue testing of full size cranks as described in 3.11. 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.

Tests made with the core material only will not be representative since the tensile residual stresses at the transition are lacking. Alternatively, the surface fatigue strength can be determined empirically as follows where HV is the surface Vickers hardness. The Equation 4.1 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:

$$\sigma_{\mathrm{Fsurface}} = 400 + 0.5\,(\mathrm{HV} - 400) \qquad [\mathrm{MPa}] \qquad (4.1)$$

It has to be noted also that the mean stress influence of induction-hardened steels may be significantly higher than that for QT steels.

The fatigue strength in the transition zone, without taking into account any possible local hardness drop, shall be determined by the equation introduced in 3.7. For journal and respectively to crankpin fillet applies:

$$\sigma_{\mathrm{Ftransition,cpin}} = \pm\mathrm{K}\,(0{,}42\,\sigma_B + 39.3)\left[0{,}264 + 1{,}073\,\mathrm{Y}^{-0,2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{\mathrm{X}}}\right] \tag{4.2}$$

where:

$Y=D_G$ and $X=R_G$ for journal fillet

$Y=D$ and $X=R_H$ for crankpin fillet

$Y=D$ and $X=D_O/2$ for oil bore outlet

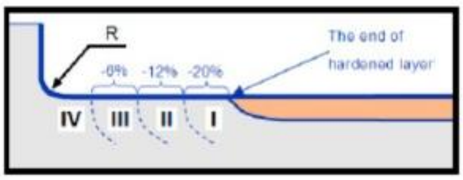
The influence of the residual stress is not included in Equation 4.2.

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.

When the residual stresses are known to be lower, also smaller value of subtraction shall be used. For low-strength steels the percentage chosen should be higher.

For the purpose of considering surface fatigue near the end of the hardened zone – i.e. in the heat-affected zone shown in the Figure 2.3.26 – the influence of the tensile residual stresses can be considered by subtracting a certain percentage, in accordance with Table 2.3.2, from the value determined by the above formula.

Table 2.3.2: The influence of tensile residual stresses at a given distance from the end of the hardening towards the fillet

1	0 to 1,0 of the max. hardening depth:	20%	
2	1,0 to 2,0 of the max. hardening depth:	12%	
3	2,0 to 3,0 of the max. hardening depth:	6%	
4	3,0 or more of the max. hardening depth:	0%	

3.12.5 Nitriding

The hardness specification shall include the surface hardness range (min and max) and the minimum and maximum depth. Only gas nitriding is considered. The referenced Vickers hardness is considered to be HV0.5.

The depth of the hardening is defined in different ways in the various standards and the literature. The most practical method to use in this context is to define the nitriding depth tN as the depth to a hardness of 50 HV above the core hardness.

The hardening profile should be specified all the way to the core. If this is not known, it may be determined empirically via the following formula:

$$\mathrm{HV(t)} = \mathrm{HV_{core}} + (\mathrm{HV_{surface}} - \mathrm{HV_{core}}) \left(\frac{50}{\mathrm{HV_{surface}} - \mathrm{HV_{core}}}\right)^{\left(\frac{t}{tN}\right)^2} \qquad (5.1)$$

where:

t = the local depth

$\mathrm{HV(t)}$ = hardness at depth t

$\mathrm{HV_{core}}$ = core hardness (minimum)

$\mathrm{HV_{surface}}$ = surface hardness (minimum)

tN = nitriding depth as defined above (minimum)

3.12.5.1 Local fatigue strength

It is important to note that in nitrided crankshaft cases, fatigue is found either at the surface or at the transition to the core. This means that the fatigue strength can be determined by tests as described in 3.11.

Alternatively, the surface fatigue strength (principal stress) can be determined empirically and conservatively as follows. This is valid for a surface hardness of 600 HV or greater:

$$\sigma_{\mathrm{Fsurface}} = 450\ \mathrm{MPa} \qquad (5.2)$$

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.

The fatigue strength in the transition zone can be determined by the equation introduced in 3.7. For crankpin and respectively to journal applies:

$$\sigma_{\mathrm{Ftransition,cpin}} = \pm \mathrm{K}\ (0{,}42\ \sigma_B + 39.3) \left[0{,}264 + 1{,}073\ \mathrm{Y}^{-0{,}2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{\mathrm{X}}}\right] \qquad (5.3)$$

where:

$Y=D_G$ and $X=R_G$ for journal fillet

$Y=D$ and $X=R_H$ for crankpin fillet

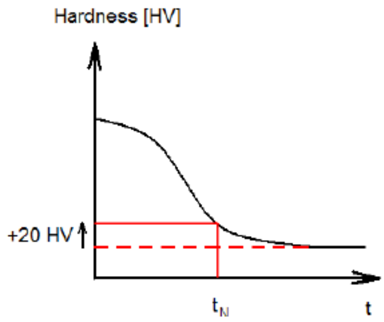
$Y=D$ and $X=D_O/2$ for oil bore outlet

Note that this fatigue strength is not assumed to include the influence of the residual stresses.

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. 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.

Although in principle the calculation should 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. 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 should be used. This means that the stresses at the transition to the core can be found by using the local SCF formulae mentioned earlier when inserting t=1,2 tN.

Figure 2.3.26: Sketch of the location for the artificial transition point in the depth direction



3.12.6 Cold forming

The advantage of stroke peening or cold rolling of fillets is the compressive residual stresses introduced in the high-loaded area. Even though surface residual stresses can be determined by X-ray diffraction technique and subsurface residual stresses can be determined through neutron diffraction, the local fatigue strength is virtually non-assessable on that basis since suitable and reliable correlation formulae are hardly known.

Therefore, the fatigue strength has to be determined by fatigue testing; see also 3.11. Such testing is normally carried out as four-point bending, with a working stress ratio of R = -1. 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.

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.

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 must be proportional to the fillet radius.

3.12.6.1 Stroke peening by means of a ball

The fatigue strength obtained can be documented by means of full size crank tests or by empirical methods if applied on the safe side. If both bending and torsion fatigue strengths have been investigated and differ from the ratio $\sqrt{3}$, the von Mises criterion should be excluded.

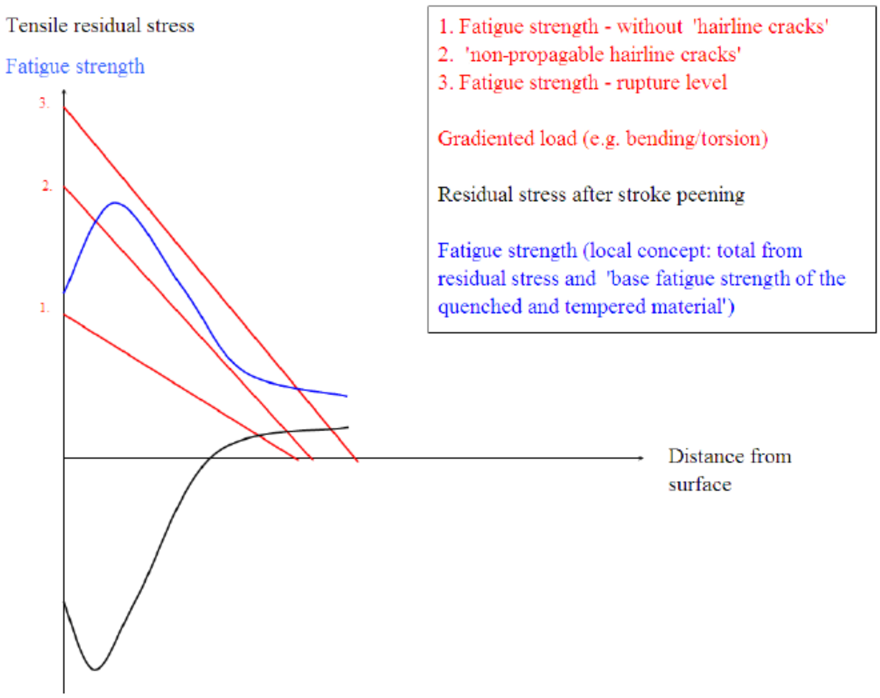
If only bending fatigue strength has been investigated, the torsional fatigue strength should be assessed conservatively. If the bending fatigue strength is concluded to be x% above the fatigue strength of the non-peened material, the torsional fatigue strength should not be assumed to be more than 2/3 of x% above that of the non-peened material.

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. This is illustrated in Figure 2.3.27 as gradient load 2.

In fatigue testing with full-size crankshafts these small “hairline cracks” should 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, should be considered for determination of the failure load level. This also applies if induction-hardened fillets are stroke-peened.

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. The effect on the fatigue strength of induction hardening and stroke peening the fillets shall be determined by a full-size crankshaft test.

Figure 2.3.27: Working and residual stresses below the stroke-peened surface. Straight lines 1…3 represent different possible load stress gradients



(a) 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:

- Ball size relative to fillet radius within ±10% in comparison to the tested crankshaft
- At least the same circumferential extension of the stroke peening
- 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
- Similar base material, e.g. alloyed quenched and tempered
- Forward feed of ball of the same proportion of the radius
- Force applied to ball proportional to base material hardness (if different)
- Force applied to ball proportional to square of ball radius

3.12.6.2 Cold rolling

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. If both, bending and torsion fatigue strengths have been investigated, and differ from the ratio $\sqrt{3}$, the von Mises criterion should be excluded.

If only bending fatigue strength has been investigated, the torsional fatigue strength should 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 should not be assumed to be more than 2/3 of x% above that of the non-rolled material.

(a) 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:

- At least the same circumferential extension of cold rolling
- 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
- Similar base material, e.g. alloyed quenched and tempered
- Roller force to be calculated so as to achieve at least the same relative (to fillet radius) depth of treatment

3.13 Guidance for Calculation of Stress Concentration Factors in the Oil Bore Outlets of crankshafts through utilisation of the Finite Element Method

3.13.1 General

The objective of the analysis described in this document 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. Because use of these formulae beyond any of the validity ranges can lead to erroneous results in either direction, the FEM-based method is highly recommended.

The SCF calculated according to the rules set forth in this document is defined as the ratio of FEM-calculated stresses to nominal stresses calculated analytically. In use in connection with the present method in this Section, principal stresses shall 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.

3.13.2 Model requirements

The basic recommendations and assumptions for building of the FE-model are presented in 3.13.2.1. The final FE-model must meet one of the criteria in 3.13.2.3.

3.13.2.1 Element mesh recommendations

For the mesh quality criteria to be met, construction of the FE model for the evaluation of stress concentration factors according to the following recommendations is advised:

- The model consists of one complete crank, from the main bearing centre line to the opposite side's main bearing centre line.
- The following element types are used in the vicinity of the outlets:
  - 10-node tetrahedral elements
  - 8-node hexahedral elements
  - 20-node hexahedral elements
- The following mesh properties for the oil bore outlet are used:
  - 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)
  - Recommended manner for element size in the fillet depth direction
    - First layer's thickness equal to element size of a
    - Second layer's thickness equal to element size of 2a
    - Third -layer thickness equal to element size of 3a
- In general, the rest of the crank should be suitable for numeric stability of the solver
- Drillings and holes for weight reduction have to be modelled

Submodeling may be used as long as the software requirements are fulfilled.

3.13.2.2 Material

This Section does not consider material properties such as Young's modulus (E) and Poisson's ratio (ν). In the FE analysis, these material parameters are required, as primarily strain is calculated and stress is derived from strain through the use of Young's modulus and Poisson's ratio. Reliable values for material parameters have to be used, either as quoted in the literature or measured from representative material samples.

For steel the following is advised: $E= 2{,}05 \cdot 10^5$ MPa and $\nu=0{,}3$.

3.13.2.3 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.

(a) Principal -stresses criterion
The quality of the mesh should 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 must be met:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0{,}03 \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

(b) Averaged/unaveraged -stresses criterion
The averaged/unaveraged –stresses criterion is based on observation of the discontinuity of stress results over elements at the fillet for the calculation of the SCF:

- Unaveraged nodal stress results calculated from each element connected to a $node_i$ should differ less than 5% from the 100% averaged nodal stress results at this $node_i$ at the location examined.

3.13.3 Load cases and assessment of stress

For substitution of the analytically determined SCF in this Section, calculation shall be performed for the following load cases.

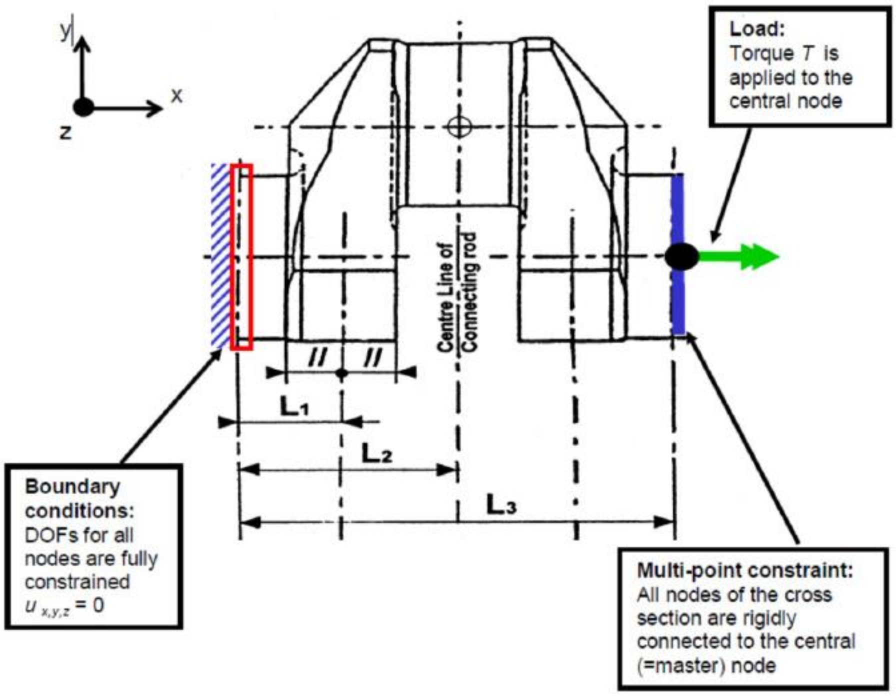
3.13.3.1 Torsion

The structure is loaded in pure torsion. The surface warp at the end faces of the model is suppressed.

Torque is applied to the central node, on the crankshaft axis. This node acts as the master node with six degrees of freedom, and is connected rigidly to all nodes of the end face.

The boundary and load conditions are valid for both in-line- and V- type engines.

Figure 2.3.28: Boundary and load conditions for the torsion load case



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 evaluated per 3.3.2.(2) of this SECTION with torque T:

$$\tau_N = \frac{T}{W_P}$$

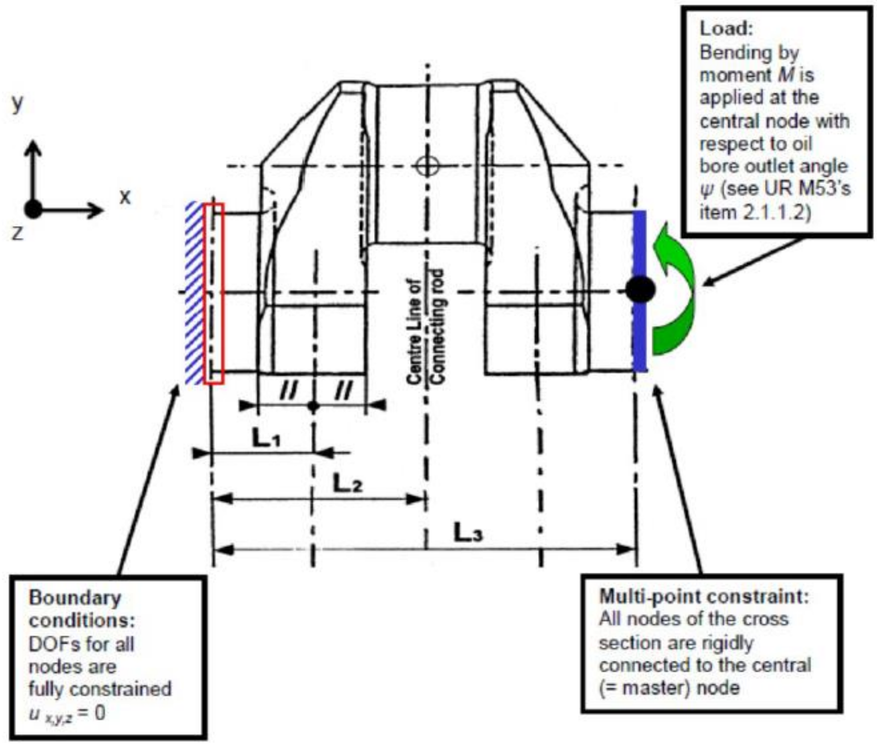
3.13.3.2 Bending

The structure is loaded in pure bending. The surface warp at the end faces of the model is suppressed.

The bending moment is applied to the central node on the crankshaft axis. This node acts as the master node, with six degrees of freedom, and is connected rigidly to all nodes of the end face.

The boundary and load conditions are valid for both in-line- and V- type engines.

Figure 2.3.29: Boundary and load conditions for the pure bending load case



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 per 3.3.1(2)b) of this SECTION, with bending moment M:

$$\sigma_N = \frac{M}{W_e}$$

SECTION 4 Construction of engine body

4.1 Crankcases

4.1.1 Crankcases are to be of solid construction with solid and securely fastened doors.

4.2 Welded structures

4.2.1 Complicated weldments should be avoided by using steel castings.

4.2.2 The intersection of weldments should be avoided.

4.2.3 The use welded joints in the construction of the major parts of engine structures should be restricted as far as possible.

4.2.4 The use of double welded butt joints is required wherever possible because of their superior fatigue strength.

4.2.5 Due care is to be taken for the consideration of stress concentrations. Sharp corners and abrupt changes in section should be avoided.

4.2.6 In the construction of girder and frame assemblies the welding of many small pieces should be avoided.

4.3 Welded joints

4.3.1 Joints subject to heavy loads should be designed as continuous full-strength welds. Their arrangement should allow adequate inspection.

4.3.2 In girders where stiffeners are used the attachment to the flange should be made by using full penetration welds.

SECTION 5 Safety devices

5.1 Cylinder overpressure monitoring of internal combustion engines

5.1.1 Means are to be provided to indicate a predetermined overpressure in the cylinders of engines having a bore exceeding 230 mm.

5.1.2 For indication of overpressure in the cylinder the sentinel valves may be replaced by effective warning devices (audible or visible), type approved by LHR.

5.2 Alarm devices of internal combustion engines (IACS UR M2 (1971))

5.2.1 Main and auxiliary engines, above 37 kW, must be fitted with an alarm device with audible and luminous signals for failure of the lubricating oil system.

5.3 Speed governor and overspeed protective device (IACS UR M3 Rev.6 (2018))

5.3.1 Speed governor and overspeed protective device for main internal combustion engines.

(1) Each main engine is to be fitted with a speed governor so adjusted that the engine speed cannot exceed the rated speed by more than 15%.

(2) In addition to this speed governor each main engine having a rated power of 220 kW and above, and which can be declutched or which drives a controllable pitch propeller, is to be fitted with a separate overspeed protective device so adjusted that the engine speed cannot exceed the rated speed by more than 20%. Equivalent arrangements may be accepted upon special consideration. The overspeed protective device, including its driving mechanism, has to be independent from the required governor.

(3) When electronic speed governors of main internal combustion engines form part of a remote-control system, they are to comply with Part 8, Chapter 1, SECTION 5, 5.3 and namely with the following conditions:

– if lack of power to the governor may cause major and sudden changes in the present speed and direction of thrust of the propeller, back-up power supply is to be provided;

– local control of the engines is always to be possible, as required by Part 8, Chapter 1, SECTION 5, 5.3.10, and, to this purpose, from the local control position it is to be possible to disconnect the remote signal, bearing in mind that the speed control according to 5.3.1(1), is not available unless an additional separate governor is provided for such local mode of control.

– in addition, electronic speed governors and their actuators are to be type tested according to Part 8, Chapter 1, SECTION 7, 7.2.

NOTE: The rated power and corresponding rated speed are those for which classification of the installation has been requested.

5.3.2 Speed governor, overspeed protective and governing characteristics of generator prime movers

(1) Prime movers for driving generators of the main and emergency sources of electrical power are to be fitted with a speed governor which will prevent transient frequency variations in the electrical network in excess of ±10% of the rated frequency with a recovery time to steady state conditions not exceeding 5 seconds, when the maximum electrical step load is switched on or off.
In the case when a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10% of the rated speed may be acceptable, provided this does not cause the intervention of the overspeed device as required by 5.3.2(5).

(2) At all loads between no load and rated power the permanent speed variation should not be more than 5% of the rated speed.

(3) Prime movers are to be selected in such a way that they will meet the load demand within the ship’s mains.
Application of electrical load should be possible with 2 load steps and must be such that prime movers - running at no load - can suddenly be loaded to 50% of the rated power of the generator followed by the remaining 50% after an interval sufficient to restore the speed to steady state. Steady state conditions should be achieved in not more than 5 seconds.
Steady state conditions are those at which the envelope of speed variation does not exceed + 1% of the declared speed at the new power.
Application of electrical load in more than 2 load steps can only be permitted, if the conditions within the ship's mains permit the use of such prime movers which can only be loaded in more than 2 load steps (see Figure 2.5.1 for guidance on 4-stroke diesel engines expected maximum possible sudden power increase) and provided that this is already allowed for in the designing stage. This is to be verified in the form of system specifications to be approved and to be demonstrated at ship's trials. In this case, due consideration is to be given to the power required for the electrical equipment to be automatically switched on after black-out and to the sequence in which it is connected. This applies analogously also for generators to be operated in parallel and where the power has to be transferred from one generator to another in the event of any one generator has to be switched off.

(4) Emergency generator sets must satisfy the governor conditions as per items 5.3.2(1) and 5.3.2(2) even when:
(a) their total consumer load is applied suddenly, or
(b) their total consumer load is applied in steps, subject to:
– the total load is supplied within 45 seconds since power failure on the main switchboard
– the maximum step load is declared and demonstrated

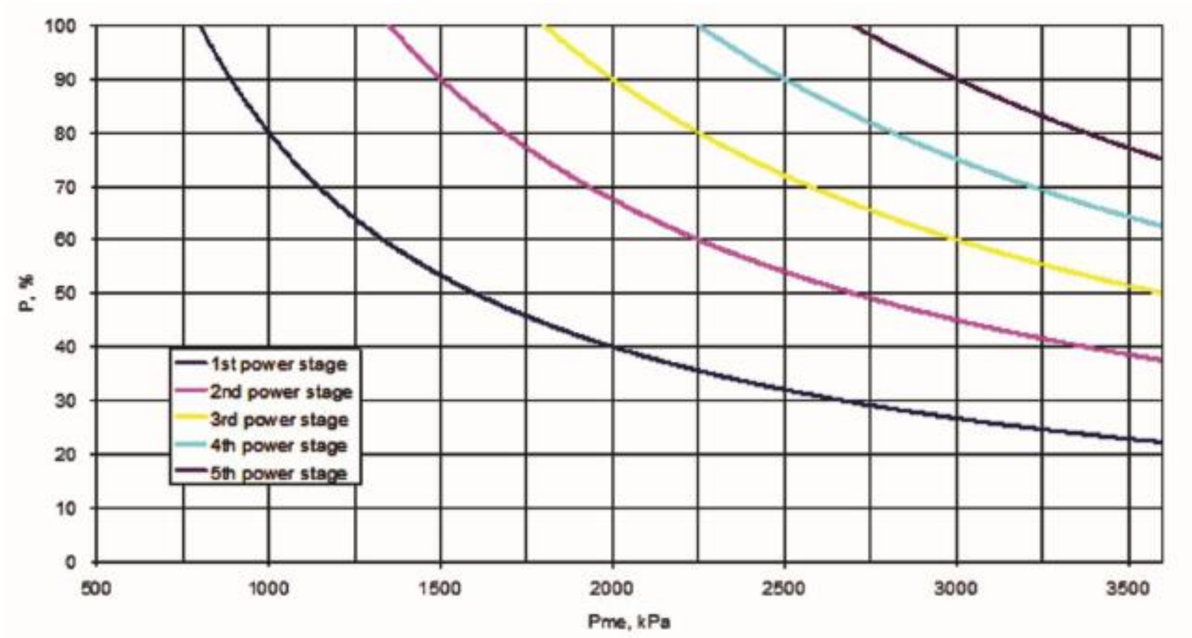
- the power distribution system is designed such that the declared maximum step loading is not exceeded
- the compliance of time delays and loading sequence with the above is to be demonstrated at ship's trials.

(5) In addition to the speed governor, each prime mover driving an electric generator and having a rated power of 220 kW and above must be fitted with a separate overspeed protective device so adjusted that the speed cannot exceed the rated speed by more than 15%

(6) For a.c. generating sets operating in parallel, the governing characteristics of the prime movers shall be such that within the limits of 20% and 100% total load the load on any generating set will not normally differ from its proportionate share of the total load by more than 15% of the rated power of the largest machine or 25% of the rated power of the individual machine in question, whichever is the less.
For an a.c. generating set intended to operate in parallel, facilities are to be provided to adjust the governor sufficiently fine to permit an adjustment of load not exceeding 5% of the rated load at normal frequency.

NOTE: For guidance, the loading for 4-stroke diesel engines may be limited as given by Figure 2.5.1.

Figure 2.5.1: 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)



Legend:

P_{me}: declared power mean effective pressure

P: power increase referred to declared power at site conditions

1: first power stage

2: second power stage

3: third power stage

4: fourth power stage

5: fifth power stage

5.4 Crankcase explosion relief valves for crankcases of internal combustion engines (IACS UR M9 Cor.2 (2007))

5.4.1 Internal combustion engines having a cylinder bore of 200 mm and above or a crankcase volume of 0,6 m^3 and above shall be provided with crankcase explosion relief valves in accordance with 5.4.2 to 5.4.13 as follows:

(1) Engines having a cylinder bore not exceeding 250 mm are to have at least one valve near each end, but, over eight crankthrows, an additional valve is to be fitted near the middle of the engine.
(2) Engines having a cylinder bore exceeding 250 mm but not exceeding 300 mm are to have at least one valve in way of each alternate crankthrow, with a minimum of two valves.
(3) Engines having a cylinder bore exceeding 300 mm are to have at least one valve in way of each main crankthrow.

NOTE:

1. The total volume of the stationary fixed parts within the crankcase may be discounted deduced in estimating the crankcase gross volume (rotating and reciprocating components are to be included in the gross volume).
2. Engines are to be fitted with components and arrangements complying with 5.4, except for 5.4.8, 5.4.9 and the second bullet point in 5.4.10, when:
   (a) an application for certification of an engine is dated on/after 1 January 2006; or
   (b) installed in new ships for which the date of contract for construction is on or after 1 January 2006.

   The requirements of 5.4.8, 5.4.9 and the second bullet point in 5.4.10 apply, in both cases above, from 1 January 2008.
3. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contract for construction”, refer to IACS Procedural Requirement (PR) No. 29.

5.4.2 The free area of each relief valve is to be not less than 45 cm^2.

5.4.3 The combined free area of the valves fitted on an engine must not be less than 115 cm^2 per cubic meter of the crankcase gross volume.

5.4.4 Crankcase explosion relief 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.

5.4.5 The valve discs in crankcase explosion relief valves are to be made of ductile material capable of withstanding the shock of contact with stoppers at the full open position.

5.4.6 Crankcase explosion relief valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than 0.02 N/mm^2 (0.2bar).

5.4.7 Crankcase explosion relief 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.

5.4.8 Crankcase explosion relief valves are to type tested in a configuration that represents the installation arrangements that will used on an engine in accordance with 9.8.

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

5.4.10 Crankcase explosion relief valves are to be provided with a copy manufacturer's installation and maintenance manual that is pertinent to the size and type of valve being supplied for installation on a particular engine. The manual is to contain the following information:

– Description of valve with details of function and design limits.
– Copy of type test certification.
– Installation instructions.
– Maintenance in service instructions to include testing and renewal of any sealing arrangements.
– Actions required after a crankcase explosion.

5.4.11 A copy of the installation and maintenance manual required by 5.4.10 is to be provided on board ship.

5.4.12 Plans of showing details and arrangements of crankcase explosion relief valves are to be submitted for approval in accordance with 1.3.

5.4.13 Valves are to be provided with suitable markings that include the following information:

– Name and address of manufacturer
– Designation and size
– Month/Year of manufacture
– Approved installation orientation

5.5 Protection of internal combustion engines against crankcase explosions (IACS M10 Rev.4 (2013))

5.5.1 Crankcase construction and crankcase doors are to be of sufficient strength to withstand anticipated crankcase pressures that may arise during a crankcase explosion taking into account the installation of explosion relief valves required by 5.4. Crankcase doors are to be fastened sufficiently securely for them not be readily displaced by a crankcase explosion.

5.5.2 Additional relief valves 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 exceeds 0,6 m^3.

5.5.3 Scavenge spaces in open connection to the cylinders are to be fitted with explosion relief valves.

5.5.4 Crankcase explosion relief valves are to comply with 5.4.

5.5.5 Ventilation of crankcase, and any arrangement which could produce a flow of external air within the crankcase, is in principle not permitted except for dual fuel engines where crankcase ventilation is to be provided in accordance with 8.4.3(b)(1).

(1) Crankcase ventilation pipes, where provided, are to be as small as practicable to minimize the inrush of air after a crankcase explosion.

(2) If a forced extraction of the oil mist atmosphere from the crankcase is provided (for mist detection purposes for instance), the vacuum in the crankcase is not to exceed $2{,}5 \cdot 10^{-4}$ N/mm², (2,5 mbar).

(3) To avoid interconnection between crankcases and the possible spread of fire following an explosion, crankcase of ventilation pipes and oil drain pipes for each engine are to be independent of any other engine.

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

5.5.7 A warning notice is to be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. 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 before a reasonable time, sufficient to permit adequate cooling after stopping the engine.

5.5.8 Oil mist detection arrangements (or engine bearing temperature monitors or equivalent devices) are required:

– for alarm and slow down purposes for low-speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore
– for alarm and automatic shutoff purposes for medium and high-speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore

Oil mist detection arrangements are to be of a type approved by classification societies and tested in accordance with 9.9 and comply with 5.5.9 to 5.5.20. Engine bearing temperature monitors or equivalent devices used as safety devices have to be of a type approved by classification societies for such purposes.

For the purpose of this section, the following definitions apply:

Low-Speed Engines means diesel engines having a rated speed of less than 300 rpm.

Medium-Speed Engines means diesel engines having a rated speed of 300 rpm and above,

but less than 1400 rpm.

High-Speed Engines means diesel engines having a rated speed of 1400 rpm and above.

NOTE: For equivalent devices for high-speed engines, refer to UI SC 133.

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

– Schematic layout of engine oil mist detection and alarm system showing location of engine crankcase sample points and piping or cable arrangements together with pipe dimensions to detector.
– Evidence of study to justify the selected location of sample points and sample extraction rate (if applicable) in consideration of the crankcase arrangements and geometry and the predicted crankcase atmosphere where oil mist can accumulate.
– The manufacturer's maintenance and test manual.
– Information relating to type or in-service testing of the engine with engine protection system test arrangements having approved types of oil mist detection equipment.

5.5.10 A copy of the oil mist detection equipment maintenance and test manual required by 5.5.9 is to be provided on board ship.

5.5.11 Oil mist detection and alarm information is to be capable of being read from a safe location away from the engine.

5.5.12 Each engine is to be provided with its own independent oil mist detection arrangement and a dedicated alarm.

5.5.13 Oil mist detection and alarm systems are to be capable of being tested on the test bed and board under engine at standstill and engine running at normal operating conditions in accordance with test procedures that are acceptable to LHR.

5.5.14 Alarms and shutdowns for the oil mist detection system are to be in accordance with Part 8, Chapter 1, SECTION 3, 3.1 and Part 8, Chapter 1, SECTION 4, 4.1 and the system arrangements are to comply with Part 8, Chapter 1, SECTION 6, 6.1 and Part 8, Chapter 1, SECTION 6, 6.2.

5.5.15 The oil mist detection arrangements are to provide an alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangements.

5.5.16 The oil mist detection system is 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.

5.5.17 Where oil mist detection equipment includes the use of programmable electronic systems, the arrangements are to be in accordance with individual LHR requirements for such systems.

5.5.18 Plans of showing details and arrangements of oil mist detection and alarm arrangements are to be submitted for approval in accordance with 1.3, Table 2.1.1 under item 28.

5.5.19 The equipment together with detectors is to be tested when installed on the test bed and on-board ship to demonstrate that the detection and alarm system functionally operates. The testing arrangements are to be to the satisfaction of LHR.

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

5.5.21 Where alternative methods are provided for the prevention of the build-up of oil mist that may lead to a potentially explosive condition within the crankcase details are to be submitted for consideration to LHR. The following information is to be included in the details to be submitted for consideration:

– Engine particulars – type, power, speed, stroke, bore and crankcase volume.
– Details of arrangements prevent the build-up of potentially explosive conditions within the crankcase, e.g., bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring, recirculation arrangements.
– Evidence to demonstrate that the arrangements are effective in preventing the build-up of potentially explosive conditions together with details of in-service experience.
– Operating instructions and the maintenance and test instructions.

5.5.22 Where it is proposed to use the introduction of inert gas into the crankcase to minimize a potential crankcase explosion, details of the arrangements are to be submitted to LHR for consideration.

5.6 Fire extinguishing systems for scavenge manifolds (IACS M12 (1972))

5.6.1 For crosshead type engines, scavenge spaces in open connection to the cylinder must be connected to an approved fire extinguishing system, which is to be entirely separate from the fire extinguishing system of the engine room.

SECTION 6 Piping

6.1 Lubricating oil filters

6.1.1 Suitable lubricating oil filters should be used in lubricating oil lines located in the main oil flow on the delivery side of the pumps.

6.1.2 The arrangement of that main flow filters should be such that to ensure easy cleaning or substitution, without interrupting operation.

6.1.3 With automatic filters, by-pass systems can be approved by LHR if simplex filters in accordance with 6.1.4 are fitted downstream. By-pass systems are not permitted for switch-over duplex filters.

6.1.4 Where simplex filters are used after the main filters the simplex filters are normally to be provided with a by-pass and a differential pressure alarm.

6.1.5 On main engines the lubricating oil for which is supplied from the engine oil sump and which have rated power less than 220 kW the use of simplex filters is permitted only if the latter are equipped with a pressure alarm and can be substituted during operation.

6.2 Exhaust systems

6.2.1 Exhaust gas lines are to be fitted with expansion compensators.

6.2.2 The exhaust gas lines of main and auxiliary engines are to be fitted with suitable silencers.

6.2.3 The surface temperature of the exhaust gas lines should normally not to exceed 220°C. This may be obtained by using proper insulation or cooling means.

6.2.4 Where lagging covering the exhaust piping system including flanges is oil-absorbing or may permit penetration of oil, the lagging is to be encased in sheet metal or equivalent.

6.2.5 Where the exhaust is led overboard near the waterline, means are to be provided to prevent water from being siphoned back to the engine. Where the exhaust is cooled by water spray, the exhaust pipes are to be self-draining overboard.

6.2.6 Where the exhausts of two or more engines are led to a common silencer or exhaust gas-heated boiler or economizer, an isolating device is to be provided in each exhaust pipe.

6.2.7 In two stroke engines with exhaust gas turbo-blowers operating on the impulse systems, due care is to be taken for preventing broken piston rings entering the turbine casing.

6.3 Starting air pipe systems and safety fittings

6.3.1 The compressor air inlets will be located in space free from oil vapor.

6.3.2 The air discharge pipe from the compressor is to be led direct to the starting air receivers.

6.3.3 Care is to be taken for intercepting oil and/or water in the air discharge. This may be obtained by using suitable separators or filters.

6.3.4 Drain valves for removing accumulations of oil and water are to be fitted on compressors, separators, filters and receivers.

6.3.5 The starting air pipe system from receivers to main and auxiliary engines is to be entirely separate from the compressor discharge pipe system. Stop valves on the receivers are to permit slow opening to avoid sudden pressure rises in the piping system. Valve chests and fittings in the piping system are to be of ductile material.

6.4 Protective devices for starting air mains (IACS M11 (1972))

6.4.1 In order to protect starting air mains against explosion arising from improper functioning of starting valves, the following devices must be fitted:

(1) an isolation non-return valve or equivalent at the starting air supply connection to each engine

(2) a bursting disc or flame arrester in way of the staring valve of each cylinder for direct reversing engines having a main starting manifold at the supply inlet to the starting air manifold for non-reversing engines.

Devices under (2) above may be omitted for engines having a bore not exceeding 230 mm.

SECTION 7 Starting arrangements of Internal Combustion Engines

7.1 Mechanical starting arrangements (IACS UR M61 Rev.1 (2022))

7.1.1 The arrangement for air starting is to be such that the necessary air for the first charge can be produced on board without external aid.

7.1.2 Where the main engine is arranged for starting by compressed air, two or more air compressors are to be fitted. At least one of the compressors is to be driven independent of the main propulsion unit and is to have a capacity not less than 50% of the total required.

7.1.3 The total capacity of air compressors is to be sufficient to supply within 1 hour the quantity of air needed to satisfy 7.1.5 by charging the receivers from atmospheric pressure. The capacity is to be approximately equally divided between the number of compressors fitted, excluding an emergency compressor which may be installed to satisfy 7.1.1.

7.1.4 Where the main engine is arranged for starting by compressed air, at least two starting air receivers of about equal capacity are to be fitted which may be used independently.

7.1.5 The total capacity of air receivers is to be sufficient to provide, without their being replenished, not less than 12 consecutive starts alternating between Ahead and Astern of each main engine of the reversible type, and not less than 6 starts of each main non-reversible type engine connected to a controllable pitch propeller or other device enabling the start without opposite torque. When other consumers such as auxiliary engines starting systems, control systems, whistle, etc., are to be connected to starting air receivers, their air consumption is also to be taken into account. Regardless of the above, for multi-engine installations the number of starts required for each engine may be reduced upon agreement with LHR depending upon the arrangement of the engines and the transmission of their output to the propellers.

7.2 Electric starting (IACS UR M61 Rev.1 (2022))

7.2.1 Where the main engine is arranged for electric starting, two separate batteries are to be fitted. The arrangement is to be such that the batteries cannot be connected in parallel. Each battery is to be capable of starting the main engine when in cold and ready to start conditions. The combined capacity of the batteries is to be sufficient without recharging to provide within 30 minutes the number of starts of main engines are required above in the case of air starting.

7.2.2 Electric starting arrangements for auxiliary engines are to have two separate batteries or may be supplied by separate circuits from the main engine batteries when such are provided. In the case of a single auxiliary engine only one battery may be required. The capacity of the batteries for starting the auxiliary engines is to be sufficient for at least three starts for each engine.

7.2.3 The starting batteries are to be used for starting and for the engine's own monitoring purposes only. Provisions are to be made to maintain continuously the stored energy at all times.

7.3 Starting of the emergency source of power

7.3.1 Emergency generators shall be capable of being readily started in their cold conditions down to a temperature of 0°C. If this is impracticable, or if lower temperatures are likely to be encountered, consideration is to be given to the provision and maintenance of heating arrangements, so that ready starting will be assured.

7.3.2 Each emergency generator that is arranged to be automatically started shall be equipped with approved starting devices with a storage energy capability of at least three consecutive starts. A second source of energy shall be provided for an additional three starts within 30 minutes unless hand (manual) starting can be demonstrated to be effective.

7.3.3 Provisions shall be made to maintain continuously the stored energy at all times, and for this purpose:

(1) Electrical and hydraulic starting systems shall be maintained from the emergency switchboard.
(2) Compressed air starting systems may be maintained by the main or auxiliary compressed air receivers, through a suitable non-return valve, or by an emergency air compressor energized by the emergency switchboard.
(3) All these starting, charging and energy storing devices shall be located in the emergency generator room.

7.3.4 When automatic starting is not required by the Rules and where it can be demonstrated as being effective, hand (manual) starting is permissible, such as manual cranking, inertial starters, manual hydraulic accumulators, powder charge cartridges.

7.3.5 When hand (manual) starting is not practicable, the provisions under 7.3.2 and 7.3.3 shall be complied with except that starting may be manually initiated.

SECTION 8 Control equipment

8.1 Engine room control platform

8.1.1 The main propulsion plant, along with reversing gear or controllable pitch propeller if exist, may be controlled from a control platform in the engine room.

8.1.2 The engine room control platform is to be equipped at least with the following main engine indicators:

- Lubricating oil pressure at engine inlet
- Engine speed/direction
- Cylinder cooling water pressure
- Fuel pressure at engine inlet
- Piston coolant pressure
- Starting air pressure
- Charge air pressure
- Control air pressure at engine inlet

8.1.3 On the control platform and/or directly on the engines indicators for the following are required:

- Fuel temperature at engine inlet
- Lubricating oil temperature
- Exhaust gas temperature
- Coolant temperature

– Only for engines running on heavy fuel oil

8.1.4 In case of geared transmissions or controllable pitch propellers, the scope of the control equipment mentioned in 8.1.2 and 8.1.3 is to be extended accordingly.

8.1.5 A red area should be marked on the pressure gauges and tachometers in order to indicate the non-permissible range of pressure or speed values.

8.1.6 A machinery alarm system is to be installed for the pressures and temperatures specified in 8.1.2 and 8.1.3 with the exception of the charge air pressure, the control air pressure, the exhaust gas temperature and the fuel pressure and temperatures.

8.2 Engine control from the bridge

8.2.1 If the engine plant is to be controlled only from the bridge, fault signals emitted by the alarm system specified in 8.1.6 are to be transmitted to the bridge. Faults may be signalled in accordance with Part 8. Indicators are to be mounted in the engine room and on the bridge control stand to show that the alarm system is operational.

8.2.2 Additional indicators are to be mounted on the bridge showing the starting air and control air pressures.

8.2.3 For engines with a power of up to 750 kW relaxations may be agreed with LHR.

8.3 Auxiliary engines

8.3.1 At least the following instruments are to be ergonomically fitted on the engine:

- Pressure gauge for cooling water pressure
- Pressure gauge for lubricating oil pressure
- Pressure gauge for fuel pressure
- Thermometer for cooling water temperature
- Tachometer

8.3.2 Engines of over 50 kW power are to be equipped with an engine alarm system responding to the lubricating oil pressure and to the pressure or flow-rate of the cooling water.

SECTION 9 Tests and trials

9.1 Manufacturing inspections

9.1.1 The manufacture of all engines with LHR classification is subject to supervision by LHR.

9.1.2 Where engine manufacturers have been approved by LHR as "Suppliers of Mass-Produced Engines", these engines are to be tested in accordance with 9.2.

9.2 Type testing of I.C. Engines (IACS UR M71 Cor.1 (2016))

9.2.1 General

9.2.1.1 Type approval of I.C. engine types consists of drawing approval, specification approval, conformity of production, approval of type testing programme, type testing of engines, review of the obtained results, and the issuance of the Type Approval Certificate. The maximum period of validity of a Type Approval Certificate is 5 years. The requirements for drawing approval and specification approval of engines and components are specified in Chapter 2 of this Part.

9.2.1.2 For the purpose of this UR, the following definitions apply:

- Low-Speed Engines means diesel engines having a rated speed of less than 300 rpm.
- Medium-Speed Engines means diesel engines having a rated speed of 300 rpm and above, but less than 1400 rpm.
- High-Speed Engines means diesel engines having a rated speed of 1400 rpm or above.

9.2.2 Objectives

9.2.2.1 The type testing, documented in 9.2, is to be arranged to represent typical foreseen service load profiles, as specified by the engine builder, as well as to cover for required margins due to fatigue scatter and reasonably foreseen in-service deterioration.

9.2.2.2 This applies to:

- Parts subjected to high cycle fatigue (HCF) such as connecting rods, cams, rollers and spring tuned dampers where higher stresses may be provided by means of elevated injection pressure, cylinder maximum pressure, etc.
- Parts subjected to low cycle fatigue (LCF) such as “hot” parts when load profiles such as idle - full load - idle (with steep ramps) are frequently used.
- Operation of the engine at limits as defined by its specified alarm system, such as running at maximum permissible power with the lowest permissible oil pressure and/or highest permissible oil inlet temperature.

9.2.3 Validity

9.2.3.1 Type testing is required for every new engine type intended for installation onboard ships subject to classification.

9.2.3.2 A type test carried out for a particular type of engine at any place of manufacture will be accepted for all engines of the same type built by licensees or the licensor, subject to each place of manufacture being found to be acceptable to LHR.

9.2.3.3 A type of engine is defined by:

- bore and stroke
- injection method (direct or indirect)
- valve and injection operation (by cams or electronically controlled)
- kind of fuel (liquid, dual-fuel, gaseous)
- working cycle (4-stroke, 2-stroke)
- turbo-charging system (pulsating or constant pressure)
- the charging air cooling system (e.g. with or without intercooler)
- cylinder arrangement (in-line or V) (1)
- cylinder power, speed and cylinder pressures (2)

De-rated engine

If an engine has been design approved, and internal testing per Stage A is documented to a rating higher than the one type tested, the Type Approval may be extended to the increased power/mep/rpm upon submission of an Extended Delivery Test Report at:

- Test at over speed (only if nominal speed has increased)
- Rated power, i.e. 100% output at 100% torque and 100% speed corresponding to load point 1., 2 measurements with one running hour in between
- Maximum permissible torque (normally 110%) at 100% speed corresponding to load point 3 or maximum permissible power (normally 110%) and speed according to nominal propeller curve corresponding to load point 3a., ½ hour
- 100% power at maximum permissible speed corresponding to load point 2, ½ hour

Integration Test

An integration test demonstrating that the response of the complete mechanical, hydraulic and electronic system is as predicted maybe carried out for acceptance of sub-systems (Turbo Charger, Engine Control System, Dual Fuel, Exhaust Gas treatment…) separately approved. The scope of these tests shall be proposed by the designer/licensor taking into account of impact on engine.

Notes:

1. One type test will be considered adequate to cover a range of different numbers of cylinders. However, a type test of an in-line engine may not always cover the V-version. Subject to the individual Societies' discretion, separate type tests may be required for the V-version. On the other hand, a type test of a V-engine covers the in-line engines, unless the bmep is higher.
   Items such as axial crankshaft vibration, torsional vibration in camshaft drives, and crankshafts, etc. may vary considerably with the number of cylinders and may influence the choice of engine to be selected for type testing.
2. The engine is type approved up to the tested ratings and pressures (100% corresponding to MCR). Provided documentary evidence of successful service experience with the classified rating of 100% is submitted, an increase (if design approved*) may be permitted without a new type test if the increase from the type tested engine is within:
   - 5% of the maximum combustion pressure, or
   - 5% of the mean effective pressure, or
   - 5% of the rpm

   Providing maximum power is not increased by more than 10%, an increase of maximum approved power may be permitted without a new type test provided engineering analysis and evidence of successful service experience in similar field applications (even if the application is not classified) or documentation of internal testing are submitted if the increase from the type tested engine is within:
   - 10% of the maximum combustion pressure, or
   - 10% of the mean effective pressure, or
   - 10% of the rpm

   * Only crankshaft calculation and crankshaft drawings, if modified.

9.2.4 Safety precautions

9.2.4.1 Before any test run is carried out, all relevant equipment for the safety of attending personnel is to be made available by the manufacturer/shipyard and is to be operational, and its correct functioning is to be verified.

9.2.4.2 This applies especially to crankcase explosive conditions protection, but also overspeed protection and any other shut down function.

9.2.4.3 The inspection for jacketing of high-pressure fuel oil lines and proper screening of pipe connections (as required in 9.2.8.9 fire measures) is also to be carried out before the test runs.

9.2.4.4 Interlock test of turning gear is to be performed when installed.

9.2.5 Test programme

9.2.5.1 The type testing is divided into 3 stages:

1. Stage A - internal tests.
   This includes some of the testing made during the engine development, function testing, and collection of measured parameters and records of testing hours. The results of testing required by LHR or stipulated by the designer are to be presented to LHR before starting stage B.
2. Stage B - witnessed tests.
   This is the testing made in the presence of LHR personnel.
3. Stage C - component inspection.
   This is the inspection of engine parts to the extent as required by LHR.

9.2.5.2 The complete type testing program is subject to approval by LHR. The extent the Surveyor's attendance is to be agreed in each case, but at least during stage B and C.

9.2.5.3 Testing prior to the witnessed type testing (stage B and C), is also considered as a part of the complete type testing program.

9.2.5.4 Upon completion of complete type testing (stage A through C), a type test report is to be submitted to LHR for review. The type test report is to contain:

- overall description of tests performed during stage A. Records are to be kept by the builders QA management for presentation to LHR.
- detailed description of the load and functional tests conducted during stage B.
- inspection results from stage C.

9.2.5.5 As required in 9.2.2 the type testing is to substantiate the capability of the design and its suitability for the intended operation. Special testing such as LCF and endurance testing will normally be conducted during stage A.

9.2.5.6 High speed engines for marine use are normally to be subjected to an endurance test of 100 hours at full load. Omission or simplification of the type test may be considered for the type approval of engines with long service experience from non-marine fields or for the extension of type approval of engines of a well-known type, in excess of the limits given in 9.2.3.

Propulsion engines for high speed vessels that may be used for frequent load changes from idle to full are normally to be tested with at least 500 cycles (idle - full load - idle) using the steepest load ramp that the control system (or operation manual if not automatically controlled) permits. The duration at each end is to be sufficient for reaching stable temperatures of the hot parts.

9.2.6. Measurements and recordings

9.2.6.1 During all testing the ambient conditions (air temperature, air pressure and humidity) are to be recorded.

9.2.6.2 As a minimum, the following engine data are to be measured and recorded:

- Engine r.p.m.
- Torque
- Maximum combustion pressure for each cylinder (1)
- Mean indicated pressure for each cylinder (1)
- Charging air pressure and temperature
- Exhaust gas temperature
- Fuel rack position or similar parameter related to engine load
- Turbocharger speed

- All engine parameters that are required for control and monitoring for the intended use (propulsion, auxiliary, emergency).

Notes:

1. For engines where the standard production cylinder heads are not designed for such measurements, a special cylinder head made for this purpose may be used. In such a case, the measurements may be carried out as part of Stage A and are to be properly documented. Where deemed necessary e.g. for dual fuel engines, the measurement of maximum combustion pressure and mean indicated pressure may be carried out by indirect means, provided the reliability of the method is documented.
   Calibration records for the instrumentation used to collect data as listed above are to be presented to - and reviewed by the attending Surveyor.
   Additional measurements may be required in connection with the design assessment.

9.2.7 Stage A - internal tests

9.2.7.1 During the internal tests, the engine is to be operated at the load points important for the engine designer and the pertaining operating values are to be recorded. The load conditions to be tested are also to include the testing specified in the applicable type approval programme.

9.2.7.2 At least the following conditions are to be tested:

- Normal case:
  The load points 25%, 50%, 75%, 100% and 110% of the maximum rated power for continuous operation, to be made along the normal (theoretical) propeller curve and at constant speed for propulsion engines (if applicable mode of operation i.e. driving controllable pitch propellers), and at constant speed for engines intended for generator sets including a test at no load and rated speed.
- The limit points of the permissible operating range. These limit points are to be defined by the engine manufacturer.
- For high speed engines, the 100 hr full load test and the low cycle fatigue test apply as required in connection with the design assessment.
- Specific tests of parts of the engine, required by the LHR or stipulated by the designer.

9.2.8 Stage B - witnessed tests

9.2.8.1 The tests listed below are to be carried out in the presence of a Surveyor. The achieved results are to be recorded and signed by the attending Surveyor after the type test is completed.

9.2.8.2 The over-speed test is to be carried out and is to demonstrate that the engine is not damaged by an actual engine overspeed within the overspeed shutdown system set-point. This test may be carried out at the manufacturer’s choice either with or without load during the speed overshoot.

9.2.8.3 Load points

The engine is to be operated according to the power and speed diagram (see Figure 1). The data to be measured and recorded when testing the engine at the various load points have to include all engine parameters listed in 9.2.6. The operating time per load point depends on the engine size (achievement of steady state condition) and on the time for collection of the operating values. Normally, an operating time of 0,5 hour can be assumed per load point, however sufficient time should be allowed for visual inspection by the Surveyor.

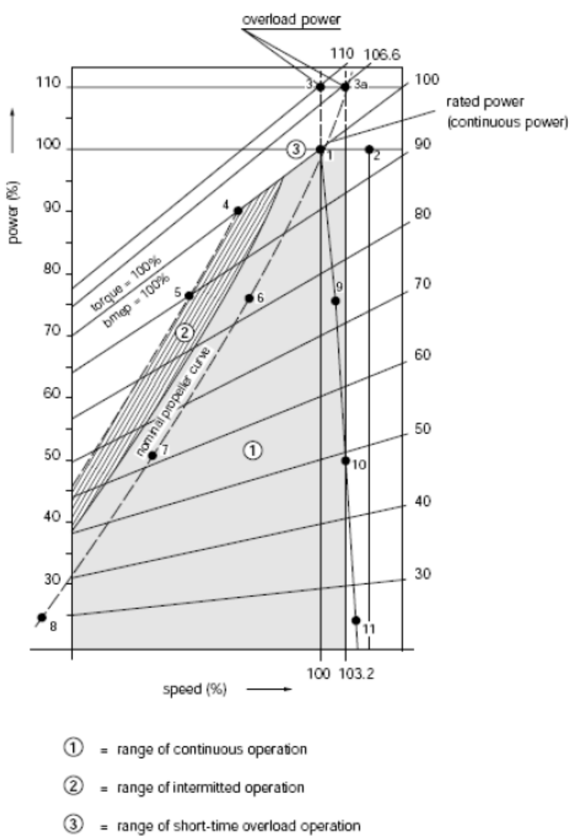
9.2.8.4 The load points are:

- Rated power (MCR), i.e. 100% output at 100% torque and 100% speed corresponding to load point 1, normally for 2 hours with data collection with an interval of 1 hour. If operation of the engine at limits as defined by its specified alarm system (e.g. at alarm levels of lub oil pressure and inlet temperature) is required, the test should be made here.

- 100% power at maximum permissible speed corresponding to load point 2.
- Maximum permissible torque (at least and normally 110%) at 100% speed corresponding to load at point 3, or maximum permissible power (at least and normally 110%) and 103,2% speed according to the nominal propeller curve corresponding to load point 3a. Load point 3a applies to engines only driving fixed pitch propellers or water jets. Load point 3 applies to all other purposes. Load point 3 (or 3a as applicable) is to be replaced with a load that corresponds to the specified overload and duration approved for intermittent use. This applies where such overload rating exceeds 110% of MCR. Where the approved intermittent overload rating is less than 110% of MCR, subject overload rating has to replace the load point at 100% of MCR. In such case the load point at 110% of MCR remains.
- Minimum permissible speed at 100% torque, corresponding to load point 4.
- Minimum permissible speed at 90% torque corresponding to load point 5. (Applicable to propulsion engines only).
- Part loads e.g. 75%, 50% and 25% of rated power and speed according to nominal propeller curve (i.e. 90,8%, 79,3% and 62,9% speed) corresponding to points 6, 7 and 8 or at constant rated speed setting corresponding to points 9, 10 and 11, depending on the intended application of the engine.
- Crosshead engines not restricted for use with C.P. propellers are to be tested with no load at the associated maximum permissible engine speed.

9.2.8.5 During all these load points, engine parameters are to be within the specified and approved values.

Figure 9.2.1: Load points



9.2.8.6 Operation with damaged turbocharger

For 2-stroke propulsion engines, the achievable continuous output is to be determined in the case of turbocharger damage. Engines intended for single propulsion with a fixed pitch propeller are to be able to run continuously at a speed (r.p.m.) of 40% of full speed along the theoretical propeller curve when one turbocharger is out of operation. (The test can be performed by either by-passing the turbocharger, fixing the turbocharger rotor shaft or removing the rotor.)

9.2.8.7 Functional tests

- Verification of the lowest specified propulsion engine speed according to the nominal propeller curve as specified by the engine designer (even though it works on a water-brake). During this operation, no alarm shall occur.
- Starting tests, for non-reversible engines and/or starting and reversing tests, for reversible engines, for the purpose of determining the minimum air pressure and the consumption for a start.
- Governor tests: tests for compliance with subsection 5.3.1 and 5.3.2 od this Chapter are to be carried out.

9.2.8.8 Integration test

For electronically controlled diesel engines, integration tests are to verify that the response of the complete mechanical, hydraulic and electronic system is as predicted for all intended operational modes. The scope of these tests is to be agreed with LHR for selected cases based on the FMEA required in SECTION 13 of this Chapter.

9.2.8.9 Fire protection measures

Verification of compliance with requirements for jacketing of high-pressure fuel oil lines, screening of pipe connections in piping containing flammable liquids and insulation of hot surfaces:

- The engine is to be inspected for jacketing of high-pressure fuel oil lines, including the system for the detection of leakage, and proper screening of pipe connections in piping containing flammable liquids.
- Proper insulation of hot surfaces is to be verified while running the engine at 100% load, alternatively at the overload approved for intermittent use. Readings of surface temperatures are to be done by use of Infrared Thermoscanning Equipment. Equivalent measurement equipment may be used when so approved by LHR. Readings obtained are to be randomly verified by use of contact thermometers.

9.2.9 Stage C - Opening up for Inspections

9.2.9.1 The crankshaft deflections are to be measured in the specified (by designer) condition (except for engines where no specification exists).

9.2.9.2 High speed engines for marine use are normally to be stripped down for a complete inspection after the type test.

9.2.9.3 For all the other engines, after the test run the components of one cylinder for in-line engines and two cylinders for V-engines are to be presented for inspection as follows (engines with long service experience from non-marine fields can have a reduced extent of opening):

- piston removed and dismantled
- crosshead bearing dismantled
- guide planes
- connecting rod bearings (big and small end) dismantled (special attention to serrations and fretting on contact surfaces with the bearing backsides)

- main bearing dismantled
- cylinder liner in the installed condition
- cylinder head, valves disassembled
- cam drive gear or chain, camshaft and crankcase with opened covers. (The engine must be turnable by turning gear for this inspection.)

9.2.9.4 For V-engines, the cylinder units are to be selected from both cylinder banks and different crank throws.

9.2.9.5 If deemed necessary by the surveyor, further dismantling of the engine may be required.

9.3 Program for trials of internal combustion engines to assess operational capability (IACS M51 Cor.1 (2018))

9.3.1 Safety precautions

9.3.1.1 Before any test run is carried out, all relevant equipment for the safety of attending personnel is to be made available by the manufacturer / shipyard and is to be operational.

9.3.1.2 This applies especially to crankcase explosive conditions protection, but also to overspeed protection and any other shut down function.

9.3.1.3 The overspeed protective device is to be set to a value, which is not higher than the overspeed value that was demonstrated during the type test for that engine. This set point shall be verified by the surveyor.

9.3.2 General

9.3.2.1 Before any official testing, the engines shall be run-in as prescribed by the engine manufacturer.

9.3.2.2 Adequate test bed facilities for loads as required in 9.3.5 shall be provided. All fluids used for testing purposes such as fuel, lubrication oil and cooling water are to be suitable for the purpose intended, e.g. they are to be clean, preheated if necessary and cause no harm to engine parts. This applies to all fluids used temporarily or repeatedly for testing purposes only.

9.3.2.3 The testing consists of workshop and shipboard (quay and sea trial) testing.

9.3.2.4 Engines are to be inspected for:

- Jacketing of high-pressure fuel oil lines including the system used for the detection of leakage.
- Screening of pipe connections in piping containing flammable liquids.
- Insulation of hot surfaces by taking random temperature readings that are to be compared with corresponding readings obtained during the type test. This shall be done while running at the rated power of engine. Use of contact thermometers may be accepted at the discretion of the attending Surveyor. If the insulation is modified subsequently to the Type Approval Test, LHR may request temperature measurements as required by 9.2.8.9.

9.3.2.5 These inspections are normally to be made during the works trials by the manufacturer and the attending surveyor, but at the discretion of LHR parts of these inspections may be postponed to the shipboard testing.

9.3.3 Works trials (Factory acceptance test)

The purpose of the works trials is to verify design premises such as power, safety against fire, adherence to approved limits (e.g. maximum pressure), and functionality and to establish reference values or base lines for later reference in the operational phase.

9.3.4 Records

9.3.4.1 The following environmental test conditions are to be recorded:

- Ambient air temperature
- Ambient air pressure
- Atmospheric humidity

9.3.4.2 For each required load point, the following parameters are normally to be recorded:

- Power and speed
- Fuel index (or equivalent reading)
- Maximum combustion pressures (only when the cylinder heads installed are designed for such measurement).
- Exhaust gas temperature before turbine and from each cylinder (to the extent that monitoring is required in 9.7 of this Chapter and Part 8, Chapter 1, Sections 3.1 and 4.1)
- Charge air temperature
- Charge air pressure
- Turbocharger speed (to the extent that monitoring is required in 9.7 of this Chapter)

9.3.4.3 Calibration records for the instrumentation are, upon request, to be presented to the attending Surveyor.

9.3.4.4 For all stages at which the engine is to be tested, the pertaining operational values 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 engine manufacturer. This also includes crankshaft deflections if considered necessary by the engine designer.

9.3.4.5 In each case, all measurements conducted at the various load points are to be carried out at steady state operating conditions. However, for all load points provision should be made for time needed by the Surveyor to carry out visual inspections. The readings for MCR, i.e. 100% power (rated maximum continuous power at corresponding rpm) are to be taken at least twice at an interval of normally 30 minutes.

9.3.5 Test loads

9.3.5.1 Test loads for various engine applications are given below. In addition, the scope of the trials may be expanded depending on the engine application, service experience, or other relevant reasons.

Note: Alternatives to the detailed tests may be agreed between the manufacturer and LHR when the overall scope of tests is found to be equivalent.

9.3.5.2 Propulsion engines driving propeller or impeller only.

a. 100% power (MCR) at corresponding speed n_0 : at least 60 min.

b. 110% power at engine speed 1,032n_0 : Records to be taken after 15 minutes or after steady conditions have been reached, whichever is shorter.

Note: Only required once for each different engine/turbocharger configuration.

c. Approved intermittent overload (if applicable) : testing for duration as agreed with the manufacturer.

d. 90% (or normal continuous cruise power), 75%, 50% and 25% power in accordance with the nominal propeller curve, the sequence to be selected by the engine manufacturer.

e. Reversing manoeuvres (if applicable).

Note: After running on the test bed, the fuel delivery system is to be so adjusted that overload power cannot be given in service, unless intermittent overload power is approved by LHR. In that case, the fuel delivery system is to be blocked to that power.

9.3.5.3 Engines driving generators for electric propulsion.

a. 100% power (MCR) at corresponding speed n_0 : at least 60 min.

b. 110% power at engine speed n_0 : 15 min- after having reached steady conditions.

c. Governor tests for compliance with Chapter 2, SECTION 5, 5.3.1 and 5.3.2 of this Part are to be carried out.

d. 75%, 50% and 25% power and idle, the sequence to be selected by the engine manufacturer.

Note: After running on the test bed, the fuel delivery system is to be adjusted so that full power plus a 10% margin for transient regulation can be given in service after installation onboard. The transient overload capability is required so that the required transient governing characteristics are achieved also at 100% loading of the engine, and also so that the protection system utilised in the electric distribution system can be activated before the engine stalls.

9.3.5.4 Engines driving generators for auxiliary purposes.

Tests to be performed as in 9.3.5.3.

9.3.5.5 Propulsion engines also driving power take off (PTO) generator.

a. 100% power (MCR) at corresponding speed n_0 : at least 60 min.

b. 110% power at engine speed n_0 : 15 min. - after having reached steady conditions.

c. Approved intermittent overload (if applicable) : testing for duration as agreed with the manufacturer.

d. 90% (or normal continuous cruise power), 75%, 50% and 25% power in accordance with the nominal propeller curve or at constant speed n_0, the sequence to be selected by the engine manufacturer.

Note: After running on the test bed, the fuel delivery system is to be adjusted so that full power plus a margin for transient regulation can be given in service after installation onboard. The transient overload capability is required so that the electrical protection of downstream system components is activated before the engine stalls. This margin may be 10% of the engine power but at least 10% of the PTO power.

9.3.5.6 Engines driving auxiliaries.

a. 100% power (MCR) at corresponding speed n_0 : at least 30 min.

b. 110% power at engine speed n_0 : 15 min. - after having reached steady conditions.

c. Approved intermittent overload (if applicable) : testing for duration as agreed with the manufacturer.

d. For variable speed engines, 75%, 50% and 25% power in accordance with the nominal power consumption curve, the sequence to be selected by the engine manufacturer.

Note: After running on the test bed, the fuel delivery system is normally to be so adjusted that overload power cannot be delivered in service, unless intermittent overload power is approved. In that case, the fuel delivery system is to be blocked to that power.

9.3.6 Turbocharger matching with engine

9.3.6.1 Compressor chart

Turbochargers shall have a compressor characteristic that allows the engine, for which it is intended, to operate without surging during all operating conditions and also after extended periods in operation.

For abnormal, but permissible, operation conditions, such as misfiring and sudden load reduction, no continuous surging shall occur. In this section, surging and continuous surging are defined as follows:

Surging means the phenomenon, which results in a high pitch vibration of an audible level or explosion-like noise from the scavenger area of the engine.

Continuous surging means that surging happens repeatedly and not only once.

9.3.6.2 Surge margin verification

Category C turbochargers used on propulsion engines are to be checked for surge margins during the engine workshop testing as specified below. These tests may be waived if successfully tested earlier on an identical configuration of engine and turbocharger (including same nozzle rings).

For 4-stroke engines:

The following shall be performed without indication of surging:

- With maximum continuous power and speed (=100%), the speed shall be reduced with constant torque (fuel index) down to 90% power.
- With 50% power at 80% speed (= propeller characteristic for fixed pitch), the speed shall be reduced to 72% while keeping constant torque (fuel index).

For 2-stroke engines:

The surge margin shall be demonstrated by at least one of the following methods:

- The engine working characteristic established at workshop testing of the engine shall be plotted into the compressor chart of the turbocharger (established in a test rig). There shall be at least 10% surge margin in the full load range, i.e. working flow shall be 10% above the theoretical (mass) flow at surge limit (at no pressure fluctuations).
- Sudden fuel cut-off to at least one cylinder shall not result in continuous surging and the turbocharger shall be stabilised at the new load within 20 seconds. For applications with more than one turbocharger the fuel shall be cut-off to the cylinders closest upstream to each turbocharger.

This test shall be performed at two different engine loads:

- The maximum power permitted for one cylinder misfiring.
- The engine load corresponding to a charge air pressure of about 0.6 bar (but without auxiliary blowers running).

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

9.3.7 Integration tests

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 for all intended operational modes and the tests considered as a system are to be carried out at the works. If such tests are technically unfeasible at the works, however, these tests may be conducted during sea trial. The scope of these tests is to be agreed with LHR for selected cases based on the FMEA required in Part 5, Chapter 2, Section 1.2.

9.3.8 Component inspections

Random checks of components to be presented for inspection after works trials are left to the discretion of LHR.

9.3.9 Shipboard trials

9.3.9.1 Objectives

The purpose of the shipboard testing is to verify compatibility with power transmission and driven machinery in the system, control systems and auxiliary systems necessary for the engine and integration of engine / shipboard control systems, as well as other items that had not been dealt with in the FAT (Factory Acceptance Testing).

9.3.9.2 Starting capacity

Starting manoeuvres are to be carried out in order to verify that the capacity of the starting media satisfies the required number of start attempts.

9.3.9.3 Monitoring and alarm system

The monitoring and alarm systems are to be checked to the full extent for all engines, except items already verified during the works trials.

9.3.9.4 Test loads

1. Test loads for various engine applications are given below. In addition, the scope of the trials may be expanded depending on the engine application, service experience, or other relevant reasons.
2. The suitability of the engine to operate on fuels intended for use is to be demonstrated.

Note: Tests other than those listed below may be required by statutory instruments (e.g. EEDI verification).

3. Propulsion engines driving fixed pitch propeller or impeller.
   a. At rated engine speed n_0 : at least 4 hours.
   b. At engine speed $1.032n_0$
      (if engine adjustment permits, see 9.3.5.1) : 30 min.
   c. At approved intermittent overload (if applicable) : testing for duration as agreed with the manufacturer.
   d. Minimum engine speed to be determined.
   e. The ability of reversible engines to be operated in reverse direction is to be demonstrated.

Note: During stopping tests according to Resolution MSC.137 (76), see 9.3.10.1 for additional requirements in the case of a barred speed range.

4. Propulsion engines driving controllable pitch propellers.
   a. At rated engine speed n_0 with a propeller pitch leading to rated engine power (or to the maximum achievable power if 100% cannot be reached) : at least 4 hours.
   b. At approved intermittent overload (if applicable) : testing for duration as agreed with the manufacturer.
   c. With reverse pitch suitable for manoeuvring, see 9.3.10.1 for additional requirements in the case of a barred speed range.

5. Engine(s) driving generator(s) for electrical propulsion and/or main power supply.

a. At 100% power (rated electrical power of generator) : at least 60 min.
b. At 110% power (rated electrical power of generator) : at least 10 min.

Note: Each engine is to be tested 100% electrical power for at least 60 min and 110% of rated electrical power of the generator for at least 10 min. This may, if possible, be done during the electrical propulsion plant test, which is required to be tested with 100% propulsion power (i.e. total electric motor capacity for propulsion) by distributing the power on as few generators as possible. The duration of this test is to be sufficient to reach stable operating temperatures of all rotating machines or for at least 4 hours. When some of the gen. set(s) cannot be tested due to insufficient time during the propulsion system test mentioned above, those required tests are to be carried out separately.

c. Demonstration of the generator prime movers' and governors' ability to handle load steps as described in Chapter 2, SECTION 5, 5.3.2.

6. Propulsion engines also driving power take off (PTO) generator.

a. 100% engine power (MCR) at corresponding speed n_0 : at least 4 hours.
b. 100% propeller branch power at engine speed n_0 (unless already covered in a) : 2 hours.
c. 100% PTO branch power at engine speed n_0 : at least 1 hour.

7. Engines driving auxiliaries

a. 100% power (MCR) at corresponding speed n_0 : at least 30 min.
b. Approved intermittent overload : testing for duration as approved.

9.3.10 Torsional vibrations

9.3.10.1 Barred speed range

Where a barred speed range (bsr) is required, passages through this bsr, both accelerating and decelerating, are to be demonstrated. The times taken are to be recorded and are to be equal to or below those times stipulated in the approved documentation, if any. This also includes when passing through the bsr in reverse rotational direction, especially during the stopping test.

Note: Applies both for manual and automatic passing-through systems.

The ship's draft and speed during all these demonstrations is to be recorded. In the case of a controllable pitch propeller, the pitch is also to be recorded.

The engine is to be checked for stable running (steady fuel index) at both upper and lower borders of the bsr. Steady fuel index means an oscillation range less than 5% of the effective stroke (idle to full index).

9.4 Type Testing Procedure for Crankcase Explosion Relief Valves (IACS UR M66 Cor.1 (2021))

9.4.1 Scope

To specify type tests and identify standard test conditions using methane gas and air mixture to demonstrate that the LHR's requirements are satisfied for crankcase explosion relief valves intended to be fitted to engines and gear cases.

This test procedure is only applicable to explosion relief valves fitted with flame arresters.

NOTE: Where internal oil wetting of a flame arrester is a design feature of an explosion relief valve, alternative testing arrangements that demonstrate compliance with 9.4 may be proposed by the manufacturer. The alternative testing arrangements are to be agreed by LHR.

9.4.2 Recognized Standards

(a) ISO 16852:2016 as amended
(b) ISO/IEC 17025:2017 as amended
(c) ISO 12100:2010 as amended
(d) VDI 3673-1:2002
(e) IMO MSC/Circ.677 as amended by MSC/Circ.1009 and MSC.1/Circ.1324

NOTES:

1. Engines are to be fitted with components and arrangements complying with 9.4 when:
   (i) the engine is installed on existing ships (i.e. ships for which the date of contract for construction is before 1 July 2008) and the date of application for certification of the engine (i.e. the date of whatever document LHR requires/accepts as an application or request for certification of an individual engine) is on or after 1 July 2008; or
   (ii) the engine is installed on new ships (i.e. ships for which the date of contract for construction is on or after 1 July 2008).

2. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contract for construction”, refer to IACS Procedural Requirement (PR) No. 29.

9.4.3 Purpose

The purpose of type testing crankcase explosion relief valves is fourfold:

(a) To verify the effectiveness of the flame arrester.
(b) To verify that the valve closes after an explosion.
(c) To verify that the valve is gas/air tight after an explosion.
(d) To establish the level of over pressure protection provided by the valve.

9.4.4 Test facilities

Test houses for carrying out type testing of crankcase explosion relief valves are to meet the following requirements:

(a) The test houses where testing is carried out are to be accredited to a National or International Standard, e.g. ISO/IEC 17025:2017 as amended, and are to be acceptable to LHR.
(b) The test facilities are to be equipped so that they can perform and record explosion testing in accordance with this procedure.
(c) The test facilities are to have equipment for controlling and measuring a methane gas in air concentration within a test vessel to an accuracy of ± 0.1%.
(d) The test facilities are to be capable of effective point-located ignition of methane gas in air mixture.
(e) The pressure measuring equipment is to be capable of measuring the pressure in the test vessel in at least two positions, one at the valve and the other at the test vessel centre. The measuring arrangements are to be capable of measuring and recording the pressure changes throughout an explosion test at a frequency recognizing the speed of events during an explosion. The result of each test is to be documented by video recording and by recording with a heat sensitive camera.

(f) The test vessel for explosion testing is to have documented dimensions. The dimensions are to be such that the vessel is not "pipe like" with the distance between dished ends being not more than 2.5 times its diameter. The internal volume of the test vessel is to include any standpipe arrangements.
(g) The test vessel is to be provided with a flange, located centrally at one end perpendicular to the vessel longitudinal axis, for mounting the explosion relief valve. The test vessel is to be arranged in an orientation consistent with how the valve will be installed in service, i.e., in the vertical plane or the horizontal plane.
(h) A circular plate is to be provided for fitting between the pressure vessel flange and valve to be tested with the following dimensions:
 – Outside diameter of 2 times the outer diameter of the valve top cover.
 – Internal bore having the same internal diameter as the valve to be tested.
(i) The test vessel is to have connections for measuring the methane in air mixture at the top and bottom.
(j) The test vessel is to be provided with a means of fitting an ignition source at a position specified in 9.8.5(c).
(k) The test vessel volume is to be as far as practicable, related to the size and the capability of the relief valve to be tested. In general, the volume is to correspond to the requirement in 5.4.3, of this Chapter, for the free area of explosion relief valve to be not less than 115cm²/m³ of crankcase gross volume.

NOTES:

1. This means that the testing of a valve having 1150cm² of free area would require a test vessel with a volume of 10m³.
2. Where the free area of relief valves is greater than 115 cm²/m³ of the crankcase gross volume, the volume of the test vessel is to be consistent with the design ratio.
3. In no case is the volume of the test vessel to vary by more than ±15% to -15% from the design cm²/m³ volume ratio.

9.4.5 Explosion test process

(a) All explosion tests to verify the functionality of crankcase explosion relief valves are to be carried out using an air and methane mixture with a volumetric methane concentration of 9.5% ±0.5%. The pressure in the test vessel is to be not less than atmospheric and is not to exceed the opening pressure of the relief valve.
(b) The concentration of methane in the test vessel is to be measured at the top and bottom of the vessel and these concentrations are not to differ by more than 0.5%.
(c) The ignition of the methane and air mixture is to be made at the centreline of the test vessel at a position approximately one third of the height or length of the test vessel opposite to where the valve is mounted.
(d) The ignition is to be made using a 100 joule explosive charge.

9.4.6 Valves to be tested

(a) The valves used for type testing (including testing specified in 9.4.6(c) are to be selected from the manufacturer's normal usual production line for such valves by LHR witnessing the tests.
(b) For approval of a specific valve size, three valves are to be tested in accordance with 9.4.6(c) and 9.4.7. For a series of valves 9.4.9 refers.
(c) The valves selected for type testing are to have been previously tested at the manufacturer's works to demonstrate that the opening pressure is in accordance with the specification within a tolerance of ± 20% and that the valve is air tight at a pressure below the opening pressure for at least 30 seconds.

NOTE: This test is to verify that the valve is air tight following assembly at the manufacturer's works and that the valve begins to open at the required pressure demonstrating that the correct spring has been fitted.

(d) The type testing of valves is to recognize the orientation in which they are intended to be installed on the engine or gear case. Three valves of each size are to be tested for each intended installation orientation, i.e. in the vertical and/or horizontal positions.

9.4.7 Method

(a) The following requirements are to be satisfied at explosion testing:
   (i) The explosion testing is to be witnessed by a LHR's surveyor.
   (ii) Where valves are to be installed on an engine or gear case with shielding arrangements to deflect the emission of explosion combustion products, the valves are to be tested with the shielding arrangements fitted.
   (iii) Successive explosion testing to establish a valve's functionality is to be carried out as quickly as possible during stable weather conditions.
   (iv) The pressure rise and decay during all explosion testing is to be recorded.
   (v) The external condition of the valves is to be monitored during each test for indication of any flame release by video and hot heat sensitive camera.

(b) The explosion testing is to be in three stages for each valve that is required to be approved as being type tested.

Stage 1:

Two explosion tests are to be carried out in the test vessel with the circular plate described in 9.4.4(h) fitted and the opening in the plate covered by a 0.05mm thick polythene film. These tests establish a reference pressure level for determination of the capability of a relief valve in terms of pressure rise in the test vessel, see 9.4.8(f).

Stage 2:

(i) Two explosion tests are to be carried out on three different valves of the same size. Each valve is to be mounted in the orientation for which approval is sought i.e., in the vertical or horizontal position with the circular plate described in 9.4.4(h) located between the valve and pressure vessel mounting flange.

(ii) The first of the two tests on each valve is to be carried out with a 0.05mm thick polythene bag having a minimum diameter of three times the diameter of the circular plate and volume not less than 30% of the test vessel, enclosing the valve and circular plate. Before carrying out the explosion test the polythene bag is to be empty of air. The polythene bag is required to provide a readily visible means of assessing whether there is flame transmission through the relief valve following an explosion consistent with the requirements of the standards identified in 9.4.2.

NOTE:

During the test, the explosion pressure will open the valve and some unburned methane/air mixture will be collected in the polythene bag. When the flame reaches the flame arrester and if there is flame transmission through the flame arrester, the methane/air mixture in the bag will be ignited and this will be visible.

(iii) Provided that the first explosion test successfully demonstrated that there was no indication of combustion outside the flame arrester and there are no visible signs of damage to the flame arrester or valve, a second explosion test without the polythene bag arrangement is to be carried out as quickly as possible after the first test. During the second explosion test, the valve is to be visually monitored for any indication of combustion outside the flame arrester and video records are to be kept for subsequent analysis. The second test is required to demonstrate that the valve can function in the event of a secondary crankcase explosion.

(iv) After each explosion, the test vessel is to be maintained in the closed condition for at least 10 seconds to enable the tightness of the valve to be ascertained. The tightness of the valve can be verified during the test from the pressure/time records or by a separate test after completing the second explosion test.

Stage 3:

Carry out two further explosion tests as described in Stage 1. These further tests are required to provide an average baseline value for assessment of pressure rise, recognizing that the test vessel ambient conditions may have changed during the testing of the explosion relief valves in Stage 2.

9.4.8 Assessment and records

For the purposes of verifying compliance with the requirements of 9.4, the assessment and records of the valves used for explosion testing is to address the following:

(a) The valves to be tested are to have evidence of design appraisal/approval by LHR witnessing tests.
(b) The designation, dimensions and characteristics of the valves to be tested are to be recorded. This is to include the free area of the valve and of the flame arrester and the amount of valve lift at 0.2bar.
(c) The test vessel volume is to be determined and recorded.
(d) For acceptance of the functioning of the flame arrester there is to be no indication of flame or combustion outside the valve during an explosion test. This should be confirmed by the test laboratory taking into account measurements from the heat sensitive camera.
(e) The pressure rise and decay during an explosion is to be recorded with indication of the pressure variation showing the maximum overpressure and steady under-pressure in the test vessel during testing. The pressure variation is to be recorded at two points in the pressure vessel.
(f) The effect of an explosion relief valve in terms of pressure rise following an explosion is ascertained from maximum pressures recorded at the centre of the test vessel during the three stages. The pressure rise within the test vessel due to the installation of a relief valve is the difference between average pressure of the four explosions from Stages 1 and 3 and the average of the first tests on the three valves in Stage 2. The pressure rise is not to exceed the limit specified by the manufacturer.
(g) The valve tightness is to be ascertained by verifying from the records at the time of testing that an under-pressure of at least 0.3bar is held by the test vessel for at least 10 seconds following an explosion. This test is to verify that the valve has effectively closed and is reasonably gas-tight following dynamic operation during an explosion.
(h) After each explosion test in Stage 2, the external condition of the flame arrester is to be examined for signs of serious damage and/or deformation that may affect the operation of the valve.
(i) After completing the explosion tests, the valves are to be dismantled and the condition of all components ascertained and documented. In particular, any indication of valve sticking or uneven opening that may affect operation of the valve is to be noted. Photographic records of the valve condition are to be taken and included in the report.

9.4.9 Design series qualification

(i) The qualification of quenching devices to prevent the passage of flame can be evaluated for other similar devices of identical type where one device has been tested and found satisfactory.

(ii) The quenching ability of a flame arrester depends on the total mass of quenching lamellas/mesh. Provided the materials, thickness of materials, depth of lamellas/thickness of mesh layer and the quenching gaps are the same, then the same quenching ability can be qualified for different sizes of flame arresters subject to (a) and (b) being satisfied.

(a) $\frac{n_1}{n_2} = \sqrt{\frac{S_1}{S_2}}$

(b) $\frac{A_1}{A_2} = \frac{S_1}{S_2}$

where:

n_1 = total depth of flame arrester corresponding to the number of lamellas of size 1 quenching device for a valve with a relief area equal to S_1

n_2 = total depth of flame arrester corresponding to the number of lamellas of size 2 quenching device for a valve with a relief area equal to S_2

A_1 = free area of quenching device for a valve with a relief area equal to S_1

A_2 = free area of quenching device for a valve with a relief area equal to S_2

(iii) The qualification of explosion relief valves of larger sizes than that which has been previously satisfactorily tested in accordance with 9.4.7 and 9.4.8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:
   (a) The free area of a larger valve does not exceed three times + 5% that of the valve that has been satisfactorily tested.
   (b) One valve of the largest size, subject to (a), requiring qualification is subject to satisfactory testing required by 9.4.6(c) and Stage 2 except that a single valve will be accepted in Stage 2 (i) and the volume of the test vessel is not to be less than ⅓ of the volume required by 9.4.4(k)
   (c) The assessment and records are to be in accordance with 9.4.8 noting that 9.4.8(f) will only be applicable to Stage 2 for a single valve.

(iv) The qualification of explosion relief valves of smaller sizes than that which has been previously satisfactorily tested in accordance with 9.4.7 and 9.4.8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:
   (a) The free area of a smaller valve is not less than one third of the valve that has been satisfactorily tested.
   (b) One valve of the smaller size, subject to (a), requiring qualification is subject to satisfactory testing required by 9.4.6(c) and Stage 2 except that a single valve will be accepted in Stage 2, (i) and the volume of the test vessel is not to be more than the volume required by 9.4.4(k).
   (c) The assessment and records are to be in accordance with 9.4.8 noting that 9.4.8(f) will only be applicable to Stage 2 for a single valve.

9.4.10 The Report

The test facility is to deliver a full report that includes the following information and documents:

(i) Test specification.
(ii) Details of test pressure vessel and valves tested.
(iii) The orientation in which the valve was tested (vertical or horizontal position).
(iv) Methane in air concentration for each test.
(v) Ignition source
(vi) Pressure curves for each test.

(vii) Video recordings of each valve test.
(viii) The assessment and records stated in 9.4.8.

9.8.11 Approval

The approval of an explosion relief valve is at the discretion of LHR based on the appraisal plans and particulars and the test facility's report of the results of type testing.

9.5 Type Testing Procedure for Crankcase Oil Mist Detection and Alarm Equipment (IACS UR M67 Rev.2 (2015))[4]

9.5.1 Scope

To specify the tests required to demonstrate that crankcase oil mist detection and alarm equipment intended to be fitted to diesel engines satisfy the LHR's requirements.

This test procedure is also applicable to oil mist detection and alarm equipment intended for gear cases.

9.5.2 Recognized Standards

See Part 8, Chapter 1, SECTION 7, 7.2 Test Specification for Type Approval

9.5.3 Purpose

The purpose of type testing crankcase oil mist detection and alarm equipment is seven fold:

(a) To verify the functionality of the system.
(b) To verify the effectiveness of the oil mist detectors.
(c) To verify the accuracy of oil mist detectors.
(d) To verify the alarm set points.
(e) To verify time delays between oil mist leaving the source and alarm activation.
(f) To verify functional failure detection.
(g) To verify the influence of optical obscuration on detection.

9.5.4 Test facilities

Test houses carrying out type testing of crankcase oil mist detection and alarm equipment are to satisfy the following criteria:

(i) A full range of facilities for carrying out the environmental and functionality tests required by this procedure shall be available and be acceptable to LHR.

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[4] NOTE:

1. Engines are to be fitted with crankcase oil mist detection and alarm equipment complying with 9.9 when:
   (i) an application for certification of an engine is dated on/after 1 January 2007; or
   (ii) installed in new ships for which the date of contract for construction is on or after 1 January 2007.

2. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of "contract for construction", refer to IACS Procedural Requirement (PR) No. 29.

(ii) The test house that verifies the functionality of the equipment is to be equipped so that it can control, measure and record oil mist concentration levels in terms of mg/l to an accuracy of ±10% in accordance with this procedure.

(iii) When verifying the functionality, test houses are to consider the possible hazards associated with the generation of the oil mist required and take adequate precautions. LHR will accept the use of low toxicity, low hazard oils as used in other applications, provided it is demonstrated to have similar properties to SAE 40 monograde mineral oil specified.

9.5.5 Equipment testing

The range of tests is to include the following:

i. For the alarm/monitoring panel:
   (a) Functional tests described in 9.5.6.
   (b) Electrical power supply failure test.
   (c) Power supply variation test.
   (d) Dry heat test.
   (e) Damp heat test.
   (f) Vibration test.
   (g) EMC test.
   (h) Insulation resistance test.
   (i) High voltage test.
   (j) Static and dynamic inclinations, if moving parts are contained.

ii. For the detectors:
   (a) Functional tests described in 9.5.6.
   (b) Electrical power supply failure test.
   (c) Power supply variation test.
   (d) Dry heat test.
   (e) Damp heat test.
   (f) Vibration test.
   (g) EMC test, where susceptible.
   (h) Insulation resistance test.
   (i) High voltage test.
   (j) Static and dynamic inclinations.

9.5.6 Functional tests

9.5.6.1 All tests to verify the functionality of crankcase oil mist detection and alarm equipment are to be carried out in accordance with 9.5.6.2 to 9.5.6.6 with an oil mist concentration in air, known in terms of mg/l to an accuracy of ±10%.

9.5.6.2 The concentration of oil mist in the test chamber is to be measured in the top and bottom of the chamber and these concentrations are not to differ by more than 10%.

9.5.6.3 The oil mist detector monitoring arrangements are to be capable of detecting oil mist in air concentrations of between

a. 0 and 10% of the Lower Explosive Limit (LEL) or
b. between 0 and a percentage of weight of oil in air determined by the Manufacturer based on the sensor measurement method (e.g. obscuration or light scattering) that is acceptable to LHR taking into account the alarm level specified in 9.5.6.4..

NOTE: The LEL corresponds to an oil mist concentration of approximately 50mg/l (≈4.1% weight of oil in air mixture).

9.5.6.4 The alarm set point for oil mist concentration in air is to provide an alarm at a maximum level corresponding to not more than 5% of the LEL or approximately 2.5mg/l.

9.5.6.5 Where alarm set points can be altered, the means of adjustment and indication of set points are to be verified against the equipment manufacturer's instructions.

9.5.6.6 The performance of the oil mist detector in mg/l is to be demonstrated. This is to include the following:

- range (oil mist detector)
- resolution (oil mist detector)
- sensitivity (oil mist detector)

Note: Sensitivity of a measuring system: quotient of the change in an indication of a measuring system and the corresponding change in a value of a quantity being measured. Resolution: smallest change in a quantity being measured that causes a perceptible change in the corresponding indication.

9.5.6.7 Where oil mist is drawn into a detector via piping arrangements, the time delay between the sample leaving the crankcase and operation of the alarm is to be determined for the longest and shortest lengths of pipes recommended by the manufacturer. The pipe arrangements are to be in accordance with the manufacturer's instructions/recommendations. Piping is to be arranged to prevent pooling of oil condensate which may cause a blockage of the sampling pipe over time.

9.5.6.8 It is to be demonstrated that the openings of detector equipment does not become occluded or blocked under continuous splash and spray of engine lubricating oil, as may occur in the crankcase atmosphere. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by LHR. The temperature, quantity and angle of impact of the oil to be used is to be declared and their selection justified by the manufacturer.

9.5.6.9 Detector equipment may be exposed to water vapor from the crankcase atmosphere which may affect the sensitivity of the equipment and it is to be demonstrated that exposure to such conditions will not affect the functional operation of the detector equipment. Where exposure to water vapor and/or water condensation has been identified as a possible source of equipment malfunctioning, testing is to demonstrate that any mitigating arrangements such as heating are effective. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by LHR.

NOTE:
This testing is in addition to that required by 9.9.5(ii)(e) and is concerned with the effects of condensation caused by the detection equipment being at a lower temperature than the crankcase atmosphere.

9.5.6.10 It is to be demonstrated that an indication is given where 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 as required by 5.5.16 of this Chapter.

9.5.7 Detectors and alarm equipment to be tested

9.5.7.1 The detectors and alarm equipment selected for the type testing are to be selected from the manufacturer's normal production line by LHR witnessing the tests.

9.5.7.2 Two detectors are to be tested. One is to be tested in clean condition and the other in a condition representing the maximum level of lens obscuration specified by the manufacturer

9.5.8 Method

The following requirements are to be satisfied at type testing:

(a) Oil mist generation is to satisfy (i) to (v).

- (i) The ambient temperature in and around the test chamber is to be at the standard atmospheric conditions defined in Part 8, Chapter 1, Section 7.2 before any test run is started.
- (ii) Oil mist is to be generated with suitable equipment using an SAE 40 monograde mineral oil or equivalent and supplied to a test chamber. The selection of the oil to be used is to take into consideration risks to health and safety, and the appropriate controls implemented. A low toxicity, low flammability oil of similar viscosity may be used as an alternative. The oil mist produced is to have an average (or arithmetic mean) droplet size not exceeding 5 μm. The oil droplet size is to be checked using the sedimentation method or an equivalent method to a relevant international or national standard. If the sedimentation method is chosen, the test chamber is to have a minimum height of 1m and volume of not less than $1m^3$.
  NOTE: The calculated oil droplet size using the sedimentation method represents the average droplet size.
- (iii) The oil mist concentrations used are to be ascertained by the gravimetric deterministic method or equivalent. Where an alternative technique is used its equivalence is to be demonstrated.
  NOTE:
  For this test, the gravimetric deterministic method is a process where the difference in weight of a 0,8 μm pore size membrane filter is ascertained from weighing the filter before and after drawing 1 litre of oil mist through the filter from the oil mist test chamber. The oil mist chamber is to be fitted with a recirculating fan.
- (iv) Samples of oil mist are to be taken at regular intervals and the results plotted against the oil mist detector output. The oil mist detector is to be located adjacent to where the oil mist samples are drawn off.
- (v) The results of a gravimetric analysis are considered invalid and are to be rejected if the resultant calibration curve has an increasing gradient with respect to the oil mist detection reading. This situation occurs when insufficient time has been allowed for the oil mist to become homogeneous. Single results that are more than 10% below the calibration curve are to be rejected. This situation occurs when the integrity of the filter unit has been compromised and not all of the oil is collected on the filter paper.
- (vi) The filters require to be weighed to a precision of 0,1mg and the volume of air/oil mist sampled to 10ml.

(b) The type approval by LHR, the testing is to be witnessed by personnel authorised by LHR.

(c) Oil mist detection equipment is to be tested in the orientation (vertical, horizontal or inclined) in which it is intended to be installed on an engine or gear case as specified by the equipment manufacturer.

(d) Type testing is to be carried out for each type of oil mist detection and alarm equipment for which a manufacturer seeks classification approval. Where sensitivity levels can be adjusted, testing is to be carried out at the extreme and mid-point level settings.

9.5.9 Assessment

Assessment of oil mist detection equipment after testing is to address the following points:

(i) The equipment to be tested is to have evidence of design appraisal/approval by LHR witnessing tests.

(ii) Details of the detection equipment to be tested are to be recorded and are to include:
- name of manufacturer
- type designation
- oil mist concentration assessment capability and alarm settings
- the maximum percentage level of lens obscuration used in 9.5.7.2

(iii) After completing the tests, the detection equipment is to be examined and the condition of all components ascertained and documented. Photographic records of the monitoring equipment condition are to be taken and included in the report.

9.5.10 Design series qualification

The approval of one type of detection equipment may be used to qualify other devices having identical construction details. Proposals are to be submitted for consideration.

9.5.11 The Report

The test house is to provide a full report which includes the following information and documents:

(a) Test specification.

(b) Details of equipment tested.

(c) Results of tests.
To include a declaration by the manufacturer of the oil mist detector of its:
- Performance, in mg/L
- Accuracy, of oil mist concentration in air
- Precision, of oil mist concentration in air
- Range, of oil mist detector
- Resolution, of oil mist detector
- Response time, of oil mist detector
- Sensitivity, of oil mist detector
- Obscuration of sensor detection, declared as percentage of obscuration. 0% totally clean, 100% totally obscure
- Detector failure alarm

9.5.12 Acceptance

9.9.12.1 Acceptance of crankcase oil mist detection equipment is at the discretion of LHR based on the appraisal plans and particulars and the test house report of the results of type testing.

9.5.12.2 The following information is to be submitted to LHR for acceptance of oil mist detection equipment and alarm arrangements:

(a) Description of oil mist detection equipment and system including alarms.

(b) Copy of the test house report identified in 9.5.11.

(c) Schematic layout of engine oil mist detection equipment arrangements showing location of detectors/sensors and piping arrangements and dimensions.

(d) Maintenance and test manual which is to include the following information:
  (i) Intended use of equipment and its operation.
  (ii) Functionality tests to demonstrate that the equipment is operational and that any faults can be identified and corrective actions notified.

(iii) Maintenance routines and spare parts recommendations.
(iv) Limit setting and instructions for safe limit levels.
(v) Where necessary, details of configurations in which the equipment is and is not to be used.

9.6 Certification of Engine Components (IACS UR M72 Rev.2 (2019))

9.6.1 General

9.6.1.1 The engine manufacturer is to have a quality control system that is suitable for the actual engine types to be certified by LHR. The quality control system is also to apply to any sub-suppliers. LHR reserves the right to review the system or parts thereof. Materials and components are to be produced in compliance with all the applicable production and quality instructions specified by the engine manufacturer. LHR requires that certain parts are verified and documented by means of LHR Certificate (LHRC), Work Certificate (W) or Test Report (TR).

9.6.1.2 LHR Certificate (LHRC)

This is a document issued by LHR stating:

- conformity with Rule requirements
- that the tests and inspections have been carried out on:
  - the finished certified component itself; or
  - on samples taken from earlier stages in the production of the component, when applicable.
- that the inspection and tests were performed in the presence of the Surveyor or in accordance with special agreements, i.e. Alternative Certification Scheme (ACS).

9.6.1.3 Work’s Certificate (W)

This is a document signed by the manufacturer stating:

- conformity with requirements
- that the tests and inspections have been carried out on:
  - the finished certified product component itself; or
  - on samples taken from earlier stages in the production of the component, when applicable
- that the tests were witnessed and signed by a qualified representative of the applicable department of the manufacturer.

A Work’s Certificate may be considered equivalent to LHR Certificate and endorsed by LHR under the following cases if:

- the test was witnessed by the LHR Surveyor; or
- an ACS agreement is in place between the LHR and the manufacturer or material supplier; or
- the Work’s certificate is supported by tests carried out by an accredited third party that is accepted by LHR and independent from the manufacturer and/or material supplier.

9.6.1.4 Test Report (TR)

This is a document signed by the manufacturer stating:

- conformity with requirements.
- that the tests and inspections have been carried out on samples from the current Production batch.

9.6.1.5 The documents above are used for product documentation as well as for documentation of single inspections such as crack detection, dimensional check, etc. If agreed to by LHR, the documentation of single tests and inspections may also be arranged by filling in results on a control sheet following the component through the production.

9.6.1.6 The Surveyor is to review the TR and W for compliance with the agreed or approved specifications. LHRC means that the Surveyor also witnesses the testing, batch or individual, unless an ACS provides other arrangements.

9.6.1.7 The manufacturer is not exempted from responsibility for any relevant tests and inspections of those parts for which documentation is not explicitly requested by LHR. The manufacturing process and equipment is to be set up and maintained in such a way that all materials and components can be consistently produced to the required standard. This includes production and assembly lines, machining units, special tools and devices, assembly and testing rigs as well as all lifting and transportation devices

9.6.2 Parts to be documented

9.6.2.1 The extent of parts to be documented depends on the type of engine, engine size and criticality of the part.

9.6.2.2 Symbols used are listed in Table 9.6.1. A summary of the required documentation for the engine components is listed in Table 9.6.2.

Table 9.6.1: Symbols used in Table 9.6.2

Symbol	Description
C	chemical composition
CD	crack detection by MPI or DP
CH	crosshead engines
D	cylinder bore diameter (mm)
GJL	gray cast iron
GJS	spheroidal graphite cast iron
GS	cast steel
M	mechanical properties
LHRC	LHR certificate
TR	test report
UT	ultrasonic testing
W	work certificate
X	visual examination of accessible surfaces by the Surveyor

9.6.2.3 For components and materials not specified in Table 9.6.2, consideration will be given by LHR upon full details being submitted and reviewed.

Table 9.6.2: Summary of required documentation for engine components

Item	Part (4), (5), (6), (7), (8)	Material properties (1)	Non-destructive examination (2)	Hydraulic testing (3)	Dimensional inspection, including surface condition	Visual inspection (surveyor)	Applicable to engines:	Component certificate
1	Welded bedplate	W(C+M)	W(UT+CD)			fit-up + post-welding	All	LHRC
2	Bearing transverse girders GS	W(C+M)	W(UT+CD)			X	All	LHRC
3	Welded frame box	W(C+M)	W(UT+CD)			fit-up + post-welding	All	LHRC
4	Cylinder block GJL			W (10)			>400 kW/cyl	
5	Cylinder block GJS			W (10)			>400 kW/cyl	
6	Welded cylinder frames	W(C+M)	W(UT+CD)			fit-up + post-welding	CH	LHRC
7	Engine block GJL			W (10)			>400 kw/cyl	
8	Engine block GJS	W(M)		W (10)			>400 kw/cyl	
9	Cylinder liner	W(C+M)		W (10)			D>300 mm	
10	Cylinder head GJL			W			D>300 mm	
11	Cylinder head GJS			W			D>300 mm	
12	Cylinder head GS	W(C+M)	W(UT+CD)	W		X	D>300 mm	LHRC
13	Forged cylinder head	W(C+M)	W(UT+CD)	W		X	D>300 mm	LHRC
14	Piston crown GS	W(C+M)	W(UT+CD)			X	D>400 mm	LHRC
15	Forged piston crown	W(C+M)	W(UT+CD)			X	D>400 mm	LHRC
16	Crankshaft: made in one piece	LHRC(C+M)	W(UT+CD)		W	Random, of fillets and oil bores	All	LHRC
17	Semi-built Crankshaft (Crankthrow, forged main journal and journals with flange)	LHRC(C+M)	W(UT+CD)		W	Random, of fillets and shrink fittings	All	LHRC
18	Exhaust gas valve cage			W			CH	

Item	Part [4], [5], [6], [7], [8]	Material properties [1]	Non-destructive examination [2]	Hydraulic testing [3]	Dimensional inspection, including surface condition	Visual inspection (surveyor)	Applicable to engines:	Component certificate
19	Piston rod	LHRC(C+M)	W(UT+CD)			Random	D>400 mm CH	LHRC
20	Cross head	LHRC(C+M)	W(UT+CD)			Random	CH	LHRC
21	Connecting rod with cap	WSC(C+M)	W(UT+CD)		W	Random, of all surfaces, in particular those shot peened	All	LHRC
22	Coupling bolts for crankshaft	LHRC(C+M)	W(UT+CD)		W	Random, of interference fit	All	LHRC
23	Bolts and studs for main bearings	W(C+M)	W(UT+CD)				D>300 mm	
24	Bolts and studs for cylinder heads	W(C+M)	W(UT+CD)				D>300 mm	
25	Bolts and studs for connecting rods	W(C+M)	W(UT+CD)		TR of thread making		D>300 mm	
26	Tie rod	W(C+M)	W(UT+CD)		TR of thread making	Random	CH	LHRC
27	High pressure fuel injection pump body	W(C+M)		W			D>300 mm	
		W(C+M)		TR			D≤300 mm	
28	High pressure fuel injection valves (only for those not autofretted)			W			D>300 mm	
				TR			D≤300 mm	
29	High pressure fuel injection pipes including common fuel rail	W(C+M)		W for those that are not autofretted			D>300 mm	
		W(C+M)		TR for those that are not autofretted			D≤300 mm	

Item	Part [4], [5], [6], [7], [8]	Material properties [1]	Non-destructive examination [2]	Hydraulic testing [3]	Dimensional inspection, including surface condition	Visual inspection (surveyor)	Applicable to engines:	Component certificate
30	High pressure common servo oil system	W(C+M)		W			D>300 mm	
		W(C+M)		TR			D≤300 mm	
31	Cooler, both sides [9]	W(C+M)		W			D>300 mm	
32	Accumulator	W(C+M)		W			All engines with accumulators with a capacity of >0,5 l	
33	Piping, pumps, actuators, etc. for hydraulic drive of valves, if applicable	W(C+M)		W			> 800 kW/cyl	
34	Engine driven pumps (oil, water, fuel, bilge) other than pumps referred to in item 27 and 33			W			> 800 kW/cyl	
35	Bearing for main, crosshead, and crankpin	TR(C)	TR (UT for full contact between base material and bearing meatal)		W		> 800 kW/cyl	

9.7 Turbochargers (IACS UR M73 Rev.1 (2022))

9.7.1 Scope

9.7.1.1 These requirements are applicable for turbochargers with regard to design approval, type testing and certification and their matching on engines. Turbochargers are to be type approved, either separately or as a part of an engine. The requirements are written for exhaust gas driven turbochargers, but apply in principle also for engine driven chargers.

9.7.1.2 The requirements escalate with the size of the turbochargers. The parameter for size is the engine power (at MCR) supplied by a group of cylinders served by the actual turbocharger, (e.g. for a V-engine with one turbocharger for each bank the size is half of the total engine power).

9.7.1.3 Turbochargers are categorised in three groups depending on served power by cylinder groups with:

- Category A: ≤ 1000 kW
- Category B: > 1000 kW and ≤ 2500 kW
- Category C: > 2500 kW

9.7.2 Documentation to be submitted

9.7.2.1 Category A:

On request

- Containment test report.
- Cross sectional drawing with principal dimensions and names of components.
- Test program.

9.7.2.2 Category B and C:

- Cross sectional drawing with principal dimensions and materials of housing components for containment evaluation.
- Documentation of containment in the event of disc fracture, see 9.7.3.2.
- Operational data and limitations as:
- Maximum permissible operating speed (rpm)
- Alarm level for over-speed
- Maximum permissible exhaust gas temperature before turbine
- Alarm level for exhaust gas temperature before turbine
- Minimum lubrication oil inlet pressure
- Lubrication oil inlet pressure low alarm set point
- Maximum lubrication oil outlet temperature
- Lubrication oil outlet temperature high alarm set point
- Maximum permissible vibration levels, i.e. self- and externally generated vibration

(Alarm levels may be equal to permissible limits but shall not be reached when operating the engine at 110% power or at any approved intermittent overload beyond the 110%.)

- Arrangement of lubrication system, all variants within a range
- Type test reports
- Test program

9.7.2.3 Category C:

- Drawings of the housing and rotating parts including details of blade fixing.
- Material specifications (chemical composition and mechanical properties) of all parts mentioned above.

- Welding details and welding procedure of above mentioned parts, if applicable.
- Documentation* of safe torque transmission when the disc is connected to the shaft by an interference fit, see M73.3.3.
- Information on expected lifespan, considering creep, low cycle fatigue and high cycle fatigue.
- Operation and maintenance manuals*).

Note*: Applicable to two sizes in a generic range of turbochargers.

9.7.3 Design requirements and corresponding type testing

9.7.3.1 General

a) The turbochargers shall be designed to operate under conditions given in Chapter 1, SECTION 3, 3.1 and 3.2 of this Part. The component lifetime and the alarm level for speed shall be based on 45°C air inlet temperature.
b) The air inlet of turbochargers shall be fitted with a filter.

9.7.3.2 Containment

a) Turbochargers shall fulfil containment in the event of a rotor burst. This means that at a rotor burst no part may penetrate the casing of the turbocharger or escape through the air intake. For documentation purposes (test/calculation), it shall be assumed that the discs disintegrate in the worst possible way.
b) For category B and C, containment shall be documented by testing. Fulfilment of this requirement can be awarded to a generic range** of turbochargers based on testing of one specific unit. Testing of a large unit is preferred as this is considered conservative for all smaller units in the generic range. In any case, it must be documented (e.g. by calculation) that the selected test unit really is representative for the whole generic range.
Note**: A generic range means a series of turbocharger which are of the same design, but scaled to each other.
c) The minimum test speeds, relative to the maximum permissible operating speed, are:
    - For the compressor: 120%.
    - For the turbine: 140% or the natural burst speed, whichever is lower.
d) Containment tests shall be performed at working temperature.
e) A numerical analysis (simulation) of sufficient containment integrity of the casing based on calculations by means of a simulation model may be accepted in lieu of the practical containment test, provided that:
    - The numerical simulation model has been tested and its suitability/accuracy has been proven by direct comparison between calculation results and the practical containment test for a reference application (reference containment test). This test shall be performed at least once by the manufacturer for acceptance of the numerical simulation method in lieu of tests.
    - The corresponding numerical simulation for the containment is performed for the same speeds as specified for the containment test.
    - Material properties for high-speed deformations are to be applied in the numeric simulation. The correlation between normal properties and the properties at the pertinent deformation speed are to be substantiated.
    - The design of the turbocharger regarding geometry and kinematics is similar to the turbocharger that was used for the reference containment test. In general, totally new designs will call for a new reference containment test.

9.7.3.3 Disc-shaft shrinkage fit

a) Applicable to Category C

b) In cases where the disc is connected to the shaft with interference fit, calculations shall substantiate safe torque transmission during all relevant operating conditions such as maximum speed, maximum torque and maximum temperature gradient combined with minimum shrinkage amount.

9.7.3.4 Type testing

a) Applicable to Categories B and C
b) The type test for a generic range of turbochargers may be carried out either on an engine (for which the turbocharger is foreseen) or in a test rig.
c) Turbochargers are to be subjected to at least 500 load cycles at the limits of operation. This test may be waived if the turbocharger together with the engine is subjected to this kind of low cycle testing, see 9.2 of this Chapter.
d) The suitability of the turbocharger for such kind of operation is to be preliminarily stated by the manufacturer.
e) The rotor vibration characteristics shall be measured and recorded in order to identify possible sub-synchronous vibrations and resonances.
f) The type test shall be completed by a hot running test at maximum permissible speed combined with maximum permissible temperature for at least one hour. After this test, the turbocharger shall be opened for examination, with focus on possible rubbing and the bearing conditions.
g) The extent of the surveyor's presence during the various parts of the type tests is left to the discretion of LHR.

9.7.4 Certification

9.7.4.1 The manufacturer shall adhere to a quality system designed to ensure that the designer's specifications are met, and that manufacturing is in accordance with the approved drawings.

9.7.4.2 For category C, this shall be verified by means of periodic product audits of an Alternative Certification Scheme (ACS) by LHR.

9.7.4.3 These audits shall focus on:

- Chemical composition of material for the rotating parts.
- Mechanical properties of the material of a representative specimen for the rotating parts and the casing.
- UT and crack detection of rotating parts.
- Dimensional inspection of rotating parts.
- Rotor balancing.
- Hydraulic testing of cooling spaces to 4 bars or 1.5 times maximum working pressure, whichever is higher.
- Overspeed test of all compressor wheels for a duration of 3 minutes at either 20% above alarm level speed at room temperature or 10% above alarm level speed at 45°C inlet temperature when tested in the actual housing with the corresponding pressure ratio. The overspeed test may be waived for forged wheels that are individually controlled by an approved non-destructive method.

9.7.4.4 Turbochargers shall be delivered with:

- For category C, a LHR certificate, which at a minimum cites the applicable type approval and the ACS, when ACS applies.
- For category B, a work's certificate, which at a minimum cites the applicable type approval, which includes production assessment.

9.7.4.5 The same applies to replacement of rotating parts and casing.

9.7.4.6 Alternatively to the above periodic product audits, individual certification of a turbocharger and its parts may be made at the discretion of LHR. However, such individual certification of category C turbocharger and its parts shall also be based on test requirements specified in the above mentioned bullet points.

9.7.5 Alarms & Monitoring

9.7.5.1 For all turbochargers of Categories B and C, indications and alarms as listed in the table are required.

9.7.5.2 Indications may be provided at either local or remote locations

Pos.	Monitored Parameters	Category of Turbochargers				Notes
		B		C		
		Alarm	Indication	Alarm	Indication	
1	Speed	High (4)	X (4)	High (4)	X (4)	
2	Exhaust gas at each turbocharger inlet temperature	high (4)	X (1)	high	X	High temp. alarms for each cylinder at engine is acceptable (2)
3	Lub. oil at turbocharger outlet, temperature			high	X	If not forced system, oil temperature near bearings
4	Lub. oil at turbocharger inlet, pressure	low	X	low	X	Only for forced lubrication systems (3)

Notes:

1. For Category B turbochargers, the 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.
2. Alarm and indication of the exhaust gas temperature at turbocharger inlet may be waived if alarm and indication for individual exhaust gas temperature is provided for each cylinder and the alarm level is set to a value safe for the turbocharger.
3. Separate sensors are to be provided if the lubrication oil system of the turbocharger is not integrated with the lubrication oil system of the diesel engine or if it is separated by a throttle or pressure reduction valve from the diesel engine lubrication oil system.
4. On turbocharging systems where turbochargers are activated sequentially, speed monitoring is not required for the turbocharger(s) being activated last in the sequence, provided all turbochargers share the same intake air filter and they are not fitted with waste gates.

9.8 Alternative Certification Scheme (ACS) (IACS UR Z26 New (2015))

9.8.1 Definitions

9.8.1.1 ACS is a certification scheme involving a manufacturer (and associated sub-suppliers, if needed) in the inspection, testing and certification of the manufacturer’s products.

9.8.1.2 An ACS will clarify:

- The extent of the required inspection and testing.
- To which extent and under which conditions the manufacturer may perform all or parts of the required inspection and testing without the presence of a Surveyor from LHR when LHR Certificate is required.

9.8.1.3 The extent to which the manufacturer is given permission to carry out inspections and testing without the presence of a Surveyor is to be agreed on a case by case basis, e.g. for a specific product production line or for specific parts.

9.8.2 Scope

9.8.2.1 An ACS may be arranged with product manufacturers and/or sub-suppliers.

9.8.2.2 An ACS with a manufacturer must define the handling of subcontracted parts (those that require LHR or work certificates or in any other way are addressed in the LHR's Rules). The sub-supplier may be included in the ACS of the manufacturer or have his own ACS or deliver parts that are inspected and certified by LHR.

9.8.2.3 An ACS that permits the manufacturer to carry out all or parts of required inspection and testing without the presence of a Surveyor may be arranged in two versions with regard to traceability:

- The ACS describes inspection, testing and certification additional to the manufacturer's standard quality control in order to meet the Rules. The components are to be stamped with a special stamp supplied by LHR or identified as required by LHR.
- The manufacturer has a standard quality control that covers all required inspection, testing and certification in compliance with the Rules. Traceability and the required type of product document for components or products will be defined in the ACS.

9.8.3 Conditions

9.8.3.1 The conditions for the manufacturer to be granted the permission to carry out inspection and testing without the presence of a Surveyor are that:

- The manufacturer has an implemented Quality System according to a national or international standard approved by an accredited certification body or recognised by LHR.
- The manufacturer has a quality control system, current drawings, and Rules and standards that cover the product to be certified.
- The inspection and testing required by the Rules are either standard procedures in the Quality System and recognized by LHR or specified in detail in the ACS.
- LHR initially ascertains the manufacturer's compliance with the ACS requirements by verifying the required product and process approvals and performing an initial audit. Follow-up and renewal audits are conducted by LHR on a regular basis to verify that conditions of the ACS are continuously maintained by the manufacturer.
- If work certificates (W) or test reports (TR) are found not to fulfil the standards agreed with LHR, the component may not be accepted.
- The agreed ACS may be suspended or cancelled when / if found justified by LHR.
- LHR may carry out unscheduled inspections at the manufacturer and/or subcontractor at its own discretion.
- The manufacturers (and designers, if producing under license) commit themselves to involve LHR when changes to the design, manufacturing process or testing are made as well as when any major production problems or any major product delivery problems have occurred.
- The validity of an ACS is to be a maximum of 5 years. The ACS may be renewed subject to an audit. The scope of the renewal audit shall:
  - verify the conditions of the ACS are still met
  - verify that the current products and processes are appropriately controlled

9.8.4 Information to be submitted

9.8.4.1 For admission to an alternative certification scheme for a product, the manufacturer is to submit an application enclosing the following documentation:

- Product details.
- Existing class approvals of the manufacturer's products as far as required.
- The procedures relevant to the manufacturing process.
- A list of material suppliers with an indication of their class approval (as far as required by the Rules) and the type of material certification in each case.
- Quality control plans relevant to the products and relevant components to be certified through the alternative certification scheme. Said plans are to detail the inspections and tests required by the Rules with an indication of which inspections and tests are delegated to the manufacturer and which are to be done in the presence of an LHR representative.
- The procedures relevant to the quality control and inspections, their methods, frequency and certification.
- The list of suppliers of materials and main components of the product, including certificates.
- The quality system details.
- List of nominated personnel for:
  - Marking/stamping of products
  - Tests and Inspection (responsible)
  - Provision of data and information (e.g. declaration of conformity, test reports etc.)
- Any other additional documents that LHR may require in order to evaluate the manufacturing processes and product quality control.

9.8.5 Audit procedure

9.8.5.1 Upon satisfactory examination of the complete documentation for application an initial audit shall be carried out at the manufacturer's works. This audit is to verify that the manufacture of the product and the relevant controls are performed in accordance with the documents submitted and are in compliance with the requirements laid down in the ACS documentation and the LHR Rules.

9.8.5.2 Upon satisfactory outcome of the audits, the extent, duration and conditions of the ACS are documented.

9.8.5.3 At least one intermediate audit during the period of validity of the ACS is to be carried out. Additional audits may be required at the discretion of LHR.

SECTION 10 Engine alignment

10.1 General

10.1.1 The crankshaft alignment is to be checked every time an engine has been aligned on its foundation by measurement of the crank web deflections. Note is to be taken of:

(a) The loading condition of the ship, and
(b) The condition of the engine (whether hot or cold).

SECTION 11 Auxiliary systems

11.1 Lubricating oil system

11.1.1 General requirements with regard to lubricating oil systems and to the filtering, cooling etc. of the lubricating oil are contained in Part 5, Chapter 10.

11.1.2 On engines where sumps are used as oil reservoirs there is to be equipment for the establishment of the oil level and, if necessary, the completion of the reservoir without interrupting operation. There is also to be appropriate equipment for the complete drainage of the oil sump.

11.1.3 The equipment of engines driving lubricating oil pumps is to comply with Part 5, Chapter 10. In addition, where main lubricating oil pumps are driven by the main engine, special care is to be taken for the following:

(1) There is to be sufficient supply of lubricating oil over the whole operating range of the engine.
(2) The main engines should be equipped with independently driven stand-by pumps or with means for connecting the lubricating oil system to independently driven stand-by pumps.

11.1.4 Where more than one engines with separate lubricating oil systems are used for the propulsion LHR may approve the carriage on board of reserve pumps provided that the means required for their installation are available on board.

11.2 Cooling system

11.2.1 The equipment of engines with cooling water systems is to comply with Part 5, Chapter 10. In addition, where main cooling water pumps are driven by the main engine, care is to be taken for the following:

(1) There is to be sufficient supply of cooling water over the whole operating range of the engine.
(2) The main engines should be equipped with independently driven stand-by pumps or with means for connecting the cooling water system to independently driven stand-by pumps.

11.2.2 Where more than one engines with separate fresh cooling water systems are used for the propulsion, LHR may approve the carriage on board of reserve pumps provided that the means required for their installation are available on board.

11.2.3 Where engines are installed in spaces in which oil-firing equipment is operated, their operation must not be impaired by variations in the air pressure.

11.3 Charge air system

11.3.1 Exhaust gas turbochargers

The design, construction and testing of exhaust gas turbochargers are to comply with Part 5, Chapter

3b. In addition, the following should be taken into account:

(1) In the case of a turbocharger failure emergency operation must be possible
(2) No critical speeds are permitted over the whole operating range of the engine.
(3) The lubricating oil supply must be ensured during start-up and run-down of the exhaust gas turbochargers.
(4) Main engines must be supplied with sufficient charge air over the whole range of operation.
(5) Where the engine can be operated only with a charge air compressor driven independently of the engine, a stand-by charge air compressor or an equivalent device is to be installed.

11.3.2 Charge air cooling

The design, construction and testing of charge air coolers are to comply with Part 5, Chapter 7. In addition means are to be provided for regulating the temperature of the charge air within the temperature range specified by the engine manufacturer.

SECTION 12 Air compressors

12.1 General

12.1.1 The Rules of the present section apply to reciprocating compressors of the normal marine types. Where the installation of a different compressor is intended LHR preserves the right to ask for proof of its appropriateness for the purposed use.

12.1.2 Documents for approval

The following documents are to be submitted to LHR in triplicate for approval:

– drawings of longitudinal cross-sections,
– drawings of transverse cross-sections,
– drawings of the crankshaft and of the connecting rod.

12.2 Materials

12.2.1 Normally, steel, cast steel or nodular cast iron should be used in the construction of the crankshafts and connecting rods of reciprocating compressors. If a different cast iron alloy is intended to be used the approval of LHR is required.

12.2.2 In case of crankshafts with a crank pin diameter greater than 50 mm, material tests are required. If the crank pin diameters is equal or less than 50 mm works certificates are sufficient.

12.3 Crankshaft dimensions

12.3.1 The diameters of journals and crank pins are to be determined as follows:

$$d_p = \sqrt[3]{k \cdot k_m \cdot p_n \cdot D^2 \cdot (2 \cdot h + k_c \cdot s)}$$

where:

d_n = minimum pin or journal diameter, mm,

k = coefficient according to Table 2.12.1,

k_m = coefficient correlated to the material according to Table 2.12.2 or Table 2.12.3,

p_n = design pressure, applicable up to 40 bar, bar,

D = cylinder bore for single-stage compressors, mm,

= D_{Hd} = cylinder bore of the second stage in two-stage compressors with separate pistons, mm

= $1{,}4 \cdot D_{Hd}$ for two stage compressors with a stepped piston as in Figure 2.12.1

= $\sqrt{D_{Nd}^2 - D_{Hd}^2}$ for two-stage compressors with a differential piston as in Figure 2.12.2, mm,

h = piston stroke , mm,

k_C = 1,0 where the cylinders are in line,

= 1,8 for V or W type with cylinders at 45°,

= 1,5 for V or W type with cylinders at 60°,

= 1,2 for V or W type with cylinders at 90°,

s = distance, mm, between main bearing centres where one crank is located between two bearings. s is to be substituted by $s_1 = 0{,}85 \cdot s$ Where two cranks at different angles are located between two main bearings, or by $s_2 = 0{,}95 \cdot s$ where two or three connecting rods are mounted on the same crank.

Table 2.12.1: Values of the coefficient k

Number of cylinders	1	2	4	6	≥ 8
k	0,0020	0,0022	0,0024	0,0026	0,0028

Table 2.12.2: Values of the coefficient km for steel shafts

Minimum testing strength (N/mm²)	400	440	480	520	560	600	640	≥ 680	720(*)	≥ 760(*)
k_m	1,03	0,94	0,91	0,85	0,79	0,77	0,74	0,70	0,66	0,64

Table 2.12.3: Values of the coefficient km for nodular cast iron shafts

Minimum testing strength (N/mm²)	370	400	500	600	700	≥ 800
k_m	1,20	1,10	1,08	0,98	0,94	0,90

Figure 2.12.1

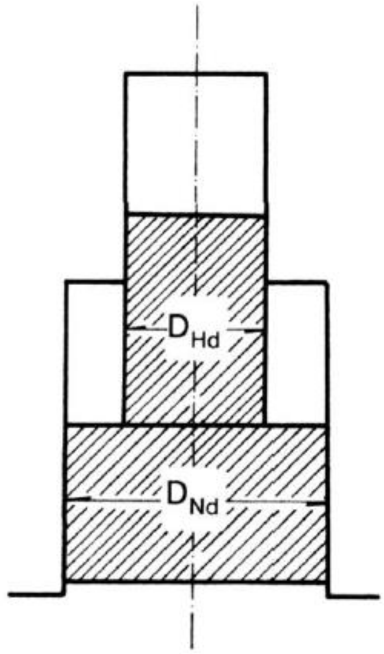
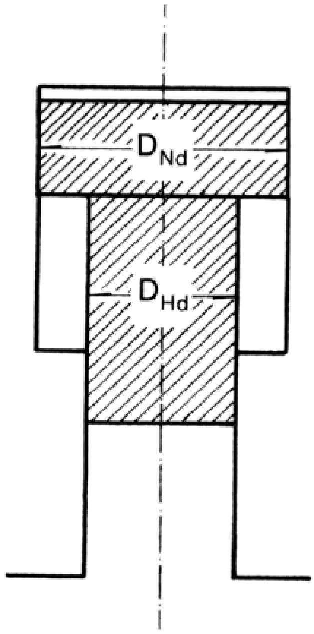


Figure 2.12.2



* Only for drop-forged crankshafts

12.3.2 Where increased strength is achieved by a favourable configuration of the crankshaft, smaller values of d_p may be approved.

12.4 Construction and fittings

12.4.1 General

(1) In the design calculation of a cooler, in case of water cooling the seawater temperature should be taken equal to 32°C and in case of air cooling the air temperature should be taken equal to 45°C, unless higher temperatures are dictated by the temperature conditions of the ship's operational area.
(2) If fresh water is used as cooling mean its inlet temperature should not exceed 40°C.
(3) High-pressure stage air coolers shall not be located in the compressor cooling water space.
(4) The cooling water spaces of compressors and coolers must be fitted with safety valves or rupture discs of sufficient cross-sectional area unless they are provided with open discharges.

12.4.2 Safety valves and pressure gauges

(1) A safety valve preventing the maximum permissible working pressure from being exceeded by more than 10%, is to be fitted in every stage of the compressor.
(2) Every stage of the compressor is to be provided with an appropriate pressure gauge.
(3) In case of compressor stages comprising more than one cylinders which can be shut off individually, a safety valve and a pressure gauge is to be fitted on each cylinder.

12.4.3 Air compressors with oil-lubricated pressure spaces

(1) The compressed air temperature at the discharge from the individual stages, may be less than 160°C for multi-stage compressors or 200°C for single-stage compressors. For discharge pressures less than 10 bar, temperatures may be higher by 20 K.
(2) All compressors are to be equipped with a water trap and an aftercooler fitted after the final stage.
(3) The air spaces between the stages must be provided with means of drainage.
(4) Where the power consumption of the compressor exceeds 20 kW thermometers should be fitted at the individual discharge connections. Alternatively, the thermometers may be mounted at the inlet end of the pressure line, if this is more practicable.

12.4.4 Name plate

The following information should be shown on a name plate fitted on each compressor:

- Manufacturer
- Year of construction
- Power consumption
- Effective suction rate
- Discharge pressure
- Speed of rotation

12.5 Tests

12.5.1 Pressure tests

(1) Cylinders and cylinder liners are to be subjected to hydraulic pressure tests at 50% over the final pressure of the stage concerned.
(2) The compressed air chambers of the intercoolers and aftercoolers of air compressors are to be subject to hydraulic pressure tests at 50% over the final pressure of the stage concerned.

12.5.2 Compressors are to be subjected to a performance test at the manufacturer's works under supervision of LHR and are to be presented for final inspection.

SECTION 13 Documents for the approval of diesel engines (IACS UR M44, Cor.1 2022)

13.1 Scope

The documents necessary to approve a diesel engine design for conformance to the Rules and for use during manufacture and installation are listed. The document flow between engine designer, LHR approval centre, engine builder/licensee and LHR's Surveyors is provided.

13.2 Definitions

Definitions relating to approval of diesel engines are given in Table 2.13.1.

Table 2.13.1: GLOSSARY

Term	**Definition**
Acceptance criteria	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
Accepted	Status of a design, product, service or process, which has been found to conform to specific acceptance criteria
Alternative Certification Scheme (ACS	A system, by which LHR evaluates a manufacturer's quality assurance and quality control arrangements for compliance with Rule requirements, then authorizes a manufacturer to undertake and witness testing normally required to be done in the presence of a Surveyor. The Alternative Certification Scheme as presently administrated by the Member Societies is generally known as: ABS: Product Quality Assurance BV: Alternative Survey Scheme CCS: Type Approval-A CRS: Examination of the manufacturing process and quality assurance system DNV-GL: Manufacturing Survey Arrangement IRS: IRS Quality Assurance Scheme KR: Quality Assurance System LR: LR Quality Schemes NK: Approval of Manufacturers PRS: Alternative Certification Scheme RINA: Alternative Survey Scheme RS: Agreement on Survey
Appraisal	Evaluation by a competent body
Approval	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
Assembly	Equipment or a system made up of components or parts
Assess	Determine the degree of conformity of a design, product, service, process, system or organization with identified specifications, Rules, standards or other normative documents
Audit	Planned systematic and independent examination to determine whether the activities are documented, the documented activities are implemented, and the results meet the stated objectives
Auditor	Individual who has the qualifications and experience to perform audits
Certificate	A formal document attesting to the compliance of a design, product, service or process with acceptance criteria
Certification	A procedure whereby a design, product, service or process is approved in accordance with acceptance criteria
Class	Short for LHR
Class approval	Approved by LHR
Classification	Specific type of certification, which relates to the Rules of LHR

Competent body	Organization recognized as having appropriate knowledge and expertise in a specific area
Component	Part, member of equipment or system
Conformity	Where a design, product, process or service demonstrates compliance with its specific requirements
Contract	Agreement between two or more parties relating to the scope of service
Contractor	see Supplier
Customer	Party who purchases or receives goods or services from another
Design	All relevant plans, documents, calculations described in the performance, installation and manufacturing of a product
Design analysis	Investigative methodology selectively used to assess the design
Design appraisal	Evaluation of all relevant plans, calculations and documents related to the design
Design review	Part of the appraisal process to evaluate specific aspects of the design
Drawings approval/ plan approval	Part of the design approval process which relates to the evaluation of drawings and plans
Equipment	Part of a system assembled from components
Equivalent	An acceptable, no less effective alternative to specified criteria
Evaluation	Systematic examination of the extent to which a design, product, service or process satisfies specific criteria
Examination	Assessment by a competent person to determine compliance with requirements
Inspection	Examination of a design, product service or process by an Inspector
Inspection plan	List of tasks of inspection to be performed by the Inspector
Installation	The assembling and final placement of components, equipment and subsystems to permit operation of the system
Manufacturer	Party responsible for the manufacturing and quality of the product
Manufacturing process	Systematic series of actions directed towards manufacturing a product
Manufacturing process approval	Approval of the manufacturing process adopted by the manufacturer during production of a specific product
Material	Goods supplied by one manufacturer to another manufacturer that will require further forming or manufacturing before becoming a new product
Modification	A limited change that does not affect the current approval
Modification notice	Information about a design modification with new modification index or new drawing number replacing the earlier drawing
Performance test	Technical operation where a specific performance characteristic is determined
Producer	See manufacturer

Product	Result of the manufacturing process
Prototype test	Investigations on the first or one of the first new engines with regard to optimization, fine tuning of engine parameters and verification of the expected running behaviour
Quality assurance	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 as amended
Regulation	Rule or order issued by an executive authority or regulatory agency of a government and having the force of law
Repair	Restore to original or near original condition from the results of wear and tear or damages for a product or system in service
Requirement	Specified characteristics used for evaluation purposes
Information	Additional technical data or details supplementing the drawings requiring approval
Revision	Means to record changes in one or more particulars of design drawings or specifications
Specification	Technical data or particulars which are used to establish the suitability of materials, products, components or systems for their intended use
Substantive modifications or major modifications or major changes	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
Subsupplier/subcontractor	One who contracts to supply material to another supplier
Supplier	One who contracts to furnish materials or design, products, service or components to a customer or user
Test	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
Traceability	Ability to follow back through the design and manufacturing process to the origin
Type approval	The establishment of the acceptability of a product through the systematic: 1. Evaluation of a design to determine conformance with specifications 2. Witnessing manufacture and testing of a type of product to determine compliance with the specification 3. Evaluation of the manufacturing arrangements to confirm that the product can be consistently produced in accordance with the specification
Type approval test	Last step of the type approval procedure. Test program in accordance with UR M71
Witness	Individual physically present at a test and being able to record and give evidence about its outcome

13.3 Overview

13.3.1 Approval process

13.3.1.1 Type approval certificate

For each type of engine that is required to be approved, a type approval certificate is to be obtained by the engine designer. The process details for obtaining a type approval certificate are in 13.4. This process consists of the engine designer obtaining:

- drawing and specification approval
- conformity of production
- approval of type testing programme,
- type testing of engines
- review of the obtained type testing results
- evaluation of the manufacturing arrangements
- issue of a type approval certificate upon satisfactorily meeting the Rule requirements

13.3.1.2 Engine certificate

Each diesel engine manufactured for a shipboard application is to have an engine certificate. The certification process details for obtaining the engine certificate are in 13.5. This process consists of the engine builder/licensee obtaining design approval of the engine application specific documents, submitting a comparison list of the production drawings to the previously approved engine design drawings referenced in 13.3.1.1, forwarding the relevant production drawings and comparison list for the use of the Surveyors at the manufacturing plant and shipyard if necessary, engine testing and upon satisfactorily meeting the Rule requirements, the issuance of an engine certificate.

13.3.2 Document flow for diesel engines

13.3.2.1 Document flow for obtaining a type approval certificate

a. For the initial engine type, the engine designer prepares the documentation in accordance with requirements in Tables 2.13.2 and 2.13.3 and forwards to LHR according to the agreed procedure for review.

b. Upon review and approval of the submitted documentation (evidence of approval), it is returned to the engine designer.

c. The engine designer arranges for a Surveyor to attend an engine type test and upon satisfactory testing LHR issues a type approval certificate.

d. A representative document flow process for obtaining a type approval certificate is shown in Figure 2.13.1.

13.3.2.2 Document flow for engine certificate

a. The engine type must have a type approval certificate. For the first engine of a type, the type approval process and the engine certification process (ECP) may be performed simultaneously.

b. Engines to be installed in specific applications may require the engine designer/licensor to modify the design or performance requirements. The modified drawings are forwarded by the engine designer to the engine builder/licensee to develop production documentation for use in the engine manufacture in accordance with Table 2.13.4.

c. The engine builder/licensee develops a comparison list of the production documentation to the documentation listed in Tables 2.13.2 and 2.13.3. An example comparison list is provided in Figure

2.13.3. If there are differences in the technical content on the licensee's production drawings/documents compared to the corresponding licensor's drawings, the licensee must obtain agreement to such differences from the licensor using the template in Table 2.13.6.

If the designer acceptance is not confirmed, the engine is to be regarded as a different engine type and is to be subjected to the complete type approval process by the licensee.

d. The engine builder/licensee submits the comparison list and the production documentation to LHR according to the agreed procedure for review/approval.

e. LHR returns documentation to the engine builder/licensee with confirmation that the design has been approved. This documentation is intended to be used by the engine builder/licensee and their subcontractors and attending Surveyors. As the attending Surveyors may request the engine builder/licensee or their subcontractors to provide the actual documents indicated in the list, the documents are necessary to be prepared and available for the Surveyors.

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

g. The engine builder/licensee assembles the engine, tests the engine with a Surveyor present. An engine certificate is issued by the Surveyor upon satisfactory completion of assembly and tests.

h. A representative document flow process for obtaining an engine certificate is shown in Figure 2.13.2.

13.3.3 Approval of diesel engine components

Components of engine designer's design which are covered by the type approval certificate of the relevant engine type are regarded as approved whether manufactured by the engine manufacturer or sub-supplied. 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.).

13.3.4 Submission format of documentation

LHR determines the documentation format: electronic or paper. If documentation is to be submitted in paper format, the number of copies is determined by LHR.

13.4 Type approval process

The type approval process consists of the steps in 13.4.1 to 13.4.4. The document flow for this process is shown in Figure 2.13.1. The documentation, as far as applicable to the type of engine, to be submitted by the engine designer/licensor to LHR is listed in Tables 2.13.2 and 2.13.3.

13.4.1 Documents for information Table 2.13.2

Table 1 lists basic descriptive information to provide LHR an overview of the engine's design, engine characteristics and performance. Additionally, there are requirements related to auxiliary systems for the engine's design including installation arrangements, list of capacities, technical specifications and requirements, along with information needed for maintenance and operation of the engine.

13.4.2 Documents for approval or recalculation Table 2.13.3

Table 2 lists the documents and drawings, which are to be approved by LHR.

13.4.3 Design approval/appraisal (DA)

DA's are valid as long as no substantial modifications have been implemented. Where substantial modifications have been made the validity of the DA's may be renewed based on evidence that the design is in conformance with all current Rules and statutory regulations (e.g. SOLAS, MARPOL). See also 13.4.6.

13.4.4 Type approval test

A type approval test is to be carried out in accordance with Part 5, Chapter 2, SECTION 9, 9.2 and is to be witnessed by LHR. The manufacturing facility of the engine presented for the type approval test is to be assessed in accordance with Part 5, Chapter 2, SECTION 9, 9.6.

13.4.5 Type approval certificate

After the requirements in 13.4.1 through 13.4.4 have been satisfactorily completed LHR issues a type approval certificate (TAC).

13.4.6 Design modifications

After LHR has approved the engine type for the first time, only those documents as listed in the tables, which have undergone substantive changes, will have to be resubmitted for consideration by LHR.

13.4.7 Type approval certificate renewals

A renewal of type approval certificates will be granted upon submission of information in either 13.4.7(a) or 13.4.7(b):

a. The submission of modified documents or new documents with substantial modifications replacing former documents compared to the previous submission(s) for DA.
b. A declaration that no substantial modifications have been applied since the last DA issued.

13.4.8 Validity of type approval certificate

LHR reserves the right to limit the duration of validity of the type approval certificate. The type approval certificate will be invalid if there are substantial modifications in the design, in the manufacturing or control processes or in the characteristics of the materials unless approved in advance by LHR.

13.4.9 Document review and approval

The assignment of documents to Table 2.13.2 for information does not preclude possible comments by LHR.

Where considered necessary, LHR may request further documents to be submitted. This may include details or evidence of existing type approval or proposals for a type testing programme in accordance with Part 5, Chapter 2, SECTION 9, 9.2.

13.5 Certification process

The certification process consists of the steps in 13.5.1 to 13.5.5. This process is illustrated in Figure 2.13.2 showing the document flows between the:

- engine designer/licensor
- engine builder/licensee
- component manufacturers
- LHR approval centre
- LHR site offices

For those cases when a licensor – licensee agreement does NOT apply, an “engine designer” shall be understood as the entity that has the design rights for the engine type or is delegated by the entity having the design rights to modify the design.

The documents listed in Table 2.13.4 may be submitted by:

- the engine designer (licensor)
- the manufacturer/licensee.

13.5.1 Document development for production

Prior to the start of the engine certification process, a design approval is to be obtained per 13.4.1 through 13.4.3 for each type of engine. Each type of engine is to be provided with a type approval certificate obtained by the engine designer/licensor prior to the engine builder/licensee beginning production manufacturing. For the first engine of a type, the type approval process and the certification process may be performed simultaneously.

The engine designer/licensor reviews the documents listed in Tables 2.13.2 and 2.13.3 for the application and develops, if necessary, application specific documentation for the use of the engine builder/licensee in developing engine specific production documents. If substantive changes have been made, the affected documents are to be resubmitted to LHR as per 13.4.6.

13.5.2 Documents to be submitted for inspection and testing

Table 2.13.4 lists the production documents, which are to be submitted by the engine builder/licensee to LHR following acceptance by the engine designer/licensor. The Surveyor uses the information for inspection purposes during manufacture and testing of the engine and its components. See 13.3.2.2(c) through 13.3.2.2(f).

13.5.3 Alternative execution

If there are differences in the technical content on the licensee’s production drawings/documents compared to the corresponding licensor’s drawings, the licensee must provide to LHR approval centre a “Confirmation of the licensor’s acceptance of licensee’s modifications” approved by the licensor and signed by licensee and licensor. Modifications applied by the licensee are to be provided with appropriate quality requirements. See Table 2.13.6 for a sample format.

13.5.4 Manufacturer approval

LHR assesses conformity of production with the LHR’s requirements for production facilities comprising manufacturing facilities and processes, machining tools, quality assurance, testing facilities, etc. See Part 5, Chapter 2, SECTION 9, 9.6. Satisfactory conformance results in the issue of a class approval document.

13.5.5 Document availability

In addition to the documents listed in Table 2.13.4, the engine builder/licensee is to be able to provide to the Surveyor performing the inspection upon request the relevant detail drawings, production quality control specifications and acceptance criteria. These documents are for supplemental purposes to the survey only.

13.5.6 Engine assembly and testing

Each engine assembly and testing procedure required according to relevant Rules are to be witnessed by LHR unless an Alternative Certification Scheme meeting the requirements of Part 5, Chapter 2, SECTION 9, 9.8 is agreed between manufacturer and LHR.

Table 2.13.2: Documentation to be submitted for information, as applicable

No.	Item
1	Engine particulars (e.g. Data sheet with general engine information (see Table 2.13.5), Project Guide, Marine Installation Manual)
2	Engine cross section
3	Engine longitudinal section
4	Bedplate and crankcase of cast design
5	Thrust bearing assembly (1)
6	Frame/framebox/gearbox of cast design (2)
7	Tie rod
8	Connecting rod
9	Connecting rod, assembly (3)
10	Crosshead, assembly (3)
11	Piston rod, assembly (3)
12	Piston, assembly (3)
13	Cylinder jacket/ block of cast design (2)
14	Cylinder cover, assembly (3)
15	Cylinder liner
16	Counterweights (if not integral with crankshaft), including fastening
17	Camshaft drive, assembly (3)
18	Flywheel
19	Fuel oil injection pump
20	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
	For electronically controlled engines, construction and arrangement of:
21	Control valves
22	High-pressure pumps
23	Drive for high pressure pumps
24	Operation and service manuals (4)
25	FMEA (for engine control system) (5)
26	Production specifications for castings and welding (sequence)
27	Evidence of quality control system for engine design and in service maintenance

28	Quality requirements for engine production
29	Type approval certification for environmental tests, control components (6)

FOOTNOTES:

1. If integral with engine and not integrated in the bedplate.
2. Only for one cylinder or one cylinder configuration.
3. Including identification (e.g. drawing number) of components.
4. 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.
5. Where engines rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves, a failure mode and effects analysis (FMEA) is to be submitted to demonstrate that failure of the control system will not result in the operation of the engine being degraded beyond acceptable performance criteria for the engine.
6. Tests are to demonstrate the ability of the control, protection and safety equipment to function as intended under the specified testing conditions per Part 8, Chapter 1, Section 7, 7.2.

Table 2.13.3: Documentation to be submitted for approval, as applicable

No.	Item
1	Bedplate and crankcase of welded design, with welding details and welding instructions (1), (2)
2	Thrust bearing bedplate of welded design, with welding details and welding instructions (1)
3	Bedplate/oil sump welding drawings (1)
4	Frame/framebox/gearbox of welded design, with welding details and instructions (1), (2)
5	Engine frames, welding drawings (1), (2)
6	Crankshaft, details, each cylinder No.
7	Crankshaft, assembly, each cylinder No.
8	Crankshaft calculations (for each cylinder configuration) according to the attached data sheet and Part 5, Chapter 2, Section 3
9	Thrust shaft or intermediate shaft (if integral with engine)
10	Shaft coupling bolts
11	Material specifications of main parts with information on non-destructive material tests and pressure tests (3)
	Schematic layout or other equivalent documents on the engine of:
12	Starting air system
13	Fuel oil system
14	Lubricating oil system
15	Cooling water system
16	Hydraulic system
17	Hydraulic system (for valve lift)
18	Engine control and safety system
19	Shielding of high pressure fuel pipes, assembly (4)

20	Construction of accumulators (for electronically controlled engine)
21	Construction of common accumulators (for electronically controlled engine)
22	Arrangement and details of the crankcase explosion relief valve (see Part 5, Chapter 2, SECTION 5, 5.4) (5)
23	Calculation results for crankcase explosion relief valves (see Part 5, Chapter 2, SECTION 5, 5.4)
24	Details of the type test program and the type test report) (7)
25	High pressure parts for fuel oil injection system (6)
26	Oil mist detection and/or alternative alarm arrangements (see Part 5, Chapter 2, SECTION 5, 5.5)
27	Details of mechanical joints of piping systems (see Part 5, Chapter 8, SECTIONS 1, 2 and 4)
28	Documentation verifying compliance with inclination limits (see Part 5, Chapter 1, SECTION 3, 3.2)
29	Documents as required in IACS UR E22, as applicable

FOOTNOTES:

1. For approval of materials and weld procedure specifications. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.
2. For each cylinder for which dimensions and details differ.
3. For comparison with LHR requirements for material, NDT and pressure testing as applicable.
4. All engines.
5. Only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m3 or more.
6. The documentation to contain specifications for pressures, pipe dimensions and materials.
7. The type test report may be submitted shortly after the conclusion of the type test.

Table 2.13.4: Documentation for the inspection of components and systems

- Special consideration will be given to engines of identical design and application
- For engine applications refer to Part 5, Chapter 2, SECTION 9, 9.6

No.	Item
1	Engine particulars as per data sheet in Table 2.13.5
2	Material specifications of main parts with information on non-destructive material tests and pressure tests (1)
3	Bedplate and crankcase of welded design, with welding details and welding instructions (2)
4	Thrust bearing bedplate of welded design, with welding details and welding instructions (2)
5	Frame/framebox/gearbox of welded design, with welding details and instructions (2)
6	Crankshaft, assembly and details
7	Thrust shaft or intermediate shaft (if integral with engine)
8	Shaft coupling bolts
9	Bolts and studs for main bearings

10	Bolts and studs for cylinder heads and exhaust valve (two stroke design)
11	Bolts and studs for connecting rods
12	Tie rods
	Schematic layout or other equivalent documents on the engine of: (3)
13	Starting air system
14	Fuel oil system
15	Lubricating oil system
16	Cooling water system
17	Hydraulic system
18	Hydraulic system (for valve lift)
19	Engine control and safety system
20	Shielding of high pressure fuel pipes, assembly (4)
21	Construction of accumulators for hydraulic oil and fuel oil
22	High pressure parts for fuel oil injection system (5)
23	Arrangement and details of the crankcase explosion relief valve (see Part 5, Chapter 2, Section 5, 5.4) (6)
24	Oil mist detection and/or alternative alarm arrangements (see Part 5, Chapter 2, SECTION 5, 5.5)
25	Cylinder head
26	Cylinder block, engine block
27	Cylinder liner
28	Counterweights (if not integral with crankshaft), including fastening
29	Connecting rod with cap
30	Crosshead
31	Piston rod
32	Piston, assembly (7)
33	Piston head
34	Camshaft drive, assembly (7)
35	Flywheel
36	Arrangement of foundation (for main engines only)
37	Fuel oil injection pump
38	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
39	Construction and arrangement of dampers
	For electronically controlled engines, assembly drawings or arrangements of:
40	Control valves
41	High-pressure pumps

42	Drive for high-pressure pumps
43	Valve bodies, if applicable
44	Operation and service manuals (8)
45	Test program resulting from FMEA (for engine control system) (9)
46	Production specifications for castings and welding (sequence)
47	Type approval certification for environmental tests, control components (10)
48	Quality requirements for engine production

FOOTNOTES:

1. For comparison with LHR requirements for material, NDT and pressure testing as applicable.
2. For approval of materials and weld procedure specifications. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.
3. Details of the system so far as supplied by the engine manufacturer such as: main dimensions, operating media and maximum working pressures.
4. All engines.
5. The documentation to contain specifications for pressures, pipe dimensions and materials.
6. Only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m^3 or more.
7. Including identification (e.g. drawing number) of components.
8. 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.
9. Required for engines that rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves.
10. Documents modified for a specific application are to be submitted to LHR for information or approval, as applicable. See 13.3.2.2(b), Figure 2.13.3 and Table 2.13.6.

Figure 2.13.1: Type approval document flow

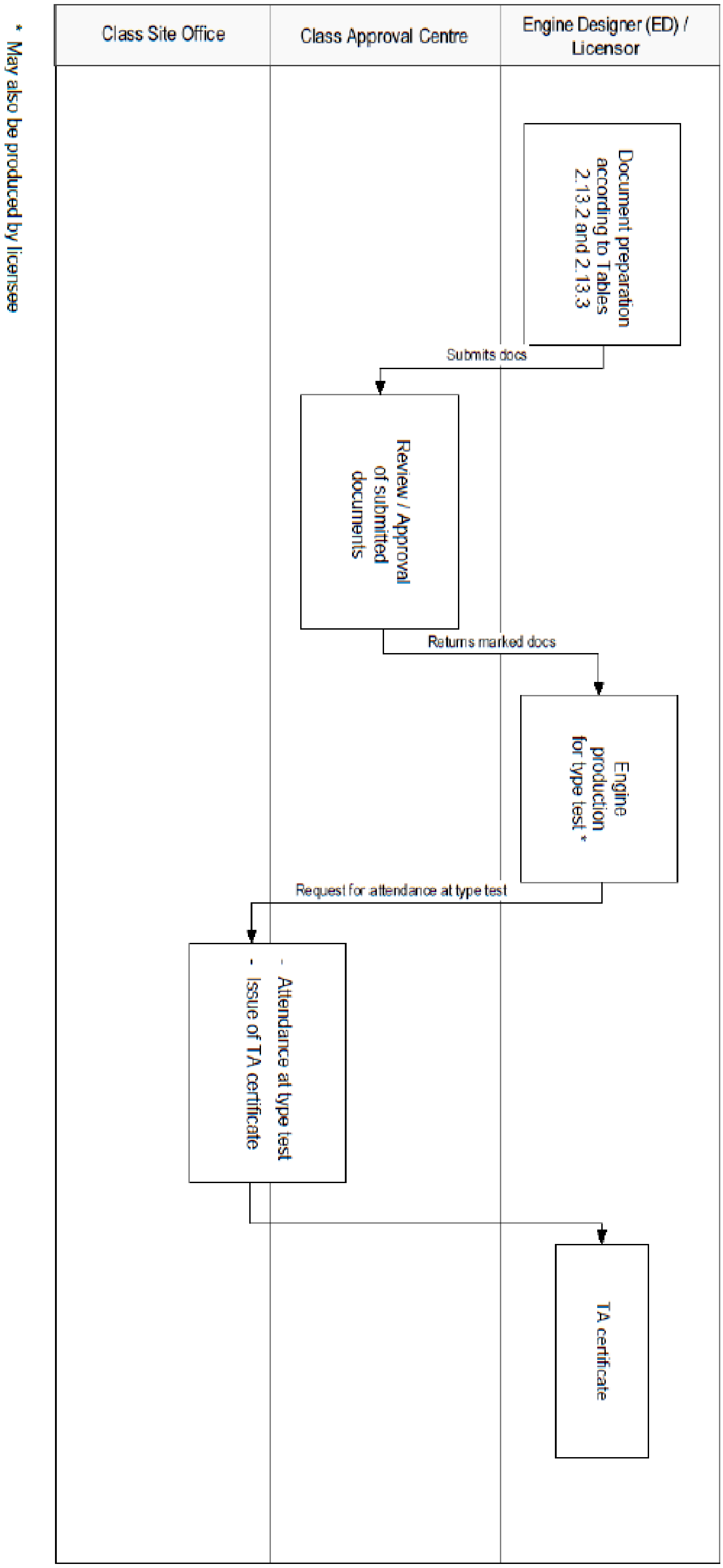


Figure 2.13.2: Engine certificate document flow

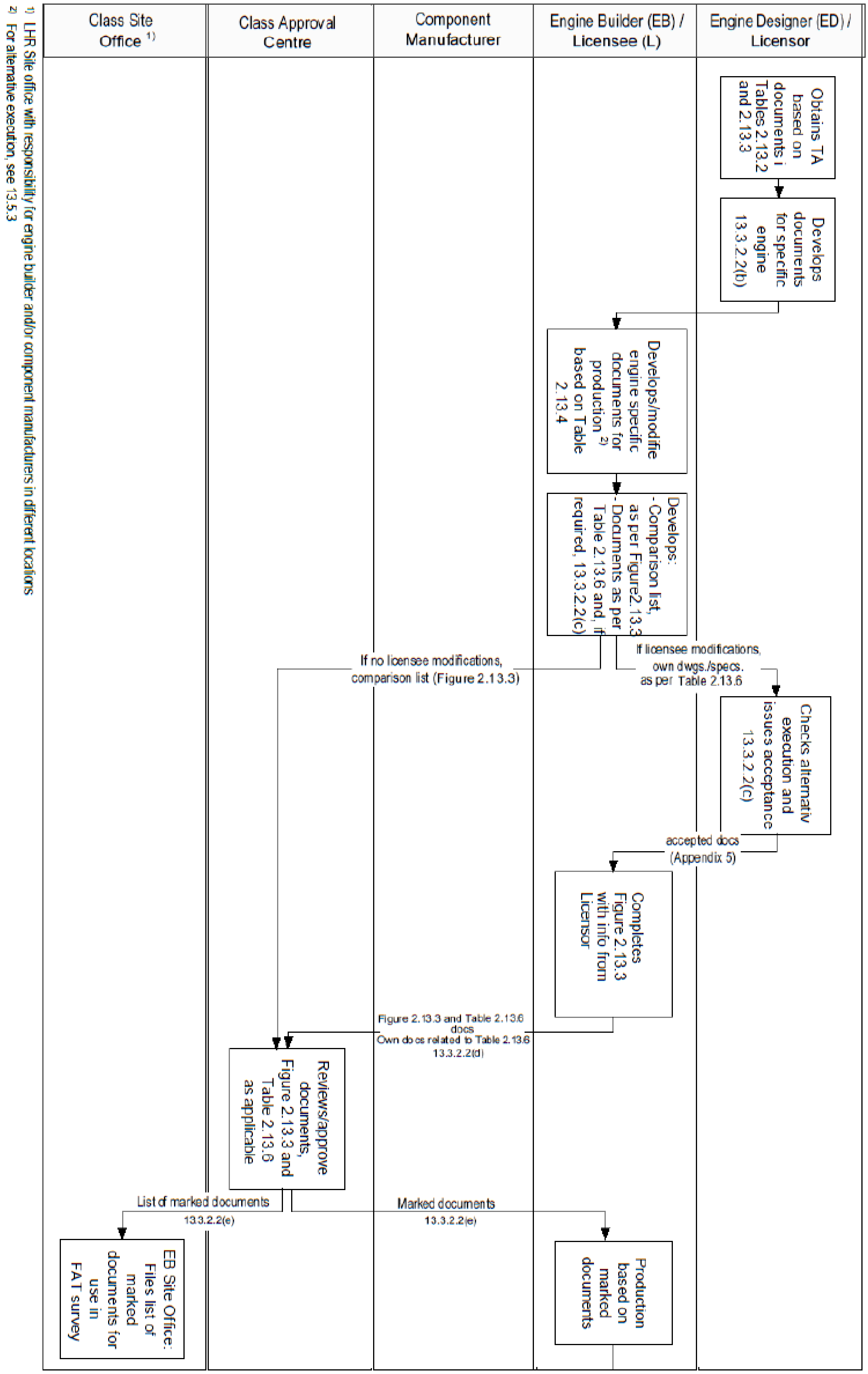


Figure 2.13.2 (continue): **Engine certificate document flow**

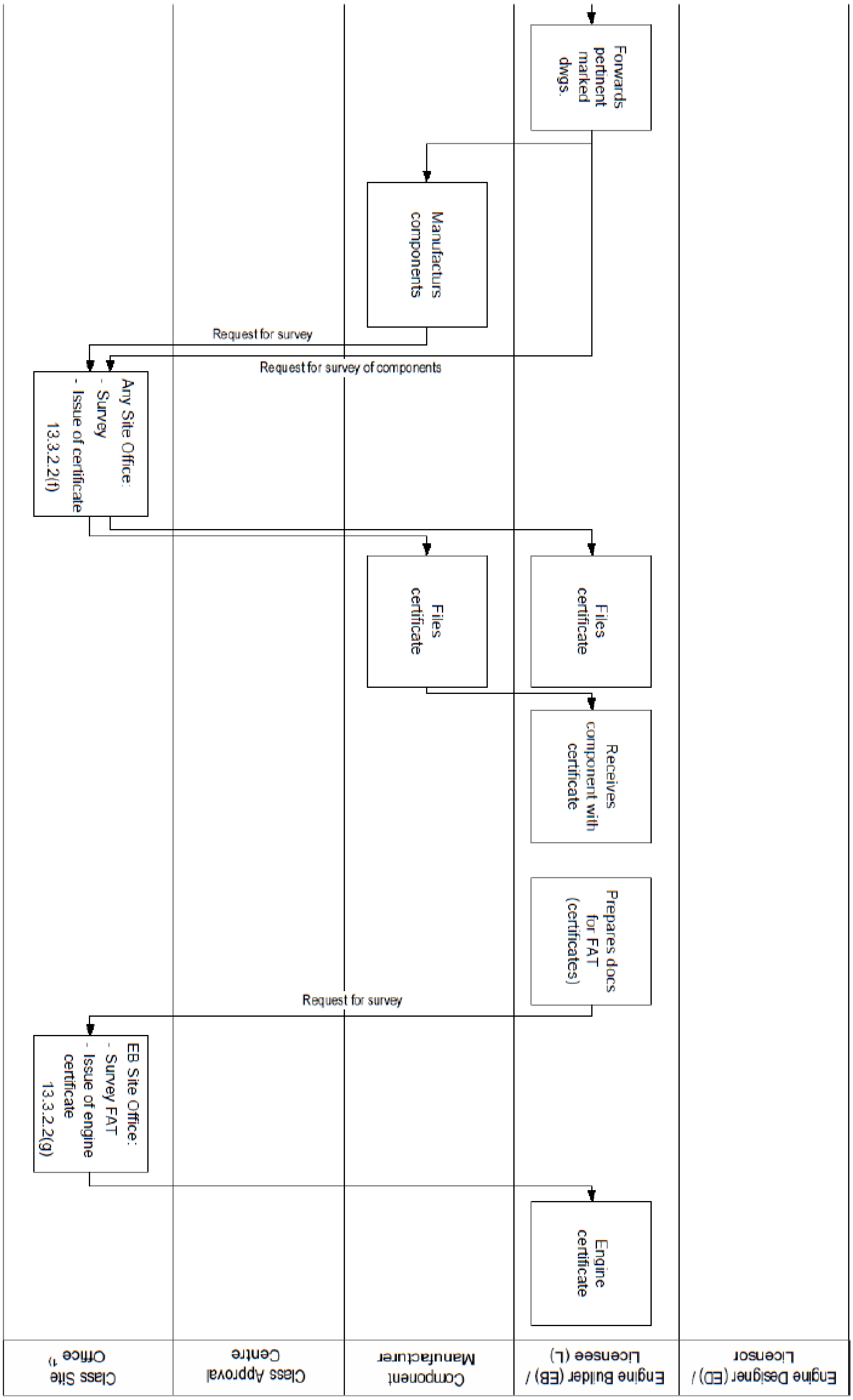


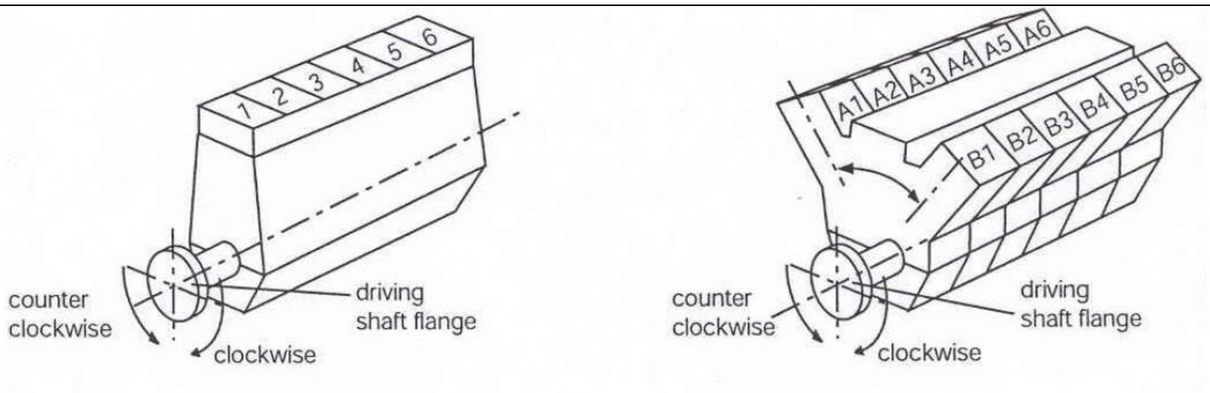
Table 2.13.5: Internal Combustion Engine Approval Application Form and Data Sheet

Class Application number (if applicable):		Engine Manufacturer's Application Identification Number:	
General Data			
Engine Designer:		Engine Manufacturer(s), Licensee(s) and/or Manufacturing Sites' Name Country	
Contact Person:			
Address:			
1. Document propose (select options from either 1a or 1b)			
1a. Type Approval Application			
Service Requested	Required activities^		
□ New Type Approval	• DA, TT, CoP		
□ Renew Type Approval	• CoP, if design change then amended or new certificate process to be followed		
□ Amend Type Approval	• DA & CoP, Further TT if previously approved engine has been substantively modified (as required by Part 5, Chapter 2, SECTION 9, 9.2)		
□ Design Evaluation	• DA, TT, applicable where designer does not have production facilities, Type Approval to be granted to specific production facility once associated CoP has been completed		
□ Update Ta Supplement	• Update to Supplement, only for minor changes not affecting the Type Approval Certificate		
□ Other	• e.g. National/Statutory Administration requirements i.e. MSC.81(70), for emergency engines		
For TA Cert amendments or Supplement updates, details of what is to be changed:			
For 'Other', Details of the requirements to be considered:			
1b. Addendum for Individual Engine FAT and Certification			
□	Individual engine requiring FAT and Certification, only where the performance data for the engine being certified differs from the details provided on the original Type Approval Application. Only section 3b requires completion. Where changes to other sections are necessary, a new Type Approval Application may be required.		
Reference number of Internal Combustion Engine Approval Application Form previously submitted and reference number of the Type Approval Certificate.	(Copy of original application form to be attached to this document)		
2. Existing documentation			
Previous Class Type Approval Certificate No. or related Design Approval No. (if applicable)			
Formerly issued documentation for engine	Issuing Body:	Document Type:	Document No.:
(E.g. previous type test reports, in-service experience justification reports, etc.)			
Existing certification	Issuing Body:	Document Type:	Document No.:
(E.g. Manufacturer's quality certification ISO 9001:2015 etc.)			
3. Design (mark all that apply)			
3a. Engine Particulars			
Engine Type		Number of delivered marine engines‡:	
Manufactured since‡:			

Application	□ Direct drive Propulsion (□ Single engine / Multi-engine installation)		□ Auxiliary (□ Aux. Services / □ Electric Propulsion)	□ Emergency
Mechanical Design	□ 2-stroke □ Cross-head Cylinder bore (mm)	□ 4-stroke □ Trunk-piston	□ In-line □ Reversible Length of piston stroke (mm)	□ Vee (V-angle °) □ Other ( ) □ Non-reversible
Supercharging	□ Without supercharging	□ With supercharging □ Without charge air cooling □ Constant pressure charging system		□ With charge air cooling □ Pulsating pressure charging system
Valve operation	□ Cam control	□ Electronic control		
Fuel injection	□ Direct injection	□ Indirect injection	□ Cam controlled injection	□ Electronically controlled injection
Fuel Types (Classification according to ISO 8216-1:2017)	□ Marine residual fuel cSt (Max. kinematic viscosity at 50 °C) □ Marine distillate fuel DMA, DMB, DMC □ Marine distillate fuel DMX □ Low flashpoint liquid fuel (specify fuel type) □ Gas (specify gas type) □ Other (specify) □ Dual Fuel (specify combinations of fuels to be used simultaneously)			
3b. Performance Data (Related to: Barometric pressure 1,000 mbar; Air temperature 45°C; Relative humidity 60%; Seawater temperature 32°C)				
Model reference No. (if applicable)				
Max. continuous rating	kW/cyl			
Rated speed	1/min			
Mean indicated pressure	MPa			
Mean effective pressure	MPa			
Max. firing pressure	MPa			
Charge air pressure	MPa			
Compression ratio	-			
Mean piston speed	m/s			
3c. Crankshaft				
Design	□ Solid	□ Semi-built	□ Built	
Method of Manufacture	□ Cast	□ Forged □ Slab Forged	□ Approved die forged	□ Continuous grain flow process
State approved forge/works name:				
Is the crankshaft hardened by an approved process which includes the fillet radii of crankpins and journals? □ Yes □ No If yes, state process:				
Crankshaft material specification: U.T.S. (N/mm^2) Hardness value (Brinell/Vickers)		Yield strength (N/mm^2) Elongation (%)		
Dimensional Data				

If shrunk on webs, state shrinkage allowance (mm)	Yield strength of crankweb material (N/mm^2)
Centre of gravity of connecting rod from large end centre (mm)	Radius of gyration of connecting rod (mm)
Mass of each crankweb (kg)	Centre of gravity of web from journal axis (mm)
Mass of each counterweight (kg)	Centre of gravity of each counterweight from journal axis (mm)
Axial length of main bearing (mm)	Main bearing working clearance (mm)
Mass of flywheel at driving end (kg)	Mass of flywheel at opposite end (kg)
Nominal alternating torsional stress in crankpin (N/mm^2)	Radius of gyration of connecting rod (mm)
Length between centres (Total length) (mm)	Nominal alternating torsional stress in crank journal (N/mm^2)

3d. Firing order



State numbering system of cylinders from left to right as per above diagrams (as applicable)

Number of cylinders	Clockwise firing order	Counter-clockwise firing order

4. Design (mark all that apply)

4a. Turbochargers	□ Fitted	□ Not Fitted
Turbocharger oil supply by:	□ Engine lub. oil system	□ TC internal lub. oil system

No. of cylinders	No. of aux blowers	No. of charge air coolers	No. of TC	TC manufacturer & type	TC type approval certificate No.
				/	
				/	
				/	
				/	
				/	
				/	
				/	

4b. Speed governor

Engine application	Manufacturer / type	Mode of operation	Type approval cert. No.

(Main/Aux/Emergency)			(if electric / electronic gov.)
	/ / / /		

4c. Overspeed protection
Independent overspeed protection available ◻ Yes ◻ No Mode of operation: Manufacturer / type, if electronic: / Type approval certificate No.
4d. Electronic systems
Engine control and management system Note: use Remarks section to identify when a different engine control system will be used for Type Test Hardware: Manufacturer & Model: / Type approval certificate No. Software: Name & Version: / Software conformity certificate No.
Additional electronic system 1: System function Manufacturer & type: / Type approval certificate No.
Additional electronic system 2: System function Manufacturer & type: / Type approval certificate No.
Additional electronic system 3: System function Manufacturer & type: / Type approval certificate No.
4e. Starting Systems
Type:
4f. Starting devices/functions
A flame arrestor or a bursting disk is installed in the starting air system: before each starting valve ◻ Yes ◻ No In the starting air manifold ◻ Yes ◻ No
Crank relief valves available ◻ Yes ◻ No Manufacturer / type: / Type approval certificate No. ◻ Yes ◻ No

No. of cyl.	Total crankcase gross volume incl. attachments (m3)	Type & size (mm) of relief valve	Relief area per relief valve (mm2)	No. of relief valves
		/ / / / /		

Method used for detection of potentially explosive crankcase condition: ◻ Oil mist detector: Manufacturer / type: / Type of approval certificate No.
◻ Alternative Method: ◻ crankcase pressure monitoring ◻ bearing temperature monitoring ◻ other: (mark all that apply) ◻ oil splash temperature monitoring ◻ recirculation arrangements
Cylinder overpressure warning device available ◻ Yes ◻ No Type: Opening pressure (bar):
4g. Attached ancillary equipment (Mark all that apply)

Engine driven pumps:

□ Main lubricating oil pump □ Sea cooling water pump □ LT-fresh cooling water pump

□ HT-fresh cooling water pump □ Fuel oil booster pump □ Hydraulic oil pump □ Other ( )

Engine attached motor driven pumps:

□ Lubricating oil pump □ Cooling fresh water pump □ Fuel oil booster pump

□ Hydraulic oil pump □ Other ( )

Engine attached cooler or heater:

□ Lubricating oil cooler	□ Lubricating oil heater	□ Fuel oil valve cooler
□ Hydraulic oil cooler	□ Cooling water fresh water cooler	

Engine attached filter:

Lubricating oil filter	□ Single	□ Duplex	□ Automatic
Fuel oil filter	□ Single	□ Duplex	□ Automatic

5. Inclination limits

(engine operation is safeguarded under the following limits)	Athwartships		Fore and aft	
	Static	Dynamic	Static	Dynamic
Main & Auxiliary machinery	□ 15.0°	□ 22.5°	□ 5.0°	□ 7.5°
Emergency machinery	□ 22.5°	□ 22.5°	□ 10.0°	□ 10.0°
Emergency machinery on ships for the carriage of liquefied gas and liquid chemicals	□ 30.0°	□ 30.0°		

6. Main engine emergency operation

At failure of one auxiliary blower, engine can be started and operated at partial load	□ Yes	□ No
At failure of one turbocharger, engine operation can be continued	□ Yes	□ No

7. References: Additional Information Attached to Application

Document Name/Number	Summary of information contained in document

8. Further Remarks:

Notes: * All parties that affect the final complete engine (e.g. manufacture, modify, adjust) are to be listed. All sites where such work is carried out may be required to complete CoP assessment. ^ DA = Design Appraisal, TT = Type Test, CoP = Assessment of Conformity of Production. See 'Definitions' at the end of this application form for more information. ‡ Only in case of TA Extension.	
Completed By:	Signature:
Company:	
Job Title:	Stamp:
Date:	

Definitions:

Design Appraisal: Evaluation of all relevant plans, calculations and documents related to the design to determine compliance with LHR technical requirements. This includes requirements for all associated ancillary equipment and systems essential for the safe operation of the engine i.e. the Complete Engine. The Design Appraisal is recorded on a Supplement to the Type Approval Certificate.

Type Testing requires satisfactory completion of testing of the Complete Engine against the requirements of LHR' applicable engine Type Testing programme. Type testing is only applicable to the first in series; all engines are to complete factory acceptance and shipboard trials as defined by DBS requirements (Part 5, Chapter 2, Section9, 9.3).

Design Evaluation Certification may be granted upon satisfactory completion of Design Appraisal and Type Testing.

Assessment of Conformity of Production means the assessment of quality assurance, manufacturing facilities and processes and testing facilities, to confirm the manufacturer's capability to repeatedly produce the complete engine in accordance with the approved and type tested design.

Type Approval Certification will be granted upon satisfactory completion of Design Appraisal, Type Testing and assessment of Conformity of Production of the complete engine. The Type Approval Certificate will incorporate outputs from the Design Appraisal, the Type Test and the Assessment of Conformity of Production.

Complete Engine includes the control system and all ancillary systems and equipment referred to in the Rules that are used for safe operation of the engine and for which there are rule requirements, this includes systems allowing the use of different fuel types. The exact list of components/items that will need to be tested in together with the bare engine will depend on the specific design of the engine, its control system and the fuel(s) used but may include, but are not limited to, the following:

a) Turbocharger(s)
b) Crankcase explosion relief devices
c) Oil mist detection and alarm devices
d) Piping
e) Electronic monitoring and control system(s) – software and hardware
f) Fuel management system (where dual fuel arrangements are fitted)
g) Engine driven pumps
h) Engine mounted filters

Fuel Types: All fuels that the engine is designed to operate with are to be identified on the application form as this may have impact on the requirements that are applicable for Design Appraisal and the scope of the tests required for Type Testing. Where the engine is to operate in a Dual Fuel mode, the combinations of fuel types are to be detailed. E.g. Natural Gas + DMA, Natural Gas + Marine Residual Fuel, the specific details of each fuel are to be provided as indicated in the relevant rows of the Fuel Types part of section 3a of this form.

Figure 2.13.3: Tabular Listing of Licensor's and Licensee's Drawing and Data

Licensee: ______________________ **Licensor:** ______________________

Licensee Engine No.: ______________________ **Engine type:** ______________________

No.	Components or System	Licensor			Licensee		Has Design Been modified By Licensee?		If Yes, indicate following information	
		Dwg. No. & Title	Rev. No.	Date of Class Approval or Review	Dwg. No.	Rev. No.	Yes	No	Identification Of Alternative approved by Licensor	Date of Class Approval or Review of Licensee Dwg.
1										
2										
3										
4										
5										
6										
7										
8										
9										
...										

I attest the above information to be correct and accurate.

Person in Charge (Licensee):

______________________ ______________________

Printed Name **Signature**

Date: ______________________

Table 2.13.6: Template for confirmation of the licensor's acceptance of licensee's modifications

Engine Licensee Proposed Alternative to Licensor's Design			
Licensee information			
Licensee:		Ref No.:	
Description:		Info No.:	
Engine type:		Main Section:	
Engine No.:		Plant Id.:	
Design Spec:	□ General	□ Specific Nos:	
Licensor design:	State relevant part or drawing. numbers. Insert drawing clips or pictures. Add any relevant information		Licensee Proposed Alternative
		For example: • Differences in geometry • Differences in the functionality • Material • Hardness • Surface condition • Alternative standard • Licensee production information introduced on the drawing • Weldings or castings • etc.	
Reason: □ Licensee's production □ Sub-supplier's production □ Cost down □ Tools	Interchangeability w. licensor design □ Yes □ No	Non-conformity Report Research, Assessment, Evaluation □ NCR □ RAE	Certified by licensee: Initials: Date:
Licensor comments			
LoAE: □ Accepted as alternative execution (Licensor undertakes responsibility)	**NCR:** □ Approved		Certified by licensor: Initials: Date:
□ No objection | □ Not acceptable (Licensee undertakes responsibility)	□ Conditionally approved □ Rejected		
Licensor ref.:			Date:
Licensee ref.:			Date:

CHAPTER 3 (a) Steam Turbines

CONTENTS

SECTION 1 General

1.1 Application

1.1.1 The requirements of this Chapter are applicable to steam turbines used for main propulsion and essential auxiliary services where power needs exceed 110 kW.

1.2 Documents for approval

1.2.1 The following documents, for each turbine installation, are to be submitted to LHR in triplicate for approval:

(1) Assembly and sectional drawings of the turbines.
(2) Detail drawings of rotors, casings, diaphragms, blades, valves, bed frames and main condenser (for gearing, see Part 5, Chapter 4a).
(3) Details of operating characteristics and critical speeds.
(4) On request, calculations relating to blade vibration, details of the welding conditions applicable to welded components, proof of a sufficient safety margin in the components subject to the severest loads, etc.

1.2.2 In case of small auxiliary turbines with a steam inlet temperature less than 250 °C it is sufficient to submit sectional drawings of the turbines only.

SECTION 2 Materials

2.1 General

2.1.1 Grey cast iron is not to be used for temperatures exceeding 260 C.

2.1.2 Creep strength, corrosion resistance and scaling properties at working temperatures of the materials to be used are to be submitted for consideration.

2.2 Materials for forgings

2.2.1 Forged steel is to be used in the construction of turbine rotors and discs. For carbon and carbon-manganese steel forgings, the specified minimum tensile strength is to be selected within the limits of 400 and 600 N/mm². For alloy steel rotor forgings, the specified minimum tensile strength is to be selected within the limits of 500 and 800 N/mm². For discs and other alloy steel forgings, the specified minimum tensile strength is to be selected within the limits of 500 and 1000 N/mm².

2.2.2 For steel alloys, details of the proposed chemical composition, heat treatment and mechanical properties are to be submitted for approval.

2.2.3 When it is proposed to use material of higher tensile strength, full details are to be submitted for approval.

2.3 Testing of materials

2.3.1 The following parts are to comply with LHR's Rules for Materials:

(1) Stationary parts (e.g. casings, diaphragms, nozzles and nozzle chests, guide vanes, turbine casing bolts, bed frames, bearing pedestals).
(2) Rotating parts (e.g. discs, shafts, shrink rings, blades, toothed couplings, other dynamically loaded components, valve spindles and cones).
(3) Condenser tubes and tube plates.

2.3.2 For small auxiliary turbines with steam inlet temperature less than 250 C, the disc and shaft materials are to be tested only.

SECTION 3 Design and construction

3.1 General

3.1.1 Due care is to be taken for the relative thermal expansion of the various turbine parts during the design of the turbine installation. Attention is also to be given to the minimization of casing and rotor distortion under the whole range of operating conditions.

3.1.2 Effective means are to be used for preventing excessive accumulation and leakage of oil. In the design of turbine installation special attention is to be given to preventing the contact of lubricating oil with hot parts of the turbine.

3.2 Welded components

3.2.1 Welding procedure approval tests are to be in accordance with the requirements of Part 2, Chapter 9.

3.2.2 In case of components fabricated by means of welding, before the commencement of the work, manufacturers are to submit for consideration details of proposed welding procedures and proposals for the way of examination of joints by non-destructive means.

3.2.3 In subsequent production, check mechanical tests are to be carried out at the Surveyors' discretion.

3.3 Stress raisers

3.3.1 Where abrupt changes of section of rotors, spindles, discs, blade roots and tenons occur smooth fillets are required. The rivet holes in blade shrouds are to be rounded and radiused on top and bottom surfaces, and tenons are to be reduced at their junction with blade tips.

3.3.2 Due attention is to be given that the blade tenons are free from cracks and also to the workmanship and riveting of blades. Test samples are to be sectioned and examined, and pull-off tests made if considered necessary by the Surveyors.

3.4 Vibration

3.4.1 Turbine rotors, rotor discs and blades are to be tested for excessive vibration within the whole operating speed range.

3.4.2 For the vibration and alignment of main propulsion systems formed by the turbines geared to the line shafting, see Part 5, Chapter 6.

3.5 External influences

3.5.1 No excessive thrust loads or moments are to be applied to the turbines due to pipes and ducts connected to turbine casings. Gratings and any fittings in way of sliding feet or flexible-plate supports are to be so arranged that casing expansion is not restricted. In case of a tank structure incorporated in main turbine seatings consideration is to be given to the temperature variation of the tank in service.

3.6 Steam supply and water system

3.6.1 Every precaution is to be taken to prevent condensed steam entering the glands and turbines.

3.7 Turning gear

3.7.1 Main propulsion turbines are to be equipped with turning gear for both directions of rotation. The turning gear for all propulsion turbines is to be power-driven and, if electric, is to be continuously rated.

3.7.2 An interlock is to be provided to ensure that the main turbine cannot be started up when the turning gear is engaged.

SECTION 4 Safety equipment

4.1 Manoeuvering equipment and emergency operation of propulsion steam turbines

4.1.1 The simultaneous admission of steam to the ahead and astern turbines is to be prevented by interlocks. Brief overlapping of the ahead and astern valves during manoeuvering can be allowed.

4.1.2 Devices for emergency operation of propulsion steam turbines

In single screw ships fitted with cross compound steam turbines, the arrangements are to be such as to enable safe navigation when the steam supply to any one of the turbines is required to be isolated. For this emergency operation purpose the steam may be led directly to the L.P. turbine and either the H.P. or M.P. turbine can exhaust direct to the condenser. Adequate arrangements and controls are to be provided for these operating conditions so that the pressure and temperature of the steam will not exceed those which the turbine and condenser can safely withstand.

The necessary pipes and valves for these arrangements are to be readily available and properly marked. A fit-up test of all combinations of pipes and valves is to be performed prior to the first sea trials.

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

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

4.2 Governors and speed control (IACS M26.1 Cor.1 (2005))

4.2.1 All main and auxiliary turbines are to be provided with overspeed protective devices to prevent the design speed from being exceeded by more than 15%. Where two or more turbines are coupled to the same gear wheel set, LHR may agree that only one overspeed protective device be provided for all the turbines.

4.2.2 Arrangement is to be provided for shutting off the steam to the main turbines by suitable hand trip gear situated at the manoeuvering stand and at the turbine itself. Hand tripping for auxiliary turbines is to be arranged in the vicinity of the turbine overspeed protective device.

4.2.3 Where the main turbine installation incorporates a reverse gear, electric transmission, controllable pitch propeller or other free-coupling arrangement, a separate speed governor in addition to the overspeed protective device is to be fitted and is to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

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

4.2.5 Auxiliary turbines driving electric generators are to have both:

- a speed governor which, with fixed setting, is to control the speed within the limit of 10% for momentary variation and 5% permanent variation when the full load is suddenly taken off, and
- an overspeed protective device which is to be independent of speed governor, and is to prevent the design speed from being exceeded by more than 15% when the full load is suddenly taken off (see 4.2.1).

4.3 Miscellaneous safety arrangements (IACS M26.2 Cor.1 (2005))

4.3.1 Main ahead turbines are to be provided with a quick acting device which will automatically shut off the steam supply in the case of dangerous lowering of oil pressure in the bearing lubricating system. This device is to be so arranged as not to prevent the admission of steam to the astern turbine for braking purposes. Where deemed necessary by LHR appropriate means are to be provided to protect the turbines in case of:

- abnormal axial rotor displacement,
- excessive condenser pressure,
- high condensate level.

4.3.2 Auxiliary turbines having governors operated other than hydraulically in which the lubricating oil is inherent in the system, are to be provided with an alarm device and a means of shutting off the steam supply in the case of lowering of oil pressure in the bearing lubricating oil system.

4.3.3 Main turbine are to be provided with a satisfactory emergency supply of lubricating oil which will come into use automatically when the pressure drops below a predetermined value. The emergency supply may be obtained from a gravity tank containing sufficient oil to maintain adequate lubrication until the turbine is brought to rest or by equivalent means. If emergency pumps are used these are to be so arranged that their operation is not affected by failure of the power supply. Suitable arrangement for cooling the bearings after stopping may also be required.

4.3.4 To provide a warning to personnel in the vicinity of the exhaust end steam turbines of excessive pressure, a sentinel valve or equivalent is to be provided at the exhaust end of all turbines. The valve discharge outlets are to be visible and suitably guarded if necessary. When, for auxiliary turbines, the inlet steam pressure exceeds the pressure for which the exhaust casing and associated piping up to exhaust valve are designed, means to relieve the excess pressure are to be provided.

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

4.3.6 Efficient steam strainers are to be provided close to the inlets to ahead and astern high-pressure turbines or alternatively at the inlets to manoeuvering valves.

NOTE:
The hand trip gear is understood as any device which is operated manually irrespective of the way the action is performed, i.e. mechanically or by means of external power.

SECTION 5 Emergency equipment

5.1 Single main boiler

5.1.1 Ships intended for unrestricted service, fitted with steam turbines and having a single main boiler, are to be provided with means ensuring emergency propulsion in case of failure of the main boiler.

5.2 Axial displacement of rotors

5.2.1 Main propulsion turbines must be equipped with quick-closing devices which automatically shut off the steam supply in case of unacceptable axial displacement of the rotor.

In single screw ships fitted with cross compound steam turbines, the arrangements are to be such as to enable safe navigation when the steam supply to any one of the turbines is required to be isolated. For this emergency operation purpose, the steam may be led directly to the L.P. turbine and either the H.P. or M.P. turbine can exhaust direct to the condenser. Adequate arrangements and controls are to be provided for these operating conditions so that the pressure and temperature of the steam will not exceed those which the turbines and condenser can safely withstand.

The necessary pipes and valves for these arrangements are to be readily available and properly marked. A fit-up test of all combinations of pipes and valves is to be performed prior to the first sea trials. The permissible power/speeds when operating without one of the turbines (all combinations) is to be specified and information provided on board.

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

SECTION 6 Tests and trials

6.1 Stability testing of turbine rotor

6.1.1 Every turbine rotor used for main propulsion service and having inlet steam temperature exceeding 400 °C is to be subjected to at least one thermal stability test. The test may be carried out at the forge or turbine builders works:

(1) after heat treatment and rough machining of the forging, or

(2) after final machining, or

(3) after final machining and blading of the rotor.

The stabilizing test temperature is to be not less than 28 °C above the maximum steam temperature to which the rotor will be exposed, and not more than the tempering temperature of the rotor material.

For details of a recommended test procedure and limits of acceptance, see LHR's Rules for Materials. Other test procedures may be adopted if approved.

6.1.2 Where main turbine rotors are subjected to thermal stability tests at both forge and turbine builders' works, the foregoing requirements are applicable to both tests. It is not required that auxiliary turbine rotors be tested for thermal stability, but if such tests are carried out, the requirements for main turbine rotors will be generally applicable.

6.2 Balancing

6.2.1 All rotors as finished-bladed and complete with half-coupling are to be dynamically balanced to the Surveyor's satisfaction, by using a sensitive machine.

6.3 Cold overspeed test

6.3.1 Turbine rotors are to be tested at high speeds. A testing speed at least 150% of the rated speed for not less than three minutes is considered as sufficient. Alternatively, LHR may accept mathematical proof of the stresses in the rotating parts at overspeed instead of the overspeed test itself.

6.4 Hydraulic tests

6.4.1 The cylinders of all turbines are to be tested to a pressure 50% over the working pressure in the casing, or to 2,0 bar, whichever is the greater.

6.4.2 Manoeuvering valves are to be tested to a pressure 100% over the working pressure. The nozzle boxes of impulse turbines are to be tested to a pressure 50% over the working pressure.

6.4.3 The steam space of the condensers is to be tested to a pressure of 1,0 bar. The water space is to be tested to the maximum pressure which the pump can develop at ship's full draught with the discharge valve closed plus 0,7 bar, with a minimum test pressure of 2,0 bar. Where the operating conditions are unspecified, the test pressure is to be not less than 3,4 bar. (See Part 5, Chapter 10).

6.5 Shipboard trials

6.5.1 Main turbines are to be subjected to a dock trial and thereafter, during a trial voyage, to the following tests:

(1) Operation at rated RPM for at least 6 hours,

(2) Reversing manoeuvres, and,

(3) During the dock or sea trials, astern revolutions equal to at least 70% of the rated ahead rpm for about 20 minutes.

During astern and subsequent forward operation, the steam pressures and temperatures and the relative expansion must not assume magnitudes liable to endanger the operational safety of the plant.

6.5.2 Turbines used for auxiliary services are to be run for at least 4 hours at their rated power and for 30 minutes at 110% of the rated power.

6.5.3 LHR reserves the right to call for additional tests in individual cases.

CHAPTER 3 (b) Gas Turbines

CONTENTS

SECTION 1 General

1.1 Application

1.1.1 The requirements of this Chapter are applicable to gas turbines used for main propulsion and for essential auxiliary services if power needs exceed 110 kW. The requirements do not apply to exhaust gas turbo-blowers.

1.2 Documents for approval

1.2.1 The following documents for each gas turbine installation are to be submitted in triplicate to LHR for approval:

(1) Assembly and sectional drawings.
(2) Diagrammatic lay-out of the fuel system, including control and safety devices.
(3) Diagrammatic lay-out of the lubricating oil system.
(4) Detailed drawings of rotors, casings, blades, combustion chambers and heat exchangers.
(5) Detailed description of the operating conditions, including the pressure and temperature curves in the turbine and compressor at the rated power and corresponding rotational speeds, and detail of permissible temporary operation beyond the values for the rated power.
(6) Proof, by analytical calculation or some other acceptable method that a sufficient safety margin has been provided for in the components subject to the severest loads.
(7) In case of welded components, detailed description of the welding conditions.

In addition, a complete set of operating instructions for each turbine type is to be submitted to LHR. In individual cases, LHR reserves the right to call for additional documentation.

SECTION 2 Materials

2.1 General

2.1.1 All materials used in the construction of gas turbine installations must satisfy the requirements imposed by the operating conditions on the individual engine components. Due account is to be taken of corrosion, creep, thermal fatigue and oxidation to which parts are subject when in service. Detailed description of the chemical and mechanical properties of the used materials are to be submitted to LHR for consideration and of the heat treatment applied.

2.1.2 In case of use of composite materials, in addition to 2.1.1, description of the method of manufacture is required.

2.2 Materials for forgings

2.2.1 Forged steel is to be used in the construction of rotors and discs. For carbon and carbon-manganese steel forging, the specified minimum tensile strength is to be selected within the limits of 400 and 600 N/mm^2. For alloy steel rotor forgings, the specified minimum tensile strength is to be selected within the limits of 500 and 800 N/mm^2. For discs and other alloy steel forgings, the specified minimum tensile strength is to be selected within the limits of 500 and 1000 N/mm^2.

2.2.2 For alloy steels, description of the proposed chemical composition and heat treatment is to be submitted for approval.

2.3 Testing of materials

2.3.1 The materials of the following parts are to comply with LHR's Rules for materials or with test specifications laid down by the gas turbine manufacturer and recognised by LHR as part of the approval procedure:

- turbine and compressor wheels
- shafts
- turbine and compressor casings
- guide vanes and blades
- combustion chambers and heat exchangers.

SECTION 3 Design and construction

3.1 General

3.1.1 Due care is to be taken for the relative thermal expansion of the various turbine parts during the design of the turbine installation. Attention is also to be given to the minimization of casing and rotor distortion under the whole range of operating conditions.

3.1.2 Effective means are to be used for preventing excessive accumulation and leakage of oil. In the design of turbine installation special attention is to be given to preventing the contact of lubricating oil with hot parts of the turbine.

3.2 Design basis

3.2.1 Analytical calculations of the steady state stresses in the turbine and compressor rotors and blading at the maximum speed and temperature in service, are to be submitted for consideration.

3.3 Welded components

3.3.1 Welding procedure approval tests are to be in accordance with the requirements of Part 2, Chapter 9.

3.3.2 In case of components fabricated by means of welding, before the commencement of the work, manufacturers are to submit for consideration details of proposed welding procedures and proposals for the way of examination of joints by non-destructive means.

3.3.3 In subsequent production, check mechanical tests are to be carried out at the Surveyors' discretion.

3.4 Vibration

3.4.1 Turbine and compressor rotors, rotor discs and blades are to be tested for excessive vibration within the whole operating speed range.

3.4.2 Analytic calculation of the critical speeds is to be submitted for consideration. In case of presence of critical speeds within the operating speed range, vibration tests may be requested in order to verify the calculations.

3.5 External influences

3.5.1 No excessive thrust loads or moments are to be applied to the turbines due to pipes and ducts connected to turbine casings. Gratings and any fittings in way of sliding feet or flexible-plate supports are to be so arranged that casing expansion is not restricted. In case of a tank structure incorporated in main turbine seatings consideration is to be given to the temperature variation of the tank in service.

SECTION 4 Emergency operation

4.1 Multi-shaft installations

4.1.1 In multi-shaft installations, the failure of one shaft must not hinder the continued, independent operation of the remaining units.

4.2 Single-shaft installations

4.2.1 In single-shaft installations with two or more main gas turbines, care is to be taken to ensure that, in the event of one of the gas turbines failing, the others are able to continue operation independently.

4.2.2 In the case of single-shaft installations with only one main gas turbine, special agreement is to be reached with LHR concerning the emergency operating equipment.

SECTION 5 Control and Safety of Gas Turbines for Marine Propulsion Use (IACS UR M60 Rev.1 (2021))

5.1 Governor and Over speed protective devices

5.1.1 Main gas turbines are to be provided with over speed protective devices to prevent the turbine speed from exceeding more than 15% of the maximum continuous speed.

5.1.2 Where a main gas turbine incorporates a reverse gear, electric transmission, controllable pitch propeller or other free-coupling arrangement, a speed governor independent of the over speed protective device is to be fitted and is to be capable of controlling the speed of the unloaded gas turbine without bringing the over speed protective device into action.

5.2 Miscellaneous automatic safety devices

5.2.1 Details of the manufacturer’s proposed automatic safety devices to safeguard against hazardous conditions arising in the event of malfunctions in the gas turbine installation are to be submitted to LHR together with the failure mode and effect analysis (FMEA).

Unless the FMEA required by this Section proves otherwise, the shutdown functions for gas turbines are to be provided in accordance with Table 3.51 in addition to the general monitoring and safety system functions given by LHR.

5.2.2 Main gas turbines are to be equipped with a quick closing device (shut-down device) which automatically shuts off the fuel supply to the turbines at least in case of:

(a) Over speed
(b) Unacceptable lubricating oil pressure drop
(c) Loss of flame during operation
(d) Excessive vibration
(e) Excessive axial displacement of each rotor (Except for gas turbines with rolling bearings)
(f) Excessive high temperature of exhaust gas
(g) Unacceptable lubricating oil pressure drop of reduction gear
(h) Excessive high vacuum pressure at the compressor inlet

5.2.3 The following turbine services 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:

(a) Lubricating oil supply
(b) Oil fuel supply (or automatic control of oil fuel viscosity as alternative)
(c) Exhaust gas

5.2.4 Automatic or interlocked means are to be provided for clearing all parts of the main gas turbine of the accumulation of liquid fuel or for purging gaseous fuel, before ignition commences on starting or recommences after failure to start.

5.2.5 Hand trip gear for shutting off the fuel in an emergency is to be provided at the manoeuvering station.

5.2.6 Starting devices are to be so arranged that firing operation is discontinued and main fuel valve is closed within pre-determined time, when ignition is failed.

5.3 Alarming devices

5.3.1 Although in principle alarming devices listed in Table 3.5.1 are to be provided, they can be added or omitted, taking into account the results of FMEA specified in 5.2.1.

5.3.2 Suitable alarms are to be operated by the activation of shutdown devices.

Table 3.5.1: List of alarm and shutdown

Monitoring parameter	Alarm	Shutdown
Turbine speed	high	x
Lubricating oil pressure	low*	x
Lubricating oil pressure of reduction gear	low*	x
Differential pressure across lubricating oil filter	high	
Lubricating oil temperature	high	
Oil fuel supply pressure	low	
Oil fuel temperature	high	
Cooling medium temperature	high	
Bearing temperature	high	
Flame and ignition Failure	x	x
Automatic starting Failure	x	
Vibration	high*	x
Axial displacement of rotor	high	x

Exhaust gas temperature	high*	x
Vacuum pressure at the compressor inlet	high*	x
Loss of control system	x	
Notes: Alarms marked with "*" are to be activated at the suitable setting points prior to arriving the critical condition for the activation of shutdown devices.		

SECTION 6 Starting equipment

6.1 Purging before ignition

6.1.1 Means are to be provided, preferably automatic or interlocked, to clear all parts of the gas turbine of the accumulation of liquid fuel, or for purging gaseous fuel, before ignition commences on starting, or recommences after failure to start.

6.2 Starting system

6.2.1 The starting arrangements and starting air pipe systems and safety fittings are to comply with the requirements of Part 5, Chapter 2, SECTION 7, 7.1 and 7.2 where applicable.

SECTION 7 Tests and trials

7.1 Balancing

7.1.1 All rotors as finished-bladed and complete with half-coupling are to be dynamically balanced to the Surveyors' satisfaction, by using a sensitive machine.

7.2 Cold overspeed test

7.2.1 Turbine and compressor rotors are to be tested at high speeds. A testing speed at least 150% of the rated speed for not less than three minutes is considered as sufficient. Alternatively, LHR may accept mathematical proof of the stresses in the rotating parts at overspeed instead of the overspeed test itself.

7.3 Hydraulic tests

7.3.1 All casings are to be tested to a hydraulic pressure 50% over the highest pressure in the casing during normal operation, or 50% over the pressure during starting, whichever is the higher.

7.3.2 Intercoolers and heat exchangers are to be tested to a pressure 50% over the maximum working pressure on each side separately.

7.3.3 Alternatively, where hydraulic tests is impossible to be carried out, the manufacturers must prove, in any satisfactory way, the strength of the component.

7.4 Shipboard trials

7.4.1 Main turbines are to be subjected to a dock trial and thereafter, during a trial voyage, to the following tests:

(1) Operation at rated RPM for at least 6 hours,
(2) Start-stop operation,
(3) Reversing manoeuvres, and,
(4) During the dock or sea trials, astern revolutions equal to at least 70% of the rated ahead rpm for about 20 minutes.

7.4.2 Turbines used for auxiliary services are to be run for at least 4 hours at their rated power and for 30 minutes at 110% of the rated power.

7.4.3 LHR reserves the right to call for additional tests in individual cases.

CHAPTER 4 (a) Gearing

CONTENTS

SECTION 1 General

1.1 Scope

1.1.1 These Rules apply to enclosed gears, both intended for main propulsion and for auxiliary services as required by the Rules of LHR.

1.1.2 For the sizing of gears and couplings for ship with ice classes, see Part 5, Chapter 13.

1.2 Documents for approval

1.2.1 The following documents are to be submitted to LHR in triplicate for approval:

- assembly drawings,
- sectional drawings,
- necessary detailed drawings.

The documents submitted for approval should contain all the information required for the check of the load calculations. LHR reserves the right to ask for any complementary information.

SECTION 2 Materials

2.1 General

2.1.1 In the selection of materials of the different parts special care is to be given to their compatibility in operation. Forged steel is preferably to be used in the construction of shafts, pinions wheels and wheel rims of gears in the main propulsion plant. Rolled steel bar may also be used for plain, flangeless shafts. Couplings in the main propulsion plant must be made of steel, cast steel or nodular cast iron with a mostly ferritic matrix. Suitable cast aluminium alloys may also be permitted for the lightly stressed external components of couplings in main propulsion plans. The rotors and casings of hydraulic slip couplings may also be made of grey cast iron or of a suitable cast aluminium alloy. The requirements of the present paragraph are also to be applied to gears of important auxiliary machinery according to Part 5, Chapter 1.

2.1.2 For gears used in auxiliary machinery installation other than that mentioned in Part 5, Chapter 1, alternatively, other materials of equivalent strength may be used, after the approval of LHR.

2.1.3 The peripheral speed of cast iron gear wheels shall generally not exceed 60m/s, that of cast iron coupling clamps or bowls, 40 m/s.

2.2 Testing of materials

2.2.1 LHR's Rules for materials are to be applied for the testing of all gear and coupling components which are used for the main propulsion. For the major components of the couplings and gears of all important auxiliary machines suitable documentation is to be submitted to LHR.

2.2.2 Where the execution of the tests prescribed in the Rules for Materials is impracticable due to any essential reason, modified tests may be considered at the discretion of LHR.

SECTION 3 Design and construction (IACS M56 Cor.1 (2021))

3.1 Basic principles - introduction and general influence factors

3.1.1 Introduction

The following definitions are mainly based on the ISO 6336 series standard, as amended, (hereinafter called "reference standard") for the calculation of load capacity of spur and helical gears.

3.1.2 Scope and field of application

The following requirements apply to enclosed gears, both intended for main propulsion and for essential auxiliary services, which accumulate a large number of load cycles (several millions), whose gear set is intended to transmit a maximum continuous power equal to, or greater than:

- 220 kW for gears intended for main propulsion

- 110 kW for gears intended for essential auxiliary services

These requirements, however, may be applied to the enclosed gears, whose gear set is intended to transmit a maximum continuous power less than those specified above at the request of LHR.

The following definitions deal with the determination of load capacity of external and internal involute spur and helical gears, having parallel axis, with regard to surface durability (pitting) and tooth root bending strength and to this purpose the relevant basic equations are provided in 3.2 and 3.3. The influence factors common to said equations are described in 3.1. The others, introduced in connection with each basic equation, are described in 3.2 and 3.3. All influence factors are defined regarding their physical interpretation. Some of the influence factors are determined by the gear geometry or have been established by conventions. These factors are to be calculated in accordance with the equations provided. Other factors, which are approximations, can be calculated according to methods acceptable to LHR.

3.1.3 Symbols and units

The main symbols used are listed below. Other symbols introduced in connection with the definition of influence factors are described in the appropriate sections.

SI units have been adopted.

a	=	centre distance, mm,
b	=	common face width, mm,
$b_{1,2}$	=	face width 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,

F_{bt} = nominal tangential load on base cylinder in the transverse section, N,

h = tooth depth, mm,

m_n = normal module, mm,

m_t = transverse module, mm,

$n_{1,2}$ = rotational speed of pinion, wheel, revs/min (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 velocity 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, degrees,

α_t = transverse pressure angle at ref. cylinder, degrees,

α_{tw} = transverse pressure angle at working pitch cylinder, degrees,

β = helix angle at reference cylinder, degrees,

β_b = helix angle at base cylinder, degrees,

ε_α = transverse contact ratio,

ε_β = overlap ratio,

ε_γ = total contact ratio,

3.1.4 Geometrical definitions

For 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 ration, defined as follows, is always greater or equal to the unity:

$$u = z_2/z_1 = d_{w2}/d_{w1} = d_2/d_1$$

For external gears u is positive, for internal gears u is negative. In the equation of surface durability b is the common face width on the pitch diameter. In the equation of tooth root bending stress b_1 or b_2 are the face widths at the 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 face width, b, may be used also in the equation of teeth root bending stress if significant crowning or end relief have been adopted.

$\tan(\alpha_t)$ = $\tan(\alpha_n)/\cos(\beta)$

$\tan(\beta_b)$ = $\tan(\beta) \cdot \cos(\alpha_t)$

$d_{1,2}$ = $z_{1,2} \cdot m_n/\cos(\beta)$

$d_{b1,2}$ = $d_{1,2} \cdot \cos(\alpha_t)$

$$d_{w1} = \frac{2a}{u+1}$$

$$d_{w2} = \frac{2au}{u+1}$$

where $\alpha = 0{,}5(\mathrm{d_{w1}} + \mathrm{d_{w2}})$

$$z_{n1,2} = \frac{z_{1,2}}{cos^2\beta_b \cdot cos\beta}$$

m_t = $\mathrm{m_n}/\cos(\beta)$

$\mathrm{inv}(\alpha)$ = $\tan(\alpha) - \pi \cdot \alpha/180;\ \ \alpha[]$

$\mathrm{inv}(\alpha_{tw})$ = $\mathrm{inv}(\alpha_t) + 2 \cdot \tan(\alpha_\eta)\ \cdot\ (x_1 + x_2)/(z_1 + z_2)$ or $cosa_{tw} = \frac{m_t(z_1+z_2)}{2a}\ cosa_t$

$$\varepsilon_\alpha = \frac{0{,}5 \cdot \sqrt{\mathrm{d_{a1}}^2 - \mathrm{d_{b1}}^2} \pm 0{,}5 \cdot \sqrt{\mathrm{d_{a2}}^2 - \mathrm{d_{b2}}^2} - a \cdot \sin(\alpha_{tw})}{\pi \cdot m_t \cdot \cos(\alpha_t)}$$

the positive sign is used for external gears, the negative sign for internal gears.

ε_β = $\mathrm{b}\ \cdot \frac{\sin(\beta)}{\pi\, m_n}$, mm

for double helix, b is to be taken as the width of one helix

ε_γ = $\varepsilon_\alpha + \varepsilon_\beta$, mm

$$\mathrm{v} = \frac{\pi\ \cdot \mathrm{d_{1,2}}\ \cdot\ \mathrm{n_{1,2}}}{60 \cdot 10^3}$$

3.1.5 Nominal tangential load, F_t

The nominal tangential load, F_t, tangential to the reference cylinder and perpendicular to the relevant axial plane, is calculated directly from the maximum continuous power transmitted by the gear set by means of the following equations:

$$T_{1,2} = (30000/\pi)\ \cdot (P/n_{1,2})$$

$$F_t = 2000\ \cdot\ T_{1,2}/d_{1,2}$$

3.1.6 General influence factors

(1) Application factor, K_A (1)

The application factor, K_A, accounts for dynamic overloads from sources external to the gearing. K_A for gears designed for infinite life is defined as the ratio between the maximum repetitive cyclic torque applied to the gear set and the nominal rated torque. The nominal rated torque is defined by the rated power and speed and is the torque used in the rating calculations. The factor mainly depends on:

- characteristics of driving and driven machines,
- ratio of masses,
- type of couplings,
- operating conditions (overspeeds, changes in propeller load conditions, etc.).

When operating near a critical speed of the drive system, a careful analysis of conditions must be made.

The application factor, K_A, should be determined by measurements or by system analysis acceptable to LHR. Where a value determined in such a way cannot be supplied, the following values can be considered:

Main propulsion	diesel engine with hydraulic or electromagnetic slip coupling	1,00
	diesel engine with high elasticity coupling	1,30
	diesel engine with other couplings	1,50
Auxiliary gears	electric motor, diesel engine with hydraulic or electromagnetic slip coupling	1,00
	diesel engine with high elasticity coupling	1,20
	diesel engine with other couplings	1,40

NOTE:

1. Where the vessel, on which the reduction gear is being used, is receiving an Ice Class notation, the Application Factor or the Nominal Tangential Force should be adjusted to reflect the ice load associated with the requested Ice Class.

(2) Load sharing factor, K_γ

The load sharing factor, K_γ accounts for the maldistribution of load in multiple path transmissions (dual tandem, epicyclic, double helix, etc.). K_γ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor mainly depends on accuracy and flexibility of the branches.

The load sharing factor, K_γ, should be determined by measurements or by system analysis. Where a value determined in such a way cannot be supplied, the following values can be considered for epicyclic gears.

up to 3 planetary gears	1,00
4 planetary gears	1,20
5 planetary gears	1,30
6 planetary gears and over	1,40

(3) Internal dynamic factor, K_V

The internal dynamic factor, K_V, accounts for internally generated dynamic loads due to vibrations of pinion and wheel against each other. K_V is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load (F_t, K_A, K_γ). The factor mainly depends on:

- transmission errors (depending on pitch and profile errors),
- masses of pinion and wheel,
- gear mesh stiffness variation as the gear teeth pass through the meshing cycle,
- transmitted load including application factor,
- pitch line velocity,
- dynamic unbalance of gears and shaft,
- shaft and bearing stiffnesses,
- damping characteristics of the gear system.

The dynamic factor, K_V, is to be calculated as follows:

This method may be applied only to cases where all the following conditions are satisfied:

- running velocity in the subcritical range, i.e.:

$$\frac{\mathrm{v}\, z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 10\ m/s$$

- spur gears (β=0º) and helical gears with β ≤ 30º
- pinion with relatively low number of teeth, z_1 < 50
- solid disc wheels or heavy steel gear rim

This method may be applied to all types of gears if

$$\frac{\mathrm{v}\, z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3\ m/s$$

as well as to helical gears where β > 30^0.

For gears other than the above, reference can be made to Method B outlined in the reference standard ISO 6336-1:2019 as amended.

a) For spur gears and for helical gears with overlap ratio $\varepsilon_\beta \geq 1$

$$K_V = 1 + \left(\frac{K_1}{K_A\,\frac{F_t}{b}} + K_2\right)\cdot\frac{\mathrm{v}\, z_1}{100}\, K_3\sqrt{\frac{u^2}{1+u^2}}$$

If $K_A \cdot F_t/b$ is less than 100 N/mm, this value is assumed to be equal to 100 N/mm.

Numerical values for the factor K_1 are to be as specified in the Table 4.3.1.

Table 4.3.1: Values of the factor K_1 for the calculation of K_v

	K_1 ISO accuracy grades (1)					
	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

NOTE 1:

ISO accuracy grades according to ISO 1328-2:2020 as amended. In case of mating gears with different accuracy grades, the grade corresponding to the lower accuracy should be used.

For all accuracy grades the factor K_2 is to be in accordance with the following:

- for spur gears, K_2=0,0193

- for helical gears, K_2=0,0087

If $\frac{v\,z_1}{100}\sqrt{\frac{u^2}{1+u^2}} \leq 0{,}2$ then $K_3 = 2{,}0$

If $\frac{v\,z_1}{100}\sqrt{\frac{u^2}{1+u^2}} > 0{,}2$ then $K_3 = 2{,}071 - 0{,}357\,\frac{v\,z_1}{100}\sqrt{\frac{u^2}{1+u^2}}$

b) For helical gears with overlap ratio ε_β<1 the value K_V is determined by linear interpolation between values determined for spur gears ($K_{v\alpha}$) and helical gears ($K_{v\beta}$) in accordance with:

$$K_V = K_{\mathrm{v}a} - \varepsilon_\beta\left(K_{\mathrm{v}a} - K_{\mathrm{v}\beta}\right)$$

where:
$K_{\mathrm{v}a}$ is the K_V value for spur gears, in accordance with a)
$K_{\mathrm{v}\beta}$ is the K_V value for helical gears, in accordance with a)

(4) Face load distribution factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress, $K_{F\beta}$ for tooth root bending stress, account for the effects of non-uniform distribution of load across the facewidth. $K_{H\beta}$ and $K_{F\beta}$ are defined as follows:

$$K_{H\beta} = \frac{\text{maximum load per unit facewidth}}{\text{mean load per unit facewidth}}$$

$$K_{F\beta} = \frac{\text{maximum bending stress at tooth root per unit facewidth}}{\text{mean bending stress at tooth root per unit facewidth}}$$

The mean bending stress at tooth root relates to the considered facewidth b_1 resp. b_2. $K_{F\beta}$ can be expressed as a function of the factor $K_{H\beta}$. The factors $K_{H\beta}$ and $K_{F\beta}$ mainly depend on:

- gear tooth manufacturing accuracy,
- errors in mounting due to bore errors,
- bearing clearances,
- wheel and pinion shaft alignment errors,
- elastic deflections of gear elements, shafts, bearings, housing and foundations which support the gear elements,
- thermal expansion and distortion due to operating temperature,
- compensating design elements (tooth crowning, end relief, etc.).

The face load distribution factors, $K_{H\beta}$ for contact stress, and $K_{F\beta}$ for tooth root bending stress are to be determined according to the Method C outlined in the reference standard ISO 6336-1:2019 as amended.

Alternative methods acceptable to LHR may be applied. In case the hardest contact is at the end of the facewidth $K_{F\beta}$ is given by the following equations:

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

$$N = \frac{(b/h)^2}{1 + (b/h) + (b/h)^2}$$

where:

b/h = facewidth/tooth height ratio, the minimum of b_1/h_1 or b_2/h_2. For double helical gears, the facewidth of only one helix is to be used. When b/h<3 the value b/h=3 is to be used.

(a) In case of gears where the ends of the facewidth are lightly loaded or unloaded (end relief or crowning):

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

(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. The factors $K_{H\alpha}$ and $K_{F\alpha}$ mainly depend on:

- total mesh stiffness,
- total tangential load F_t, K_A, K_γ, K_v, $K_{H\beta}$,
- base pitch error,
- tip relief,
- running-in-allowances

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, can be determined according to Method B outlined in the reference standard ISO 6336-1:2019 as amended.

3.2 Surface durability (pitting)

3.2.1 Scope and general remarks

The criterion for surface durability is based on the Hertz pressure, on the operating pitch point or at the inner point of single pair contact. The contact stress σ_H must be equal to or less than the permissible contact stress σ_{HP}.

3.2.2 Basic equations

(1) Contact stress

$$\sigma_H = \sigma_{HO} \cdot \sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \leq \sigma_{HP}$$

where:

σ_{HO} = basic value of contact stress for pinion and wheel,

σ_{HO} = $Z_B \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t \cdot (u+1)}{d_1 \cdot b \cdot u}}$, for pinion

σ_{HO} = $Z_D \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_T \cdot (u+1)}{d_1 \cdot b \cdot u}}$, for wheel

where:

Z_B = single pair tooth contact for pinion, (see 3.2.3),

Z_D = single pair tooth contact for wheel, (see 3.2.3),

Z_H = zone factor, (see 3.2.4),

Z_E = elasticity factor, (see 3.2.5),

Z_ε = contact ratio factor, (see 3.2.6),

Z_β = helix angle factor, (see 3.2.7),

F_t = nominal tangential load at reference cylinder in the transverse section (see 3.1.5),

b = common facewidth,

d_1 = reference diameter of pinion,

u = gear ration (for external gears u is positive, for internal gears u is negative)

Regarding factors K_A, K_γ, K_v, $K_{H\alpha}$ and $K_{H\beta}$ see 3.1.6.

(2) Permissible contact stress

The permissible contact stress σ_{HP} is to be evaluated separately for pinion and wheel:

$$\sigma_{HP} = (\sigma_{Hlim} \cdot Z_N / S_H) \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X$$

where:

Z_N	=	endurance limit for contact stress, (see 3.2.8),
σ_{Hlim}	=	life factor for contact stress, (see 3.2.9),
Z_L	=	lubrication factor, (see 3.2.10),
Z_v	=	speed factor, (see 3.2.10),
Z_R	=	roughness factor, (see 3.2.10),
Z_w	=	hardness ratio factor, (see 3.2.11),
Z_X	=	size factor for contact stress, (see 3.2.12),
S_H	=	safety factor for contact stress, (see 3.2.13)

3.2.3 Single pair mesh factors, Z_B and Z_D

The single pair mesh factors, Z_B for pinion and Z_D for wheel, account for the influence of the tooth flank curvature on contact stresses at the inner point of single pair contact in relation to Z_H. The factors transform the contact stresses determined at the pitch point to contact stresses considering the flank curvature at the inner point of single pair contact.

The single pair mesh factors Z_B for pinions and Z_D for wheels can be determined as follows:

For spur gears, $\varepsilon_\beta = 0$

Z_B = M_1 or 1 whichever is the larger value

Z_D = M_2 or 1 whichever is the larger value

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

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

For helical gears when $\varepsilon_\beta \geq 1$: $Z_B = Z_D = 1$

For helical gears when $\varepsilon_\beta < 1$, the values of Z_B and Z_D are determined by linear interpolation between Z_B and Z_D for spur gears and Z_B and Z_D for helical gears having $\varepsilon_\beta \geq 1$.

Thus:

$$Z_B = M_1 - \varepsilon_\beta \cdot (M_1 - 1) \text{ and } Z_B \geq 1$$

$$Z_D = M_2 - \varepsilon_\beta \cdot (M_2 - 1) \text{ and } Z_D \geq 1$$

For internal gears, Z_D shall be taken as equal to 1.

3.2.4 Zone factor, Z_H

The zone factor, Z_H, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and transforms the tangential load at the reference cylinder to the normal load at the pitch cylinder. The zone factor, Z_H, is to be calculated as follows:

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

3.2.5 Elasticity factor, ZE

The elasticity factor, Z_E, accounts for the influence of the material properties E (modulus of elasticity) and v (Poisson's ratio) on the contact stress. The elasticity factor Z_E for steel gears ($E = 206000\ N/mm^2, \mathrm{v} = 0{,}3$) is equal to $Z_E = 189{,}8$,. In other cases, reference is to be made to the reference standard ISO 6336-2:2019 as amended.

3.2.6 Contact ratio factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of the transverse contact ratio and the overlap ratio on the specific surface load of gears. The contact ratio factor, Z_ε, is to be calculated as follows:

Spur gears:

$$Z_\varepsilon = \sqrt{\frac{4 - \varepsilon_\alpha}{3}}$$

Helical gears:

– for $\varepsilon_\beta < 1$

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

– for $\varepsilon_\beta \geq 1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}}$$

3.2.7 Helix angle factor, Z_β

The helix angle factor, Z_β, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Z_β is dependent only on the helix angle. The helix angle factor, Z_β, is to be calculated as follows:

$$Z_\beta = \sqrt{\frac{1}{\cos\left(\beta\right)}}$$

Where β is the reference helix angle.

3.2.8 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. The value of σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least $50\cdot10^7$ load cycles. For this purpose, pitting is defined by:

- for not surface hardened gears:

  pitted area > 2% of total active flank area

- for surface hardened gears:

  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:

- material composition, cleanliness and defects,
- mechanical properties,
- residual stresses,
- hardening process, depth of hardened zone, hardness gradient,
- material structure (forged, rolled bar, cast).

The endurance limit for contact stress σ_{Hlim} is to be determined, in general, making reference to values indicated in the standard ISO 6336-5:2016 as amended, for material quality MQ.

3.2.9 Life factor, Z_N

The life factor, Z_N accounts for the higher permissible contact stress in case a limited life (number of cycles) is required. The factor mainly depends on:

- material and hardening,
- number of cycles,
- 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, as amended.

3.2.10 Influence factors of lubrication film on contact stress, Z_L, Z_v, and Z_R

The lubricant factor, Z_L, accounts for the influence of the type of lubricant and its viscosity. The velocity factor, Z_v, accounts for the influence of the pitch line velocity and the roughness factor, Z_R, accounts for the influence of the surface roughness on the surface endurance capacity. The factors may be determined for the softer material where gear pairs are of different hardness. The factors mainly depend on:

- viscosity of lubricant in the contact zone,
- the sum of the instantaneous velocities of the tooth surfaces,
- load,
- relative radius of curvature at the pitch point,
- surface roughness of teeth flanks,
- hardness of pinion and gear.

The lubricant factor Z_L, the velocity factor Z_v, and the roughness factor Z_R are to be calculated as follows:

(a) Lubricant factor Z_L

The factor, Z_L, is to be calculated from the following equation:

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

where:

v_{40} = nominal kinematic viscosity of the oil at 40°C, mm²/s

In the range 850 N/mm² ≤ σ_{Hlim} ≤ 1200 N/mm², C_{ZL} is to be calculated as follows:

$$C_{ZL} = \left(\frac{\sigma_{Hlim} - 850}{350} \cdot 0{,}08\right) + 0{,}83$$

If $\sigma_{Hlim} < 850\ N/mm^2$, take $C_{ZL} = 0{,}83$

If $\sigma_{Hlim} > 1200\ N/mm^2$, take $C_{ZL} = 0{,}91$

(b) Velocity factor, Z_V

The velocity factor, Z_V is to be calculated from the following equations

$$Z_V = C_{ZV} + \frac{2 \cdot (1 - C_{ZV})}{\sqrt{0{,}8 + 32/v}}$$

In the range 850 N/mm² ≤ σ_{Hlim} ≤ 1200 N/mm², C_{ZV} can be calculated as follows:

$$C_{ZV} = C_{ZL} + 0{,}02$$

(c) Roughness factor Z_R

The roughness factor, Z_R is to be calculated from the following equation:

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

where:

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

The peak-to-valley roughness determined for the pinion R_{Z1} and for the wheel R_{Z2} are mean values for the peak-to-valley roughness R_Z measured on several tooth flanks (R_Z as defined in the reference standard ISO 6336-2:2019 as amended).

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

relative radius of curvature:

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

where:

$\rho_{1,2}$= 0,5 $d_{b1,2}$ tan(α_{tw}), (also for internal gears, d_b negative sign)

If the roughness stated is an arithmetic mean roughness, i.e., R_a value (=CLA value) (=AA value) the following approximate relationship can be applied:

$$R_a = CLA = AA = R_Z/6$$

In the range 850 N/mm² ≤ σ_{Hlim} ≤ 1200 N/mm², C_{ZR} can be calculated as follows:

$$C_{ZR} = 0{,}32 - 0{,}0002 \cdot \sigma_{Hlim}$$

If $\sigma_{Hlim} < 850\ N/mm^2$, take $C_{ZR} = 0{,}150$

If $\sigma_{Hlim} > 1200\ N/mm^2$, take $C_{ZR} = 0{,}080$

3.2.11 Hardness ratio factor, Z_W,

The hardness ratio factor, Z_W, accounts for the increase of surface durability of a soft steel gear meshing with a significantly harder gear with a smooth surface in the following cases:

a) Surface-hardened pinion with through-hardened wheel

If HB< 130 $\quad Z_W = 1{,}2\left(\frac{3}{R_{zH}}\right)^{0{,}15}$

If 130 ≤ HB ≤ 470 $\quad Z_W = \left(1{,}2 - \frac{HB-130}{1700}\right)\left(\frac{3}{R_{zH}}\right)^{0{,}15}$

If HB >470 $\quad Z_W = \left(\frac{3}{R_{zH}}\right)^{0{,}15}$

where:

HB = Brinell hardness of the tooth flanks of the softer gear of the pair

R_{zH} = equivalent roughness, µm

$$R_{zH} = \frac{R_{z1}(10/\rho_{red})^{0{,}33}(R_{z1}/R_{z2})^{0{,}66}}{(v \cdot v_{40}/1500)^{0{,}33}}$$

ρ_{red} = relative radius of curvature (see 3.2.10 c)

b) 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.

If $HB_1/HB_2 < 1{,}2$ $Z_w = 1$

If $1{,}2 \leq HB_1/HB_2 \leq 1{,}7$ $Z_w = 1 + \left(0{,}00898\frac{HB_1}{HB_2} - 0{,}00829\right)(u-1)$

If $HB_1/HB_2 > 1{,}7$ $Z_w = 1 + 0{,}00698\,(u-1)$

If gear ratio u>20 then the value u=20 is to be used.

In any case, if calculated Z_W <1 then the value Z_W = 1.0 is to be used.

3.2.12 Size factor, Z_X

The size factor, Z_X, accounts for the influence of tooth dimensions on permissible contact stress and reflects the non-uniformity of material properties. The factor mainly depends on:

- material and heat treatment,
- tooth and gear dimensions,
- ratio of case depth to tooth size,
- ratio of case depth to equivalent radius of curvature.

For through-hardener gears and for surface-hardened gears with adequate casedepth relative to tooth size and radius of relative curvature Z_X = 1. When the casedepth is relatively shallow then a smaller value of Z_X should be chosen.

3.2.13 Safety factor for contact stress, S_H

The safety factor for contact stress, S_H, can be assumed by LHR taking into account the type of application. The following guidance values can be adopted:

- Main propulsion gears: $1{,}20\ to 1{,}40$
- Auxiliary gears: $1{,}15\ to 1{,}20$

For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of LHR.

3.3 Tooth root bending strength

3.3.1 Scope and general remarks

The criterion for tooth root bending strength is the permissible limit of local tensile strength in the root fillet. The root stress σ_F and the permissible root stress σ_{FP} shall be calculated separately for the pinion and the wheel. σ_F must not exceed σ_{FP}. The following formulae and definitions apply to gears having rim thickness greater than 3.5 m_n.

The result of rating calculations made by following this method are acceptable for normal pressure angles up to 25° and reference helix angles up to 30°. For larger pressure angles and large helix angles, the calculated results should be confirmed by experience as by Method A of the reference standard ISO 6336-3:2019 as amended.

3.3.2 Basic equations

(1) Tooth root bending stress for pinion and wheel

$$\sigma_F = (F_t/(b \cdot m_n)) \cdot (Y_F \cdot Y_S \cdot Y_\beta \cdot Y_B \cdot Y_{DT} \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta}) \leq \sigma_{FP}$$

where:

Y_F = tooth form factor, (see 3.3.3),

Y_S = stress correction factor, (see 3.3.4),

Y_β = helix angle factor, (see 3.3.5),

Y_B = rim thickness factor (see 3.3.6),

Y_{DT}= deep tooth factor (see 3.3.7),

F_t, K_A, K_γ, K_V, $K_{F\alpha}$, $K_{F\beta}$, (see 3.1.6),

b= (see 3.1.4),

m_n = (see 3.1.3).

(2) Permissible tooth root bending stress for pinion and wheel

$$\sigma_{FP} = (\sigma_{FE} \cdot Y_d \cdot Y_N / S_F) \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X$$

where:

σ_{FE}	=	bending endurance limit, (see 3.3.8),
Y_d	=	design factor, (see 3.3.9),
Y_N	=	life factor, (see 3.3.10),
$Y_{\delta relT}$	=	relative notch sensitivity factor, (see 3.3.11),
Y_{RrelT}	=	relative surface factor, (see 3.3.12),
Y_X	=	size factor, (see 3.3.13),
S_F	=	safety factor for tooth root bending stress (see 3.3.14).

3.3.3 Tooth form factor, Y_F

The tooth form factor, Y_F, represents the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F shall be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing shall be determined in the normal section, i.e. for the virtual spur gear with virtual number of teeth, z_n.

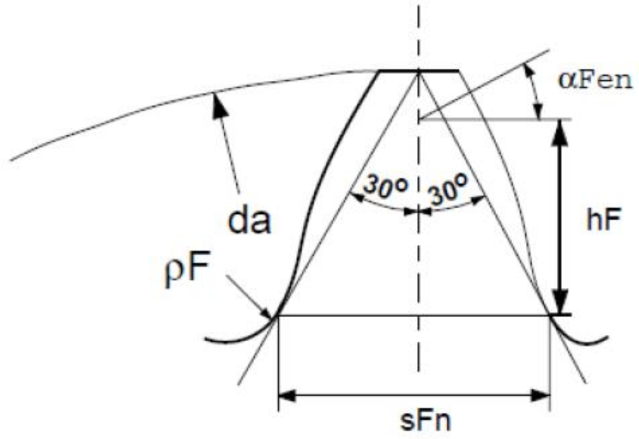
The tooth form factor Y_F can be calculated as follows:

$$Y_F = \frac{6 \cdot \frac{h_F}{m_n} \cdot \cos(\alpha_{Fen})}{\left(\frac{s_{Fn}}{m_n}\right)^2 \cdot \cos(\alpha_n)}$$

where:

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 normal chord in the critical section, mm
α_{Fen}	=	pressure angle at the outer point of single tooth pair contact in the normal section, degrees

Figure 4.3.3 **Dimensions of h_F, s_{Fn} and α_{Fen} for external gear**



For the calculation of h_F, s_{Fn} and α_{Fen}, the procedure outlined in the reference standard ISO 6336-3:2019 (Method B) as amended, is to be used.

3.3.4 Stress correction factor, Y_S

The stress correction factor, Y_S, is used to convert the nominal bending stress to the local tooth root stress, taking into account that not only bending stresses arise at the root. Y_S applies to the load application at the outer point of single tooth pair contact. Y_S shall be determined separately for the pinion and for the wheel.

The stress correction factor, Y_S, is to be determined with the following equation (having range of validity: $1 \le q_s \le 8$):

$$Y_S = (1{,}2 + 0{,}13L) q_s^{\left(\frac{1}{1{,}21 + 2{,}3/L}\right)}$$

where:

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

q_S = notch parameter

ρ_F = root fillet radius in the critical section, mm

L = s_{Fn}/h_f

For h_F and s_{Fn} see clause 3.3.3.

For the calculation of ρ_F the procedure outlined in the reference standard ISO 6336-3:2019, as amended, is to be used.

3.3.5 Helix angle factor, Y_β

The helix angle factor, Y_β, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth. The helix angle factor, Y_β, is to be calculated as follows:

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

Where:

β = reference helix angle in degrees.

The value one (1,0) is substituted for ε_β when ε_β > 1,0, and 30° is substituted for β > 30°.

3.3.6 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 should be replaced by a more comprehensive analysis. Factor Y_B is to be determined as follows:

a) for external gears:

If $s_R/h \geq 1{,}2$ $\quad\quad$ $\mathrm{Y_B} = 1$

If $0{,}5 < s_R/h < 1{,}2$ $\quad\quad$ $\mathrm{Y_B} = 1{,}6 \cdot \ln\left(2{,}242 \frac{h}{s_R}\right)$

where:

s_R = rim thickness of external gears, mm

h = tooth height, mm

The case $s_R/h \leq 0{,}5$ id to be avoided.

b) for internal gears

If $s_R/m_n \geq 3{,}5$ $\quad\quad$ $\mathrm{Y_B} = 1$

If $1{,}75 < s_R/m_n < 3{,}5$ $\quad\quad$ $\mathrm{Y_B} = 1{,}15 \cdot \ln\left(8{,}324 \frac{m_n}{s_R}\right)$

where:

s_R = rim thickness of internal gears, mm

The case $s_R/m_n \leq 1{,}75$ is to be avoided.

3.3.7 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 ≤ ε_{an} ≤ 2,5 , where:

$$\varepsilon_{an} = \frac{\varepsilon_a}{cos^2 \beta_b}$$

Factor Y_{DT} is to be determined as follows:

If ISO accuracy grade ≤ 4 and $\varepsilon_{an} > 2{,}5$ $\quad\quad$ $Y_{DT} = 0{,}7$

If ISO accuracy grade ≤ 4 and $2{,}05 < \varepsilon_{an} \leq 2{,}5$ $\quad\quad$ $Y_{DT} = 2{,}366 - 0{,}666 \cdot \varepsilon_{an}$

In all other cases $\quad\quad$ $Y_{DT} = 1{,}0$

3.3.8 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 as amended, the number of $3 \cdot 10^6$ cycles is regarded as the beginning of the endurance limit. σ_{FE} is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d. The σ_{FE} values are to correspond to a failure probability 1% or less. The endurance limit mainly depends on:

- material composition, cleanliness and defects,
- mechanical properties,
- residual stresses,

– hardening process, depth of hardened zone, hardness gradient,
– material structure (forged, rolled bar, cast).

The bending endurance limit, σ_{FE} is to be determined, in general, making reference to values indicated in the reference standard ISO 6336-5:2016 as amended, for material quality MQ.

3.3.9 Design factor, Y_d

The design factor, Y_d, takes into account the influence of load reversing and shrinkfit prestressing on the tooth root strength, relative to the tooth root strength with unidirectional load as defined for σ_{FE}. The design factor, Y_d for load reversing, can be determined as follows:

Y_d = 1,0 in general

Y_d = 0,9 for gears with occasional part load in reverse direction, such as main wheel in reversing gear boxes

Y_d = 0,7 for idler gears

3.3.10 Life factor, Y_N

The life factor, Y_N, accounts for the higher tooth root bending stress permissible in case a limited life (number of cycles) is required. The factor mainly depends on:

– material and heat treatment,
– number of load cycles (service life),
– influence factors ($Y_{\delta relT}$, Y_{RrelT}, Y_X).

The life factor, Y_N, is to be determined according to Method B outlined in the reference standard ISO 6336-3:2019 as amended.

3.3.11 Relative notch sensitivity factor, $Y_{\delta relT}$

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit. The factor mainly depends on material and relative stress gradient.

The relative notch sensitivity factor, $Y_{\delta relT}$, is to be determined as follows:

$$Y_{\delta\mathrm{relT}} = \frac{1 + \sqrt{0{,}2\rho'(1 + 2q_s)}}{1 + \sqrt{1{,}2\rho'}}$$

where:

q_s = notch parameter (see 3.3.4)

ρ' = slip-layer thickness, mm, from the following table:

Material		ρ', mm
Case hardened steels, flame or induction hardened steels		0,0030
Through-hardened steels [(1)], yield point R_e=	500 N/mm²	0,0281
	600 N/mm²	0,0194
	800 N/mm²	0,0064
	1000 N/mm²	0,0014
Nitrided steels		0,1005
Note: 1. The given values of ρ' can be interpolated for values of R_e not stated above		

3.3.12 Relative surface factor, Y_{RrelT}

The relative surface factor, Y_{RrelT}, takes into account the dependence of the root strength on the surface condition in the tooth root fillet, mainly the dependence on the peak to valley surface roughness.

The relative surface factor, Y_{RrelT} is to be determined as follows:

R_Z < 1	**1 ≤ R_Z ≤ 40**	Material
1,120	$1{,}674-0{,}529 \cdot (R_Z+1)^{0,1}$	case hardened steels, through-hardened steels ($\sigma_B \geq 800$ N/mm²)
1,070	$5{,}306-4{,}203 \cdot (R_Z+1)^{0,01}$	normalized steels ($\sigma_B < 800$ N/mm²)
1,025	$4{,}299-3{,}259 \cdot (R_Z+1)^{0,0058}$	nitrided steels

where:

R_Z = mean peak-to-valley roughness of tooth root fillets, µm

σ_B = tensile strength, N/mm²

The method applied here is only valid when scratches or similar defects deeper than $2R_Z$ are not present. If the roughness stated is an arithmetic mean roughness, i.e., R_a value (=CLA value) (=AA value) the following approximate relationship can be applied:

$$\mathrm{Ra} = \mathrm{CLA} = \mathrm{AA} = \mathrm{R_Z} / 6$$

3.3.13 Size factor, Y_X

The size factor, Y_X, takes into account the decrease of the strength with increasing size. The factor mainly depends on:

– material and heat treatment,
– tooth and gear dimensions,
– ratio of case depth to tooth size.

The size factor, Y_X is to be determined as follows:

$Y_X = 1{,}00$	For $m_n \leq 5$	generally
$Y_X = 1{,}03 - 0{,}06 \cdot m_n$	For $5 < m_n < 30$	normalised and through - hardened steels
$Y_X = 0{,}85$	For $m_n \geq 30$	
$Y_X = 1{,}05-0{,}010 \cdot m_n$	For $5 < m_n < 25$	surface hardened steels
$Y_X = 0{,}80$	For $m_n \geq 25$	

3.3.14 Safety factor for tooth root bending stress, S_F.

The safety factor for tooth root bending stress, S_F, can be assumed by LHR taking into account the type of application.

The following guidance values can be adopted:

– Main propulsion gears: 1,55 to 2,00
– Auxiliary gears: 1,40 to1,45

For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of LHR.

SECTION 4 Tests

4.1 Balancing

4.1.1 Pinions, gear wheels and flexible couplings whose maximum rotational speed exceeds 1000 rpm are to be dynamically balanced. Components whose rotational speed is less or equal to 1000 rpm are to be statically or, alternatively, dynamically balanced. Relaxations to the balancing requirements may be allowed where the pinions, gear wheels or flexible couplings have small dimensions or where they are solid forged and entirely machined.

4.1.2 The maximum permissible residual imbalance I per balancing plane of gears for which static or dynamic balancing is required is to be determined by the formula:

$$I = \frac{m \cdot b}{r \cdot N} \quad [kg \cdot mm]$$

where:

m = mass of the body to be balanced, kg,

b = 60, for gear shafts, pinions and coupling members for engine gears;

= 24, for torsion shafts and gear couplings, pinions and gear wheels involved to turbine transmissions

r = speed of rotation of the body to be balanced, rpm,

N = number of balancing planes.

4.2 Testing of gears

4.2.1 After the testing of the materials and component tests gearing systems used for the main propulsion and for important auxiliary services as prescribed in Part 5, Chapter 1, are to be inspected and tested during operation by LHR in the manufacturer's works or, if it is impracticable, on-board ship. The final inspection is to be followed by a trial run of adequate duration under part or full-load conditions, on which occasion the tooth clearance and contact pattern are to be checked. Where necessary tightness tests are to be carried out.

4.2.2 Tests during sea trials

(1) Before the commencement of sea trials, the teeth of the gears of the main propulsion plant should be properly marked in order to enable the contact pattern to be established.
(2) During the sea trials, the correct operation of the gears are to be checked over the whole range of speed values.
(3) After the end of the sea trials, the gears should be inspected. Due care is to be given to the contact pattern.
(4) During the sea trials the bearing temperatures and the freedom from contamination of the lubricating oil should be checked.

SECTION 5 Design and construction of couplings

5.1 Tooth couplings

5.1.1 Adequate loading capacity of the tooth flanks of straight-flanked tooth couplings is to be provided.

5.1.2 Adequate lubrication of the coupling teeth are to be ensured. Where the following condition is satisfied:

$$d \cdot r^2 < 6 \cdot 10^9, \mathrm{mm/min^2}$$

where:

d = pitch circle diameter, mm,

r = rotational speed, rpm,

a constant oil level maintained in the coupling may be regarded as adequate. For higher values of $d\ r^2$, the existence of a circulating system of lubrication is required.

5.1.3 The design of the sleeves, flanges and bolts of gear couplings are to be in accordance with Part 5, Chapter 4b.

5.2 Flexible couplings

5.2.1 Flexible couplings are to be so constructed that they have sufficient strength to transmit the load and is capable of being operated safely and efficiently.

5.2.2 Flexible couplings should be so designed in order to operate with one engine cylinder out of service for a reasonable time interval. For ships with ice classes, additional dynamic loads are to be taken into account.

5.2.3 LHR reserves the right to call for the performance of special tests where it is required.

5.2.4 The casings, flanges and bolts of flexible couplings are to comply with the requirements specified in Part 5, Chapter 4b, SECTION 3, 3.6.

5.2.5 Where a flexible coupling produces an axial thrust on the coupled members special care is to be taken for the absorption of this thrust.

5.2.6 Flexible couplings with elastomer or spring type flexible members and which represent the sole source of transmitting propulsive power such as in a line shaft on a single screw vessel are to be provided with torsional limit arrangement (coupling will lock beyond its limit) or positive means of locking the coupling for emergency operation.

5.3 Flange and clamp-type couplings

5.3.1 The design of the coupling bodies, flanges and bolts of flange and clamp-type couplings is to be in accordance with the Rules specified in Part 5, Chapter 4b.

CHAPTER 4 (b) Propulsion Shafting

CONTENTS

SECTION 1 General

1.1 Scope

1.1.1 The requirements of this Chapter are applicable to normal and accepted types of main shafting. In the present Chapter formulas for determining the diameters for shafting for main propulsion installations are included along with requirements for the design of sternbushes, keys, keyways, couplings, coupling bolts and other relevant parts. Any modification to the diameters determined from the formulas of the present Chapter due to alignment requirements is acceptable.

1.2 Documents for approval

1.2.1 The following plans are to be submitted to LHR for consideration before the commencement of the work:

- general arrangement of the entire shafting from main engine coupling to propeller,
- detail drawing of the final gear shaft,
- detail drawing of the thrust shaft,
- detail drawing of the intermediate shafting,
- drawing of the tube shaft, where applicable,
- drawings of the screwshaft and screwshaft oil gland
- drawing of the sternbursh.

1.2.2 The following data are also to be submitted for approval:

- material specifications,
- strength calculations,
- vibration calculations.

1.2.3 LHR reserves the right to ask for any additional information.

SECTION 2 Materials

2.1 Materials for shafts (IACS UR M68.1 Rev.3 (2021))

2.1.1 This Section applies to propulsion shafts such as intermediate and propeller shafts of traditional straight forged design and which are driven by rotating machines such as diesel engines, turbines or electric motors.

2.1.2 For shafts that are integral to equipment, such as for gear boxes, podded drives, electrical motors and/or generators, thrusters, turbines and which in general incorporate particular design features, additional criteria in relation to acceptable dimensions have to be taken into account. For the shafts in such equipment, the requirements of this Section may only be applied for shafts subject mainly to torsion and having traditional design features. Other limitations, such as design for stiffness, high temperature etc. are to be addressed by specific rules of LHR.

2.1.3 Explicitly the following applications are not covered by this Section:

- additional strengthening for shafts in ships classed for navigation in ice
- gearing shafts
- electric motor shafts
- generator rotor shafts
- turbine rotor shafts

– diesel engine crankshafts (see Part 5, Chapter 2, SECTION 3)
– unprotected shafts exposed to sea water

2.2 Material limitations (IACS UR M68.3 Rev.3 (2021))

2.2.1 Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the materials are to have a specified minimum ultimate tensile strength (σ_B) of 500 N/mm². Otherwise, materials having a specified minimum ultimate tensile strength (σ_B) of 400 N/mm² may be used.

2.2.2 For use in the following formulae in this Section, σ_B is limited as follows:

– For carbon and carbon manganese steels, a minimum specified tensile strength not exceeding 600 N/mm² for use in Part 5, Chapter 6, SECTION 2, 2.4.3 and not exceeding 760 N/mm² in 3.1.
– For alloy steels, a minimum specified tensile strength not exceeding 800 N/mm².
– For propeller shafts in general a minimum specified tensile strength not exceeding 600 N/mm² (for carbon, carbon manganese and alloy steels).

2.2.3 Where materials with greater specified or actual tensile strengths than the limitations given above are used, reduced shaft dimensions or higher permissible vibration stresses are not acceptable when derived from the formulae in this Section unless LHR verifies that the materials exhibit similar fatigue life as conventional steels (see 2.6 of this Chapter).

2.3 Ultrasonic tests

2.3.1 Ultrasonic tests are required on shaft forgings where the diameter is 250 mm or greater.

2.4 Alternative calculation methods (IACS UR M68.2 Rev.3 (2021))

2.4.1 Alternative calculation methods may be considered by LHR. Any alternative calculation method is to include all relevant loads on the complete dynamic shafting system under all permissible operating conditions. Consideration is to be given to the dimensions and arrangements of all shaft connections.

2.4.2 Moreover, an alternative calculation method is to take into account design criteria for continuous and transient operating loads (dimensioning for fatigue strength) and for peak operating loads (dimensioning for yield strength). The fatigue strength analysis may be carried out separately for different load assumptions, for example as given in 2.5.1.

2.5 Notes (IACS UR M68.7 Rev.3 (2021))

2.5.1 Shafts complying with this Section satisfy the following:

(a) Low cycle fatigue criterion (typically < 10^4), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque if applicable.

This is addressed by the formula in 3.1.

(b) High cycle fatigue criterion (typically >> 10^7), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses.

The limits for torsional vibration stresses are given in Part 5, Chapter 6, SECTION 2, 2.4.3.

The influence of reverse bending stresses is addressed by the safety margins inherent in the formula in 3.1.

(c) The accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation is addressed by the criterion for transient stresses in Part 5, Chapter 6, SECTION 2, 2.4.3.

2.5.2 Explanation of k and c_K.

The factors k (for low cycle fatigue) and c_K (for high cycle fatigue) take into account the influence of:

(a) The stress concentration factors (scf) relative to the stress concentration for a flange with fillet radius of $0{,}08 \cdot d_o$ (geometric stress concentration of approximately 1,45).

$$c_k \approx \frac{1{,}45}{scf} \text{ and } k \approx \left[\frac{scf}{1{,}45}\right]^x$$

where the exponent x considers low cycle notch sensitivity.

(b) The notch sensitivity. The chosen values are mainly representative for soft steels ($\sigma_B < 600$), while the influence of steep stress gradients in combination with high strength steels may be underestimated.
(c) The size factor c_D being a function of diameter only does not purely represent a statistical size influence, but rather a combination of this statistical influence and the notch sensitivity.

The actual values for k and c_K are rounded off.

2.5.3 Stress concentration factor of slots

The stress concentration factor (scf) at the end of slots can be determined by means of the following empirical formulae using the symbols in 3.1.2, NOTE 6:

$$scf = \alpha_{t(hole)} + 0{,}8 \cdot \frac{(l-e)/d}{\sqrt{\left(1-\frac{d_i}{d}\right)\cdot\frac{e}{d}}}$$

This formula applies to:

– slots at 120 or 180 or 360 degrees apart.
– slots with semi-circular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.
– slots with no edge rounding (except chamfering), as any edge rounding increases the scf slightly.

$\alpha_{t(hole)}$ represents the stress concentration of radial holes (in this context e = hole diameter) and can be determined as:

$$\alpha_{t(hole)} = 2{,}3 - 3\cdot\frac{e}{d} + 15\cdot\left(\frac{e}{d}\right)^2 + 10\cdot\left(\frac{e}{d}\right)^2\cdot\left(\frac{d_i}{d}\right)^2$$

or simplified to $\alpha_{t(hole)} = 2{,}3$

2.6 Special approval of alloy steel used for intermediate shaft material (IACS UR M68 Appendix I, Rev.3 (2021))

2.6.1 Application

Paragraph 2.6 is applied to the approval of alloy steel which has a minimum specified tensile strength greater than 800 N/mm^2, but less than 950 N/mm^2 intended for use as intermediate shaft material.

2.6.2 Torsional fatigue test

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 permissible torsional vibration stress τ_C given by the formulae in Chapter 6, SECTION 2, 2.4 of this Part.

The test is to be carried out with notched and unnotched specimens respectively. For calculation of the stress concentration factor of the notched specimen, fatigue strength reduction factor β should be evaluated in consideration of the severest torsional stress concentration in the design criteria.

2.6.3 Test conditions

Test conditions are to be in accordance with Table 4b.21. Mean surface roughness is to be <0,2μm Ra with the absence of localised machining marks verified by visual examination at low magnification (x20) as required by Section 8.4 of ISO 1352:2011, as amended. Test procedures are to be in accordance with Section 10 of ISO 1352:2011, as amended.

Table 4b.2.1: Test condition

Loading type	Torsion
Stress ratio	R=-1
Load waveform	Constant-amplitude sinusoidal
Evaluation	S-N curve
Number of cycles for test termination	$1x10^7$ cycles

2.6.4 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:

$$\tau_{C1} \geq \tau_{C,\lambda=0} = \frac{\sigma_B + 160}{6} C_K \, C_D$$

$$\tau_{C2} \geq 1{,}7 \frac{1}{\sqrt{C_K}} \tau_{C1}$$

where:

C_K = factor for the particular shaft design features, see 2.5 of this SECTION

scf = stress concentration factor, see 2.5.1(c) of this SECTION (For unnotched specimen, 1.0.)

C_D = size factor, see Chapter 6, SECTION 2, 2.4

σ_B = specified minimum tensile strength in N/mm² of the shaft material

2.6.5 Cleanliness requirements

The steels are to have a degree of cleanliness as shown in Table 4b.2.2 when tested according to ISO 4967:2013 method A, as amended. Representative samples are to be obtained from each heat of forged or rolled products.

The steels are generally to comply with the minimum requirements of Part 2, Chapter 5, SECTION 2m Table 5.2.2, 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 LHR.

Table 4b.2.2: 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

2.6.6 Inspection

The ultrasonic testing required by Part 2, Chapter 5, SECTION 2 is to be carried out prior to acceptance. The acceptance criteria are to be in accordance with IACS Recommendation No. 68 or a recognized national or international standard.

SECTION 3 Design

3.1 Shaft diameters (IACS UR M68.4, M68.6, Rev.3 (2021))

3.1.1 Shaft diameters are not to be less than that calculated from the following formula:

$$d = F \cdot k \cdot \sqrt[3]{\frac{P}{n_o} \cdot \frac{1}{1 - \left(\frac{d_i}{d_o}\right)^4} \cdot \frac{560}{\sigma_B + 160}}$$

where:

d = minimum required diameter, mm

d_i = actual diameter of shaft bore, mm

d_o = outside diameter of shaft, mm. If the bore of the shaft is ≤ $0{,}40 \cdot d_o$, the expression $1 - \left(\frac{d_i}{d_o}\right)^4$ may be taken as 1,0

F = factor for the type of propulsion installation,

= 95 for intermediate shafts in turbine installation, diesel installation with hydraulic (slip type) coupling, electric propulsion installation;

= 100 for all other diesel installations and all propeller shafts;

k = factor for the particular shaft design features, see 3.1.2,

n_o = speed of shaft at rated power, rpm,

P = rated power transmitted through the shaft (losses in gearboxes and bearings are to be disregarded), kW,

σ_B = specified minimum tensile strength of the shaft material, N/mm², see 2.2.

The diameter of the propeller shaft located forward of the inboard stern tube seal may be gradually reduced to the corresponding diameter required for the intermediate shaft using the minimum specified tensile strength of the propeller shaft in the formula and recognizing any limitations given in 2.2.

3.1.2 k and c_K factors for different shaft design features (see 2.5.2) (IACS UR M68.6)

(1)

for intermediate shafts with						**for thrust shafts external to engines**		**propeller shafts**		
Integral coupling flange (Note 1) and straight sections	Shrink fit coupling (Note 2)	Keyway, tapered connection (3) (4)	Keyway, cylindrical connection (Note 3,4)	radial hole (Note 5)	Longitudinal slot (Note 6)	On both sides of thrust collar (Note 1)	In way of bearing when a roller bearing is used	Flange mounted or keyless taper fitted propellers (Note 8)	Key fitted propellers (Note 8)	Between forward end of aft most bearing and forward stern tube seal
k= 1.0	1.0	1.10	1.10	1.10	1.20	1.10	1.10	1.22	1.26	1.15
c_K = 1.0	1.0	0.60	0.45	0.50	0.30 (7)	0.85	0.85	0.55	0.55	0.80

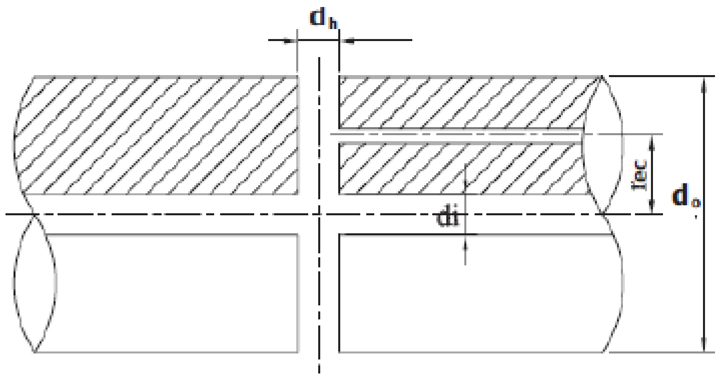
Transitions of diameters are to be designed with either a smooth taper or a blending radius. For guidance, a blending radius equal to the change in diameter is recommended.

NOTES:

1. Filet radius is not less than 0,08·d.
2. k and c_K refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter is to be provided, e.g. a diameter increase of 1 to 2 % and a blending radius as described in Note (3).
3. At a distance of not less than 0,2·d_o from the end of the keyway the shaft diameter may be reduced to the diameter calculated with k=1,0.
4. Keyways are in general not to be used in installations with a barred speed range.

5. Diameter of radial bore (d_h) not to exceed $0,3 \cdot d_o$.

   The intersection between a radial and an eccentric (r_{ec}) axial bore (see below) is not covered by this Section.



6. Subject to limitations as slot length (l)/outside diameter < 0,8 and inner diameter (d_i)/outside diameter < 0,7 and slot width (e)/outside diameter > 0,15. The end rounding of the slot is not to be less than e/2. An edge rounding should preferably be avoided, as this increases the stress concentration slightly.
   The k and c_K values are valid for 1, 2 and 3 slots, i.e. with slots at 360° respectively 180° and respectively 120° apart.
7. $c_K = 0,3$ is an approximation within the limitations in (6). More accurate estimate of the stress concentration factor (scf) may be determined from 2.5.3 of this Chapter, or by direct application of FE calculation. In which case:

$$c_K = 1,45/scf$$

   Note that the scf is defined as the ration between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress (determined for the bored shaft without slots).
8. Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2,5 times the required diameter.

(2) The determination of k-factors for shaft design features other than those given above is left to the discretion of LHR.

(3) Transitions of diameters are to be designed with either a smooth taper or a blending radius. For guidance, a blending radius equal to the change in diameter is recommended.

3.2 Gear quill shafts

3.2.1 The diameter d_q of the quill shaft is to be not less than given by the following formula:

$$d_q = 754 \cdot \sqrt{\frac{P}{n \cdot \sigma_B}} \quad [mm]$$

where:

P, n are as defined in Part 5, Chapter 1, SECTION 3, 3.6,

σ_B = specified minimum tensile strength of the material, in N/mm^2 but is not to exceed 1100 N/mm^2.

3.3 Final gear wheel shafts

3.3.1 The diameter of the shaft at the final wheel and the adjacent journals is to be greater than 1,15 times that required for the intermediate shaft, where there is only one pinion geared into the final wheel, or where there are two pinions which are set to subtend an angle at the centre of the shaft of less than 120 degrees.

3.3.2 The diameter of the shaft at the final wheel and the adjacent journals is to be greater than 1,10 times that required for the intermediate shaft, where there are two pinions geared into the final wheel opposite, or nearly opposite, to each other.

3.3.3 In 3.3.1 and 3.3.2, abaft the journals, the shaft may be gradually tapered down to the diameter required for an intermediate shaft determined according to 3.1 where $_B$ is to be taken as the specified minimum tensile strength of the final wheel shaft material, in N/mm^2.

3.4 Thrust shafts

3.4.1 The diameter at the collars of the thrust shaft is to be not less than that required for the intermediate shaft as follows from 3.1 with a k-factor value of 1,10.

3.4.2 Outside a length equal to the thrust shaft diameter from the collars, the diameter may be gradually reduced to that required for the intermediate shaft with a k-factor value of 1,0.

3.4.3 In the above calculations $_B$ is to be taken as the minimum tensile strength of the thrust shaft material, in N/mm^2.

3.5 Scantlings of coupling flanges (IACS M34 (1980))

3.6.1 For intermediate, thrust and propeller shaft couplings having all fitted coupling bolts, the coupling bolt diameter is to be not less than that given by the following formula:

$$d_b = 0{,}65 \sqrt{\frac{d^3\,(T + 160)}{iD\ T_b}}$$

where:

d_b = diameter of fitted coupling bolt, mm,
d = Rule diameter, i.e. minimum required diameter of intermediate shaft made of material with tensile strength T, taking into account ice strengthening requirements where applicable, mm,
i = number of fitted coupling bolts,
D = pitch circle diameter of coupling bolts, mm
T = tensile strength of the intermediate shaft material taken for calculation, N/mm^2,
T_b = tensile strength of the fitted coupling bolts material taken for calculation, N/mm^2, while:
$T \le T_b \le 1{,}7 \cdot T$, but not higher than 1000 N/mm^2.

3.6.2 The designs of coupling bolts in the shaftline other than that covered by 3.6.1 are to be considered and approved by LHR.

3.6.3 For intermediate shafts, thrust shafts and inboard end of propeller shafts the flange is to have a minimum thickness of 0,20 times the Rule diameter d of the intermediate shaft or the thickness of the coupling bolt diameter calculated for the material having the same tensile strength as the corresponding shaft, whichever is greater. Special consideration will be given by LHR for flanges having non-parallel faces, but in no case is the thickness of the flange to be less than the coupling bolt diameter.

3.6.4 Fillet radii at the base of the flange should in each case be not less than 0,08 times the actual shaft diameter. Fillets are to have a smooth finish and should not be recessed in way of nuts and bolt heads. The fillet may be formed of multiradii in such a way that the stress concentration factor will not be greater than that for a circular fillet with radius 0,08 times the actual shaft diameter.

3.6 Bronze or gunmetal liners on shafts

3.7.1 The thickness, of liners fitted on screwshafts or on tube shafts, in way of the bushes, is to be greater than:

$$t = \frac{D + 230}{32}$$

where:

t = thickness of the liner, mm

D = diameter of the screwshaft or tube shaft under the liner, mm.

3.7.2 The thickness of a continuous liner between the bushes is to be not less than 75 per cent of t, where t is given in 3.7.1.

3.7.3 Normally, continuous liners should be cast in one piece. Where this is impracticable, they are to be built welded together by using leed-free electrodes or filler rods. The weldings are to be to the Surveyor's satisfaction.

3.7.4 Due care is to be taken in order to ensure the water-tightness of the part of the shaft between the after end of the liner and the propeller boss.

3.7.5 The composition of the gunmetal of each length forming a butt-welded liner should be such that the lead content is not to exceed 1 per cent.

3.7.6 The use of pins for securing the liners is not allowed. Liners are to be shrunk on, or forced on, to the shafts by hydraulic pressure.

3.7.7 Every continuous liner or length of liner is to be tested by hydraulic pressure to 2,0 bar after rough machining.

3.7 Keys and keyways

3.8.1 The keyways in the propeller boss and cone of the screwshaft are to be provided with a smooth fillet at their bottom. The keys should be round or sled-runner ended. The edges at the top of the keyways are also to be smooth. The forward end of the keyway is to be so cut in the shaft as to give a gradual rise from the bottom of the keyway to the surface of the shaft.

3.8.2 In general, due care is to be taken for the reduction of the stress concentration as far as practicable.

3.8.3 The key is to fit tightly and safely in the keyway. To this purpose, two screwed pins are to be provided for securing the key in the keyway, and the forward pin is to be placed at least one-third of the length of the key from the end. The depth of the tapped holes for the screwed pins is not to exceed the pin diameter.

3.8.4 The distance between the top of the cone and the forward end of the key way is to be not less than 0,2 of the diameter of the screwshaft at the top of the cone.

3.8.5 The key is to be of sufficient size to transmitt the full torque of the shaft.

3.8 Length of aft stern bush bearing (IACS M52 Rev.2 (2019))

3.9.1 Oil lubricated bearings of white metal

(1) The length of white metal lined bearings is to be not less than 2,0 times the rule diameter of the shaft in way of the bearing.
(2) The length of the bearing may be less provided the nominal bearing pressure is not more than 8 bar as determined by 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. However, the minimum length is to be not less than 1,5 times the actual diameter.

3.9.2 Oil lubricated bearings of synthetic rubber, reinforced resin or plastic materials

(1) For bearings of synthetic rubber, reinforced resin or plastics materials which are approved for use as oil lubricated stern bush bearings, the length of the bearing is to be not less than 2,0 times the rule diameter of the shaft in way of the bearing.
(2) The length of bearing may be less provided the nominal bearing pressure is not more than 6 bar as determined by 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. However, the minimum length is to be not less than 1,5 times the actual diameter.
(3) Where the material has proven satisfactory testing and operating experience, consideration may be given to an increased bearing pressure.
(4) Synthetic materials for application as oil lubricated stern tube bearings are to be Type Approved.

3.9.3 Water lubricated bearings

(1) The length of the bearing is to be not less than 4,0 times the rule diameter of the shaft in way of the bearing.

(2) For a bearing of synthetic material, consideration may be given to a bearing length not less than 2,0 times the rule diameter of the shaft in way of the bearing, provided the bearing design and material is substantiated by experiments to the satisfaction of LHR.
(3) Synthetic materials for application as water lubricated stern tube bearings are to be Type Approved.

3.9.4 Grease lubricated bearings

(1) The length of a grease lubricated bearing is to be not less than 4,0 times the rule diameter of the shaft in way of the bearing.

3.9 Vibration and alignment

3.10.1 For the requirements for torsional, axial and lateral vibration, and for alignment of the shafting, see Part 5, Chapter 6.

CHAPTER 5 Propellers

CONTENTS

SECTION 1 General

1.1 Documents for Approval

1.1.1 A plan, in triplicate, of the propeller is to be submitted for approval to LHR. The following particulars, using the symbols shown, are also to be submitted:

(1) Propeller diameter, D, m.
(2) Number of blades, N.
(3) Revolutions per minute of the propeller at maximum power, n, rpm.
(4) Maximum shaft power (see Part 5, Chapter 1, SECTION 3, 3.6), P, kW.
(5) Length of blade section of the expanded cylindrical section at 25% radius (for solid propellers only), l_{25}, mm.
(6) Length of blade section of the expanded cylindrical section at 35% radius (for controllable pitch propellers only), l^{35}, mm.
(7) Length of blades section of the expanded cylindrical section at 60% radius, l_{60}, mm.
(8) Pitch at 25% radius (for solid propellers only), p_{25}, m.
(9) Pitch at 35% radius (for controllable pitch propellers only), p_{35}, mm.
(10) Pitch at 60% radius, p_{60}, m.
(11) Pitch at 70% radius, p_{70}, m.
(12) Rake at blade tip measured at shaft axis (backward rake positive, forward rate negative), r, mm.
(13) Maximum blade thickness of the expanded cylindrical section considered, t, mm.
(14) Developed area ratio, f.
(15) Type and specified minimum tensile strength of the material.

1.1.2 Where the propeller is fitted to the screwshaft without the use of a key, plans of the following items are to be submitted:

– Boss
– Tapered end of screwshaft
– Propeller nut
– Sleeve, where applicable

1.1.3 Where a sleeve is fitted, details of the proposed type of material and mechanical properties are also to be submitted.

SECTION 2 Materials

2.1 Casting

2.1.1 Castings for propellers and propeller blades are to comply with the requirements of LHR's Rules for Materials. The specified minimum tensile strength is to be not less than stated in Table 5.2.1.

2.1.2 For materials not included in Table 5.2.1, details of the chemical composition, mechanical properties and density are to be submitted for approval.

Table 5.2.1:

Material	Special minimum Tensile strength (N/mm^2)	ρ Density (g/m^3)	σ_a Allowable stress (N/mm^2)
Grey cast iron	250	7,2	17,2
Spheroidal or nodular graphite cast iron	400	7,3	20,6
Carbon and low alloy steels	400	7,9	20,6
13% chromium stainless steels	500	7,7	41
Chromium – nickel austenitic stainless steel	450	7,9	41
Grade Cu 1 Manganese bronze (high tensile brass)	440	8,3	39
Grade Cu 2 Ni – Manganese bronze (high tensile brass)	440	8,3	39
Grade Cu 3 Ni – Aluminium bronze	590	7,6	56
Grade Cu 4 Ni – Aluminium bronze	630	7,5	46

SECTION 3 Design

3.1 Calculation of minimum blade thickness

3.1.1 The thickness t of the propeller blades at 25% radius for solid propellers, neglecting any increase due to fillets, and at 60% radius, is to be not less than:

$$t = t_o + 5610 \cdot t_1$$

where:

t_o = $\dfrac{k_1 \cdot k_2}{k_3^2}$

t_1 = $\dfrac{k_4}{k_3} \cdot \sqrt{\dfrac{P}{n}}$

k_2 = $\rho \cdot f \cdot r \cdot n^2 \cdot D^3$

k_3 = $\sqrt{m \cdot c \cdot I \cdot N \cdot \sigma_a}$

I = I_{25}, I_{35} or I_{60}, as appropriate

ρ = density, g/cm^3 , see Table 5.2.1

σ_a = allowable stress, N/mm^2, see 3.1.1, 3.1.2, 3.1.4 and Table 5.2.1.

m = $11,1 \cdot Fm/(t^2 \cdot I)$. For aerofoil sections with and without trailing edge washback, m may be taken as 1,0 and 1,25 respectively.

F_m = actual face modulus.

For solid propellers at 25% radius:

m = $11,1 \cdot F_m/(t^2 \cdot I)$. For aerofoil sections with and without trailing edge washback, m may be taken as 1,0 and 1,25 respectively.

F_m = actual face modulus.

For solid propellers at 25% radius:

$c = 0{,}8 + \frac{p_{25}}{D}$

$k_1 = 0{,}015$

$k_4 = \sqrt{2{,}8 \cdot \frac{p_{25}}{D} + 3{,}75 \cdot \frac{D}{p_{70}} + 1}$

For controllable pitch propellers at 35% radius:

$c = 1{,}6 + \frac{p_{35}}{D}$

$k_1 = 0{,}002$

$k_4 = \sqrt{2{,}6 \cdot \frac{p_{35}}{D} + 5 \cdot \frac{D}{p_{70}} + 1{,}35}$

For all propellers at 35% radius:

$c = 4{,}5 + \frac{p_{60}}{D}$

$k_1 = 0{,}024$

$k_4 = \sqrt{1{,}35 \cdot \frac{p_{60}}{D} + 5 \cdot \frac{D}{p_{70}} + 1{,}35}$

3.1.2 The fillet radius between the root of a blade and the boss of a propeller is to be not less than the Rule thickness of the blade or equivalent at this location. Composite radiused fillets or elliptical fillets which provide a greater effective radius to the blade are acceptable and are to be preferred. Where fillet radii of the required size cannot be provided, the value of σ_a is to be multiplied by $(r/t)^{0,2}$, where r is the proposed fillet radius at the root, mm, and t is the thickness of the blade at the root,

- Where a propeller has bolted-on blades, consideration is also to be given to the distribution of stress in the palms of the blades. In particular, the fillets of recessed bolt holes and the lands between bolt holes are not to induce stresses which exceed those permitted at the outer end of the fillet radius between the blade and the palm.

3.1.3 Where the composition of the propeller material is not specified in Table 5.2.1, or where propellers of the cast irons and carbon and low alloy steels shown in this Table are provided with an approved method of cathodic protection, special consideration will be given to the value of σ_a.

3.1.4 The value of σ_a may be increased by 10% for twin screw and outboard propellers of triple screw ships.

3.1.5 Where the design of a propeller has been based on analysis of reliable wake survey data in conjunction with a detailed fatigue analysis and is deemed to permit scantlings less than required by 3.1.1 and also for propellers of unusual design, or where the propeller is intended for more than one operating regime, such as towing or trawling, a detailed stress computation for the blades is to be submitted for consideration.

3.2 Keyless Fitting of Propellers without Ice Strengthening (IACS UR K3 Cor.2 (1998))

3.2.1 Requirements to be satisfied

(1) The formulae, etc., given herein are not applicable for propellers where a sleeve is introduced between shaft and boss.

(2) The taper of the propeller shaft cone should not exceed 1/15.

(3) Prior to final pull-up, the contact area between the mating surfaces is to be checked and should not be less than 70% of the theoretical contact area (100%). Non-contact bands extending circumferentially around the boss or over the full length of the boss are not acceptable.
(4) After final pull-up, the propeller is to be secured by a nut on the propeller shaft. The nut should be secured to the shaft.
(5) The factor of safety against friction slip at 35°C is not to be less than 2,8 under the action of rated torque (based on rated power r.p.m.) plus torque due to torsionals as defined in 3.2.4.
(6) For the oil injection method the coefficient of friction should be 0,13 for bosses made in copper-based alloy and steel.
(7) The maximum equivalent uniaxial stress in the boss at 0°C based on the Mises-Hencky criterion (E) should not exceed 70% of the yield point or 0,2% proof-stress (0,2% offset yield strength) for the propeller material based on the test piece value. For cast iron the value should not exceed 30% of the nominal tensile strength.

3.2.2 Material constants

<u>Modulus of elasticity</u>	
Cast and forged steel	2,1 x 10^4 kgf/mm^2
Cast iron	1,0 x 10^4 kgf/mm^2
Copper based alloys, Cu 1 and Cu 2	1,1 x 10^4 kgf/mm^2
Copper based alloys, Cu 3 and Cu 4	1,2 x 10^4 kgf/mm^2
<u>Poisson's ratio</u>	
Cast and forged steel	0,29
Cast iron	0,26
All copper-based alloys	0,33
<u>Coefficient of linear expansion</u>	
Cast and forged steel and cast iron	12,0 x 10^{-6} mm/mm°C
All copper-based alloys	17,5 x 10^{-6} mm/mm°C

3.2.3 Formulae used

The formulae given below, for the ahead condition, will also give sufficient safety in the astern condition.

The formulae are applicable for solid shafts only.

Minimum required surface pressure at 35°

$$P_{35} = \frac{S \cdot T}{A \cdot B} \cdot \left[-S \cdot \vartheta + \sqrt{\mu^2 + B \cdot \left(\frac{F_v}{T}\right)^2} \right]$$

where:

$B = \mu^2 - S^2 \cdot \vartheta^2$

Corresponding minimum pull-up length at 35°C

$$\delta_{35} = P_{35} \cdot \frac{D_s}{2 \cdot \vartheta} \cdot \left[\frac{1}{E_b} \cdot \left(\frac{K^2 + 1}{K^2 - 1} + v_b\right) + \frac{1}{E_s} \cdot (1 - v_s)\right]$$

Minimum pull-up length at temperature t (t < 35°C)

$$\delta_\tau = \delta_{35} + \frac{D_s}{2 \cdot \vartheta} \cdot (\alpha_b - \alpha_s) \cdot (35 - t)$$

Corresponding minimum surface pressure at temperature t

$$P_t = P_{35} \frac{\delta_t}{\delta_{35}}$$

Minimum push-up load at temperature t

$$W_t = A \cdot P_t \cdot (\mu + \vartheta)$$

Maximum permissible surface pressure at 0°C

$$P_{max} = \frac{0{,}7 \cdot \sigma_y \cdot (K^2 - 1)}{\sqrt{3 \cdot K^4 + 1}}$$

Corresponding maximum permissible pull-up length at 0°C

$$\delta_{max} = \frac{P_{max}}{P_{35}} \cdot \delta_{35}$$

Shear force at interface

$$F_v = \frac{2 \cdot c \cdot Q}{D_s}$$

Rated thrust developed for free running vessels (if not given)

$$T = 132 \cdot \frac{H}{V_s} \quad or \quad T = 4{,}3 \cdot 10^6 \cdot \frac{H}{P \cdot N}$$

3.2.4 Nomenclature

A	=	100% theoretical contact area (mm^2) between boss and shaft, as read from drawings and disregarding oil grooves
Ds	=	diameter (mm) of propeller shaft at the midpoint of the taper in the axial direction
D_b	=	mean outer diameter (mm) of propeller boss at the axial position corresponding to D_s
K	=	D_b/D_s
F_v	=	shear force at interface = $2 \cdot c \cdot Q/D_s$ (kgf)
Q	=	rated torque (kgf·mm) transmitted according to rated horsepower, H, and speed of propeller shaft
T	=	rated thrust (kgf)
c	=	constant,
	=	1,0 for turbines, geared diesel drives, electric drives and for direct diesel drives with a hydraulic or an electromagnetic or high elasticity coupling
	=	1,2 for a direct diesel drive.
	=	LHR reserves the right to increase the c constant if the shrinkage has to absorb an extreme high pulsating torque.
H	=	rated brake horsepower (PS)
P	=	mean propeller pitch (mm)
N	=	propeller speed (r.p.m.) at rated brake horsepower
V_s	=	ship speed (knots) at rated horsepower
S	=	factor of safety against friction slip at 35°C

θ	=	half taper of propeller shaft, e.g. taper = 1/15. θ = 1/30
μ	=	coefficient of friction between mating surfaces
P_{35}	=	surface pressure (kgf/mm²) between mating surfaces at 35°C
Pt	=	surface pressure (kgf/mm²) between mating surfaces at temperature t°C
P_0	=	surface pressure (kgf/mm²) between mating surfaces at temperature 0°C
P_{max}	=	maximum allowable surface pressure (kgf/mm²) at 0°C
δ_{35}	=	pull-up length (mm) at temperature 35°C
δ_t	=	pull-up length (mm) at temperature t°C
δ_{max}	=	maximum allowable pull-up length (mm) at temperature 0°C
W_t	=	push-up load (kgf) at temperature t°C
σ_E	=	equivalent uniaxial stress (kgf/mm²) in the boss according to the Mises-Hencky criterion
α_s	=	coefficient of linear expansion (mm/mm°C) of shaft material
α_b	=	coefficient of linear expansion (mm/mm°C) of boss material
E_s	=	modulus of elasticity (kgf/mm²) of shaft material
E_b	=	modulus of elasticity (kgf/mm²) of boss material
v_s	=	Poisson's ratio for shaft material
v_b	=	Poisson's ratio for boss material
σ_y	=	yield point or 0,2% proof stress (0,2% offset yield strength) of propeller material (kgf/mm²)

SECTION 4 Fitting of Propeller

4.1 Propeller Boss

The propeller boss is to be a good fit on the screwshaft cone. The forward edge of the propeller bosses is to be rounded to about a 6 mm radius. In the case of keyed propellers, the length of the forward fitting surface is to be about one diameter and, where the fitting is by means of a hydraulic nut, the requirements of 4.2 and 4.3, where appropriate, are applicable.

4.2 Shop tests of Keyless Propellers

4.2.1 The bedding of the propeller, or the sleeve where applicable, with the shaft is to be demonstrated in the shop to the satisfaction of the Surveyors. Sufficient time is to be allowed for the temperature of the components to equalize before bedding. Alternative means for demonstrating the bedding of the propeller will be considered.

4.2.2 Means are to be provided to indicate the relative axial position of the propeller boss on the shaft taper.

4.3 Final Fitting of Keyless Propellers

4.3.1 After verifying that the propeller and shaft are at the same temperature and the mating surfaces are clean and free from oil or grease, the propeller is to be fitted on the shaft to the satisfaction of the Surveyors. The propeller nut is to be securely locked to the shaft.

4.3.2 Permanent reference marks are to be made on the propeller boss, nut and shaft to indicate angular and axial positioning of the propeller. Care is to be taken in marking the inboard end of the shaft taper to minimize stress raising effects.

4.3.3 The outside of the propeller boss is to be hard stamped with the following details:

(a) For the oil injection method of fitting, the start point load and the axial pull-up at 0 °C and 35 °C.
(b) For the dry fitting method, the push-up load at 0 °C and 35 °C.

4.3.4 A copy of the fitting curve relative to temperature and means for determining any subsequent movement are to be placed on board.

SECTION 5 Tests

5.1 Balancing

5.1.1 The finished propeller and the blades of controllable pitch propellers are required to undergo static balancing.

5.2 Testing

5.2.1 Fixed pitch propellers, controllable pitch propellers and controllable pitch propeller systems are to be presented to LHR for final inspection and verification of the dimensions. In addition, controllable pitch propeller systems are required to undergo pressure, tightens and operational tests. LHR reserves the right to require non-destructive tests to be conducted to detect surface cracks or casting defects. With regard to the assessment and the repair of defects on propellers, see the Rules for Materials, Part 2, Chapter 7, SECTION 1, 1.9.

CHAPTER 6 Shaft Vibration and Alignment

CONTENTS

SECTION 1 General

1.1 Scope

1.1.1 The requirements of this Chapter are applicable to the following systems:

(1) Main propulsion systems formed by oil engines, turbines or electric motors, directly driven or geared to the shafting.
(2) Machinery driven at constant speed by oil engines, developing 110 kW and over, for essential auxiliary services including generator sets which are the source of power for main electric propulsion motors.

Unless otherwise advised, it is the responsibility of the Shipbuilder as main contractor to ensure, in co-operation with the Engine builders, that the information required by this Chapter is prepared and submitted.

1.2 Basic requirements

1.2.1 The systems are to be free from excessive torsional, axial, lateral and linear vibration, and are to be aligned in accordance with accepted tolerances and taking into account the requirements of 5.3.

1.2.2 System designs are to take account of the potential effects of engine and component malfunction and variability in characteristic values such as stiffness and damping of flexible couplings and dampers or engine misfire conditions.

1.2.3 Where torques, stresses or amplitudes are found to exceed the limits for continuous operation, restrictions in speed and/or power will be imposed.

1.2.4 Where significant changes are subsequently made to a dynamic system which has been approved, (e.g. by changing the original design parameters of the prime movers and/or propulsion shafting system, or by fitting a propeller or flexible coupling of different design from the previous), revised calculations may require to be submitted for consideration. Details of all such changes are to be submitted.

1.3 Resilient mountings

1.3.1 The dynamic angles of inclination in Part 5, Chapter 1, Table 1.3.2 may be exceeded in certain circumstances dependent upon ship type and operation. The Shipbuilder is, therefore, to ensure that the vibration levels of flexible pipe connections, shaft couplings and mounts remain within the limits specified by the component manufacturer for the conditions of maximum dynamic inclinations to be expected during service, start-stop operation and the natural frequencies of the system. Due account is to be taken of any creep that may be inherent in the mount.

1.3.2 Anti-collision chocks are to be fitted together with positive means to ensure that manufacturers limits are not exceeded. Suitable means are to be provided to accommodate the propeller thrust.

1.3.3 A plan showing the arrangement of the machinery together with documentary evidence of the foregoing is to be submitted.

SECTION 2 Torsional vibrations

2.1 General

2.1.1 Further to the Scope of this Chapter, the requirements of this Section are not applicable to ships that are not:

(a) required to comply with the International Convention for the Safety at Sea, 1974, as amended, (SOLAS); or
(b) where a main engine does not have a power output exceeding 500 kW.

2.2 Documents for approval

2.2.1 Analytic calculations of the torsional vibration characteristics along with the assessment of the vibratory torques and stresses are to be submitted for approval.

2.2.2 In piston engine installations involving more than one engine, or in turbine installations, documentation with regard to power division over the whole speed range is required.

2.2.3 Any special speed requirements are to be precisely indicated.

2.2.4 In case of piston engine installations, the vibration calculations should be in compliance with Part 5, Chapter 2. Upon the discretion of LHR, alternative methods of calculation may be considered.

2.3 Items sensitive to vibratory torque

2.3.1 In installation involving items sensitive to vibratory torque special care is to be taken for the case of engine malfunction.

2.3.2 Detailed description of the properties of such items relative to their capability to withstand the effects of vibratory conditions are to be submitted.

2.4 Permissible torsional vibration stresses (IACS UR M68.5 Rev.3 (2021))

2.4.1 Alternating torsional vibration stresses are stresses resulting from the alternating torque which is superimposed on the mean torque. The alternating torsional stress amplitude is understood as $(\tau_{max} - \tau_{min})/2$ as can be measured on a shaft in a relevant condition over a repetitive cycle.

2.4.2 Torsional vibration calculations are to include normal operation and operation with any one cylinder misfiring (i.e. no injection but with compression) giving rise to the highest torsional vibration stresses in the shafting.

2.4.3 For continuous operation the permissible stresses due to alternating torsional vibrations are not to exceed the values given by the following formulae:

$$\tau_c = \pm\frac{\sigma_B + 160}{18} \cdot c_k \cdot c_d \cdot (3 - 2 \cdot \lambda^2), \qquad for\ \lambda < 0{,}9$$

$$\tau_c = \pm\frac{\sigma_B + 160}{18} \cdot c_k \cdot c_d \cdot 1{,}38, \qquad for\ 0{,}9 \le \lambda \le 1{,}05$$

where:

τ_c = permissible stress amplitude due to torsional vibrations for continuous operation, N/mm²
σ_B = specified minimum ultimate tensile strength of the shaft material, N/mm², see also Part 5, Chapter 4b, SECTION 2, 2.2

c_k = factor for the particular shaft design features, see Part 5, Chapter 4b, SECTION 3, 3.1.2(1),
c_D = size factor,
= $0{,}35 + 0{,}93.d_o^{-0{,}2}$
d_o = shaft outside diameter, mm,
λ = speed ratio
= n/n_o
n = speed under consideration, rpm,
n_o = speed of shaft at rated power, rpm.

Where the stress amplitudes exceed the limiting values of τ_C for continuous operation, including one cylinder misfiring conditions if intended to be continuously operated under such conditions, restricted speed ranges are to be imposed which are to be passed through rapidly.

Restricted speed ranges in normal operating conditions are not acceptable above $\lambda = 0{,}8$.

Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships are to enable safe navigation.

The limits of the barred speed range are to be determined as follows:

(a) The barred speed range is to cover all speeds where the acceptance limits (τ_C) are exceeded. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered. Additionally, the tachometer tolerance has to be added. At each end of the barred speed range the engine is to be stable in operation.

(b) In general and subject to (a) the following formula may be applied, provided that the stress amplitudes at the border of the barred speed range are less than τ_C under normal and stable operating conditions.

$$\frac{16 \cdot n_c}{18 - \lambda_c} \leq n \leq \frac{(18 - \lambda_c) \cdot n_c}{16}$$

where:

n_C = critical speed (resonance speed), rpm

λ_c = speed ratio

= n_c/n_o

n, n_o = as defined in 2.4.3

2.4.4 For the passing of the barred speed range the torsional vibrations for steady state condition are not to exceed the value given by the formula:

$$\tau_T = \pm 1{,}7 \cdot \frac{\tau_c}{\sqrt{C_k}}, \qquad for\ \lambda \leq 0{,}8$$

where:

τ_T = permissible stress amplitude due to steady state torsional vibration in a bared speed range, N/mm^2.

Other terms in the formula are as defined in 2.4.3.

2.5 Permissible limits of stress for crankshafts

2.5.1 For permissible limits of alternating stresses for crankshafts reference is made to Part 5, Chapter 2, SECTION 3 and in particular to 3.6, 3.7 and 3.9.

2.6 Measurements

2.6.1 Measurements are required when analytical calculations show excessive vibration levels within the range of operational speeds for the purpose of determining the need for restricted speed ranges.

2.7 Tachometer accuracy

2.7.1 Tachometer's accuracy should be checked, in the presence of a Surveyor, against the counter readings, especially where restriction of the speed range is imposed. Deviations up to 2% are treated as acceptable.

2.8 Auxiliary machinery and main engine driven generators

2.8.1 The following requirements are applicable to piston engines, developing 110 kW and over, driving auxiliary machinery used for essential services, and to propulsion main engine driven generating sets operating at constant speed.

2.8.2 Within the speed range from $0{,}95 \cdot n_o$ and $1{,}10 \cdot n_o$ the value of the vibration stresses in the crankshafts and transmission shafting are not to exceed the limiting values given by the formula:

$$\tau_1 = \pm(21 - 0{,}014 \cdot d), \quad \text{N/mm}^2$$

where:

d = the minimum diameter of shaft considered. For systems where more than one excitation order is present (e.g. most diesel engine driven applications) the overall torsional vibratory stress, including the flanks of resonances above and below the service speed is subject to special consideration but in no case should exceed 1,4 times the above allowable stress.

2.8.3 Vibration stresses in the crankshaft and transmission shafting due to critical speeds which have to be passed through in starting and stopping, are not to exceed the values given by the following formula:

$$\tau_2 = 5{,}5 \cdot \tau_1, \quad \text{N/mm}^2$$

where:

τ_1 = as specified in 2.8.2.

2.8.4 The amplitudes of the total vibratory inertia torques imposed on the generator rotors are to be limited to $2{,}0 \cdot M_o$ in general, or to $2{,}5 \cdot M_o$ for close-coupled revolving field alternating current generators, over the speed range from $0{,}95 \cdot n_o$ to $1{,}10 \cdot n_o$. Below $0{,}95 \cdot n_o$, the amplitudes are to be limited to $6{,}0 \cdot M_o$, where M_o is the torque at rated power and rated speed. Where two or more generators are driven from one engine, each generator is to be considered separately in relation to its own rated torque.

2.8.5 The rotor shaft and structure are to be designed to withstand these magnitudes of vibratory torque. Where it can be shown that they are capable of withstanding a higher vibratory torque, special consideration will be given.

2.8.6 In addition to withstanding the vibratory conditions over the speed range from 0,95·no to 1,10·no, flexible couplings, if fitted, are to be capable of withstanding the vibratory torques and twists arising from transient critical and short-circuit currents.

2.8.7 In the case of alternating current generators, resultant vibratory amplitudes at the rotor are not to exceed 3,5 electrical degrees under both full load working conditions and the malfunction condition mentioned in 2.3.1.

SECTION 3 Axial vibrations

3.1 General

3.1.1 The Shipbuilders are to ensure that amplitudes due to axial vibration of all main propulsion shafting systems are within acceptable limits. Effective means may be used for the reduction of the amplitudes. If the required reduction is not possible, restriction of the speed range will be imposed.

3.2 Documents for approval

3.2.1 In case of shaft systems with a propeller:

(1) driven directly by a reciprocating internal combustion engine, or
(2) driven via gears, or directly by an electric motor, and where the total length of shaft between propeller and thrust bearing is in excess of 60 times the intermediate shaft diameter, calculations including the effects of flexibility of the thrust bearing, are to be submitted.

3.2.2 For the systems defined in 3.2.1(2) the propeller speed at which the critical frequency occurs may be estimated using the following formula:

$$\frac{1}{N}\cdot\sqrt{\frac{0{,}9604\cdot f\cdot w}{f+w}}, \qquad \text{rpm}$$

where:

N = number of propeller blades,

f = $\frac{E}{\rho\cdot l^2}\left[66{,}2+97{,}5\cdot\left(\frac{m}{m_p}\right)-8{,}884\cdot\left(\frac{m}{m_p}\right)^2\right]^2,$ rpm

w = $91{,}2\cdot\frac{k}{m_e},$ rpm

M = mass of shaft line considered, kg,
 = $0{,}785\cdot(D^2 - d^2)\cdot\rho\cdot l$,

m_p = dry mass of propeller, kg,

m_e = $m+2\cdot m_p$,

k = estimated stiffness at thrust block bearing, N/m,

D = Outside diameter of shaft, taken as an average over length l, mm,

d = Internal diameter of shaft, mm

l = length of shaft line between propeller and thrust bearing, mm,

E = modulus of elasticity of shaft material, N/mm^2,

P = density of shaft material, kg/mm^3.

Calculations based on a recognized method may be considered alternatively.

3.2.3 The requirement for calculations to be submitted may be waived upon request provided evidence of satisfactory service experience of similar dynamic installations is submitted.

3.3 Measurements

3.3.1 Measurements are required when analytical calculations show excessive vibration levels within the range of operational speeds for the purpose of determining the need for restricted speed ranges.

SECTION 4 Bending vibrations

4.1 General

4.1.1 The Shipbuilders are to ensure that amplitudes due to lateral vibration of all main propulsion shafting systems are restricted within acceptable limits over the whole speed range.

4.2 Documents for approval

4.2.1 Normally, calculations of the lateral, or bending, vibration characteristics of shafting systems having supports outboard of the hull or incorporating cardan shafts are to be carried out. Relaxations may be considered in case of previous experience on similar installations. These calculations, taking account of bearing and structure dynamic stiffnesses, are to cover the frequencies giving rise to all critical speeds which may result in significant amplitudes within the speed range, and are to indicate relative deflections and bending moments throughout the shafting system.

4.2.2 The result of these calculations, or the evidence of previous experience, is to be submitted for consideration.

4.2.3 Requirements for calculations may be waived upon request provided evidence of satisfactory service experience of similar dynamic installations is submitted.

4.3 Measurements

4.3.1 Measurements are required when analytical calculations show excessive vibration levels within the range of operational speeds for the purpose of determining the need for restricted speed ranges.

SECTION 5 Shaft alignment

5.1 General

5.1.1 Shaft alignment is required for all main propulsion installations in order to avoid excessive bearing reactions and bending moments at all operational conditions. The Builder is to position the bearings and construct the bearing seatings to minimize the effects of movements under all operating conditions.

5.2 Documents for approval

5.2.1 For geared installations and where a flexibly mounted raft is used for the engine installation detailed calculations are required. In case of geared installations, the proposed shafting alignment indicating the limits for differentials in the gear bearing reactions, in the cold static and hot running conditions recommended by the manufacturers, is to be submitted for consideration. The calculations are to take account of thermal effects, gear tooth loadings where appropriate, and the effect of the aft section of the hull, where known.

5.2.2 The calculations are to include bearing reactions, the effect on the reactions of linear movements at the bearings, and bending moments along the shaft.

5.2.3 For single engine geared installations with a screwshaft diameter less than 300mm calculations are not required.

5.3 Measurements

5.3.1 Where the system presents excessive sensitivity to changes in alignment under service conditions, the shaft alignment is to be checked by measurements.

5.4 Alignment couplings

5.4.1 If couplings are used in order to maintain the alignment within the permissible range required by individual transmission components, under service conditions, the effects of such couplings are to be taken into account on the various modes of vibration.

CHAPTER 7 (a) Boilers and Other Pressure Vessels

CONTENTS

SECTION 1 General

1.1 Application

1.1.1 Chapter 7a provides the requirements for the design and construction, as well as for the mountings or fittings, of pressure vessels, either seamless or welded, intended for use in:

(1) fired boilers,
(2) exhaust gas heated boilers,
(3) economizers, superheaters, reheaters and steam receivers for, and associated with (1) and (2),
(4) steam heated steam or hot water generators.

1.2 Additional requirements

1.2.1 Plants incorporating pressure vessels as defined in 1.1 are also required to comply with the applicable statutory requirements and regulations of the ship's country of registration.

1.3 Compliance with recognized standards

1.3.1 Where arrangements or details of pressure vessels, equipment or other parts associated with steam raising plants, comply with recognized standards, LHR will give its consideration to these details, provided the aforementioned standards are not less effective than these Rules.

1.4 Plans and data for submission

1.4.1 The manufacturers are to submit drawings of the following:

- General arrangement,
- Sectional assembly,
- Seating arrangements,
- Steam and water drum and header details,
- Waterwall details,
- Steam and superheater tubing and the tube support arrangements,
- Economizer arrangement, header details and element details,
- Casing arrangement,
- Boiler mountings including steam stop valves, safety valves and relieving capacities, feedwater connections, blow-off arrangements, water-gauges, try cocks etc.
- Integral piping,
- Reheat section (when fitted),
- Fuel oil burning arrangements including burners, registers etc. Forced draft system.

1.4.2 The data to be submitted are to include heating surface, steam capacity, design pressure and design temperature, superheater header and tube mean wall temperatures, pressures for the settings of the safety valves, particulars of the safety valves, specifications of the materials to be used, feed and firing system.

1.4.3 Before the commencement of construction, plans are to be submitted for consideration, showing full constructional arrangements of welded pressure vessels and dimensional details of the weld preparation for longitudinal and circumferential seams and attachments. Particulars of the welding consumables, mechanical properties of the materials and generally all the necessary details for an appropriate evaluation are to be contained in the above-mentioned plans. Welding procedure specifications are also to be submitted.

SECTION 2 General Requirements

2.1 Materials

2.1.1 Materials intended to be used in the construction of boiler plants are to be manufactured and tested according to the requirements of LHR's Rules for the Manufacture, Testing and Certification of Materials (Part 2) with respect, primarily, to the high-temperature strength and good weldability.

2.1.2 Material grades and corresponding product forms are to be as in Table 7a.2.1.

2.1.3 Where it is proposed to use materials other than those specified in LHR's Rules for the Manufacture, Testing and Certification of Materials, details of the chemical composition, heat treatment and mechanical properties are to be submitted for approval. In such cases the values for the mechanical properties used for deriving the allowable stresses are to be subject to agreement by LHR.

2.1.4 Parts not subject to material testing, such as external supports, lifting brackets etc. must be made of materials suitable for the intended purpose and in accordance with accepted engineering practice.

Table 7a.2.1: Materials corresponding to product forms

Material and Product form	Material grades in accordance with the Rules of Materials
Steel plates and steel strips	Plates and strips of high temperature Part 2, Chapter 3, SECTION 4
Steel pipes	Seamless and welded pipes of ferritic steels Part 2, Chapter 6, SECTION 6
Forgings and formed parts (a) drums, headers and similar hollow components without longitudinal seam (b) covers, flanges, branch pipes end plates	Forgings for boilers and pressure vessels Part 2, Chapter 5, SECTION 4
Steel Castings	Steel castings for boilers, pressure vessels and piping equipment Part 2, Chapter 4, SECTION 5

2.2 Welded pressure vessels

2.2.1 The execution of welds, the approval of welding shops and the welding procedures, as well as the qualification testing of welders are to be in accordance with Part 2, Chapters 9, 10 and 12.

2.2.2 Where the welded pressure vessels meet the requirements set forth in Part 2, Chapter 12 in every respect and where the welds are subjected to inspections and non-destructive tests performed in accordance with Part 2, Chapter 12, SECTION 2, the efficiency of welded joints is to be evaluated as 1,0. An efficiency of 0,85 is applicable where non-destructive tests are to be performed at random, covering a percentage specified in the same Section.

2.2.3 The longitudinal and circumferential joints are to be butt joints.

2.2.4 In the cases where a fusion welded pressure vessel is intended to be made of alloy steel and approval of the scantlings is required on the basis of the high temperature properties of the material, particulars of the welding consumables to be used, including typical mechanical properties and chemical composition of the deposited weld metal, are to be submitted for approval.

SECTION 3 Principles Applicable to the Manufacture

3.1 Manufacturing processes

3.1.1 Materials must be checked for defects during the manufacturing process. Different materials are not to be confused. During the course of manufacture attention should be paid to ensure that LHR's and manufacturer's markings remain intact on the materials.

3.2 Tube expansion

3.2.1 Tube holes must be carefully drilled and deburred. Sharp edges are to be chamfered. Tube ends to be expanded are to be cleaned and checked for size and possible defects. Where necessary, tube ends are to be annealed before being expanded.

3.3 Pipe connections and flanges

3.3.1 All pipe connections and flanges are to be properly welded to the shell. The wall thickness of welded-in pipe connections are to be appropriate to the wall thickness of the part into which they are welded. Welding-neck flanges must be made of forged material with favourable grain orientation.

3.4 Cleaning and inspection openings

3.4.1 Steam boilers are to be provided with openings through which the space inside can be cleaned and inspected. Boiler vessels with an inside diameter of more than 1200mm and those measuring over 800mm in diameter and 2000mm in length are to be provided with means of access.

3.4.2 Inspection and access openings are required to have the following minimum dimensions:

(1) Manholes: 300x400 or 400mm diameter. Where the annular height exceeds 150mm, the opening is to measure: 320x420mm.
(2) Holes for the head: 220 x 320mm or 320mm diameter
(3) Handholes: 87x103mm
(4) Sight holes: they are allowable only when the design of equipment makes a handhole impracticable. Sight holes are required to have a diameter of at least 50mm.

3.4.3 The edges of manholes and other openings, e.g. for domes, are to be effectively strengthened if the plate has been unacceptably weakened by the cut-outs. The edges of openings closed with covers are to be reinforced by flanging or by welding on edge stiffeners if it is likely that the tightening of the crossbars etc. would otherwise cause undue distortion of the edge of the opening.

3.4.4 Circular flat cover plates are to be fitted to raised circular manhole frames not exceeding 400mm diameter, and for an approved design pressure not exceeding 18 bar. Thicknesses of the frames are to be not less than 19mm in all parts. The thickness of circular cover plates and joint flanges for such frames is to be not less than:

(1) 25mm for a design pressure 8 bar
(2) 29mm for a design pressure > 8 bar and 14 bar
(3) 32mm for a design pressure > 14 bar and ≤ 18 bar.

SECTION 4 Design Principles

4.1 Design pressure, P

4.1.1 Generally, the design pressure is the maximum allowable working pressure. It is not to be less than the highest set pressure of any safety valve.

4.1.2 The design pressure is used in the various formulas for the determination of pressure part scantlings. Where necessary, the design pressure is to be adjusted to take account of pressure variations due to the most severe operational conditions. The hydrostatic pressure is to be taken into account in the design pressure calculations when it exceeds 0,5 bar.

4.1.3 In designing desuperheaters in boiler drums or other boiler parts which are subject in operation to both internal and external pressure, the differential pressure is to be taken into account, provided that it is certain that in service both pressures will invariably occur simultaneously. The design is also required to take account of the loads imposed during the hydraulic pressure test.

4.2 Design temperature, T

4.2.1 The determination of allowable stresses is based upon the design temperature, T, which is not to be taken less than the mean wall metal temperature (through thickness) expected under operating conditions for the part considered. The design temperature is to be stated by the manufacturer on the drawings of the pressure parts submitted for consideration. LHR may ask for submission of heat transfer calculations to verify the design temperature or ask for measurements of temperature in service under equivalent operating conditions.

4.2.2 When the occurrence of different metal temperatures during operation can be definitely predicted for different zones of a vessel, the design of the different zones may be based on their predicted temperatures. When sudden cyclic changes in temperature are apt to occur in normal operation with only minor pressure fluctuations, the design is to be governed by the highest probable operating temperature.

4.2.3 In no case the temperature at the surface of the metal is to exceed the maximum temperature listed in the stress tables for materials, nor exceed the temperature limitations specified elsewhere in these Rules.

4.2.4 The design temperatures for the several pressure parts are not to be taken less than those mentioned in Table 7a.4.1 nor less than 250°C.

Table 7a.4.1: Minimum design temperatures

<table>
<tr><th rowspan="3">Pressure part</th><th rowspan="3">Pressure Part Internal fluid</th><th rowspan="3">Reference temperature</th><th rowspan="3">Unheated parts</th><th colspan="3">Temperature allowance to be added</th></tr>
<tr><th colspan="2">Heated parts heated mainly by</th><th rowspan="2">Preheated from hot gases</th></tr>
<tr><th>contact</th><th>radiation</th></tr>
<tr><td>Boiler tubes shells, end plates headers, tube plates</td><td>Water or steam and water</td><td>Saturation temperature at max allow work pressure</td><td>0</td><td>15+2t≤50
t = thickness</td><td>50</td><td>20</td></tr>
<tr><td>Header steam Superheater tubes</td><td>Superheated steam</td><td>Superheated steam temperature</td><td>15 (Note 1)</td><td>35</td><td>50</td><td>20</td></tr>
<tr><td rowspan="2">Economizer tubes</td><td rowspan="2">water</td><td rowspan="2">Max feedwater Temp = saturation Temp.(at the working Pressure) – 20°C</td><td rowspan="2">-</td><td>25 plain tube economizers</td><td rowspan="2">-</td><td rowspan="2">-</td></tr>
<tr><td>35 fired tube economizers</td></tr>
<tr><td>Corrugated furnaces</td><td></td><td>Max of internal fluid</td><td>-</td><td>-</td><td>75</td><td>-</td></tr>
<tr><td>Plain furnaces Combustion and Other chambers</td><td>feedwater</td><td>Max. Temperature of Internal fluid</td><td>-</td><td>-</td><td>90</td><td>-</td></tr>
<tr><td>Pressure parts</td><td></td><td>Max. temperature of feedwater</td><td>0</td><td>-</td><td>-</td><td>-</td></tr>
<tr><td>Hot water Water generators</td><td>water</td><td>Temp. of water of generators Inlet (Note 2)</td><td>-</td><td>0</td><td>-</td><td>-</td></tr>
<tr><td colspan="7">Notes:
1. The temperature allowance may be reduced to 7°C provided that special measures are taken to ensure that the design temperature cannot be exceeded.
2. Provided that there is appropriate thermostat to control the temperature.</td></tr>
</table>

4.2.5 The walls of pressure parts are considered to be non-heated in the following cases:

(1) the walls are separated from the combustion space or uptake by fire-resistant insulation. The distance between walls and insulation is to be not less than 300mm, or

(2) the walls are protected with fire-resistant insulation not exposed to radiant heat.

4.2.6 Walls are considered protected from hot gases when:

(1) the walls are protected with fire-resistant insulation or

(2) the walls are protected by closely spaced rows of tubes with a maximum clearance between rows of 3mm.

4.2.7 Generally, any parts of boiler drums or headers not protected by tubes, and exposed to radiation from the fire or to the impact of hot gases, are to be protected by a shield or good refractory material or by other approved means.

4.2.8 Drums or headers of thickness greater than 30mm are not to be exposed to combustion gases having an anticipated temperature in excess of 650°C unless the gases are efficiently cooled by closely arranged tubes.

4.3 Allowable stress, S

4.3.1 The design of pressure parts is to be based on the allowable stress, S, in N/mm².

4.3.2 The allowable stress, S, is to be the minimum of the following values:

$$\frac{R_{e,t}}{1,6},\frac{R_{m,20}}{2,7},\frac{S_{r,100000}}{1,5}$$

where:

$R_{e,t}$ = Specified lower yield stress, $R_{e,l}$, or 0,2% proof stress at the design temperature, N/mm²

$R_{m,20}$ = Specified minimum tensile strength at ambient temperature, i.e. 20 C, N/mm².

$S_{r,100000}$ = Average stress to produce rupture after 100000 h, at the design temperature, N/mm².

4.3.3 The allowable stress for steel castings is to be taken as 80% of the value determined by the method in 4.3.2, using the appropriate values for cast steel.

4.3.4 Stresses imposed by effects other than the working pressure or the hydrostatic head (see also 4.1.2), which increase the average stress are also to be taken into account. These effects may be the static and dynamic weight of the pressure vessel and its content, method of support, loads due to piping expansion, superimposed loads such as other pressure vessels, operating equipment, insulation, corrosion-resistant or erosion-resistant linings and piping, impact loads, including rapidly fluctuating pressures, the effect of temperature gradients on maximum stress etc.

4.4 Allowance for corrosion and wear, c

4.4.1 The allowance for corrosion and wear is to be 1mm. To prolong the life of a pressure vessel against corrosion, the corrosion allowance may be increased.

4.5 Special cases

4.5.1 Where pressure parts cannot be designed in accordance with the formulae of the following sections or according to general engineering principles, the scantlings in each individual case must be determined by tests e.g. by strain measurements, or by hydraulic proof test of a prototype or by an agreed alternative method.

SECTION 5 Boiler Tubes Subject to Internal Pressure

5.1 Scope

5.1.1 The design calculations of this Section apply to tubes subject to internal pressure.

5.2 Minimum thickness of tubes

5.2.1 The minimum thickness of straight tubes is to be determined as follows:

$$t=\frac{P\cdot D_o}{20\cdot S\cdot E+P}+c$$

where:

P = design pressure equal to the specified test pressure, bar,

t = minimum thickness, mm,

D_o = outside diameter, mm,

S = allowable stress, N/mm^2

E = weld efficiency of longitudinally welded tubes

c = corrosion allowance, mm; see 4.4.

5.2.2 The thickness is in no case to be less than the minimum shown in Table 7a.5.1. The thickness of intensely heated boiler tubes (e.g where the temperature of the heating gas exceeds 800°C) is not to be greater than 6,3mm. This requirement may be dispensed with in special cases, e.g. for superheater support tubes.

5.2.3 The design temperature T used is to be as specified in 4.2.

5.2.4 The weld efficiency of longitudinally welded tubes is to be taken from the corresponding approval test.

Table 7a.5.1: Minimum thicknesses of tubes

Nominal outside diameter of tube (mm)	Minimum thickness (mm)	Minimum Thickness (1) (mm)
≤ 38	1,75	2,95
38 < ... ≤ 50	2,16	
50 < ... ≤ 70	2,40	
70 < ... ≤ 75	2,67	3,28
75 < ... ≤ 95	3,05	
95 < ... ≤ 100	3,28	
100 < ... ≤ 125	3,50	3,50

Note:

1. Applicable to tubes subject to internal pressure and fitted in cylindrical boilers, and also for the tubes of low-pressure water tube boilers having a design pressure of 17,2 bar, or lower, with open feed systems.

5.3 Tube bending

5.3.1 Where boiler, superheater, reheater and economizer tubes are bent, the resulting thickness of the tubes at the thinnest part is to be not less than that required for straight tubes, unless it can be demonstrated that the method of forming the bend results in no decrease in strength at the bend. The manufacturer is to demonstrate, in connection with any new method of tube bending, that this condition is satisfied.

5.3.2 Tube bending, and subsequent heat treatment, where necessary, is to be carried out so as to ensure that residual stresses do not adversely affect the strength of the tube for the design purpose intended.

SECTION 6 Cylindrical Shells under Internal Pressure

6.1 Scope

6.1.1 The following design rules apply to drums, shell rings and headers up to a diameter ratio $D_o/D_i \leq 1.5$.

6.2 Symbols

P = Design pressure, bar.

T = Design temperature, °C.

D_i = Inside diameter, mm.

D_o = Outside diameter, mm.

c = Allowance for corrosion and wear, mm.

E = Efficiency factor, see 6.4.

d = Diameter of opening or cut-out, mm, see 6.5.

S = Maximum allowable stress, N/mm².

6.3 Minimum thickness

6.3.1 The minimum thickness, t, of a cylindrical shell subject to internal pressure is to be determined by the following formula:

$$t = \frac{P \cdot D_o}{20 \cdot S \cdot E + P} + c$$

6.3.2 For welded and seamless shell rings, the minimum permissible wall thickness is 6mm. For non-ferrous metals, stainless steels and cylinder diameters up to 200mm, smaller wall thicknesses may be permitted. The wall thickness of drums into which tubes are expanded is to be such as to provide a cylindrical expansion length of at least 16mm.

6.4 Efficiency factor

6.4.1 The efficiency factor, E, is the welding efficiency of the longitudinal joint or of ligaments between tube holes or other openings, whichever is the least.

6.4.2 For seamless shells the maximum efficiency factor is to be E = 1,0.

6.4.3 For welded shells, the efficiency of welded joints is to be 1,0 or 0,85 in accordance with 6.2.

6.4.4 Where tube holes are drilled in a cylindrical shell, as in the Figure 7a.6.1, the ligament efficiency is to be taken as the lowest of the longitudinal and the circumferential ligament efficiency which are defined as follows:

(1) The longitudinal ligament efficiency, E_L, is the efficiency of the plate through the ligaments in cylindrical shells pierced by a single row or by several well-separated rows of tubes, such as indicated in Figure 7a.6.2, and it is to be determined by the following formulae:

$$E_L = \frac{p_L - d}{p_L}, \qquad \text{where the holes are equally} - \text{pitched}$$

or

$$E_L = \frac{p_{L1} - n \cdot d}{p_{L1}}, \qquad where\ the\ holes\ are\ unequally - pitched$$

where:

p_L, p_{L1} = longitudinal pitch of tube holes, mm,

d = diameter of tube hole, mm; See 6.5,

n = number of tube holes in pitch P_{L1}..

(2) The circumferential ligament efficiency, E_C, is to be determined by the formulae:

$$E_c = \frac{2 \cdot (p_c - d)}{p_c}, \qquad where\ the\ holes\ are\ equally - pitched$$

Or

$$E_c = \frac{2 \cdot (p_{c1} - n \cdot d)}{p_{c1}}, \qquad where\ the\ holes\ are\ unequally - pitched$$

where:

P_C, P_{C1} = circumferential pitch, mm.

The circumferential pitch between tube holes is measured on the mean of the external and internal cylindrical shell diameters.

Figure 7a.6.1:

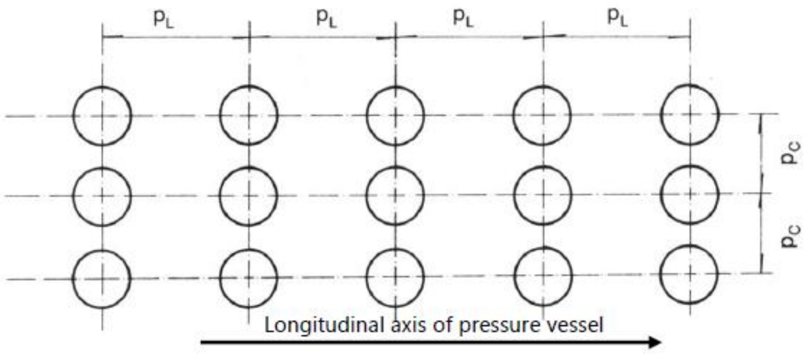
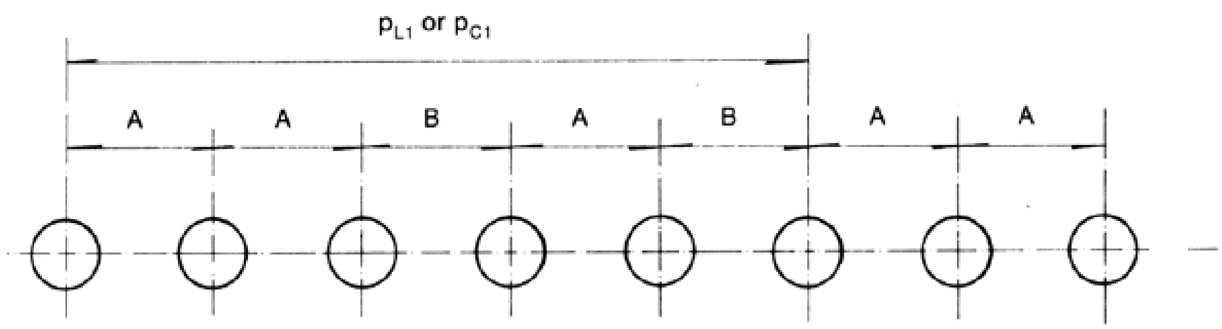


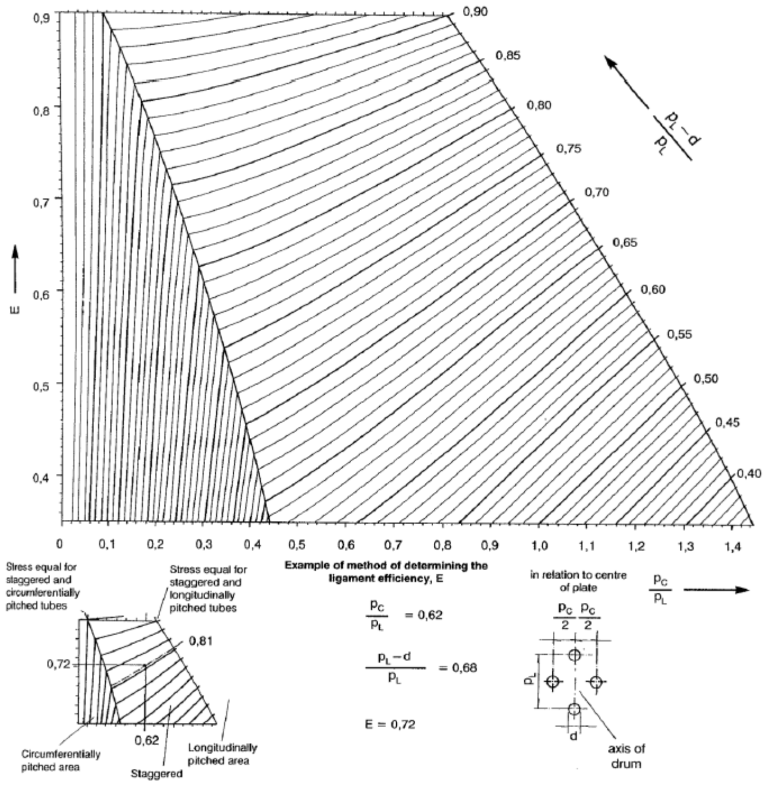
Figure 7a.6.2:



6.4.5 To determine the ligament efficiency for boiler drums with symmetrically staggered rows of holes, the procedure depicted in Figure 7a.6.3 is to be applied.

6.4.6 The longitudinal or circumferential ligament efficiency is not to be less than 0,35.

Figure 7a.6.3: Ligament efficiency, E, for boiler drums with symmetrically staggered rows of holes

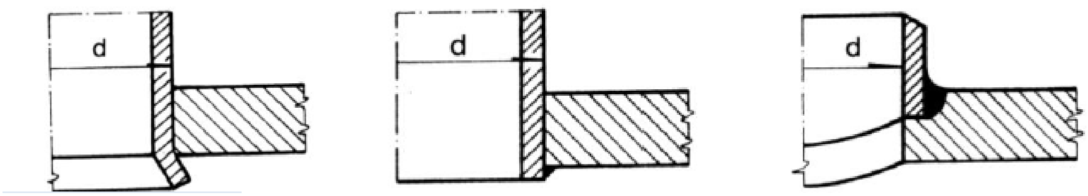


6.5 Diameter of opening

6.5.1 Where in the various formulae the diameter, d, of tube holes or other opening is used, it is the mean effective diameter of the tube holes after allowing for any serrations, counterboring or recessing, or the compensating effect of the tube stub. As a consequence, d is the hole diameter for expanded tubes and for expanded and seal-welded tubes as in Figure 7a.6.4(a),(b) or the equivalent diameter of the hole where tube stubs are fitted. See Figure 7a.6.4(c).

Figure 7a.6.4:

(a) (b) (c)



6.6 Unreinforced openings

6.6.1 Openings in cylindrical shells either single or in a definite pattern such as tube holes are to be unreinforced provided the diameter of the opening or the largest diameter in the group of holes does not exceed that permitted by the following formula:

$$d = 8{,}08 \cdot \sqrt[3]{D_o \cdot t_a \cdot (1-k)}$$

where:

d = maximum allowable diameter of opening, mm,

D_o = outside diameter of the shell, mm,

t_a = actual thickness of the shell, mm,

k = factor as determined from the following formula but not more than 0,99

$$k = \frac{P \cdot D_o}{1{,}82 \cdot S \cdot t_a}$$

S = maximum allowable stress, N/mm²,

P = design pressure, bar.

6.6.2 No unreinforced opening is to exceed 200mm in diameter.

6.6.3 For elliptical or oval holes, d refers to the major axis when this lies longitudinally or to the mean of the major and minor axes when the minor axis lies longitudinally.

6.7 Reinforced openings

6.7.1 Openings either single or in a definite pattern are to be reinforced in accordance with SECTION 10 provided the diameter of opening is greater than that permitted by the formula in 6.6.1.

SECTION 7 Spherical Shells Subject to Internal Pressure

7.1 Minimum thickness

7.1.1 The minimum thickness of a spherical shell is to be determined by the following formula:

$$t = \frac{D_o \cdot P}{40 \cdot S \cdot E + P}$$

where:

P = Design pressure, bar,

D_o = Outside diameter of the shell, mm,

S = Maximum allowable stress, N/mm²

E = Efficiency factor. See 6.4.

7.1.2 The formula in 7.1.1 is applicable only where $D_o/t > 8$.

SECTION 8 Dished Ends Subject to Internal Pressure

8.1 Scope

8.1.1 The following rules apply to the design of unstayed, dished-end plates under internal pressure.

8.1.2 The dished plates may be semi-ellipsoidal, torispherical or hemispherical. Referring to Fig.7A.8.(1), the following symbols are defined where applicable:

D_o = external diameter of the parallel portion of the end, mm,

R_i = internal radius of the dished end, mm,

r_i = internal knuckle radius, mm,

H = external height, mm.

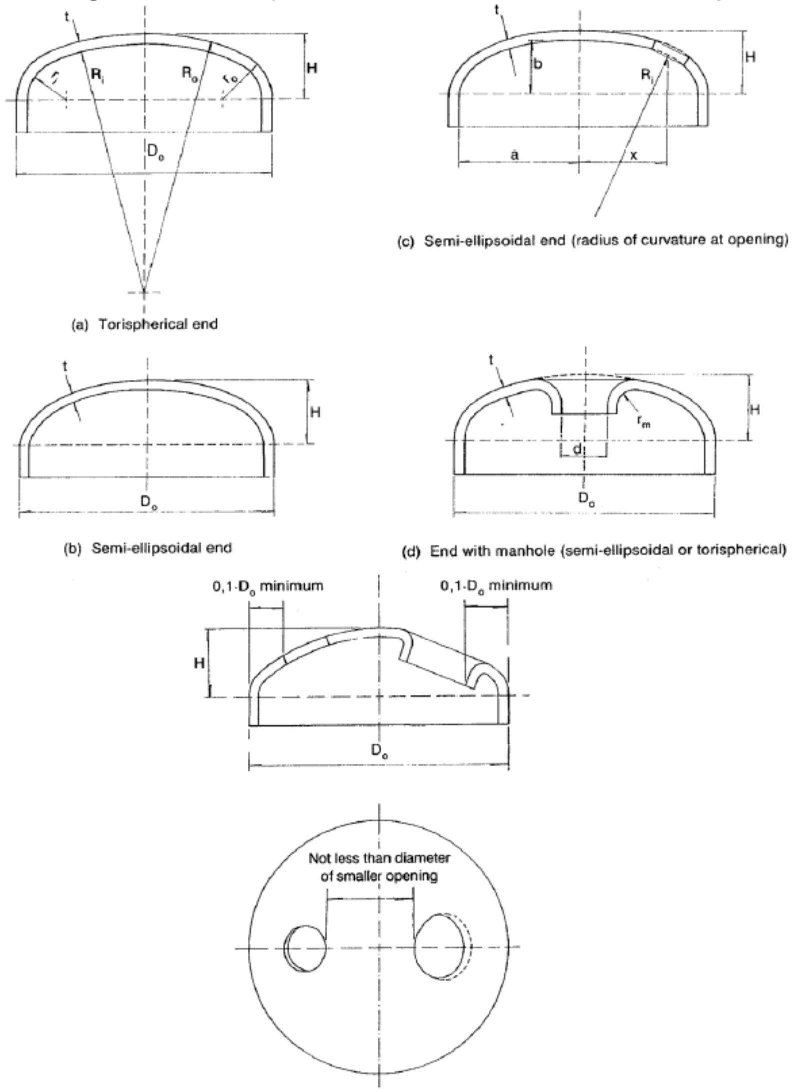
8.1.3 Concerning the geometry the following requirements are to be applied:

$Ri \le Do$

$r_i \ge 0{,}1 \cdot D_o$ and $r_i \ge 3 \cdot t$

$H \ge 0{,}18 \cdot Do$

Figure 7a.8.1: Typical dished ends and location of opening



8.2 Minimum thickness

8.2.1 The minimum wall thickness, t, of semi-ellipsoidal, torispherical and hemispherical unstayed ends, dished from plate, having pressure in the concave side and satisfying the conditions stated above, is to be determined as follows:

$$t = \frac{P \cdot D_O \cdot K}{40 \cdot S \cdot E} + c, \quad \text{mm}$$

where:

P = Design pressure, bar,

D_o = Outside diameter of the shell, mm,

S = Maximum allowable stress, N/mm

E = Efficiency factor

c = corrosion allowance, mm

K = shape factor, see 8.3.

8.2.2 The wall thickness, t, of the dished end is to be not less than the thickness derived from the formula:

$$t = \frac{P \cdot R_i}{20 \cdot S \cdot E - 0{,}5 \cdot P} + c \quad [mm]$$

where:

R_i = the internal radius of the cylindrical portion.

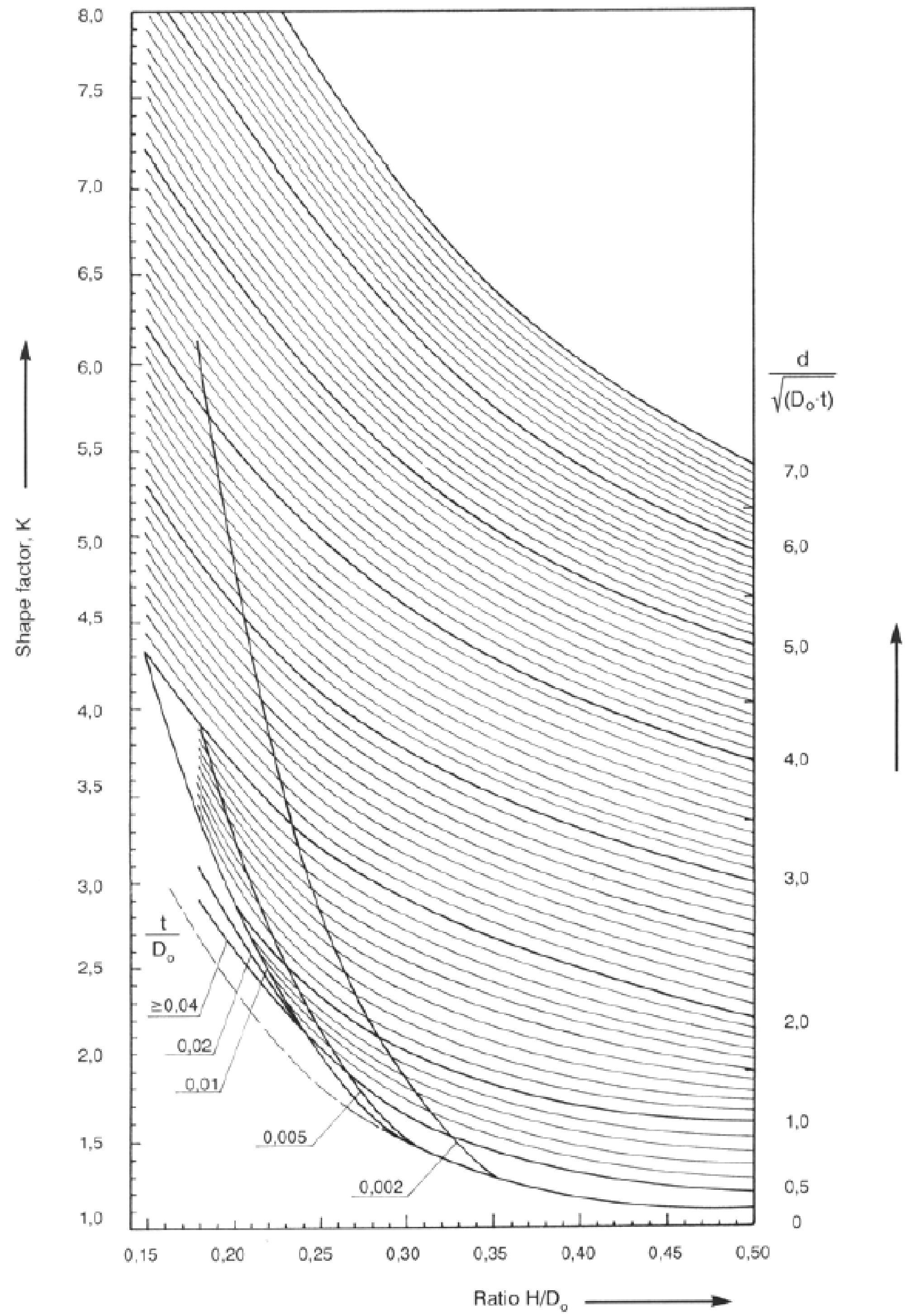
8.2.3 The wall thickness of dished-end plates is not to be less than 6mm.

8.2.4 For ends which are butt welded to the drum shell, the thickness of the edge of the flange for connection to the shell is to be not less than the thickness of an unpierced seamless or welded shell, whichever is applicable, of the same diameter and material and determined by 6.3.

8.3 Shape factor, K, for dished ends

8.3.1 The shape factor, K, to be used in the determination of wall thickness of dished end-plates is to be taken from Figure 7a.8.2 as a function of the ratio H/D_O. This figure provides K either for unpierced ends or for ends with openings. The lowest curves correspond to the former. For values of H/D_O lower than 0,35, K depends on the t/D_O as well. In these cases, a trial calculation is necessary, supposing an initial value of t, using the figure to find a K and calculate a new t.

Figure 7a.8.2: Values of shape factor K for the design of dished ends



8.4 Dished end with unreinforced openings

8.4.1 Dished ends may have openings either circular or approximately elliptical.

8.4.2 The upper curves in Figure 7a.8.2 provide values of K to be used in 8.2.1 for ends with unreinforced openings. The following procedure is to be applied:

– Suppose a thickness, t, meaning the minimum thickness after dishing.
– Calculate the value of:

$$\frac{d}{\sqrt{(D_o \cdot t)}}$$

where:

d = diameter of the largest opening in the end plate, mm (in the case of an elliptical opening, the larger axis of the opening),

D_o = outside diameter of dished end, mm

– Use Figure 7a.8.2 to arrive at a K.
– Then calculate a new t using the formula in 8.2.1

8.4.3 The following requirements must in any case be satisfied:

$$\frac{t}{D_o} \leq 0{,}1$$

$$\frac{d}{D_o} \leq 0{,}7$$

8.4.4 From Figure 7a.8.2 for any selected ratio of H/D_o, the curve of unpierced ends gives a value for $\frac{d}{\sqrt{D_o \cdot t}}$ as well as for K. Openings giving a value of $\frac{d}{\sqrt{D_o \cdot t}}$ not greater than the value so obtained, may be pierced through an end designed as unpierced without any increase in thickness

8.5 Flanged openings in dished ends

8.5.1 Either flanged or unflanged openings are to conform to 8.3. No reduction may be made in the plate thickness on account of flanging.

8.5.2 Where openings are flanged, the radius r_m of the flanging is to be not less than 25mm. The thickness of the flanged portion may be less than the calculated thickness.

8.6 Location of unreinforced and flanged openings in dished ends.

8.6.1 Unreinforced and flanged openings in dished ends are to be so arranged that the distance from the edge of the hole to the outside edge of the plate and the distance between openings, are not less than those shown in Figure 7a.8.1.

8.7 Dished ends with reinforced openings

8.7.1 Where it is desired to use a large opening on a dished end of thickness smaller than that would be required by 8.3, the end is to be reinforced as prescribed in SECTION 10.

8.7.2 The shape factor, K, for a dished end having a reinforced opening can be read from Figure 7a.8.2 using as parameter the value obtained from:

$$\frac{d_o - \frac{A_r}{t}}{\sqrt{(D_o \cdot t)}}$$

instead of

$$\frac{d}{\sqrt{(D_o \cdot t)}}$$

where:

A_r = the effective cross-sectional area of reinforcement as defined in 10.7,

d_o = external diameter of ring or standpipe, mm.

SECTION 9 Conical Ends Subject to Internal Pressure

9.1 General

9.1.1 These Rules apply to the design of conical ends and conical reducing sections subject to internal pressure.

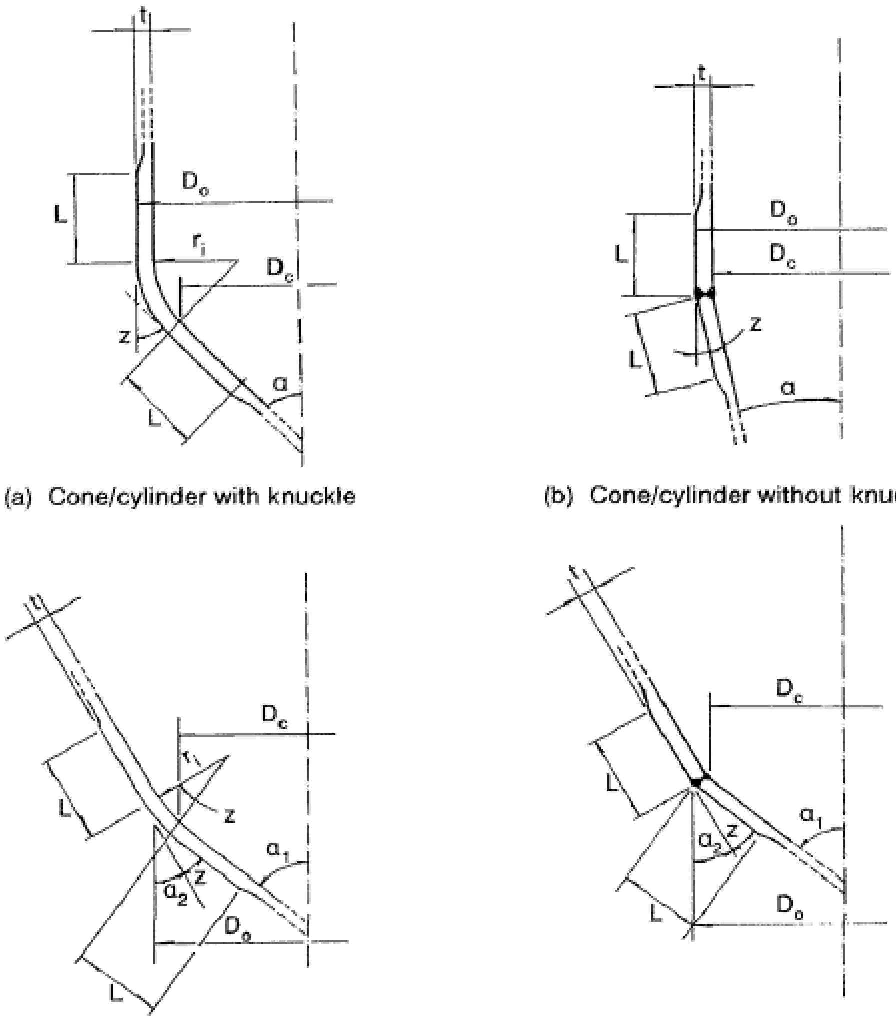
9.1.2 Connections between cylindrical shells and conical sections or conical ends or conical sections and conical ends are recommended to be by means of a knuckle transition radius, r. However, as an alternative, conical sections and ends may be butt welded to the cylindrical shells or other conical sections and ends without a knuckle radius where the change in angle of slope, z, between the two elements under consideration does not exceed 30 degrees. The weldment is to be carried out on both sides. Typical permitted details are shown in Figure 7a.9.1.

9.1.3 Conical ends may be constructed of several ring sections of decreasing thickness, as determined by the corresponding decreasing diameter.

9.1.4 The thickness of conical sections having an angle of inclination to the vessel axis of more than 75 degrees is to be determined as for a flat plate.

9.1.5 Holes and openings in conical walls are to be reinforced according to SECTION 10.

Figure 7a.9.1:



(a) Cone/cylinder with knuckle (b) Cone/cylinder without knuckle

(c) Cone/cone with knuckle (d) Cone/cone without knuckle

9.2 Minimum thickness

9.2.1 The minimum thickness, t, is to be not less than the greater of the values derived from the following formulae:

$$t_1 = \frac{P \cdot D_o \cdot K}{40 \cdot S \cdot E} + c \quad [mm]$$

where:

t_1 = the minimum thickness of the cylindrical shell, knuckle and conical section at the junction with the cylinder or other conical section.

t_2 = the minimum thickness of those parts of conical section not less than a distance, L, from the junction with a cylinder or other conical section,

P = design pressure, bar

D_o = outside diameter of the conical section, mm; see Figure 7a.9.1,

D_c = inside diameter of conical section taken where it is shown in Figure 7a.9.1,

S = allowable stress, N/mm²,

K = shape factor, taking into account the stress in the knuckle, see Table 7a.9.1,

r = knuckle transition radius, mm; to derive the value of K in the case of but welded elements r is to be taken as $0,01 \cdot D_o$,

E = efficiency factor; where the distance of a circumferential seam from the knuckle or junction is not less than L, then E = 1; otherwise, E is to be taken as the efficiency factor of the circumferential seam,

L = distance, mm, from knuckle or junction within which meridional stresses determine the required thickness, see Figure 7a.9.1.

$$L = 0{,}5 \cdot \sqrt{D_o \cdot \frac{1}{cosz}}$$

z = difference between angle of slopes of two adjoining conical sections, see Figure 7a.9.1, a, a1,a2=angle of slope of conical section; see Figure 7a.9.1,

c = corrosion allowance and wear, mm

Table 7a.9.1: Values of K depending on Z and r/Do

Z,deg	**Shape factor K at r/D_o**											
	0,01	0,02	0,03	0,04	0,06	0,08	0,10	0,15	0,20	0,30	0,40	0,50
10°	1,4	1,3	1,2	1,1	1,1	1,1	1,1	1,1	1,1	1,1	1,1	1,1
20°	2,0	1,8	1,7	1,6	1,4	1,3	1,2	1,1	1,1	1,1	1,1	1,1
30°	2,7	2,4	2,2	2,0	1,8	1,7	1,6	1,4	1,8	1,1	1,1	1,1
40°	4,1	3,7	3,3	3,0	2,6	2,4	2,2	1,9	1,8	1,4	1,1	1,1
60°	6,4	5,7	5,1	4,7	4,0	3,5	3,2	2,8	2,5	2,0	1,4	1,1
75°	13,6	11,7	10,7	9,5	7,7	7,0	6,3	5,4	4,8	3,1	2,0	1,1

SECTION 10 Reinforcement of openings in cylindrical and spherical shells or conical and dished ends

10.1 General

10.1.1 Openings in elements such as cylindrical and spherical shells or conical and dished ends are to be reinforced where the opening dimensions are larger than the maximum permitted ones for unreinforced elements mentioned in the corresponding Sections.

10.1.2 The reinforcement requirements apply to openings whose largest dimension does not exceed 500mm. Reinforcement of larger openings requires special consideration by LHR.

10.1.3 The reinforcement of an opening may be one of the following types or combination of them:

(1) Compensation of opening by providing surplus thickness. See Figure 7a.10.1.

(2) Reinforcement of opening by means of reinforcing plates (pads) attached by welding to the wall to be reinforced. See Figure 7a.10.2.

(3) Reinforcement of opening by means of welded-on tubular elements, such as nozzles, sleeves, branch pieces etc. See Figure 7a.10.3.

Figure 7a.10.1:

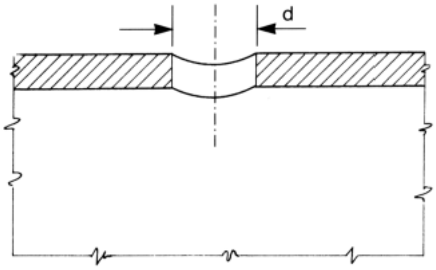


Figure 7a.10.2:

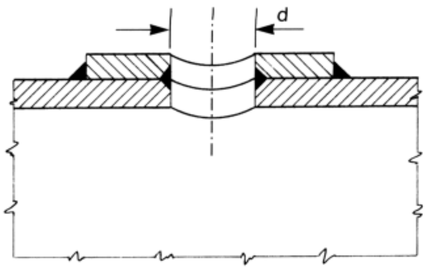
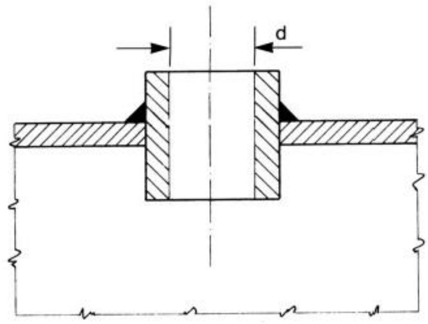


Figure 7a.10.3:



10.1.4 Forged reinforcements may be used. The materials used for the wall to be reinforced and for the reinforcing elements are to have identical strength characteristics as far as possible. Where use is made of reinforcing materials with inferior strength characteristics compared to those of the wall to be reinforced, the area of the reinforcing sections is to be increased accordingly. See 10.8.

10.1.5 The welds attaching standpipes and reinforcing plates to the shell are to be of sufficient size to transmit the full strength of the reinforcing area and all other loadings to which they may be subjected.

10.2 Reinforcement requirements

10.2.1 Reinforcement is to be provided in amount and distribution such that the area requirements for reinforcement are satisfied for all planes through the centre of the opening and normal to the shell or end plate surface. The total cross-sectional area of reinforcement is to be not less than obtained by the formula:

$$A = d \cdot t_r \cdot F$$

where:

A = Required reinforcement area, mm^2,
d = Diameter of the finished opening in the given plane, mm

t_r = Required thickness exclusive of the corrosion allowance of a seamless unpierced shell or end, mm,
F = Correction factor which compensates for the variation in pressure stresses on different planes with respect to the axis of the shell. A value of 1,00 is to be used for all configurations, except those values determined from Figure 7a.10.4 may be used for integrally reinforced openings in cylindrical shells and cones.

10.3 Limits where the reinforcement is to be extended

10.3.1 The limits of reinforcement, measured along the pressure vessel wall, are to be at a distance on each side of the axis of the opening equal to the greater of the following requirements:

- the diameter of the finished opening
- the radius of the finished opening plus the thickness of the container wall, plus the thickness of the nozzle wall.

10.3.2 The limits of reinforcements, measured normal to the pressure vessel wall, are to conform to the contour of the surface at a distance from each surface equal to the smaller of the following requirements:

- 2,5 times the shell thickness,
- 2,5 times the nozzle wall thickness, plus the thickness of any added reinforcement exclusive of the weld metal on the side of the shell under consideration.

10.4 Metal having reinforcing value as surplus thickness

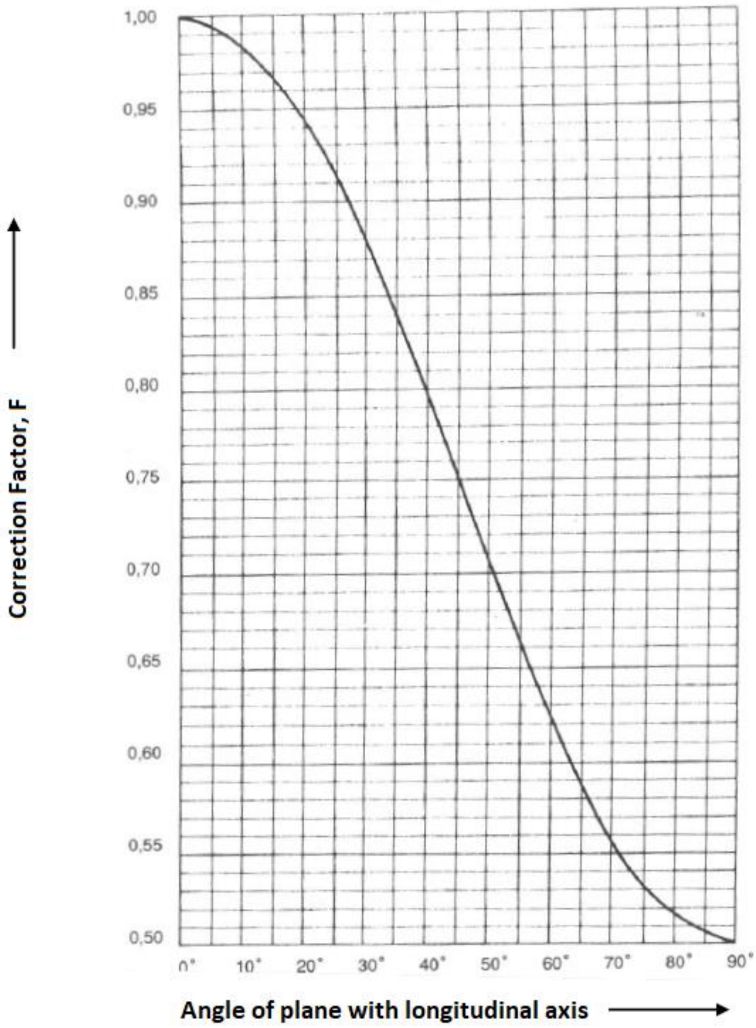
10.4.1 The metal in the wall additional to the thickness required to resist pressure and exclusive of the corrosion allowance may be considered as reinforcement within the reinforcing limits given in 10.3. The area of the additional thickness, available as reinforcement is to be the greater of the following:

$$A_1 = (E \cdot t_a - F \cdot t_r) \cdot d$$

or

$$A_1 = 2 \cdot (E \cdot t_a - F \cdot t_r) \cdot (t_a + t_N)$$

Figure 7a.10.4:



where:

A_1 = Area in the excess thickness in the pressure vessel wall available for reinforcement, mm².

E = 1,00 when an opening is in the plate or when the opening passes through a circumferential joint in a shell (exclusive of head to shell joints). Where any part of the opening passes through any other welded joint, E is to be the longitudinal joint efficiency.

t_a = Actual thickness of the pressure vessel wall, exclusive of the corrosion allowance, mm

t_r = Required thickness exclusive of the corrosion allowance of a seamless unpierced shell or end, mm.

t_N = Thickness of nozzle wall exclusive of the corrosion allowance, mm.

d Diameter in the plane under consideration of the finished opening, mm.

F = Factor from Figure 7a.10.4.

10.5 Reinforcement available in nozzles.

10.5.1 The metal additional to the thickness required to resist pressure, and exclusive of the corrosion allowance in that part of a nozzle wall extending outside the pressure vessel wall, may be considered as reinforcement within the reinforcement limits given in 10.3. The maximum area on the nozzle wall available as reinforcement is the smaller of the values of A_2 given by the following formulae:

$$A_2 = 5 \cdot (t_N - t_{NREQ}) \cdot t_a$$

or

$$A_2 = (t_N - t_{NREQ}) \cdot (5 \cdot t_N - 2 \cdot t_e)$$

where:

A_2	=	area in excess thickness in the nozzle wall available for reinforcement, mm^2,
t_a	=	actual thickness of the pressure vessel wall, exclusive of the corrosion allowance, mm
t_r	=	required thickness exclusive of the corrosion allowance of a seamless unpierced shell or end, mm
t_N	=	thickness of nozzle wall exclusive of the corrosion allowance, mm,
t_{NREQ}	=	required thickness of a seamless nozzle wall, mm, found by the formula used for the shell, less the corrosion allowance
t_e	=	thickness of reinforcing element, mm.

10.5.2 The thickness of a nozzle neck used in a reinforced opening is not to be less than the smaller of the following:

- thickness of the shell or head,
- thickness of standard-wall pipe for pipe nozzles or the required minimum thickness of a nipple based on 41,1 bar internal pressure.

10.5.3 The metal exclusive of the corrosion allowance in the nozzle wall extending inside the pressure vessel and within the reinforcing limits are to be included. The area on the nozzle wall available as reinforcement is the smaller of the values of A_3 given by the following formulae:

$$A_3 = 5 \cdot t_{Na} \cdot t_a$$

or

$$A_3 = 5 \cdot t_{Na} \cdot t_N$$

where :

A_3	=	area of the nozzle wall inside the pressure vessel, mm^2,
t_{Na}	=	actual thickness of the nozzle wall exclusive of the corrosion allowance inside the pressure vessel, mm
t_a	=	actual thickness of the pressure vessel wall exclusive of the corrosion allowance, mm,
t_N	=	thickness of nozzle wall exclusive of the corrosion allowance, mm.

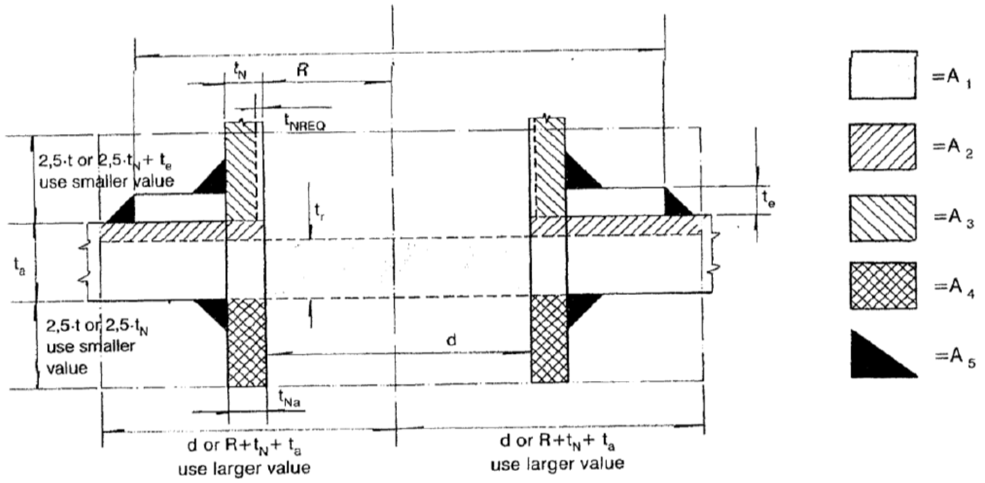
10.6 Added reinforcement

10.6.1 Metal added as reinforcement may be considered as reinforcement when extending to the reinforcement limits.

10.7 Reinforcement area

10.7.1 The reinforcement area, A, is the sum of A_1, A_2, A_3, A_4 and A_5 as shown in Figure 7a.10.5.

Figure 7a.10.5:



10.8 Strength of reinforcement

10.8.1 Where material of lower strength compared to the pressure vessel material strength is used to reinforce the opening as a nozzle or reinforcing plate, the area of reinforcement A_2, A_3 or A_4, whatever applicable, is to be multiplied by the ratio:

$$\frac{allowable\ stress\ of\ nozzle\ or\ reinforcing\ plate\ at\ design\ temperature}{allowable\ stress\ of\ pressure\ vessel\ material\ at\ design\ temperature}$$

Where material of higher allowable stress is used to reinforce the opening, no credit is to be taken for the additional strength.

10.8.2 Deposited weld metal used as reinforcement is to be credited with an allowable stress value equivalent to the weaker of the materials connected by the weld.

10.9 Reinforcement of multiple openings

10.9.1 Two adjacent openings are to have a distance between centres not less than 1,3 times their average diameter.

10.9.2 When adjacent openings are so spaced that their limits of reinforcement overlap, the opening is to be reinforced in accordance with 10.2 with a reinforcement that has an area equal to the combined area of the reinforcement required for the separate openings. No portion of the cross section is to be considered as applying to more than one opening or be evaluated more than once in a combined area.

SECTION 11 Stay Tubes, Stay Bars and Stay Bolts

11.1 General

11.1.1 These Rules apply to longitudinal stays, gusset stays, stay tubes, stay bolts and stiffening girders of steel or copper.

11.1.2 Symbol definitions

P = Design pressure, bar.

F = Load on a stay tube, stay bar or stay bolt, N.

A_p = Plate area supported by one stay tube, stay bar or stay bolt, mm².

A_{stay} = Calculated required cross-sectional area of stay tubes, stay bars and stay bolts, mm².

d_o = Outside diameter of stay tube, stay bar or stay bolt, mm.

d_i = Inside diameter of stay tube, mm.

S = Allowable stress of stay material, N/mm².

11.2 Design

11.2.1 The necessary cross-sectional area is given by the formula:

$$A_{stay} = \frac{F}{S}$$

where:

$$F = P \cdot A_p$$

11.2.2 The plate area to be supported per stay, A_p, is to be the plate area bounded by lines passing at right angles through the centres of the lines joining the centre of the stay with the adjacent points of support. The cross-sectional areas of the stay tubes, stay bars and stay bolts within this area are to be deducted from the above-mentioned area. Where the boundary areas of flanged end-plates are concerned, calculation of the plate area, A_p, is to be based on the flat surface extending to the beginning of the end-plate flange. In the case of flat end plates, up to half the load may be assumed to be supported by the directly adjacent boiler wall. Where there are no stay tubes in the tube nest, the area to be supported by a bar stay is to extend to the tangential boundary of the tube nest.

11.2.3 The minimum thickness for stay tubes is 5mm at its thinnest part. The thickness of stay tubes welded to tube plates is to be such that the maximum allowable stress, S, on the thinnest part of the tube is to be 69 N/mm².

11.2.4 The minimum diameter of stay bars is to be 19mm. Moreover, the minimum diameter of longitudinal stays at any section is not to be less than 25mm.

11.2.5 The maximum allowable stress, S, in combustion chamber and other similar stays, calculated on the minimum sectional area, is to be 62 N/mm².

11.2.6 The maximum allowable stress, S, in longitudinal stays, on the minimum cross-sectional area, is to be:

$$\frac{minimum\ specified\ tensile\ strength}{5{,}3} \quad [N/mm^2]$$

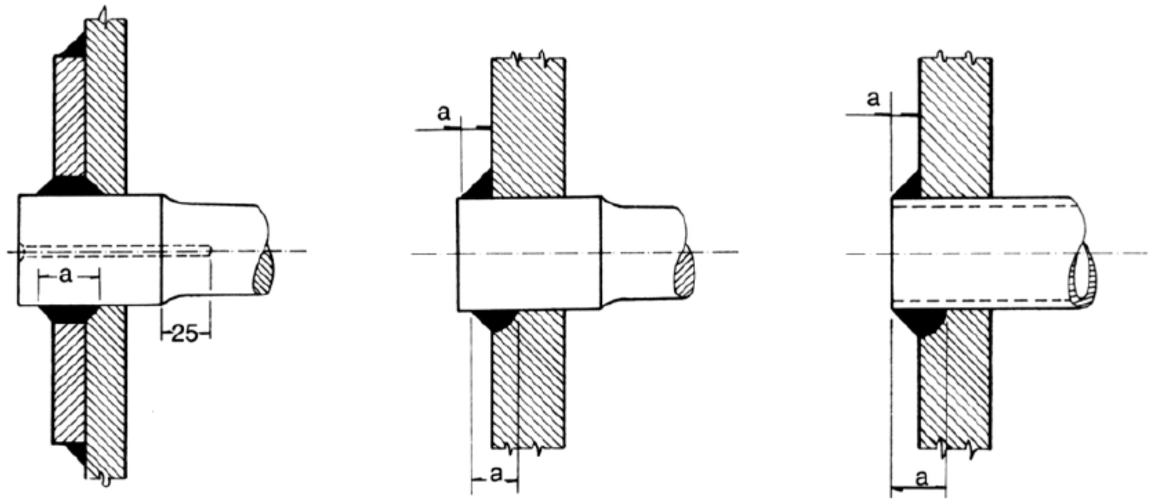
11.3 Welded connections

11.3.1 Examples of welded stay tube attachments are shown in Figure 7a.11.1.

11.3.2 Where longitudinal stays, stay tubes or stay bolts are welded in, the cross-section of the fillet weld subject to shear is to be at least 1,25 times the required cross-section, i.e.:

$$d_o \cdot \pi \cdot a \geq 1{,}25 \cdot A_{stay}$$

Figure 7a.11.1:



SECTION 12 Flat Surfaces

12.1 General

12.1.1 These Rules apply to stayed and unstayed flat, flanged or unflanged, circular, rectangular or elliptical end-plates, which are simply supported or welded at their periphery and which are subjected to internal or external pressure.

12.1.2 Definitions

P = Design pressure, bar.

t = Minimum thickness, mm.

D = Inside diameter of a flat, flanged, end-plate, mm.

D_C = Design diameter, mm.

D_B = Bolt-hole circle diameter of a plate, mm.

r = Inner corner radius of a flange, or radius of a stress relieving groove, mm.

K = Design coefficient. See 12.6.

S = Allowable stress, N/mm^2.

c = Allowance for corrosion and wear, mm.

a,b = Clear supporting or design widths of rectangular or elliptical plates, be always designates the shorter side or axis

y = Ratio, See 12.5.

12.2 Design calculations of unstayed flat surfaces

12.2.1 The minimum thickness of circular flat end-plates unsupported by stays, flanged or unflanged, is derived from the formula:

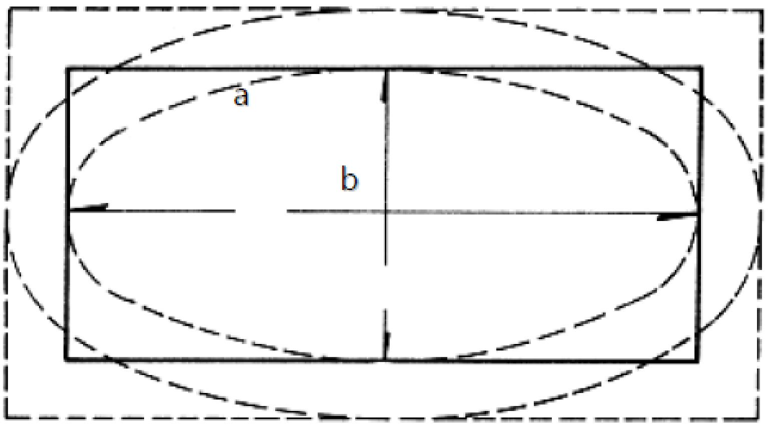
$$t = K \cdot D_C \cdot \sqrt{\frac{P}{10 \cdot S}} + c$$

12.2.2 The minimum thickness of rectangular and elliptical flat end-plates is derived from the following formula:

$$t = K \cdot y \cdot b \cdot \sqrt{\frac{P}{10 \cdot S}} + c$$

The designation of a and b is shown in Figure 7a.12.1. Values for y are to be obtained from 12.5.

Figure 7a.12.1:



12.2.3 Several types of flat unstayed end-plates are given in Table 7a.12.1 with the corresponding D_C designations and K values. The types which may be either circular or rectangular or elliptical plates are notified. LHR may give its consent to the use of other types provided that satisfactory strength calculations are submitted to prove the corresponding K value.

12.2.4 Welding-neck end plates are the cases 9, 10 shown in Table 7a.12.1. Where there is a stress relieving groove (case 10), the formulas in 12.2.1 and 12.2.2 are to be applied. Additionally, the following conditions are to be satisfied:

(1) for circular end-plates:

$$\frac{0{,}13}{S} \cdot P \cdot \left(\frac{D_C}{2} - r\right) < t_G < 0{,}77 \cdot t_a$$

(2) for rectangular end-plates:

$$\frac{0{,}13}{S} \cdot P \cdot \frac{\mathrm{a.b}}{a + b} < t_G < 0{,}55 \cdot t_a$$

where:

t_a = the actual pressure vessel thickness, mm

t_G = the thickness in the stress relieving groove, mm.

The minimum value of t_G is to be 5mm. Radius r is to be at least $0{,}2 \cdot t$ and not less than 5mm. Where welding-neck end-plates as in cases 9 and 10, are manufactured from plates, the area connected with the shell is to be tested for laminations, e.g. by ultrasonic methods.

Table 7a.12.1 Flat end surfaces

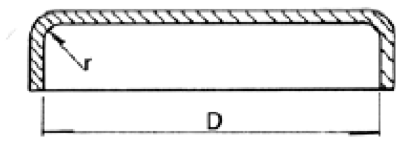
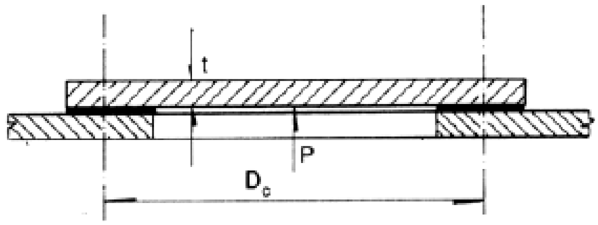
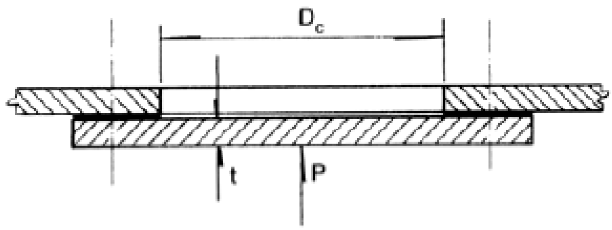
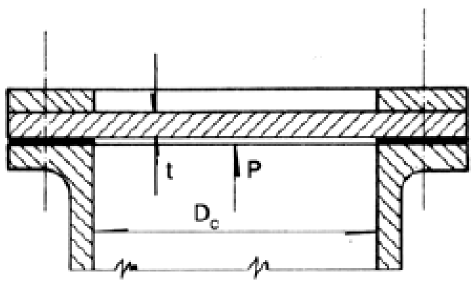
Unstayed flat surface	D_c: Design diameter, mm	K: design coefficient
case 1  circular	$D_c = D - r$	0,35
case 2  circular or rectang. or ellip.	D_c	0,42
case 3  circular or rectang. or ellip.	D_c	0,42
case 4  circular or rectang. or ellip.	D_c	0,36

Table 7a.12.2 Flat end surfaces (continuation)

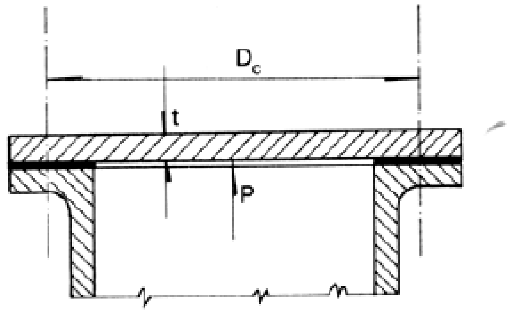
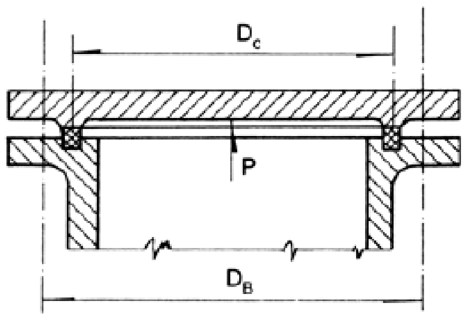
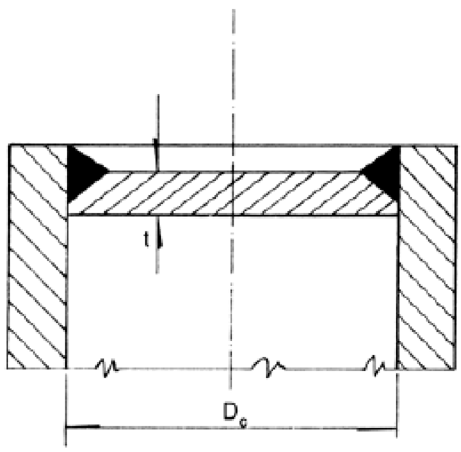
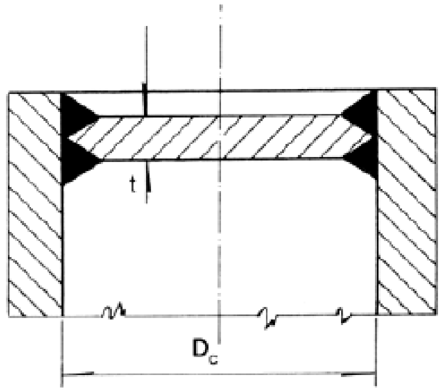
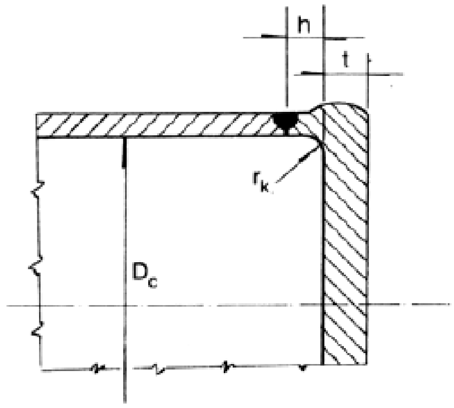
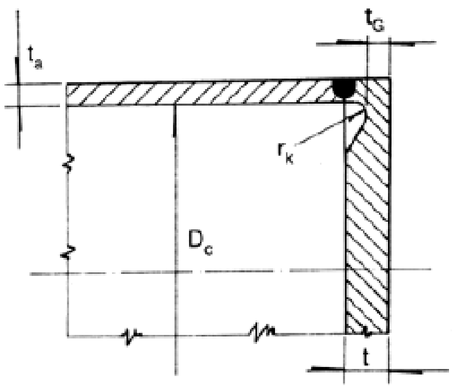
Unstayed flat surface	D_c: Design diameter, mm	K: design coefficient
case 5  circular or rectang. or ellip.	D_c	0,36
case 6  circular or rectang. or ellip.	D_c	D_B/D_c=1,0 K=0,45 D_B/D_c=1,1 K=0,50 D_B/D_c=1,2 K=0,55 D_B/D_c=1,3 K=0,60
case 7  circular or rectang. or ellip.	D_c	0,45

Table 7a.12.3 Flat end surfaces (continuation)

Unstayed flat surface	D_c: Design diameter, mm	K: design coefficient
case 8  circular or rectang. or ellip.	D_c	0,36
case 9  circular or rectang. or ellip.	D_c	0,40
case 10  circular or rectang. or ellip.	D_c	0,40

12.3 Design calculations of stayed flat surfaces

12.3.1 The thickness, t, of flat plates supported by stays bars, stay bolts or stay tubes is to be derived from the following formulae:

(1) when flat surfaces are uniformly supported by stay bars, stay bolts, stay tubes:

$$t = K \cdot \sqrt{(p_1^2 + p_2^2) \cdot \frac{P}{10 \cdot S}} + c$$

where:

p_1, p_2 = pitches shown in Figure 7a.12.2

(2) when flat surfaces are non-uniformly supported by stay bars, stay bolts, stay tubes:

$$t = K \cdot \sqrt{\frac{e_1 + e_2}{2} \cdot \frac{P}{10 \cdot S}} + c$$

where:

e_1, e_2 = stay to stay distances shown in Figure 7a.12.3.

(3) When flat surfaces are supported by corner stays or other means and flat plates between arrays of stays and tubes, the greater of the values of the following formulae is to be applied:

$$t = K \cdot d_c \cdot \sqrt{\frac{P}{10 \cdot S}} + c$$

Or

$$t = K \cdot y \cdot b \cdot \sqrt{\frac{P}{10 \cdot S}} + c$$

where:

d_C = diameter of the largest circle which can be inscribed in the free unstiffened area, mm; see Figure 7a.12.4,

b = shorter length of a rectangle which can be inscribed in the free unstiffened area, mm.

12.3.2 For flat annular plates with central longitudinal staying the minimum thickness, t, is given by the formula:

$$t = K \cdot (D_1 - D_2 - r_1 - r_2) \cdot \sqrt{\frac{P}{10 \cdot S}} + c$$

where:

D_1, D_2, r_1, r_2 are shown in Figure 7a.12.5. For K values see 12.6.

Figure 7a.12.2:

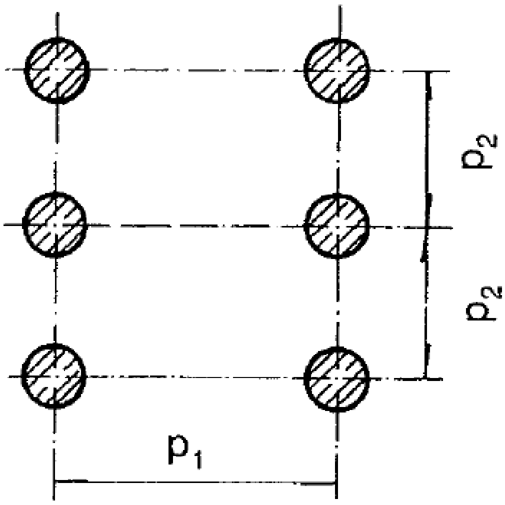


Figure 7a.12.3:

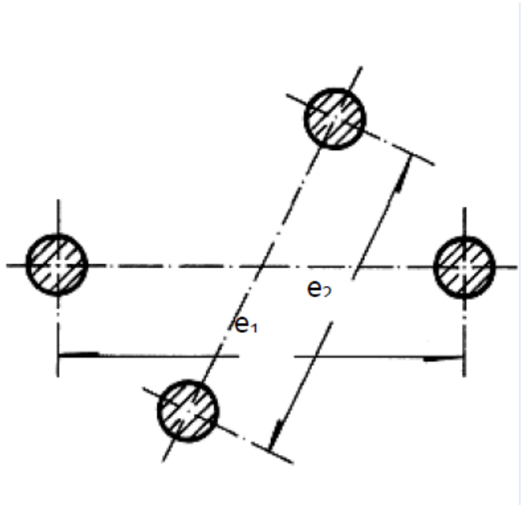


Figure 7a.12.4:

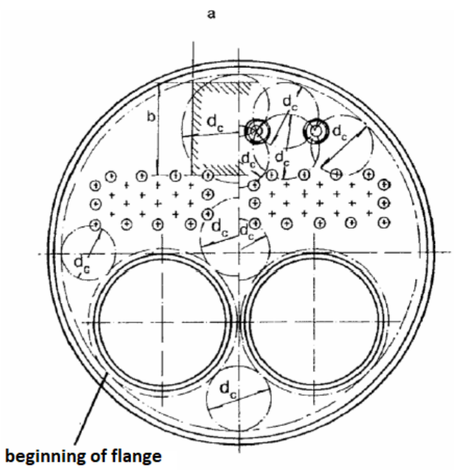
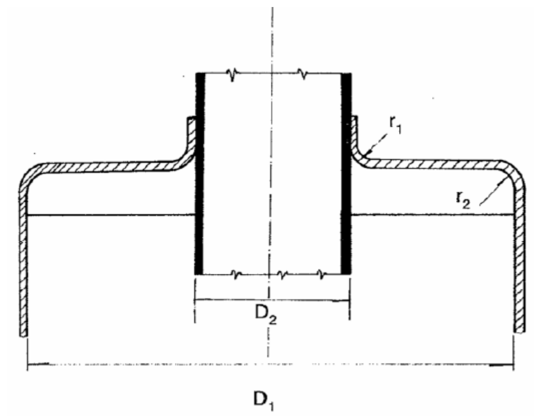


Figure 7a.12.5:



12.4 Additional requirements for flanged end-plates

12.4.1 The above formulae are to be applied to flanged end-plates provided that the corner radii, r, of the flanges should have the minimum values in relation to the outside diameter of the end-plate mentioned in Table 7a.12.4. The flange radii must be equal to at least 1,3 times the wall thickness.

12.4.2 In the case of welding-neck end-plates without a stress relieving groove for headers, the flange radius must be r>1/3 t, subject to a minimum of 8mm, and the inside depth of the end-plate must be h ≥ t, t for end-plates with openings being the thickness of an unpierced end-plate of the same dimensions.

Table 7a.12.4:

Outside diameter of end plate (mm)	Corner radius of flanges (mm)
up to 500	30
over 500 up to 1400	35
over 1400 up to 1600	40
over 1600 up to 1900	45
over 1900	50

12.5 Ratio y

12.5.1 The ratio y takes into account the increase in stress, as compared with circular plates, depending on the ratio b/a of unstayed, rectangular and elliptical plates and of the rectangles inscribed in the free unstayed areas of stayed, flat surfaces. Values for y are to be obtained from Table 7a.12.5 as a function of b/a.

Table 7a.12.5:

Type of plate	Ratio b/a				
	1,0	0,75	0,5	0,25	· 0,1
Rectangle	1,10	1,26	1,40	1,52	1,56
Ellipse	1,00	1,15	1,30	-	-

Note:

Intermediate values are to be interpolated lineary

12.6 Design coefficient, K

12.6.1 Design coefficient, K, depends on the type of support, the edge connection and the type of stiffening. Table 7a.12.6 provides K values for stayed flat surfaces.

Table 7a.12.6:

Type of stiffening and/or plate	**K**
Annular, flat-end plates flanged on both sides and with a longitudinal central stay	0,25
Stay bolts in arrays with maximum stay bolt centre distance of 200mm	0,40
Round stays and tubes outside tube arrays irrespective of whether they are welded-in, bolted or expanded	0,45
Boiler shell, header or combustion chamber wall, stay plate or tube area calculated according to the largest circle of diameter d which can be inscribed or the side length b of the largest rectangle which can be inscribed	0,35

12.6.2 Where different values of coefficient K are applicable to parts of a plate, an average value K is to be used in the calculations.

12.7 Minimum and maximum wall thickness

12.7.1 The minimum plate thickness is 12mm. The wall thickness of flat end-plates should not exceed 30mm in the heated portion.

12.8 Reinforcement of openings

12.8.1 Where the edges of opening are not reinforced, special allowance is to be made when calculating thickness for cut-outs, branches etc. in flat surfaces which lead to undue weakening of the plate.

SECTION 13 Headers of Rectangular Section

13.1 Minimum thickness

13.1.1 The following formula determines the minimum required thickness of the ligaments between the tube holes in a row on a header. Where there are several different rows of holes the minimum required thickness is to be determined for each row. The greatest calculated wall thickness is to govern the thickness of the entire header.

$$t = \frac{P \cdot h}{20 \cdot S \cdot E} + \sqrt{\frac{4{,}5 \cdot z \cdot P}{10 \cdot S \cdot E_1}}$$

where:

P = design pressure, bar,

t = thickness, mm,

S = allowable stress, N/mm^2,

h = half of the width of the header perpendicular to the side being calculated, mm, see Figure 7a.13.1,

g = half of the width of the header parallel to the side being calculated, mm; see Figure 7a.13.1,

z = calculated from the following formula:

$$z = \frac{1}{3} \cdot \frac{g^3 - h^3}{(g + h)} - \frac{1}{2} \cdot (g^2 - l^2)$$

When z has a negative value, the sign is to be disregarded when incorporating z in the formula determining the thickness

l = distance between the row of holes under consideration and the centre line of side,

E, E_1 = tube holes efficiency factors. See 13.2.

13.2 Tube hole efficiency factors

13.2.1 Where there is only one row of holes or where there are several parallel rows not staggered, the efficiency factors E and E_1 are as follows:

$$E = \frac{p-d}{p}$$

$$E_1 = E = \frac{p-d}{p}, \qquad for\ holes\ with\ d < 0{,}6\,m$$

$$E_1 = \frac{p-0{,}6}{p}, \qquad for\ holes\ with\ d \geq 0{,}6\,m$$

where:

p = pitch of holes, mm,

d = tube hole diameter, mm.

13.2.2 For the determination of E, E_1 for elliptical holes, d is to be considered as the clear width of the holes in the longitudinal direction of the header. However, for the purpose of deciding which formula is to be used for determining E_1, the value of d in the expressions d ≥ 0,6 m and d <0,6 m is to be the clear width of the hole perpendicular to the longitudinal axis.

13.2.3 For staggered rows of holes, when determining E_1, p is to be substituted by p_1 for the oblique ligaments. See Figure 7a.13.2. For staggered holes the value of z is that determined by applying the formula in 13.1 with l = 1 and by multiplying by cos(a).

Figure 7a.13.1:

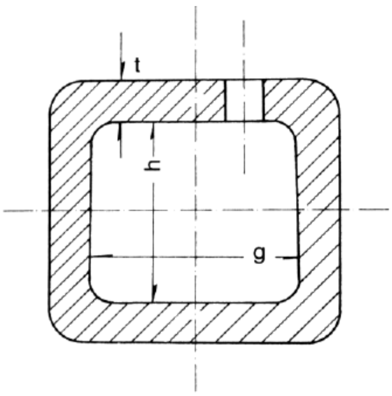
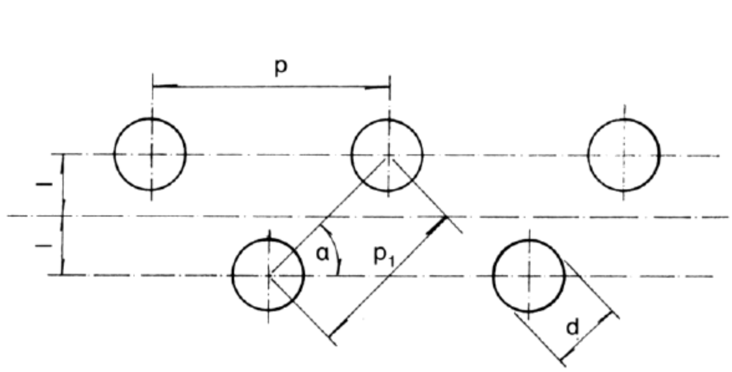


Figure 7a.13.2:



13.3 Stresses at corners

13.3.1 To avoid undue stresses at corners, the following conditions are to be satisfied:

r ≥ 1,2·t,

subject to a minimum of:

- 3mm for rectangular tubes with a clear width of up to 50mm
- 8mm for rectangular tubes with a clear width of 80mm or over.

Intermediate values are to be taken by linear interpolation. The radius is to be governed by the arithmetical mean value of the nominal wall thicknesses on both sides of the corner. The wall thickness at corners may not be less than the wall thickness determined by applying the formula in 13.1.

13.4 Minimum wall thickness and ligament width

13.4.1 The minimum wall thickness for expanded tubes is to be 14mm.

13.4.2 The width of a ligament between two openings or tube holes may not be less than ¼ of the distance between the tube centres.

SECTION 14 Straps and Girders for Combustion Chamber Crowns

14.1 General

14.1.1 These rules apply to steel girders welded to the combustion chamber crowns.

14.2 Design calculations

14.2.1 The girders, welded to the combustion chamber crowns, are to be treated as simply supported beams of length l. The required section modulus of a combustion chamber crown is given by the following formula:

$$SM = \frac{M_{max}}{1{,}3 \cdot S \cdot z} \le \frac{b \cdot h^2}{6}$$

where:

SM = section modulus of one girder, mm³,

M_{max} = bending moment on one girder, Nmm,

b = thickness of girder, mm; see Figure 7a.14.1;

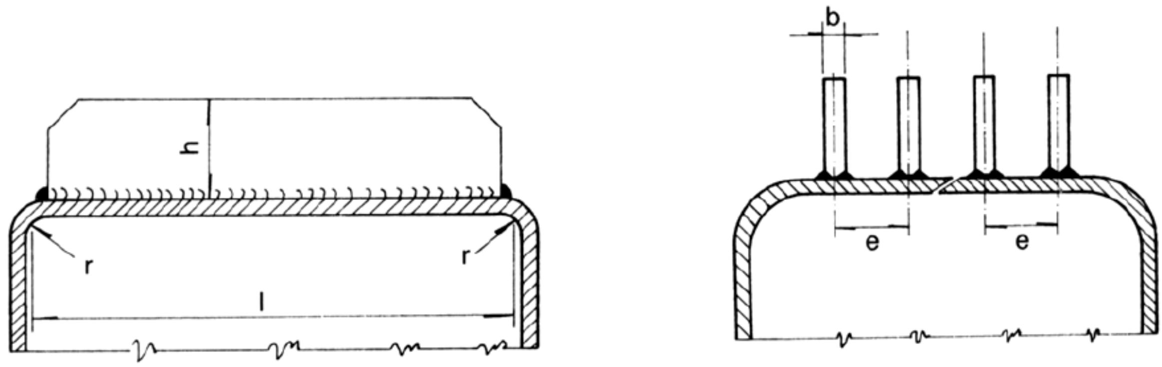
h = depth of girder, mm; see Figure 7a.14.1;

z = coefficient,

S = allowable stress, N/mm².

The coefficient z for the section modulus takes account of the increase in the section modulus due to the combustion chamber wrapper plate forming part of the girder, it may be taken as z =5/3. The depth h is not to exceed 8·b.

Figure 7a.14.1:



14.2.2 The maximum bending moment is given by the formula:

$$M_{max} = \frac{F \cdot l}{8}$$

where:

P = design pressure, bar,

F = load carried by one girder, N,

F = $\frac{P}{10} \cdot l \cdot e$

l = length between girder supports, mm; see Figure 7a.14.1;

e = distance between centre lines of girders, mm.

14.3 General welding requirements

14.3.1 The girders are to be welded to the combustion chamber crowns at all points. The positioning of girders is to be in such a way that the welding can be competently executed and the circulation of water is not obstructed.

SECTION 15 Cylindrical Furnaces Subject to External Pressure

15.1 Corrugated furnaces

15.1.1 The minimum thickness of corrugated furnaces is to be derived from the following formula:

$$t = \frac{P \cdot D_e}{F} + c$$

where:

P	=	external pressure, bar,
D_e	=	external diameter of the furnace measured at the bottom of corrugations, mm,
t	=	thickness of the furnace plate measured at the bottom of corrugations, mm,
F	=	1060, for Fox, Morrison and Deighton corrugations
	=	1130, for Suspension Bulb corrugations.

15.2 Plain furnaces, flue sections and combustion chamber bottoms

15.2.1 The minimum thickness, t, of plain furnaces or furnaces strengthened by stiffening rings, or flue sections and of the cylindrical bottoms of combustion chambers is to be the greater of the values derived from the following formulae:

$$t = \sqrt{\frac{P \cdot D_o \cdot (L + 610)}{102400}} + c \quad [mm]$$

$$t = \frac{z \cdot P \cdot D_o}{1100} + \frac{L}{320} + c \quad [mm]$$

where:

P = design pressure, bar,

D_o = external diameter of the furnace, flue or combustion chamber, mm,

L = length of section between the centres of points of substantial support, mm.

z = $\frac{2 \cdot S_1}{S_1 + S_2}$

S_1 = Specified minimum 0,2% proof stress in N/mm², at a temperature 90°C above the saturated steam temperature corresponding to the design pressure for carbon and carbon-manganese steel with a specified minimum tensile strength of 400 N/mm².

S_2 = Specified minimum 0,2% proof stress in N/mm², at a temperature 90°C above the saturated steam temperature corresponding to the design pressure, of the steel actually used.

15.2.2 For tubes exposed to fire, the length L between two stiffeners is not to exceed 6·Do. The greatest unsupported length is not to exceed 6m or, in the first pass from the front end-plate, 5m.

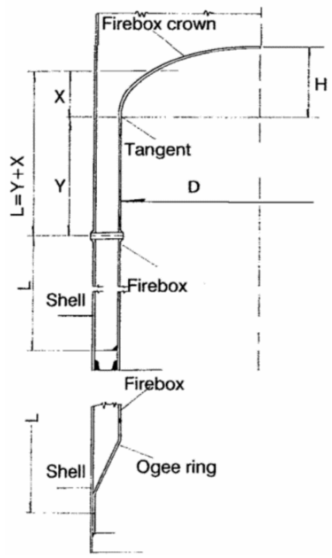
15.3 Maximum thickness

15.3.1 The maximum acceptable thickness of plain or corrugated furnaces is 22mm.

15.4 Plain furnaces of vertical boilers

15.4.1 For plain furnaces with an external diameter not greater than 1700mm, the thickness is to be determined as in 15.2, where D_O is taken equal to the external diameter of the furnace, in mm, and, L is taken equal to the effective length, in mm, of the furnace between the points of substantial support as in Figure 7a.15.1. Where the furnace is tapered, the diameter to be used in the formula is to be the mean of that at the top and that at the bottom where it meets the substantial support from flange, ring or row of stays.

Figure 7a.15.1



H/D	X/D
0,169	0,07
0,2	0,08
0,25	0,1
0,3	0,12
0,4	0,16
0,5	0,2

15.4.2 For furnaces under 760mm in external diameter, the thickness is to be not less than 8mm, and for furnaces 760mm in external diameter and over, the thickness is to be not less than 9,5mm. Furnace exceeding 1700mm in external diameter will be the subject of special consideration.

15.4.3 A circumferential row of stays connecting the furnace to the shell will be considered to provide substantial support to the furnace, provided that the diameter of the stay is not less than 22,5mm or twice the thickness of the furnace, whichever is the greater. In the case of screwed stays the diameter is to be measured over the threads. The pitch of the stays at the furnace does not exceed 14 times the thickness of the furnace.

15.5 Hemispherical furnaces

15.5.1 Unsupported hemispherical furnaces subject to pressure on the convex side must have a minimum thickness determined by the following formula:

$$t = \frac{z \cdot P \cdot R_o}{608} + c$$

where:

R_O = outer radius of curvature of the furnace, mm

z = $\frac{2 \cdot S_1}{S_1 + S_2}$

S_1 and S_2 as defined in 15.2.1.

15.5.2 The minimum thickness t, is not to exceed 22,5mm nor the ratio.

15.6 Uptakes of vertical boilers

15.6.1 The minimum thickness, t, of internal uptakes of vertical boilers is to be determined by the following formulae, the greater of the two thicknesses obtained to be used:

$$t = \sqrt{\frac{P \cdot D_o \cdot (L + 610)}{102400}} + 4 \quad [mm]$$

$$t = \frac{P \cdot D_o}{1100} + \frac{L}{320} + 4 \quad [mm]$$

where:

D_o = external diameter of uptake, mm,

L = length of uptake between the centers of points of substantial support, mm.

SECTION 16 Boiler Tubes Subject to External Pressure

16.1 Minimum thickness

16.1.1 The minimum thickness t is given by the formula:

$$t = \frac{P \cdot D_o}{205 + P}$$

where:

P = design pressure, bar,

T = thickness, mm,

D_o = outside diameter of tube, mm,

S = allowable stress, N/mm².

16.2 Allowable stress

16.2.1 The allowable stress is:

$$t = \frac{R_{e,t}}{2}$$

where:

$R_{e,t}$ = the specified lower yield stress $R_{e,l}$ or proof stress $R_{e,0,2}$ at the design temperature, in N/mm².

16.3 Wall thickness of intensely heated boiler tubes

16.3.1 The wall thickness of intensely heated boiler tubes (e.g. where the temperature of the heating gas exceeds 800°C) is not to be greater than 6,3mm.

SECTION 17 Mountings

17.1 General requirements of valves and fittings

17.1.1 Valves and fittings for boilers are to be made of ductile materials, able to withstand the loads imposed in operation as well as thermal loads or stresses due to vibration. Valves and fittings are to be of steel suitable for the expected service temperature. However, where the working pressure is not greater than 10 bar and the steam temperature not greater than 220°C, the use of cast iron may be permitted by LHR. Nevertheless, valves and fittings which are subjected to shock loads e.g. safety valves or blow-down valves, are not to be made of grey cast iron.

17.1.2 All boiler mountings are to be fitted on welded-on branches, nozzles and pads, generally secured thereto by bolts or studs. Where mountings are attached to the boilers by means of studs not perforating the boiler plates, the depth of the thread is to be at least a stud diameter. Where the studs penetrate the whole thickness of the plate, they should have a nut on the inside. Where bolts are used, they are to be fitted with the head on the inside of the boiler.

17.1.3 All valves are to be closed by a clock-wise rotation of their handles. They are to be located and connected in such a manner that it readily can be seen whether they are shut-off or not.

17.2 Safety valves - general

17.2.1 Each boiler or unfired steam generator is to be provided with at least two spring-loaded safety valves. The orifice diameter in way of each safety valve seat is not to be less than 38mm. Where the total heating surface does not exceed 45mm², LHR may permit one safety valve having an orifice diameter not less than 50mm.

17.2.2 The safety valves are to be of identical construction and equal size, installed at the saturation steam portion of the boiler drum. They are to be mounted on strong bodies directly connected to the boiler without any stopping device, separated from other valve bodies. Moreover, all the safety valves of each boiler and steam generator may be fitted on one strong body, separated from other valve bodies as well. The cross-sectional area of the orifice of this valve body is not to be less than 1,15 times the aggregate cross-sectional area of the valves installed.

17.2.3 In order to remove the water of condensation a drain pipe is to be provided at the lowest part of the discharge space of the valve body. The condensated water is to be led by gravity to the bilge or to a tank clear of the boilers. The internal diameter of drain pipe is not to be less then 20mm. No valves or cocks are to be fitted on these drain pipes.

17.2.4 Safety valves are to be provided with an effective safeguard to prevent the valves from being adjusted by unauthorized personnel.

17.2.5 Each safety valve is to be fitted with an efficient mechanical means, such as an easing gear, to lift the valve disk from its seat. This mechanism is to be so arranged that the valve is to be operated from a safe position in the boiler or engine room either by hand or by any approved power arrangement.

17.2.6 The springs of the safety valves are to be protected from direct exposure to steam and, consequently, from excessive heating. In order to protect springs, spring casings are to have appropriate vents or other approved arrangements. The springs of the safety valves as well as the sealing surfaces of seats and valves are to be manufactured from heat and corrosion resistant material. Valves are to be so designed that in the event of fracture of springs they cannot lift out of the seats.

17.2.7 Safety valves are to be so designed that the moving parts not only have adequate clearances to ensure complete freedom of movement but also, they must not lock even though subjected to different temperatures. For efficient operation maker's specified clearances are to be maintained. Safety valves are to be so constructed that their setting is not to be altered during operation.

17.2.8 Common blow-off lines shared by a number of safety valves may not unduly impair the discharge capability of the valves.

17.3 Design of boilers safety valves

17.3.1 The minimum aggregate area of the orifices of the safety valves of boilers in way of the seat is to be determined by the formula:

where:

A = the minimum cross-sectional area at the valve seat, for the flow of the steam, mm^2

P = the absolute pressure on which the safety valves has been set, bar,

x = the sound speed in steam of pressure, P and temperature, T equal to the boiler's design temperature, m/sec

E = maximum steam production based on all evaporating surfaces, kg/hour. In no case is the value of E to be based on evaporating capacities (referred to evaporating surfaces of the concerned boiler) less than:

$$14\frac{kg}{m^2 \cdot hour}, \qquad for\ exhaust\ gas\ heated\ boilers$$

$$29\frac{kg}{m^2 \cdot hour}, \qquad for\ oil\ fired\ boilers$$

a_w = flow coefficient, determined experimentally. It is the ratio of actual flow over the theoretical flow resulting for constant speed over the area A.

17.4 Safety valves on superheaters

17.4.1 Each superheater is to be provided with at least one safety valve. One safety valve must be located at the superheater's outlet. This valve may be mounted on the superheater's outlet fitting. Nevertheless, the boiler and superheater valves are to be so located and their relieving capacity so proportioned, between saturated steam drum and superheater outlet that sufficient steam is being forced through the superheater to prevent damage to the heater under all service conditions.

17.4.2 Where a superheater is fitted as an integral part of the boiler, without any intervening valve between them, the relieving capacity of the superheater safety valve(s), based on the reduced pressure at the superheater outlet, may be included as part of the total relieving capacity required for the boiler. The relieving capacity of the superheater safety valve(s) is not to be credited for more than 25% of the total capacity required. Where the National Authority of the country in which the ship is to be registered sets forth requirements to limit the proportion of the superheater safety valve(s) capacity to the total capacity required, this limit should be taken into account, provided that sufficient quantity of steam is circulated through the superheater to avoid damages.

17.4.3 Where it is proposed to fit full bore safety valves pilot-operated from the steam drum, the valves are to be submitted for consideration.

17.4.4 When designing the safety valves for superheaters, the following formula is applicable:

$$A = \frac{x \cdot E_s}{a_w \cdot P} \cdot 360$$

where:

A, a_w, x and P are as defined in 17.3.1,

E_S = the steam relieving capacity of the valves fitted at the superheater, kg/h, as defined in 17.4 and 17.4.2.

17.5 Pressure setting on boiler and superheater safety valves

17.5.1 All safety valves are to be set under steam at the presence of an LHR Surveyor to a pressure not greater than the maximum pressure for which the boiler has been approved. A working tolerance of 3% of the approved pressure is given to the lifting of safety valves. However, in no case, is the final pressure to be greater than that for which the piping and machinery are designed.

17.5.2 When setting the superheater safety valves, the pressure drop through the superheater is to be taken in account so that, when the safety valves are relieving steam the pressure on the boiler will not exceed the approved boiler pressure. In no case is the setting pressure to be greater than the maximum pressure in which the piping and machinery of the superheater have been designed.

17.6 Accumulation tests

17.6.1 After the safety valves have been set, a steam accumulation test of 15 minutes, in case of fire tube boilers, and of 7 minutes. In case of water tube boilers, is to be carried out under full firing conditions with the steam stop valves closed. During this test the water feed is to be the minimum necessary to maintain the water level in the boiler. The test is considered satisfactory if for the whole duration of the test the pressure in the boiler is not to exceed by more than 10% the maximum working pressure.

17.6.2 LHR may not require the above test in fired boilers where it is possible to compromise the superheater. LHR will consider the waiving of these tests provided that the relative application has been made when the boiler plan and sizes of safety valves are submitted for approval, and that the safety valves are of an approved type for which the capacity has been measured by tests effected according to a procedure approved by LHR and at the presence of a Surveyor or an approved independent authority or when previous experience of accumulation tests indicates that the relieving capacity of the safety valves is adequate for that particular boiler. When it is agreed to waive accumulation tests, it will be required that the valve makers provide a certificate for each safety valve stating its rated capacity at the approved working conditions of the boilers and that the boiler makers provide a certificate for each boiler stating its maximum steam production.

17.6.3 The safety valves are to be found satisfactory in operation under working conditions during the trials of the machinery on board ship.

17.7 Waste steam pipes of safety valves

17.7.1 The cross-sectional area of the waste steam branch of the valve body and the pipe connected there to, is not to be less than 1,1 times the aggregate cross - sectional area of the valves.

17.7.2 The cross-sectional area of the main waste steam pipe is to be at least equal to the combined cross-sectional area of the waste steam pipes discharging into it.

17.7.3 Waste steam pipes are to be so supported and provided with suitable expansion joints, bends or other means to ensure that the bodies of the safety valves are not to be subjected to any appreciable stress.

17.7.4 Waste steam pipes are to be led to the atmosphere. Exhaust steam vents to atmosphere or external drains are not to be led to waste steam pipes. Waste steam pipes are to be provided with means of cleaning at their lowest part.

17.7.5 Silencers fitted on waste steam pipes are to have a free passage area not less than that required for the pipes.

17.7.6 The strength of waste steam pipes and relative silencers and fittings is to be such that they withstand the maximum pressure to which they may be subjected, which in no case is to be taken less than 1/4 of the setting pressure of the safety valves.

17.7.7 The waste steam pipes fitted on the safety valves of each gas heated economizer and exhaust gas heated boiler which may be used as economizer are to be independent of each other.

17.7.8 Waste steam pipes are to be led clear of electric cables and any parts of structure sensitive to heat and likely to distort. Where necessary they are to be effectively insulated.

17.8 Steam stop valves

17.8.1 Each boiler is to be fitted with a main steam stop valve directly secured to the boiler. Where the boiler supplies steam to auxiliary steam piping systems, each separate piping system is to be fitted with an auxiliary steam stop valve directly mounted on the boiler.

17.8.2 To avoid excessive piercing of the steam drum, the various steam stop valves may be mounted on a strong manifold directly connected to the boiler.

17.8.3 Where two or more boilers are connected together, the steam stop valves are to be of the non-return type.

17.8.4 Where two or more boilers are fitted, the essential services are to be supplied from at least two boilers.

17.9 Water level indicators

17.9.1 Each boiler, designed to contain water at a specified level, is to be provided with at least two means of indicating the water level in it, one of which is to be a direct reading gauge glass. The second water level indicator is to be either an additional gauge glass or an equivalent device type approved by LHR.

17.9.2 Continuous-flow boilers are to be provided with two independent warning devices to signal a shortage of water supply to the boiler.

17.9.3 On double-ended cylindrical boilers two water level indicators are to be provided on both ends.

17.9.4 On cylindrical boilers, producing steam not used for main propulsion and having a design pressure not greater than 10 bar, try cocks may be considered as one of the water level indicators.

17.9.5 The water level indicators are to be so fitted that the water level is to be observed despite the movements and inclinations of the ship when at sea. The water level indicators are to be readily accessible and located so that the water level is to be clearly visible. Where cylindrical glass gauges are used, protective devices are to be fitted to safeguard the operating personnel in the event of glass breakage.

17.9.6 The water level indicators may be:

(1) water glass gauges directly attached to the boiler by cocks, not valves, or
(2) water gauges fitted on pillars which are bolted directly to the boiler shell or
(3) water gauges fitted on pillars which are connected to the boiler shell by means of pipes.
These pipes are to be directly fitted to the shell by terminal cocks, not valves. The internal diameters and pipes according to the size of cylindrical boilers are to be as follows:

Cylindrical boiler diameters	Pillar Internal diameters	Connecting pipes diameters
3m ≤ d	d > 63mm	d ≥ 38mm
2,3m < d < 3m	d > 50mm	d > 32mm
d ≤ 2,3 m	d > 45mm	d > 25mm

17.9.7 The connecting pipes are to be so installed without sharp bends and pockets in order to avoid possible water accumulation. Moreover, the connecting pipes are to be protected from exposure to hot gases, radiant heat and intense cooling.

17.9.8 The cocks of all water gauges are to be operable from safe positions in the event of glass breakage.

17.9.9 Each boiler is to be provided with a device, approved by LHR, suitable to give an audible and visual alarm when the water level in the boiler falls below a pre-determined value. In addition, it is to shut-off automatically the fuel supply to the burners. The device is to be independent of any other mounting. However, each boiler which is intended to operate without manual supervision is to have safety arrangements which shut off the fuel supply and give an alarm in the case of low water level, air supply failure or flame failure.

17.9.10 Water tube boilers serving turbine propulsion machinery are to be fitted with a high-water level alarm.

17.10 Lowest water level and water level indicators

17.10.1 The lowest specified water level is to be indicated on every boiler water level indicator with a reference line drawn on the gauge frame. Additionally, the lowest specified water level is to be marked on a plate with a reference line and to be written as "minimum permissible water level". The plate is to be attached to the boiler shell close to the water level indicators. The reference line and the plate should not be covered by insulation.

17.10.2 The following requirements, relating with the highest point of heating surface do not apply to waste-heat boilers, forced-circulation boilers, economizers and steam superheaters. The lowest water level in the boiler is to be not less than 170mm above the highest point of heating surface. This distance is to be maintained even when the ship is listed up to 5 degrees either side and under all possible service trim conditions. The highest point of heating surface is being defined as follows:

(1) in the case of water tube boilers is the upper edge of downcomers connected to the boiler drum,
(2) for horizontal cylindrical boilers it is the combustion chamber top,
(3) for a vertical boiler is the furnace crown.

Cylindrical boilers are to be fitted with a position indicator for the highest point of heating surface which is to be clearly marked adjacent to the water gauge.

17.10.3 Water level indicators are to be so installed that the lower limit at which the water is visible is to be at least 50mm below the lowest specified water level in the boiler and at least 50mm above the highest point of heating surface, where it is applicable. The lowest specified water level in the boiler should not be above the centre line of the visible portion of the water level indicator.

17.11 Feed check valves

17.11.1 Each boiler or auxiliary boiler intended for essential services is to be fitted with two feed check valves, each of them connected to two separate feed water lines. Only one feed check valve may be accepted on exhaust gas heated boilers and on steam heated steam generators.

17.11.2 The valves are to be attached, as far as practicable, directly to the boiler or to the economizer, if the latter is an integral part of the boiler. If this is not practicable, and the arrangements require standpipes between feed check valves and the boilers or economizers, these standpipes are to be of steel or other approved material to withstand shock loading, such as water hammering etc. Where the arrangements require a common inlet pipe on the economizer for both valves and the valves are to be fitted in such a manner that anyone of the two feeding lines can be shut off without excluding the possibility of feeding the boiler.

17.11.3 Where economizers may be by-passed and cut off from the boiler, they are to be fitted with a relief valve, unless the arrangement is such that an excessive pressure may not arise in the economizer.

17.11.4 Feed water is not to be discharged into a boiler in such a manner that it comes in direct contact with the heated part of the boiler. It is to enter the boiler in such a manner as to prevent damage to the boiler surfaces.

17.11.5 At least one of the feed water systems is to have an approved feed water regulator whereby the water level in the boilers is controlled automatically.

17.11.6 The feed check valves are to be fitted with efficient gearing, so that they can be satisfactorily worked from a convenient position as an operating platform.

17.11.7 The feedwater connection to the drum is to be fitted with a sleeve or other suitable device to reduce the effects of metal temperature differentials between the feed pipe and the shell or head of the drum.

17.12 Quality of feedwater and salinometer valve or cock

17.12.1 Each boiler is to be provided with means to supervise and control the salinity of the feed water.

17.12.2 Suitable arrangements are to be provided to preclude, as far as practicable, the entry of oil or other contaminants which may adversely affect the boiler.

17.12.3 Each boiler is to be provided with a salinometer valve or cock attached direct to the shell. This valve or cock is not to be on the water gauge standpipe.

17.13 Blow-down valves or cocks

17.13.1 Each boiler is to be fitted with at least one blow-down valve or cock. Where necessary, a similar valve or cock is to be fitted for scumming from the surface. These valves or cocks are to be secured direct to the lower part of the boiler and to be connected to piping to discharge overboard.

17.13.2 Where, on water tube boilers, it is not practicable to connect the blow-down valves directly to the boiler, the valve may be located just clear of the boiler casing and insulation, at the end of a pipe of substantial thickness suitably supported and protected from the heat of the combustion chamber.

17.13.3 The diameter of valves or cocks and of the connected piping to the sea, is not to be less than 20mm but not more than 40mm. However, the size of blow-down valves on cylindrical boilers may be 0,0085 times the diameter of the boiler.

17.13.4 Where two or more boilers have the blow-down and surface scumming-off valves connected to the same piping, the relevant valves or cocks are to be of the non-return type to prevent the possibility of discharging the water of one boiler to another.

17.14 Pressure and temperature gauges

17.14.1 Each boiler is to be fitted with at least two pressure gauges connected with the steam space by separate pipes fitted with stop valves or stop cocks. Double -ended boilers are to have one such gauge at each end.

17.14.2 Pressure gauges are to be located where they can be easily seen at well-lit places. They are to be well protected from the heat emitted by the hot boiler surfaces.

17.14.3 The pressure gauges are to indicate pressure correctly up to at least 1,5 times the pressure at which the safety valves are set. The pressure gauge scale is to have a red line to mark the working pressure of the boiler.

17.14.4 Temperature gauges are to be fitted to superheater outlets and/or between the individual superheated stages and, in particular, to cooler inlets and outlets and to continuous flow outlets where it is necessary to judge the behaviour of the materials used.

SECTION 18 Hydraulic Tests

18.1 Hydraulic test on boiler mountings

18.1.1 All valves and fittings are to be subjected to a hydraulic pressure test at twice the approved design pressure before they are fitted. For feed check valves and other fittings connected to the main feed system, the test pressure is to be the greater of 2,5 times the approved boiler design pressure, or twice the maximum pressure which can be developed in the feed line in normal service. During this procedure, the ratio of the yield stress at 20°C to the maximum stress imposed by twice the working pressure is not to fall under 1,1.

18.2 Hydraulic test on boilers

18.2.1 Each boiler or steam generator, after completion of manufacture and attachment of all fittings, and before the insulation is fitted, is to be subjected to a hydraulic pressure test with test pressure equal to 1.5 times the approved boiler design pressure. The test is considered satisfactory when any sign of leakage, defect or permanent deformation will not appear.

18.2.2 Where only some of the component parts are sufficiently accessible to allow proper visual inspection, the hydraulic pressure test may be performed in stages.

18.2.3 Depending on the boiler construction, the following procedure may be applied: all components of boilers and of other steam generators, after completion of manufacture and heat treatment carried out, are to be subjected to a hydraulic test with test pressure equal to 1,5 times the design pressure. The completed boiler after assembly is to be tested to 1,5 times the design pressure. Particularly, in components, such as water or steam drums or headers, which are to be drilled for tube holes, the hydraulic test may be carried out before drilling the tube holes, but in this case has to be repeated after welding is compulsory after the attachment of all fittings and the heat treatment carried out.

SECTION 19 Superheaters, exhaust gas economizers - Additional requirements

19.1 Superheaters

19.1.1 The requirements regarding to the design, material and construction are the same as the applicable requirements for boilers. Concerning the piping equipment, the superheaters are designed for boiler pressure, but, due to the pressure drop through the heater, the pressure at the outlet may permit piping design at a lower pressure than the boiler's pressure, so the outlet flange on the superheater may be made to fit such a standard.

19.1.2 Superheaters are to have valves or cocks fitted to permit drainage of their headers.

19.1.3 Suitable arrangements are to be fitted to the superheater for efficient venting.

19.2 Exhaust gas economizers

19.2.1 The requirements regarding to the design, material and construction are the same as the applicable requirements for boilers. Economizers exposed to the pressure of the feed-pump discharge are to be designed to withstand the increased pressure. The formulae given in the previous Sections are to be applied where P is the maximum permissible working pressure which is the economizer pressure to which the relative safety valve is set.

19.2.2 The inlet size of each economizer is to be provided with a reliable pressure gauge. On its scale the maximum permissible working pressure is to be marked by a red line.

19.2.3 Each economizer is to be fitted with a spring-loaded safety valve, of a diameter at least 15mm, set at the maximum permissible working pressure. Even if the shut off valve between the economizer and the boiler is closed, the maximum permissible working pressure of the economizer is not to be exceeded by more than 10%.

19.2.4 Each economizer is to be fitted with a shut off valve at the feedwater inlet and outlet. The boiler feed valve may be regarded as one of these shut off devices.

19.2.5 Each economizer is to be provided with means of drainage and with vents.

19.2.6 Steam generation must not be carried out in the economizer even when the main engine is suddenly stopped. This may be achieved by installing a by-pass line enabling the economizer to be completely isolated from the flow of exhaust gas.

19.2.7 Before they are installed, finished economizers are to be subjected to a hydraulic pressure test at 1,5 times the maximum allowable working pressure.

SECTION 20 Installation of Boilers

20.1 Location

20.1.1 Boilers are to be far enough from oil tanks so that sufficient room for maintenance of the structure is to be provided and to prevent the temperature in oil tanks from approaching the flash point of the oil.

20.2 Mounting

20.2.1 Boilers must be installed in the ship in such a way that they cannot be displaced under any circumstances.

20.2.2 Boilers and their seatings must be easily accessible for examination and repair.

20.2.3 Means are to be provided to accommodate the thermal expansion of the boiler in service.

20.3 Operation of valves and fittings

20.3.1 Safety valves and valves or cocks in general, are to be so arranged that they will be operated without danger. Where it is not possible for them to be directly reached by hand from the floor plates or a platform, they must be fitted with connecting rods or chains enabling them to be operated from boiler control platforms.

20.4 Name plate

20.4.1 Each boiler must be fitted with a plate, permanently attached and visible even after the insulation has been installed. This plate is to give the following details:

- manufacturer's name and address,
- serial number and year of manufacture,
- maximum allowable working pressure, bar,
- steam generation, kg/h or t/h,
- allowable superheated steam temperature in degrees Celsius if the boiler is equipped with a superheater which cannot be isolated.

CHAPTER 7 (b) Boilers and Other Pressure Vessels

CONTENTS

SECTION 1 General

1.1 Application

1.1.1 The following rules apply to pressure vessels other than those intended for steam raising plants whose design and construction requirements are contained in Part 5, Chapter 7a. These rules apply to pressure vessels involved in the operation of the main propulsion plant and its auxiliary machinery. These pressure vessels may be seamless or fusion welded.

1.1.2 Pressure vessels for the carriage of liquefied gases in bulk are not covered by these rules.

SECTION 2 General Requirements

2.1 General

2.1.1 The general requirements set forth in Part 5, Chapter 7a, SECTION 2 are to be applied on pressure vessels covered by these rules.

2.2 Classes of pressure vessels

2.2.1 Pressure vessels are classified as in Table 7b.2.1 according to operating conditions.

2.2.2 Where the pressure vessels are not fully filled by liquid but air or gas is contained as well, or which are blown out by air or gases, such as pressure tanks in drinking water or sanitary systems and reservoirs, are to be classified as pressure vessels containing air or gas.

Table 7b.2.1

Operating medium	Design pressure, bar Temperature, ºC		
Liquefied gases (propane, butane, etc.) Toxic gas cargo, refrigerants	all	-	-
Steam,	$p > 16$	$p \le 16$	$p \le 17$
Compressed air	or		
Gases	$T > 300$	$T \le 300$	$T \le 170$
Thermal oil	$p > 16$	$p \le 16$	$p \le 17$
Liquefied fuels	or		
	$T > 150$	$T \le 150$	$T \le 160$
Water and oils	$p > 40$	$p < 40$	$p \le 16$
	or		
	$T > 300$	$T < 300$	$T \le 200$
Pressure Vessel Class	I	II	III

SECTION 3 Principles Applicable to Manufacture

3.1 General

3.1.1 The requirements set forth in Part 5, Chapter 7a, SECTION 3 are to be applied to pressure vessels covered by these rules.

3.2 Identification and marking

3.2.1 Each pressure vessel is to be provided with a plate or another permanent note on which the following are to be mentioned:

- Manufacturer's name and address
- Serial number
- Year of manufacture
- Maximum allowable working pressure
- Capacity

On smaller items of equipment, an indication of the working pressure is sufficient

SECTION 4 Design principles

4.1 Design pressure, P

4.1.1 Design pressure is the maximum allowable working pressure and is to be not less than the highest set pressure of any safety valve.

4.1.2 The pressure which should be used in designing the pressure vessels scantlings, is the design pressure adjusted where necessary to take account of pressure variations corresponding to the most severe operational conditions and the hydrostatic pressure when it exceeds 0,5 bar.

4.2 Design temperature, T

4.2.1 The design temperature is to be taken not less than the mean wall metal temperature expected under operating conditions for the pressure part concerned. It is to be mentioned on the plans when submitted for consideration.

4.2.2 The minimum design temperature, T, is not to be taken less than 50 C.

4.3 Allowable stress, S

4.3.1 The determination of scantlings of pressure parts is to be based on an allowable stress, S.

4.3.2 The allowable stress, S, is to be the minimum of the values corresponding to each material used, as follows:

(1) Rolled and forged steels:

- $$\frac{R_{e,t}}{2{,}7}, \frac{R_{m,20}}{1{,}5}, \frac{S_{r,100000}}{1{,}5}$$

where:

$R_{e,t}$ = Specified lower yield stress, $R_{e,l}$, or 0,2% proof stress at the design temperature, N/mm²,

$R_{m,20}$ = the specified minimum tensile strength at ambient temperature, i.e. 20 C,

$S_{r,100000}$ = the average stress to produce rupture after 100000h at the design temperature, N/mm².

(2) Steel castings
The allowable stress for steel castings is to be taken as 80% of the value determined by the method in (1), using the appropriate values for cast steel. Where steel castings manufactured and tested according to LHR's rules, are also subjected to non-destructive tests, LHR may permit the increasing of allowable stress to 90%, instead of 80%, of that referred in (1).

(3) Modular cast iron:

$$\frac{R_{e,t}}{4{,}8}, \frac{R_{m,20}}{3}$$

(4) Grey cast iron:

$$\frac{R_{m,20}}{4}$$

4.4 Allowance for corrosion and wear

4.4.1 The allowance for corrosion and wear is to be 1mm. To prolong the life of a pressure vessel against corrosion, the corrosion allowance may be increased.

4.5 Pressure parts scantlings

4.5.1 The formulas of Part 5, Chapter 7a are to be used in the design of pressure parts of pressure vessels covered by these rules, in so far as they are applicable, and otherwise according to the general rules of engineering practice.

4.6 Minimum wall thickness

4.6.1 The wall thickness of the shells and end plates of seamless and welded vessels is not to exceed the following value:

$$3 + \frac{internal\ diameter}{1500}$$

SECTION 5 Mountings

5.1 Shut-off device

5.1.1 Each pressure vessel is to be fitted with a shut-off valve located as close as possible to the pressure vessel.

5.1.2 Where several pressure vessels are grouped together, it is not necessary that each pressure vessel is to have its own shut-off device. The whole group may have one shut-off device but not more than three vessels are to be grouped together. Starting air receivers must be capable of being shut-off individually.

5.2 Drainage

5.2.1 Pressure receivers and bottles for compressed air are to be fitted with valves or cocks for water or oil drainage. The drainage devices are to be located as to permit complete drainage in its final position on board.

5.3 Safety valves

5.3.1 Pressure vessels are to be fitted with a safety valve and a pressure gauge.

5.3.2 Where two or more pressure vessels are interconnected by a piping system of adequate size so that no branch of piping may be shut off, it will be sufficient to provide for them one safety valve and one pressure gauge only.

SECTION 6 Hydraulic Tests

6.1 General

6.1.1 Each pressure vessel, after completion of manufacture, is to be subjected to a hydraulic test with a test pressure equal to 1,5 times the design pressure without showing any leakage or failure.

SECTION 7 Gas Cylinders

7.1 General

7.1.1 Gas cylinders are pressure vessels with a maximum capacity of 150 l with an outside diameter not more than 420mm, and a length not more than 2000mm filled with gases in special stations outboard the ship and then brought for use on board.

7.2 Documents

7.2.1 The manufacture of gas cylinders is to be carried out under well-established methods using appropriate materials to withstand the expected loads. LHR is to consider the manufacturing method for approval purposes so the following are to be submitted:

(1) Manufacturing process and quality control description
(2) Chemical composition, yield stress, tensile strength, notch impact strength, heat treatment of the materials used and relative tests and certificates.

7.3 Design calculations

7.3.1 The minimum required thickness for the following parts of a gas cylinder is:

(1) cylindrical surfaces: $t = \frac{D_O \cdot P}{20 \cdot S \cdot E + P}$

(2) dished ends: $t = \frac{D_O \cdot P}{40 \cdot S \cdot E}$

(3) spherical ends: $t = \frac{D_o \cdot P}{40 \cdot S \cdot E}$

where:

P = design pressure equal to the specified test pressure, bar,

t = wall thickness, mm,

D_o = outside diameter, mm,

K = design coefficient for dished ends, see Part 5, Chapter 7a, SECTION 8

E = efficiency factor, see Part 5, Chapter 7a, SECTION 6, 6.4

S = allowable stress = R/1,33, N/mm²,

R = the minimum of the following values:

i. R_e , and
ii. $0{,}75 \cdot R_m$ for normalized or normalized and tempered cylinders or $0{,}85 \cdot R_m$ for quenched and tempered cylinders

R_m = guaranteed minimum tensile strength, N/mm²,

R_e = yield point which is to be either

R_e = $R_{e,L}$, or

R_e = $0{,}92 \cdot R_{e,H}$, or

R_e = $R_{p,0,2}$,

$R_{e,L}$ = guaranteed lower yield point, N/mm²,

$R_{e,H}$ = guaranteed upper yield point, N/mm²,

$R_{p,0,2}$ = guaranteed minimum tensile strength, N/mm².

7.4 Test pressure

7.4.1 For CO_2, the specified test pressure with a filling factor of 0,66 kg/l is 250 bar relative pressure. For other gases the test pressure is to be agreed with LHR.

7.5 Testing of gas cylinders

7.5.1 For normalized cylinders: one sample cylinder from each 200 originating from one melt and one heat treatment. For quenched and tempered cylinders: one sample cylinder from each 100 originating from one melt and one heat treatment batch.

7.5.2 The obligatory mechanical tests are the following:

- one longitudinal tensile test specimen,
- three transverse bending test specimens,
- one set of ISO V-type notched bar impact test specimens from the sample cylinders,
- the notched bar impact test specimens are to be tested at -20 C.

7.5.3 The wall thickness of all sample cylinders is to be measured in transverse planes at three levels: neck, middle and base. The end plate is also to be swan through and the thickness measured.

7.5.4 All the inner surfaces of portions of the sample cylinders are to be examined to detect possible manufacturing defects.

7.5.5 All cylinders submitted for testing are to be subjected to hydrostatic test.

7.5.6 All cylinders are to be visually inspected. The weight and the volumetric capacity of the cylinders is to be verified of about 10% cylinders submitted for testing.

7.5.7 The manufacturer must establish the volumetric capacity and weight of each cylinder. Cylinders which have been quenched and tempered are to be submitted by the manufacturer to 100% hardness testing.

7.5.8 All tests are to be carried out by LHR, or by other bodies provided that the tests are established by them as being equivalent to those prescribed above.

7.6 Marking

7.6.1 Each gas cylinder is to be marked with the following:

– Manufacturer.
– Serial number.
– Gas type.
– Capacity,l.
– Test pressure, bar.
– Empty weight, kg.
– Date of test.
– Test stamp.

In the case of halon cylinders or pressure vessels, the following details are to be marked:

– Maximum permissible charge weight.
– Maximum permissible total pressure after nitrogen super pressurization at 15°C.

CHAPTER 8 Piping General Requirements

CONTENTS

SECTION 1 General (IACS UR P2 (2021))

1.1 Application

1.1.1 This Chapter presents the general requirements which should govern the materials selection, the design and construction of piping systems installed on ships.

1.1.2 Air, vapour, gas (excluding liquefied gas cargo and process piping), water, lubricating oil, fuel oil, hydraulic fluid systems for steering gear, toxic gas and liquids, cargo oil and tank cleaning piping systems together with open ended lines such as drains, overflows, vents and boiler escape pipes are the applications which should follow this Chapter's requirements.

1.1.3 Cargo piping systems of ships carrying chemicals in bulk, piping systems dealing with cargo and processing of liquefied gases are not covered by these requirements. Requirements for open end exhaust gas lines from internal combustion engines and gas turbines are contained in Part 5, Chapter 2, Section 6 and Chapter 10, Section 8.

1.1.4 Requirements for pipes forming the integral part of a boiler are contained in Part 5, Chapter 7. Hydraulic fluid systems other than those for steering gear shall be specially considered in Part 5, Chapter 9, Section 15. Piping systems intended for liquefied gases (cargo and process) are dealt with in Part 4, Chapter 3, Section 4 and Part 2, Chapter 9, Section 3.

1.2 Definitions

1.2.1 The maximum allowable working pressure of a piping system component is the maximum pressure which the component can sustain in continuous use with regard to the materials used, design, working temperature and undisturbed operation.

1.2.2 The design pressure is the maximum allowable working pressure for which a component or a piping system is designed and is not to be less than the pressure at which the safety equipment will become active (e.g. activation of safety valves, opening of return lines of pumps, operating of over pressure safety arrangements, opening of relief valves) or at which the pumps will operate against closed valves.

1.2.3 The design temperature is the maximum temperature of the internal fluid and is not to be taken less than 50°C. In the case of pipes of superheated steam, the temperature is to be taken at the designed operating steam temperature for the pipeline, provided that the temperature as the superheat-er outlet is closely controlled. Where temperature fluctuations exceeding 15°C above the designed temperature are to be expected in normal service, the steam temperature to be used for determining the allowable stress is to be increased by this amount of excess.

1.3 Classes of pipes

1.3.1 For the purpose of testing, the type of joint to be adopted, heat treatment and welding procedure, pipes are subdivided into three Classes as indicated in Table 8.1.1 and Figure 8.1.1.

Table 8.1.1: Classes of pipes

Piping system for	Class I	Class II	Class III
Toxic or corrosive media	without special safeguards	with special safeguards (1, 2)	---
Flammable media heated above flash point or with flash point below 60°C	without special safeguards	with special safeguards (1)	---

liquefied gas (3)			
Steam	P>16 or T>300	Any pressure - temperature combination not belonging to Class I or III	P≤ 7 and T≤170
Thermal oil	P>16 or T>300		P≤ 7 and T≤150
Fuel oil, Lubricating oil, Flammable hydraulic oil	P>16 or T>150		P≤ 7 and T≤ 60
Other media (4, 5)	P>40 or T>300		P≤16 and T≤200

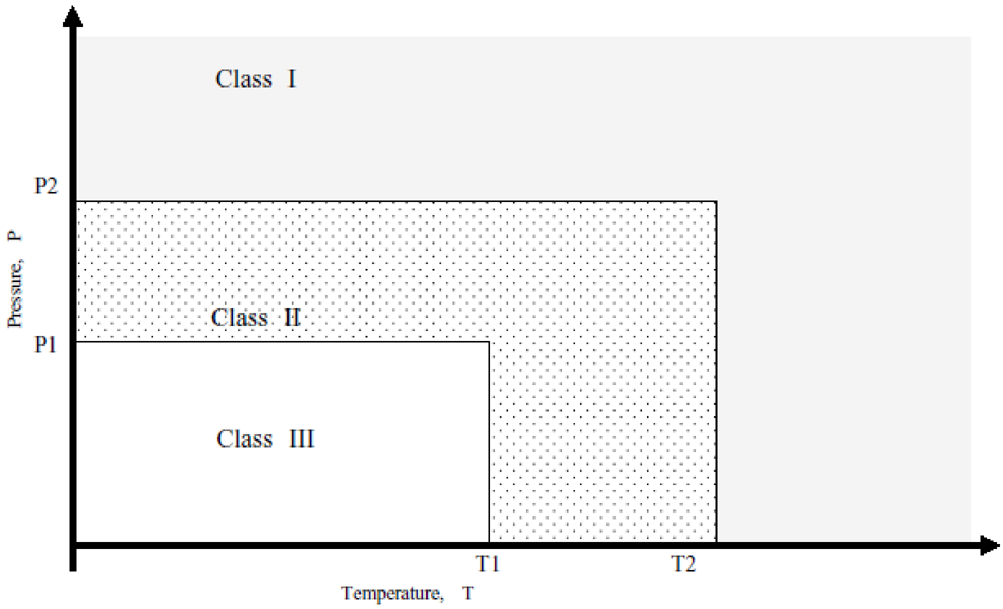
Notes:

P = Design pressure, bar, as defined in 1.2.2

T = Design temperature, °C, as defined in 1.2.3

1. Safeguards for reducing leakage possibility and limiting its consequences: e.g. pipes led in positions where leakage of internal fluids will not cause a potential hazard or damage to surrounding areas which may include the use of pipe ducts, shielding, screening etc.
2. Class II pipes are not to be used for toxic media.
3. Cargo oil pipes belong to Class III.
4. These media include water, air, gases, non-flammable hydraulic oil.
5. Open ended pipes (drains, overflows, vents, exhaust gas lines, boiler escape pipes) irrespective of the temperature T, belong to Class III.

Figure 8.1.1:



SECTION 2 Materials and hydraulic tests of piping systems (IACS UR P2 (2021))

2.1 General

2.1.1 The materials to be used for the various pipes, valves and fittings are to be suitable for the medium and service for which the piping is intended (see 2.2 to 2.5). In the case of especially corrosive media, the materials for the piping system will be considered by LHR in each particular case.

2.1.2 Plastic pipes and flexible hoses are being dealt with in SECTION 5 and SECTION 6 respectively.

2.2 Steel pipes, valves and other fittings

2.2.1 Pipes belonging to Classes I and II are to be seamless drawn steel pipes or pipes fabricated with a welding procedure considered by LHR to be equivalent to the seamless pipes.

2.2.2 Materials for piping systems relating to Class I and II, also for shipside valves and fittings and valves on the collision bulkhead, are to be manufactured and tested in accordance with the appropriate requirements of the LHR's Rules for Materials, Part 2.

2.2.3 Materials for piping systems labelled as Class III are to be manufactured and tested in accordance with the requirements of acceptable national or international specifications. Pipes having forge butt welded longitudinal seams are not to be used for fuel oil systems, for heating coils in oil tanks, or for pressures exceeding 4,0 bar. The manufacturer must provide LHR with the appropriate test certificate.

2.2.4 Generally, carbon and carbon manganese steel pipes, valves and other fittings are not to be employed for temperatures above 400°C. Nevertheless, they may be used for higher temperatures if the proposed materials have been manufactured and tested in accordance with national or international codes or standards and their metallurgical behaviour and mechanical properties such as time dependent strength (UTS after 100000 hours) are in conformity with those standards, and if such values are guaranteed by the steel manufacturer. Otherwise, special alloy steel pipes, valves and fittings should be employed according to the LHR's Rules for Materials.

2.3 Copper and copper alloy pipes, valves and other fittings

2.3.1 Copper and copper alloy piping shall be of seamless drawn material or other type approved by LHR.

2.3.2 Copper pipes for Classes I and II are to be seamless. Materials for piping systems relating to Class I and II, for shipped valves and fittings and valves on the collision bulkhead are to be manufactured and tested in accordance with the requirements of Part 2, Chapter 7 of the LHR's Rules for Materials.

2.3.3 Materials for piping systems labelled as Class III are to be manufactured and tested in accordance with the requirements of acceptable national or international specifications. The manufacturer must provide LHR with the appropriate test certificate.

2.3.4 Generally, copper and copper alloy piping, valves and fittings shall not be used for media having temperature above the following limits:

(1) Copper and aluminium brass: 200°C

(2) Copper nickel alloys: 300°C

Special bronze suitable for high temperature services may be accepted in general up to 260°C.

2.3.5 Brazing and welding materials are to be suitable for the operating temperature and for the medium being carried. All brazing and welding are to be carried out to the satisfaction of the Surveyors.

2.3.6 Pipes which have been hardened by cold bending are to be suitably heat treated on completion of fabrication and prior to being tested by hydraulic pressure. Copper pipes are to be annealed and copper alloy pipes are to be either annealed or stress relief heat treated.

2.4 Nodular cast iron pipes, valves and other fittings

2.4.1 Nodular cast iron of the ferric type according to the requirements indicated in Part 2, Chapter 8 of the LHR's Rules for Materials, may be accepted for bilge, ballast and cargo oil piping.

2.4.2 Ferritic nodular cast iron valves and other fittings may be accepted for media having temperatures not exceeding 350°C. The use of this material for pipes, valves and fittings for other services, in principle Classes II and III, will be subject to special consideration.

2.4.3 Nodular cast iron for pipes and valves fitted on the ship’s side should have specified properties to LHR's satisfaction, according to Regulation 22 of the 1966 International Convention on Load Lines.

2.5 Ordinary cast iron pipes, valves and fittings

2.5.1 Ordinary cast iron pipes, valves and fittings may be accepted in principle for Class III at the LHR's judgement.

2.5.2 Ordinary cast iron piping may be accepted for cargo oil lines within cargo tanks of tankers.

2.5.3 Ordinary cast iron is not to be used for pipes, valves and other fittings handling media having temperature above 220°C and for piping subject to pressure shock, excessive strains and vibrations.

2.5.4 Ordinary cast iron may be accepted for pressures up to 16 bar for cargo oil pipelines on weather decks of oil tankers except for manifolds and their valves and fittings connected to cargo handling hoses.

2.5.5 Ordinary cast iron shall not be used for sea valves and pipes fitted on the ship sides, and for valves fitted on the collision bulkhead.

2.5.6 The use of cast iron for other services will be subject to special consideration in each case.

2.5.7 Grey cast iron pipes and fittings for Class III piping systems are to comply with acceptable national specifications. When used for Class II piping systems, grey cast iron pipes and fittings are to be manufactured and tested in accordance with the requirements of Part 2, Chapter 5 of LHR's Rules for Materials.

2.6 Hydrostatic tests of piping

2.6.1 All Class I and II pipes and integral fittings and, in all cases, all steam pipes, feed pipes, compressed air pipes and fuel oil pipes having a design pressure greater than 3,5 bar and relative integral fittings, after completion of manufacture but before insulation and coating, if, any shall be subject to a hydrostatic test in the presence of the Surveyor at the value of pressure as per 2.6.2.

2.6.2 The test pressure is to be calculated by the following formula:

$$P_t = 1{,}5 \cdot P$$

where:

P_t = test pressure, bar,

P = design pressure as defined in 3.5 of this Chapter, bar.

2.6.3 For steel pipes and integral fittings for temperatures above 300°C, the test pressure is to be derived from the following formula but it is not to exceed 2P.

$$P_t = 1{,}5 \cdot \left(\frac{K_{100}}{K_t}\right) \cdot P$$

where:

K_{100} = permissible stress at 100°C,

K_t = permissible stress at the design temperature

The value of the test pressure may be reduced, with the approval of LHR, to 1,5·P in order to avoid excessive stress in way of bends, T-pieces, etc. In no case is the membrane stress to exceed 90% of the yield stress at the testing temperature.

2.6.4 When the hydrostatic test of piping is carried out on board, these tests may be carried out in conjunction with the test required under paragraph 2.7.

2.6.5 When, for technical reasons, it is not possible to carry out complete hydrotesting before assembly on board, proposals are to be submitted for approval to LHR for testing the closing lengths of piping, particularly in respect of closing seams.

2.6.6 Pressure testing of small pipes (less than about 15 mm) may be waived at the discretion of LHR depending on the application.

2.7 Pressure tests of piping after assembly on board

2.7.1 All the piping systems are to be checked for leakage under operational conditions after assembly on board and, if necessary, using special techniques other than hydrostatic testing, in the presence of the Surveyor.

2.7.2 Heating coils in tanks and liquid or gas fuel lines are to be tested by hydraulic pressure, after assembly on board, to not less than 1,5 times the design pressure but in no case less than 4 bar.

2.7.3 In the case of pipes specified in 2.6.1 being butt welded together during installation on board, they are to be subjected to a hydraulic test in accordance with 2.6 after welding.

2.7.4 The hydraulic test required by 2.7.3 may be omitted provided non-destructive tests by ultrasonic or radiographic methods are carried out on the entire circumference of all butt welds with satisfactory results. The manufacturer is to provide the Surveyor with a signed statement confirming that ultrasonic examination has been carried out by an approved operator containing the results of tests. It is at the discretion of LHR to judge if the indications correspond to defects which could be expected to have prejudicial effect on the service performance of the piping.

2.7.5 Where bilge pipes are accepted in way of double bottom tanks or deep tanks the pipes after fitting are to be tested by hydraulic pressure to the same pressure as the tanks through which they pass.

2.8 Hydrostatic tests on valves and fittings

2.8.1 All valves and fittings non-integral with the piping system, intended for Class I and Class II, are to be tested in accordance with recognised standards, but to not less than 1,5 times the design pressure.

2.8.2 Valves and cocks intended to be fitted on the ship side below the load waterline are to be tested by hydraulic pressure not less than 5 bar.

SECTION 3 Design (IACS UR P1 Rev. 5 (2001))

3.1 Required minimum wall thickness

3.1.1 This requirement is applicable to all piping systems covered by classification unless superseded by other regulation and interpretation of LHR applicable to specific piping systems. Chemical cargo and process piping are excluded from the scope of the present requirement.

3.1.2 The minimum wall thickness of pipes is not to be less than the greater of the values obtained by 3.2 and as indicated in Tables 8.3.1 to 8.3.4.

3.1.1 The required minimum wall thicknesses are not to be less than either those derived by stress analysis or the minimum wall thicknesses assigned to standard pipe sizes as stated in Table 8.3.1, in the case of steel pipes or in Table 8.3.2, in the case of copper and copper alloy pipes.

3.1.2 The minimum wall thicknesses, listed in Table 8.3.1 and Table 8.3.2 are the nominal wall thicknesses and they need no allowance to be made for negative tolerance and reduction in thickness due to bending. The outside diameters and the thicknesses are based on Common International Standards.

For larger diameters, the minimum thicknesses will be specially considered.

3.1.3 For venting, bilge, ballast, fuel, overflow and sounding pipes as listed Table 8.3.1, provided that they are efficiently protected against corrosion, the thickness may be reduced by not more than 1mm, at the discretion of LHR.

3.1.4 Protective coatings such as hot-dip galvanising, may be recognised as an effective corrosion protection for steel pipes provided that the preservation of the protective coating during installation is guaranteed.

3.1.5 Pipes made of stainless steel shall have minimum wall thicknesses determined after special consideration approved by LHR.

3.1.6 The minimum internal diameter for bilge, sounding, venting and overflow pipe shall be:

Bilge pipes:	50mm
Sounding pipes:	32mm
Venting and overflow pipes:	50mm

Table 8.3.1: Minimum wall thicknesses for steel pipes (all dimensions in mm)

Nominal size	Outside diameter D (mm)	Pipes in general (A)	Vent, overflow and sounding pipes for integral tanks (B)	Bilge, ballast and sea water pipes (C)	Bilge, ballast, vent, overflow and sounding pipes passing through fuel tanks. Bilge, vent, overflow, sounding and fuel pipes passing through ballast tanks (D)
6	10,2	1,6			
	12	1,6			
8	13,5	1,8			
10	17,2	1,8			
	19,3	1,8			
	20	2			
15	21,3	2		3,2	
	25	2		3,2	
20	26,9	2		3,2	
25	33,7	2		3,2	
	38	2	4,5	3,6	6,3
32	42,4	2	4,5	3,6	6,3
	44,5	2	4,5	3,6	6,3
40	48,3	2,3	4,5	3,6	6,3
	51	2,3	4,5	4	6,3
50	60,3	2,3	4,5	4	6,3
	63,5	2,3	4,5	4	6,3
	70	2,6	4,5	4	6,3
65	76,1	2,6	4,5	4,5	6,3
	82,5	2,6	4,5	4,5	6,3
80	88,9	2,9	4,5	4,5	7,1
90	101,6	2,9	4,5	4,5	7,1
	108	2,9	4,5	4,5	7,1
100	114,3	3,2	4,5	4,5	8
	127	3,2	4,5	4,5	8
	133	3,6	4,5	4,5	8
125	139,7	3,6	4,5	4,5	8
	152,4	4	4,5	4,5	8,8
150	168,3	4	4,5	4,5	8,8
	177,8	4,5	5	5	8,8
175	193,7	4,5	5,4	5,4	8,8
200	219,1	4,5	5,9	5,9	8,8
225	244,5	5	6,3	6,3	8,8
250	273	5	6,3	6,3	8,8
	298,5	5,6	6,3	6,3	8,8
300	323,9	5,6	6,3	6,3	8,8
350	355,6	5,6	6,3	6,3	8,8
	368	5,6	6,3	6,3	8,8
400	406,4	6,3	6,3	6,3	8,8
450	457,2	6,3	6,3	6,3	8,8

Notes for Table 8.3.1:

1. The nominal sizes, pipe diameters and wall thicknesses given in the table are many of the common sizes based on international standards. Notwithstanding the requirements of Table 8.3.1, diameter and thickness according to other national or international standards may be accepted.
2. Where pipes and any integral pipe joints are protected against corrosion by means of coating, lining etc. at the discretion of LHR, the thickness may be reduced by not more than 1 mm.
3. For sounding pipes, except those for flammable cargoes, the minimum wall thickness in column B is intended to apply only to the part outside the tank.
4. The minimum thicknesses listed in this table are the nominal wall thickness. No allowance needs to be made for negative tolerance or for reduction in thickness due to bending.
5. For threaded pipes, where allowed, the minimum wall thickness is to be measured at the bottom of the thread.

6. The minimum wall thickness for bilge lines and ballast lines through day tanks is be subject to special consideration by LHR. The minimum wall thickness for ballast lines through oil cargo tanks is not to be less than that specified by Chapter 9, Section 14, 14.4.
7. The minimum wall thickness for pipes larger than 450mm nominal size is to be in accordance with a national or international standard and in any case not less than the minimum wall thickness of the appropriate column indicated for 450 mm pipe size.
8. The minimum internal diameter for bilge, sounding, venting and overflow pipe shall be:

Bilge	50mm bore
Sounding	32mm bore
Venting and overflow pipes	50mm bore

9. Exhaust gas pipe minimum wall thickness will be subject to special consideration by LHR.
10. The minimum wall thickness for cargo oil lines will be subject to special consideration by LHR.

Table 8.3.2: Minimum wall thickness for copper and copper alloy pipes

External Diameter D (mm)	Minimum wall thickness (mm)	
	Copper	Copper alloy
8-10	1	0,8
12-20	1,2	1
25-44,5	1,5	1,2
50-76,1	2	1,5
88,9-108	2,5	2
133-159	3	2,5
193,7-267	3,5	3
273-457,2	4	3,5
(470)	4	3,5
508	4,5	4

Note: The external diameters and the thicknesses have been selected from ISO Standards. Diameter and thickness according to other national or international standards may be accepted.

Table 8.3.3: Minimum wall thickness for austenitic stainless steel pipes

External diameter D (mm)	Minimum wall thickness (mm)
10,2 – 17,2	1,0
21,3 – 48,3	1,6
60,3 – 88,9	2,0
114,3 – 168,3	2,3
219,1	2,6
273,0	2,9
323,9 – 406,4	3,6
Over 406,4	4,0

Note: Diameters and thicknesses according to national or international standards may be accepted.

Table 8.3.4: Minimum wall thickness for steel pipes for CO_2 fire extinguishing

External diameter D (mm)	From bottles to distribution station	From distribution station to nozzles
21,3 -26,9	3,2	2,6
30 – 48,3	4	3,2
51 – 60,3	4,5	3,6
63,5 – 76,1	5	3,6
82-5 – 88,9	5,6	4
101,6	6,3	4
108 – 114,3	7,1	4,5
127	8	4,5
133 – 139,7	8	5
152,4 – 168,3	8,8	5,6

Notes:

1. Pipes are to be galvanized at least inside, except those fitted in the engine room where galvanizing may not be required at the discretion of LHR.
2. For threaded pipes, where allowed, the minimum wall thickness is to be measured at the bottom of the thread.

3. The external diameters and thicknesses have been selected from ISO Recommendations R336 for smooth welded and seamless steel pipes. Diameter and thickness according to other national or international standards may be accepted.
4. For larger diameters the minimum wall thickness will be subject to special consideration by LHR.
5. In general the minimum thickness is the nominal wall thickness and no allowance need be made for negative tolerance or reduction in thickness due to bending.

3.2 Calculation of pipe wall thickness

3.2.1 The following formula is to be used for calculating the minimum wall thicknesses of straight or bent pressure pipes subject to internal pressure:

$$t = (t_o + b + c) \cdot \left(\frac{100}{100 - a}\right)$$

where:

t	=	Minimum calculated wall thickness, mm
t_o	=	Thickness calculated by the following basic formula, mm, $t_o = \frac{P \cdot D}{20 \cdot Ke + P}$
P	=	Design pressure, bar
D	=	Outside diameter of pipes, mm
K	=	Permissible stress (from 3.3 and 3.4), N/mm^2
e	=	efficiency factor
b	=	Allowance for bending, mm
c	=	Corrosion allowance (from Tables 8.3.5 and 8.3.6), mm
a	=	Percentage negative manufacturing tolerance, mm

3.2.2 The efficiency factor, E, is to be taken as 1 for seamless pipes and for welded pipes derived by manufacturers approved for making welded pipes which are considered as equivalent to seamless pipes. For other welded pipes LHR will consider an efficiency factor value depending upon the service and the welding procedure.

3.2.3 The value of b, for this allowance, is to be chosen in such a way that the calculated stress in the bend, due to internal pressure only, does not exceed the permissible stress. Where this allowance is not determined by a more accurate procedure, it is to be taken as not less than:

$$b = \frac{D \cdot t_o}{2{,}5 \cdot R}$$

where:

R = mean radius of curvature of a pipe bend, at the centre line of the pipe, mm

3.2.4 The corrosion allowance is to be taken from Table 8.3.5 for steel pipes and Table 8.3.6 for non-ferrous metal pipes.

Table 8.3.5: Corrosion allowance c for steel pipes

Piping Service	c, mm
Superheater steam systems	0,3
Saturated steam systems	0,8
Steam oil systems in cargo tanks	2,0
Feed water for boilers in open circuit syst	1,5
Feed water for boilers in closed circuit systems	0,5
Blow down (for boilers) systems	1,5
Compressed air systems	1,0
Hydraulic oil systems	0,3
Lubricating oil systems	0,3
Fuel oil systems	1,0
Cargo oil systems	2,0
Refrigerating plants	0,3
Fresh water systems	0,8
Sea water systems in general	3,0

Notes:

1. When pipes passing through tanks, an additional corrosion allowance is to be considered according to the figures given in the Table, and depending on the external medium, in order to account for the external corrosion.
2. The corrosion allowance may be reduced where pipes and any integral pipe joints are protected against corrosion by means of coating, lining, etc.
3. When special alloy steel is used with sufficient corrosion resistance, the corrosion allowance may be reduced to zero.

Table 8.3.6: Corrosion allowance c for non-ferrous metal pipes

Pipe material	c, mm
Copper, brass and similar alloys, Copper-tin alloys except those with lead contents	0,8
Copper-nickel alloys (with Ni≥10%)	0,5

NOTES:

For media without corrosive action in respect of the material employed and in the case of special alloys with sufficient corrosion resistance the corrosion allowance may be reduced to zero.

3.3 Permissible design stress K for carbon steel and alloy steel pipes

3.3.1 The permissible stress, K, for carbon steel and alloy steel pipes to be considered in 3.2, is to be taken as the lowest of the following values:

$$\frac{R_{20}}{2{,}7}, \frac{E_T}{1{,}6} \text{ to } \frac{E_T}{1{,}8}, \frac{\sigma_{r/100000}}{1{,}6} \text{ up to } \frac{\sigma_{r/10000}}{1{,}8}, \frac{\sigma_{1/100000}}{1} \text{ accordingly}$$

where:

R_{20} = the specified minimum tensile strength (N/mm^2) at room temperature, i.e. 20ºC

E_T = the specified minimum yield stress or 0,2% proof stress (N/mm^2) at the design temperature (see 3.6)

$\sigma_{r/100000}$ = the average stress (N/mm^2) to produce rupture in 100000 hours at the design temperature (see 3.6)

$\sigma_{1/100000}$ = average stress (N/mm^2) to produce 1% creep in 100000 hours at the design temperature (see 3.6)

Notes:

1. The values of yield stress or 0,2% proof stress given by national and international standards for steel pipes may be adopted.
2. The values in the range between 1,6 and 1,8 are to be chosen at the discretion of LHR.
3. The value of $\sigma_{1/100000}/1$ may be used at discretion of LHR on the basis of its reliability, and if deemed necessary.

3.4 Permissible stress K for copper and copper alloys

3.4.1 The permissible stress K for copper and copper alloy pipes to be considered in 3.2 is to be taken from Table 8.3.7, depending upon design temperature (see 3.6).

Table 8.3.7: Permissible stress limits K for copper and copper alloys

Pipe material		Copper	Aluminium brass	Copper nickel Cu Ni 5 Fe 1 Mn Cu Ni 10 Fe 1 Mn	Copper nickel Cu Ni 30
Material condition		Annealed	Annealed	Annealed	Annealed
Minimum tensile strength (N/mm^2)		215	325	275	365
Permissible Stress K, (N/mm^2)	50°C	41	78	68	81
	75°C	41	78	68	79
	100 C	40	78	67	77
	125°C	40	78	65,5	75
	150°C	34	78	64	73
	175°C	27,5	51	62	71

	200°C	18,5	24,5	59	69
	225°C	-	-	56	67
	250°C	-	-	52	65,5
	275°C	-	-	48	64
	300°C	-	-	44	62

Notes:

1. Intermediate values may be determined by linear interpolation.
2. For materials not included in the Table 8.3.6, the permissible stress shall be specially considered by LHR.

3.5 Design pressure

3.5.1 The design pressure P, to be considered in 3.2, is the maximum working pressure and it is not to be less than the highest set pressure of any safety valve relief valve. For special cases, the design pressure will be specially considered. For pipes containing fuel oil, the design pressure is to be taken in accordance with Table 8.3.8.

Table 8.3.8: Definition of the design pressure for fuel oil systems

Working Pressure	Working temperature	
	$T \leq 60^0$ C	$T > 60^0$ C
$P \leq 7$ bar	3 bar or max. working pressure, whichever is the greater	3 bar or max. working pressure, whichever is the greater
$P > 7$ bar	max. working pressure	14 bar or max. working pressure, whichever is the greater

3.6 Design temperature

The design temperature to be considered for determining the permissible stress in 3.3 and 3.4 is in general the maximum temperature of the medium inside the pipes. For special cases, the design temperature will be specially considered.

3.7 Flanges

The dimensions of flanges and relative bolts are to be chosen in accordance with the national standards. For special application the dimensions of flanges and relative bolts will be subject to special consideration*.

* For special applications, when the temperature, the pressure and the size of the flange have values above certain limits, to be fixed, the complete calculation of bolts and flanges is to be carried out.

3.8 Valves and fittings

Valves and fittings in piping systems are to be compatible with the pipes to which they are attached in respect of their strength (see 3.5 for design pressure) and are to be suitable for effective operation at the maximum working pressure they will experience in service.

SECTION 4 Steel pipe connections (IACS UR P2 (2021))

4.1 Direct connections of pipe lengths

4.1.1 Direct connections of pipe lengths may be made by direct welding, flanges, threaded joints or mechanical joints, and should be to a recognised standard or of a design proven to be suitable for the intended purpose and acceptable to LHR. The expression "mechanical joints" means devices intended for direct connection of pipe lengths other than by welding, flanges or threaded joints described in 4.1.2 to 4.17, 4.2 and 4.3 below.

4.1.2 Butt-welded joints shall be of full penetration type with or without special provision for a high quality of root side, i.e. butt welds were accomplished as double welded or by use of a backing ring or inert gas back-up on first pass, or other similar methods accepted by LHR. Joints of this type with special provision for a high quality of root side are applicable to pipes of any Class irrespective of the outside diameter. Butt-welded joints without special provision for a high quality of root side are applicable to pipes of Classes II and III of any outside diameter.

4.1.3 Socket weld fittings are to be of forged steel in accordance with a recognised standard. They may be used with carbon steel pipes not exceeding 60,3 mm outside diameter. They are not to be used in systems involving corrosive service or where fatigue is liable to occur.

4.1.4 The thickness of the socket weld fittings is to be extracted by the calculations of 3.2 but is not to be less than 1,25 times the nominal thickness of the pipe or tube. The diametrical clearance between the outside diameter of the pipes and the bore of the fitting is not to exceed 0,8 mm, and a gap of approximately 1,5 mm is to be provided between the end of the pipe and the bottom of the socket. The leg lengths of the fillet weld connecting the pipe to the socket weld fitting are to be such that the throat dimension of the weld is not less than the nominal thickness of the pipe or tube. Slip-on sleeve and socket welded joints are to have sleeves, sockets and weldments of adequate dimensions conforming to LHR Rules or recognized standard.

4.1.5 Slip-on sleeve and socket welded joints are applicable to Class III systems of any outside diameter. The particulars of those joints are to be submitted to LHR for approval.

4.1.6 Sleeve threaded joints are to be in accordance with a standard recognised by LHR. They shall be used for services other than conveying combustible media on pipes belonging to Class III and of outside diameter not more than 57 mm. They are allowed only for subordinate systems (e.g. sanitary and hot water heating systems). Screwed pipe connections and pipe couplings may be used subject to special approval. In particular cases, slip-on sleeve and socket welded joints may be allowed by LHR for piping systems of Class I and II having outside diameter ≤ 88,9 mm except for piping systems conveying toxic media or services where fatigue, severe erosion or crevice corrosion is expected to occur.

4.1.7 Slip-on joints, sleeve threaded joints, socket weld joints and other types of direct connection of pipe lengths may be allowed by LHR in each particular case for small diameter and depending upon the service conditions.

4.2 Flange connections

4.2.1 Flanges may be cut from plates or may be forged or cast. The selection of material is based upon the design temperature. The design pressure helps in determining the appropriate particulars of the flange. Flanges differ in method of attachment to the pipe, that is, whether they are screwed and expanded or welded. Alternative methods of flange attachment may be accepted provided details are submitted for consideration. The dimensions and configuration of flanges and bolts are to be chosen in accordance with recognized standards.

4.2.2 The dimensions and materials of flanges and corresponding bolting, and the pressure-temperature rating of bolted flanges in pressure pipelines in accordance with national or other established standards will be accepted. Gaskets are to be suitable for the media being conveyed under design pressure and temperature conditions and their dimensions and configuration are to be in accordance with recognised standards.

For non-standard flanges the dimensions of flanges and bolts are to be subject to special consideration.

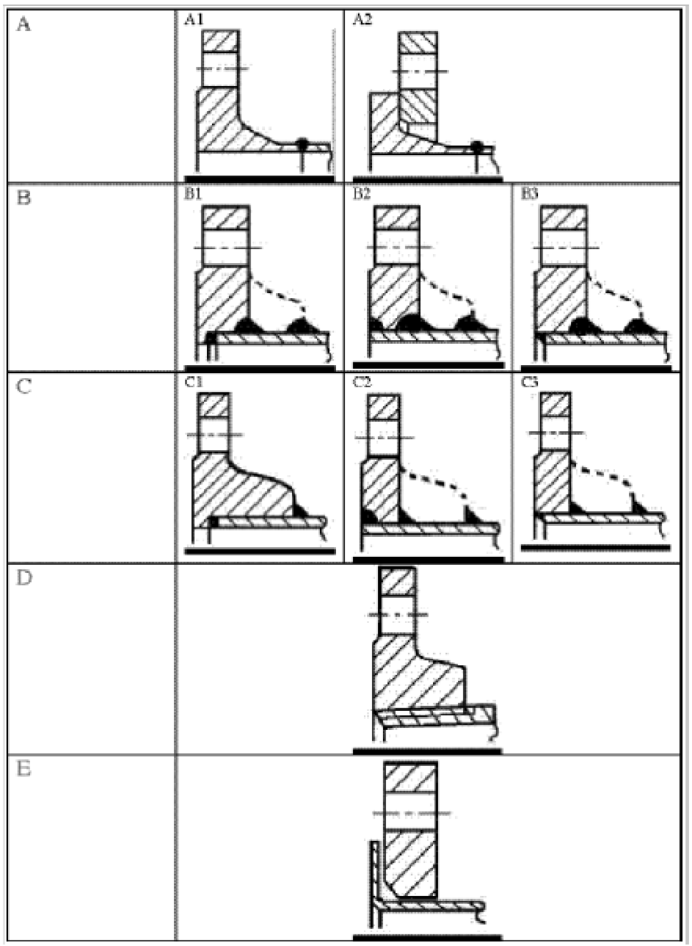
4.2.3 Examples of flange attachments are shown in Figure 8.4.1. However, other types of flange attachments may be considered LHR in each particular case.

4.2.4 Flange attachments are to be in accordance with national or international standards that are applicable to the piping system and are to recognize the boundary fluids, design pressure and temperature conditions, external or cyclic loading and location.

4.2.5 Welded-on flanges are not to be a tight fit on the pipes. The maximum clearance between the bore of the flange and the outside diameter of the pipe is to be 3 mm, at any point, and the sum of the clearances diametrically opposed is not to exceed 5mm.

4.2.6 The outside diameter of flange of Type A corresponding to the flange end is to be not less than the outside diameter of pipe. Where backing rings are used with flange type A they are to fit closely to the bore of the pipe and should be removed after welding. The rings are to be made of the same material as the pipes or of mild steel having a sulphur content not greater than 0,05%.

Figure 8.4.1: Acceptable flange connections



Note for Figure 8.4.1: For type D, the pipe and flange are to be screwed with a tapered thread and the diameter of the screw portion of the pipe over the thread is not to be appreciably less than the outside diameter of the unthreaded pipe. For certain types of thread, after the flange has been screwed hard home, the pipe is to be expanded into the flange.

4.3 Miscellaneous requirements of pipe connections

4.3.1 Where butt welds are employed in the attachment of flange Type A, in pipe-to-pipe joints or in the construction of branch pieces and the parts to be joined differ in wall thickness, the thicker wall is to be gradually tapered to the thickness of the thinner at the butt joint.

4.3.2 Branches may be attached to pressure pipe by means of welding provided that the pipe is reinforced at the branch by a compensating plate or collar or other approved means, or alternatively that the thickness of pipe and branch are increased to maintain the strength of the pipe. The requirements also apply to fabricated branch pieces.

4.3.3 Welding may be carried out by means of the shielded metal arc, inert gas metal arc, oxy-acetylene or other approved process, but in general oxy-acetylene is suitable only for flange Type A and is not to be applied to pipes exceeding 100 mm in diameter or 9,5 mm in thickness.

4.3.4 The welding necks of valve chests are to be sufficiently long to ensure that the valves are not distorted as the result of welding and subsequent heat treatment of the joints.

Table 8.4.1:

Class of Piping	Toxic or corrosive media (4) Flammable media (4) Liquefied gases (LG)	Lubricating and fuel oil	Steam (3) and thermal Oil	Other Media (1,2,3)
I	A,B (6)	A,B,	A,B (6)	A,B
II	A,B,C	A,B,C,E (7)	A,B,C,D (5),E (5)	
III		A,B,C,E	A,B,C,D,E	A,B,C,D,E,F

NOTES:

1. Including water, air, gases, hydraulic oil.
2. Type F for water pipes and open ended lines only.
3. Only type A when design temperature exceeds 400°C.
4. Only type A when design pressure exceeds 10 bar.
5. Types D and E are not to be used when design temperatures exceeds 250°C.
6. Type B for outside diameter <150mm only.
7. Type E for oil piping when design temperature <150°C and design pressure <16 bar only.

4.3 Slip-on threaded joints

4.3.1 Slip-on threaded joints having pipe threads where pressure-tight joints are made on the threads with parallel or tapered threads, shall comply with requirements of a recognized national or international standard.

4.3.2 Slip-on threaded joints may be used for outside diameters as stated below except for piping systems conveying toxic or flammable media or services where fatigue, severe erosion or crevice corrosion is expected to occur.

4.3.3 Threaded joints in CO_2 systems shall be allowed only inside protected spaces and in CO_2 cylinder rooms.

Threaded joints for direct connectors of pipe lengths with tapered thread are to be allowed for:

a) Class I, outside diameter not more than 33,7 mm

b) Class II and Class III, outside diameter not more than 60,3 mm

Threaded joints with parallel thread are to be allowed for Class III, outside diameter not more than 60,3 mm.

4.3.4 In particular cases, sizes in excess of those mentioned above may be accepted by LHR if in compliance with a recognized national and/or international standard.

4.4 Mechanical joints

4.4.1 Due to the great variations in design and configuration of mechanical joints, no specific recommendation regarding calculation method for theoretical strength calculations is given in these requirements. The Type Approval is to be based on the results of testing of the actual joints. These requirements are applicable to pipe unions, compression couplings, slip-on joints as shown in Table 8.4.2. Similar joints complying with these requirements may be acceptable.

The application and pressure ratings of different mechanical joints are to be approved by LHR. The approval is to be based on the Type Approval procedure in SECTION 8. Mechanical joints including pipe unions, compression couplings, slip-on joints and similar joints are to be of approved type for the service conditions and the intended application.

4.4.2 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.

4.4.3 Construction of mechanical joints is to prevent the possibility of tightness failure affected by pressure pulsation, piping vibration, temperature variation and other similar adverse effects occurring during operation on board.

4.4.4 Material of mechanical joints is to be compatible with the piping material and internal and external media.

4.4.5 Mechanical joints are to be tested where applicable, to a burst pressure of 4 times the design pressure. For design pressures above 200 bar the required burst pressure will be specially considered by LHR.

4.4.6 Where appropriate, mechanical joints are to be of fire-resistant type as required by Table 8.4.3.

4.4.7 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 bulkhead deck of passenger ships and freeboard deck of cargo ships or tanks containing flammable fluids.

4.4.8 The mechanical joints are to be designed to withstand internal and external pressure as applicable and where used in suction lines are to be capable of operating under vacuum.

4.4.9 The number of mechanical joints in oil systems is to be kept to a minimum. In general, flanged joints conforming to recognised standards are to be used.

4.4.10 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.

4.4.11 Slip-on joints are not to be used in 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.LHR Usage of slip type slip-on joints as the main means of pipe connection is not permitted except for cases where compensation of axial pipe deformation is necessary.

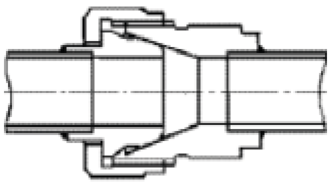
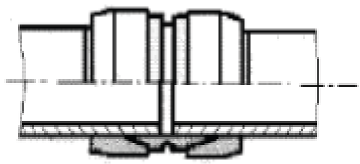
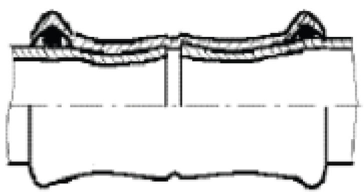
4.4.12 Application of mechanical joints and their acceptable use for each service is indicated in Table 8.4.3; dependence upon the Class of piping and pipe dimensions is indicated in Table 8.4.4. In particular cases, sizes in excess of those mentioned above may be accepted by LHR if in compliance with a recognized national and/or international standard.

4.4.13 Mechanical joints are to be tested in accordance with a program approved by LHR, which is to include at least the following:

1. leakage test
2. vacuum test (where necessary)
3. vibration (fatigue) test
4. fire endurance test (where necessary)
5. burst pressure test
6. pressure pulsation test (where necessary)
7. assembly test (where necessary)
8. pull out test (where necessary)

4.4.14 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.

Table 8.4.2: Examples of mechanical joints

Pipe Unions	
Welded and Brazed Types	
Compression Couplings	
Swage Type	
Press Type	

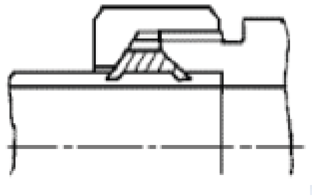
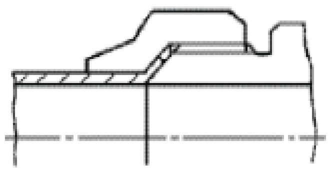
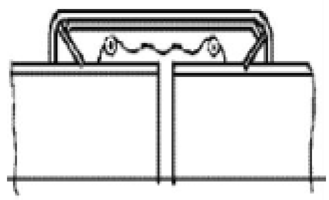
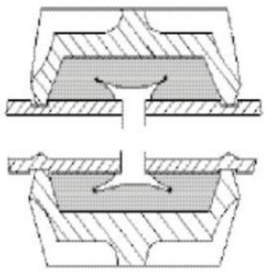
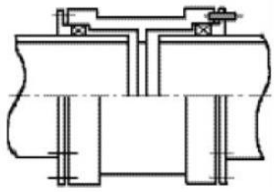
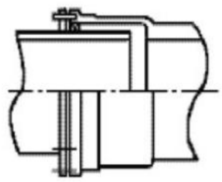
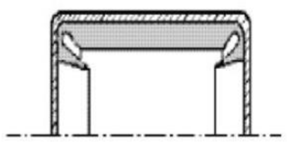
Bite Type	
Flared Type	
Slip-on Joints	
Grip Type	
Machine Grooved Type	
Slip Type	  

Table 8.4.3: Application of mechanical joints

Systems		Kind of connections				
		Pipe Unions	**Compression Couplings**	**Slip-on Joints**	**Classification of pipe system**	**Fire endurance test condition (7)**
Flammable fluids (Flash point ≤ 60°C)						
1.	Cargo oil lines (1)	+	+	+	dry	30 min dry (*)
2.	Crude oil washing lines (1)	+	+	+	dry	
3.	Vent lines (3)	+	+	+	dry	
Inert gas						
4.	Water seal effluent lines	+	+	+	wet	30 min wet (*)
5.	Scrubber effluent lines	+	+	+	wet	30 min wet (*)
6.	Main lines (1), (2)	+	+	+	dry	30 min dry (*)
7.	Distributions lines (1)	+	+	+	dry	30 min dry (*)
Flammable fluids (Flash point > 60ºC)						
8.	Cargo oil lines (1)	+	+	+	dry	30 min dry (*)
9.	Fuel oil lines (2), (3)	+	+	+	wet	30 min wet (*)
10.	Lubricating oil lines (2), (3)	+	+	+	wet	
11.	Hydraulic oil (2), (3)	+	+	+	wet	
12.	Thermal oil (2), (3)	+	+	+	wet	
Sea Water						
13.	Bilge lines (4)	+	+	+	dry/wet	8 min dry + 22 min wet (*)
14.	Permanent water filled fire extinguishing systems, e.g. fire main, sprinkler systems (3)	+	+	+	wet	30 min wet (*)
15.	Non-permanent water filled fire extinguishing systems, e.g. foam, drencher systems and fire main (3)	+	+	+	dry/wet	8 min dry + 22 min wet (*) For foam systems FSS Code Chapter 6 to be observed
16.	Ballast system (4)	+	+	+	wet	30 min wet (*)
17.	Cooling water system (4)	+	+	+	wet	30 min wet (*)
18.	Tank cleaning services	+	+	+	dry	Fire endurance test not required
19.	Non-essential systems	+	+	+	dry dry/wet wet	Fire endurance test not required
Fresh Water						

20.	Cooling water system (4)	+	+	+	wet	30 min wet (*)
21.	Condensate return (4)	+	+	+	wet	30 min wet (*)
22.	Non-essential system	+	+	+	dry dry/wet wet	Fire endurance test not required
Sanitary/Drains/Scuppers						
23.	Deck drains (internal) (5)	+	+	+	dry	Fire endurance test not required
24.	Sanitary drains	+	+	+	dry	
25.	Scuppers and discharge (overboard)	+	+	-	dry	
Sounding/Vent						
26.	Water tanks/Dry spaces	+	+	+	dry, wet	Fire endurance test not required
27.	Oil tanks (f.p.>60°C) (2), (3)	+	+	+	dry	
Miscellaneous						
28.	Starting/Control air (1)	+	+	-	dry	30 min dry (*)
29.	Service air (non-essential)	+	+	+	dry	Fire endurance test not required
30.	Brine	+	+	+	wet	
31	CO_2 system (outside protected space)	+	+	-	dry	30 min dry (*)
32.	CO_2 system (inside protected space)	+	+	-	dry	Mechanical joints shall be constructed of materials with melting point above 925 ^{0}C. Ref. to FSS Code Chapter 5.
33.	Steam	+	+	+ (6)	wet	Fire endurance test not required

Abbreviations:

+ Application is allowed

- Application is not allowed

* Fire endurance test as specified in P2.11.5.5.6

Notes - Fire resistance capability

If mechanical joints include any components which readily deteriorate in case of fire, the following footnotes are to be observed:

1. Fire endurance test shall be applied when mechanical joints are installed in pump rooms and open decks.
2. Slip on joints are not accepted inside machinery spaces of category A or accommodation spaces. May be accepted in other machinery spaces provided the joints are located in easily visible and accessible positions (refer to MSC/Circ.734).
3. Approved fire-resistant types except in cases where such mechanical joints are installed on open decks, as defined in SOLAS II-2/Reg. 9.2.3.3.2.2(10) and not used for fuel oil lines.
4. Fire endurance test shall be applied when mechanical joints are installed inside machinery spaces of

category A.

Notes - General

5. Only above bulkhead deck of passenger ships and freeboard deck of cargo ships.
6. Slip type slip-on joints as shown in Table 6. May be used for pipes on deck with a design pressure of 10 bar or less.
7. 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 8.4.3 indicates systems where the various kinds of joints may be accepted. However, in all cases, acceptance of the joint type is to be subject to approval for the intended application, and subject to conditions of the approval and applicable Rules. Further, relevant statutory requirements must be taken into consideration. In cases exposure time (t_T) is greater than 30 minutes the dry-wet test conditions are 8 minutes dry and, accordingly, the wet period t_T-8 min.

Table 8.4.4: Application of mechanical joints depending upon the class of piping

Type of joints	**Classes of piping systems**		
	Class I	**Class II**	**Class III**
Pipe Unions			
Welded and brazed type	+ (OD ≤ 60,3 mm)	+ (OD ≤ 60,3 mm)	+
Compression coupling			
Swage type	+	+	+
Bite type Flared type	+ (OD ≤ 60,3 mm) + (OD ≤ 60,3 mm)	+ (OD ≤ 60,3 mm) + (OD ≤ 60,3 mm)	+ +
Press type	-	-	+
Slip-on joints			
Machine grooved type	+	+	+
Grip type	-	+	+
Slip type	-	+	+

Abbreviations:

+ Application is allowed
- Application is not allowed

SECTION 5 Production and Application of Plastic Pipes on Ships (IACS UR P4, Rev.7 (2022))[5]

5.1 Terms and Definitions

5.1.1 "Plastic(s)" 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.

5.1.2 "Pipes/piping systems" means those made of plastic(s) and include the pipes, fittings, system joints, method of joining and any internal or external liners, coverings and coatings required to comply with the performance criteria.

5.1.3 "Joint" means the location at which two pieces of pipe and a fitting are connected together. The joint may be made by adhesive bonding, laminating, welding, flanges and mechanical joints according to Table 8.4.2 in Section 5 of this Chapter.5.1.4 "Fittings" means bends, elbows, fabricated branch pieces etc. of plastic materials.

5.1.5 "Nominal pressure" means the maximum permissible working pressure which should be determined in accordance with the requirements in 5.3.1.

5.1.6 "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.

5.1.7 "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.

5.1.8 "Essential to the safety of ship" means all piping systems that in event of failure will pose a threat to personnel and the ship. Examples for piping systems essential to the safety are provided in Table 8.5.1.

5.1.9 "Essential services" are those services essential for propulsion and steering and safety of the ship as specified in IACS UI SC134.

5.2 Scope

5.2.1 These requirements are applicable to piping systems on ships, including pipe joints and fittings, made predominately of other material than metal.

5.2.2 The use of mechanical joints approved for the use in metallic piping systems only are not permitted.

5.2.3 Piping systems intended for non-essential services are to meet only the requirements of recognized standards and paragraphs 5.5 and 5.6 of this Section.

5.3 General Requirements

5.3.1 The specification of piping is to be in accordance with a recognised national or international standard acceptable to LHR. In addition, the following requirements apply:

(1) Strength

[5] NOTE:
1. This Section addresses the provisions of IMO Resolution A.753(18) as amended by IMO Resolutions MSC.313(18) and MSC.399(95).

(a) The strength of the pipes is to be determined by a hydrostatic test failure pressure of a pipe specimen under the standard conditions: atmospheric pressure equal to 100 kPa, relative humidity 30%, environmental and carried fluid temperature 298 kPa (25°C).
(b) The strength of fittings and joints is to be not less than that of the pipes.
(c) The nominal pressure is to be determined from the following conditions:

(i) Internal Pressure

For an internal pressure the following is to be taken whichever is smaller:

$$Pn_{int} \leq P_{sth}/4 \;\; or \; Pn_{int} \leq P_{lth}/2{,}5$$

where:

P_{sth} = short-term hydrostatic test pipe failure pressure;
P_{lth} = long-term hydrostatic test pipe failure pressure (> 100,000 h)

(ii) 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 and for any pipe installation required to remain operational in case of flooding damage, as per SOLAS II-1/8-1, or for any pipes that would allow progressive flooding to other compartments through damaged piping or through open ended pipes in the compartments).

For an external pressure:

$$Pn_{ext} \leq P_{col}/3$$

where:

P_{col}= pipe collapse pressure.

(d) In no case is the collapse pressure to be less than 3 bar.
(e) The maximum working external pressure is a sum of the vacuum inside the pipe and a head of liquid acting on the outside of the pipe.
(f) Notwithstanding the requirements of c(i) or c(ii) above as applicable, the pipe or pipe layer minimum wall thickness is to follow recognized standards. In the absence of standards for pipes not subject to external pressure, the requirements of c(ii) above are to be met.
(g) The maximum permissible working pressure is to be specified with due regard for maximum possible working temperatures in accordance with Manufacturer's recommendations.

(2) Axial Strength
(a) The sum of the longitudinal stresses due to pressure, weight and other loads is not to exceed the allowable stress in the longitudinal direction.
(b) 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 (see 5.3.1).

(3) Impact Resistance
(a) Plastic pipes and joints are to have a minimum resistance to impact in accordance with recognised national or international standards.
(b) After the test the specimen is to be subjected to hydrostatic pressure equal to 2.5 times the design pressure for at least 1 hour.

(4) Temperature

(a) 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°C 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.
(b) The minimum heat distortion/deflection temperature is to be not less than 80°C.

5.4 Requirements for Pipes/Piping Systems Depending on Service and/or Locations

5.4.1 Fire Endurance

(1) Pipes and their associated joints and fittings whose integrity is essential to the safety of ships, including plastic piping required by SOLAS II-1/21.4 to remain operational after a fire casualty, are required to meet the minimum fire endurance requirements of Appendix 1 or 2, as applicable, of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95).

(2) Unless instructed otherwise by the Flag Administration, fire endurance tests are to be carried out with specimen representative for pipes, joints and fittings[2]:
   a. Pipes:
      - for sizes with outer diameter < 200 mm the minimum outer diameter and wall thickness [3]
      - for sizes with outer diameter ≥ 200 mm one test specimen for each category of t/d (D = outer diameter, t = structural wall thickness). A scattering of ±10% for t/D is regarded as the same group. Minimum size approved is equal to the diameter of specimen successfully tested.
   b. Joints:
      - Each type of joint applicable for applied fire endurance level tested on pipe to pipe specimen

Notes:
(2): A test specimen incorporating several components of a piping system may be tested in a single test.
(3): Test conditions are most demanding for minimum wall thickness and thus larger wall thickness is covered. A key factor determining the fire performance of a pipe component variant is the thickness-to-diameter (t/D) ratio and whether it is larger or smaller than that of the variant which has been fire-tested. If fire-protective coatings or layers are included in the variant used in the fire test, only variants with the same or greater thickness of protection, regardless of the (t/D) ratio, shall be qualified by the fire test.

(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 the IMO Res.A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95). During the test it is not permitted to replace media drained by fresh water or nitrogen.

(4) Depending on the capability of a piping system to maintain its strength and integrity, there exist three different levels of fire endurance for piping systems.
   (i) Level 1. Piping having passed the fire endurance test specified in Appendix 1 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95) for a duration of a minimum of one hour without loss of integrity in the dry condition is considered to meet level 1 fire endurance standard (L1). Level 1W-Piping systems similar to Level 1 systems except these systems do not carry flammable fluid or any gas and a maximum 5% flow loss in the system after exposure is acceptable (L1W).
   (ii) Level 2. Piping having passed the fire endurance test specified in Appendix 1 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95) for a duration of a minimum of 30 minutes in the dry condition is considered to meet level 2 fire endurance standard (L2). Level 2W – Piping systems similar to Level 2 systems except a maximum 5% flow loss in the system after exposure is acceptable (L2W).
   (iii) Level 3. Piping having passed the fire endurance test specified in Appendix 2 of IMO Resolution A.753(18) as amended by IMO Resolutions MSC.313(88) and MSC.399(95) for a

duration of a minimum of 30 minutes in the wet condition is considered to meet level 3 fire endurance standard (L3).

(5) Permitted use of piping depending on fire endurance, location and piping system is given in Table 8.5.1.

(6) For Safe Return to Port purposes (SOLAS II-2/21.4), piping can be considered to remain operational after a fire casualty if the plastic pipes and fittings have been tested to L1 standard.

Table 8.5.1: Fire Endurance Requirements Matrix

N	Piping Systems	Location										
		A	**B**	**C**	**D**	**E**	**F**	**G**	**H**	**I**	**J**	**K**
		Machinery spaces of category A	Other spaces & Pump rooms	Cargo pump rooms	Ro/Ro cargo holds	Other dry cargo holds	Cargo tanks	Fuel oil tanks	Ballast water tanks	Cofferdams void spaces pipe tunnel & ducts	Accommodation service & control spaces	Open decks
1	2	3	4	5	6	7	8	9	10	11	12	13
CARGO (FLAMMABLE CARGOES (f.p. • 60°C)												
1.	Cargo lines	NA	NA	L1	NA	NA	O	NA	O(10)	O	NA	L1(2)
2.	Crude Oil washing lines	NA	NA	L1	NA	NA	O	NA	O(10)	O	NA	L1(2)
3.	Vent lines	NA	NA	NA	NA	NA	O	NA	O(10)	O	NA	X
INERT GAS												
4.	Water seal effluent line	NA	NA	O(1)	NA	NA	O(1)	O(1)	O(1)	O(1)	NA	O
5.	Scrubber effluent line	O(1)	O(1)	NA	NA	NA	NA	NA	O(1)	O(1)	NA	O
6.	Main Line	O	O	L1	NA	NA	NA	NA	NA	O	NA	L1(6)
7.	Distribution lines	NA	NA	L1	NA	NA	O	NA	NA	O	NA	L1(2)
FLAMMABLE LIQUIDS (f.p. > 60°C)												
8.	Cargo lines	X	X	L1	X	X	NA(3)	O	O(10)	O	NA	L1
9.	Fuel oil	X	X	L1	X	X	NA(3)	O	O	O	L1	L1
10.	Lubricating	X	X	L1	X	X	NA	NA	NA	O	L1	L1
11.	Hydraulic oil	X	X	L1	X	X	O	O	O	O	L1	L1
SEAWATER (1)												
12.	Bilge main & branches	L1(7)	L1(7)	L1	X	X	NA	O	O	O	NA	L1
13.	Fire main & water spray	L1	L1	L1	X	NA	NA	NA	O	O	X	L1
14.	Foam system	L1	L1	L1	NA	NA	NA	NA	NA	O	L1	L1
15.	Sprinkler system	L1	L1	L3	X	NA	NA	NA	O	O	L3	L3
16.	Ballast	L3	L3	L3	L3	X	O(10)	O	O	O	L2	L2
17.	Cooling water, essential services	L3	L3	NA	NA	NA	NA	NA	O	O	NA	L2
18.	Tank cleaning services fixed machines	NA	NA	L3	NA	NA	O	NA	O	O	NA	L3(2)
19.	Non-essential systems	O	O	O	O	O	NA	O	O	O	O	O

FRESHWATER											
20. Cooling water essential services	L3	L3	NA	NA	NA	NA	O	O	O	L3	L3
21. Condensate return	L3	L3	L3	O	O	NA	NA	NA	O	O	O
22. Non-essential systems	O	O	O	O	O	NA	O	O	O	O	O
SANITARY/DRAINS/ SCUPPERS											
23. Deck drains (internal)	L1(4)	L1(4)	NA		O	NA	O	O	O	O	O
24. Sanitary drains (internal)	O	O	NA	O	O	NA	O	O	O	O	O
25. Scuppers and discharges (over-board)	O(1,8)	O(1,8)	O(1,8)	O(1,8)	O(1,8)	O	O	O	O	O(1,8)	O
SOUNDING/AIR											
26. Water tanks/dry spaces	O	O	O	O	O	O(10)	O	O	O	O	O
27. Oil tanks (f.p. > 60°C)	X	X	X	X	X	X(3)	O	O(10)	O	X	X
MISCELLANEOUS											
28. Control air	L1(5)	L1(5)	L1(5)	L1(5)	L1(5)	NA	O	O	O	L1(5)	L1(5)
29. Service air (non-essential)	O	O	O	O	O	NA	O	O	O	O	O
30. Brine	O	O	NA	O	O	NA	NA	NA	O	O	O
31. Auxiliary low-pressure steam (≤7 bar)	L2	L2	O(9)	O(9)	O(9)	O	O	O	O	O(9)	O(9)

ABBREVIATIONS:

L1 Fire endurance test (appendix 1 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95)) in dry conditions, 60 min

L1W Fire endurance test (section P.4.4.1.2)

L2 Fire endurance test (appendix 1 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95)) in dry conditions, 30 min

L2W Fire endurance test (section P.4.4.1.2)

L3 Fire endurance test (appendix 2 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and MSC.399(95)) in wet conditions, 30 min

O No fire endurance test required

NA Not applicable

X Metallic materials having a melting point greater than 925°C

NOTES:

1. Where non-metallic piping is used, remotely controlled valves to be provided at ship's side (valve is to be controlled from outside space).
2. Remote closing valves to be provided at the cargo tanks.
3. When cargo tanks contain flammable liquids with f.p. > 60°C, "O" may replace "NA" or "X".
4. For drains serving only the space concerned, "O" may replace "L1 W".
5. When controlling functions are not required by statutory requirements or guidelines, "O" may replace "L1".
6. For pipe between machinery space and deck water seal, "O" may replace "L1".
7. For passenger vessels, "X" is to replace "L1".
8. Scuppers serving open decks in positions 1 and 2, as defined in Regulation 13 of Protocol of 1988 relating to the International Convention on Load Lines, 1966, should be "X" 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. For essential services, such as fuel oil tank heating and ship's whistle, "X" is to replace "O".
10. For tankers where compliance with paragraph 3.6 of Regulation 19 of MARPOL Annex I is required, "NA" is to replace "O".
11. L3 in service spaces, NA in accommodation and control spaces.

12. Type Approved plastic piping without fire endurance test (O) 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.
13. For Passenger Ships subject to SOLAS II-2/21.4 (Safe return to Port), plastic pipes for services required to remain operative in the part of the ship not affected by the casualty thresholds, such as systems intended to support safe areas, are to be considered essential services. In accordance with MSC.1/Circ.1369, interpretation 12, for Safe Return to Port purposes, plastic piping can be considered to remain operational after a fire casualty if the plastic pipes and fittings have been tested to L1 standard.

LOCATION DEFINITIONS

Location	**Definition**	
A -	Machinery spaces of category A	Machinery spaces of category A as defined in SOLAS II-2/3.31.
B -	Other machinery spaces and pump rooms	Spaces, other than category A machinery spaces and cargo pump rooms, containing propulsion machinery, boilers, steam and internal combustion engines, generators and major electrical machinery, pumps, oil filling stations, refrigerating, stabilizing, ventilation and air-conditioning machinery, and 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 SOLAS II-2/3.41 and SOLAS II-2/3.46.
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 SOLAS II-2/3.1, SOLAS II-2/3.45 and SOLAS II-2/3.18.
K -	Open decks	Open deck spaces as defined in SOLAS II-2/9.2.2.3.2(5).

5.4.2 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 not exceeding average values listed in Appendix 3 of IMO Resolution A.753(18), as amended by IMO Resolutions MSC.313(18) and MSC.399(95).
(2) Surface flame spread characteristics are to be determined using the procedure given in the 2010 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 Resolution A.753(18), as amended by IMO Resolutions MSC.313(88) and 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.

5.4.3 Fire Protection Coatings

(1) Where a fire protective coating of pipes and fittings is necessary for achieving the fire endurance level required, it is to meet the following requirements:

(i) The pipes are generally to be delivered from the manufacturer with the protective coating on.
(ii) The fire protection properties of the coating are not to be diminished when exposed to salt water, oil or bilge slops. It is to be demonstrated that the coating is resistant to products likely to come into contact with the piping.
(iii) In considering fire protection coatings, such characteristics as thermal expansion, resistance against vibrations, and elasticity are to be taken into account.
(iv) The fire protection coatings are to have sufficient resistance to impact to retain their integrity.

5.4.4 Electrical Conductivity

Where electrical conductivity is to be ensured, the resistance of the pipes and fittings is not to exceed 1×10^5 Ω/m.

5.5 Material approval and Quality Control During Manufacture

5.5.1 Except as required in paragraph 5.2.3, prototypes of pipes and fittings are to be tested to determine short-term and long-term design strength, fire endurance and low surface flame spread characteristics (if applicable), electrical resistance (for electrically conductive pipes), impact resistance in accordance with this Section.

5.5.2 For prototype testing representative samples of pipes and fittings are to be selected to the satisfaction of LHR.

5.5.3 The Manufacturer is to have quality system that meets ISO 9001:2015 as amended or equivalent. 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.

5.5.4 Each pipe and fitting is to be tested by the Manufacturer at a hydrostatic pressure not less than 1.5 times the nominal pressure. Alternatively, for pipes and fittings not employing hand layup techniques, the hydrostatic pressure test may be carried out in accordance with the hydrostatic testing requirements stipulated in the recognized national or international standard to which the pipe or fittings are manufactured, provided that there is an effective quality system in place.

5.5.5 Piping and fittings are to be permanently marked with identification. Identification is to include pressure ratings, the design standards that the pipe or fitting is manufactured in accordance with, and the material of which the pipe or fitting is made.

5.5.6 In case the Manufacturer does not have an approved quality system complying with ISO 9001:2015 as amended or equivalent, pipes and fittings are to be tested in accordance with this Section to the satisfaction of the LHR's surveyors for every batch of pipes.

5.5.7 Depending upon the intended application LHR may require the pressure testing of each pipe and/or fitting.

5.6 Installation

5.6.1 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 the pipe Manufacturer's recommended spacing. The selection and spacing of pipe supports are to take into account pipe dimensions, length of piping, mechanical and physical properties of the pipe

material, mass of pipe and contained fluid, external pressure, operating temperature, thermal expansion effects, loads due to external forces, thrust forces, water hammer, vibrations, maximum accelerations to which the system 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 minimize wear of the pipes where they contact the supports.

(3) Heavy components in the piping system such as valves and expansion joints are to be independently supported.

5.6.2 Expansion

(1) Suitable provision is to be made in each pipeline to allow for relative movement between pipes made of plastic and the steel structure, having due regard to:
   (i) the difference in the coefficients of thermal expansion;
   (ii) deformations of the ship's hull and its structure.

(2) When calculating the thermal expansions, account is to be taken of the system working temperature and the temperature at which assembly is performed.

5.6.3 External Loads

(1) When installing the piping, allowance is to be made for temporary point loads, 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 100mm nominal outside diameter.

(2) Besides for providing adequate robustness for all piping including open-ended piping a minimum wall thickness, complying with 5.3.1 (1), may be increased upon the demand of LHR taking into account the conditions encountered during service on board ships.

(3) Pipes are to be protected from mechanical damage where necessary.

5.6.4 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 a permanent seal between the pipes and fittings throughout the temperature and pressure range of the intended application.

(4) Tightening of joints is to be performed in accordance with Manufacturer's instructions.

5.6.5 Installation of Conductive Pipes

(1) In piping systems for fluids with conductivity less than 1000 pico siemens per meter (pS/m) such as refined products and distillates use is to be made of conductive pipes.

(2) Regardless of the fluid being conveyed, plastic piping is to be electrically conductive if the piping passes through a hazardous area. The resistance to earth from any point in the piping system is not to exceed 1×10^6 Ω. It is preferred that pipes and fittings be homogeneously conductive. Pipes and fittings having conductive layers are to be protected against a possibility of spark damage to the pipe wall. Satisfactory earthing is to be provided.

(3) After completion of the installation, the resistance to earth is to be verified. Earthing wires are to be accessible for inspection.

5.6.6 Application of Fire Protection Coatings

(1) Fire protection coatings are to be applied on the joints, where necessary for meeting the required fire endurance as for 5.4.3, after performing hydrostatic pressure tests of the piping system.
(2) The fire protection coatings are to be applied in accordance with Manufacturer's recommendations, using a procedure approved in each particular case.

5.6.7 Penetration of Divisions

(1) Where plastic pipes pass through "A" or "B" class divisions, arrangements are to be made to ensure that the fire endurance is not impaired. These arrangements are to be tested in accordance with Recommendations for fire test procedures for "A", "B" and "F" bulkheads specified in Part 3 of Annex 1 to the 2010 FTP Code.
(2) When plastic pipes pass through watertight bulkheads or decks, the watertight integrity of the bulkhead or deck is to be maintained. For pipes not able to satisfy the requirements in 5.3.1 (1) (ii) of this Section, a metallic shut-off valve operable from above the freeboard deck should be fitted at the bulkhead or deck.
(3) If the bulkhead or deck is also a fire division and destruction by fire of plastic pipes may cause the inflow of liquid from tanks, a metallic shut-off valve operable from above the freeboard deck should be fitted at the bulkhead or deck.

5.6.8 Control During Installation

(1) Installation is to be in accordance with the Manufacturer's guidelines.
(2) Prior to commencing the work, joining techniques are to be approved by LHR.
(3) The tests and explanations specified in this Section are to be completed before shipboard piping installation commences.
(4) The personnel performing this work are to be properly qualified and certified to the satisfaction of LHR.
(5) The procedure of making bonds is to include:
   (i) materials used,
   (ii) tools and fixtures,
   (iii) joint preparation requirements,
   (iv) cure temperature,
   (v) dimensional requirements and tolerances, and
   (vi) tests acceptance criteria upon completion of the assembly.
(6) Any change in the bonding procedure which will affect the physical and mechanical properties of the joint is to require the procedure to be requalified.

5.6.9 Bonding Procedure Quality Testing

(1) A test assembly is to be fabricated in accordance with the procedure to be qualified and it is to consist of at least one pipe-to-pipe joint and one pipe-to-fitting joint.
(2) When the test assembly has been cured, it is to be subjected to a hydrostatic test pressure at a safety factor 2.5 times the design pressure of the test assembly, for not less than one hour. No leakage or separation of joints is allowed. The test is to be conducted so that the joint is loaded in both longitudinal and circumferential directions.
(3) Selection of the pipes used for test assembly, is to be in accordance with the following:
   (i) When the largest size to be joined is 200mm nominal outside diameter, or smaller, the test assembly is to be the largest piping size to be joined.
   (ii) When the largest size to be joined is greater than 200mm nominal outside diameter, the size of the test assembly is to be either 200mm or 25% of the largest piping size to be joined, whichever is greater.

(4) 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 above.

5.6.10 Testing After Installation on Board

(1) Piping systems for essential services are to be subjected to a test pressure not less than 1.5 times the design pressure or 4 bar whichever is greater. Notwithstanding the requirement above, the requirement in 5.6.10 (2) may be applied to open ended pipes (drains, effluent, etc.).
(2) Piping systems for non-essential services are to be checked for leakage under operational conditions.
(3) For piping required to be electrically conductive, earthing is to be checked and random resistance testing is to be conducted.

5.7 Test Specification for Plastic Pipes

5.7.1 Scope

Section 5.7 contains requirements for the Type Approval of plastic pipes. It is applicable to piping systems, including pipe joints and fittings made predominately of other material than metal.

5.7.2 Documentation

The following information for the plastic pipes, fittings and joints is to be submitted for consideration and approval:

(1) General information
    (a) Pipe and fitting dimensions.
    (b) Maximum internal and external working pressure.
    (c) Working temperature range.
    (d) Intended services and installation locations.
    (e) The level of fire endurance.
    (f) Electrically conductive.
    (g) Intended fluids.
    (h) Limits on flow rates.
    (i) Serviceable life.
    (j) Installation instructions.
    (k) Details of marking.
(2) Drawings and supporting documentation:
    (a) Certificates and reports for relevant tests previously carried out.
    (b) Details of relevant standards.
    (c) All relevant design drawings, catalogues, data sheets, calculations and functional descriptions.
    (d) Fully detailed sectional assembly drawings showing pipe, fittings and pipe connections.
(3) Materials (as applicable)
    (a) The resin type.
    (b) Catalyst and accelerator types, and concentration employed in the case of reinforced polyester resin pipes or hardeners where epoxide resins are employed.
    (c) A statement of all reinforcements employed where the reference number does not identify the mass per unit area or the tex number of a roving used in a filament winding process, these are to be detailed.
    (d) Full information regarding the type of gel-coat or thermoplastic liner employed during construction, as appropriate.
    (e) Cure/post-cure conditions. The cure and post cure temperatures and times employ resin/reinforcement ratio.

(f) Winding angle and orientation.
(g) Joint bonding procedures and qualification tests results, see 5.6.8 (5).

5.7.3 Testing

Testing is to demonstrate compliance of the pipes, fittings and joints for which Type Approval is sought with this Section.

Pipes, joints and fittings are to be tested for compliance with the requirements of standards (for the list of standards refer to IACS Recommendation 86) acceptable to LHR.

SECTION 6 Flexible hoses (IACS UR P2.12 Rev.3 (2021))

6.1 Definition

6.1.1 Flexible hose assembly - short length of metallic or non-metallic hose normally with prefabricated end fittings ready for installation.

Note: Flexible hose assemblies for essential services or containing either flammable or toxic media are not to exceed 1,5 m in length.

6.2 Scope

6.2.1 The requirements 6.3 to 6.6 apply to flexible hoses of metallic or non-metallic material intended for a permanent connection between a fixed piping system and items of machinery. The requirements may also be applied to temporary connected flexible hoses or hoses of portable equipment.

6.2.2 Flexible hose assemblies as defined in 6.1.1 may be accepted for use in oil fuel, lubricating, hydraulic and thermal oil systems, fresh water and sea water cooling systems, compressed air systems, bilge and ballast systems, and Class III steam systems where they comply with 6.3 to 6.6. Flexible hoses in high pressure fuel oil injection systems are not to be accepted.

6.2.3 These requirements for flexible hose assemblies are not applicable to hoses intended to be used in fixed fire extinguishing systems.

6.3 Design and construction

6.3.1 Flexible hoses are to be designed and constructed in accordance with recognised National or International standards acceptable to LHR. Flexible hoses constructed of rubber materials and intended for use in bilge, ballast, compressed air, oil fuel, lubricating, hydraulic and thermal oil systems are to incorporate a single, double or more, closely woven integral wire braid or other suitable material reinforcement.

Flexible hoses of plastics materials for the same purposes, such as Teflon or Nylon, which are unable to be reinforced by incorporating closely woven integral wire braid are to have suitable material reinforcement as far as practicable.

Where rubber or plastics materials hoses are to be used in oil supply lines to burners, the hoses are to have external wire braid protection in addition to the reinforcement mentioned above. Flexible hoses for use in steam systems are to be of metallic construction.

6.3.2 Flexible hoses are to be complete with approved end fittings in accordance with manufacturer's specification. The end connections that do not have a flange are to comply with 4.4 of this Chapter as applicable and each type of hose/fitting combination is to be subject to prototype testing to the same standard as that required by the hose with particular reference to pressure and impulse tests.

6.3.3 The use of hose clamps and similar types of end attachments is not acceptable for flexible hoses in piping systems for steam, flammable media, starting air systems or for sea water systems where failure may result in flooding. In other piping systems, the use of hose clamps may be accepted where the working pressure is less than 5 bar and provided there are double clamps at each end connection.

6.3.4 Flexible hose assemblies intended for installation in piping systems where pressure pulses and/or high levels of vibration are expected to occur in service, are to be designed for the maximum expected impulse peak pressure and forces due to vibration. The tests required by 6.5 are to take into consideration the maximum anticipated in-service pressures, vibration frequencies and forces due to installation.

6.3.5 Flexible hose assemblies constructed of non-metallic materials intended for installation in piping systems for flammable media and sea water systems where failure may result in flooding, are to be of fire-resistant type except in cases where such hoses are installed on open decks, as defined in Regulation 9.2.3.3.2.2(10) of SOLAS Chapter II-2 as amended by IMO resolutions up to MSC.421(98) and not used for fuel oil lines. Fire resistance is to be demonstrated by testing to ISO 15540:2016 and ISO 15541:2016 as amended.

6.3.6 Flexible hose assemblies are to be selected for the intended location and application taking into consideration ambient conditions, compatibility with fluids under working pressure and temperature conditions consistent with the manufacturer's instructions and any requirements of LHR.

6.4 Installation

6.4.1 In general, flexible hoses are to be limited to a length necessary to provide for relative movement between fixed and flexibly mounted items of machinery/equipment or systems.

6.4.2 Flexible hose assemblies are not to be installed where they may be subjected to torsion deformation (twisting) under normal operating conditions.

6.4.3 The number of flexible hoses, in piping systems mentioned in 6.2.2 is to be kept to minimum and to be limited for the purpose stated in 6.2.1.

6.4.4 Where flexible hoses are intended to be used in piping systems conveying flammable fluids that are in close proximity of heated surfaces the risk of ignition due to failure of the hose assembly and subsequent release of fluids is to be mitigated as far as practicable by the use of screens or other similar protection to the satisfaction of LHR.

6.4.5 Flexible hoses are to be installed in clearly visible and readily accessible locations.

6.4.6 The installation of flexible hose assemblies is to be in accordance with the manufacturer's instructions and use limitations with particular attention to the following:

- Orientation
- End connection support (where necessary)
- Avoidance of hose contact that could cause rubbing and abrasion
- Minimum bend radii

6.5 Tests

6.5.1 Acceptance of flexible hose assemblies is subject to satisfactory prototype testing. Prototype test programmes for flexible hose assembles are to be submitted by the manufacturer and are to be sufficiently detailed to demonstrate performance in accordance with the specified standards.

6.5.2 The tests are, as applicable, to be carried out on different nominal diameters of hose type complete with end fittings for pressure, burst, impulse resistance and fire resistance in accordance with the requirements of the relevant standard. The following standards are to be used as applicable.

- ISO 6802:2018 as amended - Rubber and plastics hoses and hose assemblies with wire reinforcements - Hydraulic impulse test with flexing.
- ISO 6803:2017 as amended - Rubber or plastics hoses and hose assemblies – Hydraulic-pressure impulse test without flexing.
- ISO 15540:2016 as amended - Ships and marine technology - Fire resistance of hose assemblies - Test methods.
- ISO 15541:2016 - Ships and marine technology - Fire resistance of hose assemblies - Requirements for test bench.
- ISO 10380:2012 as amended - Pipework - Corrugated metal hoses and hose assemblies.

Other standards may be accepted where agreed by the classification society.

Note:

Prototype tests are to be carried out for each size of hose assembly. However, for ranges with more than 3 different diameters, the prototype tests are to be carried out for at least:

- the smallest diameter
- the largest diameter
- Intermediate diameters selected based on the principle that prototype tests carried out for a hose assembly with a diameter D are considered valid only for the diameters ranging between 0,5·D and 2·D.

For fire resistance tests the specimens shall be selected in accordance with ISO 15540:2016 as amended.

6.5.3 All flexible hose assemblies are to be satisfactorily prototype burst tested to an international standard* to demonstrate they are able to withstand a pressure not less than four times its design pressure without indication of failure or leakage.

Note *: The international standards, e.g. EN or SAE for burst testing of non-metallic hoses, require the pressure to be increased until burst without any holding period at 4xMWP.

6.6 Marking

6.6.1 Flexible hoses are to be permanently marked by the manufacturer with the following details:

- Hose manufacturer's name or trademark
- Date of manufacture (month/year)
- Designation type reference
- Nominal diameter
- Pressure rating
- Temperature rating

Where a flexible hose assembly is made up of items from different manufacturers, the components are to be clearly identified and traceable to evidence of prototype testing.

SECTION 7 Metal pipes for sea water systems

7.1 General

7.1.1 The following can serve as a guide to the selection of materials, design and construction of metal piping systems intended to be used in sea water services.

7.2 Selection of materials

7.2.1 Water can cause damages to piping systems through the corrosion mechanism, so the materials selected to construct such piping systems must have proven ability to resist general and localised corrosion throughout all the flow velocities likely to be encountered. They must also be compatible with the other materials of the system such as valve bodies and casings in order to minimize bimetallic corrosion. Moreover, they must also resist stress corrosion and corrosion fatigue.

7.2.2 Materials selection should also be based on their amenability to fabrication by normal practices.

7.2.3 The following materials may be used in sea water systems with respect to their individual characteristics:

Galvanized steel

– steel pipes lined with rubber, plastics or stoved coatings.

Copper

– 90/10 Copper-nickel-iron,
– 70/30 Copper-nickel-iron.

Aluminium brass.

7.3 Steel pipes

7.3.1 Steel pipes should be, in any case, protected against corrosion. The minimum required protection for all sea water piping systems, including bilge and ballast lines is to galvanize the internal sur-faces and flanges of steel pipes. Protective coatings should also be applied at the end of any fabrication process, i.e. bending, forming and we lading of the steel pipes.

7.3.2 Welds should be free of imperfections. The surfaces should be dressed to remove slag and spatter and this should be done before coating. The coating should be continuous around the ends of the pipes and on the faces of flanges.

7.3.3 Austenitic stainless-steel pipes are not recommended for salt water services because they dis-play a tendency to pitting, particularly in polluted waters.

7.3.4 Rubber lined pipes or pipes lined with plastics are effective against corrosion and suitable for higher water velocities. The rubber lining should be free from defects, e.g. discontinuities, pinholes, etc., and it is essential that the bonding of the rubber to the internal surface of the pipe and flange face is sound.

7.3.5 Stove coating of pipes as protection against corrosion should only be used where the pipes will be efficiently protected against mechanical damage.

7.3.6 The water velocity in steel pipes, valves and fittings should neither be less than 1 m/sec, to avoid fouling and subsequent pitting, nor more than 3,0 m/sec to avoid corrosion because of excessive turbulence.

7.4 Copper and copper alloy pipes

7.4.1 Copper pipes are particularly tender to perforation by corrosion/erosion and should only be used for low water velocities and where there is no excessive local turbulence, meaning that a good water velocity selection for copper pipes, valves and fittings may vary from 1,0 m/sec to 3,0 m/sec for aluminium-brass, to 3,5 m/sec for 90/10 copper-nickel-iron and to 5,0 m/sec for 70/30 copper-nickel.

7.4.2 Aluminium brass and copper-nickel-iron alloy pipes give good service in reasonably clean sea water. For service with polluted river or harbour waters, copper-nickel-iron alloy pipes with at least 10% Ni are preferable. Alpha-brasses, i.e. those containing 70% or more copper, must be inhibited effectively against dezincification by suitable additions to the composition. Alpha beta-brasses, i.e. those containing less than 70% copper should not be used for pipes and fittings.

7.4.3 New copper alloy pipes should not be exposed initially to polluted water. To allow the pipes develop protective films clean sea water should be used first in such pipes. If this is not practicable, the system should be filled with inhibited town mains water.

7.5 Metal pipes for fresh water services

7.5.1 Mild steel or copper pipes may be employed in fresh water applications. Mild steel should be used for hot water services after consideration because it is liable to be corroded unless the hardness and pH of water are controlled.

7.5.2 Mild steel pipes are not to be used where water, even with a slight salinity, is to be left stagnant, because the low salinity and the limited supply of oxygen in such conditions promote the formation of black iron oxide, and this may give rise to severe pitting. Where it is unavoidable to have stagnant water, steel pipes should be galvanized or pipes of suitable non-ferrous materials are to be used.

7.5.3 Brass fittings and flanges in contact with water should be made of an alpha-brass effectively inhibited against dezincification by suitable additions to the composition.

7.5.4 Aluminium brass has been widely used as material for heat exchangers and condenser tubes, but its use in "once through" systems is not recommended since, under certain conditions, it is prone to pitting and cracking.

7.6 Installation of metal sea water piping systems

7.6.1 Abrupt changes in the direction of flow, protrusions into the bores and generally any kind of restrictions to the flow should be avoided. Branches in continuous flow lines should be set at a shallow angle to the main pipe and the junction should be smooth.

7.6.2 Jointing should be flush with the bore surfaces. The bores should be smooth and clean.

7.6.3 Pipe bends should have as large a radius as possible and have no puckerings in these positions.

7.6.4 The number, position and kind of supports is to be rationally designed in order to prevent the system from excessive vibrations and bending moments.

7.6.5 Systems should not be left idle for long periods, especially where the water is polluted.

7.6.6 Strainers should be provided at the inlet to sea water systems.

7.6.7 Where pipes are exposed to sea water on both external and internal surfaces, flanges should be made, preferably, of the same material as the pipe. Where sea water is confined to the bores of pipes, flanges may be of the same material or of less noble metal than that of the pipes in order to be sacrificed instead of the pipe.

7.6.8 Inert gas shielded arc welding is the preferred process to connect flanges to pipes but metal arc welding may be used on copper-nickel-iron alloy pipes.

7.6.9 Mild steel flanges may be attached by argon arc welding to copper-nickel-iron pipes and give satisfactory service, provided that no part of the steel is exposed to the sea water.

7.6.10 Where silver brazing is used, strength should be obtained by means of the bond in a capillary space over the whole area of the mating surfaces. A fillet braze at the back of the flange or at the face is undesirable. The alloy used for silver brazing should contain not less than 49% silver.

7.6.11 The use of a copper-zinc-brazing alloy is not permitted.

SECTION 8 Type Approval of Mechanical Joints

8.1 General

This specification describes the type testing condition for type approval of mechanical joints intended for use in marine piping systems. Conditions outlined in these requirements are to be fulfilled before Type Approval Certificates are issued. LHR may specify more severe testing conditions and additional tests if considered necessary to ensure the intended reliability and also accept alternative testing in accordance with national or international standards where applicable to the intended use and application.

8.2 Scope

This specification is applicable to mechanical joints defined in 4.4 of this SECTION including compression couplings and slip-on joints of different types for marine use.

8.3 Documentation

Following documents and information are to be submitted by Manufacturer for assessment and/or approval:

a) product quality assurance system implemented
b) complete description of the product
c) typical sectional drawings with all dimensions necessary for evaluation of joint design
d) complete specification of materials used for all components of the assembly
e) proposed test procedure as required in 8.5 and corresponding test reports or other previous relevant tests
f) initial information
   - maximum design pressures (pressure and vacuum)
   - maximum and minimum design temperatures
   - conveyed media
   - intended services
   - maximum axial, lateral and angular deviation, allowed by manufacturer
   - installation details

8.4 Materials

The materials used for mechanical joints are to comply with the requirements of 4.4.4 of this Chapter. The manufacturer has to submit evidence to substantiate that all components are adequately resistant to working the media at design pressure and temperature specified.

8.5 Testing, procedures and requirements

8.5.1 The aim of tests is to demonstrate ability of the pipe joints to operate satisfactory under intended service conditions. The scope and type of tests to be conducted e.g. applicable tests, sequence of testing, and the number of specimen, is subject to approval and will depend on joint design and its intended service in accordance with the requirements of this SECTION. Unless otherwise specified, the water or oil as test fluid is to be used.

8.5.2 Test program

Testing requirements for mechanical joints are to be as indicated in Table 8.8.1.

8.5.3 Selection of Test Specimen

Test specimens are to be selected from production line or at random from stock. Where there is a variety of size of joints requiring approval, a minimum of three separate sizes, representative of the range, from each type of joint to be tested in accordance with Table 8.8.1 are to be selected.

Table 8.8.1:

1	Tightness test	+	+	+	
2	Vibration (fatigue) test	+	+	-	
3	Pressure pulsation test[1]	+	+	-	
4	Burst pressure test	+	+	+	
5	Pull-out test	+	+	-	
6	Fire endurance test	+	+	+	
7	Vacuum test	+[3]	+	+	
8	Repeated assembly test	+[2]	+	-	

Abbreviations:
+ test is required
- test is not required

Footnotes:
1. for use in those systems where pressure pulsation other than water hammer is expected.
2. except press type and swage type.
3. except joints with metal-to-metal tightening surfaces

8.5.4 Mechanical Joint Assembly

Assembly of mechanical joints should consist of components selected in accordance with 8.5.3 and the pipe sizes appropriate to the design of the joints. Where pipe material would affect the performance of mechanical joints, the selection of joints for testing is to take the pipe material into consideration.

Where not specified, the length of pipes to be connected by means of the joint to be tested is to be at least five times the pipe diameter. Before assembling the joint, conformity of components to the design requirements, is to be verified. In all cases the assembly of the joint shall be carried out only according to the manufacturer's instructions. No adjustment operations on the joint assembly, other than that specified by the manufacturer, are permitted during the test.

8.5.5 Test Results Acceptance Criteria

Where a mechanical joint assembly does not pass all or any part of the tests in Table 8.8.1, two assemblies of the same size and type that failed are to be tested and only those tests which the mechanical joint assembly failed in the first instance, are to be repeated. In the event where one of the assemblies fails the second test, that size and type of assembly is to be considered unacceptable. The methods and results of each test are to be recorded and reproduced as and when required.

8.5.6 Methods of tests

(1) Tightness test

In order to ensure correct assembly and tightness of the joints, all mechanical joints are to be subjected to a tightness test, as follows.

a) The mechanical joint assembly test specimen is to be connected to the pipe or tubing in accordance with the requirements of 8.5.4 and the manufacturer's instructions, filled with test fluid and de-aerated.
Mechanical joints assemblies intended for use in rigid connections of pipe lengths, are not to be longitudinally restrained.
The pressure inside the joint assembly is to be slowly increased to 1,5 times the design pressure. This test pressure is to be retained for a minimum period of 5 minutes.
In the event of a drop in pressure or visible leakage, the test (including fire test) is to be repeated for two further specimens.
If during the repeat test one test piece fails, the coupling is regarded as having failed.
An alternative tightness test procedure, such as a pneumatic test, may be accepted.

b) For compression couplings a static gas pressure test is to be carried out to demonstrate the integrity of the mechanical joints assembly for tightness under the influence of gaseous media. The pressure is to be raised to maximum pressure or 70 bar whichever is less.

c) Where the tightness test is carried out using gaseous media as permitted in (a) above, then the static pressure test mentioned in (b) above need not be carried out.

(2) Vibration (fatigue) test

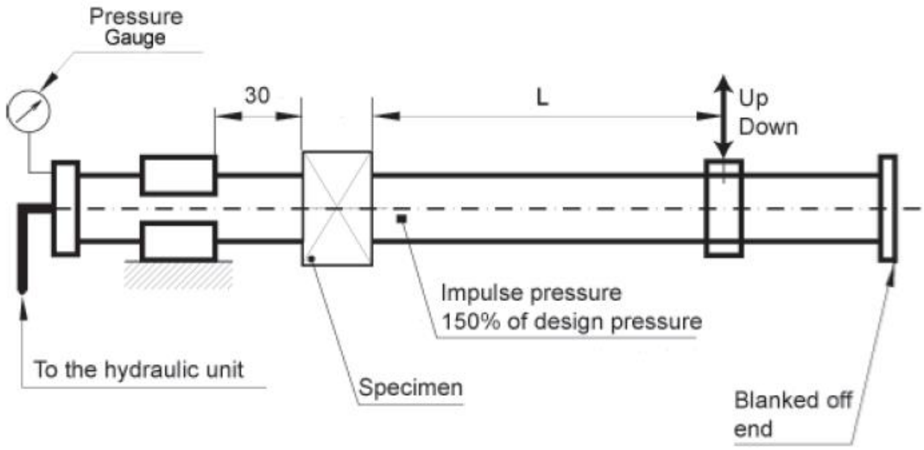
In order to establish the capability of the mechanical joint assembly to withstand fatigue, which is likely to occur due to vibrations under service conditions, mechanical joint assemblies are to be subject to the following vibration test. Conclusions of the vibration tests should show no leakage or damage.

a) Testing of compression couplings and pipe unions

Compression couplings and pipe unions intended for use in rigid pipe connections are to be tested as follows. Rigid connections are joints, connecting pipe length without free angular or axial movement.

Two lengths of pipe are to be connected by means of the joint to be tested. One end of the pipe is to be rigidly fixed while the other end is to be fitted to the vibration rig. The test rig and the joint assembly specimen being tested are to be arranged as shown in Fig.8.8.1.

Figure 8.8.1



The joint assembly is to be filled with test fluid, de-aerated and pressurised to the design pressure of the joint.

Pressure during the test is to be monitored. In the event of a drop in the pressure and visible leakage the test is to be repeated as described in 8.5.5.

Visual examination of the joint assembly is to be carried out.

Re-tightening may be accepted once during the first 1000 cycles.

Vibration amplitude is to be within 5% of the value calculated from the following formula:

$$A = \frac{2 \cdot S \cdot L^2}{3 \cdot E \cdot D}$$

where:

A = single amplitude, mm

L = length of the pipe, mm

S = allowable bending stress in N/mm2 based on 0,25 of the yield stress

E = modulus of elasticity of tube material (for mild steel, E = 210 kN/mm2)

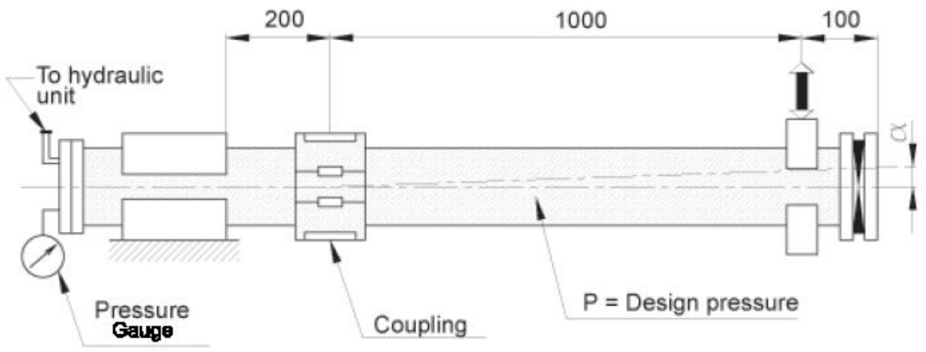
D = outside diameter of tube, mm

Test specimen is to withstand not less than 10^7 cycles with frequency 20 - 50 Hz without leakage or damage.

b) Grip type and Machine grooved type joints

Grip type joints and other similar joints containing elastic elements are to be tested in accordance with the following method. A test rig of cantilever type used for testing fatigue strength of components may be used. The test specimen being tested is to be arranged in the test rig as shown in Fig. 8.8.2.

Figure 8.8.2



Two lengths of pipes are to be connected by means of joint assembly specimen to be tested. One end of the pipe is to be rigidly fixed while the other end is to be fitted to the vibrating element on the rig. The length of pipe connected to the fixed end should be kept as short as possible and in no case exceed 200 mm.

Mechanical joint assemblies are not to be longitudinally restrained.

The assembly is to be filled with test fluid, de-aerated and pressurized to the design pressure of the joint. Preliminary angle of deflection of pipe axis is to be equal to the maximum angle of deflection, recommended by the manufacturer. The amplitude is to be measured at 1 m distance from the center line of the joint assembly at free pipe end connected to the rotating element of the rig. (See Fig. 8.8.2)

Parameters of testing are to be as indicated below and to be carried out on the same assembly:

Number of cycles	Amplitude, mm	Frequency, Hz
$3 \cdot 10^6$	± 0,06	100
$3 \cdot 10^6$	± 0,5	45
$3 \cdot 10^6$	± 1,5	10

Pressure during the test is to be monitored. In the event of a drop in the pressure and visual signs of leakage the test is to be repeated as described in 8.5.5. Visual examination of the joint assembly is to be carried out for signs of damage which may eventually cause leakage.

(3) Pressure pulsation test

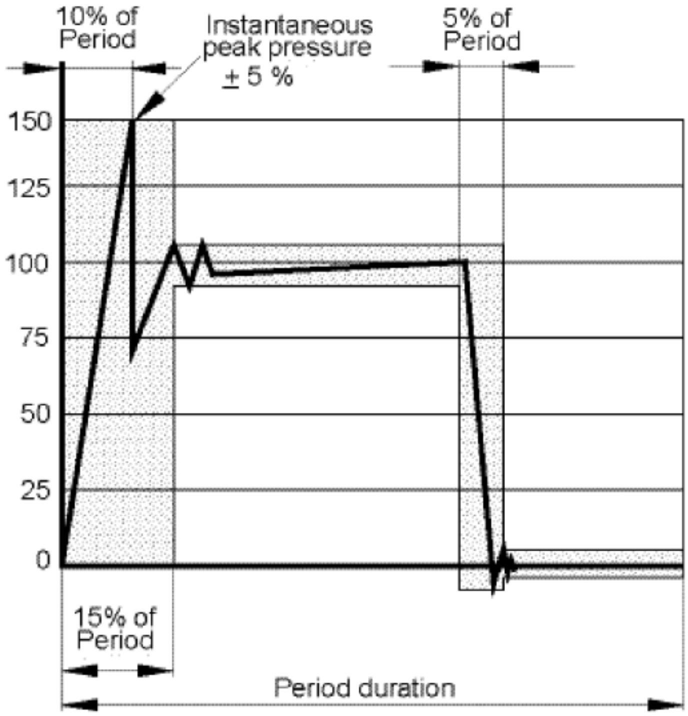
In order to determine capability of mechanical joint assembly to withstand pressure pulsation likely to occur during working conditions, joint assemblies intended for use in rigid connections of pipe lengths, are to be tested in accordance with the following method.

The mechanical joint test specimen for carrying out this test may be the same as that used in the test in 8.5.6.1(a) provided it passed that test.

The vibration test in 8.5.6.2 and the pressure pulsation test are to be carried out simultaneously for compression couplings and pipe unions.

The mechanical joint test specimen is to be connected to a pressure source capable of generating pressure pulses of magnitude as shown in Fig. 8.8.3.

Figure 8.8.3: Impulse pressure diagram



Impulse pressure is to be raised from 0 to 1,5 times the design pressure of the joint with a frequency equal to 30-100 cycles per minute. The number of cycles is not to be less than 5 x 105.

The mechanical joint is to be examined visually for sign of leakage or damage during the test.

(4) Burst pressure test

In order to determine the capability of the mechanical joint assembly to withstand a pressure as stated by 4.4.5 of this Chapter, the following burst test is to be carried out.

Mechanical joint test specimen is to be connected to the pipe or tubing in accordance with the requirements of 8.5.4, filled with test fluid, de-aerated and pressurized to test pressure with an increasing rate of 10% per minute of test pressure. The mechanical joint assembly intended for use in rigid connections of pipe lengths is not to be longitudinally restrained.

Duration of this test is not to be less than 5 minutes at the maximum pressure.

Where considered convenient, the mechanical joint test specimen used in the tightness test in 8.5.6.1, may be used for the burst test provided it passed the tightness test.

The specimen may exhibit a small deformation whilst under test pressure, but no leakage or visible cracks are permitted.

(5) Pull-out test

In order to determine the ability of a mechanical joint assembly to withstand the axial loading likely to be encountered in service without the connecting pipe becoming detached, following pull-out test is to be carried out.

Pipes of suitable length are to be fitted to each end of the mechanical joint assembly test specimen. The test specimen is to be pressurized to design pressure. When pressure is attained, an external axial load is to be imposed with a value calculated using the following formula:

$$L = \frac{\pi}{4} D^2 p$$

where:

L = applied axial load, N

D = pipe outside diameter, mm

p = design pressure, N/mm^2

The pressure and axial load are to be maintained for a period of 5 minutes.

During the test, pressure is to be monitored and relative movement between the joint assembly and the pipe measured. The mechanical joint assembly is to be visually examined for drop in pressure and signs of leakage or damage. There is to be no movement between the mechanical joint assembly and the connecting pipes.

(6) Fire endurance test

In order to establish capability of the mechanical joints to withstand effects of fire which may be encountered in service, mechanical joints are to be subjected to a fire endurance test. The fire endurance test is to be conducted on the selected test specimens as per the following standards.

- ISO 19921:2005 as amended: Ships and marine technology – Fire resistance of metallic pipe components with resilient and elastomeric seals – Test methods
- ISO 19922:2005: Ships and marine technology – Fire resistance of metallic pipe components with resilient and elastomeric seals – Requirements imposed on the test bench.

Clarifications to the standard requirements in ISO19921:2005 as amended, Paragraphs 7.2, 7.4, 7.6 and 7.7:

1. If the fire test is conducted with circulating water at a pressure different from the design pressure of the joint (however of at least 5 bar) the subsequent pressure test is to be carried out to 1,5 times the design pressure.
2. If the fire test is required in Table 7 to be “8 min dry + 22 min wet” or “30 min dry”, i.e. conducted for a period of time without circulating of water, the following test conditions apply:

   <u>Test condition “8 min dry + 22 min wet”</u>

   The test piece is not required to be rinsed with the test medium (water) in preparation for the test as required in Paragraph 7.2 of ISO 19921:2005 as amended. The exposure to fire is to be started and continued for 8 minutes with the sample dry; after 8 minutes of dry test condition the piping system is to be filled with water and test pressure is to be increased up to at least 5 bar within 2 minutes, then maintained to at least 5 bar. After further 22 minutes (i.e. 30 minutes from initial exposure to fire) the exposure to fire is to be stopped and a hydrostatic pressure test as specified in 1. is to be carried out.

Test condition “30 min dry”

The exposure to fire is to be started and continued for 30 minutes with the sample dry. After 30 minutes the exposure to fire is to be stopped and a hydrostatic pressure test as specified in 1. is to be carried out.

Note

For fire tests in dry condition the pressure inside the test specimen is to be monitored for a rise due to heating of the enclosed air. Means of pressure relief should be provided where deemed necessary.

High pressures created during this test can result in failure of the test specimen. Precautions shall be taken to protect personnel and facilities.

Paragraph 7.5 of ISO 19921:2005 as amended does not apply to the dry tests and no forced air circulation is to be arranged.

For fire endurance test requiring exposure time greater than 30 minutes test conditions are adjusted to meet the extended required total exposure time. In all cases for dry-wet test the minimum dry test exposure time is 8 minutes.

3. A selection of representative nominal bores may be tested in order to evaluate the fire resistance of a series or range of mechanical joints of the same design. When a mechanical joint of a given nominal bore (D_n) is so tested then other mechanical joints falling in the range D_n to $2xD_n$ (both inclusive) are considered accepted.
4. Alternative test methods and/or test procedures considered to be at least equivalent may be accepted at the discretion of LHR in cases where the test pieces are too large for the test bench and cannot be completely enclosed by the flames.
5. Where thermal insulation is acceptable as a means of providing fire resistance, following requirements apply:
   a) Thermal insulation materials applied on couplings are to be non-combustible according to ISO 1182:2010 as amended, as required by the Fire Test Procedures Code defined in Regulation 3 of SOLAS Chapter II-2 as amended by IMO resolutions up to MSC.421(98). Precautions are to be taken to protect the insulation from being impregnated with flammable oils.
   b) At least the fire endurance and the vibration testing in table 9 are to be carried out with thermal insulation in place.
   c) A service restriction is to be stated on the type approval certificate that the mechanical joints are to be fitted with thermal insulation during the installation in cases where the mechanical joints are used where fire resistance is required, unless mechanical joints are delivered already fitted with thermal insulation before installation.

(7) Vacuum test

In order to establish the capability of the mechanical joint assembly to withstand internal pressures below atmospheric, similar to the conditions likely to be encountered under service conditions, the following vacuum test is to be carried out.

The mechanical joint assembly is to be connected to a vacuum pump and subjected to a pressure of 170 mbar absolute. Once this pressure is stabilized, the specimen under test is to be isolated from the vacuum pump and the pressure is to be maintained for a period of 5 minutes.

No internal pressure rise is permitted.

(8) Repeated assembly test

The mechanical joint test specimen is to be dismantled and reassembled 10 times in accordance with manufacturer's instructions and then subjected to a tightness test as defined in 8.5.6.1.

CHAPTER 9 Piping Systems

CONTENTS

SECTION 1 General

1.1 Scope

1.1.1 The Rules of this Chapter apply to piping systems, including valves, fittings and pumps, used on all types of ships, except where otherwise stated.

1.1.2 The requirements conform, where relevant, with those of the International Convention on Load Lines, 1966. Reference should also be made to any additional requirements of the National Authority of the country in which the ship is to be registered and to the relevant regulations of the International Convention for the Safety of Life at Sea, 1974 and applicable amendments.

1.1.3 LHR shall give special consideration to cases or arrangements, submitted for approval, which are presented by the designer as equivalent to those required by these Rules.

1.1.4 Consideration will also be given to the pumping arrangements of small ships and ships to be assigned class notations for restricted or special services.

1.2 Documents for approval

1.2.1 Before proceeding with the work, plans in triplicate are to be submitted, showing clearly the arrangements and details of the following:

1) General arrangements of pumps and piping, including any cross-flooding pipes and fittings.
2) Vent, sounding, and overflow pipes.
3) Bilge and ballast systems. In the case of passenger ships, the criterion numeral, as defined in the International Convention for the Safety of Life at Sea, 1974 and applicable amendments, is to be stated, together with the number of flooded compartments which the ship is required to withstand under damage conditions. In cases where subdivision and stability matters are controlled by National Authorities, one copy of all the approved thereby relevant documents, booklets etc. should be submitted for information purposes to LHR.
4) Fuel oil systems (bunkering, transfer and supply lines) including fuel oil settling, service and other fuel oil tanks not forming part of the ship's structure.
5) Fuel oil overflow systems, where these are fitted.
6) Boiler feed water systems.
7) Steam and exhaust piping systems.
8) Lubricating oil systems and arrangements of other flammable liquids.
9) Cooling water systems for main and auxiliary services.
10) Compressed air systems for main and auxiliary services.
11) Hydraulic power piping systems.
12) Fire main and fire extinguishing systems.
13) Steering gear piping systems.
14) Inert gas system arrangements (where fitted).
15) Crude oil washing arrangements (where fitted).
16) Drainage of decks.

1.2.2 All plans are to consist of a diagrammatic drawing of each system accompanied by lists of material showing size, wall thickness, maximum working pressure and material of all pipes and the type, size and material of valves and fittings. Where steam is the working fluid, temperatures and pressures are also to be given. References to weld attachments may also be required.

1.2.3 For steam lines with working temperatures greater than 400°C, the relevant stress calculations together with isometric data are to be submitted.

1.3 Piping systems location for ships with load lines based on damage stability calculations

1.3.1 Where assignment of load lines is based on damage stability considerations, in order to avoid progressive flooding between compartments due to a possible damage of piping systems the following arrangements are to be made:

1) Bilge and other piping systems are to be located inboard of B/5 measured from the ship's side at right angles to the centerline at the level of the summer load waterline, where B is the moulded breadth of the ship.
2) Where it is not practicable to locate piping systems as indicated in (1), bilge suction pipes are to be provided with non-return valves of an approved type. Other piping systems are to be provided with valves capable of being operated from accessible positions above the bulkhead deck. These valves are to be located in the compartment containing the open ends of the pipes and are to effectively isolate the compartment.

SECTION 2 Designs

2.1 Materials

2.1.1 Pipes, valves and fittings are to be made of steel, cast iron, copper, copper alloys or other approved material appropriate for the intended service.

2.1.2 Where applicable, materials used for pipes, valves and fittings are to comply with the relevant requirements contained in Part 5, Chapter 8.

2.1.3 Materials sensitive to heat such as aluminium, lead or plastics, are not to be used in systems whose failure may bring the safe operation of a ship into trouble or for containing combustible liquids or sea water where leakage or failure could cause fire or flooding of watertight compartments.

2.1.4 Special consideration will be given to cases where alloys or other materials not covered by these Rules are proposed for use.

2.2 Pipe wall thicknesses

2.2.1 The minimum nominal wall thicknesses of steel, copper and copper alloy pipes are to be in accordance with Part 5, Chapter 8 or the wall thicknesses of pipes made of materials other than steel, copper and copper alloy, the requirements of 2.1.4 are to be applied.

2.3 General requirements for the installation of piping systems

2.3.1 Piping systems shall be adequately identified according to their purpose. Moreover, valves are to be permanently and clearly marked.

2.3.2 As far as is practicable, piping systems are not to be led close to electrical switchboards and appliances where the drip or escape of liquid, gas or steam from joints or fittings could cause damage to the electrical installation. Where it is not practicable to comply with these requirements, drip trays or shields are to be provided as found necessary. Short sounding pipes to tanks are not to terminate near electrical appliances.

2.3.3 Piping systems are to be so arranged that they can be completely emptied and drained. Piping systems in which the accumulation of liquids during operation could cause damage must be equipped with special drain arrangements.

2.4 Valves and other shut-off devices

2.4.1 Valves are to be constructed and tested according to a standard accepted by LHR (e.g. ISO 4126-1981 "Safety valves-General requirements", ISO 5996-1984 "Cast iron gate valves" etc.).

2.4.2 Valves over 40mm diameter in Class I and II piping systems are to be of the outside screw type.

2.4.3 All valves are to be permanently marked. The marking must contain at least the following details:

– Material of valve body
– Nominal diameter
– Nominal pressure

2.4.4 Valves with screw-on covers are to be constructed as to prevent the possibility of these covers or glands being slackened or loosened when the valves are operated.

2.4.5 Unless otherwise specifically mentioned in the Rules, the valves are to be fitted in places where they are at all times readily accessible.

2.4.6 All valves are to be arranged to shut with a right-hand (clockwise) motion of the wheels and are to be provided with nameplates and indicators showing whether they are open or shut unless this is readily obvious.

2.4.7 Change-over devices in piping systems in which a possible intermediate position of the device could be dangerous in service must not be used.

2.5 Remote - controlled valves

2.5.1 These Rules apply to hydraulically, pneumatically or electrically operated valves in piping systems. The remote control of valves by mechanical means such as rich rods is to follow these Rules as far as it is practicable.

2.5.2 All valves which are provided with remote control are to be arranged for local manual operation, independent of the remote operating mechanism. Controlling the valve by local manual means is not to render the remote-control system inoperable.

2.5.3 Valves and control lines of bilge and ballast piping systems are to be located as far away as possible from the bottom and sides of the ship, in order to be protected in case of grounding and local damage of the hull.

2.5.4 Where remote-controlled valves are arranged inside the ballast tanks, the valves should always be located in the tank adjoining that to which they relate.

2.5.5 Remote controlled valves mounted on fuel tanks located above the double bottom must be capable of being closed from outside the compartment in which they are installed.

2.5.6 Where remote-controlled valves are arranged inside the cargo tanks, the valves should always be located in the tank adjoining that to which they relate. A direct arrangement of the remote-controlled valves in the tanks concerned is allowed only if each tank is fitted with two suction lines each of which is provided with a remote-controlled valve.

2.5.7 The control devices of remote-controlled valves are to be arranged together in one control stand. The controlled devices are to be clearly identified and marked.

2.5.8 On passenger ships, the control stand for remote controlled bilge valves is to be located outside the machinery space and above the bulkhead deck.

2.5.9 The control devices of valves for tanks with alternative uses are to be interlocked to ensure that only the valve relating to the tank concerned can be operated. The same also applies to the valves of cargo holds and tanks in which dry cargo and ballast water are carried alternately.

2.5.10 The energy required for the closing of valves which are not closed by spring power is to be supplied by a pressure accumulator.

2.5.11 Pneumatically operated valves, notwithstanding any special relevant requirements applicable in each system, may be supplied with air from the general compressed air system. Where the quick-closing valves of fuel tanks are closed pneumatically, a separate pressure accumulator is to be provided. This is to be of adequate capacity and is to be located outside the engine room. Filling of this accumulator by a direct connection to the general compressed air system is allowed. A non-return valve is to be arranged in the filling connection of the pressure accumulator. The accumulator is to be provided either with a pressure control device with a visual and acoustic alarm or with a hand-compressor as a second filling appliance. The hand-compressor is to be located outside the engine room.

2.5.12 Additional requirements for remote controlled bilge valves in passenger ships are mentioned in 8.9.

2.5.13 After installation on board, the entire system dealing with remote controlled valves is to be subjected to an operational test.

2.6 Valves, fittings and pipes attached to watertight plating

2.6.1 Valves, fittings or pipes attached direct to tank plating, bulkheads, flats or tunnels which are required to be of watertight and oil-tight construction, are to be secured by means of studs screwed through the plating or by tap bolts, and not by bolts drilling the plating. Alternatively, the studs or the bulkhead piece may be welded to the plating.

2.6.2 For requirements relating to valves on the collision bulkhead, see 3.6.3.

2.7 Ship side valves and fittings (other than those on scuppers and sanitary discharges)

2.7.1 All sea inlet and overboard discharge pipes are to be fitted with valves secured direct to the shell plating or to the plating of fabricated steel water boxes attached to the shell plating. Fittings bolted to the plating are to have the bolt heads countersunk and the bolts tapped into the plating. Where a reinforcing ring is welded to the inside of the plating, studs may be used not penetrating the plating.

2.7.2 Distance pieces of short, rigid construction, and made of approved material, may be fitted between the valves and the shell plating. They may be welded to the shell plating. Details of the welded end connections and of fabricated steel water boxes are to be submitted to LHR for approval.

2.7.3 All ship side openings for sea inlet valves and inlet water boxes are to be provided with gratings. The net area through the gratings is to be not less than twice that of the valves connected to the sea inlets, and provision is to be made for clearing the gratings by use of low-pressure steam or compressed air.

2.7.4 Sea inlet and overboard discharge valves and cocks are to be fitted in easily accessible positions and so far as practicable, are to be readily visible. Indicators are to be provided local to the valves and cocks showing whether they are open or shut. Provision is to be made for preventing any discharge of water into lifeboats. The valve spindles are to extend above the lower platform, and the hand wheels of the main cooling water sea inlet and emergency bilge suction valves are to be situated not less than 460mm above this platform.

2.7.5 Ship side valves and fittings, if made of steel or other approved material with low corrosion resistance, are to be suitably protected against wastage.

2.7.6 The scantlings of valves and valve stools fitted with steam or compressed air clearing connections are to be suitable for the maximum pressure to which the valves and stools are subjected.

2.7.7 All suction and discharge valves and cocks secured direct to the shell plating of the ship are to be fitted with spigots passing through the plating, but the spigots on the valves or cocks may be omitted if these fittings are attached to pads or distance pieces which themselves form spigots in way of shell plating. Blow-down valves or cocks are also to be fitted with a protection ring through which the spigot is to pass, the ring being on the outside of the shell plating. Where alternative forms of attachment are proposed, details are to be submitted for consideration.

2.7.8 Blow-down valves or cocks on the ship side are to be fitted in accessible positions above the level of the working platform, and are to be provided with indicators showing whether they are open or shut. Cock handles are not to be capable of being removed unless the cocks are shut, and, if valves are fitted, the hand wheels are to be suitably retained on the spindle.

2.8 Installation and expansion of piping systems

2.8.1 Bilge, ballast and cooling water suction and discharge pipes are to be permanent, made in readily removable lengths with flanged joints. Exception to this Rule is imposed where a possible leakage in a flange may cause troubles such as a leakage in bilge pipes in way of deep tanks (see 7.5.2). For joints in oil fuel piping systems the requirements of Part 5, Chapter 10 are applicable.

2.8.2 All pipes are to be efficiently secured in position to prevent chafing or lateral movement. Long or heavy lengths of pipes are to be supported by bearers so that no undue load is carried by the flanged connections of the pumps or fittings to which they are attached. Where lack of space prevents the use of normal circular flanges, details of alternative methods of joining the pipes are to be submitted.

2.8.3 Each range of pipes may have suitable provision for expansion. Expansion pieces, where fitted, are to be of an approved type incorporating special quality oil resistant rubber or other synthetic material when they are used in cooling water lines in machinery spaces. Where fitted in sea water lines, they are to be provided with guards which will effectively enclose, but not interfere with the action of the expansion pieces and will reduce to the minimum practicable any flow of water into the machinery spaces in the event of failure of the flexible elements. Proposals to use such fittings in water lines for other services, including: ballast lines in machinery spaces, in duct keels and inside double bottom water ballast tanks, and bilge lines inside duct keels only, will be specially considered when plans of the pumping systems are submitted for approval.

2.8.4 All pipes situated in cargo spaces, fish holds, chain lockers or other positions where they are liable to mechanical damage are to be efficiently protected.

2.8.5 Wash deck pipes and discharge pipes from the pumps to domestic water tanks are not to be led through cargo holds.

2.8.6 All pipes, including scupper pipes, air pipes and sounding pipes which pass through chambers intended for the carriage of refrigerated products are to be well insulated.

2.8.7 Where the pipes referred to in 2.8.6 pass through chambers intended for temperatures of 0°C or below, they are also to be insulated from the steel structure, except in positions where the temperature of the structure is mainly controlled by the external temperature and will normally be above freezing point. Pipes passing through a deck plate within the ship side insulation, where the deck is fully insulated below and has an insulation riband on top, are to be attached to the deck plating. In the case of pipes adjacent to the shell plating, metallic contact between the pipes and the shell plating or frames is to be arranged as far as practicable.

2.8.8 The air refreshing pipes to and from refrigerated compartments need not, however be insulated from the steel work.

2.8.9 In order to avoid unacceptable overpressures, the following piping systems are to be fitted with safety valves:

1) Piping systems and valves in which liquids can be enclosed and heated;
2) Piping systems which may be exposed in service to pressures in excess of the design pressure.

Safety valves must be capable of discharging the medium at a maximum pressure increase of 10%.
Safety valves are to be fitted on the low-pressure side of reducing valves.

SECTION 3 Drainage of compartments other than machinery spaces

3.1 General requirements

3.1.1 An efficient pumping plant is to be provided in all ships capable of draining any compartment when the vessel is on an even keel and either upright or listed 5° through at least one suction. For this purpose wing suctions will often be necessary, except in narrow compartments where one suction can provide effective drainage.

3.1.2 The pumping plants installed in all passenger ships are to be as efficient as needed in order to provide effective drainage of any watertight compartment under all practicable conditions after a casualty, whether the ship is upright or listed.

3.1.3 The bilge arrangements of small ships or ships intended for or special services will be specially considered in each case.

3.2 Cargo holds

3.2.1 Cargo holds are to be efficiently drained by suctions connected to the main bilge line (named branch bilge suctions) located in well considered positions for good drainage.

3.2.2 In ships having only one hold, and this over 30m in length, bilge suctions are to be fitted in suitable positions in the fore and after sections of the hold.

3.2.3 Cargo holds with bilge gutterways are to be fitted with bilge suctions fore and aft. Where close ceiling or continuous gusset plates are fitted over the bilges, arrangements are to be made whereby water in a hold compartment may find its way to the suction pipes.

3.2.4 Where the inner bottom plating extends to the ship side, the bilge suctions are to be led to wells placed at the wings. If the tank top plating has inverse camber, a well is also to be fitted at the centerline, but in the case of trawlers and fishing vessels, a single well fitted at the centre may be accepted.

3.3 Tanks, cofferdams, void spaces and pipe tunnels

3.3.1 All tanks (including double bottom tanks), whether used for water ballast, fuel oil or liquid cargoes are to be provided with suction pipes from the after end of each tank, in the light of the general requirements of 3.1.

3.3.2 Where the tanks are divided by longitudinal watertight bulkheads or girders into two or more tanks, a single suction pipe, led to the after end of each tank, will normally be acceptable.

3.3.3 Cofferdams, pipe tunnels and void spaces adjoining to the ship's shell are to have suctions located as specified in 3.1, 3.2 and 3.3 leading to the main bilge line.

3.4 Refrigerated cargo spaces

3.4.1 Refrigerated cargo spaces and thawing trays are to be provided with drains which should be prevented from being blocked frozen, by local heating or other suitable means. Each drain pipe is to be fitted at its discharge end with a trap to prevent the transfer of heat and odours.

3.5 Spaces which may be used for the alternative carriage of liquid or dry cargo

3.5.1 Where dry cargo holds or deep tanks are also intended for the alternative carriage of water ballast, liquid or dry cargo, the drainage arrangements are to be according with the following:

1) for dry cargoes, see 3.1, 3.2
2) for water ballast, fuel oil or cargo oil having a flash point of 60°C or above, see 3.3
3) for fuel oil or cargo oil having a flash point below 60°C, see Part 5, Chapter 10.

3.5.2 For the above mentioned multi-used spaces the branch bilge pipes from these spaces may be connected to the ballast or cargo pipe system by change-over valves. When the tank or hold is being used for the carriage of dry cargo, water ballast filling and suction pipes are to be blank flanged. If the tank or hold is being used for the carriage of water ballast the bilge suction is to be blank flanged.

3.5.3 Where a ship is designed for the alternative carriage of dry cargo or oil having a flash point below 60°C, the blanking arrangements will be specially considered.

3.6 Fore and after peaks

3.6.1 Connection of the fore and after peak tanks to the general bilge system is not permitted. Where the peak tanks are not connected to the ballast system, separate means of pumping are to be provided having been approved by LHR.

3.6.2 In the case of trawlers and fishing vessels, where the after peak has a common bulkhead with the engine room, its drainage to the engine room bilge may be affected by means of a self-closing cock fitted in a well-lit and readily accessible position.

3.6.3 In passenger ships, the collision bulkhead may be pierced below the bulkhead deck by not more than one pipe for dealing with fluid in the forepeak tank, provided that the pipe is fitted with a screwdown valve capable of being operated from above the bulkhead deck, the valve chest being secured inside the forepeak to the collision bulkhead. LHR may however give its permission for the fitting of this valve on the after side of the collision bulkhead provided that the valve is readily accessible under all service conditions and the space in which it is located is not a cargo space, taking into consideration the relevant requirements of the National Authorities. An indicator is to be provided to show whether the valve is open or closed.

3.6.4 Where the fore peak is divided into two compartments, the collision bulkhead may be pierced below the bulkhead deck (i.e. one for each compartment). Each pipe is to be provided with a screw-down valve, fitted and controlled as in 3.6.3. In passenger ships the arrangement will be required to be specially approved. It must be proved to LHR 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.

3.6.5 In ships other than passenger ships, pipes piercing the collision bulkhead are to be fitted with suitable valves operable from above the freeboard deck and the valve chests are to be secured to the bulkhead inside the fore peak. The valves may be fitted on the after side of the collision bulkhead, provided that the valve is readily accessible under all service conditions and the space in which it is located is not a cargo space.

3.7 Spaces above fore peaks, after peaks and machinery spaces

3.7.1 Chain lockers and watertight compartments above the fore peak may be drained either through the main bilge system or by means of hand pumps. These spaces may not be drained to the fore peak.

3.7.2 Steering gear compartments or other spaces above the after peak are to be provided with suitable means of drainage, either by hand pumps or through the main bilge system. These spaces may be drained to the shaft tunnel or machinery space, provided that the drain line is fitted with a self-closing shutoff valve at a clearly visible and easily accessible position. The drain pipes shall have an inside diameter of at least 40mm. In passenger ships the arrangement will be required to be specially approved with regard to subdivision considerations.

3.7.3 In the case of trawlers and fishing vessels, accommodation spaces which overhang the machinery space, may also be drained as in 3.7.2.

3.8 Integrity of bulkheads

3.8.1 Steering gear compartments or other spaces above the after peak as mentioned in 3.7.2, may be drained to the shaft tunnel or the machinery space by drain pipes. However, the integrity of the machinery space bulkheads or the shaft tunnel plating when it is required to be watertight, must not be impaired by the fitting of these drain pipes or scuppers. They may be led into a strongly constructed scupper drain tank situated in the machinery space or shaft tunnel, but closed to these spaces and drained by means of a suction of appropriate size led from the main bilge line through a screw-down non-return valve.

3.8.2 The scupper tank vent pipe is to be led to above the bulkhead deck, and provision is to be made for ascertaining the level of the water in the tank.

3.8.3 Where one tank is used for the drainage of several watertight compartments, the scupper pipes are to be provided with screw-down non-return valves.

3.8.4 No drain valves or cocks are to be fitted to watertight bulkheads if other means of drainage are practicable. No drain valves or cocks are to be fitted to the collision bulkhead.

3.9 Draining and Pumping Forward Spaces in Bulk Carriers (IACS UR M65 Rev.1 (2004))

3.9.1 Application

This requirement applies to bulk carriers constructed generally with single deck, top-side tanks and hopper side tanks in cargo spaces intended primarily to carry dry cargo in bulk, and includes such types as ore carriers and combination carriers, which are contracted for construction on or after 1 January 2005 [(1)].

3.9.2 Dewatering capacity

The dewatering system for ballast tanks located forward of the collision bulkhead and for bilges of dry spaces any part of which extends forward of the foremost cargo hold [(2)] is to be designed to remove water from the forward spaces at a rate of not less than $320A\,m^3/h$, where A is the cross-sectional area in m^2 of the largest air pipe or ventilator pipe connected from the exposed deck to a closed forward space that is required to be dewatered by these arrangements.

NOTE:

1. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contract for construction”, refer to IACS Procedural Requirement (PR) No. 29.
2. Reference is made to SOLAS regulation XII/13 and Unified Interpretation SC 179 "Dewatering of forward spaces of bulk carriers".

SECTION 4 Bilge drainage of machinery spaces

4.1 General requirements

4.1.1 The bilge drainage arrangement in the machinery space should contain:
1) at least one branch bilge suction i.e. a suction connected to the main bilge line
2) at least one direct bilge suction i.e. a suction led direct to an independent power pump
3) an emergency bilge suction draining from a suitable low level in the machinery space which is to be fitted with a screw-down non-return valve having the spindle extended so that the hand wheel is not less than 460mm above the bottom platform.

4.1.2 Any water which may enter this compartment must be pumped out through at least the (1) and (2) bilge suctions referred in 4.1.1 when the ship is on an even keel and is either upright or has a list of not more than 5°.

4.1.3 In passenger ships, the drainage arrangements are to be such that machinery spaces can be pumped out under all practical conditions after a casualty, whether the ship is upright or listed.

4.2 Suctions in the machinery space

4.2.1 Where the double bottom extends the full length of the machinery space and forms bilges at the sides, at least one branch and one direct bilge suction must be provided at each side.

4.2.2 Where the double bottom plating extends the full breadth and length of the compartment, at least one bilge well must be situated in each side with one branch bilge suction and one direct bilge suction inside.

4.2.3 Where there is no double bottom and the rise of floor is not less than 5°, one branch and one direct bilge suction are to be led to accessible positions as near the centerline as practicable.

4.2.4 In ships without double bottom in the machinery space, where the rise of floor in the machinery space is less than 5°, and on all passenger ships, additional bilge suctions are to be provided at the wings.

4.2.5 Additional bilge suctions may be required because of discontinuities or recesses in the structure of the machinery space.

4.2.6 When the machinery space is situated in the after end of the ship, it will generally be necessary for bilge suctions to be fitted in the forward wings as well as in the after end of the machinery space. However, the positioning of bilge suctions in every machinery space wherever located in a ship, will be determined with regard to the structural arrangements of the compartment.

4.2.7 Ships with electrical propulsion will have special means in order to prevent accumulation of bilge water under generators and motors.

4.3 Separate machinery spaces

4.3.1 In ships where the machinery space is divided by watertight bulkheads to isolate the boiler room(s) or auxiliary engine room(s) from the main engine room, the number and position of branch bilge suctions in the boiler room(s) or auxiliary engine room(s) are to be the same as for cargo holds.

4.3.2 In addition to 4.3.1, at least one direct bilge suction led to an independent power pump, is to be fitted in each compartment. Similar provision is to be made in separate motor rooms of electrically propelled ships.

4.3.3 In passenger ships, each independent bilge pump is to have a direct bilge suction from the space in which it is situated, but not more than two such suctions are required in any one space. Where two or more such suctions are provided, there is to be at least one suction on each side of the space.

4.3.4 An emergency bilge suction is to be provided in each machinery space where the main engine or part of the propulsion plant is situated.

4.4 Emergency bilge suction connection

4.4.1 The emergency bilge suction is normally led to the main cooling water pump. Where two or more cooling water pumps are provided, each capable of supplying cooling water for normal power, only one pump need be fitted with the emergency bilge suction.

4.4.2 In ships with main cooling water pumps are of a capacity lower than that required to handle bilge pumping duties or where there are not the suitable piping arrangements, the emergency bilge suction is to be led to the largest available pump which is not a bilge pump as prescribed in 6.1 and 6.2. This pump is to have a capacity not less than that required for the bilge pump. The emergency bilge suction is to be of same size as that of the pump suction.

4.4.3 Where the pump to which the emergency bilge suction is connected is of the self-priming type, the direct bilge suction on the same side of the ship as the emergency bilge suction may be omitted, except in passenger ships.

4.4.4 Emergency bilge suction valve nameplates are to be marked: FOR EMERGENCY USE ONLY

4.5 Shaft tunnel drainage

4.5.1 A bilge suction located in a tunnel well is to be arranged at the after end of the shaft tunnel. Where the structure of the bottom of the tunnel or its length requires this, additional bilge suctions are to be provided at the forward end.

4.5.2 Bilge valves for the shaft tunnel are to be arranged outside the tunnel in the engine room. The bilge suction(s) for the drainage of shaft tunnel are to be connected to the main bilge system.

SECTION 5 Bilge suction design

5.1 General

5.1.1 The values derived from the expression given in this Section are to be rounded up to the next higher nominal diameter.

5.2 Dry cargo and passenger ships

5.2.1 The diameter d_M of the main bilge line is to be neither less than that required from the following expression nor that required for any branch bilge suction:

$$d_M = 1{,}68 \cdot \sqrt{L \cdot (B + D)} + 25 \ \ [mm]$$

where:

d_M = internal diameter of main bilge line, mm

L = length of ship between perpendiculars, m

B = moulded breadth of ship, m

D = moulded depth to bulkhead deck, m

5.2.2 The diameter, d_B of branch bilge suction pipes in either the cargo spaces or the machinery spaces is to be not less than that required by the following formula:

$$d_B = 2{,}15 \cdot \sqrt{C \cdot (B + D)} + 25 \ \ [mm]$$

where:

d_B = internal diameter of branch bilge suction, mm

C = length of the compartment, m

B,D = as defined in 5.2.1.

5.2.3 No main bilge piping is to be less than 63mm internal diameter. No branch piping is to be less than 50mm.

5.3 Direct bilge suctions other than emergency suctions

5.3.1 The direct bilge suctions in the main engine room, and the direct bilge suctions in large separate boiler rooms, motor rooms of ships with electrical propulsion, and auxiliary engine rooms are not to be of a diameter less than that required for the main bilge line.

5.3.2 Where the separate machinery spaces are of small dimensions, the sizes of the direct bilge suctions to these spaces will be specially considered.

5.4 Emergency bilge suctions

5.4.1 When ships have steam propulsion the emergency bilge suction is to have an internal diameter of at least two-thirds that of the pump suction.

5.4.2 In ships other than steam ships the internal diameter of the emergency bilge suction is to be the same as the suction of the connected pump.

5.5 Tunnel suctions

5.5.1 The bilge suction pipe to the tunnel well is to be not less than 65mm bore, except in ships not exceeding 60m in length, in which case it may be 50mm bore.

5.6 Distribution chest branch pipes

5.6.1 Several branch bilge suctions may be led to a distribution chest. This is to be connected with the main bilge line by branch pipes. The area of such a branch pipe is to be not less than the sum of the areas required by the Rules for the two largest branch bilge suction pipes connected to that chest, but need not be greater than that required for the main bilge line.

5.7 Main bilge line of tankers and similar ships

5.7.1 The diameter of the main bilge pipe in the engine rooms of tankers and similar ships is given by:

$$d_M = 3{,}0 \cdot \sqrt{C \cdot (B + D)} + 35 \quad [mm]$$

where:

d_M	=	internal diameter of main bilge line, mm
C	=	length of the compartment, m
B	=	moulded breadth of ship, m
D	=	moulded depth to bulkhead deck, m.

SECTION 6 Bilge pumps

6.1 Required number of pumps

6.1.1 Cargo ships are to be provided with at least two independent power-driven bilge pumping units connected with the main bilge system. On ships up to 90 m in length, one of these units may be driven from the main engines. In larger ships both units are to be independently driven.

6.1.2 Each unit may consist of one or more pumps connected to the main bilge line, provided that their combined capacity is adequate.

6.1.3 In passenger ships, at least three power-driven pumps shall be fitted connected to the main bilge system, one of which may be attached to the propulsion unit. Where the criterion numeral as derived from Regulation 6.3 of Chapter II-1 of the International Convention for the Safety of Life at Sea, 1974 and applicable amendments, is 30 or more, one additional independent power-driven pump shall be provided.

6.1.4 Special consideration will be given to the number of pumps for small ships and, in general, if there is a class notation restricting a small ship to harbour or river service, a hand pump may be accepted in lieu of one of the bilge pumping units.

6.2 Types of pumps for bilge duties

6.2.1 Sanitary, ballast and general service pumps may be accepted as independent power bilge pumps if fitted with the necessary connections to the bilge pumping system provided that they are of the required capacity and they are immediately available for bilge duty when required.

6.2.2 Fuel and oil pumps may not be connected to the bilge system.

6.2.3 Bilge ejectors combined with a high-pressure sea water pump may be accepted as a substitute for an independent bilge pump in ships other than passenger ships.

6.2.4 All power-driven pumps which are essential for bilge duties are to be of the self-priming type, unless an approved central priming system is provided for these pumps. Details of this system are to be submitted.

6.2.5 Cooling water pumps having emergency bilge suctions need not be of the self-priming type.

6.3 Capacity of pumps

6.3.1 Each bilge pump is to be connected to the main bilge line. The capacity of each bilge pump is to be not less than required by the following formula:

$$Q = 0{,}00575 \cdot d_M^2, \quad m^3/hour$$

where:

Q = capacity, m^3/hour,

d_M = rule internal diameter of main bilge line, mm.

In cases where instead of one pump a pumping unit is available, the aggregate capacity of the pumps of that unit is to be of the above-mentioned capacity.

6.3.2 In ships other than passenger ships, where one bilge pump or pumping unit is of slightly less than Rule capacity, the deficiency is acceptable provided that the other pump is designed for a correspondingly larger capacity. However, the capacity of the smaller bilge pump shall not be less than 30% of the calculated capacity.

6.4 Pump connections

6.4.1 Bilge pumps are to be connected in a way that one pump may continue to be in operation when the other pump is being opened up for overhaul.

6.4.2 Pumps required for essential services are not to be connected to a common suction or discharge chest or pipe unless the arrangements are such that the working of any pumps so connected is not affected by the other pumps being in operation at the same time.

6.5 Direct bilge suctions

6.5.1 The direct bilge suctions in the machinery space(s) are to be led to independent power-driven pumps and the arrangements are to be such that these direct suctions can be used independently of the main bilge line suctions.

SECTION 7 Pipe systems and fittings

7.1 Main bilge line suctions

7.1.1 Branch bilge suctions are to be arranged to draw water from any watertight compartment other than a space permanently used for the carriage of fresh water, water ballast, fuel oil or liquid cargo and for which other efficient means of pumping are provided or spaces mentioned in 3.7 where hand pumps are acceptable.

7.2 Bilge piping systems

7.2.1 Bilge pipes are to be entirely distinct from sea inlet pipes or from pipes which may be used for filling or emptying spaces where water or oil is carried. This does not, however, exclude a bilge ejection connection, a connecting pipe from a pump to its suction valve chest, or a deep tank suction pipe suitably connected through a change-over device to a bilge, ballast or oil line.

7.2.2 The arrangement of the bilge system including its valves, fittings and pipes is to be such as to prevent any possibility of one watertight compartment being placed in communication with another, or of dry cargo spaces, machinery spaces or other dry compartments being placed in communication with the sea or with tanks. In order to satisfy this requirement, screw down non-return valves are to be fitted in the following:

(1) In bilge distribution chests
(2) In bilge suction connections, whether fitted direct to the pump or on the main bilge line.
(3) In direct bilge suctions.
(4) In bilge pumps connections to the main bilge line.

7.3 Mud boxes, strum boxes and bilge wells

7.3.1 Bilge lines in machinery spaces and shaft tunnels, other than emergency bilge suctions, are to be fitted with mud boxes easily accessible from the floor plates. They shall have straight tail pipes to the bilges and covers secured in such a manner as to permit their being expeditiously opened or closed. Strum boxes are not to be fitted to the lower ends of these tail pipes or to the emergency bilge suctions.

7.3.2 The open ends of bilge suctions in holds and other compartments outside machinery spaces and shaft tunnels are to be enclosed in strum boxes having perforations of not more than 10mm diameter, whose combined area is not less than twice that required for the suction pipe. The boxes are to be so constructed that they can be cleared without breaking any joint of the suction pipe.

7.3.3 Bilge wells are to be formed of steel plates and are to be not less than 0,15m^3 capacity. In small compartments, steel bilge hats of reasonable capacity may be fitted.

7.3.4 In passenger ships, the depth of bilge wells in double bottom tanks will be specially considered.

7.3.5 Where access manholes to bilge wells are necessary, they are to be fitted as near to the suction strums as practicable.

7.3.6 The distance between the foot of all bilge tail pipes and the bottom of the bilge well is to be adequate to allow a full flow of water to facilitate cleaning.

7.4 Location of fittings

7.4.1 Bilge valves, cocks and mud boxes are to be fitted in easily accessible spaces at, or above, the machinery space and shaft tunnel platforms. Where it is not practicable, they may be situated just below the platform, provided readily removable traps or covers are fitted, and nameplates indicate the presence of these fittings.

7.4.2 Where relief valves are fitted to pumps having sea connections, these valves are to be fitted in readily visible positions above the platform and in such an arrangement that any possible discharge will be readily visible.

7.5 Bilge pipes passing through or leading to tanks

7.5.1 In order to prevent the possibility of flooding after grounding, bilge pipes are not to be led through double bottom tanks as far as it is practicable.

Where it is unavoidable to do so, bilge pipes which have to pass through double bottom tanks shall be made of steel, having a wall thickness in accordance to Table 8.3.(1).

7.5.2 In way of deep tanks, in order to prevent any damage to them bilge pipes should be led through pipe tunnels. Where this is not practicable, bilge pipes are to be made of steel having a wall thickness in accordance to Table 8.3.(1). Furthermore, they shall have welded joints or heavy flanged joints. The number of joints is to be kept to a minimum.

7.5.3 Expansion joints, not glands, are to be fitted to these pipes, within the tanks. The pipes are to be tested, after installation, to a pressure not less than that to which the tanks can be subjected in service.

7.6 Non-return valves in bilge suctions in the holds and deep tanks used as dry cargo storage spaces

7.6.1 In order to prevent the possibility of flooding after a casualty, non-return valves are to be fitted to the open ends of bilge suction pipes in cargo holds regardless if they are inboard of B/5 or not.

7.6.2 In the case of deep tanks which may be used for the carriage of dry-cargo non-return valves are to be fitted at the open ends of bilge suctions.

SECTION 8 Additional requirements for bilge drainage and cross flooding arrangements for passenger ships

8.1 Additional requirements

8.1.1 Where practicable, the power-driven bilge pumps shall be placed in separate watertight compartments and so arranged or situated that these compartments will not be flooded by the same damage. If the main propulsion machinery, auxiliary machinery and boilers are in two or more watertight compartments, the pumps available for bilge service shall be distributed as far as possible throughout these compartments.

8.1.2 On a passenger ship of 91,5 m in length and upwards or having a criterion numeral of 30 or more, the arrangements shall be such that at least one power bilge pump shall be available for use in all flooding conditions which the ship is required to withstand. This requirement will be satisfied if:

(1) One of the required bilge pumps shall be an emergency pump of a reliable submersible type having a source of power situated above the bulkhead deck, or
(2) The pumps and their sources of power are so distributed throughout the length of the ship that at least one pump in an undamaged compartment will be available.

8.1.3 The bilge main is to be located inboard of B/5 measured at right angles to the centerline at the level of the deepest subdivision load line, where B is the breadth of the ship.

8.1.4 Where a part of the bilge suction pipe connections are situated outside the B/5 line or in the duct keel and in order to prevent flooding of the compartment containing the open end, a non-return valve is to be fitted to the pipe in that compartment. This valve is to be in addition to the remotely controlled valve used to communicate the main bilge line with the suction from the relevant compartment.

8.1.5 Where any bilge pump or its pipe connections to the bilge main is situated outboard of the B/5 line, then a non-return valve is to be provided in the piping connection at the junction with the bilge main. The emergency bilge pump and its connections to the bilge main are to be so arranged that they are situated inboard of the B/5 line.

8.1.6 Distribution boxes, cocks and valves in connection with the bilge pumping system shall be arranged so that in the event of flooding, one of the bilge pumps may be operative in any compartment. If there is only one system of pipes common to all the pumps, the necessary valves for controlling the bilge suctions must be capable of being operated from the bulkhead deck. Where in addition to the main bilge pumping unit an emergency bilge pumping system is provided, it shall be independent of the main system and so arranged that a pump is capable of operating in any compartment under flooding conditions. In that case only the valves necessary for the operation of the emergency system need be capable of being operated from above the bulkhead deck.

8.1.7 All cocks and valves referred to in 8.1.6 which can be operated from above the bulkhead deck shall have their controls at their place of operation clearly marked and shall be provided with means to indicate whether they are open or closed.

8.1.8 Where divided deep tanks or side tanks are provided with cross flooding arrangements to limit the angle of heel after side damage, the arrangements are to be self-acting where practicable. In any case, where controls to cross flooding fittings are provided, they are to be operable from above the bulkhead deck.

8.1.9

(1) General

In passenger ships, where bilge valves required to be operated from above the bulkhead deck are remotely operated by power the following provisions are applicable:

(a) Actuators and indicating mechanism are to operate satisfactorily when submerged by a head of water up to the ship's bulkhead deck.
(b) The complete installation is to be fitted within the ship's B/5 lines, and the remote control situated above the bulkhead deck.
(c) The remote control of bilge valves is to be able to override local manual control.
(d) Valve position indicators are to be provided above the bulkhead deck.

(e) Indicators are to be provided showing "valve open" and "valve closed" position for each bilge valve and in the event of damage between the valve and the control station, the last position of the valve.

(f) The indicator board is to be provided with means to show that a power supply is available. Provisions are to be made to check the functioning of the "open" and "shut" position indicator lights.

The air supply for pneumatic valve actuators should be taken from an independent source, situated above the bulkhead deck and within the ship's B/5 lines. This independent source should consist of an air receiver and air compressor capable of being operating satisfactorily in an emergency. Where the air supply for the valve actuators is also taken from the main air receivers in the machinery spaces, provision should be made to ensure that a loss of pressure in the main air receivers will not adversely affect the supply from the independent source. In electric hydraulic systems the pumping unit should be supplied from the main and emergency sources of power. The working pressure in the system should not be less than 25% greater than the pressure required to open a valve to the full open position, particularly when pneumatic opening is employed and the valve is kept closed by a spring or a similar device. Control pipes should be of steel, copper or other equivalent material, copper being preferred in pneumatic systems. Only pipe couplings of an accepted type should be used. Electric valve position indicators should take their electrical supply from both main and emergency sources of electric power.

(2) Prototype tests

Valves should be capable of operating satisfactorily when the compartments in which they are situated are flooded up to bulkhead deck level. When a spring is used to keep the valve closed, it should be non-corrodible, and sufficient to keep the valve closed when the underside of the valve lid is subjected to the head of water in the flooded compartment, whilst at the same time the upperside is subjected to the pump suction pressure, to simulate these conditions the following tests should be carried out:

(a) The valve, complete with actuator and position switch should be placed in a tank of water pressurized to at least an equivalent head of water front the tank top to the bulkhead deck, and operated not less than one hundred times to test the control and indicating systems.

(b) When a spring is employed to keep the valve closed, the underside of the valve lid should be subjected to a pressure test to determine the pressure at which the valve begins to lift. After installation a selected valve should also be tested to determine the minimum operating air or fluid pressure required to lift the spindle to the fully open position, and also the minimum pressure required to just raise the valve off its seat.

SECTION 9 Drainage arrangements in ships without propulsion machinery

9.1 Hand pumps

9.1.1 Where auxiliary power is not provided, hand pumps are to be fitted, in number and position, as may be required for the efficient drainage of the ship. Generally, one hand pump is to be provided in each compartment. Alternatively, two pumps connected to a bilge main, having at least one branch to each compartment, are to be provided. The pumps are to be capable of being worked from the upper deck or from positions above the load waterline which are at all times readily accessible.

9.1.2 Details of the pumping arrangements are to be submitted for special consideration.

9.1.3 Table 9.9.1 gives the minimum sizes of hand pumps. Where the ship is closely subdivided into small watertight compartments, 50mm bore suctions will be accepted.

9.2 Ships with auxiliary power

9.2.1 In ships in which auxiliary power is available on board, power pump suctions are to be provided for dealing with the drainage of tanks and of the bilges of the principal compartments.

9.2.2 The pumping arrangements are to be as required for self-propelled ships, so far as these requirements are applicable, duly modified to suit the size and service of the ship.

9.2.3 Details of the pumping arrangements are to be submitted for special consideration.

Table 9.9.1: Sizes of hand pumps

Tonnage under upper deck	Diameter of barrel of bucket pump (mm)	Bore of suction pipe of bucket pumps and semirotary pumps (mm)
Not exceeding 500 tons	100	50
Above 500 tons but not exceeding 1000 tons	115	57
Above 1000 tons but not exceeding 2000 tons	125	65
Above 2000 tons	140	70

SECTION 10 Vent, overflow and sounding pipes

10.1 General requirements

10.1.1 Vent, overflow and sounding pipes are to be made of steel or other approved material.

However, the lengths of pipes fitted above the weather deck are to be of steel or equivalent material.

10.1.2 Nameplates are to be affixed to the upper ends of all vent and sounding pipes.

10.1.3 All vent and overflow pipes on the open deck are to terminate by way of 180°C bends or equivalent, equipped with non-return devices.

10.1.4 All vent, sounding and overflow pipes passing through cargo holds are to be protected against damage.

10.1.5 Wherever in this Section cargo oil is mentioned refers to cargo oil with a flash point of 60°C or above (closed cup test).

10.2 Vent pipes -general arrangement

10.2.1 Vent pipes are to be fitted to all tanks, cofferdams, void spaces, tunnels and other compartments which are not fitted with alternative ventilation arrangements.

10.2.2 The structural arrangement in double bottom and all other tanks is to be such as to permit the free passage of air and gases from all parts of the tanks to the vent pipes. The vent pipes are to be fitted at the opposite end of the tank to that which the filling pipes are placed and/or at the highest part of tank, laid to the vertical direction.

10.3 Termination of vent pipes

10.3.1 Vent pipes to double bottom tanks, deep tanks extending to the shell plating or tanks which can be run up by the sea are to be led above the bulkhead deck. Furthermore, when the tanks contain fuel oil or cargo oil or when they are cofferdams or generally they are tanks which can be pumped up, vent pipes are to be led to the open.

10.3.2 Vent pipes of lubricating oil tanks may terminate at clearly visible positions in the machinery space in such locations as to preclude the possibility of overflowing oil on electrical equipment or high-temperature piping. Where these tanks form part of the ship's shell, they are to terminate in the engine room casing above the bulkhead deck.

10.4 Height of vent pipes

10.4.1 The height of vent pipes from the upper surface of decks exposed to the weather, to the point where water may have access below is normally to be not less than:

- 760mm, at Position 1,
- 450mm, at Position 2,

these heights being measured above deck sheathing, where fitted. The above mentioned Positions are corresponding to the positions of hatches on weather decks, as the International Convention on Load Lines, 1966, defines:

- Position 1: exposed freeboard and raised quarterdecks and exposed superstructure decks within the forward 0,25L.
- Position 2: exposed superstructure decks abaft the forward 0,25L.

10.4.2 Lower heights may be approved in cases where these reduced heights are essential for the operation of the ship, provided that the design and arrangements are otherwise satisfactory. In such cases, efficient, permanently attached closing appliances of an approved automatic type will generally be required.

10.4.3 The height of vent pipes may be required to be increased on ships of Type "A", Type "B-100" and Type "B-60" where this is shown to be necessary by the float ability calculations required by the International Convention on Load Lines, 1966. An increase in height may also be required or recommended by individual Administrations when vent pipes to fuel oil and settling tanks are situated in positions where sea water could be temporarily entrapped, e.g. in recesses in the sides and ends of superstructures or deckhouses, between hatch ends, behind high sections of bulwark, etc.

10.4.4 Vent pipes are generally to be led to an exposed deck, but alternative arrangements will be considered in the case of ferries where such an arrangement is not practicable.

10.4.5 The minimum wall thickness of vent pipes in positions indicated in 10.4.1 is to be:

- 6,0mm for pipes of 80mm external diameter or smaller 8,5mm for pipes of 165mm external diameter or greater Intermediate minimum thicknesses are to be determined by linear interpolation.

10.5 Vent pipes accessories

10.5.1 All openings of vent pipes are to be provided with permanently attached, satisfactory means of closing to prevent the free entry of water.

10.5.2 The closing appliances fitted to tank vent pipes are to be of an automatic opening type which will allow the free passage of air to prevent excessive pressure or vacuum coming on the tanks. The air pipe closing devices should comply with the requirements of 10.14.

10.5.3 Vent outlets from fuel oil tanks are to be fitted with a flame arrestor of an approved type. A flame arrest may consist of single screen of corrosion resistant wire of at least 12x12 mesh (wires per square cm), or two screens of at least 8x8 mesh, spaced not less than 12,5mm and not more than 38mm apart, or equivalent means. The above screen where necessary are to be supported and protected by a mechanically stronger wire screen of non-corrosive material of approximately 1x1 mesh. The clear area of the flame arresting device is to be not less than the cross-sectional area required for the vent pipe, and the outlets of flammable vapours are not to be led in the vicinity of areas where the possibility of ignition exists.

10.5.4 Wood plugs and other devices which can be secured closed are not to be fitted at the outlets.

10.5.5 In a ship to which timber freeboards are assigned, vent pipes which will be inaccessible when the deck cargo is carried are to be provided with automatic closing appliances.

10.6 Size of vent pipes

10.6.1 Where tanks are to be filled by pump pressure either by the ship's pumps or by shore pumps, the aggregate area of the vents in the tank is to be at least 125% of the effective area of the tank filling line. However, vent pipes are not to be less than 50mm internal diameter. Notwithstanding the above, the pump capacity and pressure head are to be considered in the sizing of vent pipes because a vent pipe can serve also as an overflow pipe.

10.6.2 Where tanks are fitted with cross flooding connections, the vent pipes are to be of adequate area for these connections.

10.7 Overflow pipes

10.7.1 It is essential for all tanks which can be pumped up to be fitted with overflow pipes in order to prevent a pressure in the tank greater than the design pressure of the tank.

10.7.2 The overflow pipes fitted to fuel oil, cargo oil and lubricating oil tanks are to be led to a corresponding overflow tank of adequate capacity or to a storage tank having a space reserved for overflow purposes. A sight glass is to be provided in the overflow pipe to indicate when the tanks are overflowing or, alternatively, an alarm device is to be provided to give warning either when the tanks are overflowing or when the contents reach a predetermined level in the tanks.

10.7.3 Tanks other than fuel oil, cargo oil and lubricating oil tanks are to be led to the open or to suitable overflow tanks arranged as in 10.7.2. Overflow pipes led to the open are to be furnished with closing appliances of an automatic opening type to prevent excessive pressure coming on the tanks.

10.8 Size of overflow pipes

10.8.1 The pump(s) capacity and pressure head, the type of the closing appliance, the height of the overflow pipe are to be taken into consideration when sizing overflow pipes in order to prevent an excessive pressure of that for which the tank has been constructed from coming on any part of the tank when it is being pumped up and is overflowing. Indicatively, the overflow pipes are to be at least 125% of the effective area of the filling line and the vents need not exceed their minimum sizes.

10.8.2 Where it is impracticable to comply with 10.8.1 or in addition to, a notice board is to be displayed at the pumping control station(s) advising the operator to reduce the pumping rate to a safe level before the tanks are full. Alternatively, the tanks are to be fitted with high level alarms and arrangements for automatically closing the tank filling valves before the tanks are liquid full.

10.9 Vent and overflow systems

10.9.1 Where a pipe is used for both vent and overflow purposes the provisions of 10.5, 10.6, 10.7 and 10.8 are to be complied with.

10.9.2 Vent and overflow pipes are to be arranged to be self-draining under normal conditions of trim.

10.9.3 The arrangement of the vent and/or overflow system is to be such that in the event of any one of the tanks being bilged, tanks situated in other watertight compartments of the ship cannot be flooded from the sea through combined vent pipes or the overflow main. In the case of trawlers and fishing vessels, the arrangement is to be such that in the event of any of the tanks being bilged, the other tanks cannot be flooded from the sea through the combined vent pipes or the overflow main.

10.9.4 Where overflows from tanks which are used for the alternative carriage of oil and water ballast are connected to an overflow system, arrangements are to be made to prevent water ballast overflowing into tanks containing oil.

10.10 Sounding pipes-general arrangement

10.10.1 Sounding pipes are to be provided for all tanks, cofferdams and void spaces with bilge connections and generally for the bilges of those compartments which are not at all times readily accessible.

10.10.2 It is at the discretion of LHR to allow not to install sounding pipes in bilge wells in permanently accessible spaces. Furthermore, when the tanks are furnished with remote level indicators type-approved by LHR, the arrangement of the sounding pipes can be dispensed with.

10.10.3 When means other than a sounding pipe is used to indicate the level of liquid in tanks containing fuel oil, lubricating oil or other flammable liquid, care is to be taken to prevent any leakage of the tank contents because of a failure of such means or overfilling of the tank.

10.10.4 Sounding pipes are to be located as near the suction pipes as practicable, and are to be led as straight as possible from the lowest part of the tank. If curved to suit the structure of the ship, the curvature must be sufficiently easy to permit the ready passage of the sounding rod or chain.

10.10.5 Tanks independent of the hull structure or integral tanks located above the deepest load water line, except those containing liquids having a flash point of 60°C or less, may be fitted with suitable gauge glasses, provided they are fitted with a valve at each end of the gauge and adequately protected from mechanical damage. Gauge glasses on tanks containing fuel oil, lubricating oil or other flammable liquid are to be of the flat type of heat-resisting quality, fitted with self-closing valves at each end.

10.10.6 In passenger ships, sounding devices for fuel oil tanks, lubricating oil tanks and other tanks which may contain flammable liquids are to be of a type which does not require penetration below the top of the tank.

10.11 Sounding pipes - termination

10.11.1 Sounding pipes are normally to be terminated above the bulkhead deck at positions accessible at all times. The sounding pipes of fuel oil, cargo oil and lubricating oil tanks pipes are to be led to safe positions to the open deck and their openings must be fitted with watertight closures. Where such arrangements, in the case of ferries with enclosed tween deck space on the main vehicle deck, are not practicable, alternative arrangements will be specially considered.

10.11.2 Where it is impracticable in machinery spaces and tunnels to extend the sounding pipes as prescribed in 10.10.1, short sounding pipes leading from double bottom tanks may extend to readily accessible positions above the platform. Installation of short sounding pipes from tanks other than double bottom tanks terminating in the machinery space is to be specially approved by LHR.

10.11.3 In the case of fuel oil, cargo oil and lubricating oil tanks short sounding pipes are to be fitted with self-closing cocks having parallel plugs. Such sounding pipes are to be located in positions as far as possible from any heated surface or electrical equipment and, where necessary, effective shielding is to be provided in way of such surfaces and/or equipment.

10.11.4 Short sounding pipes to tanks other than those with contents mentioned in 10.11.3, are to be fitted with shut-off cocks or with screw cap secured to the pipe by chains.

10.11.5 In passenger ships, short sounding pipes are permissible for sounding cofferdams and double bottom tanks situated in the machinery space and in all cases are to be fitted with self-closing cocks as described in 10.11.4.

10.12 Sizes of sounding pipes

10.12.1 Sounding pipes are not to be less than 32mm in internal diameter and not less than 4,5mm in thickness.

10.12.2 All sounding pipes, whether for compartments or tanks, which pass through refrigerated spaces or the insulation thereof, in which the temperatures are 0°C or below, are not to be less than 65mm internal diameter.

10.13 Striking plates

10.13.1 Striking plates of adequate thickness and size are to be fitted under open ended sounding pipes.

10.13.2 Where slotted sounding pipes having closed ends are used, the closing plugs are to be of substantial construction.

10.14 Air Pipe Closing Devices (IACS UR P3 Rev.5 (2021))

10.14.1 General requirements

Where air pipes are required by the Rules of LHR, the International Convention on Load Lines, 1966 or the Protocol of 1988 relating to the International Convention on Load Lines, 1966, as amended by IMO resolutions up to MSC.375(93) to be fitted with automatic closing devices, they are to comply with the following:

10.14.2 Design

10.14.2.1 Air pipe automatic closing devices are to be so designed that they will withstand both ambient and working conditions, and be suitable for use at inclinations up to and including ±40°.

10.14.2.2 Air pipe automatic closing devices are to be constructed to allow inspection of the closure and the inside of the casing as well as changing the seals.

10.14.2.3 Efficient ball or float seating arrangements are to be provided for the closures. Bars, cage or other devices are to be provided to prevent the ball or float from contacting the inner chamber in its normal state and made in such a way that the ball or float is not damaged when subjected to water impact due to a tank being overfilled.

10.14.2.4 Air pipe automatic closing devices are to be self-draining.

10.14.2.5 The clear area through an air pipe closing device in the open position is to be at least equal to the area of the inlet.

10.14.2.6 An automatic closing device is to:

- Prevent the free entry of water into the tanks,
- Allow the passage of air or liquid to prevent excessive pressure or vacuum coming on the tank

10.14.2.7 In the case of air pipe closing devices of the float type, suitable guides are to be provided to ensure unobstructed operation under all working conditions of heel and trim as specified in 10.14.2.1.

10.14.2.8 The maximum allowable tolerances for wall thickness of floats should not exceed ± 10% of thickness.

10.14.2.9 The inner and the outer chambers of an automatic air pipe head is to be of a minimum thickness of 6 mm. Where side covers are provided and their function is integral to providing functions of the closing device as outlined in 10.14.2.6, they shall have a minimum wall thickness of 6 mm. If the air pipe head can meet the tightness test in 10.14.4.1(b) without the side covers attached, then the side covers are not considered to be integral to the closing device, in which case a wall less than 6 mm can be acceptable for side covers.

10.14.3 Materials

10.14.3.1 Casings of air pipe closing devices are to be of approved metallic materials adequately protected against corrosion.

10.14.3.2 For galvanised steel air pipe heads, the zinc coating is to be applied by the hot method and the thickness is to be 70 to 100 microns.

10.14.3.3 For areas of the head susceptible to erosion (e.g. those parts directly subjected to ballast water impact when the tank is being pressed up, for example the inner chamber area above the air pipe, plus an overlap of 10° or more either side) an additional harder coating should be applied. This is to be an aluminium bearing epoxy, or other equivalent, coating, applied over the zinc.

10.14.3.4 Closures and seats made of non-metallic materials are to be compatible with the media intended to be carried in the tank and to seawater and suitable for operating at ambient temperatures between -25°C and 85°C.

10.14.4 Type Testing

10.14.4.1 Testing of Air Pipe Automatic Closing Devices

Each type and size of air pipe automatic closing device shall be surveyed and type tested at the manufacturer's works or other acceptable location according to the LHR's practice. The minimum test requirements for an air pipe automatic closing device are to include the following:

(a) Determination of the Flow Characteristics. The flow characteristics of the air pipe closing device are to be determined. Measuring of the pressure drop versus rate of volume flow is to be carried out using water and with any intended flame or insect screens in place.
(b) Tightness test during immersion/emerging in water. An automatic closing device is to be subjected to a series of tightness tests involving not less than two (2) immersion cycles under each of the following conditions:
    (i) The automatic closing device is to be submerged slightly below the water surface at a velocity of approximately 4 m/min. and then returned to the original position immediately. The quantity of leakage is to be recorded.

(ii) The automatic closing device is to be submerged to a point slightly below the surface of the water. The submerging velocity is to be approximately 8 m/min and the air pipe vent head is to remain submerged for not less than 5 minutes. The quantity of leakage shall be recorded.

(iii) Each of the above tightness tests shall be carried out in the normal position as well as at an inclination of 40 degrees under the strictest conditions for the device. In cases where such strictest conditions are not clear, tests shall be carried out at an inclination of 40 degrees with the device opening facing in three different directions: upward, downward, sideways (left or right). (See Figure 9.10.1 to 9.10.4).

The maximum allowable leakage per cycle shall not exceed 2 ml/mm of nominal diameter of inlet pipe during any individual test.

Figure 9.10.1: Example of normal position

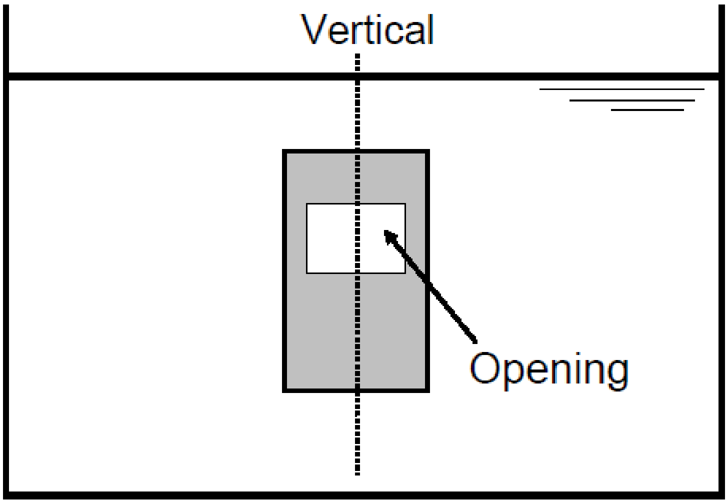


Figure 9.10.2: Example of inclination 40 degrees opening facing upward

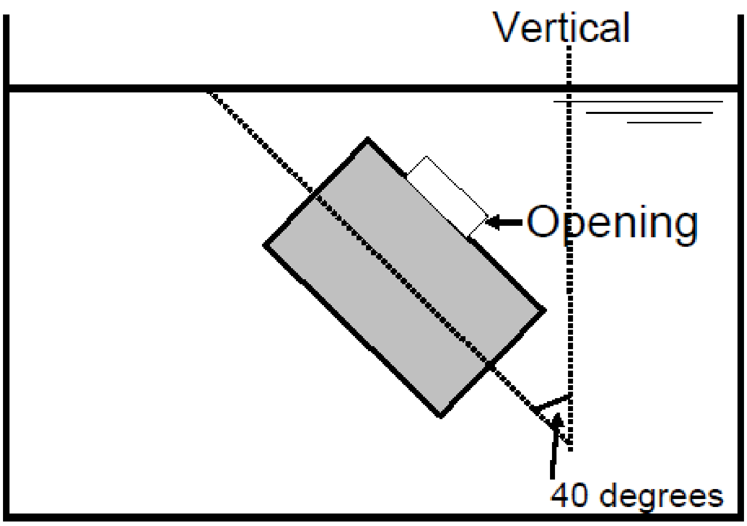


Figure 9.10.3: Example of inclination 40 degrees opening facing downward

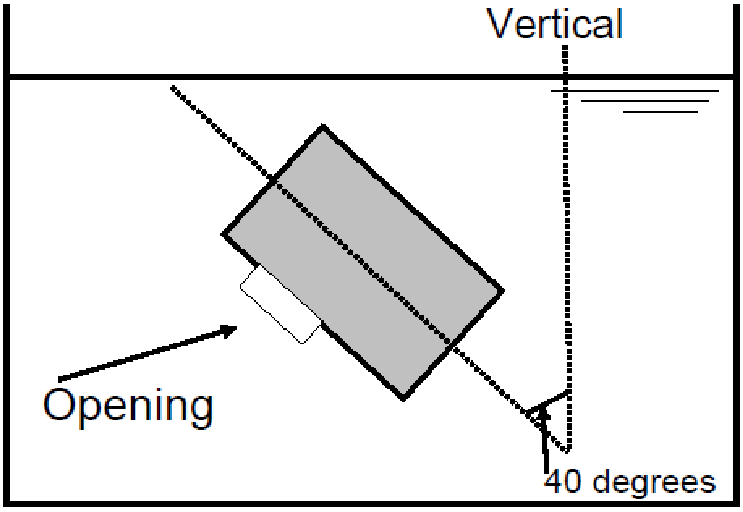
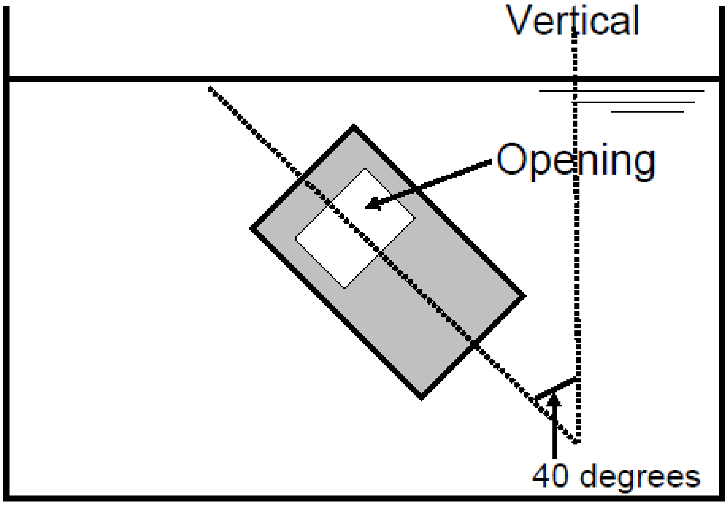


Figure 9.10.4: Example of inclination 40 degrees opening facing sideways



(c) Discharge / Reverse flow test. The air pipe head shall allow the passage of air to prevent excessive vacuum developing in the tank.

(i) Reverse flow test

1. A re verse flow test shall be performed. A vacuum pump or another suitable device shall be connected to the opening of the air pipe leading to the tank. The flow velocity shall be applied gradually at a constant rate until the float gets sucked and blocks the flow.
2. The velocity at the point of blocking shall be recorded. 80% of the value recorded will be stated in the certificate.

(ii) Alternative to the reverse flow test

1. For pipe heads of 400 mm nominal diameter and above, as an alternative to the reverse flow test, a numerical simulation test based on computational fluid dynamics (CFD), to be carried out in conjunction with limited representative testing to establish the validity of the CFD modelling and results, may be accepted.
2. CFD predictions for air pipe heads can be validated against the available actual reverse flow test results of same size and type of air pipe heads.

3. The accuracy of the CFD modelling and the major assumptions used for the calculation are to be documented.
4. Mesh convergence studies are to be carried out and documented.
5. The requirement as per the preceding (i) 2. applies.

10.14.4.2 Testing of non-metallic Floats

Impact and compression loading tests shall be carried out on the floats before and after pre-conditioning as shown in Table 9.10.5:

Table 9.10.5:

Test condition	**Test temperature °C**		
	-25°C	**20°C**	**85°C**
Dry	+	+	+
After immerging in water	+	+	+
After immerging in fuel oil	-	+	-
Immerging in water and fuel oil is to be for at least 48 hours			

(a) Impact Test

The test may be conducted on a pendulum type testing machine. The floats shall be subjected to 5 impacts of 2.5 Nm each and shall not suffer permanent deformation, cracking or surface deterioration at this impact loading. Subsequently the floats shall be subjected to 5 impacts of 25 Nm each. At this impact energy level some localized surface damage at the impact point may occur. No permanent deformation or cracking of the floats shall appear.

(b) Compression Loading Test

Compression tests shall be conducted with the floats mounted on a supporting ring of a diameter and bearing area corresponding to those of the float seating with which it is intended that float shall be used. For ball type float, loads shall be applied through a concave cap of the same internal radius as the test float and bearing on an area of the same diameter as the seating. For a disc type float, loads are to be applied through a disc of equal diameter as the float. A load of 350 kg shall be applied over one minute and maintained for 60 minutes. The deflection shall be measured at intervals of 10 minutes after attachment of the full load. The record of deflection against time is to show no continuing increase in deflection and, after release of the load, there shall be no permanent deflection.

10.14.4.3 Testing of Metallic Floats

Tests shall be conducted in accordance with 10.14.4.2(a). The tests shall be carried out at room temperature and in the dry condition.

SECTION 11 Ballast systems

11.1 Arrangement of ballast piping

11.1.1 Because of the importance of the ballast piping system to the safety of the ship, suctions in ballast water tanks are to be so arranged that the tanks can be emptied despite unfavorable conditions of trim and list.

11.1.2 Ballast water pipes may not pass-through drinking water, feed water or lubricating oil tanks.

11.1.3 With regard to the installation on ships of oily water separators, filter plants, oil collecting tanks, oil discharge lines and a monitoring and control system or a 15 ppm alarm device in the water outlet of oily water separators, compliance is required with the provisions of the International Convention for the Prevention of Pollution from Ships, 1973 (MARPOL), and the Protocol of 1978.

11.1.4 Ballast piping which runs outboard of B/5 is to have the open ends of the pipes in each compartment controlled by valves capable of being operated from accessible positions above the bulkhead deck. These valves are to effectively isolate the compartment.

11.1.5 The requirements of 3.6.3 through 3.6.5 are applicable.

11.1.6 Where ballast water tanks may be used exceptionally as dry cargo holds, such tanks are also to be connected to the bilge system. The requirements specified in 3.5 are applicable.

11.1.7 The ballast arrangement is to be subjected to operational testing under LHR's supervision.

11.2 Ballast tank valves

11.2.1 Ballast tank valves are to be arranged so that they will remain closed at all times except when ballasting. For this purpose manual screw thread operated valves or positive holding arrangements for butterfly type valves or other approved arrangement may be accepted. Where installed, remote controlled valves are to be arranged so that they will close and remain closed upon loss of control power. Remote control of ballast valves is to be clearly marked at the control station and means are to be provided to indicated whether the valve is closed or open.

11.3 Ballast pumps

11.3.1 The number and capacity of the pumps must satisfy the operational requirements of the ship.

11.4 Additional requirements for passenger ships

11.4.1 As far as possible, anti-heeling arrangements for equalizing unsymmetrical flooding should operate automatically. Where the arrangement does not operate automatically, any shutoff valves and other devices must be capable of being operated from above the bulkhead deck.

11.4.2 The cross-flooding arrangements must ensure that in case of flooding, equalization is achieved within 15 minutes. Separation of the tanks is to be ensured during normal operation.

11.4.3 Plans for anti-heeling arrangements, intended to equalize unsymmetrical flooding, approved by the National Authority, should be submitted to LHR for information.

SECTION 12 Drinking water systems

12.1 Drinking water connections

12.1.1 Filling connections are not to be fitted to vent pipes.

12.1.2 Sounding pipes must terminate sufficiently high above deck.

12.1.3 Vent and overflow pipes are to be extended above the open deck. The upper openings of vent pipes are to be fitted with automatic closing devices and are to be protected against the entry of the insects.

12.2 Drinking water pipe lines

12.2.1 Drinking water pipe lines are not to be connected to pipe lines carrying other media.

12.2.2 Drinking water pipe lines are not to be laid through tanks which do not contain drinking water.

12.2.3 Drinking water supply to tanks which do not contain drinking water (e.g. expansion tanks of the fresh water cooling system) is to be made by means preventing back flow.

12.3 Drinking water pumps

12.3.1 Separate drinking water pumps are to be provided for drinking water systems.

12.3.2 The pressure lines of the pumps of drinking water pressure tanks are to be fitted with screw down non-return valves.

12.4 Drinking water generation

12.4.1 Where the distillate produced by the ship's own evaporator unit is used for the drinking water supply, the treatment of the distillate has to comply with the requirements of National Health Authorities.

SECTION 13 Scuppers and sanitary systems

13.1 General

13.1.1 All decks are to be provided with scuppers, sufficient in number and size for effective drainage.

13.1.2 Scuppers draining weather decks and spaces within superstuctures or deckhouses not fitted with efficient weathertight doors are to be led overboard.

13.1.3 Scuppers and discharges which drain spaces below the freeboard deck, or spaces within intact superstructures or deckhouses on the freeboard deck fitted with efficient weathertight doors, may be led to bilges in the case of scuppers, or to suitable sanitary tanks in the case of sanitary discharges.

Alternatively, they may be led overboard provided that the spaces drained above the load waterline, and the pipes are fitted with efficient and accessible means of preventing water from passing inboard.

13.1.4 The drainage arrangements of enclosed tweendeck spaces in ships with small freeboards are to be specially considered.

13.1.5 In the enclosed tweendeck space on the main vehicle deck of a ferry scuppers are to be provided port and starboard spaced not more than 20m apart. On each side of the ship, the total area of all drainage openings in the vehicle deck is not to be less than the greater of 0,00012 times the vehicle deck area and 0,075m^2. The capacity of scuppers may be required to be increased, depending on the discharge rate of the drencher system, where provided. The perforated area of above drainage openings is to have a total opening area not less than 4 times that of the drainage pipe. It is recommended that the drainage inlets are to made rectangular with the longer side in the longitudinal direction of the ship and to have mud boxes. The area of the opening above any partition plate in the mud box is not to be less than twice the opening area of the drainage pipe.

13.1.6 The minimum wall thickness of pipes not indicated in 13.2.5 is to be:

– 4,5mm for pipes of 155mm external diameter or smaller
– 6,0mm for pipes of 230mm external diameter or greater.

Intermediate minimum thicknesses are to be determined by linear interpolation.

13.1.7 Scuppers and discharge pipes should not normally pass-through fuel oil or cargo oil tanks. Where scuppers and discharge pipes pass, unavoidably, through fuel oil or cargo oil tanks, and are led through the shell within the tanks, the thickness of the piping should be at least the same thickness as the rule shell plating in way, derived from the appropriate Chapters, but need not exceed 19mm.

13.1.8 If a tween deck space on the main vehicle deck of a ferry is not totally enclosed, scuppers or freeing ports are to be provided consistent with the requirements of Part 3, Chapter 16.

13.1.9 Piping within tanks is to be tested.

13.1.10 All piping is to be adequately supported.

13.2 Closing appliances

13.2.1 Generally, each separate overboard discharge is to be fitted with a screw-down non-return valve capable of being operated from a position always accessible and above the freeboard deck. An indicator is to be fitted at the control position showing whether the valve is open or closed.

13.2.2 In an enclosed tweendeck space on the main vehicle deck of a ferry, such controls are to be operable from spaces above the main vehicle deck and to be protected from mechanical damage.

13.2.3 Where the vertical distance from the summer load waterline to the inboard end of the discharge pipe exceeds the 1% of the waterline length of the ship, the discharge may be fitted with two automatic non-return valves without positive means of closing, instead of the screw-down non-return valve, provided that the inboard valve is always accessible for examination under service conditions.

13.2.4 Where the vertical distance from the summer load waterline to the inboard end of the discharge pipe exceeds the 2% of the waterline length of the ship, a single automatic non-return valve without positive means of closing may be fitted.

13.2.5 Scuppers and discharge pipes originating at any level which penetrate the shell either more than 450mm below the freeboard deck or less than 600mm above the summer load waterline, are to be fitted with an automatic non-return valve at the shell. This valve, unless required by 13.1.3, may be omitted provided the piping has a minimum wall thickness of: 7,0mm for pipes of 80mm external diameter or smaller 10,0mm for pipes of 180mm external diameter 12,5mm for pipes of 220mm external diameter or greater Intermediate minimum thicknesses are to be determined by linear interpolation. Unless required by 13.1.7, the maximum thickness need not exceed 12,5mm.

13.2.6 If in association with 13.1.8, a valve is required by 13.1.3, this valve should preferably be fitted as close as possible to the point of entry of the pipe into the tank. If fitted below the freeboard deck, the valve is to be capable of being controlled from an easily accessible position above the freeboard deck. Local control is also to be arranged, unless the valve is inaccessible. An indicator is to be fitted at the control position showing whether the valve is open or closed.

13.2.7 In a ship to which timber freeboard is assigned, the summer load waterline is to be regarded as that corresponding to the timber summer freeboard.

13.3 General arrangement of sanitary systems

13.3.1 Sanitary discharge pipes located in cargo holds are to be specially protected.

13.3.2 The discharge lines of the sewage pumps are to be fitted with a storm valve and a gate valve which is to be fitted in the discharge line directly at the ship side. A second means of preventing backflow is to be arranged either in the suction or in the pressure line of sewage pumps.

13.4 Sewage tanks

13.4.1 Vent pipes are to be extended above the open deck and are to be fitted with automatic closing devices.

13.4.2 A flushing connection is to be provided.

13.4.3 If, when the tank is full, sewage can flow out through the sanitary arrangements connected, the tank is to be fitted with a level alarm.

13.4.4 Ballast and bilge pumps may not be used for emptying sewage tanks.

SECTION 14 Piping systems for oil tankers

14.1 Scope

14.1.1 The requirements of this Section apply to ships intended to carry oil in bulk and they are additional to the ones of previous sections. It is assumed that the ships are of the usual type where the main propelling machinery is located aft. Other arrangements will be specially considered.

14.1.2 The requirements of this Section are mainly to apply to ships intended to carry flammable liquids having a flash point not exceeding 60°C (closed cup test). In cases of carriage of non-flammable liquids or cargoes having a flash point exceeding 60°C, the requirements will be altered to take account of this fact.

14.2 Documents for approval

14.2.1 In addition to the documents, plans or specifications mentioned in SECTION 1, the following plans are to be submitted for approval:

(1) Cargo piping arrangements
(2) Pumping arrangements at the fore and aft ends as well as drainage arrangements for cofferdams and pump rooms.
(3) Venting arrangements for cargo tanks
(4) Venting arrangements of cargo and/or ballast pump rooms and other enclosed spaces where cargo handling equipment is located.

14.3 Definition of "Dangerous Spaces"

14.3.1 Cargo tanks, slop tanks, pump rooms, cofferdams or other spaces which could possibly contain petroleum or explosive vapours, or spaces immediately adjacent to cargo oil or slop tanks are defined as dangerous spaces and by no way any equipment which could be a source of ignition is to be located within them. The temperature of steam, or other fluid, in pipes (or heating coils) in these spaces is not to exceed 220°C.

14.3.2 For principles governing the definition of dangerous zones or spaces and requirements for electrical equipment within such spaces, see Part 6.

14.4 Design of piping systems

14.4.1 Materials

The materials used in piping and cargo pumping systems or any other part to come in contact with the specified cargoes, are to be suitable for these cargoes. Where applicable, the material requirements set forth in Part 5, Chapter 8 are to be fulfilled.

14.4.2 Design

All piping systems and their associate valve and fittings are to be able to withstand the maximum pressure to which the system can be subjected. Generally, all the piping systems are to comply with the design requirements of Part 5, Chapter 8.

14.4.3 Piping passing through dangerous zones (IACS UR F15 Cor.1 (2022))

(1) Ballast piping passing through cargo tanks and cargo oil pipes passing through segregated ballast tanks, as permitted by Regulation 19.3.6 of MARPOL Annex I, are to comply with the following requirements:
   - The pipes are to be of heavy gauge steel of minimum wall thickness according to Table 9.14.1 with welded or heavy flanged joints the number of which is to be kept to a minimum. Expansion bends only (not glands) are permitted in these lines within cargo tanks for serving the ballast tanks and within the ballast tanks for serving the cargo tanks.

Table 9.14.1:

Nominal diameter (mm)	Minimum wall thickness (mm)
50	6,3
100	8,6
125	9,5
150	11,0
200 and above	12,5

(2) The thicknesses shown in Table 9.14.1 refer to carbon steel.

(3) Connection between cargo piping and ballast piping referred to above is not permitted except for emergency discharge as specified in the Unified Interpretation to Regulation 1.18 of MARPOL Annex I. Nevertheless, provision may be made for emergency discharge of the segregated ballast by means of a connection to a cargo pump through a portable spool piece. In this case non-return valves should be fitted on the segregated ballast connections to prevent the passage of oil to the ballast tanks. The portable spool piece should be mounted in a conspicuous position in the pump

room and a permanent notice restricting its use should be prominently displayed adjacent to it. Shut-off valves shall be provided to shut off the cargo and ballast lines before the spool piece is removed.

(4) The ballast pump is to be located in the cargo pump room, or a similar space within the cargo area not containing any source of ignition.

14.5 Cargo pump room

14.5.1 The cargo pump rooms are to be well separated from the machinery space having no direct communication with it. They must be easily accessible from the open deck. The cargo pump rooms are to be located within or adjacent the cargo tank area.

14.5.2 Efficient ventilation is to be provided in cargo pump rooms and other closed spaces containing cargo handling equipment and requiring easy access during ship's operation. The ventilation is to be of the mechanical extraction type.

14.5.3 The ventilation system is to be capable of being operated from outside the compartment concerned. Adjacent to the entrance of the compartment a notice is to be fixed stating that:

"NO PERSON IS TO ENTER INSIDE UNLESS THE VENTILATION IS OPENED FOR AT LEAST 15 MINUTES".

14.5.4 The design of the ventilation system is to be based on 20 air changes per hour for the gross volume of the pump room or space.

14.5.5 The suctions of main ventilating ducts are to be located close to the pump room bilges, immediately above the transverse floor plates or bottom longitudinals. For emergency purposes an additional intake is to be fitted at a height of 2 m above the pump room lower platform. This intake is to be fitted with a damper capable of being operated from the open deck and the lower platform as well. The design of ventilating system is to be such that at least the minimum required number of air changes is to be circulated through the lower inlets. On the other hand, when the lower inlets are not in use because of flooding of the bilges, then at least 75% of the required number of air changes is to be obtainable through the upper suctions. The gases are to be directed through the lower platform to the duct intakes.

14.5.6 The outlets of ventilating ducts are to be located at a distance at least 3 m above the deck, while the same distance is to be maintained from the nearest air inlets or openings to accommodation areas or other enclosed spaces, as well as from possible sources of ignition.

14.5.7 On the outlets of ventilating ducts protection screens are to be fitted of not more than 13mm, square mesh. Ventilating duct inlets from the open are to be located at an adequate distance from ventilating discharges to avoid the recycling of hazardous vapours.

14.6 Fans

14.6.1 Materials

The material combinations between impeller and the housing adjacent to the impeller are to be such as to provide safe operation, with no danger of sparking. The following combinations are permissible:

(1) impellers and housings of non-ferrous metals,
(2) impellers and housings of austenitic stainless steels,
(3) impellers and/or housings of non-metallic material, taking care of the elimination of static electricity,
(4) impellers of aluminium alloys or magnesium alloys and a ferrous housing provided that a ring of suitable thickness of non-ferrous material is fitted in way of the impeller,

(5) any combination of impeller and housing materials where it is established that it is spark proof,
(6) any combination of ferrous impellers and housings with not less than 13mm tip clearance.

14.6.2 The following combinations are not permissible, considered as "not spark proof":

(1) impellers of an aluminium alloy or magnesium alloy and a ferrous housing, irrespective of tip clearance
(2) housing made of an aluminium alloy or a magnesium alloy and a ferrous impeller, regardless of tip clearance
(3) any combination of ferrous impeller and housing with less than 13mm design tip clearance

14.6.3 Antistatic materials or other effective means are to be used in the impeller body as well as in the housing to prevent electrostatic charges on the surface of the material.

14.6.4 Miscellaneous. Protection screens of not more than 13mm square mesh are to be fitted in the inlet and outlet ventilation openings on the open deck to prevent the entry of objects into the fan housing.

14.6.5 Type tests of the complete fan are to be carried out to the Surveyor's satisfaction.

14.6.6 The ventilation units are to be installed in such a way as to ensure safe bonding to the hull.

14.7 Slop tanks

14.7.1 Slop tanks are provided in order to collect oil residues until the ship discharges them at an appropriate port.

14.7.2 Slop tanks are to be provided with an approved independent venting system. Efficient ventilation is to be provided for spaces adjacent to slop tanks.

14.7.3 At least two portable instruments are to be available on board for gas detection.

14.7.4 To satisfy the requirements of certain National and/or Terminal Authorities, it may be necessary to provide an inert gas system for blanking the slop tanks.

14.7.5 Pump room piping to the slop tanks is to be efficiently isolated when not in use. A valve followed by a spectacle flange or a spool piece with blank flanges are efficient means. This arrangement is to be located adjacent to the slop tanks, but where this is impracticable it may be located within the pump room directly after the piping penetrates the bulkhead. Separate pumping and piping arrangements are to be used to discharge the contents of the slop tanks directly over the open deck when the ship is in the dry cargo mode.

14.7.6 Warning notices are to be installed at suitable positions detailing precautions to be observed prior to the ship loading or unloading, or when the ship is carrying dry cargo with liquid in the slop tanks.

14.8 Steam connections to cargo tanks

14.8.1 Steaming out and/or fire-extinguishing lines connections provided in cargo tanks or cargo pipe lines are to be fitted with screw-down non-return valves. The main supply to these connections is to be fitted with a master valve placed in a readily accessible position clear of the cargo tanks.

14.9 Bilge, ballast and fuel oil piping systems

14.9.1 Pumping arrangements at the ends of ship. Bilge, ballast and fuel oil lines in the machinery space and at the forward end of the ship are to follow the requirements for general cargo ships as well as the requirements of this subsection.

14.9.2 Bilge, ballast and fuel oil lines connected to pumps, tanks or compartments at the ends of the ship are not to pass through cargo tanks or have any part of them inside cargo tanks, but may be led through clean ballast tanks or void spaces within the region of the cargo tanks.

14.9.3 The bunkering system for fuel oil is to be entirely separate from the cargo piping system.

14.10 Cargo pump room drainage

14.10.1 The cargo pump room is to have its own bilge suctions separate from the machinery space ones. The bilge drainage of the cargo pump rooms is to be carried out by pump or bilge ejectors. When the bilge drainage is serviced by the cargo pump or cargo stripping pumps, this is permitted provided that the bilge suctions are fitted with screw-down non-return valves and, additionally, an isolating valve or cock is to be fitted on the pump connection to the bilge chest.

14.11 Cofferdam drainage

14.11.1 The cofferdams, located at the fore or aft ends of cargo spaces, are to be provided with appropriate drainage arrangements corresponding to the manner they are used:

(1) where the cofferdams can be filled with water ballast, the drainage of the aft one may be carried out by a ballast pump located in the main engine room. Where there is a ballast pump in the forward pump room this may be used to drain the fore cofferdam. In all cases the suctions are to be led directly to the pump and not to connect with other pipes.

(2) in the case the cofferdams are used as dry compartments: the aft cofferdams, adjacent to the pump room, may be drained by a cargo pump, provided that the bilge suctions of that pump are fitted with screw-down non-return valves and, additionally, an isolating device is to be fitted on the pump connection to the bilge chest. The fore cofferdams may be drained by a bilge and ballast pump in a forward pump room. Cofferdams may be drained by bilge ejectors or, in the case of small ships, by hand pumps.

Where there is no suitable power pump to be provided at the forward end of the ship, a clean ballast pump in the cargo pump room may be used to deal with the forward ballast tanks and to drain the forward cofferdams, which can be filled with water ballast. Clean ballast piping system is to follow the requirements of 14.13.

14.11.2 The cofferdams are not to have any direct connections to the cargo tanks or cargo lines.

14.12 Ballast tanks and void spaces within the region of the cargo tanks and fore peak ballast system

14.12.1 The clean ballast tanks and void spaces within the cargo space are not to have any connection to the cargo piping system. For ballast and bilge purposes, a ballast/bilge pump is to be provided for dealing with the contents of these spaces, located in the cargo pump room or other suitable space within the region of the cargo tanks.

14.12.2 Where it is proposed to connect double bottom and/or wing tanks, located in the range of cargo tanks, with pumps in the machinery space, this may be considered for approval provided the tanks are completely separated from the cargo tanks by cofferdams, heating ducts or containment spaces, etc.

14.12.3 The fore peak tank can be ballasted with the system serving other ballast tanks within the cargo area, provided:

- The fore peak tank is considered as hazardous;
- The vent pipe openings are located on open deck 3 m away from sources of ignition;

- Means are provided, on the open deck, to allow measurement of flammable gas concentrations within the fore peak tank by a suitable portable instrument;
- The sounding arrangement to the fore peak tank is direct from open deck;
- The access to the fore peak tank is direct from open deck. Alternatively, indirect access from the open deck to the fore peak tank through an enclosed space may be accepted provided that:

(1) In case the enclosed space is separated from the cargo tanks by cofferdams, the access is through a gas tight bolted manhole located in the enclosed space and a warning sign is to be provided at the manhole stating that the fore peak tank may only be opened after:
   - it has been proven to be gas free; or
   - any electrical equipment which is not certified safe in the enclosed space is isolated.

(2) In case the enclosed space has a common boundary with the cargo tanks and is therefore hazardous, the enclosed space can be well ventilated.

14.13 Clean ballast piping in way of cargo tanks

14.13.1 Ballast piping are to be entirely separate from the cargo piping system. However, for safety purposes, emergency de-ballasting of clean ballast tanks may be carried out by means of a portable spool connection to a cargo oil pump, provided that its ballast suction is to be fitted with a non-return valve.

14.13.2 Where ballast lines led from the clean ballast pumps in the cargo pump room to the forward ballast tanks pass through cargo tanks, the pipes are to be of heavy gauge steel, not less than 16mm in thickness, having welded or heavy flanged joints. The number of joints is to be kept to a minimum. Where the proposed material is to be more corrosion resistant than carbon or carbon-manganese steel. the pipe thickness will be specially considered. For suitable expansion, expansion bends are to be fitted to these pipes inside cargo tanks.

14.14 Vent and sounding pipes

14.14.1 The cofferdams at the fore and aft ends of the cargo space, and other tanks or cofferdams within the cargo space, not for cargo, are to be fitted with vent and sounding pipes led to the open deck. The vent pipes are to be fitted with wire gauze diaphragms at their outlets.

14.14.2 The vent and sounding pipes are not to pass through cargo tanks. Where this cannot be avoided, the pipes are to be of steel having a wall thickness of not less than 12,5mm and they are to be in continuous lengths or with welded joints.

14.15 Cargo piping arrangements - general requirements

14.15.1 The piping system dealing with the cargo is to be made up from pipes, pumps and fittings whose general and specific requirements are contained in this subsection.

14.15.2 Stand-by means for pumping out each cargo tank are to be provided.

14.15.3 Where cargo tanks are provided with single deep well pumps, or submerged pumps, it is necessary to provide alternative means for emptying the tanks in the event of the failure of a pump. For this purpose, portable submersible pumps may be provided on board, ensuring at the same time that such a pump could be safely introduced into a full or partly full tank. The related documents describing such arrangements are to submitted for approval.

14.15.4 Cargo tank openings e.g. hatches, tank cleaning openings etc. are to be located on the open deck.

14.15.5 All cargo oil tanks and all compartments adjacent to cargo oil tanks, are to be efficiently ventilated and subjected to gas freeing procedures after the cargo has been discharged. It is recommended that suitable arrangements be provided to enable double bottom tanks situated below cargo tanks to be filled with water ballast to assist in the gas freeing of these tanks. See also 14.4.2.

14.16 Cargo pumps requirements

14.16.1 Cargo pumps are to be used only for cargo handling, except, as provided in 14.10 and 14.11, dealing with pump room and cofferdam drainage within the region of cargo tanks.

14.16.2 Means are to be provided for stopping the cargo oil pumps from a position outside the pump rooms as well as at the pumps.

14.16.3 The cargo pumps must be protected against overpressure by suitable means e.g. relief valves located at the pump's discharge which are to discharge at the suction side. Alternative proposals are to be specially considered.

14.16.4 Where cargo pumps are driven by shafting which penetrates a pump room bulkhead or deck, gastight glands are to be fitted to the shaft at the pump room plating. The lubrication of the glands must be carried out from outside the pump room. The seal parts of the glands are to be of materials that will not initiate sparks. The glands are to be of an approved type and are to be attached to the bulkhead in accordance with 9.2.6. Where a bellows piece is proposed for suitable expansion, it is to be hydraulically tested to 3.4 bar, before fitting.

14.16.5 Where cargo pumps are driven by hydraulic motors which are located inside cargo tanks, suitable means are to be taken to exclude any contamination of the operating medium with cargo liquid at least under normal operating conditions.

14.17 Cargo piping system-general

14.17.1 Cargo lines are to be entirely located inside the cargo area.

14.17.2 Cargo lines and cargo pumps must have suitable arrangements so as to be drained to an appropriate tank which might be a cargo tank.

14.17.3 Where it is necessary, expansion joints of an approved type or bends are to be fitted in the cargo lines. Expansion pieces of an approved type, using oil-resisting rubber or other suitable material may be accepted in cargo lines.

14.17.4 In combination carriers where cargo wing tanks are provided, cargo oil lines below deck are to be installed inside these tanks. However, LHR may permit cargo oil lines to be placed in special ducts which are to be capable of being adequately cleaned and ventilated to the satisfaction of LHR's Surveyors. Where cargo wing tanks are not provided, cargo oil lines below deck are to be placed in special ducts.

14.17.5 Shore connections. The pipes, valves and fittings of the shore connection are to be of steel or other approved ductile material. The whole construction is to be strongly supported. To each shore connection a manually operated shut-off valve is to be fitted.

14.17.6 Shore connections outside the cargo area. For ships, where the cargo is loaded or discharged at the bow and/or stern part of the ship, outside the cargo area, the piping equipment forward and/or aft of the cargo area are to have only welded joints and are to be provided with spectacle flanges or removable spool pieces, where branched off from the main line, and a blank flange at the bow and/or stern end connection, irrespective of the number and type of valves in the line. The spaces within 3 m of discharge manifolds are to be considered as dangerous spaces with regard to electrical equipment.

14.18 Connections to cargo tanks

14.18.1 The loading pipes to each cargo tank, are to be led at a level as low as practicable inside the tank.

14.18.2 Where cargo pipe lines are led through cargo tanks or through other spaces situated below the weather deck, the connection to each tank is to be provided with a valve situated inside the tank, capable of being operated from the deck. In the case of cargo tanks which are located adjacent to below-deck pump rooms or pipe tunnels, the deck operated valves may be located in these spaces at the bulkhead.

14.18.3 In any case, not less than two isolating shut-off valves are to be provided in the pipe lines between the tanks and the cargo pumps.

14.19 Remote control valves

14.19.1 The remotely controlled valves either on deck or in pump rooms, are to be able to be operated manually as well, independent of the remote operating mechanism. Valves provided with facilities for being remotely controlled, are to be able of being operated also manually, independent of the remote operating mechanism.

14.19.2 Where the valves and their actuators are located inside the cargo tanks, two separate suctions are to be provided in each tank, or alternative means of emptying the tank, in the event of a defective actuator, are to be provided.

14.19.3 All actuators are to be of a type which will prevent the valves from opening inadvertently in the event of loss of pressure in the operating medium. Indication is to be provided at the remote control station showing whether the valve is open or not.

14.19.4 The piping equipment as well as the actuators inside the cargo tanks are to be made of materials suitable for use with the intended cargoes.

14.19.5 The actuators inside cargo tanks must not be operated by compressed air.

14.19.6 The actuator operating medium in hydraulic systems is to have a flash point of 60°C or more (closed cup test) and is to be compatible with the intended cargoes.

14.19.7 The design of the actuator is to be such as to avoid any contamination of the operating medium with cargo at least under normal operating conditions.

14.19.8 Where the operating medium is fluid, the supply tank is to be located as high as possible above the level of the top of the cargo tanks. All actuator supply lines are to enter the cargo tanks through the highest part of the tanks. The supply tank is to be of the closed type with an air pipe led to a safe space on the open and fitted with a flameproof wire gauze diaphragm as its open end. This tank is to have a high and low level audible and visual alarm. The requirements of this paragraph are not mandatory where actuators and piping are located outside the cargo tanks.

14.19.9 Remotely controlled valves not arranged for manual operation, are to be provided with efficient means to operating them in the event of damage to the main hydraulic circuits on deck. In the case of valves located inside cargo tanks, this could be achieved by ensuring that the supply lines to the actuators are led vertically inside the tanks from deck, and that connections with the necessary isolating valves are provided on deck for coupling to a portable pump carried on board.

14.20 Cargo handling controls

14.20.1 All electrical circuits (measuring, monitoring or communication circuits) located in dangerous spaces are to be intrinsically safe.

14.20.2 The controls are to be grouped at a number of control stations or at one main control station.

14.20.3 The open deck, the bridge, the machinery space and the cargo handling stations must be provided with means for efficient communication.

14.20.4 The cargo handling controls are to be separate from the propulsion and auxiliary machinery controls.

14.21 Cargo tank venting - general

14.21.1 Each cargo tank is to be provided with venting arrangements in order to protect the tanks from overpressure or vacuum.

14.21.2 In order to ensure that the pressure in any tank does not exceed the test pressure of this tank, appropriate measures are to be taken, such as high-level alarms, overflow control systems or other equivalent means together with gauging devices and tank filling procedures.

14.21.3 Cargo tank venting arrangements are to be designed to provide:

(1) pressure/vacuum release of small volumes of vapor/air mixtures flowing during a normal voyage and
(2) venting of large volumes of vapor/air mixtures during cargo handling and gas-freeing operations.

14.22 Pressure/vacuum and venting systems

14.22.1 The pressure/vacuum system and venting systems may be separate or combined and may be connected to an inert gas system.

14.22.2 The venting systems may be designed to permit the free flow of vapor/air mixtures or, alternatively, the egress of vapor/air mixtures at velocities not less than 30 m/sec. The requirements given in this subsection will vary according to the system fitted.

14.22.3 The vent and/or pressure/vacuum valve stand pipes are to be connected to the highest part of each tank, and, where combined systems are adopted, a means of isolation is to be provided between each tank and a common main. Where stop valves are fitted, they are to be provided with locking arrangements which are to be under the control of a responsible ship's officer. Any isolation must continue to permit the flow caused by thermal variations in a cargo tank in accordance with 14.21.3.

14.22.4 Means are to be provided to prevent any tank being subjected to excessive pressure or vacuum during any phase of the cargo handling or ballasting operations.

14.22.5 Pressure/vacuum valves are to be set at a positive pressure of not more than 0,2 bar above atmospheric pressure and a negative pressure of not more than 0,07 bar below atmospheric. Higher positive pressures not exceeding 0,7 bar gauge may be permitted in specially designed integral tanks.

14.22.6 In no case are shut-off valves to be fitted either above or in the pipe leading to a pressure/vacuum valve. However, by-pass valves may be fitted or provision may be made to enable the tank pressure/vacuum valves to be held in an open position. The arrangements are to be such that clear indication is given when the by-pass valve is open or the pressure/vacuum valve is secured in the open position. In any case, means are to be provided to enable the functioning of the pressure/vacuum valve to be easily checked.

14.22.7 The area of the venting system used during cargo loading is to be based on the maximum design loading rate and a gas evolution factor of 1,25.

14.22.8 Suitable drainage arrangements are to be provided in the vapor lines. In general, the venting arrangements are to be self-draining to the cargo tanks under all normal conditions of trim and list of the ship. Where it may not be possible to provide self-draining lines, permanent arrangements are to be provided to drain the vent lines to a cargo tank.

14.23 Arrangement of vapor outlets/inlets

14.23.1 Outlets from vent pipes and, where necessary outlets/inlets from pressure/vacuum valves are to be provided with readily renewable wire gauzes or safety heads of approved type. Material of wire gauzes is to be resistant to corrosion.

14.23.2 Vent outlets and pressure/vacuum valve outlets, if used during loading, are to be arranged to discharge the vapor in an upward vertical direction. All outlets are to be arranged to prevent the entrance of water into the cargo tanks.

14.24 Location of cargo tank vent outlets

14.24.1 The height of vapor outlets is to be not less than 6m above the weather deck or above the fore and aft gangway if fitted within 4 m of the gangway. The vapor outlets are also to be arranged not less than 10 m from the nearest air intakes or openings to accommodation and enclosed working spaces, or to possible sources of ignition. This distance may be reduced to 5 m in the case of outlets from pressure vacuum valves as required by 14.22.3.

14.24.2 The height above the weather deck of the outlets from pressure/vacuum valves, which are used for breathing purposes only, (see 14.22.3 (a)) and high velocity venting heads, may be reduced to 2m.

14.24.3 Where a reduced height of vent outlets is permitted on account of the fitting of high velocity vents the venting heads are to be of an approved type and the design is to be such that, when in operation, the vapours will be expelled upwards in an unimpeded jet at a velocity of not less than 30 m/sec. The effectiveness of the venting heads is to be demonstrated.

14.25 Cargo tank sounding arrangements

14.25.1 Each cargo tank is to be fitted with suitable means for ascertaining the liquid level in the tank in accordance with the following requirements.

14.25.2 Sounding pipes or other approved devices which may permit a limited amount of vapor to escape to atmosphere when being used, would be accepted for those tanks which are not required to be fitted with closed sounding devices, see 14.26. The devices are to be so designed as to minimize the sudden release of vapor or liquid under pressure and the possibility of liquid spillage on deck. Means are also to be provided for relieving tank pressure before the device is operated.

14.25.3 Separate ullage openings may be fitted as a reserve means for sounding cargo tanks.

14.25.4 Arrangements which permit the escape of vapor to the atmosphere are not to be fitted in enclosed spaces.

14.26 Close sounding devices

14.26.1 In all tankers fitted with a fixed inert gas system, the cargo tanks are to be fitted with closed sounding devices of an approved type, which do not permit the escape of cargo to the atmosphere when being used.

14.26.2 Proposals to use indirect sounding or measuring devices which do not penetrate the tank plating will be specially considered.

14.27 Cargo heating arrangements

14.27.1 Where heating systems are provided for the cargo tanks, the arrangements are to comply with the following requirements.

14.27.2 Spectacle flanges or spool pieces are to be provided in the heating medium supply and return pipes to the cargo heating system at a suitable position within the cargo area, so that lines can be blanked off in circumstances where the cargo does not require to be heated or where the heating coils have been removed from the tanks. Alternatively, blanking arrangements may be provided for each tank heating circuit.

14.27.3 Where a combustible liquid is used as the heating medium it is to have a flash point of 60°C or above (closed cup test).

14.27.4 Generally, the temperature of the heating medium is not to exceed 220°C.

14.27.5 The heating medium supply and return lines are not to penetrate the cargo tank plating other than at the top of the tank, and the main supply lines are to be run above the weather deck.

14.27.6 Isolating shut-off valves or cocks are to be provided at the inlet and outlet connections to the heating circuits of each tank and means are to be provided for regulating the flow.

14.27.7 Where steam or water is employed in the heating circuits the returns are to be led to an observation tank which is to be in a well ventilated and well-lit part of the machinery space remote from the boilers.

14.27.8 Where a thermal oil is employed in the heating circuits, the arrangements will be specially considered but, in any case, they are to be such that contamination of the thermal oil with cargo liquid cannot take place under normal operating conditions. Generally, the arrangements are at least to comply with 14.19.8 in so far as they are applicable.

14.27.9 In any heating system, a higher pressure is to be maintained within the heating circuit that the maximum pressure head which can be exerted by the contents of the cargo tank on the circuit. Alternatively, when the heating circuit is not in use it may be drained and blanked.

14.27.10 Means are to be provided for measuring the cargo temperature. Where overheating could result in a dangerous condition an alarm system which monitors the cargo temperature is to be provided.

14.28 Inert gas systems - general

14.28.1 The requirements of 14.29 to 14.37 to apply where an inert gas system, based on flue gas, is fitted on board ships intended for the carriage of oil in bulk having a flash point not exceeding 60°C (closed cup test). Any proposal to use an inert gas other than flue gas e.g. nitrogen will be specially approved.

14.28.2 For this Section purposes the term "cargo tank" includes also "slop tanks".

14.29 Inert gas systems - gas supply

14.29.1 The inert gas may be treated flue gas from the main or auxiliary boiler(s), gas turbine(s) or from a separate inert gas generator. In all cases automatic combustion control capable of producing suitable inert gas under all service conditions, is to be fitted.

14.29.2 Two fuel oil pumps are to be fitted to the inert gas generator. One fuel pump only may be accepted provided sufficient spares for the fuel oil pump and its prime mover are carried on board to enable any failure of the fuel oil pump and its prime mover to be rectified by the ship's crew.

14.29.3 The inert gas system is to be capable of:

(1) inerting empty cargo tanks by reducing the oxygen content of the atmosphere in each tank to a level at which combustion cannot be supported.
(2) maintaining the atmosphere in any part of any cargo tank with an oxygen content not exceeding 8% by volume and at a positive pressure at all times in port and at sea except when it is necessary for such a tank to be gas free.
(3) eliminating the need for air to enter a tank during normal operations except when it is necessary for such a tank to be gas free.
(4) purging empty cargo tanks of hydrocarbon gas, so that subsequent gas freeing operations will at no time create a flammable atmosphere within the tank.

14.29.4 The system is to be capable of delivering inert gas to the cargo tanks at a rate of at least 125% of the maximum rate of discharge capacity of the ship expressed as a volume. The system is to be capable of delivering inert gas with an oxygen content of not more than 5% by volume in the inert gas supply main to the cargo tanks at any required rate or flow.

14.29.5 Flue gas isolating valves are to be fitted to the inert gas supply mains between the boiler uptakes and the flue gas scrubber. These valves are to be provided with indicators to show whether they are open or shut and precautions are to be taken to maintain them gastight and keep the seating clear of soot. Arrangements are to be made to ensure that boiler soot blowers cannot be operated when the corresponding flue gas valve is open.

14.30 Inert gas systems - gas scrubber

14.30.1 A flue gas scrubber is to be fitted which will effectively cool the volume of gas specified in

14.29.4 and remove solids and sulphur combustion products. The cooling water arrangements are to be such that an adequate supply of water will always be available without interfering with any essential services on the ship. Provision is also to be made for alternative supply of cooling water.

14.30.2 Filters or equivalent devices are to be fitted to minimize the amount of water carried over to the inert gas blowers.

14.30.3 The scrubber is to be located aft of all cargo tanks, cargo pump rooms and cofferdams separating these spaces from machinery spaces.

14.31 Inert gas systems - gas blowers

14.31.1 At least two blowers are to be fitted which together are to be capable of delivering to the cargo tanks at least the volume of gas required by 14.29.4. In the system with gas generators one blower only may be accepted if that system is capable of delivering the total volume of gas required by 14.29.4 to the protected cargo tanks, provided that sufficient spares for the blower and its prime mover are carried on board to enable any failure of the blower and its prime mover to be rectified by the ship's crew.

14.31.2 The inert gas system is to be so designed that the maximum pressure which it can exert on any cargo tank will not exceed the test pressure of any cargo tank. Suitable shut-off arrangements are to be provided on the suction and discharge connections of each blower. Arrangements are to be provided to enable the functioning of the inert gas plant to be stabilized before commencing cargo discharge. If the blowers are to be used for gas-freeing their air inlets are to be provided with blanking arrangements.

14.32 Inert gas systems - gas distribution lines

14.32.1 Special consideration is to be given to the design and location of scrubber and blowers with relevant piping and fittings in order to prevent flue gas leakages into enclosed spaces.

14.32.2 To permit safe maintenance, an additional water seal or other effective means of preventing flue gas leakage is to be fitted between the flue gas isolating valves and scrubber or incorporated in the gas entry to the scrubber.

14.32.3 A gas regulating valve is to be fitted in the inert gas supply main. This valve is to be automatically controlled to close as required in 14.34.9 and 14.34.10. It is also to be capable of automatically regulating the flow of inert gas to the cargo tanks unless means are provided to automatically control the speed of the inert gas blowers required in 14.31.1.

14.32.4 The valve referred to 14.32.3 is to be located at the forward bulkhead of the forward most gas safe space through which the inert gas supply main passes.

14.32.5 At least two non-return devices, one of which is to be a water seal are to be fitted in the inert gas supply main in orders to prevent the return of hydrocarbon vapor to the machinery space uptakes or to any gas safe spaces under all normal conditions of trim list and motion of the ship. They are to be located between the automatic valves required by 14.32.3 and the aftermost connection to any cargo tank or cargo pipeline.

14.32.6 The devices referred to in 14.32.5 are to be located in the cargo area on deck.

14.32.7 The water seal referred to in 14.32.5 is to be capable of being supplied by two separate pumps, each of which is to be capable of maintaining an adequate supply at all times.

14.32.8 The arrangement of the seal and its associated fittings is to be such that it will prevent backflow of hydrocarbon vapours and will ensure the proper functioning of the seal under operating conditions.

14.32.9 Provision is to be made to ensure that the water seal is protected against freezing, in such a way that the integrity of seal is not impaired by overheating.

14.32.10 A water loop or other approved arrangement is also to be fitted to each associated water supply and drain pipe and each venting or pressure-sensing pipe leading to gas safe spaces. Means are to be provided to prevent such loops from being emptied by vacuum.

14.32.11 The deck water seal and all loop arrangements are to be capable of preventing return of hydrocarbon vapours at a pressure equal to the test pressure of the cargo tanks.

14.32.12 The second non-return device is to be a non-return valve or equivalent capable of preventing the return of vapours or liquids and fitted forward of the deck water seal required in 14.32.5. It is to be provided with positive means of closure. As an alternative to positive means of closure an additional valve having such means of closure may be provided forward of the non-return valve to isolate the deck water seal from the inert gas main to cargo tanks.

14.32.13 As an additional safeguard against the possible leakage of hydrocarbon liquids or vapours back from the deck main means are to be provided to permit this section of the line between the valve having positive means of closure referred in 14.32.12 and the valve referred to in 14.32.3 to be vented in a safe manner when the first of these valves is closed.

14.32.14 The inert gas main may be divided into two or more branches forward of the non-return devices required by 14.32.5.

14.32.15 The inert gas supply mains are to be fitted with branch piping leading to each cargo tank. Branch piping for inert gas is to be fitted with either stop valves or equivalent means of control for isolating each tank. Where stop valves are fitted, they are to be provided with locking arrangements which are to be under the control of a responsible ship's officer.

14.32.16 In combination carriers the arrangement to isolate the slop tanks containing oil or oil residues from other tanks is to consist of blank flanges which will remain in position at all times when cargoes other than oil are being carried, except as provided for in 14.7.

14.32.17 Means are to be provided to protect cargo tanks against the effect of overpressure or vacuum caused by thermal variations when the cargo tanks are isolated from the inert gas mains.

14.32.18 Piping systems are to be so designed as to prevent the accumulation of cargo or water in the pipelines under all normal conditions.

14.32.19 Suitable arrangements are to be provided to enable the inert gas main to be connected to an external supply of inert gas.

14.33 Inert gas systems - venting arrangements

14.33.1 The arrangements for the venting of all vapours displaced from the cargo tanks during loading and ballasting are to comply with 14.15 and are to consist of either one or more mast risers, or a number of high velocity vents The inert gas supply mains may be used for such venting.

14.33.2 The arrangements for inerting, purging or gas freeing of empty tanks as required in 14.29.3 are to be such that the accumulation of hydrocarbon vapours in pockets formed by the internal structural members in a tank is minimized and that:

(1) on individual cargo tanks the gas outlet pipe if fitted is to be positioned as far a practicable from the inert gas/air inlet and in accordance with 14.29.3. The inlet of such outlet pipes may be located either at deck level or at not more than 1 m above the bottom of the tank.
(2) the cross-sectional area of such gas outlet pipes referred to in (1) is to be such that an exit velocity of at least 20 m/sec can be maintained when any three tanks are being simultaneously supplied with inert gas. Their outlets are to extend not less than 2 m above deck level.
(3) each gas outlet referred to in (2) is to be fitted with suitable blanking arrangements.
(4) if a connection is fitted between the inert gas supply mains and the cargo piping system, arrangements are to be made to ensure an effective isolation having regard to the large pressure difference which may exist between the systems. This is to consist of two shut-off valves with an arrangement to vent the space between the valves in a safe manner or an arrangement consisting of a spool piece with associated blanks. The valve separating the inert gas supply main from the cargo main and which is on the cargo main side is to be a non-return valve with a positive means of closure.

14.33.3 One or more pressure-vacuum breaking devices are to be provided to prevent the cargo tanks from being subject to:

(1) a positive pressure in excess of the test pressure of the cargo tank if the cargo were to be loaded at the maximum rated capacity and all other outlets were left shut; and
(2) a negative pressure in excess of 700mm water gauge if cargo were to be discharged at the maximum rated capacity of the cargo pumps and the inert gas blowers were to fail.

Such devices shall be installed on the inert gas main unless they are installed in the venting system required by 14.29.3 or on individual cargo tanks.

14.33.4 The location and design of the devices referred to in 14.33.3 are to be in accordance with 14.21 to 14.24.

14.34 Inert gas systems - instrumentation and alarms

14.34.1 Means are to be provided for continuously indicating the temperature and pressure of the inert gas at the discharge side of the gas blowers, whenever the gas blowers are operating.

14.34.2 Instrumentation is to be fitted for continuously indicating and permanently recording when the inert gas is being supplied:

(1) the pressure of the inert gas supply mains forward of the non-return devices required by 14.32.5 and
(2) the oxygen content of the inert gas in the inert gas supply mains on the discharge side of the gas blowers.

14.34.3 The devices referred to in 14.34.2 are to be placed in the cargo control room where provided. But where no cargo control room is provided, they are to be placed in a position accessible to the officer in charge of cargo operations.

14.34.4 In addition to 14.34.2 meters are to be fitted:

(1) in the navigating bridge to indicate at all times the pressure referred to in 14.34.2(1) and the pressure in the slop tanks of combination carriers whenever those tanks are isolated from the inert gas supply main and
(2) in the machinery control room or in the machinery space to indicate the oxygen content referred to in 14.34.2(2).

14.34.5 Portable instruments for measuring oxygen and flammable vapour concentration are to be provided. In addition suitable arrangements is to be made on each cargo tank such that the condition of the tank atmosphere can be determined using these portable instruments.

14.34.6 Suitable means are to be provided for the zero and span calibration of both fixed and portable gas concentration measurement instruments referred to in 14.34.2, 14.34.4 and 14.34.5.

14.34.7 Audible and visual alarms are to be provided to indicate:

(1) low water pressure or low water flow rate to the flue gas scrubber as referred to in 14.30.1
(2) high water level in the flue gas scrubber as referred to in 14.30.1.
(3) high gas temperature as referred to in 14.34.1.
(4) failure of the inert gas blowers referred to in 14.31.
(5) oxygen content in excess of 8% by volume as referred to 14.34.2(2)
(6) failure of the power supply to the automatic control system for the gas regulating valve and to the indicating devices as referred to in 14.32.3 and 14.34.2.
(7) low water level in the water seal as referred to in 14.32.5 (h) gas pressure less than 100mm water gauge as referred to in 14.34.2(1). The alarm arrangement is to be such as to ensure that the pressure in slop tanks in combination carriers can be monitored at all times and
(8) high gas pressure as referred to in 14.34.2(1).

14.34.8 In the system with gas generators, audible and visual alarms are to be provided in accordance with 14.34.7(1), (3), (5) to (8) and additional alarms to indicate:

(1) insufficient fuel oil supply
(2) failure of the power supply to the generator
(3) failure of the power supply to the automatic control system for the generator.

14.34.9 Automatic shut-down of the inert gas blowers and gas regulating valve is to be arranged on predetermined limits being reached in respect of (1), (2) and (3) of 14.34.7.

14.34.10 Automatic shut-down of the gas regulating valve is to be arranged in respect of 14.34.7(4).

14.34.11 In respect of 14.34.7(5) when the oxygen content of the inert gas exceeds 8% by volume immediate action is to be taken to improve the gas quality. Unless the quality of the gas improves, all cargo tank operations are to be suspended so as to avoid air being drawn into the tanks and the isolation valve referred to in 14.32.12 is to be closed.

14.34.12 The alarms required in 14.34.7(5), (6), and (8) are to be fitted in the machinery space and cargo control room, where provided, but in each case in such a position that they are immediately received by responsible members of the crew.

14.34.13 In respect of 14.34.7(7) where a semi-dry or dry water seal is fitted, the arrangements are to be such that the maintenance of an adequate reserve of water will be ensured at all times and that the water seal will be automatically formed when the gas flow ceases. The audible and visual alarm on the low level of water in the water seal is to operate when the inert gas mains is not being supplied.

14.34.14 An audible alarm system independent of that required in 14.34.7(8) or automatic shut-down of cargo pumps is to be provided to operate on predetermined limits of low pressure in the inert gas mains being reaches.

14.34.15 Detailed instruction manuals are to be provided on board, covering the operation, safety and maintenance requirements and occupational health hazards relevant to the inert gas system and its application to the cargo tank system. The manuals are to include guidance on procedures to be followed in the event of a fault or failure of the inert gas system.

14.34.16 The inert gas system including alarms and safety devices is to be installed on board and tested under working conditions to the satisfaction of the Surveyors.

14.35 Tank cleaning equipment

14.35.1 Tank cleaning equipment may be installed provided that LHR has given its consent.

14.35.2 Tank cleaning equipment is to be made of steel or other approved material.

14.35.3 Tank washing equipment is to be so supported as to withstand the reaction forces arisen during operation. The installation of the tank cleaning system is to be such that normal natural resonance does not occur under any operating conditions.

14.35.4 Tanks equipped for washing with crude oil are to be fitted with an inert gas system as well, in accordance with 14.28 to 14.34.

14.36 Gas-freeing procedures for tanks and cofferdams

14.36.1 The tanks and the cofferdams of tankers are to be provided with means for gas-freeing.

14.36.2 The inlet openings in cargo tanks used for gas-freeing must be located either immediately below deck or not higher than 1 m above the tank bottom. The outlet openings used to extract the vapours and gases are to be located as far as possible from air or inert gas openings at a height of at least 2m above the deck. The discharge of gas/air mixtures is to be carried out vertically. The gas/air mixture is to be extracted by an outlet velocity of not less than 20 m/sec. The openings for gas-freeing are to be secured with screw-down covers.

14.36.3 Where fans for the gas-freeing are connected to cargo oil lines suitable arrangements are to be provided to ensure that when the ventilation system is not in use, neither cargo or vapours can be inserted into the fans.

14.36.4 Where the ships have inert gas systems, the free area of the vent openings are to be so designed that an outlet velocity of at least 20 m/sec is maintained if 3 cargo tanks are simultaneously are filled with inert gas.

14.36.5 Where the ships are not provided with inert systems, the event openings used for gas-freeing are to be fitted with flame arresters. Where LHR has given its consent, the fitting of flame arresters may be dispensed with if proof is provided of a velocity of at least 30 m/sec in the vent openings.

14.37 Instrumentation for gas detection

14.37.1 At least two portable instruments are to be provided on board for gas detection purposes.

14.38 Design of integrated cargo and ballast systems on tankers (IACS UR M64 Rev.1 (2004))

14.38.1 Application

These requirements are applicable to integrated cargo and ballast systems installed on tankers (i.e. cargo ships constructed or adapted for the carriage of liquid cargoes in bulk) contracted for construction on or after 1 January 2004 (1)(2), irrespective of the size or type of the tanker. Within the scope of these requirements, integrated cargo and ballast system means any integrated hydraulic and/or electric system used to drive both cargo and ballast pumps (including active control and safety systems and excluding passive components, e.g. piping).

NOTES:

1. The requirements of 14.38 are to be uniformly implemented by all IACS Societies on tankers (as defined in 14.38.1) contracted for construction on or after 1 January 2004.
2. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of "contract for construction", refer to IACS Procedural Requirement (PR) No. 29.

14.38.2 Functional Requirements

The operation of cargo and/or ballast systems may be necessary, under certain emergency circumstances or during the course of navigation, to enhance the safety of tankers. As such, measures are to be taken to prevent cargo and ballast pumps becoming inoperative simultaneously due to a single failure in the integrated cargo and ballast system, including its control and safety systems.

14.38.3 Design features

The following design features are, inter alia, to be fitted:

(1) the emergency stop circuits of the cargo and ballast systems are to be independent from the circuits for the control systems. A single failure in the control system circuits or the emergency stop circuits are not to render the integrated cargo and ballast 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 power pack making ballast pumps inoperable,
(3) the control systems are to be provided with 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 alarm activation at each location where the control panel is fitted,
(4) in the event of failure of the automatic or remote-control systems, a secondary means of control is to be made available for the operation of the integrated cargo and ballast system. This is to be achieved by manual overriding and/or redundant arrangements within the control systems.

SECTION 15 Hydraulic systems

15.1 General

15.1.1 Scope

The rules contained in this Section apply to hydraulic systems used e.g. to operate hatch covers, closing appliances in the ship's shell and bulkheads, and hoists. The rules are to be applied in analogous manner to the ship's other hydraulic systems except where covered by the rules of Part 5, Chapter 8.

15.1.2 Documents for approval

The diagram of the hydraulic system together with drawings of the cylinders containing all the data necessary for accessing the system, e.g. operating data, descriptions, materials used etc., are to be submitted in triplicate for approval.

15.1.3 Dimensional design

For the design of pressure vessels, see Part 5, Chapter 7. For the dimensions of pipes and hose assemblies, see Part 5, Chapter 8.

15.2 Materials

15.2.1 Approved materials

Components fulfilling a major function in the power transmission system are normally to be made of steel or cast steel in accordance with Part 2. The use of other materials is subject to special agreement with LHR. Cylinders are preferably to be made of steel, cast steel or nodular cast iron (with a predominantly ferritic matrix). Pipes are to be made of seamless or longitudinally welded steel tubes. The pressure- loaded walls of valves, fittings, pumps, motors etc. are subject to the requirements of Part 5, Chapter 8.

15.2.2 Testing of materials

The following components are to be tested according to the Rules for Materials:

(1) Pressure pipes
(2) Cylinders where the product of the pressure times the diameter:

$$P \cdot D_i > 20000$$

where:

P = maximum allowable working pressure, bar

D_i = inside diameter of tube, mm

(3) For testing the materials of hydraulic accumulators, see Part 5, Chapter 7.

Testing of materials by LHR may be dispensed with in the case of cylinders for secondary applications provided that evidence in the form of a works test certificate is supplied.

15.3 Hydraulic operating equipment for hatch covers-design and construction.

15.3.1 Hydraulic operating equipment for hatch covers may be served either by one common power station for all hatch covers or by several power stations individually assigned to a single hatch cover. Where a common power station is used, at least two pump units are to be fitted. Where the systems are supplied individually, change-over valves or fittings are required so that operation can be maintained should two pump unit fail.

15.3.2 Movement of hatch covers may not be initiated merely by starting of the pumps. Special control stations are to be provided for controlling the opening and closing of hatch covers. The controls are to be designed so that, as soon as they are released, movement of the hatch covers stops immediately. The hatches should normally be visible from the control stations. Should this, in exceptional cases, be impossible, opening and closing of the hatches is to be signaled by an audible alarm. In addition, the control stations, must then be equipped with indicators for monitoring the movement of hatch covers. At the control stations, the controls governing the opening and closing operations are to be appropriately marked.

15.3.3 Suitable equipment must be fitted in or immediately adjacent to each power unit (cylinder or similar) used to operate hatch covers to enable the hatches to be closed slowly in the event of a power failure, e.g. due to a pipe rupture.

15.4 Hydraulic operating equipment for hatch covers - pipes

15.4.1 Pipes are to be installed and secured in such a way as to be protected from damage while enabling them to be properly maintained from outside. Pipes may be led through tanks in pipe tunnels only. The laying of such pipes through cargo spaces is to be restricted to the essential minimum. The piping system is to be fitted with relief valves to limit the pressure to the maximum allowable working pressure.

15.4.2 The piping system is to be fitted with filters for cleaning the hydraulic fluid. Equipment is to be provided to enable the hydraulic system to be vented.

15.4.3 The accumulator space of the hydraulic accumulator must have permanent access to the relief valve of the connected system. The gas chamber of the accumulator may be filled only with inert gases. Gas and operating medium are to be separated by accumulator bags, diaphragms or similar.

15.4.4 Connection between the hydraulic system used for hatch cover operation and other hydraulic systems is permitted only with the consent of LHR.

15.4.5 Tanks forming part of the hydraulic system are to be fitted with oil level indicators.

15.4.6 The construction of hose assemblies is to conform to Part 5, Chapter 8. The requirements that hose assemblies should be of flame-resistant construction may be waived for hose lines in spaces not subject to a fire hazard and in systems not important to the safety of the ship.

15.4.7 Emergency operation It is recommended that devices be fitted which are independent of the main system and which enable hatch covers to be opened and closed in the event of failure of the main system. Such devices may, for example, take the form of loose rings enabling hatch covers to be moved by cargo winches, warping winches etc.

15.5 Hydraulically operated closing appliances in the ship's shell-design and construction

15.5.1 The following rules apply to power equipment of hydraulically operated closing appliances in the ship's shell such as shell and landing doors which are not normally operated while at sea.

15.5.2 Local control, inaccessible to unauthorized persons, is to be provided for every closing appliance in the ship's shell. As soon as the controls are released movement of the appliance must stop immediately

15.5.3 Closing appliances in the ship's shell should normally be visible from the control stations. If the movement cannot be observed, audible alarms are to be fitted. In addition, the control stations are then to be equipped with indicators enabling the execution of the movement to be monitored.

15.5.4 Closing appliances in the ship's shell are to be fitted with devices which prevent them from moving into their end positions at excessive speed. Such devices are not to cause the power unit to be switched off. As far as is required, mechanical means must be provided for locking closing appliances in the open position.

15.5.5 Every power unit driving horizontally hinged or vertically operated closing appliances is to be fitted with throttle valves or similar devices to prevent sudden dropping of the closing appliance.

15.5.6 It is recommended that the driving power be shared between at least two mutually independent pump sets.

15.6 Hydraulically operated closing appliances in the ship's shell-piping

15.6.1 The requirements of 15.4.5 and 15.4.6 are to be applied in analogous manner to the pipes and hose lines of hydraulically operated closing appliances in the ship's shell.

15.7 Hydraulic operating systems for watertight doors-general

15.7.1 The following rules apply to the power equipment of hydraulically operated watertight doors in passenger ships. They are also applicable in analogous manner to the corresponding equipment on cargo ships.

15.7.2 The number, design and arrangement of bulkhead doors are to comply in all respects also with the applicable requirements of SOLAS in force.

15.8 Hydraulic operating systems for watertight doors-design

15.8.1 Hydraulically operated watertight doors are to be sliding doors moving horizontally. Other designs are subject to LHR's approval and may necessitate additional safety measures.

15.8.2 Hydraulic operating gear must be fully capable of closing all watertight doors despite permanent inclination of the ship to 15°.

15.8.3 Wherever applicable, the pipes of hydraulic systems for operating watertight doors are governed by the requirements of 15.4, but hose lines are not permitted.

15.8.4 Pipes and hydraulic power stations must be located at a distance of at least 0,2 B from the shell plating. The hydraulic system used to operate watertight doors is not to be connected to other hydraulic systems.

15.8.5 The hydraulic fluids must give satisfactory service even at low temperature.

15.8.6 Every watertight door operated by a manual hydraulic system must be capable of being closed from both sides of the door as well as from a readily accessible control station located above the bulkhead or freeboard deck and outside the machinery space. The controls at the doors must close and open the door, while those on the control station above deck must at least serve to close the door.

15.8.7 Every watertight door with a power-operated hydraulic system must be provided in addition with a manual actuating mechanism completely separate from the power-operated hydraulic system and complying with the requirements set out in 15.8.2 and 15.8.6. This emergency operating mechanism may be either mechanical or a manually operated hydraulic system.

15.8.8 Manually operated hydraulic and mechanical actuating mechanisms and emergency systems must enable a door to be closed completely within 90 sec when the ship is in the upright position.

15.8.9 Closed or open watertight doors must not automatically start moving if the actuating system fails.

15.8.10 The control stations mentioned in 15.8.6 must be fitted with visual indicators showing whether the door in question is fully open or closed. Electrical indicators must be capable of being supplied from the emergency power source.

15.8.11 During the closing operation of a door, an audible alarm must sound in its vicinity.

15.9 Additional requirements for power hydraulic systems

15.9.1 Two mutually independent power pump units are to be installed, if possible above the bulkhead or freeboard deck and outside the machinery spaces.

15.9.2 Each pump unit must be capable of closing all the connected watertight doors simultaneously in not more than 60 sec. On the other hand, the closing time should never be less than 20 sec.

15.9.3 The hydraulic system must incorporate accumulators with sufficient capacity to operate all the connected doors three times, i.e. to close, open and re-close them.

15.9.4 The movement of watertight doors may not be initiated simply by the starting of the pumps but shall necessitate that additional equipment be operated.

15.9.5 A central control station is to be located on the bridge. This must comprise:

(a) control and monitoring devices for the pumps and
(b) optical indicators showing the fully open or closed condition of each watertight door.

It is recommended that the controls for the individual and joint operation of all watertight doors should also be located in the central control station.

15.9.6 Where the controls are located at operating stations away from the central control station it is necessary to:

(1) locate these operating stations above the bulkhead or freeboard deck and outside the machinery space,
(2) equip the operating stations with the visual indicators mentioned in 15.9.5 for the doors concerned, and,
(3) provide a means of communication between the operating stations and the central control station.

15.9.7 Next to each door, and on both sides of the bulkhead, controls for the power hydraulic system are to be fitted, actuation of which enables a closed door to be opened. When they are released, these controls must return to their original position, thereby causing the door to close again. The controls are to be fitted in such a way that a person passing through the door is able to hold both elements in the opened position. In addition, local means must be provided for locking each door in such a way that it can no longer be opened by remote control.

15.9.8 The watertight door closing system including the control, indicating and alarm systems, must be capable of being supplied from the emergency power source. See also Part 6, Chapter 11.

15.10 Hoists

15.10.1 For the purpose of these rules, hoists include hydraulically operated appliances such as wheel house hoists, lifts and similar equipment.

15.10.2 Hoists may be supplied either by a combined power station or individually by several power stations. In the case of a combined power supply and hydraulic drives whose piping system is connected to other hydraulic systems, a second pump unit is to be fitted.

15.10.3 The movements of hoists is not to be capable of being initiated merely by starting the pumps. The movement of hoists is to be controlled from special operating stations. The controls are to be arranged so that, as soon as they are released, the movement of the hoist ceases immediately.

15.10.4 Local controls, inaccessible to unauthorized persons are to be fitted. The movement of hoists should normally be visible from the operating stations. If the movement cannot be observed, audible and or visual warning devices are to be fitted. In addition, the operating stations are then to be equipped with indicators for monitoring the movement of the hoist.

15.10.5 Devices are to be fitted which prevent the hoist from reaching its end position at excessive speed. These devices are not to cause the power unit to be switched off. As far as is necessary, mechanical means must be provided for locking the hoist in its end positions.

15.10.6 The requirements of 15.3.3 apply in analogous manner to those devices which, if the power unit fails or a pipe ruptures, ensure that the hoist is slowly lowered.

15.10.7 The requirements of 15.3.2 and 15.3.3 apply in analogous manner to the pipes and hose lines of hydraulically operated hoists.

15.10.8 After installation the equipment is to undergo an operational test. The operational test of watertight doors is to include the emergency operating system and determination of the closing times.

15.11 Works tests

15.11.1 Testing of power units.

The power units are required to undergo test on a test stand in the manufacturer's works. Where the drive power is 50 kW or over, the test is to be performed in the presence of an LHR Surveyor.

15.11.2 Pressure and tightness tests.

Pressure components are to undergo a pressure test. The test pressure is to be 50% over the maximum allowable working pressure. However, for working pressures above 200 bar the test pressure need not exceed P+100 bar, where P the maximum allowable working pressure. A tightness test is to be performed on components for which it is appropriate.

15.12 Shipboard trials

15.12.1 After installation, the equipment is to undergo an operational test. The operational test of watertight doors is to include the emergency operating system and determination of the closing times.

CHAPTER 10 Machinery Piping Systems

CONTENTS

SECTION 1 General

1.1 General requirements

1.1.1 The rules of this Chapter apply to piping systems serving the ship propulsion and auxiliary power generation systems.

1.1.2 The general requirements contained in Part 5, Chapter 9, SECTION 1 and the requirements related to the design of piping systems included in Part 5, Chapter 9, SECTION 2 are also applicable to the machinery piping systems.

1.1.3 The requirements of Part 5, Chapter 9, SECTION 3 are also to be complied with, as far as they are applicable, for the drainage of tanks, oil bilges and cofferdams, etc.

SECTION 2 Fuel oil systems

2.1 Fuel oil

2.1.1 Ships classed for unrestricted service should use fuel oil with a flash point, determined by the closed cup test, not less than 60°C. A flash point of not less than 43°C is to be allowed only for emergency generators.

2.1.2 Ships classed for service in restricted geographical limits may use fuel oil with a lower flash point, not less than 43°C, only where it can be ensured that the temperature of the machinery and boiler spaces will always be 10°C below the flash point of the fuel and the relative storage and pumping arrangements, as well as the safety precautions, are specially approved.

2.1.3 LHR may approve the use of fuel oil with a flash point lower than 60°C for ships classed for unrestricted service or lower than 43°C for ships classed for service in restricted geographical limits, provided that such fuel is not stored in any machinery space and that drawings of all the arrangements for the complete installation are to be submitted for special consideration.

2.1.4 The approval to carry fuel having a flash point below 43°C for special services, e.g. aviation spirit for use in helicopters, may be granted provided that full particulars of the proposed arrangements are submitted for special consideration.

2.1.5 Fuel oil in storage and service tanks is not to be heated to a temperature exceeding 10°C below its flash point. Heating to higher temperatures will be specially considered.

2.2 Bunker lines

2.2.1 The effective bunkering of fuel oil is to be done by permanently installed bunker lines extended to both sides of the ship, terminated to bunker stations either on the open deck or located below deck, isolated from other spaces and efficiently ventilated and drained. The bunker lines are to be fitted with blind flanges at the end intended for shore connection. In order to avoid any over-pressure in the bunker lines, relief valves are to be fitted which will discharge to an overflow tank or other safe position.

2.2.2 Where bunker lines are led through the tank top, a screw down non-return valve at the tank top is sufficient.

2.2.3 Bunker lines are to extend to the bottom of the tanks. The use of bunker lines as suction lines is permitted.

2.2.4 Filters are to be fitted to bunker lines.

2.3 Suction lines and their valves

2.3.1 Every fuel oil suction pipe from a double bottom tank is to be fitted with a valve.

2.3.2 Every fuel oil suction pipe from a storage, settling or daily service tank situated above the double bottom is to be fitted with a valve secured to the tank. Where the tank is situated in the machinery space the valve is to be capable of being closed locally and from positions outside this space which will be always accessible. Where the tank is situated outside the machinery space, above the double bottom, a valve is to be fitted in the pipeline inside the machinery space capable of being controlled locally and from positions outside this space, always accessible. The ability of remote control may be omitted where the valve on the tank is already capable of being closed from an accessible position above the bulkhead deck.

2.3.3 In the case of very small tanks consideration will be given to the omission of remote controls.

2.3.4 The connections of suction lines are to be arranged far enough from the drains of the tanks so that water and impurities which have settled down will not enter the suctions.

2.4 Pipe layout

2.4.1 Fuel oil lines which pass through ballast tanks are to have an increased wall thickness, as indicated in Part 5, Chapter 8, SECTION 3, Table 8.3.1.

2.4.2 Pipes conveying vegetable oils or similar cargo oils are not to be led through fuel oil tanks, nor are fuel oil pipes to be led through cargo oil tanks.

2.4.3 Fuel oil lines or oily water lines are to be entirely distinct and separate from lines dealing with fresh water. Fuel oil lines are not to be led through tanks which contain fresh water, nor pipes conveying fresh water to be led through fuel oil tanks.

2.4.4 The location of pipes, valves and other fittings is to be such that it can be ensured that oil cannot enter tanks not intended for the carriage of oil. All valves are to be located so that they can be controlled from readily accessible positions which, in the machinery space, are to be above the working platform.

2.4.5 Glass and plastic components are not permitted in fuel systems.

2.4.6 Transfer, suction and other low pressure oil pipes and all pipes passing through oil storage tanks are to be made of cast iron or steel, having flanged joints suitable for a working pressure of not less than 6.9 bar. The flanges are to be machined and the jointing material is to be impervious to oil. Where the pipes are of 25mm internal diameter or less, they may be of seamless copper or copper alloy, except those which pass through oil storage tanks. Oil pipes within the machinery space are to be so positioned that they can be readily inspected and repaired.

2.4.7 Pipes transferring heated oil under pressure are to be of seamless steel or other approved material having flanged or welded joints, and are to be placed above the platform in well lit parts of the machinery space. The number of flanged joints is to be kept to a minimum. The flanges are to be machined, and the jointing material, which is to be impervious to oil heated to 150°C, is to be the thinnest possible, so that flanges are practically metal to metal. The scantlings of the pipes and their flanges are to be suitable for a pressure of at least 13,7 bar or of the design pressure, whichever is the greater.

2.4.8 Relief valves are to be fitted on the oil side of heaters and are to be adjusted to operate at a pressure of 3,4 bar above that of the supply pump relief valve. The discharge from the relief valves is to be led to a safe position.

2.4.9 The use of flexible hoses of approved material and design may be permitted for the burner pipes, provided that spare lengths, complete with couplings, are carried on board.

2.5 Fuel transfer, feed and booster pumps

2.5.1 Fuel transfer, feed and booster pumps are to be designed for the proposed operating temperature of the medium pumped.

2.5.2 To transfer fuel oil from fuel oil bunker tanks a power-driven fuel transfer pump is to be provided. A standby pump is also to be provided and connected ready for use or, alternatively, emergency connections may be made to another suitable power-driven pump.

2.5.3 Where a feed or booster pump is required to supply fuel oil to main or auxiliary engines, a standby pump is to be provided. Where, in the case of auxiliary engines, the pumps are attached to the engine, a standby pump may be dispensed with.

2.5.4 All pumps which are capable of developing a pressure exceeding the design pressure of the system are to be provided with relief valves. Each relief valve is to be in a close circuit, i.e. arranged to discharge back to the suction side of the pump and to effectively limit the pump discharge pressure to the design pressure of the system.

2.5.5 Simplex filters are to be fitted in the suction side of fuel oil transfer, feed and booster pumps. Immediately after the pumps, screw-down non-return valves are to be fitted at the discharge lines.

2.5.6 The power supply of all independently driven fuel oil transfer and pressure pumps is to be capable of being stopped from a position outside the compartment in which they are situated, which will always be accessible in the event of fire occurring in the compartment in which they are situated, as well as from the compartment itself.

2.5.7 A number of valves is to be positioned along the pipeline in a well-considered way, between the pumps and the suction and discharge positions in order to ensure that any pump may be overhauled while the system is in operation.

2.5.8 The supply lines to the fuel injection pumps are to be fitted with heatable duplex filters with a change-over valve or with automatic back-flushing filters fitted with a differential pressure control device. The back-flushing intervals of the automatic back-flushing filters are to be manually controlled. For auxiliary engines simplex filters are sufficient.

2.5.9 In order to ensure that the fuel oil will enter the main engine at a proper temperature and viscosity, two mutually independent preheaters are to be installed after the booster pumps.

2.5.10 A viscosity control is to be provided connected with an oil viscosity regulator and local temperature indicators effecting the heating medium supply to the preheaters.

2.5.11 A deaerating valve is to be fitted in the pipeline conveying return fuel oil from the main engine.

2.5.12 At suitable positions, flow meters, pressure and temperature indicators are to be provided.

2.5.13 Provision is to be made for the transfer of fuel oil from any fuel oil storage or settling tank to any other fuel oil storage or settling tank.

2.6 Heating arrangements in fuel oil systems

2.6.1 Heating arrangements are to be installed in fuel oil tanks, wherever there is a need, especially in heavy fuel oil tanks, to bring the viscosity at an acceptable level so that the fuel oil can easily be pumped under the operating conditions depending on the quality of fuel and the operating requirements. It is LHR's discretion to permit not to fit heating systems in storage tanks provided it can be guaranteed that the proposed quality of fuel oil can be pumped under all ambient and environmental conditions.

2.6.2 Where steam is used for heating fuel oil in tanks, heaters or separators drains are to discharge the condensate into an observation tank in a well-lit and accessible position where it can be readily seen whether or not it is free from oil.

2.6.3 The fuel oil pipelines are to be insulated and heated as necessary.

2.6.4 The heating pipes in contact with oil are to be of iron, steel, approved aluminium alloy or approved copper alloy and after being fitted on board are to be tested by hydraulic pressure.

2.6.5 Where electric heating elements are fitted, means are to be provided to ensure that all elements are submerged at all times when electric current is flowing and that their surface temperature cannot exceed 220°C.

2.6.6 Where hot water is used for heating, means are to be provided to detect the presence of oil in the return lines from the heating coils.

2.6.7 Tanks and heaters in which oil is heated are to be provided with suitable means for ascertaining the temperature of the oil. Where thermometers or temperature sensing devices are not fitted in blind pockets, a warning notice, in raised letters, is to be affixed adjacent to the fittings stating " Do not remove unless tank/heater is drained".

2.6.8 Controls are to be fitted to limit oil temperatures in oil storage and service tanks in accordance with 2.1.5 and in oil heaters to the maximum approved operating temperature.

2.6.9 Where it is necessary to preheat injection valves of engines running with heavy fuel oil, the injection valve cooling system is to be provided with additional means for heating.

2.6.10 Heating coils are to be appropriately subdivided or arranged in group with their own valves. Where necessary, suction pipes are to be provided with a heating arrangement. At the tank outlet, heating coils are to be fitted with means of closing together with an upstream device for testing the condensate for oil. Heating coil connections in tanks should normally be welded. The provision of detachable connections is permitted only in exceptional cases. Inside tanks, heating coils are to be supported in such a way that they cannot be subjected to unacceptable stresses due to vibration, particularly at their point of clamping.

2.6.11 The temperature of any heating medium is not to exceed 220°C.

2.7 Fuel oil tanks and general provisions for fuel oil systems safety against fire

2.7.1 The fuel is to be stored in several tanks so that, even in the event of damage to one of the tanks, the fuel will not be entirely lost. No fuel tanks or tanks for the transport of combustible liquids may be arranged forward of the collision bulkhead.

2.7.2 Fuel tanks are not to be located above engines, boilers, turbines and other equipment with a high surface temperature (above 220°C) as far as is practicable. If this is impossible, they may be located in that way only if adequate spill trays are provided below such tanks and they are protected against heat radiation. Surface temperatures of the elements without insulation and lagging will be considered. Oil-tight drip trays, having suitable drainage arrangements, are to be provided for oil tanks which do not form part of the hull structure and at pumps, valves and other fittings, where there is a possibility of leakage.

2.7.3 The fuel oil pipes are to be led, wherever practicable, at a considerable distance from heated surfaces, electrical appliances and switchboards, but where this is impracticable, any detachable pipe connections are to be at a safe distance from them or effectively shielded with suitable drainage arrangements and the pipes are to be led in well-lit and readily visible positions. Drip trays will not be required where pumps, valves and other fittings are placed in special compartments, either inside or outside the machinery space, with approved overall drainage arrangements or for valves which are so positioned that, any leakage will drain directly into the bilges.

2.7.4 Fuel tanks are to be designed as an integral part of the ship's structure. If this is not possible, the tanks shall be located adjacent to an engine room bulkhead and the tank top of the double bottom. The arrangement of free-standing fuel tanks inside engine room is recommended to be avoided. Non-integral tank locations must be specially considered by LHR.

2.7.5 Fuel oil tanks must be subjected to a hydraulic test. They are to be tested by a head of water of 2,5 m above top of tank, or to the level of the load water line, if this line is more than 2,5 m above top of the tank. The test water head is to be at least at the same level with the highest point of the overflow or air pipe.

2.7.6 Reference shall also be made to Part 9, Chapter 1, SECTION 13, 13.2.

2.8 Fuel oil tanks non-integral with the ship's structure

2.8.1 Generally, the minimum thickness of the plating of service, settling and other oil tanks, where they do not form part of the structure of the ship, is to be 5mm, but in the case of very small tanks, the minimum thickness may be 3mm.

2.8.2 For rectangular steel tanks of welded construction, the plate thicknesses are to be not less than those indicated in Table 10.2.1. The stiffeners are to be of approved dimensions.

2.8.3 The dimension given in Table 10.2.1 for the breadth of the panel is the maximum distance allowable between continuous lines of support, which may be stiffeners, wash-plates or the boundary of the tank.

2.8.4 Where necessary, stiffeners are to be provided, and if the length of the stiffener exceeds twice the breadth of the panel, transverse stiffeners are also to be fitted, or, alternatively, tie bars are to be provided between stiffeners on opposite sides of the tank.

Table 10.2.1: Plate thickness of separate fuel oil tanks

Thickness of plate, mm	Heat from bottom of tank to top of overflow pipe, m				
	2,5	3,0	3,7	4,3	4,9
	Breadth of panel, mm				
5	585	525	-	-	-
6	725	645	590	-	-
7	860	770	700	650	-
8	1000	900	820	750	700
10	1280	1140	1040	960	900

2.8.5 On completion, the tanks are to be tested by a head of water equal to the maximum to which the tanks may be subjected, but not less than 2,5 m above the top of the tank.

2.9 Settling and daily service tanks

2.9.1 Two settling tanks are to be provided for heavy fuel oil. The capacity of each settling tank is to be sufficient for at least one day's consumption of all consumers. Deviating arrangements may, in special cases, be approved after special consideration of LHR.

2.9.2 The capacity of the daily service tanks shall be such that, should the treatment plant fail, the supply to all connected consumers can be maintained for at least 8 hours.

2.9.3 Where the overflow pipe of the daily service tank is terminated in the settling tanks, suitable means are to be provided to ensure that no untreated heavy fuel oil can penetrate into the daily service tank in case of overfilling of a settling tank.

2.9.4 Settling and daily service tanks are to be provided with valves of the self-closing type, fitted at the bottom of the tanks, for the effective drainage of them. Suitable provision is to be made for collecting the oily discharge.

2.10 Alternative carriage of fuel oil and water ballast

2.10.1 Where it is intended to carry fuel oil and water ballast in the same compartment alternatively, the valves connecting the suction pipes of these compartments with the ballast pump and those connecting them with the fuel oil transfer pump, are to be so arranged that the oil may be pumped from any one compartment by the fuel oil pump at the same time as the ballast pump is being used on any other compartment. In passenger ships the arrangement will require to be specially approved.

2.10.2 Where settling or service tanks are fitted, each having a capacity sufficient to permit 12 hours normal service without replenishment, the above requirement may be dispensed with.

2.10.3 Attention is drawn to the statutory regulations issued by National Authorities in connection with the International Convention for the Prevention of Pollution at Sea by Oil, 1973/1978.

2.11 Deep tanks for the alternative carriage of oil water ballast or dry cargo

2.11.1 In the case of deep tanks which can be used for the carriage of fuel oil, cargo oil, water ballast or dry cargo, provision is to be made for blank flanging the oil and water ballast filling and suction pipes, also for retaining the steam heating coils in place, when the tank is used for dry cargo. When the tanks are used for oil or water ballast, the bilge suction pipes are to be blank flanged.

2.11.2 If the deep tanks are connected to an overflow system, the arrangements are to be such that liquid or vapour from other tanks cannot enter the deep tanks when dry cargo is carried in them.

2.12 Fuel oil burning installations

2.12.1 Where steam is required for the main propulsion machinery engines, for auxiliary machinery for essential services, or for heating of heavy fuel and is generated by burning fuel oil under pressure, there are to be not less than two oil burning units, each unit comprising a pressure pump, a suction filter, a discharge filter and a heater. For auxiliary boilers, a single oil burning unit may be accepted, provided that alternative means, such as an exhaust gas boiler or a composite boiler, are available for supplying steam for essential purposes.

2.12.2 In two-unit installations, each unit is to be capable of supplying fuel, for generating all the steam required for essential services.

2.12.3 In installations of three or more units, the capacities and arrangements of the units are to be such that all the steam required for essential services can be maintained with any one unit out of action.

2.12.4 Unit pressure pumps are to be entirely separate from the feed, bilge or ballast system.

2.12.5 In systems where oil is fed to the burners by gravity, duplex filters are to be fitted in the supply pipeline to the burners and so arranged that one filter can be opened up when the other is in use.

2.12.6 A starting-up fuel oil unit, including an auxiliary heater and hand pump, or other suitable starting-up device, which does not require power from shore, is to be provided.

2.12.7 Alternatively, where auxiliary machinery requiring compressed air or electric power is used to bring the boiler plant into operation, the arrangements for starting such machinery are to comply with Part 5, Chapter 2, SECTION 8.

2.12.8 Where burners are provided with steam purging and/or atomizing connections, the arrangements are to be such that fuel oil cannot find its way into the steam system in the event of leakage.

2.12.9 The burner arrangements are to be such that a burner cannot be withdrawn unless the fuel oil supply to that burner is shut off, and that the oil cannot be turned on unless the burner has been correctly coupled to the supply line.

2.12.10 A quick-closing master valve is to be fitted to the oil supply to each boiler manifold, suitably located so that the valve can be readily operated in an emergency, either directly or by means of remote control, having regard to the machinery arrangements and location of controls.

2.12.11 In the case of top-fired boilers, means are to be provided so that, in the event of flame failure, the fuel oil supply to the burners is shut off automatically, and audible and visual warnings are given. Any proposal to depart from this requirement in the case of small auxiliary top-fired boilers, will be specially considered.

2.12.12 Provision is to be made, by suitable non-return arrangements, to prevent oil from spill systems being returned to the burners when the oil supply to these burners has been shut off.

2.12.13 For alternately fired furnaces of boilers using exhaust gases and fuel oil, the exhaust gas inlet pipe is to be provided with an isolating device and interlocking arrangements whereby fuel oil can only be supplied to the burners when the isolating device is closed to the boiler.

2.13 Oil-fired galleys

2.13.1 The fuel oil tank is to be located outside the galley and is to be fitted with approved means of filling and venting.

2.13.2 The fuel supply to the burners is to be controlled from a position which will always be accessible in the event of a fire occurring in the galley.

2.13.3 The galley is to be well ventilated.

2.13.4 Where liquefied petroleum gas is used, similar provisions are to be made.

2.14 Ventilation

2.14.1 The spaces in which the fuel oil burning appliances and the fuel oil settling and service tanks are located, are to be well ventilated and easy to access.

2.15 Boiler insulation and air circulation in boiler room

2.15.1 The boilers are to be suitably lagged. The clearance spaces between the boilers and tops of the double bottom tanks, and between the boilers and the sides of the storage tanks in which fuel oil and cargo oil is carried, are to be adequate for the free circulation of the air necessary to keep the temperature of the stored oil sufficiently below its flash point.

2.15.2 Where water tube boilers are installed, there is to be a space of at least 760mm between the tank top and the underside of the pans forming the bottom of the combustion space.

2.16 Requirements concerning use of crude oil or slops as fuel for tanker boilers (IACS UR M24 Rev.1 (1976))

2.16.1 In tankers crude oil or slops may be used as fuel for main or auxiliary boilers according to the following requirements. For this purpose all arrangement drawings of a crude oil installation with pipeline layout and safety equipment are to be submitted for approval in each case.

2.16.2 Crude oil or slops may be taken directly from cargo tanks or flow slop tanks or from other suitable tanks. These tanks are to be fitted in the cargo tank area and are to be separated from non-gas-dangerous areas by means of cofferdams with gas-tight bulkheads.

2.16.3 The construction and workmanship of the boilers and burners are to be proved to be satisfactory in operation with crude oil.

The whole surface of the boilers shall be gas-tight separated from the engine room. The boilers themselves are to be tested for gas-tightness before being used. The whole system of pumps, strainers, separators and heaters, if any, shall be fitted in the cargo pump room or in another room, to be considered as dangerous, and separated from engine and boiler room by gas-tight bulkheads. When crude oil is heated by steam or hot water the outlet of the heating coils should be led to a separate observation tank installed together with above mentioned components. This closed tank is to be fitted with a venting pipe led to the atmosphere in a safe position according to the rules for tankers and with the outlet fitted with a suitable flame proof wire gauze of corrosion resistant material which is to be easily removable for cleaning.

2.16.4 Electric, internal combustion and steam (when the steam temperature is higher than 220°C) prime movers of pumps, of separators (if any), etc., shall be fitted in the engine room or in another non-dangerous room.

Where drive shafts pass through pump room bulkhead or deck plating, gas-tight glands are to be fitted.

The glands are to be efficiently lubricated from outside the pump room.

2.16.5 Pumps shall be fitted with a pressure relief bypass from delivery to suction side and it shall be possible to stop them by a remote control placed in a position near the boiler fronts or machinery control room and from outside the engine room.

2.16.6 When it is necessary to preheat crude oil or slops, their temperature is to be automatically controlled and a high temperature alarm is to be fitted.

2.16.7 The piping for crude oil or slops and the draining pipes for the tray defined in 2.16.9 are to have a thickness according to Table 10.2.2.

Table 10.2.2:

External diameter of pipes, d_e (mm)	Thickness, t (mm)
$d_e \leq 82,5$	$t \geq 6,3$
$88,9 < d_e \leq 108$	$t \geq 7,1$
$114,3 < d_e \leq 139,7$	$t \geq 8,0$
$152,4 \leq d_e$	$t \geq 8,8$

Their connections (to be reduced to a minimum) are to be of the heavy flange type. Within the engine room and boiler room these pipes are to be fitted within a metal duct, which is to be gas-tight and tightly connected to the fore bulkhead separating the pump room and to the tray. This duct (and the enclosed piping) is to be fitted at a distance from the ship's side of at least 20% of the vessel's beam amidships and be at an inclination rising towards the boiler so that the oil naturally returns towards the pump room in the case of leakage or failure in delivery pressure. It is to be fitted with inspection openings with gas-tight doors in way of connections of pipes within it, with an automatic closing drain-trap placed on the pump room side, set in such a way as to discharge leakage of crude oil into the pump room. In order to detect leakages, level position indicators with relevant alarms are to be fitted on the drainage tank defined in 2.16.9. Also a vent pipe is to be fitted at the highest part of the duct and is to be led to the open in a safe position. The outlet is to be fitted with a suitable flame proof wire gauze of corrosion resistant material which is to be easily removable for cleaning. The duct is to be permanently connected to an approved inert gas system or steam supply in order to make possible:

– injection of inert gas or steam in the duct in case of fire or leakage
– purging of the duct before carrying out work on the piping in case of leakage.

2.16.8 In way of the bulkhead to which the duct defined in 2.16.7 is connected, delivery and return oil pipes are to be fitted on the pump room side, with shut-off valves remotely controlled from a position near the boiler fronts or from the machinery control room. The remote-control valves should be interlocked with the hood exhaust fans (defined in 2.16.10) to ensure that whenever crude oil is circulating the fans are running.

2.16.9 Boilers shall be fitted with a tray or gutterway of a height to the satisfaction of LHR and be placed in such a way as to collect any possible oil leakage from burners, valves and connections. Such a tray or gutterway shall be fitted with a suitable flame proof wire gauze, made of corrosion resistant material and easily dismountable for cleaning. Delivery and return oil pipes shall pass through the tray or gutterway by means of a tight penetration and shall then be connected to the oil supply manifolds. A quick closing master valve is to be fitted on the oil supply to each boiler manifold. The tray or gutterway shall be fitted with a draining pipe discharging into a collecting tank in pump room. This tank is to be fitted with a venting pipe led to the open in a safe position and with the outlet fitted with wire gauze made of corrosion resistant material and easily dismountable for cleaning. The draining pipe is to be fitted with arrangements to prevent the return of gas to the boiler or engine room.

2.16.10 Boilers shall be fitted with a suitable hood placed in such a way as to enclose as much as possible of the burners, valves and oil pipes, without preventing, on the other side, air inlet to burner register. The hood, if necessary, is to be fitted with suitable doors placed in such a way as to enable inspection of and access to oil pipes and valves placed behind it. It is to be fitted with a duct leading to the open in a safe position, the outlet of which is to be fitted with a suitable flame wire gauze, easily dismountable for cleaning. At least two mechanically driven exhaust fans having spark proof impellers are to be fitted so that the pressure inside the hood is less than that in the boiler room. The exhaust fans are to be connected with automatic change over in case of stoppage or failure of the one in operation. The exhaust fan prime movers shall be placed outside the duct and a gas-tight bulkhead penetration shall be arranged for the shaft. Electrical equipment installed in gas dangerous areas or in areas which may become dangerous (i.e. in the hood or duct in which crude-oil piping is placed) is to be of certified safe type as required by LHR.

2.16.11 When using fuel oil for delivery to and return from boilers fuel oil burning units in accordance with LHR's Rules shall be fitted in the boiler room. Fuel oil delivery to, and returns from, burners shall be effected by means of a suitable mechanical interlocking device so that running on fuel oil automatically excludes running on crude oil or vice versa.

2.16.12 The boiler compartments are to be fitted with a mechanical ventilation plant and shall be designed in such a way as to avoid the formation of gas pockets. Ventilation is to be particularly efficient in way of electrical plants and machinery and other plants which may generate sparks. These plants shall be separated from those for service of other compartments and shall be in accordance with LHR's requirements.

2.16.13 A gas detector plant shall be fitted with intakes in the duct defined in 2.16.7, in the hood duct (downstream of the exhaust fans in way of the boilers) and in all zones where ventilation may be reduced. An optical warning device is to be installed near the boiler fronts and in the machinery control room. An acoustical alarm, audible in the machinery space and control room, is to be provided.

2.16.14 Means are to be provided for the boiler to be automatically purged before firing.

2.16.15 Independent of the fire extinguishing plant as required by LHR's Rules, an additional fire extinguishing plant is to be fitted in the engine and boiler rooms in such a way that it is possible for an approved fire extinguishing medium to be directed on to the boiler fronts and on to the tray defined in 2.16.9. The emission of extinguishing medium should automatically stop the exhaust fan of the boiler hood (see 2.16.8).

2.16.16 A warning notice must be fitted in an easily visible position near the boiler front. This notice must specify that when an explosive mixture is signalled by the gas detector plant defined in 2.16.13 the watchkeepers are to immediately shut off the remote controlled valves on the crude oil delivery and return pipes in the pump room, stop the relative pumps, inject inert gas into the duct defined in 2.16.7 and turn the boilers to normal running on fuel oil.

2.16.17 One pilot burner in addition to the normal burning control is required.

SECTION 3 Steam lines

3.1 General

3.1.1 Steam lines are to be laid out and arranged so that important consumers can be supplied with steam from every main boiler as well as from a standby boiler or boiler for emergency operation.

3.1.2 Important consumers are all consuming units for the propulsion, manoeuvrability, safety of the ship as well as the important auxiliary machines.

3.2 Pipe layout

3.2.1 Steam lines are to be so installed and supported that expected stresses due to thermal expansion, external loads and shifting of the supporting structure under both normal and interrupted service conditions, will be safely compensated. See also Part 5, Chapter 9, SECTION 2, 2.8.3.

3.2.2 Steam lines are to be so installed that water pockets are avoided.

3.2.3 Means are to be provided for the reliable drainage of the piping system even when the ship is in normal trim and has a list of up to 5 degrees.

3.2.4 Pipe penetrations through bulkheads and decks are to be insulated to prevent heat conduction.

3.2.5 Steam lines are to be effectively insulated to prevent heat losses.

3.2.6 At points where there is a possibility of contact, the surface temperature of the insulated steam lines may not exceed 55°C.

3.2.7 Steam pipes are not to be led through spaces which may be used for cargo, but where it is impracticable to avoid this, plans are to be submitted for consideration. Pipe joints are to be as few as practicable and preferably butt-welded.

3.2.8 Wherever necessary, machines and devices in steam systems are to be protected against solid particles by steam strainers.

3.2.9 Steam lines for superheated steam at above 500°C are to be provided with means for verification that the stresses due to expansion of the piping will not exceed the allowable predetermined limits.

3.2.10 Where a system can be entered from a system with higher pressure, pressure gauges and relief valves of adequate capacity are to be fitted after the pressure reducer. Where low pressure air serves control or safety systems, an equivalent stand-by pressure reducing system should be provided.

SECTION 4 Sea water cooling systems

4.1 General

4.1.1 Adequate sea water supply is to be used as the cooling medium in lubricating oil coolers, charge air coolers, jacket cooling water coolers and other coolers employed in servicing main engines or other machinery systems.

4.2 Sea chests and sea valves

4.2.1 Not less than two sea chests are to be provided, located as low as possible on each side of the ship. To ensure efficient sea-water supply when the ship operates in shallow waters, an additional sea chest located in higher position is to be provided.

4.2.2 The size of sea chest and the relative piping arrangement is to be such as to allow the total sea water supply to be taken only from one sea chest.

4.2.3 Each sea-chest is to be provided with an effective vent to ensure free flow of water. The installation of a vent pipe of at least 32mm internal diameter, fitted with a valve on the chest, extending above the bulkhead deck or of adequately dimensioned ventilation slots in the shell plating, may be approved as a vent.

4.2.4 Sea chests should have well designed gratings to ensure the adequate inlet of water, arranged in a way to be easily cleaned during dry docking. The clear area of gratings is in no case to be less than two times the total area of the valves served by the sea chests. To provide effective clearing of sea chest gratings during operation, steam or compressed air connections are to be installed with valves fitted directly to the sea chest. Compressed air pressure for blowing through sea chest gratings may exceed 2 bar only if the sea chests are constructed for higher pressures.

4.2.5 Sea valves are to be so arranged that they can be operated from 460mm above platform.

4.2.6 Discharge pipes for sea water cooling systems are to be fitted with a non-return valve at the shell plating.

4.3 Sea water cooling pumps for ships with diesel engine

4.3.1 Main and standby cooling water pumps are to be employed in sea water cooling systems in diesel engines of ships. The main cooling water pump may be attached to the propulsion plant only if it is guaranteed that the attached pump is of sufficient capacity for the cooling water required by main and auxiliary engines over the whole speed range of the propulsion plant. The standby cooling water pump is to be driven independently of the main engine. Main and standby cooling water pumps are each to be of sufficient capacity to meet the maximum cooling water requirements of the plant.

4.3.2 Ballast pumps or other suitable sea water pumps may be used as standby cooling water pumps.

4.3.3 Where cooling water is supplied by means of a scoop, the main and standby cooling water pumps are to be of a capacity which will ensure reliable operation of the plant under partial load conditions. The main cooling water pump is to be automatically started as soon as the speed falls below that required for the operation of the scoop.

4.3.4 Where more than one engine is employed, each with its own pump, a complete spare pump of each type may be accepted instead of standby pumps, provided that the main sea water cooling pumps are so arranged that they can be replaced with the means available on board within reasonable time.

4.4 Steam turbine plants

4.4.1 Steam turbine plants are to be provided with a main and a standby cooling water pump. The main cooling water pump is to be of sufficient capacity to supply the maximum cooling water requirements of the turbine plant. The capacity of the standby cooling water pump is to be such as to ensure reliable operation of the plant also during astern operation.

4.4.2 Where cooling water is supplied by means of a scoop, the main cooling water pump is to be of sufficient capacity for the cooling water requirements of the turbine plant under conditions of maximum astern output. The main cooling water pump is to start automatically as soon as the speed falls below that required for the operation of the scoop.

4.5 Cooling water for auxiliary engines

4.5.1 Where each auxiliary engine is fitted with a cooling water pump, standby means of cooling need not be provided. Where, however, a group of auxiliaries is supplied with cooling water from a common system, a standby cooling water pump is to be provided for this system.

4.6 Piping layout

4.6.1 The sea inlets may be connected to a suction line available to main and standby pumps.

4.6.2 The cooling water suction pipes are to be provided with strainers which can be cleared without interruption to the cooling water supply.

4.6.3 Where cooling water pumps can develop a pressure head greater then the design pressure of the system, they are to be provided with relief valves on the pump discharge to effectively limit the pump discharge pressure to the design pressure of the system.

4.7 Selection of standby pumps

4.7.1 When selecting a pump for standby purposes, consideration is to be given to the maximum pressure which it can develop if the overboard discharge valve is partly or fully closed and, when necessary, condenser doors, water boxes, etc. are to be protected by an approved device against inadvertent over-pressure.

SECTION 5 Fresh water-cooling systems

5.1 General

5.1.1 Fresh water-cooling systems are to be so arranged that the engines can be sufficiently cooled under all operating conditions.

5.2 Coolers

5.2.1 The coolers of cooling water systems, engines and equipment are to be so constructed to ensure that the specified cooling water temperatures can be maintained under all operating conditions. Cooling water temperatures are to be adjusted to meet the requirements of engines and equipment.

5.2.2 Valves are to be provided at the inlet and outlet of all coolers.

5.2.3 Every heat exchanger and cooler is to be provided with a vent and a drain.

5.3 Expansion tanks

5.3.1 Expansion tanks are to be arranged at sufficient height for every cooling water circuit to prevent cavitation of the pump.

5.3.2 Expansion tanks are to be fitted with filling connection, vent pipes, water level indicators and a drain arrangement.

5.4 Fresh water-cooling pumps

5.4.1 Main and standby cooling water pumps are to be provided for each fresh water-cooling system.

5.4.2 Main cooling water pumps may be driven directly by the main or auxiliary engines which they are intended to cool, provided that a sufficient supply of cooling water is assured under all operating conditions.

5.4.3 Standby cooling water pumps are to be driven independently of main engines. They are to have the same capacity as main cooling water pumps.

5.4.4 Main engines are to be fitted with at least one main and one standby cooling water pump. Where according to the construction of the engines, more than one water cooling circuit is necessary, a standby pump is to be fitted for each main cooling water pump.

5.4.5 A standby fresh water pump need not be fitted if there are suitable emergency connections from a salt water system. Where each auxiliary is fitted with a cooling water pump standby means of cooling need not be provided. Where, however, a group of auxiliaries is supplied with cooling water from a common system, a standby cooling water pump is to be provided for this system. This pump is to be connected ready for immediate use and maybe a suitable general service pump.

SECTION 6 Lubricating oil systems

6.1 General

6.1.1 Lubricating oil systems are to be so constructed to ensure reliable lubrication over the whole range of speed and to ensure adequate heat transfer while in starting and manoeuvering.

6.1.2 All the necessary equipment (purifiers, automatic back-flushing filters, filters etc.) for the effective treatment of lubricating oil is to be provided.

6.1.3 All main and auxiliary engines and turbines intended for essential services are to be provided with means of indicating the lubricating oil pressure in them.

6.1.4 The arrangements for lubricating bearings and for draining crankcase and other oil sumps of main and auxiliary engines, gearcases, electric generators, motors, and other running machinery, are to be so designed that lubricants will remain efficient with the ship inclined under the conditions referred in Part 5, Chapter 1.

6.2 Lubricating oil pumps to main engines

6.2.1 Main lubricating oil pumps are to be independently or main engine driven. Where they are driven by the main engine account should be taken to comply with requirements of 6.1.1.

6.2.2 A standby lubricating oil pump is to be provided where the following conditions apply (see also Part 5, Chapter 2).

1) The lubricating oil pump is independently driven and the total output of the main engine(s) exceeds 370 kW (500 shp).
2) One main engine with a built-in lubricating oil pump is fitted and the output of the main engine exceeds 370 kW (500 shp).
3) Two or more main engines each with a built-in lubricating pump are fitted and the output of each engine exceeds 370 kW(500 shp).

6.2.3 The standby pumps is to be of sufficient capacity to maintain the supply of oil for normal conditions with any one pump out of action. The pump is to be fitted and connected ready for immediate use except that where the conditions mentioned in 6.2.2(3) apply, a complete spare pump may be accepted.

6.2.4 Independently driven pumps of rotary type are to be fitted with a non-return valve on the discharge side of the pump.

6.2.5 The power supply to all independently driven lubricating oil pumps is to be capable of being stopped from a position located outside the space and always accessible in the event of fire occurring in the compartment in which the lubricating oil pumps are situated, as well as from the compartment itself.

6.2.6 A relief valve in close circuit is to be fitted on the pump discharge if the pump is capable of developing a pressure exceeding the design pressure of the system.

6.2.7 For propulsion turbines and propulsion turbo-generators a suitable emergency supply of lubricating oil is to be arranged to come automatically into use in the event of a failure of the supply from the pump. The emergency supply may be obtained from a gravity tank containing sufficient oil to maintain adequate lubrication for not less than 6 minutes and, in the case of propulsion turbo-generators, until the unloaded turbine comes to rest from its maximum rated running speed. Alternatively, the supply may be provided by the standby pump or by an emergency pump. These pumps are to be so arranged that their availability is not affected by a failure in the power supply.

6.2.8 Where more than one diesel generator is available, stand-by pumps are not required. Where only one generator is available (e.g. on turbine-driven vessels where the diesel generator is needed for start-up etc.) a complete pump is to be carried on board.

6.2.9 Turbogenerators and turbines used for driving important auxiliaries such as boiler feed water pumps etc. are to be equipped with a main pump and an independent auxiliary pump. The auxiliary pump is to be designed to ensure a sufficient supply of lubricating oil during the start-up and run-down operation.

6.3 Piping layout

6.3.1 Extreme care is to be taken to ensure that lubricating oil pipes and fittings , before installation are free from scale, sands, metal particles and other foreign matter.

6.3.2 Outlet valves on lubricating oil service tanks, other than double bottom tanks, situated in machinery spaces are to be capable of being closed locally and from positions outside the space which will always be accessible in the event of fire occurring in these spaces. Remote controls need only be fitted to outlet valves which are open in normal service and are not required for other outlets such as those on storage tanks. Instructions for closing the valves are to be indicated at the valves and at the remote-control positions.

6.3.3 In the case of very small tanks, consideration will be given to the omission of remote controls.

6.3.4 Where an engine lubricating oil drain tank extends to the bottom shell plating in ships that are required to be provided with a double bottom, a valve is to be fitted in the drain pipe between the engine casing and the double bottom tank. This valve is to be capable of being closed from an accessible position above the level of the lower platform.

6.3.5 The suction connections of lubricating oil pumps to lubrication oil tanks are to be located as far away as possible from drain pipes.

6.3.6 The gravity tank is to be fitted with an overflow pipe which leads to the drain tank. Arrangements are to be made for observing the flow of excess oil in the overflow pipe.

6.4 Filters

6.4.1 Lubricating oil filters are to be arranged in the pressure lines of the pumps. The fineness and the size of the filters are to be in accordance with the requirements of the engine. The filters are to be capable of being cleared without stopping the engine or reducing the supply of filtered oil to the engine. Proposals for an automatic by-pass for emergency purposes in high-speed engines are to be submitted for special consideration.

SECTION 7 Compressed air lines

7.1 General

7.1.1 Pressure lines connected to air compressors are to be fitted with non-return valves at the compressor outlet.

7.1.2 Efficient oil and water traps are to be provided in the filling lines of compressed air receivers.

7.1.3 Typhoons are to be connected to at least two compressed air receivers.

7.1.4 A safety valve is to be fitted behind each pressure-reducing valve. Each compressor and each air-receiver is to be fitted with a relief valve.

7.1.5 The starting air line to each engine is to be fitted with a non-return valve and a drain.

SECTION 8 Exhaust gas lines

8.1 Pipe layout

8.1.1 Exhaust gas pipes from engines are to be installed separately from each other with regard to structural fire protection. The same applies to boiler exhaust gas pipes.

8.1.2 When laying out and suspending the lines, efficient means for their thermal expansion are to be fitted.

8.1.3 Engine exhaust pipes are to be fitted with effective silencers provided with an inspection opening.

8.1.4 Exhaust lines and silencers are to be provided with suitable water drains of adequate size.

8.1.5 Exhaust gas lines, silencers and exhaust gas boilers, are to be effectively insulated to prevent the ignition of any combustible material which could possibly come in contact with them. Insulating materials must be incombustible. Exhaust gas lines inside engine rooms are to be provided with a metal sheathing or other approved type impervious to oil.

8.1.6 Where exhaust gas lines discharge near water level, provisions are to be taken to prevent water from entering the engines.

8.2 Shell Type Exhaust Gas Heated Economizers That May Be Isolated From The Steam Plant System (IACS UR P6 Rev.1 (2015))

8.2.1 Application

This paragraph is applicable to shell type exhaust gas heated economisers that are intended to be operated in a flooded condition and that may be isolated from the steam plant system. All shell type exhaust gas heated economisers that may be isolated from the steam plant system in a flooded condition and which are fitted on board ships contracted for construction on or after 1 January 2007 are to comply with this paragraph.

8.2.2 Design and Construction

Design and construction of shell type economisers are to pay particular attention to the welding, heat treatment and inspection arrangements at the tube plate connection to the shell.

8.2.3 Pressure Relief

a) Where a shell type economizer is capable of being isolated from the steam plant system, it is to be provided with at least one safety valve, and when it has a total heating surface of 50 m^2 or more, it is to be provided with at least two safety valves in accordance with the LHR's requirements.
b) To avoid the accumulation of condensate on the outlet side of safety valves , the discharge pipes and/or safety valve housings are to be fitted with drainage arrangements from the lowest part, directed with continuous fall to a position clear of the economizer where it will not

pose threats to either personnel or machinery. No valves or cocks are to be fitted in the drainage arrangements.

c) Full details of the proposed arrangements to satisfy 8.2.3(a) to 8.2.3(b) are to be submitted for approval.

8.2.4 Pressure Indication

Every shell type economizer is to be provided with a means of indicating the internal pressure. A means of indicating the internal pressure is to be located so that the pressure can be easily read from any position from which the pressure may be controlled.

8.2.5 Lagging

Every shell type economizer is to be provided with removable lagging at the circumference of the tube end plates to enable ultrasonic examination of the tube plate to shell connection.

8.2.6 Feed Water

Every economizer is to be provided with arrangements for pre-heating and de-aeration, addition of water treatment or combination thereof to control the quality of feed water to within the manufacturer’s recommendations.

8.2.7 Operating Instructions

The manufacturer is to provide operating instructions for each economizer which is to include reference to:

- Feed water treatment and sampling arrangements.
- Operating temperatures – exhaust gas and feed water temperatures.
- Operating pressure.
- Inspection and cleaning procedures.
- Records of maintenance and inspection.
- The need to maintain adequate water flow through the economizer under all operating conditions.
- Periodical operational checks of the safety devices to be carried out by the operating personnel and to be documented accordingly.
- Procedures for using the exhaust gas economizer in the dry condition.
- Procedures for maintenance and overhaul of safety valves.

CHAPTER 11 Steering Gear

CONTENTS

Steering Gear (IACS M42 Rev.6 (2022))

Preamble

In addition to the requirements contained in SOLAS II-1/29 and SOLAS II-1/30 as well as related Guidelines (see Annex 2 of IMCO document MSC XLV/4), the following requirements apply to new ocean-going vessels of 500 gross tonnage and upwards. These requirements may be applied to other vessels at the discretion of LHR.

SECTION 1 Plans and specifications

1.1 Documents for approval

1.1.1 Before starting construction, all relevant plans and specifications are to be submitted to LHR for approval.

SECTION 2 Definitions

2.1 Main definitions

2.1.1

(1) Steering gear control system means the equipment by which orders are transmitted from the navigating bridge to the steering gear power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables. Steering gear control system is also understood to cover "the equipment required to control the steering gear power actuating system".
(2) Main steering gear means the machinery, rudder actuator(s), the steering gear power units, if any, and ancillary equipment and the means of applying torque to the rudder stock (e.g. tiller or quadrant) necessary for effecting movement of the rudder for the purpose of steering the ship under normal service conditions.
(3) Steering gear power unit means:
  (a) in the case of electric steering gear, an electric motor and its associated electrical equipment;
  (b) in the case of electrohydraulic steering gear, an electric motor and its associated electrical equipment and connected pump;
  (c) in the case of other hydraulic steering gear, a driving engine and connected pump.
(4) Auxiliary steering gear means the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not including the tiller, quadrant or components serving the same purpose.
(5) Power actuating system means the hydraulic equipment provided for supplying power to turn the rudder stock, comprising a steering gear power unit or units, together with the associated pipes and fittings, and a rudder actuator. The power actuating systems may share common mechanical components. i.e. tiller, quadrant and rudder stock, or components serving the same purpose.
(6) Maximum ahead service speed means the greatest speed which the ship is designed to maintain in service at sea at her deepest sea going draught at maximum RPM and corresponding engine MCR.
(7) Rudder actuator means the component which converts directly hydraulic pressure into mechanical action to move the rudder.
(8) Maximum working pressure means the maximum expected pressure in the system when the steering gear is operated to comply with the SOLAS II-1/29.3.2.
(9) Hydraulic locking means all situations where two hydraulic systems (usually identical) oppose each other in such a way that it may lead to loss of steering. It can either be caused by pressure in the

two hydraulic systems working against each other or by hydraulic "by-pass" meaning that the systems puncture each other and cause pressure drop on both sides or make it impossible to build up pressure.

SECTION 3 Power piping arrangements

3.1 General

3.1.1 The power piping for hydraulic steering gears is to be arranged so that transfer between units can be readily effected.

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

3.1.3 For all vessels with non-duplicated actuators, isolating valves are to be fitted at the connection of pipes to the actuator, and are to be directly fitted on the actuator.

3.1.4 Arrangements for bleeding air from the hydraulic system are to be provided where necessary.

3.1.5 Piping, joints, valves, flanges and other fittings are to comply with the LHR's requirements for Class 1 components. The design pressure is to be in accordance with 6.1.8.

SECTION 4 Rudder Angle Limiters

4.1 General

4.1.1 Power-operated 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.

SECTION 5 Materials

5.1 Strength requirements

5.1.1 Ram cylinders; pressure housings of rotary vane type actuators; hydraulic power piping valves, flanges and fittings; and all steering gear components transmitting mechanical forces to the rudder stock (such as tillers, quadrants, or similar components) should be of steel or other approved ductile material, duly tested in accordance with the requirements of LHR. In general, such material should not have an elongation of less than 12% nor a tensile strength in excess of 650N/mm². Grey cast iron may be accepted for redundant parts with low stress levels, excluding cylinders, upon special consideration.

SECTION 6 Design

6.1 General principles

6.1.1 The construction should be such as to minimize local concentrations of stress.

6.1.2 Welds

a) The welding details and welding procedures should be approved.
b) All welded joints within the pressure boundary of a rudder actuator or connecting parts transmitting mechanical loads should be full penetration type or of equivalent strength.

6.1.3 Oil seals

a) Oil seals between non-moving parts, forming part of the external pressure boundary, should be of the metal upon metal type or of an equivalent type.
b) Oil seals between moving parts, forming part of the external pressure boundary, should be duplicated, so that the failure of one seal does not render the actuator inoperative. Alternative arrangements providing equivalent protection against leakage may be accepted at the discretion of the Administration.

6.1.4 All steering gear components transmitting mechanical forces to the rudder stock, which are not protected against overload by structural rudder stops or mechanical buffers, are to have a strength at least equivalent to that of the rudder stock in way of the tiller (see Part 3, Chapter 11, SECTION 2). Relevant calculations must be submitted for consideration.

6.1.5 For piping, joints, valves, flanges and other fittings see 3.1.4 of this SECTION.

6.1.6 Rudder actuators other than those covered by the SOLAS II-1/29.17 and relating Guidelines should be designed in accordance with Class 1 pressure vessels (notwithstanding any exemptions for hydraulic cylinders).

6.1.7 In application of such rules the permissible primary general membrane stress is not to exceed the lower of the following values:

$$\frac{\sigma_B}{A} \ or \ \frac{\sigma_y}{B}$$

where:

σ_B = specified minimum tensile strength of material at ambient temperature [N/mm²]

σ_y = specified minimum yield stress or 2% proof stress of the material, at ambient temperature [N/mm²]

A and B are given by Table 11.6.1.

Table 11.6.1:

	Steel	Cast Steel	Nodular Cast iron
A	3,5	4	5
B	1,7	2	3

6.1.8 The design pressure is to be at least equal to the greater of the following:
1) 1,25 times the maximum working pressure,
2) the relief valve setting.

6.1.9 Accumulators, if any are to comply with LHR's requirements for pressure vessels.

SECTION 7 Dynamic loads for fatigue and fracture mechanic analysis

7.1 Specification

7.1.1 The assumed dynamic loading in the fatigue and fracture mechanics analysis considering SOLAS II-1/29.2.2 and SOLAS II-1/29.17.1 as well as relating Guidelines, will be established at the discretion of LHR. Both the case of high cycle and cumulative fatigue are to be considered.

SECTION 8 Hoses

8.1 Requirements

8.1.1 Hose assemblies of type approved by LHR may be installed between two points where flexibility is required but should not be subjected to torsional deflection (twisting) under normal operating conditions. In general, the hose should be limited to the length necessary to provide for flexibility and for proper operation of machinery.

8.1.2 Hoses should be high pressure hydraulic hoses according to recognized standards and suitable for the fluids, pressures, temperatures and ambient conditions in question.

8.1.3 Burst pressure of the hoses should not be less than four times the design pressure.

SECTION 9 Relief valves

9.1 Requirements

9.1.1 Relief valves for protecting any part of the hydraulic system which can be isolated, as required by SOLAS II-1/29.2.3 should comply with the following:

1) The setting pressure should not be less than 1,25 times the maximum working pressure.
2) The minimum discharge capacity of the relief valve(s) should not be less than the total capacity of the pumps, which can deliver through it (them), increased by 10%.

Under such conditions the rise in pressure should not exceed 10% of the setting pressure. In this regard, due consideration should be given to extreme foreseen ambient conditions in respect of oil viscosity. LHR may require, for the relief valves, discharge capacity tests and/or shock tests.

SECTION 10 Electrical installations

10.1 Requirements

10.1.1 Electrical installations should comply with the requirements of Part 6 and Part 8 of these Rules.

SECTION 11 Alternative source of power

11.1 General

11.1.1 Where the alternative power source required by the SOLAS II-1/29.14 is a generator, or an engine driven pump, automatic starting arrangements are to comply with the requirements relating to the automatic starting arrangements of emergency generators.

SECTION 12 Monitoring and alarm systems

12.1 Requirements

12.1.1 Monitoring and alarm systems, including the rudder angle indicators, should be designed, built and tested to the satisfaction of LHR (see also Part 8, Chapter 1).

12.1.2 Where hydraulic locking, caused by a single failure, may lead to loss of steering, an audible and visual alarm, which identifies the failed system, shall be provided on the navigating bridge.

This alarm should be activated whenever:

– position of the variable displacement pump control system does not correspond with given order, or
– incorrect position of three-way full flow valve or similar in constant delivery pump system is detected.

SECTION 13 Operating instructions

13.1 General

13.1.1 Where applicable, following standard signboard should be fitted at a suitable place on steering control post on the bridge or incorporated into operating instruction on board:

CAUTION
IN SOME CIRCUMSTANCES WHEN 2 POWER UNITS ARE RUNNING SIMULTANEOUSLY THE RUDDER MAY NOT RESPOND TO HELM. IF THIS HAPPENS STOP EACH PUMP IN TURN UNTIL CONTROL IS REGAINED.

The above signboard is related to steering gears provided with two identical units intended for simultaneous operation, and normally provided with either their own control systems or two separate (partly or mutually) control systems which are/may be operated simultaneously.

SECTION 14 Testing and trials

14.1 Testing

14.1.1 The requirements of LHR relating to the testing of Class 1 pressure vessels, piping, and relating fittings including hydraulic tenting apply.

14.1.2 A power unit pump is to be subjected to a type test. The type test shall be for a duration of not less than 100 hours, the test arrangements are to be such that the pump may run in idling conditions, and at maximum delivery capacity at maximum working pressure. During the test, idling periods are to be alternated with periods at maximum delivery capacity at maximum working pressure. The passage from one condition to another should occur at least as quickly as on board. During the whole test no abnormal heating, excessive vibration or other irregularities are permitted. After the test, the pump should be disassembled and inspected. Type tests may be waived for a power unit which has been proven to be reliable in marine service.

14.1.3 All components transmitting mechanical forces to the rudder stock should be tested according to the requirements of LHR.

14.1.4 After installation on board the vessel the steering gear is to be subjected to the required hydrostatic and running tests.

14.2 Trials

14.2.1 The steering gear should be tried out on the trial trip in order to demonstrate to the Surveyor's satisfaction that the requirements of the Rules have been met. The trial should include the operation of the following:

1) the steering gear, including demonstration of the performances required by the SOLAS II-1/29.3.2 and SOLAS II-1/29.4.2. For controllable pitch propellers, the propeller pitch is to be at the maximum design pitch approved for the maximum continuous ahead R.P.M. at the main steering gear trial. If the vessel cannot be tested at the deepest draught, steering gear trials shall be conducted at a displacement as close as reasonably possible to full-load displacement as required by Section 6.1.2 of ISO 19019:2005, as amended, on the conditions that either the rudder is fully submerged (zero speed waterline) and the vessel is in an acceptable trim condition, or the rudder load and torque at the specified trial loading condition have been predicted and extrapolated to the full load condition. In any case for the main steering gear trial, the speed of ship corresponding to the number of maximum continuous revolution of main engine and maximum design pitch applies.
2) the steering gear power units, including transfer between steering gear power units.
3) the isolation of one power actuating system, checking the time to regain steering capability.
4) the hydraulic fluid recharging system.
5) the emergency power supply required by the SOLAS II-1/29.14.
6) the steering gear controls, including transfer of control and local control.
7) the means of communication between the wheel house, engine room, and the steering gear compartment.
8) the alarms and indicators required by the SOLAS II-1/29 and SOLAS II-1/30 as well as SECTION 12 of this Chapter; these tests may be effected at dockside.
9) where steering gear is designed to avoid hydraulic locking this feature shall be demonstrated.

CHAPTER 12 Spares for Machinery

CONTENTS

SECTION 1 General

1.1 Application

1.1.1 Adequate spare parts for the main and essential auxiliary machinery with the necessary tools for maintenance and repair shall be readily available for use on board.

1.1.2 The spare parts to be supplied and their location is to be the responsibility of the Owner but must take into account the design and arrangements of the machinery and the intended service and operation of the ship. Account should also be taken of the recommendations of the manufacturers and any applicable statutory requirement of the country of registration of the ship.

SECTION 2 Recommended spare parts

2.1.1 For general guidance purposes lists of the minimum recommended spare parts for ships for unrestricted service, are given in the following tables:

Table 12.2.1: List of minimum recommended spare parts for main internal combustion engines of ships for unrestricted service

Table 12.2.2: List of minimum recommended spare parts for each type of auxiliary internal combustion engine driving electric generators for essential services on board ships for unrestricted service

Table 12.2.3: List of minimum recommended spare parts for essential auxiliary machinery of ships for unrestricted service

Table 12.2.4: List of minimum recommended spare parts for main steam turbines of ships for unrestricted service

Table 12.2.5: List of minimum recommended spare parts for auxiliary steam turbines driving electric generators for essential services of ships for unrestricted service

Table 12.2.6: Spare parts for boilers supplying steam for propulsion and for essential services

Table 12.2.1: List of minimum recommended spare parts for main internal combustion engines of ships for unrestricted service (IACS Recommendation No. 26)

Item	Spare parts	Number recommended
1. Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1
2. Main thrust block	Pads for one face of Michell type thrust block, or	1 set
	Complete white metal thrust shoe of solid ring type, or	1
	Inner and outer race with rollers, where roller thrust bearings are fitted	1
3. Cylinder liner	Cylinder liner, complete with joint rings and gaskets	1
4. Cylinder cover	Cylinder cover, complete with valves, joint rings and gaskets.	1
	Cylinder cover bolts and nuts, for one cylinder	½ set
5. 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
	Cylinder overpressure sentinel valve, complete	1
	Fuel valves of each size and type fitted, complete with all fittings, for one engine	1 set (1)
6. 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
7. 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
8. Piston rings	Piston rings, for one cylinder	1 set
9. Piston cooling	Telescopic cooling pipes and fittings or their equivalent, for one cylinder unit	1 set
10. Cylinder lubricators	Lubricator, complete, of the largest size, with its chain drive or gear wheels, or equivalent spare part kit	1
11. Fuel injection pumps	Fuel pump complete or, when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valves,	1

	springs, etc.), or equivalent high pressure fuel pump	
12. Fuel injection piping	High pressure double wall fuel pipe of each size and shape fitted, complete with couplings	1
13. Scavenge blower (including turbochargers)	Rotors, rotor shafts, bearings, nozzle rings and gear wheels or equivalent working parts if other types	1 set (2)
14. Scavenging system	Suction and delivery valves for one pump of each type fitted	1 set
15. Reduction and/or reverse gear	Complete bearing bush, of each size fitted in the gear case assembly	1 set
	Roller or ball race, of each size fitted in the gear case assembly	1 set

FOOTNOTES:

1.

(a) Engines with one or two fuel valves per cylinder: one set of fuel valves, complete.

(b) Engines with three or more fuel valves per cylinder: two fuel valves complete per cylinder, and a sufficient number of valve parts, excluding the body, to form, with those fitted in the complete valves, a fuel engine set.

2. The spare parts may be omitted where it has been demonstrated, at the Builder's test bench or one engine of the type concerned, that the engine can be manoeuvred satisfactorily with one blower out of action.
The requisite blanking and blocking arrangements for running with one blower out of action are to be available on board.

NOTES:

1. The availability of other spare parts, such as gears and chains for camshaft drive, should be specially considered and decided upon by the owner.
2. It is assumed that the new crew has on board the necessary tools and equipment.
3. When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.
4. In case of multi-engine installations, the minimum recommended spares are only necessary for one engine.
5. For electronically controlled engines spare parts as recommended by the engine designer/manufacturer.

Table 12.2.2: List of minimum recommended spare parts for each type of auxiliary internal combustion engine driving electric generators for essential services on board ships for unrestricted service (IACS Recommendation No. 27)

Item	Spare parts	Number recommended
1. Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1
2. 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 casing, seat, springs and other fittings	1
	Cylinder overpressure sentinel valve, complete	1
	Fuel valves of each size and type fitted, complete with all fittings, for one engine	½ set
3. 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
	Trunk piston type: gudgeon pin with bush for one cylinder	1 set
4. Piston rings	Piston rings, for one cylinder	1 set
5. Piston cooling	Telescopic cooling pipes and fittings or their equivalent, for one cylinder	1 set
6. Fuel injection pumps	Fuel pump complete or, when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valve springs, etc.), or equivalent high pressure fuel pump	1
7. Fuel injection piping	High pressure double wall fuel pipe of each size and shape fitted, complete with couplings	1
8. Gaskets and Packings	Special gaskets and packings of each size and type fitted, for cylinder covers and cylinder liners for one cylinder	1 set

NOTES:

- The availability of other spare parts should be specially considered and decided upon by the Owner.
- It is assumed that the crew has on board the necessary tools and equipment.
- When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.
- Where the number of generators of adequate capacity fitted for essential services exceeds the required number, spare parts may be omitted.
- For electronically controlled engines spare parts as recommended by the engine designer/manufacturer.

Table 12.2.3: List of minimum recommended spare parts for essential auxiliary machinery of ships for unrestricted service (IACS Recommendation No. 30)

(a) Auxiliary internal combustion engines and steam turbines driving essential service machinery other than generators

The number of minimum recommended spare parts for auxiliary internal combustion engines and steam turbines driving essential service machinery is to be in accordance with that recommended for internal combustion engines and turbines driving electric generators. When an additional unit for the same purpose and of adequate capacity is fitted, spare parts may be omitted.

(b) Pumps (1)

	Spare parts	Number recommended
1.	Piston pumps	
	1.1 Valve with seats and springs, each size fitted	1 set
	1.2 Piston rings, each type and size for one piston	1 set
2.	Centrifugal pumps	
	2.1 Bearing of each type and size	1
	2.2 Rotor sealings of each type and size	1
3.	Gear type pumps	
	3.1 Bearings of each type and size	1
	3.2 Rotor sealings of each type and size	1

NOTES:

- When a sufficiently rated standby pump is available, the spare parts may be dispensed with.

(c) Compressors for essential service

	Spare parts	Number recommended
1.	Suction and delivery valves complete of each size fitted one unit	½ set
2.	Piston rings of each type and size fitted for one piston	1 set

(d) General

It is recommended that where, for maintenance or repair work of the essential machinery, special tools or equipment are to be used, there are available on board. When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.

Table 12.2.4: List of minimum recommended spare parts for main steam turbines of ships for unrestricted service

(IACS Recommendation No. 29)

Item	Spare parts	Number recommended
1. Turbine shaft	Carbon sealing rings, where fitted, with springs for each size sealing rings and type of gland	1 set
2. Oil filters	Strainer baskets or inserts for filters of special design, of each type and size	1 set

NOTES:

- The availability of other spare parts should be specially considered and decided upon by the owner.
- It is assumed that the crew has on board the necessary tools and equipment.
- When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.
- In the case of multi-engine installations, the minimum recommended spare parts are only necessary for one engine.

Table 12.2.5: List of minimum recommended spare parts for auxiliary steam turbines driving electric generators for essential services of ships for unrestricted service
(IACS Recommendation No. 28)

Item	Spare parts	Number recommended
1. Turbine shaft	Carbon sealing rings, where fitted, with springs, for each size and sealing rings type of gland, for one turbine	1 set
2. Oil filters	Strainer baskets or inserts, for filters of special design, of each type and size	1 set

NOTES:

- The availability of other spare parts should be specially considered and decided upon by the Owner.
- It is assumed that the crew has on board the necessary tools and equipment.
- When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.
- Where the number of generators of adequate capacity fitted for essential services exceeds the required number, spare parts may be omitted.

Table 12.2.6: Spare parts for boilers supplying steam for propulsion and for essential services

Item	Spare parts	Number	
		Ships for unrestricted service	Ships for restricted service
1. Tube stoppers or plugs	Tube stoppers or plugs, of each size used, for boiler, superheater and economizer rubes	10	6
2. Fuel oil burners	Fuel oil burners complete or a complete set of wearing parts for the burners, for one boiler	1 set	1 set
3. Gauge glasses	Gauge glasses of round type	2 sets per boiler	2 sets per boiler
	Gauge glasses of flat type	1 set for every two boilers	1 set for every two boilers

CHAPTER 13 Strengthening for Navigation on Ice

CONTENTS

SECTION 1 General

1.1 Notations affixed to the character of classification

1.1.1 When propulsion power and machinery items of ships strengthened for navigation in ice, comply with requirements of "Finish - Swedish Ice Class Rules 1985", an additional notation "I" may be designated. Other type of Ice class reinforcement may also be considered by LHR and a relevant special notation is to be assigned.

1.1.2 The LHR machinery Ice class notations, corresponding to those of the "Finish - Swedish Ice Class Rules 1985" are given in Table 13.1.1.

Table 13.1.1: Corresponding ice class

LHR ice class	I	I1	I2	I3	I4
Finnish-Swedish Ice Class Rules	II	IC	IB	IA	IA super