TÜRK LOYDU

RULES FOR THE CLASSIFICATION OF NAVAL SHIPS

Chapter 104 - Propulsion Plants

January 2022

This latest edition incorporates all rule changes. The latest revisions are shown with a vertical line. The section title is framed if the section is revised completely. Changes after the publication of the rule are written in red colour.

Unless otherwise specified, these Rules apply to ships for which the date of contract for construction as defined in TL- PR 29 is on or after 1st of January 2022. New rules or amendments entering into force after the date of contract for construction are to be applied if required by those rules. See Rule Change Notices on TL website for details.

"General Terms and Conditions" of the respective latest edition will be applicable (see Rules for Classification and Surveys).

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TÜRK LOYDU

Head Office Postane Mah. Tersaneler Cad. No:26 Tuzla 34944 İSTANBUL / TÜRKİYE

Tel : (90-216) 581 37 00

Fax : (90-216) 581 38 00

E-mail : info@turkloydu.org

http://www.turkloydu.org

Regional Offices

Ankara Eskişehir Yolu Mustafa Kemal Mah. 2159. Sokak No : 6/4 Çankaya - ANKARA / TÜRKİYE

Tel : (90-312) 219 56 34 - 219 68 25

Fax : (90-312) 219 69 72

E-mail : ankara@turkloydu.org

İzmir Atatürk Cad. No :378 K.4 D.402 Kavalalılar Apt. 35220 Alsancak - İZMİR / TÜRKİYE

Tel : (90-232) 464 29 88

Fax : (90-232) 464 87 51

E-mail : izmir@turkloydu.org

Adana Çınarlı Mah. Atatürk Cad. Aziz Naci İş Merkezi No:5 K.1 D.2 Seyhan - ADANA / TÜRKİYE

Tel : (90- 322) 363 30 12

Fax : (90- 322) 363 30 19

E-mail : adana@turkloydu.org

Chapter 104 - Propulsion Plants

Section 7 A Propeller

Section 7 B Azimuthing Propulsors

Section 8 Torsional Vibrations

Section 9 Machinery for Ships with Ice Classes

Section 10 Spare Parts

AMENDMENTS

Revision	RCS No.	EIF Date*
Section 03	03/2024	01.01.2025
Section 07	03/2024	01.01.2025
Section 10	01/2024	01.07.2024
Section 4B	01/2024	01.07.2024
Appendix IV	01/2024	01.07.2024
Section 03	01/2024	01.07.2024
Section 03	04/2023	01.01.2024
Section 4B	02/2023	01.07.2023

* Entry into Force (EIF) Date is provided for general guidance only, EIF dates given in Rule Change Summary (RCS) are considered valid. In addition to the above stated changes, editorial corrections may have been made.

SECTION 1

GENERAL RULES AND INSTRUCTIONS

A. General

1. These Rules apply to the propulsion plant of seagoing surface ships and craft intended for naval activities.

The following types of propulsion plants are not included in these Rules:

- Nuclear power plants
- Plants with fuel cell technology **(1)**
- Steam boilers for main propulsion
- Steam turbines
- Low speed diesel engines with crossheads
- Reversible two-stroke diesel engines
- Plants for operation with heavy fuel oil and its pretreatment
- Thermal oil systems

However, on application, plants of a type listed above may be included in a design review and classification procedure, where relevant for the overall concept of a naval project.

2. Apart from machinery and equipment detailed below, these Rules are also applicable individually to other machinery and equipment where this is necessary for the safety of the ship and its crew.

3. Designs which deviate from these Rules may be approved, provided that such designs have been recognized as equivalent.

4. Machinery installations which have been developed on novel principles and/or which have not yet been sufficiently tested in shipboard service require special **TL** approval.

In such cases **TL** is entitled to require additional documentation to be submitted and special trials to be carried out. Such machinery may be marked by the Notation **EXP** affixed to the Character of Classification.

5. In addition to these Rules, **TL** reserve the right to impose further requirements in respect of all types of machinery where this is unavoidable due to new findings or operational experience, or **TL** may permit deviations from the Rules where these are specially warranted.

6. Reference to further regulations and standards

6.1 If the requirements for propulsion plants and operating agents are not defined in these Rules, the application of other regulations and standards has to be defined as far as necessary.

6.2 The regulations of the "International Convention for the Safety of Life at Sea 1974/1978" **(SOLAS),** as amended are considered in these Rules as far as they appear to be applicable to naval surface combat ships. The definite scope of application has to be defined in the building specification by the Naval Authority and the shipyard.

These Rules are also in compliance with the provisions of the "International Convention for the Prevention of Pollution from Ships" of 1973 and the relevant Protocol of 1978 **(MARPOL 73/78).**

6.3 For ships of NATO states the Nato Agreement for Standardisation (STANAG) may be considered.

6.4 Besides of these Rules national regulations, international standards and special definitions in the building specification respectively in the mission statement of the actual ship have to be considered. The application of such regulations is not affected by the **TL** Rules.

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(1) *For auxiliary power to be produced with fuel cell technology see **TL** Rules Guidelines for the Use of Fuel Cell Systems on Board of Ships .*

7. Design

The design of the propulsion plant has to fulfill the following conditions:

7.1 The operation of the naval ship and the habitual conditions foreseen on board as well as the operation of all systems under the operational conditions of combat, wartime cruise, peacetime cruise and peacetime in-port readiness must be ensured at all times.

7.2 The power distribution network shall be designed to ensure operability in case of network failure.

7.3 The operation of certain systems and equipment, which are necessary for safety, is to be guaranteed under defined emergency conditions.

7.4 The risks for crew and ship from operation of the propulsion plant shall be minimized.

7.5 High working reliability shall be achieved by simple and clearly arranged operation processes as well as by application of type-approved products.

7.6 The requirements concerning design, ar-rangement, installation and operation which are de-fined in Chapter 101 - Classification and Surveys and Chapters 102 - Hull Structures and Ship Equipment , 105 - Electrical Installations, 106 - Automation and 107 - Ship Operation Installations and Auxiliary Systems, must be fulfilled.

7.7 A high degree of survivability of the ship should be achieved by redundancies in the design and functioning of essential equipment.

7.8 The principles of ergonomic design of machinery and equipment have to be considered.

7.9 Where in a class of naval ships, originally planned to be identical, deviations become necessary, **TL** shall be duly informed and changes properly documented.

7.10 One failure principle

The single failure concept assumes that only one (single) failure is the initiating event for an undesired occurrence. The simultaneous occurrence of independent failures is not considered.

8. Equivalence

8.1 Naval ships deviating from the **TL** Rules in their type, equipment or in some of their parts may be classed, provided that their structures or equipment are found to be equivalent to the **TL** requirements for the respective Class.

8.2 In this respect, **TL** can accept alternative design, arrangements and calculation/analyses (FE, FMEA, etc.) which are suitable to satisfy the intent of the respective **TL** requirements and to achieve the equivalent safety level

B. Definitions

1. Auxiliary Electrical Power

The auxiliary electrical power [kVA] is defined as the continuous electrical power at continuous speed v_0, which is not directly used for propulsion of the ship, but for driving all kinds of auxiliary devices and equipment. The degree of redundancy shall be defined in the building specification.

2. Black-Out Condition

Black-out condition means that the complete machinery plant including the main source of electrical power are out of operation, but auxiliary energy as compressed air, starting current from batteries, etc. are still available for restoration of power supply.

3. Dead Ship Condition

"Dead ship" condition means that the complete machinery plant including the main source of electrical power are out of operation and auxiliary energy as compressed air, starting current from batteries, etc. are not available for the restoration of the main power supply, for the restart of the auxiliaries and for the start- up of the propulsion plant. It is however assumed that special mobile or fixed equipment for start-up will be available on board of a naval ship.

4. Draught T

The draught **T** is the vertical distance at the middle of the length **L**, from base line to the deepest design water line, as estimated for the lifetime of the ship.

5. Essential Equipment

5.1 Essential for ship operation are all main propulsion plants.

5.2 Essential (operationally important) are the following auxiliary machinery and plants, which:

- Are necessary for propulsion and manoeuvrability of the ship
- Are required for maintaining ship safety
- Are required to maintain the safety of human life at sea

as well as

- Equipment according to special Characters of Classification and Class Notations

5.3 Essential auxiliary machinery and plants are comprising e.g.:

- Generator units
- Steering gear plant
- Fuel oil supply units
- Lubricating oil pumps
- Cooling water/cooling media pumps
- Starting and control-air compressor
- Starting installations for auxiliary and main engines
- Charging air blowers
- Exhaust gas turbochargers
- Controllable pitch propeller installation
- Azimuth drives
- Engine room ventilation fans
- Steam, hot and warm water generation plants
- Oil firing equipment
- Pressure vessels and heat exchangers in essential systems
- Hydraulic pumps
- Fuel oil treatment units
- Fuel oil transfer pumps
- Lubrication oil treatment units
- Bilge and ballast pumps
- Heeling compensation systems
- Fire pumps and fire fighting equipment
- Anchor windlasses and capstans
- Transverse thrusters
- Ventilation fans for hazardous areas
- Turning gears for main engines
- Bow and stern ramps as well as shell openings, if applicable
- Bulkhead door closing equipment
- Weapon systems (effectors)

- Equipment considered necessary to maintain endangered spaces in a safe condition

- NBC fans and passage heaters

- Decontamination equipment

- Parts of the shipboard aircraft installations

5.4 For ships with equipment according to special Characters of Classification and Notations certain type-specific plants may be classed as essential equipment

6. Rated Driving Power P

The rated driving power [kW] is defined as continuous power to be delivered by the propulsion machinery when running at rated speed v_0.

7. Ship speeds

7.1 Rated speed v_0

Expected maximum, continuous ahead speed v_0 [kn] of the ship in calm water at the draught T, when the total available rated driving power is exclusively used for propulsion purposes.

7.2 Maximum speed v_{max}

Expected maximum ahead speed v_{max} [kn] of the ship in calm water at the draught T, when the total available maximum driving power is exclusively used for propulsion devices. This speed is related to an overload condition, permissible only for a defined, relatively short time period.

7.3 Cruising speed v_M

Expected economic, continuous ahead cruising speed v_M [kn] of the ship, which provides the maximum radius of action.

7.4 Minimum speed v_{min}

Expected minimum ahead speed v_{min} [kn] of the ship in calm water at the draught T, when the total available driving power is acting at its technically possible minimum power output.

C. Document for Approval

1. All documents have to be submitted for approval to **TL** in Turkish or English language.

2. The survey of the ship's construction will be carried out on the basis of approved documents. The drawings must contain all data necessary for approval. Where necessary, calculations and descriptions of the ship's elements are to be submitted. Any non-standard symbols used are to be explained in a key list. All documents have to indicate the number of the project and the name of the Naval Authority and/or shipyard.

The drawings and documents have to give sufficient evidence for proving that the requirements set out in this Chapter have been complied with.

3. The supporting calculations shall contain all necessary information concerning reference documents. Literature used for the calculations has to be cited, important but not commonly known sources shall be added in copy.

The choice of computer programs according to the "State of the Art" is free. The programs may be checked by **TL** through comparative calculations with predefined test examples. A generally valid approval for a computer program is, however, not given by **TL**.

The calculations have to be compiled in a way which allows to identify and check all steps of the calculation in an easy way. Hand written, easily readable documents are acceptable.

Comprehensive quantities of output data shall be presented in graphic form. A written comment to the main conclusions resulting from the calculations has to be provided.

4. A summary of the required documents is contained in Chapter 101 - Classification and Surveys, Table 4.1. Further details are defined in the following Sections of this Chapter.

5. **TL** reserve the right to demand additional documentation if that submitted is insufficient for an assessment of the naval ship.

This may especially be the case for plants and equipment related to new developments and/or which are not tested on board to a sufficient extent.

6. The drawings are to be submitted in triplicate, all calculations and supporting documentation in one copy for examination at a sufficiently early date to ensure that they are approved and available to the Surveyor at the beginning of the manufacture of or installation on the naval ship.

7. Once the documents submitted have been approved by **TL** they are binding for the execution of the work. Subsequent modifications and extensions require the approval of **TL** before being put into effect.

8. At the commissioning of the naval ship or after considerable changes or extensions of the propulsion plant, the documentation for approval as defined in the different Sections, showing the final condition of the systems, has to be given on board. All documents have to indicate the name of the ship, the newbuilding number of the shipyard and the date of execution.

D. Ambient Conditions

1. General operating conditions

1.1 The selection, layout and arrangement of the ship's structure and all shipboard machinery shall be such as to ensure faultless continuous operation under defined standard ambient conditions.

More stringent requirements must be observed for Class Notation **AC1** (see Chapter 101 - Classification and Surveys, Section 2, C.).

For the Class Notation **ACS** variable requirements for unusual types and/or tasks of naval ships can be discussed case by case, but shall not be less than the standard requirements.

Components in the machinery spaces or in other spaces which comply with the conditions for the Notations **AC1** or **ACS** must be approved by **TL**.

1.2 Inclinations and movements of the ship

The design conditions for static and dynamic inclinations of a naval ship have to be assumed independently from each other. The standard requirements and the requirements for Class Notation **AC1** are defined in Table 1.1.

TL may consider deviations from the angles of inclination defined in Table 1.1 taking into consideration type, size and service conditions of the naval ship.

The effects of elastic deformation of the ship's hull on the machinery installation have to be considered.

1.3 Environmental conditions

The standard requirements and the requirements for Class Notation **AC 1** are defined in Table 1.2.

2. Vibrations

2.1 General

2.1.1 Machinery, equipment and hull structures are normally subject to vibration stresses. Design, construction and installation must in every case take account of these stresses.

The fault-free long-term service of individual components shall not be endangered by vibration stresses.

If a naval ship is designed to create only a limited influence of vibrations on the fatigue of the hull structures, the mast mounted electronic equipment, etc. and the habitability of the crew the Class Notation **VIBR** may be assigned. For details see **TL** Rules for Hull Structures and Ship Equipment, Section 16, C.

2.1.2 Where a machine or a piece of equipment generates vibrations when in operation, the intensity of the vibration shall not exceed defined limits. The purpose is to protect the vibration generators, the connected assemblies, peripheral equipment and hull components from additional, excessive vibration stresses liable to cause premature failures or malfunctions.

2.1.3 The following provisions relate to vibrations in the frequency range from 2 to 300 Hz. The underlying assumption is that vibrations with oscillation frequencies below 2 Hz can be regarded as rigid-body vibrations while vibrations with oscillation frequentcies above 300 Hz normally occur only locally and may be interpreted as structure-borne noise. Where, in special cases, these assumptions are not valid (e.g. where the vibration is generated by a gear pump with a tooth meshing frequency in the range above 300 Hz) the following provisions are to be applied in analogous manner.

2.1.4 Attention has to be paid to vibration stresses over the whole relevant operating range of the vibration exciter.

Where the vibration is generated by an engine, consideration must cover the whole available working speed range and, where appropriate, to the whole power range.

2.1.5 The procedure described below is largely standardized. Basically, a substitution quantity is formed for the vibration stress or the intensity of the exciter spectrum (cf. 2.2.1). This quantity is then compared with permissible or guaranteed values to check that it is admissible.

2.1.6 The procedure mentioned in 2.1.5 takes the physical facts into account only incompletely. The aim is to evaluate the true alternating stresses or alternating forces. No simple relationship exists between the actual load and the substitution quantities: vibration amplitude, vibration velocity and vibration acceleration at the external parts of the frame. Nevertheless, this procedure is adopted since at present, it appears to be the only one which can be implemented in a reasonable way. For these reasons it is expressly pointed out that the magnitude of the substitution quantities applied in relation to the relevant limits enables no conclusion to be drawn concerning the reliability or load of components as far as these limits are not exceeded. It is, in particular, inadmissible to compare the load of components of different reciprocating machines by comparing the substitution quantities measured at the engine frame.

2.1.7 For reciprocating machinery, the following statements are only applicable for outputs over 100 kW and speeds below 3 000 min^{-1}.

2.1.8 The special rules concerning torsional vibrations according to Section 8 have to be considered.

2.2 Assessment

2.2.1 In assessing the vibration stresses imposed on machinery, equipment and hull structures, the vibration velocity v is generally used as a criterion for the prevailing vibration stress. The same criterion is used to evaluate the intensity of the vibration spectrum produced by a vibration exciter (cf. 2.1.2).

In the case of a purely sinusoidal oscillation, the effective value of the vibration velocity v_{eff} can be calculated by the formula:

$$v_{eff} = \frac{1}{\sqrt{2}} \cdot \hat{s} \cdot \omega = \frac{1}{\sqrt{2}} \cdot \hat{v} = \frac{1}{\sqrt{2}} \cdot \frac{\hat{a}}{\omega} \qquad \textbf{(1)}$$

in which

$\hat{s}$ = vibration displacement amplitude,

$\hat{v}$ = vibration velocity amplitude,

v_{eff} = effective value of vibration velocity,

$\hat{a}$ = vibration acceleration amplitude,

ω = angular velocity of vibration.

For any periodic oscillation with individual harmonic components 1,2,...n, the effective value of the vibration velocity can be calculated by the formula:

$$v_{effi} = \sqrt{v_{eff1}^2 + v_{eff2}^2 + ... + v_{effn}^2} \qquad \textbf{(2)}$$

in which v_{eff} is the effective value of the vibration velocity of the i-th harmonic component. Using formula (1), the individual values of v_{effi} are to be calculated for each harmonic.

Depending on the prevailing conditions, the effective value of the vibration velocity is given by formula (1) for purely sinusoidal oscillations or by formula (2) for any periodic oscillation.

2.2.2 The assessment of vibration loads is generally based on areas A, B and C, which are enclosed by the boundary curves shown in Fig. 1.1. The boundary curves of areas A, B, and C are indicated in Table 1.3. If the vibration to be assessed comprises several harmonic components, the effective value according to 2.2.1 must be applied. The assessment of this value shall take account of all important harmonic components in the range from 2 to 300 Hz.

2.2.3 Area A can be used for the assessment of all machines, equipment and appliances. Machines, equipment and appliances for use on board a ship shall as a minimum requirement be designed to withstand a vibration load corresponding to the boundary curve of area A.

Otherwise, with **TL**'s consent, steps must be taken (vibration damping etc.) to reduce the actual vibration load to the permissible level.

2.2.4 Reciprocating machines must be separately considered, because they act as vibration exciters. Both the vibration generated by reciprocating machines and the stresses consequently imparted to directly connected peripheral equipment (e.g. governors, exhaust gas turbochargers and lubricating oil pumps) and adjacent machines or apparatus (e.g. generators, transmission systems and pipes) may, for the purpose of these Rules and with due regard to the limitations stated in 2.1.6, be assessed using the substitution quantities presented in 2.2.1.

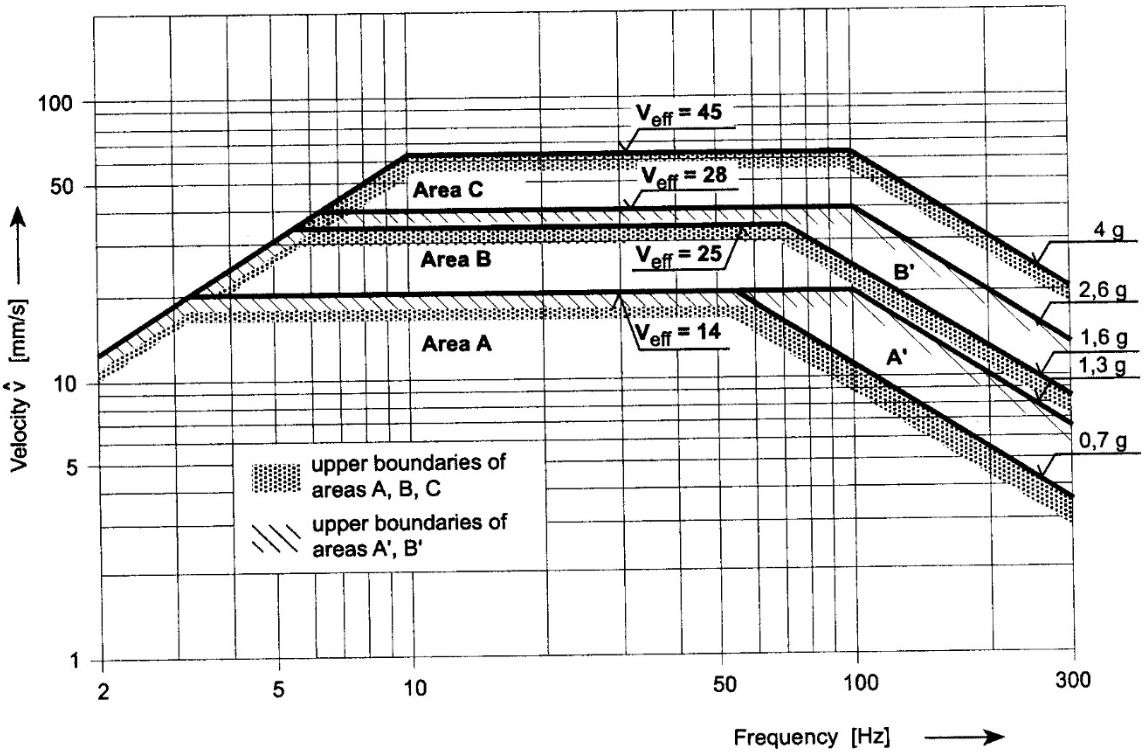

Fig. 1.1 Areas for the assessment of vibration loads

Table 1.1 Design conditions for ship inclinations and movements

Type of movement	Type of inclination and affected equipment	Design conditions	
		Standard requirements	Notation AC 1
Static condition	**Inclination athwartships (1)** Main and auxiliary machinery	15°	25°
	Other installations **(2)**	22,5°	25°
	Ship's structure	acc. to stability requirements	acc. to stability requirements
	Inclinations fore and aft: (1) Main and auxiliary machinery	5°	5°
	Other installations **(2)**	10°	10°
	Ship's structure	acc. to stability requirements	acc. to stability requirements
Dynamic condition	**Rolling: (1)** Main and auxiliary machinery	22,5°	30°
	Other installations **(2)**	22,5°	30°
	Pitching: (1) Main and auxiliary machinery	7,5°	10°
	Other installations **(2)**	10°	10°
	Accelerations: Vertical (pitch and heave)	a_z [g] **(3)**	pitch: 32 °/s^2 heave: 1,0 g
	Transverse (roll, yaw and sway)	a_y [g] **(3)**	roll: 48 °/s^2 yaw: 2 °/s^2
	Longitudinal (surge) Combined acceleration	a_x [g] **(3)** acceleration ellipse **(3)**	sway: a_y [g] a_x [g] **(4)** direct calculation

(1) *Athwartships and fore and aft inclinations may occur simultaneously*
(2) *Ship's safety equipment, e.g. emergency power installations, emergency fire pump and their device, switch gear and electric/electronic equipment*
(3) *Defined in Chapter 102 - Hull Structures and Ship Equipment, Section 5, B.*
(4) *To be defined by direct calculation*

Table 1.2 Design environmental conditions

Environmental area	Parameters	Design conditions	
		Standard requirements	Notation AC1
Outside the ship/air	Temperature For partially open spaces	- 25 °C to + 45 °C **(1)** -	- 30 °C to + 55 °C **(1)** - 10 °C to + 50 °C **(1)**
	Temperatures related to: - atmospheric pressure - max. relative humidity	 1000 mbar 60 % **(2)**	 900 mbar to 1100 mbar 100 %
	Salt content	1 mg/m³	1 mg/m³
		withstand salt-laden spray	withstand salt-laden spray
	Dust/sand	to be considered	filters to be provided
	Wind velocity (systems in operation)	43 kn **(3)**	90 kn
	Wind velocity (systems out of operation)	86 kn **(3)**	100 kn
Outside the ship/ seawater	Temperature **(4)**	-2°Cto +32°C	-2°C to +35°C
	Density acc. to salt content	1,025 t/m³	1,025 t/m³
	Flooding	withstand temporarily	withstand temporarily
Outside the ship/ icing of surface	Icing on ship's surfaces up to 20 m above waterline	see Chapter 102, Section 2, B.3.4	see Chapter 102, Section 2, B.3.4
Outside the ship/ navigation in ice	Ice class B	drift ice in mouth of rivers and coastal regions	drift ice in mouth of rivers and coastal regions
Entrance to the ship/ for design of heating/ cooling systems	Air temperature	-15°Cto +35 °C	-15°Cto +35 °C
	Max. heat content of the air	100kJ/kg	100 kJ/kg
	Seawater temperature	- 2 °C to + 32 °C	-2°C to +35°C
Inside the ship/ all spaces **(5)**	Air temperature	0 °C to + 45 °C	0 °C to + 45 °C
	Atmospheric pressure	1000 mbar	1000 mbar
	Max. relative humidity	up to 100% (+45 °C)	100 %
	Salt content	1 mg/m³	1 mg/m³
	Oil vapour	withstand	withstand
	Condensation	to be considered	to be considered
Inside the ship/ air-conditioned areas	Air temperature	0 °C to + 40 °C	0 °C to + 40 °C
	Max. relative humidity	80%	100 %
	Recommended ideal climate for manned computer spaces	-	air temperature + 20 °C to + 22 °C at 60 % rel. humidity
Inside the ship/ in electrical devices with higher degree of heat dissipation	Air temperature	0 °C to + 55 °C	0 °C to + 55 °C
	Max. relative humidity	100 %	100 %

(1) *Higher temperatures due to radiation and absorption heat have to be considered*
(2) *100 % for layout of electrical installations*
(3) *For lifting devices according to **TL** Rules , Guidelines for the Construction and Survey of Lifting Appliances,*
(4) ***TL** may approve lower limit water temperatures for ships operating only in special geographical areas*
(5) *For recommended climatic conditions in the ship's spaces see also Chapter 107 — Ship Operation Installations and Auxiliary Systems, Section 11, F.*

Table 1.3 Numerical definition of the area boundaries shown in Fig. 1.1

Areas		A	B	C	A'	B'
$\hat{s}$	[mm]	< 1	< 1	< 1	< 1	< 1
$\hat{v}$	[mm/s]	<20	< 35	< 63	< 20	< 40
v_{eff}	[mm/s]	< 14	< 25	< 45	< 14	< 28
â	[g]	< 0.7	< 1.6	< 4	< 1.3	< 2.6

2.2.4.1 In every case the manufacturer of reciprocating machines has to guarantee permissible vibration loads for the important directly connected peripheral equipment. The manufacturer of the reciprocating machine is responsible to **TL** for proving that the vibration loads are within the permissible limits in accordance with 2.3.

2.2.4.2 Where the vibration loads of reciprocating machines lie within the A' area, separate consideration or verifications relating to the directly connected peripheral equipment (cf. 2.2.4) are not required. The same applies to machines and apparatus located in close proximity to the vibration exciter (2.2.4).

In these circumstances directly connected peripheral appliances shall in every case be designed for at least the limit loads of area B', and machines located nearby for the limit loads of area B.

If the permissible vibration loads of individual directly connected peripheral appliances in accordance with 2.2.4.1 lie below the boundary curve of area B, permissibility must be proved by measurement of the actually occurring vibration load.

2.2.4.3 If the vibration loads of reciprocating machines lie outside area A' but are still within area B', it must be proved by measurement that directly connected peripheral appliances are not loaded above the limits for area C.

In these circumstances directly connected peripheral appliances shall in every case be designed for at least the limit loads of area C, and machines located nearby for the limit loads of area B.

Proof is required that machines and appliances located in close proximity to the main exciter are not subject to higher loads than those defined by the boundary curve of area B.

If the permissible vibration loads of individual, directly connected peripheral appliances or machines in accordance with 2.2.4.1 lie below the stated values, permissibility must be proved by measurement of vibration load which actually occurs.

2.2.4.4 If the vibration loads of reciprocating machines lie outside area B' but are still within area C, it is necessary to ensure that the vibration loads on the directly connected peripheral appliances still remain within area C. If this condition cannot be met, the important peripheral appliances must, in accordance with 2.3, be demonstrably designed for the higher loads.

Suitable measures (vibration damping etc.) are to be taken to ensure reliable prevention of excessive vibration loads on adjacent machines and appliances. The permissible loads stated in 2.2.4.3 (area B or a lower value specified by the manufacturer) continue to apply to these units.

2.2.4.5 For directly connected peripheral appliances, **TL** may approve higher values than those specified in 2.2.4.2, 2.2.4.3 and 2.2.4.4, if these are guaranteed by the manufacturer of the reciprocating machine in accordance with 2.2.4.1 and are proved in accordance with 2.3.

Analogously, the same applies to adjacent machines and appliances, if the relevant manufacturer guarantees higher values and provides proof of these in accordance with 2.3.

2.2.5 For appliances, equipment and components which, because of their installation in steering gear compartments or bow thruster compartments, are exposed to higher vibration stresses, the permissibility of the vibration load may, notwithstanding 2.2.3, be assessed according to the limits of area B. The design of such equipment shall allow for the above mentioned increased loads.

2.3 Proofs

2.3.1 Where in accordance with 2.2.4.1, 2.2.4.4 and 2.2.4.5 **TL** is asked to approve higher vibration load values, all that is normally required for this is the binding guarantee of the admissible values by the manufacturer or the supplier.

2.3.2 **TL** reserve the right to call for detailed proofs (calculations, design documents, measurements, etc.) in cases where this is justified.

2.3.3 Type testing in accordance with the **TL** Rules - Test Requirements for Electrical/Electronic Equipment and Systems, are regarded as proof of permissibility of the tested vibration load.

2.3.4 **TL** may recognize long-term trouble free operation as sufficient proof of the required reliability and operational dependability.

2.3.5 The manufacturer of the reciprocating machine is in every case responsible to **TL** for any proof which may be required concerning the level of the vibration spectrum generated by the reciprocating machine.

2.4 Measurement

2.4.1 Proof based on measurements is normally required only for reciprocating machines with an output of more than 100 kW, provided that the other conditions set out in 2.2.4.2 - 2.2.4.4 are met. Where circumstances justify this, **TL** may also require proofs based on measurements for smaller outputs.

2.4.2 Measurements are to be performed in every case under realistic service conditions at the installa-tion location. During verification, the output supplied by the reciprocating machine shall be not less than 80 % of the rated value. The measurement shall cover the entire available speed range in order to facilitate the detection of any resonance phenomena.

2.4.3 **TL** may accept proofs based on measurements which have not been performed at the installation location (e.g. test bed runs), but under different mounting conditions, provided that the transferability of the results can be proved.

The results are normally regarded as transferable in the case of flexibly mounted reciprocating machines of customary design.

If the reciprocating machine is not flexibly mounted, the transferability of the results may still be acknowledged if the essential conditions for this (similar bed construction, similar installation and pipe routing etc.) are fulfilled.

2.4.4 For assessment of the vibration stresses affecting or generated by reciprocating machines normally the location in which the vibration loads are greatest. Fig. 1.2 indicates the points of measurement which are normally required for an in-line piston engine. The measurement has to be performed in all three directions. In justified cases exceptions can be made to the inclusion of all measuring points.

2.4.5 The measurements may be performed with mechanical manually-operated instruments provided that the instrument setting is appropriate to the measured values bearing in mind the measuring accuracy.

Directionally selective, linear sensors with a frequency range of at least 2 to 300 Hz should normally be used. Non-linear sensors can also be used provided that the measurements take account of the response characteristic

2.4.6 The records of the measurements for the points at which the maximum loads occur are to be submitted to **TL** together with a tabular evaluation.

3. Shock

Naval ships may also be exposed to shock forces created by air or underwater explosions from conventional or nuclear weapons.

For naval ships with special ability to withstand shock loads Class Notation **SHOCK** may be assigned. Details for shock requirements are described in Chapter 102 - Hull Structures and Ship Equipment, Section 16.

E. Materials

1. Approved Materials

1.1 The materials used for propulsion plants have to fulfill the quality requirements defined in **TL** Material Rules and Welding Rules. The approved materials for the different systems are defined in the following Sections.

1.2 Materials deviating from the defined quality requirements may only be used with special approval of **TL**. The suitability of the materials has to be proven.

F. Fuels and Consumables for Operation

1. All fuels and consumables used for the operation of propulsion plants must be in accordance with the requirements of the manufacturers.

2. The flash point **(2)** of liquid fuels for the operation of boilers and diesel engines may not be lower than 60°.

For emergency power generating sets, however, use may be made of fuels with a flash point of ≥ 43 °C. The fuel must enable a starting of the emergency generating set at ambient temperatures of - 15 °C and above.

3. In exceptional cases, for ships intended for operation in limited geographical areas or where special precautions subject to **TL** approval are taken, fuels with flash points between 43 °C and 60 °C may also be used. This is conditional upon the requirement that the temperatures of the spaces in which fuels are stored or used must invariably be 10 °C below the flash point.

4. The fuel must be filterable.

5. The fresh cooling water for internal combus-tion engines has to be treated from freshwater and corrosion protection agent.

Fresh water must comply with the requirements of the engine manufacturer with respect to:

- Water hardness [dGH]
- p_H value (at 20 °C)
- Chloride content [mg/l]

6. The storage of fuel and consumables for operation has to follow the requirements of Chapter 107 - Ship Operation Installations and Auxiliary Systems, Section 7.

G. Safety Equipment and Protective Measures

Machinery is to be installed and safeguarded in such a way that the risk of accidents is largely ruled out. It has to be ensured that physical arrangements for machinery and equipment do not pose a risk to personnel. Besides of national regulations particular attention is to be paid to the following:

1. Moving parts, flywheels, chain and belt drives, linkages and other components which could constitute an accident hazard for the operating personnel are to be fitted with guards to prevent contact. The same applies to hot machine parts, pipes and walls for which no thermal insulation is provided, e.g. pressure lines to air compressors.

2. The design and installation of all systems and equipment has to guarantee that elements, which have to be used during normal operation of the ship by the crew and where no thermal insulation is provided, are kept within the following restrictions concerning accidental contact of hot surfaces.

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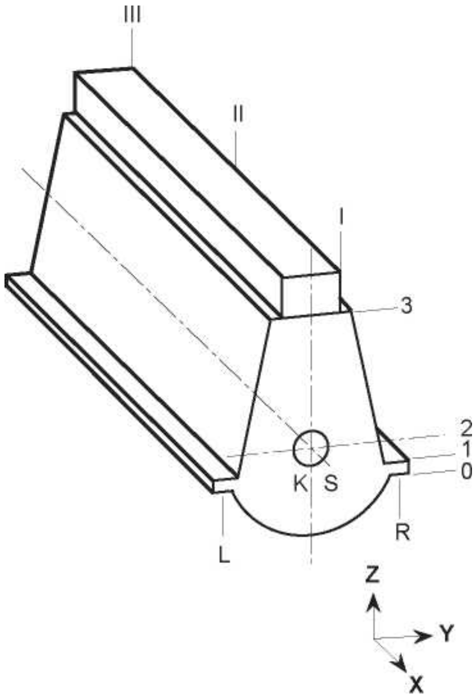
(2) *Based, up to 60 °C, on determination of the flash point in a closed crucible (cup test).*



Sides for measurement

L left side looking towards coupling flange
R right side looking towards coupling flange

Measuring height
0 bed
1 base
2 crankshaft height
3 frame top

Measuring point over engine length
I coupling side (KS)
II engine center
III opposite side to coupling (KGS)

Fig. 1.2 Schematic representation of in-line piston engine

2.1 No skin contact is possible with elements warmed up under operating conditions to surface temperature above 70 °C.

2.2 Components, which may be used without body protection, e.g. protective gloves and with a contact time up to 5 s, are to have no higher surface temperature than 60 °C.

2.3 Components made of materials with high thermal conductivity, which may be used without body protection and a contact time of more than 5 s are not to achieve a surface temperature above 45 °C.

2.4 Exhaust gas lines and other apparatus and lines transporting hot media have to be insulated effectively. Insulation material must be non combustible. Locations where inflammable liquids or moisture may penetrate into the insulation are to be protected in a suitable way by coverings, etc.

3. When using hand cranks for starting internal combustion engines, steps are to be taken to ensure that the crank disengages automatically when the engine starts.

Dead-Man's circuits are to be provided for rotating equipment.

4. Blowdown and drainage facilities are to be designed in such a way that the discharged medium can be safely drained off.

5. In operating spaces, anti-skid floor plates and floor coverings must be used.

6. Service gangways, operating platforms, stairways and other areas open to access during operation are to be safeguarded by guard rails. The outside edges of platforms and floor areas are to be fitted with coamings unless some other means is adopted to prevent persons and objects from sliding off.

7. Glass water level gauges for auxiliary steam boilers are to be equipped with protection devices.

Devices for blowing through water level gauges shall be capable of safe operation and observation.

8. Safety valves and shutoffs must be capable of safe operation. Fixed steps, stairs or platforms are to be fitted where necessary.

9. Safety valves are to be installed to prevent the occurrence of excessive operating pressures.

10. Steam and feedwater lines, exhaust gas ducts, auxiliary boilers and other equipment and pipelines carrying steam or hot water are to be effectively insulated. Insulating materials are to be incombustible. Points at which combustible liquids or moisture can penetrate into the insulation are to be suitably protected, e.g. by means of shielding.

11. For machinery which is operated in hazardous areas, provisions are to be established to avoid the risk of ignition. These might be design measures, monitoring of criterial parameters and/or bring the machinery in a safe condition.

H. Survivability

1. Definition

Survivability of a naval ship is to be regarded as the degree of ability to withstand a defined weapon threat and to maintain at least a basic degree of safety and operability of the ship.

It is obvious that survivability is an important characteristic of a naval ship which may be endangered by:

- Loss of global strength of the hull structure
- Loss of buoyancy and/or stability
- Loss of manoeuvrability
- Fire in the ship and ineffective fire protection or fire fighting capability
- Direct destruction of machinery, equipment or control systems
- Direct destruction of weapons and sensors
- Threat to the crew

2. Measures for improved survivability

The design of a ship which is classed as naval ship has to consider a series of possible measures to improve survivability. These **TL** Rules for naval surface ships offer in the different Chapters various measures and Class Notations to achieve improved survivability. The degree of including such measures in an actual project has to be defined by the Naval Authority.

3. Measures for the propulsion plant

In this Chapter the following main measure to improve survivability is included.

3.1 Redundant propulsion

The requirements for the Class Notations **RP1 x%** to **RP3 x%** concerning redundant propulsion and manoeuvrability are defined in Section 2, K.

SECTION 2

DESIGN AND CONSTRUCTION OF THE MACHINERY INSTALLATION

A. General

1. As far as necessary for function and safe operation, the design of the propulsion plant has to consider the size and experience of the crew according to the mission statement of the actual naval ship.

2. Normally it can be assumed, that:

- No personnel is permanently present in machinery rooms

- In machinery rooms inspection patrols will be made in regular time intervals

- The machinery control centre (MCC) is permanently manned

3. The operating and maintenance instructions, warning signs, etc. have to be prepared in English and in the user's language.

B. Dimensions of Components

1. General

1.1 All parts must be capable of withstanding the stresses and loads peculiar to shipboard service, e.g. those due to movements of the ship, vibrations, intensified corrosive attack, temperature changes and wave impact, and must be dimensioned in accordance with the requirements set out in the present Chapter. The ambient conditions acc. to Section 1, D. and Chapter 101 - Classification and Surveys and Chapters 105 - Electrical Installations, 106 - Automation and 107 - Ship Operation Installations and Auxiliary Systems, have to be considered.

In the absence of rules governing the dimensions of parts, the recognized rules of engineering practice are to be applied.

1.2 Where connections exist between systems or plant items which are designed for different forces, pressures and temperatures (stresses), safety devices are to be fitted which prevent the over-stressing of the system or plant item designed for the lower design parameters. To preclude damage, such systems are to be fitted with devices affording protection against excessive pressures and temperatures and/or against overflow.

2. Materials

All components subject to these Rules must comply with **TL** Material Rules.

3. Welding

The fabrication of welded components, the approval of companies and the testing of welders are subject to **TL** Welding Rules.

4. Screw connections

4.1 Screws are to be designed according to proven and acknowledged principles.

4.2 The design of screw connections between equipment and foundations has to avoid shear forces or bending moments in the screws, only tensional forces in axial direction are allowed. Therefore, the flanges where the screw forces are acting have to be stiffened with a suitable number of brackets to avoid flange bending, which would create bending moments to the screws.

4.3 Only closed holes in flanges shall be provided for screws, because under shock influence screws might slip out of a hole in form of a half open slot. In addition friction connections are not safe under shock influence.

Note :

If expansion screws are used, the peak stresses at the core of the thread are reduced, because the maximum stress will occur at the location of the reduced shaft section. If such screws are long enough, they will be able to protect the mounted equipment from shock loads to some extent, because the elongation of the shaft reduces acceleration peaks.

It is also recommendable to use cap nuts together with expansion screws for connections. With this form of nut the load on the different turns of the thread is nearly the same, *which helps to avoid excessive local stresses.*

If a normal screw is fitted into a threaded hole, the breaking of the screw because of the severe transverse dynamic shock loads will be evident at the first turn of the thread coming out of the hole. This is the case because of the adding up of the bending stress to the maximum axial stress which occurs at this turn. The problem can be solved by a flat necking of the shaft above the last turn of the thread or by a special form of shaft and hole. The bolt is in the latter case provided with a small collar fitting in the upper part of the hole. Thus the biggest part of the bending stress will be transmitted at the collar and is separated from the axial stress at the thread.

5. Tests

5.1 Machinery and its component parts are subject to constructional and material tests, pressure and leakage tests, and trials. All the tests prescribed in the following Sections are to be conducted under supervision of **TL**.

In case of parts produced in series, other methods of testing may be agreed with **TL** instead of the tests prescribed, provided that the former are recognized as equivalent by **TL**.

5.2 **TL** reserve the right, where necessary, to increase the scope of tests and also to subject to testing those parts which are not expressly required to be tested according to the Rules.

5.3 Components subject to mandatory testing are to be replaced by tested parts.

5.4 After installation on board of the main and auxiliary machinery, operational functioning of the machinery including associated ancillary equipment is to be verified. All safety equipment is to be tested, unless adequate testing has already been performed at the manufacturer's works in the presence of the **TL** Representative.

In addition, the entire machinery installation is to be tested during sea trials, as far as possible under the intended service conditions.

6. Corrosion protection

Parts which are exposed to corrosion are to be safeguarded by being manufactured of corrosion-resistant materials or provided with effective corrosion protection.

C. Availability of Machinery

1. Ship's machinery is to be so arranged and equipped that it can be brought into operation from the "dead ship" condition with the means available on board.

The "dead ship" condition means that the entire machinery installation including the electrical power supply is out of operation and auxiliary sources of energy such as starting air, battery-supplied starting current etc. are not available for restoring the ship's electrical system, restarting auxiliary operation and bringing the propulsion installation back into operation.

To overcome the "dead ship" condition use may be made of an emergency generator set provided that it is ensured that the electrical power for emergency services is available at all times.

2. In case of "dead-ship" condition it must be ensured that it will be possible for the propulsion system and all necessary auxiliary machinery to be restarted within a period of 30 minutes, see Chapter 107 - Ship Operation Installations and Auxiliary Systems, Section 6,A.4.

D. Control and Regulating

1. Machinery must be so equipped that it can be controlled in accordance with operating requirements in such a way that the service conditions prescribed by the manufacturer can be met.

1.1 For the control equipment of main engines and systems essential for operation see Chapters 105 - Electrical Installations, 106 - Automation and 107 - Ship Operation Installations and Auxiliary Systems.

2. In the event of failure or fluctuations of the supply of electrical, pneumatic or hydraulic power to regulating and control systems, or in case of a break in a regulating or control circuit, steps must be taken to ensure that:

- The appliances remain at their present operational setting or, if necessary, are changed to a setting which will have the minimum adverse effect on operation (fail-safe conditions)

- The power output or engine speed of the machinery being controlled or governed is not increased

- No unintentional start-up sequences are initiated

3. Each driving machinery has to be provided with an emergency stopping device.

4. Manual Operation

Every functionally important, automatically or remote controlled system must also be capable of manual operation, see also the Rules in Chapter 106 - Automation.

A manual emergency stopping device has to be provided.

E. Propulsion Plant

1. General

1.1 All devices forming a part of the propulsion plant have to be provided with peripheral components which ensure a faultless handling as well as a simple and safe operation and control, even if they are not specified in detail.

1.2 All auxiliary machinery and the control units directly needed for the operation of the propulsion plant are to be located as near as possible to the driving machinery.

1.3 Manoeuvring equipment

Every engine control stand is to be equipped in such a way that:

- the propulsion plant can be adjusted to any setting

- the direction of propulsion can be reversed

- the propulsion unit or the propeller shaft can be stopped

1.4 If required in the building specification, the driving machinery has to be designed and equipped for operation in a sound protection capsule.

Using sound protection capsules the relative movement between the driving machinery and the shipside piping system has to be compensated by flexible connections. If applicable, the elastic mounting of the driving machinery has to be taken into account.

1.5 For connected pumps no electrically driven stand by pumps are to be provided if the propulsion is ensured by several driving machines.

1.6 The remote control of the propulsion plant from the bridge is subject to the **TL** Rules for Automation.

2. Multiple Shaft and Multi-Engine Systems

2.1 Definitions

2.1.1 CODAD

CODAD is the abbreviation for COmbined Diesel engine And Diesel engine. In this configuration a propulsion shaft can be driven alternatively by one diesel engine or several diesel engines, see Fig. 2.1.

2.1.2 CODAG

CODAG is the abbreviation for COmbined Diesel engine And Gas turbine. In this configuration a propulsion shaft can be driven alternatively by a diesel engine or by a gas turbine or by both, see Fig. 2.2.

2.1.3 CODOG

CODOG is the abbreviation for COmbined Diesel engine Or Gas turbine. In this configuration a propulsion shaft can be driven alternatively by a diesel engine or a gas turbine, see Fig. 2.3.

2.1.4 COGAG

COGAG is the abbreviation for COmbined Gas turbine And Gas turbine. In this configuration a propulsion shaft can be driven alternatively by one gas turbine or by several gas turbines, see Fig. 2.4.

2.1.5 COGOG

COGOG is the abbreviation for COmbined Gas turbine Or Gas turbine. In this configuration a propulsion shaft can be driven alternatively by one or the other gas turbine, see Fig. 2.5.

2.1.6 CODELAG

CODELAG is the abbreviation for COmbined Diesel ELectric And Gasturbine. In this configuration a propeller shaft can be driven alternatively by one or two electric motors, only by the gas turbine or by electric motor and gas turbine together. The electric drive will be chosen if the radiated noise has to be minimized and for cruising speed v_M. The combination of electric motors and gas turbine has to be applied for maximum speed v_{max}, see Fig. 2.6.

2.1.7 IEP

IEP is the abbreviation for Integrated Electric Propulsion. In this configuration the electric power is generated by several diesel gen sets and/or gas turbine gen sets.

According to the actual speed requirements the required power for propulsion will be delivered to the electric propulsion motor of each propeller shaft, see Fig. 2.7.

2.1.8 AZP

AZP is the abbreviation for AZimuthing Propulsion. In this configuration the electric power is generated by several diesel gen sets and/or gas turbine gen sets. The drive of the propulsors is done directly by electric motors in the azimuthing pods (without transmitting shaft), see Fig. 2.8.

2.2 If a ship is equipped with several propulsion machines, it must be possible to disengage the different propulsion engines from the power transmitting unit. At manual or automatic emergency stops of an engine, the relevant clutch must disconnect automatically.

2.3 Steps are to be taken to ensure that in the event of a failure of a propulsion engine, operation can be maintained with remaining engines, where appropriate by a simple change-over system.

2.4 Multiple shaft systems

A space separation between driving engines and driving gear shall be provided, if possible.

All necessary provisions have to be made, that:

- Only one propulsion shaft can be used and no overloading occurs

- Starting and operation is possible with every driving engine intended to drive a propulsion shaft, independently from the other propulsion shafts

For Class Notation redundant propulsion **(RP)** see K.

For multiple-shaft systems, each shaft is to be provided with a locking device which prevents dragging of the shaft.

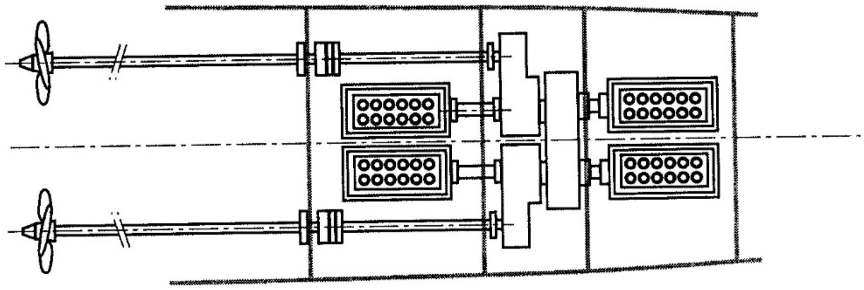


Figure 2.1 Principle arrangement for CODAD propulsion plants

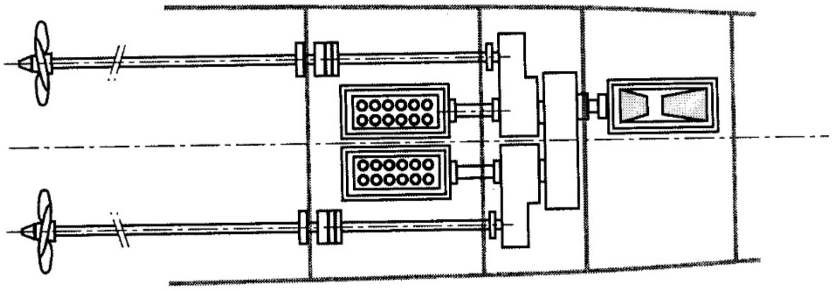


Figure 2.2 Principle arrangement for CODAG propulsion plants

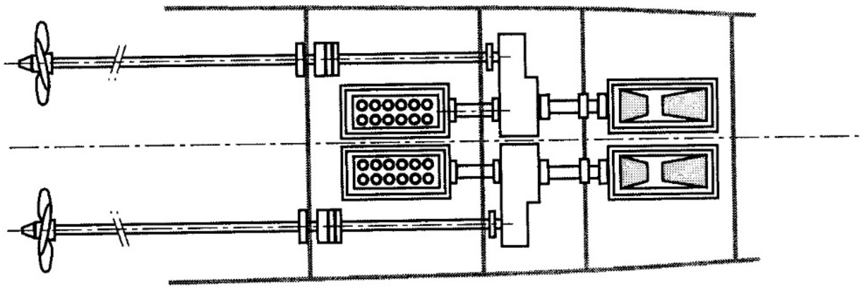


Figure 2.3 Principle arrangement for CODOG propulsion plants

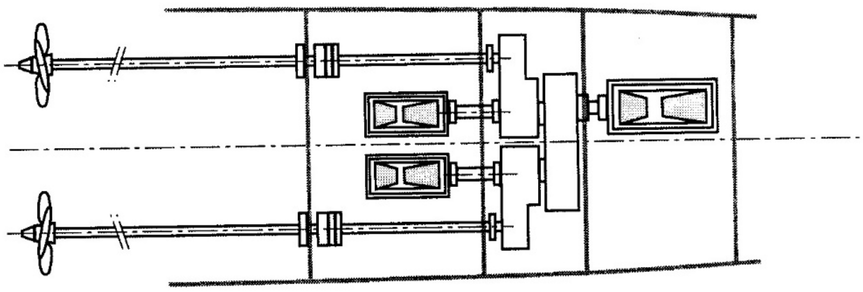


Figure 2.4 Principle arrangement for COGAG propulsion plants

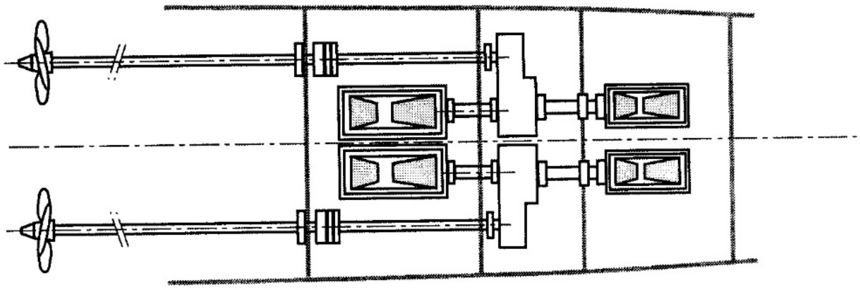


Figure 2.5 Principle arrangement for COGOG propulsion plants

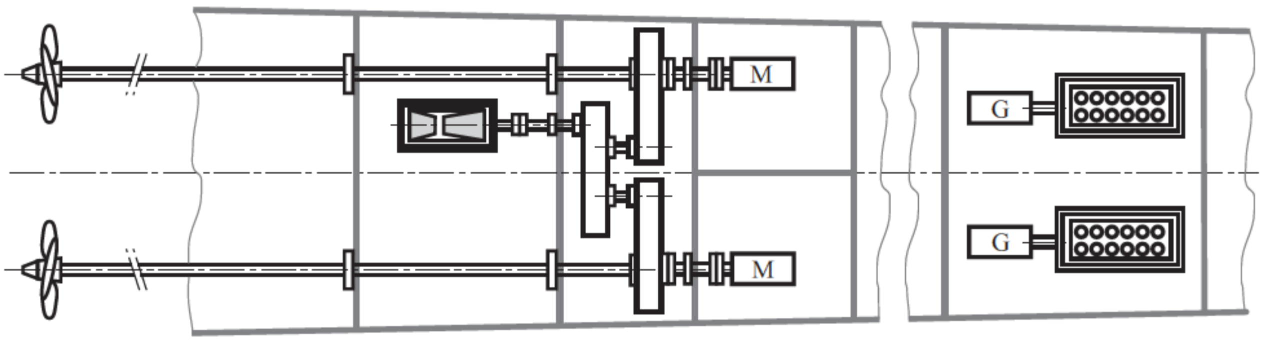


Figure 2.6 Principle arrangement for CODELAG plants

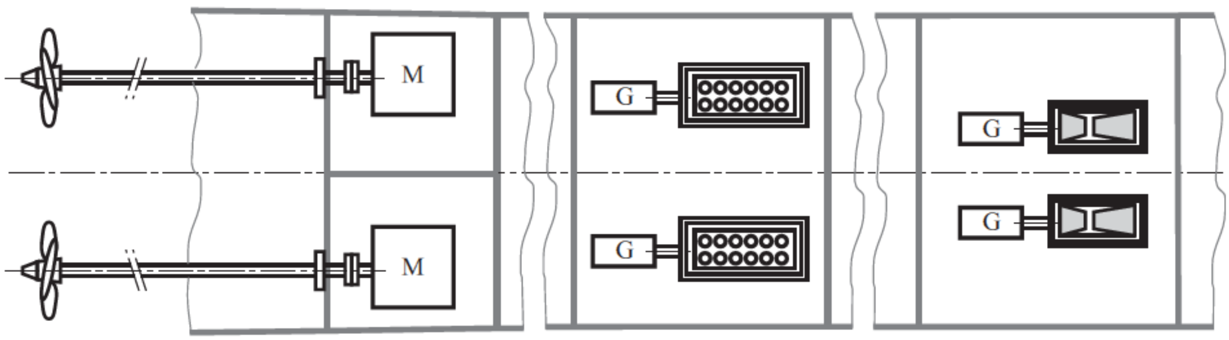


Figure 2.7 Principle arrangement for Integrated Electric Propulsion (IEP)

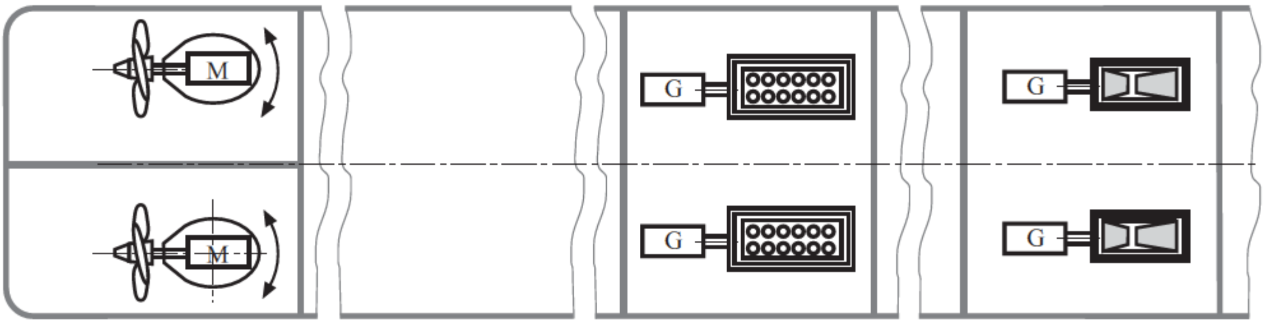
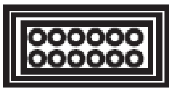
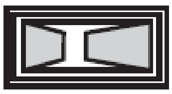


Figure 2.8 Principle arrangement for Azimuthing Propulsors (AZP)

Explanation of symbols used in Figures 2.1 to 2.8


Diesel engine


Gas turbine

Electric generator

Electric motor


Clutch


Bulkhead penetration of shafts

F. Turning Appliances

1. Machinery is to be equipped with suitable and adequately dimensioned turning appliances.

2. The turning appliances are to be of self-locking type. Electric motors are to be fitted with suitable retaining brakes.

3. An automatic interlocking device is to be provided to ensure that the propulsion and auxiliary prime movers cannot start up while the turning gear is engaged. In case of manual turning installations warning devices may be provided alternatively.

G. Operating and Maintenance Instructions

Manufacturers of machinery, boilers and auxiliary equipment must supply a sufficient number of operating and maintenance notices and manuals together with the equipment.

In addition, an easily legible board is to be mounted on boiler operating platforms giving the most important operating instructions for boilers and oil-firing equipment.

H. Markings, Identification

In order to avoid unnecessary operating and switching errors, all parts of the machinery which function is not immediately apparent are to be adequately marked and labelled.

I. Engine Room Equipment

1. Operating and Monitoring Equipment

1.1 Instruments, warning and indicating systems and operating appliances are to be clearly displayed and conveniently sited. Absence of dazzle, particularly on the bridge, is to be ensured.

Operating and monitoring equipment is to be grouped in such a way as to facilitate easy supervision and control of all important parts of the installation.

The following requirements are to be observed when installing systems and equipment:

- Protection against humidity and the effects of dirt

- Avoidance of excessive temperature variations

- Adequate ventilation

In consoles and cabinets containing electrical or hydraulic equipment or lines carrying steam or water the electrical gear is to be protected from damage due to leakage. Redundant ventilation systems are to be provided for air-conditioned machinery and control rooms.

1.2 Pressure gauges

The scales of pressure gauges are to be dimensioned up to the specified test pressure. The maximum permitted operating pressures are to be marked on the pressure gauges for boilers, pressure vessels and in systems protected by safety valves. Pressure gauges must be installed in such a way that they can be isolated.

Lines leading to pressure gauges must be installed in such a way that the readings cannot be affected by liquid heads and hydraulic hammering.

2. Accessibility of Machinery and Boilers

2.1 Machinery and boiler installations, apparatuses, control and operating devices must be easily accessible and visible for operation and maintenance. National regulations have to be observed.

2.2 In the layout of machinery spaces (design of foundation structures, laying of pipelines and cable conduits, etc.) and for the design of machinery and equipment (mountings for filters, coolers, etc.) 2.1 is to be complied with.

For the installation and the dismantling of the propulsion machinery, openings of sufficient size have to be provided; for all other parts of the equipment installation and dismantling routes have to be planned.

Fixtures for transport devices have to be provided.

The installation of components within the installation routes has to be avoided or the components must be dismountable.

3. Machinery Control Centre

Machinery control centres (MCC) are to be provided with at least two exits, one of which can also be used as an escape route.

4. Lighting

All operating spaces must be adequately lit to ensure that control and monitoring instruments can be easily read. In this connection see the rules defined in Chapter 105 - Electrical Installation, Section 11.

5. Bilge wells/ bilges

See rules defined in Chapter 107 - Ship Operation Installations and Auxiliary Systems, Section 8.

6. Ventilation

The design and construction of ventilation systems are subject to the requirements defined in Chapter 107 -Ship Operation Installations and Auxiliary Systems, Section 11.

7. Noise abatement

In compliance with the relevant national regulations, care is to be taken to ensure that operation of the ship is not unacceptably impaired by engine noise, e.g. by means of shielding.

J. Communication and Signalling Equipment

1. Voice Communication

Means of voice communication are to be provided between the ship's manoeuvring station, the engine room and the steering gear compartment, and these means shall allow fully satisfactory intercommunication independent from the shipboard power supply under all operating conditions (see also Chapter 105 - Electrical Installations, Section 9).

2. Duty Alarm System

From the engine room or the machinery control centre a duty alarm system for the engineer officers has to be established for the off-duty period. See also Chapter 105 – Electrical Installation, Section 9.

3. Engine Telegraph

Machinery operated from the engine room must be equipped with a telegraph.

In the case of multiple-shaft installations, a telegraph must be provided for each unit.

Local control stations are to be equipped with an emergency telegraph.

4. Shaft revolution indicator

The speed and direction of rotation of the propeller shafts are to be indicated on the bridge, in the engine room respectively in all control stations. In the case of small propulsion units, the indicator may be dispensed with.

Barred speed ranges are to be marked on the shaft revolution indicators.

5. Design of communication and signaling equipment

Reversing, command transmission and operating controls etc. are to be grouped together at a convenient point on the control platform.

The current status, "Ahead" or "Astern", of the reversing control is to be clearly indicated on the propulsion plant control platform.

Signalling devices must be clearly perceptible from all parts of the engine room when the machinery is in full operation.

For details of the design of electrically operated command transmission, signalling and alarm systems, see Chapter 105 - Electrical Installations and 106 - Automation.

K. Redundant Systems

1. General

1.1 The Rules relating to redundant propulsion and steering systems apply to ships, which are classified by **TL** and are to receive the Notation **RP1 x%**, **RP2 x%** or **RP3 x%** affixed to the Character of Classification.

1.2 The Rules for redundant propulsion and steering systems stipulate the level of redundancy for the propulsion and steering systems. It is characterized by the appropriate Notation to be affixed to the Character of Classification as defined in Chapter 101 - Classification and Surveys, Section 2, C.

1.3 The Rules are based on the single-failure concept.

1.4 Documents for approval

1.4.1 Compliance in accordance with the Notation applied for must be demonstrated by block diagrams, schematic drawings, descriptions of system functions and operation, calculations and arrangement plans.

Model tests or calculations shall be used to show the speed and manoeuvring qualities that have to be attained during sea trials in order to demonstrate compliance with the requirements set out in 2.

1.4.2 A failure mode and effects analysis (FMEA) or an equivalent analysis must be conducted for the propulsion and steering systems, and for the auxiliary systems and control systems needed to operate them.

The analysis must demonstrate that a single failure cannot lead to any loss in propulsion and/or in steering ability in accordance with the requirements set out in 2.

The analysis shall further demonstrate that measures are in place for failure detection and control of possible effects and that these measures are adequate to ensure in particular that the propulsion and steering of the ship can be rapidly restored.

In addition, the analysis must deal with the identification of possible failure conditions, which have a common cause. The identification of technical elements and/or operational procedures, which could undermine the redundancy concept, must also be accounted for.

For the Notation **RP1 x%**, the FMEA only has to be performed for the redundant propulsion machines and their requisite auxiliary systems. The events of water ingress or fire in a machinery compartment, and a failure of any of the common elements of the propulsion train related to this Notation do not have to be considered.

For the Notation **RP2 x%**, the FMEA has to be performed for the redundant propulsion and steering systems and their requisite auxiliary systems. The events of water ingress or fire in a machinery compartment and water ingress in a steering gear compartment do not have to be considered.

1.4.3 A programme of tests to be conducted during sea trials must be submitted for approval.

2. General Requirements

In accordance with the requirements set out in these Rules, it must be ensured that when a failure in a propulsion or steering system occurs,

2.1 the manoeuvrability of the ship can be maintained so that even under unfavourable weather conditions **(1)** the ship can be manoeuvred into a position of less resistance to the weather and can be maintained in this position,

2.2 a minimum speed can be maintained to keep the ship under control and ensure that it is able to make speed over the ground in waters where there is a strong current. The alternative propulsion system power capacity shall be such that it will enable the vessel to maintain a speed of not less than 7 knots.

2.3 the requirements stated in 2.1 and 2.2 can be met for a minimum period of 72 hours **(2)** .

2.4 the requirements stated in 2.1, 2.2 and 2.3 can be met irrespective of the ship's loading condition,

2.5 the redundant propulsion systems and steering systems are ready for operation at any time and can be activated on demand,

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(1) *Within the context of these Rules, unfavourable weather conditions are regarded as being a wind speed of up to and including 21 m/s (8 on the Beaufort scale) and a significant wave height of 5,4 m with an average wave period of 8,3 s.*

(2) *For ships, which normally spend less than 72 hours cruising at sea, the period specified may be limited to the maximum time of a voyage.*

2.6 the redundant propulsion system is capable of taking up operation from a still standing propulsion plant.

Compliance with the above requirements must be demonstrated by calculations and/or model tests and verified in a suitable manner during sea trials.

3. Requirements for auxiliary systems

3.1 Auxiliary systems for redundant propulsion systems which function have a direct effect on the propulsion system, for example fuel, lubrication oil, cooling water, control air and uninterrupted power supply systems, must be provided for each propulsion system independently of each other.

Where standby units are specified for these systems in accordance with **TL** Rules, these must be provided for each of the systems in question.

3.2 Auxiliary systems for redundant propulsion systems which failure do not have a direct effect on the propulsion system, such as fuel treatment, starting air supply systems etc. are to be designed to be separate from each other. For these systems no additional standby units have to be provided if interconnection lines are provided between the systems and if the units are designed so that the propulsion systems can be supplied with power simultaneously without restriction. In the connection lines shut-off valves are to be provided which must be kept closed during normal operation.

On ships with Class Notation **RP3 x%**, a shut-off valve must be fitted on either side of the partition bulkhead between the machinery compartments.

3.3 In fuel oil systems, the heating facilities for preheating the fuel oil must be designed so that if one propulsion system fails, the required preheating of the fuel oil for the redundant propulsion system can be ensured.

It is not necessary to provide a redundant heating facility if diesel oil storage tanks are provided which allow unrestricted operation for the redundant propulsion system for the period of time specified in 2.3.

3.4 Supply lines from fuel oil service tanks of redundant propulsion systems must be provided with an interconnection fitted between service tank and pump of each system. The interconnection is to be provided with a shut-off device, which must be kept closed during normal operation.

On ships with Class Notation **RP3 x%**, a shut-off valve must be fitted on either side of the partition bulkhead between the machinery compartments.

3.5 The seawater supply of redundant propulsion systems may be achieved via a common sea chest connection by means of a pump assigned to each propulsion system. The systems must be capable of being isolated by means of a shut-off valve in the connection line.

On ships with Class Notation **RP3 x%** the sea chests are to be installed in separate compartments in accordance with 6.1. The shut-off valve in the connection line must be fitted to the partition bulkhead and be capable of being operated either from both machinery compartments or from a position outside the machinery compartments.

3.6 On ships with Class Notation **RP3 x%** it must be possible to operate the redundant propulsion system when one of the seawater cooling systems fails, in accordance with the compartment separation requirements specified in 6.1.

3.7 Electric propulsion

3.7.1 In electric propulsion systems the main and excitation converter systems, and where appropriate, their supply transformers, their protection and control facilities and the corresponding Uninterrupted power supply systems (UPS) is to be designed in such a way that the redundant propulsion power of the ship remains available when a single failure occurs. Auxiliary systems (e.g. re-cooling devices and auxiliary power supplies) are to be designed so that they are separate from one another.

3.8 Redundant electric propulsion systems of naval ships shall be supplied from the switchboards of at least two electrical power generation plants, which are linked by an interconnection feeder. If one of the switchboard fails, the remaining one shall supply the propulsion system and its auxiliary power supplies, compare the **TL** Rules for Electrical Installations, Chapter 105, Section 4, H.

All equipment, which is primary essential for operation, shall be connected to at least two switchboards of electrical power generating plants.

The two switchboards shall be capable of being controlled and monitored independently of each other. Transitional power supplies necessary for this purpose have to be of redundant design.

Where power management systems are required to provide a reliable power supply to the propulsion systems, these shall also be of redundant design.

4. Control and Monitoring Systems

4.1 Controls

4.1.1 The redundant propulsion system is to be capable of being controlled by means of a simple control from the ship's bridge. A local control shall also be provided for emergency operation.

4.1.2 Common controls, e.g. joystick controls that operate redundant propulsion systems are to be designed so that a single failure does not affect an intact system, and the control remains possible without restriction by means of another method of control (individual control or emergency control).

4.1.3 In the case of multiple propulsion systems, a central emergency control facility is to be provided, for example from the machinery control room, at which it is possible to adjust the speed and direction of rotation of the propulsion machines centrally.

4.2 Monitoring devices

The redundant propulsion machines and their auxiliary systems are to be monitored by independent alarms.
Alarms and status indicators are to be provided.

5. Requirements for Steering Systems

5.1 Rudders

Every redundant steering system must consist of a main and an auxiliary steering gear, each with independent control.

The rudder position must be indicated by means of electrically independent rudder position indicators.

The ship's steering capability must be ensured even when the rudder is blocked at maximum deflection. If the steering ability is impaired to the extent that the requirements set out in 2. cannot be met, it must be possible to move and lock the failed rudder into the midships position.

5.2 Azimuth propulsion units as steering systems

Where ship steering is exclusively performed by azimuth propulsion systems, at least two azimuth propulsion systems must be provided, each with independent controls.

The position of the individual azimuth propulsion systems must be indicated by electrically independent indicators.

If the ship's steering ability is impaired, even when the propulsion of a defective azimuth propulsion system is disconnected, to the extent that the requirements stated in 2. cannot be met, it must be possible to move and to lock the defective azimuth propulsion unit into the midships position.

For all further details, see Section 7b.

6. Compartment Separation Requirements for RP3 x%

6.1 Bulkheads and partitions

6.1.1 Redundant propulsion systems and steering systems must be separated from each other by watertight bulkheads.

6.1.2 Partitions between machinery compartments containing redundant propulsion systems must comply with a fire resistance, the level of which depends on the fire potential of the machinery compartments. The partitions must keep their structural integrity in case of fire for at least 60 minutes. Fire insulation shall be provided if the function of essential machinery and equipment could be adversely affected.

6.1.3 Partition walls of machinery compartments, which are isolated from each other by tanks, cofferdams or other void spaces, must keep their structural integrity in case of fire for at least 60 minutes. The length or depths of that spaces shall be at least 500 mm.

6.1.4 Watertight doors may be permitted in accordance with **SOLAS** II-1 / Reg. 13 and Reg. 22. These have to be equipped with an open/closed status indication and a remote control facility on the bridge.

Watertight doors must not be regarded as emergency exits for machinery compartments.

6.2 Ventilation

Machinery compartments are to be fitted with independent ventilation systems.

7. Tests

Tests are to be performed during sea trials in accordance with an approved sea trial program. The tests are designed to prove that:

- the ship is able to meet the requirements defined
- the propulsion and steering systems have the necessary redundancy in line with the Notation applied for
- the conclusions drawn in the FMEA regarding the effects of failure conditions and measures to detect and control these failure conditions are correct and adequate

8. Further details

Further details concerning redundant systems are defined in **TL** Rules for Classification and Construction, Chapter 23 - Redundant Propulsion and Steering Systems.

SECTION 3

INTERNAL COMBUSTION ENGINES

A. General

1. Application

The requirements in this Section apply to internal combustion engines used as main propulsion units and auxiliary units (including emergency units) as well as to air compressors.

For the purpose of these requirements, internal combustion engines are:

- Diesel engines, fuelled with liquid fuel oil,
- Dual-fuel engines, fuelled with liquid fuel oil and/or gaseous fuel,
- Gas engines, fuelled with gaseous fuel.

Requirements for dual-fuel engines and gas engines are specified in N.

2. Definitions

2.1 Diesel engine type

- 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.

2.2 Rated power

2.2.1 Diesel engines are to be designed such that their rated power when running at rated speed according to the definitions of the engine manufacturer at ambient conditions as defined in Section 1, D. can be delivered as continuous power. Diesel engines are to be capable of continuous operation within power range ① in Fig. 3.1A and of short period operation in power range ②. The extent of the power ranges are to be stated by the engine manufacturer.

2.2.2 Continuous power is understood to mean the service standard 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**.**

2.2.3 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 **TL.**

2.2.4 Engines, which have to meet the requirements of a permanent low-load operation according to the mission statement of the naval ship, have to be designed with regard to bad combustion and low temperatures. Relevant measures and additional equipment have to be approved by **TL**.

2.2.5 After running on the test bed, the fuel delivery system of main engines is to be so adjusted that after installation on board overload power cannot be given delivered. The limitation of the fuel delivery system has to be secured permanently.

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

2.2.7 Subject to the approval by **TL**, diesel engines may be designed for a continuous power (fuel stop power) which cannot be exceeded.

2.2.8 For main engines, a power diagram (Fig. 3.1A) is to be prepared showing the power ranges within which the engine is able to operate continuously and for short periods under service conditions.

2.3 Ambient conditions

For all engines, which are used on ships with unrestricted range of service, the definition of the performance has to be based on the ambient conditions according to Section 1, D.

The defined seawater temperature has especially to be considered as inlet temperature to coolers for charge air coolant operating with seawater

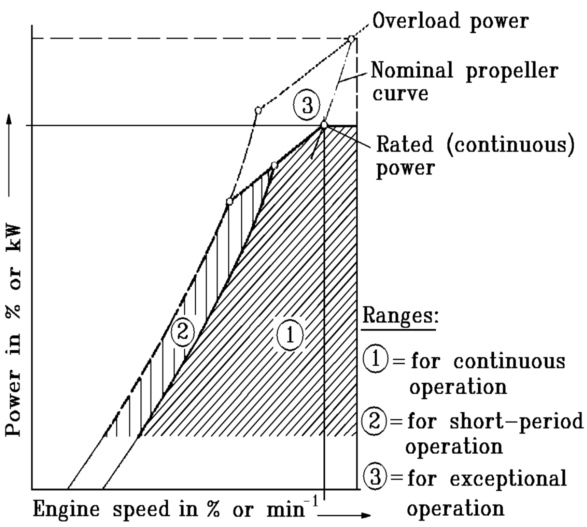


Fig.3.1A Example of a power diagram

3. Fuels

3.1 The use of liquid fuels is subject to the Rules contained in Section 1, F.

3.2 For fuel treatment and supply, see Chapter 107 - Ship Operation Installations and Auxiliary Systems, Section 7, B. and Section 8, G.

4. Accessibility of engines

Engines are to be so arranged in the engine room that all the assembly holes and inspection ports provided by the engine manufacturer for inspections and maintenance are accessible. A change of components, as far as practicable on board, must be possible. Requirements related to space and construction have to be considered for the installation of the engines.

5. Electronic components and systems

5.1 For electronic components and systems which are necessary for the control of internal combustion engines the following items have to be observed:

5.1.1 Electronic components and systems have to be type approved according to the **TL** Rules - Test Requirements for Electrical/Electronic Equipment and Systems.

5.1.2 For computer system the Rules, Chapter 105 - Electrical Installations, Section 10 have to be observed.

5..1.3 For main propulsion engines one failure of an electronic control system shall not result in a total loss or sudden change of the propulsion power. In individual cases, **TL** may approve other failure conditions, whereby it is ensured that no increase in ship's speed occurs.

5.1.4 The non-critical behaviour in case of a failure of an electronic control system has to be proven by a structured analysis (e.g. FMEA), which has to be provided by the system's manufacturer. This investigation shall include the effects on persons, environment and technical condition.

5.1.5 Where the electronic control system incorporates a speed control, F.2.3 of this Section and Chapter 105 - Electrical Installations, Section 9 have to be observed.

6. 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 TL-R M44 (Section 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 (see B.), forwarding the relevant production drawings and comparison list for the use of the Surveyors at the manufacturing plant and shipyard if necessary, engine testing (see E.) and upon satisfactorily meeting the Rule requirements, the issuance of an engine certificate.

B. Documents for Approval

1. General

For each engine type the drawings and documents listed in Table 3.1, Table 3.2 and Table 3.3 shall, wherever applicable, be submitted by the engine manufacturer to **TL** for information or approval or inspection of components and systems.

Where considered necessary, **TL** may request further documents to be submitted. This also applies to the documentation of design changes according to 3.

2. Engines Manufactured Under License

For each engine type manufactured under license, the licensee shall submit to **TL**, as a minimum requirement, the following documents:

- Comparison of all the drawings and documents as per Table 3.1, Table 3.2 and Table 3.3 - where applicable-indicating the relevant drawings used by the licensee and the licensor.

- All drawings of modified components, if available, as per Table 3.1, Table 3.2 and Table 3.3 together with the licensor's declaration of consent to the modifications,

- A complete set of drawings shall be put at the disposal of the local inspection office of **TL** as a basis for the performance of tests and inspections.

3. Design Modifications

Following initial approval of an engine type by **TL**, only those documents listed in Table 3.1, Table 3.2 and Table 3.3 are to be resubmitted for examination which embody important design modification.

4. Approval of Engine Components

The approval of exhaust gas turbochargers, heat exchangers, engine-driven pumps, etc. is to be requested from **TL** by the respective manufacturer.

5. Mass/Serial Produced Engines

Trunk engines manufactured in mass or in series, can be produced according to agreed survey arrangement in accordance with Classification and Survey Rules, Section 2, F. Alternative Certification Scheme. The scope and extent of the application of Alternative Certification Scheme are to be agreed on a case by case basis.

C. Materials

1. Approved Materials

1.1 The mechanical characteristics of materials used for the components of diesel engines shall conform to the **TL** Material Rules. The materials approved for the various components are shown in Table 3.4 together with their minimum required characteristics and material certificates.

1.2 Materials with properties deviating from the Rules specified may be used only with **TL**'s special approval. **TL** requires proof of the suitability of such materials.

1.3 If shock loads gain great importance for the naval ship, cast iron with lamellar graphit (GG) is not recommended for components exposed to such loads. It may only be used if it is proven that the shock loads are sufficiently reduced by adequate mountings

Table 3.1 Documentation to be submitted for information, as applicable

No.	**Item**	**Quantity**
1	Engine particulars (e.g. Data sheet with general engine information **(7)**, Project Guide, Marine Installation Manual)	3
2	Engine cross section	3
3	Engine longitudinal section	3
4	Bedplate and crankcase of cast design	1
5	Thrust bearing assembly **(1)**	3
6	Frame/framebox/gearbox of cast design **(2)**	1
7	Tie rod	1
8	Connecting rod	3
9	Connecting rod, assembly **(3)**	3
10	Crosshead, assembly **(3)**	3
11	Piston rod, assembly **(3)**	3
12	Piston, assembly **(3)**	3
13	Cylinder jacket/ block of cast design(2)	1
14	Cylinder cover, assembly **(3)**	1
15	Cylinder liner	1
16	Counterweights (if not integral with crankshaft), including fastening	3
17	Camshaft drive, assembly **(3)**	1
18	Flywheel	1
19	Fuel oil injection pump	1
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	3
	For electronically controlled engines, construction and arrangement of:	
21	Control valves	3
22	High-pressure pumps	3
23	Drive for high pressure pumps	3
24	Operation and service manuals**(4)**	1
25	FMEA (for engine control system) **(5)**	3
26	Production specifications for castings and welding (sequence)	1
27	Evidence of quality control system for engine design and in service maintenance	1
28	Quality requirements for engine production	1
29	Type approval certification for environmental tests, control components**(6)**	1

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. Only necessary if sufficient details are not shown on the transverse cross section and longitudinal section.*

(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. 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.*

(5) *And the system, where this is supplied by the engine manufacturer. 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. The FMEA reports required will not be explicitly approved by **TL**. For FMEA process of diesel engine control systems see TL- G 138.*

(6) *Tests are to demonstrate the ability of the control, protection and safety equipment to function as intended under the specified testing conditions per **TL** Additional Rules, Regulations for the Performance of the Type Tests Part 1 – Test Specification for Type Approval.*

(7) *According to TL- R M44, "Internal Combustion Engine Approval Application Form and Data Sheet" should be filled and submitted to **TL***

Table 3.2 Documentation to be submitted for approval, as applicable

No.	Item	Quantity
1	Bedplate and crankcase of welded design, with welding details and welding Instructions**(1) (2)**	3
2	Thrust bearing bedplate of welded design, with welding details and welding instructions **(1)**	3
3	Bedplate/oil sump welding drawings **(1)**	3
4	Frame/framebox/gearbox of welded design, with welding details and instructions**(1) (2)**	3
5	Engine frames, welding drawings **(1) (2)**	3
6	Crankshaft, details, each cylinder No.	3
7	Crankshaft, assembly, each cylinder No.	3
8	Crankshaft calculations (for each cylinder configuration) according to the TL-R M44 Appendix 3 and TL- R M53.	3
9	Thrust shaft or intermediate shaft (if integral with engine)	3
10	Shaft coupling bolts	3
11	Material specifications of main parts with information on non-destructive material tests and pressure tests **(3)**	3
	Schematic layout or other equivalent documents on the engine of:	
12	Starting air system	3
13	Fuel oil system	3
14	Lubricating oil system	3
15	Cooling water system	3
16	Hydraulic system	3
17	Hydraulic system (for valve lift)	3
18	Engine control and safety system	3
19	Shielding of high pressure fuel pipes, assembly **(4)**	3
20	Construction of accumulators (for electronically controlled engine)	3
21	Construction of common accumulators (for electronically controlled engine)	3
22	Arrangement and details of the crankcase explosion relief valve **(5)**	3
23	Calculation results for crankcase explosion relief valves	3
24	Details of the type test program and the type test report) **(7)**	1
25	High pressure parts for fuel oil injection system **(6)**	3
26	Oil mist detection and/or alternative alarm arrangements (see Table 3.7)	3
27	Details of mechanical joints of piping systems	3
28	Documentation verifying compliance with inclination limits	3
29	Documents as required in **TL** Chapter 5, Section 10 as applicable	3

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 **TL** 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 m^3 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 3.3 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 IACS UR M72.

No.	Item	Quantity
1	Engine particulars**(11)**	3
2	Material specifications of main parts with information on non-destructive material tests and pressure tests **(1)**	3
3	Bedplate and crankcase of welded design, with welding details and welding instructions **(2)**	3
4	Thrust bearing bedplate of welded design, with welding details and welding instructions **(2)**	3
5	Frame/framebox/gearbox of welded design, with welding details and instructions **(2)**	3
6	Crankshaft, assembly and details	3
7	Thrust shaft or intermediate shaft (if integral with engine)	3
8	Shaft coupling bolts	3
9	Bolts and studs for main bearings	3
10	Bolts and studs for cylinder heads and exhaust valve (two stroke design)	3
11	Bolts and studs for connecting rods	3
12	Tie rods	1
	Schematic layout or other equivalent documents on the engine of **(3)**	
13	Starting air system	3
14	Fuel oil system	3
15	Lubricating oil system	3
16	Cooling water system	3
17	Hydraulic system	3
18	Hydraulic system (for valve lift)	3
19	Engine control and safety system	3
20	Shielding of high pressure fuel pipes, assembly **(4)**	3
21	Construction of accumulators for hydraulic oil and fuel oil	3
22	High pressure parts for fuel oil injection system **(5)**	3
23	Arrangement and details of the crankcase explosion relief valve **(6)**	3
24	Oil mist detection and/or alternative alarm arrangements (see Table 3.7)	3
25	Cylinder head	1
26	Cylinder block, engine block	1
27	Cylinder liner	1
28	Counterweights (if not integral with crankshaft), including fastening	3
29	Connecting rod with cap	3
30	Crosshead	3
31	Piston rod	3
32	Piston, assembly **(7)**	3
33	Piston head	3
34	Camshaft drive, assembly **(7)**	1
35	Flywheel	1
36	Arrangement of foundation (for main engines only)	3
37	Fuel oil injection pump	1
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	3

No.	Item	Quantity
39	Construction and arrangement of dampers	3
	For electronically controlled engines, assembly drawings or arrangements of:	3
40	Control valves	3
41	High-pressure pumps	3
42	Drive for high pressure pumps	3
43	Valve bodies, if applicable	3
44	Operation and service manuals **(8)**	1
45	Test program resulting from FMEA (for engine control system) **(9)**	1
46	Production specifications for castings and welding (sequence)	1
47	Type approval certification for environmental tests, control components**(10)**	1
48	Quality requirements for engine production	1

Footnotes:

(1) *For comparison with **TL** 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. 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.*

(9) *Required for engines that rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves. For FMEA process of diesel engine control systems see TL- G 138.*

(10) *Documents modified for a specific application are to be submitted to **TL** for information or approval, as applicable. See TL- R M44 Item 3.2.2.2, Appendix 4 and Appendix 5.*

(11) *According to TL- R M44, Appendix 3 - "Internal Combustion Engine Approval Application Form and Data Sheet" should be filled and submitted to **TL**.*

Table 3.4 Approved materials and type of test certificate

Approved materials	TL's Rules (*)	Components	Test certificate (**) A	B	C
Forged steel $R_m \geq 360$ N/mm^2	Section 5,	Crankshafts	X	-	-
		Connecting rods	X	-	-
		Piston rods	X **(3)**	X **(4)**	-
		Crossheads	X **(3)**	X **(4)**	-
		Pistons and piston crowns	X **(3)**	X **(4)**	-
		Cylinder covers/heads	X	-	-
		Camshaft drive wheels	X **(3)**	X **(4)**	-
Rolled or forged steel rounds $R_m \geq 360$ N/mm^2	Section 5,	Tie rods	X	-	-
		Bolts and studs	X **(1)**	X **(2)**	-
Special grade cast steel $R_m \geq 440$ N/mm^2 and Special grade forged steel $R_m \geq 440$ N/mm^2	Section 6, Section 5,	Throws and webs of built-up crankshafts	X	-	-
Cast steel	Section 6,	Bearing transverse girders (weldable)	X	-	-
		Pistons and piston crowns	X **(3)**	X **(4)**	-
		Cylinder covers/heads	X **(1)**	X **(2)**	-
		Camshaft drive wheels	X **(3)**	X **(4)**	-
Nodular cast iron, preferably ferritic grades $R_m \geq 370$ N/mm^2	Section 7,	Engine blocks	-	X **(1)**	-
		Bed plates	-	X **(1)**	-
		Cylinder blocks	-	X **(1)**	-
		Pistons and piston crowns	X **(3)**	X **(4)**	-
		Cylinder covers/heads	-	X **(1)**	-
		Flywheels	-	X **(1)**	-
		Valve bodies	-	X **(1)**	-
Lamellar cast iron $R_m \geq 200$ N/mm^2	Section 7,	Engine blocks	-	-	X
		Bedplates	-	-	X
		Cylinder blocks	-	-	X
		Cylinder liners	-	-	X
		Cylinder covers/heads	-	-	X
		Flywheels	-	-	X
Shipbuilding steel, all **TL** grades for plate thickness ≤ 35 mm	Section 3,	Welded cylinder blocks	X	-	-
		Welded bedplates	X	-	-
Shipbuilding steel, **TL** grade B for plate thickness > 35 mm		Welded frames	X	-	-
		Welded housings	X	-	-
Structural steel, unalloyed, for welded assemblies	Section 3,				

(*) *All details refer to the **TL** Material Rules*

(**) *Test certificates are to be issued in accordance with **TL** Material Rules, Test Procedures, with the following abbreviations: A : **TL** Material Certificate, B: Manufacturer Inspection Certificate, C: Manufacturer Test Report*

(1) *Only for cylinder bores > 300 mm.*

(2) *For cylinder bores ≤ 300 mm.*

(3) *Only for cylinder bores > 400 mm.*

(4) *For cylinder bores ≤ 400 mm.*

D. Crankshaft Design

1. General

1.1 These Rules for the scantlings of crankshafts are to be applied to diesel engines for main propulsion and auxiliary purposes, where the engines are so designed as to be capable of continuous operation at their rated power when running at rated speed.

Crankshafts which cannot satisfy these Rules will be subject to special consideration as far as detailed calculations or measurements can be submitted.

In case of:

- Surface treated fillets,
- Tested parameters influencing the fatigue behaviour,
- Measured working stresses,

these data can be considered on special request.

1.2 Outside the end bearings, crankshafts designed according to the requirements specified in this section may be adapted to the diameter of the adjoining shaft by a generous fillet ($r \geq 0{,}06 \cdot d$) or a taper.

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

1.4 The scantlings of crankshafts are based on an evaluation of safety against fatigue in the highly stressed areas.

The calculation is also based on the assumption that the fillet transitions between the crankpin and web as well as between the journal and web are the areas exposed to the highest stresses.

The outlets of oil bores into crankpins and journals are to be formed in such a way that the safety margin against fatigue at the oil bores is not less than that acceptable in the fillets. The engine manufacturer, if requested by **TL** should submit a documentation supporting his oil bore design.

Calculation of crankshaft strength consists initially in determining the nominal alternating bending and nominal alternating torsional stresses which, multiplied by the appropriate stress concentration factors using the theory of constant energy of distortion (v. Mises' Criterion), result in an equivalent alternating stress (uni-axial stress). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material. This comparison will then show whether or not the crankshaft concerned is dimensioned adequately.

1.5 For the calculation of crankshafts, the documents and particulars listed in the following are to be submitted:

- Crankshaft drawing which must contain all data in respect of the geometrical configuration 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 [min^{-1}]
- Sense of rotation (see Fig. 3.1)
- Ignition sequence with the respective ignition intervals and, where necessary, V-angle α_V (see Fig. 3.1)

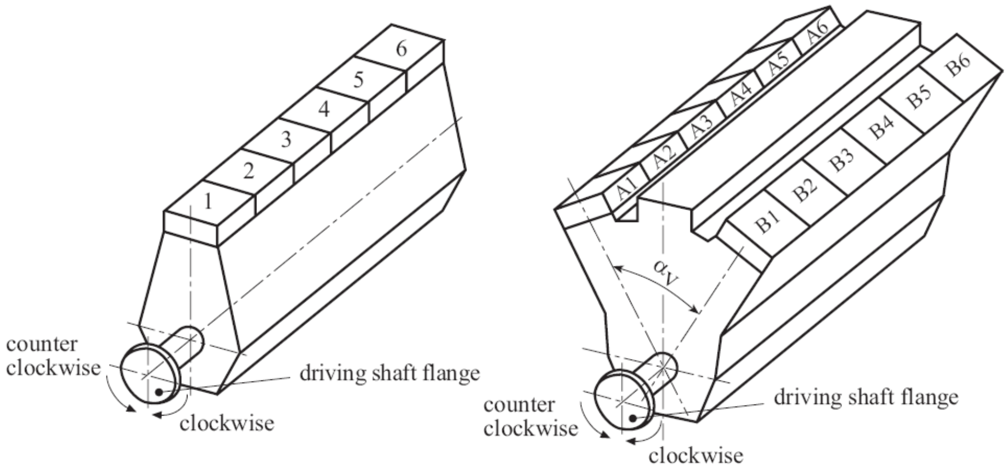


Fig. 3.1 Designation of the cylinders

- Cylinder diameter [mm]
- Stroke [mm]
- Maximum cylinder pressure P_{max} [bar]
- Charge air pressure [bar] (before inlet valves or scavenge ports, whichever applies)
- Connecting rod length L_H [mm]
- Oscillating weight of one crank gear [kg] (in case of V-type engines, where necessary, also for the cylinder unit with master and articulated type connecting rod or forked and inner connecting rod)
- Digitalized gas pressure curve presented at equidistant intervals (bar versus crank angle, but not more than 5° CA)
- For engines with articulated-type connecting rod (see Fig. 3.2)
  - Distance to link point L_A [mm]
  - Link angle α_N [°]
  - Connecting rod length L_N [mm]
- Details of crankshaft material
  - Material designation (according to DIN, AISI, etc.)
  - Mechanical properties of material (minimum values obtained from longitudinal test specimens)
    - Tensile strength [N/mm^2]
    - Yield strength [N/mm^2]
    - Reduction in area at fracture [%]
    - Elongation A5 [%]
    - Impact energy – KV [J]
  - 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 specified so as to enable calculation according to Appendix V
- Particulars for alternating torsional stresses, see 2.2.

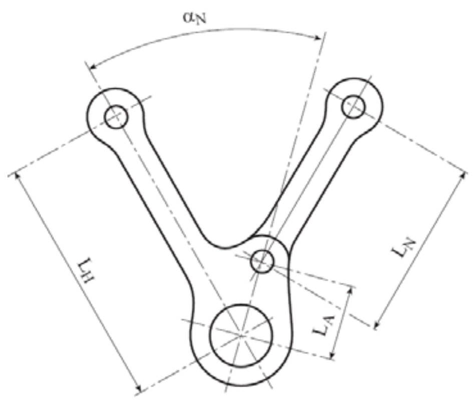


Fig. 3.2 Articulated-type connecting rod

2. Calculation of Stresses

2.1 Calculation of alternating stresses due to bending moments and shearing forces

2.1.1 Assumptions

The calculation is based on a statically determinate system, so that only one single crank throw is considered of which the journals are supported in the centre of adjacent bearings and which is subject to gas and inertia forces. The bending length is taken as the length between the two main bearings (distance L3) see Figs. 3.3 and 3.4.

The bending moments, M_{BR} and 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 Fig. 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 Fig. 3.4)

2.1.1.1 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 L1) 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. 3.6).

Mean stresses are neglected.

2.1.1.2 Bending acting in outlet of crankpin oil bore

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

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

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

The alternating stresses due to these bending moments are to be related to the crosssectional area of the axially bored crankpin.

Mean bending stresses are neglected.

2.1.2 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 2.1.1.1 and 2.1.1.2 - 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 = Alternating force, moment or stress

X_{max} = Maximum value within one working cycle

X_{min} = Minimum value within one working cycle

2.1.2.1 Nominal alternating bending and compressive stresses in web cross section

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

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

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

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 fig. 3.3 and 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:

Ke = 0.8 for 2-stroke engines

Ke = 1.0 for 4-stroke engines

$$Q_{RFN}=\pm\frac{1}{2}\left(Q_{RF,max}-Q_{RF,min}\right)$$

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 (see fig. 3.3 and 3.4) [N]

$$F=B.W$$

F = Area related to cross-section of web [mm²]

2.1.2.2 Nominal alternating bending stress in outlet of crankpin oil bore

The calculation of nominal alternating bending stress is as follows:

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

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,[Nm]

$$M_{BON}=\pm\frac{1}{2}\left(M_{BO,max}-M_{BO,min}\right)$$

With

$$M_{BO}=(M_{BTO}\cos\psi+M_{BRO}\sin\psi)$$

and ψ [º] angular position (see fig. 3.5)

W_e = Section modulus related to cross-section of axially bored crankpin, [mm³]:

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

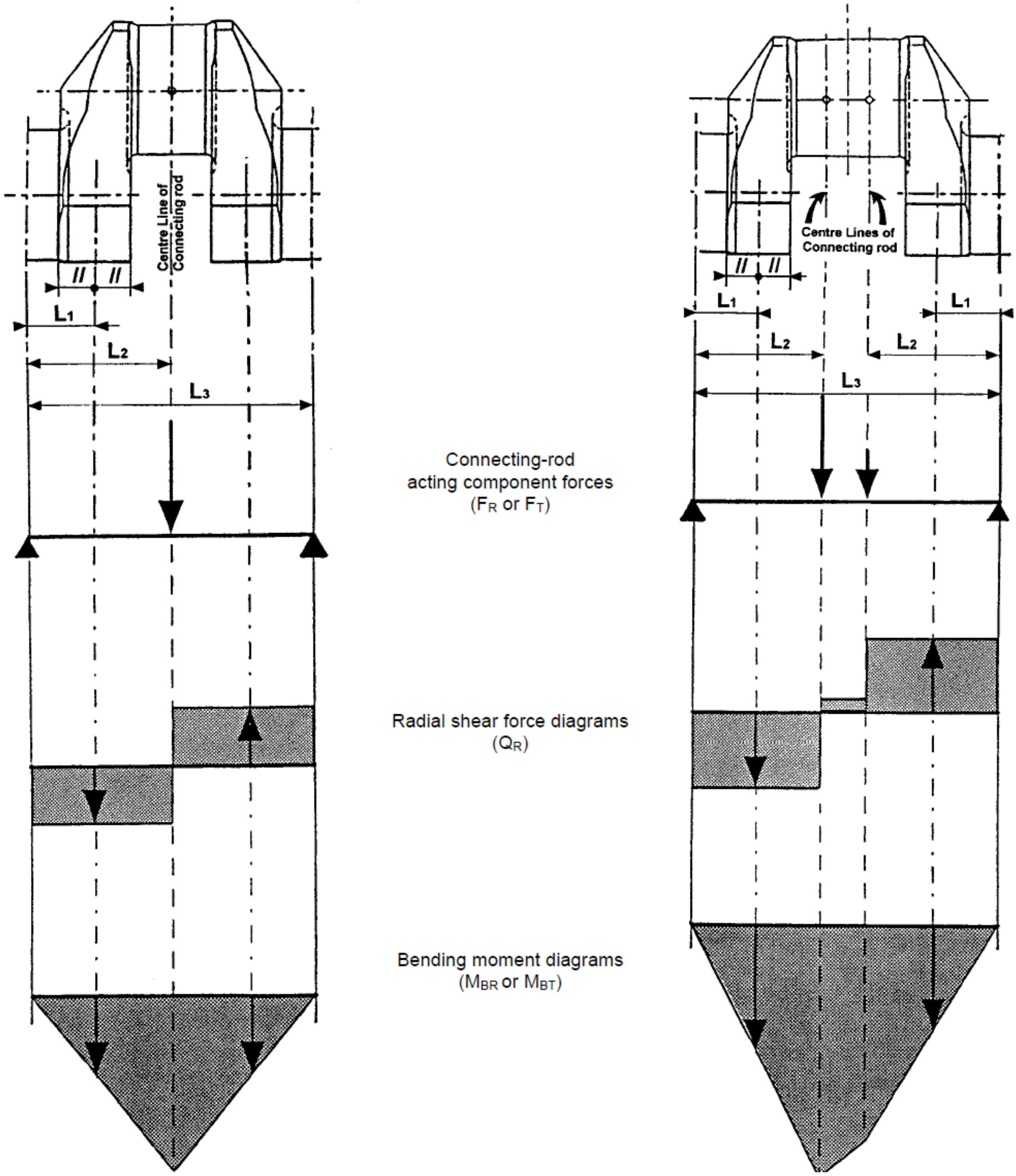


Fig. 3.3 Crankthrow for in line engine

Fig. 3.4 Crankthrow for Vee engine with 2 adjacent connecting-rods

L1 = Distance between main journal centre line and crankweb centre (see also Fig 3.5 for crankshaft without overlap)
L2 = Distance between main journal centre line and connecting-rod centre
L3 = Distance between two adjacent main journal centre lines

2.1.3 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:

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

σ_{BH} = Alternating bending stress in crankpin fillet [N/mm²],

α_B = Stress concentration factor for bending in crankpin fillet [–] (determination, see 3.).

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

$$\sigma_{BG} = \pm\left(\beta_B \cdot \sigma_{BN} + \beta_Q \cdot \sigma_{QFN}\right)$$

σ_{BG} = Alternating stresses in journal fillet [N/mm²],

β_B = Stress concentration factor for bending in journal fillet [–] (determination, see 3.),

β_Q = Stress concentration factor for shearing [–] (determination, see 3.).

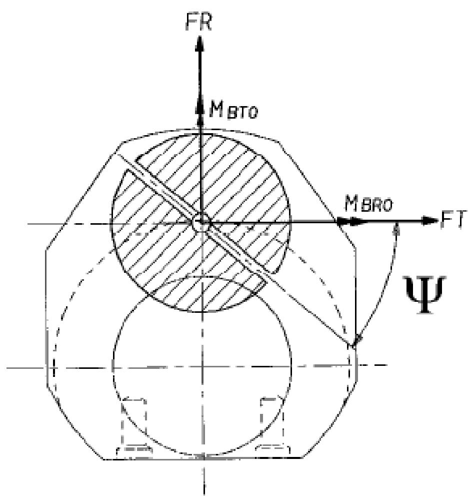


Fig. 3.5 Crankpin section through the oil bore

2.1.4 Calculation of alternating bending stresses in outlet of crankpin oil bore

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

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 item 3)

2.2 Calculation of alternating torsional stresses

2.2.1 General

The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in 2.1.2. The manufacturer shall specify the maximum nominal alternating torsional stress.

The maximum value obtained from such calculations will be used by **TL** when determining the equivalent alternating stress, according to 5. In the absence of such a maximum value it will be necessary for **TL** to incorporate a fixed value in the calculation for the crankshaft dimensions on the basis of an estimation.

In case **TL** is entrusted with carrying out a forced vibration calculation on behalf of the engine manufacturer to determine the torsional vibration stresses to be expected in the engine and possibly in its shafting, the following data are to be submitted to **TL** additionally to 1.5:

- Equivalent dynamic system of the engine comprising
  - Mass moment of inertia of every mass point [kgm²]
  - Inertialess torsional stiffnesses [Nm/rad]
- Vibration dampers

- Type designation
- Mass moments of inertia [kgm^2]
- Inertialess torsional stiffnesses [Nm/rad]
- Damping coefficients [Nms]

- Flywheels
  - Mass moment of inertia [kgm^2]

If the whole installation is to be considered, the above information is to be extended by the following:

- Coupling
  - Dynamic characteristics and damping data
- Gearing data
  - Shaft diameter of gear shafts, thrust shafts, intermediate shafts and propeller shafts
- Shafting
  - Diameter of thrust shafts, intermediate shafts and propeller shafts
- Propellers
  - Propeller diameter
  - Number of blades
  - Pitch and area ratio
- Natural frequencies with their relevant modes of vibration and the vector sums for the harmonics of the engine excitation.
- Estimated torsional vibration stresses in all important elements of the system with particular reference to clearly defined resonance speeds of rotation and continuous operating ranges.

2.2.2 Calculation of nominal alternating torsional stresses

The maximum and minimum alternating torques are to be ascertained for every mass point of the 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 exist in the system and for unfavourable conditions (misfiring 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 and the transient response can be recorded with sufficient accuracy. Misfiring is defined as cylinder condition when no combustion occurs but only compression cycle.

The values received from such calculation are to be submitted.

The nominal alternating torsional stress in every mass point, which is essential to the assessment, results from the following equation:

$$\tau_N = \pm \frac{M_T}{W_p} \cdot 10^3$$

$$M_{TN} = \pm \frac{1}{2} (M_{Tmax} - M_{Tmin})$$

$$W_p = \frac{\pi}{16} \left(\frac{D^4 - D_{BH}^4}{D} \right)$$

or

$$W_p = \frac{\pi}{16} \left(\frac{D_G{}^4 - D_{BG}^4}{D_G} \right)$$

τ_N = Nominal alternating torsional stress referred to crankpin or journal [N/mm^2],

M_{TN} = Maximum alternating torque [Nm],

W_p = Polar section modulus related to cross section of axially bored crankpin or bored journal [mm^3],

M_{Tmax} = Maximum value of the torque [Nm].

M_{Tmin} = Minimum value of the torque [Nm].

The assessment of the crankshaft is based on the torsional stress which in conjunction with the associated bending stress, results in the lowest acceptability factor. Where barred speed ranges are necessary, the torsional stresses within these ranges are to be neglected in the calculation of the acceptability factor.

Barred speed ranges are to be so arranged that satisfactory operation is possible despite of their existence. There are to be no barred speed ranges above a speed ratio of $\lambda \geq 0{,}8$ of the rated speed.

The approval of crankshafts is to be based on the installation having the lowest acceptability factor.

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.

2.2.3 Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore

The calculation of stresses is to be carried out for both the crankpin and the journal fillet.

For the crankpin fillet:

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

τ_H = Alternating torsional stress in crankpin fillet [N/mm^2],

α_T = Stress concentration factor for torsion in crankpin fillet [–] (determination, see C.).

τ_N = Nominal alternating torsional stress related to crankpin diameter [N/mm^2]

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

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

τ_G = Alternating torsional stress in journal fillet [N/mm^2],

β_T = Stress concentration factor for torsion in journal fillet [–] (determination, see C.).

τ_N = Nominal alternating torsional stress related to journal diameter [N/mm^2],

For the outlet of crankpin oil bore:

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

σ_{TO} = Alternating stress in outlet of crankpin oil bore due to torsion[N/mm^2]

γ_T = Stress concentration factor for torsion in outlet of crankpin oil bore[-] (determination-see item C)

τ_N = Nominal alternating torsional stress related to crankpin diameter [N/mm^2]

3. Evaluation of Stress Concentration Factors

3.1 General

The stress concentration factors are evaluated by means of the formulae according to items 3.2, 3.3 and 3.4 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 item D.3.5 may be undertaken.

The stress concentration factors for bending (α_B, β_B) are defined as the ratio of the maximum bending stress – occurring in the fillets under bending load acting in the central cross-section of a crank – to the nominal stress related to the web cross-section.

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 torsion (α_T, β_T) are defined as the ratio of the maximum torsional stress occurring under torsional load in the fillets to the nominal stress related to the bored crankpin or journal 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.

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 **TL** 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 3.2 to 3.4.

Appendix III and VI describes how FE analyses can be used for the calculation of the stres concentration factors. Care should be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

All crank dimensions necessary for the calculation of stress concentration factors are shown in Fig. 3.6.

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 [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:

  $W_{red} = W - (T_H - R_H)$ (refer to fig. 3.6)

- Web width B must be taken in way of crankpin fillet radius centre according to fig. 3.6

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

Table 3.5 Stress concentration factors

Crankpin Fillets	Journal Fillets
$r = R_H / D$	$r = R_G / D$
$s = S/D$	
$w = W/D$ crankshafts with overlap	
$w = W_{red}/D$ 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. Ranges are as follows:

$s \leq 0.5$

$0.2 \leq w \leq 0.8$

$1.1 \leq b \leq 2.2$

$0.03 \leq r \leq 0.13$

$0 \leq d_G \leq 0.8$

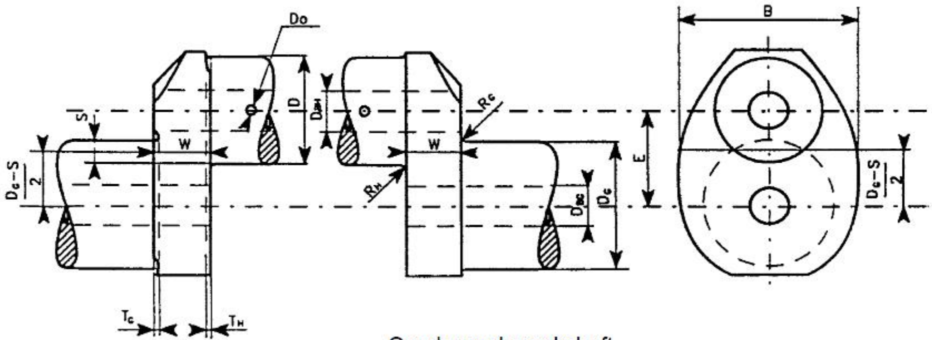
$0 \leq d_H \leq 0.8$

$0 \leq d_O \leq 0.2$

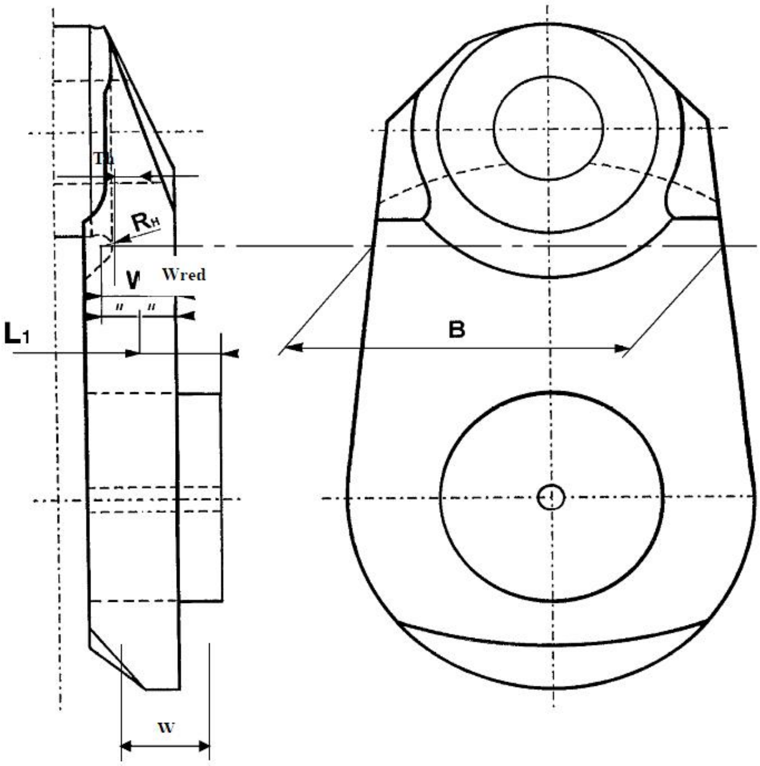
Low range of s can be extended down to large negative values provided that:

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.



Overlapped crankshaft



Crankshaft without overlap

Fig. 3.6 Crank dimensions necessary for the calculation of stress concentration factors

3.2 Crankpin fillet

The stress concentration factor for bending α_B is:

α_B = $2.6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(recess)$

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:

α_T = $0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$

f(r,s) = $r^{[-0.322 + 0.1015 \cdot (1 - s)]}$

f(b) = $7.8955 - 10.654 . b + 5.3482 \cdot b^2 - 0.857 \cdot b^3$

f(w) = $w^{-0.145}$

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

The stress concentration factor for bending β_B is:

β_B = $2.7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(recess)$

$f_B(s,w)$ = $-1.7625 + 2.9821 \cdot w - 1.5276 \cdot w^2 + (1 - s) \cdot (5.1169 - 5.8089 \cdot w + 3.1391 \cdot w^2) + (1 - s)^2 \cdot (- 2.1567 + 2.3297 \cdot w - 1.2952 \cdot w^2)$

$f_B(w)$ = $2.2242 \cdot w^{0,7548}$

$f_B(b)$ = $0.5616 + 0.1197 \cdot b + 0.1176 \cdot b^2$

$f_B(r)$ = $0.1908 \cdot r^{-0.5568}$

$f_B(d_G)$ = $1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$

$f_B(d_H)$ = $1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$

f(recess) = $1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$

The stress concentration factor for shearing β_Q is:

β_Q = $3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(recess)$

$f_Q(s)$ = $0.4368 + 2.6130 \cdot (1 - s) - 1.5212 \cdot (1 - s)^2$

$f_Q(w)$ = $w / (0.0637 + 0.9369 \cdot w)$

$f_Q(b)$ = $- 0.5 + b$

$f_Q(r)$ = $0.5331 \cdot r^{-0.2038}$

$f_Q(d_H)$ = $0.9937 - 1.1949 \cdot d_H + 1.7373 \cdot d_H^2$

f(recess) = $1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$

The stress concentration factor for torsion β_T is:

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

β_T = α_T

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

β_T = $0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$f(r,s), f(b) and f(w) are to be determined in accordance with 3.2. (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 Outlet of crankpin oil bore

The stress concentration factor for bending (γ_B) is:

γ_B = $3 - 5{,}88\ d_O + 34{,}6\ d_O^2$

The stress concentration factor for torsion (γ_T) is:

γ_T = $4 - 6\ d_O + 30\ d_O^2$

4. Additional Bending Stresses

In addition to the alternating bending stresses in fillets (see 2.1.3) 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:

Table 3.6 Additional stress

Type of engine	σ_{add} [N/mm²]
Crosshead engines	± 30 **(*)**
Trunk piston engines	± 10

(*) *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.

5. Calculation of Equivalent Alternating Stress

5.1 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 Appendix I).

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.

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 Appendix II).

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.

5.2 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:

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

σ_V = Equivalent alternating stress [N/mm²]

For other parameters, see D.2.1.3, D.2.2.3 and D.4.

6. Calculation of Fatigue Strength

The fatigue strength is to be understood as that value of alternating bending stress which a crankshaft can permanently withstand at the most highly stressed points of the fillets: Where the fatigue strength for a crankshaft cannot be furnished by reliable measurements, the fatigue strength may be evaluated by means of the following formulae:

Related to the crankpin diameter:

$$\sigma_{DW} = \pm K \cdot (0{,}42 \cdot \sigma_B + 39{,}3) \cdot \left[0{,}264 + 1{,}073 \cdot D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{R_X}}\right]$$

R_X = R_H in the fillet area

R_X = $D_o/2$ in the oil bore area

Related to the journal diameter:

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

σ_{DW} = Allowable fatigue strength of crankshaft [N/mm^2],

K = Factor for different types of forged and cast 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 a **TL** approved cold rolling process.

σ_B = Minimum tensile strength of crankshaft material [N/mm^2].

For other parameters see D.3.1.

When a surface treatment process is applied, it must be approved by **TL**. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Appendix V.

However, it is to be considered that, the 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 and for calculation purposes R_H, R_G and R_X are not to be taken 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 Appendix IV.

7. Acceptability Criteria

The sufficient dimensioning of 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, the outlet of crankpin oil bore and is based on the formula:

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

Q = Acceptability factor [–]

Adequate dimensioning of the crankshaft is ensured if the smallest of all acceptability factors satisfies the criteria:Q ≥ 1.15

8. Calculation of Shrink-fits of Semi-Built Crankshafts

8.1 General

All crank dimensions necessary for the calculation of the shrink-fit are shown in Fig. 3.7.

D_S = Shrink diameter [mm],

L_S = Length of shrink-fit [mm],

D_A = Outside diameter of web [mm],
or
twice the minimum distance x between centreline of journals and outer contour of web, whichever is less.

y = Distance between the adjacent generating lines of journal and pin [mm].

$y \geq 0{,}05\ D_S$

Where y is less than 0,1 . D_S, special consideration is to be given to the effect of the stress due to the shrink on the fatigue strength at the crankpin fillet.
For other parameter, see D.3.1 (Fig. 3.6).

Regarding the radius of the transition from the journal to the shrink diameter, the following must be observed:

$R_G \geq 0.015 \cdot D_G$ 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 items 8.2. and 8.3.

In the case where 8.2 condition cannot be fulfilled then 8.3 and 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.

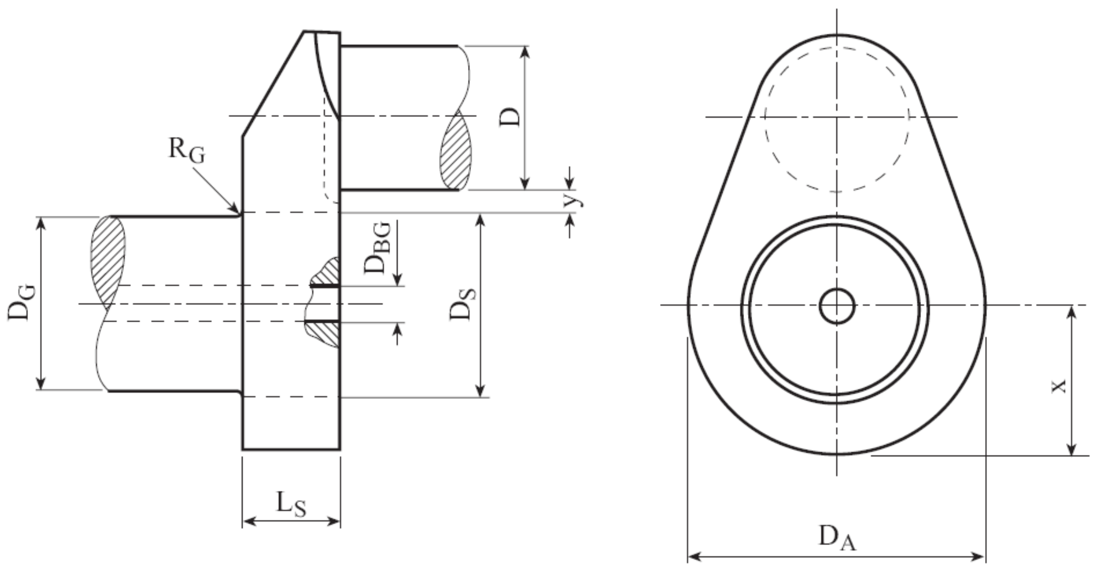


Fig. 3.7 Crank throw of semi-built crankshaft

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:

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

S_R = Safety factor against slipping, [-], 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} [Nm] in accordance with 2.2.2

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/mm^2]

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

8.3 Necessary minimum oversize of shrink-fit

The necessary minimum oversize is determined by the greater value calculated in accordance with 8.3.1 and 8.3.2.

8.3.1 The calculation of the minimum oversize is to be carried out for the crank throw with the absolute maximum torque M_{max}. The torque M_{max} corresponds to the maximum value of the torque M_{Tmax}, ascertained as per D.2.2 for the various mass points of the crankshaft.

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

$$Q_A=\frac{D_S}{D_A}$$

$$Q_S=\frac{D_{BG}}{D_S}$$

Z_{min} = Minimum oversize [mm],

S_R = Safety factor against slipping, however a value not less than 2 is to be taken [–],

Q_A, Q_S = Ratio of different diameters [–],

μ = Coefficient for static friction, see 8.2 [–],

E_m = Young's modulus [N/mm^2].

8.3.2 In addition to 8.3.1 the minimum oversize is also to be calculated according to the following formula:

$$Z_{min}\geq\frac{\sigma_{SW}\cdot D_S}{E_m}$$

σ_{SW} = Minimum yield strength of material for crank web [N/mm^2].

8.4 Maximum permissible oversize of shrink-fit

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

$$Z_{max}\leq\frac{\sigma_{SW}\cdot D_S}{E_m}+\frac{0.8\cdot D_S}{1000}$$

Z_{max} = Maximum oversize [mm].

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

E. Tests and Trials

1. Engine Manufacturer’s Workshop and Manufacturing Inspections

1.1 The manufacture of all engines with **TL** classification is subject to supervision by **TL** and the manufacturer’s works are to be audited by **TL**. The scope should be agreed between manufacturer and **TL**.

1.2 Where engine manufacturers have been approved by the **TL** as "Suppliers of Mass Produced Internal Combustion Engines", these engines are to be tested in accordance with 4.

1.3 Every workshop where engines are assembled and tested are to be approved by **TL** if the workshop is newly set up or a new production line is set up or a new engine type is introduced or a new production process is implemented.

1.4 Manufacturer's works have to have suitable production and testing facilities, competent staff and a quality management system, which ensures a uniform production quality of the products according to the specification.

- Manufacturing plants shall be equipped in such a way that all materials and components can be machined and manufactured to a specified standard. Production facilities and assembly lines, including machining units, welding processes, special tools, special devices, assembly and testing rigs as well as lifting and transportation devices shall be suitable for the type and size of engine, its components, and the purpose intended. Materials and components shall be manufactured in compliance with all production and quality instructions specified by the manufacturer and recognised by **TL**.

- Suitable test bed facilities for load tests have to be provided, if required also for dynamic response testing. All liquids used for testing purposes such as fuel oil, lubrication oil and cooling water shall be suitable for the purpose intended, e.g. they shall be clean, preheated if necessary and cause no harm to engine parts.

- Trained personnel shall be available for production of parts, assembly, testing and partly dismantling for shipping, if applicable.

- Storage, reassembly and testing processes for diesel engines at shipyards shall be such that the risk of damage to the engine or its parts is minimized.

  Engine manufacturer's workshops shall have in place a Quality Management System recognized by **TL**.

1.5 Pressure Tests

The individual components of internal combustion engines are to be subjected to pressure tests at the pressures specified in Table 3.6C.

2. Certification of Engine Components

2.1. General

2.1.1 The engine manufacturer is to have a quality control system that is suitable for the actual engine types to be certified by **TL**. The quality control system is also to apply to any sub-suppliers. **TL** 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. **TL** requires that certain parts are verified and documented by means of **TL** Certificate (SC), Work Certificate (W) or Test Report (TR).

2.1.2 TL Certificate (SC)

This is a document issued by **TL** 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).

2.1.3 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 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 a **TL** Certificate and endorsed by **TL** if:

- The test was witnessed by **TL** Surveyor; or

- An Alternative Certification Scheme (ACS) agreement is in place between **TL** 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 the Society and independent from the manufacturer and/or material supplier.

2.1.4 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.

2.1.5 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 **TL,** 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.

2.1.6 The Surveyor is to review the TR and W for compliance with the agreed or approved specifications. SC means that the Surveyor also witnesses the testing, batch or individual, unless an ACS provides other arrangements.

2.1.7 The manufacturer is not exempted from responsibility for any relevant tests and inspections of those parts for which documentation is not explicitly requested by **TL.**

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.

2.2 Parts to be documented

2.2.1 The extent of parts to be documented depends on the type of engine, engine size and criticality of the part.

2.2.2 Symbols used are listed in Table 3.6.A. A summary of the required documentation for the engine components is listed in Table 3.6.B.

2.2.3 For components and materials not specified in Table 3.6.B, consideration will be given by **TL** upon full details being submitted and reviewed.

3. Type Tests of Diesel Engines

3.1 General

3.1.1 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.

3.2. Objectives

3.2.1 The type testing 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.

Table 3.6.A Symbols used in Table 3.6.B

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
SC	**TL** certificate
TR	Test report
UT	Ultrasonic testing
W	Work certificate
X	Visual examination of accessible surfaces by the Surveyor

Table 3.6.B 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	SC
2	Bearing transverse girders GS	W (C+M)	W (UT+CD)			X	All	SC
3	Welded frame box	W (C+M)	W (UT+CD)			Fit-up+ post welding	All	SC
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	SC
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	SC
13	Forged cylinder head	W (C+M)	W (UT+CD)	W		X	D> 300 mm	SC
14	Piston crown GS	W (C+M)	W (UT+CD)			X	D> 400 mm	SC
15	Forged piston crown	W (C+M)	W (UT+CD)			X	D> 400 mm	SC
16	Crankshaft: made in one piece	SC (C+M)	W (UT+CD)		W	Random, of fillets and oil bores	All	SC
17	Semi-built crankshaft (Crankthrow, forged main journal and journals with flange)	SC(C+M)	W(UT+CD)		W	Random, of fillets and shrink fittings	All	SC

Table 3.6.B Summary of required documentation for engine components (continued)

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
18	Exhaust gas valve cage			W			CH	
19	Piston rod	SC (C+M)	W (UT+CD)			Random	D > 400 mm CH	SC
20	Cross head	SC (C+M)	W (UT+CD)			Random	CH	SC
21	Connecting rod with cap	SC (C+M)	W (UT+CD)		W	Random, of all surfaces, in particular those shot peened	All	SC
22	Coupling bolts for crankshaft	SC (C+M)	W (UT+CD)		W	Random, of interference fit	All	SC
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	SC
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	
30	High pressure common servo oil system	W (C+M)		W			D> 300 mm	
		W (C+M)		TR			D≤ 300 mm	

Table 3.6.B Summary of required documentation for engine components (continued)

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
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 metal		W		> 800 kW/cyl	

Notes :

(1) *Material properties include chemical composition and mechanical properties, and also surface treatment such as surface hardening (hardness, depth and extent), peening and rolling (extent and applied force).*

(2) *Non-destructive examination means e.g. ultrasonic testing, crack detection by MPI or DP. When certain NDE method on the finished component is impractical (for example UT for items 12/13), the NDE method can be performed at earlier appropriate stages in the production of the component, see E.2.1.2.*

(3) *Hydraulic testing is applied on the water/oil side of the component. Items are to be tested by hydraulic pressure at the pressure equal to 1.5 times the maximum working pressure. High pressure parts of the fuel injection system are to be tested by hydraulic pressure at the pressure equal to 1.5 maximum working pressure or maximum working pressure plus 300 bar, whichever is the less. Where design or testing features may require modification of these test requirements, special consideration may be given.*

(4) *Material certification requirements for pumps and piping components are dependent on the operating pressure and temperature. Requirements given in this Table apply except where alternative requirements are explicitly given in Chapter 4, Sections 14 and 16.*

(5) *For turbochargers, see Chapter 4, Section 3.*

(6) *Crankcase explosion relief valves are to be type tested..*

(7) *Oil mist detection systems are to be type tested.*

(8) *For speed governor and overspeed protective devices.*

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

(10) *Hydraulic testing is also required for those parts filled with cooling water and having the function of containing the water which is in contact with the cylinder or cylinder liner.*

Table 3.6C Pressure tests (1)

Component		Test pressure, p_p [bar] **(2)**
Cylinder cover, cooling water space **(3)**		7
Cylinder liner, over whole length of cooling water space **(5)**		7
Cylinder jacket, cooling water space		4, at least 1,5 . p_{eperm} **(2)**
Exhaust valve, cooling water space		4, at least 1,5 . p_{eperm}
Fuel injection system	Pump body, pressure side	1,5 . p_{eperm} or p_{eperm} + 300 (whichever is less)
	Valves	1,5 . p_{eperm} or p_{eperm} + 300 (whichever is less)
	Pipes	1,5 . p_{eperm} or p_{eperm} + 300 (whichever is less)
Hydraulic system	High pressure piping for hydraulic drive of exhaust gas valves	1,5 . p_{eperm}
Exhaust gas turbocharger, cooling water space		4, at least 1,5 . p_{eperm}
Exhaust gas line, cooling water space		4, at least 1,5 . p_{eperm}
Coolers, both sides **(4)**		4, at least 1,5 . p_{eperm}
Engine-driven pumps (oil, water, fuel and bilge pumps)		4, at least 1,5 . p_{eperm}
Starting and control air system before installation		1,5 . p_{eperm}

(1) *in general, items are to be tested by hydraulic pressure as indicated in the Table. Where design or testing features may require modification of these test requirements, special consideration will be given*

(2) . *p_{eperm} [bar] = maximum working pressure in the part concerned*

(3) *for forged steel cylinder covers test methods other than pressure testing may be accepted, e.g. suitable non-destructive examination and dimensional control properly recorded*

(4) *charge air coolers need only be tested on the water side.*

(5) *for centrifugally cast cylinder liners, the pressure test can be replaced by a crack test*

3.2.2 The type testing 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.

3.3. Validity

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

3.3.2 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 **TL**.

3.3.3 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

---

(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. According to **TL**, 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.*

- 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.

3.4. Safety precautions

3.4.1 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.

3.4.2 This applies especially to crankcase explosive conditions protection, but also over-speed protection and any other shut down function.

3.4.3 The inspection for jacketing of high-pressure fuel oil lines and proper screening of pipe connections (as required in 3.8.9 fire measures) is also to be carried out before the test runs.

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

3.5 Test programme

3.5.1 The type testing is divided into 3 stages:

3.5.1.1. 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 **TL** or stipulated by the designer are to be presented to **TL** before starting stage B.

3.5.1.2. Stage B - witnessed tests

This is the testing made in the presence of **TL** Surveyor.

3.5.1.3. Stage C - component inspection

This is the inspection of engine parts to the extent as required by **TL**.

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

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

3.5.4 Upon completion of complete type testing (stage A through C), a type test report is to be submitted to **TL** 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 quality assurance management for presentation to **TL**.

- Detailed description of the load and functional tests conducted during stage B.

- Inspection results from stage C.

3.5.5 As required in 3.2 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.

3.5.6 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 3.3.

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.

3.6 Measurements and recordings

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

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

- Engine r.p.m.
- Torque
- Maximum combustion pressure for each cylinder **(3)**
- Mean indicated pressure for each cylinder **(3)**
- 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).

3.7 Stage A - internal tests

3.7.1 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.

3.7.2 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 **TL** or stipulated by the designer.

(3) *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.

3.8 Stage B - witnessed tests

3.8.1 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.

3.8.2 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.

3.8.3 Load points

The engine is to be operated according to the power and speed diagram (see Fig. 3.8). The data to be measured and recorded when testing the engine at the various load points have to include all engine parameters listed in 3.6. 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.

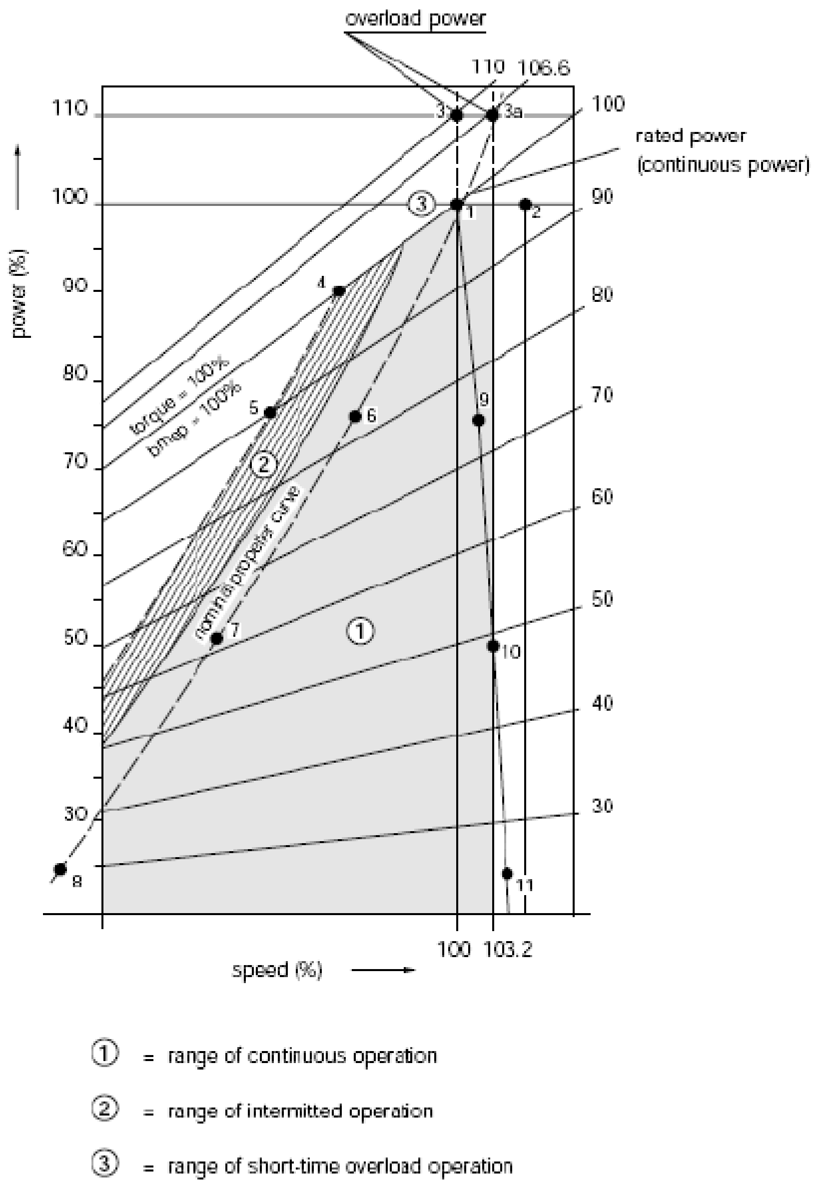


Fig. 3.8 – Load points

3.8.4 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.

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

3.8.6 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.)

Note:

The engine manufacturer is to state whether the achievable output is continuous. If there is a time limit, the permissible operating time is to be indicated.

3.8.7 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 item F. are to be carried out.

3.8.8 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 **TL** for selected cases based on the FMEA required in Table 2.1.

3.8.9 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 **TL**. Readings obtained are to be randomly verified by use of contact thermometers.

3.9 Stage C - Opening up for Inspections

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

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

3.9.3 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.)

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

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

4. Factory Acceptance Test and Shipboard Trials of I.C. Engines

4.1 Safety precautions

4.1.1 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.

4.1.2 This applies especially to crankcase explosive conditions protection, but also to over-speed protection and any other shut down function.

4.1.3 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.

4.2 General

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

4.2.2 Adequate test bed facilities for loads as required in Item 4.3.3 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.

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

4.2.4 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, TL may request temperature measurements as required by 3.8.9.

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

4.3 Works trials (Factory Acceptance Test)

4.3.1 Objectives

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.

4.3.2 Records

4.3.2.1 The following environmental test conditions are to be recorded:

- Ambient air temperature

- Ambient air pressure

- Atmospheric humidity

4.3.2.2 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 Chapter 4, Section 2 - Table 2.7, Section 4 - Item B.7, Section 8 - Table 8.1 and Table 8.2)

- Charge air temperature

- Charge air pressure

- Turbocharger speed (to the extent that monitoring is required in Chapter 4, Section 4, Item B.7)

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

4.3.2.4 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.

4.3.2.5 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.

4.3.3 Test loads

4.3.3.1 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 **TL** when the overall scope of tests is found to be equivalent.*

4.3.3.2 Propulsion engines driving propeller or impeller only

4.3.3.2.1 100% power (MCR) at corresponding speed n_0:

at least 60 min.

4.3.3.2.2 110% 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.

4.3.3.2.3 Approved intermittent overload (if applicable):

testing for duration as agreed with the manufacturer.

4.3.3.2.4 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.

4.3.3.2.5 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 the **TL**. In that case, the fuel delivery system is to be blocked to that power.*

4.3.3.3 Engines driving generators for electric propulsion

4.3.3.3.1 100% power (MCR) at corresponding speed n_0:

at least 60 min.

4.3.3.3.2 110% power at engine speed n_0:

15 min. - after having reached steady conditions.

4.3.3.3.3 Governor tests for compliance with Item F are to be carried out.

4.3.3.3.4 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.

4.3.3.4 Engines driving generators for auxiliary purposes

Tests to be performed as in Item 4.3.3.3.

4.3.3.5 Propulsion engines also driving power take off (PTO) generator

4.3.3.5.1 100% power (MCR) at corresponding speed n_0:

at least 60 min.

4.3.3.5.2 110% power at engine speed n_0:

15 min. - after having reached steady conditions.

4.3.3.5.3 Approved intermittent overload (if applicable):

testing for duration as agreed with the manufacturer.

4.3.3.5.4 90% (or normal continuous cruise power), 75%, 50% and 25% power in accordance with the nominal propeller curve or at constant speed n0, 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.

4.3.3.6 Engines driving auxiliaries

4.3.3.6.1 100% power (MCR) at corresponding speed n_0:

at least 30 min.

4.3.3.6.2 110% power at engine speed n_0:

15 min. - after having reached steady conditions.

4.3.3.6.3 Approved intermittent overload (if applicable):

testing for duration as agreed with the manufacturer.

4.3.3.6.4 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.

4.3.4 Turbocharger matching with engine

4.3.4.1 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.

4.3.4.2 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).

- 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.

4.3.5 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 **TL** for selected cases based on the FMEA required in Table 2.1, Table 2.2 and Table 2.3.

4.3.6 Component inspections

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

4.4 Shipboard trials(*)

4.4.1 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).

4.4.2 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.

*(*) The requirements of Item 4.4 Shipboard trials are to be uniformly implemented by **TL** to engines:*

i) with an application for certification dated on or after 1 July 2016; or

ii) installed on ships contracted for construction on or after 1 July 2016.

4.4.3 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.

4.4.4 Test loads

4.4.4.1 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.

4.4.4.2 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).

4.4.4.3 Propulsion engines driving fixed pitch propeller or impeller

4.4.4.3.1 At rated engine speed n_0:

at least 4 hours.

4.4.4.3.2 At engine speed $1.032n_0$ (if engine adjustment permits, see 4.3.3.1):

30 min.

4.4.4.3.3 At approved intermittent overload (if applicable):

testing for duration as agreed with the manufacturer.

4.4.4.3.4 Minimum engine speed to be determined.

4.4.4.3.5 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 4.4.5.1 for additional requirements in the case of a barred speed range.

4.4.4.4 Propulsion engines driving controllable pitch propellers.

4.4.4.4.1 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.

4.4.4.4.2 At approved intermittent overload (if applicable):

testing for duration as agreed with the manufacturer.

4.4.4.4.3 With reverse pitch suitable for manoeuvring, see 4.4.5.1 for additional requirements in the case of a barred speed range.

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

4.4.4.5.1 At 100% power (rated electrical power of generator):

at least 60 min.

4.4.4.5.2 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.

4.4.4.5.3 Demonstration of the generator prime movers' and governors' ability to handle load steps as described in Item F.

4.4.4.6 Propulsion engines also driving power take off (PTO) generator

4.4.4.6.1 100% engine power (MCR) at corresponding speed n_0:

at least 4 hours.

4.4.4.6.2 100% propeller branch power at engine speed n_0 (unless already covered in A):

2 hours.

4.4.4.6.3 100% PTO branch power at engine speed n_0:

at least 1 hour.

4.4.4.7 Engines driving auxiliaries

4.4.4.7.1 100% power (MCR) at corresponding speed n_0:

at least 30 min.

4.4.4.7.2 Approved intermittent overload:

testing for duration as approved.

4.4.5 Torsional vibrations

4.4.5.1 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).

4.4.6 Earthing

It is necessary to ensure that the limits specified for main engines by the engine manufacturers for the difference in electrical potential (Voltage) between the crankshaft/shafting and the hull are not exceeded in service. Appropriate earthing devices including limit value monitoring of the permitted voltage potential are to be provided.

5. Certification of AC Generating Sets

5.1 General

5.1.1 This item provides requirements for AC generating sets (i.e. Reciprocating Internal Combustion engines a), b), alternators c) and couplings) in addition to those stated in TL-R E13, TL-R M3, TL-R M51, and TL-R M53.

a) Reciprocating Internal Combustion engines are to comply with the requirements in TL-R M51 and M53.

b) The reciprocating internal combustion engine speed governor and overspeed protective device are to comply with the requirements of TL-R M3.

c) Alternators are to comply with the requirements in TL-R E13

5.1.2 The requirements are applicable to AC generating sets driven by reciprocating internal combustion engines irrespective of their types (i.e. diesel engine, dual fuel engine, gasfuel engine), except for those sets consisting of a propulsion engine which also drives power take off (PTO) generator(s).

5.2 Generating sets - requirements

5.2.1 For diesel generator sets with a mechanical output of more than 110 kW torsional vibration calculations must be submitted to TL for approval. (See, Chapter 4, Section 6, F.2).

5.2.2 The rated power shall be appropriate for the actual use of the generator set.

5.3 Marking

The entity responsible of assembling the generating set shall install a rating plate marked with at least the following information:

(i) the generating set manufacturer’s name or mark;

(ii) the set serial number;

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

(iv) the rated power (both in kW and KVA) with one of the prefixes COP, PRP (or, only for emergency Generating sets, LTP) as defined in ISO 8528-1:2018;

(v) the rated power factor;

(vi) the set rated frequency (Hz);

(vii) the set rated voltage (V);

(viii) the set rated current (A);

(ix) the mass (kg).

F. Safety Devices

1. Speed Control and Engine Protection Against Over speed

1.1 Main and auxiliary engines

1.1.1 Each diesel engine not used to drive an electric generator must be equipped with a speed governor or regulator so adjusted that the engine speed cannot exceed the rated speed by more than 15%.

1.1.2 In addition to the normal governor, each main engine with a rated power of 220 kW or over which can be declutched in service or which drives a variable-pitch propeller must be fitted with an independent over speed protection device so adjusted that the engine speed cannot exceed the rated speed by more than 20%. Equivalent equipment may be approved by **TL**. The overspeed protective device, including its driving mechanism, has to be independent from the required governor.

1.1.3 When electronic speed governors of main internal combustion engines form part of a remote control system, they are to comply 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, 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 subparagraph 1.1.1, 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 **TL** Add. Rules Regulations for the Performance of Type Tests, Part 1

Note:

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

1.2 Engines driving electric generators

1.2.1 Each diesel engine used to drive an electric main or emergency generator must be fitted with a 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 the 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 1.2.2.

1.2.2 In addition to the normal governor, each diesel engine with a rated power of 220 kW or over must be equipped with an over speed protection device independent of the normal governor which prevents the engine speed from exceeding the rated speed by more than 15%.

1.2.3 The diesel engine must be suitable and designed for the special requirements of the ship's electrical system.

Where connection of loads is envisaged in two stages, the following procedure is to be applied: Sudden loading from no-load to 50%, followed by the remaining 50% of the rated generator power, duly observing the requirements of 1.2.1 and 1.2.4.

Application of the load in more than two steps (see Fig. 3.9 for guidance on 4-stroke diesel engines expected maximum possible sudden power increase) is acceptable on condition that:

- The design of the ship's electrical system enables the use of such generator sets;

- Load application in more than two steps is considered in the design of the ship's electrical system and is approved when the drawings are reviewed;

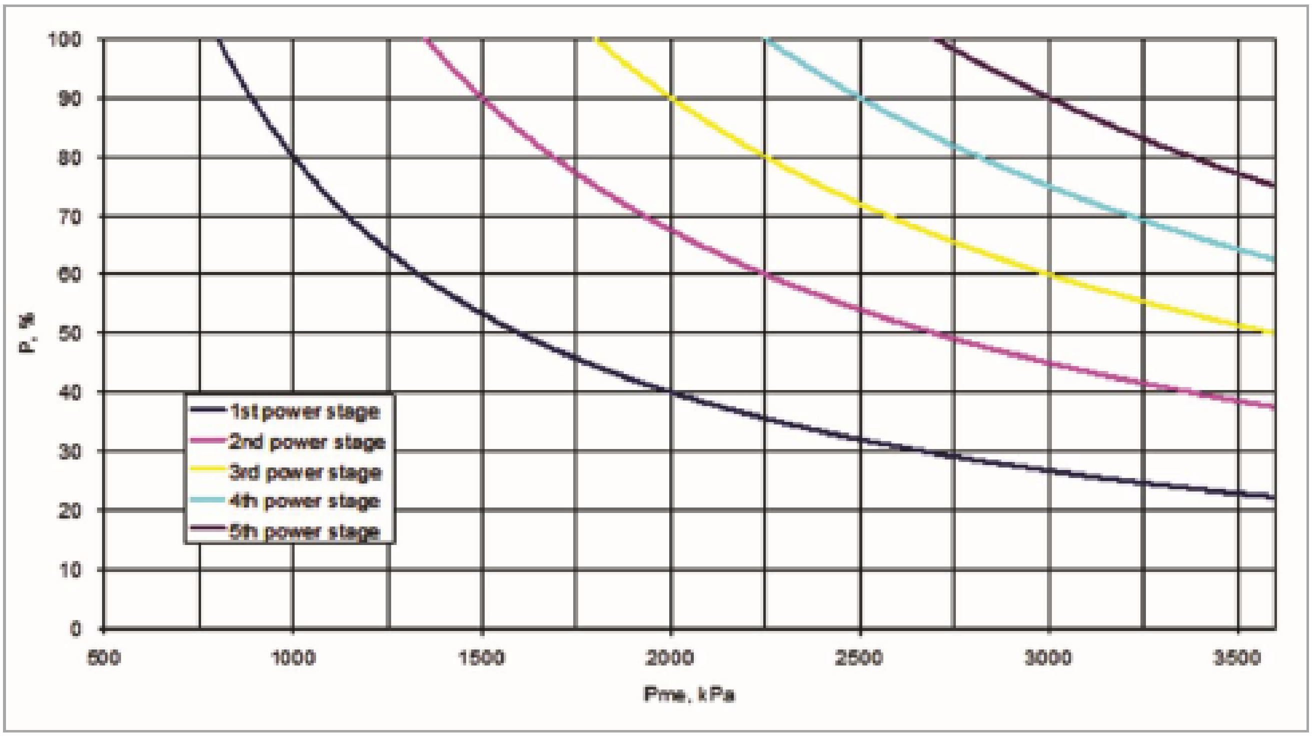
- During shipboard trials the functional tests are carried out without objections. Here the loading of the ship’s electrical net while sequentially connecting essential equipment after breakdown and during recovery of the net is to be taken into account.

- The safety of the ship's electrical system in the event of parallel generator operation and failure of a generator is to be demonstrated.

1.2.4 Speed must be stabilized and in steady-state condition within 5 seconds, inside the permissible range for the permanent speed variation δ_r. The steady-state condition n is considered to have been reached when the residual speed variation does not exceed $\mp$ 1 % of the speed associated with the set power.

1.2.5 The characteristic curves of the governors of diesel engines of generator sets operating in parallel must not exhibit deviations larger than those specified in the Rules for Electrical Installation.

1.2.6 Generator sets which are installed to serve stand-by circuits must satisfy the corresponding requirements even when the engine is cold. The start-up and loading sequence is to be concluded in about 30 seconds.



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

Fig. 3.9 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)

1.2.7 Emergency generator sets must satisfy the above governor conditions even when:

- Their total consumer load is applied suddenly, or
- 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.

1.2.8 The governors of the engines mentioned in 1.2 must enable the rated speed to be adjusted over the entire power range with a maximum deviation of 5%.

1.2.9 The rate of speed variation of the adjusting mechanisms must permit satisfactory synchronization in a sufficiently short time.

The speed characteristic should be as linear as possible over the whole power range. The permanent deviation

from the theoretical linearity of the speed characteristic may, in the case of generating sets intended for parallel operation, in no range exceed 1% of the rated speed.

1.2.10 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.

Notes relating to 1.1 and 1.2:

- *The rated power and the corresponding rated speed relate to the conditions under which the engines are operated in the system concerned.*
- *An independent over speed protection device means a system all of whose component parts, including the drive, function independently of the normal governor.*

1.3 Use of electrical / electronic governors

1.3.1 The governor and the associated actuator must, for controlling the respective engine, be suitable for the operating conditions laid down in the Construction Rules and for the requirements specified by the engine manufacturer. For single propulsion drives it has to be ensured that in case of a failure of the governor or actuator the control of the engine can be taken over by another control device.

The regulating conditions required for each individual application as described in 1.1 and 1.2 are to be satisfied by the governor system.

Electronic governors and the associated actuators are subject to type testing.

For the power supply, see the Rules for Electrical Installation, Chapter 5, Section 9, B.8.

1.3.2 Requirements applying to main engines

For single propulsion installations, to ensure continuous speed control or immediate resumption of control after a fault at least one of the following requirements is to be satisfied:

- The governor system has an independent back-up system, or,

- There is a redundant governor assembly for manual change-over with a separately protected power supply, or,

- The engine has a manually operated fuel admission control system suitable for manoeuvring.

In the event of a fault in the governor system the operating condition of the engine must not become dangerous, that is, the engine speed and power must not increase.

Alarms to indicate faults in the governor system are to be fitted.

1.3.3 Requirements applying to auxiliary engines for driving generators

Each auxiliary engine must be equipped with its own governor system.

In the event of a fault in the governor system, the fuel admission in the injection pumps must be set to "0". Alarms to indicate faults in the governor system are to be fitted.

1.3.4 The special conditions necessary to start operation from the dead ship condition are to be observed (see **TL** Rules for Electrical Installation).

2. Cylinder Overpressure Warning Device

2.1 All the cylinders of engines with a cylinder bore of > 230 mm. are to be fitted with cylinder overpressure warning devices. The response threshold of these valves shall be set at not more than 40% above the combustion pressure at the rated power.

2.2 A warning device may be dispensed with if it is ensured by an appropriate engine design or by control functions that an increased cylinder pressure cannot create danger.

3. Crankcase Ventilation

3.1 The ventilation of crankcases and any arrangement which could produce air intake within the crankcase is not allowed. For gas engines, see **TL** Part C, Chapter 10 – Liquefied Gas Carriers, Section 16.

3.2 Crankcase ventilation pipes

3.2.1 Where crankcase ventilation pipes are provided, their clear opening is to be dimensioned as small as possible, to minimize the inrush of air after a crankcase explosion.

3.2.2 Where forced provision has been made for extracting the lubricating oil vapours, e.g. for monitoring the oil vapour concentration, the vacuum in the crankcase is not to exceed 2.5 mbar.

3.2.3 The crankcase ventilation pipes for each engine are to be independent of any other engine. Exemptions may be approved if an interaction of the combined systems is inhibited by suitable means and possible spread of fire is prevented.

3.2.4 In the case of two-stroke engines the lubricating oil vapours from the crankcase must not be admitted into the scavenge manifolds respectively the air intake pipes of the engine.

4. Crankcase Safety Devices

4.1 Relief valves

4.1.1 Crank safety case devices shall be type approved. See **TL** “Type Testing Procedure for Crankcase Explosion Relief Valves”.

4.1.2 Safety valves to safeguard against overpressure in the crankcase are to be fitted to all engines with a cylinder bore of > 200 mm. or a crankcase volume of ≥ 0.6 m^3.

All separated spaces within the crankcase, e.g. gear or chain casings for camshafts or similar drives, if the volume of these spaces exceeds 0.6 m^3, and scavenge spaces in open connection to the cylinders are to be equipped with additional safety devices.

4.1.3 Engines with a cylinder bore of > 200 mm. ≤250 mm. must be equipped with at least one safety valve at each end of the crankcase. If the crankshaft has more than 8 throws, an additional safety valve is to be fitted near the middle of the crankcase.

Engines with a cylinder bore of > 250 mm. ≤ 300 mm. must have at least one safety valve close to every second crank throw, subject to a minimum number of two.

Engines with a cylinder diameter of > 300 mm. must have at least one safety valve close to each crank throw.

4.1.4 Each safety valve must have a free relief area of at least 45 cm^2.

The total free cross-sectional area of the safety valves fitted to an engine to safeguard against overpressure in the crankcase may not be less than 115 cm^2/m^3 of crankcase volume.

Notes relating to 4.1.2 and 4.1.4:

- *In estimating the gross volume of the crankcase, the volume of the fixed parts which it contains may be deducted.*

- *A space communicating with the crankcase via a total free cross-sectional area of > 115 cm^2/m^3 of volume need not be considered as a separate space. In calculating the total free cross-sectional area, individual sections of < 45 cm^2 are to be disregarded.*

- *Each safety valve required may be replaced by not more than two safety valves of smaller cross-sectional area provided that the free cross-sectional area of each safety valve is not less than 45 cm^2.*

4.1.5 The safety devices are to be quick acting and self closing devices to relief a crankcase of pressure at a crankcase explosion. In service they shall be oil tight when closed and have to prevent air inrush into the crankcase. The gas flow caused by the response of the safety device must be deflected, e. g. by means of a baffle plate, in such a way as not to endanger persons standing nearby. Is has to be demonstrated that the baffle plate does not adversely affect the operational effectiveness of the device. For relief valves the discs are to be made of ductile material capable of withstanding the shock load at the full open position of the valve.

Relief valves shall be fully opened at a differential pressure in the crankcase not greater than 0.2 bar.

4.1.6 The relief valves are to be provided with a flame arrester that permits crankcase pressure relief and prevents passage of flame following a crankcase explosion.

4.1.7 Safety devices are to be provided with suitable markings that include the following information:

- Name and address of manufacturer,

- Designation and size,

- Relief area,

- Month/year of manufacture,

- Approved installation orientation.

4.2 Crankcase doors and sight holes

4.2.1 Crankcase doors and their fittings shall be so dimensioned as not to suffer permanent deformation due to the overpressure occurring during the response of the safety equipment. Crankcase doors are to be fastened sufficiently securely for them not be readily displaced by a crankcase explosion.

4.2.2 Crankcase doors and hinged inspection ports are to be equipped with appropriate latches to effectively prevent unintended closing.

4.2.3 A warning notice is to be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. The 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.

4.3 Oil mist detection/monitoring and alarm system (Oil mist detector)

4.3.1 Engines with a cylinder diameter > 300 mm or a rated power of 2250 kW and above are to be fitted with crankcase oil mist detectors or engine bearing temperature monitors (all bearings i.e. journal and connecting rod bearings) or equivalent devices (See also TL- I SC228). The oil mist detectors are to be type tested in accordance with Type Testing Procedure for Crankcase Oil Mist Detection and Alarm Equipment.

Engine bearing temperature monitors or equivalent devices used as safety devices have to be of a type approved by **TL** for such purposes.

Measures applied to high speed engines where specific design features to preclude the risk of crankcase explosions are incorporated, can be accepted by **TL** as equivalent device.

4.3.2 For multiple engine installations each engine is to be provided with a separate oil mist detector and a dedicated alarm.

4.3.3 Oil mist detectors are to be type approved. Oil mist detection arrangements are to be tested in accordance with “Type Testing Procedure for Crankcase Oil Mist Detection and Alarm Equipment” and comply with 4.3.2 to 4.3.13.

Alarms and shutdowns for the oil mist detection system are to be in accordance with Chapter 4-1 - Automation, Section 4 and Section 8, also Table 3.7 and the system arrangements are to comply with Chapter 4-1 - Automation.

4.3.4 The oil mist detector is to be installed in accordance with the engine designer’s and the system manufacturer’s instructions and recommendations.

4.3.5 Function tests and equipment together with detectors to demonstrate that the detection and alarm system functionally operates are to be performed on the engine test bed at manufacturer’s workshop and on board under the conditions of "engine at standstill" and "engine running at normal operating conditions" in accordance with test procedures to be agreed with **TL**.

4.3.6 Alarms and shutdowns for the detector are to be in accordance with Table 3.7.

4.3.7 Functional failures at the devices and equipment are to be alarmed.

4.3.8 The oil mist detector has to indicate that the installed lens, which is used in determination of the oil mist concentration has been partly obscured to a degree that will affect the reliability of the information and alarm indication.

4.3.9 Where the detector includes the use of programmable electronic systems, the arrangements are in accordance with the requirements of **TL** Rules for Electrical Installations, Section 10.

4.3.10 Where sequential oil mist detection / monitoring arrangements are provided, the sampling frequency and time are to be as short as reasonably practicable.

4.3.11 Plans of showing details and arrangements of the oil mist detector are to be submitted for approval.

The following particulars are to be included in the documentation:

- Schematic layout of engine oil mist detector showing location of engine crankcase sample points and piping arrangement together with pipe dimensions to detector/monitor.

- 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.

- Maintenance and test manuals

- Information about type approval of the detection/monitoring system or functional tests at the particular engine

4.3.12 A copy of the documentation supplied with the system such as maintenance and test manuals are to be provided on board ship.

4.3.13 The readings and the alarm information from the oil mist detector are to be capable of being read from a safe location away from the engine.

4.3.14 Where alternative methods are provided for the prevention of build-up a potentially explosive condition within the crankcase (independent of the reason, e.g. oil mist, gas, hot spots, etc.), details are to be submitted for consideration of **TL**. The following information is to be included in the details to be submitted for approval:

- Engine particulars - type, power, speed, stroke, bore and crankcase volume (including volumes of all divisions integrated with crankcase, if existing),

- Details of arrangements preventing the build-up of potentially explosive conditions within the crankcase, e.g. bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring, recirculation arrangements, crankcase atmosphere monitoring,

- Evidence that the arrangements are effective in preventing the build-up of potentially explosive conditions together with details of in service experience

- Operating instructions and maintenance and test instructions

4.4 Active safety measures where it is proposed to use alternative active technologies to minimise the risk for a potential crankcase explosion, details of the arrangement and the function description are to be submitted to **TL** for approval.

4.5 Crankcase safety devices have to be type approved.

4.6 Plans showing details and arrangements of safety devices are to be submitted for approval.

4.7 Safety devices 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.

A copy of this manual is to be kept on board of the ship.

4.8 Where it is proposed to use the introduction of inert gas into the crankcase to minimise a potential crankcase explosion, details of the arrangements are to be submitted to **TL** for consideration.

5. Safety Devices in the Starting Air System

The following equipment is to be fitted to safeguard starting air system against explosions due to failure of starting valves:

5.1 An isolation non-return valve is to be fitted to the starting air line serving each engine.

5.2 Engines with cylinder bores of > 230 mm. are to be equipped with flame arresters as follows:

- On directly reversible engines immediately in front of the start-up valve of each cylinder;

- On non-reversible engines, immediately in front of the intake of the main starting air line to each engine.

5.3 Equivalent safety devices may be approved by **TL**.

6. Safety Devices in the Lubricating Oil System

Each engine with a rated power of 220 kW or over is to be fitted with devices which automatically shut down the engine in the event of failure of the lubricating oil supply. This is not valid for engines serving solely for the drive of emergency generator sets and emergency fire pumps. For these engines an alarm has to be provided.

7. Safety Devices in Scavenging Air Ducts

For two-stroke engines scavenging air ducts are to be protected against overpressure by safety valves.

G. Auxiliary Systems

1. General

For piping systems and accessory filter arrangements Chapter 4, Section 16 is to be applied, additionally.

2. Fuel Oil System

2.1 General

2.1.1 Only pipe connections with metal sealing surfaces or equivalent pipe connections of approved design may be used for fuel injection lines.

2.1.2 Feed and return lines are to be designed in such a way that no unacceptable pressure surges occur in the fuel supply system. Where necessary, the engines are to be fitted with surge dampers approved by **TL**.

2.1.3 All components of the fuel system are to be designed to withstand the maximum peak pressures which will be expected in the system.

2.1.4 If fuel oil reservoirs or dampers with a limited life cycle are fitted in the fuel oil system the life cycle together with overhaul instructions is to be specified by the engine manufacturer in the corresponding manuals.

2.1.5 Oil fuel lines are not to be located immediately above or near units of high temperature, steam pipelines, exhaust manifolds, silencers or other equipment required to be insulated by 7.1. as for as practicable, oil fuel lines are to be arranged far apart from hot surfaces, electrical installations or other potential sources of ignition and are to be screened or otherwise suitably protected to avoid oil spray or oil leakage onto the sources of ignition. The number of joints in such piping systems are to be kept to a minimum.

2.2 Shielding

2.2.1 Regardless of the intended use and location of internal combustion engines, all external fuel injection lines (high pressure lines between injection pumps and injection valves) are to be shielded by jacket pipes in such a way that any leaking fuel is:

- Safely collected,
- Drained away unpressurized, and
- Efficiently monitored and alarmed.

2.2.2 If pressure variations of > 20 bar occur in fuel feed and return lines, these lines are also to be shielded.

2.2.3 The high pressure fuel pipe and the outer jacket pipe have to be permanent assembly.

2.2.4 Where, pipe sheaths in the form of hoses are provided as shielding, the hoses must be suitable for this purpose and approved by **TL**.

2.3 Fuel leak drainage

Appropriate design measures are to be introduced to ensure generally that leaking fuel is drained efficiently and cannot enter into the engine lube oil system.

2.4 Heating, thermal insulation, re-circulation

Fuel lines, including fuel injection lines, to engines which are operated with preheated fuel are to be insulated against heat losses and, as far as necessary, provided with heating.

Means of fuel circulation are also to be provided.

2.5 Fuel oil emulsions

For engines operated on emulsion of fuel oil and other liquids, it has to be ensured that engine operation can be resumed after failures to the fuel oil treatment system.

3. Filter Arrangements for Fuel Oil and Lubricating Oil Systems

3.1 Fuel and lubricating oil filters which are to be mounted directly on the engine are not to be located above rotating parts or in the immediate proximity of hot components.

3.2 Where the arrangement stated in 3.1 is unfeasible, the rotating parts and the hot components are to be sufficiently shielded.

3.3 Filters have to be so arranged that fluid residues can be collected by adequate means. The same applies to lubricating oil filters if oil can escape when the filter is opened.

3.4 Switch-over filters with two or more filter chambers are to be fitted with devices which safely ensure a relief of pressure before opening and venting when a chamber is placed in service. Shutoff valves shall normally be used for this purpose. It must be clearly discernible which filter chambers are in service and which are out of operation at any time.

3.5 Oil filters fitted parallel for the purpose of enabling cleaning without disturbing oil supply to engines (e.g. duplex filters) are to be provided with arrangements that will minimize the possibility of a filter under pressure being opened by mistake. Filters/ filter chambers shall be provided with suitable means for:

- Venting when put into operation.
- Depressurizing before being opened.

Valves or cocks with drain pipes led to a safe location shall be used for this purpose.

For oil filters of generator diesel engines for ships with more than a single generator set, requirements in Chapter 4, Section 16, H-3.4.1 may be applied.

3.6 For filters, requirements in Chapter 4, Section 14 shall be taken into consideration.

4. Lubricating Oil System

4.1 General requirements relating to lubricating oil systems and to the cleaning, cooling etc. of the lubricating oil are contained in Chapter 4, Section 16, H. For piping arrangement 2.1.3 is to be applied.

4.1.1 Engines whose sumps serve as oil reservoirs must be so equipped that the oil level can be established and, if necessary, topped up during operation. Means must be provided for completely draining the oil sump.

4.1.2 The oil drain pipes for each engine are to be independent of any other engine.

4.1.3 Drain lines from the engine sump to the drain tank are to be submerged at their outlet ends.

4.2 The equipment of engines fitted with lubricating oil pumps is subject to Chapter 4, Section 16, H.3.

4.2.1 Main lubricating oil pumps driven by the engine are to be designed to maintain the supply of lubricating oil over the entire operating range

4.2.2 Main engines which drive main lubricating oil pumps are to be equipped with independently driven stand-by pumps.

4.2.3 In multi-engine installations having separate lubricating oil systems approval may be given for the carriage on board of reserve pumps ready for mounting provided that the arrangement of the main lubricating oil pumps enables the change to be made with the means available on board.

4.2.4 Lubricating oil systems for cylinder lubrication which are necessary for the operation of the engine and which are equipped with electronic dosing units have to be approved by **TL**.

5. Cooling System

5.1 For the equipment of engines with cooling water pumps and for the design of cooling water systems, see Chapter 4, Section 16, I and K.

5.1.1 Main cooling water pumps driven by the engine are to be designed to maintain the supply of cooling water over the entire operating range.

5.1.2 Main engines which drive main cooling water pumps are to be equipped with independently driven stand-by pumps or with means for connecting the cooling water system to independently driven stand-by pumps.

5.1.3 In installations comprising more than one main engine and with separate fresh cooling water systems approval may be given for the carriage on board of reserve pumps ready for mounting provided that the arrangement of the main fresh cooling water pumps enables the change to be made with the means available on board. Shutoff valves must be provided enabling the main pumps to be isolated from the fresh cooling water system.

5.2 If cooling air is drawn from the engine room, the design of the cooling system is to be based on a room temperature of at least 45°C.

The exhaust air of air-cooled engines may not cause any unacceptable heating of the spaces in which the plant is installed. The exhaust air is normally to be led to the open air through special ducts.

5.3 Where engines are installed in spaces which oil-firing equipment is operated, Chapter 4, Section 15, A.5. is also to be complied with.

6. Charge Air System

6.1 Exhaust gas turbochargers

6.1.1 The construction and testing of exhaust gas turbochargers are subject to Chapter 4, Section 4.

6.1.2 Exhaust gas turbochargers may exhibit no critical speed ranges over the entire operating range of the engine.

6.1.3 The lubricating oil supply must also be ensured during start-up and run-down of the exhaust gas turbochargers.

6.1.4 Even at low engine speeds, main engines must be supplied with charge air in a manner to ensure reliable operation.

Where necessary, two-stroke engines are to be equipped with directly or independently driven scavenging air blowers.

6.1.5 If, in the lower speed range or when used for manoeuvring, an engine can be operated only with a charge air blower driven independently of the engine, as stand-by charge air blower is to be installed or an equivalent device of approved design.

6.1.6 With main engines emergency operation must be possible in the event of a turbocharger failure.

6.2 Charge air cooling

6.2.1 The construction and testing of charge air coolers are subject to Chapter 4, Section 14.

6.2.2 Means are to be provided for regulating the temperature of the charge air within the temperature range specified by the engine manufacturer.

6.2.3 The charge air lines of engines with charge air coolers are to be provided with sufficient means of drainage.

6.3 Fire extinguishing equipment

The charge air receivers of crosshead engines which have open connection to the cylinders are to be connected to an approved fire extinguishing system which is independent of the engine room fire extinguishing system. (See Chapter 4, Section 18, Table 18.2)

7. Exhaust Gas Lines

7.1 Exhaust gas lines are to be insulated and/or cooled in such a way that the surface temperature cannot exceed 220°C at any point.
Insulating materials must be non-combustible.

7.2 General rules relating to exhaust gas lines are contained in Chapter 4, Section 16, M.

H. Control Equipment

1. General

For unmanned machinery installations, Chapter 4-1 - Automation is to be observed in addition to the following requirements.

2. Main Engines

2.1 Local control station

To provide emergency operation of the propulsion plant a local control station is to be installed from which the plant can be operated and monitored.

2.1.1 Indicators according to Table 3.7 are to be clearly sited on the local main engine control station.

2.1.2 Temperature indicators are to be provided on the local control station or directly on the engine.

2.1.3 In the case of gear and controllable pitch propeller systems, the local control indicators and control equipment required for emergency operation are to be installed at the main engines local control station.

2.1.4 Critical speed ranges are to be marked in red on the tachometers.

2.2 Machinery control room / control centre

For remotely operated or controlled machinery installations the indicators listed in Table 3.7 are to be installed, see Chapter 4-1 - Automation

2.3 Bridge / navigation center

2.3.1 The essential operating parameters for the propulsion system are to be provided in the control station area.

2.3.2 The following stand-alone control equipment is to be installed:

- Speed/direction of rotation of main engine,
- Speed/direction of rotation of shafting,
- Propeller pitch (controllable pitch propeller),
- Starting air pressure,
- Control air pressure.

2.3.3 In the case of engine installations up to a total output of 600 kW, simplifications can be agreed with **TL**.

Table 3.7 Alarm and indicators

Description	Propulsion engines	Auxiliary engines	Emergency engines
Speed / direction of rotation	I		
Engine overspeed **(5)**	A, S	A, S	A, S
Lubricating oil pressure at engine inlet	I, L**(9)**, S	I, L**(9)**, S	I, L**(9)**
Lubricating oil temperature at engine inlet	I, H	I **(5)**, H **(5)**	I **(5)**, H **(5)**
Fuel oil pressure at engine inlet	I	I	
Fuel oil temperature at engine inlet **(1)**	I	I	
Fuel oil leakage from high pressure pipes	A	A	A
Cylinder cooling water pressure at engine inlet	I, L	I**(4)**, L **(4)**	I**(4)**, L **(4)**
Cylinder cooling water /air temperature at engine outlet	I, H	I, H	I, H
Cylinder pressure **(10)**	H	H	H
Piston coolant pressure at engine inlet	I, L		
Piston coolant temperature at engine outlet	I, H		
Charge air pressure at cylinder inlet	I		
Charge air temperature at charge air cooler inlet	I		
Charge air temperature at charge air cooler outlet	I, H		
Starting air pressure	I, L		
Control air pressure	I, L		
Exhaust gas temperature **(2)**	I, H **(3)**		
Oil mist concentration in crankcase or alternative monitoring system **(6) (7) (8)**	I, H	I, H	I, H

(1) *For engines running on heavy fuel oil only.*
(2) *Wherever the dimensions permit, at each cylinder outlet and at the turbo charger inlet and outlet.*
(3) *At turbo charger outlet only.*
(4) *Cooling water pressure or flow.*
(5) *Only for an engine output ≥220 kW.*
(6) *For engines having an output >2250 kW or a cylinder bore >300 mm.*
(7) *Alternative methods of monitoring may be approved by **TL**. See F-4.3.1.*
(8) *Engine slowdown function for low speed engines and shutdown function for medium and high speed engines to be provided.*
(9) *Only for an engine output > 37 kW*
(10) *Only for engines having cylinder bore > 230 mm.*

I : Indicator
A : Alarm
H : Alarm for upper limit
L : Alarm for lower limit
S : Shutdown

3. Auxiliary Engines

For auxiliary engines and emergency application engines the controls according to Table 3.7 are to be provided as a minimum.

I. Alarms

1. General

1.1 The following requirements apply to machinery installations which have been designed for conventional operation without any degree of automation.

1.2 Within the context of these Rules, the word alarm is understood to mean the visual and audible warning of abnormal operating parameters.

2. Scope of Alarms

Alarms have to be provided for main, auxiliary and emergency engines according to Table 3.7.

J. Engine Alignment / Seating

For engine alignment / seating see **TL** Additional Rules, Seating of Propulsion Plant.

1. Crankshaft Alignment

The crankshaft alignment is to be checked every time an engine has been aligned on its foundation by measurement of the crank web deflection and/or other suitable means.

For the purpose of subsequent alignments, note is to be taken of:

- The draught / load condition of the vessel
- The condition of the engine-cold / preheated / hot

2. Permissible Crank Web Deflection

Where the engine manufacturer has not specified values for the permissible crank web deflection, assessment is to be based on **TL**'s reference values.

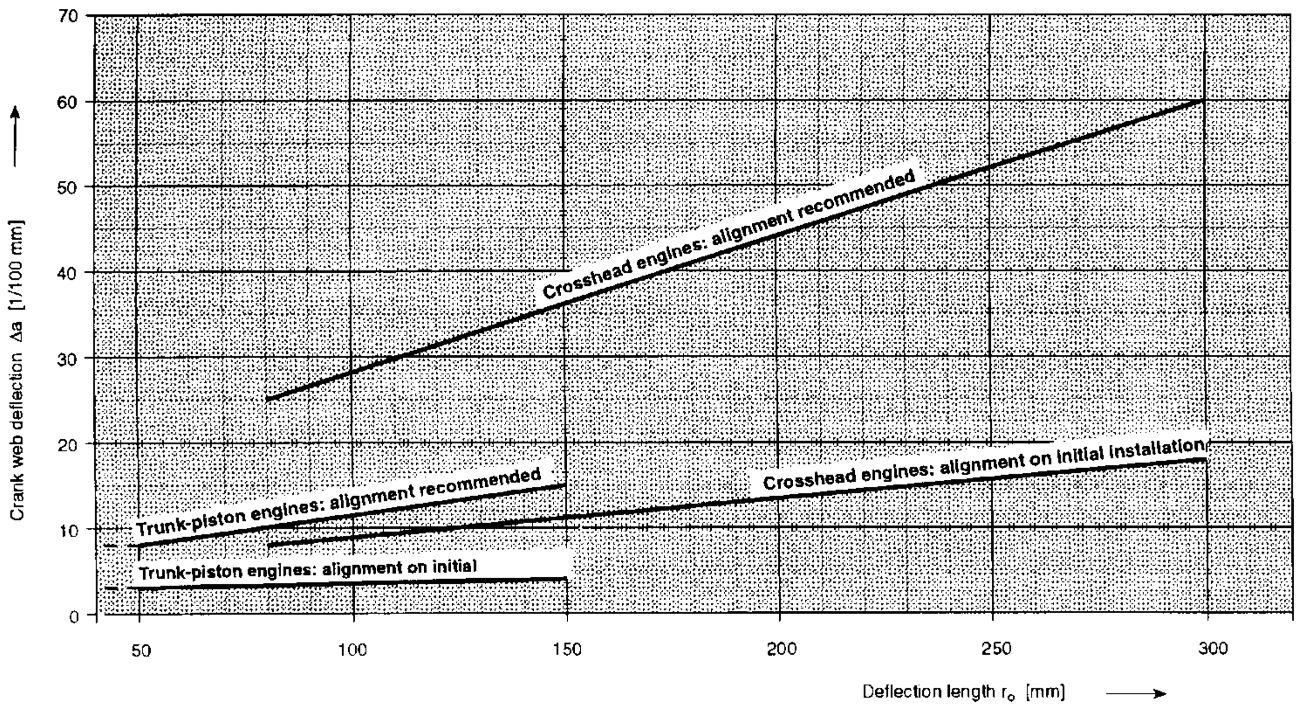


Fig. 3.10 Reference values for crank web deflection

3. Reference Values for Crank Web Deflection

3.1 Irrespective of the crank web deflection figures quoted by the manufacturers of the various engine types, reference values for assessing the crank web deflection in relation to the deflection length r_o can be taken from Fig. 3.11.

Provided that these values are not exceeded, it may be assumed that neither the crankshaft nor the crankshaft bearings are subjected to any unacceptable additional stresses.

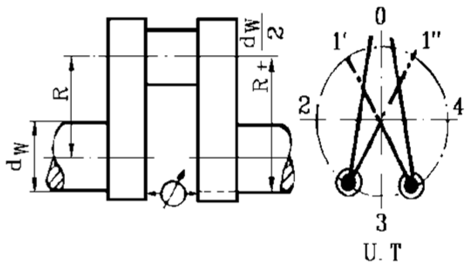


Fig. 3.11 Measurements of crank web deflections

3.2 Notes on the measurement of crank web deflections

Crank web deflections are to be measured at distance R + $d_w/2$ from the crankpin centre line (see Fig. 3.11).

Crank web deflection Δa is only meaningful as measured between opposite crank positions (see Fig. 3.11), i.e. between 0-3 for evaluating vertical alignment and bearing location, and between 2-4 for evaluating lateral bearing displacement when aligning the crankshaft and assessing the bearing wear.

For measuring point 0, which is obstructed by the connecting rod, the mean value of the measurements made at 1' and 1'' is to be applied.

3.3 Determining the crank web deflection length r_o

- Solid-forged and drop-forged crankshafts in Fig. 3.12, parts A, B and C;
- Semi-built crankshafts, Fig 3.12, D.

Symbols:

R = Crank radius, [mm]

H = Stroke (2R), [mm]

d_k = Crank pin diameter, [mm]

d_w = Journal diameter, [mm]

d_N = Shrink annulus diameter, [mm]

W = Axial web thickness, [mm]

B = Web width at distance R/2, [mm]

T_i = Depth of web undercut (on crank pin side), mm]

T_a = Depth of web undercut (on journal side), [mm]

s = Pin/journal overlap [mm].

$$= \frac{(d_k + d_w)}{2} - R$$

Where there is a negative pin/journal overlap (s<0), the deflection length r_o in accordance with Fig. 3.12, A is determined by applying the following crank web deflection length formula:

$$r_0 = 0{,}5 \cdot (H + d_k + d_w) - W\left[\left(\sqrt{\frac{2 \cdot d_k}{W}} - 1 + \sqrt{\frac{2 \cdot d_w}{W}} - 1\right)\right]$$

In case of web undercut, W in crank web deflection length formula is to be replaced by the following value:

$$W^* = W - \frac{(T_i + T_a)}{2}$$

In the case of semi-built crankshafts in accordance with Fig. 3.12, D, the value d_w under the root sign only in crank web deflection length formula is to be replaced by the following value.

$$d_W^* = 1/3(d_N - d_W) + d_W$$

Where there is a positive pin/journal overlap (s≥0) according to Fig 3.12, C, the value in crank web deflection length formula is to be replaced by:

$$W^* = \sqrt{(W - T_i - T_a)^2 + [0,5\,(d_k + d_W - H)\,]^2}$$

For the conventional design, where

B/d_w = 1.37 to 1.51 in case of solid-forged crankshafts, and

B/d_w = 1.51 to 1.63 in case of semi-built crankshafts,

the influence of B in the normal calculations of r_o is already taken into account in the values of Δ_a in Fig. 3.10. Where the values of B/d_w depart from the above (e.g. in the case of discs, oval webs etc.), the altered stiffening effect of B is to be allowed for by a fictitious web thickness W^{**}, which is to be calculated by applying the following equations and is to be substituted for W in crank web deflection length formula:

$$W^{**} = W^* \cdot \sqrt[3]{B/\,d_W - 0,44}$$ for solid - forged crankshafts

$$W^{**} = W^* \cdot \sqrt[3]{B/\,d_W - 0,57}$$ for semi - built crankshafts

K. Exhaust Gas Cleaning Systems

1. General

Exhaust gas cleaning systems shall comply with the applicable statutory requirements. In case of sea going ships requirements stipulated in the MARPOL Convention are to be observed. In case of wet exhaust gas cleaning systems (scrubber systems) IMO Resolution MEPC 259(68) applies.

In case of Exhaust Gas Recirculation (EGR) method is used Resolution MEPC 307(73)* should be considered. In case of engines fitted with Selective Catalytic Reduction system, Resolution MEPC.291(71) as amended by MEPC.313(74) should be taken into account in addition to NOx Technical Code 2008.

1.1 Application

The following requirements apply to exhaust gas cleaning systems which reduce the amount of nitrogen oxides (NOx), sulphur oxides (SOx) or particulate matter from the exhaust gases of internal combustion engines, incinerators or steam boilers.

2. Approval

Where an exhaust gas cleaning system is installed details of the arrangement and a description of the function are to be submitted to **TL** for approval.

2.1 Documents for approval

For approval, drawings showing the main dimensions of the systems shall be submitted including documentation concerning installation requirements and operational features. An operation manual shall include instructions for emergency operation, if applicable.

2.2 Approval certificate

After successful appraisal of the required documents and successful conclusion of the shipboard test in presence of a Surveyor **TL** issues an Approval Certificate.

(*) The resolution should apply to a marine diesel engine fitted with an EGR device having a bleed-off water discharge arrangement, for which the EIAPP Certificate is first issued on or after 1 June 2019.

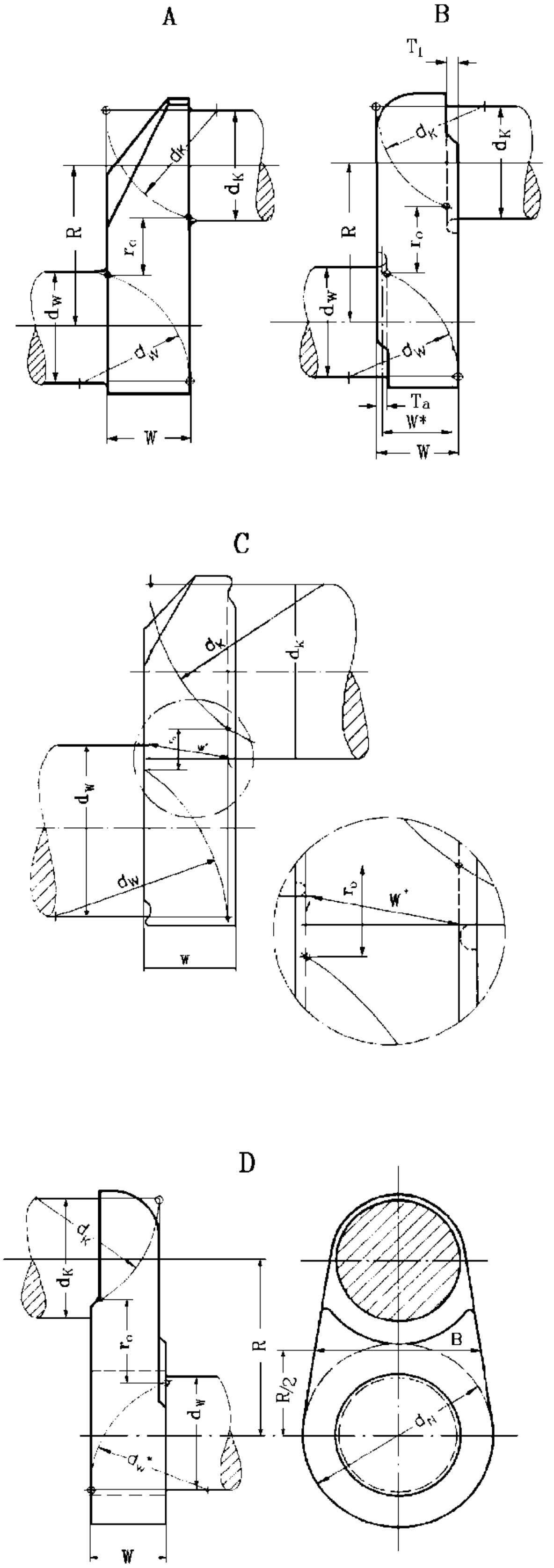


Fig. 3.12 Solid-forged (A, B ve C) and semi-built (D) crankshafts

3. Layout

3.1 System layout and installation

Exhaust gas cleaning systems shall be independent for each combustion engine or combustion plant. General requirements on the use of combustible materials and on structural fire protection are to be observed. In cases where urea or sodium hydroxide solution tanks are installed in a space separated from engine room, for fire integrity of solution tank space see Chapter 1 – Hull, Section 21. Thermal expansion of the system and its mechanical connections to both the ship's structure and the exhaust pipes has to be considered. The requirements for exhaust gas lines set out in Chapter 4, Section 16, M shall be taken into account. The aftertreatment system is to be equipped with at least one inspection port.

Exhaust gas cleaning systems are to be accessible for inspection and maintenance. A change or removal of internal components shall be possible, where applicable.

3.2 Bypass

Where an exhaust gas cleaning system is installed with a single main propulsion engine a bypass, controlled by flap valves or other suitable cut-off devices, is required in order to allow unrestricted engine operation in case of system failure. The bypass shall be designed for the maximum exhaust gas mass flow at full engine load.

In case of an exhaust gas cleaning system installed on an engine of a multi engine plant a bypass system may be dispensed with.

3.3 Additional pressure loss

The total pressure loss in the exhaust gas system, including the additional pressure loss from the exhaust gas cleaning system, shall not exceed the maximum allowable exhaust gas back pressure as specified by the engine manufacturer at any load condition.

3.4 Maximum gas pressure

The maximum pressure in the system of the exhaust pipes as specified by the manufacturer shall not be exceeded. Care is to be taken in particular where the exhaust gas cleaning system is located upstream of the turbocharger of the combustion engine (e.g. Selective Catalytic Reduction systems in conjunction with large bore 2-stroke Diesel engines).

3.5 Oscillation characteristics of the exhaust gas column

The installation and operation of the exhaust gas cleaning system shall not have an adverse effect on the oscillation characteristics of a combustion engine's exhaust gas column in order to avoid unsafe engine operation.

3.6 Deposition of soot

The deposition of soot within or in the proximity of the exhaust gas cleaning system should be avoided. Where this may lead to additional fire hazards the deposition of soot is not acceptable.

3.7 Vibrations in piping system

The design and installation of the exhaust gas cleaning system including the exhaust gas piping system shall account for vibrations induced by the ship's machinery, the pulsation of the exhaust gas or vibrations transmitted through the ship's structure in order to prevent mechanical damage to the piping system. Consideration should be given to the installation of damping systems and/or compensators.

3.8 Monitoring of the operating parameters

The main operating parameters of the exhaust gas cleaning system have to be monitored and should serve as indicators for possible abnormalities. As a minimum, the following operating parameters shall be monitored:

- Gas temperature upstream of the exhaust gas cleaning system
- Gas temperature downstream of the exhaust gas cleaning system
- Pressure drop across the exhaust gas cleaning system
- Engine exhaust gas back pressure
- Position of flap valves

4. Materials

All materials of the exhaust gas cleaning system, connecting pipes and chemically reactive agent dosing units shall be non-combustible. The requirements relating to exhaust gas lines as contained in Chapter 4, Section 16, M are to be observed, as applicable.

5. Chemically reactive agents

5.1 Reducing agent

For Selective Catalytic Reduction (SCR) type exhaust gas cleaning systems the reducing agent (Ammonia, dissolved Ammonia, Urea or the like) has to be stored and pumped in tanks and pipes made of approved materials for these types of agents, see Chapter 4, Section 16.

For more details see TL- I MPC 105.

5.2 Ammonia slip

Where Selective Catalytic Reduction (SCR) type exhaust gas cleaning systems are applied excessive slip of ammonia has to be prevented.

5.3 Washwater criteria

Where the exhaust gases are washed with water, discharged wash water has to comply with criteria as specified in IMO Resolution MEPC 259(68).

5.4 Storage and use of SCR reductants

5.4.1 General

The NOx Technical Code, in 2.2.5 and elsewhere, provides for the use of NOx Reducing Devices of which Selective Catalytic Reduction (SCR) is one option. SCR requires the use of a reductant which may be a urea/water solution or, in exceptional cases, aqueous ammonia or even anhydrous ammonia. These requirements apply to the arrangements for the storage and use of SCR reductants.

5.4.2 Reductant using urea based ammonia (e.g. 40%/60% urea/water solution)

The following requirements apply to SCR reductants with tank volume over 500 litres.

5.4.2.1 Where urea based ammonia (e.g. AUS 40 – aqueous urea solution specified in ISO 18611-1) is introduced, the storage tank is to be arranged so that any leakage will be contained and prevented from making contact with heated surfaces. All pipes or other tank penetrations are to be provided with manual closing valves attached to the tank. Tank and piping arrangements are to be approved.

5.4.2.2 The storage tank may be located within the engine room.

5.4.2.3 The storage tank is to be protected from excessively high or low temperatures applicable to the particular concentration of the solution. Depending on the operational area of the ship, this may necessitate the fitting of heating and/or cooling systems. The physical conditions recommended by applicable recognized standards (such as ISO 18611-3) are to be taken into account to ensure that the contents of the aqueous urea tank are maintained to avoid any impairment of the urea solution during storage.

5.4.2.4 If a urea storage tank is installed in a closed compartment, the area is to be served by an effective mechanical ventilation system of extraction type providing not less than 6 air changes per hour which is independent from the ventilation system of accommodation, service spaces, or control stations. The ventilation system is to be capable of being controlled from outside the compartment . A warning notice requiring the use of such ventilation before entering the compartment shall be provided outside the compartment adjacent to each point of entry.

Alternatively, where a urea storage tank is located within an engine rooma separate ventilation system is not required when the general ventilation system for the space is arranged so as to provide an effective movement of air in the vicinity of the storage tank and is to be maintained in operation continuously except when the storage tank is empty and has been thoroughly ventilated.

5.4.2.5 Each urea storage tank is to be provided with temperature and level monitoring arrangements. High and low level alarms together with high and low temperature alarms are also to be provided.

5.4.2.6 Where urea based ammonia solution is stored in integral tanks, the following are to be considered during the design and construction:

- These tanks may be designed and constructed as integral part of the hull, (e.g. double bottom, wing tanks).

- These tanks are to be coated with appropriate anti-corrosion coating and cannot be located adjacent to any fuel oil and fresh water tank.

- These tanks are to be designed and constructed as per the structural requirements applicable to hull and primary support members for a deep tank construction.

- These tanks are to be included in the ship's stability calculation.

5.4.2.7 The requirements specified in 5.4.2.4 also apply to closed compartments normally entered by persons:

- when they are adjacent to the urea integral tanks and there are possible leak points (e.g. manhole, fittings) from these tanks, or

- when the urea piping systems pass through these compartments, unless the piping system is made of steel or other equivalent material with melting point above 925 degrees C and with fully welded joints.

5.4.2.8 The reductant piping and venting systems are to be independent of other ship service piping and/or systems. Reductant piping systems are not to be located in accommodation, service spaces, or control stations. The vent pipes of the storage tank are to terminate in a safe location on the weather deck and the tank venting system is to be arranged to prevent entrance of water into the urea tank.

5.4.2.9 Reductant tanks are to be of steel or other equivalent material with a melting point above 925 degrees C.

Pipes/piping systemsare to be of steel or other equivalent material with melting point above 925 degrees C, except downstream of the tank valve, provided this valve is metal seated and arranged as fail-to-closed or with quick closing from a safe position outside the space in the event of fire; in such case, type approved plastic piping may be accepted even if it has not passed a fire endurance test. Reductant tanks and pipes/piping systems are to be made with a material compatible with reductant or coated with appropriate anti-corrosion coating.

5.4.2.10 For the protection of crew members, the ship is to have on board suitable personnel protective equipment. Eyewash are to be provided, the location and number of these eyewash stations and safety showers are to be derived from the detailed installation arrangements.

5.4.2.11 Urea storage tanks are to be arranged so that they can be emptied of urea and ventilated by means of portable or permanent systems.

5.4.3 Reductant using aqueous ammonia (28% or less concentration of ammonia)

Aqueous ammonia is not to be used as a reductant in a SCR except where it can be demonstrated that it is not practicable to use a urea based reductant. Where an application is made to use aqueous ammonia as the reductant then the arrangements for its loading, carriage and use are to be derived from a risk based analysis.

5.4.4 Reductant using anhydrous ammonia (99.5% or greater concentration of ammonia by weight)

Anhydrous ammonia is not to be used as a reductant in a SCR except where it can be demonstrated that it is not practicable to use a urea based reductant and where the Flag Administration agrees to its use. Where it is not practicable to use a urea reductant then it is also to be demonstrated that it is not practicable to use aqueous ammonia. Where an application is made to use anhydrous ammonia as the reductant then the arrangements for its loading, carriage and use are to be derived from a risk based analysis.

6. Shipboard testing

The exhaust gas cleaning and bypass system is subject to inspection and functional tests in each case in the presence of a Surveyor.

L. Gas or Other Low-Flashpoint Fuels Fuelled Engines

1. Scope and application

1.1 For internal combustion engines using gas or other low-flashpoint fuels as fuel the following requirements are to be observed. These requirements are applicable to gas or other low-flashpoint fuels fuelled engines meeting the following criteria:

- Engines using natural gas or other low-flashpoint fuels as fuel

- Engines burning fuel gas and fuel oil (dual-fuel engines), or single gas fuel engines (operating on gas-only)

- Engines with low or high pressure gas supply systems

1.2 Special design features will be considered on a case by case basis, taking into account the basic engine design and the engine safety concept.

2. Further Rules and Guidelines

2.1 The gas or other low-flashpoint fuels fuelled engine requirements defined in **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuelsare generally to be fulfilled.

2.2 Requirements for internal combustion engines as defined in Machinery rules, Chapter 4, from A to N are to be followed for gas-fuelled engines as far as applicable.

2.3 **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels apply to gas fuel supplied from gas fuel storage tanks.

2.4 **TL** Part C, Chapter 10 – Liquefied Gas Carriers apply to gas fuel supplied from liquefied gas carrier cargo boil-off.

3. Definitions

3.1 Definitions addressing gas as fuel as given in **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels apply.

3.2 Gas admission valve: Valve or injector on the engine which controls gas supply to the engine according to the engine's actual gas demand.

3.3 Safety concept: The safety concept is a document describing the safety philosophy with regard to gas as fuel.

It describes how risks associated with this type of fuel are controlled under normal operating conditions as well as possible failure scenarios and their control measures.

4. General and operational availability

4.1 The safety, operational reliability, and dependability of a gas-fuelled engine shall be equivalent to that of a conventional oil-fuelled marine diesel engine.

4.2 The engine shall be capable of safe and reliable operation throughout the entire power range under all expected operation conditions.

4.3 Composition and minimum methane number of gas fuel supplied to the engine shall be in accordance with the engine manufacturer's specification. If gas composition or methane number exceeds specified limits, no dangerous situation shall arise.

4.4 General requirements regarding redundancy of essential systems (main propulsion, electrical power generation, etc.) are to be considered. The same basic requirements apply to gas-fuelled engine installations as for oil-fuelled engine installations.

4.5 Arrangements of the gas-fuelled installation for sustained or restored operation following blackout and dead ship condition shall be carefully evaluated.

4.6 Overall operational availability of the gasfuelled engine installation shall not be reduced by engine safety functions, such as automatic shutdown of external gas supply, to a level lower than achieved by oil-fuelled engine installations. Furthermore, gas leakages anywhere in the gas storage system, gas supply system, or gas engine components shall not cause automatic shutdown of other engines in order to maintain essential functions such as main propulsion power and electrical power generation.

4.7 For single engine main propulsion plants the entire system, including gas supply, machinery space safety concept, and gas engine design shall be evaluated with regard to operational availability and redundancies.

4.8 In general, dual-fuel engines suitable for change-over to oil fuel mode in case of failure in the gas supply system are considered to be the only gas fuelled engines practicable for single engine main propulsion plants.

5. Documents to be submitted

5.1 In addition to the documents defined in B and **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels the documents as listed in Table 3.8 shall be submitted for approval respectively review. Following prior agreement with **TL** they shall be submitted in paper form in triplicate.

6. General requirements

Requirements as specified in the **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels , Part A-1 shall be observed.

6.1 Gas supply concept

6.1.1 Gas-fuelled engines shall either be designed according to Emergency Shut-down Concept (ESD) or Gas Safe Concept (definition and requirements see **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels.

6.1.2 The general design principle (ESD or Gas Safe Concept) will influence the range of acceptable applications with regard to engine room arrangements, engine room safety concept, redundancy concept, propulsion plant, etc.

6.2 Requirements for single gas fuel engines

6.2.1 In general, single gas fuel engines are only considered suitable for electric power generating plants.

6.2.2 The application of single gas fuel engines for mechanical propeller drives requires special evaluation and consideration.

6.3 Requirements for dual-fuel engines

6.3.1 Dual-fuel engines are to be of the dual-fuel type employing pilot fuel ignition and to be capable of immediate change-over to oil fuel only.

Table 3.8 Documents to be submitted for gas-fuelled engines

Item No.	Description
1	General engine concept with regard to gas as fuel (description)
2	Engine specification sheet and technical data
3	Specification of permissible fuel gas properties
4	Engine safety concept, including system FMEA with regard to gas as fuel
5	Definition of hazardous areas
6	General installation manual for the engine type with regard to machinery space layout and equipment
7	Fuel gas system for the engine, including double wall piping system and ventilation system (schematic layout, details, assembly, functional description)
8	Charge air system (schematic layout, functional description, assembly)
9	Engine exhaust gas system (schematic layout, assembly)
10	Explosion relief valves for crankcase, air intake manifold and exhaust manifold (specification, arrangement, determination of minimum number and size required, operating parameters of protected manifolds) refer also to 8.3.3.4
11	Engine control system (schematic layout, functional description, specification)
12	Ignition system (schematic layout, functional description, specification)
13	Combustion monitoring system (schematic layout, functional description, specification)
14	Engine monitoring system (schematic layout, functional description, specification)
15	Engine alarm and safety system (schematic layout, functional description, specification)
16	Gas detection system for the engine (schematic layout, functional description)
17	Electronic components of engine control-, ignition-, alarm-, safety-, monitoring system, etc. (specification, type approvals)
18	List of type approved equipment
19	List of explosion-proof electrical equipment incl. specification of certifications
20	Testing procedure for gas detection system
21	Testing procedure for gas tightness
22	General concept regarding training measures for operating personnel

6.3.2 Only oil fuel is to be used when starting the engine.

6.3.3 Only oil fuel is, in principle, to be used when the operation of an engine is unstable, and/or during manoeuvring and port operations.

6.3.4 In case of shut-off of the gas fuel supply or engine failure related to gas operation, engines are to be capable of continuous operation by oil fuel only.

6.3.5 In general, engine power and speed shall not be influenced during fuel change-over process. An automatic system shall provide for a change-over procedure with minimal fluctuations in engine power and speed.

6.3.6 The change-over process from gas mode to oil mode shall be possible at all operating conditions.

7. Systems

Requirements as specified in the **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels shall be observed.

7.1 Cooling water system

7.1.1 Means are to be provided to degas the cooling water system from fuel gas if the possibility is given that fuel gas can leak directly into the cooling water system.

7.1.2 Suitable gas detectors are to be provided.

7.1.3 Flame arrestors are to be provided at the vent pipes.

7.2 Lubrication oil system

7.2.1 Means are to be provided to degas the lubrication oil system from fuel gas if the possibility is given that fuel gas can leak directly into the lubrication oil system.

7.2.2 Suitable gas detectors are to be provided.

7.2.3 Flame arrestors are to be provided at the vent pipes.

7.3 Fuel oil system

7.3.1 Means are to be provided to degas the fuel oil system from fuel gas if the possibility is given that fuel gas can leak directly into the fuel oil system.

7.3.2 Suitable gas detectors are to be provided.

7.3.3 Flame arrestors are to be provided at the vent pipes.

7.4 External gas supply system

7.4.1 The external gas supply system shall be designed such that the required gas conditions and properties (temperature, pressure, etc.) as specified by the engine maker at engine inlet are adhered to under all possible operating conditions.

7.4.2 Arrangements are to be made to ensure that no gas in liquid state is supplied to the engine, unless the engine is designed to operate with gas in liquid state.

7.4.3 In addition to the automatic shut off supply valve a manually operated valve shall be installed in series in the gas supply line to each engine.

7.5 Gas system on the engine

7.5.1 General requirements

7.5.1.1 Gas piping on an engine shall be designed and installed taking due account of vibrations and movements during engine operation.

7.5.1.2 In case of rupture of a gas pipe or excessive pressure loss, automatic shutdown of the gas supply shall be activated.

7.5.2 Low pressure gas supply

7.5.2.1 Flame arresters shall be provided in the gas supply system on the engine as determined by the system FMEA.

7.5.2.2 Gas admission valves shall be located directly at each cylinder inlet. In general, mixing of fuel gas with combustion air shall not take place before the cylinder inlet.

7.5.2.3 Gas admission by a common gas admission valve and mixing of gas with combustion air before the cylinder inlet may be acceptable subject to an acceptable level of risk being determined in the safety concept and system FMEA.

7.5.3 High pressure gas supply

7.5.3.1 Flame arresters shall be provided at the inlet to the gas supply manifold of dual-fuel engines.

7.5.3.2 The high pressure gas is to be blown directly into the cylinders without prior mixing with combustion air.

7.5.3.3 High pressure gas pipes on the engine shall be carried out in double wall design with leakage detection. The outer pipe is to be designed to withstand serious leakage of the inner high pressure pipe. Gas pressure and temperature is to be considered.

7.5.4 Gas admission valve

7.5.4.1 The gas admission valve shall be controlled by the engine control system according to the actual gas demand of the engine.

7.5.4.2 Uncontrolled gas admission shall be prevented by design measures or indicated by suitable detection and alarm systems. Measures to be taken following detection and alarm are to be examined as part of the system FMEA.

7.6 Ignition system

7.6.1 General requirements

Ignition systems commonly use either electrical spark plugs (single gas fuel engines) or pilot fuel oil injection (dual fuel engines).

7.6.1.1 The ignition system has to ensure proper ignition of the gas at all operating conditions and must be able to provide sufficient ignition energy.

7.6.1.2 Before starting the engine, the engine has to be ventilated without injection or supplying any fuel.

7.6.1.3 Before activating the gas admission to the engine, the ignition system has to be checked automatically to verify correct functioning.

7.6.1.4 Combustion of each cylinder is to be monitored. Misfiring and knocking combustion is to be detected.

7.6.1.5 Safe and reliable operation of the ignition system shall be demonstrated and documented by a system FMEA.

7.6.1.6 During stopping of the engine the fuel gas supply shall be shut off automatically before the ignition source.

7.6.2 Spark ignition

For a spark ignition engine, if ignition has not been detected on each cylinder by the engine monitoring system within an engine specific time after operation of the gas admission valve, gas supply shall be automatically shut off and the starting sequence terminated. Any unburned gas mixture is to be purged from the exhaust system.

7.6.3 Ignition by pilot injection

7.6.3.1 Prior to admission of fuel gas the correct operation of the pilot oil injection system on each cylinder shall be verified.

7.6.3.2 An engine shall always be started using fuel oil only.

7.7 Electrical systems

7.7.1 Care shall be taken to prevent any possible sources of ignition caused by electrical equipment, electrical sensors, etc. installed in hazardous areas.

7.7.2 For electrical equipment and sensors in hazardous areas the explosion protection requirements in the **TL** Rules for Electrical Installations, Section 1 are to be observed.

7.7.3 Systems that shall remain operational when the safety system triggers shut off of the gas supply are to be determined by the system FMEA. Systems to be considered shall include, but not be limited to, the ventilation system, inert gas system and gas detection system.

7.8 Engine control-, monitoring-, alarm-, and safety systems

7.8.1 General requirements

7.8.1.1 General requirements regarding gas supply and automatic activation of gas supply valves (double block and bleed valves, master gas valve) to the engine as defined in the **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels and **TL** Part C, Chapter 10 – Liquefied Gas Carriers shall be observed.

7.8.1.2 Knocking combustion and misfiring is to be detected and combustion conditions are to be automatically controlled to prevent knocking and misfiring.

7.8.1.3 The engine operating mode shall always be clearly indicated to the operating personnel.

7.8.1.4 Guidance for the scope of instrumentation for monitoring, alarm, and safety systems is given in Table 3.9. Depending on engine design, safety concept, and system FMEA examining all possible failure modes, deviations from Table 3.9 may be agreed.

7.8.2 Gas detection

7.8.2.1 A continuous gas detection system shall be provided (see **TL** Rules Chapter 78 Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Part A-1).

7.8.2.2 The gas detection system shall be in operation as long as fuel gas is supplied to the engine.

7.8.2.3 As guidance, the gas detection system shall cover the spaces of the engine as specified in Table 2.12. Depending on engine design, safety concept, and system FMEA deviations from Table 3.9 may be agreed.

7.8.2.4 Manual gas detection may be installed in lieu of continuous gas detection for certain spaces if this is shown to be acceptable by the system FMEA.

7.8.3 Speed control and load acceptance

7.8.3.1 In general, the requirements in F.1 shall be observed.

7.8.3.2 The basic requirements of F.1.2.3 regarding design of the ship's power management system apply.

7.8.3.3 Exemptions from minimum required step loading capability of engines driving electrical generators can be agreed for gasfuelled engines of limited step loading capability.

7.9 Exhaust gas system and ventilation system

7.9.1 Exhaust gas pipes from gas-fuelled machinery are to be installed separately from each other, taking into account structural fire protection requirements.

Table 3.9 Indicative scope of instrumentation for gas-fuelled engines

	Indicator, alarm shutdown (1)	**Shut off of gas supply to individual engine (double block and bleed valves) (1)**	**Shut off of gas supply to machinery space (master gas valve) (1)**	**Comment**
Gas supply				
Gas pressure	I.L.H			
Gas temperature	I.L.H			
Gas admission valve(s) failure	A.S **(2)**	X		Incl. failure of sealing oil cooling etc.
Pressure of inert gas supply	I.L			
Rupture of gas pipe or excessive gas leakage	A.S	X	X	
Failure containment or vacuum of shielded gas piping system	A.S **(2)**	X	X	Gas safe concept
Gas detection				
Gas concentration in air manifold	H			
Gas concentration in crankcase	H			
Gas concentration in exhaust manifold	H			
Gas concentration in below each piston **(3)**	H			
Gas concentration in shielded gas piping system	H.S **(2)**	X	X	
Gas concentration in engine room	H.S	X	X	
Crankcase				
Pressure	H.S	X	X	
Temperature **(4)**	H.S	X	X	
Oil mist concentration	H.S	X	X	
Combustion monitoring				
Misfiring each cylinder	A.S **(2)**	X		
Knocking each cylinder	A.S **(2)**	X		
Cylinder pressure	H.L.S **(2)**	X		
Load deviation	A.S **(2)**	X		
Spark ignition system or pilot injection system failure	A.S **(2)**	X		
Exhaust gas				
Exhaust gas temperature turbocharger inlet and outlet	I.H			
Exhaust gas temperature each cylinder	I.L.H.S **(2)**	X		
Deviation from exhaust gas mean temperature	L.H.S **(2)**	X		
Miscellaneous				
Failure in gas combustion control system	A.S **(2)**	X		
Failure ventilation of shielded gas piping system	A			Gas safe concept
Failure exhaust gas ventilation system	A			
Engine shutdown	A.S	X		Externally or manually activated

I : Indicator
A : Alarm
L : Alarm for lower limit
H : Alarm for upper limit
S : Shutdown
X : Activation

(1) *In general, shut off gas supply and engine shutdown shall not be activated at initial trigger level without pre-alarm.*
(2) *Automatic shutdown shall be replaced by automatic change-over to fuel oil mode for dual-fuel engines subjected to a continued safe*
(3) *Cross-Head type engines*
(4) *Temperature of liners and bearings*

7.9.2 Machinery, including the exhaust gas system, is to be ventilated:

- Prior to each engine start,
- After starting failure,
- After each gas operation of gas-fuelled machinery not followed by an oil fuel operation.

7.9.3 Control of the ventilation system shall be included in the automation system. Failures shall be alarmed.

8. Safety equipment and safety systems

Basic requirements as specified in the **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels shall be observed.

8.1 Safety concept and system FMEA

8.1.1 The safety concept shall describe the safety philosophy with regard to gas as fuel and in particular address how risks associated with this type of fuel are controlled. The safety concept shall also describe possible failure scenarios and the associated control measures.

8.1.2 In the system FMEA possible failure modes related to gas as fuel shall be examined and evaluated in detail with respect to their consequences on the engine and the surrounding systems as well as their likelihood of occurrence and mitigating measures.

Verification tests are to be defined. Aspects to be examined include, but shall not be limited to:

- Gas leakage, both engine internal and release of gas to the engine room – shut off of gas supply (inter alia with respect to systems that shall remain operational, refer 7.7.3)
- Incomplete/ knocking combustion
- Deviation from the specified gas composition
- Malfunction of the ignition system
- Uncontrolled gas admission to engine
- Switch over process from gas to fuel and vice versa for dual fuel engines
- Explosions in crankcase, scavenging air system and exhaust gas system
- Uncontrolled gas air mixing process, if outside cylinder
- Interfaces to other ship systems, e.g. control system, gas supply

8.2 Crankcase safety equipment

8.2.1 Piston failure

Piston failure and abnormal piston blow-by shall be detected and alarmed.

8.2.2 Crankcase

8.2.2.1 Crankcase venting pipes are to be equipped with flame arrestors.

8.2.2.2 A detailed evaluation regarding the hazard potential of fuel gas accumulation in the crankcase is to be carried out and included in the safety concept (see 8.1).

8.2.3 Removal of fuel gas from crankcase and inert gas injection

8.2.3.1 Means shall be provided to measure the fuel gas concentration in the crankcase.

8.2.3.2 Suitable measures, such as inert gas injection, shall be provided to remove fuel gas – air mixtures from the crankcase at engine standstill.

8.2.3.3 Suitable means shall be available to purge inert gas from the crankcase before opening the crankcase for maintenance.

8.2.3.4 Signs requiring a fuel and inert gas free atmosphere in the crankcase before opening of crankcase doors shall be placed in conspicuous locations.

Note:

Means for automatic injection of inert gas into the crankcase are recommended, e.g. in case of:

- *Engine emergency shutdown*
- *Oil mist detection as well as bearing and liner temperature alarm*
- *Fire detection in engine room*

8.3 Explosion relief valves

8.3.1 General requirements

8.3.1.1 Explosion relief devices shall close firmly after an explosion event.

8.3.1.2 The outlet of explosion relief devices shall discharge to a safe location remote from any source of ignition. The arrangement shall minimize the risk of injury to personnel.

8.3.1.3 For type testing procedure of explosion relief devices for combustion air inlet and exhaust gas manifolds of internal combustion engines using gas as fuel see IACS UR M82.

8.3.2 Crankcase explosion relief valves

8.3.2.1 For crankcase safety devices (e.g. explosion relief valves, oil mist detection, etc.) the requirements specified in F.4. are to be observed.

8.3.2.2 Crankcase explosion relief valves are to be provided at each crank throw.

8.3.2.3 The minimum required total relief area of crankcase explosion relief valves is to be evaluated by engine maker considering explosions of fuel gas –air mixtures and oil mist.

8.3.3 Other explosion relief valves

8.3.3.1 As far as required in the **TL** Rules Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, explosion relief valves are to be provided for combustion air inlet manifolds and exhaust manifolds.

8.3.3.2 Explosion relief valve shall generally be approved by **TL** for the application on inlet manifolds and exhaust manifolds of gas-fuelled engines.

8.3.3.3 For the approval of relief valves the following documentation is to be submitted (usually by the maker of explosion relief valve):

- Drawings of explosion relief valve (sectional drawings, details, assembly, etc.)
- Specification data sheet of explosion relief valve (incl. specification of operation conditions such as max. working pressure, max. working temperature, opening pressure, effective relief area, etc.)
- Test reports

8.3.3.4 In addition to the approval under 8.3.3.3 the arrangement of explosion relief valves shall be approved for each engine type. The following documents are to be submitted (usually by the engine manufacturer):

- Drawing of arrangement of explosion relief valves (incl. number, type, locations, etc.)
- Drawings of protected component (air inlet manifold, exhaust manifold, etc.) (incl. specification of max. working pressure, max. working temperature, max. permissible explosion pressure, etc.)
- Evidence for effectiveness of flame arrestor at actual arrangement
- Evidence for effectiveness of pressure relief at explosion (sufficient relief velocity, sufficient relief pressure)

Note:

Evidence can be provided by suitable tests or by theoretical analysis.

9. Tests

9.1 Type approval test for gas-fuelled engines

9.1.1 Gas-fuelled engines shall be type approved by **TL**.

9.1.2 The scope of type approval testing stated in E.4. applies as far as pertinent also to gas-fuelled engines. Additional or differing requirements reflecting gas specific aspects are listed below. The type test program is to be agreed with **TL**.

9.1.3 Tests:

- Load acceptance test and load cut off
- Fuel change-over procedures (for dual fuel engines)
- Combustion monitoring
- Safety system
- Alarm system
- Monitoring system
- Control system
- Gas detection
- Tightness tests of gas piping and double wall pipes and ducts
- Ignition system
- Automatic gas shut off
- Turbocharger waste gate, by-pass, etc.
- Ventilation system
- Start, stop, emergency stop
- Verification tests resulting from the system FMEA

9.2 Works trials

In addition to the requirements of E.5., the following items shall be tested during works trials of gas fuelled engines:

- Tightness test of gas system
- Testing of systems for combustion monitoring
- Testing of gas shut off and fuel change-over (dual-fuel engines) procedures

9.3 Shipboard trials

In addition to the requirements of E.6., during shipboard trials the following items shall be tested:

- Tightness test of gas system
- Testing of systems for combustion monitoring
- Testing of gas shut off and fuel change-over (dual-fuel engines) procedures
- Testing of ventilation systems and gas detection systems

10. Machinery spaces

10.1 Sufficient air exchange and air flow shall be ensured around the engine to prevent accumulation of explosive, flammable, or toxic gas concentrations.

10.2 Direction of air flow in machinery spaces shall be directed in such way as to avoid flow of any leaking gas towards potential sources of ignition.

10.3 Machinery spaces shall have sufficient openings to the outside to allow pressure relief from the machinery space in case of an explosion event inside a gas-fuelled engine installed in the space.

10.4 Sign plates shall be fixed at adequate locations to make notice of gas-fuelled machinery to persons entering the relevant machinery spaces. Instructions regarding operation as well as behavior in case of gas leaks and failure of machinery are to be provided at prominent positions in machinery spaces.

11. Training

Personnel operating gas-fuelled engines aboard a vessel shall be duly trained regarding operation of the specific engine, gas supply systems, safety- and control systems, etc. installed on the vessel.

12. Spare parts

Spare parts, which are of major importance for the safety and operational reliability of the gas-fuelled engine, as well as parts with limited lifetime, shall be provided on board in addition to those required in Section 17.

13. Retrofit

Acceptance criteria and procedure for conversion of existing oil-fuelled diesel engines into gas-fuelled or dual-fuel engines are to be individually agreed with **TL**.

M. Reciprocating Internal Combustion Engines Fuelled by Natural Gas

1. General

1.1 Scope

1.1.1 Type of engines

This subsection addresses the requirements for marine reciprocating internal combustion engines supplied with natural gas as fuel.

The scope of the subsection is intended for natural gas fuelled engines. It may also be referred for engines using similar fuels with main component methane such as bio-methane or synthetic methane.

It shall be ensured by the gas supply system that the gas supplied to the engine is always in gaseous state.

This subsection does not cover requirements for liquid or cryogenic gas.

The engines can be dual fuel engines (hereinafter referred to as DF engines), gas fuel only engines (hereinafter referred to as GF engines), or any variations thereof including fuel sharing capability.

DF engines and GF engines may not be permitted for emergency applications.

This subsection is to be applied in association with other relevant **TL** internal combustion engine requirements, as far as found applicable to the specific engine design.

The mandatory international codes for gas carriers (IGC Code) and for other ships burning low flashpoint fuels (IGF Code) must also be considered, as applicable.

Specific requirements of the IGF Code as referenced in this subsection shall be applied to engine types covered by this subsection installed on any ship, regardless of type, size and trading area, as long as the IGC Code is not referenced or explicitly specified otherwise.

1.2 Definitions

1.2.1 Certified safe equipment is equipment certified by an independent national test institution or competent body to be in accordance with a recognised standard for electrical apparatus in hazardous areas.

Note:

Refer to IEC 60079 series, Explosive atmospheres and IEC 60092-502:1999 Electrical Installations in Ships – Tankers – Special Features.

1.2.2 Double block and bleed valves means the set of valves referred to in:

- Chapter 10 - Liquefied Gas Carriers, Section 16, Item 16.4.5 (IGC Code, 16.4.5)
- Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 2.2.9 and Items 9.4.4 to 9.4.6 (IGF Code, 2.2.9 and 9.4.4 to 9.4.6)

1.2.3 Dual fuel engine ("DF engine") means an engine that can burn natural gas as fuel simultaneously with liquid fuel, either as pilot oil or bigger amount of liquid fuel (gas mode), and also has the capability of running on liquid diesel fuel oil only (Diesel mode).

1.2.4 Explosion relief device means a device to protect personnel and component against a determined overpressure in the event of a gas explosion. The device may be a valve, a rupture disc or other, as applicable.

1.2.5 Gas means natural gas used as fuel consisting primarily of methane.

Note:

Gas may also be bio-methane or synthetic methane etc. with methane as main component.

1.2.6 Gas admission valve is a valve or injector on the engine, which controls gas supply to the cylinder(s) according to the engine's gas demand.

1.2.7 Gas engine means a DF engine, a GF engine, or any variations thereof.

1.2.8 Gas fuel only engine ("GF engine") means an engine capable of operating on gas fuel only and not able to switch over to oil fuel operation.

1.2.9 Gas piping means piping containing gas or air / gas mixtures.

1.2.10 High pressure gas means gas with a maximum working pressure greater than 10 bar gauge.

1.2.11 IGC Code means the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk, as amended.

1.2.12 IMO means the International Maritime Organisation

1.2.13 IGF Code means the International Code of Safety for Ships Using Gases or other Low-Flashpoint Fuels (IMO Resolution MSC.391(95), as amended).

1.2.14 Low pressure gas means gas with a maximum working pressure lower or equal to 10 bar gauge.

1.2.15 Lower Heating Value ("LHV") means the amount of heat produced from the complete combustion of a specific amount of fuel, excluding latent heat of vaporization of water.

1.2.16 Methane Number is a measure of resistance of a gas fuel to knock, which is assigned to a test fuel based upon operation in knock testing unit at the same standard knock intensity.

Note:

Pure methane is used as the knock resistant reference fuel, that is, methane number of pure methane is 100, and pure hydrogen is used as the knock sensitive reference fuel, methane number of pure hydrogen is 0.

1.2.17 Pilot fuel means the fuel oil that is injected into the cylinder to ignite the main gas-air mixture on DF engines.

1.2.18 Pre-mixed engine means an engine where gas is supplied in a mixture with air through a common manifold for all cylinders, e.g. mixed before or after the turbocharger.

1.2.19 Recognized standards means applicable international or national standards acceptable to **TL** or standards laid down and maintained by an organisation which complies with the standards adopted by IMO and which are recognized by **TL** .

1.2.20 Safety Concept is a document describing the safety philosophy with regard to gas as fuel. It describes how risks associated with this type of fuel are controlled under reasonably foreseeable abnormal conditions as well as possible failure scenarios and their control measures. The results of the risk analysis, see 1.4, shall be reflected in the safety concept.

Note:

A detailed evaluation regarding the hazard potential of injury from a possible explosion is to be carried out and reflected in the safety concept of the engine.

1.3 Documents and drawings to be submitted

1.3.1 Documents and drawings to be submitted for the approval of DF and GF engines.

The following documents are to be submitted for the approval of DF and GF engines, in addition to those required in Table 3.1, 3.2 and 3.3.

No.	Item
1	Schematic layout or other equivalent documents of gas system on the engine
2	Gas piping system (including double-walled arrangement where applicable)
3	Parts for gas admission system **(3)**
4	Arrangement of explosion relief valves (crankcase **(1)**, charge air manifold, exhaust gas manifold and exhaust gas system on the engine) as applicable
5	List of certified safe equipment and of relevant certification
6	Safety concept (for information)
7	Report of the risk analysis **(2)** (for information)
8	Gas used as fuel specification (for information)

(1) *If required by Table 3.2 and 3.3, see also 2.2.5.1.*
(2) *See 1.4.*
(3) *The documentation to contain specification of design pressures, working pressure, pipe dimensions and materials.*

1.3.2 Documents and drawings to be submitted for the approval of DF engine

No.	Item
9	Schematic layout or other equivalent documents of pilot fuel system
10	Shielding of high pressure fuel pipes for pilot fuel system, assembly
11	High pressure parts for pilot fuel oil injection system **(3)**

(3) *The documentation to contain specification of design pressures, working pressure, pipe dimensions and materials.*

1.3.3 Documents and drawings to be submitted for the approval of GF engine

No.	Item
12	Schematic layout or other equivalent documents of the ignition system

1.3.4 Where considered necessary, **TL** may request further documents to be submitted.

1.4 Risk analysis

1.4.1 Scope of the risk analysis

The risk analysis is to address:

- a failure or malfunction of any system or component involved in the gas operation of the engine
- a gas leakage downstream of the double block and bleed valves
- the safety of the engine in case of emergency shutdown or blackout, when running on gas
- the inter-actions between the gas fuel system and the engine.

Note:
With regard to the scope of the risk analysis it shall be noted that failures in systems external to the engine, such as fuel storage or fuel gas supply systems, may require action from the engine control and monitoring system in the event of an alarm or fault condition. Conversely failures in these external systems may, from the vessel perspective, require additional safety actions from those required by the engine limited risk analysis required by this subsection.

1.4.2 Form of the risk analysis

The risk analysis is to be carried out in accordance with international standard IEC 31010 Risk management - Risk assessment techniques, or other recognized standards.

The required analysis is to be based on the single failure concept, which means that only one failure needs to be considered at the same time. Both detectable and non-detectable failures are to be considered. Consequences failures, i.e. failures of any component directly caused by a single failure of another component, are also to be considered.

1.4.3 Procedure for the risk analysis

The risk analysis is to:

a) Identify all the possible failures in the concerned equipment and systems which could lead:

- to the presence of gas in components or locations not designed for such purpose, and/or

- to ignition, fire or explosion.

b) Evaluate the consequences (see also 2.1.2)

c) Where necessary, identify the failure detection method

d) Where the risk cannot be eliminated, identify the corrective measures:

- in the system design, such as:

  - redundancies

  - safety devices, monitoring or alarm provisions which permit restricted operation of the system

- the system operation, such as:

  - initiation of the redundancy

  - activation of an alternative mode of operation.

The results of the risk analysis are to be documented.

1.4.4 Equipment and systems to be analysed

The risk analysis required for engines is to cover at least the following aspects:

a) failure of the gas-related systems or components, in particular:

- gas piping and its enclosure, where provided

- gas admission valves

Note:

Failures of the gas supply components not located directly on the engine, such as block-and-bleed valves and other components of the gas supply system, are not to be considered in the analysis.

b) failure of the ignition system (oil fuel pilot injection or sparking plugs, glow plugs)

c) failure of the air to fuel ratio control system (charge air by-pass, gas pressure control valve, etc.)

d) for engines where gas is supplied upstream of the turbocharger compressor, failure of a component likely to result in a source of ignition (hot spots)

e) failure of the gas combustion or abnormal combustion (misfiring, knocking)

f) failure of the engine monitoring, control and safety systems

Note:

Where engines incorporate electronic control systems, a failure mode and effects analysis (FMEA) is to be carried out in accordance with Table 3.1, Footnote 5.

g) presence of gas in engine components (e.g. air inlet manifold or scavenge space and exhaust manifold) and in the external systems connected to the engines (e.g. exhaust duct, cooling water system, hydraulic oil system, etc.).

h) changes of operating modes for DF engines

i) hazard potential for crankcase fuel gas accumulation, for trunk-piston engines, refer to Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 10.3.1.2 (IGF Code 10.3.1.2) and IACS UR M10.

j) risk of crankcase explosion in connection with active crankcase ventilation which produces a flow of external air into the crankcase, (see IACS UR M10).

2. Design Requirements

2.1 General Principles

2.1.1 The manufacturer is to declare the allowable gas composition limits for the engine and the minimum and (if applicable) maximum methane number.

2.1.2 Components containing or likely to contain gas are to be designed to:

a) minimise the risk of fire and explosion so as to demonstrate an appropriate level of safety commensurate with that of an oil-fuelled engine;

b) mitigate the consequences of a possible explosion to a level providing a tolerable degree of residual risk, due to the strength of the component(s) or the fitting of suitable explosion relief devices of an approved type.

The strength of the component(s) of arrangement of explosion relief devices shall be documented (e.g., as part of risk analysis) or otherwise demonstrated to be sufficient for a worst-case explosion.

Also refer to the Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 10.2 and 10.3 (IGF Code 10.2 and 10.3).

2.1.3 Discharge from explosion relief devices shall prevent the passage of flame to the machinery space and be arranged such that the discharge does not endanger personnel or damage other engine components or systems"

2.1.4 Explosion relief devices shall be fitted with a flame arrester.

2.2 Design Requirements

2.2.1 Gas piping

2.2.1.1 General

The requirements of this section apply to engine-mounted gas piping. The piping shall be designed in accordance with the criteria for gas piping (design pressure, wall thickness, materials, piping fabrication and joining details etc.) as given in the Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Section 7 (IGF Code chapter 7) or Chapter 10 - Liquefied Gas Carriers, Section 5, Item 5.1 to 5.9 and Section 16 (IGC Code chapter 5.1 to 5.9 and 16) as applicable.

Other connections as mentioned in Chapter 78, 7.3.6.4.4 (IGF Code 7.3.6.4.4) may be accepted subject to type approval in accordance with the requirements of IACS UR P2.7 and P2.11.

All single walled or high-pressure gas pipes should be considered as Class I.

Low pressure double walled gas pipes should be considered as Class II.

All secondary enclosures for gas pipes should be considered as Class II.

Single walled gas vent pipes, if permitted, should be considered as Class I, except it is justified that the maximum built up pressure is less than 5 bar gauge, in which case it should be considered as Class II.

Gas vent pipes protected by a secondary enclosure should be considered as Class II.

Secondary enclosure for vent pipes should be considered as Class III.

Table 3.10 Design pressure for gas pipes

	Design pressure	
Gas pipe, low pressure	see Chapter 78 (IGF Code), 7.3.3.1	see Chapter 10 (IGC Code), 5.4.1
Gas pipe, high pressure	see Chapter 78 (IGF Code), 7.3.3.1	see Chapter 10 (IGC Code), 5.4.1
outer pipe, low pressure	see Chapter 78 (IGF Code), 9.8.1	see Chapter 10 (IGC Code), 5.4.4
outer pipe, high pressure	see Chapter 78 (IGF Code), 9.8.2	see Chapter 10 (IGC Code), 5.4.4
Open ended pipe	see Chapter 78 (IGF Code), 7.3.3.2	see Chapter 10 (IGC Code), 5.4.1

Flexible bellows used in the fuel gas system on the engine shall be approved based on the requirements of Chapter 78, item 16.7.2 (IGF Code 16.7.2), and Chapter 10, item 5.13.1.2 (IGC Code 5.13.1.2), as applicable.

The number of cycles, pressure, temperature, axial movement, rotational movement and transverse movement which the bellow will encounter in actual service on the engine should be specified by the engine designer.

Endurance against high cycle fatigue due to vibration loads shall be verified by testing or alternatively be documented by the Expansion Joint Manufacturers

Association, Inc. (EJMA) calculation or equivalent (i.e., more than 10^7 cycles).

Note:
The fatigue test due to ship deformations in Chapter 78, item 16.7.2.4 (IGF 16.7.2.4) is considered not relevant for bellows which are an integral part of the engine.

2.2.2 Arrangement of the gas piping system on the engine

Pipes and equipment containing fuel gas are defined as hazardous area Zone 0 (refer to Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 12.5.1 (IGF Code 12.5.1)).

The space between the gas fuel piping and the wall of the outer pipe or duct is defined as hazardous area Zone 1 (refer to Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 12.5.2.6 (IGF Code 12.5.2.6)).

2.2.2.1 Normal “double wall” arrangement

The gas piping system on the engine shall be arranged according to the principles and requirements of Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 9.6 (IGF Code 9.6). For gas carriers, Chapter 10 - Liquefied Gas Carriers, Item 16.4.3 (IGC Code 16.4.3) applies.

The design criteria for the double pipe or duct are given in the Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 9.8 and 7.4.1.4 (IGF Code 9.8 and 7.4.1.4).

In case of a ventilated double wall, the ventilation inlet is to be located in accordance with the provisions of Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 13.8.3 (IGF Code, regulation 13.8.3). For gas carriers, Chapter 10 - Liquefied Gas Carriers, Item 16.4.3.2 (IGC Code 16.4.3.2) applies.

The pipe or duct is to be pressure tested at 1.5 x design pressure to ensure gas tight integrity and to show that it can withstand the expected maximum pressure at gas pipe rupture.

2.2.2.2 Alternative arrangement

Single walled gas piping is only acceptable

a) for engines supplied with low pressure gas and installed in ESD protected machinery spaces, as defined in Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 5.4.1.2 (IGF Code 5.4.1.2) and in compliance with other relevant parts of the Chapter 78 (e.g. 5.6) (IGF Code (e.g. 5.6));

b) in the case as per footnote(19) of to item 9.6.2 of Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels (footnote 18 to paragraph 9.6.2 of IGF Code).

For gas carriers, the IGC Code applies.

In case of gas leakage in an ESD-protected machinery space, which would result in the shutdown of the engine(s) in that space, a sufficient propulsion and manoeuvring capability including essential and safety systems is to be maintained.

Therefore the safety concept of the engine is to clearly indicate application of the “double wall” or “alternative” arrangement.

Note:
The minimum power to be maintained is to be assessed on a case-by-case basis from the operational characteristics of the ship.

2.2.3 Charge air system on the engine and exhaust gas system

The charge air system and the exhaust gas system on the engine are to be designed in accordance with 2.1.2 above. In case of a single engine installation, the engine is to be capable of operating at sufficient load to maintain power to essential consumers after opening of the explosion relief devices caused by an explosion event. Sufficient power for propulsion capability is to be maintained.

Note:
Load reduction is to be considered on a case by case basis, depending on engine configuration (single or multiple) and relief mechanism (self-closing valve or rupture disk).

2.2.4 Continuous relief of exhaust gas (through open rupture disc) into the engine room or other enclosed spaces is not acceptable.

Suitable explosion relief system for air inlet manifolds, scavenge spaces and exhaust system should be provided unless designed to accommodate the worst-case overpressure due to ignited gas leaks or justified by the safety concept of the engine. A detailed evaluation regarding the hazard potential of overpressure in air inlet manifolds, scavenge spaces and exhaust system should be carried out and reflected in the safety concept of the engine.

Explosion relief devices for air inlet and exhaust manifold shall be type approved according to IACS UR M82.

The necessary total relief area and the arrangement of the explosion relief devices shall be determined taking into account:

- The worst-case explosion pressure depending on initial pressure and gas concentration,
- the volume and geometry of the component, and
- the strength of the component.

The arrangement shall be determined in the risk analysis (see item 1.4.4.g) and reflected in the safety concept.

2.2.5 Engine crankcase

2.2.5.1 Crankcase explosion relief valves

Crankcase explosion relief valves are to be installed in accordance with F.4. Refer also to Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 10.3.1.2 (IGF Code 10.3.1.2).

For engines not covered by IACS UR M9, the detailed evaluation as required in 1.4.4.i is to determine if crankcase explosion relief valves are necessary.

2.2.5.2 Inerting

For maintenance purposes, a connection, or other means, are to be provided for crankcase inerting and ventilating and gas concentration measuring.

2.2.5.3 Crankcase ventilation

Ventilation of crankcase (either supply or extraction), if arranged, is to comply with IACS UR M10. Relevant evidence is to be documented in Safety Concept.

The ventilation systems for crankcase, sump and other similar engine spaces are to be independent from the systems on the other engines.

2.2.6 Gas ignition in the cylinder

2.2.6.1 Requirements of Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels, Item 10.3(IGF Code 10.3) apply. For gas carriers, Chapter 10 - Liquefied Gas Carriers, Item 16.7 (IGC Code 16.7) applies.

2.2.7 Control, monitoring, alarm and safety systems

The engine control system is to be independent and separate from the safety system.

The gas admission valves are to be controlled by the engine control system or by the engine gas demand.

Combustion is to be monitored on an individual cylinder basis.

In the event that poor combustion is detected on an individual cylinder, gas operation may be allowed in the conditions specified in item 10.3.1.6 of Chapter 78 – Rules for Classification of Ships Using Gases or Other Low-Flashpoint Fuels (IGF Code 10.3.1.6).

If monitoring of combustion for each individual cylinder is not practicable due to engine size and design, common combustion monitoring may be accepted.

Unless the risk analysis required by 1.4 of this subsection proves otherwise, the monitoring and safety system functions for DF or GF engines are to be provided in accordance with Table 3.11 of this subsection in addition to the general monitoring and safety system functions given by **TL**.

Note:

For DF engines, Table 3.11 applies only to the gas mode.

Table 3.11 Monitoring and Safety System Functions for DF and GF Engines

Parameter	Alarm	Alarm Automatic activation of the double block-andbleed valves	Automatic switching over to oil fuel mode (1)	Engine shutdown
Abnormal pressures in the gas fuel supply line	X	X	X	X **(5)**
Gas fuel supply systems - malfunction	X	X	X	X **(5)**
Pilot fuel injection or spark ignition systems - malfunction	X	X **(2)**	X	X **(2)(5)**
Exhaust gas temperature after each cylinder - high	X	X **(2)**	X	X **(2)(5)**
Exhaust gas temperature after each cylinder, deviation from average – low **(3)**	X	X **(2)**	X	X **(2)(5)**
Cylinder pressure or ignition - failure, including misfiring, knocking and unstable combustion	X	X **(2)(4)**	X **(4)**	X **(2)(4)(5)**
Oil mist concentration in crankcase or bearing temperature **(6)** - high	X	X		X**(9)**
Pressure in the crankcase – high **(8)**	X	X	X	
Engine stops - any cause	X	X		
Failure of the control-actuating medium of the block and bleed valves	X	X	X	
Failure of crankcase ventilation system, if applicable	X	X **(7)**	X **(7)**	

Footnotes:

(1) *DF engine only, when running in gas mode*

(2) *For GF engines, the double block-and-bleed valves and the engine shutdown may not be activated in case of specific failures affecting only one cylinder, provided that the concerned cylinder can be individually shutoff and the safe operation of the engine in such conditions is demonstrated by the risk analysis.*

(3) *Required only if necessary for the detection of misfiring*

(4) *In the case where the failure can be corrected by an automatic mitigation action, only the alarm may be activated. If the failure persists after a given time, the safety actions are to be activated.*

(5) *GF engine only*

(6) *Where required by TL-R M10*

(7) Automatic safety actions to be activated as specified by the engine manufacturer, see IACS UR M10

(8) Only for trunk piston engines. This pressure sensor cannot replace or substitute a gas detector.

(9) Only for trunk piston engines. For crosshead engines slow down shall apply (see IACS UR M35 Tab.1)

2.2.8 Gas admission valves

Electrically operated gas admission valves shall be certified safe as follows:

- The inside of the valve contains gas and shall therefore be certified for Zone 0.
- When the valve is located within a pipe or duct in accordance with 2.2.2.1, the outside of the valve shall be certified for Zone 1.
- When the valve is arranged without enclosure in accordance with the "ESD-protected machinery space" (see 2.2.2.2) concept, no certification is required for the outside of the valve, provided that the valve is de-energized upon gas detection in the space.

However, if they are not rated for the zone they are intended for, it shall be documented that they are suitable for that zone. Documentation and analysis are to be based on IEC 60079-10- 1 or IEC 60092-502.

Gas admission valves operated by hydraulic oil system are to be provided with sealing arrangement to prevent gas from entering the hydraulic oil system.

3. Specific Design Requirements

3.1 DF Engines

3.1.1 General

The maximum continuous power that a DF engine can develop in gas mode may be lower than the approved MCR of the engine (i.e. in oil fuel mode), depending in particular on the gas composition and its quality or the engine design.

This maximum continuous power available in gas mode and the corresponding conditions shall be stated by the engine manufacturer.

3.1.2 Starting, changeover and stopping

DF engines are to be arranged to be started using either oil fuel or gas fuel with pilot oil fuel for ignition. The engines are to be arranged for rapid changeover from gas use to fuel oil use. In the case of changeover to either fuel supply, the engines are to be capable of continuous operation using the alternative fuel supply without interruption to the power supply.

Changeover to gas fuel operation is to be only possible at a power level and under conditions where it can be done with acceptable reliability and safety as demonstrated through testing.

Changeover from gas fuel operation mode to oil fuel operation mode is to be possible at all situations and power levels.

The changeover process itself from and to gas operation is to be automatic but manual interruption is to be possible in all cases.

If the power level or other conditions do not allow safe and reliable gas operation, changeover to oil fuel mode shall be automatically performed.

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

3.1.3 Pilot injection

Gas supply to the combustion chamber is not to be possible without operation of the pilot oil injection.

Note:
Pilot injection is to be monitored for example by fuel oil pressure and combustion parameters.

3.2 GF Engines

3.2.1 Spark ignition system

In case of failure of the spark ignition, the engine is to be shut down except if this failure is limited to one cylinder, subject to immediate shut off of the cylinder gas supply and provided that the safe operation of the engine is substantiated by the risk analysis and by tests.

3.3 Pre-Mixed Engines

3.3.1 Charge air system

Inlet manifold, turbo-charger, charge air cooler, etc. are to be regarded as parts of the fuel gas supply system.Failures of those components likely to result in a gas leakage are to be considered in the risk analysis (see 1.4).

Flame arresters are to be installed before each cylinder head, unless otherwise justified in the risk analysis, considering design parameters of the engine such as the gas concentration in the charge air system, the path length of the gas-air mixture in the charge air system, etc.

3.4 Two-stroke engines

3.4.1 Scavenge air system

The risk analysis required in 1.4 is to cover the possible gas accumulation in a scavenge space.

3.4.2 Crankcase

The risk analysis required in 1.4 is to cover the possible failure of a piston rod stuffing box.

4. Type Testing, Factory Acceptance Tests and Shipboard Trials

4.1 Type Testing

4.1.1 General

Type approval of DF and GF engines is to be carried out in accordance with E.3, taking into account the additional requirements below.

4.1.2 Type of engine

In addition to the criteria given in E.3.3.3 the type of engine is defined by the following:

- gas admission method (cylinder injection after compression stroke, cylinder-individual injection before compression stroke, or pre-mixed)

- gas admission valve operation (mechanical or electronically controlled)

- ignition system (pilot injection, spark ignition, glow plug or gas self-ignition)

- ignition system (mechanical or electronically controlled)

Note:
Cylinder-individual injection before compression stroke may be port injection into the air inlet channel before the cylinder inlet valve, injection into the cylinder before or during compression stroke, or similar arrangements.

4.1.3 Safety precautions

In addition to the safety precautions mentioned in E.3.4, measures to verify that gas fuel piping on engine is gas tight are to be carried out prior to start-up of the engine.

4.1.4 Test programme

The type testing of the engine is to be carried out in accordance with E.3.5, taking into account the additional requirements of this subsection.

The 110% load tests are not required in the gas mode.

The influence of the methane number and LHV of the fuel gas is not required to be verified during the Stage B type tests. It shall however be justified by the engine designer through internal tests or calculations and documented in the type approval test report.

4.1.5 Measurements and records

In addition to the measurements and records required in E.3.6, the following engine data are to be measured and recorded:

- Each fuel index for gas and diesel as applicable (or equivalent reading)

- Gas pressure and temperature at the inlet of the gas manifold

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

- Gas concentration in the crankcase

Note:
The gas concentration in the crankcase should normally be measured inside the crankcase or at the crankcase outlet (crankcase vent pipe).

Gas concentration measurements may be carried out as part of Stage A if the method and the results are properly documented.

Additional measurements may be required in connection with the design assessment.

4.1.6 Stage A – internal tests

In addition to tests required in E.3.7, the following conditions are to be tested:

- DF engines are to run the load points defined in E.3.7 in both gas and diesel modes (with and without pilot injection in service) as found applicable for the engine type.

- For DF engines with variable liquid / gas ratio, the load tests are to be carried out at different ratios between the minimum and the maximum allowable values.

- For DF engines, switch over between gas and diesel modes are to be tested at different loads.

- The influence of the methane number and LHV of the fuel gas on the engine's maximum continuous power available in gas mode is to be verified.

4.1.7 Stage B – witnessed tests

4.1.7.1 General

Gas engines are to undergo the different tests required in E.3.8.

In case of DF engine,

- all load points must be run in both gas and diesel modes that apply for the engine type as defined by the engine designer. The independent overspeed protection device has to be tested both in gas and diesel mode (E.3.8.2).

- For engines with variable liquid / gas ratio, selected load tests are to be carried out at different ratios between the minimum and the maximum allowable value. (most relevant and critical loads and ratios should be selected for the test)

- The maximum continuous power available in gas mode (see 3.1.1) is to be demonstrated.

- Overload testing is not required in gas mode for DF engines, provided that changeover to oil fuel mode is automatically performed in case of overload.

- The load tests are to be carried out in diesel mode and in gas mode at the different percentages of the engine's MCR.

4.1.7.2 Functional tests

In addition to the functional tests required in E.3.8.3, the following tests are to be carried out:

- For DF engines, the lowest specified speed is to be verified in diesel mode and gas mode.

- For DF engines, switch over between gas and diesel modes are to be tested at different loads.

- For DF engines, verification of automatic changeover to diesel mode when the load demand exceeds the maximum continuous power available in gas mode (see 3.1.1 and 3.1.2)

- The efficiency of the ventilation arrangement or other approved principal of the double walled gas piping system is to be verified.

Engines intended to produce electrical power are to be tested as follows:

- Capability to take sudden load and loss of load in accordance with the provisions of TL-R M3.2.

- For GF and premixed engines, the influences of LHV, methane number and ambient conditions on the dynamic load response test results are to be theoretically determined and specified in the test report. Referring to the limitations as specified in 2.1.1, the margin for satisfying dynamic load response is to be determined.

Notes:

1. For DF engines, switchover to oil fuel during the test is acceptable.

2. Application of electrical load in more than 2 load steps can be permitted in the conditions stated in TL-R M3.2.3.

4.1.7.3 Integration Tests

GF and DF engines are to undergo integration tests to verify that the response of the complete mechanical, hydraulic and electronic engine system is as predicted for all intended operational modes. The scope of these tests is to be agreed with **TL** for selected cases based on the risk analysis required in 1.4 of this subsection, and shall at least include the following incidents:

- Failure of ignition (spark ignition or pilot injection systems), both for one cylinder unit and common system failure
- Failure of a gas admission valve
- Failure of the combustion (to be detected by e.g. misfiring, knocking, exhaust temperature deviation, etc.)
- Abnormal gas pressure
- Abnormal gas temperature **(4)**

4.1.8 Stage C – Component inspection

Component inspection is to be carried out in accordance with the provisions of E.3.9.

The components to be inspected after the test run are to include also:

- gas admission valve including pre-chamber as found applicable
- spark igniter (for GF engines)
- pilot fuel injection valve (for DF engines)

4.1.9 Engine type approval certificate

For DF engines, the maximum continuous power available in gas mode should be specified on the type approval certificate in addition to the maximum continuous rating in diesel mode if differing.

(4) *This test may be carried out using a simulation signal of the temperature*

.

4.2 Factory Acceptance Test

4.2.1 General

Factory acceptance tests of DF and GF engines are to be carried out in accordance with E.4, taking into account the additional requirements below.

For DF engines, the load tests referred to in E.4.3.3 are to be carried out in diesel mode and in gas mode at the different percentages of the engine's MCR. Maximum continuous power available in gas mode is to be demonstrated (see 3.1.1). The 110% load test is not required in the gas mode.

4.2.2 Safety precautions

In addition to the safety precautions mentioned in E.4.1, measures to verify that gas fuel piping on engine is gas tight are to be carried out prior to start-up of the engine.

4.2.3 Records

In addition to the records required in E.4.3.2, the following engine data are to be recorded:

- Fuel index, both gas and diesel as applicable (or equivalent reading)
- Gas pressure and temperature
- Pilot fuel temperature and pressure (supply or common rail as appropriate)

4.2.4 Test loads

Test loads for various engine applications are given in E.4.3.3. DF engines are to be tested in both diesel and gas mode as found applicable. In addition the scope of the trials may be expanded depending on the engine application, service experience, or other relevant reasons.

4.2.5 Integration tests

GF and DF engines are to undergo integration tests 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 **TL** for selected cases based on the risk analysis required in 1.4 of this subsection and shall at least include the following incidents:

- Failure of ignition (spark ignition or pilot injection systems), for one cylinder unit
- Failure of a gas admission valve
- Failure of the combustion (to be detected by e.g. misfiring, knocking, exhaust temperature deviation, etc.)
- Abnormal gas pressure
- Abnormal gas temperature

The above tests may be carried out using simulation or other alternative methods, subject to special consideration by **TL**.

4.3 Shipboard Trials

A leak test is to be carried out for the gas piping system (Chapter 78, item 16.7.3.3(IGF Code 16.7.3.3)) after assembly on board.

Shipboard trials are to be carried out in accordance with the provisions of E.4.4, considering the additional requirements below.

For DF engines, the test loads required in E.4.4.4 are to be carried out in all operating modes (gas mode, diesel mode, etc.) as applicable (see 3.1.1).

The maximum continuous power available in gas mode is to be demonstrated.

Note:
If a test load is performed in all applicable operation modes without interruption (direct changeover at same power and speed), the duration as required in E.4.4.4 may be considered as the total duration demonstrated in all fuel modes. However, demonstration at each mode shall not be less than one hour.

.

The starting maneuvers required in E.4.4.2 are to be carried out in diesel mode and gas mode, if applicable

For DF engines, automatic switching over to oil fuel mode is to be tested.

Further, manual change over from diesel to gas mode and vice versa is to be tested.

5. Certification of Engine Components

The principals, definitions, and general requirements of E.2 apply.

In addition to those components specified in E.2, the engine components listed in Table 3.12 shall be documented as listed in the table.

Table 3.12: Required documentation for engine components

Part	Material properties	Non-destructive examination	Pressure testing	Visual inspection of welds	Component certificate
Gas Pipe Low-pressure double walled	W(C+M)	W 2), 6)	W 4)	X	
Single walled Gas pipes	W(C+M)	W 1)	W 4)	X	SC
High-pressure gas pipes	W(C+M)	W 1)	W 4)	X	SC
Secondary enclosure for gas pipes	W(C+M)	W 2)	W 3)	X	
Gas pipe Low-pressure, Flanges*	W(C+M)	W 2), 6)		X	
Gas pipe High-pressure, Flanges*	W(C+M)	W 1)		X	SC
Gas pipe Low-pressure, Fittings and other components	W(C+M)		W 4)	X	
Gas pipe High-pressure, Fittings and other components	W(C+M)		W 4)	X	SC
Gas pipe Low-pressure Bodies of valves, 7)	W(C+M)		W 4)		
Gas pipe High-pressure Bodies of valves	W(C+M)		W 4)		SC
Gas venting pipes and flanges*, build up pressure less than 5.0bar	TR(C+M)	W 2)	W 4)	X	
Gas venting pipes and flanges*, build up pressure at 5.0bar or more with secondary enclosure	TR(C+M)	W 2)	W 4)	X	
Gas venting pipes and flanges*, build up pressure at 5.0bar or more	W(C+M)	W 1)	W 4)	X	SC
Gas venting pipes Secondary enclosure			W 5)	X	

Footnotes:

1) 100 % radiographic or ultrasonic inspection of all butt-welded joints (IGF Code 16.6.3.1)

2) 10 % radiographic or ultrasonic inspection of butt-welded joints (IGF Code 16.6.3.4)

3) Pressure test at 1.5 x design pressure to ensure gas tight integrity, not less than the expected maximum pressure at gas pipe rupture (as per IGF 16.7.3.4, and 9.8.4)

4) Pressure test at 1.5 x design pressure

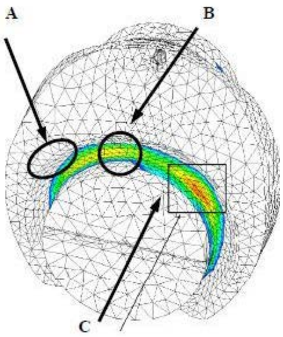
5) Leak test.

6) If inside diameter > 75 mm or wall thickness > 10 mm: 100 % radiographic or ultrasonic inspection of all butt-welded joints (IGF Code 16.6.3.1)

7) If nominal diameter > 25 mm

(*) "Flanges" limited to the final connection to the engine.

APPENDIX I -DEFINITION OF STRESS CONCENTRATION FACTORS IN CRANKSHAFTFILLETS



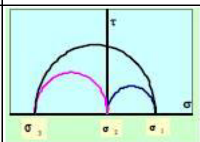
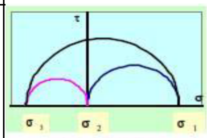
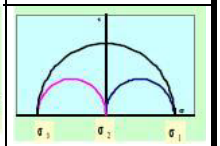
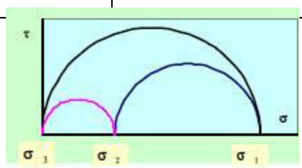
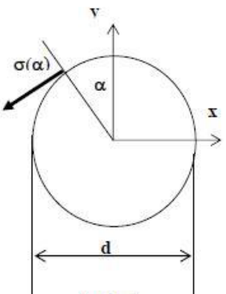
	Stress	Max $\|\sigma_3\|$	Max σ_1	
Torsional loading	Location of maximal stresses	**A**	**C**	**B**
	Typical principal stress System Mohr's circle diagram With $\sigma_2 = 0$	 $\|\sigma_3\| > \sigma_1$	 $\sigma_1 > \|\sigma_3\|$	 $\sigma_1 \approx \|\sigma_3\|$
	Equivalent stress and **S.C.F.**	$\tau_{\text{equiv}} = \dfrac{\sigma_1 - \sigma_3}{2}$ $\text{S.C.F.} = \dfrac{\tau_{equiv}}{\tau_n} \; for \alpha_T, \beta_T$		
Bending loading	Location of maximal stresses	**B**	**B**	**B**
	Typical principal Stress system Mohr's circle diagram With $\sigma_3 = 0$			$\sigma_2 \neq 0$
	Equivalent stress And **S.C.F**	$\sigma_{equiv} = \sqrt{\sigma_1^2 + \sigma_2{}^2 - \sigma_1.\sigma_2}$ $S.C.F. = \dfrac{\sigma_{equiv}}{\sigma_n} \; for \alpha_B, \beta_B, \beta_Q$		

Fig. 1.1 Definition of stress concentration factors in crankshafts fillets

APPENDIX II -STRESS CONCENTRATION FACTORS AND STRESS DISTRIBUTION AT THE EDGE OF OIL DRILLINGS



d = hole diameter

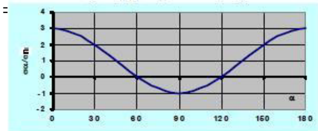
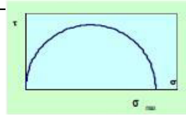
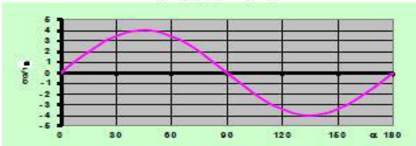
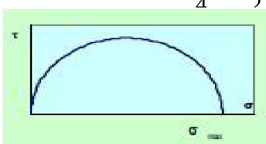
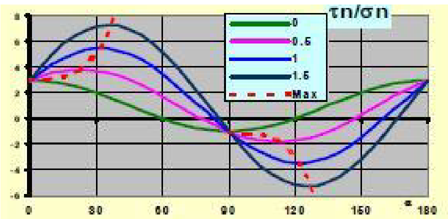
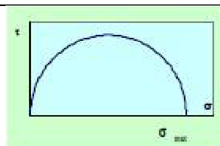
Stress type	Nominal stress tensor	Uniaxial distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha =$ 	 $\gamma_B = \sigma_{max}/\sigma_n$ for $\alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_T \tau_n \sin(2\alpha)$ 	$\gamma_T = \sigma_{max}/\tau_n$ for $\alpha = \frac{\pi}{4} + k\frac{\pi}{2}$ 
Tension+shear	$\begin{bmatrix} \sigma n & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3}\sigma_n\left\{1+2\left[\cos(2\alpha)+\frac{3}{2}\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\sin(2\alpha)\right]\right\}$ 	 $\sigma_{max} = \frac{\gamma_B}{3}\sigma_n\left[1+2\sqrt{1+\frac{9}{4}\left(\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\right)^2}\right]$ for $\alpha \frac{1}{2} tg^{-1}\left(\frac{3\gamma_T\tau_n}{2\gamma_B\sigma_n}\right)$

Fig. 2.1 Stress concentration factors and stress distribution at the edge of oil drillings

APPENDIX III - GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD

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 in this section or the alternative methods, von Misses 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.

2. Model requirements

The basic recommendations and perceptions for building the FE-model are presented in 2.1. It is obligatory for the final FE-model to fulfill the requirement in 2.3.

2.1 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 2.1.

- Drillings and holes for weight reduction have to be modeled.

- Sub-modeling may be used as far as the software requirements are fulfilled.

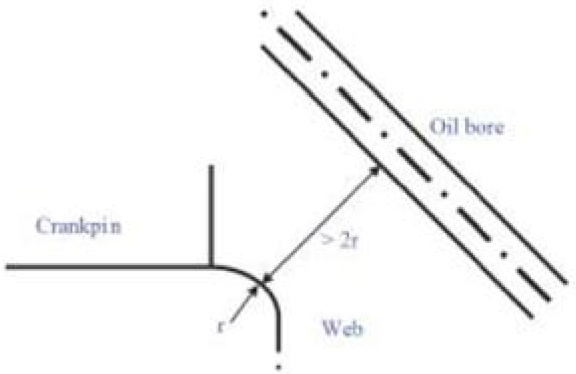


Fig. 2.1 Oil bore proximity to fillet

2.2 Material

In FE analysis, Young's Modulus (E) and Poisson's ratio (ν) are required, as strain is primarily calculated and stress is derived from strain using those material parameters.

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 $\nu = 0.3$.

2.3 Element mesh quality criteria

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

2.3.1 Principal stresses criterion

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:

$$\min(|\sigma 1|, |\sigma 2|, |\sigma 3|) < 0.03.max(|\sigma 1|, |\sigma 2|, |\sigma 3|)$$

2.3.2 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. Load cases

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

3.1 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.

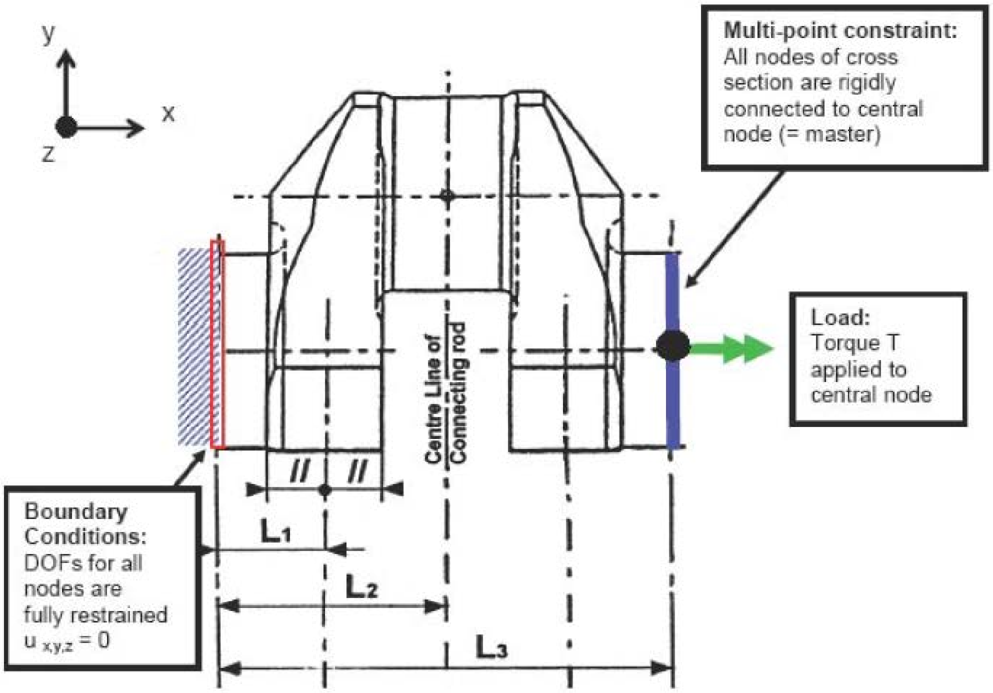


Fig. 3.1 Boundary and load conditions for the torsion load case

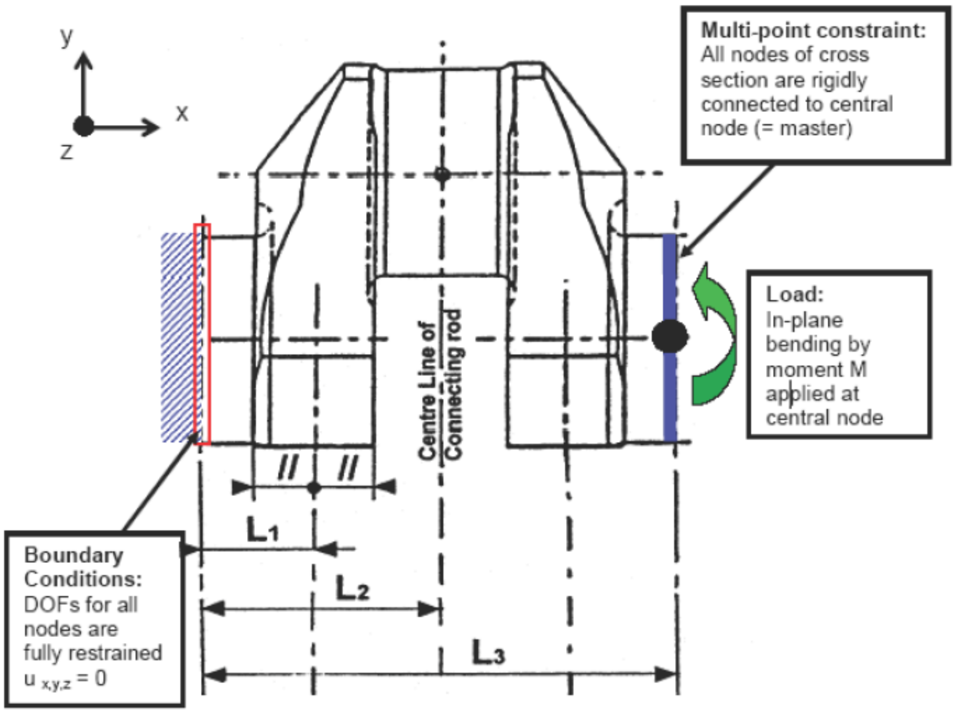


Fig. 3.2 Boundary and load conditions for the pure bending load case

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

$$\tau_{equiv} = \max\left(\frac{|\sigma 1 - \sigma 2|}{2}, \frac{|\sigma 2 - \sigma 3|}{2}, \frac{|\sigma 1 - -\sigma 3|}{2}\right)$$

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

$$\alpha_{\tau} = \frac{\tau_{equiv,\alpha}}{\tau_n}$$

$$\beta_T = \frac{\tau_{equiv,\beta}}{\tau_N}$$

where τ_N is nominal torsional stress referred to the crankpin and respectively journal as per D-2.2.2 with the torsional torque T:

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

3.2 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.

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:

$$\alpha_{\beta} = \frac{\sigma_{equiv,\alpha}}{\sigma_N}$$

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

Nominal stress σ_N is calculated as per D-2.1.2.1 with the bending moment M:

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

3.3 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 below.

3.3.1 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 3.4.

β_B = As determined in paragraph 3.2.

$$\sigma_{Q3P} = \frac{Q_{3P}}{(B.W)}$$

where Q_{3P} is the radial (shear) force in the web due to the force F_{3P} [N] applied to the centre-line of the actual connecting rod, see also figures 2.3 and 2.4.

3.3.2 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 3.3.1.

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 D-2.1.3 is then evaluated:

$$\sigma_{BG} A_{-} \pm \left| \beta_{BQ} \cdot \sigma_{BFN} \right|$$

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.

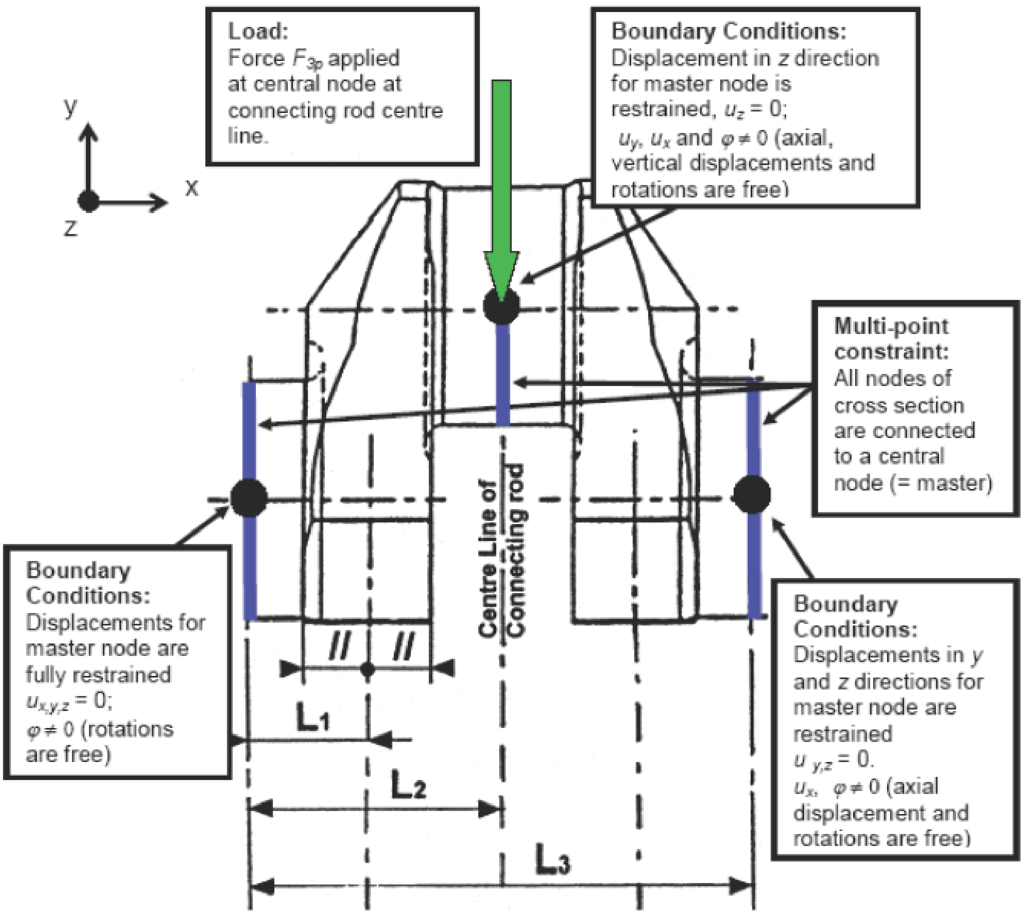


Fig. 3.3 Boundary and load conditions for the 3-point bending load case of an inline engine

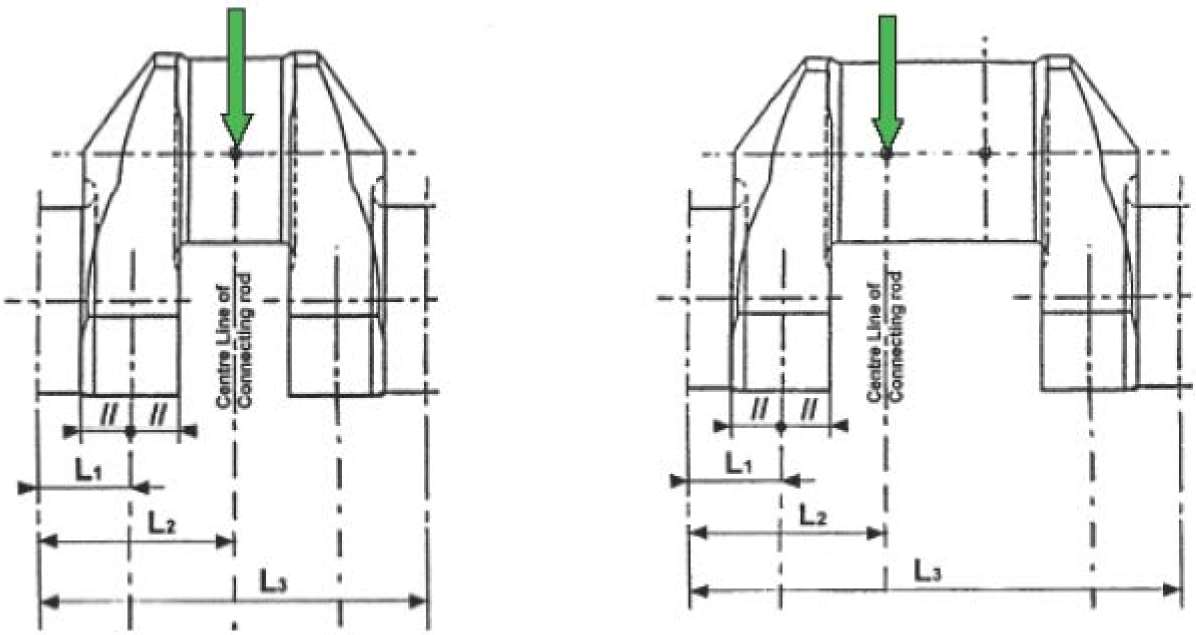


Fig. 3.4 Load applications for in-line and V-type engines

APPENDIX IV - GUIDANCE FOR EVALUATION OF FATIGUE TESTS

A. 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.

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.

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.

B. Evaluation of Test Results

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 Section 2 item D (UR M53), 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 item 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.

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. 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.

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:

$$\bar{S}_a = S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2} \right) \quad \text{when C=1}$$

$$\bar{S}_a = S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2} \right) \quad \text{when C=2}$$

The standard deviation can be found by

$$s = 1.62 \cdot d \cdot \left(\frac{F \cdot \bar{B} - A^2}{F^2} + 0.029 \right)$$

where:

S_{a0} is the lowest stress level for the less frequent occurrence

d is the stress increment

$F = \Sigma fi$

$A = \Sigma i \cdot fi$

$B = \Sigma\, i^2 \cdot fi$

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

$$\frac{B \cdot F - A^2}{F^2} > 0{,}3$$

and

$$0{,}5 \cdot s < d < 1{,}5 \cdot s$$

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.

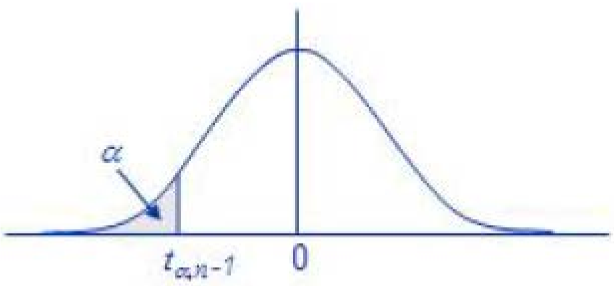
5. Confidence interval for mean fatigue limit

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) which is a distribution symmetric around the average.

If S_a is the empirical mean and s is the empirical standard deviation over a series of n samples, in which the variable values are normally distributed with an unknown sample mean and unknown variance, the $(1 - \alpha) \cdot 100\%$ confidence interval for the mean is:

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



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 shows the *t*value for $(1 - \alpha)$ 100 confidence interval for the sample mean.

Figure 2.1. Student's t-distribution

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

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

which is the mean fatigue limit (population value) to be used to obtain the reduced fatigue limit where the limits for the probability of failure are taken into consideration.

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.

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:

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

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

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

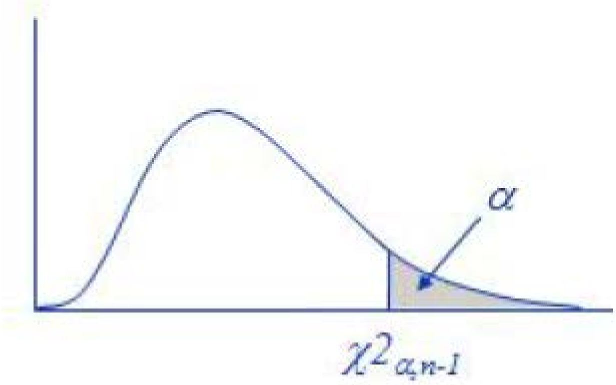
This standard deviation (population value) is to be used to obtain the fatigue limit, where the limits for the probability of failure are taken into consideration.

C. 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 3.1.

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 utilising the finite element method.

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



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 chisquare for (1 - α) · 100% confidence interval for the variance.

Figure 2.2. Chi-square distribution

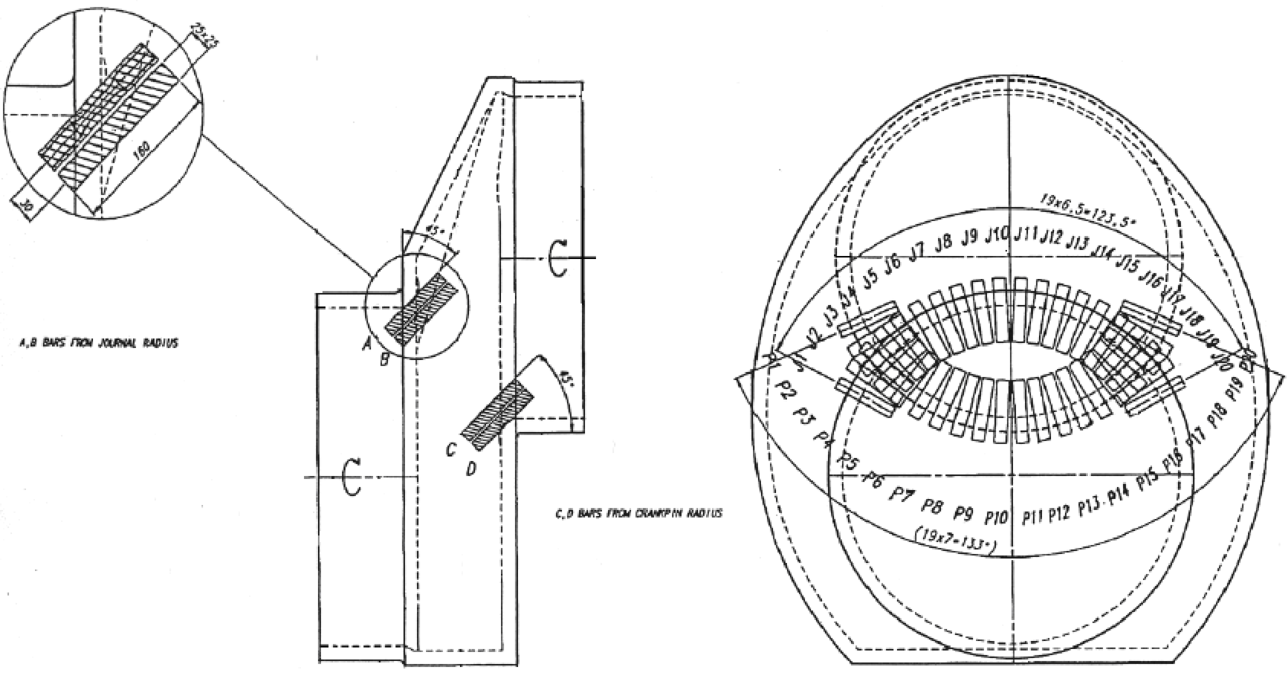


Figure 3.1. Specimen locations in a crank throw

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 3.1.

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 3.1.

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.

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 fullsize 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. 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.

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 recognised 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.

D. Full size testing

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 Appendix III (section 3.1 to 3.3).

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

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 4.1 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.

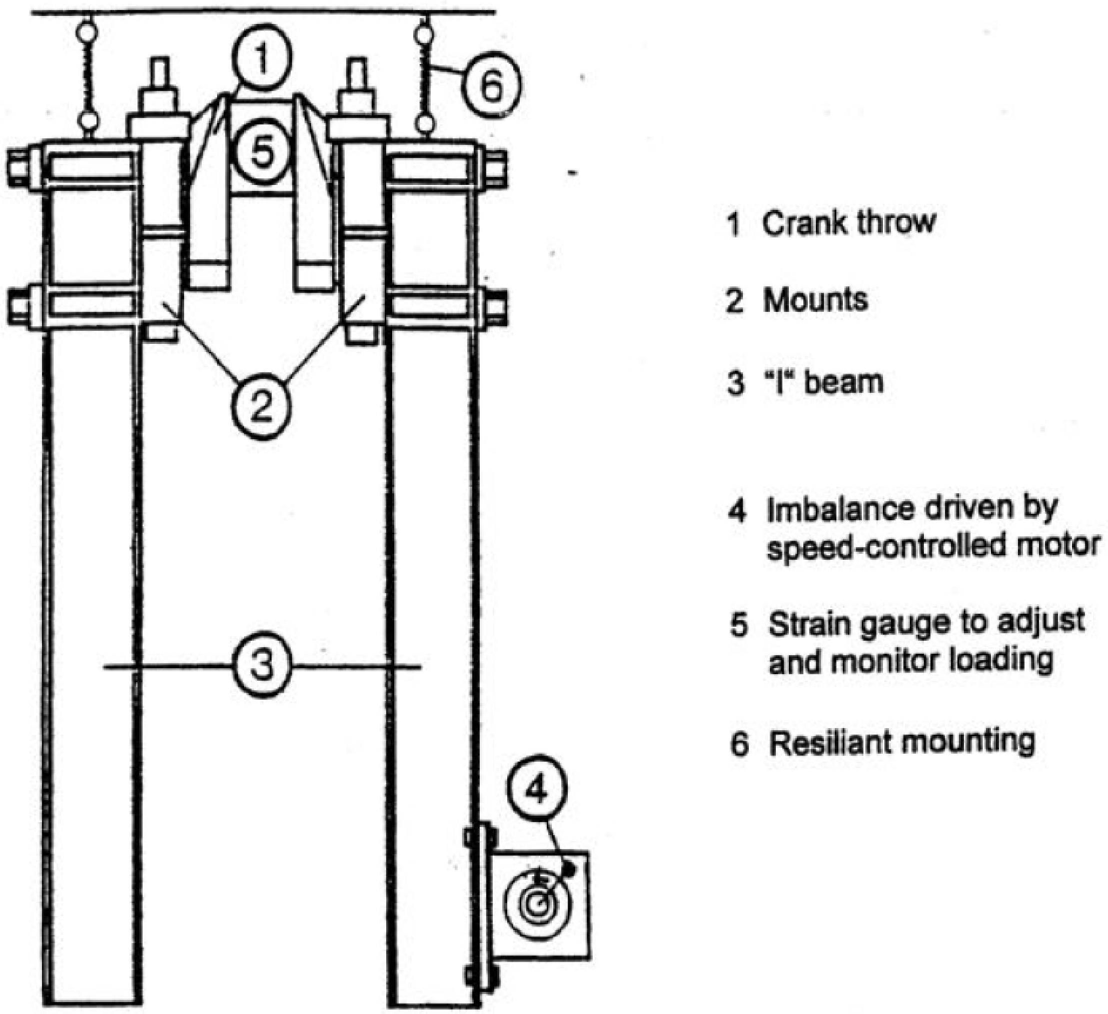


Figure 4.1. An example of testing arrangement of the resonance tester for bending loading

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 4.2. 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.

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.

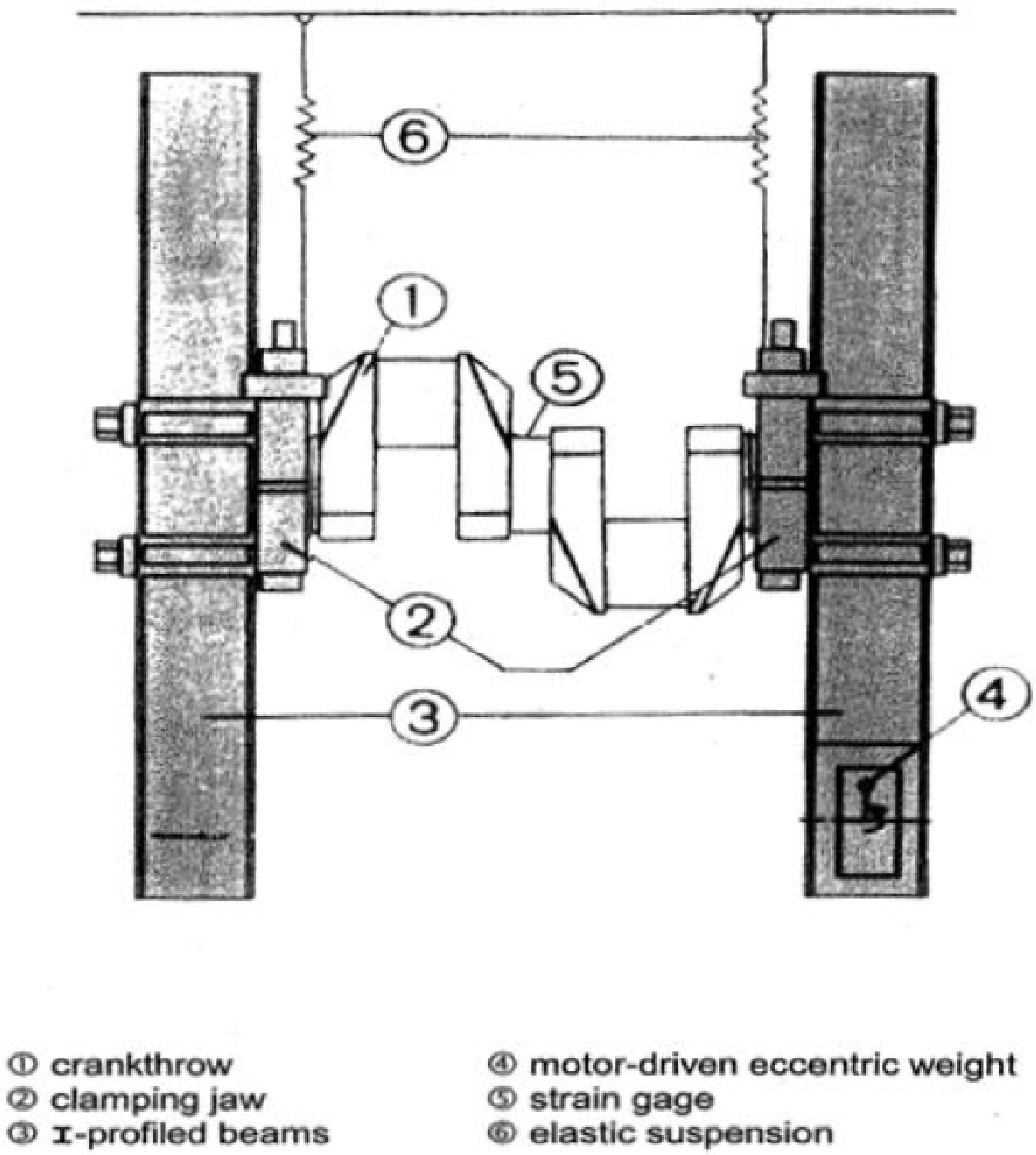


Figure 4.2. An example of testing arrangement of the resonance tester for torsion loading with double crank throw section

3. Use of results and crankshaft acceptability

In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability, see Section 3 item D.7, the Gough-Pollard approach and the maximum principal equivalent stress formulation can be applied for the following cases:

At the crankpin fillet:

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

where:

σ_{DWCT} fatigue strength by bending testing

τ_{DWCT} fatigue strength by torsion testing

For other parameters see Section 3 items D.2.1.3, D.2.2.3 and D.4.

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

At the journal fillet:

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

where:

σ_{DWJT} fatigue strength by bending testing

τ_{DWJT} fatigue strength by torsion testing

For other parameters see Section 3 items D.2.1.3, D.2.2.3 and D.4.

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.

E. 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 item C.
- **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.

APPENDIX V -GUIDANCE FOR CALCULATION OF SURFACE TREATED FILLETS AND OIL BORE OUTLETS

A. Introduction

This appendix deals with surface treated fillets and oil bore outlets. The various treatments are explained and some empirical formulae are given for calculation purposes. 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.

B. 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.

1. Surface treatment methods

The following list covers possible treatment methods and how they influence the properties that are decisive for the fatigue strength.

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.

Table 2.1. 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

C. 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 section F.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 stres concentration as well as to a depth of about 1.2 times the depth of the treatment. The following figure 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.

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.

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 Appendix III. 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 Section 2 item D.3 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 $\alpha_{B\text{-local}}$ and $\alpha_{T\text{-local}}$ have different shapes due to the different stress gradients.

The SCFs α_B and α_T are valid at the surface. The local $\alpha_{B\text{-local}}$ and $\alpha_{T\text{-local}}$ 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 3.2 for bending and respectively for torsion in Equation 3.2 and Figure 3.3.

$$\alpha_{B-local} = (\alpha_B - 1)\cdot e^{\frac{-2\cdot t}{R_H}} + 1 - \left(\frac{2\cdot t}{\sqrt{W^2 + S^2}}\right)^{\frac{0.6}{\sqrt{\alpha_B}}}$$

(3.1)

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

(3.2)

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.

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 Appendix III. 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 Appendix III item 3.1 – and the bending moment, with four-point bending as in Appendix III item 3.2.

If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from Section 2 item D.3 if within its applicability range. Bending and torsional stresses at the point of peak stresses are combined as in Section 2 item D.5.

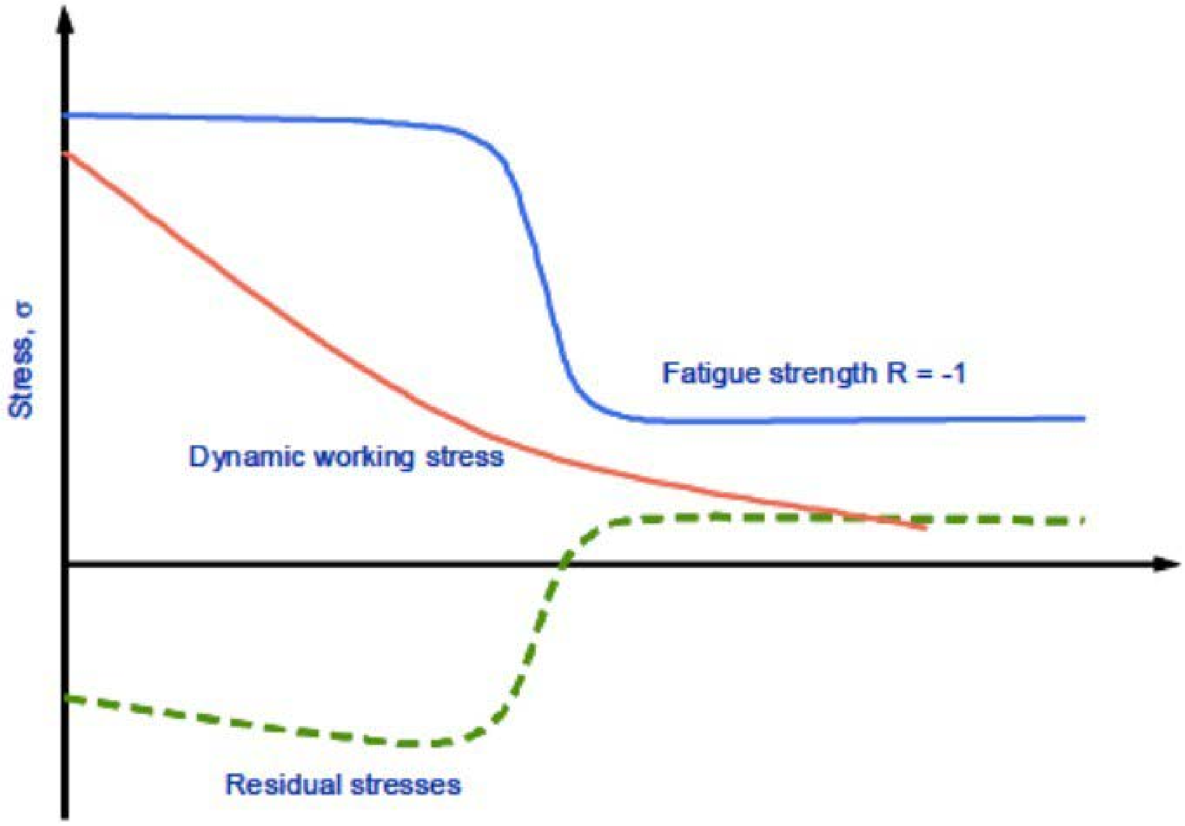


Figure 3.1. Stresses as functions of depth, general principles

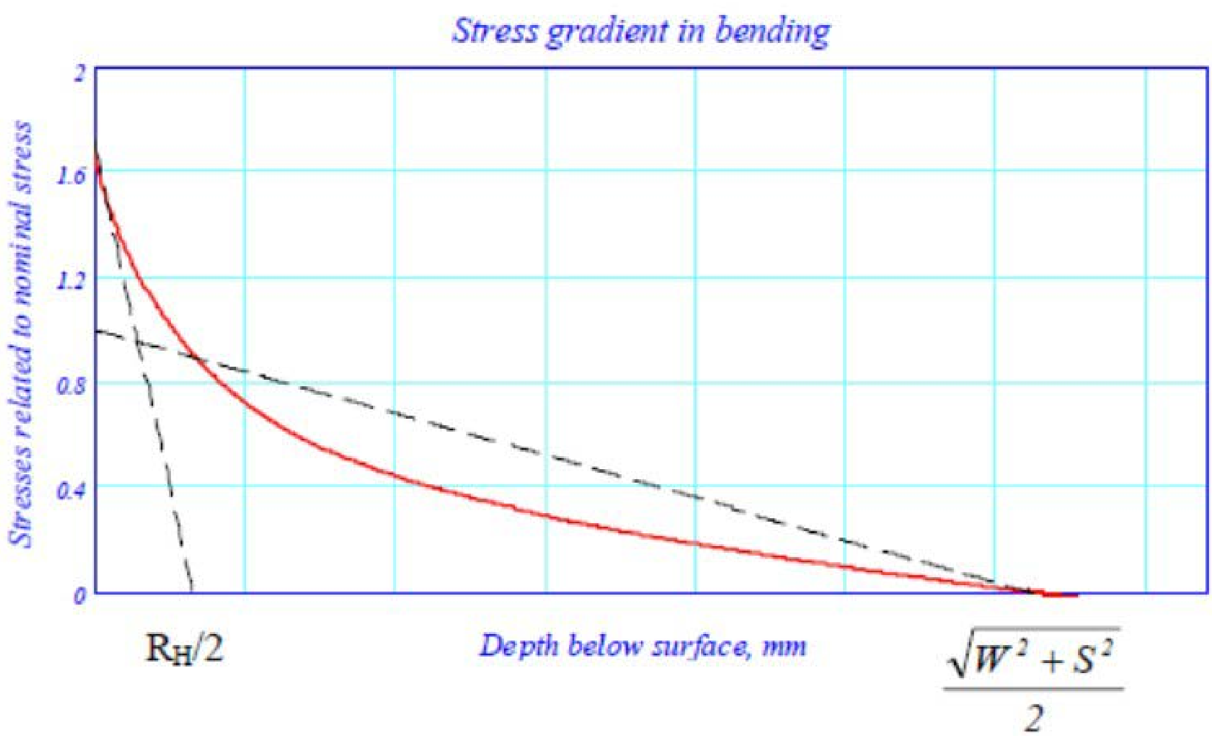


Figure 3.2. Bending SCF in the crankpin fillet as a function of depth. The corresponding SCF for the journal fillet can be found by replacing R_H with R_G

Figure 3.4 indicates a local drop of the hardness in the transition zone between a hard and soft material. Whether this drop occurs depends also on the tempering temperature after quenching in the QT process.

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 3.4, 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 (1-2*tH/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 $\gamma_{B\text{-}local}$ and $\gamma_{T\text{-}local}$ have different shapes, because of the different stress gradients.

The stress concentration factors γ_B and γ_T are valid at the surface. The local SCFs $\gamma_{B\text{-}local}$ and $\gamma_{T\text{-}local}$ 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*:

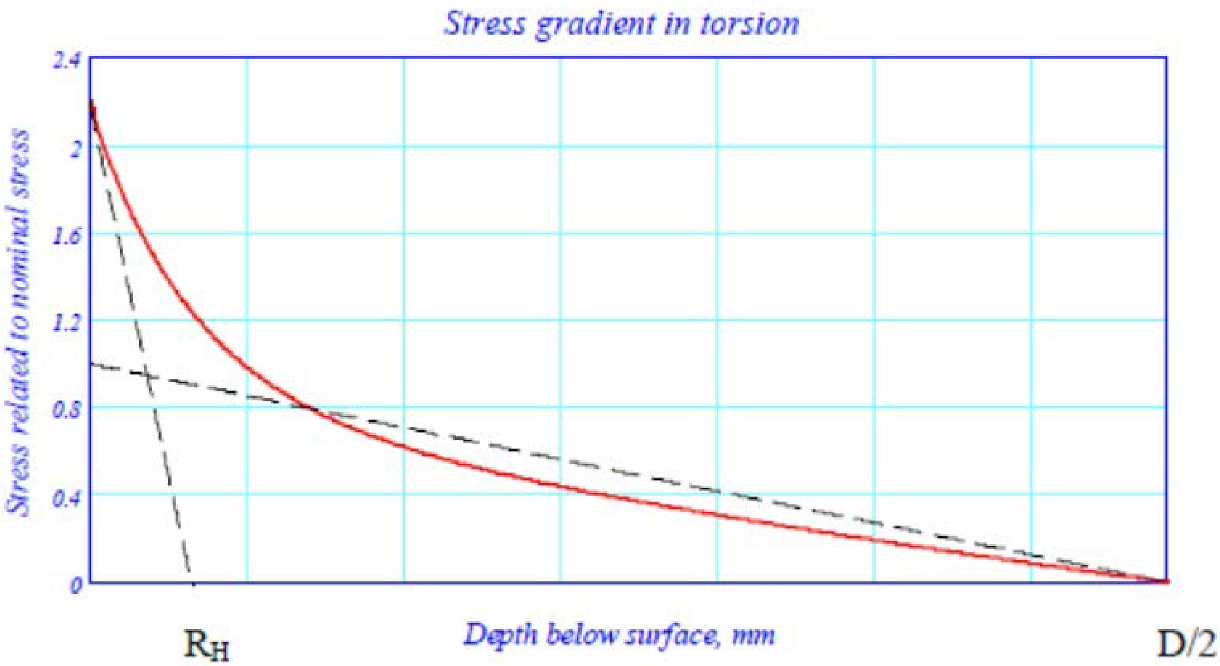


Figure 3.3. Torsional SCF in the crankpin fillet as a function of depth. The corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G

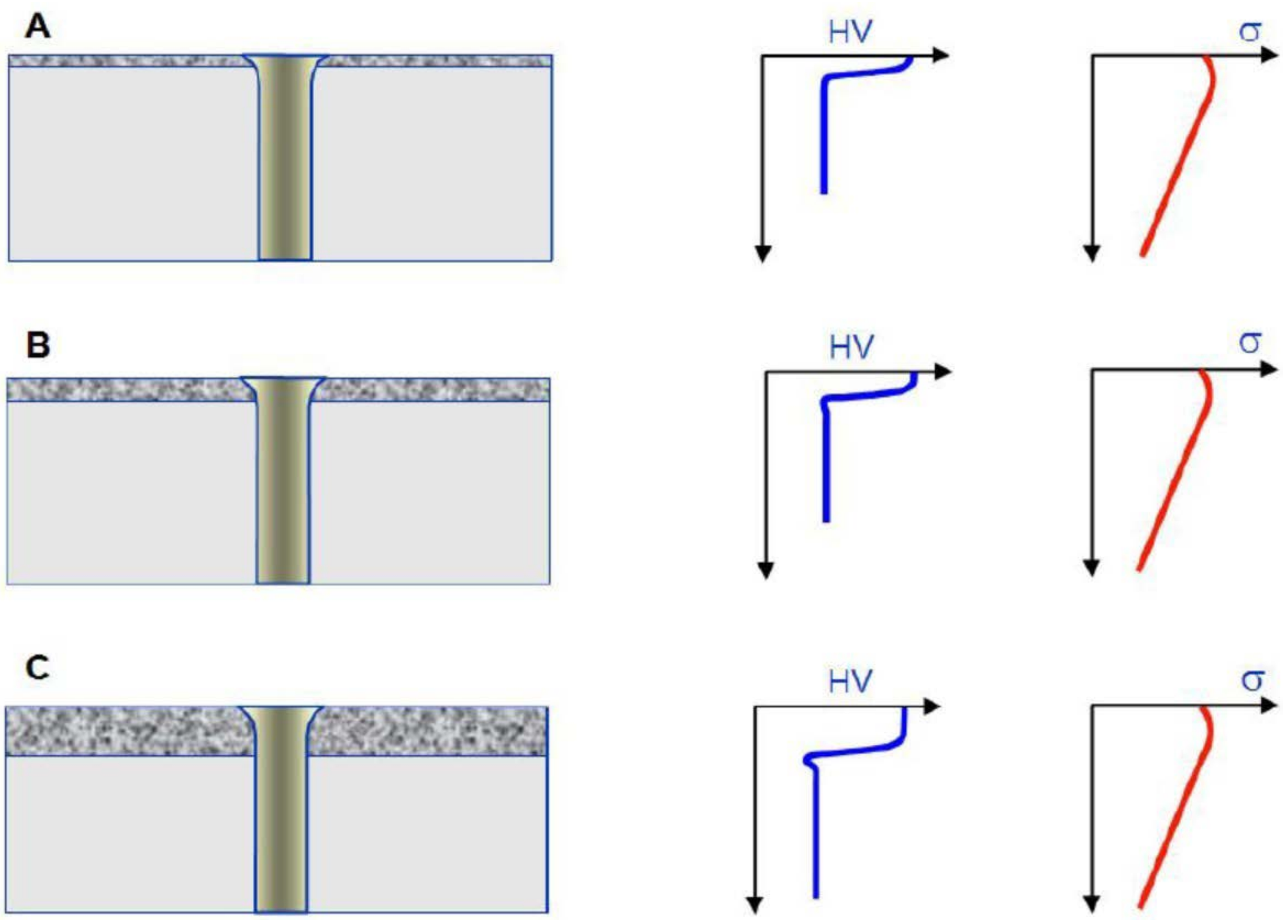


Figure 3.4. Stresses and hardness in induction hardened oil holes

$$\gamma_{B-local} = (\gamma_B - 1) \cdot e^{\frac{-4 \cdot t}{D_0}} + 1 \quad \textbf{(3.3)}$$

$$\gamma_{B-local} = (\gamma_T - 1) \cdot e^{\frac{-2 \cdot t}{D_0}} + 1 \quad \textbf{(3.4)}$$

3. Acceptability criteria

Acceptance of crankshafts is based on fatigue considerations; Section 2 item D (UR M53) compares the equivalent alternating stress and the fatigue strength ratio to an acceptability factor of $Q \geq 1.15$ 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.

D. 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.

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 4.2.

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 4.1):
    - Constant hardness from the surface to the transition zone

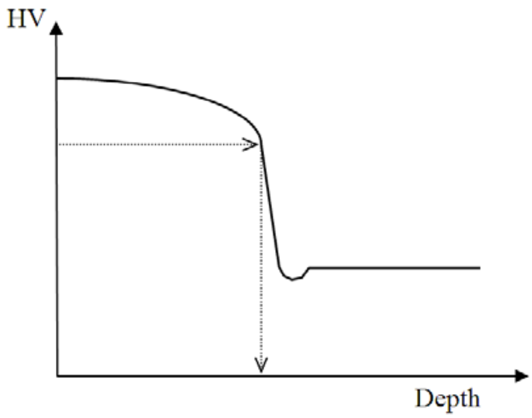


Figure 4.1. Typical hardness as a function of depth. The arrows indicate the defined hardening depth. Note the indicated potential hardness drop at the transition to the core. This can be a weak point as local strength may be reduced and tensile residual stresses may occur.

-
    - 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.

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 Appendix IV. 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:

$$\sigma_{Fsurface} = 400 + 0.5 \cdot (HV - 400) \quad [MPa] \tag{4.1}$$

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 UR M53.6.

For journal and respectively to crankpin fillet applies:

$$\sigma_{Ftransition,cpin} = \pm K \cdot (0.42 \cdot \sigma_B + 39{,}3) \cdot \left[0.264 + 1.073 \cdot \gamma^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{X}}\right] \tag{4.2}$$

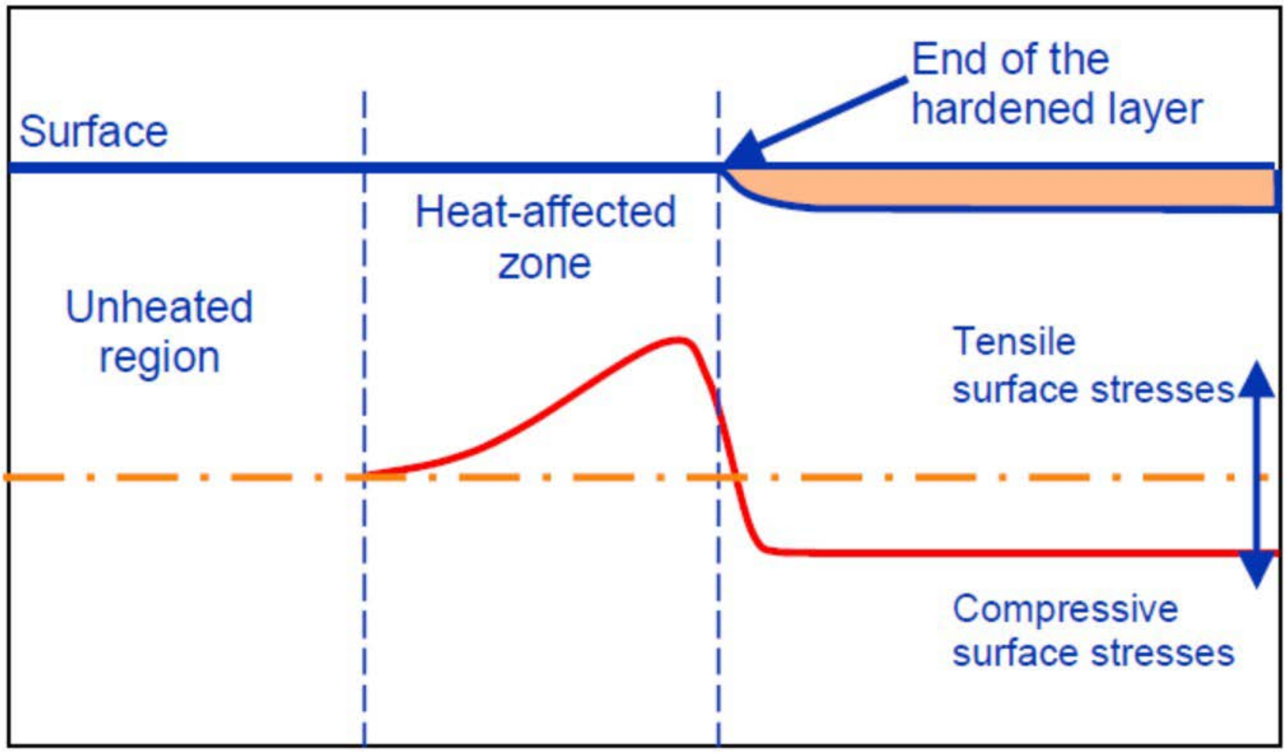


Figure 4.2. Residual stresses along the surface of a pin and fillet

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 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 4.2 – the influence of the tensile residual stresses can be considered by subtracting a certain percentage, in accordance with Table 4.1, from the value determined by the above formula.

E. 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:

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

where:

t	=	The local depth
HV(t)	=	Hardness at depth *t*
HVcore	=	Core hardness (minimum)
HVsurface	=	Surface hardness (minimum)
tN	=	Nitriding depth as defined above (minimum)

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 Appendix IV.

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:

$$\sigma_{Fsurface} = 450 \text{ MPa} \quad \textbf{(5.2)}$$

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 UR M53.6. For crankpin and respectively to journal applies:

$$\sigma_{\text{Ftransition,cin}} = \pm K \cdot (0.42 \cdot \sigma_B + 39{,}3) \cdot \left[0.264 + 1.073 \cdot \gamma^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{\chi}}\right]$$

(5.3)

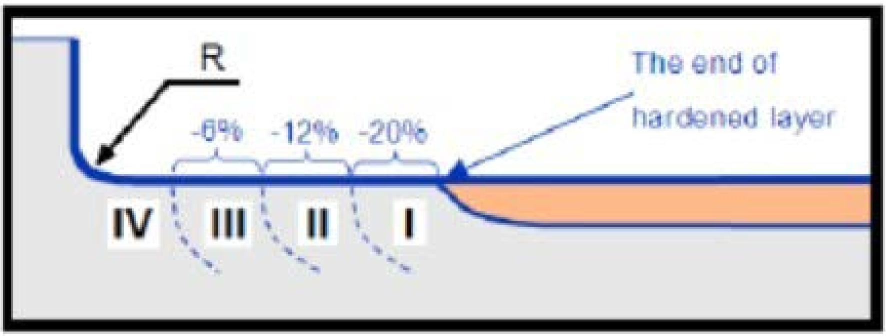
where:

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.



I.	0 to 1.0 of the max. hardening depth:	20%
II.	1.0 to 2.0 of the max. hardening depth:	12%
III.	2.0 to 3.0 of the max. hardening depth:	6%
IV.	3.0 or more of the max. hardening depth:	0%

Table 4.1. The influence of tensile residual stresses at a given distance from the end of the hardening towards the fillet

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=1.2t_N.

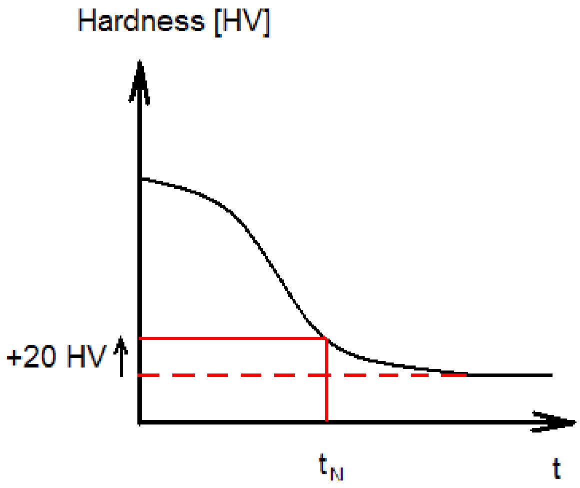


Figure 5.1. Sketch of the location for the artificial transition point in the depth direction

F. 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 Appendix IV.

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.

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 stres 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 6.1 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.

1.1 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

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.

2.1. 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

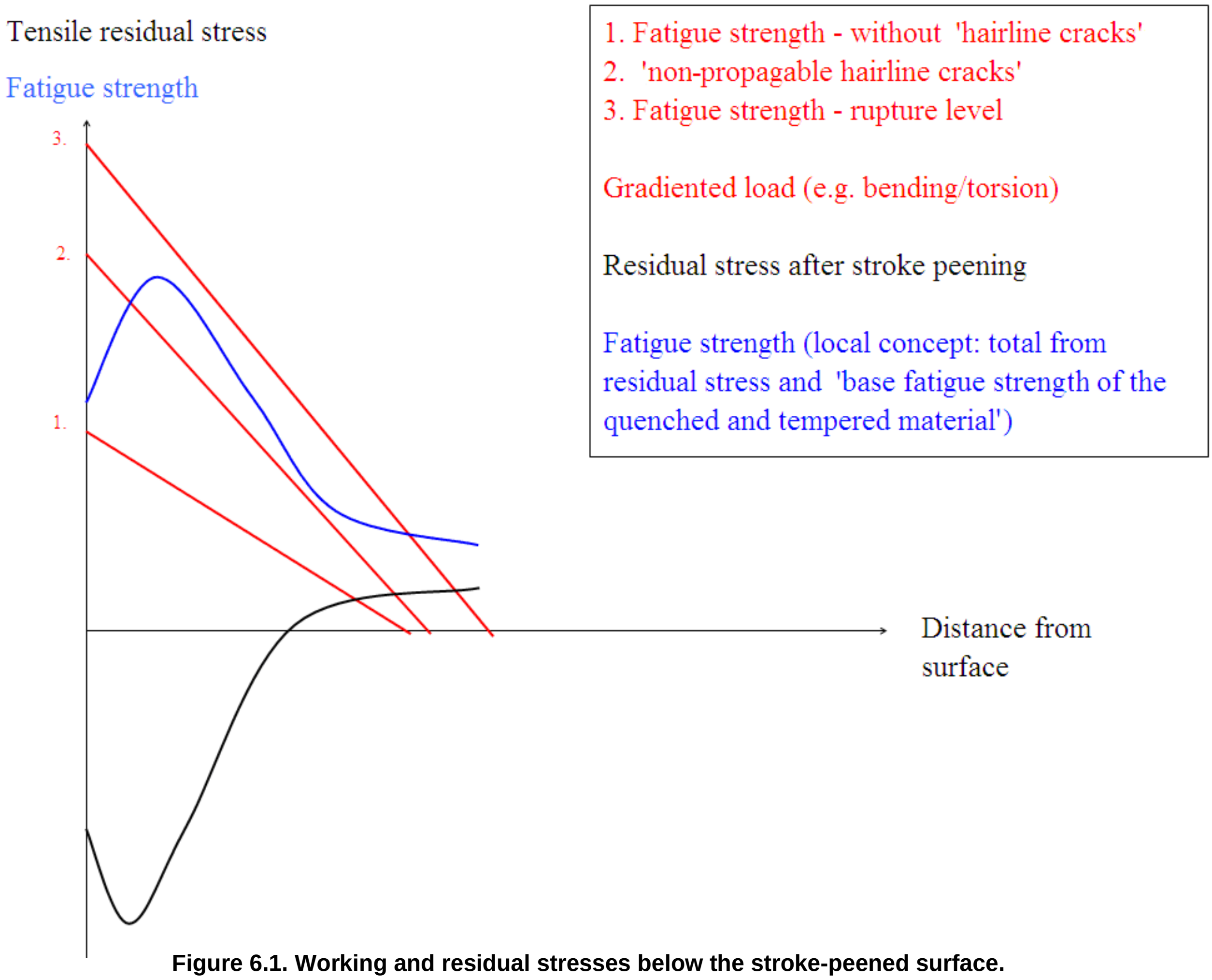


Figure 6.1. Working and residual stresses below the stroke-peened surface. Straight lines 1…3 represent different possible load stress gradients.

APPENDIX VI- GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE OIL BORE OUTLETS OF CRANKSHAFTS THROUGH UTILISATION OF THE FINITE ELEMENT METHOD

A. 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 UR M53, 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.

B. Model Requirements

The basic recommendations and assumptions for building of the FE-model are presented in Subsection 1. The final FE-model must meet one of the criteria in Subsection 3.

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.

2. Material

UR M53 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. 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.

3.1. 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:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0.03 \cdot \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

3.2. 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.

C. Load cases and assessment of stress

For substitution of the analytically determined SCF in UR M53, calculation shall be performed
for the following load cases.

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.

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:

$$\gamma_T = \cdot\frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\tau_N}$$

where the nominal torsion stress τ_N referred to the crankpin is evaluated per Section 2 item D.2.2.2 with torque T:

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

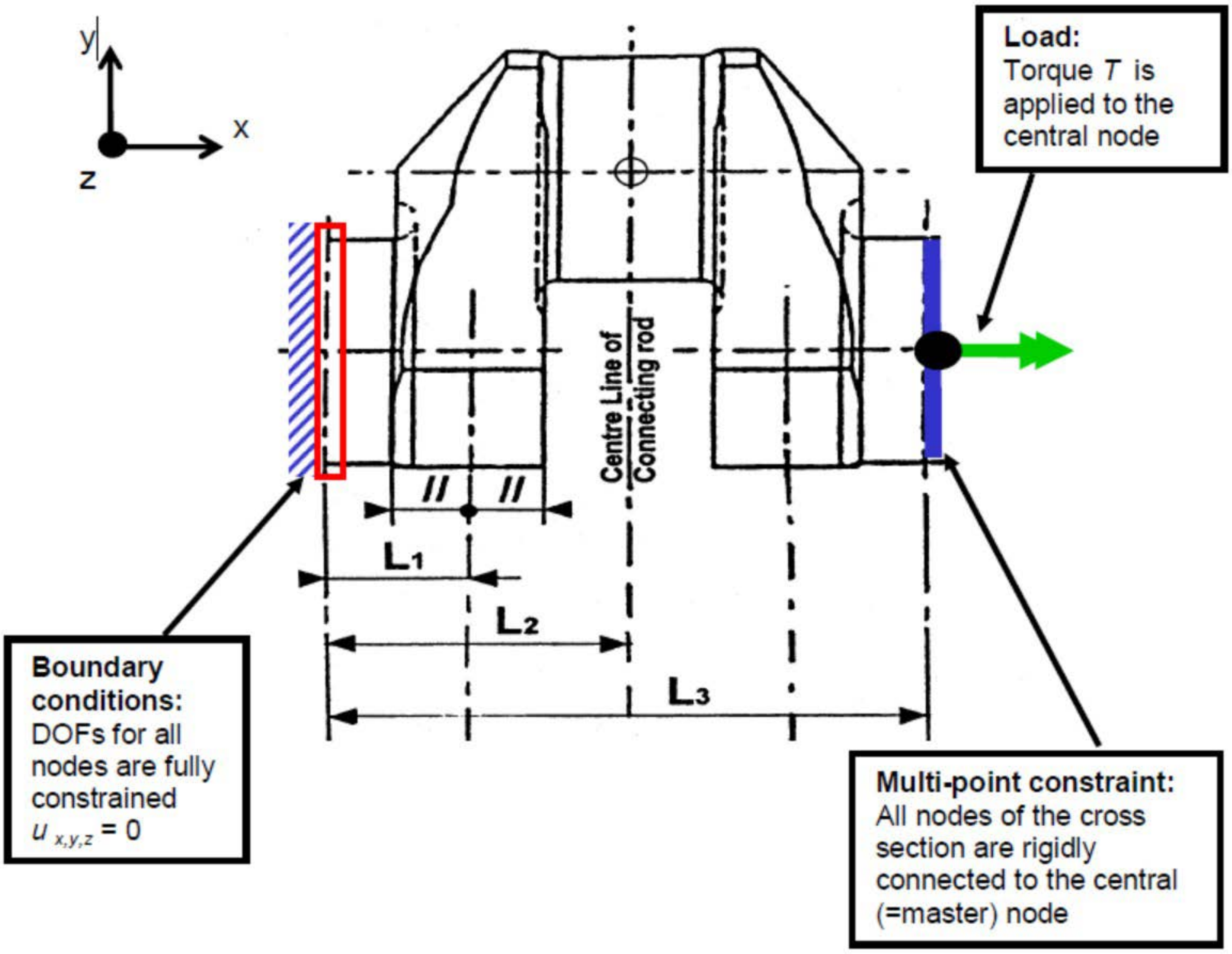


Figure 3.1 Boundary and load conditions for the torsion load case

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.

For all nodes in the oil bore outlet, principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_B = \frac{\max\left(|\sigma_1|, |\sigma_2|, |\sigma_3|\right)}{\sigma_N}$$

where the nominal bending stress σ_N referred to the crankpin is calculated per Section 2 item D.2.1.2.2 with bending moment M:

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

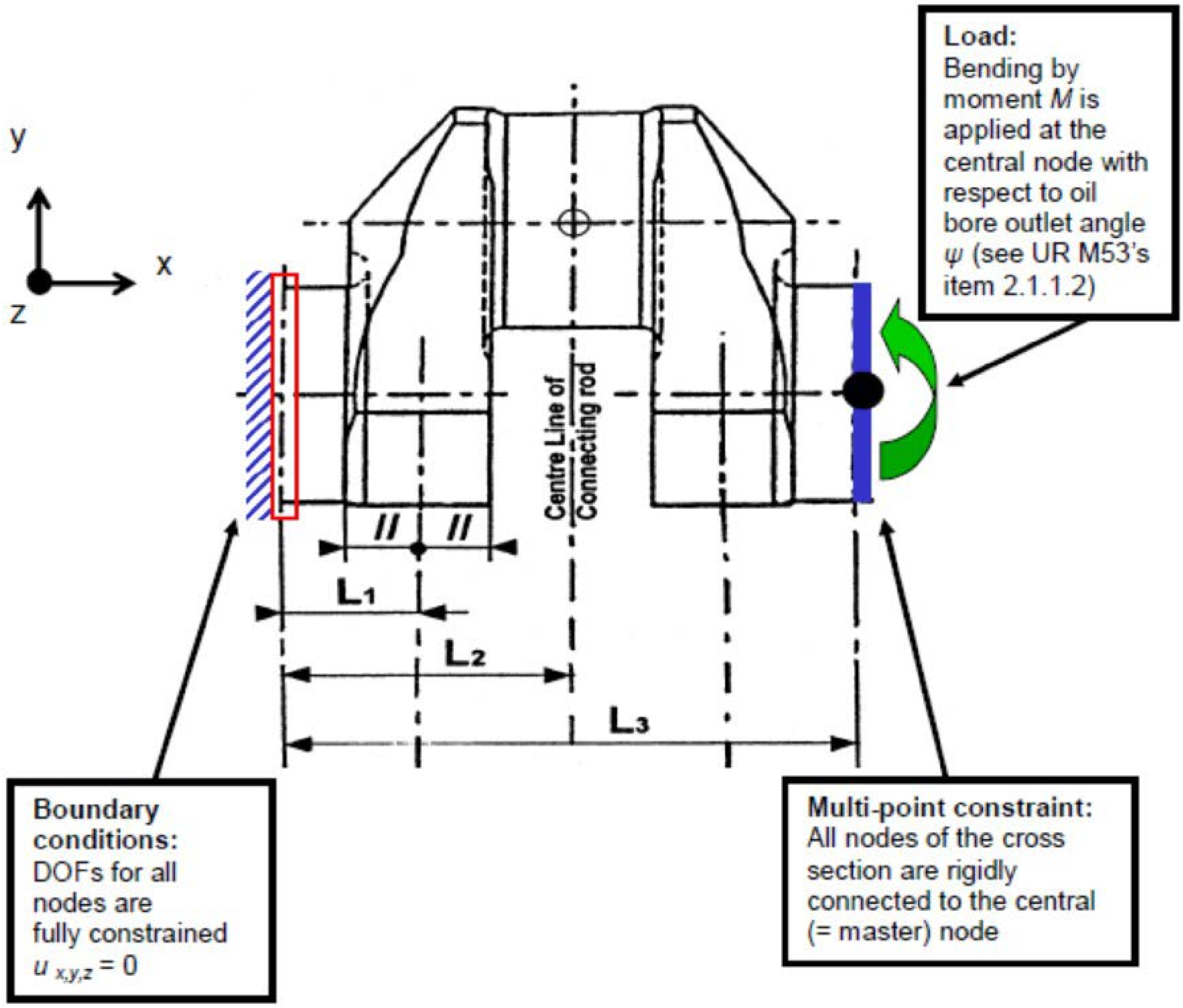


Figure 3.2. Boundary and load conditions for the pure bending load case

SECTION 4 A

THERMAL TURBOMACHINERY/GAS TURBINES

A. General

1. Application

1.1 Gas turbines on naval ships for the applications

- Essential main propulsion
- Non-essential propulsion
- Driving of auxiliaries

as defined in 2. are subject to these Rules. According to the special purpose of the type of naval ship, they are to be designed, constructed, tested, certified and installed on board in accordance with the requirements of this Section.

1.2 Gas turbines installed on **TL** classed ships and not fulfilling purposes as described under 1.1 are not subjected to the rules of this Section. They have nevertheless to fulfil all requirements related to passive safety (fire protection, containment, safety devices for rendering to a safe state after malfunction or failure of components).

1.3 Gas turbines with a power less than 500 kW are not required to comply with the rules of this Section, but have to be designed, constructed and equipped in accordance to good practice for marine applications.

1.4 Acceptance is based on test bed results for a prototype (manufacturer´s documentation) and successful performance test after installation in presence of a surveyor. Further documentation may be requested for appraisal purposes upon agreement between **TL**, manufacturer and shipyard.

1.5 Subject to approval is the complete gas turbine plant including foundation, piping, fuel, lubrication, cooling as well as safety, control, monitoring and alarm systems. The rules of this Section, especially for plants as described under 1.1, apply for the gas turbine unit itself, as well as for the complete installation. Typically the gas turbine unit with manifold integrated feeding systems, internal clutches, combustion chambers, etc. is subject to type approval, the installation, connection and foundation of the complete manifold on the ship´s side are project specific.

So far the gas turbine type/version has been subjected to a type approval, the relevant drawings are to be submitted, so far project related and therefore not covered up by the type approval (see 3.3 and 3.4).

2. Definitions

The following definitions are to be applied:

2.1 Essential main propulsion

The gas turbine serves as exclusive drive for the propulsion elements and is essential for the safety and manoeuvrability of the ship.

2.2 Non-essential propulsion

Other propulsion drives, e.g. internal combustion engines or other gas turbines, secure the safety and manoeuvrability of the ship and the gas turbine in question is only temporarily used as booster to achieve maximum ahead speed.

2.3 Driving of auxiliaries

The gas turbine is not directly involved in the propulsion of the ship, but drives a generator or other auxiliary units of the machinery systems.

2.4 Maximum permissible power (type related)

The maximum permissible power is the maximum power of the applied type of gas turbine in the actual upgrade version, independent of project dedicated application. This is the base for design calculations and type approval procedure.

2.5 Maximum continuous rating (project related)

Maximum continuous rating means 100 % gas turbine power (MCR condition). The gas turbine is to be limited to the specified maximum continuous rating after performance test in the manufacturer’s facilities and/or onboard after installation.

This rating is the relevant power for dimensioning of the driven devices, such as gear box, shaftings, etc. for a specific application.

3. Documents for Approval

3.1 General

Design drawings of gas turbines are to be submitted to **TL** for approval. The drawings shall contain all the details necessary to carry out an examination in accordance with the following requirements.

3.2 Gas turbine unit (manifold)

The following drawings are to be supplied by the gas turbine manufacturer within the type approval procedure:

- General arrangement drawing of complete manifold with manufacturer´s term definitions

- Assembly and sectional drawings

- Rotor assembly drawings including general dimensions and materials

- Detailed drawings of rotating parts (turbine and compressor, discs, shaftings, blades for gas generator and power turbine), including dimensions, materials and treatment (coating, heat treatment, etc.)

- Casings

- Containment calculations for the failure mode "blade loss", so far available. This may be replaced by documented experimental data for a prototype of the gas turbine. Instead also experimental verification of integrity at higher speeds than 115 % n_{rated} may be accepted (for existing and proven designs).

- Bearings, thrust bearing arrangement

- Sealing system

- Combustion chambers including burners with dimensions and materials

- Foundation and fastening (frames)

- Material specifications including their mechanical and chemical properties for the rotating parts, combustor and casing.

3.3 Gas turbine systems (partly manifold integrated, partly project specific)

The majority of the following documents are to be adapted to the needs of the specific project and may be not part of the type approval hold by the gas turbine manufacturer.

- General arrangement drawing of complete system with block indication and technical nomenclature (Schematic indication of gas turbine unit with all connections to the feeding / controlling systems)

- Arrangement within engine room with fire and safety details (location of doors, fans, fire dampers, materials and insulation, location of electrical and control panels)

- Documentation of starting system with arrangement drawings, capacity data, a proof of capability to perform the required start attempts, starting logic and set points

- Drawings of the fuel system including piping, instrumentation diagrams, purification and filtering equipment, specification of fuel as required by the turbine manufacturer

- Specification of the fuel metering valve including material specification, flow rates, control parameter and instrumentation, calibrating process

- Shielding of fuel oil system piping

- Lubrication system in form of schematic piping diagrams, including pumps, valves, tanks, indicators, filtering equipment, etc.

- Drawings shall indicate maximum and minimum pressure and temperature in the system (idling and MCR values)

- Type of recommended lubricating oil and approved list of lubricants

- Maximum permissible amount of water in the oil

- Control oil system

- Air-intake and exhaust gas system (including filtration and silencer system)

- Bleed / cooling / seal air system

- Heat balance and design of the cooling air system to be documented on a separate drawing indicating the maximal/minimal flow rates, the corresponding pressures of cooling air and design temperatures at rated power / speed of the power turbine.

- Water wash system - blade cleaning system

- Appended manifolds including gears, clutches, couplings

- Fire protection system. Complete documentation of fire safety procedures e.g. type and capacity of fire extinguishing medium, location and specification of fire (flame) detection, specification of control system action sequences

- Foundation and fastening (frames to the ship´s structures, resilient mountings if applicable)

- Electrical installations, instrumentation, alarms, safety system

- Governor arrangement/system

- Failure Mode and Effect Analysis (FMEA) for the control system (extent to be agreed upon individually between manufacturer and **TL**)

3.4 Design data for a specific project

For each specific project related application of gas turbines the following basic design data has to be specified:

- Maximum continuous power, corresponding speed for gas generator and power turbine (project related)

- Maximum permissible rates (limitation for project, but in any case lower than maximum permissible power as per type approval)

- Rated power turbine inlet temperature (and limit)

- Rated compressor discharge temperature (and limit) permissible (tolerable) combustor outlet temperature deviation

- Maximum intake air temperature at which rated power can be achieved (to be not less than 15 °C)

- Balancing data sheet (minimum requirements)

- Scheme with moment of inertias and stiffnesses, so far applicable, of rotating parts

- Schedule and description of the intended performance test (FAT = Factory Acceptance Test) with list of parameters to be registered and documented.

3.5 Maintenance

A plan is to be submitted to **TL** specifying the time intervals and extent of works for planned maintenance. In case of Continuous Machinery Survey agreement the maintenance scheme shall be adapted accordingly.

4. References to Further Rules

4.1 TL Rules

The latest issue of the following **TL** Rules have to be observed:

- Materials

- Welding

4.2 Other rules and regulations

- ISO 21940-11: Mechanical vibration – Rotor Balancing – Part 11: Procedures and tolerances for rotors with rigid behaviour

- ISO 2314: Gas turbines - Acceptance tests

- DIN 86009: Exhaust gas lines on ships; steel tubes

5. Certification

If the requirements of this Section are met and the type tests defined in F.2.3 were carried out successfully, a Type Approval Certificate will be issued. For further gas turbines of the same type and applications mentioned under A.1. work's tests (FAT) as defined in F.2.4 shall be carried out and a Certificate will be issued.

B. Materials

1. Gas turbine materials have to fulfil the requirements specified by the operating conditions for the individual engine component. In the choice of materials the effects of creep, thermal fatigue, oxidation and corrosion, to which the different components of the gas turbine unit are exposed, have to be considered.

2. For materials the information about suitability of mechanical properties such as chemical composition, yield strength, elongation, fatigue properties as well as the applied heat treatment is required. Where composite material is used, their manufacturing procedures are to be specified.

For the turbine blades only corrosion and heat-resistant materials are to be applied.

3. The production procedures including welding are to be qualified according to a recognized standard or the relevant Rules of **TL**

4. For welding seams full details as filler metal particulars, heat treatment after welding and NDT for high stress seams are required and welding procedure specifications shall be included in the submitted, relevant drawings.

5. For the air pipes steel or another equivalent material has to be used. Flexible connections / hoses have to be non-flammable.

6. As material for exhaust gas pipes heat resistant steel has to be used. The choice of the steel type has to consider the operation temperatures and the corrosion resistance, see also DIN 86009.

7. Pipes and fittings are to be manufactured from stainless steel.

C. Design and Construction Principles

1. General

The gas turbine design shall be suitable for marine application and enable the full manoeuvrability of the ship.

Gas turbines shall be designed to permit fast start-up from cold conditions. Further possibilities are to be provided enabling manual re-start despite control system constraints after acknowledgement of the cause of tripping or start failure.

Gas turbines shall be designed for minimum times between overhauls (TBO) of > 1000 running hours. So far classified as unique main propulsion unit, the gas turbine is expected to operate without major interruptions. The machinery class intervals, if relevant, shall be adapted in such case to the required planned maintenance scheme of the propulsion system.

2. Blades

A blade strength calculation for the maximum permissible output/speed shall be submitted by means of finite element analysis (FEM) or other sound engineering methods for review. The following loads shall be taken into account: centrifugal and axial forces, gas pressure as well as thermal loads. The calculations may contain the design equivalent full load operation hours or (if applicable) the assumed load profile. Cleaning equipment is to be provided during gas turbine operation for removal of blade deposits from compressor and turbine.

3. Rotor Assembly

Each shaft as well as the complete rotating assembly of compressor and turbine has to be individually dynamically balanced in accordance with the approved quality control procedure. The balancing specification and the results of the balancing procedure are subject of the documentation in accordance to A.3.4. The requirements as set out in ISO 21940-11 apply, specifically class G2.5 or equivalent standards.

4. Casing

4.1 A blade or impeller loss may not result in casing penetration and consequential loss of other components, injury or other hazardous conditions for the ship.

Casing integrity has to be maintained in case of blade loss at the most critical speed but not higher than 115 % of the rated speed, which is to be demonstrated by containment calculations or other methods.

4.2 The casing shall have sufficient openings to enable boroscope inspection of the combustor as well as for the compressor´s and turbine´s individual stages.

5. Burners and Combustors

5.1 Fuel nozzles are to be removable without disassembly of the combustor system. Regarding burners the following parameters are to be observed:

- Maximum/minimum temperature and supply pressure of fuel
- Maximum mass flow rate of the fuel and the expected air flow (fuel/air ratio)

The burner lifetime shall be specified as well as the recommended replacement / maintenance intervals.

5.2 The combustor shall have a dual ignition system. During operation the igniters shall not be exposed to the high temperatures of the primary combustion zone.

5.3 Optical and/or thermal flame sensors shall enable inspection of the flame during operation.

6. Adjustable Vanes

6.1 So far the compressor air flow is controlled by means of variable guide vanes (VGV), inlet guide vanes (IGV) or/ and variable stator vanes (VSV) the corresponding mass flow charts over the guide vane angle / travel have to be indicated in the form of a chart or table.

6.2 The actuator is to be designed in a way to be capable to operate the adjusting mechanism of the vanes under all conditions. Corresponding charts / tables shall be available on request.

7. Internal air Cooling System

The design shall be such to enable an adequate air flow capacity to keep the temperature in the power turbine safely within the design limits under full load conditions.

8. Bleed Valves

8.1 The arrangement of bleed valves shall be documented in a drawing, indicating their position and size. The associated (maximal) power loss of the turbine shall be specified for the maximal flow rates of the bleed valves.

8.2 In case that bleed air is used for anti-icing purposes, this system and its associated technical parameters such as flow rate, supply pressure, maximum air temperature and simplified heat balance, shall be submitted for information purposes.

9. Bearings

9.1 Bearings are generally to be designed in accordance to the manufacturer's standards for loads resulting from the turbine's full output operation for an adequate life time (compare 1.). They are to be equipped with adequate, replaceable sealing devices and shall be reliably lubricated to withstand also short time operation under normal or exceptional transient conditions, such as shut-down due to trip, blackout, etc.

9.2 Vibration monitoring for the bearing condition may be required by **TL**, depending on the application.

10. Gas Turbine Enclosure

10.1 The enclosure shall include a system for fire detection and automatic fire extinguishing.

10.2 The enclosure ventilation or cooling air of the gas turbine shall be supplied from redundant fans with separated electrical power supply. The distribution of ventilation air shall ensure that an acceptable temperature profile of the gas turbine is maintained, and that any local accumulation of combustible gas mixtures is prohibited.

In case of emergency, closing of the ventilation ducts of the plant shall be feasable in a controlled way and within short time, avoiding further damages due to local overheating.

10.3 The enclosure shall be equipped with at least two exits, arranged on opposite sides of the enclosure. The exits shall provide an easy escape from all relevant positions inside the enclosure.

Interlocks on doors shall be provided to ensure that fire extinguishing medium hazardous to personnel is not released, when personnel are inside the enclosure. A gas turbine start is to be interlocked when personnel are inside the enclosure and the enclosure doors are open. During operation the doors have to be locked.

10.4 If NBC protection of the machinery rooms is required the air pressure in the enclosure should be kept lower by not less than 0,5 mbar than the pressure in the machinery room.

11. Fire Safety

11.1 In addition to the machinery room fire fighting system, an approved automatic fire extinguishing system is to be provided for each gas turbine enclosure.
Enclosure ventilation ducts are to be automatically closed when a confirmed fire is detected.

11.2 Inside of enclosure four flame detectors shall be arranged at different locations. A type approval is required for flame detectors.

Detected fire in enclosure or engine room has to release an alarm and automatic cut off of the fuel supply.

11.3 The accumulation of flammable fluid inside of enclosure bottom has to be prevented by draining.

11.4 Purging of all internal gas turbine parts is required in order to discharge liquid or gaseous fuel before ignition. Purging shall be automatically initiated for an adequate time before ignition signal is released.

11.5 Where surface temperatures exceed 220 °C, insulation of oil non-absorbent material has to be applied. In case that the insulation material may be penetrated by fuel, it is to be additionally shielded by sheet steel or equivalent.

12. Starting Systems

12.1 The starting system, electrically, pneumatically or hydraulically driven, shall have redundancy regarding technical design and physical arrangement. A redundancy of starting system is not required if redundant propulsion gas turbines are available each one with its own starting systems. The same applies for turbines installed in plants in combination with other power units for back up, such as diesel engines or electric motors.

12.2 The capacity of the starting system is to be designed in order to enable six (6) consecutive starts of gas turbines for essential main propulsion duties (controllable pitch propellers or other device enabling the start without opposite torque) respectively three (3) consecutive starts of gas turbines for non-essential propulsion and driving of auxiliaries. The recovery time of the starting system depends on minimum time available between start attempts based on starting and control cycles.

12.3 Prior to ignition process, automatic purging is required for all starts and restarts. The purge phase is to be of sufficient duration in order to remove all the accumulated fuel.

12.4 The starting control system is to be fitted with ignition detection devices. If light-off will not be achieved within a preset time, the control system has to abort the ignition automatically, shut-off the main fuel valve and release a purge cycle.

12.5 The start system shall have its own protective system to ensure prevention of damage due to overspeed or failure to reach ignition speed. The starting system is to be protected prior to mechanical and electrical overload.

13. Lubricating Oil System

13.1 For multiple gas turbine arrangements each turbine shall be supplied by a separate and independent lubrication oil system.

13.2 Bearing lubrication may not be impaired by hot gases or by adjacent hot components.

13.3 The lubricating oil system has to be equipped with sufficient means for filtering, heat exchange, magnetic chip detection and water separation in accordance to usual marine practice.

The lubrication equipment of a gas turbine is to be arranged and protected in a way, that in case of leakage the lubricating oil will not be spoiled over surfaces with a temperature of above 220 °C and will not reach any rotating parts.

The lubricating oil system has to be equipped with a filter device, for which cleaning is possible without interruption of operation of the turbine. For redundant plants and other applications as described under 12.1 double filters are not required. The condition of the filter (s) is to be monitored by indication of the pressure difference or other adequate means. Tanks are to be equipped with oil stand indications combined with a low level alarm. Re-filling of oil shall be possible without interrupting the operation. Means for taking of representative samples for analysis purposes are to be provided.

13.4 Especially in case of application of synthetic lubricants attention shall be given to the compatibility to the materials of sealing arrangements and heat exchangers. Leakages within heat exchangers should not lead rapidly to a total contamination of the lubrication oil.

14. Fuel Oil Systems

Gaseous fuel or combined gas / liquid fuel is excluded for the scope of these Rules.

14.1 Fuel nozzles are to be replaceable as complete units without requirement of major adjustments works after replacement.

14.2 The system is to be equipped with suitable drain facilities for the fuel manifold and fuel nozzle to safely handle excessive fuel originating from shutdown (normal and emergency) of the engine fuel system.

14.3 The combustors are to be equipped with a separate drainage system, preventing accumulation of fuel after a failed start.

14.4 The day tank for the fuel supply of the gas turbine shall have adequate capacity referenced to the ship´s destination and the purpose of installation of a gas turbine for essential main propulsion. The day tank has to contain fuel in accordance to the specifications of the gas turbine maker adequately conditioned for immediate use.

15. Turning Gear

Essential main propulsion turbines are to be equipped with turning gear both for the gas generator as well as for the power turbine.

The rotors of turbines for driving of auxiliaries shall at least be capable of being turned by hand.

16. Emergency Operation

16.1 In multi-shaft installations, further operation of at least one propulsion train has to be feasible in the case that one train is unavailable.

16.2 In single-shaft installations with two or more essential main gas turbine driving units, care is to be taken to enable further propulsion on a reduced power level , in the event of failure or unavailability of one of the gas turbines. Each gas turbine is to be designed in a way to be able to drive the shaft on a reduced power level, still enabling at least secure manoeuvring of the ship.

16.3 In the case of single-shaft installations with only one essential main gas turbine, special provisions are to be met in agreement with **TL** to ensure additional adequate redundancy on the level of components / systems.

D. Control and Monitoring

1. Control Logic

1.1 Gas turbines are to be installed with a type approved control and monitoring system.

1.2 The control logic shall include the following:

- Monitoring of relevant operational parameters for control purposes, such as vibration, temperature, speed, etc. including limits and set points

- Normal and emergency stop and start sequence

- Load control

- Fuel control for normal running

- Fuel control for shut-down

- Alarms and shut-down

- Automatic purge cycle

- Dther systems (auxiliary supply systems and safety systems)

- Override functions

1.3 A Failure Mode and Effect Analysis (FMEA) is required for the verification of the logical interconnections within the control system of gas turbines for essential main propulsion. Single failure of any system or control during operation at any mode shall not lead to loss of control of safety related properties of the ship, e.g. loss of control of propulsion, manoeuvrability for propulsion units, loss of electrical supply for auxiliary turbines, etc. Safe operation of the ship is to be demonstrated within the FMEA after partly or complete failure or malfunction of a gas turbine unit, subjected to these rules.

2. Operation Characteristics

2.1 Automation

Automation shall be used to simplify operation and control and exclude operational mistakes by an automatic initiation of procedures operationally connected to each other. In addition automation shall enable a centralised handling and control of the propulsion plant. The equipment for local manual handling and control has to exist to a full extent independently from the degree of automation.

The procedures to prepare the ship for sea shall not be included in the automation.

2.2 Starting

Start-up shall take place automatically in a defined sequence. Interlocks are to be provided to ensure that this sequence (attainment of ignition speed, ignition, flame monitoring) is followed.

Starting sequence is to be disconnected and main fuel valve to be closed within a pre-determined time, when ignition has failed.

The purging mode is to be integrated in the control system both for normal start-up as well as after start failure. The duration of required purging should normally be sufficient to displace the exhaust system volume three times before attempting re-start.

2.3 Speed control

2.3.1 Gas turbines within the scope of these Rules are to be fitted with a speed governor which, in the event of a sudden load drop, prevents the revolutions from increasing to the trip speed.

2.3.2 The speed increase of gas turbines driving electric generators subsequent to a load drop from 100 % to idling may not exceed 10 % of the nominal speed and shall return to the steady state with a maximal deviation of 5 % of the nominal value within 2 s. The transient increase shall in any case remain safely within the overspeed margin.

2.3.3 Gas turbine control systems are to be provided with overspeed protection preventing the turbine speed from exceeding 115 % of the maximum continuous speed (project related speed).

2.4 The safety system has to work independently from the gas turbine control. In the case of activation of a safety device, the gas turbine has to be blocked against a new start, before manual acknowledgement. The devices for de-blocking have to be arranged in a way enabling a quick re-start attempt of the gas turbine.

3. Control Stations

3.1 The operation of the gas turbine shall be possible by remote control from the machinery control centre (MCC). The wiring for control purposes shall be independent and free of cross-connections to other systems.

3.2 An additional manual control is to be provided directly at the gas turbine´s enclosure and shall include a shut-down release. This shall interrupt the fuel supply instantaneously.

Suitable operating devices have to be provided at a good accessible position.

3.3 The operation of the gas turbine control system shall be independent of the common ship control system.

4. Power Supply

The control system is to be equipped with an uninterruptible power supply designed to maintain supply also under blackout conditions. Total loss of control system power shall lead to a controlled and safe turbine shutdown.

5. Monitoring

An instrumentation list showing sensor, type, set point and measuring limit for essential main propulsion gas turbines is required according to Table 4A.1 for approval.

6. Further Requirements

For all equipment of control, operating and watch stands and centres the requirements of the **TL** Rules for Automation, especially Section 1 have to be applied. If an extended scope of the operating and control equipment is required by the Naval Authority, it has to be defined in the building specification.

E. Arrangement and Installation

1. Alignment and Mounting

1.1 Shaft alignment calculations of the complete drive train are to be submitted for approval. Both hot and cold conditions are to be included in the calculation. The shaft alignment calculation for propulsion purposes as required by **TL** is limited to the part between gear box and propeller. The internal alignment of the turbine is to be carried out in accordance to the manufacturer´s recommendations.

1.2 In general the gas turbine foundation in combination with the resilient mountings and the frames shall be designed in a way, that ship deflections do not cause distortions within the integral gas turbine manifold. Design inherent deflections between gas turbine unit and gear box are to be compensated by a suitable flexible element, such as elastic, tooth, reinforced plastic membrane, steel plate coupling, etc.

1.3 Further calculations, which may also be required for naval ships, are not subject to the basic classification procedure. Such calculations are:

- Extreme loads due to given profiles of accelerations
- Forces transferred to foundation structure due to deflection of the ship structure
- Application of specific external acceleration loads transferred via the foundation to the turbine
- Crash stop loads

Table 4A.1 Alarms and indicators for essential main propulsion gas turbines

Signal **F = Fault** **L = Low limit** **H = High limit** **S = Shut-down**	**Individual Alarm Indication**	**Group Alarm Indication**	**Shut-down**
Fuel temperature		H	
Fuel oil supply pressure	L		
Fuel filter, differential pressure **(2)**	H		
Level in lubrication oil sump		LH	
Lubrication oil pressure	L		LS
Lubrication oil temperature inlet	H		
Lubrication oil filter, differential pressure	H		
Cooling water temperature		H	
Cooling water pressure	L		
Compressor inlet pressure or air intake filter, differential pressure	HL		HS LS
Anti-icing system failure		F	
Power turbine overspeed, speed sensor	H		HS
Failure to reach idle speed **(2)**	F		
Failure to ignite	F		FS
Flame out detection	F		FS
Power turbine inlet temperature **(1)**	L		
Exhaust temperature (power turbine outlet)	H		HS
Bearing temperature	H		
Thrust bearing temperature	H		
Vibration, vibration sensor (for each bearing) **(3)**	H		HS
Axial displacement of power turbine, thrust bearing value	H		HS
Power loss of monitoring system	F		
Power loss of control system	F		
Loss of cooling air supply (pressure or flow)	L		
Starting system failure	F		
Fire detection inside gas turbine enclosure	F		FS

(1) *Not less than 6 temperature sensors per turbine*

(2) *Not required for generator driving turbines*

(3) *Details in accordance with monitoring concept of the gas turbine manufacturer*

For non-essential propulsion gas turbines and gas turbines driving auxiliaries the extent of alarms and indicators has to be agreed upon with **TL** considering the safe operation of these units.

- Forces caused from blade loss due to blade fracture
- Any other operational load significant for the individual application

Such calculations may be reviewed by **TL** on special request and in accordance with given specifications.

1.4 The gas turbine (power part) shall be connected to any consumers, propulsion train or gear box by flexible couplings fulfilling also safety functions.
Additional separating devices (clutches) are to be provided when more than one gas turbine is driving a common gear box or no controllable pitch propeller is installed.

1.5 Applied resilient mountings, elastic or other compensating couplings and clutches shall be preferably type approved. Due to the specific requirements for the connecting components of turbine units (high revolutions, major displacements) a case by case dedicated design/approval taking into account the recommendations of the manufacturer shall be considered.

2. Air inlet and Exhaust Gas Outlet System

2.1 The air inlet shall be equipped with air filters in order to avoid the ingestion of dirt or harmful particles including sea salt deposits to the gas turbine. The pressure drop across air filter shall be monitored and indicated in relation to the maximal permissible value.

Air filter icing is to be prevented by an anti-icing system, so far required by the specific application. The suction of air intakes is to be arranged and designed in a way to limit intrusion of spraying water to a minimum. The air inlet shall be equipped with means to enable drainage of water.

2.2 Air intakes and exhaust outlet are to be arranged in a way that re-ingestion of exhaust gases is minimised.

2.3 Multi engine installations shall be equipped with separate inlets and outlets for each gas turbine.

2.4 Inlet and exhaust silencers are to be fitted, if needed, in order to limit the sound power level. The maximal sound power at a distance of one meter from the gas turbine system shall not exceed 110 dB for unmanned machinery spaces (typically in the gas turbine's enclosure) or 80 dB for manned machinery control centres (MCC).

3. Vibration Analysis

3.1 For propulsion systems driving a propeller or water jet a torsional vibration of the complete system including gas turbine and propulsion train is to be submitted for approval. As main excitation for the system the periodical forces of the propeller / impeller may be regarded (1st and 2nd blade excitation).

3.2 Excitations generated by the power turbine may be neglected, unless specifically given by the manufacturer. Excitations exceeding 100 Hz can be neglected for the scope of torsional vibration calculations.

3.3 Further vibration analyses such as bending vibrations may be required by the manufacturer of the gas turbine. The recommendations of the manufacturer in this respect shall be applied.

F. Tests and Trials

1. Material Tests

1.1 Material certificates for the components as listed in Table 4A.2 are to be supplied on request. Special agreements related to the manufacturing procedures and suppliers have to be met on a case by case base. The extent of tests shall at least comply with the approved quality scheme of the manufacturer

1.2 Non-destructive examination

Non-destructive examination shall be applied for the rotors, blades, disks and welded joints of rotating parts. An other production control process may be accepted as equivalent for welded joints. The examination shall be performed by the manufacturer and the results together with details of the test method are to be evaluated according to recognised criteria of acceptability and documented in an acceptance protocol.

2. Tests during construction at the manufacturer's works

2.1 Tests on components

The following component tests have to be performed for every gas turbine:

2.1.1 Pressure and tightness test

Turbine casings are to be tested with 1,5 times the design pressure. Design pressure is the highest expected pressure within the casing under nominal operating conditions (at least 1,25 times maximum allowable working pressure under nominal conditions).

The tightness test of the turbine casing may be replaced, so far not practicable, by other alternative means with the agreement of **TL**.

Further pressure vessels, such as coolers, heat exchangers, etc. are to undergo a pressure test with 1,5 times of their design pressure.

2.1.2 Rotor balancing

Before final installation all completed rotors including mounted discs shall be dynamically balanced. The balance procedure as well as the results before and after balancing shall be documented. For assessment ISO 21940-11 or comparable regulations may be used.

2.1.3 Cold overspeed test

Turbine and compressor wheels are to be tested at a speed at least 15 % above the rated speed for not less than three minutes.

TL may accept mathematical proof of the stresses in the rotating parts at overspeed as a substitute for the overspeed test itself provided that the design is such that reliable calculations are possible and the rotating parts have been subjected to thorough non-destructive testing to ascertain their freedom from defects.

2.2 General requirements for testing of gas turbines

2.2.1 The planned procedures for the type test, work's test as well as sea trials of complete gas turbine units are to be submitted for approval. The tests shall be be witnessed by a **TL** Surveyor.

Table 4A.2 Type of material certificate required for essential components

Component	Material certificate
Blades	**TL** Material Certificate or Manufacturer Inspection Certificate to be decided case by case by **TL** dependent on the installed quality assurance system of the manufacturer
Impellers	
Shafts	
Disks	
Tie bolts	
Combustors	
Fuel nozzles	
Gas generator casing	
Turbine generator casing	
Labyrinth seals	Manufacturer inspection certificate
Bearings, hydrodynamic	

2.2.2 Type test is referring to one specific type of engine (new or upgraded design) and does not cover a range of substantial design variations. The maximum speed of compressor and power turbine, firing temperature, turbine inlet temperature, exhaust temperature, mass flow rates and power are typically fixed design values for a type of turbine. Such a list of characteristic parameters will be listed for reference purposes in the type approval Certificate.

2.2.3 Every gas turbine, besides the extensively type tested prototype, shall undergo, before delivery the work's tests (Factory Acceptance Test) according to 2.4 and the sea trials according to 4., witnessed by a **TL** Surveyor and documented accordingly.

2.2.4 Besides the characteristic gas turbine parameters as listed in A.3., additionally the following parameters indicating performance and consumption under deviating ambient condition parameters shall be recorded:

- Ambient air temperature
- Ambient air pressure
- Relative humidity
- Lower heating value for liquid or gaseous fuel
- Torque measured by absorption dynamometer or shaft torque meter

The measured ambient parameters as listed above shall be used in combination with the reference conditions as listed in Table 4A.3 for calculation of referenced values for specific fuel consumption, so far required by the Naval Authority.

Table 4A.3 Standard reference conditions according to ISO 2314

Temperature	15 °C
Humidity	60% relative
Barometric pressure	1013 mbar

2.2.5 The monitoring and control system used during FAT (manufacturer test bed facilities) of the gas turbine manifold has to deliver equivalent and comparable parameter lists to the system´s, which will be applied on the ship or during containerization of the manifold. Latter systems are to be approved by the manufacturer of the gas turbine and have to be type approved by **TL.**

2.2.6 After running on the test bed, the fuel delivery system of gas turbines for propulsion is normally to be adjusted and limited to 100 % power. The fuel system for gas turbines driving generators may be limited to a higher value, but not higher than 110 % output power. The limit value shall not exceed the "maximum permissible power" as defined under A.2.

2.3 Testing within type test procedure (FAT Prototype)

2.3.1 General

Type approval of gas turbines includes an extended type test for the first gas turbine of a family. The type test is to be carried out at the manufacturAşaer's facilities and is normally to be attended by a **TL** Surveyor. For already existing and proven design a well documented and complete type test procedure performed by the manufacturer, and attended by another independent body or recognized Classification Society may be accepted by **TL** after thorough review of the documentation.

2.3.2 Scope of tests

The type test has to include:

- Start test
- Emergency shutdown test
- Performance test
- Emergency operation test

The test procedure is to be approved by **TL** prior testing.

The following tests shall be carried out preferably with the actual control system installed and fully active. It is recommended to check the control system provisionally during type test and work´s test, but additionally in combination with the finally installed system on the ship before or during the sea trials.

2.3.3 Start test

There has to be at least one false engine start simulation, followed by the manufacturer's specified minimum fuel drainage time before attempting a normal start.

Three normal restarts after an emergency shutdown are to be performed.

2.3.4 Emergency shut-down test

Emergency shutdown may be caused by:

- Hot shut-down, at full load; restart is to be achieved before lockout and within 30 minutes.

- Failure to ignite, resulting in aborted start sequence

- Flame out

During this test the gas turbine is to accelerate to the overspeed limit (115 %) in order to verify the function of the overspeed shut-down.

2.3.5 Performance test (type test)

2.3.5.1 Test sequence

The test sequence is as follows:

- 100 hours total at different part loads and time increments between maximum continuous turbine rotational speed and minimum idle speed for propulsion applications (Propeller curve) or synchronous speed and different loads for generator applications (Load Variation). For details see 2.3.5.2 and 2.3.5.3 .

- Thereof at least 1 hour total at rated maximum continuous power (100 % output at 100 % torque and 100 % speed).

- 30 min. with 110 % power.

The performance test as roughly described herein and in 2.4 represents minimal requirements. It is taken for granted that the manufacturer has performed long term running trials for the first engine, which are documented and available on request.

In case that the gas turbine shall be offered for both propulsion and generator purposes, the performance test shall include both load variation at variable speed (propeller curve) as well as constant speed. The type approval then covers up both applications, so far successfully contacted.

2.3.5.2 Propulsion plants

The performance test takes mainly reference to the 100 % and 110 % output. Further investigation points under steady conditions shall be required in accordance to the available facilities. Commonly the 90 %, 75 %, 50 %, 25 % output point at variable speed corresponding to the propeller curve shall be demonstrated and documented.

2.3.5.3 Generator plants

For generator plants further tests concerning load drop and load raise as well as part load running will be required. This will be done on individual base, when approving the proposed test procedure, in order to take into account the special conditions of the specific turbine resp. manufacturer.

In general for generator plants, load drop from 100 % to 0%, load increase from 0 % to 100 % and other load variations due to power demand, are not to result in a transient variation in rpm higher than +/- 10 %. The permanent speed fluctuation under stationary conditions shall not exceed +1 % of the rated speed.

2.3.6 Emergency operation test

For emergency operating situation, the following tests are to be performed:

- Quick start

- Override functions

The manufacturer has to demonstrate by proven experience or design calculations that this emergency operation will not cause malfunctions or damages to the gas turbine.

2.3.7 Type test report

The gas turbine manufacturer's records including all monitored performance data shall be documented in a type test report, which is to be submitted for approval.
In case that, irregularities, such as failures of components, seizing of parts, etc., occur, the cause has to be analysed and eliminated. The report on the damage/irregularities and the introduced countermeasures has to be part of the report.

2.4 Testing within work's test procedure (FAT individual gas turbine)

2.4.1 General

Every gas turbine of a type approved type, subject to certification in accordance to A.5., shall undergo a work´s test in the facilities of the manufacturer before certification and delivery for installation.

The work's test procedure is to be approved by **TL** prior testing.

2.4.2 Scope of tests

The work's test has to include:

- Start test according to 2.3.3
- Emergency shut-down test according to 2.3.4
- Performance work's test according to 2.4.3
- Emergency operation test according to 2.3.6

2.4.3 Performance work's test

The test sequence has to be as follows:

- 1 hour total at rated maximum continuous power, 100 % output at 100 % torque and 100 % speed
- 30 min. with 110 % power
- Further operation at part load (totally 1h) with variable speed (for the case of essential main propulsion turbine) or at synchronous speed (for generator applications).

For special applications such as naval craft further or "in lieu of" testing may be required and will be agreed upon between **TL** and the manufacturer.
This depends on the special requirements of the Naval Authority.

For further details see 2.3.5.

3. Boroscope inspection

3.1 A boroscope inspection is to be conducted after the type test according to 2.3 and the work's tests according to 2.4.

After the type test normally the turbine unit will be disassembled and the major components will be checked thoroughly. If such a procedure is applied, a boroscope inspection is not necessary.

A boroscope inspection may be required also after sea trials, on specific demand of an involved party or if irregularities are detected.

Boroscope inspection shall be conducted or witnessed by an attending **TL** Surveyor, if required by **TL**.

3.2 Boroscope inspection of the following parts is to be conducted, if inspection ports are available:

- Compressor (blades and nozzles)
- Combustor
- Fuel burners
- High pressure turbine (blades and nozzles)
- Power turbine (blades and nozzles)

3.3 In general no cracks or major wear shall be seen in rotating parts after testing of a new gas turbine.

Minor cracks, indents or tears in uncritical parts may be accepted based on documented acceptance criteria.

4. Sea Trials

4.1 Trial procedure

The sea trials have to simulate the conditions in which the engine is expected to operate in service on board of the naval ship, including typical start-stop cycles, idling, acceleration, deceleration.

The sea trial procedure is to be approved by **TL** prior to testing.

4.2 No load running, adjustments

Prior to the start of the sea trials, the engine and the control and monitoring system are to demonstrate trouble free running at no load for 20 minutes.

For generator gas turbines testing and adjustment of the load sharing characteristics, as far as applicable, are to be carried out. Such test adjustments may also be carried out in the facilities of the Generator Set Maker or during containerization.

4.3 Performance test

As a minimum the engine is to be run for at least 6 hours as follows:

- At least 4 hours at rated speed
- At least 2 hours at engine speed corresponding to normal continuous ahead cruising speed v_M

For special purpose gas turbines such as for nonessential propulsion, special considerations in accordance with specifications of the Naval Authority may be applicable.

4.4 Special tests

Crash-stop conditions shall be tested from full speed. This is to be performed in the fastest time permitted by the controls of the gas turbine.

There shall be at least one simulation of a false turbine start with the following purging time before attempting a normal start. Minimum time required for restart of the turbine is to be checked in order to verify that start can be achieved before thermal interlock occurs.

4.5 Vibration measurements

Vibration measurements are to be recorded during sea trials. The vibration signals are to be recorded for conditions taking into consideration the possible operation modii of the propulsion plant, e.g. controllable pitch propeller and constant turbine speed or fixed pitch propeller and variable turbine speed. Clutch-in procedures, starting and stopping, etc. shall be investigated on request separately.

4.6 Final inspection

Boroscope inspection according to 3. may be required by **TL** after sea trials.

SECTION 4 B

THERMAL TURBOMACHINERY/EXHAUST GAS TURBOCHARGERS

A. General

1. Scope

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.

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).

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

2. Definitions

Regarding turbocharger speed conditions, the following definitions are to be applied:

- maximum permissible speed:
  maximum turbocharger speed, independent of application.

- maximum operational speed:
  speed at 110 % diesel engine output.

- operational speed:
  speed at 100 % diesel engine output representing MCR (maximum continuous rating) condition

The maximum operational speed and maximum permissible speed may be equal.

3. Type Approval

3.1 In general turbochargers are type approved. A Type Certificate valid for 5 years will be issued in accordance with 3.2.

3.2 Documents for approval

The documents listed in the following are to be submitted to **TL.** (I – For Information, A – For approval)

Category A:

On request

- Containment test report. (A)
- Cross sectional drawing with principal dimensions and names of components. (I)
- Test program. (A)

Category B and C:

- Cross sectional drawing with principal dimensions and materials of housing components for containment evaluation. (I)
- Documentation of containment in the event of disc fracture. (A)
- Operational data and limitations as (I):
- 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 (A)

- Type test reports. (A)

- Test program. (A)

Category C:

- Drawings of the housing and rotating parts including details of blade fixing. (I)

- Material specifications (chemical composition and mechanical properties) of all parts mentioned above. (A)

- Welding details and welding procedure of above mentioned parts, if applicable. (A)

- Documentation* of safe torque transmission when the disc is connected to the shaft by an interference fit, see C.3. (I)

- Information on expected lifespan, considering creep, low cycle fatigue and high cycle fatigue. (I)

- Operation and maintenance manuals*. (I)

** Applicable to two sizes in a generic range of turbochargers.*

Additional information is to be submitted to **TL** with documents:

- Details (name and address) of the subcontractors for rotating parts and casings.

- Details (name and address) of the licensees, if applicable, who are authorized by the licensor to produce and deliver turbochargers of a certain type.

4. Certification

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.

4.2 For category C, this shall be verified by means of periodic product audits of an Alternative Certification Scheme (ACS; see Classification and Surveys, Section 2, item F) by **TL**.

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.

4.4 Turbochargers shall be delivered with:

- For category C, a society certificate, which as a minimum cites the applicable type approval and the ACS, when ACS applies.

- For category B, a work's certificate, which as a minimum cites the applicable type approval, which includes production assessment.

4.5 The same applies to replacement of rotating parts and casing.

4.6 Alternatively to the above periodic product audits, individual certification of a turbocharger and its parts may be made at the discretion of **TL**. 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.

B. Design and Installation

1. General

Turbochargers are to be designed to operate at least under the ambient conditions given in TL-R M46 and TL-R M28. The component lifetime and the alarm level for speed shall be based on 45°C air inlet temperature.

2. Basic Design Considerations

Basis of acceptance and subsequent certification of a turbocharger is the drawing approval and the documented type test as well as the verification of the containment integrity.

The turbocharger rotors need to be designed according to the criteria for natural burst speed. In general the burst speed of the turbine shall be lower than the burst speed of the compressor in order to avoid an excessive turbine overspeed after compressor burst due to loss of energy absorption in the compressor.

3. Air Inlet

The air inlet of the turbocharger is to be fitted with a filter in order to minimise the entrance of dirt or water.

4. Hot Surfaces

4.1 Parts with surface temperatures above 220 °C are to be properly insulated in order to minimise the risk of fire if flammable oils, lubrication oils, or fuel come into contact with these surfaces.

4.2 Pipe connections have to be located or shielded with collars in such a way that either spraying or dripping leak oil may not come into contact with hot surfaces of more than 220 °C.

4.3 Hot components in range of passageways or within the working area of turbochargers shall be insulated or protected so that touching does not cause burns.

4.4 Pipe connections in lubrication oil lines for turbochargers with pressure above 1.8 bar shall be screened or otherwise suitably protected to avoid as far as practicable oil spray or oil leakage onto potentially hot surfaces, into machinery air intakes, or other sources of ignition. The number of joints in such piping shall be kept at a minimum.

5. Bearing Lubrication

5.1 Bearing lubrication shall not be impaired by exhaust gases or by adjacent hot components.

5.2 Leakage oil and oil vapours are to be evacuated in such a way that they do not come into contact with parts at temperatures equal or above their selfignition temperature.

5.3 For turbochargers which share a common lubrication system with the diesel engine and which have got an electrical lubrication oil pump supply, it is recommended to install an emergency lubrication oil tank.

5.4 A gas flow from turbocharger to adjacent components containing explosive gases, e.g. crankshaft casing shall be prevented by an adequate ventilating system.

6. Pipe and Duct Connections

Pipe or duct connections to the turbocharger casing are to be made in such a way as to prevent the transmission of excessive loads or moments to the turbochargers.

7. Alarms & Monitoring

7.1 For all turbochargers of Categories B and C, indications and alarms as listed in Table 4B.1 are required.

7.2 Indications may be provided at either local or remote locations.

Table 4B.1

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**(1)**	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)**

(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.*

C. Tests

1. Material Tests

1.1 General

Material testing is required for casings, shaft, compressor and turbine wheel, including the blades. The materials used for the components of exhaust gas turbochargers shall be suitable for the intended purpose
and shall satisfy the minimum requirements of the approved manufacturer's specification.

All materials shall be manufactured by sufficiently proven techniques according to state of the art, whereby it is ensured that the required properties are achieved. Where new technologies are applied, a preliminary proof of their suitability is to be submitted to **TL.** According to the decision of **TL**, this may be done in terms of special tests for procedures and/or by presentation of the work's own test results as well as by expertises of independent testing bodies.

The turbocharger casings are to be from ductile materials (minimum 90 % ferritic structure) and properly heat-treated in order to achieve the required microstructure and ductility as well as to remove residual stresses. Deviations from the standard heat-treatment have to be approved separately by **TL**.

1.2 Condition of supply and heat treatment

Materials are to be supplied in the prescribed heat-treated condition. Where the final heat treatment is to be performed by the supplier, the actual condition in which the material is supplied shall be clearly stated in the relevant Certificate. The final verification of material properties for components needs to be adapted and coordinated according to production procedure. Deviations from the heat treatment procedures have to be approved by **TL** separately.

1.3 Chemical composition and mechanical properties

Materials and products have to satisfy the requirements relating to chemical composition and mechanical properties specified in the **TL** Rules for Materials or, where applicable, in the relevant manufacturer's specifications approved for the type in each case.

1.4 Non-destructive testing

Non-destructive testing shall be applied for the wheels, blades and welded joints of rotating parts. Another equal production control may be accepted for welded joints. The testing shall be performed by the manufacturer and the results together with details of the test method are to be evaluated according to recognized quality criteria and documented in a Certificate.

1.5 Material Certificates

Material Certificates shall contain at least the following information:

- Quantity, type of product, dimensions where applicable, types of material, supply condition and weight
- Name of supplier together with order and job numbers, if applicable
- Construction number, where known
- Manufacturing process
- Heat numbers and chemical composition
- Supply condition with details of heat treatment
- Identifying marks
- Results of mechanical property tests carried out on material at ambient temperature

Depending on the produced component material Certificates are to be issued by **TL** respectively the manufacturer. The required Certificates are summarized in Table 4B.2.

The materials are to conform to specifications approved in connection with the approval of the type in each case.

Table 4B.2 Material Certificates

Components	Type of Certificate
Shaft	**TL** Material Certificate
Rotors (compressor and turbine)	**TL** Material Certificate
Blades	**TL** Material Certificate
Casing	Manufacturer Test Report

If the manufacturer is approved according to D.2. as manufacturer of mass produced exhaust gas turbochargers fitted on diesel engines having a cylinder bore ≤ 300 mm, the material properties of these parts may be covered by Manufacturer Inspection Certificates and need not to be verified by a **TL** Surveyor.

2. Containment Test

2.1 The turbocharger has to fulfil containment requirements in case of rotor burst. This requires that at 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.

2.2 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.

*****A generic range means a series of turbocharger which are of the same design, but scaled to each other.***

The following requirements in item 2.3 and 2.4 are applicable for an approval of the type of turbochargers.

2.3 The minimum speeds for the containment test are defined as follows:

Compressor: ≥ 120 % of its maximum permissible speed

Turbine: ≥ 140 % of its maximum permissible speed or the natural burst speed (whichever is lower).

2.4 The containment test has to be performed at a temperature which is not lower than the maximum allowable temperature of the turbocharger to be specified by the manufacturer. The theoretical (design) natural burst speeds of compressor and turbine have to be submitted for information.

2.5 Manufacturers are to determine whether cases more critical than those defined in items 2.3 and 2.4 exist with respect to containment safety. Where such a case is identified, evidence of containment safety shall also be provided for that case.

2.6 A numerical prove such as Finite Element Method (FEM) of sufficient containment integrity of the casing based on calculations by means of a simulation model may be accepted, provided that:

- The numerical simulation model has been tested and it's applicability/accuracy has been proven by direct comparison between calculation results and practical containment test for a reference application (reference containment test). This proof has to be provided once by the manufacturer who wants to apply for acceptance of numerical simulation

- The corresponding numerical simulation for the containment is performed for the same speeds, as specified for the containment test (see above)

- The design of the turbocharger regarding the geometry and kinematics is to be similar to the turbocharger that was used for the reference 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.

2.7 In cases where a totally new design*** is adopted for a turbocharger for which an application for type approval certification has been requested, new reference containment tests are to be performed.

*** Totally new design means the principal differences between a new turbocharger and previous ones are related to geometry and kinematics. The turbochargers are to be regarded as having a totally new design if the structure and/or material of the turbocharger casings are changed, or any of, but not limited to, the following items is changed from the previous design.

- Maximum permissible exhaust gas temperature
- Number of bearings
- Number of turbine blades
- Number of turbine wheels and/or compressor wheels
- Direction of inlet air and/or exhaust gas (e.g., axial flow orientation, radial flow orientation)
- Type of the turbocharger drive (e.g., axial turbine type, radial turbine type, mixed flow turbine type)

2.8 In general a **TL** Surveyor or the Head Office has to be involved for the containment test. The documentation of the physical containment test as well as the report of the simulation results are to be submitted to **TL** within the scope of the approval procedure.

3. Disc-shaft shrinkage fit

3.1 Applicable to Category C

3.2 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.

4. Type Testing

4.1 Applicable to Categories B and C

4.2 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.

4.3 Turbochargers for the low, medium, and high-speed engines 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.

4.4 The suitability of the turbocharger for such kind of operation is to be preliminarily stated by the manufacturer.

4.5 The rotor vibration characteristics shall be measured and recorded in order to identify possible sub-synchronous vibrations and resonances.

4.6 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.

4.7 The extent of the surveyor's presence during the various parts of the type tests is left to the discretion of **TL**.

5. Spare Parts

The rotating assembly parts (rotor, wheels and blades) as well as turbocharger casings have to be replaced by spare parts which are manufactured by **TL** approved manufacturers according to the previously approved drawings and material specifications. The manufacturer is to be recognized by the holder of the original type approval.

D. Shop Approvals

1. Materials and Production

The manufacturers of the material as well as the production procedures for the rotating parts and casings have to be approved by **TL**

2. Mass Produced Exhaust Gas Turbochargers

2.1 Manufacturers of mass-produced turbo-chargers who operate a quality management system and are manufacturing exhaust gas turbochargers fitted on **TL** approved mass produced diesel engines having a cylinder bore of ≤ 300 mm may apply for the shop approval by **TL** Head Office.

The shop approval is valid for 3 years with annual follow up audits.

2.2 Upon satisfactory shop approval, the material tests according to C.1. for these parts may be covered by a Manufacturer Inspection Certificate and need not to be verified by a Surveyor.

2.3 No **TL** Certificate will be issued for massproduced turbochargers. Mass-produced turbochargers will be mentioned with the serial number in the final Certificate intended for the diesel engine.

3. Manufacturing of Exhaust Gas Turbochargers Under License Agreement

3.1 Manufacturers who are manufacturing exhaust gas turbochargers under a license agreement shall have a shop recognition of **TL** Head Office.

The shop recognition can be issued in addition to a valid license agreement if the following requirements are fulfilled:

- The manufactured turbochargers have a valid **TL** approval of the type for the licensor.

- The drawings and the material specification as well as the working procedures comply with the drawings and specifications approved in connection with the turbocharger approval of the type for the licensor.

3.2 Upon satisfactory assessment in combination with a bench test carried out on a sample basis with **TL** Surveyor's attendance, the drawing approval and tests are not required.

3.3 The shop recognition is valid for three years with annual follow up audits and can be granted, if required in combination with an approval as manufacturer of mass-produced turbochargers.

The shop recognition becomes invalid if the licence agreement expires. The licensor is obliged to inform the **TL** Head Office about the date of expiry.

SECTION 5

MAIN SHAFTING

A. General

1. Scope

1.1 The following Rules apply to standard and established types of shafting for main and auxiliary propulsion as well as lateral thrusters. Deviating designs require special approval by **TL.**

1.2 For difficult or special operating conditions adequate reinforcements have to be provided.

1.3 **TL** reserve the right to call for propeller shaft dimensions in excess of those specified in this Section if the propeller arrangement results in increased bending stresses.

1.4 In case of ships with ice classes, the strengthening factors given in Section 9 are to be complied with.

2. Documents for Approval

2.1 The following drawings are to be submitted in triplicate, all calculations and supporting documentation in one copy for approval:

- Arrangement of the entire shafting, from the main engine coupling flange to the propeller.

- Detailed drawings and material data for all torque transmitting components, especially shafts, couplings and other components.

- Arrangement of the shaft bearings

- Arrangement and detail drawings of the stern tube as well as bush bearings including stern tube sealing and the corresponding lubricating oil system.

- Calculation of the shaft alignment considering all static and dynamic external forces acting on the shaft during operation (e.g. weight of couplings, propeller weight and propeller forces, toothing forces of gears, etc.). With consent of **TL** for shafting with intermediate shaft diameter < 200 mm the alignment calculation may be waived.

- Calculation of torsional vibrations

- In special cases separate bending and axial vibration calculations may be required

- For cast resin foundation of shaft components arrangement and design of the adapting pieces and bolts.

The submitted documentation must contain all data necessary to enable the stresses to be evaluated.

B. Materials

1. Approved Materials

1.1 Propeller, intermediate and thrust shafts together with flange and clamp couplings are to be made of forged steel; as far as applicable, couplings may be made of cast steel. Rolled round steel may be used for plain, flangeless shafts.

In general, the tensile strength of steels used for shafting (shafts, flange couplings, bolts/fitted bolts) shall be between 400 N/mm^2 and 800 N/mm^2. For dynamically loaded parts of the shafting, designed in accordance to the formulas as given under C. and D., and explicitly for the shafts themselves as well as for connecting / fitted bolts for flanged connections in general quenched and tempered steels shall be used with a tensile strength of more than 500 N/mm^2.

However, the value of R_m used for the calculation of the material factor C_w in accordance with formula (2) shall not exceed

– 600 N/mm^2 for propeller shafts made of carbon, carbon manganese and alloy steels, For all other applications;

– 760 N/mm^2 for carbon and carbon manganese steels .
– 800 N/mm^2 for alloy steels.

Where materials with higher specified or actual tensile strengths than the limitations given above are used, the shaft dimensions derived from formulae (1) and (2) are not to be reduced accordingly unless **TL** verifies that the materials exhibit similar fatigue life as conventional steels (see Appendix 1).

1.2 Where in special cases wrought copper alloys resistant to seawater are to be used for the shafting, consent of **TL** shall be obtained.

1.3 For shafts made of fibre reinforced plastics the **TL** Material Rules, Chapter 2 - Fibre Reinforced Plastics and Bonding, are applicable.

2. Testing of Materials

2.1 All components of the shafting which are transmitting the torque from the ship's propulsion are subject to the **TL** Material Rules, Chapter 2 and are to be tested. This requirement is also applicable for metal propeller shaft liners.

2.2 Where propeller shafts running in seawater are to be protected against seawater penetration not by a metal liner, but by plastic coatings, the coating technique used is to be approved by **TL.**

C. Shaft Dimensioning

1. General

1.1 The following requirements apply 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.

1.2 For shafts that are integral to equipment, such as for gear boxes (see Section 6), 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 following requirements 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 considered additionally.

1.3 Explicitly it will be emphasized that the following applications are not covered by the requirements in this Section:

- Additional strengthening for shafts in ships, which are strengthened for navigation in ice (see Section 9)

- Gearing shafts (see Section 6)

- Electric motor and generator rotor shafts

- Turbine rotor shafts (see Section 4A)

- Crankshafts for internal combustion engines (see Section 3).

Additionally, all parts of the shafting are to be designed to comply with the requirements relating to torsional vibrations set out in Section 8.

1.4 In general dimensioning of the shafting shall be based on the total rated installed power.

1.5 Where the geometry of a part is such that it cannot be dimensioned in accordance with these formulae, special evidence of the mechanical strength of the part concerned is to be furnished to **TL.**

2. Alternative Calculation

TL may accept alternative shaft calculations, e.g. according to DIN 743. In such cases complete calculations based on the applied standard are to be submitted to **TL** for approval.

Any alternative calculation has to include all relevant dynamic loads on the complete shafting system under all permissible operating conditions. Consideration has to be given to the dimensions and arrangements of all shaft connections. Moreover, an alternative calculation has 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:

- Low cycle fatigue criterion (typically < 10^4), i.e. the primary cycles represented by zero to full load and back, including reversing torque if applicable. This is addressed by formula (1)
- 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 Section 8. The influence of reverse bending stresses is addressed by the safety margins inherent in formula (1).
- The accumulated fatigue due to torsional vibration when passing through barred speed ranges or other transient operational conditions with stresses beyond the permitted limits for continuous operation is addressed by the criterion for transient stresses in Section 8.

3. Minimum Diameter

The minimum shaft diameter is to be determined by applying formula (1).

$$d_a \geq d \geq F \cdot k \cdot \sqrt[3]{\frac{P_W}{n \cdot \left[1 - \left(\frac{d_i}{d_a}\right)^4\right]} \cdot C_W} \qquad \textbf{(1)}$$

d = Minimum required outside diameter of shaft, [mm]

d_i = Actual diameter of shaft bore [mm]. If the bore in the shaft is ≤ 0,4 · d, the expression,

$$1 - \left(\frac{d_i}{d_a}\right)^4 \text{ may be set to 1,0}$$

d_a = Actual outer diameter shaft [mm]

P_w = Rated power of propulsion motor [kW], gear-box and bearing losses are not to be subtracted.

n = Shaft speed at rated power, [min^{-1}]

F = Factor for the type of propulsion installation [-]

a) Intermediate and thrust shafts

= 95 for turbine installations, diesel engine installations with hydraulic (slip type) couplings and electric propulsion installations,

= 100 for all other propulsion installations,

b) Propeller shafts,

= 100 for all types of installations,

C_w = Material factor, [-]

$$= \frac{560}{R_m + 160} \qquad \textbf{(2)}$$

R_m = Tensile strength of the shaft material [N/mm^2]

k = Factor for the type of shaft [–] (See Table 5.1)

Table 5.1 Form factors for intermediate and propeller shafts

c_K [–]	k [-]	**Shaft Type / Design**
		Intermediate shafts with
1.00	1.00	Integral coupling flange and straight sections **(1)**
1.00	1.00	Shrink fit couplings **(2)**
0.60	1.10	Keyway, tapered connection (not valid with bared speed ranges) **(3) (4)**
0.45	1.10	Keyway, cylindrical connection **(3) (4)**
0.50	1.10	Radial holes of standard design (for example OD shaft of CP plants) **(5)**
0.30 **(7)**	1.20	with longitudinal slots of standard design (for example OD shaft of CP plants) **(6)**
		Thrust shafts external to engine
0.85	1.10	transmitting thrust, additionally to the torque, by means of a collar (bending)
0.85	1.10	in way of axial bearing where a roller bearing is used as a thrust bearing
		Propeller shafts
0.80	1.15	between forward end of the aft most bearing and forward stern tube
0.55	1.22	with flange mounted or keyless taper fitted propellers **(8)**
0.55	1.26	with key fitted propellers (in general not to be used for plants with barred speed ranges) **(8)**

Note:
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.
Footnotes

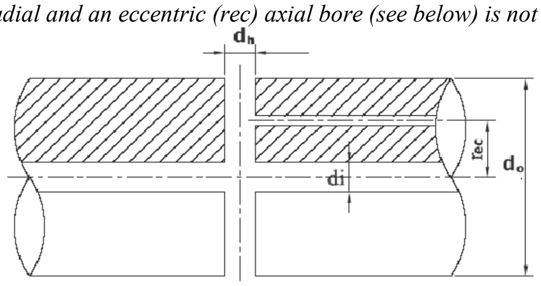
(1) *Fillet radius is not to be less than $0.08d_0$.*

(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 the table note.*

(3) *At a distance of not less than $0.2d_0$ 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 $0.3d_0$.*
The intersection between a radial and an eccentric (rec) axial bore (see below) is not covered here.



(6) *Subject to limitations as slot length (l)/outside diameter < 0.8 and inner diameter (di)/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 degrees 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 2.6 or by direct application of FE calculation. In which case:*
$c_K = 1.45/scf$
Note that the scf is defined as the ratio between the maximum local principal stress and √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.*

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:

- The stress concentration factors (scf) relative to the stress concentration for a flange with fillet radius of $0.08d_0$ (geometric stress concentration of approximately 1.45).

$$c_K = \frac{1.45}{scf} \qquad k = [\frac{scf}{1.45}]^x$$

where the exponent x considers low cycle notch sensitivity.

- The notch sensitivity. The chosen values are mainly representative for soft steels (σ_B <600), while the influence of steep stress gradients in combination with high strength steels may be underestimated.
- 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.

4. Lower Values of Shaft Diameter

An approval of a shaft diameter lower than calculated according to formula (1) is possible under the following conditions:

- the fatigue strength values of the used material in the operating medium have to be submitted

- an advanced calculation method (such as mentioned in 2.) has to be applied

5. Shafts Made of Pipes

For pipe shafts with relative thick walls the problem of buckling needs generally not to be investigated. For thin wall and large diameter shafts buckling behaviour must be checked additionally. For isotropic materials the following formula for the critical torque applies:

$$M_{tcrit} = C \cdot \frac{0{,}272 \cdot E \cdot 2 \cdot r_m^{0{,}5} \cdot t^{2{,}5} \cdot \pi}{(1 - v^2)^{0{,}75}} \quad [\mathrm{Nm}] \qquad \textbf{(2a)}$$

C = Factor for special conditions
  = 1,0 generally

E = Modulus of elasticity [N/mm^2]

ν = Poisson's ratio

t = Thickness of pipe wall
  = $(d_a - d_i) \cdot 0{,}5$ [mm]

r_m = Average radius of the pipe [mm]
  = $0{,}25(d_a + d_i)$

The design criterion is:

$$3{,}5 \cdot M_t \leq M_{crit} \qquad \textbf{(2b)}$$

M_t = Nominal torque at maximum continuous rating [Nm]

6. Consideration of Shock Loads

If the Class Notation **SHOCK** shall be assigned to the naval ship, the influence of the additional accelerations caused by shock loads (shock spectra), are to be defined by the Naval Authority.

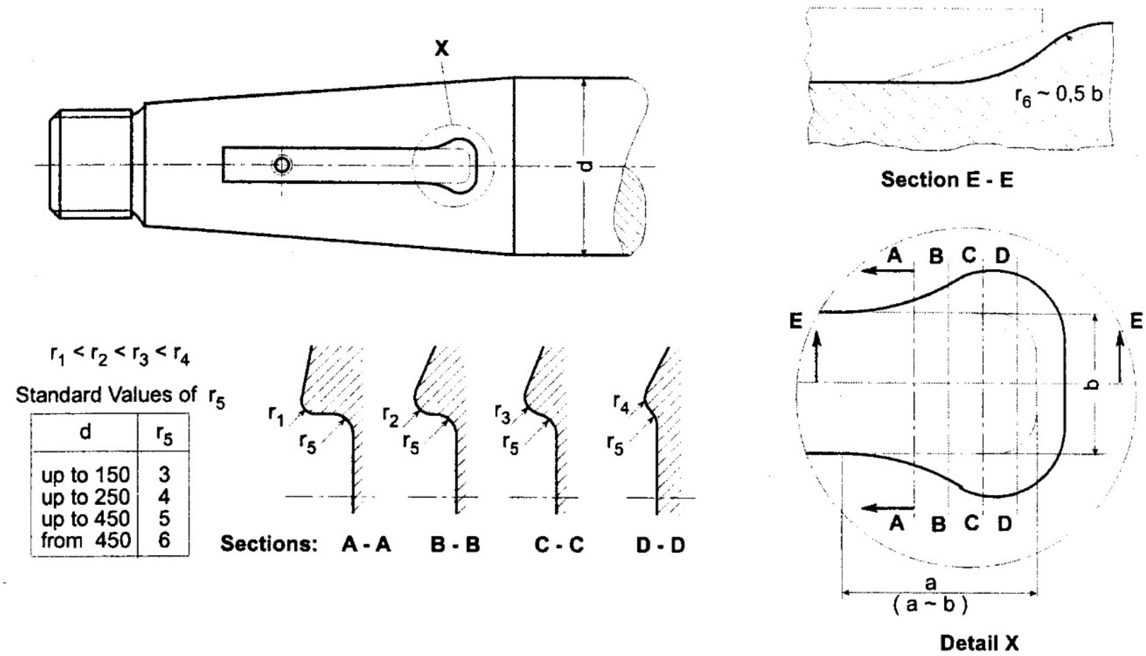


Fig. 5.1 Design of keyway in propeller shaft

D. Design

1. General

The design of the shafts should aim to achieve smooth stress distribution avoiding high stress concentration spots.

Changes in diameter are to be realised by smooth tapering or by providing ample radii. Radii are to be at least equal to the change in diameter.

For intermediate and thrust shafts, the radius at forged flanges is to be at least 8 % of the calculated minimum diameter for a full shaft at the relevant location. The radius at the aft propeller shaft flange shall be at least 12,5 % of the calculated minimum diameter for a full shaft at the relevant location.

The surface quality of the shaft has to be chosen according to the type of loads and the notch sensitivity of the material. In the areas between the bearings a minimum quality of the arithmetic mean roughness of R_a= 10 - 16 μm, at bearing running surfaces and transition zones a value of R_a = 1,6 - 2,5 μm will be in general required.

2. Shaft Tapers and Propeller Nut Threads

2.1 Keyways in the shaft taper for the propeller are to be designed in a way that the forward end of the groove makes a gradual transition to the full shaft section. In addition, the forward end of the keyway shall be spoon-shaped. The edges of the keyway at the surface of the shaft taper for the propeller are not to be sharp. The forward end of the rounded keyway has to lie well within the seating of the propeller boss. Threaded holes for securing screws for propeller keys shall be located only in the aft half of the keyway (see Fig. 5.1)

2.2 In general, tapers for securing flange couplings which are jointed with keys shall have a conicity of between 1 : 12 and 1 : 20. See Section 7A for details of propeller shaft tapers on the propeller side.

2.3 The outside diameter of the threaded end of the propeller retaining nut should not be less than 60 % of the calculated bigger taper diameter.

3. Propeller Shaft Protection

3.1 Sealing

At the stern tube ends propeller shafts with oil or grease

lubrication are to be fitted with seals of proven efficiency and approved by **TL**, see also the requirements applicable to the external sealing of the stern tube in the context of the propeller shaft survey described in Chapter 101 - Classification and Surveys, Section 3.

For protection of the sealing a rope guard should be provided.

The propeller boss seating is to be effectively protected against the ingress of seawater. This seal can be dispensed with if the propeller shaft is made of corrosion-resistant material.

In the case of Class Notation **IWS**, the seal is to be fitted with a device by means of which the bearing clearance can be measured when the vessel is afloat.

3.2 Shaft liners

3.2.1 Propeller shafts which are not made of corrosion-resistant material and which run in seawater are to be protected against ingress with seawater by sea-water-resistant metal liners or other liners approved by **TL** and by proven seals at the propeller.

3.2.2 Metal liners in accordance with 3.2.1, which run in seawater, must be made in a single piece. Only with the expressed consent of **TL** the liner may consist of two or more parts, provided that the abutting edges of the parts are additionally sealed and protected, after fitting, by a method approved by **TL** to guarantee water-tightness. Such joints will be subject of special tests to prove their effectiveness.

3.2.3 Minimum wall thickness of shaft liners

The minimum wall thickness s [mm] of metal shaft liners in accordance with 3.2.1 is to be determined using the following formula:

$$s = 0{,}03 \cdot d + 7{,}5 \qquad (3)$$

d = shaft diameter under the liner [mm]

In the case of continuous liners, the wall thickness between the bearings may be reduced to 0,75 · s.

4. Coupling

4.1 Definitions

In the formulae (4), (5), (6) and (7), the following symbols are used:

A = Effective area of shrink-fit seating, [mm^2]

c_A = Coefficient for shrink-fitted joints, depending on the kind of driving unit [-]

= 1,0 for geared oil engine and turbine drives,

= 1,2 for direct coupled diesel engine drives,

C = Conicity of shaft ends [-]

$$= \frac{\text{difference in taper diameters}}{\text{length of } cone}$$

d = Shaft diameter in area of clamp-type coupling, [mm]

d_s = Diameters of fitted bolts, [mm]

d_k = Inner throat diameter of necked-down bolts, [mm]

D = Diameter of pitch circle of bolts, [mm]

E = Modules of elasticity, [N/mm^2]

f = Coefficient for shrink-fitted joints, [-]

Q = Peripheral force at the mean joint diameter of a shrink fit, [N]

n = Shaft speed, [min^{-1}]

p = Interface pressure of shrink fits, [N/mm^2]

P_w = Rated power of the driving motor(s) [kW]

s_{fl} = Flange thickness in area of bolt pitch circle, [mm]

S = Safety factor against slipping of shrink fits in the shafting, [-]

= 3,0 between motor and gear

= 2,5 for all other applications,

T = Propeller thrust respectively axial force, [N]

z = Number of fitted or necked-down bolts, [-]

R_m = Nominal tensile strength of fitted or necked-down bolt material, [N/mm^2]

μ_o = Coefficient of static friction, [-]

= 0,15 for hydraulic shrink fits,

= 0,18 for dry shrink fits,

θ = Half taper of shaft ends, [-]

= C/2

4.2 Coupling flanges

The thickness of coupling flanges of intermediate and thrust shafts as well as of the forward end of the propeller shaft must be not less than 20 % of the calculated minimum diameter of a solid shaft at the relevant location.

Where propellers are connected by means of a forged flange with the propeller shaft, the thickness of this flange must not be less than 25 % of the calculated minimum diameter of a solid shaft at the relevant location.

The thickness of mentioned flanges shall not be less than the Rule diameter of the fitted bolts, as far as their calculation is based on the same material tensile strength as applied for the shafting.

4.3 Bolts

4.3.1 The bolts used to connect flange couplings are normally to be designed as fitted bolts. The minimum diameter d_s of fitted bolts at the coupling flange faces is to be determined by applying the formula:

$$d_s = 16 \cdot \sqrt{\frac{10^6 \cdot P_W}{n \cdot z \cdot D \cdot R_m}} \qquad (4)$$

The coupling bolts shall be tightened so that flange contact will not be lost under both shaft bending moment and astern thrust.

4.3.2 Where, in special circumstances, the use of fitted bolts is not feasible, **TL** may agree to the use of an application of an equivalent frictional transmission.

4.3.3 The minimum thread root diameter d_k of the connecting bolts used for clamp-type couplings is to be determined using the formula:

$$d_k = 12 \cdot \sqrt{\frac{10^6 \cdot P_W}{n \cdot d \cdot z \cdot R_m}} \qquad (5)$$

4.3.4 The shaft of necked-down bolts shall not be less than 0,9 times the thread root diameter. If, besides the torque, the bolted connection is also required to transmit considerable additional forces, the size of the bolts must be increased accordingly.

4.3.5 Nuts for fitted coupling bolts and shaft nuts for coupling flanges shall be properly secured against unintentional loosening. Shaft nuts for keyless fitted couplings shall be secured to the shaft.

4.4 Shrink-fitted couplings

Where shafts are connected by keyless shrink fitted couplings (flange or sleeve type), the dimensioning of these shrink fits should be chosen in a way that the maximum von Mises equivalent stress in the all parts will not exceed 80 % of the yield strength of the specific materials during operation and 95 % during mounting and dismounting.

4.4.1 Normal operation

For the calculation of the safety margin of the connection against slippage, the maximal clearance will be applied. This clearance has to be derived as the difference between the lowest respectively highest diameter for the bore and the shaft according to the manufacturing drawings. The contact pressure p [N/mm^2] in the shrunk-on joint to achieve the required safety margin may be determined by applying formulae (6) and (7).

$$p = \frac{\sqrt{\theta^2 \cdot T^2 + f \cdot (c_A{}^2 \cdot Q^2 + T^2)} - \theta \cdot T}{A \cdot f} \qquad (6)$$

T has to be introduced as positive value if the propeller thrust increases the surface pressure at the taper. Change of direction of propeller thrust is to be neglected as far as power and thrust are essentially less.

T has to be introduced as negative value if the propeller thrust reduces the surface pressure at the taper, e.g. for tractor propellers.

$$f = \left(\frac{\mu_o}{S}\right)^2 - \Theta^2 \qquad (7)$$

4.4.2 Operation at a resonance

For direct coupled propulsion plants with a barred speed range it has to be confirmed by separate calculation that the vibratory torque in the main resonance is transmitted safely. For this proof the safety against slipping for the transmission torque shall be at least S = 1.8, the coefficient c_A may be set to 1.0. For this additional proof the respective influence of the thrust shall be disregarded.

5. Shafting Bearings

5.1 Arrangement of shaft bearings

Drawings showing all shaft bearings, like stern tube bearings, intermediate bearings and thrust bearings shall be submitted for approval separately, if the design details are not visible at the shafting arrangement drawings. The permissible bearing loads are to be indicated. The lowest permissible shaft speed also has to be considered.

Shaft bearings both inside and outside the stern tube are to be so arranged that each bearing is subjected to positive reaction forces, irrespective of the ship's loading when the plant is at operating state temperature.

By appropriate spacing of the bearings and by the alignment of the shafting in relation to the coupling flange at the engine or gearing, care is to be taken to ensure that no undue shear forces or bending moments are exerted to the crankshaft or gear shafts when the plant is at operating state temperature. By spacing the bearings sufficiently far apart, steps are also to be taken to ensure that the reaction forces of line or gear shaft bearings are not significantly affected should the alignment of one or more bearings be altered by hull deflections or by displacement or wear of the bearings themselves.

Guide values for the maximum permissible distance between bearings ℓ_{max} [mm] can be determined using formula (8):

$$\ell_{max} = K_1 \cdot \sqrt{d} \qquad (8)$$

d = Diameter of shaft between bearings, [mm]

K_1 = 450 for oil-lubricated white metal bearings,

= 280 for grey cast iron, grease-lubricated stern tube bearings,

= 280–350 for water-lubricated rubber bearings in stern tubes and shaft brackets (upper values for special designs only).

Where the shaft speed exceeds 350 min^{-1} it is recommended that the maximum bearing spacing is determined in accordance with formula (9) in order to avoid excessive loads due to bending vibrations. In limiting cases a bending vibration analysis for the shafting system is recommended.

$$\ell_{max} = K_2 \cdot \sqrt{\frac{d}{n}} \qquad (9)$$

n = Shaft speed, [min^{-1}]

K_2 = 8400 for oil-lubricated white metal bearings,

= 5200 for grease-lubricated, grey cast iron bearings and for rubber bearings inside stern tubes and tail shaft brackets.

In general, the distance between bearings should not be less than 60 % of the maximum permissible distance as calculated using formula (8) or (9) respectively.

5.2 Stern tube bearings

5.2.1 Inside the stern tube the propeller shaft shall normally be supported by two bearing points. In short stern tubes the forward bearing may be dispensed with. In such cases generally at least one freestanding journal shaft bearing should be provided.

5.2.2 Where the propeller shaft inside the stern tube runs in oil-lubricated white metal bearings or in synthetic rubber or reinforced resin or plastic materials approved for use in oil-lubricated stern tube bearings, the lengths of the after and forward stern tube bearings should be approximately 2 · d_a and 0,8 · d_a respectively.

The length of the after stern tube bearing may be reduced to 1,5 · d_a where the contact load, which is calculated from the static load and considering the weight of the propeller, is less than 0,8 MPa in the case of shafts supported on white metal bearings and less than 0,6 MPa in the case of bearings made of synthetic materials.

For approved materials higher surface pressure values may be applied.

5.2.2.1 Oil lubricated white metal bearings

The length of white-metal-lined, oil-lubricated propeller-end bearings fitted with an approved oil-seal gland is to be not less than two times the required tail shaft diameter. The length of the bearing may be reduced, provided the nominal bearing pressure is not more than 0.80 N/mm^2, 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 bearing surface. The minimum length, however, is not to be less than 1.5 times the actual diameter.

5.2.2.2 Oil lubricated synthetic material bearings

The length of synthetic rubber, reinforced resin or plastic oil-lubricated propeller end bearings fitted with an approved oil-seal gland is to be not less than two times the required tail shaft diameter. The length of bearing may be reduced, provided the nominal bearing pressure is not more than 0.60 N/mm^2, 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 bearing surface. The minimum length, however, is not to be less than 1.5 times the actual diameter. Where the material has demonstrated satisfactory testing and operating experience, consideration may be given to increased bearing pressure.

Synthetic materials for application as oil lubricated stern tube bearings are to be Type Approved.

5.2.2.3 Oil lubricated cast iron or bronze bearings

The length of oil-lubricated cast iron or bronze bearings which are fitted with an approved oil-seal gland is to be not less than four times the required tail shaft diameter.

5.2.2.4 Stern tube bearing oil lubricating system sampling arrangement

An arrangement for readily obtaining accurate oil samples is to be provided. The sampling point is to be taken from the lowest point in the oil lubricating system, as far as practicable. Also, the arrangements are to be such as to permit the effective removal of contaminants from the oil lubricating system.

5.2.3 Water lubricated bearings

Where the propeller shafts inside the stern tube runs in bearings approved for use in water-lubricated stern tube bearings, the length of the after bearing is to be not less than $4 \cdot d_a$ and at that of the forward bearing is to be not less $1.5.d_a$.

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 **TL.**

Synthetic materials for application as water lubricated stern tube bearings are to be Type Approved.

Note: *In a closed fresh water system lubricated stern tube, the sample is to be drawn from the same agreed position in the system which should be positively identified. The sample should be representative of the water circulating within the stern tube (also refer to TL- G 143 Recommended procedure for the determination of contents of metals and other contaminants in a closed fresh water system lubricated stern tube).*

5.2.4 Grease lubricated bearings

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.

5.2.5 If roller bearings are provided, the requirements of 5.3.2 have to be considered.

5.3 Intermediate bearings

5.3.1 Plain bearings

For intermediate bearings shorter bearing lengths or igher specific loads as defined in 5.2 may be agreed with **TL.**

5.3.2 Roller bearings

For the case of application of roller bearings for shaft lines the design is to be adequate for the specific requirements. For shaft lines significant deflections and inclinations have to be taken into account. Those shall not have adverse consequences.

For application of roller bearings the required minimum loads as specified by the manufacturer are to be observed.

The minimum L_{10a} (acc. ISO 281) lifetime has to be suitable with regard to the specified overhaul intervals.

5.4 Bearing lubrication

5.4.1 Lubrication and matching of materials of the plain and roller bearings for the shafting have to meet the operational demands of seagoing ships.

5.4.2 Lubricating oil or grease is to be introduced into the stern tube in such a way as to ensure a reliable supply of oil or grease to the forward and after stern tube bearing.

With grease lubrication, the forward and after bearings are each to be provided with a grease connection. Wherever possible, a grease gun driven by the shaft is to be used to secure a continuous supply of grease. Where the shaft runs in oil inside the stern tube, a header tank is to be fitted at a sufficient height above the ship's load line. It shall be possible to check the filling of the tank at any time.

The temperature of the after stern tube bearing (in general near the lower aft edge of the bearing) is to be indicated. Alternatively, with propeller shafts less than 400 mm in diameter the stern tube oil temperature may be indicated. In this case the temperature sensor is to be located in the vicinity of the after stern tube bearing.

5.4.3 In the case of ships with automated machinery, **TL** Rules for Automation have to be complied with.

5.5 Stern tube connections

Oil-lubricated stern tubes are to be fitted with filling, testing and drainage connections as well as with a vent pipe.

Where the propeller shaft runs in seawater, a flushing line is to be fitted in front of the forward stern tube bearing instead of the filling connection. If required, this flushing line shall also act as forced water lubrication.

5.6 Condition monitoring of propeller shaft at stern tube

5.6.1 Where the propeller shaft runs within the stern tube in oil the possibility exists to prolong the intervals between shaft withdrawals. For this purpose the following design measures have to be provided:

- A device for measurement of the temperature of the stern tube bearings and the sea water temperature (and regular documentation of measured values), compare 5.4.2

- a possibility to determine the oil consumption within the stern tube (and regular documentation)

- An arrangement to measure the wear down of the aft bearing

- A system to take representative oil samples at the rear end of the stern tube under running conditions for analysis of oil quality (aging effects and content of H_2O, iron, copper, tin, silicon, bearing metal, etc.) and suitable receptacles to send samples to accredited laboratories. (The samples shall be taken at least every six months.)
- a written description of the right procedure to take the oil samples

- A test device to evaluate the water content in the lubricating oil on board (to be used once a month)

- If roller bearings are provided, additional vibration measurements have to be carried out regularly and to be documented. The scope of the measurements and of the documentation has to be agreed with **TL** specifically for the plant.

5.6.2 The requirements for the initial survey of this system as well as for the checks at the occasion of annual and Class Renewal surveys are defined in the relevant CM-PS Record File.

5.6.3 If the requirements according to 5.6.1 and 5.6.2 are fulfilled, the Class Notation **CM-PS** may be assigned.

5.7 Cast resin mounting

The mounting of stern tubes and stern tube bearings made of cast resin and also the seating of intermediate shaft bearings on cast resin parts is to be carried out by **TL** approved companies in the presence of a **TL** Surveyor.

Only **TL**-approved cast resins may be used for seatings.

The installation instructions issued by the manufacturer of the cast resin must be observed.

5.8 Shaft alignment

5.8.1 It has to be verified by alignment calculation that the requirements for shaft-, gearbox- and engine bearings are fulfilled in all relevant working conditions of the propulsion plant. At this all essential static, dynamic and thermal effects have to be taken into account.

The calculation reports to be submitted are to include the complete scope of used input data and have to disclose the resulting shaft deflection, bending stress and bearing loads and have to document the compliance with the specific requirements of the component manufacturer.

5.8.2 For the execution of the alignment on board an instruction has to be created which lists the permissible gap and sag values for open flange connections respectively the "Jack-up" loads for measuring the bearing loads.

5.8.3 Before the installation of the propeller shaft the correct alignment of the stern tube bearings is to be checked.

The final alignment on board has to be checked by suitable methods in afloat condition in presence of the **TL** Surveyor.

5.9 Shaft locking devices

5.9.1 A locking device acc. to Section 2, E.2.4 has to be provided at each shaftline of multiple-shaft systems.

5.9.2 The locking device is at least to be designed to prevent the locked shaft from rotating while the ship is operating with the remaining shafts at reduced power. This reduced power has to ensure a ship speed that maintains the manoeuvring capability of the ship in full scope, in general not less than 8 kn.

5.9.3 If the locking device is not designed for the full power/speed of the remaining shafts, this operational restriction has to be recognizable for the operator by adequate signs.

5.10 Shaft earthing

Shaft earthing has to be provided according to Section 3, E, 5.5.

E. Balancing and Testing

1. General

The imbalance of the shafts, e.g. because of eccentric drilling hole of hollow shafts has to be within the quality range G 16 according to ISO 21940-11, as far as applicable.

2. Shaft Liners

Prior to fitting, shaft liners are to be subjected to a hydraulic tightness test at 2 bar pressure in the finish-machined condition.

3. Stern Tubes

Prior to fitting, cast stern tubes are to be subjected to a hydraulic tightness test at 2 bar pressure in the finish-machined condition. A further tightness test is to be carried out after fitting.

For stern tubes fabricated from welded steel plates, it is sufficient to test for tightness during the pressure tests applied to the hull spaces passed by the stern tube.

F. Special Requirements for Fibre Laminate Shafts

1. Theoretical Strength Calculation

The strength calculation must at least cover the following failure modi in conjunction with the given corresponding load cases:

- Statical failure

  Dimensioning to be performed against nominal torque with a safety of 3.

- Failure due to fatigue (high cycle)

  As far as the shaft is not exposed to bending stresses fatigue analysis may be carried out for nominal torque plus 30 % torsional vibration torque.

- Buckling failure mode

  Dimensioning may be estimated for a load of 3 times the nominal torque and in accordance to the formulas in 2.

For the strength analysis the nominal strength of the material has to be reduced by the factor 0,7 in order to compensate random influence factors such as geometrical and production inaccuracies as well as environmental factors (moisture, temperature).

The calculation of the stress may be performed on the basis of accepted analytical methods such as CLT (Classical Laminate Theory) or FEM models. With these stresses as input a set of failure modi in relation to fibre and interfibre failure must be checked. This set of failure modi must be coherent, i.e. a complete and accepted theory .

2. Buckling Failure

For shafts made of anisotropic materials, such as winded shafts of fibre laminate, buckling strength can be checked for the critical torque by the following formula:

$$M_{tcrit} = C_s \cdot \frac{\pi^3}{6000} \cdot \frac{r_m^{5/4} \cdot t^{9/4} \cdot E_x^{3/8}}{\ell^{0,5}} \cdot \left(\frac{E_y}{1 - \nu_{xy} \cdot \nu_{yz}} \right)^{5/8} \; [\mathrm{Nm}]$$

C_s = Factor depending on boundary conditions of support

= 0,800 for free ends

= 0,925 ends simply supported

E_x = Modulus of elasticity in x-direction [N/mm²]

E_y = Modulus of elasticity in transverse direction [N/mm²]

ℓ = Unsupported length of shaft [mm]

r_m, t = See C.5.

ν_{xy} = Poisson's ratio of the laminate in longitudinal direction

ν_{yx} = Poisson's ratio of the laminate in peripheral direction

The design criteria is:

$$3{,}5 \, . \, M_t \leq M_{tcrit}$$

M_t = Nominal torque at maximum continuous rating [Nm]

3. Experimental Strength Investigation

Experimental strength investigation has to be provided on request. Specifically:

- Testing of samples, if necessary for verification of material data

- Prototype testing/process checking for verification of the theoretical analysis in presence of a **TL** Surveyor

- After a year or 3000 operating hours, whichever is reached earlier, a visual examination and optionally a crack or delamination check of the fibre laminate components is to be carried out by a **TL** Surveyor.

4. Fire Protection

If fire protection requirements are relevant for composite shafting, specifically in the cases of penetration of fire protection bulkheads and/or redundant propulsion, appropriate provisions shall be taken to ensure the required properties in consent with **TL**.

5. Final Documentation

After finalising manufacturing of the components an updated documentation in the form of a list of all definitive valid analyses and documents is to be submitted to **TL**. The documentation must refer to the status quo and take into account all alterations or optimisations introduced during designing and manufacturing process as well as the achieved and measured properties.

Appendix 1 Special approval of alloy steel used for intermediate shaft material

1. Application

This appendix 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. 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_1 given by the formulae in Section 8, C.1.

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.

2.1 Test conditions

Test conditions are to be in accordance with Table 5.1. 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.

Test procedures are to be in accordance with Section 10 of ISO 1352.

Table 5.1 Test condition

Loading type	Torsion
Stress ratio	R= -1
Load waveform	Constant-amplitude sinusoidal
Evaluation	S-N curve
Number of cycles for test termination	1 x 10^7 cycles

2.2 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:

$$\tau_{c1} \geq \tau_{c\lambda-0} = \frac{\sigma_B + 160}{6} \cdot C_K \cdot C_D$$

$$\tau_{c2} \geq 1.7 \cdot \frac{1}{\sqrt{c_K}} \tau_{c1}$$

where

C_K = Factor for the particular shaft design features, see Section 8, C.1.

scf = Stress concentration factor, see below (*) (For unnotched specimen, 1,0.)

C_D = Size factor, see Section 8, C.1.

σ_B = Specified minimum tensile strength in N/mm^2 of the shaft material

(*) 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:

$$s_{cf} = \alpha_{t(hole)} + 0{,}8 \cdot \frac{(1-e)/d}{\sqrt{\left(1-\frac{d_i}{d}\right) \cdot \frac{e}{d}}}$$

This formula applies to:

- Slots at 120 or 180 or 360 degrees apart.

- Slots with semicircular 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.

3. Cleanliness requirements

The steels are to have a degree of cleanliness as shown in Table 5.2 when tested according to ISO 4967 method A. 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 **TL** Rules, Chapter 2, Section 5, Table 5.2, 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 **TL.**

Table 5.2 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

4. Inspection

The ultrasonic testing required by **TL** Rules, Chapter 2, Section 5 is to be carried out prior to acceptance. The acceptance criteria are to be in accordance with TL- G 68 or a recognized national or international standard.

SECTION 6

GEARS, COUPLINGS

A. General

1. Scope

1.1 These Rules apply to spur, planetary and bevel gears and to all types of couplings for incorporation in the main propulsion plant or essential auxiliary machinery as specified in Section 1, B.4. The design requirements laid down here may also be applied to gears and couplings of auxiliary machinery other than mentioned in Section 1, B.4.

1.2 Application of these requirements to the auxiliary machinery couplings mentioned in 1.1 may normally be limited to a general approval of the particular coupling type by **TL.** Regarding the design of elastic couplings for use in generator sets, reference is made to G.4.4.6.

1.3 For the dimensional design of gears and couplings with ice class, see Section 9.

2. Documents for Approval

Assembly and sectional drawings together with the necessary detail drawings and parts lists are to be submitted to **TL** in triplicate for approval. They must contain all the data necessary to enable the load calculations to be checked.

B. Materials

1. Approved Materials

1.1 Shafts, pinions, wheels and wheel rims of gears in the main propulsion plant should preferably be made of forged steel. Rolled steel bar may also be used for plain, flangeless shafts. Gear wheel bodies may be made of grey cast iron **(1)** or nodular cast iron or may be fabricated from welded steel plate with steel or cast steel hubs. For the material of the gearings the requirements according to ISO 6336, Part 5 are to be considered.

1.2 Couplings in the main propulsion plant must be made of steel, cast steel or nodular cast iron with a mostly ferritic matrix. Grey cast iron or suitable cast aluminium alloys may also be permitted for lightly stressed external components of couplings and the rotors and casings of hydraulic slip couplings.

1.3 The gears of essential auxiliary machinery according to Section 1, B.4. are subject to the same requirements as those specified in 1.1 as regards the materials used. For gears intended for auxiliary machinery other than that mentioned in Section 1, B.4. other materials may also be permitted.

1.4 Flexible coupling bodies for essential auxiliary machinery according to Section 1, B.4. may generally be made of grey cast iron, and for the outer coupling bodies a suitable aluminium alloy may also be used. However, for generator sets use should only be made of coupling bodies preferably made of nodular cast iron with a mostly ferritic matrix, of steel or of cast steel, to ensure that the couplings are well able to withstand the shock torques occasioned by short circuits. **TL** reserve the right to impose similar requirements on the couplings of particular auxiliary drive units.

2. Testing of Materials

All gear and coupling components which are involved in the transmission of torque and which will be installed in the main propulsion plant must be tested under surveillance of **TL** in accordance with the **TL** Material Rules and a **TL** certificate has to be provided. The same applies to the materials used for gear components with major torque transmission function of gears and couplings in generator drives.

Suitable proof is to be submitted for the materials used for the major components of the couplings and gears of all other functionally essential auxiliary machines in accordance with Section 1, B.4. This proof may take place by a Manufacturer Inspection Certificate of steelmaker.

(1) *The peripheral speed of cast iron gear wheels shall generally not exceed 60 m/s, that of cast iron coupling clamps or bowls, 40 m/s.*

C. Calculation of the Load-Bearing Capacity of Gear Theeth

1. General

1.1 Components of the gearing system are:

- Gear
- Equipment for the gear lubrication and control oil
- Cooling water equipment
- System for engaging/disengaging

1.2 Gears have to be designed to meet the sound level defined in the building specification. The following design principles for low-noise gears are to be considered:

- Elastic mounting of the gear directly on the ship structure
- Fixed mounting of gear and drive unit on an intermediate framing which is elastically mounted on the ship structure
- Flexible couplings to the drive unit and to the propeller shaft
- Use of low-noise types of toothing, such as double helical gearing
- Installation of low-noise components, e.g. rotary screw pumps, etc. for the gear lubrication. Installation of pumps with small suction height to avoid cavitation noise

2. Calculation of Load Capacity for Spur and Bevel Gears

2.1 General

2.1.1 The sufficient load capacity of the gear-tooth system of main and auxiliary gears in ship propulsion systems is to be demonstrated by load capacity calculations according to the international standards ISO 6336, ISO 9083 or DIN 3990 for spur gear tooth systems respectively ISO 10300 or DIN 3991 for bevel gears while maintaining the safety margins stated in Table 6.1 for flank and root load.

2.1.2 For gears in the main propulsion plant proof of the sufficient mechanical strength of the roots and flanks of gear teeth in accordance with the formulae contained in this Section is linked to the requirement that the accuracy of the teeth should ensure sufficiently smooth gear operation combined with satisfactory exploitation of the dynamic loading capacity of the teeth.

For this purpose, the magnitude of the individual pitch error f_p and of the total profile error F_f for peripheral speeds at the pitch circle up to 25 m/s shall generally conform to at least quality 5 as defined in DIN 3962 or 4 to ISO 1328, and in the case of higher peripheral speeds generally to at least quality 4 as defined in DIN 3962 or 3 to ISO 1328. The total error of the tooth trace fHß should conform at least to quality 5 to DIN 3962, while the parallelism of axis shall at least meet the requirements of quality 5 according to DIN 3964 or 4 according to ISO 1328 respectively.

Prior to running-in the surface roughness R_z of the tooth flanks of gears made by milling or by shaping shall generally not exceed 10 μm. In the case gears where the tooth profile is achieved by e.g. grinding or lapping, the surface roughness should generally not exceed 4 μm. The tooth root radius ρ_{ao} on the tool reference profile is to be at least $0{,}25 \cdot m_n$.

TL reserves the right to call for proof of the manufacturing accuracy of the gear-cutting machines used and for testing of the method used to harden the gear teeth.

3. Symbols, terms and summary of input data

3.1 Indices

1 = pinion
2 = wheel
m = in the mid of the face width
n = normal plane
t = transverse plane
o = tool

Table 6.1 Minimum safety margins for flank and root stress

Case	Application	Boundary conditions	S_H	S_F
1.1	Gearing in ship propulsion systems and generator drive systems	Modulus $m_n \le 16$	1,3	1,8
1.2		Modulus $m_n > 16$	$0{,}024\ m_n + 0{,}916$	$0{,}02\ m_n + 1{,}48$
1.3		In the case of two mutually independent main propulsion systems up to an input torque of 8000 Nm	1,2	1,55
2.1	Gears in auxiliary drive systems which are subjected to dynamic load		1,2	1,4
2.2	Gears in auxiliary drive systems used for dynamic positioning (Class Notation DK)		1,3	1,8
2.3	Gears in auxiliary drive systems which are subjected to static load	$N_L \le 10^4$	1,0	1,0

Note

*If the fatigue bending stress of the tooth roots is increased by special technique approved by **TL**, e.g. by shot peening, for case-hardened toothing with modulus $m_n \le 10$ the minimum safety margin S_F may be reduced up to 15 % with consent of **TL.***

3.2 Parameters

a = Centre distance, [mm]

b = Face width, [mm]

b_{eh} = Effective face width (bevel gears), [mm]

Bz_O = Measure for shift of datum line

d = Standard pitch diameter [mm]

d_a = Tip diameter, [mm]

d_f = Root diameter, [mm]

F_t = Circular force at reference circle, [N]

$F_{\beta x}$ = Initial equivalent misalignment, [µm]

f_{pe} = Normal pitch error, [µm]

f_f = Profile form error, [µm]

h_{a0}^* = Addendum coefficient of tool, [-]

h_{f0}^* = Dedendum coefficient of tool, [-]

h_{FfP0}^* = Utilized dedendum coefficient of tool, [-]

K_A = Application factor, [-]

$K_{F\alpha}$ = Transverse load distribution factor (root stress), [-]

$K_{F\beta}$ = Face load distribution factor (root stress), [-]

$K_{H\alpha}$ = Transverse load distribution factor (contact stress), [-]

$K_{H\beta}$ = Face load distribution factor (contact stress), [-]

$K_{H\beta\text{-}be}$ = Bearing factor (bevel gears), [-]

K_v = Dynamic factor, [-]

K_γ = Load distribution factor. [-]

m_n = Normal modul, [mm]

m_{nm} = Mean normal modul (bevel gear), [mm]

n = Number of revolutions, [min^{-1}]

N_L = Number of load cycles, [min^{-1}]

P = Transmitted power, [kW]

pr = Protuberance at tool, [mm]

Q = Toothing quality, acc. to DIN, [-]

q = Machining allowance, [mm]

R_a = Arithmetic mean roughness, [μm]

R_{zF} = Mean peak to valley roughness of root [μm]

R_{zH} = Mean peak to valley roughness of flank, [μm]

S_F = Safety factor against tooth breakage, [-]

S_{FG} = Tooth root stress limit, [N/mm^2]

S_H = Safety factor against pittings, [-]

T = Torque, [Nm]

u = Gear ratio, [-]

x = Addendum modification coefficient, [-]

x_{hm} = Mean addendum modification coefficient, [-]

x_{sm} = Thickness modification coefficient,(bevel gears) [-]

Y_F = Tooth form factor (root), [-]

Y_{NT} = Live factor (root), [-]

$Y_{\delta\ rel\ T}$ = Relative notch sensitivity factor, [-]

$Y_{R\ rel\ T}$ = Relative surface condition factor, [-]

Y_S = Stress correction factor, [-]

Y_{ST} = Stress correction factor for reference test gears, [-]

Y_X = Size factor for tooth root stress, [-]

Y_β = Helix angle factor for tooth root stress, [-]

z = Number of teeth, [-]

Z_E = Elasticity factor, [-]

Z_H = Zone factor (contact stress), [-]

Z_L = Lubricant factor, [-]

Z_{NT} = Live factor (contact stress), [-]

Z_v = Speed factor, [-]

Z_R = Roughness factor, [-]

ZW = Work-hardening factor, [-]

ZX = Size factor (contact stress), [-]

Z_β = Helix angle factor (contact stress), [-]

Z_ε = Contact ratio factor (contact stress), [-]

α_n = Normal pressure angle, [°]

α_{pr} = Protuberance angle, [°]

β = Helix angle, [°]

β_m = Mean helix angle (bevel gears), [-]

υoil = Oil temperature, [°C]

ν_{40} = Kinematic viscosity of the oil at 40°C [mm^2/s]

ν_{100} = Kinematic viscosity of the oil at 100°C [mm^2/s]

ρ_{a0}^* = Coefficient of tip radius of tool, [-]

Σ = Shaft angle (bevel gears), [°]

σ_F = Root bending stress, [N/mm^2]

σ_{FE} = Root stress, [N/mm^2]

σ_{FG} = Root stress limit, [N/mm^2]

σ_{F0} = Nominal root stress, [N/mm^2]

σ_{Flim} = Endurance limit for bending stress, [N/mm^2]

σ_{FP} = Permissible root stress, [N/mm^2]

σ_H = Calculated contact stress, [N/mm^2]

σ_{HG} = Modified contact stress limit, [N/mm^2]

σ_{Hlim} = Endurance limit for contact stress, [N/mm^2]

σ_{HP} = Permissible contact stress, [N/mm^2]

σ_{H0} = Nominal contact stress, [N/mm^2]

3.3 The input data required to carry out load-bearing capacity calculations are summarized in Table 6.2.

4. Influence Factors for Load Calculations

4.1 Rated torque

The calculation of the rated torque has to be based on the planned maximum continuous rating.

4.2 Application factor K_A

The application factor K_A takes into account the increase in rated torque caused by superimposed dynamical or impact loads. K_A is determined for main and auxiliary systems in accordance with Table 6.3.

4.3 Load distribution factor K_γ

The load distribution factor K_γ takes into account deviations in load distribution, e.g. in gears with dual or multiple load distribution or planetary gearing with more than three planet wheels.

The following values apply in respect of planetary gearing:

Gear with:

- up to 3 planet wheels $K_\gamma = 1{,}0$
- 4 planet wheels $K_\gamma = 1{,}2$
- 5 planet wheels $K_\gamma = 1{,}3$
- 6 planet wheels $K_\gamma = 1{,}6$

In gears which have no load distribution $K_\gamma = 1{,}0$ is applied.

For all other cases K_γ is to be agreed with **TL**.

4.4 Face load distribution factors $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors take into account the effects of uneven load distribution over the tooth flank on the contact stress ($K_{H\beta}$) and on the root stress ($K_{F\beta}$).

In the case of flank corrections which have been determined by recognized calculation methods, the KHß and KFß values can be preset. Hereby the special influence of ship operation on the load distribution has to be taken into account.

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

The transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$ take into account the effects of an uneven distribution of force of several tooth pairs engaging at the same time.

In the case of gears in main propulsion systems with a gear tooth system of a quality described in 2.1.2.

$$K_{H\alpha} = K_{F\alpha} = 1{,}0$$

can be applied. For other gears the transverse load distribution factors are to be calculated in accordance with DIN/ISO explicitly quoted under 2.1.1 and 2.1.2.

Table 6.2 List of input data for evaluating load-bearing capacity

Yard./Newb. No.						Reg. No.			
Manufacturer					Type				
Application					Cylindrical gear ☐		Bevel gear **(1)** ☐		
Nominal rated power	P			kW	Ice class				-
No. of revolutions	n_1			1/min	No. of planets				-
Application factor	K_A			-	Dynamic factor	K_v			-
Face load distribution factors	$K_{H\beta}$				Load distribution factor	K_γ			
	$K_{H\beta\text{-}be}$**(1)**			-	Transversal load distribution factors	$K_{H\alpha}$			
	$K_{F\beta}$			-		$K_{F\alpha}$			
Geometry Data		Pinion	Wheel		**Tool Data**		Pinion	Wheel	
Number of teeth	z		-	-	Addendum modification coefficient	x/x_{hm}			-
Normal modul	m_n/m_{nm} **(1)**			mm					-
Mean normal modul	m_{nm}**(1)**			mm	Thickness modification coeff.	x_{sm}**(1)**			-
Normal pres. angle	α_n			°	Coefficient of tool tip radius	ρ_{a0}^*			-
Centre distance	a			mm	Addendum coefficient of tool	h_{a0}^*			-
Shaft angle	Σ**(1)**			°	Dedendum coefficient of tool	h_{f0}^*			-
Relative effective face width	b_{eh}/b**(1)**			-	Utiliz. ded. coeff. of tool	h_{FfPO}^*			
Helix angle	β			°	Protuberance	pr			mm
Mean helix angle	β_m**(1)**			°	Protuberance angle	α_{pr}			°
Face width	b			mm	Machining allowance	q			mm
Tip diameter	d_a			mm	Measure at tool	Bz_O			mm
Root diameter	d_{fe}			mm	Backlash allowance/tolerance				-
Lubrication Data					**Quality**				
Kin.viskosity 40°C	ν_{40}			mm²/s	Quality acc. to DIN	Q			-
Kin.viskosity 100°C	ν_{100}			mm²/s	Mean peak to valley roughness of flank	R_{zH}			µm
Oil temperature	ν_{Oil}			°C	Mean peak to valley roughness of root	R_{zF}			µm
FZG load stage				-	Initial equivalent misalignment	$F_{\beta x}$			µm
Material Data					Normal pitch error	f_{pe}			µm
Material type					Profile form error	f_f			µm
Endurance limit for contact stress	σ_{Hlim}			N/mm²					
Endurance limit for bending stress	σ_{Flim}			N/mm²	Date :				
Surface hardness				HV					
Core hardness				HV	Signature :				
Heat treatment method				-					

(1) *Declaration for bevel gear.*

Table 6.3 Application factor K_A

System type	K_A
Main system:	
Turbines and electric drive systems	1,1
Diesel engine drive systems with fluid clutch between engine and gears	1,1
Diesel engine drive systems with highly flexible coupling between engine and gears	1,3
Diesel engine drive systems with no flexible coupling between engine and gears	1,5
Generator drives	1,5
Auxiliary system:	
Thruster with electric drive	1,1 (20000 h) **(1)**
Thruster drives with diesel engines	1,3 (20000 h) **(1)**
Windlasses	0,6 (300 h) **(1)** 2,0 (20 h) **(1)**
Combined anchor and mooring winches	0,6 (1000 h) **(1)** 2,0 (20 h) **(2)**

Note

(1) *Assumed operating hours*

(2) *Assumed maximum load of windlasses*

For other types of system K_A is to be stipulated separately.

5. Contact stress

5.1 The calculated contact stress σ_H shall not exceed the permitted flank stress σ_{HP} (Hertzian flank stress).

$$\sigma_H = \sigma_{Ho} \cdot \sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha}} \leq \sigma_{HP} \qquad (1)$$

$$\sigma_{Ho} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_i}{d_1 \cdot b} \cdot \frac{u+1}{u}}$$

5.2 The permissible contact stress σ_{HP} shall include a safety margin S_H as given in Table 6.1 against the contact stress limit σ_{HG} which is determined from the material-dependent endurance limit σ_{Hlim} as shown in Table 6.4 **(2)** allowing for the influence factors Z_{NT}, Z_L, Z_V, Z_R, Z_W, Z_X.

$$\sigma_{HP} = \frac{\sigma_{HG}}{S_H} \qquad (2)$$

$$\sigma_{HG} = \sigma_{Hlim} \cdot Z_{NT} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X$$

Table 6.4 Endurance limits (3) for contact stress σ_{Hlim}

Material	$\sigma_{H\,lim}$ [N/mm²]
Case-hardening steels, case-hardened	1500
Nitriding steels, gas nitrided	1250
Alloyed heat treatable steels, bath or gas nitrided	850-1000
Alloyed heat treatable steels, induction hardened	0,7HV10 + 800
Alloyed heat treatable steel	1,3HV10 + 350
Unalloyed heat treatable steel	0,9HV10 + 370
Structural steel	1,0 HB + 200
Cast steel, cast iron and nodular cast graphite	1,0 HB + 150

6. Tooth root bending stress

6.1 The calculated maximum root bending stress σ_F of the teeth shall not exceed the permissible root stress σ_{FP} of the teeth.

The tooth root stress is to be calculated separately for pinion and wheel.

$$\sigma_F = \sigma_{F0} \cdot K_A \cdot K_V \cdot K_\gamma \cdot K_{F\beta} \cdot K_{F\alpha} \leq \sigma_{FP} \qquad (3)$$

$$\sigma_{F0} = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta$$

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(2) *With consent of **TL** for case hardened steel with proven quality higher endurance limits may be accepted.*

(3) *With consent of **TL** for case hardened steel with or over proven quality application of higher values for fatigue strength may be accepted.*

Table 6.5 Endurance limit (2) for tooth root bending stress $\sigma_{FE} = \sigma_{Flim} \cdot Y_{ST}$, with Y_{ST} =2

Material	$\sigma_{FE}=\sigma_{Flim}^{*} \cdot Y_{ST}$ [N/mm²]
Case-hardened steels, case-hardened	860 - 920
Nitriding steels, gas nitrided	850
Alloyed heat treatable steel, bath or gas nitrided	740
Alloyed heat treatable steel, induction hardened	700
Alloyed heat treatable steels	0,8HV10 + 400
Unalloyed heat treatable steels	0,6 HV10 + 320
Structural steel	0,8HB+180
Cast steel, cast iron with nodular graphite	0,8HB+140
Note *For alternating stressed toothing only 70 % of these values are permissible.*	

6.2 The permissible root bending stress σ_{FP} shall have a safety margin S_F as indicated in Table 6.1 against the root stress limit σ_{FG} which is determined from the material-dependent fatigue strength σ_{FE} or σ_{Flim} accordance with Table 6.5 **(2),** allowing for the stress correction factors Y_{ST}, Y_{NT}, $Y_{\delta\,rel\,T}$, $Y_{R\,rel\,T}$, Y_X.

$$\sigma_{FP} = \frac{\sigma_{FG}}{S_F} \qquad (4)$$

$$\sigma_{FG} = \sigma_{F\,lim} \cdot Y_{ST} \cdot Y_{NT} \cdot Y_{\delta\,rel\,T} \cdot Y_{R\,rel\,T} \cdot Y_X$$

D. Gear Shafts

1. Minimum diameter

The dimensions of shafts of reversing and reduction gears are to be calculated by applying the following formula:

$$d \geq F \cdot k \cdot \sqrt[3]{\frac{P}{n \cdot \left[1 - \left(\frac{d_i}{d_a}\right)^4\right]}} \cdot C_W \qquad (5)$$

for $\frac{d_i}{d_a} \leq 0{,}4$ the expression

$\left[1 - \left(\frac{d_i}{d_a}\right)^4\right]$ may be set to 1,0

d = Required outside diameter of shaft, [mm]

d_i = Diameter of shaft bore for hollow shafts [mm]

d_a = Actual shaft diameter, [mm]

P = Driving power of shaft, [kW]

n = Shaft speed, [min⁻¹]

F = Factor for the type of drive, [-]

= 95 for turbine plants, electrical drives and engines with slip couplings,

= 100 for all other types of drive. **TL** reserve the right to specify higher F values if this appears necessary in view of the loading of the plant.

C_W = Material factor

$\frac{500}{R_m + 160}$ =

R_m = tensile strength of the shaft material

For whell shafts no higher value than 800 N/mm².shall be used. For pinion shafts the actual tensile strength may generally be substituted for R_m.

k = 1,10 for gear shafts

= 1,15 in the area of the pinion or wheel body is this is keyed to the shaft and for multiple-spline shafts.

Higher values of k may be specified by **TL** where increased bending stresses in the shaft are expected because of the bearing arrangement, the casing design, the tooth pressure, etc.

E. Equipment

1. Gear Lubrication

1.1 General

The gear system has to be designed to enable a start with a lubrication oil temperature from 0°C upwards without restrictions.

For engageable couplings the guidelines according to G.5. are valid.

Suitable equipment has to be provided, which limits the water content in the gear lubricant and the humidity within the gear in a way to exclude corrosion in the gear.

1.2 Oil level indicator

For monitoring the lubricating oil level in main and auxiliary gears, equipment must be fitted to enable the oil level to be determined.

1.3 Pressure and temperature control

Temperature and pressure gauges are to be fitted to monitor the lubricating oil pressure and the lubricating oil temperature at the oil-cooler outlet before it enters the gears.
Plain journal bearings are also to be fitted with temperature indicators.

Where gears are fitted with anti-friction bearings, a temperature indicator is to be mounted at a suitable point. For gears rated up to 2 000 kW, special arrangements may be agreed with **TL.**

Where ships are equipped with automated machinery, the requirements of Chapter 106 - Automation are to be complied with.

1.4 Lubricating oil pumps

Lubricating oil pumps driven by the gearing must be mounted in such a way that they are accessible and can be replaced easily by using common board available tools.

The supply of lubricating oil has to be ensured by a main pump and an independent stand by pump. If a reduction gear is approved for sufficient self lubrication at 75 % of the driving torque, the stand by pump may be abolished up to a performance relation P/n_1 [kW/min^{-1}]≤ 3,0.

2. Equipment for Operation, Control and Safety

Equipment for operation and control has to enable a safe operation of the gear by remote control as well as directly at the gear. All parameters which are important for a regular operation must be indicated directly at the gear. These indicating instruments shall be put together in an auxiliary control stand, as far as possible.

3. Gear Casings

The casings of gears belonging to the main propulsion plant and to essential auxiliaries must be fitted with removable inspection covers to enable the toothing to be inspected, the thrust bearing clearance to be measured and the oil sump to be cleaned.

4. Seating of Gears

It has to be taken care that no inadmissible forces caused by deformation of the foundation as part of the hull structure are transfered to the toothing.

The seating of gears on steel or cast resin chocks is to conform to the **TL** Rules , Guidelines for the Seating of Propulsion Plants.

For the seating of gears on casting resin chocks the thrust must be absorbed by stoppers. The same applies for casting resin foundations of separate thrust bearings.

F. Balancing and Testing

1. Balancing Quality

1.1 Gear wheels, pinions, shafts, gear couplings and, where applicable, high-speed flexible couplings must be assembled in a properly balanced condition.

1.2 The generally permissible residual imbalance U per balancing plane of gears for which static or dynamic balancing is rendered necessary by the method of manufacture and by the operating and loading conditions can be determined by applying the formula:

$$U=9.6\frac{Q \cdot G}{z \cdot n} \quad [\text{kgmm}] \qquad \textbf{(6)}$$

G = Mass of component to be balanced [kg]

n = Operating speed of component to be balanced [min-1]

z = Number of balancing planes [-]

Q = Degree of balance [-]

- 6,3 for gear shafts, pinion and coupling members for engine gears

- 2,5 for torsion shafts and gear couplings, pinions and gear wheels belonging to turbine transmissions

2. Testing

2.1 Testing in the manufacturer's works

When the testing of materials and component tests have been carried out, gearing systems for the main propulsion plant and for essential auxiliaries in accordance with Section 1, B.4 are to be presented to **TL** for final inspection and operational testing in the manufacturer's works.

The final inspection is to be combined with a trial run lasting several hours under part or full-load conditions, on which occasion the tooth clearance and contact pattern are to be checked. In the case of a trial at full-load, any necessary running-in of the gears must have been completed beforehand. Where no test facilities are available for the operational and on-load testing of large gear trains, these tests may also be performed on board ship on the occasion of the dock trials.

Tightness tests are to be performed on those components to which such testing is appropriate.

Reductions in the scope of the tests require the consent of **TL.**

2.2 Tests during sea trials

2.2.1 Prior to the start of sea trials, the teeth of the gears belonging to the main propulsion plant are to be coloured with suitable dye to enable the contact pattern to be established. During the sea trials, the gears are to be checked at all forward and reverse speeds to establish their operational efficiency and smooth running as well as the bearing temperatures and the freedom from contamination of the lubricating oil. At the latest on conclusion of the sea trials, the gearing is to be examined via the inspection openings and the contact pattern checked. If possible the contact pattern has to be checked after conclusion of every load step. Assessment of the contact pattern is to be based on the guide values for the proportional area of contact in the axial and radial directions of the teeth given in Table 6.6 and shall take account of the running time and loading of the gears during the sea trial.

2.2.2 In the case of multistage gear trains and planetary gears manufactured to a proven high degree of accuracy and if an appropriate check at the manufacturer's workshop was successful, checking of the contact pattern after sea trials may, with the consent of **TL**, be reduced in scope.

2.2.3 For checking of gears of Z-drive, e.g. rudder-propellers and azimuth propulsors, as main propulsion, see Section 7B, D.8..

2.3 Acoustic properties to be proven in compliance with Chapter 102 - Hull Structures and Ship Equipment, Section 16, B.3.

Table 6.6 Percentage area of contact

Material, manufacturing of toothing	Working tooth depth (without tip relief)	Width of tooth (without end relief)
heat-treated, milled, shaped	33% average	70%
surface-hardened, grinded, scarped	40% average	80%

G. Design and Construction of Couplings

1. General

It must be possible to disengage respectively to install and dismantle all couplings outside of gears using the tooling on board without displacing of major system's components such as gear, thrust bearing, engine etc.

2. Flange and Clamp-Type Couplings

In the dimensional design of the coupling bodies, flanges and bolts of flange and clamp-type couplings, the Rules specified in Section 5 are to be complied with.

3. Tooth Couplings

3.1 Torsionally stiff couplings, such as multi-tooth couplings may be used to compensate deviations in radial and axial direction.

3.2 Adequate loading capacity of the tooth flanks of straight-flanked tooth couplings requires that the following conditions are satisfied:

$$p = \frac{2{,}55 \cdot 10^7 \cdot P \cdot K_A}{b \cdot h \cdot d \cdot z \cdot n} \le p_{perm} \qquad (7)$$

p = Actual contact pressure of the tooth flanks [N/mm²]

P = Driving power at coupling, [kW]

d = Standard pitch diameter, [mm]

K_A = Application factor in accordance with C.4.2, [-]

z = Number of teeth, [-]

n = Speed in rev/min, [min^{-1}]

h = Working depth of toothing, [mm]

b = Load-bearing tooth width, [mm]

σ_{HP} = Permissible Hertzian stress, [N/mm²]

p_{perm} = 0,7 . R_{eH} for ductile steels [N/mm²]

p_{perm} = 0,7 . R_m for brittle steels [N/mm²]

Where methods of calculation recognized by **TL** are used for determining the Hertzian stress on the flanks of tooth couplings with convex tooth flanks, the permissible Hertzian stresses are equal to 75 % of the values of σ_{HP} shown in C.5.2 with influence factors Z_{NT} to Z_X set to 1,0:

p_{perm} =400 - 600 N/mm²
for tooth systems of quenched and tempered steel; the higher values apply to high tensile steels with superior tooth manufacturing and surface finish quality.

=800 - 1000 N/mm²
for toothing made of hardened steel (case or nitrogen). Higher values apply for superior tooth manufacturing and surface finish quality

3.3 The coupling teeth are to be effectively lubricated. For this purpose a permanent oil or grease lubrication in the coupling may generally be regarded as adequate where

$$d \cdot n^2 < 6 \cdot 10^9 \quad [\text{mm/min}^2] \qquad (8)$$

For higher values of $d \cdot n^2$, couplings in main propulsion plants are to be provided with a circulating lubrication oil system.

3.4 For the dimensional design of the sleeves, flanges and bolts of gear couplings the formulae given in Section 5 are to be applied.

4. Flexible couplings

4.1 Flexible couplings must be approved for the loads specified by the manufacturer and for use in main propulsion plants and essential auxiliary machinery. In general, flexible couplings shall be type-approved.

4.2 Documentation

The documentation to be submitted shall include:

- Assembly drawings
- Detailed drawings including material characteristics
- Definition of main parameters
  - rubber Shore hardness
  - nominal torque T_{KN}
  - permissible torque T_{Kmax1} for normal transient conditions like starts/stops, passing through resonances, electrical or mechanical engagements, ice impacts, etc.
  - permissible torque T_{Kmax2} for abnormal impact loads like short circuits, emergency stops, etc.
  - permissible vibratory torque $\pm T_{KW}$ for continuous operation
  - permissible power loss P_{KV} due to heat dissipation
  - permissible rotational speed n_{max}
  - dynamic torsional stiffness c_{Tdyn}, radial stiffness c_{rdyn}
  - relative damping ψ respectively damping characteristics
  - permissible axial, radial and angular displacement
  - permissible permanent twist
- Design calculations
- Test reports

4.3 Tests

The specifications mentioned in 4.2 are to be proven and documented by adequate measurements at test establishments.

For single approvals the scope of tests may be reduced by agreement with **TL**.

4.4 Design

4.4.1 With regard to casings, flanges and bolts the requirements specified in Section 5, D. are to be complied with.

4.4.2 The flexible element of rubber couplings shall be so designed that the average shear stress in the rubber/metal bonding surface relating to T_{KN} does not exceed a value of 0,5 N/mm^2.

4.4.3 For the shear stress within the rubber element due to T_{KN} it is recommended not to exceed a value subjected to the Shore hardness according to Table 6.7.

Higher values can be accepted if appropriate strengths of rubber materials have been documented by means of relevant tests and calculations.

Table 6.7 Limits of shear stress

Shore hardness [-]	Limit of shear stress [N/mm^2]
40	0,4
50	0,5
60	0,6
70	0,7

For special materials, e.g. silicon, corresponding limit values shall be derived by experiments and experiences.

4.4.4 Flexible couplings in the main propulsion plant and in power-generating plants shall be so dimensioned that they are able to withstand for a reasonable time operation with any one engine cylinder out of service, see Section 8, C.4.2. Additional dynamic loads for ships with ice class are to be taken into account according to Section 9.

4.4.5 If a flexible coupling is so designed that it exerts an axial thrust on the coupled members of the driving mechanism, provision shall be made for the absorption of this thrust.

If torsional limit devices are applicable, the functionality shall be verified.

4.4.6 Flexible couplings for diesel generator sets shall be capable of absorbing impact moments due to electrical short circuits up to a value of 6 times the nominal torque of the plant.

5. Clutches

5.1 General

5.1.1 Definition and application

Clutches are couplings which can be engaged and disengaged mechanically, hydraulically or pneumatically. The following requirements apply for their use in shaft lines and as integrated part of gear boxes. Clutches intended for trolling operation are subject to special consideration.

5.1.2 Documentation

For all new types of clutches a complete documentation has to be submitted to **TL** for approval. This documentation has to include e.g.:

- Assembly drawings
- Detail drawings of torque transmitting components including material properties
- Documentation of the related system for engaging/disengaging
- Definition of the following main technical parameters
  - Maximum and minimum working pressure for hydraulic or pneumatic systems [bar]
  - Static and dynamic friction torque [kNm]
  - Time diagram for clutching procedure
  - Operating manual with definition of the permissible switching frequency
- For special cases calculation of heat balance, if requested by **TL**

5.2 Materials

The mechanical characteristics of materials used for the elements of the clutch shall conform to the **TL** Material Rules.

5.3 Design requirements

5.3.1 Safety factors

For the connections to the shafts on both sides of the clutch and all torque transmitting parts the requirements of Section 5 have to be considered.

The mechanical part of the clutch may be of multiple disc type. The multiple disc package shall be kept free of external axial forces. All components shall be designed for static loads with a friction safety factor between 1,8 and 2,5 in relation to the nominal torque of the driving plant.

A dynamic switchable torque during engaging of 1,3 times the nominal torque of the driving plant has generally to be considered. In case of combined multiple engine plants the actual torque requirements will be specially considered.

5.3.2 Ice class

For clutches used for the propulsion of ships with ice class the reinforcements according to Section 9 have to be considered.

5.3.3 Measures for a controlled switching of the coupling and an adequate cooling in all working conditions have to be provided.

5.3.4 Auxiliary systems for engaging/ disengaging

If hydraulic or pneumatic systems are used to engage/ disengage a clutch within the propulsion system of a ship with a single propulsion plant an emergency operation shall be possible. This may be done by a redundant power system for engagement/ disengagement or in a mechanical way, e.g. by installing connecting bolts. For built-in clutches this would mean that normally the connecting bolts shall be installed on the side of the driving plant equipped with turning facilities.

The procedure to establish emergency service has to be described in the operating manual of the clutch and has to be executed in a reasonable time.

5.3.5 Controls and alarms

Local operation of remotely controlled clutches for the propulsion plants shall be possible.

The pressure of the clutch activating medium has to be indicated locally. Alarms according to the **TL** Rules for Automation, Section 9, E. have to be provided.

5.4 Tests

5.4.1 Tests at the manufacturer's works

Magnetic particle or dye penetrant inspection shall be applied for crack detection at surface hardened zones with increased stress level as well as at shrinkage surfaces. The manufacturer shall issue a Manufacturer Inspection Certificate.

Clutches for ship propulsion plants, for generator sets and transverse thrusters are to be presented to **TL** for final inspection and, where appropriate, for the performance of functional and tightness tests.

If a type approval is requested the requirements will be defined on a case by case basis by **TL** Head Office.

5.4.2 Tests on board

As part of the sea trials the installed clutches will be tested for correct functioning on board in presence of a **TL** Surveyor,

6. Hydraulic Couplings/Torque Converters

The torque characteristic of the hydraulic coupling has to be adjusted to the operating conditions. As hydraulic oil the lubricating oil required for the diesel engines or the gears shall be used.

While clutch is opened and reduction gear is in free condition, drive engine can operate in all speed range. The actual operation condition has to be indicated (for this see also Section 2).

7. Mechanical Clutches for Multi-Engine Synchronization

7.1 Basic description

For propulsion plants driven by multiple power sources of different type (such as diesel engines and turbines) generally a switching over to alternative operational modes without intermediate shut-down or reduction of the speed of the driving engine may be required. For such purposes a synchronisation of the speed of the engines in duty and the idling engine should take place before introduction of mechanical clutch in procedure.
The synchronisation aims to minimise the clutch in shock and induced peak torques, but also to enable an undisturbed and smooth continuous operation while changing the operational mode. The speed difference before introduction of clutch in procedure should not be more than 10%, or depending on the moment of inertias of the driving and driven parts the transient speed drop or increase after clutch in should be less than 5%.

7.2 Design requirements

The mechanical part of the clutch may be of multiple disc or mechanical teeth type. All components must be designed for the nominal transmitted torque with a safety factor of 2,5. In case that the speed is not synchronised in accordance to 7.1 a safety factor of 3,5 must be reckoned with. In case that multiple disc plates or other frictional devices are applied, they should be set in a way that the slip point is reached for torque values between 150 % and 250 % of the nominal torque, depending on the requirements of the manufacturer.

Additional functions, like lock-in/lock-out control or position indication may be required depending on the overall design of the propulsion plant.

8. Testing

Couplings for ship propulsion plants and couplings for generator sets and transverse thrusters are to be presented to **TL** for final inspection and, where appropriate, for the performance of functional and tightness tests.

SECTION 7 A

PROPELLER

A. General

1. Scope

These Rules apply to screw propellers (controllable and fixed pitch) as well as miscellaneous propulsion systems. See Section 9 for information on propeller sizes and materials for ships navigating in ice.

2. Documents for Approval

2.1 Design drawings of propellers in main propulsion systems having an engine output in excess of 300 kW and in transverse thrust systems of over 500 kW, are to be submitted to **TL** in triplicate for approval. The drawings are required to contain all the details necessary to carry out an examination in accordance with the following requirements.

2.2 In the case of controllable pitch propeller systems, general drawings and sectional drawings are to be submitted in addition to the design drawings for blade, boss and pitch control mechanisms. Control and hydraulic diagrams are to be submitted. In the case of new designs or controllable pitch propeller systems which are being installed for the first time on a vessel with **TL** Class, a description of the controllable pitch propeller system has also to be provided.

3. Design

3.1 Propellers are to be designed in a way that their power consumption lies within the family of characteristics of the driving machinery.

- The propeller of controllable pitch plants shall absorb the continuous rating of the driving machinery at nominal speed.

- A fixed pitch propeller shall reach the absorbed power/speed relation defined in the building specification.

3.2 For determination of the propeller parameters well known and established methods may be used. The lay out of the propeller has to consider the noise requirements in the building specification, e.g. minimum circumferential speed, etc. Nominal speed ahead and propeller diameter are to be selected for a maximum ship speed without cavitation. The limit curve for cavitation noise in the Sigma n - C_{Th} diagram in J. may be used. The noise emission from the propeller shall be estimated already in the concept phase.

3.3 Noise behaviour

3.3.1 Cavitation noise

The propeller has to be designed to develop low noise, especially at the nominal acoustic operation point. Cavitation noise has to be avoided.

Note

Guiding values for cavitation avoidance are contained in J.

3.3.2 Singing behaviour

By calculating the natural frequencies of the propeller and by a comparison with the hydrodynamic excitation at the trailing edges the risk of singing of the propeller has to be estimated and minimized. If singing is observed during trials, relevant counter measures have to be applied.

Note:

The singing of a propeller is a strong peak tone in the noise spectrum. It is created by the excitation of natural frequencies as a consequence of vortex shedding at the trailing edge.

3.3.3 Air blow-out device

3.3.3.1 Blow-out devices for air may be considered for naval ships for which low-noise operation is required in velocity ranges with cavitating propeller. If an air blow-out device is required in the building specification, the system has to be agreed with the Naval Authority and **TL.** The details of the design have to be included in the building specification.

3.3.3.2 The exits of the blown-out air have to be arranged in a way that a compact air veil covers the blade surface. The feed of the air into the propulsion system should be done at the front end of the propeller shaft. The air temperature at the shaft entrance should not be higher than 40°C. By special measures in the construction it has to be secured that no air penetrates into the hydraulic system. On the other hand no oil or water shall get into the air system.

3.3.3.3 For the creation of an effective air veil at the propeller blade the required air volume is depending on the propeller speed. Therefore speed regulated air compressors have to be provided.

3.3.3.4 All components and pipes of the air system have to be made of stainless steel.

3.3.3.5 For ships with NBC protection the air should be sucked from outside of the citadel. Ventilation and drainage pipes shall be conducted in closed form to the outside.

3.3.4 Measurements of water-borne noise

3.3.4.1 The measurements of water-borne noise have to be executed in deep and shallow water according to the building specification.

3.3.4.2 For the assessment of cavitation noise the following test criteria will be applied simultaneously:

- Recording audible noise in the frequency range 30 Hz - 20 kHz
- Comparative evaluation of the relevant third filter analyses. Checking if a clear increase of noise level in the complete frequency range above 1 kHz is to be observed or if with increased speed a transfer of the noise peak to the range of 100 Hz is happening
- Creation of a DEMON ("Demolition of Envelop Modulation on Noise") spectrum
- Measurement of the target level with a linear antenna

3.3.4.3 The execution of the measurements and the documentation of results has to be coordinated with **TL.**

4. Designation

Each propeller and the essential components for torque transmission and for blade adjustment of controllable pitch propellers respectively have to be definitely marked with the appropriate steel-stamp. The markings have to be executed by steel-stamp numbers with rounded edges to avoid notch effects.

B. Materials

1. Propellers and Propeller Hubs

The material for the propulsion device has to be selected according to the actual functional requirements. Special materials are not prescribed

Materials and their parameters may be taken from the **TL** Material Rules.

Metallic propellers are to be made of sea-water-resistant copper cast alloys or steel cast alloys with a minimum tensile strength R_m of 440 N/mm^2 and with sufficient bending fatigue strength.

For the purpose of the following design Rules governing the thickness of the propeller blades, the requisite resistance to seawater of a cast copper alloy or cast steel alloy is considered to be achieved if the alloy can withstand a fatigue test under alternating bending stresses comprising 10^8 load cycles amounting to about 20 % of the minimum tensile strength and carried out in a 3 % NaCl solution, and if it can be proven that the fatigue strength under alternating bending stresses in natural seawater is not less than about 65 % of the values established in 3 % NaCl solution. Sufficient fatigue strength under alternating bending stresses must be proven by a method recognized by **TL.**

2. Components for Controllable Pitch Propellers and Assembled Fixed Pitch Propellers

The materials of the major components of the pitch control mechanism and also the blade and boss retaining bolts must comply with the **TL** Material Rules.

The blade retaining bolts of assembled fixed pitch propellers or controllable pitch propellers are to be manufactured of seawater-resistant materials if they are not protected against contact with seawater.

3. Material Parameters

The material has to be documented according to the **TL** Material Rules, Sections 12.

If materials shall be used which are not corresponding to the **TL** grades, the following parameters have to be delivered to **TL**:

- Designation of the material (acronym)
- Chemical composition
- Tensile strength, yield stress and elongation of a material sample (serves especially for the identification of the material)
- Impact energy, if required
- Bending fatigue strength in sea water spray fog (serves as limit value for the strength calculations)
- Density of material, modulus of elasticity, thermal expansion coefficient
- Magnetic characteristics, if applicable
- Information concerning repairing ability (weldability, heat treatment)
- Information concerning resistance against corrosion and erosion

4. Novel Materials

Where propeller materials with not sufficient experience for their reliability are applied, the suitability has to be proven particularly to **TL.**

5. Material Testing

The material of propellers, propeller bosses and all other major components involved in the transmission of torque and pitch setting is to be tested in accordance with the **TL** Material Rules. This also applies to components which are used to control the blades and also to propellers in main propulsion systems smaller than 300 kW and transverse thrust systems of less than 500 kW.

C. Design and Dimensioning of Propellers

Alternative design methods which guarantee the same sufficient safety may be submitted to **TL** for approval.

1. Symbols and Terms

A = Effective area of a shrink fit, [mm^2]

A_D = Propeller plane area [m^2]

B = Developed blade width of cylindrical sections at radii 0,25R, 0,35R and 0,6R, [mm]

c_A = Coefficient for shrunk joints, [-]

= 1,0 for geared diesel engine and turbine plants as well as for electric motor drives,

= 1,2 for direct diesel engine drives,

C_G = Size factor in accordance with formula (2), [-]

C_{Dyn} = Dynamic load factor in accordance with formula (3), [-]

C_w = Characteristic value for propeller material as shown in Table 7A.1 corresponds to the minimum tensile strength R_m of the propeller material where sufficient fatigue strength under alternating bending stresses according to B.1 is proven [-].

C = Conicity of shaft ends, [-]

$$= \frac{\text{difference in taper diameter}}{\text{length of taper}}$$

C_{Th} = Thrust load coefficient

$$= \frac{T}{0{,}5 \cdot \rho \cdot v_A^2 \cdot A_D}$$

d = Pitch circle diameter of blade or propeller-fastening bolts, [mm]

Table 7A.1 Characteristic values C_w

Material	Description (1)	C_w
Cu 1	Cast manganese brass	440
Cu 2	Cast manganese nickel brass	440
Cu 3	Cast nickel aluminium bronze	590
Cu 4	Cast manganese aluminium bronze	630
Fe 3	Martensitic cast chrome steel 13/1-6	660
Fe 4	Martensitic-austenitic cast steel 17/4	600
Fe 5	Ferritic-austenitic cast steel 24/8	600
Fe 6	Austenitic cast steel 17/8-11	500
(1) *For the chemical composition of the alloys, see **TL** Rules - Materials*		

d_i = Diameter of shaft bore [mm]

d_k = Root diameter of blade or propeller-fastening bolts, [mm]

D = Diameter of propeller, [mm]

= 2 · R

d_m = Mean taper diameter, [mm]

D_N = Mean outer diameter of propeller hub [mm]

e = Blade rake acc. Fig. 7A.1 [mm]

= R · tanε

E_N = Modulus of elasticity for hub material [N/mm^2]

E_W = Modulus of elasticity for shaft material [N/mm^2]

E_T = Thrust stimulating factor in accordance with formula (5). [-]

f, f_1, f_2 = Factors in formulae (2), (4) and (11), [-]

H = Pressure side pitch of propeller blade at radii 0,25R, 0,35R and 0,6R [mm]

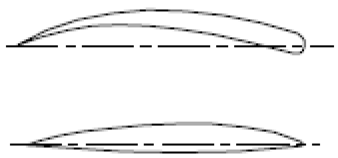
H_m = Mean effective pressure side for pitch varying with the radius, [mm]

$$= \frac{\sum(R \cdot B \cdot H)}{\sum(R \cdot B)}$$

R, B and H are the corresponding measures of the various sections.

k = Coefficient for various profile shapes in accordance with Table 7A.2, [-]

Table 7A.2 Values of k for examples of various profile shapes

Profile shape	k		
	0,25 R	**0,35 R**	**0,6 R**
Segmental profiles with circular arced suction side 	73	62	44
Segmental profiles with parabolic suction side 	77	66	47
Blade profiles as for Wageningen B series propellers 	80	66	44

K_N = Diameter ratio of hub [-] = d_m/D_N

K_W = Diameter ratio of shaft [-] = d_i/d_m

L = Pull-up length of propeller on cone, [mm]

L_{act} = Chosen pull-up distance [mm]

L_M = 2/3 of the leading-edge part of the blade width at 0,9R, but at least 1/4 of the total blade width at 0,9R for propellers with high skew blades [mm].

L_{mech} = Pull-up length at t = 35°C [mm]

L_{temp} = Temperature-related portion of pull-up length at t < 35°C [mm]

n_2 = Rotational speed of propeller at MCR[min^{-1}]

p = Surface pressure in the shrink joint between propeller and shaft [N/mm^2]

p_{act} = Surface pressure in the shrink joint at L_{act} [N/mm^2]

p_L = Local pressure at the propeller blade surface [N/mm^2]

p_S = Static pressure at the propeller axis of rotation, [N/mm^2]

p_V = Vapour pressure, [N/mm^2]

p_w = Nominal power of driving engine at MCR [kW]

Q = Peripheral force at mean taper diameter, [N]

$R_{p0,2}$ = 0,2% proof stress of propeller material, [N/mm^2]

R_{eH} = Minimum nominal upper yield strength, [N/mm^2]

R_m = Tensile strength, [N/mm^2]

r = Filet radius, [mm]

S = Margin of safety against propeller slipping on cone = 2,8, [-]

$SIGMA_n$ = Cavitation inception number

t = Maximum blade thickness of developed cylindrical section at radii 0,25R ($t_{0,25}$), 0,35R ($t_{0,35}$), 0,6R ($t_{0,6}$) and 1,0 R ($t_{1,0}$) [mm]

T = Propeller thrust, [N]

T_M = Impact moment, [Nm]

v_A = Average water velocity to the propeller [m/s]

v_S = Speed of ship at MCR, [kn]

w = Wake fraction, [-]

$W_{0,35R}$, $W_{0,6R}$ = Section modulus of cylindrical section at radii 0,35R and 0,6R, [mm^3]

W_x = Section modulus of cylindrical section at the radius x [mm^3]

Z = Total number of bolts used to retain one blade or propeller, [-]

z = Number of blades, [-]

α = Pitch angle of profile at radii 0,25R, 0,35R and 0,6R, [-]

$$\alpha_{0,25} = \arctan \frac{1,27 \cdot H}{D}$$

$$\alpha_{0,35} = \arctan \frac{0,91 \cdot H}{D}$$

$$\alpha_{0,6} = \arctan \frac{0,53 \cdot H}{D}$$

α_A = Tightening factor for retaining bolts depending on the method of tightening used (see VDI 2230 or equivalent standards) [-].

Guidance values:

= 1,2 for angle control

= 1,3 for bolt elongation control

= 1,6 for torque control

σ_N = Coefficient of linear thermal expansion of hub material [1/°C]

σ_W = Coefficient of linear thermal expansion of shaft material [1/°C]

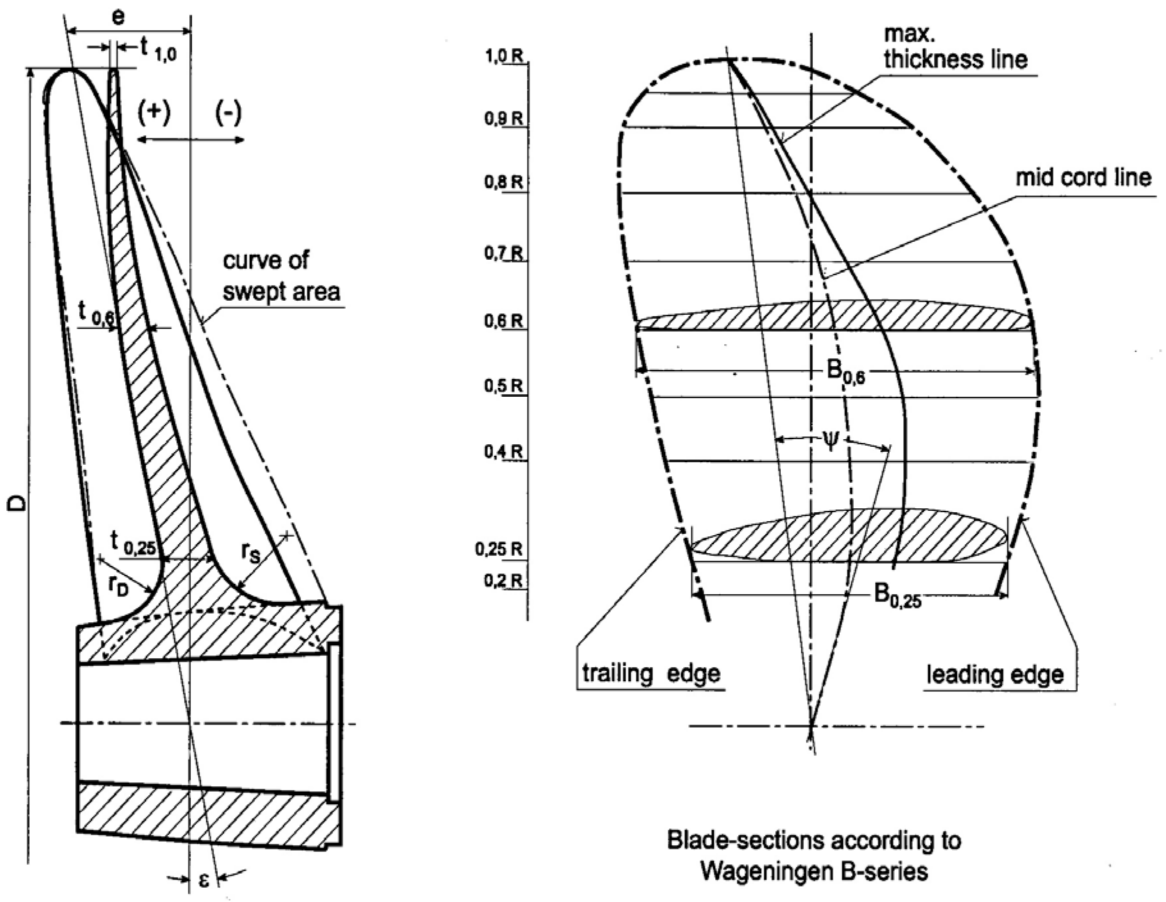


Figure 7A.1 Illustration of blade geometry

ε = Angle between lines of face generatrix and normal [-]

θ = Half-conicity of shaft ends, [-]

= C / 2

μ_o = Coefficient of static friction, [-]

= 0,13 for hydraulic oil shrunk joints

= 0,15 for dry fitted shrink joints - bronze/steel

= 0,18 for dry shrunk joints steel/steel

ν_N = Poisson's ratio of hub material [-]

ν_W = Poisson's ratio of shaft material [-]

Ψ = Skew angle acc. to Fig. 7A.1, [°]

ρ = Density of water, [kg/m^3]

σ_{max}/σ_m = Ratio of maximum to mean stress at pressure side of blades [-]

σ_V = von Mises equivalent stress [N/mm^2]

2. Calculation of blade thickness

2.1 At radii 0,25 R ($t_{0,25}$) and 0,6 R ($t_{0,6}$), the blade thicknesses of fixed pitch propellers must, as a minimum requirement, comply with the formula (1):

$$t = K_o \cdot k \cdot K_1 \cdot C_G \cdot C_{Dyn} \qquad (1)$$

$$K_o = 1 + \frac{e \cdot \cos\alpha}{H} + \frac{n_2}{15000}$$

k = as in Table 7A.2

= k' for other profiles as defined in Table 7A.2.

$$k' = k \cdot \sqrt{\frac{\beta_x}{\beta_x'}}$$

β_x = factor for the section modulus of the cylindrical section related to the pitch line of the blade for profile shapes defined in Table 7A.2

β_x' = factor for the section modulus of the cylindrical section related to the pitch line of the blade for profile shapes other than defined in Table 7A.2

$$= \frac{W_x}{t^2 \cdot B}$$

$$K_1 = \sqrt{\frac{P_W \cdot 10^5 \cdot [2 \cdot (D/H_m) \cdot \cos\alpha + \sin\alpha]}{n_2 \cdot B \cdot z \cdot C_W \cdot \cos^2\varepsilon}}$$

C_G = Size factor, [-]

$$= 1{,}1 \geq \sqrt{\frac{f_1 + 0{,}001D}{12{,}2}} \geq 0{,}85 \qquad (2)$$

f_1 = 7,2 for monobloc propellers,

= 6,2 for separately casted blades of variable-pitch or built-up propellers,

C_{Dyn} = Dynamic load factor, [-]

$$= \sqrt{\frac{(\sigma_{max}/\sigma_m - 0{,}8)}{0{,}7}} \geq 1{,}0 \qquad (3)$$

for $\dfrac{\sigma_{max}}{\sigma_m} > 1{,}5$ otherwise

= 1,0

σ_{max}/σ_m is generally to be taken from the detailed calculation according to 2.5. If, in exceptional cases, no such calculation exists, the stress ratio may be calculated approximately according to formula (4)

$$\frac{\sigma_{max}}{\sigma_m} = f_2 \cdot E_T + 1 \quad \text{with} \qquad (4)$$

$$E_T \approx \frac{4{,}3 \cdot 10^{-9} \cdot \nu_s \cdot n_2 \cdot (1 - w) \cdot D^3}{T} \qquad (5)$$

f_2 = 0,4 - 0,6 for single-screw ships, the lower value applies to stern shapes with a wide propeller tip clearance and no rudder heel, and the larger value to sterns with little clearance and with rudder heel. Intermediate values are to be selected accordingly.

= 0,2 for twin-screw ships

2.2 The blade thicknesses of controllable pitch propellers are to be determined at radii 0,35 · R and 0,6 · R by applying formula (1).

For ships other than tugs, trawlers, etc. the diameter/pitch ratio D/H_m applicable to open-water navigation at maximum engine power (MCR = Maximum Continuous Rating) can be used in formula (1).

2.3 The blade thicknesses calculated by applying formula (1) are minima for the finish-machined propellers.

2.4 The fillet radii at the transition from the pressure and suction side of the blades to the propeller boss should correspond, in the case of three and four-bladed propellers, to about 3,5 % of the propeller diameter. For propellers with a larger number of blades the maximum fillet radii allowed by the propeller design should be aimed at, and the radii shall not in any case be made smaller than 40 % of the blade root thickness.

Variable fillet radii which are aiming at a uniform stress distribution, may be applied if an adequate proof of stress is given case by case. The resulting calculated maximum stress shall not exceed the values occurring from a design with constant fillet radius in accordance with the first paragraph of 2.4.

2.5 For special designs such as propellers with skew angle Ψ ≥ 25°, tip fin propellers, special profiles etc, special mechanical strength calculations are to be submitted to **TL.**

A geometry data file and details on the measured make are to be submitted to **TL** together with the design documents to enable the evaluation of the blade stress of special designs to be carried out.

2.6 If the propeller is subjected to an essential wear, e.g. by abrasion in tidal flats or rivers, a wear addition has to be provided to the thickness determined under 2.1 to achieve an equivalent life time. If the actual thickness in service drops below 50 % at the blade tip or 90 % at other radii of the rule thickness obtained from 2.1, effective counter measures have to be taken. For unconventional blade geometries as defined in 2.5, the design thickness as shown on the approved drawings replaces the thickness requested according to 2.1.

D. Controllable Pitch Propellers

1. General

For multi-shaft propulsion plants separated hydraulic systems have to be provided for each controllable pitch propeller.

2. Design

For the design of the components, the following aspects have to be considered.

2.1 The adjustment of the controllable pitch propeller has to be done in a way that at a position "zero" of the operating lever and minimum operation speed, zero thrust is developed. If deviations because of adjustment tolerances are occurring, only thrust ahead shall be created.

Note

The maximum adjustable pitch ahead shall be at least 108 % of the design pitch [mm], the maximum pitch astern shall be at least 70 % of the design pitch [mm].

2.2 The construction of controllable pitch propeller plants has to be flood safe, if required in the building specification.

2.3 The installation and the dismantling of the controllable pitch propellers shall be possible without axial displacement of the shafts. The leading edge of the blade shall lead without overhang to the blade root disc.

2.4 The hub shall be tightened reliably. The space before the blade gaskets shall stay under constant pressure to avoid pressure and volume variations at the gaskets during pitch variation. A device has to be provided which allows flushing of the hub content at the floating ship. The blade position for this procedure has to be marked at a flange of the propeller shaft in the hull with "P" (purging).

2.5 The following blade positions of each blade have to be marked permanently at the hub:

- Maximum pitch ahead
- Design pitch
- Zero thrust position
- Maximum pitch ahead and astern

3. Emergency Control

3.1 Controllable pitch propeller plants are to be equipped with means for emergency control to maintain the function of the controllable pitch propeller in case of failure of the remote control system. It is recommended to provide a device enabling the propeller blades to be locked in the “ahead” setting position.

This pitch has to be selected in a way that a start of the propulsion system is possible with the weakest driving machinery and at a standstill of the ship. Afterwards it shall be possible to operate the system like a fixed pitch propeller.

3.2 Suitable devices have to prevent that an alteration of the blade pitch setting can lead to an overload or stall of the propulsion engine.

It has to be ensured that, in the event of failure of the control system, the setting of the blades

- Does not change, or

- Drifts to a final position slowly enough to allow the emergency control system to be put into operation

4. Hydraulic Control Equipment

4.1 Where the pitch-control mechanism is operated hydraulically, two mutually independent, power driven pump sets are to be fitted. For propulsion plants up to 200 kW, one power-driven pump set is sufficient provided that, in addition, a hand-operated pump is fitted for controlling the blade pitch and that this enables the blades to be moved from ahead to the astern position in a short enough time.

The selection and arrangement of filters has to ensure an uninterrupted supply with filtered oil, also during filter cleaning or exchange. In general, main filters are to be arranged on the pressure side directly after the pump. An additional coarse filtration of the hydraulic oil at the suction side, before the pump, should be provided.

For all operating conditions the adjusting time between design pitch and maximum astern pitch shall be defined in building specification. Guidance values are:

- 22 s maximum for propellers with a diameter D≤3,0 m

- 30 s maximum for propellers with a diameter D>3,0 m

4.2 The lay-out of the hydraulic system shall ensure that the electrically driven pumps are activated in case that:

- The mechanically driven pump fails

- Parallel operation with the mechanically driven pump in the lower speed range is required

- Short adjustment times are necessary, e.g. at manoeuvring operation

4.3 Each pump for the control oil has to be designed for an suitable angle velocity for blade adjustment a guide is 2,5° per second. This velocity is valid for:

- Operation with the electrically driven pump for the complete speed range

- Operation with the mechanically driven pump at nominal speed of the propeller

4.4 For hydraulic pipes and pumps Chapter 107 - Ship Operation Installations and Auxiliary Systems, Section 8 has to be applied.

5. Pitch Control Mechanism

For the pitch control mechanism proof is to be furnished that the individual components when subjected to impact loads still have a safety factor of 1,5 against the yield strength of the materials used. The impact moment T_M has to be calculated according to formula (6) and the resulting equivalent stresses at the different components are to be compared with their yield strength.

$$T_M = 1{,}5 \frac{R_{P\,0,2} \cdot W_{0,6R}}{\sqrt{\left(\frac{0{,}15\,D}{L_M}\right)^2 + 0{,}75}} .10^{-3} \qquad (6)$$

$W_{0,6R}$ can be calculated by applying the formula (7).

$$W_{0,6R} = 0{,}11\,(B \cdot t^2)_{0,6R} \qquad (7)$$

The components have to be designed fatigue-resistant for the maximum pressure.

6. Blade Retaining Bolts

6.1 The blade retaining bolts shall be designed in such a way as to withstand the forces induced in the event of plastic deformation of the blade at 0,35 R caused by a force acting on the blade at 0,9 R. The bolt material shall have a safety margin of 1,5 against its minimum nominal upper yield stress.

The thread core diameter of the blade retaining bolts shall not be less than

$$d_k = 2{,}6 \cdot \sqrt{\frac{M_{0,35R} \cdot \alpha_A}{d \cdot Z \cdot R_{eH}}} \qquad (8)$$

$M_{0,35R} = W_{0,35R} \cdot R_{p0,2}$

$W_{0,35R}$ may be calculated analogously to formula (7) or (9).

For nearly elliptically sections at the root area of the blade the following formula may be used instead:

$$W_{0,35R} = 0{,}10 \cdot (B \cdot t2)_{0,35R} \qquad (9)$$

6.2 The blade retaining bolts are to be tightened in a controlled manner in such a way that the tension on the bolts is about 60 - 70 % of their minimum nominal upper yield stress.

The shank of blade retaining bolts may be designed with a minimum diameter equal to 0,9 times the root diameter of the thread.

6.3 Blade retaining bolts must be secured against unintentional loosening.

7. Indicators

7.1 Controllable pitch propeller systems are to be provided with a direct acting indicator inside the engine room showing the actual setting of the blades. Further blade position indicators are to be mounted on the bridge and in the machinery control centre (see also **TL** Rules for Automation, Section 4 and Electrical Installations, Section 9, B.).

7.2 Hydraulic pitch control systems are to be provided with means to monitor the oil level. A pressure gauge for the pitch control oil pressure is to be fitted. A suitable indicator for filter clogging shall be provided. An oil temperature indicator is to be fitted at a suitable position.

Where ships are equipped with automated machinery, the requirements of **TL** Rules for Automation are to be complied with.

E. Propeller Mounting

1. Tapered Mountings for Fixed Propellers

1.1 Where the cone connection between the shaft and the propeller is fitted with a key, the propeller is to be mounted on the tapered shaft in such a way that approximately 120 % of the mean torque can be transmitted from the shaft to the propeller by friction.

Keyed connections are in general not to be used in installations with a barred speed range..

1.2 Where the connection between propeller shaft cone and propeller is realised by the hydraulic oil technique without the use of a key, the necessary pull-up distance L on the tapered shaft is to be determined according to formula (10). Where appropriate, allowance is also to be made for surface smoothing when calculating L.

$$L = L_{mech} + L_{temp} \qquad (10)$$

where L_{mech} is determined according to the formulae of elasticity theory applied to shrunk joints for a specific pressure p [N/mm²] at the mean taper diameter found by applying formula (11) and for a water temperature of 35 °C .

$$p = \frac{\sqrt{\theta^2 \cdot T^2 + f \cdot (c_A^2 \cdot Q^2 + T^2)} - \theta \cdot T}{A \cdot f} \qquad (11)$$

T has to be introduced as positive value if the propeller thrust increases the surface pressure at the taper. Change of direction of propeller thrust is to be neglected as far as absorbed power and thrust are essential less.

T has to be introduced as negative value if the propeller thrust reduces the surface pressure at the taper, e.g. for tractor propellers.

$$f = (\mu_o / S)^2 - \theta^2 \qquad (11a)$$

$$L_{temp} = (d_m / C) \cdot (\alpha_N - \alpha_W) \cdot (35 - t) \qquad (12)$$

t = The temperature at which the propeller is mounted. [°C]

Values for α_N and α_W can be taken from Table 7A.3.
At least the temperature range between 0 °C and 35 °C has to be considered..

1.3 For keyless propeller fittings without intermediate sleeve, the required pull-up distance and related stresses in the propeller hub and shaft can be calculated as follows.
Joint stiffness factor:

Values for E_N, E_W, ν_N and ν_W can be taken from Table 7A.3.

Minimum required pull-up distance at mounting temperature 35 °C:

$$K_{el} = \frac{d_m}{C} \cdot \left[\frac{1}{E_N} \cdot \left(\frac{1+K_N^2}{1-K_N^2} + \nu_N \right) + \frac{1}{E_W} \cdot \left(\frac{1+K_W^2}{1-K_W^2} - \nu_W \right) \right]$$

$L_{mech} = p \cdot K_{el}$

Minimum required pull-up distance at mounting temperature t [°C]:

$L = L_{mech} + L_{temp}$

Surface pressure at the mean taper diameter at chosen pull-up distance L_{act} [mm]:

$p_{act} = L_{act} / K_{el}$

Related von Mises' equivalent stresses:

$$\sigma_V = \frac{p_{act}}{1 - K_N^2} \cdot \sqrt{3 + K_N^4} \text{ (hub)}$$

$\sigma_V = p_{act}$ (solid shaft)

$\sigma_V = p_{act} \cdot 2 / (1 - K_W^2)$ (hollow shaft)

1.4 The von Mises' equivalent stress resulting from the maximum surface pressure p and the tangential stress in the bore of the propeller hub shall not exceed 75 % of the 0,2 % proof stress or yield strength of the propeller material in the installed condition and 90 % during mounting and dismounting..

1.5 The cones of propellers which are mounted on the propeller shaft with the aid of the hydraulic oil technique should not be more than 1 : 15 and not be less than 1 :25. For keyed connections the cone shall not be steeper than 1:10.

Table 7A.3 Material values

Material	Modulus of elasticity E [N/mm²]	Poisson's ratio ν [-]	Coefficient of linear thermal expansion α [1/°C]
Steel	205000	0,29	$12{,}0 \cdot 10^{-6}$
Copper based alloys CU1 and CU2	105000	0,33	$17{,}5 \cdot 10^{-6}$
Copper based alloys CU3 and CU4	115000	0,33	$17{,}5 \cdot 10^{-6}$
Note *For austenitic stainless steel see manufacturer's specification*			

1.6 The propeller nut must be strongly secured to the propeller shaft.

2. Flange connections

2.1 Flanged propellers and the hubs of controllable pitch propellers are to be connected by means of fitted pins and retaining bolts (preferably necked down bolts).

2.2 The diameter of the fitted pins is to be calculated by applying formula (4) given in Section 5, D.4.3.

2.3 The propeller retaining bolts are to be designed according to D.6., however the thread core diameter shall not be less than

$$d_k = 4{,}4. \sqrt{\frac{M_{0,35R} \cdot \alpha_{A\cdot}}{d \cdot Z \cdot R_{eH}}}$$

2.4 The propeller retaining bolts have to be secured against unintentional loosening.

F. Balancing and Testing

1. Balancing

All propellers including monoblock propellers ready for mounting as well as the blades of controllable and built up fixed pitch propellers are required to undergo static balancing in accordance with specified ISO 484 tolerance class (or equivalent) as specified in the approved drawing in presence of a surveyor. The mass difference between blades of controllable or built-up fixed pitch propellers has to be not more than 1,5 %.

Dynamic balancing is required for propellers with an operating speed of more than 500 rpm or propellers with tip speed exceeding 60 m/s. The manufacturer shall demonstrate that the assembled propeller shall be within the specified limits.

For built-up propellers, the required static balancing may be replaced by an individual control of blade weight and gravity centre position.

2. Testing

Fixed pitch propellers, controllable pitch propellers and controllable pitch propeller systems are to be presented to **TL** for final inspection and verification.

The verification of dimensions, the dimensional and geometrical tolerances is the responsibility of the manufacturer.

The report on the relevant examinations is to be submitted to the Surveyor, who may require checks to be made in his presence.

TL reserves the right to require non-destructive tests to be conducted to detect surface cracks or casting defects.

In addition, controllable pitch propeller systems shall undergo pressure, tightness and functional tests.

2.2 Casted propeller boss caps, which also serve as corrosion protection, have to be tested for tightness at the manufacturer's workshop. **TL** reserves the right to require a tightness test of the aft propeller boss sealing in assembled condition.

2.3 If the propeller is mounted onto the shaft by a hydraulic shrink fit connection, a blue print test showing at least a 70 % contact area has to be demonstrated to the Surveyor. The blue print pattern shall not show any larger areas of contact, especially not at the forward cone end. The proof has to be demonstrated using the original components.

If alternatively a male/female calibre system is used, between the calibres a contact area of at least 80 % of the cone area has to be demonstrated and certified. After ten applications or five years the blue print proof has to be renewed.

2.4 The propeller blades shall be manufactured according to the specified tolerance class (ISO 484). As a minimum, verification of the following is required:

- Surface finish
- Pitch (local and mean pitch)
- Thickness and length of blade sections
- Form of blade sections
- Location of blades, reference line and blade contour
- Balancing (see also [1])
- For propellers running in nozzle or tunnel:

  - extreme radius of blades (for controllable pitch propellers with outer section at zero pitch).

Verification of blade section form may include the use of edge templates as specified for manufacturing tolerance classes S and I in ISO 484. Equivalent methods can be accepted, for instance the use of multi-axial milling machines, which have proven to be capable of producing the specified geometry with such an accuracy that only a slight grinding is necessary to obtain the specified surface finish.

3. Shipboard Testing

3.1 Purpose

The purpose of the tests is to ascertain that the pitch control system of CP propellers for main propulsion is working correctly.

3.2 Application

The requirements in this item apply to all new buildings and to all replacements, modifications, repairs, or re-adjustments that may affect the pitch control or response characteristics for main propulsion.

3.3 Scope of the tests

3.3.1 Pitch response test

A full range of tests is to be carried out to get the pitch response and verify that it coincides with the combinator curve of the propeller (see note). The tests are to be carried out for at least three positions of the control lever in ahead and astern directions (e.g., dead slow ahead / astern, half ahead / astern, full ahead / astern).

The tests are to be carried out in normal and emergency operating conditions.

Tests that are not affected by the control position may be carried out from one control position only.

Note: *The combinator curve is the relationship between the propeller pitch setting and the propeller speed.*

3.3.2 Test of the fail-to-safe characteristics

A test of the fail-to-safe characteristics of the propeller pitch control system is to be carried out to demonstrate that failures in the pitch command and control or feedback signals are alarmed and do not cause any change of thrust. Such failures are to be clearly identified and included in the test procedure.

3.3.3 Test procedure

Test procedure is to be prepared and proposed by the pitch control system manufacturer or integrator and agreed with **TL**.

3.4 Parameters to be recorded

The list of the parameters to be recorded during the pitch response test within this requirements is to be established by the pitch control system manufacturer or integrator and agreed with **TL**. This should include at least the following parameters:

- Position of the control handle,
- Actual pitch indication (local indication, remote indications),
- Rotational speed of the propeller,
- Response time between the pitch change order (modification of the lever position) and the instant when the pitch and propeller speed have reached their final position,
- Propelling thrust variation during the transfer of the control from one location to another one.

3.5 Test Result

Tests are to demonstrate:

- that the propelling thrust is not significantly altered when transferring control from one location to another and in case of failures in the pitch command and control or feedback signals.
- that the pitch response times measured during the test do not exceed the maximum value to be defined by the pitch control system manufacturer or integrator.

4. Quality Classes of Propellers

4.1 The requirements for the quality classes used in 4.1 are given in the international standards ISO 484/1 and 484/2.

4.2 The quality of manufacturing and the accuracy of the dimensions of propellers shall be adequate to their use.

G. Lateral Thrust Units

1. General

1.1 Scope

The requirements contained in G. apply to the lateral thrust unit, the control station and all the transmission elements from the control station to the lateral thrust unit.

1.2 Documents for approval

Assembly and sectional drawings for lateral thrust units with an input power of 100 kW and more together with detail drawings of the gear mechanism and propellers containing all the data necessary for checking are each to be submitted to **TL** for approval. In the case of propellers, this only applies to propulsive power levels above 500 kW.

2. Materials

Materials are subject, as appropriate, to the provisions of Section 6, B. G. applies analogously to the materials and the material testing of propellers.

3. Dimensioning and Design

3.1 General requirements

The dimensional design of the driving mechanisms of lateral thrust units is governed by Section 5 and 6, that of the propellers by C.

The pipe connections of hydraulic drive systems are subject to the applicable requirements contained in Section 7B, F.

Lateral thrust units must be capable of being operated independently of other connected systems.

Windmilling of the propeller during sea passages has to be taken into account as an additional load case. Otherwise effective countermeasures have to be introduced to avoid windmilling, e.g. a shaft brake.

In the propeller area, the thruster tunnel is to be protected against damages caused by cavitation erosion by effective measures, such as stainless steel plating.

For monitoring the lubricating oil level, equipment shall be fitted to enable the oil level to be determined.

For the electrical equipment of lateral thrust units, see Chapter 105 - Electrical Installations, Section 7, B.

3.2 Additional requirements for lateral thrust units for dynamic positioning

Bearings, sealings, lubrication, hydraulic system and all other aspects of the design are to be suitable for continuous, uninterrupted operation.

Gears shall comply with the safety margins for DK as specified in Section 6, Table 6.1. The lubrication system for the gearbox shall comply with Section 6, E.

For units with controllable pitch propellers, the hydraulic system has to comply with D.7.2. The selection and arrangement of filters has to ensure an uninterrupted supply with filtered oil, also during filter cleaning and exchange.

Where ships are equipped with automated machinery, the thruster unit has to comply with the requirements for main gears and main propellers in the **TL** Rules for Automation.

4. Tests in the Manufacturer's Works

Section 7B,H. is applicable as appropriate.

For hydraulic pumps and motors with a drive power of 100 kW or more, the tests are to be conducted in the presence of a **TL** Surveyor.

For lateral thrust units with an input power of less than 100 kW final inspection and function tests may be carried out by the manufacturer, who will then issue the relevant Manufacturer Inspection Certifacate.

5. Shipboard Trials

Testing is to be carried out during sea trials during which the operating times are to be established.

6. Casting Defects

Propellers shall be free of cracks. The correction of defects has to be executed for copper cast alloys according to the **TL** Material Rules, Section 11, A and Section 11, B for stainless steel.

H. Special Forms of Propulsion Systems

1. General

The investigation for a propulsion system most suitable for an actual naval ship should include special forms of propulsion. Lay-out and arrangement of these propulsion systems have to be coordinated closely with the manufacturer of the system.

1.2 Requirements for design and manufacturing have to be specified in the building specification. Concerning strength and model tests the rules for propellers have to be applied accordingly. Model tests have to be performed in order to proof the reliability and check special performance characteristics.

1.3 In the following the design characteristics of several propulsion systems are defined. The scope does not fulfill the demand for entirety.

2. Cycloidal Propellers

2.1 Cycloidal propellers are propulsion devices with blades rotating around a vertical axis and are able to change the thrust in size and direction.

2.2 If low noise emission is required, the following principles should be observed:

- The size of the device with a minimum circumferential speed of the wheel body at the nominal acoustic layout point has to be chosen

- An operation mode has to be used where the adjustment of the thrust is achieved by adjusting the number of revolutions

- The arrangement in the hull shall enable a distance to the sonar system which is as big as possible (stern arrangement)

- For the reduction of the intake speed a low-noise gear has to be provided

- The connection of the housing to the ship foundation shall be established by an elastic mounting. The foundation as part of the hull structure shall be of appropriate stiffness

- The surface of the wheel body at the same level as the ship's bottom has to be coated with antidrumming material

2.3 If a low magnetic signature is required, a high content of non-magnetizable materials has to be provided, for reference see Chapter 103 - Special Materials for Naval Ships.

3. Supercavitating Propellers

The blades of supercavitating propellers are to be designed to achieve a stable cavitation layer over the complete blade surface at the nominal lay-out point. Lay-out and design have to observe the foreseen operation characteristics and the demanded reversing and steering ability.

4. Partially Submerged Propellers

Partially submerged propellers are designed for an arrangement at the ship where the propeller blades break through the water surface.

Because of the high transverse forces this type of propulsion is only applicable for multi-shaft propulsion systems. The degree of submerging has to be chosen to achieve a stable ventilation at the suction side.

5. Water Jet Propulsion Systems

5.1 Scope

These requirements apply to all devices producing a thrust by jet. This includes pump jets as well as water jets and comparable drives.

5.2 Design

5.2.1 The lay-out data have to be coordinated with the manufacturer of the water jet propulsion aggregate.

At all operating conditions no cavitation shall occur in the pump which can lead to damage of the components. To achieve this already in the design phase, the manufacturer has to indicate the areas of erosive cavitation danger.

5.2.2 Requirements for the manoeuvring abilities have to be specified in the building specification.

5.2.3 If it is not planned to equip all propulsion units of a multi-shaft ship with manoeuvring devices, it has to be ensured in the building specification that all requirements concerning rudder effect, stopping time, survivability and redundancy are defined and can be met.

5.3 Arrangement

Pump shaft and entrance duct have to be arranged parallel to the longitudinal midship plane of the ship. The pump shaft must be parallel to the waterline in the trimmed condition of the ship. The pump is to be arranged in a height that a quick start of the system is ensured for the minimum draught of the ship

5.4 Construction requirements

5.4.1 The pump has to be integrated in the hull structure in a way that the maximum longitudinal and transverse forces occurring during acceleration or crash-stop operation and during manoeuvres with the jet in the maximum steering position are safely withstood.

Detachable connections to the ship's hull shall enable an easy installation and dismounting of the pump unit.

5.4.2 The form of the entrance duct has to be coordinated with the manufacturer of the water jet unit. The stream towards the pump has to be continuous and cavitation shall be avoided in the complete inlet area. The entrance duct has to be arranged to exclude the possibility of air ingress.

5.4.3 To protect the pump from flotsam a protection grid has to be arranged at a suitable location of the entrance duct. Before this grid an inspection port has to be provided. It must be possible to open this port and have access to it at floating ship. It shall be possible to clean the intake protection grid by reversing the flow direction.

5.5 Manoeuvring facilities

5.5.1 Steering nozzle

The adjusting angle of the steering nozzle shall reach 30° to each side. If no time for adjusting is defined in the building specification a value less than 8 s has to be assumed for the complete range of 60°.

5.5.2 Thrust reversing device

If no time for adjusting is defined in the building specification a value of not more than 10 s has to be assumed for the adjusting from "full ahead" to "full astern". The neutral position of the thrust reversing system (zero thrust position) has to be marked at the control device.

5.6 Hydraulic system

5.6.1 In case of multi-shaft propulsion each propulsion unit has to be equipped with its own, independently working hydraulic system.

5.6.2 Control oil pumps

The lay-out of the control oil pumps shall follow D.4.

I. Dynamic Positioning Systems (DK Systems)

1. General

1.1 Scope

The **TL** Rules concerning Dynamic Positioning Systems apply to ships, which are classed by **TL** and are to receive the Class Notation DK 1 to DK 3 affixed to the Character of Classification.

In the following an overview for general information about these Rules shall be given.

The Rules are based on the single failure concept.

1.2 Definitions

Control mode

Possible control modes of a DK control system may be:

- Automatic mode (automatic position and heading control)

- Joystick mode (manual position control with selectable automatic or manual heading control)

- Auto track mode (considered as a variant of automatic position control, with programmed movement of the reference point)

- Manual mode (individual control of pitch and speed, azimuth, start and stop of each thruster)

DK-system

A DK system consists of components and systems acting together to achieve sufficiently reliable position keeping capability.

The complete installation comprises the sub-systems:

- Power system

- Thruster system and

- Control system

The requirements of the DK system configuration for the different Class Notations are shown in Table 7A.4.

Position keeping

Maintaining a desired position and heading or following a predefined track within the critical excursions or otherwise the specified excursions as specified in the DK operation manual of the DK system and under the defined environmental conditions.

Single failure

The single failure concept assumes that only one (single) failure is the initiating event for an undesired occurrence. The simultaneous occurrence of independent failures is not considered. However, common mode failures are to be examined.

WCF - Worst Case Failure

The identified single failure mode in the DK system resulting in maximum effect on DK capability as determined through FMEA study.

This worst case failure may be used in a consequence analysis.

1.3 Documents for approval

1.3.1 The documents and drawings specified below are to be submitted for approval to **TL:**

- General description of the system

- Test programs for factory acceptance test and DK control trial

- Documentation for control, safety and alarm systems including test program

- Thruster documentation

- Electric power system documentation

- Operation and maintenance manual

1.3.2 Failure Mode and Effect Analysis (FMEA)

A failure mode and effect analysis (FMEA) concerning availability of the DK system after a single failure shall be provided for Class Notation DK 2 and DK 3.

2. Requirements for Class Notations

2.1 Reliability

The necessary reliability is determined by the consequence of a loss of position keeping capability. The larger the consequence, the more reliable the DK system shall be.

Consequently the requirements have been grouped into three Class Notations. For each Class Notation the associated single failure criteria shall be defined.

The Class Notation of the ship required for a particular operation is based on a risk analysis of the consequence of a loss of position.

2.2 Class Notations

2.2.1 For Class Notation **DK 1**, loss of position may occur in the event of a single fault. The redundancy requirements according to Table 7A.4 are to be fulfilled.

2.2.2 For Class Notation DK 2, a loss of position may not occur in the event of a single fault in any active component or system. Static components will not be considered to fail where adequate protection from damage is demonstrated and reliability is deemed acceptable by **TL.**

2.2.3 For Class Notation DK 3, a loss of position may not occur in the event of a single fault in any active or static component or system. This applies also for the total failure of one compartment due to fire or flooding.

2.2.4 For Class Notations **DK 2** and **DK 3**, a single inadvertent action shall be considered as a single fault, if such an action is reasonably probable.

2.2.5 If the DK control system is tested with a special “hardware-in-the-loop” test (during FAT and on board) a respective entry in the Technical File of the Class Certificate is possible.

2.2.6 DK systems which exceed the requirements for Class Notation DK 2 or DK 3 (e.g. separate fuel, cooling water system for each diesel engine) a respective entry in the Technical File of the Class Certificate is possible.

3. Functional Requirements

3.1 Ships with Class Notation DK 1 are able to keep their position at least in automatic mode and joystick mode.

3.2 Ships with Class Notation DK 2 fulfil the requirements of DK 1 and are able to keep their position after a single failure in an active component.

3.3 Ships with class notation DK 3 fulfil the requirements of DK 2 and are able to keep their position after a single failure in an active or static component. This applies also for the total loss of the equipment in one compartment due to fire or flooding.

3.4 In order to meet the single failure criteria redundancy of components will normally be necessary as follows:

- for Class Notation **DK 2**, redundancy of all active components

- for Class Notation **DK 3**, redundancy of all active and static components and physical separation of DK relevant systems.

4. Tests

4.1 Factory acceptance test (FAT)

Before a new installation is surveyed and tested factory acceptance tests shall be carried out at the manufacturer's work. These tests are to be based on the approved program and shall demonstrate full compliance with the redundancy concept, if applicable.

4.2 Newbuilding survey

4.2.1 Newbuilding survey, which shall include a complete survey of the DK system to ensure full compliance with the rules.

4.2.2 This survey includes a complete test of all DK relevant systems and components (DK control trial). For details see **TL** Rules according to item 5.

5. Full Information

The full and binding requirements concerning dynamic positioning are defined in the **TL** Rules for Dynamic Positioning Systems (Chapter 22).

J. Cavitation Noise of Propellers

Fig. 7A.2 shows the probable and unprobable ranges for cavitation depending on the thrust load coefficient.

Cavitation inception number:

$$\mathrm{SIGMA}_n = \frac{p_S - p_V}{0,5 \cdot \rho \cdot (\pi \cdot D \cdot n_2/60)^2}$$

D has to be inserted in metres

Thrust load coefficient:

$$C_{Th} = \frac{T}{0,5 \cdot \rho \cdot v_A^2 \cdot A_D}$$

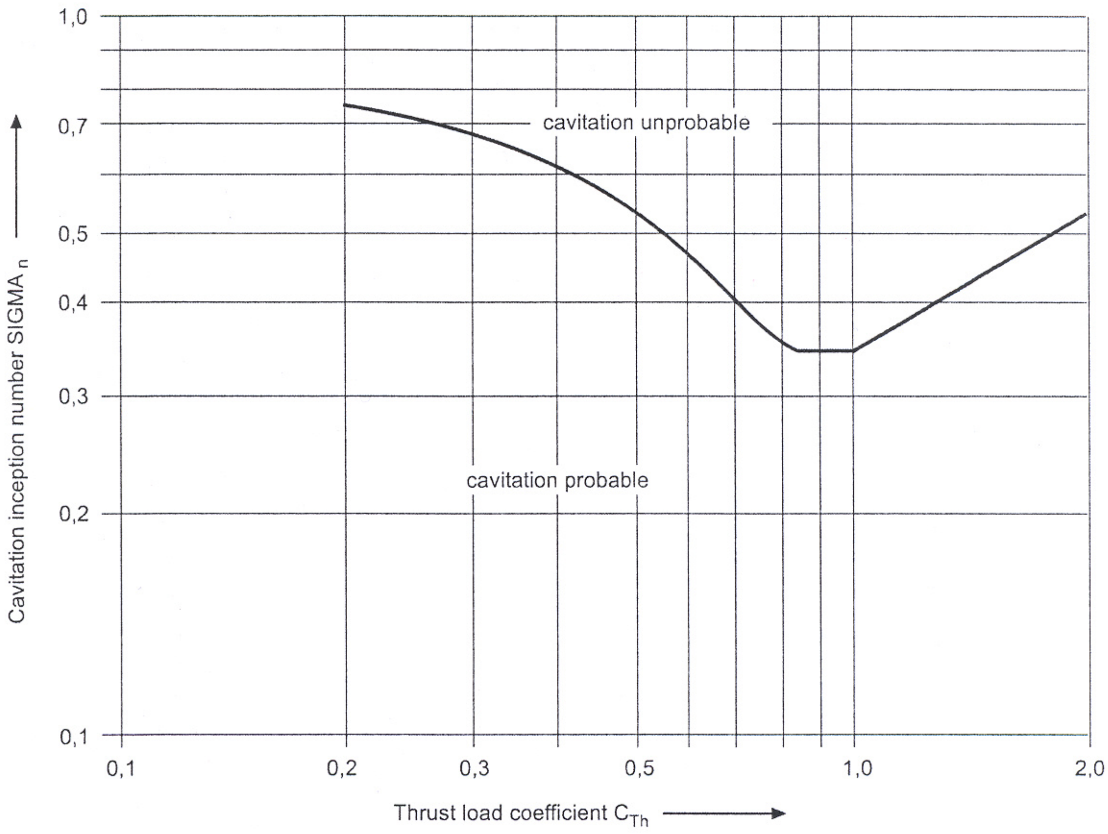


Fig. 7A.2 SIGMA$_n$ - C$_{Th}$ - diagram for cavitation noise of propellers

Table 7A.4 Minimum requirements for DK systems

<table>
<tr><th colspan="3" rowspan="2">Subsystem or component</th><th colspan="4">Minimum requirements for Class Notation</th></tr>
<tr><th>DK1</th><th>DK 2</th><th colspan="2">DK 3</th></tr>
<tr><td rowspan="5">Power system</td><td colspan="2">Generators and prime mover</td><td>Non-redundant</td><td>redundant</td><td colspan="2">redundant, separate compartments</td></tr>
<tr><td colspan="2">Main switchboard</td><td>1</td><td>1</td><td colspan="2">2
in separate compartments</td></tr>
<tr><td colspan="2">Bus-tie breaker</td><td>0</td><td>1</td><td colspan="2">2</td></tr>
<tr><td colspan="2">Distribution system</td><td>Non-redundant</td><td>redundant</td><td colspan="2">redundant, through separate compartments</td></tr>
<tr><td colspan="2">Power management</td><td>No</td><td>redundant</td><td colspan="2">redundant, separate compartments</td></tr>
<tr><td>Thruster system</td><td colspan="2">Arrangement of thrusters</td><td>Non-redundant</td><td>redundant</td><td colspan="2">Redundant, separate compartments</td></tr>
<tr><td colspan="3">DK relevant auxiliary systems</td><td>-</td><td>redundant (3)</td><td colspan="2">redundant, separate compartments, provided WCF is not exceeded</td></tr>
<tr><td rowspan="2">DK-Control system</td><td colspan="2">Auto control;
no. of computer systems</td><td>1</td><td>2</td><td colspan="2">2+1
in backup control station</td></tr>
<tr><td colspan="2">Manual control;
joystick with auto heading</td><td>Yes</td><td>Yes</td><td colspan="2">Yes</td></tr>
<tr><td rowspan="4">Sensors</td><td colspan="2">Position reference systems</td><td>2</td><td>3</td><td colspan="2">3
whereof 1 connected to back-up control system</td></tr>
<tr><td rowspan="3">Vessel's sensors</td><td>Wind</td><td>1</td><td>3</td><td>3</td><td rowspan="3">one of each connected to back-up control system</td></tr>
<tr><td>Vertical reference sensor (VRS)</td><td>1</td><td>3</td><td>3</td></tr>
<tr><td>Heading reference system</td><td>1</td><td>3 (1)</td><td>3</td></tr>
<tr><td colspan="3">Essential non-DK systems (2)</td><td>Non-redundant</td><td>redundant</td><td colspan="2">redundant,
separate compartments</td></tr>
<tr><td colspan="3">Printer</td><td>yes</td><td>yes</td><td colspan="2">yes</td></tr>
<tr><td colspan="3">UPS for DK control system</td><td>1</td><td>2</td><td colspan="2">2+1
In separate compartments</td></tr>
</table>

(1) *The heading reference system(s) shall comply with IMO Res. A424(XI) performance standards for gyro-compasses. When three heading reference systems are required one of the three may be replaced by a heading measuring device based upon another principle, as long as this heading device is type approved as a THD (transmitting heading device) as specified in IMO Res. MSC.116 (73).*

(2) *See Section 2.B.6*

(3) *when active components are used*

SECTION 7 B

AZIMUTHING PROPULSORS

A. General

1. Scope

1.1 Types of propulsors

Azimuthing propulsors cover all steerable devices with geared torque transmission (rudder propeller) and main propulsion motor in an underwater gondola (podded drives). Fixed podded drives (booster) have to comply with this Section as far as applicable.

The requirements of this Section are valid for azimuthing devices as main propulsion systems. The ship's manoeuvring station and all transmission elements from the manoeuvring station to the azimuthing propulsor are specified in the **TL** Rules for Electrical Installations (Chapter 105), Section 13.

1.2 Operating conditions

A faultless continuous operation under the specified ambient and operational conditions is required. Components which can not easily be changed without drydocking the ship and without going out of service or having enough redundancies have to be designed for a life- and/or service time cycle of 5 respectively 6 years minimum. If this period cannot be guaranteed, **TL** has to be informed about the respective faultless continuous service time already at the time of class application.

Azimuthing propulsors serve as driving and steering device. As long as they are used for steering, a certain fraction of the thrust is not available for driving in forward direction. The concept of the ship, created by the Naval Authority and shipyard, has to take into account, that even under severe weather conditions while keeping the course, the remaining thrust fraction for driving the ship forward is sufficient. This has to be fulfilled for all loading conditions. **TL** reserves the right to request for an appropriate demonstration.

The spaces, where the azimuthing propulsors are mounted, are regarded as steering gear room. The respective environmental conditions apply.

For the design of podded drives, it has to be taken into account that most of the components are not accessible.

2. Documents for Approval

2.1 General

Design drawings of azimuth propulsors in main propulsion systems are to be submitted to **TL** for approval. The drawings are required to contain all the details necessary to carry out an examination in accordance with the following requirements.

Drawings shall contain necessary dimensions and material specification / mechanical properties.

A general brief functional description including definition of load cases as a basis for strength calculation for the individual components is to be submitted.

The following drawings and documents are to be submitted for approval:

2.2 Overall system

- Arrangement drawing of the complete azimuthing propulsor and definition of forces acting on the hull
- Sectional drawing of the entire azimuthing propulsor
- Protection concept including list of sensors and their initial limits (document)
- Complete list of measuring points, type of sensor safety and alarm system, incl. detailed description (drawing and document)
- Fire protection (document) podded drives only
- Ventilation system (drawing) podded drives only
- Heat balance and cooling system (document and drawing) podded drives only
- Description of test procedures at different assembly stages, dock and sea trial (document)

- Quality assurance and inspection plan (document) podded drives only

- Definition of overall control system

- Failure Mode and Effects Analysis podded drives only (where required)

2.3 Structural components

- Scantling of structural parts of the housing, steering pipe/strut and well (drawings)

- Loading cases to represent highest loads in normal sailing, manoeuvring and emergency conditions (document)

- Stress calculation esp. ratio calculated/permissible (document)

- Fatigue analyses, as far as requested (calculation)

- Foundation and connection with ship structure (drawing)

2.4 Mechanical components

- Propeller (drawings)

- Shafting (drawings)

- Shaft roller bearings (life time calculation)

- Connecting elements as bolts, couplings, clutches, etc. (drawings and calculation)

- Slewing and slip ring bearings

- Lubrication and hydraulic systems (diagram)

- Seals, sealing systems including emergency seal system (if applicable) (drawings and functional description)

- Torsional vibration calculations as required (see Section 8), additionally for the load case:

  - Blade loss case

- Shaft locking device (drawing and calculation):

  - Calculation of brake moment, description of locking procedure

- Bilge system for drainage, podded drives only (diagram)

- Steering gear:

  - Assembly and detailed drawings of all important load and torque transmitting single components

  - Gear calculation

  - Hydraulic diagram for steering

  - Hydraulic power pack (or equivalent for azimuth rotation)

  - Fechanical key on slewing bearing, locking device (drawing)

  - Functional description for podded drives only and other new designs

2.5 Electrical components

The documentation of electrical components is specified in the **TL** Rules for Electrical Installations (Chapter 105), Section 13.

2.6 Maintenance

(podded drives only and propulsors above 2500 kW)

- Maintenance plan (expected life time)

- Inspection arrangement (underwater, inside pod)

- Clearance of slewing bearing, normal and critical

- Condition monitoring arrangement, upper and lower bounds

3. Reference to further Rules

3.1 TL Rules

- Chapter 107, Ship Operation Installations and Auxiliary Systems
- Chapter 105, Electrical Installations
- Chapter 106, Automation
- Chapter 104, Propulsion Plants, Section 7A

3.2 Other Rules and regulations

- IEC 60092: Electrical Installations in Ships, Part 501: Special features - Electric propulsion plant, Podded Drives

B. Materials

1. Approved materials

The selection of materials is subject, as and where applicable, to the provisions for approved materials defined in the **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 2, B.2.1 and to those of Sections 5, 6 and 7A.
Materials have to comply with the **TL** Rules for Materials (Chapter 2). Special requirements are contained in the respective Chapter.

2. Testing of materials

All important components of the azimuthing propulsor involved in the transmission of torque, thrust, bending moment or hydraulic pressure are to be tested under the supervision of **TL** in accordance with the **TL** Rules for Materials (Chapter 2).

C. Design of Azimuthing Propulsors

1. Number of azimuthing propulsors

Each ship shall be equipped with at least two azimuthing propulsors. Both units are to be capable of being operated independently of the other. The failure of one azimuthing propulsor or its control system, converter or motor shall not cause the failure of other devices (see IEC standard 60092-501, 4.1.4). This has to be demonstrated by a FMEA.

2. Load cases for scantling

For scantling of components the respective worst case has to be considered, except for life time calculations. The maximum attainable torque at respective rpm has to be considered. The following load cases should be taken into account at minimum:

- full ahead (maximum torque including possible overload of an electric motor)
- bollard ahead and astern, if applicable
- loss-off one propeller blade at 0,35 radius (unbalance at full rpm; bending moment)
- maximum steering at full speed (sailing into turning circle)
- maximum steering at maximum manoeuvring speed
- crash stop

Additionally for electric main propulsion motor (IEC 60092-501, 4.3):

- highest steady state torque in three phase shortcircuit of the motor, if exceeding crash stop values
- highest oscillating torque during two phase short-circuit of the motor, if exceeding crash stop values

If the naval ship is assigned an ice class, additional requirements are defined in Section 9.

3. Screw propeller

Screw propellers have to comply with the requirements defined in Section 7A.

For propeller shrink fit calculation the thrust has to be taken into account with a negative sign in case of pulling propellers.

4. Electric propulsion motor

The requirements of the electric propulsion motor are specified in the **TL** Rules for Electrical Installations (Chapter 105), Section 13. The required supervision of construction has to be extended to the respective power supply systems.

The air humidity of electric motors, operating in a closed ventilation loop, has to be monitored.

5. Gears and couplings

5.1 Gear design

5.1.1 Gears or toothed connections have to comply with the requirements defined in Section 6.

In case of electrically driven azimuthing propulsors, a safety of S_F = 1,35 and S_H = 1,15 respectively against static load (less than 1000 load cycles) of 1,6 M nominal has to be demonstrated for the tooth root and flank.

5.1.2 Curved tooth couplings have to be dimensioned according to the Rules of Section 6, G.3. for tooth couplings.

5.2 Lubrication

The lubricating and/or cooling oil supply is to be ensured by a main pump and an independent standby pump.

In the case of separate lubricating systems in which the main lubricating oil pumps can be replaced by means available on board, the standby pump may be replaced by a complete spare pump. This spare pump is to be carried on board and is to be ready for mounting.

Oil level indicators and filters are subject to the provisions of F.4. and F.5. wherever relevant.

The oil temperature has to be measured. A heat exchanger has to be provided, if higher than allowable temperatures can be expected under unfavourable environmental conditions.

6. Shafting

6.1 Shaft

6.1.1 For the dimensioning of the propeller shaft between propeller and gear wheel, see Section 5. For the dimensioning of the remaining part of this shaft and all other gear shafts see Section 6.

6.1.2 In case that the propeller shaft coincides with the electric motor shaft, it has to be ensured that in hot and cold condition and while acting all reasonable forces, the shaft is stiff enough to guarantee an appropriate gap between rotor and stator of the motor.

6.2 Shaft bearings

6.2.1 Shaft bearings are normally of roller type. They have to be so arranged that a sufficient load in all operational conditions is applied. The oil temperature has to be measured.

6.2.2 The expected lifetime has to be given and proven by a lifetime calculation. Instructions for maintenance have to be provided. An inspection of the bearings, e.g. by endoscope has to be enabled.

6.2.3 For podded drives the wear of bearings has to be monitored, if they are not accessible. It shall be possible to take an oil sample.

6.2.4 For podded drives the thermal expansion of the shaft and housing has to be taken into account. The temperature of the bearing has to be monitored redundantly and independent from the alarm system and a two stage alarm has to be delivered. In exceptional cases where a sufficiently forced lubricating oil flow is applied, the oil temperature can be measured instead of. Sufficient redundancy of forced lubrication has to be ensured, e.g. by a second pump (see also IEC 60092-501, 12.2.1) and the flow has to be monitored. Switching over to the redundant system shall be performed automatically and shall be alarmed.

6.2.5 For podded drives, a continuous control of the lubrication has to be provided redundantly and independent from the alarm system (see IEC 60092- 501, 13.2,2). A two stage alarm has to be delivered.

In case of oil lubricated bearings, the maximum and minimum level has to be monitored. The monitoring of the minimum level shall be operable under all normal operational conditions.

6.2.6 For podded drives, measures should be taken to prevent the circulation of electrical current between the shaft and the bearings.

6.3 Shaft sealing

6.3.1 For podded drives, the shaft seal has to protect the motor and other functionally essential components against a liquid ingress with a sufficient safety and redundancy. In case of malfunction of one of the tightening parts of the seal, measures have to be taken to get the leakage information as an alarm into the ships information system. The seal has still to protect the functionally essential components safely.

6.3.2 The shaft seal has to cope with the axial expansion and movements of the shaft.

6.3.3 The header tank has to be equipped with a minimum level sensor. A maximum level sensor is highly recommended. The piping system with header tank, position and kind of level sensors has to be submitted for approval. If the seal is type approved a schematic sketch with piping is sufficient, otherwise full drawings have to be submitted.

6.3.4 The seal has to protect the sea water against pollution if there is an oil leakage. The oil leakage has also to be indicated as soon as a specified maximum permissible leakage is exceeded.

6.3.5 For podded drives, an emergency sealing device, suitable to seal the propeller shaft while the shaft locking device is active, has to be provided (see IEC 60092-501, 13.2.3). The operability of the system shall be indicated and shall be checked on a regular basis. Any failure of the system has to be alarmed.

6.4 Locking device

6.4.1 Each azimuthing propulsor is to be provided with locking devices to prevent the unintentional rotation of the propeller shaft in case of a failure (see also IEC 60092-501, 13.2.3). The locking device of the propeller shaft is to be designed to securely lock the non-operated propeller while operating the ship with the maximum power of the remaining propulsors, however at a ship speed of at least 8 kn.

6.4.2 Clutch / Brake

Where a clutch /brake is introduced to serve as a locking device, an unintentional operation has to be avoided. In case the clutch / brake is not dimensioned according to the maximum motor torque, a preventive measure has to be introduced to avoid an overloading of the device. An engaged clutch / brake has to be indicated at the control station.

7. Bilge system (for podded drives only)

7.1 The gondola and other separately closed rooms, e.g. for the electric propulsion motor, of the podded drive have to be equipped with an effective bilge system in order to prevent liquid ingress into the motor or other functionally essential components. The gondola bilge has to be large enough in order to collect the liquid safely if a seal element fails and has to ensure that the pipe(s) gets the liquid reliably out of the gondola. The system has to be dimensioned so that a sufficient suction capacity of at minimum 5 times the maximum allowable leakage rate of all seals feeding the bilge system is continuously guaranteed.

The suction pipe is to be positioned in such a way that due to a failure at the highest level the flushing of the fluid inside the pipe does not cause any harm for the motor or other equipment.

A main and auxiliary bilge system has to be provided for the bilges protecting the propulsion motor.

7.2 The bilge level shall be monitored by at minimum two independent sensors. Any abnormal leakage has to be indicated with an alarm. A two stage alarm level is required, indicating “high” and “high high” level.

The bilge in the gondola has to be equipped with an analogous sensor, monitoring the bilge level and its change. If the pipe(s) cannot empty the bilge, the high high level alarm has to indicate the possible flooding of the gondola and a shut-down of the motor should be requested. An automatic shut-down is allowed.

8. Ship foundation, support pipe and gondola

8.1 The design of the support pipe and gondola and their attachment to each other and the ship's hull has to take account of the loads due to the propeller and nozzle thrust including the dynamic components and due to steering. For load case scenarios see 2.

8.2 Additionally the ships foundation and propulsor casing has to be stiff enough to limit the deflections at the connecting points to rotating mechanical parts in order to enable a reliable operation under relevant thermal and mechanical load conditions.

D. Design of Steering Device

The following paragraphs are applicable, so far no other means are installed to enable the ship's steering capabilities, as required by **SOLAS** Chapter II-1, 29.

For podded drives see also IEC 60092-501, 13.7.

Azimuthing propulsors absorbing 2500 kW or more installed on one ship are regarded as steering devices having a rudder stock diameter in excess of 250 mm in order to comply with **SOLAS** Reg. 29.

In case of a sudden inoperability of all steering devices (e.g. black out) the steering angle has to be kept under all weather and sea conditions until an emergency steering is possible.

1. Number of steering devices

Each azimuthing propulsor has to be equipped with at least one main and one auxiliary steering device. Both steering devices are to be independent of each other and, wherever possible, act separately upon the propulsor (slewing bearing). A fault in one steering device shall not influence the remaining one(s). **TL** may agree to components being used jointly by the main and auxiliary steering device. A comparison of conventional rudder components and the respective parts of an azimuthing propulsor is given in the Table 7B.1.

For podded propulsors it has to be demonstrated by a comprehensive FMEA that no single fault in one steering device leads to an inoperability of both devices.
No single fault and ship speed and steering position should prevent the remaining steering devices from fulfilling the **SOLAS** requirements (see 3. and 4.)

2. Overload protection and steering angle limitation

2.1 Power-operated steering device systems are to be fitted with overload protection (slip coupling, relief valve or comparable) to ensure that the steering device is not harmed. The overload protection device is to be secured to prevent later adjustment by unauthorized persons. Means shall be provided for checking the setting while the ship is in service.

2.2 The steering angle in normal operation (sea mode) has to be limited to the declared steering angle, e.g. by limit switches. Since azimuthing propulsors are intended to turn 360°, any mechanical stoppers, required for normal rudders, have to be replaced by adequate alternative means. Those alternative means have to be independent from the declared steering angle limit switches.

This limitation can be switched off (harbour mode), as soon as means are active (e.g. speed, power limitation) to avoid manoeuvring which endangers the ships safety.

If any unintentional use of the steering device at the most unfavourable ships condition (e.g. maximum speed) does not endanger the ship (which has to be demonstrated at the sea trial), a limitation of the steering angle may be dispensed from.

How to operate the steering module, the range of operation and all related constraints, e.g. power reduction if steering angle exceeds the declared steering angle, as well as the crash stop procedure have to be described and submitted.

Table 7 B.1 Comparison between conventional steering gear and azimuthing propulsors

Required redundancy	Conventional steering gear	Azimuthing propulsor
No	Rudder stock	Slewing ring, housing (support cone)
No	Hub of tiller	Contact area of gear connection (teeth/teeth) **(1)**
Yes	Tiller arms, chamber walls (wings)	Pinion wheels and shafts (and reduction gears, if present)
Yes	Hydraulic cylinders, chambers of rotary gear	Hydraulic / electric motors
Yes	Hydraulic piping	Hydraulic piping / electric power supply
Yes	Hydraulic power unit	Hydraulic power unit / converters

(1) *With respect of redundancy as recommended by **SOLAS**, main and auxiliary steering system should act autonomously on the rudder stock. This is guaranteed if there are more than one actuating hydraulic / electric motor without inline hydraulic / electric power supply /electric motor, they can be separated and disengaged (prevention of blocking).*

2.3 If SOLAS shall be applied in the sense of required manoeuvrability for a ship a "declared steering angle" **(1)** is introduced. The maximum angle at which the azimuth propulsor can be oriented on each side when the ship navigates at its maximum speed (sea mode) is to be specified by the Naval Authority. Such maximum angle is generally to be less than 35° on each side.

2.4 Helm angle in excess of the declared steering angle may be used below a certain ship speed (manoeuvring mode) as soon as the ships safety is not endangered, while the steering device is unintentionally used. A power limitation may be required so that the safety of the ship is not endangered.

2.5 Manoeuvres possibly conflicting with design features of the azimuthing propulsor or the ship have to be blocked by automatic control.

3. Main steering device

The main steering device has to be capable to turn the azimuthing propulsor from one side at the declared steering angle to the opposite side at the declared steering angle with a speed of 2,5 s/° at maximum ship service speed.

For electric propulsion motors, the main steering device should be fed from the same switchboard part as the motor.

4. Auxiliary steering device (podded drives only)

Auxiliary steering devices shall, with the azimuthing propulsor fully immersed in calm water, be capable of putting the propulsor from 15° port to 15° starboard or vice versa within 60 seconds at 50 % of the ship's maximum speed, subject to a minimum of eight knots. Hydraulically operated auxiliary steering devices shall be fitted with their own piping system independent of that of the main steering device.

The pipe or hose connections of steering devices shall be capable of being shut-off directly at the pressurized casings.

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(1) *Declared steering angle is at minimum the helm angle at which the vessel shows a comparable manoeuvring behaviour as it would show, when equipped with a conventional rudder at 35° helm angle with maximum steering force.*

In case of a failure of the main steering system the auxiliary steering device is at least to be capable of moving the azimuthing propulsor to midship position, where the unit can be locked. Manual operation is acceptable as an emergency solution, if the auxiliary steering device is out of operation.

The auxiliary steering device has to be capable to keep the propulsor at the current position, if the main steering device fails, even the ship is sailing at full speed.

The requirements of electrically driven auxiliary steering devices have to be realised accordingly.

5. Locking mechanism

A locking mechanism has to be provided in order to fix the azimuthing propulsor in a desired position. The mechanism has to withstand all the loads which may occur during operating the unit with the maximum power. It has to be shown that the application of the locking device can be done within a reasonable time, independent from weather and sea condition.

6. Power unit

6.1 Hydraulic device

6.1.1 Where power operated hydraulic main steering gears are equipped with two or more identical power units, no auxiliary steering gear needs to be installed provided that the following conditions are fulfilled:.

- two independent and separate power actuating systems (power unit(s), hydraulic pipes, power actuator), each capable of meeting the requirements as set out in 3. and 4. or

- at least two identical power actuating systems which, acting simultaneously in normal operation, are to be capable of meeting the requirements as set out in 3. and 4.

6.1.2 In the event of failure of a single component of the main steering gear including the piping, excluding the slewing bearing means are to be provided for quickly regaining control of one steering system (podded drives only).

6.1.3 In the event of a loss of hydraulic oil, it has to be possible to isolate the damaged system in such a way that the second steering system remains fully operable.

6.1.4 At least one electrically driven hydraulic pump has to be fed from an other switchboard.

6.2 Electric device

The requirements of the **TL** Rules for Electrical Installations (Chapter 105), Section 7, A. have to be fulfilled.

The electrical main steering device has to be fed from the switchboard of an electrical power generating plant. At least one (auxiliary) steering device has to be fed from an other switchboard.

The electric device has to be protected against overcurrent and short-circuit.

7. Control and monitoring

7.1 Both the propeller drive and the steering device of each azimuthing propulsor are to be controlled from a manoeuvring station on the navigating bridge.

The controls of each individual azimuthing propulsor are to be mutually independent and so designed that the azimuthing propulsor cannot be turned unintentionally. An additional combined control for all azimuthing propulsors is permitted.

7.2 Failures of single control components (e.g. control system for variable displacement pump or flow control valve) which may lead to loss of steering shall be monitored by an audible and visible alarm on the navigating bridge, if loss of steering cannot be prevented by other measures.

7.3 Means have to be provided, fulfilling the same purpose as the steering angle limitation to the declared steering angle on both sides in normal service (sea mode). Those means shall limit the steering angle effectively and may trigger a power reduction of the propulsion motor. These may be dispensed with by **TL** in cases where no danger for the ship is caused by unintentional slewing of the azimuthing propulsors at full power and ship speed to any angle. A safety system has to be realised according to the **TL** Rules for Electrical Installations (Chapter 105), Section 9, C. for cases, where the steering angle limitation fails. The failure of the steering angle limitation has to be alarmed. It shall be possible in the sea mode that the limitations can be overruled by an emergency crash stop manoeuvre.

7.4 The failure of a single element within the control and hydraulic / electrical azimuthing system of one unit shall not lead to a failure of another unit.

7.5 Where the hydraulic / electric systems of more than one azimuthing propulsors are combined, it is to be possible in case of a loss of hydraulic oil or electrical fault to isolate the damaged system in such a way that the other control system(s) remain fully operational.

7.6 Local control

Means are also to be provided for exercising control from the propulsor machinery compartment. The transmission system is to be independent of that serving the main control station. Requirements for the local control station are defined in the **TL** Rules for Electrical Installations (Chapter 105), Section 13, H.

It shall be possible to move the azimuthing propulsor into a favourable position and to start the main propulsion motor again, e.g. after a black out.

7.7 Steering angle indication

The position of the azimuthing propulsor is to be clearly indicated at the bridge and at all steering stations. Where the steering device is operated electrically or hydraulically, the steering angle is to be indicated by a device (local steering device position indicator) which is mechanically actuated either by the steering device itself or by parts which are rigidly connected to it. In case of time-dependent control of the main and auxiliary steering device, the midship position of the steering device is to be indicated on the bridge by some additional means (signal lamp or similar). In general, this indicator is still to be fitted even if the second control system is a manually operated hydraulic system. See also the **TL** Rules for Electrical Installations (Chapter 105), Section 9, C.

The steering position of the azimuthing propulsor at any moment is also to be indicated at the steering device itself.

It is recommended that an additional steering angle indicator has to be fitted at the machinery control centre.

8. Dimensioning of components

8.1 Scantling of steering torque transmitting components

The most severe loads on the components of the steering device determined from the load cases defined in C.2. are not to exceed the yield point of the materials used. Additionally a torque of 2,5 times the maximum operational torque (corresponds to the rudder stock yielding torque) has to be used for scantling with a safety factor of 1,3 against yield strength. The design of parts of the steering device with overload protection is to be based on the loads corresponding to the response threshold of the overload protection with a safety factor of 1,3.

It is assumed that the most severe loads occur seldom. If they occur more frequently, their influence on the fatigue has to be taken into account.

8.2 Slewing Bearing

8.2.1 A lifetime calculation and a calculation of the safety factor for static load have to be provided. The lifetime calculation should show a sufficient dimensioning to obtain a lifetime period of 40000 h. A safety factor of 1,8 at minimum has to be obtained from static load calculation.

8.2.2 At least one inspection opening has to be provided in order to inspect the tooth contact pattern and lubrication situation.

8.2.3 Seals have to be provided to protect the slewing bearing from sea water ingress and from any oil / grease leakage. If the sealing is grease lubricated, a sufficient space has to be provided to accommodate the old grease of one docking period.

8.3 Slewing gear

The slewing gears are in general to be designed as cylindrical, bevel or worm gears, applying a steering moment until safety valve or overload protection device operates.

8.3.1 Scantling

The scantlings of slewing gears have to follow the calculation procedure as described in Section. 6.

The application factor K_A has to be calculated for the submitted load spectrum according to ISO 6336, Part 6.

The manufacturer has to submit the maximum torque at which either the safety valves open or the safety clutch is disengaged.

Depending on the application a respective load spectrum has to be submitted and its suitability to be demonstrated. A three step load spectrum has to be used for scantling and its suitability to be demonstrated. For the resulting equivalent load a safety of S_F = 1,8 and S_H = 1,3 has to be demonstrated for unlimited load cycles.

In case of a worm gear the minimum life cycle is not to be less than one class period with a minimum wear safety S_W = 1,3. The lubrication of the worm has to be performed by an individual separate circulation.

8.3.2 Operation

A sufficient lubrication has to be ensured and a possibility to check lubrication at reasonable time intervals has to be realised.

An opening for a visual tooth inspection has to be provided (inspection cover).

E. Auxiliary Equipment

1. Bolts and screws

1.1 General

TL may dispense from the submission of detailed drawings as defined in A.2.4 in case of using standardised screws. **TL** may also dispense from delivery of material certificate, if the screw size is below M 39 and is manufactured in a well supervised mass production.

1.2 Screws

Where necessary, a sufficient lengthening of the screw by applying a torque, which usually generates a stress of up to 90 % of yield point, has to be demonstrated by calculation for essential connections, transmitting propulsion torque, thrust and bending moments. The female thread strength has also to be considered.

The calculation of necessary diameter for mounting the propeller blades see Section 7A, D.6., for the hub and flange couplings see Section 7A, E.2. For all other screws, except foundations, a safety factor of 1,5 against yielding has to be demonstrated for the most severe load case.

Washers are normally not permitted.

Screws have to be secured against unintentional loosening.

The material requirements for screws have to be fulfilled, see the **TL** Rules for Materials (Chapter 2).

1.3 Fitted bolts and shear pins

Wherever torque or shear forces have to be transmitted safely by bolt/pin connections, fitted bolts/pins should be applied. A torque/shear transmission by friction created by screws of the same connection, may be taken into consideration at the discretion of **TL.** Dimensions of fitted bolts/pins can be calculated for connections of propeller blade according to Section 7A, D.6., for hub and flange couplings according to Section 7A, E.2. Fitted bolts/pins can be replaced by stoppers in suitable applications. For all other fitted bolts/pins a safety factor of 1,5 against yielding applying the most severe load has to be demonstrated.

Fitted bolts have to be secured against unintentional loosening.

The material requirements for forgings have to be fulfilled, see the **TL.** Rules for Materials (Section 2).

2. Fitting systems

2.1 Shrink fit

A shrink fit calculation has to be submitted for all shrunk joints for torque transmission, for fluid-tight connections or for the safety relevant or functionally essential components. A contact area of more than 70% with equally distributed contact has to be ensured. The safety factor against slippage has to be chosen according to the respective application, using the following minimum guide values:

- propellers: 2,8
- couplings between propeller and gear box: 2,5
- within gear box: 3,0
- between gear box and motor: 3,0

The safety factor against yielding has to be chosen according to the respective component (due to geometry and production process), using the following minimum guide values:

-	propeller	1,33 for service (75 % of yield strength)
		1,11 for mounting and dismantling (90 % of yield strength)
-	coupling	1,25 for service (80 % of yield strength)
		1,05 for mounting and dismantling (95 % of yield strength)

2.2 Keyed shaft connections

Requirements as set out in Section 7A for propulsors and Section 5 for main shafting have to be applied.

3. Cooling systems

3.1 The cooling system of the motor or oil has to be dimensioned according to the most critical conditions:

- full power
- highest ambient temperatures
- lowest possible ship speed

3.2 In general the most difficult situation can be described by maximum water and air temperature (see Section 1, D.) and the propulsor running at maximum shaft speed.

3.3 If the cooling system does not allow to deliver the maximum power under the described conditions, the maximum attainable power has to be considered to fulfil A.1.2. The above described situation has to be reflected in the heat balance, which has to be submitted for approval on request of **TL.**

For the cooling system of podded drive motors see also IEC 60092-501, 7.3 and 12.3.

F. Hydraulic Systems

1. Piping

1.1 The design and setting of safety valves shall be such that their response threshold does not allow the maximum allowable working pressure to be exceeded by more than 10 %.

1.2 Pipes are to be installed at a sufficient distance from the ship's shell. As far as possible, pipes should not pass through cargo spaces. Pipe flange connections are not permitted in the vicinity of electrical equipment or connections.

At points where they are exposed to danger, copper pipes for control lines are to be provided with protective shielding and are to be safeguarded against hardening due to vibration by the use of suitable fastenings.

1.3 **TL** reserves the right to permit short hoses or metallic compensators (flexible pipes) instead of pipes in case of connections to moving parts or comparable situations.

1.4 Connections to other hydraulic systems are not permitted.

2. Dimensioning

For the design and dimensions of pipes, valves, fittings, etc., see the **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 8 and for pressure vessels Section 16.

For the determination of the maximum allowable working pressure the frictional losses in the steering gear including piping are to be considered. The relief valves are to be set at this pressure value.

3. Application for steering device

3.1 If hydraulic power is used for the steering gear of the azimuthing propulsor it shall not be in direct connection with other hydraulic systems.

3.2 The pipes of hydraulic steering devices are to be made of seamless or longitudinally welded steel tubes. The use of cold-drawn, unannealed tubes is not permitted.

3.3 The pipes of hydraulic steering devices are to be installed in a way ensuring maximum protection while remaining readily accessible.

4. Filters

Filters for cleaning the operating fluid are to be located in the piping system. It shall be possible to change oil cleaning filters without interruption of oil supply.

5. Tanks

Tanks forming part of the hydraulic system are to be fitted with oil level indicators.

The lowest permissible oil level is to be monitored. Audible and visual alarms shall be given on the navigating bridge and in the machinery space. The alarms on the navigating bridge shall be individual alarms.

In power-operated hydraulic steering systems, an additional permanently installed storage tank is to be fitted which has a capacity sufficient to refill at least one of the control systems including the service tank.

This storage tank is to be permanently connected by pipes to the control systems so that the latter can be recharged from a position inside the steering compartment.

G. Electrical Installations

1. Sensors and automation

1.1 Sensors have to be of type approved design and self checking, as far as applicable.

Level sensors should give a signal independent from the kind of liquid, which is involved in the respective system. Sensors showing levels of certain liquids only are not allowed, except, where explicitly requested. Level sensors should give a signal safely in both directions.

Sensors, essential for the operation and mounted in non-accessible areas, have to be designed as double sensors and are to be independently redundant. The independent redundancy is fulfilled, if another sensor measures a different but directly dependent parameters
for measurement. A check of plausible signals has to be realised, Illogical signals have to be alarmed.

1.2 Alarms and indicators are summarized in Table 7B.2.

The following table summarises the necessary sensors and alarms as given in the text above. The table is in addition to necessary sensors and alarms, which have normally to be applied, see also the **TL** Rules for Ship Operation and Auxiliary Systems (Chapter 107) and IEC 60092-501.

Table 7 B.2 Alarms and indicators

Description **L = Low** **I = Indication** **H = High** **S = Shut-down** **HH= High High** **A = Alarm**	**Applicable for pods only**	**Parameter**	**Kind of information**	**Information transfer**
Propulsion motor:				
Electromotor with closed air system (C.4)	X	Humidity	I	Display
Closed circuit cooling (E.3)	X	Temperature	H	
		Flow	I	
Gears:				
Lubrication oil (C.5.2)		Temperature	H	
		Pressure	L	
Lubrication oil tank (C.5.2)		Min + max level	L H	
Shafting:				
Shaft bearings (C.6.2)	X	(Wear) Accelerations Metal particles detection (oil samples or online sensor)	 I I	
		Temperature	H	Alarm, two stage
Lubrication of shafting :				
Redundant lubrication oil pump (C.6.2.3)		Flow	I in two stages	Display
	X	Switching over	I	Display
Lubrication oil (C.6.2.3)		Temperature	H	Alarm
		Min + max level	L + H	L-alarm / H-display
Shaft sealing: (C.6.3)				
Header tank		Min + max level	L + H	Alarm
Any leakage		Leakage	H	Alarm
Emergency sealing device	X	Operability	I	Alarm (if fault)
Shaft movement				
Locking device for propeller shaft (C.6.4.1)		Engagement	I	Display
Clutch for power transmission (C.6.4.2)		Engagement	I	Display

Table 7 B.2 Alarms and indicators (continued)

Description **L = Low** **I = Indication** **H = High** **S = Shut-down** **HH= High High** **A = Alarm**	**Applicable for pods only**	**Parameter**	**Kind of information**	**Information transfer**
Bilge system: (C.7.2)				
Liquid in gondola	X	Bilge level	H HH	Alarm, two stage
Liquid in gondola	X	Bilge level monitoring	L HH	Alarm, S
Liquid in motor	X	Bilge level	H HH	Alarm, two stage, S
Steering device:				
Control components (D.7.2)		Failure	I	Alarm
Steering angle limitation (D.7.3)		Failure	I	Display
Hydraulic system:				
Permissible oil level in tanks (F.5)	X	Level	L H	Display
Electrical systems:				
Comparing of redundant parameters for measurement (G.1.1)		Missing plausibility	I	Display
Data transmission (G.1.4)		Signal fault	I	Display
Critical situation (G.1.5)	X	Occurrence	I	Display
Fire alarm (G.2.2)	X	Smoke detected	I	Display

1.3 Safety systems

All signals leading to or requesting a shut-down or an essential emergency action shall be produced and transferred independently from the operation and the alarm system.

The requirements of the **TL** Rules for Electrical Installations (Chapter 105), Section 9, B. have to be complied with analogously.

1.4 Data transmission (podded drives only)

Within the approval process of the slip ring unit, a check of the data transmitting system has to be included. Special emphasis shall be put on the influence of the special electromagnetic environment.

If signals are transferred via components of the slip ring unit, those components have to be realised redundantly. A fault in a signal line has to be alarmed.

1.5 If for podded drives critical situations may occur, they have to be indicated by a two stage alarm. The crew shall be enabled to identify the problem and to take actions.

2. Further requirements

2.1 For further requirements of the electrical installations see the **TL** Rules for Electrical Installations (Chapter 105), Section 13.

2.2 Fire alarm (podded drives only)

The area above the slip ring unit and accessible areas have to be equipped with smoke detectors.

2.3 Accessible areas

Accessible areas, normally used for maintenance work, have to be equipped with sufficient illumination and ventilation. The access has to be locked to avoid any hazard to the staff.

H. Testing and Trials

1. Quality assurance and inspection plan

Testing and supervision have to be performed according to the quality assurance and inspection plan approved by **TL.**

2. Testing and supervision during construction

2.1 Material certification and approval of components

The certification of the material for essential components and their inspection and testing are summarized in Table 7B.3.

2.2 Testing of power units for propulsion or steering

2.2.1 The power units are required to undergo tests on a test stand in the manufacturer's works.

For diesel engines see Section 3.

For electric motors see **TL** Rules for Electrical Installations (Chapter 105), Section 14.

2.2.2 For hydraulic pumps and motors, the **TL** Guidelines for the Design, Construction and Testing of Pumps are to be applied analogously. Where the drive power is 50 kW or more, this testing is to be carried out in the presence of a **TL** Surveyor.

2.3 Pressure and tightness tests

Pressure components are to undergo a pressure test.

The test pressure p_p is

$$p_p = 1,5 \cdot p_{e,perm}$$

$p_{e,perm}$ = is the maximum allowable working pressure [bar] = the pressure at which the relief valves open. However, for working pressures above 200 bar the test pressure need not exceed p + 100 bar

For pressure testing of pipes, their valves and fittings, see the **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 8.

Tightness tests are to be performed on components to which this is appropriate (e.g. gondola of a podded drive).

2.4 Final inspection and operational test

2.4.1 After inspection of the individual components and completion of assembly, azimuthing propulsors are to undergo a final inspection and operational test at the manufacturer’s premises. The final inspection is to be combined with a trial run lasting several hours under part or full-load conditions. A check of the tooth clearance and of the tooth contact pattern is to be carried out, as far as applicable.

2.4.2 If no suitable test bed is available for the operational and load testing of azimuthing propulsors, the tests can be carried out on the occasion of the dock test or sea trials.

2.4.3 Limitations on the scope of the tests require **TL**'s consent.

3. Sea trials

3.1 The scope of sea trials for azimuthing and podded propulsors is defined in Table 7B.4.

3.2 The faultless operation, smooth running and bearing temperatures of the gears and control system are to be checked during the sea trials under all steering conditions.

3.3 After the conclusion of the sea trials, the toothing is to be examined through the inspection openings and the contact pattern is to be checked. The tooth contact pattern is to be assessed on the basis of the reference values for the percentage area of contact given in Section 6, Table 6.6.

The scope of the check on contact pattern following the sea trials may be limited with the Surveyor's agreement provided that the checks on contact pattern called for in 2.4.1 and 2.4.2 have been satisfactory.

3.4 After successful sea trials the final motor parameters and all other relevant data of the podded drives have to be stamped on the podded drive poster. The power has to be calculated for the highest water and engine room temperature, based on the sea trial data. The calculation has to be submitted to **TL** for approval.

Table 7 B.3 Certification and approval of components

Component, assembly group (1)	Certificate		Special aprroval tests
	Material	Component, final	
Steering foundation plate	B	B	
Slewing bearing	B	A	Ultrasonic test, true running test, bearing tolerances
Planetary gear for slewing motor	B	A	Final testing and inspection
Slewing gear	A	A	Contact pattern, visual inspection
Hydraulic power pack - for steering motors - for CPP	A	A	Pipes and hoses see **TL** Rules for Machinery (Chapter 4). Function test, pressure test
Hydraulic motor Azimuthing drive	A	A	See **TL** Rules for Machinery Installations (Chapter 4). Function test, pressure test
Converter and power transmission for azimuthing motor	A	A	See **TL** Rules for Electrical Installations (Chapter 105), Section 2, C.
Electric motor Azimuthing drive	A	A	See **TL** Rules for Electrical Installations (Chapter 105), Section 14, B.
Azimuthing locking device	B	B	Final inspection and functional test
Cone, well section and major housing parts transmitting thrust	A metal plate	A	Final inspection after heat treatment and sand blasting
Pod cone complete with electrical and hydraulic installation		A	Final inspection by **TL**, FAT
Pod casing (gondola)	Raw casting: A	A	Visual inspection, pressure and ultrasonic test After machining surface crack detection test and dimension protocol
Slewing seal	B	A	Same as stern tube seal
Electric propulsion motor, complete	A	A	Supervision during construction by **TL**, inspections to be agreed based on QA plan of maker, see **TL** Rules for Electrical Installations (Chapter 105), Section 14, B.
Propulsor electric, transmission system, slip ring (podded drive)	A	A	Supervision during construction by **TL**, inspections to be agreed based on QA plan of maker, see **TL** Rules for Electrical Installations (Chapter 105), Section 13.
Gear parts, transmitting propulsion torque	A	A	See Section 6
Propeller shaft	A	A	See Section 5
Shaft coupling	A	A	Supervision of mounting by **TL**

Table 7 B.3 Certification and approval of components (continued)

Component, assembly group (1)	Certificate		Special aprroval tests
	Material	Component, final	
Propeller shaft locking device	B	B	Manufacturer Inspection Certificate, functional test by **TL**
Shaft (roller-)bearing + housing, etc	B	B	Manufacturer Inspection Certificate
Propeller shaft seal	B	A	Pressure test
Propeller, controllable, build-up fixed pitch or monoblock	A	A	See Section 7A
Complete azimuthing propulsor		A	Functional and tightness test
Auxiliary equipment			
Pressure vessel	A	A	See **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 16
Emergency propeller shaft sealing device	B	A	See Section 5
Hydraulic pipes and hoses	B / A	A	Pipes and hoses see **TL** Rules for Machinery Installations (Chapter 4). Pressure test
Sensors	Type approved		Physical functional test in mounted condition (FAT) by **TL**
Further components are to be tested and inspected as specified in the relevant **TL** Rules and to the satisfaction of the **TL** Surveyor			
(1) *As far as applicable for actual of azimuthing propulsor* *A: **TL** Material / Inspection Certificate, B: Manufacturer Material / Inspection Certificate*			

Table 7 B.4 Scope of sea trials

Test group/test	Sub-test	Test remarks
1. Slewing mechanism	a) Testing of sensor for angle measurement in sea/ harbour mode	
	b) Testing of switch over from sea to harbour mode and vice versa	Ship speed, rpm, steering angle, etc. to be checked
	c) Testing of steering angle limitation (sea / harbour mode) and associated reductions	Reductions: power, rpm
	d) Testing whether extreme, but possible manoeuvres could endanger the ship or not	Verification or correction of limitations
	e) Testing of locking of steering drive and emergency positioning	
	f) Testing of steering drive failure	Check of auxiliary drive, brake and overload clutch
	g) Wear down measurement of the slewing bearing	As basic measurement
2. Steering manoeuvres	Steering manoeuvres acc. to **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 2, B.	Compare also SOLAS II-1, Reg. 29
3. Run out test	Check of ships course stability and down to which speed an inactive azimuthing propulsor can influence the course of the ship	
4. Propeller shaft lock		
5. Loss of propulsion of one pod		
6. Crash stop	a) Testing to fulfil requirements acc. to SOLAS	
	b) Testing under adverse conditions	Manual operation
	c) Testing of the automated procedure	Only as far as applicable
7. Endurance test		
8. Test of motor reversing		
9.Manoeuvring tests according to **TL** rules		**TL** Guidelines for Sea Trials of Motor Vessels
10. Test of local control stations		
11. Test of operational condition monitoring	a) Measured and monitored data have to be recorded during the complete sea trials	Power, bearing temperature, vibrations, etc. to be measured
	b) Quality of electrical current	**TL** Rules for Electrical Installations (Chapter 105)
	c) Envelope of bearing vibrations	Podded drives only

SECTION 8

TORSIONAL VIBRATIONS

A. General

1. Scope

The requirements of this Section apply to the components of the main shafting system and to essential equipment, compare Section 1, B.5.

These rules may be applied analogously for rudder propeller units driven by internal combustion engines. They are not applicable for electrically driven azimuthing propulsors (fixed or turnable).

2. Definitions

2.1 For the purposes of these Rules, torsional vibration stresses are additional loads due to torsional vibrations. They result from the alternating torque which is superimposed on the mean torque.

2.2 The speed range in which the plant can be operated continuously is the service speed range. It covers the range between n_{min} (minimum speed) and 1,05 n_N (nominal speed).

B. Calculation of Torsional Vibrations

1. A torsional vibration analysis covering the torsional vibration stresses to be expected in the main shafting system including its branches is to be submitted to **TL** for examination. The following data shall be included in the analysis:

Input data:

- Equivalent torsional vibration system comprising moments of inertia and inertialess torsional elasticities/stiffnesses for the complete system

- Prime mover
  engine type, rated power, rated speed, cycles per revolution, design (in line, V-type, etc.), number of cylinders, firing order, cylinder diameter, stroke, stroke to connecting rod ratio, oscillating mass of one crank gear, excitation spectrum of engine in form of tangential coefficients (for new/unconventional types of engines)

- Vibration dampers
  type, damping coefficient, moments of inertia, dynamic stiffness

- Elastic couplings
  type, damping coefficient, moments of inertia, dynamic stiffness

- Reduction/power intake off (PTO) gears
  type, moment of inertia for wheels and pinions, individual gear's ratios per mesh, effective stiffness

- Shafting
  shaft diameter of crankshafts, intermediate shafts, gear shafts, thrust shafts and propeller shafts

- Propeller
  type, diameter, number of blades, pitch and expanded area ratio, moment of inertia in air, moment of inertia of entrained water (for Zero and full pitch for CP propellers)

Output data/results:

- Natural frequencies
  with their relevant vibration forms (modes)

- Forced vibratory loads (torques or stresses)
  estimated torsional vibration torques/shear stresses in all important elements of the system with particular reference to clearly defined resonance speeds for the whole operating speed range. The results shall include the synthesised values (vectorial sum over all harmonics) for the torques/stresses.

2. The calculations are to be performed both for normal operation (uniform pressure distribution over all cylinders or small deviations in the pressure distribution e.g. ± 5 %) and misfiring operation (one cylinder without ignition, compression of the cylinder still existing).

3. Where the installation allows various operation modes, the torsional vibration characteristics are to be investigated for all possible modes, e.g. in installations fitted with controllable pitch propellers for zero and full pitch, with power take off gear integrated in the main gear or at the forward crankshaft end for loaded and idling condition of the generator unit, with clutches for engaged and disengaged branches.

4. The calculation of torsional vibrations shall also take account of the stresses/torques resulting from the superimposition of several harmonics (synthesized values) so far relevant for overall assessment of the system.

5. If modifications are introduced into the system which have a substantial effect on the torsional vibration characteristics, the calculation of the torsional vibrations is to be repeated and submitted for approval.

6. Where an electrical machine, e.g. static converter controlled motors, can generate periodic excitation leading to relevant torsional vibration stresses in the system as a whole, this is to be taken into account in the calculation of the forced torsional vibration. The manufacturer of the electrical machine is responsible for defining the excitation spectrum in a suitable manner for performing forced torsional vibration calculations.

C. Permissible Torsional Vibration Stresses

1. Shafting

In no part of the shafting may the alternating torsional vibration stresses exceed the following values of τ_1 for continuous operation or of τ_2 under transient conditions. Fig. 8.1 indicates the τ_1 and τ_2 limits as a reference for intermediate and propeller shafts of common design and for the location deemed to be most severely stressed (c_K = 0,55 or c_K = 0,45 for propeller shafts, and c_K = 1,0 and c_K = 0,8 for intermediate shafts). The limits depend on the design and the location considered and may in particular cases lie outside the indicated ranges according to Fig. 8.1. They are to be determined in accordance with equations (1) - (4) and Table 5.1.

Speed ranges in the $n/n_o \leq 0{,}8$ area, in which the permissible values of τ_1 for continuous operation are exceeded shall be crossed through quickly (barred speed ranges for continuous operation), provided that the limit for transient operation τ_2 is not exceeded.

$$\tau_1 = \pm C_W \cdot C_K \cdot C_D \cdot (3 - 2 \cdot \lambda^2) \quad [\mathrm{N/mm^2}] \qquad (1)$$

For speed ratio values $\lambda < 0{,}9$

$$\tau_1 = \pm C_W \cdot C_K \cdot C_D \cdot 1{,}38 \quad [\mathrm{N/mm^2}] \qquad (2)$$

where $0{,}9 \leq \lambda \leq 1{,}05$

$$\tau_2 = \pm 1{,}7 \cdot 6{,}0 \cdot \tau_1 / \sqrt{C_K \cdot C_W} \quad [\mathrm{N/mm^2}] \qquad (3a)$$

Alternatively and depending on the material and design the following formula may be used instead (3a)

$$\tau_2 = \pm 1{,}7 \cdot \tau_1 / \sqrt{C_K} \qquad (3b)$$

d = Shaft diameter, [mm]

λ = Speed ratio [-] = n/n_o

n = Speed, [$\mathrm{min^{-1}}$]

n_o = Nominal speed, [$\mathrm{min^{-1}}$]

R_m = Tensile strength of shaft material, [$\mathrm{N/mm^2}$]

c_W = Material factor. [-]

$$= \frac{R_m + 160}{18} \qquad (4)$$

For the purpose of the formulas (1), (2), (3a), (3b) the tensile strength calculation value applied shall not exceed the following limits:

R_m = 600 N/mm²

- For propeller shafts in general

- For other shafts particularly intermediate shafts, made of forged, low alloy carbon or carbon manganese steel

R_m = 800 N/mm²

- For all shafts except propeller shafts made of forged high alloy steels. Formula (3a) should be applied in conjunction with such steels and special design features only

c_D = size factor [-]

= $0{,}35+0{,}93 \cdot d^{-0.2}$

c_K = Form factor [-]

For intermediate and propeller shafts depending on details of design and construction of applied mechanical joints in the shaft line. The value of c_K is shown in Table 5.1.

2. Crankshafts

2.1 Crankshafts have to be designed according to Section 3, D. For application of this guideline a gas pressure distribution in the cylinder over the crank angle is submitted by the maker of the engine. The maker of the engine also applies for approval of a maximal additional (vibratory) shear stress, which is referred to the crank with the highest load due to mean torque and bending forces. Normally this approved additional shear stress may be applied for first evaluation of the calculated vibratory stresses in the crankshaft via the torsional vibration model. Common values are between 30 and 90 N/mm² for medium and high speed engines; but special confirmation of the value considered for judgement by **TL** is necessary.

For further details see also Section 3, C.1.

2.2 When the generally approved limit for the vibratory stresses for the crankshaft of the engine as defined under 2.1 is exceeded, special considerations may be applied to define a higher limit for the special investigated case. For this detailed system calculations (combined axial / torsional model) and application of the actual calculated data within the model in accordance to the **TL** Guidelines for the Calculation of Crankshafts for Internal Combustion Engines.

2.3 Torsional vibration dampers which are aiming to reduce the stresses in the crankshaft shall be suitable for use for diesel engines. **TL** reserve the right to call for proof of this, compare also F.

Torsional vibration dampers shall be capable of being checked for performance ability in the assembled condition or shall be capable of being dismounted with reasonable ease for checking purposes. This requirement does not apply for small medium or high speed engines, so far the exchange of the damper is a part of the regular service of the engine and a fixed exchange interval is part of the engine's crankshaft approval.

3. Gears

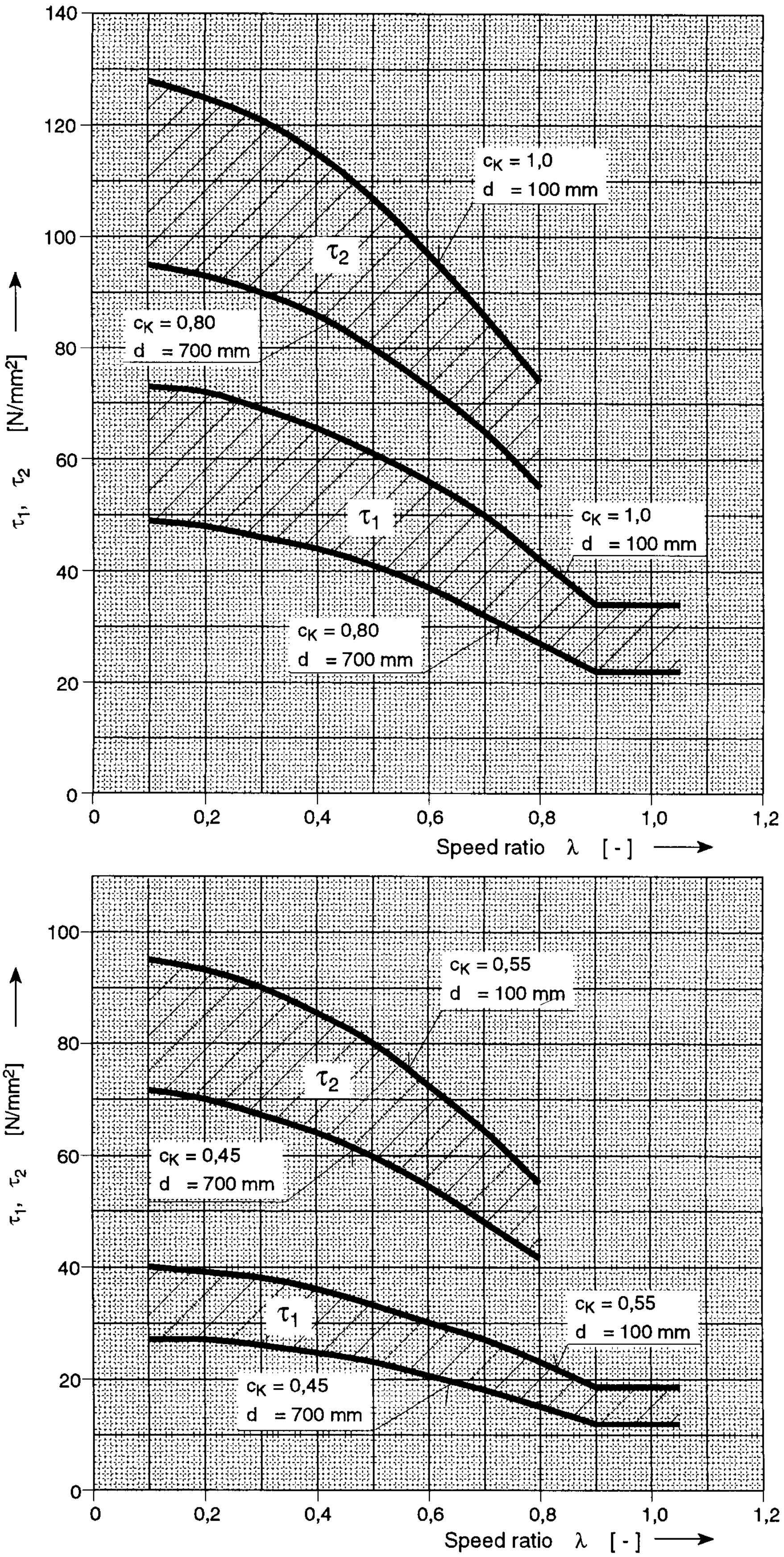
3.1 In the service speed range $0{,}9 \leq \lambda \leq 1{,}05$, no alternating torque higher than 30 % of the mean nominal torque for this stage shall normally occur in any loaded gear's mesh. In general the value for the maximum mean torque transmitted by the gear stage has to be applied for evaluation purposes as the mean nominal torque.

If the gearing is demonstrably designed for a higher power, then, in agreement with **TL,** 30 % of the design torque of the concerned gear's mesh may be applied as admissible.

3.2 When passing through resonant speeds below the operational speed range during starting and stopping of the plant, the alternating torque in the gear shall not exceed twice the nominal mean torque for which the gear has been designed.

3.3 Load reversal due to alternating torques is normally permitted only while passing through the lower speed range up to $\lambda \leq 0{,}35$.

If, in special cases, gear hammering within the operational speed range, is unavoidable, a barred speed range in accordance with E.1. is to due to be specified. This requirement does not apply to gear stages which run without load (e.g. the idling stage of a reversing gear or the idling gears of an unloaded shaft-driven generator). These are covered by the provisions in accordance to 3.4.



Intermediate shafts

Propeller shafts

Fig. 8.1 Permissible torsional vibration stresses in shafting systems is accordance with formulas (1)-(3) for shaft materials with a tensile strength of 450 N/mm^2

3.4 In installations where parts of the gear train run without load, the torsional vibration torque in continuous operation shall not exceed 20 % of the nominal torque in order to avoid unacceptable stresses due to gear hammering.

This applies not only to gear stages but also to parts which are particularly subject to torsional vibrations (e.g. multiple-disc clutch carriers). For loaded parts of the gear system the provisions in accordance to 3.1 apply.

Higher alternating torques may be approved by **TL** if proof is submitted that measures have been introduced considering these higher loadings see 3.1.

4. Flexible Couplings

4.1 Flexible couplings must be designed to with stand the torsional vibration loads which occur in the operation of the ship. In this context, the total load resulting, in accordance with B.4., from the superimposition of several orders is to be taken into account, see also Section 6.

4.2 Flexible couplings must be capable to take in higher alternating torque which can occur during deviation from normal operation according to B.2, during continuous operation within the service speed range.

Speed ranges within which, under abnormal operating conditions, continuous operation is not allowed shall be indicated in accordance with E.2.

5. Shaft-Driven Generators

5.1 In installations with generators directly coupled to the engine (free crankshaft end) it is necessary to ensure that the accelerations do not exceed the values prescribed by the manufacturer in any part of the generator.

The applicable criterion is the tangential acceleration, which is the product of the angular acceleration and the effective radius. The angular acceleration is determined by means of forced torsional vibrations calculations and is to be regarded as the synthesised value of all major orders. However, for marked points of resonance the value of the individual harmonics may be used instead as the basis for assessment.

5.2 The torsional vibration amplitude (angle) of shaft-driven generators shall not normally exceed an electrical value of ±5°. The electrical vibration amplitude is obtained by multiplying the torsional vibration amplitude by the number of pole pairs. Whether **TL** is able to permit higher values depends on the configuration of the ship's electrical system.

6. Connected Units

6.1 If further units, e.g. power turbines or compressors, are coupled positively or non-positively to the main propulsion system, due attention is to be paid to these when establishing the torsional vibration loadings.

In the assessment of their dynamic loads, the limits laid down by the respective manufacturers are to be considered in addition to the factors mentioned in 1. If these limits are exceeded, the units concerned are to be disengaged or prohibited ranges of operation in accordance with E.1 are to be declared. Disengaging of these units shall in general not lead to an overload of the main system in terms of exceeding the τ_2 limit for shafting systems, the maximum torque for flexible couplings or similar low cycle criteria for other components..

6.2 In particularly critical cases, the calculations of forced torsional vibrations, also for disturbed operation (uncoupled set), as stated in B.1. are to be submitted to **TL.** In such cases **TL** reserve the right to stipulate the performance of confirmatory measurements, see D., also for disturbed operation.

D. Torsional Vibration Measurements

1. During the ship's sea trials, the torsional vibrations of the propulsion plant are to be measured over the whole operating range. Measuring investigations shall cover the normal as well as the misfiring condition. Speed ranges, which have been declared as barred speed ranges in accordance with E.1. for misfiring operation shall not be investigated by measurements, as far as these ranges are finally declared as "barred" on the base of reliable and approved calculations and adequately documented.

Measurements are required by **TL** for all plants with a nominal torque exceeding 40 kNm. For other plants not meeting this condition, **TL** reserve the right to ask for measurements depending on the calculation results. The requirement for measurements will be communicated to the yard/engine supplier with the approval letter for the torsional vibration calculation.

Where measurements of identical propulsion plants (specifically sister ships) are available, further torsional vibration measurements for repeat ships may, with the consent of **TL**, be dispensed with. In case that the measuring results are not conclusive enough in respect to the calculations, **TL** reserves the right to ask for further investigations or new approval of a revised calculation model. .

2. Where existing propulsion plants are modified, **TL** reserve the right to require a renewed investigation of the torsional vibration characteristics.

E. Prohibited Ranges of Operation

1. Operating ranges, which due to the magnitude of the torsional vibration stresses and/or torques may only be passed through quickly (transient operation), are to be indicated as prohibited ranges of operation by red marks on the tachometer or in some other suitable manner at the operating station.

In normal operation the speed range $\lambda \geq 0{,}8$ is to be kept free of prohibited ranges of operation.

In specifying prohibited ranges of operation it has to be observed that the navigating and manoeuvring functions are not severely restricted. The width of the barred speed range(s) is (are) to be selected in a way that the stresses in the shafting do not exceed the permissible τ_1 limit for continuous operation with an adequate allowance considering the inaccuracies of the tachometers and the speed setting devices. For geared plants the barred speed ranges, if any, refer to the gear meshes and elastic couplings and are to be determined in the same way with reference to the permissible vibratory torques or permissible power loss for these components (see also C.4. and C.5.).

2. Measures necessary to avoid overloading of the propulsion plant under abnormal operating conditions are to be displayed on instruction plates to be affixed to all engine control stations.

F. Auxiliary Machinery

1. Essential auxiliary machinery such as diesel generators, bow thrusters and other units driven by internal combustion engines shall be designed in a way that the service speed range is free of unacceptable stresses due to torsional vibrations in accordance with C.

2. Generators

2.1 The generating set shall show torsional vibration levels which are compatible with the allowable limits for the alternator, shafts, coupling and damper.

2.2 The coupling selection for the generating set shall take into account the stresses and torques imposed on it by the torsional vibration of the system.
For diesel generator sets with a mechanical output of more than 110 kW torsional vibration calculations shall be submitted to **TL** for approval. The investigations shall include natural frequencies as well as forced vibration calculations. The speed range 90 % to 105 % of the nominal speed shall be investigated under full load conditions (nominal excitation).

2.3 For rigidly coupled generators (without elastic coupling) the vibratory torque in the input part of the generator's shaft shall not exceed 250 % of the nominal torque. For the purposes of these Rules nominal torque is the torque which can be calculated by applying the actual data of the diesel engine (nominal output / nominal speed).

The compliance of the limit of 250 % within the speed range 90 % to 105 % of the nominal speed shall be proven. The calculation for this speed range shall be carried out by using the excitation corresponding to the nominal torque of the engine.

Exceeding the limit of 250 % may be considered in exceptional cases, provided that the generator's manufacturer has designed the generator for a higher dynamical torque. But also in such cases a highest value of 300 % of the actual nominal torque of the set as defined above shall not be exceeded.

3. Bow thruster

3.1 For bow thrusters as well as for other essential auxiliary machinery driven by diesel engines with a mechanical output higher than 150 kW, natural as well as forced torsional vibration calculations must be submitted to **TL** for approval. The torsional vibration calculation must focus on the real load profile of the set.

3.2 For bow thrusters as well as for further essential auxiliary machinery driven by electrical motor the supplier shall take care that relevant excitation forces (e.g. propeller blade frequency or similar), may not lead to unacceptable torsional vibration loadings. In special cases **TL** may require the submission of corresponding calculations.

SECTION 9

MACHINERY FOR SHIPS WITH ICE CLASSES

A. General

1. Notation ICE-B affixed to the Character of Classification

The machinery of naval ships strengthened for navigation in drift ice in the mouth of rivers and in coastal regions is designated after the Character of Classification by the additional Notation **ICE-B**, provided that the rules in B. are satisfied.

2. Measures for Other Conditions of Navigation in Ice

2.1 The requirements for ice classes **ICE-B1, ICE-B2, ICE-B3** and **ICE-B4** are equivalent to the relevant Finnish-Swedish ice classes IC, IB, IA and IA super and are defined in the **TL** Machinery Rules, Chapter 4, Section 13.

2.2 Class Notations **PC1** to **PC7** for polar class ships may be assigned if the requirements which are defined in Part C, Chapter 33, **TL** Guidelines for the Construction of Polar Class Ships are fulfilled.

2.3 The additional requirements for special deck and machinery equipment necessary for operation in ice are defined in the **TL** Rules for Ship Operation Installations and Auxiliary Systems (Chapter 107), Section 19. Ships meeting these requirements may be assigned the Class Notation **ICEOPS** affixed to their Character of Classification.

2.4 Measures for conditions of navigation in ice, different from the conditions relevant for 1., and 2.1 to 2.3 may be agreed with **TL** case by case.

B. Requirements for Notation ICE-B

1. Necessary Propulsion Power

The rated output of the main engines in accordance with Section 3, A.3. must be such to cover the power demand of the propulsion plant for the ice class contitions under consideration and for continuous service.

2. Propeller Shafts, Intermediate Shafts, Thrust Shafts

2.1 General

The necessary propeller shaft reinforcements in accordance with formula (1), in conjunction with the formulae and factors specified in Section 5, C.3., apply to the area of the aft stern tube bearing or shaft bracket bearing from the forward end of the propeller cone or the aft propeller shaft coupling flange subject to a minimum axial distance of 2,5 · d.

The diameter of the adjoining part of the propeller shaft to the point where it leaves the stern tube may be designed by applying an ice class reinforcement factor reduced by 15 % as calculated by formula (2).

The portion of the propeller shaft located forward of the stern tube can be regarded as an intermediate shaft. Intermediate and thrust shafts do not need to be strengthened.

2.2 Reinforcements

$$d_E = C_{EW} \cdot d \qquad (1)$$

d_E = increased diameter of propeller shaft [mm]

d = Shaft diameter according to Section 5, C.3. [mm]

C_{EW} = Ice class strengthening factor, [-]

$$C_{EW} = c \cdot \sqrt[3]{1 + \frac{85 \cdot m}{P_W^{0,6} \cdot n_2^{0,2}}} \geq 1,0 \qquad (2)$$

P_W = Main engine power, [kW]

n_2 = Propeller shaft speed in rev/min, [min^{-1}]

m = Ice class factor, [-]

= 8

c = 0,7 for shrink fits in gears, [-]

= 0,71 for the propeller shafts of fixed-pitch propellers,

= 0,78 for the propeller shafts of controllable pitch propellers

In the case of ducted propellers, the values of c can be reduced by 10 %.

3. Shrunk Joints

When designing shrink fits in the shafting system and in gearboxes, the necessary pressure per unit area p_E [N/mm²] is to be calculated in accordance with the following formula (3).

$$p_E = \frac{\sqrt{\theta^2 \cdot T^2 + f \cdot (c_A^2 \cdot c_e^6 \cdot Q^2 + T^2)} - \theta \cdot T}{A \cdot f} \qquad (3)$$

T has to be introduced as positive value, if the propeller thrust increases the surface pressure at the taper. Change of direction of the axial force is to be neglected as far as performance and thrust are essentially less.

T has to be introduced as negative value, if the axial force reduces the surface pressure of the taper, e.g. for tractor propellers.

$$f = (\mu_0 / S) - \theta^2 \qquad (4)$$

c_A = see Section 5

$$c_e = 0{,}89 \cdot C_{EW} \geq 1{,}0 \qquad (5)$$

C_{EW} = To be calculated according to 2.2 the higher value of the connected shaft ends has to be taken for the coupling

Other symbols in accordance with Section 5, D.4.

4. Propellers

4.1 General

The propellers of ships with ice class **ICE-B** must be made of the cast copper alloys or cast steel alloys specified in Section 7 A.

4.2 Strengthening

4.2.1 Blade sections

$$t_E = C_{EP} \cdot t \qquad (6)$$

t = blade section thickness in accordance with Section 7A C.2 [mm]

t_E = Increased thickness of blade section, [mm]

If $C_{EP} \leq C_{Dyn}$ then

$$t_E = t$$

If $C_{EP} > C_{Dyn}$ then

$$t_E = \frac{C_{EP}}{C_{Dyn}} \cdot t$$

C_{EP} = Ice class strengthening factor, [-]

$$C_{EP} = f \cdot \sqrt{1 + \frac{21 \cdot z \cdot m}{P_W^{0,6} \cdot n_2^{0,2}}} \geq 1{,}0 \qquad (7)$$

f= 0,62 for solid propellers

f= 0,72 for controllable pitch propellers

In case of ducted propellers, the values of f may be reduced by 15%.

z = Number of blades, [-]

C_{Dyn} = Dynamic factor [-] in accordance with Section 7A, formula (3).

m, P_W, n_2 see 2.2.

4.2.2 Blade tips

$$t_{1,0E} = \sqrt{\frac{500}{C_W}} \cdot (0{,}002 \cdot D + t') \qquad (8)$$

$t_{1,0E}$ = Strengthened blade tip, [mm] t' = Increase in thickness, [mm]

= 10 for ice class **ICE-B**

D = Propeller diameter, [mm]

C_w = Material factor [N/mm²] in accordance with Section 7A, C.1, Table 7A.1.

In case of ducted propellers, the thickness of the blade tips may be reduced by 15 %.

4.2.3 Leading and trailing edges

For ice class **ICE-B** the thickness of the leading and trailing edges of solid propellers and the thickness of the leading edge of controllable pitch propellers must be equal to at least 35 % of the blade tip $t_{1,0E}$ when measured at a distance of 1,25 · $t_{1,0E}$ from the edge of the blade.

For ducted propellers, the strengthening at the leading and trailing edges has to be based on the non-reduced tip thickness according to formula (8).

4.2.4 Blade wear

If the actual thickness in service is below 50 % at the blade tip or 90 % at other radii of the values obtained from 4.2.1 and 4.2.2, respective counter measures have to be taken. Ice strengthening factors according to 4.2.1 and 4.2.2 will not be influenced by an additional allowance for abrasion.

Note

If the propeller is subjected to substantial wear, e.g. abrasion in tidal flats, a wear allowance should be added to the blade thickness determined in order to achieve an adequate service time with respect to blade wear.

4.2.5 Propeller mounting

Where the propeller is mounted on the propeller shaft by the oil injection method, the necessary pressure per unit area p_E [N/mm²] in the area of the mean taper diameter is to be determined by formula (9).

$$p_E = \frac{\sqrt{\theta^2 \cdot T^2 + f \cdot (c_A^2 \cdot c_e^6 \cdot Q^2 + T^2)} - \theta \cdot T}{A \cdot f} \qquad (9)$$

T has to be introduced as positive value, if the propeller thrust increases the surface pressure at the taper. Change of direction of the axial force is to be neglected as far as performance and thrust are essentially less.

T has to be introduced as negative value, if the propeller thrust reduces the surface pressure of the taper, e.g. for tractor propellers.

$$f = (\mu_0 / S) - \theta^2 \qquad (10)$$

c_e = ice class reinforcement factor [-] in accordance with formula (5)

Other symbols in accordance with Section 7A, C.1.

In the case of flanged propellers, the required diameter d_{sE} of the alignment pin is to be determined by applying formula (11).

$$d_{sE} = C_{EW}^{1,5} \cdot d_s \qquad (11)$$

d_{sE} = reinforced root diameter of alignment pin [mm]

d_s = diameter of alignment pin for attaching the propeller [mm] in accordance with Section 5, D.4.3.

C_{EW} = ice class reinforcement factor in accordance with formula (2) [-]

5. Gears

Gears in the main propulsion plant of ships with ice class **ICE-B** are not to be strengthened.

6. Sea Chests and Discharge Valves

Sea chests and discharge valves are to be designed in accordance with Chapter 107-Ship Operation Installations and Auxiliary Systems, Section 8.

7. Steering gear

The dimensional design of steering gear components is to take account of the rudderstock diameter specified in Chapter 102 - Hull Structures and Ship Equipment, Section 12.

8. Electric Propeller Drive

Where electric propeller drives are used, the conditions set out in Chapter 105 - Electrical Installations, Section 13 must be fulfilled.

SECTION 10

SPARE PARTS

A. General

1. In order to be able to restore engine operation and manoeuvring capacity to the ship in the event of damage at sea spare parts for the main drive and the essential equipment (see Section 1, B.5.) are to be carried on board every ship, together with the necessary tools.

These requirements are considered to be complied with if the range of spare parts corresponds to the following Tables considering the extent of the actually installed systems and components at the time of commissioning.

2. Depending on the design and arrangement of the engine plant, the intended service and operation of the ship, and also the manufacturer's recommendations, a different volume of spare parts may be agreed between the Naval Authority and **TL**.

Where the volume of spare parts is based on special arrangements between the Naval Authority and **TL,** technical documentation is to be provided.

A list of the relevant spare parts is to be carried on board.

3. In the case of propulsion systems and essential equipment which are not included in the following tables, the requisite range of spare parts is to be established in each individual case between Naval Authority, shipyard and **TL.**

4. Assessment of the recommended spare parts to be carried onboard ship can be determined through risk assessment and is to be agreed by the Naval Authority and TL. For details, refer to Part B, Chapter 4, Machinery, Section 17.

B. Volume of Spare Parts

The volume of spare parts has to be in accordance with the following Tables.

A = Unlimited range of service and **Y**
B = All other ranges of service

Explanations:

Restricted International Service – Y

This range of service is limited, in general, to operate along the coast, provided that the distance to the nearest port of refuge and the offshore distance do not exceed 200 nautical miles. This applies also to operation in the North Sea and within enclosed seas, such as the Mediterranean Sea, the Black Sea, the Caspian Sea and waters with similar seaway conditions.

Coastal Service - K50/K20

This range of service is limited, in general, to operate along the coasts, provided that the distance to the nearest port of refuge and the offshore distance do not exceed 50/20 nautical miles. This applies also to operation within enclosed seas, such as the Baltic Sea, Marmara Sea and gulfs with similar seaway conditions.

Coastal Service – K6

This range of service is limited to operate along the coasts, provided that the distance to the nearest port of refuge and the offshore distance do not exceed 6 nautical miles. This area of service is restricted to operate in shoals, bays, haffs and firths or similar waters, where heavy seas do not occur.

1. Internal Combustion Engines

For internal combustion engines, see Section 3, the volume of spare parts is defined in Tables 10.1 to 10.3.

2. Gas Turbines

For gas turbines see Section 4A, the volume of spare parts is defined in Table 10.4.

3. Gears and thrust bearings

For gears and thrust bearings, see Section 6, the volume of spare parts is defined in Table 10.5.

4. Air Compressors for Essential Services

For air compressors see the **TL** Rules, Chapter 107, Ship Operation Installations and Auxiliary Systems, Section 6, 12 and 18, the volume of spare parts is defined in Table 10.6.

Table 10.1 Spare parts for main engines (1), (4), (5)

Range of spare parts		A	B
Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1	
Main thrust block (integral)	Pads for "ahead" face of Michell type thrust block, or complete white metal thrust shoe of solid ring type	1 set	1 set
		1	1
Connecting rod bearings	Bottom end bearings or shells of each size and type fitted, complete with shims, bolts and nuts, for one cylinders	1 set	-
	Trunk piston type: Gudgeon pin complete with bush/bearing shells and securing rings for one cylinder	1 set	-
Cylinder liner	Cylinder liner, complete, fully equipped and ready for installation, including gaskets	1	-
Cylinder cover	Cylinder cover, complete, fully equipped and ready for installation, including gaskets	1	-
	Cylinder cover bolts and nuts, for one cylinder	¼ set	-
Valves	Exhaust valves, with full equipment and ready for installation, for one cylinder	1 set	1 set
	Inlet valves, with full equipment and ready for installation, for one cylinder	1 set	1 set
	Starting air valve, with full equipment and ready for installation	1	1
	Overpressure control valve, complete	1	1
	Fuel injection valves of each type, ready for installation, for one engine **(2)**	1 set	¼ set
Hydraulic valve drive	High-pressure pipe/hose of each type	1	-
Piston: Trunk piston type	Piston of each type, ready for fitting, with piston rings, gudgeon pin, connecting rod, bolts and nuts	1	-
Piston rings	Piston rings for one cylinder	1 set	-
Piston cooling	Articulated or telescopic cooling pipes and fittings for one cylinder	1 set	-
Cylinder lubricator	Scope of spare parts to be defined with regard to lubricator design and subject to approval	1	-
Fuel injection pumps	Fuel injection pump complete or, when replacement of individual components at sea is practicable, complete pump element with associated valves, seals, springs, etc. or higher pressure fuel pump	1	-
Fuel injection pipes	High pressure fuel pipe of each size and shape fitted, complete with couplings	1	-
Charge air system (3)	Auxiliary blower, complete including drive	1	-
	Exhaust-gas turbocharger: rotor complete with bearings, nozzle rings and attached lube oil pump	1 set	-
	Suction and pressure valves of each type for one cylinder	1 set	-
Gaskets and packings	Special gaskets and packings of each type for cylinder covers and cylinder liners, for one cylinder	-	1 set
Exhaust gas system (engine-related)	Compensator of each type	1	-

(1) *in the case of multi-engine installations, the minimum required spares are only necessary for one engine*

(2) *a) engines with one or two fuel-injection valves per cylinder: one set of fuel valves, complete*

b) engines with more than two fuel injection valves per cylinder: two valves complete per cylinder plus a corresponding number of valve parts (excluding the valve bodies) which make it possible to form a complete spare set by re-using the operational parts of the dismantled valves

(3) *spare parts for exhaust-gas turbocharger and auxiliary blower may be omitted if emergency operation of the main engine after failure is demonstrably possible.*
The requisite blanking and blocking arrangements for the emergency operation of the main engine are to be available on board.

(4) *the necessary tools and equipment for fitting the required spare parts must be available on board*

(5) *spare parts are to be replaced immediately as soon as they are "used-up"*

5. Pumps

For pumps see the **TL** Rules, Chapter 107, Ship Operation Installations and Auxiliary Systems, Section 8, the volume of spare parts is defined in Table 10.7.

6. Hydraulic Systems

For hydraulic systems see the **TL** Rules, Chapter 107, Ship Operation Installations and Auxiliary Systems, Section 14, the volume of spare parts is defined in Table 10.8.

7. Other Spare Parts

For other spare parts for main and auxiliary engines the volume is defined in Table 10.9.

Table 10.2 Spare parts for auxiliary engines driving electric generators for essential equipment

Range of spare parts		A
Main bearings	Bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1
Valves	Exhaust valves, complete with casings, seats, springs and other fittings for one cylinder	2 sets
	Inlet valves, complete with casings, seats, springs and other fittings for one cylin-der	1 set
	Starting air valve, complete with casing, seat, springs and other fittings	1
	Overpressure control valve, complete	1
	Fuel valves of each size and type fitted, complete, with all fittings, for one engine	¼ set
Connecting rod	Bottom end bearings or shells of each type, complete with all fittings	1
	Gudgeon pin with bush for one cylinder	1
Piston rings	Piston rings, for one cylinder	1 set
Fuel injection pumps	Fuel injection pump complete or, when replacement of individual components at sea is practicable, complete pump element with associated valves, seals, springs, etc. or equivalent high pressure fuel pump	1
Fuel injection pipes	High pressure fuel pipe of each size and shape fitted, complete with fittings	1
Gasket and packings	Special gaskets and packings of each size and type fitted, for cylinder covers and cylinder liners for one cylinder	1 set
Control, alarm and safety system	Parts essential for safe engine operation	1 set

Note

1. *Where the number of generating sets (including stand-by units) is greater than called for by the Rules, no spares are required for the auxiliary engines.*
2. *Where several diesel engines of the same type are installed by way of generator drive spare parts are required for one engine only.*
3. *No spares are required for the engines driving emergency generator sets.*

Table 10.3 Spare parts for prime movers of essential equipment other than generators

Range of spare parts
The range of spare parts required for auxiliary drive machinery for essential consumers is to be specified in accordance with Table 10.2

Note

Where an additional unit is provided for the same purpose no spare parts are required.

Table 10.4 Spare parts for gas turbines

Range of spare parts
For essential main propulsion: For each gas turbine a complete set of wear and tear parts, which can be changed on board by a crew without considerable maintenance expertise.
For non-essential propulsion: No spare parts prescribed by **TL.** If applicable, to be defined by manufacturer
For driving of auxiliaries: For two gas turbines each one set of wear and tear parts, which can be changed on board by a crew without considerable maintenance expertise.

Table 10.5 Spare parts for gears and thrust bearings in propulsion plants

Range of spare parts	A	B
Wearing parts of gear-driven pump supplying lubricating oil to gears or	1 set	-
one complete lubricating oil pump if no stand-by pump is available	1	
Thrust pads for ahead side of thrust bearings	1 set	1 set

Table 10.6 Spare parts for air compressors for essential services

Range of spare parts	A	B
Piston ring of each type and size fitted for one piston	1 set	1 set
Suction and delivery valves complete of each size fitted in one unit	½ set	½ set

Table 10.7 Spare parts for pumps

Range of spare parts		A	B
Piston pumps	Valve with seats and springs each size fitted	1 set	1 set
	Piston rings each type and size for one piston	1 set	1 set
Centrifugal pumps	Bearing of each type and size	1	1
	Rotor sealings of each type and size	1	1
Gear and screw type pumps	Bearings of each type and size	1	1
	Rotor sealings of each type and size	1	1
Note *Where, for a system a stand-by pump of sufficient capacity is available, the spare parts may be dispensed.*			

Table 10.8 Spare parts for hydraulic systems

Range of spare parts	A	B
Pressure hoses and flexible pipes, at least one of each size	20 %	20 %
Seals, gaskets	1 set	1 set
Note *For seals, this requirement is applicable only to the extent that these parts can be changed with the means available on board.* *Where a hydraulic system comprises two mutually independent sub-systems, spare parts need to be supplied for one*		

Table 10.9 Other spare parts

Range of spare parts	A	B
Safety valve or one valve cone and spring of each type for pressure vessels	1	1
Hoses and compensators	20%	20%
Testing device for fuel injection valves	1	1
Note *For carrying out maintenance and repair work, a sufficient number of suitable tools and special tools according to the size of the machinery installation must be available on board.*		