RULES FOR BUILDING AND CLASSING

STEEL VESSELS
2010

PART 4
VESSEL SYSTEMS AND MACHINERY

American Bureau of Shipping
Incorporated by Act of Legislature of
the State of New York 1862

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Rule Change Notice *(2010)*

The effective date of each technical change since 1993 is shown in parenthesis at the end of the subsection/paragraph titles within the text of each Part. Unless a particular date and month are shown, the years in parentheses refer to the following effective dates:

(2000) and after	1 January 2000 (and subsequent years)	*(1996)*	9 May 1996
(1999)	12 May 1999	*(1995)*	15 May 1995
(1998)	13 May 1998	*(1994)*	9 May 1994
(1997)	19 May 1997	*(1993)*	11 May 1993

Listing by Effective Dates of Changes from the 2009 Rules

Notice No. 1 (effective on 1 July 2009) to the 2009 Rules, which are incorporated in the 2010 Rules, are summarized below.

EFFECTIVE DATE 1 July 2009 – shown as *(1 July 2009)*
(based on the contract date for new construction between builder and Owner)

Part/Para. No.	*Title/Subject*	*Status/Remarks*
4-3-1A1/21.17	Helix Angle Factor	To align the requirements with the corrigenda to ISO 6336-2 *Calculation of Surface Durability*. (Incorporates Notice No. 1)
4-6-3/9	Manufacturing of Plastic Pipes	To allow for consideration of the requirements in recognized standards for plastic pipe manufacturing. (Incorporates Notice No. 1)
4-6-7/3.5.5	Hydraulic Power Cylinder	To clarify the requirements. (Incorporates Notice No. 1)
4-7-3/5.1.5 (New)	Fixed Foam Systems Using Inside Air	To reflect the use of MSC.1/Circ.1271 for High-Expansion Foam Systems that utilize inside air. (Incorporates Notice No. 1)
4-7-3/5.3.1	System Characteristics	To align the requirements with Annex 3 of IMO resolution MSC.217(82). (Incorporates Notice No. 1)
Appendix 4-7-3A1 (New)	IMO MSC.1/Circ.1271	To incorporate IMO MSC.1/Circ.1271 for ready reference. (Incorporates Notice No. 1)
4-7-3/7.1	Fixed Pressure Water-spraying Fire Extinguishing System	To align the requirements with IMO resolution MSC.217(82). (Incorporates Notice No. 1)
Appendix 4-7-3A2 (New)	IMO MSC/Circ.1165	To incorporate IMO MSC/Circ.1165 for ready reference. (Incorporates Notice No. 1)
4-7-3/7.1.3	Sprinkler Systems Equivalency	To align the requirements with IMO resolution MSC.265(84). (Incorporates Notice No. 1)
Appendix 4-7-3A3 (New)	IMO Resolution A.800(19), as Amended by MSC.265(84)	To incorporate IMO Resolution A.800(19), as amended by MSC.265(84) for ready reference. (Incorporates Notice No. 1)
4-7-3/15.3	Portable Foam Applicators	To align the requirements with Annex 3 of IMO resolution MSC.217(82). (Incorporates Notice No. 1)
4-7-3/Table 4	Equivalent Fire Extinguishers	To align the Table with the text of 4-7-3/15.1.1 and to include Type A dry chemical extinguishers. (Incorporates Notice No. 1)
4-8-4/21.17.1	Emergency and Essential Feeders	To clarify the requirements. (Incorporates Notice No. 1)
4-8-4/21.17.3 (New)	Electrical Cables for the Emergency Fire Pump	To clarify that the electrical cables to the emergency fire pump are not to pass through the machinery spaces containing the main fire pumps and their sources of power and prime movers (such as the main machinery space) and that where the cables pass through other high fire risk areas, the cables are to be of a fire resistant type in accordance with paragraph 2(a) of IACS UR E15. (Incorporates Notice No. 1)

EFFECTIVE DATE 1 January 2010 – shown as *(2010)*
(based on the contract date for new construction between builder and Owner)

Part/Para. No.	*Title/Subject*	*Status/Remarks*
4-1-1/Table 1	Certification Details – Prime Movers	To indicate that the Product Quality Assurance program is not available for diesel engines, steam turbines or gas turbines that are less than 100 kW (135 hp).
4-2-1/7.2.1	General	To require that all diesel engines of 2,250 kW and above or having cylinders of more than 300 mm bore along should be fitted with oil mist monitoring arrangements or equivalent arrangements, regardless of whether or not an automation notation was being pursued, in line with IACS UR M10 and IACS UI 228.
4-3-2/7.3	Shaft Alignment Calculations	To clarify the requirements for alignment verification and to expand the requirements for submission of calculations to shafts of 300 mm (11.81 in.) and greater.
4-3-2/11.1.1	Alignment	To clarify the requirements for alignment verification and to expand the requirements for submission of calculations to shafts of 300 mm (11.81 in.) and greater.
4-3-4/3.3.2	Small Parts of Rudder Actuators	To align the requirements with the material testing requirements in 4-4-1/3.5 and 4-4-1/1.1(v).
4-3-4/5.11	Power Gear Stops	To clarify the requirements.
4-3-5/15.13.2	Trials	To align the requirements with 7-9-6/3 of the ABS *Rules for Survey After Construction*, and to clarify the requirements for the Failure Modes and Effects Analysis (FMEA).
4-6-4/9.3.2(a)	Exposed to Weather	To align the requirements with Regulation 20 of the International Convention on Load Lines, IACS UI LL36, 3-2-17/Table 2, and IACS UR S27.
4-6-4/9.3.5(a)	Protection from Weather and Sea Water Ingress	To align the requirements with Regulation 20 of the International Convention on Load Lines, as amended by MSC.143(77)
4-6-7/5.5.2	Air Quality	To provide requirements for the quality of air supplied to safety and control air systems.
4-6-7/7.5.1(b)	Piping Materials	To align the requirements with NFPA 51, USCG Title 46 CFR 56.50-103, and BS 29593/ISO 9539
4-7-1/7	Plans and Data to be Submitted	To ensure that the emergency fire pumps have sufficient lift capacity (NPSHR vs. NPSHA) and performance at limited suction conditions and to verify compliance with such requirements early in the design phase of the vessel.
4-7-3/1.5.3v)	<No Title>	To provide a safety margin of at least 1 meter (3.3 feet) between the Net Positive Suction Head Available (NPSHA) and Net Positive Suction Head Required (NPSHR).
4-8-1/5.5	Equipment Plans	To clarify the requirements for essential machines of 100 kW and over.
4-8-1/Table 1	Primary Essential Services	To incorporate additional services required to be ABS certified.
4-8-1/Table 2	Secondary Essential Services	To incorporate additional services required to be ABS certified.
4-8-3/1.5	Certification of Equipment	To document that rotating machines, motor controllers, and motor control centers for services indicated in 4-8-3/Table 7 require ABS certification.
4-8-3/3.1	Application	To document that rotating machines, motor controllers, and motor control centers for services indicated in 4-8-3/Table 7 require ABS certification.
4-8-3/3.15.1	Machines to be Tested and Test Schedule	To document that rotating machines, motor controllers, and motor control centers for services indicated in 4-8-3/Table 7 require ABS certification.
4-8-3/3.17	Certification	To document that rotating machines, motor controllers, and motor control centers for services indicated in 4-8-3/Table 7 require ABS certification.

Part/Para. No.	*Title/Subject*	*Status/Remarks*
4-8-3/5.11.1	Certification	To document that rotating machines, motor controllers, and motor control centers for services indicated in 4-8-3/Table 7 require ABS certification.
4-8-3/9.1	Standard of Compliance	To clarify the requirements regarding cable glands designed for use in hazardous areas.
4-8-3/Table 7	Additional Services Requiring Electrical Equipment to be Designed, Constructed and Tested to the requirements in Section 4-8-3	To incorporate additional services required to be ABS certified.
4-9-1/9.7.5	Temporarily Disconnecting Alarms	To incorporate the requirement for reactivation of disabled loops or detectors in fire detection systems for ABCU and ACCU operations, in line with IACS UR F32.8.
4-9-1/11.7	Pneumatic	To provide requirements for the quality of air supplied to pneumatic control equipment.
4-9-4/21.3	Controls at Fire Fighting Station	To clarify the requirements for shutdown of equipment.
4-9-4/21.5.2 (New)	Temporarily Disconnecting Alarms	To incorporate the requirement for reactivation of disabled loops or detectors in fire detection systems for ABCU and ACCU operations, in line with IACS UR F32.8.
4-9-4/Table 3A	Instrumentation and Safety System Functions in Centralized Control Station – Slow Speed (Crosshead) Diesel Engines	To align the requirements with IACS M35. To reflect electronically controlled diesel engine design and the need to monitor the common rail fuel and lube oil pressures. To apply the turbocharger lube oil monitoring requirements regardless of the design type of turbocharger lubrication.
4-9-4/Table 3B	Instrumentation and Safety System Functions in Centralized Control Station – Medium and High Speed (Trunk Piston) Diesel Engines	To align the requirements with IACS M35. To reflect electronically controlled diesel engine design and the need to monitor the common rail fuel and lube oil pressures. To provide specific requirements for turbochargers.
4-9-4/Table 6B	Instrumentation and Safety System Functions in Centralized Control Station – Generator Prime Mover for electric Propulsion	To align the requirements with IACS M35. To reflect electronically controlled diesel engine design and the need to monitor the common rail fuel and servo oil pressures. To apply the turbocharger lube oil monitoring requirements regardless of the design type of turbocharger lubrication.
4-9-4/Table 8	Instrumentation and Safety System Functions in Centralized Control Station – Auxiliary Turbines and Diesel Engines	To align the requirements with IACS M35. To reflect electronically controlled diesel engine design and the need to monitor the common rail fuel and servo oil pressures.

PART

4

Vessel Systems and Machinery

CONTENTS

PART

4

CHAPTER 1 General

CONTENTS

PART
4

CHAPTER 1 General

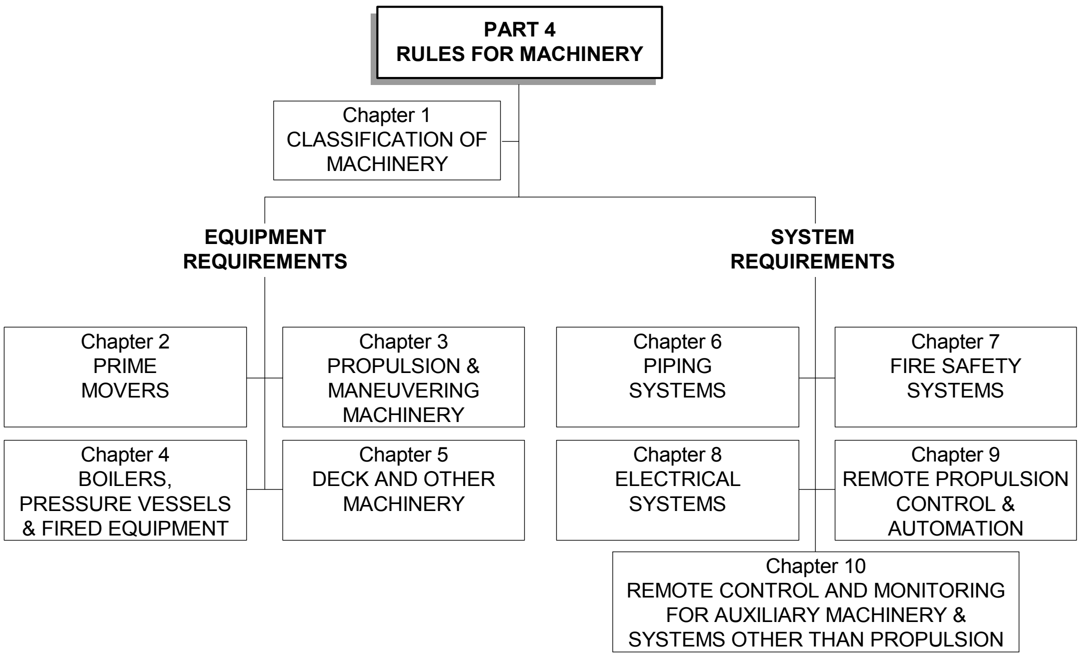
SECTION 1 Classification of Machinery

1 General

1.1 Organization of Part 4

Part 4 contains classification requirements for machinery. These requirements are organized in two broad segments: that specific to equipment, and that specific to systems. 4-1-1/Figure 1 shows the overall organization of Part 4.

FIGURE 1
Organization of Part 4



1.3 Requirements for Classification

1.3.1 Scopes of Part 4 and Part 5C

Part 4 provides the minimum requirements for machinery of self-propelled vessels of 90 meters in length and over. Compliance with Part 4 is a condition for classification of all such vessels, and for assigning the appropriate machinery class notations indicated in 4-1-1/1.5. Additional requirements for machinery, which are specific for each vessel type, are provided in Part 5C. Compliance with the provisions of Part 5C is a condition for assigning the vessel type class notation specified therein, such as **Oil Carrier**, **Passenger Vessel**, **Liquefied Gas Carrier**, etc.

1.3.2 Fundamental Intent of Machinery Rules

1.3.2(a) Propulsion and maneuvering capability. Part 4 of the Rules is intended to assure the propulsion and maneuvering capability of the vessel through specification of pertinent design, testing and certification requirements for propulsion, maneuvering and other equipment and their associated systems. See 4-1-1/Figure 1 for equipment and systems included in the scope.

1.3.2(b) Machinery hazards. Part 4 of the Rules is also intended to identify and address hazards associated with machinery aboard a vessel, particularly those hazards which are capable of causing personal injury, flooding, fire or pollution.

1.3.2(c) Cargo hazards. Hazards associated with cargoes carried (such as oil, dangerous goods, etc.) or to the specialized operations of the vessel (such as navigating in ice) are addressed in Part 5C.

1.3.3 Application

Requirements in Part 4 are intended for vessels under construction; but they are to be applied to alterations made to existing vessels, as far as practicable.

1.5 Classification Notations

Classification notations are assigned to a vessel to indicate compliance with particular portions of the Rules. The following classification notations define compliance with specific requirements of the Rules for machinery:

AMS indicates that a vessel complies with all machinery requirements in Part 4, other than the requirements associated with the other classification notations below. **AMS** is mandatory for all self-propelled vessels.

ACC indicates that in a self-propelled vessel, in lieu of manning the propulsion machinery space locally, it is intended to monitor the propulsion machinery space and to control and monitor the propulsion and auxiliary machinery from a continuously manned centralized control station. Where such a centralized control station is installed, the provisions of Section 4-9-3 are to be complied with. Upon verification of compliance, **ACC** will be assigned.

ACCU indicates that a self-propelled vessel is fitted with various degrees of automation and with remote monitoring and control systems to enable the propulsion machinery space to be periodically unattended and the propulsion control to be effected primarily from the navigation bridge. Where periodically unattended propulsion machinery space is intended, the provisions of Section 4-9-4 are to be complied with. Upon verification of compliance, **ACCU** will be assigned.

APS indicates that a self-propelled vessel is fitted with athwartship thrusters. **APS** is optional for all self-propelled vessels fitted with such thrusters and signifies compliance with applicable requirements of Section 4-3-5.

PAS indicates that a non-self-propelled vessel is fitted with thrusters for the purpose of assisting the movement or maneuvering. **PAS** is only assigned when requested by the Owner and signifies compliance with applicable requirements of Section 4-3-5.

DPS-0, **-1**, **-2**, or **-3** indicates that a vessel, self-propelled or non-self-propelled, is fitted with a dynamic positioning system. The numerals (**-0**, **-1**, **-2** or **-3**) indicates the degree of redundancy in the dynamic positioning system. **DPS** is assigned only when requested by the owners and signifies compliance with 4-3-5/15.

The above class notations, where preceded by the symbol ✠ (Maltese cross; e.g., ✠ **AMS**), signify that compliance with these Rules was verified by the Bureau during construction of the vessel. This includes survey of the machinery at the manufacturer’s plant (where required), during installation on board the vessel and during trials.

Where an existing vessel, not previously classed by the Bureau, is accepted for class, these class notations are assigned without ✠.

1.7 Alternative Standards

Equipment, components and systems for which there are specific requirements in Part 4 may comply with requirements of an alternative standard, in lieu of the requirements in the Rules. This, however, is subject to such standards being determined by the Bureau as being not less effective than the Rules. Where applicable, requirements may be imposed by the Bureau in addition to those contained in the alternative standard to assure that the intent of the Rules is met. In all cases, the equipment, component or system is subject to design review, survey during construction, tests and trials, as applicable, by the Bureau for purposes of verification of its compliance with the alternative standard. The verification process is to be to the extent as intended by the Rules. See also 1-1-1/1.

1.9 Definitions

Definitions of terms used are defined in the chapter, sections or subsections where they appear. The following are terms that are used throughout Part 4.

1.9.1 Control Station

A location where controllers or actuator are fitted, with monitoring devices, as appropriate, for purposes of effecting desired operation of specific machinery.

Control Station is defined exclusively for purposes of Part 4, Chapter 7 "Fire Safety Systems," as intended by SOLAS, in 4-7-1/11.21.

Centralized Control Station is used in Part 4, Chapter 9 "Remote Propulsion Control and Automation" to refer to the space or the location where the following functions are centralized:

- Controlling propulsion and auxiliary machinery,
- Monitoring propulsion and auxiliary machinery, and
- Monitoring the propulsion machinery space.

1.9.2 Machinery Space

Machinery Space is any space that contains propulsion machinery, boilers, oil fuel units, steam and internal combustion engines, generators and major electrical machinery, oil filling stations, air conditioning and ventilation machinery, refrigerating machinery, stabilizing machinery or other similar machinery, including the trunks to the space. Machinery space is to include "machinery space of category A", which, as defined in 4-7-1/11.15, is a space and trunks to that space which contains:

- Internal combustion machinery used for main propulsion; or
- Internal combustion machinery used for purposes other than main propulsion where such machinery has in the aggregate a total power output of not less than 375 kW (500 hp); or
- Any oil-fired boiler (including similar oil-fired equipment such as inert gas generators, incinerators, waste disposal units, etc.) or oil fuel unit (see definition in 4-7-1/11.19).

1.9.3 Essential Services *(2004)*

For definition of essential services, see 4-8-1/7.3.3.

1.9.4 Hazardous Area

Areas where flammable or explosive gases, vapors or dust are normally present or likely to be present are known as hazardous areas. Hazardous areas are, however, more specifically defined for certain machinery installations, storage spaces and cargo spaces that present such hazard, e.g.:

- Helicopter refueling facilities, see 4-8-4/27.3.3;
- Paint stores, see 4-8-4/27.3.3;
- Cargo oil tanks and other spaces of oil carriers; see 5C-1-7/31.5;
- Ro-ro cargo spaces; see 5C-10-4/3.7.2.

1.9.5 Toxic or Corrosive Substances

Toxic Substances (solid, liquid or gas) are those that possess the common property of being liable to cause death or serious injury or to harm human health if swallowed or inhaled, or by skin contact.

Corrosive Substances (solid or liquid) are those, excluding saltwater, that possess in their original stage the common property of being able through chemical action to cause damage by coming into contact with living tissues, the vessel or its cargoes, when escaped from their containment.

1.9.6 Dead Ship Condition *(2004)*

Dead ship condition means a condition under which:

i) The main propulsion plant, boilers and auxiliary machinery are not in operation due to the loss of the main source of electrical power, and

ii) In restoring propulsion, the stored energy for starting the propulsion plant, the main source of electrical power and other essential auxiliary machinery is assumed to not be available.

1.9.7 Blackout *(2004)*

Blackout situation means the loss of the main source of electrical power resulting in the main and auxiliary machinery to be out of operation.

3 Certification of Machinery

3.1 Basic Requirements

The Rules define, to varying degrees, the extent of evaluation required for products, machinery, equipment and their components based on the level of criticality of each of those items. There are three basic evaluation constituents:

- Design review; type/prototype testing, as applicable;
- Survey during construction and testing at the plant of manufacture; and
- Survey during installation on board the vessel and at trials.

Where design review is required by the Rules, a letter will be issued by the Bureau upon satisfactory review of the plans to evidence the acceptance of the design. In addition to, or independent of, design review, ABS may require survey and testing of forgings, castings and component parts at the various manufacturers' plants, as well as survey and testing of the finished product. A certificate or report will be issued upon satisfactory completion of each survey to evidence acceptance of the forging, casting, component or finished product. Design review, survey and the issuance of reports or certificates constitute the *certification* of machinery.

Based on the intended service and application, some products do not require certification because they are not directly related to the scope of classification or because normal practices for their construction within the industry are considered adequate. Such products may be accepted based on the manufacturers' documentation on design and quality.

In general, surveys during installation on board the vessel and at trials are required for all items of machinery. This is not considered a part of the product certification process. There may be instances, however, where letters or certificates issued for items of machinery contain conditions which must be verified during installation, tests or trials.

3.3 Type Approval Program *(2003)*

Products that can be consistently manufactured to the same design and specification may be Type Approved under the ABS Type Approval Program. The ABS Type Approval Program is a voluntary option for the demonstration of the compliance of a product with the Rules or other recognized standards. It may be applied for at the request of the designer or manufacturer. . The ABS Type Approval Program generally covers Product Type Approval (1-1-4/7.7.3), but is also applicable for a more expeditious procedure towards Unit-Certification, as specified in 1-1-4/7.7.2.

See the "ABS Type Approval Program" in Appendix 1-1-A3.

3.5 Non-mass Produced Machinery *(2003)*

Non-mass produced critical machinery, such as propulsion boilers, slow speed diesel engines, turbines, steering gears, and similar critical items are to be individually unit certified in accordance with the procedure described in 4-1-1/3.1. However, consideration will be given to granting Type Approval to such machinery in the categories of Acceptable Quality System (AQS) and Recognized Quality System (RQS). The category of Product Quality Assurance (PQA) will not normally be available for all products, and such limitations will be indicated in 4-1-1/Table 1 through 4-1-1/Table 6. In each instant where Type Approval is granted, in addition to quality assurance and quality control assessment of the manufacturing facilities, the Bureau will require some degree of product specific survey during manufacture.

3.7 Details of Certification of Some Representative Products

4-1-1/Table 1 through 4-1-1/Table 6 provide abbreviated certification requirements of representative machinery based on the basic requirements of the Rules for machinery. The tables also provide the applicability of the Type Approval Program for each of these machinery items.

For easy reference, the tables contain six product categories as follows:

- Prime movers
- Propulsion, maneuvering and mooring machinery
- Electrical and control equipment
- Fire safety equipment
- Boilers, pressure vessels, fired equipment
- Piping system components

5 Machinery Plans

5.1 Submission of Plans

Machinery and systems plans required by the Rules are to be submitted by the manufacturer, designer or shipbuilder, in triplicate, to the Bureau. After review and approval of the plans, one copy will be returned to the submitter, one copy will be retained for the use of the Bureau's Surveyor, and one copy will be retained by the Bureau for record. It may be necessary to submit additional copies of plans when attendance by the Bureau's Surveyor is anticipated at more than one location. Where so stated in the shipbuilding contract, the Owner may require the builder to provide copies of approved plans and related correspondence, in which case the total number of copies of each plan to be submitted to the Bureau is to be increased, correspondingly. A fee will be charged for the review of plans which are not covered by a contract of classification with the shipbuilder.

In general, all plans are to be submitted and approved before proceeding with the work.

5.3 Plans

Machinery plans required to be submitted for review and approval by the Bureau are listed in each of the sections in Part 4. In general, equipment plans are to contain performance data and operational particulars; standard of compliance where standards are used in addition to, or in lieu of, the Rules; construction details such as dimensions, tolerances, welding details, welding procedures, material specifications, etc.; and engineering calculations or analyses in support of the design. System plans are to contain a bill of material with material specifications or particulars, a legend of symbols used, system design parameters, and are to be in a schematic format. Booklets containing standard shipyard practices of piping and electrical installations are generally required to supplement schematic system plans.

7 Miscellaneous Requirements for Machinery

7.1 Construction Survey Notification

Before proceeding with the manufacture of machinery requiring test and inspection, the Bureau is to be notified that survey is desired during construction. Such notice is to contain all of the necessary information for the identification of the items to be surveyed.

7.3 Machinery Equations

The equations for rotating parts of the machinery in Part 4 of the Rules are based upon strength considerations only and their application does not relieve the manufacturer from responsibility for the presence of dangerous vibrations and other considerations in the installation at speeds within the operating range.

7.5 Astern Propulsion Power *(2005)*

7.5.1 General

Sufficient power for going astern is to be provided to secure proper control of the vessel in all normal circumstances. The astern power of the main propelling machinery is to be capable of maintaining in free route astern at least 70% of the ahead rpm corresponding to the maximum continuous ahead power. For main propulsion systems with reversing gears, controllable pitch propellers or electric propulsion drive, running astern is not to lead to overload of the propulsion machinery. The ability of the machinery to reverse the direction of thrust of the propeller in sufficient time, and so to bring the vessel to rest within a reasonable distance from maximum ahead service speed, is to be demonstrated and recorded during trials.

7.5.2 Steam Turbine Propulsion

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

7.7 Dead Ship Start *(2005)*

Means are to be provided to bring the machinery into operation from a "dead ship" condition, as defined in 4-1-1/1.9.6. See 4-8-2/3.1.3 and 4-8-4/1.13 for the required starting arrangements.

7.9 Inclinations

Machinery installations are to be designed to ensure proper operations under the conditions as shown in 4-1-1/Table 7.

7.11 Ambient Temperature

For vessels of unrestricted service, ambient temperature, as indicated in 4-1-1/Table 8, is to be considered in the selection and installation of machinery, equipment and appliances. For vessels of restricted or special service, the ambient temperature appropriate to the special nature is to be considered.

7.13 Machinery Space Ventilation *(2002)*

Suitable ventilation is to be provided for machinery spaces so as to simultaneously allow for crew attendance and for engines, boilers and other machinery to operate at rated power in all weather conditions, including heavy weather. The main propulsion machinery space is to be provided with mechanical means of ventilation.

The supply of air is to be provided through ventilators which can be used in all weather conditions. In general, ventilators necessary to continuously supply the main propulsion machinery space and the immediate supply to the emergency generator room are to have coamings of sufficient height to eliminate the need to have closing arrangements. See 3-2-17/9.3.

However, where due to the vessel size and arrangement this is not practicable, lesser heights for machinery space and emergency generator room ventilator coamings may be accepted with provision of weathertight closing appliances in accordance with 3-2-17/9.5 in combination with other suitable arrangements to ensure an uninterrupted and adequate supply of ventilation to these spaces. See also 4-7-2/1.9.5 and 4-7-2/1.9.6.

7.15 Materials Containing Asbestos *(2005)*

Installation of materials which contain asbestos is prohibited, except for the following:

i) Vanes used in rotary vane compressors and rotary vane vacuum pumps

ii) Watertight joints and linings used for the circulation of fluids when, at high temperature [in excess of 350°C (662°F)] or high pressure [in excess of 70.0 bar (71.38 kgf/cm^2, 1015.3 psi)], there is a risk of fire, corrosion or toxicity

iii) Supple and flexible thermal insulation assemblies used for temperatures above 1000°C (1832°F).

9 Sea Trials

A final underway trial is to be made of all machinery, steering gear, anchor windlass, stopping and maneuvering capability, including supplementary means for maneuvering, if any. Insofar as practicable, the vessel is to be ballasted or otherwise arranged to simulate fully laden condition so as to allow propulsion machinery to discharge its rated power. The entire machinery installation is to be operated in the presence of the Surveyor in order to demonstrate its reliability and sufficiency to function satisfactorily under operating conditions and its freedom from dangerous vibration and other detrimental operating phenomena at speeds within the operating range. All automatic controls, including tripping of all safety protective devices that affect the vessel's propulsion system, are to be tested under way or alongside the pier, to the satisfaction of the Surveyor. References are also to be made to the following for more detailed requirements:

- Steering gear trial: 4-3-4/21.7
- Anchor windlass trial: 4-5-1/9
- Remote propulsion control and automation trial: 4-9-5/5
- Shipboard trials for diesel engines: 4-2-1/15

The viscosity of the fuel used on the sea trial will be entered in the classification report.

Based on the sea trials, the following information is to be provided on board:

- Stopping time (see also 4-1-1/7.5),
- Vessel headings and distances recorded on sea trials, and
- For vessels with multiple propellers, ability to navigate and maneuver with one or more propellers inoperative.

Reference may be made to IMO Resolution A.209(VII) *Recommendation on Information to be Included in the Maneuvering Booklet* and IMO Resolution A.601(15) *Recommendation on the Provision and the Display of Maneuvering Information on board ships.*

TABLE 1
Certification Details – Prime Movers *(2010)*

Prime Movers [1]	*Individual Unit Certification* [2]	*Type Approval Program* [3]					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Diesel engines with cylinder bore; > 300 mm	d, m, s, t,	x	x	x	o	o	NA
2. Diesel engines; steam turbines; gas turbines; ≥ 100 kW (135 hp)	d, m, s, t	x	x	x	o	o	o
3. Diesel engines; steam turbines; gas turbines, < 100 kW (135 hp)	g	x	o	x	o	o	NA
4. Turbochargers for engines ≥100 kW (135 hp) and bore ≥ 300 mm (11.8 in.)	d, m, s, t	x	x	x	o	o	o
5. Turbochargers for engines ≥ 100 kW (135 hp) and bore < 300 mm (11.8 in.)	d, t	x	x	x	o	o	NA

Notes

1 For full certification details, refer to Part 4, Chapter 2.

2 See also 4-1-1/3.1. Notations used in this column are:

d – design review by ABS.

m – material tests witnessed by Surveyor.

s – survey at the plant of manufacture including witnessing acceptance tests on production unit.

t – type/prototype testing conducted on an actual sample or a prototype model is required, as applicable.

g – certification by ABS not required; acceptance based on manufacturer's guarantee.

3 For description of Type Approval Program, see 1-1-A3/5. Notations used in these columns are:

x – indicates the particular element of the program is applicable

o – indicates the particular element of the program is optional

NA – indicates the particular element of the program is not applicable.

TABLE 2
Certification Details – Propulsion, Maneuvering and Mooring Machinery *(2003)*

Propulsion, Maneuvering and Mooring Machinery [(1)]	*Individual Unit Certification* [(2)]	*Type Approval Program* [(3)]					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Propulsion shafts, couplings, coupling bolts [(4)]	d, m, s	x	NA	NA	o	o	NA
2. Cardan shafts, standard couplings and coupling bolts	d, m, s	x	x	x	o	o	o
3. Gears and Clutches ≥ 5590 kW (7500 hp)	d, m, s	x	x	x	o	o	NA
4. Gears and clutches, ≥ 100 kW (135 hp)	d, m, s	x	x	x	o	o	o
5. Gears and clutches, < 100 kW (135 hp)	g	o	o	x	o	o	NA
6. Propellers, fixed and controllable pitch [(4)]	d, m, s	x	NA	NA	o	o	NA
7. Propulsion thrusters	d, m, s	x	x	x	o	o	o
8. Steering gears	d, m, s	x	x	x	o	o	NA
9. Athwartship thrusters	d, m, s	x	x	x	o	o	o
10. Positioning thrusters [(5)]	g	x	x	x	o	o	NA
11. Dynamic positioning thrusters with **DPS** notation	d, m, s, t	x	x	x	o	o	NA
12. Anchor windlass	d or t, and s	x	x	x	o	o	o
13. Mooring winches	g	x	x	x	o	o	NA

Notes

1 For full certification details, refer to Part 4, Chapter 3 and Chapter 5.

2 See also 4-1-1/3.1. Notations used in this column are:

d – design review by ABS.

M – material tests to be witnessed by Surveyor.

s – survey at the plant of manufacture, and witness acceptance tests on production unit.

t – type/prototype testing conducted on an actual sample or a prototype model is required, as applicable.

g – certification by ABS not required; acceptance is based on manufacturer's guarantee.

3 For description of Type Approval Program, see 1-1-A3/5. Notations used in these columns are:

x – indicates the particular element of the program is applicable.

o – indicates the particular element of the program is optional.

NA – indicates the particular element of the program is not applicable.

4 Typically made to custom designs. However, manufacturing facilities may be quality assurance approved, see 4-1-1/3.5.

5 Thrusters in this category would be those not normally relied upon for maneuvering assistance.

TABLE 3
Certification Details – Electrical and Control Equipment *(2006)*

Electrical and Control Equipment (1)	*Individual Unit Certification* (2)	*Type Approval Program* (3)					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Generators and motors for essential services ≥100 kW (135 hp)	d, s, t	x	x	x	o	o	o
2. Motors ≥100 kW (135 hp) for LNG cargo or vapor handling services. (See 5C-8-10/1.8)	d, s, t	x	x	x	o	o	o
3. Generators and motors for essential services <100 kW (135 hp)	g	o	o	x	o	o	NA
4. Motors <100 kW (135 hp) for LNG cargo or vapor handling services. (See 5C-8-10/1.8)	g	o	o	x	o	o	NA
5. Propulsion generators and motors	d, m, s, t	x	x	x	o	o	NA
6. Switchboards (propulsion, main and emergency) (4)	d, s	x	NA	NA	o	o	o
7. Motor controllers for essential services ≥ 100 kW (135 hp)	d, s	x	x	NA	o	o	o
8. Motor controllers ≥ 100 kW (135 hp) for LNG cargo or vapor handling services. (See 5C-8-10/1.8)	d, s	x	x	NA	o	o	o
9. Motor control centers for essential services ≥ 100 kW (135 hp)	d, s	x	x	NA	o	o	o
10. Motor control centers ≥ 100 kW (135 hp) for LNG cargo or vapor handling services. (See 5C-8-10/1.8)	d, s	x	x	NA	o	o	o
11. Battery charging and discharging boards for essential, emergency or transitional source of power. (See 4-8-3/5.9)	d, s	x	x	NA	o	o	o
12. Power transformers and converters of low voltage	g	x	x	x	o	o	NA
13. Power transformers and converters for high voltage systems exceeding 1 kV	d, s	x	x	x	o	o	o
14. Cables	d-1, t	x	x	x	o	o	o
15. Propulsion cables	d-1, s, t	x	x	x	o	o	NA
16. Circuit breakers & fuses	g	NA	x	x	o	o	NA
17. Certified safe equipment	t	NA	x	x	o	o	NA
18. Governors	t	NA	x	x	o	o	NA
19. Control, monitoring and safety system devices, including computers, programmable logic controllers, etc., for **ACC** and **ACCU** notations	t	x	x	x	o	o	o
20. Complete assembly or subassembly units for **ACC** and **ACCU** notations	d, s, t	x	x	x	o	o	NA

TABLE 3 (continued)
Certification Details – Electrical and Control Equipment *(2006)*

Notes

1 For full certification details, see Section 4-8-3 and Section 4-8-5 for electrical equipment and Section 4-9-7 for control, monitoring and safety system equipment.

2 See also 4-1-1/3.1. Notations used in this column are:

d – design review by ABS.

d-1 – reviewed for compliance with a recognized standard.

m – material tests to be witnessed by Surveyor.

s – survey at the plant of manufacture including witnessing acceptance tests of production unit.

t – type/prototype testing conducted on an actual sample or a prototype model is required, as applicable.

g – certification by ABS not required; acceptance is based on manufacturer's guarantee.

3 For description of Type Approval Program, see 1-1-A3/5. Notations used in these columns are:

x – indicates the particular element of the program is applicable.

o – indicates the particular element of the program is optional.

NA – indicates the particular element of the program is not applicable.

4 This equipment is generally made to custom design; but manufacturing facilities may be quality assurance approved, see 4-1-1/3.5.

TABLE 4
Certification Details – Fire Safety Equipment *(2003)*

Fire Safety Equipment (1)	*Individual Unit Certification* (2)	*Type Approval Program* (3)					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Fire detection and alarm system components	d, t	x	x	x	o	o	NA
2. Fixed fire extinguishing system components	d, t	x	x	x	o	o	NA
3. Fireman's outfit	t	x	x	x	o	o	NA
4. Fire hoses	t	x	x	x	o	o	NA
5. Portable fire extinguishers	t	x	x	x	o	o	NA

Notes

1 For certification details, see Section 4-7-3.

2 See also 4-1-1/3.1. Notations used in this column are:

d – design review by ABS.

s – survey at the plant of manufacture and witness acceptance tests of production unit.

t – type/prototype testing conducted on an actual sample or a prototype model is required, as applicable; or type approval by Flag Administration.

3 For description of Type Approval Program, see 1-1-A3/5. Notations used in these columns are:

x – indicates the particular element of the program is applicable.

o – indicates the particular element of the program is optional.

TABLE 5
Certification Details – Boilers, Pressure Vessels and Fired Equipment *(2003)*

Boilers, Pressure Vessels and Fired Equipment [(1)]	*Individual Unit Certification* [(2)]	*Type Approval Program* [(3)]					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Group I boilers and pressure vessels	d, m, s	x	x	NA	o	o	NA
2. Group II pressure vessels	d, s	x	x	NA	o	o	o
3. Inert gas generators, incinerators	d	x	x	x	o	o	NA

Notes

1 For grouping of boilers and pressure vessels, see 4-4-1/1.7 and 4-4-1/1.9.

2 See also 4-1-1/3.1. Notations used in this column are:

d – design review by ABS

m – material tests to be witnessed by Surveyor

s – survey at the plant of manufacture and witness acceptance tests of production unit

3 For description of Type Approval Program, see 1-1-A3/5. Type Approval Programs are generally applicable to mass produced boilers and pressure vessels (See 4-4-1/1.11.2). Notations used in these columns are:

x – indicates the particular element of the program is applicable

o – indicates the particular element of the program is optional

NA – indicates the particular element of the program is not applicable

TABLE 6
Certification Details – Piping System Components *(2003)*

Piping System Components (1)	*Individual Unit Certification* (2)	*Type Approval Program* (3)					
		Product Design Assessment			*Manufacturing Assessment*		
		1-1-A3/5.1			*1-1-A3/5.3*		*1-1-A3/5.5*
		Design Review	*Type Exam.*	*Type Test*	*AQS*	*RQS*	*PQA*
1. Pumps related to propulsion diesel engines (bore >300 mm) (11.8 in.) and gas turbines and gears—fuel, cooling water, lube. Oil services	s	x	x	x	o	o	o
2. Pumps related to propulsion steam plant and gears—fuel oil, lube. Oil, condensate, main circulating, feed water services	s	x	x	x	o	o	o
3. Hydraulic pumps of steering gears, controllable pitch propellers, anchor windlass	s	x	x	x	o	o	o
4. Pumps for fire main, ballast, bilge, liquid cargoes	s	x	x	x	o	o	o
5. Air compressors	g	x	x	x	o	o	NA
6. Steel pipes, classes I and II	m, s	x	NA	NA	o	o	o
7. Steel pipes, class III	g	x	NA	NA	x	x	NA
8. Pipe fittings—flanges, elbows, tees, flexible joints, etc., and valves; classes I & II	d-1	x	NA	NA	o	o	NA
9. Pipe fittings—flanges, elbows, tees, flexible joints, etc., and valves; class III	g	x	NA	NA	o	o	NA
10. Plastic pipes and pipe joints	d-2, t, s (4)	x	x	x	o	o	o
11. Hoses	d-2, t	x	x	x	o	o	NA
12. Vent heads, pressure vacuum valves	d-2, t	x	x	x	o	o	NA
13. Gauges, detectors and transmitters	d-2	x	x	x	o	o	NA
14. Fluid power cylinders and systems, including valve actuators (5)	d-1	x	x	x	o	o	NA

Notes

1 For full certification details, see 4-6-1/7 and Section 4-6-2 for metallic piping and Section 4-6-3 for plastic piping.

2 See also 4-1-1/3.1. Notations used in this column are:

d-1 – verification for compliance with recognized standard or design review by ABS.

d-2 – reviewed for suitability for proposed installation.

m – material tests witnessed by Surveyor.

s – survey at the plant of manufacture, including witnessing acceptance tests of production unit.

t – type/prototype testing conducted on an actual sample or a prototype model is required, as applicable. Where, for plastic pipes, the manufacturer does not have a certified quality system in accordance with 1-1-A3/5.3, 1-1-A3/5.5 or ISO 9001 (or equivalent), and that ensures testing is carried to demonstrate the compliance of plastic pipes, fittings and joints with 4-6-3/5.1 through 4-6-3/5.15 and 4-6-3/19, as applicable, testing is to be witnessed by Surveyor.

g – certification by ABS not required; acceptance is based on manufacturer's documentation.

3 For description of Type Approval Program, see 1-1-A3/5. Notations used in these columns are:

x – indicates the particular element of the program is applicable.

o – indicates the particular element of the program is optional.

NA – indicates the particular element of the program is not applicable.

4 Where the manufacturer does not have a certified quality system, see 4-6-3/9.

5 Other than steering gear actuators.

TABLE 7
Design Angles of Inclination

	Angle of Inclination, degrees (1)			
	Athwartship		*Fore-and-Aft*	
Installations, components	Static	Dynamic	Static	Dynamic
Propulsion and auxiliary machinery	15	22.5	5 (4)	7.5
Safety equipment				
Emergency power installation (3)	22.5	22.5	10	10
Emergency fire pumps and their drives	22.5	22.5	10	10
Switchgear				
Electrical and electronic appliances and control systems	22.5 (2)	22.5 (2)	10	10

Notes

1 Athwartship and fore-and-aft inclinations occur simultaneously.

2 Up to an angle of inclination of 45 degrees, switches and controls are to remain in their last set position.

3 In vessels designed for carriage of liquefied gases and of chemicals, the emergency power installation is to remain operable with the vessel flooded to its permissible athwartship inclination up to a maximum of 30 degrees.

4 *(2004)* Where the length of the vessel exceeds 100 m (328 ft), the fore-and-aft static angle of inclination may be taken as 500/L degrees, where L is the length of the vessel in meters (1640/L degrees, where L is the length of the vessel in feet), as defined in 3-1-1/3.1.

TABLE 8
Ambient Temperatures for Unrestricted Service

	Location	*Temperature Range (°C)*
Air	Enclosed spaces (1, 2)	0 to +45
	Open deck (1)	–25 to +45

	Temperature (°C)
Seawater	+32

Notes:

1 Electronic equipment is to be suitable for operations up to 55°C.

2 Electrical equipment in machinery spaces is to be designed for 45°C, except that electric generators and motors are to be designed for 50°C. Electrical equipment outside machinery space may be designed for 40°C.

PART
4

CHAPTER 2 Prime Movers

CONTENTS

PART

4

CHAPTER 2 Prime Movers

SECTION 1 Diesel Engines

1 General

1.1 Application

Diesel engines having a rated power of 100 kW (135 hp) and over, intended for propulsion and for auxiliary services essential for propulsion, maneuvering and safety (see 4-1-1/1.3) of the vessel, are to be designed, constructed, tested, certified and installed in accordance with the requirements of this section.

Diesel engines having a rated power of less than 100 kW (135 hp) are not required to comply with the provisions of this section but are to be designed, constructed and equipped in accordance with good commercial and marine practice. Acceptance of such engines will be based on manufacturer's affidavit, verification of engine nameplate data, and subject to a satisfactory performance test after installation conducted in the presence of the Surveyor.

Diesel engines having a rated power of 100 kW (135 hp) and over, intended for services considered not essential for propulsion, maneuvering and safety, are not required to be designed, constructed and certified by the Bureau in accordance with the requirements of this section. They are to comply with safety features, such as crankcase explosion relief valve, overspeed protection, etc., as provided in 4-2-1/7, as applicable. After installation, they are subject to a satisfactory performance test conducted in the presence of the Surveyor.

Piping systems serving diesel engines, such as fuel oil, lubricating oil, cooling water, starting air, crankcase ventilation and exhaust gas systems are addressed in Section 4-6-5; hydraulic and pneumatic systems are addressed in Section 4-6-7.

Requirements for turbochargers are provided in Section 4-2-2.

1.3 Definitions

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

1.3.1 Slow-, Medium-, High-speed Diesel Engines

Slow-Speed Engines means crosshead type diesel engines having a rated speed of less than 400 rpm.

Medium-Speed Engines means trunk piston type diesel engines having a rated speed of 400 rpm and above, but less than 1400 rpm.

High-Speed Engines means trunk piston type diesel engines having a rated speed of 1400 rpm or above.

1.3.2 Rated Power

The *Rated Power* is the maximum power output at which the engine is designed to run continuously at its rated speed between the normal maintenance intervals recommended by the manufacturer.

1.5 Increased Power Rating

The rated power of an engine, which has been type tested as specified in 4-2-1/13.7 or 4-2-1/13.11 and which has proven reliable in service, may be increased by not more than 10% of the type tested power rating without performing any new type test, subject to prior approval of relevant plans and particulars.

1.7 Ambient Reference Conditions

The following ambient reference conditions are to be applied by the engine manufacturer for the purpose of determining the rated power of diesel engines used on vessels with unrestricted service. However, the engine manufacturer is not expected to provide simulated ambient reference conditions at any test.

Barometric pressure:	1 bar (1 kgf/cm^2, 15 psi)
Air temperature:	45°C (113°F)
Relative air humidity:	60%
Seawater Temperature (Charging air coolant inlet):	32°C (90°F)

1.9 Plans and Particulars to be Submitted

For a tabulated listing, see Appendix 4-2-A1.

1.9.1 Engine Construction

Engine transverse cross-section

Engine longitudinal section

Bedplate with welding details and procedures; frame/column with welding details and procedures; crankcase with welding details and procedures

Structural supporting and seating arrangements

Arrangement of foundation bolts (for main engines only)

Thrust bearing assembly

Thrust bearing bedplate

Tie rod

Cylinder cover, assembly or cylinder head

Cylinder jacket or engine block

Cylinder liner

Crankshaft, details

Crankshaft, assembly

Thrust shaft or intermediate shaft (if integral with engine)

Coupling bolts

Counterweights (if not integral with crankshaft)

Connecting rod

Connecting rod, assembly and details

Crosshead, assembly and details

Piston rod, assembly and details

Piston, assembly and details

Camshaft drive, assembly

Arrangement of crankcase explosion relief valve and breather arrangement (only for engines having a cylinder bore of 200 mm (8 in.) and above)

1.9.2 Engine Systems and Appurtenances *(2001)*

Starting air system

Fuel oil system

Lubricating oil system

Cooling water system

Governor arrangements

Schematic diagram of the engine control and safety system

Shielding and insulation of exhaust pipes, assembly

Shielding of high pressure fuel pipes, assembly as applicable

Turbochargers and superchargers, see 4-2-2/1.5

Couplings and clutches

Vibration damper assembly

Tuning wheel assembly, if fitted

Engine driven pump assembly

Scavenging pump and blower assemblies

1.9.3 Data

Type designation of engine and combustion cycle

Number of cylinders

Rated power, kW (PS, hp)

Rated engine speed, (rpm)

Sense of rotation (clockwise/counter-clockwise)

Firing order with the respective ignition intervals and, where necessary, V-angle, α_v

Cylinder diameter, mm (in.)

Stroke, mm (in.)

Maximum cylinder pressure p_{max}, bar (kgf/mm^2, psi)

Mean effective pressure, bar (kgf/mm^2, psi)

Mean indicated pressure, bar (kgf/mm^2, psi)

Charge air pressure, bar (kgf/mm^2, psi), (before inlet valves or scavenge ports, whichever applies)

Nominal compression ratio

Connecting rod length L_H, mm (in.)

Oscillating mass of one crank gear, kg (lb.), (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)

Mass of reciprocating parts, kg (lb.)

Digitalized gas pressure curve presented at equidistant intervals, bar (kgf/mm^2, psi) versus crank angle, (intervals equidistant and integrally divisible by the V-angle, but not more than 5 degrees CA)

For engines with articulated-type connecting rod:

- Distance to link point L_A, mm (in.)
- Link angle α_N (degree)
- Connecting rod length (between bearing centers) L_N, mm (in.)
- Tightening torques for pretensioned bolts and studs for reciprocating parts.
- Mass and diameter of flywheel and flywheel effect on engine

(2005) Operation and service manuals, including maintenance requirements for servicing and repair and details of any special tools and gauges that are to be used with their fittings/settings, together with any test requirements on completion of the maintenance (see also 4-6-2/9.6).

1.9.4 Materials

Crankshaft material:

- Material designation
- Mechanical properties of material (tensile strength, yield strength, elongation (with length of specimen), reduction of area, impact energy)
- Type of forging (open die forged (free form), continuous grain flow forged, close die forged (drop-forged), etc., with description of the forging process)

Crankshaft heat treatment

Crankshaft surface treatment

- Surface treatment of fillets, journals and pins (induction hardened, flame hardened, nitrided, rolled, shot peened, etc., with full details concerning hardening)
- Hardness at surface
- Hardness as a function of depth, mm (in.)
- Extension of surface hardening

Material specifications of other main parts

1.9.5 Calculations and Analyses *(2001)*

Strength analysis for crankshaft and other reciprocating parts

Strength analysis for engine supports and seating arrangements

Torsional vibration analysis for propulsion shafting systems for all modes of operation including the condition of one cylinder misfiring

Calculation demonstrating the adequacy of the bolting arrangement attaching tuning wheels or vibration dampers to the propulsion system to withstand all anticipated torsional vibration and operating loads

1.9.6 Submittals by Licensee

1.9.6(a) Plans lists. For each diesel engine manufactured under license, the licensee is to submit two listings of plans and data to be used in the construction of the engine:

- One list is to contain drawing numbers and titles (including revision status) of the licenser's plans and data of the engine as approved by the Bureau (including approval information such as location and date at which they are approved); and
- A second list, which is to contain the drawing numbers and titles (including revision status) of the licensee's plans and data, insofar as they are relevant to the construction of the engine. In the event that construction is based solely on the licenser's plans, this list will not be required.

1.9.6(b) Plans for approval. Any design change made by the licensee is to be documented, and relevant plans and data are to be submitted by the licensee for approval or for information, as appropriate. The licenser's statement of acceptance of the modifications is to be included in the submittal.

1.9.6(c) Plans for surveyor. A complete set of the licenser's or the licensee's plans and data, as approved by the Bureau, is to be made available to the Surveyor attending the licensee's plant.

3 Materials

3.1 Material Specifications and Tests

Material specifications are to be in accordance with that in Part 2, Chapter 3 or other specifications approved under 4-2-1/3.3.1. Except as noted in 4-2-1/3.3, materials intended for engines required to be constructed under survey are to be tested and inspected in accordance with 4-2-1/Table 1. The material tests, where so indicated in the table, are to be witnessed by the Surveyor. Nondestructive tests in 4-2-1/Table 1 are to be carried out by the manufacturer whose test records may be accepted by the Bureau.

Copies of material specifications or purchase orders are to be submitted to the Surveyor for information.

3.3 Alternative Materials and Tests

3.3.1 Alternative Specifications

Material manufactured to specifications other than those given in Part 2, Chapter 3 may be accepted, provided that such specifications are approved in connection with the design and that they are verified or tested in the presence of a Surveyor, as applicable, as complying with the specifications.

3.3.2 Steel-bar Stock

Hot-rolled steel bars up to 305 mm (12 in.) in diameter may be used when approved for any of the items indicated in 4-2-1/Table 1, subject to the conditions specified in Section 2-3-8

3.3.3 Material for Engines of 375 kW (500 hp) Rated Power or Less

Material for engines having a rated power of 375 kW (500 hp) or less, including shafting, couplings, and coupling bolts will be accepted on the basis of the material manufacturer's certified test reports and a satisfactory surface inspection and hardness check witnessed by the Surveyor. Coupling bolts manufactured to a recognized bolt standard will not require material testing.

3.3.4 Engines Certified Under Quality Assurance Approval

For diesel engines certified under quality assurance assessment as provided for in 4-2-1/13.13.2(b), material tests required by 4-2-1/3.1 need not be witnessed by the Surveyor; such tests are to be conducted by the engine manufacturer whose certified test reports may be accepted instead.

5 Design

5.1 Bedplate/Crankcase

The bedplate or crankcase is to be of rigid construction, oiltight, and provided with a sufficient number of bolts to secure the same to the vessel's structure. See also 4-2-1/11.1 for seating of diesel engines.

5.3 Crankcase Doors *(2006)*

Crankcase construction and crankcase doors are to be of sufficient strength to withstand anticipated crankcase pressures that may arise during a crankcase explosion taking into account the installation of explosion relief valves required by 4-2-1/7.1. Crankcase doors are to be fastened and secured so that they will not be readily displaced by a crankcase explosion.

5.5 Cylinders and Covers, Liners and Pistons

Cylinders, liners, cylinder covers and pistons, which are subjected to high temperatures or pressures, are to be of materials suitable for the stresses and temperatures to which they are exposed.

5.7 Securing of Nuts

All nuts of main bearings and of connecting-rod bolts and all other moving parts are to be secured by split pins or other effective locking means.

5.9 Crankshafts *(2007)*

5.9.1 General

5.9.1(a) Scope. These Rules for the design of crankshafts are to be applied to diesel engines for propulsion and auxiliary purposes, where the engines are being so designed as to be capable of continuous operation at their rated power when running at rated speed.

Where a crankshaft design involves the use of surface treated fillets, when fatigue testing is conducted, or when direct stress (strain) measurements are taken, the relevant documents with calculations/analysis and reliability data are to be submitted in order to substantiate the design.

5.9.1(b) Field of application. These Rules apply only to solid-forged and semi-built crankshafts of forged or cast steel, with one crank throw between main bearings.

5.9.1(c) Principles of calculation. The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas.

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

- Fillet transitions between the crankpin and web as well as between the journal and web,
- Outlets of crankpin oil bores.

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

Calculation of crankshaft strength consists of determining the nominal alternating bending (see 4-2-1/5.9.2) and nominal alternating torsional stresses (see 4-2-1/5.9.3) which, multiplied by the appropriate stress concentration factors (see 4-2-1/5.9.4), result in an equivalent alternating stress (uni-axial stress) (see 4-2-1/5.9.6). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (see 4-2-1/5.9.7). This comparison will show whether or not the crankshaft concerned is dimensioned adequately (see 4-2-1/5.9.8).

5.9.2 Calculation of Alternating Stresses Due to Bending Moments and Radial Forces

5.9.2(a) Assumptions. The calculations are based on a quasi-static model where the steady alternating loads are combined in a statically determined system. The statically determined system is composed of a single crank throw supported in the center of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints (distance L_3, as per 4-2-1/Figure 1 and 4-2-1/Figure 2).

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 4-2-1/Figure 1).

For crank throws with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see 4-2-1/Figure 2).

i) *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 center of the solid web (distance L_1) and are derived from the radial component of the connecting-rod force.

The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crank web. This reference section results from the web thickness W and the web width B (see 4-2-1/Figure 3). Mean stresses are neglected.

FIGURE 1
Crank Throw for In Line Engine *(2007)*

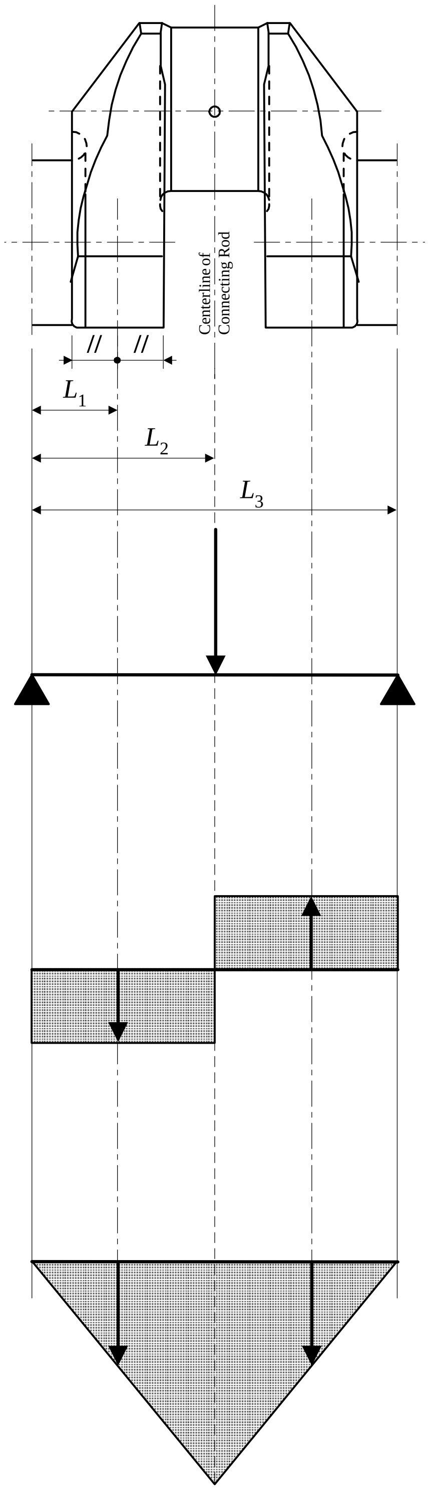
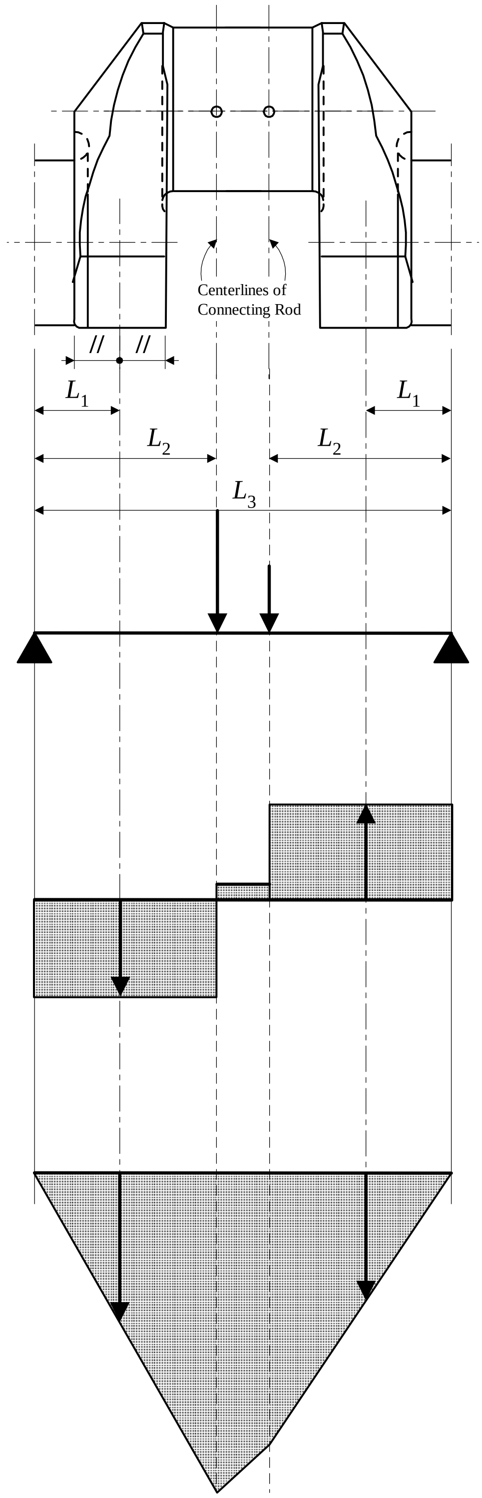
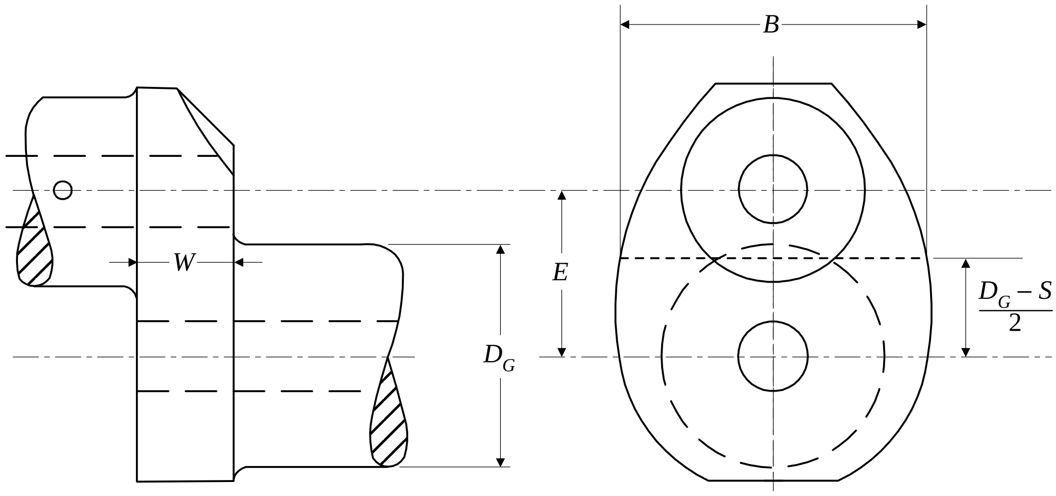


FIGURE 2
Crank Throw for Vee Engine with 2 Adjacent Connecting-Rods *(2007)*

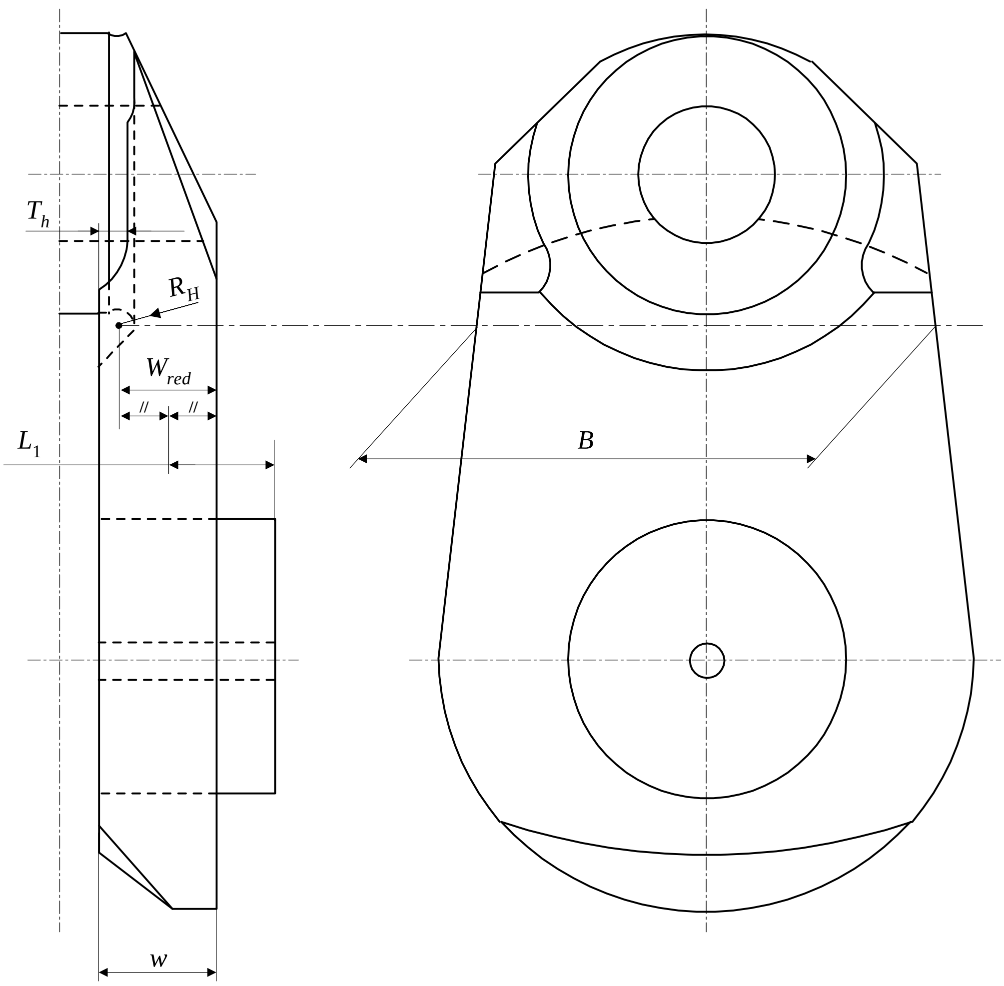


L_1	=	Distance between main journal centerline and crank web center (see also 4-2-1/Figure 3 for crankshaft without overlap)
L_2	=	Distance between main journal centerline and connecting-rod center
L_3	=	Distance between two adjacent main journal centerlines

FIGURE 3
Reference Area of Crank Web Cross Section *(2007)*



Overlapped crankshaft



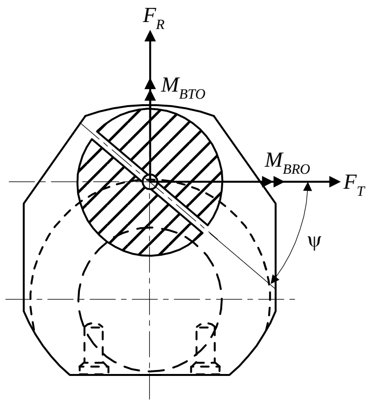
Crankshaft without overlap

ii) *Bending acting in outlet of crankpin oil bore.* The two relevant bending moments are taken in the crankpin cross-section through the oil bore.

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

Mean bending stresses are neglected.

FIGURE 4
Crankpin Section Through the Oil Bore *(2007)*



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

5.9.2(b) 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 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 4-2-1/5.9.2(a)i) and 4-2-1/5.9.2(a)ii) 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 crank throw are superposed according to phase. Different designs (forked connecting-rod, articulated-type connecting-rod or adjacent connecting-rods) are to 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:

$$X_N = \frac{1}{2}\left[X_{\max} - X_{\min}\right]$$

where

X_N = considered as alternating force, moment or stress

$X_{\max}$ = maximum value within one working cycle

$X_{\min}$ = minimum value within one working cycle *i)* *Nominal alternating bending and compressive stresses in web cross section.* The calculation of the nominal alternating bending and compressive stresses is as follows:

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

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

where

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

M_{BRFN} = alternating bending moment related to the center of the web, in N-m (see 4-2-1/Figure 1 and 4-2-1/Figure 2)

= $\frac{1}{2}\left[M_{BFN\,\max} - M_{BFN\,\min}\right]$

W_{eqw} = section modulus related to cross-section of web, in mm³

= $\frac{B \bullet W^2}{6}$

K_e empirical factor considering to some extent the influence of adjacent crank and bearing restraint

= 0.8 for 2-stroke engines

= 1.0 for 4-stroke engines

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

Q_{RFN} = alternating radial force related to the web, in N (see 4-2-1/Figure 1 and 4-2-1/Figure 2)

= $\frac{1}{2}\left[Q_{RF\,\max} - Q_{RF\,\min}\right]$

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

= $B \bullet W$

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

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

where

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

M_{BON} = alternating bending moment calculated at the outlet of crankpin oil bore, in N-m

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

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

ψ = angular position, in degrees (see 4-2-1/Figure 4)

W_e = section modulus related to cross-section of axially bored crankpin, in mm^3

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

5.9.2(c) 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 :

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

where

σ_{BH} = alternating bending stress in crankpin fillet, in N/mm^2

α_B = stress concentration factor for bending in crankpin fillet (see 4-2-1/5.9.4)

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

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

where

σ_{BG} = alternating bending stress in journal fillet, in N/mm^2

β_B = stress concentration factor for bending in journal fillet (see 4-2-1/5.9.4)

β_Q = stress concentration factor for compression due to radial force in journal fillet (determination as per 4-2-1/5.9.4)

5.9.2(d) Calculation of alternating bending stresses in outlet of crankpin oil bore.

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

where

σ_{BO} = alternating bending stress in outlet of crankpin oil bore, in N/mm^2

γ_B = stress concentration factor for bending in crankpin oil bore (determination as per 4-2-1/5.9.4)

5.9.3 Calculation of Alternating Torsional Stresses

5.9.3(a) General. The alternating torsional stresses that are to be used in determining the equivalent alternating stress in the crankshaft are to be provided by the engine manufacturer, and substantiated either by appropriate calculations, or by crankshaft fatigue testing.

Where applicable, the calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in 4-2-1/5.9.3(b). In either case supporting documentation is to be submitted for review.

5.9.3(b) Calculation of nominal alternating torsional stresses. The maximum and minimum torques are to be ascertained for every mass point of the complete dynamic system and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines. Allowance must be made for the damping that exists in the system and for unfavorable conditions (misfiring [*] in one of the cylinders). The speed step calculation is to be selected in such a way that any resonance found in the operational speed range of the engine is to be detected.

** Note:* Misfiring is defined as cylinder condition when no combustion occurs but only a compression cycle.

Where barred speed ranges are necessary, they are to be arranged so that satisfactory operation is possible despite their existence. There are to be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for normal firing conditions.

The values received from such calculation are to be submitted for review.

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

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

$$M_{TN} = \frac{1}{2}\left[M_{T\max} - M_{T\min}\right]$$

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

where

τ_N	=	nominal alternating torsional stress referred to crankpin or journal, in N/mm^2
M_{TN}	=	maximum alternating torque, in N-m
W_P	=	polar section modulus related to cross-section of axially bored crankpin or bored journal, in mm^3
$M_{T\max}$	=	maximum value of the torque, in N-m
$M_{T\min}$	=	minimum value of the torque, in N-m

For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in further calculations is the highest calculated value, according to the above method, occurring at the most torsionally loaded mass point of the crankshaft system.

Where barred speed ranges exist, the torsional stresses within these ranges are not to be considered for assessment calculations.

Approval of the crankshaft will be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by the engine manufacturer).

Thus, for each installation, it is to be ensured by suitable calculation that this approved nominal alternating torsional stress is not exceeded. This calculation is to be submitted for review.

5.9.3(c) Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore. The calculation of stresses is to be carried out for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore.

- For the crankpin fillet:

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

where

τ_H	=	alternating torsional stress in crankpin fillet, in N/mm^2
α_T	=	stress concentration factor for torsion in crankpin fillet (determination as per 4-2-1/5.9.4)
τ_N	=	nominal alternating torsional stress related to crankpin diameter, in N/mm^2

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

  $\tau_G = (\beta_T \tau_N)$

  where

  τ_G = alternating torsional stress in journal fillet, in N/mm^2

  β_T = stress concentration factor for torsion in journal fillet (determination as per 4-2-1/5.9.4)

  τ_N = nominal alternating torsional stress related to journal diameter, in N/mm^2

- For the outlet of crankpin oil bore

  $\tau_{TO} = (\gamma_T \tau_N)$

  where

  τ_{TO} = alternating stress in outlet of crankpin oil bore due to torsion, in N/mm^2

  γ_T = stress concentration factor for torsion in outlet of crankpin oil bore (determination as per 4-2-1/5.9.4)

  τ_N = nominal alternating torsional stress related to crankpin diameter, in N/mm^2

5.9.4 Evaluation of Stress Concentration Factors

5.9.4(a) General. The stress concentration factors are evaluated by means of the equations in 4-2-1/5.9.4(b), 4-2-1/5.9.4(c) and 4-2-1/5.9.4(d) applicable to the fillets and crankpin oil bore of solid forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. The stress concentration factor equations concerning the oil bore are only applicable to a radially drilled oil hole. All crank dimensions necessary for the calculation of stress concentration factors are shown in 4-2-1/Figure 5.

The stress concentration factors for bending (α_B, β_B) are defined as the ratio of the maximum equivalent stress (Von Mises) – occurring in the fillets under bending load – to the nominal bending stress related to the web cross-section (see 4-2-1/Appendix 2).

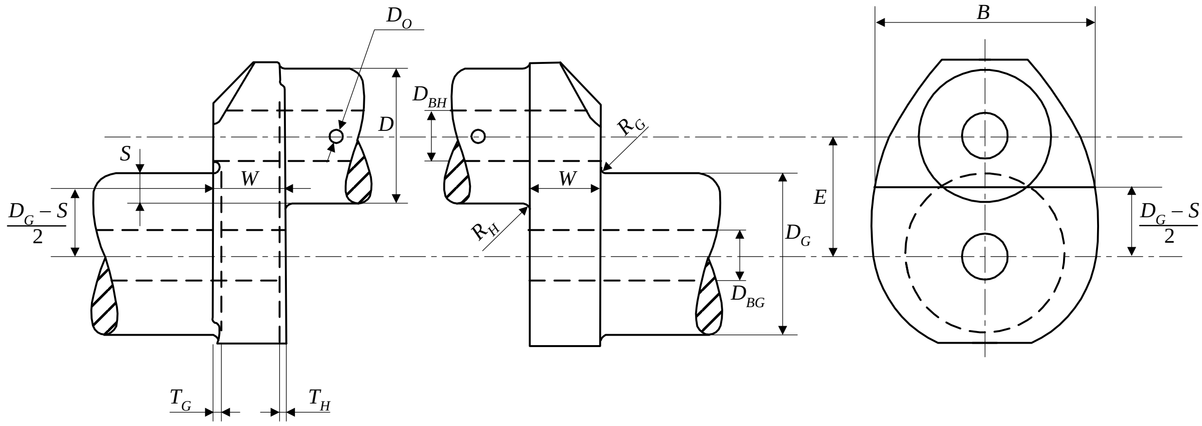
The stress concentration factor for compression (β_Q) in the journal fillet is defined as the ratio of the maximum equivalent stress (Von Mises), occurring in the fillet due to the radial force, to the nominal compressive stress related to the web cross-section.

The stress concentration factors for torsion (α_T, β_T) are defined as the ratio of the maximum equivalent shear stress, occurring in the fillets under torsional load, to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see 4-2-1/Appendix 3).

The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress, occurring at the outlet of the crankpin oil-hole under bending and torsional loads, to the corresponding nominal stress related to the axially bored crankpin cross section (see 4-2-1/Appendix 3).

When reliable measurements and/or calculations are available, which can allow direct assessment of stress concentration factors, the relevant documents and their analysis method is to be submitted in order to demonstrate their equivalence with the Rules.

FIGURE 5
Crank Dimensions *(2007)*



- Actual dimensions:

D	=	crankpin diameter, in mm
D_{BH}	=	diameter of axial bore in crankpin, in mm
D_O	=	diameter of oil bore in crankpin, in mm
R_H	=	fillet radius of crankpin, in mm
T_H	=	recess of crankpin fillet, in mm
D_G	=	journal diameter, in mm
D_{BG}	=	diameter of axial bore in journal, in mm
R_G	=	fillet radius of journal, in mm
T_G	=	recess of journal fillet, in mm
E	=	pin eccentricity, in mm
S	=	pin overlap, in mm
	=	$\dfrac{D + D_G}{2} - E$
W *	=	web thickness, in mm
B *	=	web width, in mm

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

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

  $W_{red} = W - (T_H - R_H)$ (see 4-2-1/Figure 3)

- Web width B must be taken in way of crankpin fillet radius centre according to 4-2-1/Figure 3

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

Crankpin fillet	*Journal fillet*
$r = R_H/D$	$r = R_G/D$

s	=	S/D	
w	=	W/D	crankshafts with overlap
		W_{red}/D	crankshafts without overlap
b	=	B/D	
d_o	=	D_O/D	
d_G	=	D_{BG}/D	
d_H	=	D_{BH}/D	
t_H	=	T_H/D	
t_G	=	T_G/D	

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

$s \le 0.5$

$0.2 \le w \le 0.8$

$1.1 \le b \le 2.2$

$0.03 \le r \le 0.13$

$0 \le d_G \le 0.8$

$0 \le d_H \le 0.8$

$0 \le d_o \le 0.2$

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 $f(s,w)$, $f(r,s)$ and $f_B(s,w)$ are to be evaluated replacing actual value of s by -0.5.

5.9.4(b) Crankpin fillet.

- The stress concentration factor for bending (α_B) is:

$$\alpha_B = 2.6914 \bullet f(s,w) \bullet f(w) \bullet f(b) \bullet f(r) \bullet f(d_G) \bullet f(d_H) \bullet f(recess)$$

where

$$f(s,w) = -4.1883 + 29.2004 \bullet w - 77.5925 \bullet w^2 + 91.9454 \bullet w^3 - 40.0416 \bullet w^4 + (1 - s) \bullet (9.5440 - 58.3480 \bullet w + 159.3415 \bullet w^2 - 192.5846 \bullet w^3 + 85.2916 \bullet w^4) + (1 - s)^2 \bullet (-3.8399 + 25.0444 \bullet w - 70.5571 \bullet w^2 + 87.0328 \bullet w^3 - 39.1832 \bullet w^4)$$

$$f(w) = 2.1790 \bullet w^{0.7171}$$

$$f(b) = 0.6840 - 0.0077 \bullet b + 0.1473 \bullet b^2$$

$$f(r) = 0.2081 \bullet r^{(-0.5231)}$$

$$f(d_G) = 0.9993 + 0.27 \bullet d_G - 1.0211 \bullet d_G^2 + 0.5306 \bullet d_G^3$$

$$f(d_H) = 0.9978 + 0.3145 \bullet d_H - 1.5241 \bullet d_H^2 + 2.4147 \bullet d_H^3$$

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

- The stress concentration factor for torsion (α_T) is :

$$\alpha_T = 0.8 \bullet f(r,s) \bullet f(b) \bullet f(w)$$

where

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

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

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

5.9.4(c) Journal fillet (not applicable to semi-built crankshaft).

- The stress concentration factor for bending (β_B) is:

$$\beta_B = 2.7146 \bullet f_B(s,w) \bullet f_B(w) \bullet f_B(b) \bullet f_B(r) \bullet f_B(\mathrm{d}_G) \bullet f_B(d_H) \bullet f(recess)$$

where

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

$$f_B(w) = 2.2422 \bullet w^{0.7548}$$

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

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

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

$$f_B(d_\mathrm{H}) = 1.0022 - 0.1903 \bullet d_H + 0.0073 \bullet d_H^2$$

$$f(recess) = 1 + (t_H + t_\mathrm{G}) \bullet (1.8 + 3.2 \bullet s)$$

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

$$\beta_Q = 3.0128 \bullet f_Q(s) \bullet f_Q(w) \bullet f_Q(b) \bullet f_Q(r) \bullet f_Q(d_H) \bullet f(recess)$$

where

$$f_Q(s) = 0.4368 + 2.1630 \bullet (1-s) - 1.5212 \bullet (1-s)^2$$

$$f_Q(w) = \frac{w}{0.0637 + 0.9369 \bullet w}$$

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

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

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

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

- The stress concentration factor for torsion (β_T) is:

$\beta_T = \alpha_T$	if the diameters and fillet radii of crankpin and journal are the same
$\beta_T = 0.8 \bullet f(r,s) \bullet f(b) \bullet f(w)$	if crankpin and journal diameters and/or radii are of different sizes

where $f(r,s)$, $f(b)$ and $f(w)$ are to be determined in accordance with 4-2-1/5.9.4(b) (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter:

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

5.9.4(d) Outlet of crankpin oil bore.

- The stress concentration factor for bending (γ_B) is:

$$\gamma_B = 3 - 5.88 \bullet d_o + 34.6 \bullet d_o^2$$

- The stress concentration factor for torsion (γ_T) is:

$$\gamma_T = 4 - 6 \bullet d_o + 30 \bullet d_o^2$$

5.9.5 Additional Bending Stresses

In addition to the alternating bending stresses in fillets (see 4-2-1/5.9.2(c) 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 table below:

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

Notes:

* The additional alternating stress of 30 N/mm² is composed of two components

1 An additional alternating stress of 20 N/mm² resulting from axial vibration

2 An additional alternating 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.

5.9.6 Calculation of Equivalent Alternating Stress

5.9.6(a) 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 not time phased and the corresponding peak values occur at the same location (see 4-2-1/Appendix 2).

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 the principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see 4-2-1/Appendix 3).

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.

5.9.6(b) Equivalent alternating stress. The equivalent alternating stress is calculated in accordance with the following equations.

- For the crankpin fillet:

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

- For the journal fillet:

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

- For the outlet of crankpin oil bore:

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

where

σ_v = equivalent alternating stress, in N/mm^2, for other parameters referred to in 4-2-1/5.9.2(c), 4-2-1/5.9.3(c) and 4-2-1/5.9.5

5.9.7 Calculation of Fatigue Strength

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

- Related to the crankpin diameter:

$$\sigma_{DW} = \pm K \bullet (0.42 \bullet \sigma_B + 39.3) \bullet \left[0.264 + 1.073 \bullet D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \bullet \sqrt{\frac{1}{R_X}}\right]$$

where

R_X = R_H in the fillet area

= $D_O/2$ in the oil bore area

- Related to the journal diameter:

$$\sigma_{DW} = \pm K \bullet (0.42 \bullet \sigma_B + 39.3) \bullet \left[0.264 + 1.073 \bullet D_G^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \bullet \sqrt{\frac{1}{R_G}}\right]$$

where

σ_{DW} = allowable fatigue strength of crankshaft, in N/mm^2

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

= 1.05 for continuous grain flow forged or drop-forged crankshafts

= 1.0 for free form forged crankshafts (without continuous grain flow) factor for cast steel crankshafts with cold rolling treatment in fillet area

= 0.93 for cast steel crankshafts manufactured by companies using a cold rolling process approved by the Bureau

σ_B = minimum tensile strength of crankshaft material, in nN/mm^2

For other parameters refer to 4-2-1/5.9.4(c).

When a surface treatment process is applied, it must be specially approved.

These equations are subject to the following conditions:

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

As an alternative, the fatigue strength of the crankshaft can be determined by experiment based either on full size crank throw (or crankshaft) or on specimens taken from a full size crank throw.

In any case the experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment is to be submitted for approval (method, type of specimens, number of specimens (or crank throws), number of tests, survival probability, confidence number).

5.9.8 Acceptability Criteria

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

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

where

Q = acceptability factor

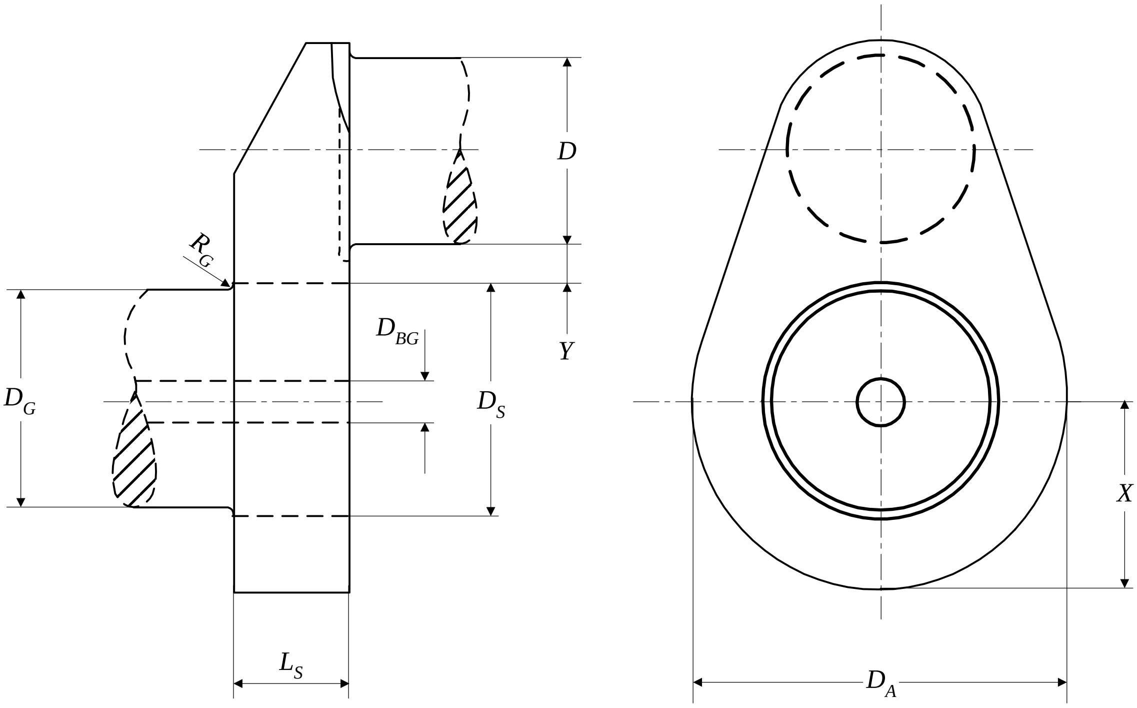
Adequate dimensioning of the crankshaft is ensured if the smallest of all acceptability factors satisfies the criteria:

$Q \geq 1.15$

5.9.9 Calculation of Shrink-fits of Semi-built Crankshaft

5.9.9(a) General. All crank dimensions necessary for the calculation of the shrink-fit are shown in 4-2-1/Figure 6.

FIGURE 6
Crank Throw of Semi-built Crankshaft *(2007)*



where

D_A = outside diameter of web, in mm, or twice the minimum distance x between center-line of journals and outer contour of web, whichever is less

D_S = shrink diameter, in mm

D_G = journal diameter, in mm

D_{BG} = diameter of axial bore in journal, in mm

L_S = length of shrink-fit, in mm

R_G = fillet radius of journal, in mm

Y = distance between the adjacent generating lines of journal and pin, in mm, $Y \geq 0.05 \bullet D_S$

where Y is less than $0.1 \bullet D_S$, special consideration is to be given to the effect of the stress due to the shrink-fit on the fatigue strength at the crankpin fillet.

With regard to the radius of the transition from the journal to the shrink diameter, the following is to be complied with:

$$R_G \geq 0.015 \bullet D_G$$

and

$$R_G \geq 0.5 \bullet (D_S - D_G)$$

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 4-2-1/5.9.9(c) and 4-2-1/5.9.9(d)

If the condition in 4-2-1/5.9.9(b) cannot be fulfilled, the calculation methods of Z_{min} and Z_{max} in 4-2-1/5.9.9(c) and 4-2-1/5.9.9(d) are not applicable due to multi-zone-plasticity problems.

In such case, Z_{min} and Z_{max} have to be established based on FEM calculations.

5.9.9(b) Maximum permissible hole in the journal pin. The maximum permissible hole diameter in the journal pin is calculated in accordance with the following equation:

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

where

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

M_{max} = absolute value of the maximum torque M_{Tmax}, N-m, in accordance with 4-2-1/5.9.3(b)

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, in N/mm^2

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

5.9.9(c) Necessary minimum oversize of shrink-fit. The necessary minimum oversize is determined by the greater value calculated according to:

$$Z_{\min} \geq \frac{\sigma_{sw} \bullet D_S}{E_m}$$

and

$$Z_{\min} \geq \frac{4000}{\mu \bullet \pi} \bullet \frac{S_R \bullet M_{\max}}{E_m \bullet D_S \bullet L_S} \bullet \frac{1 - Q_A^2 \bullet Q_S^2}{\left(1 - Q_A^2\right) \bullet \left(1 - Q_S^2\right)}$$

where

$Z_{\min}$ = minimum oversize, in mm

E_m = Young's modulus, in N/mm^2

σ_{SW} = minimum yield strength of material for crank web, in N/mm^2

Q_A = web ratio, $Q_A = \dfrac{D_S}{D_A}$

Q_S = shaft ratio, $Q_S = \dfrac{D_{BG}}{D_S}$

5.9.9(d) Maximum permissible oversize of shrink-fit. The maximum permissible oversize is calculated according to:

$$Z_{\max} \leq D_S \bullet \left(\frac{\sigma_{SW}}{E_m} + \frac{0.8}{1000} \right)$$

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

5.9.10 Other Reciprocating Components

All other reciprocating components (e.g., connecting rod) are to have acceptability factors of at least 1.15. Tightening torques are to be submitted for pretensioned bolts/studs.

5.11 Shaft Couplings and Clutches

The design and construction of fitted bolt and non-fitted bolt couplings, flexible couplings and clutches is to be in accordance with the provisions of 4-3-2/5.19.

7 Engine Appurtenances

7.1 Explosion Relief Valves

7.1.1 Application

Explosion relief valves of an approved type are to be installed on enclosed crankcases of all engines having a cylinder bore of 200 mm (8 in.) or above or having a crankcase gross volume of 0.6 m^3 (21 ft^3) or above.

7.1.2 Valve Construction and Sizing *(2006)*

The following requirements apply:

i) The free area of each explosion relief valve is not to be less than 45 cm^2 (7 in^2), and the total free area of all relief valves is to be not less than 115 cm^2 for each cubic meter (1 in^2 for each 2 ft^3) of crankcase gross volume. The total volume of the stationary parts within the crankcase may be discounted in estimating the crankcase gross volume (rotating and reciprocating components are to be included in the gross volume).

ii) Crankcase explosion relief valves are to be provided with lightweight spring-loaded valve discs or other quick-acting and self closing devices to relieve a crankcase of pressure in the event of an internal explosion and to prevent the inrush of air thereafter.

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

iv) Crankcase explosion relief valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than 0.2 bar (0.2 kgf/cm^2, 2.85 psi).

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

vi) *(2007)* Crankcase explosion relief valves are to be type tested in a configuration that represents the installation arrangements that will be used on an engine in accordance with Appendix 4-2-1A5.

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

viii) Crankcase explosion relief valves are to be provided with a copy of the 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.

ix) A copy of the installation and maintenance manual is to be provided on board.

x) Plans showing details and arrangements of the crankcase explosion relief valves are to be submitted for approval in accordance with 4-2-1/1.9.

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

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

7.1.3 Location of Valves *(2002)*

Engines having a bore of 200 mm (8 in.) and above, but not exceeding 250 mm (10 in.), are to have at least one valve near each end. However, for engines with more than 8 crank throws, an additional valve is to be fitted near the middle of the engine.

Engines having a bore exceeding 250 mm (10 in.), but not exceeding 300 mm (11.8 in.), are to have at least one valve in way of each alternate crank throw, with a minimum of two valves.

Engines having a bore exceeding 300 mm (11.8 in.) are to have at least one valve in way of each main crank throw.

7.1.4 Other Compartments of Crankcase

Additional relief valves are to be fitted on separate spaces of the crankcase such as gear or chain cases for camshaft or similar drives when the gross volume of such spaces is 0.6 m^3 (21 ft^3) and above.

7.1.5 Scavenge Spaces *(2006)*

Explosion relief valves are to be fitted in scavenge spaces which are in open connection to the cylinders.

7.2 Protection Against Crankcase Explosions *(2009)*

7.2.1 General *(2010)*

All engines rated at 2250 kW (3000 hp) and above or having cylinders of more than 300 mm (11.8 in.) bore are to be provided with one of the following arrangements as protection against crankcase explosions.

- Oil mist detection/monitoring arrangements (see 4-2-1/7.2.2), or
- Bearing temperature monitoring arrangements (see 4-2-1/7.2.3), or
- Alternative arrangements (see 4-2-1/7.2.4).

For low speed diesel engines, the above protection arrangements are to initiate an alarm and an automatic slowdown of the engine.

For medium and high speed diesel engines, they are to initiate an alarm and an automatic shutdown of the engine.

For automatic shutdown or automatic slowdown for vessels with the **ACCU** notation, see item B7 of 4-9-4/Table 3A, item B4 of 4-9-4/Table 3B, item G4 of 4-9-4/Table 6B and item A3 of 4-9-4/Table 8.

Automatic shutdown is not permitted for emergency diesel engines, see 4-8-2/5.19.2(c) and item B3 of 4-8-2/Table 1.

7.2.2 Oil Mist Detection/Monitoring Arrangements *(2006)*

7.2.2(a) General (2007). Where crankcase oil mist detection/monitoring arrangements are fitted to engines, they are to be of an approved type and tested in accordance with Appendix 4-2-1A6.

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

i) Schematic layout of engine oil mist detection/monitoring and alarm system showing location of engine crankcase sample points and piping arrangements together with pipe dimensions to detector/monitor.

ii) *(2009)* 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. All areas that have open communication with the crankcase are to be adequately monitored.

iii) The manufacturer's maintenance and test manual.

iv) Information relating to type or in-service testing of the engine with engine protection system test arrangements having approved types of oil mist monitoring equipment.

A copy of the oil mist detection/monitoring equipment maintenance and test manual required is to be provided onboard.

7.2.2(c) Arrangements. The following requirements apply:

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

ii) Where there are multi engine installations, each engine is to be provided with oil mist detection/monitoring and a dedicated alarm.

iii) Oil mist detection/monitoring and alarm systems are to be capable of being tested on the test bed and onboard under engine at standstill and engine running at normal operating conditions in accordance with manufacturer's test procedures.

iv) *(2009)* Alarms, automatic slowdowns and automatic shutdowns for the oil mist detection/monitoring system are to be in accordance with 4-2-1/7.2.1, 4-8-2/Table 1, 4-9-4/Table 3A, 4-9-4/Table 3B, 4-9-4/Table 6B and 4-9-4/Table 8, as applicable. The system arrangements are to comply with 4-9-1/9.

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

vi) The oil mist detection/monitoring system is to provide an indication that any lenses fitted in the equipment and used in determination of the oil mist level have been partially obscured to a degree that will affect the reliability of the information and alarm indication.

vii) Where oil mist detection/monitoring equipment includes the use of programmable electronic systems, the arrangements are to be in accordance with 4-9-6/3.

viii) Plans showing details and arrangements of oil mist detection/monitoring and alarm arrangements are to be submitted for approval in accordance with 4-2-1A1/28.

ix) The equipment, together with detectors/monitors, is to be tested in the presence of the Surveyor when installed on the test bed and onboard to demonstrate that the detection/monitoring and alarm system functionally operates.

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

7.2.3 Bearing Temperature Monitoring Arrangements *(2009)*

Where bearing temperature monitoring arrangements are provided, the following requirements apply:

7.2.3(a) Monitoring of bearings. All bearings (main, crank, crosshead, thrust, etc.) that have open communication with the crankcase are to be monitored for abnormal temperature.

7.2.3(b) Slow Speed Diesel Engines. For slow speed diesel engines, the lubricating oil temperature at the outlet of each main, crank, crosshead and thrust bearing may be monitored in lieu of directly monitoring the temperature of these bearings.

7.2.3(c) Alarms, automatic slowdowns and automatic shutdowns for bearing temperature monitoring arrangements are to be in accordance with 4-2-1/7.2.1, 4-8-2/Table 1, 4-9-4/Table 3A, 4-9-4/Table 3B, 4-9-4/Table 6B and 4-9-4/Table 8, as applicable. The system arrangements are to comply with 4-9-1/9.

7.2.4 Alternative Arrangements *(2009)*

Where alternative arrangements are provided for the prevention of the build-up of oil mist that may lead to a potentially explosive condition within the crankcase details are to be submitted for review, in order to determine the effectiveness of the arrangements for each specific engine design. In order to prevent the build up of a potentially explosive condition, it may be necessary to provide combinations of means, such as: oil splash temperature monitoring, crankcase pressure monitoring, recirculation arrangements. The details are to include, but not be limited to the following:

- Engine particulars – type, power, speed, stroke, bore and crankcase volume.
- Details of arrangements to prevent the build up of potentially explosive conditions within the crankcase.

- Evidence to demonstrate that the arrangements are effective in preventing the build up of potentially explosive conditions together with details of in-service experience.
- Operating instructions and the maintenance and test instructions.

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

7.3 Governors and Overspeed Protection

7.3.1 Governors

All diesel engines, except those driving electric generators (see 4-2-1/7.5), are to be fitted with governors which will prevent the engines from exceeding the rated speed by more than 15%.

7.3.2 Overspeed Protective Device

In addition to the governor, each main propulsion engine having a rated power of 220 kW (295 hp) and over and which can be declutched or which drives a controllable pitch propeller, is to be fitted with a overspeed device so adjusted that the speed cannot exceed the maximum rated speed by more than 20%. This overspeed device, including its driving mechanism, is to be independent from the normal governor.

7.3.3 Electronic Governors *(2006)*

Electronic speed governors fitted to main propulsion diesel engines, and which form part of a remote propulsion control system, are to comply with the following:

i) If lack of power to the governor control and actuator systems may cause major and sudden changes in the preset speed and/or direction of thrust of the propeller, an automatically available back up power supply is to be provided so as not to interrupt the power supply to these systems. An alarm for the failure of the main power supply is to be provided at the main and remote (if provided) propulsion control stations.

ii) Local control of the engines is to be possible. For this purpose, means are to be provided at the local control position to disconnect the remote control signal. If this will also disconnect the speed governing functions required by 4-2-1/7.3.1, an additional separate speed governor is to be provided for such local mode of control.

iii) Electronic speed governors and their electrical actuators are to be subjected to prototype environmental tests in accordance with 4-9-7/13.1. In addition, the tests required by 4-9-7/Table 10 are to be carried out in the presence of the Surveyor as prototype testing. However, no production unit certification in accordance with 4-9-7/13.3 is required.

7.5 Governors and Overspeed Protection for Engines Driving Generators

7.5.1 Speed Governing *(2004)*

Diesel engines driving propulsion, auxiliary or emergency electric generators are to be fitted with an operating governor which is capable of automatically maintaining the speed within the following limits:

7.5.1(a) (2007) The transient frequency variations in the electrical network when running at the indicated loads below are to be within ±10% of the rated frequency with a recovery time within ±1% of the final steady state condition in not more than 5 seconds when:

i) Running at full load (equal to rated output) of the generator and the maximum electrical step load is suddenly thrown off;

In the case when a step load equivalent to the rated output of a generator is thrown off, a transient frequency variation in excess of 10% of the rated frequency may be acceptable, provided the overspeed protective device fitted in addition to the governor, as required by 4-2-1/7.5.3, is not activated.

ii) Running at no load and 50% of the full load of the generator is suddenly thrown on, followed by the remaining 50% after an interval sufficient to restore the frequency to steady state.

7.5.1(b) Where the electrical power system is fitted with a power management system and sequential starting arrangements, the application of loads in multiple steps of less than 50% of rated load in 4-2-1/7.5.1(a)ii) above may be permitted, provided it is in accordance with 4-2-1/Figure 7. The details of the power management system and sequential starting arrangements are to be submitted and its satisfactory operation is to be demonstrated to the Surveyor.

7.5.1(c) *(2007)* For diesel engines driving emergency generators, the requirements of 4-2-1/7.5.1(a)i) and 4-2-1/7.5.1(d) are to be met even when:

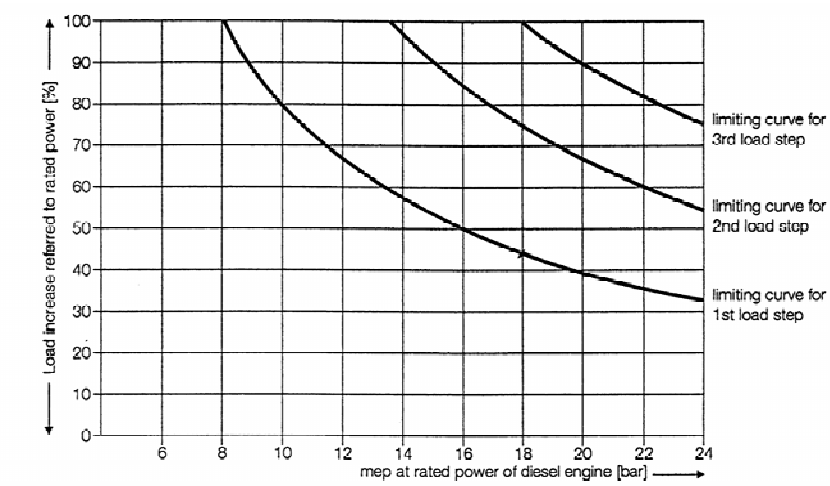
i) Their total consumer load is applied suddenly, or

ii) 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.

7.5.1(d) The permanent frequency variation is to be within ±5% of the rated frequency at any load between no load and the full load.

FIGURE 7
Limiting Curves for Loading 4-stroke Diesel Engines Step by Step from No-load to Rated Power as Function of the Brake Mean Effective Pressure



7.5.2 Generators in Parallel

For diesel engines driving AC generators that operate in parallel, the governor's characteristics are to be such that in the range between 20% and 100% of the combined rated load of all generators, the load on any individual generator will not differ from its proportionate share of the total combined load by more than 15% of the rated power of the largest generator or 25% of the individual generator, whichever is less.

Provisions are to be made to adjust the governors sufficiently fine in order to permit a load adjustment within the limits of 5% of the rated load at normal frequency.

7.5.3 Overspeed Protective Device

In addition to the governor, each engine driving an electric generator and having a rated power of 220 kW (295 hp) and over is to be fitted with a separate overspeed device so adjusted that the speed cannot exceed the maximum rated speed by more than 15%. Provision is to be made for hand tripping.

7.7 Cylinder Overpressure Monitoring *(2001)*

For diesel engines having a bore exceeding 230 mm (9 in.), each cylinder is to be fitted with a means of indicating a predetermined overpressure. This may be a relief valve, a sentinel valve, audible and visual alarms, or equivalent. The device is to be set at a pressure to be determined by the engine manufacturer.

7.9 Scavenging Blowers

At least one scavenging blower, either of the reciprocating or the rotary type, is to be provided for each two-cycle propulsion engine. Each ocean-going vessel of 300 gross tonnage and over, having a single two-cycle propulsion engine with an attached rotary scavenging blower, driven by means other than a gear train, is to be provided with at least two such blowers. The capacity of each of the two attached rotary scavenging blowers is to be sufficient to provide not less than one-half the maximum rated engine rpm when one blower is out of service. Provisions for design and construction of scavenge blowers are in Section 4-2-2.

7.11 Fire Extinguishing System for Scavenge Manifold

For crosshead type engines, scavenge spaces in open connection to the cylinder are to be permanently connected to an approved fire extinguishing system entirely separate from the fire extinguishing system of the engine room. A steam smothering system is acceptable for this purpose. Provisions for the design and installation of fixed fire-extinguishing system are in Section 4-7-3.

7.13 Warning Notices

7.13.1 Crankcase

To caution against opening hot crankcase, suitable warning notices are to be fitted, preferably on a crankcase door on each side of the engine or on the engine control stand. The notices are to specify a period of time for cooling after shutdown, based upon the size of the engine, but not less than 10 minutes in any case, before opening the door. Such notice is also to warn against restarting an overheated engine until the cause of overheating has been remedied.

7.13.2 Barred Speed Ranges

Where a barred speed range is specified in accordance with torsional vibration analysis, the engine speed indicator is to be so marked. A warning notice is to be fitted to the engine and at all local and remote propulsion control stations to the effect that operation in the barred range is to occur only while passing through the range and that operation within the barred range is to be avoided. See 4-3-2/7.5 for torsional vibration criteria.

7.15 Jacket Drain and Overpressure Protection

A drain cock is to be fitted at the lowest point of all cooling medium jackets. Means are to be provided to prevent the cooling medium jacket from being overpressurized. This will not be required if the cooling pump is of the centrifugal type, such that the no-flow pressure is no greater than the design pressure of the jacket.

7.17 Monitoring

Required monitoring for engine’s fuel oil, lubricating oil, cooling water, starting air and exhaust gas systems are provided in the system requirements, see 4-2-1/9 below. For propulsion machinery spaces intended for centralized or unattended operation, engine and engine system monitoring and safety system functions are provided in Part 4, Chapter 9; see e.g., 4-9-4/Table 3A and 4-9-4/Table 3B.

9 Piping Systems for Diesel Engines

The requirements of piping systems, essential for operation of diesel engines intended for propulsion, maneuvering, electric power generation and vessel safety, are provided in Section 4-6-5. These systems include:

Fuel oil:	4-6-5/3
Lubricating oil:	4-6-5/5
Cooling water:	4-6-5/7
Starting air:	4-6-5/9
Electric starting:	4-8-2/11.11
Crankcase ventilation:	4-6-5/13
Exhaust gas:	4-6-5/11
Hydraulic and pneumatic systems:	4-6-7/3 and 4-6-7/5

11 Installation of Diesel Engines

11.1 Seating Arrangements for Diesel Engines

Diesel engines are to be securely supported and mounted to the vessel's structure by bolted connections with consideration given to all of the static and dynamic forces imposed by the engine.

11.3 Metal Chocks

Where metal chocks are used, they are to be made of forged steel, rolled steel or cast steel.

11.5 Cast Resin Chocks

Cast resin chocks of an approved type (see 1-1-A3/5 for type approval) may be used, provided that the arrangements and installation procedures are in accordance with the manufacturer's recommendations. Arrangements of the proposed installation, along with installation parameters such as engine deadweight, holding-down bolt tightening torque, etc., and calculations showing that the manufacturer's specified pressure is not exceeded, are to be submitted for review in each case.

11.7 Resilient Mountings

Resilient mountings may be used within the limits of the manufacturer's instructions and specifications for the resilient elements' elasticity and durability under shipboard ambient conditions.

11.9 Hot Surfaces

Hot surfaces likely to come into contact with the crew during operation are to be suitably guarded or insulated. Where the temperature of hot surfaces are likely to exceed 220°C (428°F), and where any leakage, under pressure or otherwise, of fuel oil, lubricating oil or other flammable liquid is likely to come into contact with such surfaces, they are to be suitably insulated with non-combustible materials that are impervious to such liquid. Insulation material not impervious to oil is to be encased in sheet metal or an equivalent impervious sheath.

13 Testing, Inspection and Certification of Diesel Engines

13.1 Material and Nondestructive Tests

For testing and nondestructive tests of materials intended for engine construction, see 4-2-1/3.1 and 4-2-1/3.3.

13.3 Hydrostatic Tests of Diesel Engine Components

Hydrostatic tests of diesel engine parts and components are to be in accordance with 4-2-1/Table 2. These tests are to be carried out by the manufacturer whose certificate of tests will be acceptable. However, independently driven pumps for fuel oil, lubricating oil, and cooling water services of diesel engines of bores exceeding 300 mm (11.8 in.) are required to be certified by the Surveyor; see 4-6-1/7.3.

13.5 Relief and Safety Valves

All relief and safety valves are to be tested and set in the presence of the Surveyor.

13.6 Manufacturer's Quality Control *(2002)*

13.6.1 Quality Plan

Prior to commencement of construction, the manufacturer is to submit to the Surveyor a quality plan setting out the quality control that it plans to perform on, but not limited to the following:

- issuance of material specifications for purchasing
- receiving inspection of materials
- receiving inspection of finished components and parts
- dimensional and functional checks on finished components and parts
- edge preparation and fit-up tolerances
- welding procedure qualification
- welder qualification
- Weld inspection plan
- welding defect tracking
- NDT written procedures and qualification documentation
- NDT plan
- casting and weld defect resolutions
- assembly and fit specifications
- subassembly inspection: alignment and dimension checks, functional tests
- testing of safety devices
- hydrostatic testing plan
- engine test plan

The Surveyor is to review the quality plan and may, at his discretion, identify inspection hold points on the quality plan, in addition to those required in 4-2-1/13.1 through 4-2-1/13.5. In all cases, the manufacturer is to maintain traceability and documentation of materials and parts; welds, welders and NDT records; dimension and alignment measurements bolted joint pretension values; hydrostatic test records; and other quality control measurements; which are to be made available to the Surveyor during the course of the inspection. Upon completion of the construction, the quality control documents are to be compiled in a folio and a copy of which is to be provided to the Surveyor.

13.6.2 Welding on Engine Entablatures, Frames, Bedplates and Power Transmitting Component

13.6.2(a) Welding procedure. Before proceeding with welding, the manufacturer is to prove to the satisfaction of the Surveyor that the intended welding process, welding filler metal, preheat, post weld heat treatment, etc., as applicable, have been qualified for joining the base metal. In general, the intended welding procedure is to be supported by welding procedure qualification record (PQR) acceptable to or conducted in the presence of the Surveyor. The extent to which a PQR may be used to support multiple welding procedures is to be determined based on a recognized welding standard and is subject to acceptance by the Surveyor.

13.6.2(b) Welders and welding operators. Before proceeding with welding, the manufacturer is to prove to the satisfaction of the Surveyor that the welder or the welding operator is qualified for performing the intended welding procedure. In general, welders and welding operators are to be qualified in accordance with 2-4-3/11 in the presence of the Surveyor or supported by documented welder performance qualification records (WPQ) acceptable to the Surveyor. The extent to which a WPQ may be used to support multiple welding procedures is to be determined based on a recognized welding standard and is subject to acceptance by the Surveyor.

13.6.2(c) Facility-specific PQR and WPQ. To prove the capability of specific facilities, PQR and WPQ are to be conducted at and certified by the facilities where the fabrication or weld repair is to be conducted. PQR and WPQ conducted at other facilities are normally not acceptable for supporting the intended welding, without specific acceptance by the Surveyor.

13.6.2(d) Welding filler metals. All welding filler metals are to be certified by their manufacturers as complying with appropriate recognized national or international standards. Welding filler metals tested, certified and listed by the Bureau in its publication, *Approved Welding Consumables*, for meeting such a standard may be used in all cases. See Part 2, Appendix 2 for approval of filler metals. Welding filler metals not so listed may also be accepted provided that:

i) They are of the same type as that proven in qualifying the welding procedure; and

ii) They are of a make acceptable to the Surveyor; and

iii) For welding of Group I engineering structures, representative production test pieces are to be taken to prove the mechanical properties of the weld metal.

13.6.2(e) Tack welds. Tack welds, where used, are to be made with filler metal suitable for the base metal. Tack welds intended to be left in place and form part of the finished weld are to be made by qualified welders using process and filler metal the same as or equivalent to the welding procedure to be used for the first pass. When preheating is required, the same preheating should be applied prior to tack welding.

13.6.2(f) Repair of defective welds. Any weld joint imperfection disclosed by examination in 4-2-1/13.6.3(c) and deemed unacceptable is to be removed by mechanical means or thermal gouging processes, after which the joint is to be welded using the appropriate qualified welding procedure by a qualified welder. Preheat and post-weld heat treatment is to be performed, as applicable. Upon completion of repair, the repaired weld is to be re-examined by the appropriate technique that disclosed the defect in the original weld.

13.6.2(g) Repair of castings by welding. Casting surface defects and defects revealed by nondestructive tests specified in 4-2-1/Table 1 and deemed unacceptable may be repaired by welding. All welding repairs are to be conducted using qualified welding procedure and by qualified welders as per 4-2-1/13.6.2(a), 4-2-1/13.6.2(b) and 4-2-1/13.6.2(c). The welding procedure, preheat and post weld heat treatment, as applicable, are to be in accordance with engine designer's specifications and supported by appropriate PQR. The Surveyor is to be notified prior to proceeding with the repair. Where welding repair is to be conducted at the foundry, the same procedure is to be adhered to. Defects detected by nondestructive tests required by 4-2-1/Table 1 are to be re-examined by at least the same technique after completion of repair.

13.6.3 Nondestructive Tests and Inspections

13.6.3(a) Qualification of procedures and operators. Before proceeding to conduct nondestructive tests required by 4-2-1/Table 1, the manufacturer is to have a written procedure for conducting each of these tests and for qualifying the operators intended for conducting these tests. Subcontractors, if employed for this purpose, are to be similarly qualified. In general, the processes of qualifying the procedures and the operators and the necessary technical supervision and training are to be in accordance with a recognized standard.

13.6.3(b) Nondestructive test procedures of engine parts. Parts requiring ultrasonic tests by 4-2-1/Table 1 are each to be provided with a test plan. Typically, for ultrasonic testing, the plan is to specify:

– part to be tested
– ultrasonic equipment
– couplant
– reference block(s)
– scanning coverage and rate
– calibration procedure
– acceptance standards

As a minimum, dye penetrant test plans are to specify the part to be tested, penetrant type and developer, procedure for retest, allowable ambient and test piece temperatures, and acceptance standards; and magnetic particle test plans are to specify parts to be tested, magnetization technique and equipment, surface preparation, type of ferromagnetic particles, and acceptance standards.

13.6.3(c) Nondestructive tests of welds. The manufacturer's quality plan for weld inspection should include visual inspection, measurement of weld sizes, as well as nondestructive tests (dye-penetrant, magnetic particle, radiography or ultrasonic), as may be specified by the engine designer for important structural parts.

13.6.3(d) Documentation. The manufacturer is to document and certify the results of the required nondestructive tests. The number and locations of unacceptable indications found are to be reported, together with corrective action taken, preferably on a sketch, along with questionable areas and any required areas not examined, where applicable.

13.6.3(e) Witness by Surveyor. All documents required in 4-2-1/13.6.3 are to be made available to the Surveyor. Where in doubt, or for purposes of verification, the Surveyor may request for a demonstration of any nondestructive tests required by 4-2-1/Table 1 to be conducted in his presence.

13.6.4 Assembly and Fit

The manufacturer's quality plan is to require checks on important fit, alignment, tolerances, pretensioning, etc. specified by the engine designers. Data measured in the as-assembled condition are to be recorded and made available to the Surveyor, who may request for verification of the recorded data.

13.7 Type Tests of Diesel Engines

13.7.1 Application *(2009)*

Each new type of diesel engine, as defined in 4-2-1/13.7.2, is to be type tested under the conditions specified in 4-2-1/13.7, except that mass-produced engines intended to be certified by quality assurance may be type tested in accordance with 4-2-1/13.11. The testing of the engine for the purpose of determining the rated power and 110% power is to be conducted at the ambient reference conditions given in 4-2-1/1.7 of the Rules, or power corrections are to be made. A type test carried out for a type of engine at any place of manufacture will be accepted for all engines of the same type built by licensees and licensers. A type test carried out on one engine having a given number of cylinders will be accepted for all engines of the same type having a different number of cylinders.

Where a previously approved engine of the conventional type (i.e., non-electronically controlled) is modified to be an electronically controlled engine, the requirement to conduct those tests specified in 4-2-1/13.7 which were already conducted as part of the conventional engine approval and for which it can be shown that the results would not be impacted due to the addition of the electronic controls, may be subject to special consideration.

13.7.2 Engine Type Definition

For purposes of type tests, a diesel engine "type", as specified by the manufacturer's type designation, is to be defined by:

- The working cycle (2-stroke, 4-stroke)
- The cylinder arrangement (in-line, vee)
- The rated power per cylinder at rated speed and mean effective pressure
- The kind of fuel (liquid, dual fuel, gaseous)
- The method of fuel injection (direct or indirect)
- The cylinder bore
- The stroke
- The scavenging system (naturally aspirated or supercharged)
- The supercharging system (constant or pulsating pressure)
- The charged air cooling system (provided with intercooler or not, number of cooling stages)

Engines may be considered the same type if they do not differ from any of the above items.

13.7.3 Increase in Rated Power

The rated power of an engine type, which has proven reliability in service, may be increased by not more than 10% without any further type test, subject to prior approval of relevant plans and particulars.

13.7.4 Stages of Type Tests

Each type test is subdivided into three stages:

- Stage A: manufacturer's tests;
- Stage B: type assessment tests to be conducted in the presence of a Surveyor;
- Stage C: component inspection after the test by a Surveyor.

These stages are described in details as follows.

13.7.4(a) Stage A: manufacturer's tests. The manufacturer is to carry out functional tests in order to collect and record the engine's operating data. During these tests, the engine is to be operated at the load points specified by the engine manufacturer and the pertinent operating values are to be recorded. The load points may be selected according to the range of applications.

Where an engine is designed for operation without using mechanically driven cylinder lubricators, such capability is to be demonstrated and recorded.

For engines which are designed to operate on heavy fuel oil, the suitability for this is to be proven during the tests. Where not possible, this is to be demonstrated during shipboard trials of the first engine brought into service.

The tests are to include the normal and the emergency operating modes as specified below:

i) *Normal operating mode.*

The load points 25%, 50%, 75%, 100% and 110% of the rated power:

- Along the nominal (theoretical) propeller curve and at constant rated speed for propulsion engines;
- At constant rated speed for engines intended to drive electric generators;

The limit points of the permissible operating range, as defined by the engine manufacturer.

ii) *Emergency operating mode.* The manufacturer's test for turbocharged engines is to include the determination of the maximum achievable continuous power output in the following cases of simulated turbocharger damage:

- Engines with one turbocharger: with the rotor blocked or removed;
- Engines with two or more turbochargers: with one turbocharger shut off.

13.7.4(b) Stage B: type tests to be witnessed by the Surveyor. The engine is to be operated at the load points shown in 4-2-1/Figure 8. The data measured and recorded at each load point is to include all necessary parameters for the engine operation.

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. For 4-2-1/13.7.4(b)i) below, an operating time of two hours is required and two sets of readings are to be taken at a minimum interval of one hour. For 4-2-1/13.7.4(b)ii) through 4-2-1/13.7.4(b)vi) below, the operating time per load point is not to be less than 30 minutes.

i) Rated power, i.e., 100% output at 100% torque and 100% speed corresponding to load point 1.

ii) 100% power at maximum permissible speed corresponding to load point 2.

iii) 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.

iv) Minimum permissible speed at 100% torque corresponding to load point 4.

v) Minimum permissible speed at 90% torque corresponding to load point 5.

vi) Part load operation, e.g., 75%, 50%, 25% of maximum continuous rated power and speed according to the nominal propeller curve corresponding to point 6, 7 and 8, and at rated speed with constant governor setting corresponding to points 9, 10 and 11.

For turbocharged engines, maximum achievable power when operating along the nominal propeller curve and when operating with constant governor setting for rated speed under the following conditions:

i) Engines equipped with one turbocharger: with the rotor blocked or removed,

ii) Engines equipped with two or more turbochargers: with one turbocharger shut off.

Functional tests are to be performed for the following:

i) The lowest engine speed according to the nominal propeller curve

ii) The engine starting and reversing appliances, where applicable

iii) The speed governor

iv) The safety system, particularly for overspeed and low lubricating oil pressure

v) *Integration Test (2009):* 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 determined based on the FMEA as required in Appendix 4-2-1A1 of the Rules.

13.7.4(c) Stage C: component inspection by the Surveyor. Immediately after the test run, the following components of one cylinder for in-line and of two cylinders for V-engines are to be presented for the Surveyor's inspection:

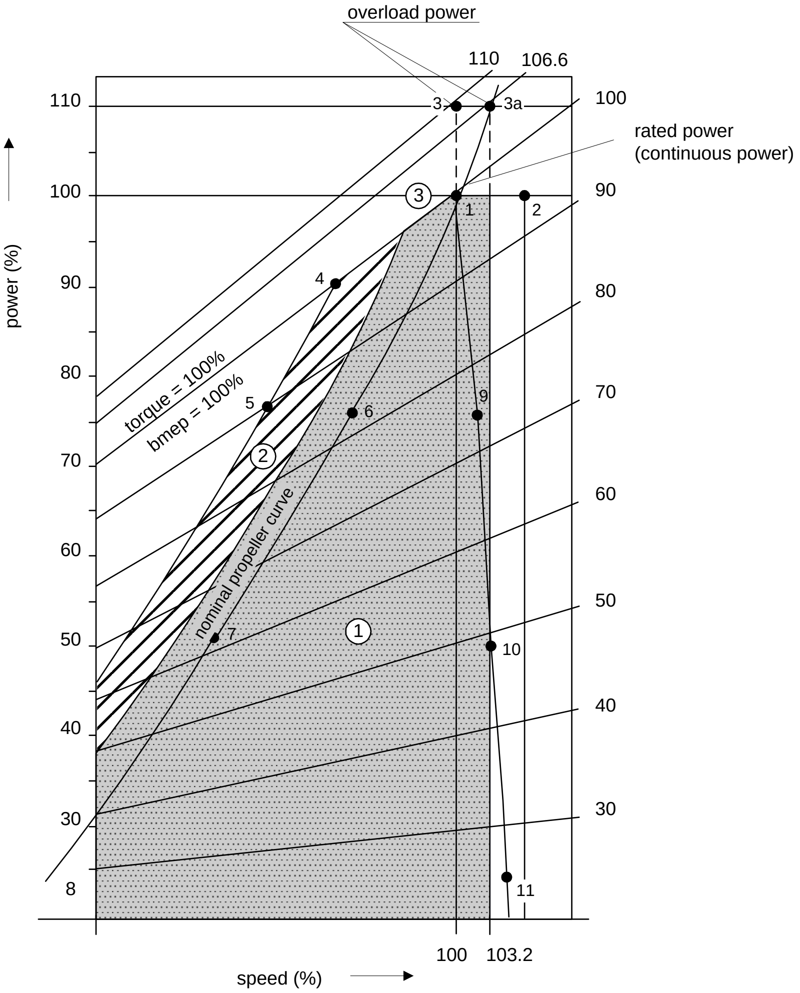
- Piston removed and dismantled
- Crosshead bearing, dismantled
- Crank bearing and main bearing, dismantled
- Cylinder liner in the installed condition
- Cylinder head, valves disassembled
- Control gear, camshaft and crankcase with opened covers

Further dismantling of the engine may be required by and at the discretion of the Surveyor.

13.7.5 Additional Tests

For engines intended to be used for emergency services, supplementary tests according to the regulations of the Administration whose flag the vessel flies may be required.

FIGURE 8
Type Test Power/Speed Diagram



(1) = range of continuous operation

(2) = range of intermittent operation

(3) = range of short-time overload operation

13.9 Shop Tests of Each Produced Diesel Engine

Each diesel engine where required to be certified by 4-2-1/1.1, except those manufactured on the basis of the Type Approval Program referred to in 4-2-1/13.13.2, is to be tested in the presence of a Surveyor, in accordance with the provisions of 4-2-1/13.9. The scope of tests may be expanded as called for by the specific engine application.

13.9.1 General

The operational data corresponding to each of the specified test load conditions are to be determined and recorded and all results are to be compiled in an acceptance protocol to be issued by the engine manufacturer. In each case, all measurements conducted at the various load points shall be carried out at steady operating conditions. The readings for 100% power (rated power at rated speed) are to be taken twice at an interval of at least 30 minutes.

13.9.2 Engines Driving Propellers

Main propulsion engines driving propellers are to be tested under the following conditions:

i) 100% of rated power at rated engine speed (n_o), for at least 60 minutes, after having reached steady conditions.

ii) 110% of rated power at an engine speed of n = 1.032n_o for 30 minutes, after having reached steady conditions.

iii) After running on the test bed, the fuel delivery system of the engine is to be adjusted so that the engine output is limited to the rated power and so that the engine cannot be overloaded under service condition.

iv) 90%, 75%, 50% and 25% of rated power, in accordance with the nominal propeller curve.

v) Starting and reversing maneuvers.

vi) Testing of governor and independent overspeed protective device.

vii) Testing of shutdown device.

13.9.3 Engines Driving Generators Dedicated for Propulsion Motors

For engines intended for driving electric propulsion generators, the tests are to be performed at the rated speed with a constant governor setting under the following conditions:

i) 100% rated power for at least 60 min., after having reached steady conditions.

ii) 110% of rated power for 30 min., after having reached steady conditions.

iii) After running on the test bed, the fuel delivery system of the engine is to be adjusted so that an overload power of 110% of the rated power can be supplied. Due regard is to be given to service conditions after installation on board and to the governor characteristics, including the activation of generator protective devices. See also 4-2-1/7.5.1(b) for governor characteristics associated with power management systems.

iv) 75%, 50% and 25% of rated power and idle run.

v) Start-up tests.

vi) Testing of governor and independent overspeed protective device.

vii) Testing of shutdown device.

13.9.4 Engines Driving Other Generators and Machinery *(2009)*

Engines intended for driving vessel service generators (consumers of which may include propulsion motors), emergency generators, or other essential machinery are to be tested as specified in 4-2-1/13.9.3. After running on the test bed, the fuel delivery system of the engine is to be adjusted so that an overload power of 110% of the rated power can be supplied. Due regard is to be given to service conditions after installation on board and to the governor characteristics including the activation of generator protective devices. See also 4-2-1/7.5.1(b) for governor characteristics associated with power management systems.

13.9.5 Electronically Controlled Engines *(2009)*

For electronically controlled engines, integration tests are to be conducted. They 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 determined based on those tests that have been established for the type testing as per 4-2-1/13.7.4(b)v).

13.9.6 Inspection After Tests

After shop tests, engine components, randomly selected at the discretion of the Surveyor, are to be presented for inspection. Where engine manufacturers require crankshaft deflection to be periodically checked during service, the crankshaft deflection is to be measured at this time after the shop test and results recorded for future reference.

13.11 Type Tests of Mass-produced Diesel Engines

13.11.1 Application *(2007)*

Each type of diesel engine mass produced (see 4-2-1/13.11.2) under the accepted quality assurance program is to be type tested in accordance with the provisions of 4-2-1/13.11. A type test carried out for a type of engine at a place of manufacture will be accepted for all engines of the same type built by licensees and licensers. A type test carried out on one engine having a given number of cylinders will qualify all engines of the same type having a different number of cylinders. The type test is to be conducted in the presence of the Surveyor.

Consideration will be given to modification of the type test requirements for existing engine designs which have proven reliability in service.

13.11.2 Definition of Mass Production of Diesel Engines

A diesel engine intended for propulsion or auxiliary service is considered mass-produced if it meets the following criteria:

- The engines are produced in quantity.
- The materials and components used for the construction of the engines are manufactured in accordance with approved quality control procedures specified by the engine builder.
- The machinery used for manufacturing of the engine components is specially calibrated and subject to regular inspection in order to meet the engine builder's specifications and quality requirements and to allow for assembly or interchangeability of components without any re-machining.
- Each assembled engine undergoes a bench test in accordance with specified procedures.
- Final testing, in accordance with specified procedures, is carried out on engines selected at random after bench testing.

13.11.3 Tests to be Witnessed by the Surveyor

13.11.3(a) Load points:

i) 80 hours at rated power;

ii) 8 hours at 110% overload and alternately 100% rpm and 103% rpm;

iii) 10 hours at partial loads (in steps of 90%, 75%, 50% and 25% of rated power);

iv) 2 hours at intermittent loads.

The tests are to be conducted in cycles, with each cycle comprising all of the above load points, and the cycle repeated until the specified durations of the tests are achieved.

13.11.3(b) Functional tests:

i) The engine starting and reversing appliances,

ii) The speed governor,

iii) The overspeed device,

iv) The lubricating oil failure indication device,

v) The condition of turbocharger out of action (where applicable),

vi) Running at minimum speed for main propulsion engines and at idle speed for auxiliary engines.

13.11.3(c) Component inspection. After the type test, all main parts of the engine are to be dismantled and examined by the Surveyor.

13.11.4 Measurements and Recordings

The following particulars are to be measured and recorded:

Ambient test conditions:

- Air temperature
- Barometric pressure
- Relative humidity
- External cooling water temperature
- Characteristics of fuel and lubricating oil

Engine measurements:

- Engine power
- Engine rpm
- Torque or brake load
- Maximum combustion pressure (indicator diagram where practicable)
- Exhaust smoke
- Lubricating oil pressure and temperature
- Cooling water pressure and temperature
- Exhaust gas temperature in exhaust manifold and, if possible, at each cylinder outlet and for supercharged engines
- Turbocharger rpm
- Air pressure and temperature at inlet and outlet of turbocharger and cooler
- Exhaust gas pressure and temperature at inlet and outlet of exhaust gas turbine and charge air cooler
- Cooling water temperature at charge air cooler inlet

Results of examination done after type tests: This should include disassembling and examination of the main parts and the parts subject to wear.

13.11.5 Additional Tests

For engines intended for different purposes and having different performances for each purpose, the type test program is to be extended to cover each performance under consideration of the most severe condition.

13.13 Certification of Diesel Engine

13.13.1 General

Each diesel engine required to be certified by 4-2-1/1.1 is:

i) To have its design approved by the Bureau; for which purpose, plans and data as required by 4-2-1/1.9 are to be submitted to the Bureau for approval; and the performance of the engine is to be verified by a type test which is to be conducted in the presence of a Surveyor in accordance with 4-2-1/13.7 or 4-2-1/13.11, as appropriate;

ii) To be surveyed during its construction, which is to include, but not limited to, material and nondestructive tests as indicated in 4-2-1/13.1, hydrostatic tests of components as indicated in 4-2-1/13.3, and tests and setting of relief valves, as applicable; and

iii) To subject the finished unit to a performance test conducted in accordance with 4-2-1/13.9 to the satisfaction of a Surveyor.

13.13.2 Certification Under Type Approval Program *(2003)*

13.13.2(a) Product design assessment. Upon application by the manufacturer, each model of a type of diesel engine may be design assessed as described in 1-1-A3/5.1. For this purpose, each design of an engine type is to be approved in accordance with 4-2-1/13.13.1i). Engines so approved may be applied to the Bureau for listing on the ABS website in the Design Approved Products Index [see 1-1-A3/5.1 (DA)]. Once listed, and subject to renewal and updating of certificate as required by 1-1-A3/5.7, engine particulars will not be required to be submitted to the Bureau each time the engine is proposed for use on board a vessel.

13.13.2(b) Mass produced engines. Manufacturer of mass-produced engines, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for a quality assurance assessment, described in 1-1-A3/5.5 (PQA).

Upon satisfactory assessment under 1-1-A3/5.5 (PQA), engines produced in those facilities will not require a Surveyor's attendance at the tests and inspections indicated in 4-2-1/13.13.1ii) and 4-2-1/13.13.1iii). Such tests and inspections are to be carried out by the manufacturer whose quality control documents will be accepted. Certification of each engine will be based on verification of approval of the design and on continued effectiveness of the quality assurance system. See 1-1-A3/5.7.1(a).

13.13.2(c) Non-mass Produced Engines. Manufacturer of non-mass produced engines, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for a quality assurance assessment, described in 1-1-A3/5.3.1(a) (AQS) or 1-1-A3/5.3.1(b) (RQS). Certification to 1-1-A3/5.5 (PQA) may also be considered in accordance with 4-1-1/Table 1.

13.13.2(d) Type Approval Program. Engine types which have their designs approved in accordance with 4-2-1/13.13.2(a) and the quality assurance system of their manufacturing facilities approved in accordance with 4-2-1/13.13.2(b) or 4-2-1/13.13.2(c) will be deemed Type Approved and will be eligible for listing on the ABS website as Type Approved Products.

15 Shipboard Trials of Diesel Engines

After the conclusion of the running-in program, diesel engines are to undergo shipboard trials in the presence of a Surveyor, in accordance with the following procedure.

15.1 Engines Driving Fixed Pitch Propellers

For main propulsion engines directly driving fixed pitch propellers, the following running tests are to be carried out:

i) At rated engine speed (n_o) for at least 4 hours.

ii) At engine speed corresponding to the normal continuous cruising power for at least 2 hours.

iii) At engine speed $n = 1.032n_o$ for 30 minutes (where engine adjustment permits).

iv) At minimum on-load speed.

v) Starting and reversing maneuvers.

vi) In reverse direction of propeller rotation at a minimum engine speed of $n = 0.7n_o$ for 10 minutes.

vii) Testing of the monitoring and safety systems.

15.3 Engines Driving Controllable Pitch Propellers

For main propulsion engines driving controllable pitch propellers or reversing gears, the tests as per 4-2-1/15.1 apply, as appropriate. In addition, controllable pitch propellers are to be tested with various propeller pitches.

15.5 Engines Driving Propulsion Generators

For engines driving electric propulsion generators, the running tests are to be carried out at the rated speed with a constant governor setting and are to be based on the rated electrical power of the driven generators, under the following conditions:

- At 100% rated power for at least 4 hours.
- At normal continuous cruising power for at least 2 hours.
- In reverse direction of propeller rotation at a minimum speed of 70% of the nominal propeller speed for 10 minutes.

Further, starting maneuvers and functional tests of the monitoring and safety systems are to be carried out. Governor characteristics associated with power management systems in 4-2-1/7.5.1(b) are to be demonstrated during the vessel's trial.

15.7 Engines Driving Generators or Essential Auxiliaries *(2006)*

Engines driving essential auxiliaries or generators, other than propulsion generators are to be subjected to an operational test for at least 4 hours. During this operational test, the set concerned is required to operate at its rated power for a period not less than 1 hour. In addition, the governor characteristics associated with power management systems in 4-2-1/7.5.1(b) are to be demonstrated.

15.9 Engines Burning Residual Fuel Oil

The suitability of propulsion and auxiliary diesel engines to burn residual fuel oils or other special fuel oils, where they are intended to burn such fuel oils in service, is to be demonstrated.

15.11 Torsional Vibration Barred Speed Range

Where torsional vibration analyses indicate that a torsional vibration critical is within the engine operating speed range, the conduct of torsiograph tests and marking of the barred speed range, as appropriate, are to be carried out in accordance with 4-3-2/11.3.1.

TABLE 1
Required Material and Nondestructive Tests of Diesel Engine Parts *(2007)*

		Nondestructive Tests (2)	
Engine Part (1)	*Material Tests* (5, 8)	*Magnetic Particle, Liquid Penetrant, or Similar Tests* (3)	*Ultrasonic Tests* (6)
Crankshafts, forged, cast, fully built, or semi-built	all	all	all
Crankshaft coupling flange (non integral) for main propulsion engines	above 400 mm (15.7 in.) bore	—	—
Coupling bolts for crankshaft	above 400 mm (15.7 in.) bore	—	—
Connecting rods and connecting rod bearing caps (7)	all	all	above 400 mm (15.7 in.) bore
Piston rods	above 400 mm (15.7 in.) bore	above 400 mm (15.7 in.) bore	above 400 mm (15.7 in.) bore
Crosshead	above 400 mm (15.7 in.) bore	—	—
Cylinder liner – steel/gray and nodular iron parts	above 300 mm (11.8 in.) bore	—	—
Steel/gray and nodular iron cylinder covers	above 300 mm (11.8 in.) bore	above 400 mm (15.7 in.) bore	all
Tie rods	all	above 400 mm (15.7 in.) bore	—
Steel piston crowns	above 400 mm (15.7 in.) bore	above 400 mm (15.7 in.) bore	all
Bolts and studs for cylinder covers, crossheads, main bearings, connecting rod bearings	above 300 mm (11.8 in.) bore	above 400 mm (15.7 in.) bore	—
Steel gear wheels for camshaft drives	above 400 mm (15.7 in.) bore	above 400 mm (15.7 in.) bore	—
Steel castings for welded bedplates (including main bearing housing)	all	all	all
Steel forgings for welded bedplates (including main bearing housing)	all	—	—
Plates for welded bedplates (including main bearing housing); frames, crankcases and entablatures of welded construction	all	—	—
Weld seams of important structural parts as determined by engine designer	—	As specified by engine designer	As specified by engine designer

Notes

1 This table does not cover turbochargers, superchargers and piping systems, such as fuel oil injection, starting air, etc. Turbochargers and superchargers are provided for in 4-2-2/3. For material tests of piping components, see 4-6-1/Table 1 and 4-6-1/Table 2.

2 Where there is evidence to doubt the soundness of any engine component, a nondestructive test by approved detecting methods may be required in any case.

3 Tests are to be at positions mutually agreed to by the Surveyor and manufacturer, where experience shows defects are most likely to occur.

4 For tie rods, the magnetic particle test is to be carried out at each threaded portion, which is twice the length of the thread.

5 Tests for these items are to be witnessed by the Surveyor. Alternatively, for engines certified under quality assurance assessment, tests by the manufacturer will suffice, see 4-2-1/3.3.4.

6 Ultrasonic test report to be certified by the engine manufacturer, see also 4-2-1/13.6.3(d).

7 Nondestructive testing is not applicable to the connecting rod bearing caps.

8 For engines < 375 kW, see 4-2-1/3.3.3.

TABLE 2
Test Pressures for Parts of Internal-combustion Engines

Engine Part	*Test Pressure* *(P = max. working pressure of engine part)*
Cylinder cover, cooling space. For forged cylinder covers, test methods other than pressure testing may be accepted, e.g., nondestructive examination and dimensional checks.	7 bar (7 kgf/cm^2, 100 psi)
Cylinder liner, over the whole length of cooling space	7 bar (7 kgf/cm^2, 100 psi)
Cylinder jacket, cooling space	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Exhaust valve, cooling space	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Piston crown, cooling space, where the cooling space is sealed by piston rod or by piston rod and skirt (test after assembly). For forged piston crowns test methods other than pressure testing may be used, e.g., nondestructive examination and dimensional checks.	7 bar (7 kgf/cm^2, 100 psi)
Fuel-injection system (pump body pressure side, valve and pipe)	1.5P or P + 300 bar (P + 306 kgf/cm^2, P + 4350 psi) which ever is less
Hydraulic System, high pressure piping for hydraulic drive of exhaust gas valve	1.5P
Scavenge-pump cylinder	4 bar (4 kgf/cm^2, 58 psi)
Turbocharger, cooling space (see 4-2-2/11.1.3)	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Exhaust pipe, cooling space	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Engine-driven air compressor, (cylinders, covers, intercoolers and aftercoolers) air side	1.5P
Engine-driven air compressor, (cylinders, covers, intercoolers and aftercoolers) water side	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Coolers, each side (charge air coolers need only be tested on the water side)	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Engine driven pumps (oil, water, fuel, bilge)	4 bar (4 kgf/cm^2, 58 psi) but not less than 1.5P
Independently driven pumps (oil, water, fuel) for engines with bores >300 mm (11.8 in.)	1.5P, for certification of pumps; see 4-6-1/7.3.

PART
4

CHAPTER 2 Prime Movers

SECTION 1 Appendix 1 – Plans and Data for Diesel Engines *(1 July 2005)*

For each type of engine that is required to be approved, the plans and data listed in the following table and as applicable to the type of engine are to be submitted for approval (A), approval of materials and weld procedure specifications (A*), or for information (R) by each engine manufacturer. After the approval of an engine type has been given by the Bureau for the first time, only those documents as listed in the table which have undergone substantive changes will have to be submitted again for consideration by the Bureau. In cases where both (R) and (A*) are shown, the first refers to cast components and the second to welded components.

No.	*A/R*	*Item*
1	R	Engine particulars as per attached sheet
2	R	Engine transverse cross-section
3	R	Engine longitudinal section
4	R/A*	Bedplate and crankcase, cast or welded with welding details and instructions (9)
5	R	Thrust bearing assembly (3)
6	R/A*	Thrust bearing bedplate, cast or welded with welding details and instructions (9)
7	R/A*	Frame/framebox, cast or welded with welding details and instructions (1,9)
8	R	Tie rod
9	R	Cylinder head, assembly
10	R	Cylinder liner
11	A	Crankshaft, details each cylinder No.
12	A	Crankshaft, assembly, each cylinder No.
13	A	Thrust shaft or intermediate shaft (if integral with engine)
14	A	Shaft coupling bolts
15	A	Counterweights (if not integral with crankshaft) including fastening
16	R	Connecting rod
17	R	Connecting rod, assembly
18	R	Crosshead, assembly (2)
19	R	Piston rod, assembly (2)
20	R	Piston, assembly
21	R	Camshaft drive, assembly
22	A	Material specifications of main parts with information on nondestructive testing and pressure tests (8)
23	R	Arrangement of foundation (for main engines only)
24	A	Schematic layout or other equivalent documents of starting air system on the engine (6)
25	A	Schematic layout or other equivalent documents of fuel oil system on the engine (6)
26	A	Schematic layout or other equivalent documents of lubricating oil system on the engine (6)
27	A	Schematic layout or other equivalent documents of cooling water system on the engine (6)
28	A	Schematic diagram of engine control and safety system on the engine (6)

No.	*A/R*	*Item*
29	R	Shielding and insulation of exhaust pipes, assembly
30	A	Shielding of high pressure fuel pipes, assembly (4)
31	A	Arrangement of crankcase explosion relief valve (5)
32	R	Operation and service manuals (7)
33	A	Schematic layout or other equivalent documents of hydraulic systems (for valve lift) on the engine
34	A	Type test program and type test report
35	A	High pressure parts for fuel oil injection system (10)

Notes:

1 Only for one cylinder

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

3 If integral with engine and not integrated in the bedplate

4 All engines

5 Only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m^3 or more

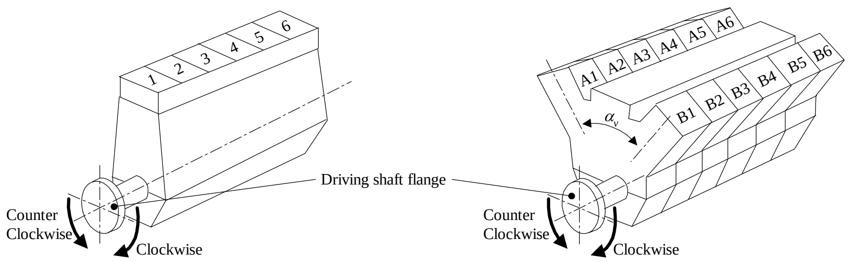
6 And the system, so far as supplied by the engine manufacturer. Where engines incorporate electronic control systems a failure mode and effects analysis (FMEA) is to be submitted to demonstrate that failure of an electronic control system will not result in the loss of essential services for the operation of the engine and that operation of the engine will not be lost or degraded beyond an acceptable performance criteria of the engine.

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

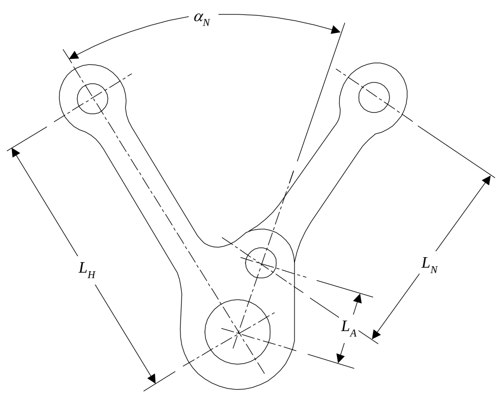
8 For comparison with requirements for material, NDT and pressure testing as applicable.

9 The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.

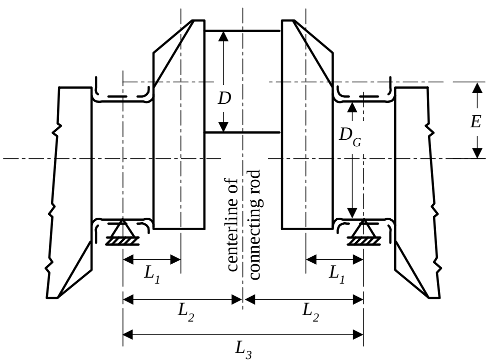
10 The documentation to contain specification of pressures, pipe dimensions and materials.

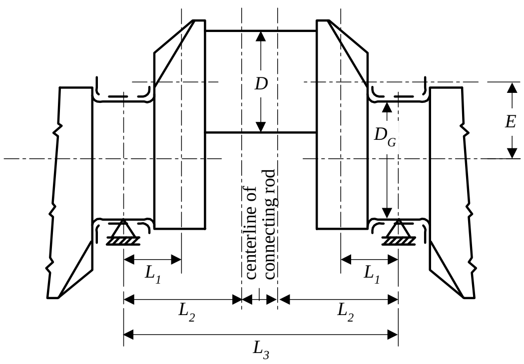
Data Sheet for Calculation of Crankshafts for Diesel Engines			
1	Engine Builder		
2	Engine Type Designation		
3	Stroke-Cycle ☐ 2 SCSA ☐ 4 SCSA		
4	Kind of engine ☐ In-line engine ☐ V-type engine with adjacent connecting rods ☐ V-type engine with articulated-type connecting rods ☐ V-type engine with forked/inner connecting rods ☐ Crosshead engine ☐ Trunk piston engine		
5	Combustion Method ☐ Direct injection ☐ Precombustion chamber ☐ Others:		
6	 FIGURE 1 Designation of the cylinders		
7	Sense of Rotation (corresponding to Item 6) ☐ Clockwise ☐ Counter clockwise		
8	Firing Order (corresponding to Item 6 and 7)		
9	Firing Intervals [deg] (corresponding to Item 8)		
10	Rated Power		kW
11	Rated Engine Speed		1/min
12	Mean Effective Pressure		bar
13	Mean Indicated Pressure		bar
14	Maximum Cylinder Pressure (Gauge)		bar

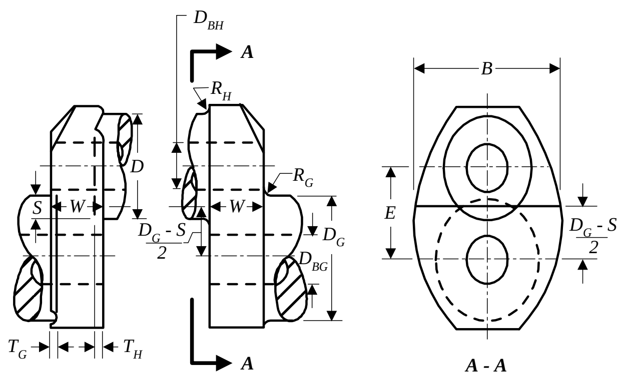
…Continued

15	Charge Air Pressure (Gauge) (before inlet valves or scavenge ports)		bar
16	Nominal Compression Ratio		–
17	Number of Cylinders		–
18	Diameter of Cylinders		mm
19	Length of Piston Stroke		mm
20	Length of Connecting Rod (between bearing centers)		mm
21	*(1 July 2005)* Oscillating Mass of One Cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg
22	Digitalized Gas Pressure Curve (Gauge) – presented at equidistant intervals [bar versus crank angle] – (intervals not more than 5° CA) ☐ given in the attachment		
Additional Data of V-type Engines			
23	V-Angle α_v(corresponding to Item 6)		deg
For the Cylinder Bank with Articulated-type Connecting Rod (Dimensions corresponding to Item 27)			
24	Maximum Cylinder Pressure (Gauge)		bar
25	Charge Air Pressure (Gauge) (before inlet valves or scavenge ports)		bar
26	Nominal Compression Ratio		
27	 Articulated-type connecting rod		
28	Distance to Link Point L_A		mm
29	Link Angle α_N		deg
30	Length of Connecting Rod L_N		mm
31	*(1 July 2005)* Oscillating Mass of One Cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg
32	Digitalized Gas Pressure Curve (Gauge) – presented at equidistant intervals [bar versus crank angle] – (intervals not more than 5° CA) ☐ given in the attachment		
For the Cylinder Bank with Inner Connecting Rod			
33	*(1 July 2005)* Oscillating Mass of One Cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg

…Continued

<table>
<tr><td colspan="3">Data of Crankshaft
(Dimensions corresponding to Item 39)
Note: For asymmetric cranks the dimensions are to be entered both for the left and right part of the crank throw.</td></tr>
<tr><td>34</td><td colspan="2">Drawing No.</td></tr>
<tr><td>35</td><td colspan="2">Kind of crankshaft (e.g., solid-forged crankshaft, semi-built crankshaft, etc.)</td></tr>
<tr><td rowspan="2">36</td><td colspan="2">Method of Manufacture (e.g., free form forged, cast steel, etc.):</td></tr>
<tr><td colspan="2">☐ Description of the forging process – if c.g.f forged or drop-forged – given in the attachment</td></tr>
<tr><td>37</td><td colspan="2">Heat treatment (e.g., tempered)</td></tr>
<tr><td rowspan="2">38</td><td colspan="2">Surface Treatment of Fillets, Journals and Pins (e.g., induction hardened, nitrided, rolled, etc.):</td></tr>
<tr><td colspan="2">☐ Full details given in the attachment</td></tr>
<tr><td>39</td><td colspan="2">

Crank throw for in-line engine


Crank throw for engine with 2 adjacent connecting rods


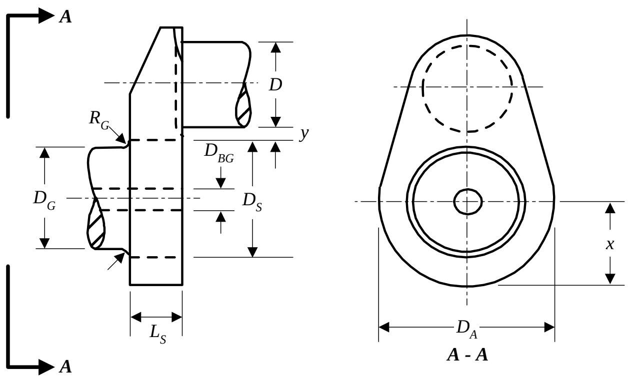
Crank dimensions necessary for the calculation of stress concentration factors</td></tr>
<tr><td>40</td><td>Crankpin Diameter D</td><td>mm</td></tr>
<tr><td>41</td><td>Diameter of Bore in Crankpin D_{BH}</td><td>mm</td></tr>
<tr><td>42</td><td>Fillet Radius of Crankpin R_H</td><td>mm</td></tr>
<tr><td>43</td><td>Recess of Crankpin T_H</td><td>mm</td></tr>
</table>

…continued

44	Journal Diameter D_G		mm
45	Diameter of Bore in Journal D_{BG}		mm
46	Fillet Radius of Journal R_G		mm
47	Recess of Journal T_G		mm
48	Web Thickness W		mm
49	Web Width B		mm
50	Bending Length L_1		mm
51	Bending Length L_2		mm
52	Bending Length L_3		mm
53	Oil Bore Design ☐ Safety margin against fatigue at the oil bores is not less than acceptable in the fillets		
54	Diameter of Oil Bore		mm
55	Smallest Edge Radius of Oil Bore		mm
56	Surface Roughness of Oil Bore Fillet		μm
57	Inclination of Oil Bore Axis related to Shaft Axis		deg

Additional Data for Shrink-Fits of Semi-Built Crankshafts (dimensions corresponding to Item 58)

58



Crank throw of semi-built crankshaft

59	Shrink Diameter D_S		mm
60	Length of Shrink-Fit L_S		mm
61	Outside Diameter of Web D_A or Twice the Minimum Distance x (the smaller value is to be entered)		mm
62	Amount of Shrink-Fit (upper and lower tolerances)		mm
			%
63	Maximum Torque (ascertained according to 4-2-1/5.9.3 with consideration of the mean torque)		N-m

…Continued

Data of Crankshaft Material Note: Minimum values of mechanical properties of material obtained from longitudinal test specimens			
64	Material Designation (according to DIN, AISI, etc.)		
65	Method of Material Melting Process (e.g., open-hearth furnace, electric furnace, etc.)		
66	Tensile Strength		N/mm^2
67	Yield Strength		N/mm^2
68	Reduction in Area at Break		%
69	Elongation A_5		%
70	Impact Energy – KV		J
71	Young's Modulus		N/mm^2
Additional Data for Journals of Semi-Built Crankshafts			
72	Material Designation (according to DIN, AISI, etc.)		
73	Tensile Strength		N/mm^2
74	Yield Strength		N/mm^2
Data according to calculation of torsional stresses			
75	Max. nominal alternating torsional stress (ascertained by means of a harmonic synthesis according to 4-2-1/5.9.3 and related to cross-sectional area of bored crank pin)		N/mm^2
76	Engine speed (at which the max. nominal alternating torsional stress occurs)		N/mm^2
77	Minimum engine speed (for which the harmonic synthesis was carried out)		N/mm^2
Data of stress concentration factors (S.C.F.) and/or fatigue strength furnished by reliable measurements Note: To be filled in only when data for stress concentration factors and/or fatigue are furnished by the engine manufacturer on the basis of measurements. Full supporting details are to be enclosed.			
78	S.C.F. for Bending in Crankpin Fillet α_B		–
79	S.C.F. for Torsion in Crankpin Fillet α_T		–
80	S.C.F. for Bending in Journal Fillet β_B		–
81	S.C.F. for Shearing in Journal Fillet β_Q		–
82	S.C.F. for Torsion in Journal Fillet β_T		–
83	Allowable Fatigue Strength of Crankshaft σ_{DW}		N/mm^2
Remarks			
Manufacturer’s Representative……………………..		Date ………………..	

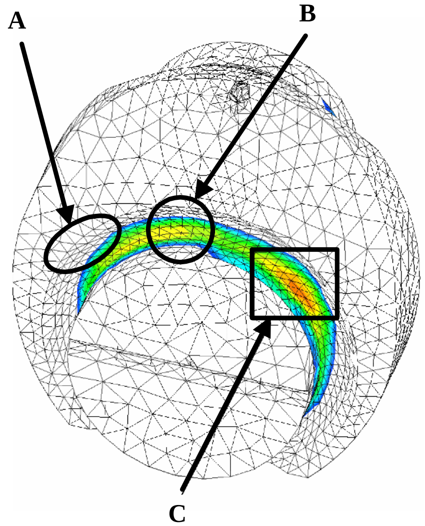
PART

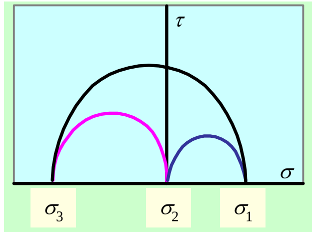
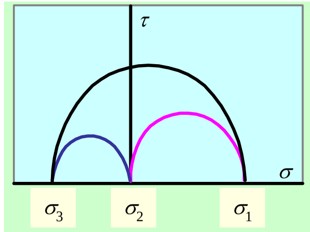
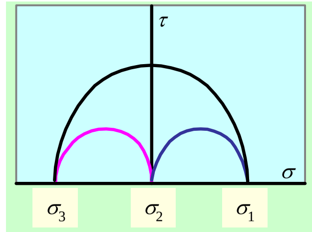
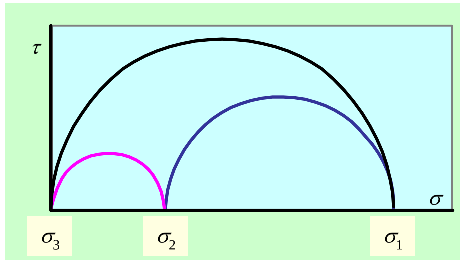
4

CHAPTER 2 Prime Movers

SECTION 1 Appendix 2 – Definition of Stress Concentration Factors in Crankshaft Fillets *(2007)*

Definition of Stress Concentration Factors in Crankshaft Fillets



	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.**	$\tau_{equiv} = \dfrac{\sigma_1 - \sigma_3}{2}$ **S.C.F.** $= \dfrac{\tau_{equiv}}{\tau_n}$ for α_T, β_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.**	$\sigma_{equiv} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2}$ **S.C.F.** $= \dfrac{\sigma_{equiv}}{\sigma_n}$ for $\alpha_B, \beta_B, \beta_Q$		

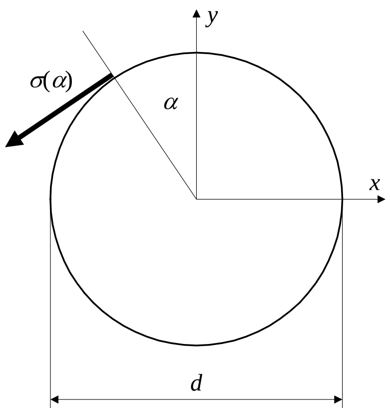
PART

4

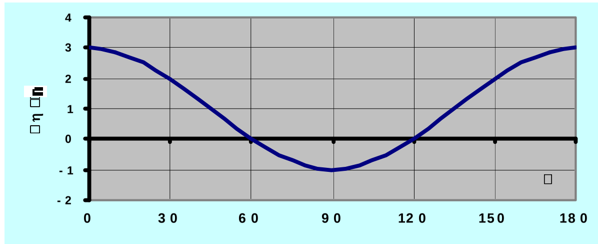
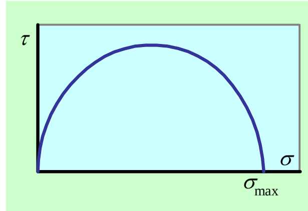
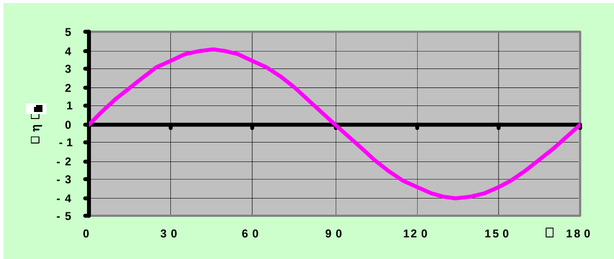
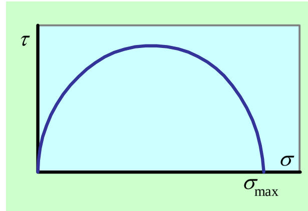
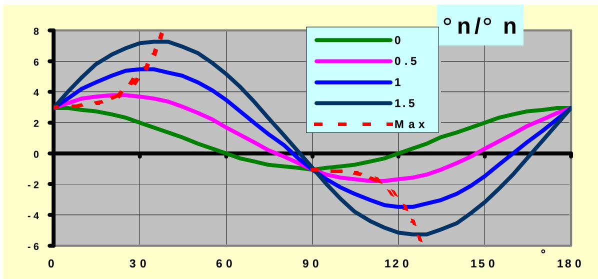
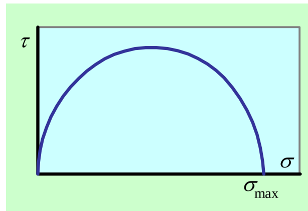
CHAPTER 2 Prime Movers

SECTION 1 Appendix 3 –Stress Concentration Factors and Stress Distribution at the Edge of Oil Drillings *(2007)*

Stress Concentration Factors and Stress Distribution at the Edge of Oil Drillings



d = hole diameter

Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's Circle diagram
Tension	$\begin{bmatrix} \sigma_n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha = \sigma_n \gamma_B / 3[1 + 2\cos(2\alpha)]$ 	 $\gamma_B = \sigma_{max}/\sigma_n$ for $\alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_B \tau_n \sin(2\alpha)$ 	 $\gamma_T = \sigma_{max}/\tau_n$ 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 $\alpha = \frac{1}{2}tg^{-1}\left(\frac{3\gamma_T\tau_n}{2\gamma_B\sigma_n}\right)$

PART

4

CHAPTER 2 Prime Movers

SECTION 1 Appendix 4 – Guidance for Spare Parts

1 General

While spare parts are not required for purposes of classification, the spare parts list below is provided as a guidance for vessels intended for unrestricted service. Depending on the design of the engine, spare parts other than those listed below, such as electronic control cards, should be considered.

3 Spare Parts for Main Propulsion Diesel Engines *(2008)*

Spare Parts	*Description*	*Number Recommended*
Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1 set
Main thrust block	Pads for one face of michell-type thrust block, or	1 set
	Complete white metal thrust shoe of solid-ring type, or	1 set
	Inner and outer race with rollers, where roller-thrust bearings are fitted	1
Cylinder liner	Cylinder liner, complete with joint ring and gaskets for one cylinder	1
Cylinder cover	Cylinder cover, complete with all valves, joint rings and gaskets.	1
	Cylinder-cover bolts and nuts, for one cylinder	1/2 set
Cylinder valves	Exhaust valves, complete with castings, seats springs and other fittings for one cylinder	2 sets
	Air inlet valves, complete with casings, seats, springs and other fittings for one cylinder	1 set
	Starting-air valve, complete with casing, seat, springs and other fittings	1
	Relief valve, complete	1
	Engines with one or two fuel valves per cylinder: fuel valves complete with springs and fittings for half the number of cylinders on one engine	As applicable
	Engines with three or more fuel valves per cylinder: two fuel valves complete per cylinder for half the number of cylinders on one engine and a sufficient number of valve parts, excluding the body, to form with those fitted on each cylinder for a complete engine set	As applicable
Cylinder cooling	Cooling pipes, fittings, and seals or their equivalent for one cylinder unit	1 set
Connecting rod bearings	Bottom-end bearings or shells of each size and type fitted, complete with shims, bolts and nuts, for one cylinder	1 set
	Top-end bearings or shells of each size and type fitted, complete with shims, bolts and nuts, for one cylinder	1 set

Spare Parts	*Description*	*Number Recommended*
Cross-head bearing lubrication	Engines fitted with a separate attached pump for lubrication of cross-head bearing, a complete pump with fittings for one cylinder unit	1
Pistons	Cross-head type: Piston of each type fitted, complete with piston rod, stuffing box, skirt, rings, studs and nuts	1
	Trunk-piston type: Piston of each type fitted, complete with skirt, rings, studs, nuts, gudgeon pin and connecting rod.	1
Piston rings	Piston rings of each type for one cylinder	1 set
Piston cooling	Telescopic cooling pipes, fittings and seals or their equivalent, for one cylinder unit	1 set
Gear and chain for camshaft drivers	Gear wheel drive: Wheels for the camshaft drive of one engine	1 set
	Chain Drive: Separate links with pins and rollers of each size and type fitted.	6
	Bearings bushes of each type fitted	1 set
Cylinder lubricators	Lubricator, complete, of the largest size, with its chain drive or gear wheels, or equivalent spare part kit	1
Fuel-injection pumps	Fuel pump complete or when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valve springs, etc.), or equivalent high pressure fuel pump	1
Fuel pump cams	Engines fitted with separate cams for ahead and astern drive for the fuel pump: One piece of cam nose with fittings for each ahead and astern drive for one cylinder unit	1 set
Fuel injection piping	High pressure double wall fuel pipe of each size and shape fitted, complete with couplings	1 set
Scavenge blowers (including turbo chargers)	Bearings, nozzle rings, gear wheels and complete attached lubricating pump or equivalent working parts of other types. [*Note:* Where an engine with one blower out of action can be maneuvered satisfactorily, spare parts, except for the necessary blanking arrangements, may be omitted.]	1 set
Scavenging system	Suction and delivery valves for one pump of each type fitted	1/2 set
Overspeed governors	A complete set of working parts for one governor	1 set
Gaskets and packings	Special gaskets and packings of each size and type fitted for cylinder covers and cylinder liners for one cylinder	1 set
Tools	Necessary special tools	1 set

5 Spare Parts for Auxiliary Diesel Engines *(2008)*

Spare Parts	*Description*	*Number Recommended*
Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1
Cylinder valves	Exhaust valves, complete with casings, seats, springs and other fittings in one cylinder	2 sets
	Air inlet valves, complete with casings, seats, springs and other fittings for one cylinder	1 set
	Starting-air valve, complete with casing, seat, springs and other fittings	1
	Relief valve, complete	1
	Fuel valves of each size and type fitted, complete with all fittings for one engine	1/2 set
Connecting-rod bearings	Bottom-end bearings or shells of each size and type fitted, complete with shims, bolts and nuts for one cylinder	1 set
	Trunk-piston type: gudgeon pin with bush for one cylinder	1 set
Piston rings	Piston rings, for one cylinder	1 set
Piston cooling	Telescopic cooling pipes and fittings or their equivalent for one cylinder unit	1 set
Fuel-injection pumps	Fuel pump complete or, when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valve springs, etc.), or equivalent high pressure fuel pump	1
Fuel-injection piping	High-pressure double wall fuel pipe of each size and shape fitted, complete with couplings	1
Gaskets and packings	Special gaskets and packings of each size and type fitted, for cylinder covers and cylinder liners for one cylinder	1 set

PART
4

CHAPTER 2 Prime Movers

SECTION 1 Appendix 5 – Type Testing Procedure for Crankcase Explosion Relief Valves *(2007)*

1 Scope

This Appendix specifies type tests and identifies standard test conditions using methane gas and air mixture to demonstrate that crankcase explosion relief valves intended to be fitted to engines and gear cases are satisfactory.

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

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

3 Recognized Standards *(1 July 2008)*

i) EN 12874:2001: Flame arresters – Performance requirements, test methods and limits for use.

ii) ISO/IEC EN 17025:2005: General requirements for the competence of testing and calibration laboratories.

iii) EN 1070:1998: Safety of Machinery – Terminology.

iv) VDI 3673: Part 1: Pressure Venting of Dust Explosions.

v) IMO MSC/Circular 677 – Revised Standards for the Design, Testing and Locating of Devices to Prevent the Passage of Flame into Cargo Tanks in Tankers

5 Purpose

The purpose of type testing crankcase explosion relief valves is:

i) To verify the effectiveness of the flame arrester.

ii) To verify that the valve closes after an explosion.

iii) To verify that the valve is gas/air tight after an explosion.

iv) To establish the level of overpressure protection provided by the valve.

7 Test Facilities *(1 July 2008)*

Test facilities carrying out type testing of crankcase explosion relief valves are to meet the following requirements in order to be acceptable to the Bureau:

i) The test facilities are to be accredited to a National or International Standard, e.g., ISO/IEC 17025 and are to be acceptable to the Bureau.

ii) The test facilities are to be equipped so that they can perform and record explosion testing in accordance with this procedure.

iii) The test facilities are to have equipment for controlling and measuring a methane gas in air concentration within a test vessel to an accuracy of ±0.1%.

iv) The test facilities are to be capable of effective point located ignition of methane gas in air mixture.

v) The pressure measuring equipment is to be capable of measuring the pressure in the test vessel in at least two positions: one at the valve and the other at the test vessel center. The measuring arrangements are to be capable of measuring and recording the pressure changes throughout an explosion test at a frequency recognizing the speed of events during an explosion. The result of each test is to be documented by video recording and by recording with a heat sensitive camera.

vi) The test vessel for explosion testing is to have documented dimensions. The dimensions are to be such that the vessel is not "pipe like" with the distance between dished ends being not more than 2.5 times its diameter. The internal volume of the test vessel is to include any standpipe arrangements.

vii) The test vessel for explosion testing is to be provided with a flange, located centrally at one end perpendicular to the vessel longitudinal axis, for mounting the explosion relief valve. The test vessel is to be arranged in an orientation consistent with how the valve will be installed in service, i.e., in the vertical plane or the horizontal plane.

viii) A circular plate is to be provided for fitting between the pressure vessel flange and valve tested with the following dimensions:

- Outside diameter of 2 times the outer diameter of the valve top cover
- Internal bore having the same internal diameter as the valve to be tested.

ix) The test vessel is to have connections for measuring the methane in an air mixture at the top and bottom.

x) The test vessel is to be provided with a means of fitting an ignition source at a position specified in 4-2-1A5/9.

xi) The test vessel volume is to be, as far as practicable, related to the size and capability of the relief valve to be tested. In general, the volume is to correspond to the requirement in 4-2-1/7.1.2 for the free area of explosion relief valve to be not less than 115 cm^2/m^3 (0.505 in^2/ft^3) of crankcase gross volume.

Notes:

1 This means that the testing of a valve having 1150 cm^2 (178.25 in^2) of free area, would require a test vessel with a volume of 10 m^3 (353.15 ft^3).

2 Where the free area of relief valves is greater than 115 cm^2/m^3 (0.505 in^2/ft^3) of the crankcase gross volume, the volume of the test vessel is to be consistent with the design ratio.

3 In no case is the volume of the test vessel to vary by more than +15% to -15% from the design cm^2/m^3 volume ratio.

9 Explosion Test Process *(1 July 2008)*

All explosion tests to verify the functionality of crankcase explosion relief valves are to be carried out using an air and methane mixture with a volumetric methane concentration of 9.5% ± 0.5%. The pressure in the test vessel is to be not less than atmospheric and not exceed the opening pressure of the relief valve.

The concentration of methane in the test vessel is to be measured at the top and bottom of the vessel and these concentrations are not to differ by more than 0.5%.

The ignition of the methane and air mixture is to be made at the centerline of the test vessel at a position approximately one third of the height or length of the test vessel opposite to where the valve is mounted.

The ignition is to be made using a maximum 100 Joule (0.0947 BTU) explosive charge.

11 Valves to be Tested

i) The valves used for type testing [including testing specified in 4-2-1A5/11iii)] are to be selected from the manufacturer's normal production line for such valves.

ii) For approval of a specific valve size, three valves are to be tested in accordance with 4-2-1A5/11iii) and 4-2-1A5/13. For a series of valves, 4-2-1A5/17 refers.

iii) The valves selected for type testing are to have been previously tested at the manufacturer's works to demonstrate that the opening pressure is in accordance with the specification within a tolerance of ±20% and that the valve is airtight at a pressure below the opening pressure for at least 30 seconds.

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

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

13 Method

13.1 General Requirements *(1 July 2008)*

The following requirements are to be satisfied during explosion testing:

i) The explosion testing is to be witnessed by the Surveyor.

ii) Where valves are to be installed on an engine or gear case with shielding arrangements to deflect the emission of explosion combustion products, the valves are to be tested with the shielding arrangements fitted.

iii) Successive explosion testing to establish a valve's functionality is to be carried out as quickly as possible during stable weather conditions.

iv) The pressure rise and decay during all explosion testing is to be recorded.

v) The external condition of the valves is to be monitored during each test for indication of any flame release by video and heat sensitive camera.

13.3 Stages of Testing

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

13.3.1 Stage 1

Two explosion tests are to be carried out in the test vessel with the circular plate described in 4-2-1A5/7viii) fitted and the opening in the plate covered by a 0.05 mm (0.002 inch) thick polythene film.

Note: These tests establish a reference pressure level for determination of the capability of a relief valve in terms of pressure rise in the test vessel [see 4-2-1A5/15vi)].

13.3.2 Stage 2

13.3.2(a) Two explosion tests are to be carried out on three different valves of the same size. Each valve is to be mounted in the orientation for which approval is sought, i.e., in the vertical or horizontal position with the circular plate described in 4-2-1A5/7viii) located between the valve and pressure vessel mounting flange.

13.3.2(b) The first of the two tests on each valve is to be carried out with a 0.05 mm (0.002 inch) thick polythene bag having a minimum diameter of three times the diameter of the circular plate and volume not less than 30% of the test vessel enclosing the valve and circular plate. Before carrying out the explosion test the polythene bag is to be empty of air. The polythene bag is required to provide a readily visible means of assessing whether there is flame transmission through the relief valve following an explosion consistent with the requirements of the standards identified in 4-2-1A5/3.

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

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

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

13.3.3 Stage 3

Carry out two further explosion tests as described in Stage 1, (see 4-2-1A5/13.3.1). These further tests are required to provide an average base line value for assessment of pressure rise recognizing that the test vessel ambient conditions may have changed during the testing of the explosion relief valves in Stage 2, (see 4-2-1A5/13.3.2).

15 Assessment and Records *(1 July 2008)*

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

i) The valves to be tested are to have been design approved.

ii) The designation, dimensions and characteristics of the valves to be tested are to be recorded. This is to include the valve free area of the valve and of the flame arrester and the amount of valve lift at 0.2 bar (0.2 kgf/cm^2, 2.85 lbf/in^2).

iii) The test vessel volume is to be determined and recorded.

iv) For acceptance of the functioning of the flame arrester there must not be any indication of flame or combustion outside the valve during an explosion test. This is to be confirmed by the test laboratory taking into account measurements from the heat sensitive camera.

v) The pressure rise and decay during an explosion is to be recorded with indication of the pressure variation showing the maximum overpressure and steady under pressure in the test vessel during testing. The pressure variation is to be recorded at two points in the pressure vessel.

vi) The effect of an explosion relief valve in terms of pressure rise following an explosion is ascertained from maximum pressures recorded at the center of the test vessel during the three stages. The pressure rise within the test vessel due to the installation of a relief valve is the difference between average pressure of the four explosions from Stages 1 and 3 (see 4-2-1A5/13.3.1 and 4-2-1A5/13.3.3) and the average of the first tests on the three valves in Stage 2, (see 4-2-1A5/13.3.2). The pressure rise is not to exceed the limit specified by the manufacturer.

vii) The valve tightness is to be ascertained by verifying from records at the time of testing that an underpressure of at least 0.3 bar (3.06 kgf/cm^2, 43.5 psi) is held by the test vessel for at least 10 seconds following an explosion. The test is to verify that the valve has effectively closed and is reasonably gas-tight following dynamic operation during an explosion.

viii) After each explosion test in Stage 2, (see 4-2-1A5/13.3.2), the external condition of the flame arrester is to be examined for signs of serious damage and/or deformation that may affect the operation of the valve.

ix) After completing the explosion tests, the valves are to be dismantled and the condition of all components ascertained and documented. In particular, any indication of valve sticking or uneven opening that may affect operation of the valve is to be noted. Photographic records of the valve condition are to be taken and included in the report.

17 Design Series Qualification

17.1 General *(1 July 2008)*

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

17.3 Flame Arrester *(1 July 2008)*

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

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

ii) $$\frac{A_1}{A_2} = \frac{S_1}{S_2}$$

where

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

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

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

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

17.5 Valves of Larger Sizes than Have Been Satisfactorily Tested

The qualification of explosion relief valves of larger sizes than that which has been previously satisfactorily tested in accordance with 4-2-1A5/13 and 4-2-1A5/15 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

17.5.1 *(1 July 2008)*

The free area of a larger valve does not exceed three times +5% that of the valve that has been satisfactorily tested.

17.5.2 *(1 July 2008)*

One valve of the largest size, subject to 4-2-1A5/17.5.1, requiring qualification is subject to satisfactory testing required by 4-2-1A5/11iii) and 4-2-1A5/13.3.2 except that a single valve will be accepted in 4-2-1A5/13.3.2(a) and the volume of the test vessel is not to be less than one third of the volume required by 4-2-1A5/7xi).

17.5.3

The assessment and records are to be in accordance with 4-2-1A5/15 noting that 4-2-1A5/15vi) will only be applicable to Stage 2 for a single valve.

17.7 Valves of Smaller Sizes than Have Been Satisfactorily Tested

The qualification of explosion relief valves of smaller sizes than that which has been previously satisfactorily tested in accordance with 4-2-1A5/13 and 4-2-1A5/15 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

17.7.1

The free area of a smaller valve is not less than one-third of the valve that has been satisfactorily tested.

17.7.2 *(1 July 2008)*

One valve of the smallest size, subject to 4-2-1A5/17.7.1, requiring qualification is subject to satisfactory testing required by 4-2-1A5/11iii) and 4-2-1A5/13.3.2 except that a single valve will be accepted in 4-2-1A5/13.3.2(a) and the volume of the test vessel is not to be more than the volume required by 4-2-1A5/7xi).

17.7.3

The assessment and records are to be in accordance with 4-2-1A5/15 noting that 4-2-1A5/15vi) will only be applicable to Stage 2 for a single valve.

19 Reporting

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

i) Test specification.

ii) Details of test pressure vessel and valves tested.

iii) The orientation in which the valve was tested, (vertical or horizontal position).

iv) Methane in air concentration for each test.

v) Ignition source

vi) Pressure curves for each test.

vii) Video recordings of each valve test.

viii) The assessment and records stated in 4-2-1A5/15.

21 Acceptance

Acceptance of an explosion relief valve will be based on design approved plans and particulars and on the test facility's report of the results of the type testing.

PART

4

CHAPTER 2 Prime Movers

SECTION 1 Appendix 6 – Type Testing Procedure for Crankcase Oil Mist Detection and Alarm Equipment *(2007)*

1 Scope

This Appendix specifies the tests required to demonstrate that crankcase oil mist detection and alarm equipment intended to be fitted to diesel engines demonstrate compliance with a defined standard for type testing.

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

3 Recognized Environmental Test Standards

Equipment tests as required in 4-2-1/9 are to in accordance with 4-9-7/Table 9

5 Purpose

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

i) To verify the functionality of the system.

ii) To verify the effectiveness of the oil mist detectors.

iii) To verify the accuracy of oil mist detectors.

iv) To verify the alarm set points.

v) To verify time delays between oil mist leaving the source and alarm activation.

vi) To verify functional failure detection.

vii) To verify the influence of optical obscuration on detection.

7 Test Facilities

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

i) A full range of provisions for carrying out the environmental and functionality tests required by this procedure are to be available and acceptable to the Bureau.

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

9 Equipment Testing

The range of tests is to include the following (see also 4-9-7/13.1 – Prototype Environmental Testing):

9.1 For the Alarm/Monitoring Panel

i) Functional tests described in 4-2-1A6/11.

ii) Electrical power supply failure test.

iii) Power supply variation test.

iv) Dry heat test.

v) Damp heat test.

vi) Vibration test.

vii) EMC test.

viii) Insulation resistance test.

ix) High voltage test.

x) Static and dynamic inclinations, if moving parts are contained.

9.3 For the Detectors

i) Functional tests described in 4-2-1A6/11.

ii) Electrical power supply failure test.

iii) Power supply variation test.

iv) Dry heat test.

v) Damp heat test.

vi*)* Vibration test.

vii) EMC test where susceptible.

viii) Insulation resistance test.

ix) High voltage test.

x) Static and dynamic inclinations.

11 Functional Tests

i) All tests to verify the functionality of crankcase oil mist detection and alarm equipment are to be carried out in accordance with 4-2-1A6/11ii) through 4-2-1A6/11vi) with an oil mist concentration in air, known in terms of mg/l to an accuracy of ±10%.

ii) The concentration of oil mist in the test chamber is to be measured in the top and bottom of the chamber and these concentrations are not to differ by more than 10%. See also 4-2-1A6/15i)a).

iii) The oil mist monitoring arrangements are to be capable of detecting oil mist in air concentrations of between 0 and 10% of the lower explosive limit (LEL) or between 0 and a percentage corresponding to a level not less than twice the maximum oil mist concentration alarm set point. Note: The LEL corresponds to an oil mist concentration of approximately 50 mg/l (~4.1% weight of oil-in air mixture).

iv) The alarm set point for oil mist concentration in air is to provide an alarm at a maximum level corresponding to not more than 5% of the LEL or approximately 2.5 mg/l.

v) Where alarm set points can be altered, the means of adjustment and indication of set points are to be verified against the equipment manufacturer’s instructions.

vi) Where oil mist is drawn into a detector/monitor via piping arrangements, the time delay between the sample leaving the crankcase and operation of the alarm is to be determined for the longest and shortest lengths of pipes recommended by the manufacturer. The pipe arrangements are to be in accordance with the manufacturer's instructions/recommendations.

vii) It is to be demonstrated that openings in detector equipment that is in contact with the crankcase atmosphere and may be exposed to oil splash and spray from engine lubricating oil do not occlude or become blocked under continuous oil splash and spray conditions. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by the Bureau.

viii) It is to be demonstrated that exposed to water vapor from the crankcase atmosphere, which may affect the sensitivity of the detector equipment, will not affect the functional operation of the detector equipment. Where exposure to water vapor and/or water condensation has been identified as a possible source of equipment malfunctioning, testing is to demonstrate that any mitigating arrangements, such as heating, are effective. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by the Bureau.

Note: This testing is in addition to that required by 4-2-1A6/9.3v) and is concerned with the effects of condensation caused by the detection equipment being at a lower temperature than the crankcase atmosphere.

13 Detectors and Alarm Equipment to be Tested

The detectors and alarm equipment selected for the type testing are to be selected by the Surveyor from the manufacturer's usual production line.

Two detectors are to be tested. One is to be tested in the clean condition and the other in a condition representing the maximum level of lens obscuration specified by the manufacturer.

15 Method

The following requirements are to be satisfied during type testing:

i) Oil mist generation is to satisfy 4-2-1A6/15i)a) to 4-2-1A6/15i)e.)

a) Oil mist is to be generated with suitable equipment using an SAE 80 monograde mineral oil or equivalent and supplied to a test chamber having a volume of not less than 1 m^3. The oil mist produced is to have a maximum droplet size of 5 μm.

Note: The oil droplet size is to be checked using the sedimentation method.

b) The oil mist concentrations used are to be ascertained by the gravimetric deterministic method or equivalent.

Note: For this test, the gravimetric deterministic method is a process where the difference in weight of a 0.8 μm pore size membrane filter is ascertained from weighing the filter before and after drawing 1 liter of oil mist through the filter from the oil mist test chamber. The oil mist chamber is to be fitted with a recirculating fan.

c) Samples of oil mist are to be taken at regular intervals and the results plotted against the oil mist detector output. The oil mist detector is to be located adjacent to where the oil mist samples are drawn off.

d) The results of a gravimetric analysis are considered invalid and are to be rejected if the resultant calibration curve has an increasing gradient with respect to the oil mist detection reading. This situation occurs when insufficient time has been allowed for the oil mist to become homogeneous. Single results that are more than 10% below the calibration curve are to be rejected. This situation occurs when the integrity of the filter unit has been compromised and not all of the oil is collected on the filter paper.

e) The filters require to be weighed to a precision of 0.1 mg and the volume of air/oil mist sampled to 10 ml.

ii) The testing is to be witnessed by the Surveyor.

iii) Oil mist detection equipment is to be tested in the orientation (vertical, horizontal or inclined) in which it is intended to be installed on an engine or gear case as specified by the equipment manufacturer.

iv) Type testing is to be carried out for each type of oil mist detection and alarm equipment for which a manufacturer seeks approval. Where sensitivity levels can be adjusted, testing is to be carried out at the extreme and mid-point level settings.

17 Assessment

Assessment of oil mist detection equipment after testing is to address the following:

i) The equipment to be tested is to have been design approved.

ii) Details of the detection equipment to be tested are to be recorded, such as name of manufacturer, type designation, oil mist concentration assessment capability and alarm settings.

iii) After completing the tests, the detection equipment is to be examined and the condition of all components ascertained and documented. Photographic records of the monitoring devices condition are to be taken and included in the report.

19 Design Series Qualification

The approval of one type of detection equipment may be used to qualify other devices having identical construction details. Proposals are to be submitted for consideration.

21 Reporting

The test facility is to provide a full report which includes the following information and documents:

i) Test specification.

ii) Details of equipment tested.

iii) Results of tests.

23 Acceptance

Acceptance of crankcase oil mist detection equipment will be based on design approved plans and particulars and on the test facility's report of the results of the type testing.

The following information is to be submitted for acceptance of oil mist detection equipment and alarm arrangements:

i) Description of oil mist detection equipment and system including alarms.

ii) Copy of the test facility's report identified in 4-2-1A6/21.

iii) Schematic layout of engine oil mist detection arrangements showing location of detectors/ sensors and piping arrangements and dimensions.

iv) Maintenance and test manual which is to include the following information:

- Intended use of equipment and its operation.
- Functionality tests to demonstrate that the equipment is operational and that any faults can be identified and corrective actions notified.
- Maintenance routines and spare parts recommendations.
- Limit setting and instructions for safe limit levels.
- Where necessary, details of configurations in which the equipment is intended to be used and in which it is not to be used.

PART
4

CHAPTER 2 Prime Movers

SECTION 2 Turbochargers

1 General

1.1 Application

Turbochargers for diesel engines rated 100 kW (135 hp) and over, intended for propulsion and for auxiliary services essential for propulsion, maneuvering and safety of the vessel [see 4-1-1/1.3.2(a)], are to be designed, constructed, tested, certified and installed in accordance with the requirements of this section.

Turbochargers for diesel engines rated less than 100 kW (135 hp) are to be designed, constructed and equipped in accordance with good commercial and marine practice. Acceptance of such turbochargers will be based on the manufacturer's affidavit, verification of turbocharger nameplate data and subject to a satisfactory performance test after installation, conducted in the presence of the Surveyor.

1.3 Definitions

1.3.1 Turbocharger

Where the term *Turbocharger* is used in this section, it refers also to superchargers, turboblowers, scavenge blowers or other similar equipment designed to charge the diesel engine cylinders with air at a higher pressure and hence higher density than air at atmospheric pressure.

1.3.2 Maximum Operating Speed

The *Maximum Operating Speed* is the maximum permissible speed for which the turbocharger is designed to run continuously at the maximum permissible operating temperature. This speed is to be used for making strength calculations.

1.5 Plans and Particulars to be Submitted

1.5.1 Turbocharger Construction

Turbochargers for engines with cylinder bores ≤ 300 mm (11.8 in.):

- Sectional assembly
- Parts list

Turbochargers for engines with cylinder bores > 300 mm (11.8 in.):

- Sectional assembly
- Parts list
- Casings
- Turbine rotors
- Compressor rotors
- Compressor and turbine discs
- Blading
- Shafts
- Bearing arrangements

1.5.2 Turbocharger System and Appurtenances

Lubrication system

Cooling system

1.5.3 Data

Turbochargers for engines with cylinder bores ≤ 300 mm (11.8 in.):

- Operating speed and temperatures
- Speed and temperature limitations
- Balancing data
- Type test data

Turbochargers for engines with cylinder bores > 300 mm (11.8 in.):

- Operating speed and temperatures
- Speed and temperature limitations
- Mass and velocity of rotating elements
- Balancing data
- Type test data

1.5.4 Materials

Turbochargers for engines with cylinder bores ≤ 300 mm (11.8 in.):

- Material specifications

Turbochargers for engines with cylinder bores > 300 mm (11.8 in.):

- Material specifications, including density, Poisson's ratio, range of chemical composition, room-temperature physical properties and, where material is subject to temperatures exceeding 427°C (800°F), the high-temperature strength characteristics as well as creep rate and rupture strength for the design service life.

1.5.5 Calculation and Analyses

Turbochargers for engines with cylinder bores > 300 mm (11.8 in.):

- Design basis and design analyses for turbine and compressor rotors and blading including calculations and test results to substantiate the suitability and strength of components for the intended service.

3 Materials

3.1 Material Specifications and Purchase Orders

Materials entered into the construction of turbochargers are to conform to specifications approved in connection with the design in each case. Copies of material specifications and purchase orders are to be submitted to the Surveyor for information and verification.

3.3 Engines with Cylinder Bore ≤ 300 mm (11.8 in.)

Materials for turbochargers intended for engines with cylinder bore ≤ 300 mm need not be verified by a Surveyor. The turbocharger manufacturer is to assure itself of the quality of the materials.

3.5 Engines with Cylinder Bore > 300 mm (11.8 in.) *(2003)*

The materials are to meet specifications in Part 2, Chapter 3 or that approved in connection with the design. Except as noted in 4-2-2/3.7, materials for turbochargers intended for engines with cylinder bore > 300 mm (11.8 in.), as specified below, are to be tested in the presence of and inspected by the Surveyor.

i) Forgings: compressor and turbine rotors and shafts.

ii) Blade material.

3.7 Alternative Material Test Requirements

3.7.1 Alternative Specifications

Material manufactured to specifications other than those given in Part 2, Chapter 3 may be accepted, provided that such specifications are approved in connection with the design and that they are verified or tested by a Surveyor as complying with the specifications.

3.7.2 Steel-bar Stock

Hot-rolled steel bars up to 305 mm (12 in.) in diameter may be used when approved for use in place of any of the forgings as per 4-2-2/3.5i) above, under the conditions outlined in Section 2-3-8.

3.7.3 Certification Under Quality Assurance Assessment

For turbochargers certified under quality assurance assessment as provided for in 4-2-2/11.3.2(b), material tests and inspections required by 4-2-1/3.1 need not be witnessed by the Surveyor. Such tests are to be conducted by the turbocharger manufacturer whose certified material test reports will be accepted instead.

5 Design

5.1 Engines with Cylinder Bores ≤ 300 mm (11.8 in.)

Turbochargers intended for engines with cylinder bore ≤ 300 mm (11.8 in.) will be accepted on the basis of manufacturer's type test data. The test data are to be in accordance with 4-2-2/5.3.3. Considerations will also be given to submittal of design criteria and engineering analyses. The manufacturer is to specify and guarantee the limits of speed and temperature.

5.3 Engines with Cylinder Bores > 300 mm (11.8 in.)

5.3.1 Casings

5.3.1(a) Castings. Castings for turbochargers intended for engines with cylinder bore > 300 mm (11.8 in.) are to be of a specification suitable for stresses and temperatures to which they are designed to be exposed. Cast iron may only be considered for operating temperatures not exceeding 232°C (450°F). Ductile cast iron designed for high temperature service is acceptable subject to review of mechanical and metallurgical properties at design temperatures. Cast steel may be considered for operating temperatures not exceeding 427°C (800°F). All castings are to be properly heat-treated to remove internal stresses.

5.3.1(b) Seals and drains. Casings are to be provided with suitable seals. Drains are to be fitted in places where water or oil may collect.

5.3.2 Shafts, Rotors and Blades

Rotors, bearings, discs, impellers and blades are to be designed in accordance with sound engineering principles. Design criteria along with engineering analyses substantiating the suitability of the design for the rated power and speed are to be submitted for review.

5.3.3 Type Test Data

The manufacturer is to submit type test data in support of the design. The type test is preferably to be witnessed and certified by a Surveyor. The type test data are to contain at least the test schedule, measurements taken during the tests and test results. Normally, the type test is to have at least one hour of hot running test at maximum permissible speed and temperature, verification of performance, tests specified in 4-2-2/11.1.5 and opening for examination after the test. Also, burst tests and containment tests are to be performed, but the submission of appropriate stress calculations may be substituted in lieu of these tests.

7 Piping Systems for Turbochargers

The lubricating oil and cooling water piping systems of turbochargers are to be in accordance with the provisions of 4-6-5/5 and 4-6-5/7, respectively.

9 Installation of Turbochargers

9.1 Air Inlet

The air inlet of the turbocharger is to be fitted with a filter to minimize the entrance of harmful foreign material or water.

9.3 Hot Surfaces

Hot surfaces likely to come into contact with the crew are to be water-jacketed or effectively insulated. Where the temperature of hot surfaces is likely to exceed 220°C (428°F) and where any leakage, under pressure or otherwise, of fuel oil, lubricating oil or other flammable liquid is likely to come into contact with such surfaces, they are to be suitably insulated with non-combustible materials that are impervious to such liquid. Insulation material not impervious to oil is to be encased in sheet metal or an equivalent impervious sheath.

9.5 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.

11 Testing, Inspection and Certification of Turbochargers

11.1 Shop Inspection and Tests

The following shop inspection and tests are to be witnessed by a Surveyor for turbochargers of engines having cylinder bores greater than 300 mm (11.8 in.).

11.1.1 Material Tests *(2002)*

Materials entered into the construction of turbines are to be tested in the presence of a Surveyor in accordance with the provisions of 4-2-2/3. This does not apply to independently driven auxiliary blowers that are not needed during continuous operation of the engine.

11.1.2 Welded Fabrication

All welded fabrication is to be conducted with qualified welding procedures, by qualified welders, and with welding consumables acceptable to the Surveyors. See Section 2-4-2.

11.1.3 Hydrostatic Tests

The cooling spaces of each gas inlet and outlet casing are to be hydrostatically tested to 1.5 times the working pressure but not to be less than 4 bar.

11.1.4 Dynamic Balancing

Rotors are to be dynamically balanced at a speed equal to the natural period of the balancing machine and rotor combined.

11.1.5 Shop Trial *(2003)*

Upon completion of fabrication and assembly, each turbocharger is to be subjected to a shop trial, either on a test bed or on a test engine, in accordance with the manufacturer's test schedule, which is to be submitted for review before the trial. During the trial, the following tests are to be conducted:

i) Impeller and inducer wheels are to be overspeed tested for 3 minutes either:

- On a test bed at 20% above the maximum operating speed at ambient temperature, or
- On a test engine at 10% above the maximum operating speed at operating temperature.

ii) A mechanical running test for at least 20 minutes at maximum operating speed and operating temperature, or a test run on the engine for which the turbocharger is intended for 20 minutes at 110% of the engine's rated output.

11.3 Certification of Turbochargers

11.3.1 General

Each turbocharger required to be certified by 4-2-2/1.1 is:

i) To have its design approved by the Bureau; for which purpose, plans and data as required by 4-2-2/1.5 are to be submitted to the Bureau for approval, and a unit of the same type is to be satisfactorily type tested (see 4-2-2/5.3.3);

ii) To be surveyed during its construction for compliance with the design approved, along with, but not limited to, material tests, hydrostatic tests, dynamic balancing, performance tests, etc., as indicated in 4-2-2/11.1, all to be carried out to the satisfaction of the Surveyor.

11.3.2 Approval Under the Type Approval Program *(2003)*

11.3.2(a) Product design assessment. Upon application by the manufacturer, each model of a type of turbocharger is to be design assessed as described in 1-1-A3/5.1. For this purpose, each design of a turbocharger type is to be approved in accordance with 4-2-2/11.3.1i). The type test specified in 4-2-2/5.3.3, however, is to be conducted in accordance with an approved test schedule and is to be witnessed by a Surveyor. Turbochargers so approved may be applied to the Bureau for listing on the ABS website as Products Design Assessed. Once listed, and subject to renewal and updating of the certificate as required by 1-1-A3/5.7, turbocharger particulars will not be required to be submitted to the Bureau each time the turbocharger is proposed for use on board a vessel.

11.3.2(b) Mass produced turbochargers. A manufacturer of mass-produced turbochargers, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance approval described in 1-1-A3/5.5 (PQA).

Upon satisfactory assessment under 1-1-A3/5.5 (PQA), turbochargers produced in those facilities will not require a Surveyor's attendance at the tests and inspections indicated in 4-2-2/11.3.1ii). Such tests and inspections are to be carried out by the manufacturer whose quality control documents will be accepted. Certification of each turbocharger will be based on verification of approval of the design and on continued effectiveness of the quality assurance system. See 1-1-A3/5.7.1(a).

11.3.2(c) Non-mass Produced Turbochargers. A manufacturer of non-mass produced turbochargers, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.3.1(a) (AQS) or 1-1-A3/5.3.1(b) (RQS). Certification to 1-1-A3/5.5 (PQA) may also be considered in accordance with 4-1-1/Table 1.

11.3.2(d) Type Approval Program. Turbocharger types which have their designs approved in accordance with 4-2-2/11.3.2(a) and the quality assurance system of their manufacturing facilities approved in accordance with 4-2-2/11.3.2(b) or 4-2-2/11.3.2(c) will be deemed Type Approved and will be eligible for listing on the ABS website as Type Approved Product.

11.5 Engine and Shipboard Trials

Before final acceptance, each turbocharger, after installation on the engine, is to be operated in the presence of the Surveyor to demonstrate its ability to function satisfactorily under operating conditions and its freedom from harmful vibrations at speeds within the operating range. The test schedules are to be as indicated in 4-2-1/13.9 for engine shop test and in 4-2-1/15 for shipboard trial.

13 Spare Parts

While spare parts are not required for purposes of classification, the spare parts listed in Appendix 4-2-1A4 are provided as a guidance for vessels intended for unrestricted service. The maintenance of spare parts aboard each vessel is the responsibility of the owner.

PART

4

CHAPTER 2 Prime Movers

SECTION 3 Gas Turbines

1 General

1.1 Application

Gas turbines having a rated power of 100 kW (135 hp) and over, intended for propulsion and for auxiliary services essential for propulsion, maneuvering and safety (see 4-1-1/1.3) of the vessel, are to be designed, constructed, tested, certified and installed in accordance with the requirements of this section.

Gas turbines having a rated power of less than 100 kW (135 hp) are not required to comply with the provision of this section but are to be designed, constructed and equipped in accordance with good commercial and marine practice. Acceptance of the gas turbines will be based on the manufacturer's affidavit, verification of gas turbines nameplate data and subject to a satisfactory performance test after installation conducted in the presence of the Surveyor.

Gas turbines having a rated power of 100 kW (135 hp) and over, intended for services considered not essential for propulsion, maneuvering and safety, are not required to be designed, constructed and certified by the Bureau in accordance with the provisions of this section. They are to comply with safety features, such as overspeed protection, etc., as provided in 4-2-3/7 hereunder, as applicable, and are subject to a satisfactory performance test after installation, conducted in the presence of the Surveyor.

Provisions for piping systems of gas turbines, in particular, fuel oil, lubricating oil, cooling water and exhaust gas systems, are addressed in Section 4-6-5.

1.3 Definitions

1.3.1 Rated Power

The *Rated Power* is the maximum power output at which the turbine is designed to run continuously at its rated speed. Gas turbine power is to be that developed at the lowest expected inlet air temperature, but in no case is this design inlet air temperature to exceed 15°C (59°F).

1.5 Plans and Particulars to be Submitted

1.5.1 Gas Turbine Construction *(2007)*

Sectional assembly

Casings

Foundation and fastening

Combustion chambers

Gasifiers

Regenerators or recuperators

Turbine rotors

Compressor rotors

Compressor and turbine discs

Blading

Shafts

Bearing arrangements

Thrust bearing

1.5.2 Gas Turbine Systems and Appurtenances *(2007)*

Couplings

Clutches

Starting arrangements

Fuel oil system

Shielding of fuel oil service piping

Lubricating oil system

Air-intake system

Exhaust system

Shielding and insulation of exhaust pipes, assembly

Governor arrangements

Safety systems and devices and associated failure modes and effects analysis

Control oil system

Bleed/cooling/seal air system

Cooling system

Electrical and instrumentation schematics

Accessory drives

Water wash

Enclosure arrangement

Fire protection (gas turbine manufacturer supplied)

1.5.3 Data *(2007)*

Rated power, maximum intermittent power for 1 hour operating time, maximum power (peak)

Rated engine speed, including gas generator and power turbine speeds and limits (rpm)

Rated compressor discharge temperature and limit

Rated power turbine inlet temperature and limit

Other engine limiting parameters

Compressor maps

Allowable combustor outlet temperature spread

Combustion fuel equivalence ratio

Sense of rotation (clockwise/counterclockwise)

Maximum temperature at which rated power can be achieved

Compressor configuration

Combustor configuration

Turbine configuration

Mass and moment of inertia of rotating elements

Balancing data

Type test schedule, measurements and data

Manufacturer's shop test schedule

Manufacturer's recommended overhaul schedule

1.5.4 Materials

Material specifications (including density, Poisson's ratio, range of chemical composition, room-temperature physical properties and, where material is subject to temperatures exceeding 427°C (800°F), the elevated temperature mechanical properties, as well as creep rate and rupture strength for the design service life).

1.5.5 Calculations and Analyses *(2007)*

Design basis for turbine and compressor rotors and blading including calculations or test results to substantiate the suitability and strength of components for the intended service.

Blade containment strength, see 4-2-3/5.9.

Design service life data

Vibration analysis of the entire propulsion shafting system; see 4-2-3/5.1.2 and 4-3-2/7.

3 Materials

3.1 Material Specifications and Tests

Materials entered into the construction of gas turbines are to conform to specifications approved in connection with the design. Copies of material specifications and purchase orders are to be submitted to the Surveyor for information and verification.

Except as noted in 4-2-3/3.3, the following materials are to be tested in the presence of, inspected and certified by the Surveyor. The materials are to meet the specifications of Part 2, Chapter 3, or to the requirements of the specifications approved in connection with the design:

i) *Forgings:* Compressor and turbine rotors, shafts, couplings, coupling bolts, integral gears and pinions.

ii) *Castings:* Compressor and turbine casings where the temperature exceeds 232°C (450°F) or where approved for use in place of any of the above forgings.

iii) *Plates:* Plates for casings of fabricated construction where the casing pressure exceeds 41.4 bar (42.9 kgf/cm^2, 600 psi) or the casing temperature exceeds 371°C (700°F).

iv) *Blade material:* Material for all turbine blades.

v) *Pipes, pipe fittings and valves:* See 4-6-1/Table 1 and 4-6-1/Table 2.

3.3 Alternative Materials and Tests

3.3.1 Alternative Specifications

Material manufactured to specifications other than those given in Part 2, Chapter 3 may be accepted, provided that such specifications are approved in connection with the design and that they are verified or tested by a Surveyor, as applicable, as complying with the specifications.

3.3.2 Steel-bar Stock

Hot-rolled steel bars up to 305 mm (12 in.) in diameter may be used when approved for use in place of any of the forgings as per 4-2-3/3.1i) above, under the conditions outlined in Section 2-3-8.

3.3.3 Materials for Turbines of 375 kW (500 hp) Rated Power or Less

Materials for turbines of 375 kW (500 hp) rated power or less, including shafting, integral gears, pinions, couplings and coupling bolts will be accepted on the basis of the material manufacturer's certified test reports and a satisfactory surface inspection and hardness check witnessed by the Surveyor. Coupling bolts manufactured to a recognized bolt standard and used as coupling bolts do not require material testing.

3.3.4 Certification Under Quality Assurance Assessment

For gas turbines certified under quality assurance assessment as provided for under 4-2-3/13.3.2(b), material tests required by 4-2-3/3.1 need not be witnessed by the Surveyor; such tests may be conducted by the turbine manufacturer whose certified material test reports will be accepted instead.

5 Design

5.1 Rotors and Blades *(1 July 2006)*

5.1.1 Criteria

Rotors, bearings, discs, drums and blades are to be designed in accordance with sound engineering principles, taking into consideration criteria such as fatigue, high temperature creep, etc. Design criteria along with engineering analyses substantiating the suitability of the design for the rated power and speed are to be submitted for review.

Design criteria are to include the design service life, which is the maximum number of hours of operation at rated power and speed. The service life between major overhauls is generally not to be less than 5000 hours or the equivalent of one year of the vessel's service.

The rated power is to be taken as that developed at the lowest expected inlet air temperature. In no case is this temperature to exceed 15°C (59°F).

5.1.2 Vibration *(1 July 2006)*

The designer or builder is to evaluate the shafting system for different modes of vibrations (torsional, axial, lateral) and their coupled effect, as appropriate.

5.3 Operation Above the Rated Speed and Power

Where operation above the rated power and speed for short duration is required in service, the design criteria for such operation, along with operating envelope, engineering analyses and type test data, are to be submitted for review.

5.5 Overhaul Interval

The manufacturer's recommended overhaul schedule is to be submitted for information and record and is to be considered with the design service life indicated in 4-2-3/5.1. As far as practicable, the overhaul schedule is to coincide with the survey cycle or Continuous Survey – Machinery cycle specified in Part 7.

5.7 Type Test Data

The manufacturer is to submit type test data in support of the design. The type test is to be witnessed and certified by a Surveyor or by an independent agency. The type test data are to contain at least the test schedule, measurements taken during the tests and test results. Properly documented, actual operational experience may be considered in lieu of type test data.

5.9 Casing *(2007)*

The gas turbine casing is to be designed such that, at overspeed up to 15% above the rated speed, any failure of blades or blade attachment devices will be contained.

Containment strength calculations, or other method such as computer simulation or impingement test, verifying the above requirement are to be submitted for review.

7 Gas Turbine Appurtenances

7.1 Overspeed Protective Devices

All propulsion and generator turbines are to be provided with overspeed protective devices to prevent the rated speed from being exceeded by more than 15%.

Where two or more turbines are coupled to the same output gear without clutches, the use of only one overspeed protective device for all turbines may be considered. This is not to prevent operation with one or more turbines uncoupled.

7.3 Operating Governors for Propulsion Gas Turbines

Propulsion turbines coupled to reverse gear, electric transmission, controllable-pitch propeller or similar are to be fitted with a separate independent speed governor system in addition to the overspeed protective device specified in 4-2-3/7.1. This governor system is to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

7.5 Operating Governors for Turbines Driving Electric Generators

7.5.1 Speed Governing *(2004)*

An operating governor is to be fitted to each gas turbine driving propulsion, vessel service or emergency electric generator. The governor is to be capable of automatically maintaining the turbine speed within the following limits.

7.5.1(a) The transient frequency variations in the electrical network when running at the indicated loads below are to be within ±10% of the rated frequency when:

i) Running at full load (equal to rated output) of the generator and the maximum electrical step load is suddenly thrown off;

In the case where a step load equivalent to the rated output of a generator is thrown off, a transient frequency variation in excess of 10% of the rated frequency may be acceptable, provided the overspeed protective device fitted in addition to the governor, as required by 4-2-3/7.1, is not activated.

ii) Running at no load and 50% of the full load of the generator is suddenly thrown on, followed by the remaining 50% after an interval sufficient to restore the frequency to steady state.

In all instances, the frequency is to return to within ±1% of the final steady state condition in not more than five (5) seconds.

7.5.1(b) For gas turbines driving emergency generators, the requirements of 4-2-3/7.5.1(a)ii) above are to be met. However, if the sum of all emergency loads that can be automatically connected is more than 50 % of the full load of the emergency generator, the sum of the emergency loads is to be used as the first applied load.

7.5.1(c) The permanent frequency variation is to be within ±5% of the rated frequency at any load between no load and the full load.

7.5.2 Load Sharing

Gas turbines driving AC generators that operate in parallel are to have the following governor characteristics. In the range between 20% and 100% of the combined rated load of all generators, the load on any individual generator will not differ from its proportionate share of the total combined load by more than the lesser of the following:

- 15% of the rated power of the largest generator or
- 25% of the individual generator.

7.5.3 Fine Adjustments

Provisions are to be made to adjust the governors sufficiently fine in order to permit a load adjustment within the limits of 5% of the rated load at normal frequency.

7.5.4 Turbines Driving Electric Propulsion Generators

For gas turbines driving electric propulsion generators, where required by the control system, this governor is to be provided with means for local hand control, as well as for remote adjustment from the control station.

7.7 Safety Systems and Devices

7.7.1 General

Gas turbines are to be fitted with automatic safety systems and devices for safeguards against hazardous conditions arising from malfunctions in their operations. The design of such systems and devices is to be evaluated with failure mode and effect analysis, which is to be submitted for review.

7.7.2 Automatic Shutdown

Gas turbines are to be fitted with a quick acting device which will automatically shut off fuel supply in the event of:

i) Overspeed;

ii) Excessive high vacuum at compressor inlet;

iii) Low lubricating oil pressure;

iv) Low lubricating oil pressure in reduction gear;

v) Loss of flame during operation;

vi) Excessive vibration;

vii) Excessive axial displacement of each rotor (except for gas turbines fitted with roller bearings); or

viii) Excessively high exhaust gas temperature;

7.7.3 Automatic Temperature Controls

Gas turbines are to be fitted with automatic control systems to maintain steady state temperatures in the following systems throughout the turbines' normal operating ranges:

i) Lubricating oil;

ii) Fuel oil (or in lieu of temperature, viscosity);

iii) Exhaust gas.

7.7.4 Starting System Safety

7.7.4(a) Automatic purging. Prior to commencing the ignition process, automatic purging is required for all starts and restarts. The purge phase is to be of sufficient duration so as to clear all parts of the turbine of accumulation of liquid or gaseous fuel.

7.7.4(b) Preset time. The starting control system is to be fitted with ignition detection devices. If light off does not occur within a preset time, the control system is to automatically abort the ignition, shutoff the main fuel valve and commence a purge phase.

7.7.5 Alarms and Shutdowns

4-2-3/Table 1 provides a summary of the required alarms and, where applicable, the corresponding requirements for shutdowns.

TABLE 1
List of Alarms and Shutdowns

Monitored Parameter	*Alarm*	*Shutdown*
Speed	High	Required [(2)]
Lubricating oil pressure	Low [(1)]	Required [(2)]
Lubricating oil pressure of reduction gear	Low [(1)]	Required [(2)]
Differential pressure across lubricating oil filter	High	
Lubricating oil temperature	High	
Fuel oil supply pressure	Low	
Fuel oil temperature	High	
Cooling medium temperature	High	
Bearing temperature	High	
Flame and ignition	Failure	Required [(2)]
Automatic starting	Failure	
Vibration	Excessive [(1)]	Required [(2)]
Axial displacement of rotor	High	Required [(2, 3)]
Exhaust gas temperature	High [(1)]	Required [(2)]
Vacuum at compressor inlet	High [(1)]	Required [(2)]
Control system power	Loss	

Notes:

1 Alarm is to be set at a point prior to that set for shutdown.

2 Each shutdown is to be accompanied by own alarm.

3 Except where fitted with roller bearings.

7.9 Hand Trip Gear

Hand trip gear for shutting off the fuel in an emergency is to be provided locally at the turbine control platform and, where applicable, at the centralized control station.

7.11 Air-intake Filters and Anti-icing

Air intake is to be provided with demisters and filters to minimize the entry of water and harmful foreign material. They are to be so designed as to prevent the accumulation of salt deposits on the compressor and turbine blades. Means are to be provided to prevent icing in the air intake.

7.13 Silencers

Inlet and exhaust silencers are to be fitted to limit the sound power level at one meter from the gas turbine system to 110 dB for unmanned machinery spaces or to 90 dB for manned machinery spaces.

9 Piping and Electrical Systems for Gas Turbines *(2007)*

The requirements of piping and electrical systems associated with operation of gas turbines for propulsion, electric power generation and vessel's safety are provided in Section 4-6-5, Section 4-6-7 and Section 4-8-2. These systems include:

Fuel oil:	4-6-5/3 (see 4-6-5/3.7 in particular)
Lubricating oil:	4-6-5/5 (see 4-6-5/5.3 and 4-6-5/5.5 in particular)
Cooling water:	4-6-5/7
Starting air:	4-6-5/9
Electric starting	4-8-2/11.11
Hydraulic system	4-6-7/3
Exhaust gas:	4-6-5/11.11

11 Installation of Gas Turbines

11.1 Pipe and Duct Connections

Pipe or duct connections to the gas turbine casing are to be made in such a way as to prevent the transmission of excessive loads or moments to the turbine.

11.3 Intake and Exhaust *(2007)*

Air inlets are to be located as high as possible to minimize water intake, and are to be fitted with baffle, demisters, anti-icing arrangements and silencers as indicated in 4-2-3/7.11 and 4-2-3/7.13. Air-intake ducting is to be arranged in accordance with the turbine manufacturer's recommendations with a view to providing the gas turbine with a uniform pressure and velocity flowfield at the compressor inlet. The exhaust outlets are to be so located as to prevent reingestion of exhaust gas into the intake.

11.5 Hot Surfaces

Hot surfaces likely to come into contact with the crew during operation are to be suitably guarded or insulated. Hot surfaces likely to exceed 220°C (428°F), and which are likely to come into contact with any leakage, under pressure or otherwise, of fuel oil, lubricating oil or other flammable liquid, are to be suitably insulated with non-combustible materials that are impervious to such liquid. Insulation material not impervious to oil is to be encased in sheet metal or an equivalent impervious sheath.

13 Testing, Inspection and Certification of Gas Turbines

13.1 Shop Inspection and Tests

The following shop tests and inspection are to be witnessed by a Surveyor on all gas turbines required to be certified by the Bureau under 4-2-3/1.1.

13.1.1 Material Tests

Materials entered into the construction of turbines are to be tested in the presence of a Surveyor in accordance with the provisions of 4-2-3/3.

13.1.2 Welded Fabrication

All welded fabrication is to be conducted with qualified welding procedures, by qualified welders, and with welding consumables acceptable to the Surveyors. See Section 2-4-2.

13.1.3 Pressure Tests

Turbine casings are to be subjected to a pressure test of 1.5 times the highest pressure in the casing during normal operation. Turbine casings may be divided by temporary diaphragms to allow for an even distribution of the test pressures. Where hydrostatic tests are not practicable, alternative tests to determine soundness and workmanship are to be submitted for consideration and approval in each case. Intercoolers and heat exchangers are to be hydrostatically tested on both sides to 1.5 times the design pressure.

13.1.4 Rotor Balancing

All finished compressor and turbine rotors are to be dynamically balanced at a speed equal to the natural period of the balancing machine and rotor, combined.

13.1.5 Shop Trial

Upon completion of fabrication and assembly, each gas turbine is to be subjected to a shop trial in accordance with the manufacturer's test schedule, which is to be submitted for review before the trial. During the trial, the turbine is to be brought up to its overspeed limit to enable the operation of the overspeed protective device to be tested.

13.3 Certification of Gas Turbines

13.3.1 General

Each gas turbine required to be certified by 4-2-3/1.1 is:

i) To have its design approved by the Bureau; for which purpose, plans and data as required by 4-2-3/1.5 are to be submitted to the Bureau for approval, and a gas turbine of the same type is to have been satisfactorily type tested or to have a documented record of satisfactory service experience (see 4-2-3/5.7);

ii) To be surveyed during its construction for compliance with the approved design, along with, but not limited to, material and nondestructive tests, pressure tests, dynamic balancing, performance tests, etc., as indicated in 4-2-3/13.1, all to be carried out to the satisfaction of the Surveyor.

13.3.2 Approval Under Type Approval Program *(2003)*

13.3.2(a) Product design assessment. Upon application by the manufacturer, each model of a type of turbine may be design assessed as described in 1-1-A3/5.1. For this purpose, each design of a turbine type is to be approved in accordance with 4-2-3/13.3.1i). The type test, however, is to be conducted in accordance with an approved test schedule and is to be witnessed by a Surveyor. Turbine so approved may be applied to the Bureau for listing on the ABS website as Products Design Assessed. Once listed, and subject to renewal and updating of certificate as required by 1-1-A3/5.7, turbine particulars will not be required to be submitted to the Bureau each time the turbine is proposed for use on board a vessel.

13.3.2(b) Mass produced turbines. A manufacturer of mass-produced turbines, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.5 (PQA).

Upon satisfactory assessment under 1-1-A3/5.5 (PQA), turbines produced in those facilities will not require a Surveyor's attendance at the tests and inspections indicated in 4-2-3/13.3.1ii). Such tests and inspections are to be carried out by the manufacturer whose quality control documents will be accepted. Certification of each engine will be based on verification of approval of the design and on continued effectiveness of the quality assurance system. See 1-1-A3/5.7.1(a).

13.3.2(c) Non-mass Produced Gas Turbines. A manufacturer of non-mass produced turbines, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.3.1(a) (AQS) or 1-1-A3/5.3.1(b) (RQS). Certification to 1-1-A3/5.5 (PQA) may also be considered in accordance with 4-1-1/Table 1.

13.3.2(d) Type Approval Program. Turbine types which have their designs approved in accordance with 4-2-3/13.3.2(a) and the quality assurance system of their manufacturing facilities approved in accordance with 4-2-3/13.3.2(b) or 4-2-3/13.3.2(c) will be deemed Type Approved and will be eligible for listing on the ABS website as Type Approved Product.

13.5 Shipboard Trials

After installation, each gas turbine, including all starting, control and safety system, is to be operated in the presence of the Surveyor to satisfactorily demonstrate function and freedom from harmful vibration at speeds within the operating range. Each gas turbine is also to operate to the overspeed limit to test the function of the overspeed governor. The means for the propulsion system to reverse are to be demonstrated and recorded.

15 Spare Parts

Spare parts are not required for purposes of classification. The maintenance of spare parts aboard each vessel is the responsibility of the owner.

PART

4

CHAPTER 2 Prime Movers

SECTION 3 Appendix 1 – Plans and Data for Gas Turbines *(2007)*

For each type of gas turbine to be approved, the drawings and data listed in the following table, and as applicable to the type of gas turbine, are to be submitted for approval (A) or for information (R) by each engine manufacturer. After the approval of an engine type has been given by the Bureau for the first time, only those documents as listed in the table, which have undergone substantive changes, will have to be submitted again for consideration by the Bureau. In cases where both (R) and (A) are shown, the first refers to cast components and the second to welded components. Bill of materials is to include material specification of the components, as listed.

No.	*A/R*	*Item*
1	A	Certified dimensional outline drawing and list of connections
2	A	Cross-sectional assembly drawing and bill of materials
3	A	Casings assembly and bill of materials
4	A	Foundations and fastening
5	A	Combustion chambers
6	A	Gasifiers
7	A	Regenerators or recuperators and Intercoolers and bill of materials
8	A	Turbine rotors and bill of materials
9	A	Compressor rotors and bill of materials
10	A	Compressor and turbine discs and bill of materials
11	A	Blading details and bill of material
12	A	Shafts and bill of materials
13	A	Bearing assembly and bill of materials
14	A	Thrust bearing assembly, performance data and bill of materials
15	A	Shaft coupling assembly including coupling alignment diagram and procedure and bill of materials
16	A	Clutches and brakes details and bill of materials
17	A	Starting arrangement
18	A	Fuel oil system including fuel injector system operational schematic and components drawing with connection schedule and bill of materials
19	A	Shielding of fuel oil service piping
20	A	Lubricating oil system schematic and bill of material
21	A	Air-intake system and air intake model test report
22	A	Exhaust system
23	A	Shielding and insulation of exhaust pipes, assembly
24	A	Governor arrangements including governor control and trip system data
25	A	Safety systems and devices and associated Failure Modes, Effects and Criticality Analysis (FMECA)
26	A	Control oil system assembly and arrangement drawing
27	R	Bleed/cooling/seal air schematic and bill of materials

No.	*A/R*	*Item*
28	A	Cooling system
29	A	Electrical and instrumentation schematics and arrangement drawings, list of terminations, and bill of materials
30	A	Accessory drive
31	A	Water wash
32	A	Enclosure arrangement
33	A	Fire protection
34	R	Gas turbine particulars (rated power, rated speed, max. temperature at which rated power can be achieved, etc.)
35	R	Speed vs power curves at site rated conditions
36	R	Ambient temperature vs power curves at site rated conditions
37	R	Output power vs shaft speed curves at site rated conditions
38	R	Heat rate correction factors
39	R	Type test schedule, measurements and data
40	R	Manufacturer's shop test schedule
41	R	Manufacturer's recommended overhaul schedule
42	R	Computer Steady State and Transient performance program (engine mounted system)
43	R	Engine Health Monitoring (EHM) equipment and program, where specified
44	R	Hot Section Repair Interval analyses
45	A	Welding procedures
46	R	B10 Bearing life analysis
47	R	Blading vibration analysis data
48	R	Lateral critical analysis
49	R	Torsional critical analysis report
50	R	Transient torsional analysis report
51	R	Allowable piping flange loading, as applicable
52	R	Spring mass model analysis, as applicable

PART
4

CHAPTER 2 Prime Movers

SECTION 4 Steam Turbines

1 General

1.1 Application

Steam turbines having a rated power of 100 kW (135 hp) and over, intended for propulsion and for auxiliary services essential for propulsion, maneuvering and safety (see 4-1-1/1.3) of the vessel, are to be designed, constructed, tested, certified and installed in accordance with the requirements of this section.

Steam turbines having a rated power of less than 100 kW (135 hp) are not required to comply with the provisions of this section but are to be designed, constructed and equipped in accordance with good commercial and marine practice. Acceptance of such steam turbines will be based on manufacturer's affidavit, verification of steam turbines nameplate data and subject to a satisfactory performance test after installation conducted in the presence of the Surveyor.

Steam turbines having a rated power of 100 kW (135 hp) and over, intended for services considered not essential for propulsion, maneuvering and safety, are not required to be designed, constructed and certified by the Bureau in accordance with the provisions of this section. However, they are to comply with safety features, such as overspeed protection, etc., as provided in 4-2-4/7 hereunder, as applicable, and are subject to a satisfactory performance test after installation, conducted in the presence of the Surveyor.

Piping systems of steam turbines, in particular, steam, condensate and lubricating oil systems are given in Section 4-6-6.

1.3 Definitions

For the purpose of this section the following definitions apply:

1.3.1 Rated Power

The *Rated Power* of a turbine is the maximum power output at which the turbine is designed to run continuously at its rated speed.

1.3.2 Rated Speed

The *Rated Speed* is the speed at which the turbine is designed to run continuously at its rated power. The rated speed is to be used for making strength calculations.

1.3.3 Turbine Overspeed Limit

The *Overspeed Limit* is the maximum intermittent speed allowed for a turbine in service. It is not to exceed the rated speed by more than 15% and is to be the maximum permissible setting of the overspeed governor.

1.3.4 Operating Temperature

The requirements for steam turbine rotors and blades in 4-2-4/5.3 and 4-2-4/5.5 are based on a maximum operating temperature at the turbine inlet of 427°C (800°F).

Installations for which this maximum operating temperature is exceeded will be subject to special consideration of the design criteria.

1.5 Plans and Particulars to be Submitted

1.5.1 Steam Turbine Construction

Sectional assembly

Casings

Foundation and fastening

Turbine rotors

Turbine discs

Blading

Shafts

Bearing arrangements

1.5.2 Steam Turbine Systems and Appurtenances

Couplings

Clutches

Steam inlet and exhaust system

Lubrication system.

Governor arrangements

Monitoring and safety arrangements

1.5.3 Data

Rated speed

Rated power

Mass and velocity of rotating elements

Area of wheel

Moment of inertia of wheel profile area

Center of gravity of blade and root

Balancing data

Manufacturer's shop operating test schedule

1.5.4 Materials

Material specifications (including density, Poisson's ratio, range of chemical composition, room-temperature physical properties and, where material is subject to temperatures exceeding 427°C (800°F), the high-temperature strength characteristics, as well as creep rate and rupture strength for the design service life).

1.5.5 Calculations and Analyses *(1 July 2006)*

Design basis for turbine rotors and blading including calculations or test results to substantiate the suitability and strength of components for the intended service.

A vibration analysis of the entire propulsion shafting system; see 4-2-4/5.3.4 and 4-3-2/7.

3 Materials

3.1 Material Specifications and Tests

Materials entered into the construction of turbines are to conform to specifications approved in connection with the design. Copies of material specifications and purchase orders are to be submitted to the Surveyor for information and verification.

Except as noted in 4-2-4/3.3, the following materials are to be tested in the presence of, inspected and certified by the Surveyor in accordance with the requirements of Part 2, Chapter 3 or to the requirements of the specifications approved in connection with the design:

i) *Forgings*: Discs and rotor drums, shafts and rotors, couplings, coupling bolts, integral gears and pinions for all turbines.

ii) *Castings*: Turbine casings and maneuvering valves where the temperature exceeds 232°C (450°F) or where approved for use in place of any of the above forgings.

iii) *Plates*: Plates for turbine casings of fabricated construction where the casing pressure exceeds 41.4 bar (42.9 kgf/cm^2, 600 psi) or the casing temperature exceeds 371°C (700°F).

iv) *Blade material*: Material for all turbine blades (hardness or chemical composition check-tested).

v) *Pipes, pipe fittings and valves*: See 4-6-1/Table 1 and 4-6-1/Table 2, except for maneuvering valves as provided for in 4-2-4/3.1ii) above.

3.3 Alternative Materials and Tests

3.3.1 Alternative Specifications

Material manufactured to specifications other than those given in Part 2, Chapter 3, may be accepted, provided that such specifications are approved in connection with the design and that they are verified or tested by a Surveyor, as applicable, as complying with the specifications.

3.3.2 Steel-bar Stock

Hot-rolled steel bars up to 305 mm (12 in.) in diameter may be used when approved for use in place of any of the forgings as per 4-2-4/3.1i) above, under the conditions outlined in Section 2-3-8.

3.3.3 Materials for Turbines of 375 kW (500 hp) Rated Power or Less

Materials for turbines of 375 kW (500 hp) rated power or less, including shafting, integral gears, pinions, couplings and coupling bolts will be accepted on the basis of the material manufacturer's certified test reports and a satisfactory surface inspection and hardness check witnessed by the Surveyor. Coupling bolts manufactured to a recognized bolt standard will not require material testing.

3.3.4 Certification Under Quality Assurance Assessment

For steam turbines certified under quality assurance assessment as provided for in 4-2-4/13.3.2(b), material tests required by 4-2-4/3.1 need not be witnessed by the Surveyor; such tests are to be conducted by the turbine manufacturer whose certified material test reports will be accepted instead.

5 Design

In lieu of the design rules provided hereunder, the Bureau will consider designs that are substantiated by sound engineering analyses conducted for all designed operating conditions, and taking into consideration strength criteria such as fatigue, high temperature creep, torsional vibration, etc., as appropriate.

5.1 Casings

5.1.1 Castings

Turbine casings and associated fixtures that are subject to pressure are to be of a design and made of material suitable for the stress and temperatures to which they may be exposed. Cast iron and cast steel may be considered suitable where the maximum operating temperature does not exceed 232°C (450°F) and 427°C (800°F), respectively. All castings are to be heat-treated to remove internal stresses.

5.1.2 Seals and Drains

Casings are to be provided with suitable seals. Drains are to be fitted in places where water or oil may collect.

5.1.3 Overpressure Protection

Turbine casings are to be fitted with means to prevent overpressure; see 4-2-4/7.11.

5.3 Rotor Shafts

5.3.1 General

The diameter of a turbine rotor shaft is to be determined by the following equations:

$$d = K \cdot \sqrt[6]{(bT)^2 + (mM)^2}$$

$$b = 0.073 + \frac{n}{Y}$$

$$m = \frac{c_1}{c_2 + Y}$$

where

d = shaft diameter at section under consideration; mm (in.)

Y = yield strength (see 2-3-1/13.3); N/mm^2 (kgf/mm^2, psi)

T = torsional moment at rated speed; N-m (kgf-cm, lbf-in)

M = bending moment at section under consideration; N-m (kgf-cm, lbf-in)

K, n, c_1 and c_2 are constants given in the following table:

	SI units	*MKS units*	*US units*
K	5.25	2.42	0.10
n	191.7	19.5	27800
c_1	1186	121	172000
c_2	413.7	42.2	60000

5.3.2 Shaft Diameters in way of Rotors

Where rotor members are fitted by a press or shrink fit, or by keying, the diameter of the shaft in way of the fitted member is to be increased not less than 10%.

5.3.3 Astern Power

In determining the required size of coupling shafting transmitting astern power, the astern torque is to be considered when it exceeds the transmitted ahead torque.

5.3.4 Vibration *(1 July 2006)*

The designer or builder is to evaluate the shafting system for different modes of vibrations (torsional, axial, lateral) and their coupled effect, as appropriate.

5.5 Blades

Blades are to be so designed as to avoid abrupt changes in section and to provide an ample amount of stiffness to minimize deflection and vibration. The area at the root of the blade is not to be less than that given in the following equation based upon either the tensile strength or yield strength of the material.

SI units	*MKS units*	*US units*
$A = \dfrac{4.39WN^2r}{F}$	$A = \dfrac{0.45WN^2r}{F}$	$A = \dfrac{114WN^2r}{F}$

Notes:

1. These equations are based solely upon centrifugal stress consideration. Designers/manufacturers are to take into consideration vibrations at speeds within the operating range.

2. Where turbine blades are designed with $F = 2Y$ resulting in a safety factor against ultimate strength of less than four, a dye-penetrant or magnetic-particle inspection is to be made of each individual rotor blade.

where

A = minimum blade root areas; cm^2 (in^2)

W = mass of one blade; kg (lb)

N = rpm at rated speed divided by 1000

r = radius of center of gravity of blade from centerline of shaft; cm (in.)

U = minimum tensile strength of material; N/mm^2 (kgf/mm^2, psi)

Y = yield strength of material (2-3-1/13.3); N/mm^2 (kgf/mm^2, psi)

F = U or optionally 2Y (See *Note* 2 under the equations above)

5.7 Discs or Drums

5.7.1 General

The following strength requirements are applicable only where creep and relaxation are not the determining factors in design, and their use does not relieve the manufacturer from responsibility for excessive creep or relaxation at normal operating temperatures. In general, they apply to installations where maximum operating temperature at the superheater outlet does not exceed 427°C (800°F).

5.7.2 Factors of Safety

The stress at any point in the disc or drum section is not to exceed the value Y/f, where Y is the yield strength of the material and f is the factor of safety given in the following table.

	Built-up rotor			*Solid rotor*	
	Propulsion	*Auxiliary*		*Propulsion*	*Auxiliary*
Radial stress, R	2.5	2.25		2.5	2.25
Tangential stress, T	2.5	2.25		2	2
Mean tangential stress [(1)], T_m	3	3		3	3

1 T_m is not to be higher than ultimate tensile strength divided by a factor of safety of 4.

5.7.3 Symbols

The symbols used in the equations are as follows [units of measure are given in the order of SI units (MKS units, US units)]:

R = radial stress; N/mm^2 (kgf/mm^2, psi)

T = tangential stress; N/mm^2 (kgf/mm^2, psi)

Y = yield strength (see 2-3-1/13.3); N/mm^2 (kgf/mm^2, psi)

U = minimum tensile strength; N/mm^2 (kgf/mm^2, psi)

S = sum of principal stresses; N/mm^2 (kgf/mm^2, psi)

D = difference of principal stresses; N/mm^2 (kgf/mm^2, psi)

ΔS = change in S caused by change in thickness

ΔD = change in D caused by change in thickness

y, y' = successive thickness of disc at step points; cm (in.)

V = tangential velocity at rated speed; m/s (ft/s)

n = Poisson's ratio

w = specific mass of material; kg/cm^3 (lb/in^3)

T_m = mean tangential stress; N/mm^2 (kgf/mm^2, psi)

N = rpm at rated speed divided by 1000

A = area of wheel profile, including the rim, on one side of axis of rotation; cm^2 (in^2)

I = moment of inertia of area A about the axis of rotation; cm^4 (in^4)

W = total mass of blades and roots; kg (lb)

$\bar{r}$ = radial distance to center of gravity of W; cm (in.)

P = Total rim load due to centrifugal force of blades; N (kgf, lbf)

5.7.4 Elastic Stress

To calculate the elastic stresses, assume R = 0 at the edge of the bore in solid rotors if the inspection hole is larger than one-fourth the basic diameter in way of the discs, and at the bottom of the keyway in the bore for separate discs. Assume R = T at the center for solid rotors if the inspection hole does not exceed one-fourth the basic diameter in way of the discs. Assume that T has the maximum permissible value at the starting point. Proceed step by step outward to the rim or bottom of the machined blade grooves, calculating S and D at the step points for the determination of R and T at all points on the disc or drum section, using the following equations.

$$S_2 = (S_1 + \Delta S_1) + k_1 w(1 + n)(V_1^2 - V_2^2)$$

$$D_2 V_2^2 = (D_1 + \Delta D_1)V_1^2 + k_2 w(1 - n)(V_2^4 - V_1^4)$$

where k_1 and k_2 are factors given in the following table:

	SI units	*MKS units*	*US units*
k_1	0.5	0.051	0.186
k_2	0.25	0.025	0.093

$$R = \frac{S - D}{2}$$

$$T = \frac{S + D}{2}$$

$$\Delta S = R(n+1)\left(\frac{y}{y'} - 1\right)$$

$$\Delta D = R(n-1)\left(\frac{y}{y'} - 1\right)$$

The calculated radial stress R at the rim or bottom of the machined blade grooves determines the maximum permissible rim load. The rim load in this calculation is the total load due to blades, roots and that portion of the rim which extends beyond the bottom of the groove, neglecting supporting effect in the rim. If in the calculation it is found that the permissible stress at any point has been exceeded, the calculation is to be repeated, assuming a value of T at the starting point sufficiently low to prevent the calculated stress from exceeding the permissible stress at any point.

5.7.5 Mean Tangential Stress

The mean tangential stress is to be calculated by the following equation:

$$T_m = T_m = \frac{c_1 w N^2 I}{A} + \frac{c_2 P}{2\pi A}$$

where

	SI units	*MKS units*	*US units*
P	$109.7W\bar{r}N^2$	$11.2W\bar{r}N^2$	$28.4W\bar{r}N^2$
c_1	1.10	0.11	28.4
c_2	0.01	0.01	1.0

7 Steam Turbine Appurtenances

7.1 Overspeed Protective Devices

All propulsion and auxiliary turbines are to be provided with a overspeed protective device to prevent the rated speed from being exceeded by more than 15%.

In addition to cutting off the main steam supply, where steam from other systems or exhaust steam are admitted to the turbine lower stages, they are also to be cut off at the activation of overspeed protective device.

Where two or more turbines are coupled to the same output gear, use of only one overspeed protective device for all turbines may be considered.

7.3 Operating Governors for Propulsion Turbines

Propulsion turbines coupled to reverse gear, electric transmission, controllable-pitch propeller or similar are to be fitted with a separate independent speed governor system, in addition to the overspeed protective device specified in 4-2-4/7.1. This governor system is to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

7.5 Operating Governors for Turbines Driving Electric Generators

7.5.1 Speed Governing *(2004)*

An operating governor is to be fitted to each steam turbine driving propulsion or vessel service electric generator. The governor is to be capable of automatically maintaining the turbine speed within the following limits.

7.5.1(a) The transient frequency variations in the electrical network when running at the indicated loads below are to be within ±10% of the rated frequency when:

i) Running at full load (equal to rated output) of the generator and the maximum electrical step load is suddenly thrown off;

In the case where a step load equivalent to the rated output of a generator is thrown off, a transient frequency variation in excess of 10% of the rated frequency may be acceptable, provided the overspeed protective device fitted in addition to the governor, as required by 4-2-1/7.5.3, is not activated.

ii) Running at no load and 50% of the full load of the generator is suddenly thrown on, followed by the remaining 50% after an interval sufficient to restore the frequency to steady state.

In all instances, the frequency is to return to within ±1% of the final steady state condition in no more than five (5) seconds.

7.5.1(b) The permanent frequency variation is to be within ±5% of the rated frequency at any load between no load and the full load.

7.5.2 Load Sharing

Steam turbines driving AC generators that operate in parallel are to have the following governor characteristics. In the range between 20% and 100% of the combined rated load of all generators, the load on any individual generator will not differ from its proportionate share of the total combined load by more than the lesser of the following:

- 15% of the rated power of the largest generator
- 25% of the individual generator

7.5.3 Fine Adjustments

Provisions are to be made to adjust the governors sufficiently fine in order to permit a load adjustment within the limits of 5% of the rated load at normal frequency.

7.5.4 Turbines Driving Electric Propulsion Generators

For steam turbines driving electric propulsion generators, where required by the control system, this governor is to be provided with means for local hand control as well as remote adjustment from the control station.

7.7 Hand and Automatic Tripping

Arrangements are to be provided for shutting off steam to propulsion turbines by suitable hand trip gear situated at the main control console and at the turbine itself. For auxiliary steam turbines, hand tripping is to be arranged in the vicinity of the turbine overspeed protective device. The hand tripping gear is to shut off both the main and exhaust steam supplies to the turbines.

Automatic means of shutting off the steam supply (including exhaust steam supply) through a quick acting device is also to be fitted for all steam turbines upon overspeed (see 4-2-4/7.1) and upon failure of the lubricating oil system (see 4-6-6/9). See also 4-2-4/7.11 for back-pressure trip.

7.9 Shaft Turning Gear

Propulsion turbines are to be equipped with a slow turning gear, providing for rotation in both directions. For auxiliary turbines, provisions are to be made that allow at least for shaft turning by hand.

For vessels fitted with remote propulsion control, the turning gear status is to be indicated at each remote propulsion control station. In addition, interlock is to be fitted to prevent operation of the turbine when the turning gear is engaged, and vice versa. See also 4-9-2/Table 1.

In the propulsion machinery space intended for centralized or unattended operation (**ACC** or **ACCU** notation), the non-rotation of the propulsion shaft in excess of a predetermined duration on a standby or stop maneuver is to be alarmed at the centralized control station and other remote control stations. In addition, for the unattended propulsion machinery space (**ACCU** notation), whenever such duration is exceeded, means for automatic roll-over of the propulsion turbine shaft is to be fitted. See also 4-9-4/Table 4.

7.11 Overpressure Protection *(2006)*

Sentinel valves or equivalent are to be fitted to all main and auxiliary steam turbine exhausts to provide a warning of excessive pressure to personnel in the vicinity of the exhaust end of steam turbines. Auxiliary steam turbines sharing a common condenser are to be fitted with back-pressure trips or other approved protective device

9 Piping Systems for Steam Turbines

The requirements of piping systems essential for operation of steam turbines for propulsion, electric power generation and vessel's safety are in Section 4-6-6. These systems are:

Steam piping for propulsion turbines:	4-6-6/3.11
Steam piping for auxiliary turbines:	4-6-6/3.13
Condensers:	4-6-6/5.3.2
Lubricating oil system:	4-6-6/9
Condenser cooling system:	4-6-6/11

11 Installation of Steam Turbines

11.1 Exhaust Steam to Turbine

If exhaust steam is admitted to a turbine, means are to be provided to prevent water from entering the turbine.

11.3 Extraction of Steam

Where provision is made for extraction of steam, approved means are to be provided for preventing a reversal of flow to the turbine.

11.5 Pipe and Duct Connections

Any pipe or duct connections to the steam turbine casing are to be made in such a manner as to prevent the transmission of excessive loads or moments to the turbine casing.

11.7 Hot Surfaces

Hot surfaces likely to come into contact with crew during operation are to be suitably guarded or insulated. Hot surfaces likely to exceed 220°C (428°F), and which are likely to come into contact with any leakage, under pressure or otherwise, of fuel oil, lubricating oil or other flammable liquid, are to be suitably insulated with non-combustible materials that are impervious to such liquid. Insulation material not impervious to oil is to be encased in sheet metal or an equivalent impervious sheath.

13 Testing, Inspection and Certification of Steam Turbines

13.1 Shop Inspection and Tests

The following shop tests and inspections are to be witnessed by a Surveyor on all steam turbines required to be certified by the Bureau under 4-2-4/1.1.

13.1.1 Material Tests

Materials entered into the construction of turbines are to be tested in the presence of a Surveyor, in accordance with the provisions of 4-2-4/3.

13.1.2 Welded Fabrication

All welded fabrication is to be conducted with qualified welding procedures, by qualified welders and with welding consumables acceptable to the Surveyors. See Section 2-4-2.

13.1.3 Nondestructive Examination of Turbine Blades

Where turbine blades are designed with $F = 2Y$ (see 4-2-4/5.5), resulting in a safety factor against ultimate strength of less than four, dye-penetrant or magnetic-particle inspection is to be made of each rotor blade.

13.1.4 Hydrostatic Tests

Turbine casings and maneuvering valves are to be subjected to hydrostatic tests of 1.5 times the working pressure. Turbine casings may be divided by temporary diaphragms to allow for an even distribution of test pressures. Where hydrostatic tests are not practicable, alternative tests to determine soundness and workmanship are to be submitted for consideration and approval in each case.

Condensers are to have both the steam side and the water side hydrostatically tested to 1.5 times the design pressure; in any case, the test pressure on the steam side is not to be less than 1 bar (1 kgf/cm^2, 15 lb/in^2). See also 4-6-6/5.11.2.

13.1.5 Safety Relief Valves

All safety relief valves are to be tested and set in the presence of the Surveyor.

13.1.6 Vibration and Balancing

Excessive vibration is not to occur within the operating speed range of turbines. Turbine rotors and discs are to be dynamically balanced at a speed equal to the natural period of the balancing machine and rotor combined.

13.1.7 Shop Trial

Upon completion of fabrication and assembly, each steam turbine is to be subjected to a shop trial in accordance with the manufacturer's test schedule, which is to be submitted for review before the trial. The test schedule is to specify the duration of tests and to include full load test, half load response tests, full load thrown-off tests, etc. During the trial, the turbine is to be brought up to its overspeed limit to enable the operation of the overspeed protective device to be tested. Where this is not practicable, the manufacturer may submit alternative testing methods for consideration.

13.3 Certification of Steam Turbines

13.3.1 General

Each steam turbine required by 4-2-4/1.1 to be certified is:

i) To have its design approved by the Bureau, for which purpose, plans and data, as required by 4-2-4/1.5 are to be submitted to the Bureau for approval.

ii) To be surveyed during its construction for compliance with the design approved, along with, but not limited to, material and nondestructive tests, hydrostatic tests, dynamic balancing, performance tests, etc., as indicated in 4-2-4/13.1, all to be carried out to the satisfaction of the Surveyor.

13.3.2 Approval Under Type Approval Program *(2003)*

13.3.2(a) Product design assessment. Upon application by the manufacturer, each model of a type of turbine may be design assessed, as described in 1-1-A3/5.1. For this purpose, each design of a turbine type is to be approved in accordance with 4-2-4/13.3.1i) and either satisfactorily type tested in a shop in the presence of a Surveyor or substantiated by documented satisfactory service experience. Turbine so approved may be applied to the Bureau for listing on the ABS website as Products Design Assessed. Once listed, and subject to renewal and updating of certificate as required by 1-1-A3/5.7, turbine particulars will not be required to be submitted to the Bureau each time the turbine is proposed for use onboard a vessel.

13.3.2(b) Mass produced turbines. A manufacturer of mass-produced turbines who operates a quality assurance system in the manufacturing facilities may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.5 (PQA).

Upon satisfactory assessment under 1-1-A3/5.5 (PQA), turbines produced in those facilities will not require a Surveyor's attendance at the tests and inspections indicated in 4-2-4/13.3.1ii). Such tests and inspections are to be carried out by the manufacturer whose quality control documents will be accepted. Certification of each turbine will be based on verification of approval of the design and on continued effectiveness of the quality assurance system. See 1-1-A3/5.7.1(a).

13.3.2(c) Non-mass Produced Turbines. A manufacturer of non-mass produced turbines who operates a quality assurance system in the manufacturing facilities may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.3.1(a) (AQS) or 1-1-A3/5.3.1(b) (RQS). Certification to 1-1-A3/5.5 (PQA) may also be considered in accordance with 4-1-1/Table 1.

13.3.2(d) Type Approval Program. Turbine types which have their designs assessed in accordance with 4-2-4/13.3.2(a) and the quality assurance system of their manufacturing facilities assessed in accordance with 4-2-4/13.3.2(b) or 4-2-4/13.3.2(c) will be deemed Type Approved and will be eligible for listing on the ABS website as Type Approved Product.

13.5 Shipboard Trials

Before final acceptance, the entire installation of each steam turbine including all control and safety equipment is to be operated in the presence of the Surveyor to demonstrate its ability to function satisfactorily under operating conditions and its freedom from harmful vibration at speeds within the operating range.

Each steam turbine is to be tested to the overspeed limit in order to operate the overspeed governor.

The reversing characteristics of propulsion turbine plants are to be demonstrated and recorded.

PART 4

CHAPTER 2 Prime Movers

SECTION 4 Appendix 1 – Guidance for Spare Parts

1 General

While spare parts are not required for purposes of classification, the spare parts list below is provided as a guidance for vessels intended for unrestricted service. Depending on the turbine design, spare parts other than those listed below, such as electronic control cards, should be considered.

3 Spare Parts for Propulsion Steam Turbines

a) One (1) set of springs for governor, relief and maneuvering valves

b) Sufficient packing rings with springs to repack one gland of each kind and size

c) One (1) set of thrust pads or rings, also springs where fitted, for each size turbine-thrust bearing

d) Bearing bushings sufficient to replace all of the bushings on every turbine rotor, pinion and gear for main propulsion, spare bearing bushings sufficient to replace all of the bushings on each non-identical auxiliary-turbine rotor, pinion and gear having sleeve-type bearings or complete assemblies consisting of outer and inner races and cages complete with rollers or balls where these types of bearings are used

e) One (1) set of bearing shoes for one face, for one single-collar type main thrust bearing where fitted. Where the ahead and astern pads differ, pads for both faces are to be provided.

f) One (1) set of strainer baskets or inserts for filters of special design of each type and size, for oil filters.

g) Necessary special tools.

5 Spare Parts for Steam Turbines Driving Electric Generators

a)	Main bearings	Bearing bushes or roller bearings of each size and type fitted for the shafts of the turbine rotor and of the reduction gearing, if any, for one engine	1 set
b)	Turbine thrust bearing	Pads for one face of tilting pad type thrust, with liners, or rings for turbine adjusting block with assorted liners, for one engine	1 set
c)	Turbine shaft sealing rings	Carbon sealing rings where fitted, with springs for each size and type of gland, for one engine	1 set
d)	Oil filters	Strainer baskets or inserts, for filters of special design, of each type and size	1 set

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

CONTENTS

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 1 Gears

1 General

1.1 Application

Gears having a rated power of 100 kW (135 hp) and over, intended for propulsion and for auxiliary services essential for propulsion, maneuvering and safety (see 4-1-1/1.3) of the vessel, are to be designed, constructed, certified and installed in accordance with the provisions of this section.

Gears having a rated power of less than 100 kW (135 hp) are not required to comply with the provisions of this section but are to be designed, constructed and equipped in accordance with good commercial and marine practice. Acceptance of such gears will be based on the manufacturer's affidavit, verification of gear nameplate data and subject to a satisfactory performance test after installation conducted in the presence of the Surveyor.

Gears having a rated power of 100 kW (135 hp) and over, intended for services considered not essential for propulsion, maneuvering and safety, are not required to be designed, constructed and certified by the Bureau in accordance with the requirements of this section, but are to be installed and tested to the satisfaction of the Surveyor.

Piping systems of gears, in particular, lubricating oil and hydraulic oil, are addressed in 4-6-5/5, 4-6-6/9 and 4-6-7/3.

1.3 Definitions

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

1.3.1 Gears

The term *Gears* as used in this section covers external and internal involute spur and helical cylindrical gears having parallel axis as well as bevel gears used for either main propulsion or auxiliary services.

1.3.2 Rated Power

The *Rated Power* of a gear is the maximum transmitted power at which the gear is designed to operate continuously at its rated speed.

1.3.3 Rated Torque

The *Rated Torque* is defined by the rated power and speed and is the torque used in the gear rating calculations.

1.3.4 Gear Rating

The *Gear Rating* is the rating for which the gear is designed in order to carry its rated torque.

1.5 Plans and Particulars to be Submitted

1.5.1 Gear Construction

General arrangement

Sectional assembly

Details of gear casings

Bearing load diagram

Quill shafts, gear shafts and hubs

Shrink fit calculations and fitting instructions

Pinions

Wheels and rims

Details of welded construction of gears

1.5.2 Gear Systems and Appurtenances

Couplings

Coupling bolts

Lubricating oil system and oil spray arrangements

1.5.3 Data

Transmitted rated power for each gear

Revolutions per minute for each gear at rated transmitted power

Bearing lengths and diameters

Length of gap between helices, if any

Distance between inner ends of bearings

Tooth form layout (see 4-3-1A1/Figure 1) or calculated data

Facewidths, net and total

Width of tooth at highest stressed section

Helix angle at reference and at pitch diameter

Helix deviation

Normal pressure angle

Transverse pressure angle at reference cylinder

Transverse pressure angle at working pitch cylinder

Reference cone angle for gears

Tip angle for gears

Cone distance for gears

Middle cone distance for gears

Normal module

Transverse module

Bending moment arm for tooth root bending stress for application of load at the point of single tooth pair contact

Working pitch diameter of gears

Tip diameter of gears

Root diameter of gears

Reference diameters

Addendum

Addendum modification coefficient of gears

Dedendum

Transverse diametral pitch

Normal base pitch

Number of gear teeth

Virtual number of spur teeth for gears

Center distance between mating gears

Length of contact in plane of rotation

Root fillet radius of gears in the critical section

Axial lead modification or lead mismatch, if any, for reference

Method of cutting and finishing gear teeth

Tooth thickness modification coefficient (midface)

Sketch of basic rack tooth form

Root radius, addendum, dedendum of basic rack

Degree of finish of tooth flank

Grade of accuracy

Tooth hardness range, including core hardness and total depth of hardness, surface to core

Mean peak-to-valley roughness of tooth root fillets

Mass of rotating parts

Balancing data

Spline data

Shrink allowance for rims and hubs

Type of coupling between prime mover and reduction gears

Type and viscosity of lubricating oil recommended by manufacturer

For a comprehensive listing of data, see Appendix 4-3-1A3.

1.5.4 Materials

The following typical properties of gear materials are to be submitted:

Range of chemical composition

Physical properties at room temperature

Endurance limits for pitting resistance, contact stress and tooth root bending stress

Heat treatment of gears, coupling elements, shafts, quill shafts, and gear cases

1.5.5 Hardening Procedure

The hardening procedure for surface hardened gears is to be submitted for review. The submittal is to include materials, details for the procedure itself, quality assurance procedures and testing procedures. The testing procedures are to include surface hardness, surface hardness depth (e.g., case depth) and core hardness. Surface hardness depth and core hardness (and their shape) are to be determined from sectioned test samples. These test samples are to be of sufficient size to provide for determination of core hardness and are to be of the same material and heat treated as the gears that they represent. Forgings are to be tested in accordance with Part 2, Chapter 3.

1.5.6 Calculations and Analyses

Bearing life calculations

Tooth coupling and spline connection calculations

3 Materials

3.1 Material Specifications and Test Requirements

3.1.1 Material Certificates

Material for gears and gear units is to conform to specifications approved in connection with the design in each case. Copies of material specifications and purchase orders are to be submitted to the Surveyor for information.

3.1.2 Material Tests *(2006)*

Except as noted in 4-3-1/3.3, the following materials are to be tested in the presence of and inspected by the Surveyor. The materials are to meet the specifications in Part 2, Chapter 3 or that approved in connection with the design.

- Forgings for shafting, couplings, coupling bolts.
- Forgings for through hardened, induction hardened, and nitrided gears.
- Forgings for carburized gears. See 2-3-7/3.9.3(g).
- Castings approved for use in place of any of the above forgings.

3.3 Alternative Material Test Requirements

3.3.1 Alternative Specifications

The Surveyor will inspect and test material manufactured to other specifications than those given in Part 2, Chapter 3, provided that such specifications are approved in connection with the designs and that they are clearly indicated on purchase orders which are provided for the Surveyor's information.

3.3.2 Gears Certified Under Quality Assurance Assessment

For gear units certified under quality assurance assessment provided for in 4-3-1/9.7 hereunder, material tests and inspections required in 4-3-1/3.1 need not be witnessed by the Surveyor; such tests and inspections are to be conducted by the gear manufacturer whose certified material test reports will be accepted instead.

3.3.3 Steel-bar Stock

Hot-rolled steel bars up to 305 mm (12 in.) in diameter may be used when approved for use in place of any of the forgings as per 4-3-1/3.3.1 above, under the conditions outlined in Section 2-3-8.

3.3.4 Gear Units of 375 kW (500 hp) or Less

Material for gear units of 375 kW (500 hp) or less, including shafting, gears, couplings, and coupling bolts will be accepted on the basis of the manufacturer’s certified material test reports and a satisfactory surface inspection and hardness check witnessed by the Surveyor. Coupling bolts manufactured to a recognized bolt standard do not require material testing.

3.3.5 Power Takeoff Gears and Couplings

Materials for power takeoff gears and couplings that are:

- For transmission of power to drive auxiliaries that are for use in port only (e.g., cargo oil pump), and
- Declutchable from propulsion shafting

may be treated in the same manner as 4-3-1/3.3.4 above, regardless of power rating.

5 Design

5.1 Gear Tooth Finish

In general the gear teeth surface finish is not to be rougher than 1.05 μm (41 μin.) arithmetic or centerline average. However, gears having a rated power below 3728 kW (5000 hp) and with a surface finish rougher than 1.05 μm (41 μin.) arithmetic or centerline average will be specially considered, taking into account the lubricant recommended by the manufacturer.

5.3 Bearings

Bearings of gear units are to be so designed and arranged that their design lubrication rate is assured in service under working conditions.

5.3.1 Journal Bearings *(2003)*

For journal bearings, the maximum bearing pressures, minimum oil film thickness, maximum predicted internal bearing temperature and the maximum static unit load are to be in accordance with an applicable standard such as ISO 12130-1:2001 (plain tilting plain thrust bearings), ISO 12131-1:2001 (plain thrust pad bearings), ISO 12167-1:2001 (plain journal bearings with drainage grooves), ISO 12168-1:2001 (plain journal bearings without drainage grooves), ANSI/AGMA 6032-A94, Table 8, etc.

5.3.2 Roller Bearings *(2003)*

The minimum L10 bearing life is not to be less than 20,000 hours for ahead drives and 5,000 hours for astern. Shorter life may be considered in conjunction with an approved bearing inspection and replacement program reflecting the actual calculated bearing life. See 4-3-5/5.9 for application to thrusters. Calculations are to be in accordance with an applicable standard such as ISO 76:1987, ISO 281:1990 (rolling bearings, for static and dynamic ratings, respectively).

5.3.3 Alternative Designs *(2003)*

Consideration will be given to bearing pressures exceeding the limits in the standards listed in 4-3-1/5.3.1 and 4-3-1/5.3.2 provided the manufacturer can demonstrate a reliable operating history with similar designs.

5.3.4 Shaft Alignment Analysis *(2004)*

For gear-shafts which are directly connected to the propulsion shafting, the load on the gear shaft bearings is to be evaluated by taking into consideration the loads resulting from the shafting alignment condition.

5.5 Gear Cases *(2003)*

Gear cases are to be of substantial construction in order to minimize elastic deflections and maintain accurate mounting of the gears. They are to be designed to withstand without deleterious deflection: the tooth forces generated by the gear elements, thrust bearing arrangement, line shaft alignment, prime mover(s), clutches, couplings and accessories, under all modes of operation. Additionally, the inertial effects of the gears within the case, due to 1 **g** horizontal and 2 **g** vertical dynamic forces of the ship in a seaway, are to be considered. Calculations, in accordance with the manufacturer's code of practice, substantiating these Rule requirements are to be submitted. For gear case designs for which the manufacturer can provide a satisfactory service history, submission of calculations may be waived.

5.7 Access for Inspection

The construction of gear cases is to be such that a sufficient number of access points are provided for adequate inspection of gears, checking of gear teeth contacts and measurement of thrust bearing clearance. Alternative methods such as use of special viewing devices may be considered.

5.9 Calculation of Shafts for Gears

5.9.1 General *(1 July 2006)*

The diameter of shafts for gears is to be determined by the following equations:

$$d = k \cdot \sqrt[6]{(bT)^2 + (mM)^2}$$

$$b = 0.073 + \frac{n}{Y}$$

$$m = \frac{c_1}{c_2 + Y}$$

where (in SI (MKS and US units), respectively):

d = shaft diameter at section under consideration; mm (in.)

Y = yield strength (see 2-1-1/13.3); N/mm^2 (kgf/mm^2, psi)

T = torsional moment at rated speed; N-m (kgf-cm, lbf-in) (See also 4-3-1/5.9.6 to account for effect of torsional vibrations, where applicable.)

M = bending moment at section under consideration; N-m (kgf-cm, lbf-in)

k, n, c_1 and c_2 are constants given in the following table:

	SI units	*MKS units*	*US units*
k	5.25	2.42	0.10
n	191.7	19.5	27800
c_1	1186	121	172000
c_2	413.7	42.2	60000

5.9.2 Shaft Diameter in way of Gear Wheel

Where gear wheels are fitted by keying, the diameter of the shaft in way of the fitted member is to be increased not less than 10%.

5.9.3 Astern Power

In determining the required size of gear, coupling and shafting transmitting astern power, the astern torque is to be considered when it exceeds the transmitted ahead torque. See also 4-1-1/7.5.

5.9.4 Quill Shafts

In the specific case of quill shafts subjected to small stress raisers and no bending moments, the least diameter may be determined by the following equation:

SI units	*MKS units*	*US units*
$d = 5.25 \cdot \sqrt[3]{bT}$	$d = 2.42 \cdot \sqrt[3]{bT}$	$d = 0.1 \cdot \sqrt[3]{bT}$
$b = 0.053 + \frac{187.8}{Y}$	$b = 0.053 + \frac{19.1}{Y}$	$b = 0.053 + \frac{27200}{Y}$

5.9.5 Shaft Couplings

For shaft couplings, coupling bolts, flexible coupling and clutches, see 4-3-2/5.19. For keys, see 4-3-2/5.7.

5.9.6 Vibration *(1 July 2006)*

The designer or builder is to evaluate:

i) The shafting system for different modes of vibrations (torsional, axial, lateral) and the their coupled effect, as appropriate,

ii) The diameter of shafts considering maximum total torque (steady and vibratory torque),

iii) The gears for gear chatter and harmful torsional vibrations stresses. See also 4-3-2/7.5.8.

5.9.7 Shrink Fitted Pinions and Wheels

For pinions and wheels shrink-fitted on shafts, preloading and stress calculations and fitting instructions are to be submitted for review. In general, the torsional holding capacity is to be at least 2.8 times the transmitted mean torque plus vibratory torque due to torsional vibration. For calculation purpose, to take account of torsional vibratory torque, the following factors may be applied to the transmitted torque, unless the actual measured vibratory torque is higher, in which case the actual vibratory torque is to be used.

- For direct diesel engine drives: 1.2
- For all other drives, including diesel engine drives with elastic couplings: 1.0

The preload stress based on the maximum available interference fit or maximum pull-up length is not to exceed 90% of the minimum specified yield strength.

The following friction coefficients are to be used:

- Oil injection method of fit: 0.13
- Dry method of fit: 0.18

5.9.8 Shrink Fitted Wheel Rim

For shrink-fitted wheel rims, preloading and stress calculations and fitting instructions are to be submitted for consideration. In general, the preloading stress based on the maximum interference fit or maximum pull-up length is not to exceed 90% of the minimum specified yield strength.

5.11 Rating of Cylindrical and Bevel Gears

The calculation procedures for the rating of external and internal involute spur and helical cylindrical gears having parallel axes and of bevel gears, with regard to surface durability (pitting) and tooth root bending strength, may be as given in Appendix 4-3-1A1. These procedures are in substantial agreement with ISO 6336 and ISO-DIS 10300 for cylinder and bevel gears, respectively.

5.13 Alternative Gear Rating Standards

Consideration will be given to gears that are rated based on any of the following alternative standards. In which case, gear rating calculations and justification of the applied gear design coefficients in accordance with the applicable design standard are to be submitted to the Bureau for review.

Cylindrical Gears	*Bevel Gears*
AGMA 6032-A94	AGMA 2003-A86
ISO 6336	ISO-DIS 10300
DIN 3990, Part 31	DIN 3991

5.15 Gears with Multiple Prime Mover Inputs *(2003)*

For single helical gears with arrangements utilizing multiple prime mover inputs, and single or multiple outputs, the following analyses for all operating modes are to be conducted:

- All bearing reactions
- Tooth modifications
- Load distributions on the gear teeth
- Contact and tooth root bending stresses

A summary of the results of these analyses for each operating mode is to be submitted for review.

7 Piping Systems for Gears

The requirements of piping systems essential for operation of gears for propulsion, maneuvering, electric power generation and vessel's safety are in Section 4-6-5 for diesel engine and gas turbine installations and in Section 4-6-6 for steam turbine installations. Additionally, requirements for hydraulic and pneumatic systems are provided in Section 4-6-7. Specifically, the following references are applicable:

Lubricating oil system:	4-6-5/5 and 4-6-6/9
Cooling system:	4-6-5/7 and 4-6-6/11
Hydraulic system:	4-6-7/3
Pneumatic system:	4-6-7/5
Piping system general requirements:	Section 4-6-1 and Section 4-6-2

9 Testing, Inspection and Certification of Gears

9.1 Material Tests

For testing of materials intended for gear construction, see 4-3-1/3.1 and 4-3-1/3.3.

9.3 Dynamic Balancing

Finished pinions and wheels are to be dynamically balanced in two planes where their pitch line velocity exceeds 25 m/s (4920 ft/min).

Where their pitch line velocity does not exceed 25 m/s (4920 ft/min) or where dynamic balance is impracticable due to size, weight, speed or construction of units, the parts may be statically balanced in a single plane.

The residual unbalance in each plane is not to exceed the value determined by the following equations:

SI units	*MKS units*	*US units*
$B = 24 \cdot W/N$	$B = 24000 \cdot W/N$	$B = 15.1 \cdot W/N$

where

B = maximum allowable residual unbalance; N-mm (gf-mm, oz-in)

W = weight of rotating part; N (kgf, lbf)

N = rpm at rated speed

9.5 Shop Inspection

Each gear unit that requires to be certified by 4-3-1/1.1 is to be inspected during manufacture by a Surveyor for conformance with the approved design. This is to include but not limited to checks on gear teeth hardness, and surface finish and dimension checks of main load bearing components The accuracy of meshing is to be verified for all meshes and the initial tooth contact pattern is to be checked by the Surveyor.

Reports on pinions and wheels balancing as per 4-3-1/9.3 are to be made available to the Surveyor for verification.

9.7 Certification of Gears

9.7.1 General

Each gear required to be certified by 4-3-1/1.1 is:

i) To have its design approved by the Bureau; for which purpose, plans and data as required by 4-3-1/1.5 are to be submitted to the Bureau for approval.

ii) To be surveyed during its construction, which is to include, but not limited to, material tests as indicated in 4-3-1/9.1, meshing accuracy and tooth contact pattern checks as indicated in 4-3-1/9.5, and verification of dynamic balancing as indicated in 4-3-1/9.3.

9.7.2 Approval Under the Type Approval Program *(2003)*

9.7.2(a) Product design assessment. Upon application by the manufacturer, each rating of a type of gear may be design assessed as described in 1-1-A3/5.1. For this purpose, each rating of a gear type is to be approved in accordance with 4-3-1/9.7.1i). The type test, however, is to be conducted in accordance with an approved test schedule and is to be witnessed by a Surveyor. Gear so approved may be applied to the Bureau for listing on the ABS website as Products Design Assessed. Once listed, and subject to renewal and updating of certificate as required by 1-1-A3/5.7, gear particulars will not be required to be submitted to the Bureau each time the gear is proposed for use onboard a vessel.

9.7.2(b) Mass produced gears. A manufacturer of mass-produced gears, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.5 (PQA).

Upon satisfactory assessment under 1-1-A3/5.5 (PQA), gears produced in those facilities will not require a Surveyor's attendance at the tests and inspections indicated in 4-3-1/9.7.1ii). Such tests and inspections are to be carried out by the manufacturer whose quality control documents will be accepted. Certification of each gear will be based on verification of approval of the design and on continued effectiveness of the quality assurance system. See 1-1-A3/5.7.1(a).

9.7.2(c) Non-mass Produced Gears. A manufacturer of non-mass produced gears, who operates a quality assurance system in the manufacturing facilities, may apply to the Bureau for quality assurance assessment described in 1-1-A3/5.3.1(a) (AQS) or 1-1-A3/5.3.1(b) (RQS). Certification to 1-1-A3/5.5 (PQA) may also be considered in accordance with 4-1-1/Table 2.

9.7.2(d) Type Approval Program. Gear types which have their ratings approved in accordance with 4-3-1/9.7.2(a) and the quality assurance system of their manufacturing facilities approved in accordance with 4-3-1/9.7.2(b) or 4-3-1/9.7.2(c) will be deemed Type Approved and will be eligible for listing on the ABS website as Type Approved Product.

9.9 Shipboard Trials

After installation on board a vessel, the gear unit is to be operated in the presence of the Surveyor to demonstrate its ability to function satisfactorily under operating conditions and its freedom from harmful vibration at speeds within the operating range. When the propeller is driven through reduction gears, the Surveyor is to ascertain that no gear-tooth chatter occurs throughout the operating range; otherwise, a barred speed range, as specified in 4-3-2/7.5.3, is to be provided.

For conventional propulsion gear units above 1120 kW (1500 hp), a record of gear-tooth contact is to be made at the trials. To facilitate the survey of extent and uniformity of gear-tooth contact, selected bands of pinion or gear teeth on each meshing are to be coated beforehand with copper or layout dye. See also 7-6-2/1.1.2 for the first annual survey after the vessel enters service.

Post trial examination of spur and helical type gears is to indicate essentially uniform contact across 90% of the effective face width of the gear teeth, excluding end relief.

The gear-tooth examination for spur and helical type gear units of 1120 kW (1500 hp) and below, all epicyclical gear units and bevel type gears will be subject to special consideration. The gear manufacturers' recommendations will be considered.

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 1 Appendix 1 – Rating of Cylindrical and Bevel Gears

1 Application

The following calculation procedures cover the rating of external and internal involute spur and helical cylindrical gears having parallel axis, and of bevel gears with regard to surface durability (pitting) and tooth root bending strength.

For normal working pressure angles in excess of 25° or helix angles in excess of 30°, the results obtained from these calculation procedures are to be confirmed by experience data which are to be submitted by the manufacturer.

The influence factors are defined regarding their physical interpretation. Some of the influence factors are determined by the gear geometry or have been established by conventions. These factors are to be calculated in accordance with the equations provided.

Other influence factors, which are approximations, and are indicated as such, may also be calculated according to appropriate alternative methods for which engineering justification is to be provided for verification.

3 Symbols and Units

The main symbols used are listed below. Symbols specifically introduced in connection with the definition of influence factors are described in the appropriate sections.

Units of calculations are given in the sequence of SI units (MKS units, and US units.)

a	center distance	mm (in.)
b	common facewidth	mm (in.)
b_1, b_2	facewidth of pinion, wheel	mm (in.)
b_{eH}	effective facewidth	mm (in.)
b_s	web thickness	mm (in.)
b_B	facewidth of one helix on a double helical gear	mm (in.)
d_1, d_2	reference diameter of pinion, wheel	mm (in.)
d_{a1}, d_{a2}	tip diameter of pinion, wheel (refer to 4-3-1A1/Figure 5)	mm (in.)
d_{b1}, d_{b2}	base diameter of pinion, wheel (refer to 4-3-1A1/Figure 5)	mm (in.)
d_{f1}, d_{f2}	root diameter of pinion, wheel (refer to 4-3-1A1/Figure 5)	mm (in.)
d_{i1}, d_{i2}	rim inside diameter of pinion, wheel (refer to 4-3-1A1/Figure 5)	mm (in.)
d_{sh}	external diameter of shaft	mm (in.)
d_{shi}	internal diameter of hollow shaft	mm (in.)
d_{w1}, d_{w2}	working pitch diameter of pinion, wheel	mm (in.)
f_{fa1}, f_{fa2}	profile form deviation of pinion, wheel	mm (in.)
f_{pb1}, f_{pb2}	transverse base pitch deviation of pinion, wheel	mm (in.)
h_1, h_2	tooth depth of pinion, wheel	mm (in.)
h_{a1}, h_{a2}	addendum of pinion, wheel	mm (in.)
h_{a01}, h_{a02}	addendum of tool of pinion, wheel	mm (in.)

h_{f1}, h_{f2}	dedendum of pinion, wheel	mm (in.)
h_{fp1}, h_{fp2}	dedendum of basic rack of pinion, wheel	mm (in.)
h_{F1}, h_{F2}	bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact for pinion , wheel	mm (in.)
ℓ	bearing span	mm (in.)
ℓ_b	length of contact	mm (in.)
m_n	normal module	mm (in.)
m_{na}	outer normal module	mm (in.)
m_t	transverse module	mm (in.)
n_1, n_2	rotational speed of pinion, wheel	rpm
p_d	outer diametral pitch	mm^{-1} (in^{-1})
p_{r01}, p_{r02}	protuberance of tool for pinion, wheel	mm (in.)
q_1, q_2	machining allowance of pinion, wheel	mm (in.)
s_{Fn1}, s_{Fn2}	tooth root chord in the critical section of pinion, wheel	mm (in.)
s	distance between mid-plane of pinion and the middle of the bearing span	mm (in.)
u	gear ratio	---
v	tangential speed at reference diameter	m/s (m/s, ft/min)
x_1, x_2	addendum modification coefficient of pinion, wheel	---
x_{sm}	tooth thickness modification coefficient (midface)	---
z_1, z_2	number of teeth of pinion, wheel	---
z_{n1}, z_{n2}	virtual number of teeth of pinion, wheel	---
B	total facewidth, of double helical gear including gap width	mm (in.)
F_{bt}	nominal tangential load on base cylinder in the transverse section	N (kgf, lbf)
F_t	nominal transverse tangential load at reference cylinder	N (kgf, lbf)
P	transmitted rated power	kW (mhp, hp)
Q	ISO grade of accuracy	-
R	cone distance	mm (in.)
R_m	middle cone distance	mm (in.)
R_{zf1}, R_{zf2}	flank roughness of pinion, wheel	μm (μin.)
R_{zr1}, R_{zr2}	fillet roughness of pinion. wheel	μm (μin.)
T_1, T_2	nominal torque of pinion, wheel	N-m (kgf-m, lbf-ft)
U	minimum ultimate tensile strength of core (applicable to through hardened, nomalized and cast gears only)	N/mm^2 (kgf/mm^2, lbf/in^2)
α_{en1}, α_{en2}	form-factor pressure angle: pressure angle at the outer point of single pair tooth contact for pinion, wheel	degrees
α_{Fen1}, α_{Fen2}	load direction angle: relevant to direction of application of load at the outer point of single pair tooth contact of pinion, wheel	degrees
α_n	normal pressure angle at reference cylinder	degrees
α_t	transverse pressure angle at reference cylinder	degrees
α_{vt}	transverse pressure angle of virtual cylindrical gear	degrees
α_{wt}	transverse pressure angle at working pitch cylinder	degrees
β	helix angle at reference cylinder	degrees
β_b	helix angle at base cylinder	degrees
β_{vb}	helix angle at base circle	degrees
δ_1, δ_2	reference cone angle of pinion, wheel	degrees
δ_{a1}, δ_{a2}	tip angle of pinion, wheel	degrees
ε_α	transverse contact ratio	---
ε_β	overlap ratio	---
ε_γ	total contact ratio	---
ρ_{a01}, ρ_{a02}	tip radius of tool of pinion, wheel	mm (in.)
ρ_c	radius of curvature at pitch surface	mm (in.)
ρ_{fp}	root radius of basic rack	mm (in.)
ρ_{F1}, ρ_{F2}	root fillet radius at the critical section of pinion, wheel	mm (in.)
Subscripts	1 = pinion; 2 = wheel; 0 = tool	

5 Geometrical Definitions

For internal gearing z_2, a, d_2, d_{a2}, d_{b2}, d_{w2} and u are negative.

The pinion is defined as the gear with the smaller number of teeth. Therefore the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity:

$$u = z_2/z_1 = d_{w2}/d_{w1} = d_2/d_1$$

In the equation of surface durability, b is the common facewidth on the pitch diameter.

In the equation of tooth root bending stress, b_1 or b_2 are the facewidths at the respective tooth roots. In any case, b_1 and b_2 are not to be taken as greater than b by more than one normal module m_n on either side.

The common facewidth b may be used also in the equation of teeth root bending stress if significant crowning or end relief have been applied.

Additional geometrical definitions are given in the following expressions.

$$\tan\alpha_t = \tan\alpha_n/\cos\beta$$

$$\tan\beta_b = \tan\beta \cdot \cos\alpha_t$$

$$d_{1,2} = z_{1,2} \cdot m_n/\cos\beta$$

$$d_{b1,2} = d_{1,2} \cdot \cos\alpha_t = d_{w1,2} \cdot \cos\alpha_{tw}$$

$$a = 0.5(d_{w1} + d_{w2})$$

$$z_{n1,2} = z_{1,2}/(\cos^2\beta_b \cdot \cos\beta)$$

$$m_t = m_n/\cos\beta$$

$$\mathrm{inv}\alpha = \tan\alpha - \pi \cdot \alpha/180,\ \alpha \text{ in degrees}$$

$$\mathrm{inv}\alpha_{wt} = \mathrm{inv}\alpha_t + 2 \cdot \tan\alpha_n \cdot \frac{(x_1 + x_2)}{(z_1 + z_2)}$$

$$\alpha_{wt} = \arccos\left(\frac{d_{b1} + d_{b2}}{2 \cdot a}\right),\ \alpha_{wt} \text{ in degrees}$$

$$x_1 + x_2 = \frac{(z_1 + z_2)\cdot(\mathrm{inv}\,\alpha_{wt} - \mathrm{inv}\,\alpha_t)}{2 \cdot \tan\alpha_n}$$

$$x_1 = \frac{h_{a0}}{m_n} - \frac{d_1 - d_{f1}}{2 \cdot m_n},\quad x_2 = \frac{h_{a0}}{m_n} - \frac{d_2 - d_{f2}}{2 \cdot m_n}$$

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

(The positive sign is to be used for external gears; the negative sign for internal gears.)

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

(For double helix, b is to be taken as the width of one helix.)

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

$$\rho_c = \frac{a \cdot \sin\alpha_{wt} \cdot u}{\pi \cdot m_n}$$

$v = d_{1,2} \cdot n_{1,2}/19099$ [SI and MKS units]

$v = d_{1,2} \cdot n_{1,2}/3.82$ [US units]

7 Bevel Gear Conversion and Specific Formulas *(2006)*

Conversion of bevel gears to virtual (equivalent) cylindrical gears is based on the bevel gear midsection.

Index *v* refers to the virtual (equivalent) cylindrical gear.

Index *m* refers to the midsection of bevel gear.

δ_1, δ_2 = pitch angle pinion, wheel

δ_{a1}, δ_{a2} = face angle pinion, wheel

Σ = shaft angle

β_m = mean spiral angle

$d_{e1,2}$ = outer pitch diameter pinion, wheel

$d_{m1,2}$ = mean pitch diameter pinion, wheel

$d_{v1,2}$ = reference diameter of virtual cylindrical gear pinion, wheel

$R_{e1,2}$ = outer cone distance pinion, wheel

R_m = mean cone distance

Number of teeth of virtual cylindrical gear:

$$z_{v1} = \frac{z_1}{\cos\delta_1}$$

$$z_{v2} = \frac{z_2}{\cos\delta_2}$$

- For $\Sigma = 90°$:

$$z_{v1} = z_1 \frac{\sqrt{u^2+1}}{u}$$

$$z_{v2} = z_2\sqrt{u^2+1}$$

Gear ratio of virtual cylindrical gear:

$$u_v = \frac{z_{v2}}{z_{v1}}$$

- For $\Sigma = 90°$:

$$u_v = u^2$$

Geometrical definitions:

$$\delta_1 + \delta_2 = \Sigma$$

$$\tan\alpha_{vt} = \frac{\tan\alpha_n}{\cos\beta_m}$$

$$\tan\beta_{bm} = \tan\beta_m \cdot \cos\alpha_{vt}$$

$$\beta_{vb} = \arcsin(\sin\beta_m \cdot \cos\alpha_n)$$

$$R_e = \frac{d_{e1,2}}{2\cdot\sin\delta_{1,2}}$$

$$R_m = R_e - \frac{b}{2}\ ,\quad b \le \frac{R}{3}$$

Reference diameter of pinion, wheel refers to the midsection of the bevel gear:

$$d_{m1} = d_{e1} - b \cdot \sin\delta_1$$

$$d_{m2} = d_{e2} - b \cdot \sin\delta_2$$

Modules:

Outer transverse module:

$$m_{et} = \frac{d_{e2}}{z_2} = \frac{d_{e1}}{z_1} = \frac{25.4}{p_d}$$

Outer normal module:

$$m_{na} = m_t \cdot \cos\beta_m$$

Mean normal module:

$$m_{mn} = m_{mt} \cdot \cos\beta_m$$

$$m_{mn} = m_{et} \cdot \frac{R_m}{R_e} \cdot \cos\beta_m$$

$$m_{mn} = \frac{d_{m1}}{z_1} \cdot \cos\beta_m$$

$$m_{mn} = \frac{d_{m2}}{z_2} \cdot \cos\beta_m$$

Base pitch:

$$p_{btm} = \frac{\pi \cdot m_{mn} \cdot \cos\alpha_{vt}}{\cos\beta_m}$$

Reference diameter of pinion, wheel refers to the virtual (equivalent) cylindrical gear:

$$d_{v1} = \frac{d_{m1}}{\cos\delta_1}$$

$$d_{v2} = \frac{d_{m2}}{\cos\delta_2}$$

Base diameter of pinion, wheel:

$$d_{vb1} = d_{v1} \cdot \cos\alpha_{vt}$$

$$d_{vb2} = d_{v2} \cdot \cos\alpha_{vt}$$

Center distance of virtual cylindrical gear:

$$a_v = 0.5 \cdot (d_{v1} + d_{v2})$$

Transverse pressure angle of virtual cylindrical gear:

$$\alpha_{vt} = \arccos\left(\frac{d_{vb1} + d_{vb2}}{2 \cdot a_v}\right),\ \alpha_{vt} \text{ in degrees}$$

Mean Addendum:

For gears with constant addendum *Zyklo-Palloid (Klingelnberg)*:

$$h_{am1} = m_{mn} \cdot (1 + x_{hm1})$$

$$h_{am2} = m_{mn} \cdot (1 + x_{hm2})$$

For gears with variable addendum *(Gleason):*

$$h_{am1} = h_{ae1} - \frac{b}{2} \cdot \tan(\delta_{a1} - \delta_1)$$

$$h_{am2} = h_{ae2} - \frac{b}{2} \cdot \tan(\delta_{a2} - \delta_2)$$

where h_{ae} is the outer addendum.

Profile shift coefficients:

$$x_{hm1} = \frac{h_{am1} - h_{am2}}{2 \cdot m_{mn}}$$

$$x_{hm2} = \frac{h_{am2} - h_{am1}}{2 \cdot m_{mn}}$$

Mean Dedendum:

For gears with constant dedendum *Zyklo-Palloid (Klingelnberg):*

$$h_{fp} = (1.25 \ldots 1.30) \cdot m_{mn}$$

$$\rho_{fp} = (0.2 \ldots 0.3) \cdot m_{mn}$$

For gears with variable dedendum (*Gleason):*

$$h_{fm1} = h_{fe1} - \frac{b}{2} \cdot \tan(\delta_{a1} - \delta_1)$$

$$h_{fm2} = h_{fe2} - \frac{b}{2} \cdot \tan(\delta_{a2} - \delta_2)$$

$$h_{f1} = h_{fm1} + x_{hm1} \cdot m_{mn}$$

$$h_{f2} = h_{fm2} + x_{hm2} \cdot m_{mn}$$

where h_{fe} is the outer dedendum and h_{fm} is the mean dedendum.

Tip diameter of pinion, wheel:

$$d_{va1} = d_{v1} + 2 \cdot h_{am1}$$

$$d_{va2} = d_{v2} + 2 \cdot h_{am2}$$

Transverse contact ratio:

$$\varepsilon_{v\alpha} = \frac{0.5 \cdot \sqrt{d_{va1}^2 - d_{vb1}^2} + 0.5 \cdot \sqrt{d_{va2}^2 - d_{vb2}^2} - a_v \cdot \sin a_{vt}}{\pi \cdot m_{mn} \cdot \dfrac{\cos a_{vt}}{\cos \beta_m}}$$

Overlap ratio:

$$\varepsilon_{v\beta} = \frac{b \cdot \sin \beta_m}{\pi \cdot m_{mn}}$$

Modified contact ratio:

$$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha}^2 + \varepsilon_{v\beta}^2}$$

Tangential speed at midsection:

$$v_{mt} = \frac{d_{m1,2} \cdot n_{1,2}}{19098} \quad \text{m/s} \quad \text{[SI and MKS units]}$$

$$v_{mt} = \frac{d_{m1,2} \cdot n_{1,2}}{3.82} \quad \text{ft/min} \quad \text{[US units]}$$

Radius of curvature (normal section):

$$\rho_{vc} = \frac{a_v \cdot \sin\alpha_{vt}}{\cos\beta_{bm}} \cdot \frac{u_v}{(1+u_v)^2}$$

Length of the middle line of contact:

$$\ell_{bm} = \frac{b \cdot \varepsilon_{v\alpha}}{\cos\beta_{vb}} \cdot \frac{\sqrt{\varepsilon_{v\gamma}^2 - [(2-\varepsilon_{v\alpha})\cdot(1-\varepsilon_{v\beta})]^2}}{\varepsilon_{v\gamma}^2} \quad \text{for } \varepsilon_{v\beta} < 1$$

$$\ell_{bm} = \frac{b \cdot \varepsilon_{v\alpha}}{\varepsilon_{v\gamma} \cdot \cos\beta_{vb}} \quad \text{for } \varepsilon_{v\beta} \geq 1$$

9 Nominal Tangential Load, F_t, F_{mt} *(2006)*

The nominal tangential load, F_t or F_{mt}, tangential to the reference cylinder and perpendicular to the relevant axial plane, is calculated directly from the rated power transmitted by the gear by means of the following equations:

	SI units	*MKS units*	*US units*
	$T_{1,2} = 9549 \cdot P/n_{1,2}$ N-m	$T_{1,2} = 716.2 \cdot P/n_{1,2}$ kgf-m	$T_{1,2} = 5252 \cdot P/n_{1,2}$ lbf-ft
Cylindrical gears	$F_t = \frac{2000T_{1,2}}{d_{1,2}} = \frac{19.1P \times 10^6}{n_{1,2}d_{1,2}}$ N	$F_t = \frac{2000T_{1,2}}{d_{1,2}} = \frac{1.4325P \times 10^6}{n_{1,2}d_{1,2}}$ kgf	$F_t = \frac{24T_{1,2}}{d_{1,2}} = \frac{126.05P \times 10^3}{n_{1,2}d_{1,2}}$ lbf
Bevel gears	$F_{mt} = \frac{2000T_{1,2}}{d_{m1,2}} = \frac{19.1P \times 10^6}{n_{1,2}d_{m1,2}}$ N	$F_{mt} = \frac{2000T_{1,2}}{d_{m1,2}} = \frac{1.4325P \times 10^6}{n_{1,2}d_{m1,2}}$ kgf	$F_{mt} = \frac{24T_{1,2}}{d_{m1,2}} = \frac{126.05P \times 10^3}{n_{1,2}d_{m1,2}}$ lbf

Where the vessel on which the gear unit is being used, is receiving an **Ice Class** notation, see 6-1-1/57 or 6-1-2/29.

11 Application Factor, K_A

The application factor, K_A, accounts for dynamic overloads from sources external to the gearing.

The application factor, K_A, for gears designed for infinite life, is defined as the ratio between the maximum repetitive cyclic torque applied to the gear set and the nominal rated torque.

The factor mainly depends on:

- Characteristics of driving and driven machines;
- Ratio of masses;
- Type of couplings;
- Operating conditions as e.g., overspeeds, changes in propeller load conditions.

When operating near a critical speed of the drive system, a careful analysis of these conditions must be made.

The application factor, K_A, should be determined by measurements or by appropriate system analysis. Where a value determined in such a way cannot be provided, the following values are to be used:

a) *Main propulsion gears:*

Turbine and electric drive:	1.00
Diesel engine with hydraulic or electromagnetic slip coupling:	1.00
Diesel engine with high elasticity coupling:	1.30
Diesel engine with other couplings:	1.50

b) *Auxiliary gears:*

Electric motor, diesel engine with hydraulic or electromagnetic slip coupling:	1.00
Diesel engine with high elasticity coupling:	1.20
Diesel engine with other couplings:	1.40

13 Load Sharing Factor, K_γ

The load sharing factor K_γ accounts for the maldistribution of load in multiple path transmissions as e.g., dual tandem, epicyclical, double helix.

The load sharing factor K_γ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor mainly depends on accuracy and flexibility of the branches.

The load sharing factor K_γ should be determined by measurements or by appropriate system analysis. Where a value determined in such a way cannot be provided, the following values are to be used:

a) *Epicyclical gears:*

Up to 3 planetary gears:	1.00
Up to 4 planetary gears:	1.20
Up to 5 planetary gears:	1.30
6 planetary gears and over:	1.40

b) *Other gear arrangements:* 1.00

c) *Bevel gears:*

For $\varepsilon_\gamma \leq 2$:	1.00
For $2 < \varepsilon_\gamma < 3.5$:	$1 + 0.2\sqrt{(\varepsilon_\gamma - 2)\cdot(5 - \varepsilon_\gamma)}$
For $\varepsilon_\gamma \geq 3.5$:	1.30

15 Dynamic Factor, K_v

The dynamic factor, K_v, accounts for internally generated dynamic loads due to vibrations of pinion and wheel against each other.

The dynamic factor, K_v, is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load ($F_t \cdot K_A \cdot K_\gamma$).

The factor mainly depends on:

- Transmission errors (depending on pitch and profile errors)
- Masses of pinion and wheel
- Gear mesh stiffness variation as the gear teeth pass through the meshing cycle
- Transmitted load including application factor
- Pitch line velocity
- Dynamic unbalance of gears and shaft
- Shaft and bearing stiffnesses
- Damping characteristics of the gear system

The dynamic factor, K_v, is to be advised by the manufacturer as supported by his measurements, analysis or experience data or is to be determined as per 4-3-1A1/15.1, except that where $vz_1/100$ is 3 m/s (590 ft/min.) or above, K_v may be obtained from Appendix 4-3-1A1/15.3.

15.1 Determination of K_v – Simplified Method

Where all of the following four conditions are satisfied, K_v may be determined in accordance with Appendix 4-3-1A1/15.1.

a) Steel gears of heavy rims sections.

b) Values of F_t/b are in accordance with the following table:

SI units	*MKS units*	*US units*
> 150 N/mm	> 15 kgf/mm	> 856 lbf/in

c) $z_1 < 50$

d) Running speeds in the subcritical range are in accordance with the following table:

	$(v \cdot z_1) / 100$	
	SI & MKS units	*US units*
Helical gears	< 14 m/s	< 2756 ft/min
Spur gears	< 10 m/s	< 1968 ft/min
All types of gears	< 3 m/s	< 590 ft/min

For gears other than specified above, the single resonance method, as per 4-3-1A1/15.3 below, may be applied.

The methods of calculation are as follows:

	SI and MKS units	*US units*
For helical gears of overlap ratio $\varepsilon_\beta \geq$ unity	$K_v = K_{vh} = 1 + K_1 \cdot v \cdot z_1/100$	$K_v = K_{vh} = 1 + K_1 \cdot v \cdot z_1/19685$
For helical gears of overlap ratio $\varepsilon_\beta <$ unity	K_v is obtained by means of linear interpolation: $K_v = K_{vs} - \varepsilon_\beta(K_{vs} - K_{vh})$	
For spur gears	$K_v = K_{vs} = 1 + K_1 \cdot v \cdot z_1/100$	$K_v = K_{vs} = 1 + K_1 \cdot v \cdot z_1/19685$
For bevel gears	In the above conditions (b, c and d) and in the above formulas: • the real z_1 is to be used instead of the virtual (equivalent) z_{v1}; • v is to be substituted by v_{mt}; (tangential speed at midsection); and • F_t is to be substituted by F_{mt}.	
For all gears	K_1 values are specified in table below.	

	K_1 for different values of Q (ISO Grades of accuracy)					
	3	4	5	6	7	8
Spur gears	0.0220	0.0300	0.0430	0.0620	0.0920	0.1250
Helical gears	0.0125	0.0165	0.0230	0.0330	0.0480	0.0700
Q is to be according to ISO 1328. In case of mating gears with different grades of accuracy, the grade corresponding to the lower accuracy should be used.						

15.3 Determination of K_v – Single Resonance Method

For single stage gears, the dynamic factor K_v may be determined from 4-3-1A1/15.3.3 through 4-3-1A1/15.3.6 for the ratio N in 4-3-1A1/15.3.1 and using the factors given in 4-3-1A1/15.3.2.

15.3.1 Resonance Ratio, N *(2006)*

$$N = \frac{n_1}{n_{E1}}$$

$$n_{E1} = \frac{30 \times 10^3}{\pi \cdot z_1} \cdot \sqrt{\frac{C_\gamma}{m_{red}}} \quad \text{rpm} \quad \text{[SI units]}$$

$$n_{E1} = \frac{93.947 \times 10^3}{\pi \cdot z_1} \cdot \sqrt{\frac{C_\gamma}{m_{red}}} \quad \text{rpm} \quad \text{[MKS units]}$$

$$n_{E1} = \frac{589.474 \times 10^3}{\pi \cdot z_1} \cdot \sqrt{\frac{C_\gamma}{m_{red}}} \quad \text{rpm} \quad \text{[US units]}$$

where:

m_{red} = relative reduced mass of the gear pair, per unit facewidth referred to the line of action:

$$m_{red} = \frac{\pi}{8} \cdot \left(\frac{d_{m1}}{d_{b1}}\right)^2 \cdot \left[\frac{d_{m1}^2}{\left(1/(1-q_1^4) \cdot \rho_1\right) + \left(1/(1-q_2^4) \cdot \rho_2 \cdot u^2\right)}\right] \quad \text{kg/mm (lb/in)}$$

$$m_{red} = \frac{J_1 \cdot J_2}{p \cdot J_1 \cdot \left(\frac{d_{b2}}{2}\right)^2 + J_2 \cdot \left(\frac{d_{b1}}{2}\right)^2} \quad \text{kg/mm (lb/in)}$$

p = for planetary gears, number of planets

$d_{m1,\,m2}$ = $\dfrac{d_{a1,a2} + d_{f1,f2}}{2}$ mm (in.)

$q_1 = \dfrac{d_{i1}}{d_{m1}}$; $q_2 = \dfrac{d_{i2}}{d_{m2}}$ for reference, see 4-3-1A1/Figure 5

J_1 = moment of inertia per unit facewidth for pinion:

$$J_1 = \frac{\pi \cdot \rho_1 \cdot d_{b1}^4}{32}$$ kg-mm^2/mm (lb-in^2/in)

J_2 = moment of inertia per unit facewidth for wheel:

$$J_2 = \frac{\pi \cdot \rho_2 \cdot d_{b2}^4}{32}$$ kg-mm^2/mm (lb-in^2/in)

$\rho_{1,2}$ = density of pinion, wheel materials

ρ = density of steel material:

= 7.83×10^{-6} kg/mm^3 [SI and MKS units]

= 2.83×10^{-1} lb/in^3 [US units]

Bevel gears:

For bevel gears, the real z_1 (not the equivalent) should be inserted in the above formulas. Determination of m_{red} is to be as follows:

$$m_{1,2}^* = \frac{\pi}{8} \cdot \rho_{1,2} \cdot \left[\frac{d_{m1,2}^2}{\cos^2 \alpha_n}\right]$$ kg/mm (lb/in.)

$$m_{red} = \frac{m_1^* \cdot m_2^*}{m_1^* + m_2^*}$$ kg/mm (lb/in.)

Mesh stiffness per unit facewidth, C_γ:

$$C_\gamma = \frac{20}{0.85} \cdot B_b$$ N/mm-μm [SI units]

$$C_\gamma = \frac{2.039}{0.85} \cdot B_b$$ kgf/mm-μm [MKS units]

$$C_\gamma = \frac{2.901}{0.85} \cdot B_b$$ lbf/in-μin [US units]

Tooth stiffness of one pair of teeth per unit facewidth (single stiffness), c':

$$c' = \frac{14}{0.85} \cdot B_b$$ N/mm-μm [SI units]

$$c' = \frac{1.428}{0.85} \cdot B_b$$ kgf/mm-μm [MKS units]

$$c' = \frac{2.031}{0.85} \cdot B_b$$ lbf/in-μin [US units]

For a combination of different materials for pinion and wheel, c' is to be multiplied by ξ, where

$$\xi = \frac{E}{E_{st}}$$

$$E = \frac{2E_1E_2}{E_1 + E_2}$$ where values of E_1 and E_2 are to be obtained from 4-3-1A1/Table 1

E_{st} = Young's modulus of steel; see 4-3-1A1/Table 1

Overall facewidth, B_b:

$$B_b = \frac{b_{eH}}{b}$$

where: b_{eH} = effective facewidth, mm (in.)

Note: Higher values than B_b = 0.85 are not to be used.

Cylindrical gears:

Mesh stiffness per unit facewidth, C_γ:

$C_\gamma = c' \cdot (0.75 \cdot \varepsilon_\alpha + 0.25)$N/mm-μm (kgf/mm-μm, lbf/in-μin)

Tooth stiffness of one pair of teeth per unit facewidth (single stiffness), c':

$$c' = 0.8 \cdot \frac{\cos\beta}{q'} \cdot C_{BS} \cdot C_R$$ N/mm-μm [SI units]

$$c' = 8.158 \cdot 10^{-2} \cdot \frac{\cos\beta}{q'} \cdot C_{BS} \cdot C_R$$ kgf/mm-μm [MKS units]

$$c' = 11.603 \cdot 10^{-2} \cdot \frac{\cos\beta}{q'} \cdot C_{BS} \cdot C_R$$ lbf/in-μin [US units]

For a combination of different materials for pinion and wheel, c' is to be multiplied by ξ, where

$$\xi = \frac{E}{E_{st}}$$

$$E = \frac{2 \cdot E_1 \cdot E_2}{E_1 + E_2}$$ where values of E_1 and E_2 are to be obtained from 4-3-1A1/Table 1

E_{st} = Young's modulus of steel; see 4-3-1A1/Table 1

$$q' = 0.04723 + \frac{0.15551}{z_{n1}} + \frac{0.25791}{z_{n2}} - 0.00635 \cdot x_1 - \frac{0.11654 \cdot x_1}{z_{n1}} - 0.00193 \cdot x_2 - \frac{0.24188 \cdot x_2}{z_{n2}} + 0.00529 \cdot x_1^2 + 0.00182 \cdot x_2^2$$

$$z_{n1} = \frac{z_1}{\cos^2\beta_b \cdot \cos\beta}$$

$$z_{n2} = \frac{z_2}{\cos^2\beta_b \cdot \cos\beta}$$

Note: For internal gears, use z_{n2} (= ∞) equal infinite and x_2 = 0.

$$C_{BS} = \left[1 + 0.5 \cdot \left(1.2 - \frac{h_{fp}}{m_n}\right)\right] \cdot [1 - 0.02 \cdot (20 - \alpha_n)]$$

When the pinion basic rack dedendum is different from that of the wheel, the arithmetic mean of C_{BS1} for a gear pair conjugate to the pinion basic rack and C_{BS2} for a gear pair conjugate to the basic rack of the wheel:

$$C_{BS} = 0.5 \cdot (C_{BS1} + C_{BS2})$$

Gear blank factor, C_R:

$$C_R = 1 + \frac{\ln(b_s / b)}{5 \cdot e^{(s_R / 5 \cdot m_n)}}$$

when: 1) $b_s/b < 0.2$, use 0.2 for b_s/b

2) $b_s/b > 1.2$, use 1.2 for b_s/b

3) $s_R/m_n < 1$, use 1.0 for s_R/m_n

For b_s and s_R see 4-3-1A1/Figure 3.

15.3.2 Factors B_p, B_j, and B_k *(2006)*

a) Values for f_{pt} and y_a according to ISO grades of accuracy Q:

	Q (1)	3	4 (3)	5	6	7	8	9	10	11 (4)	12
f_{pt}	μm	3	6	12	25	45	70	100	150	201	282
	μin	118	236	472	984	1772	2756	3937	5906	7913	11102
y_a (2)	μm	0	0.5	1.5	4	7	15	25	40	55	75
	μin	0	19.7	59.1	157	276	591	984	1575	2165	2953

Notes

1 ISO grades of accuracy according to ISO 1328. In case of mating gears with different grades of accuracy, the grade corresponding to the lower accuracy should be used.

2 For specific determination of y_a, see b) through d) below.

3 Hardened nitrided.

4 Tempered normalized.

b) Determination of y_a for structural steels, through hardened steels and nodular cast iron (perlite, bainite):

SI units	*MKS units*	*US units*
$y_a = \frac{160}{\sigma_{H\lim}} \cdot f_{pb}$	$y_a = \frac{16.315}{\sigma_{H\lim}} \cdot f_{pb}$	$y_a = \frac{2.331 \times 10^4}{\sigma_{H\lim}} \cdot f_{pb}$
For $v \le 5$ m/s: no restriction	For $v \le 5$ m/s: no restriction	For $v \le 984$ ft/min: no restriction
For 5 m/s < $v \le 10$ m/s: $y_{a\max} = \frac{12800}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 80 μm	For 5 m/s < $v \le 10$ m/s: $y_{a\max} = \frac{1.305 \times 10^3}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 80 μm	For 984 ft/min < $v \le 1968$ ft/min: $y_{a\max} = \frac{7.341 \times 10^7}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 3150 μin
For $v > 10$ m/s: $y_{a\max} = \frac{6400}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 40 μm	For $v > 10$ m/s: $y_{a\max} = \frac{652.618}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 40 μm	For $v > 1968$ ft/min: $y_{a\max} = \frac{3.67 \times 10^7}{\sigma_{H\lim}}$ and $f_{pb\max}$ = 1575 μin

c) Determination of y_a for gray cast iron and nodular cast iron (ferritic):

SI and MKS units	*US units*
$y_a = 0.275 \cdot f_{pb}$ μm	$y_a = 0.275 \cdot f_{pb}$ μin
For $v \le 5$ m/s: no restriction	For $v \le 984$ ft/min: no restriction
For 5 m/s < $v \le 10$ m/s: $y_{a\max}$ = 22 and $f_{pb\max}$ = 80 μm	For 984 ft/min < $v \le 1968$ ft/min: $y_{a\max}$ = 866 and $f_{pb\max}$ = 3150 μin
For $v > 10$ m/s : $y_{a\max}$ = 11 and $f_{pb\max}$ = 40 μm	For $v > 1968$ ft/min $y_{a\max}$ = 433 and $f_{pb\max}$ = 1575 μin

d) Determination of y_a for case hardened, nitrided or nitrocarburized steels:

SI and MKS units	*US units*
$y_a = 0.075 \cdot f_{pb}$ μm	$y_a = 0.075 \cdot f_{pb}$ μin
For all velocities but with the restriction: $y_{a\max}$ = 3 and $f_{pb\max}$ = 40 μm	For all velocities but with the restriction: $y_{a\max}$ = 118 and $f_{pb\max}$ = 1575 μin

When the material of pinion differs from that of the wheel, y_{a1} for pinion and y_{a2} for wheel are to be determined separately. The mean value:

$$y_a = 0.5(y_{a1} + y_{a2})$$

is to be used for the calculation.

For bevel gears, f_{pt} is substituted for f_{pb} when determining y_a in *b), c)* and *d).*

e) Determination of factors B_p, B_f, B_k

$$B_p = \frac{c' \cdot f_{pbeff} \cdot b}{F_t \cdot K_A \cdot K_\gamma}$$

$$B_f = \frac{c' \cdot f_{feff} \cdot b}{F_t \cdot K_A \cdot K_\gamma}$$

where

$f_{pbeff} = f_{pb} - y_a$ μm (μin)

$f_{feff} = f_{fa} - y_f$ μm (μin)

$f_{pb} = f_{pt} \cdot \cos\alpha_t$ μm (μin)

y_f can be determined in the same way as y_a when the profile deviation f_{fa} is used instead of the base pitch deviation f_{pb}.

$$B_k = \left|1 - \frac{C_a}{C_{eff}}\right|$$

where:

SI units [μm]	MKS units [μm]	US units [μin]
$C_a = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim}}{97} - 18.45\right)^2 + 1.5$	$C_a = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim}}{9.891} - 18.45\right)^2 + 1.5$	$C_a = \frac{1}{0.457} \cdot \left(\frac{\sigma_{H\lim}}{1.407 \times 10^4} - 18.45\right)^2 + 59$

When the materials differ, C_{a1} should be determined for the pinion material and C_{a2} for the wheel material using the following equations. The average value $C_a = \frac{C_{a1} + C_{a2}}{2}$ is used for the calculation.

$C_{a1} = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim 1}}{97} - 18.45\right)^2 + 1.5$	$C_{a1} = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim 1}}{9.891} - 18.45\right)^2 + 1.5$	$C_{a1} = \frac{1}{0.457} \cdot \left(\frac{\sigma_{H\lim 1}}{1.407 \times 10^4} - 18.45\right)^2 + 59$
$C_{a2} = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim 2}}{97} - 18.45\right)^2 + 1.5$	$C_{a2} = \frac{1}{18} \cdot \left(\frac{\sigma_{H\lim 2}}{9.891} - 18.45\right)^2 + 1.5$	$C_{a2} = \frac{1}{0.457} \cdot \left(\frac{\sigma_{H\lim 2}}{1.407 \times 10^4} - 18.45\right)^2 + 59$

For cylindrical gears:

$$C_{eff} = \frac{F_t \cdot K_A \cdot K_\gamma}{b \cdot c'}$$ μm (μin)

For bevel gears:

$$C_{eff} = \frac{F_{mbt} \cdot K_A}{b_{eH} \cdot c'}$$ μm (μin)

15.3.3 Dynamic Factor, K_v, in the Subcritical Range *(2006)*

Cylindrical gears: ($N \le 0.85$)

C_{v1} = 0.32

C_{v2} = 0.34 for $\varepsilon_\gamma \le 2$

$$C_{v2} = \frac{0.57}{\varepsilon_\gamma - 0.3} \quad \text{for } \varepsilon_\gamma > 2$$

C_{v3} = 0.23 for $\varepsilon_\gamma \le 2$

$$C_{v3} = \frac{0.096}{\varepsilon_\gamma - 1.56} \quad \text{for } \varepsilon_\gamma > 2$$

$K = (C_{v1} \cdot B_p) + (C_{v2} \cdot B_f) + (C_{v3} \cdot B_k)$

Bevel gears: ($N \le 0.75$)

For bevel gears, $\varepsilon_{v\gamma}$ are to be substituted for ε_γ

$$K = \frac{b \cdot f_{peff} \cdot c'}{F_{mt} \cdot K_A} \cdot c_{v1,2} + c_{v3}$$

$f_{peff} = f_{pt} - y_p$ with $y_p \approx y_a$

$c_{v1,2} = c_{v1} + c_{v2}$

$K_v = (N \cdot K) + 1$

15.3.4 Dynamic Factor, K_v, in the Main Resonance Range *(2006)*

C_{v4} = 0.90 for $\varepsilon_\gamma \le 2$

$$C_{v4} = \frac{0.57 - 0.05 \cdot \varepsilon_\gamma}{\varepsilon_\gamma - 1.44} \quad \text{for } \varepsilon_\gamma > 2$$

Cylindrical gears: ($0.85 < N \le 1.15$)

$K_{v(N=1.15)} = (C_{v1} \cdot B_p) + (C_{v2} \cdot B_f) + (C_{v4} \cdot B_k) + 1$

Bevel gears: ($0.75 < N \le 1.25$)

For bevel gears, $\varepsilon_{v\gamma}$ are to be substituted for ε_γ

$$K_{V(N=1.25)} = \frac{b \cdot f_{peff} \cdot c'}{F_{mt} \cdot K_A} \cdot c_{v1,2} + c_{v4} + 1$$

For C_{v1}, C_{v2}, $C_{v1,2}$, and f_{peff}, see 4-3-1A1/15.3.3 above.

15.3.5 Dynamic Factor, K_v, in the Supercritical Range ($N \ge 1.5$) *(2006)*

C_{v5} = 0.47

C_{v6} = 0.47 for $\varepsilon_\gamma \le 2$

$$C_{v6} = \frac{0.12}{\varepsilon_\gamma - 1.74} \quad \text{for } \varepsilon_\gamma > 2$$

C_{v7} = 0.75 for $1.0 < \varepsilon_\gamma \le 1.5$

$C_{v7} = 0.125 \cdot \sin[\pi(\varepsilon_\gamma - 2)] + 0.875$ for $1.5 < \varepsilon_\gamma \le 2.5$

C_{v7} = 1.0 for $\varepsilon_\gamma > 2.5$

Cylindrical gears:

$$K_{v(N=1.5)} = (C_{v5} \cdot B_p) + (C_{v6} \cdot B_f) + C_{v7}$$

Bevel gears:

For bevel gears, $\varepsilon_{v\gamma}$ are to be substituted for ε_γ.

$$K_{V(N=1.5)} = \frac{b \cdot f_{peff} \cdot c'}{F_{mt} \cdot K_A} \cdot c_{v5,6} + c_{v7}$$

$$c_{v5,6} = c_{v5} + c_{v6}$$

For f_{peff}, see 4-3-1A1/15.3.3 above.

15.3.6 Dynamic Factor, K_v, in the Intermediate Range *(2006)*

Cylindrical gears:

In this range, the dynamic factor is determined by linear interpolation between K_v at N = 1.15 as specified in 4-3-1A1/15.3.4 and K_v at N = 1.5 as specified in 4-3-1A1/15.3.5.

$$K_v = K_{v(N=1.5)} + \frac{K_{v(N=1.15)} - K_{v(N=1.5)}}{0.35} \cdot (1.5 - N)$$

Bevel gears:

In this range, the dynamic factor is determined by linear interpolation between K_v at N = 1.25 as specified in 4-3-1A1/15.3.4 and K_v at N = 1.5 as specified in 4-3-1A1/15.3.5.

$$K_v = K_{v(N=1.5)} + \frac{K_{v(N=1.25)} - K_{v(N=1.5)}}{0.25} \cdot (1.5 - N)$$

17 Face Load Distribution Factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress and $K_{F\beta}$ for tooth root bending stress, account for the effects of non-uniform distribution of load across the facewidth.

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

$$K_{H\beta} = \frac{\text{maximum load per unit facewidth}}{\text{mean load per unit facewidth}}$$

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

$$K_{F\beta} = \frac{\text{maximum bending stress at tooth root per unit facewidth}}{\text{mean bending stress at tooth root per unit facewidth}}$$

Note: The mean bending stress at tooth root relates to the considered facewidth b_1 or b_2

$K_{F\beta}$ can be expressed as a function of the factor $K_{H\beta}$.

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

- Gear tooth manufacturing accuracy
- Errors in mounting due to bore errors
- Bearing clearances
- Wheel and pinion shaft alignment errors
- Elastic deflections of gear elements, shafts, bearings, housing and foundations which support the gear elements

- Thermal expansion and distortion due to operating temperature
- Compensating design elements (tooth crowning, end relief, etc.)

These factors can be obtained from 4-3-1A1/17.3 and 4-3-1A1/17.5, using the factors in 4-3-1A1/17.1.

17.1 Factors Used for the Determination of $K_{H\beta}$

17.1.1 Helix Deviation F_β *(2003)*

The helix deviation, F_β, is to be determined by the designer or by the following equations and tables:

SI and MKS units, μm	*US units, μin.*
$F_\beta = 0.1\sqrt{d_{geom}} + 0.63\sqrt{b_{geom}} + 4.2$ for ISO Grade of accuracy $Q = 5$	$F_\beta = 19.8\sqrt{d_{geom}} + 125.0\sqrt{b_{geom}} + 165.4$ for ISO Grade of accuracy $Q = 5$
For other accuracy grades, multiply F_β by the following formula: $2^{0.5(Q-5)}$ where $0 \le Q \le 12$	

SI and MKS units		*US units*	
Reference diameter d mm	*Corresponding geometric mean diameter d_{geom} mm*	*Reference diameter d in*	*Corresponding geometric mean diameter d_{geom} mm*
$5 \le d \le 20$	10.00	$0.2 \le d \le 0.79$	0.3937
$20 < d \le 50$	31.62	$0.79 \le d \le 2.0$	1.245
$50 < d \le 125$	79.06	$2.0 \le d \le 4.92$	3.112
$125 < d \le 280$	187.1	$4.92 \le d \le 11.0$	7.365
$280 < d \le 560$	396.0	$11.0 \le d \le 22.0$	15.59
$560 < d \le 1000$	748.3	$22.0 \le d \le 39.37$	29.46
$1000 < d \le 1600$	1265	$39.37 \le d \le 62.99$	49.80
$1600 < d \le 2500$	2000	$62.99 \le d \le 98.43$	78.74
$2500 < d \le 4000$	3162	$98.43 \le d \le 157.5$	124.5
$4000 < d \le 6000$	4899	$157.5 \le d \le 236.2$	192.9
$6000 < d \le 8000$	6928	$236.2 \le d \le 315.0$	272.8
$8000 < d \le 10000$	8944	$315.0 \le d \le 393.70$	352.1

SI and MKS units		*US units*	
Facewidth b mm	*Corresponding geometric mean facewidth b_{geom} mm*	*Facewidth b in*	*Corresponding geometric mean facewidth b_{geom} in.*
$4 < b \le 10$	6.325	$0.16 < b \le 0.39$	0.2490
$10 < b \le 20$	14.14	$0.39 < b \le 0.79$	0.5568
$20 < b \le 40$	28.28	$0.79 < b \le 1.6$	1.114
$40 < b \le 80$	56.57	$1.6 < b \le 3.15$	2.227
$80 < b \le 160$	113.1	$3.15 < b \le 6.30$	4.454
$160 < b \le 250$	200.0	$6.30 < b \le 9.84$	7.874
$250 < b \le 400$	316.2	$9.84 < b \le 15.7$	12.45
$400 < b \le 650$	509.9	$15.7 < b \le 25.6$	20.07
$650 < b \le 1000$	806.2	$25.6 < b \le 39.37$	31.74

Rounding Rules			
For resulting F_β	$F_\beta < 5$ μm	5 μm $\le F_\beta \le$ 10 μm	$F_\beta >$ 10 μm (μin)
	Round to the nearest 0.1 μm value or integer number	Round to the nearest 0.5 μm value or integer number	Round to the nearest integer number

17.1.2 Mesh Alignment f_{ma} [μm (μin)]

Generally: $f_{ma} = 1.0 \cdot F_\beta$.

For gear pairs with well designed end relief: $f_{ma} = 0.7 \cdot F_\beta$.

For gear pairs with provision for adjustment (lapping or running-in under light load, adjustment bearings or appropriate helix modification) and gear pairs suitably crowned: $f_{ma} = 0.5 \cdot F_\beta$.

For helix deviation due to manufacturing inaccuracies: $f_{ma} = 0.5 \cdot F_\beta$.

17.1.3 Initial Equivalent Misalignment $F_{\beta\chi}$ [μm (μin)]

$F_{\beta\chi}$ is the absolute value of the sum of manufacturing deviations and pinion and shaft deflections, measured in the plane of action.

$$F_{\beta\chi} = 1.33 \cdot f_{sh} + f_{ma}$$

$$f_{sh} = f_{sh0} \cdot \frac{F_t \cdot K_A \cdot K_\gamma \cdot K_v}{b}$$

$f_{sh0\min} = 0.005$ μm-mm/N [SI units]

$f_{sh0\min} = 0.049$ μm-mm/kgf [MKS units]

$f_{sh0\min} = 0.03445$ μin-in/lbf [US units]

	SI units [*μm-mm/N*]	*MKS units* [*μm-mm/kgf*]	*US units* [*μin-in/lbf*]
For spur and helical gears without crowning or end relief	$f_{sh0} = 0.023 \cdot \gamma$	$f_{sh0} = 0.22555 \cdot \gamma$	$f_{sh0} = 0.15858 \cdot \gamma$
For spur and helical gears without crowning but with end relief	$f_{sh0} = 0.016 \cdot \gamma$	$f_{sh0} = 0.15691 \cdot \gamma$	$f_{sh0} = 0.11032 \cdot \gamma$
For spur and helical gears with crowning	$f_{sh0} = 0.012 \cdot \gamma$	$f_{sh0} = 0.11768 \cdot \gamma$	$f_{sh0} = 0.08274 \cdot \gamma$
For spur and helical gears with crowning and end relief	$f_{sh0} = 0.010 \cdot \gamma$	$f_{sh0} = 0.09807 \cdot \gamma$	$f_{sh0} = 0.06895 \cdot \gamma$

$$\gamma = \left[\left|1 + K' \cdot \frac{\ell \cdot s}{d_1^{\,2}} \cdot \left(\frac{d_1}{d_{sh}}\right)^4 - 0.3\right| + 0.3\right] \cdot \left(\frac{b}{d_1}\right)^2$$ for spur and single helical gears

$$\gamma = 2 \cdot \left[\left|1.5 + K' \cdot \frac{\ell \cdot s}{d_1^{\,2}} \cdot \left(\frac{d_1}{d_{sh}}\right)^4 - 0.3\right| + 0.3\right] \cdot \left(\frac{b_B}{d_1}\right)^2$$ for double helical gears

where $b_B = b/2$ is the width of one helix.

The constant K' makes allowances for the position of the pinion in relation to the torqued end. It can be taken from 4-3-1A1/Table 6.

17.1.4 Determination of y_β and χ_β for Structural Steels, Through Hardened Steels and Nodular Cast Iron (Perlite, Bainite)

SI units	*MKS units*	*US units*
$y_\beta = \dfrac{320}{\sigma_{H\lim}} \cdot F_{\beta\chi}$ μm	$y_\beta = \dfrac{32.63}{\sigma_{H\lim}} \cdot F_{\beta\chi}$ μm	$y_\beta = \dfrac{4.662 \cdot 10^4}{\sigma_{H\lim}} \cdot F_{\beta\chi}$ μin
$\chi_\beta = 1 - \dfrac{320}{\sigma_{H\lim}}$ with $y_\beta \le F_{\beta\chi}$; $\chi_\beta \ge 0$	$\chi_\beta = 1 - \dfrac{32.63}{\sigma_{H\lim}}$ with $y_\beta \le F_{\beta\chi}$; $\chi_\beta \ge 0$	$\chi_\beta = 1 - \dfrac{4.662 \cdot 10^4}{\sigma_{H\lim}}$ with $y_\beta \le F_{\beta\chi}$; $\chi_\beta \ge 0$
For $v \le 5$ m/s no restriction	For $v \le 5$ m/s no restriction	For $v \le 984$ ft/min no restriction
For 5 m/s < $v \le 10$ m/s: $y_{\beta\max} = \dfrac{25600}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 80$ μm	For 5 m/s < $v \le 10$ m/s: $y_{\beta\max} = \dfrac{2.61 \cdot 10^3}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 80$ μm	For 984 ft/min < $v \le 1968$ ft/min: $y_{\beta\max} = \dfrac{14.682 \cdot 10^7}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 80$ μm
For $v > 10$ m/s: $y_{\beta\max} = \dfrac{12800}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 40$ μm	For $v > 10$ m/s: $y_{\beta\max} = \dfrac{1.305 \cdot 10^3}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 40$ μm	For $v > 1968$ ft/min: $y_{\beta\max} = \dfrac{7.341 \cdot 10^7}{\sigma_{H\lim}}$ corresponding to $F_{\beta\chi} = 40$ μm

For $\sigma_{H\lim}$, see 4-3-1A1/Table 3.

17.1.5 Determination of y_β and χ_β for Gray Cast Iron and Nodular Cast Iron (Ferritic):

SI & MKS units, μm	US units, μin
$Y_\beta = 0.55 \cdot F_{\beta\chi}$	$Y_\beta = 0.55 \cdot F_{\beta\chi}$
$\chi_\beta = 0.45$	$\chi_\beta = 0.45$
For $v \le 5$ m/s, no restriction	For $v \le 984$ ft/min, no restriction
For 5 m/s < $v \le 10$ m/s: $y_{\beta\max} = 45$ corresponding to $F_{\beta\chi} = 80$ μm	For 984 ft/min < $v \le 1968$ ft/min: $y_{\beta\max} = 1771$ corresponding to $F_{\beta\chi} = 3150$ μin
For $v > 10$ m/s: $y_{\beta\max} = 22$ corresponding to $F_{\beta\chi} = 40$ μm	For $v > 1968$ ft/min: $y_{\beta\max} = 866$ corresponding to $F_{\beta\chi} = 1575$ μin

17.1.6 Determination of y_β and χ_β for Case Hardened, Nitrided or Nitrocarburized Steels*:*

SI & MKS units	*US units*
$Y_\beta = 0.15 \cdot F_{\beta\chi}$ μm	$Y_\beta = 0.15 \cdot F_{\beta\chi}$ μin
$\chi_\beta = 0.85$	$\chi_\beta = 0.85$
For all velocities but with the restriction: $y_{\beta\max} = 6$ corresponding to $F_{\beta\chi} = 40$ μm	For all velocities but with the restriction: $y_{\beta\max} = 236$ corresponding to $F_{\beta\chi} = 1575$ μin

When the material of the pinion differs from that of the wheel, $y_{\beta 1}$ and $\chi_{\beta 1}$ for pinion, and $y_{\beta 2}$ and $\chi_{\beta 2}$ for wheel are to be determined separately. The mean of either value:

$$y_\beta = 0.5 \cdot (y_{\beta 1} + y_{\beta 2})$$

$$\chi_\beta = 0.5 \cdot (\chi_{\beta 1} + \chi_{\beta 2})$$

is to be used for the calculation.

17.3 Face Load Distribution Factor for Contact Stress $K_{H\beta}$

17.3.1 $K_{H\beta}$ for Helical and Spur Gears

$$K_{H\beta} = 1 + \frac{b \cdot F_{\beta y} \cdot C\gamma}{2 \cdot F_t \cdot K_A \cdot K_\gamma \cdot K_v} < 2, \text{ for } \frac{b \cdot F_{\beta y} \cdot C\gamma}{2 \cdot F_t \cdot K_A \cdot K_\gamma \cdot K_v} < 1$$

$$K_{H\beta} = \sqrt{\frac{2 \cdot b \cdot F_{\beta y} \cdot C\gamma}{F_t \cdot K_A \cdot K_\gamma \cdot K_v}} \geq 2 \text{ for } \frac{b \cdot F_{\beta y} \cdot C\gamma}{2 \cdot F_t \cdot K_A \cdot K_\gamma \cdot K_v} \geq 1$$

where:

$$F_{\beta y} = F_{\beta x} - y_\beta \qquad \text{or} \qquad F_{\beta y} = F_{\beta x} \cdot \chi_\beta$$

Calculated values of $K_{H\beta} \geq 2$ are to be reduced by improvement accuracy and helix deviation.

17.3.2 $K_{H\beta}$ for Bevel Gears

$$K_{H\beta} = 1.5 \cdot \frac{0.85}{B_b} \cdot K_{H\beta be}$$

The bearing factor, $K_{H\beta be}$, representing the influence of the bearing arrangement on the faceload distribution, is given in the following table:

Mounting conditions of pinion and wheel		
Both members straddle mounted	One member straddle mounted	Neither member straddle mounted
1.10	1.25	1.50
Based on optimum tooth contact pattern under maximum operating load as evidenced by results of a deflection test on the gears in their mountings.		

17.5 Face Load Distribution Factor for Tooth Root Bending Stress $K_{F\beta}$

17.5.1 In Case the Hardest Contact is at the End of the Face-width $K_{F\beta}$ is Given by the Following Equations

$$K_{F\beta} = (K_{H\beta})^N$$

$$N = \frac{(b/h)^2}{1 + (b/h) + (b/h)^2} = \frac{1}{1 + (h/b) + (h/b)^2}$$

(b/h) = (facewidth/tooth depth), the lesser of b_1/h_1 or b_2/h_2. For double helical gears, the facewidth of only one helix is to be used, i.e., $b_B = b/2$ is to be substituted for b in the equation for N.

17.5.2 In Case of Gears Where the Ends of the Facewidth are Lightly Loaded or Unloaded (End Relief or Crowning)

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

17.5.3 Bevel Gears

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

$$K_{FO} = 0.211 \cdot \left(\frac{r_{eo}}{R_m}\right)^q + 0.789 \quad \text{for spiral bevel gears.}$$

$$q = \frac{0.279}{\log(\sin \beta_m)}$$

where

$K_{FO} = 1$ for straight or zero bevel gears.

r_{eo} = cutter radius, mm (in.)

R_m = mean cone distance, mm (in.)

Limitations of K_{FO}:

If $K_{FO} <$ unity, use K_{FO} = unity

If $K_{FO} > 1.15$, use $K_{FO} = 1.15$

19 Transverse Load Distribution Factors, $K_{H\alpha}$ and $K_{F\alpha}$

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, account for the effects of pitch and profile errors on the transversal load distribution between two or more pairs of teeth in mesh.

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

- Total mesh stiffness
- Total tangential load F_t, K_A, K_γ, K_v, $K_{H\beta}$
- Base pitch error
- Tip relief
- Running-in allowances

The load distribution factors, $K_{H\alpha}$ and $K_{F\alpha}$ are to be advised by the manufacturer as supported by his measurements, analysis or experience data or are to be determined as follows.

19.1 Determination of $K_{H\alpha}$ for Contact Stress $K_{F\alpha}$ for Tooth Root Bending Stress *(2006)*

$$K_{H\alpha} = K_{F\alpha} = 0.9 + 0.4 \cdot \sqrt{\frac{2 \cdot (\varepsilon_\gamma - 1)}{\varepsilon_\gamma}} \cdot \frac{C_\gamma \cdot (f_{pbe} - y_a) \cdot b}{F_{tH}} \quad \text{for } \varepsilon_\gamma > 2$$

$$K_{H\alpha} = K_{F\alpha} = \frac{\varepsilon_\gamma}{2} \cdot \left[0.9 + 0.4 \cdot \frac{C_\gamma \cdot (f_{pbe} - y_a) \cdot b}{F_{tH}}\right] \quad \text{for } \varepsilon_\gamma \le 2$$

Cylindrical gears:

$F_{tH} = F_t \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{H\beta}$ N (kgf, lbf)

$f_{pbe} = f_{pb} \cdot \cos \alpha_t$

Bevel gears:

$F_{mtH} = F_{mt} \cdot K_A \cdot K_v \cdot K_{H\beta}$ N (kgf, lbf)

For bevel gears, f_{pt}, $\varepsilon_{v\gamma}$, F_{mtH}, F_{mt} and α_{vt} (equivalent) are to be substituted for f_{pbe}, ε_γ, F_{tH}, F_t and α_t in the above formulas.

19.3 Limitations of $K_{H\alpha}$ and $K_{F\alpha}$

19.3.1 $K_{H\alpha}$ *(2006)*

When $K_{H\alpha} < 1$, use 1.0 for $K_{H\alpha}$

Cylindrical gears:

When $K_{H\alpha} > \dfrac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Z_\varepsilon^2}$, use $\dfrac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Z_\varepsilon^2}$ for $K_{H\alpha}$;

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3} \cdot (1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}},$$ contact ratio factor (pitting) for helical gears for $\varepsilon_\beta < 1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}},$$ contact ratio factor (pitting) for helical gears for $\varepsilon_\beta \geq 1$

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}},$$ contact ratio factor (pitting) for spur gears

Bevel gears:

When $K_{H\alpha} > \dfrac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} \cdot Z_{LS}^2}$, use $\dfrac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} \cdot Z_{LS}^2}$ for $K_{H\alpha}$.

For the calculation of Z_{LS}, see 4-3-1A1/21.13.

19.3.2 $K_{F\alpha}$ *(2006)*

When $K_{F\alpha} < 1$, use 1.0 for $K_{F\alpha}$.

When $K_{F\alpha} > \dfrac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Y_\varepsilon}$, use $\dfrac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Y_\varepsilon}$ for $K_{F\alpha}$;

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha},$$ contact ratio factor for $\varepsilon_\beta = 0$

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha} - \left(\frac{0.75}{\varepsilon_\alpha} - 0.375\right) \cdot \varepsilon_\beta,$$ contact ratio factor for $0 < \varepsilon_\beta < 1$

$Y_\varepsilon = 0.625$, contact ratio factor for $\varepsilon_\beta \geq 1$

or:

$$Y_\varepsilon = 0.25 + \frac{0.75 \cdot \cos^2 \beta_b}{\varepsilon_\alpha}$$ for cylindrical gears only

For bevel gears, $\varepsilon_{v\gamma}$, $\varepsilon_{v\beta}$, and $\varepsilon_{v\alpha}$, (equivalent) are to be substituted for ε_γ, ε_β, and ε_α, in the above formulas.

21 Surface Durability

The criterion for surface durability is based on the Hertzian pressure on the operating pitch point or at the inner point of single pair contact. The contact stress σ_H is not to exceed the permissible contact stress σ_{HP}.

21.1 Contact Stress *(2006)*

$$\sigma_{H1} = \sigma_{HO1} \cdot \sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \le \sigma_{HP1}$$

$$\sigma_{H2} = \sigma_{HO2} \cdot \sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \le \sigma_{HP2}$$

Cylindrical gears:

$\sigma_{HO1,2}$ = basic value of contact stress for pinion and wheel

$$\sigma_{HO1} = Z_B \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \qquad \text{for pinion}$$

$$\sigma_{HO2} = Z_D \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \qquad \text{for wheel}$$

where

Z_B = single pair mesh factor for pinion, see 4-3-1A1/21.5 below

Z_D = single pair mesh factor for wheel, see 4-3-1A1/21.5 below

Z_H = zone factor, see 4-3-1A1/21.7 below

Z_E = elasticity factor, see 4-3-1A1/21.9 below

Z_ε = contact ratio factor (pitting), see 4-3-1A1/21.11 below

Z_β = helix angle factor, see 4-3-1A1/21.17 below

F_t = nominal transverse tangential load, see 4-3-1A1/9 of this Appendix.

Gear ratio u for external gears is positive, for internal gears, u is negative.

Regarding factors K_A, K_γ, K_v, $K_{H\alpha}$ and $K_{H\beta}$, see 4-3-1A1/11, 4-3-1A1/13, 4-3-1A1/15, 4-3-1A1/19 and 4-3-1A1/17 of this Appendix.

Bevel gears:

σ_{HO1} = basic value of contact stress for pinion

$$\sigma_{HO1} = Z_{M\text{-}B} \cdot Z_H \cdot Z_E \cdot Z_{LS} \cdot Z_\beta \cdot Z_K \cdot \sqrt{\frac{F_{mt}}{d_{v1} \cdot \ell_{bm}} \cdot \frac{u_v + 1}{u_v}}$$

For the shaft angle $\Sigma = \delta_1 + \delta_2 = 90°$ the following applies:

$$\sigma_{HO1} = Z_{M\text{-}B} \cdot Z_H \cdot Z_E \cdot Z_{LS} \cdot Z_\beta \cdot Z_K \cdot \sqrt{\frac{F_{mt}}{d_{m1} \cdot \ell_{bm}} \cdot \frac{\sqrt{u^2+1}}{u}}$$

where

$Z_{M\text{-}B}$ = mid-zone factor, see 4-3-1A1/21.5 below

Z_H = zone factor, see 4-3-1A1/21.7 below

Z_E = elasticity factor, see 4-3-1A1/21.9 below

Z_{LS} = load sharing factor, see 4-3-1A1/21.13 below

Z_{β} = helix angle factor, see 4-3-1A1/21.17 below

Z_K = bevel gear factor (flank), see 4-3-1A1/21.15 below

F_{mt} = nominal transverse tangential load, see 4-3-1A1/9 of this appendix.

d_{m1} = mean pitch diameter of pinion of bevel gear

d_{v1} = reference diameter of pinion of virtual (equivalent) cylindrical gear

ℓ_{bm} = length of middle line of contact

u_v = gear ratio of virtual (equivalent) cylindrical gear

u = gear ratio of bevel gear

21.3 Permissible Contact Stress

The permissible contact stress, σ_{HP}, is to be evaluated separately for pinion and wheel.

$$\sigma_{HP} = \frac{\sigma_{H\lim}}{S_H} \cdot Z_N \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X \qquad \text{N/mm}^2 \text{ (kgf/mm}^2\text{, psi)}$$

where:

$\sigma_{H\lim}$ = endurance limit for contact stress, see 4-3-1A1/21.19 below

Z_N = life factor for contact stress, see 4-3-1A1/21.21 below

Z_L = lubrication factor, see 4-3-1A1/21.23 below

Z_V = speed factor, see 4-3-1A1/21.23 below

Z_R = roughness factor, see 4-3-1A1/21.23 below

Z_W = hardness ratio factor, see 4-3-1A1/21.25 below

Z_X = size factor for contact stress, see 4-3-1A1/21.27 below

S_H = safety factor for contact stress, see 4-3-1A1/21.29 below

For shrink-fitted wheel rims, σ_{HP} is to be at least K_S times the mean contact stress σ_H, where

K_S = safety factor available for induced contact stresses, and is to be calculated as follows:

$$1 + \frac{\delta_{\max} \cdot 2.2 \times 10^5}{Y} \cdot \frac{d_{ri}}{{d_{w2}}^2} \cdot \frac{0.25 m_n}{\rho_{F2}} \qquad \text{[SI units]}$$

$$1 + \frac{\delta_{\max} \cdot 2.243 \times 10^4}{Y} \cdot \frac{d_{ri}}{{d_{w2}}^2} \cdot \frac{0.25 m_n}{\rho_{F2}} \qquad \text{[MKS units]}$$

$$K_S = 1 + \frac{\delta_{\max} \cdot 3.194 \times 10^7}{Y} \cdot \frac{d_{ri}}{{d_{w2}}^2} \cdot \frac{0.25 m_n}{\rho_{F2}} \qquad \text{[US units]}$$

where

$\delta_{\max}$ = maximum available interference fit or maximum pull-up length; mm (in.)

d_{ri} = inner diameter of wheel rim; mm (in.)

d_{w2} = working pitch diameter of wheel; mm (in.)

m_n = normal module; mm (in.)

Y = yield strength of wheel rim material is to be as follows:

- minimum specified yield strength for through hardened (quenched and tempered) steels
- 500 N/mm^2 (51 kgf/mm^2, 72520 psi) for case hardened, nitrided steels

21.5 Single Pair Mesh Factors, Z_B, Z_D and Mid-zone Factor Z_{M-B}

The single pair mesh factors, Z_B for pinion and Z_D for wheel, account for the influence on contact stresses of the tooth flank curvature at the inner point of single pair contact in relation to Z_H.

The factors transform the contact stresses determined at the pitch point to contact stresses considering the flank curvature at the inner point of single pair contact.

21.5.1 For Cylindrical and Bevel Gears when $\varepsilon_\beta = 0$ *(2006)*

$Z_B = Z_{M-B} = M_1$ or 1, whichever is the larger value

$Z_D = Z_{M-B} = M_2$ or 1, whichever is the larger value

$$M_1 = \frac{\tan\alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a1}/d_{b1})^2 - 1} - (2\pi/z_1)\right]\cdot\left[\sqrt{(d_{a2}/d_{b2})^2 - 1} - (\varepsilon_\alpha - 1)\cdot(2\pi/z_2)\right]}} \quad \text{for cylindrical gears}$$

$$M_2 = \frac{\tan\alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a2}/d_{b2})^2 - 1} - (2\pi/z_2)\right]\cdot\left[\sqrt{(d_{a1}/d_{b1})^2 - 1} - (\varepsilon_\alpha - 1)\cdot(2\pi/z_1)\right]}} \quad \text{for cylindrical gears}$$

For bevel gears, α_{wt}, d_a, d_b, ε_a and z are to be substituted by α_{vt}, d_{va}, d_{vb}, ε_{va} and z_v, respectively, in the above formulas.

21.5.2 For Cylindrical and Bevel Gears when $\varepsilon_\beta \geq 1$

Z_B = $Z_D = 1$ for cylindrical gears

Z_{M-B} = M or 1, whichever is the larger value, for bevel gears

$$M = \frac{\tan\alpha_{vt}}{\sqrt{\left[\sqrt{(d_{va1}/d_{vb1})^2 - 1} - \varepsilon_\alpha\cdot(\pi/z_{v1})\right]\cdot\left[\sqrt{(d_{va2}/d_{vb2})^2 - 1} - \varepsilon_\alpha\cdot(\pi/z_{v2})\right]}}$$

21.5.3 For Cylindrical and Bevel Gears when $0 < \varepsilon_\beta < 1$

Cylindrical gears:

The values of Z_B, Z_A are determined by linear interpolation between Z_B, Z_A for spur gears and Z_B, Z_A for helical gears having $\varepsilon_\beta < 1$

Thus:

Z_B = $M_1 - \varepsilon_\beta(M_1 - 1)$ and $Z_B \geq 1$

Z_D = $M_2 - \varepsilon_\beta(M_2 - 1)$ and $Z_D \geq 1$

Bevel gears:

Z_{M-B} = M or 1, whichever is the larger value

where:

$$M = \frac{\tan\alpha_{vt}}{\sqrt{\left[\sqrt{(d_{va1}/d_{vb1})^2 - 1} - \left(2 + (\varepsilon_\alpha - 2)\cdot\varepsilon_\beta\right)\cdot(\pi/z_{v1})\right]\cdot\left[\sqrt{(d_{va2}/d_{vb2})^2 - 1} - \left(2(\varepsilon_\alpha - 1) + (2 - \varepsilon_\alpha)\cdot\varepsilon_\beta\right)\cdot(\pi/z_{v2})\right]}}$$

21.7 Zone Factor, Z_H *(2006)*

Cylindrical gears:

The zone factor, Z_H, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and relates the tangential force at the reference cylinder to the normal force at the pitch cylinder.

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

Bevel gears:

$$Z_H = 2 \cdot \sqrt{\frac{\cos\beta_{vb}}{\sin(2 \cdot \alpha_{vt})}}$$

21.9 Elasticity Factor, Z_E

The elasticity factor, Z_E, accounts for the influence of the material properties E (modulus of elasticity) and ν (Poisson's ratio) on the Hertzian pressure.

$$Z_E = \sqrt{\frac{1}{\pi \cdot \left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}}$$

With Poisson's ratio of 0.3 and same E and ν for pinion and wheel, Z_E may be obtained from the following:

$$Z_E = \sqrt{0.175 \cdot E}$$

with E = Young's modulus of elasticity.

The elasticity factor, Z_E, for steel gears [E_{st} = 206000 N/mm^2 (2.101×10^4 kgf/mm^2, 2.988×10^7 psi) is:

Elasticity factor Z_E		
SI units	*MKS units*	*US units*
189.8 N$^{1/2}$/mm	60.61 kgf$^{1/2}$/mm	2.286 × 10^3 lbf$^{1/2}$/in

For other material combinations, refer to 4-3-1A1/Table 1.

21.11 Contact Ratio Factor (Pitting), Z_ε

The contact ratio factor, Z_ε, accounts for the influence of the transverse contact ratio and the overlap ratio on the specific surface load of gears.

Spur gears:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}}$$

Helical gears:

For $\varepsilon_\beta < 1$: $Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3} \cdot (1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}$

For $\varepsilon_\beta \geq 1$: $Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}}$

21.13 Bevel Gear Load Sharing Factor, Z_{LS} *(2006)*

The load sharing factor, Z_{LS}, accounts for the load sharing between two or more pair of teeth in contact.

For $\varepsilon_{v\gamma} \leq 2$: $Z_{LS} = 1$

For $\varepsilon_{v\gamma} > 2$ and $\varepsilon_{v\beta} > 1$: $Z_{LS} = \left[1 + 2 \cdot \left\{1 - (2/\varepsilon_{v\gamma})^{1.5}\right\} \cdot \sqrt{1 - (4/\varepsilon_{v\gamma}^2)}\right]^{-0.5}$

For other cases: The calculation of Z_{LS} can be based upon the method used in ISO 10300-2, Annex A, Load Sharing Factor, Z_{LS}.

21.15 Bevel Gear Factor (Flank), Z_K

The bevel gear factor (flank), Z_K, accounts the difference between bevel and cylindrical loading and adjusts the contact stresses so that the same permissible stresses may apply.

$Z_K = 0.8$

21.17 Helix Angle Factor, Z_β *(1 July 2009)*

The helix angle factor, Z_β, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Z_β is dependent only on the helix angle.

Cylindrical gears: $Z_\beta = \dfrac{1}{\sqrt{\cos\beta}}$

Bevel gears: $Z_\beta = \dfrac{1}{\sqrt{\cos\beta_m}}$

21.19 Allowable Stress Number (Contact), $\sigma_{H\lim}$

For a given material, $\sigma_{H\lim}$ is the limit of repeated contact stress that can be sustained without progressive pitting. For most materials, their load cycles may be taken at 50×10^6, unless otherwise specified.

For this purpose, pitting is defined as follows:

- Not surface hardened gears: pitted area > 2% of total active flank area.
- Surface hardened gears: pitted area > 0.5% of total active flank area, or > 4% of one particular tooth flank area.

The endurance limit depends mainly on:

- Material composition, cleanliness and defects
- Mechanical properties
- Residual stresses
- Hardening process, depth of hardened zone, hardness gradient
- Material structure (forged, rolled bar, cast)

The $\sigma_{H\lim}$ values correspond to a failure probability of 1% or less. The values of $\sigma_{H\lim}$ are to be determined from 4-3-1A1/Table 3 or to be advised by the manufacturer together with technical justification for the proposed values.

21.21 Life Factor, Z_N

The life factor, Z_N, accounts for the higher permissible contact stress, including static stress in case a limited life (number of load cycles) is specified.

The factor depends mainly on:

- Material and hardening
- Number of cycles
- Influence factors (Z_R, Z_V, Z_L, Z_W, Z_X)

The life factor, Z_N, can be determined from 4-3-1A1/Table 4**.**

21.23 Influence Factors on Lubrication Film, Z_L, Z_V and Z_R

The lubricant factor, Z_L, accounts for the influence of the type of lubricant and its viscosity.

The speed factor, Z_V, accounts for the influence of the pitch line velocity.

The roughness factor, Z_R, accounts for the influence of the surface roughness on the surface endurance capacity.

The factors mainly depend on:

- Viscosity of lubricant in the contact zone
- The sum of the instantaneous velocities of the tooth surfaces
- Load
- Relative radius of curvature at the pitch point
- Surface roughness of teeth flanks
- Hardness of pinion and gear

Where gear pairs are of different hardness, the factors may be based on the less hardened material.

21.23.1 Lubricant Factor, Z_L

$$Z_L = C_{ZL} + \frac{4 \cdot (1.0 - C_{ZL})}{\left[1.2 + (134/\nu_{40})\right]^2} \qquad \text{[SI and MKS units]}$$

$$Z_L = C_{ZL} + \frac{4 \cdot (1.0 - C_{ZL})}{\left[1.2 + (0.208/\nu_{40})\right]^2} \qquad \text{[US units]}$$

or

$$Z_L = C_{ZL} + \frac{4 \cdot (1.0 - C_{ZL})}{\left[1.2 + (80/\nu_{50})\right]^2} \qquad \text{[SI and MKS units]}$$

$$Z_L = C_{ZL} + \frac{4 \cdot (1.0 - C_{ZL})}{\left[1.2 + (0.127/\nu_{50})\right]^2} \qquad \text{[US units]}$$

where $\sigma_{H\lim}$ is the allowable stress number (contact) of the softer material.

a) with $\sigma_{H\lim}$ in the range of:

850 N/mm^2 (87 kgf/mm^2, 1.23 × 10^5 psi) < $\sigma_{H\lim}$ < 1200 N/mm^2 (122 kgf/mm^2, 1.74 × 10^5 psi)

$$C_{ZL} = \left(0.08 \cdot \frac{\sigma_{H\lim} - 850}{350}\right) + 0.83 \qquad \text{[SI units]}$$

$$C_{ZL} = \left(0.08 \cdot \frac{\sigma_{H\lim} - 87}{35.7}\right) + 0.83 \qquad \text{[MKS units]}$$

$$C_{ZL} = \left(0.08 \cdot \frac{\sigma_{H\lim} - 1.23 \cdot 10^5}{5.076 \times 10^4}\right) + 0.83 \qquad \text{[US units]}$$

b) with $\sigma_{H\lim}$ in the range of $\sigma_{H\lim}$ < 850 N/mm² (87 kgf/mm², 1.23 × 10⁵ psi), C_{ZL} = 0.83;

c) with $\sigma_{H\lim}$ in the range of $\sigma_{H\lim}$ > 1200 N/mm² (122 kgf/mm², 1.74 × 10⁵ psi), C_{ZL} = 0.91.

and

ν_{40} = nominal kinematic viscosity of the oil at 40°C (104°F), mm²/s; see table below.

ν_{50} = nominal kinematic viscosity of the oil at 50°C (122°F), mm²/s; see table below.

ISO lubricant viscosity grade		VG 32 (1)	VG 46 (1)	VG 68 (1)	VG 100	VG 150	VG 220	VG 320
SI/MKS units	average viscosity ν_{40} mm²/s	32	46	68	100	150	220	320
	average viscosity ν_{50} mm²/s	21	30	43	61	89	125	180
US units	average viscosity ν_{40} in²/s	0.0496	0.0713	0.1054	0.1550	0.2325	0.3410	0.4960
	average viscosity ν_{50} in²/s	0.0326	0.0465	0.0667	0.0945	0.1380	0.1938	0.2790
1) Only for high speed (> 1400 rpm) transmission.								

21.23.2 Speed Factor, Z_V *(2006)*

$$Z_V = C_{Zv} + \frac{2(1.0 - C_{Zv})}{\sqrt{0.8 + (32/v)}} \qquad \text{[SI and MKS units]}$$

$$Z_V = C_{Zv} + \frac{2(1.0 - C_{Zv})}{\sqrt{0.8 + (6.4 \cdot 10^3/v)}} \qquad \text{[US units]}$$

where $\sigma_{H\lim}$ is the allowable stress number (contact) of the softer material.

For bevel gears, v is to be substituted by v_{mt} in the above formula.

a) with $\sigma_{H\lim}$ in the range of:

850 N/mm² (87 kgf/mm², 1.23 × 10⁵ psi) < $\sigma_{H\lim}$ < 1200 N/mm² (122 kgf/mm², 1.74 × 10⁵ psi)

$$C_{Zv} = \left(0.08 \frac{\sigma_{H\lim} - 850}{350}\right) + 0.85 \qquad \text{[SI units]}$$

$$C_{Zv} = \left(0.08 \frac{\sigma_{H\lim} - 87}{35.7}\right) + 0.85 \qquad \text{[MKS units]}$$

$$C_{Zv} = \left(0.08 \frac{\sigma_{H\lim} - 1.23 \times 10^5}{5.076 \times 10^4}\right) + 0.85 \qquad \text{[US units]}$$

b) with $\sigma_{H\lim}$ < 850 N/mm² (87 kgf/mm², 1.23 × 10⁵ psi), C_{Zv} = 0.85.

c) with $\sigma_{H\lim}$ > 1200 N/mm² (122 kgf/mm², 1.74 × 10⁵ psi), C_{Zv} = 0.93.

21.23.3 Roughness Factor, Z_R

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

The peak-to-valley roughness, R_Z, is to be advised by the manufacturer or to be determined as a mean value of R_Z, measured on several tooth flanks of the pinion and the gear, as given by the following expression:

$$R_Z = \frac{R_{Zf1} + R_{Zf2}}{2}$$

Where roughness values are not available, roughness of the pinion R_{Zf1} = 6.3 μm (248 μin) and of the wheel R_{Zf2} = 6.3 μm (248 μin) may be used.

R_{Z10} is to be given by:

$$R_{Z10} = R_Z \sqrt[3]{\frac{10}{\rho_{red}}} \qquad \text{[SI and MKS units]}$$

$$R_{Z10} = R_Z \sqrt[3]{\frac{6.4516 \cdot 10^{-6}}{\rho_{red}}} \qquad \text{[US units]}$$

and the relative radius of curvature is to be given by:

$$\rho_{red} = \frac{\rho_1 \cdot \rho_2}{\rho_1 + \rho_2} \qquad \text{for cylindrical gears}$$

$$\rho_{red} = \frac{\rho_{v1} \cdot \rho_{v2}}{\rho_{v1} + \rho_{v2}} \qquad \text{for bevel gears}$$

ρ_1 = $0.5 \cdot d_{b1} \cdot \tan\alpha_{tw}$

ρ_{v1} = $0.5 \cdot d_{vb1} \cdot \tan\alpha_{tw}$

ρ_2 = $0.5 \cdot d_{b2} \cdot \tan\alpha_{tw}$

ρ_{v2} = $0.5 \cdot d_{vb2} \cdot \tan\alpha_{tw}$

If the stated roughness is an R_a value, also known as arithmetic average (*AA*) and centerline average (*CLA*), the following approximate relationship may be applied:

R_a = $CLA = AA = R_{Zf} / 6$

Where R_{Zf} is either R_{Zf1} for pinion or R_{Zf2} for gear and $\sigma_{H\lim}$ is the allowable stress number (contact) of the softer material.

In the range of 850 N/mm² ≤ $\sigma_{H\lim}$ ≤ 1200 N/mm² (87 kgf/mm² ≤ $\sigma_{H\lim}$ ≤ 122 kgf/mm²; 1.23×10^5 psi ≤ $\sigma_{H\lim}$ ≤ 1.74×10^5 psi) , C_{ZR} can be calculated as follows:

C_{ZR} = $0.32 - 2.00 \times 10^{-4} \cdot \sigma_{H\lim}$ [SI units]

C_{ZR} = $0.32 - 1.96 \times 10^{-3} \cdot \sigma_{H\lim}$ [MKS units]

C_{ZR} = $0.32 - 1.38 \times 10^{-6} \cdot \sigma_{H\lim}$ [US units]

If $\sigma_{H\lim}$ < 850 N/mm² (87 kgf/mm², 1.23×10^5 psi), take C_{ZR} = 0.150

If $\sigma_{H\lim}$ > 1200 N/mm² (122 kgf/mm², 1.74×10^5 psi), take C_{ZR} = 0.080

21.25 Hardness Ratio Factor; Z_W

The hardness ratio factor, Z_W, accounts for the increase of surface durability of a soft steel gear when meshing with a surface hardened gear with a smooth surface.

The hardness ratio factor, Z_W, applies to the soft gear only and depends mainly on:

- Hardness of the soft gear
- Alloying elements of the soft gear
- Tooth flank roughness of the harder gear

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

where

HB = Brinell hardness of the softer material

$HV10$ = Vickers hardness with F = 98.1 N

For unalloyed steels

$HB \approx HV10 \approx U / 3.6$ [SI units]

$HB \approx HV10 \approx U / 0.367$ [MKS units]

$HB \approx HV10 \approx U / 522$ [US units]

For alloyed steels

$HB \approx HV10 \approx U / 3.4$ [SI units]

$HB \approx HV10 \approx U / 0.347$ [MKS units]

$HB \approx HV10 \approx U / 493$ [US units]

For $HB < 130$, $Z_W = 1.2$ is to be used.

For $HB > 470$, $Z_W = 1.0$ is to be used.

21.27 Size Factor, Z_X

The size factor, Z_X , accounts for the influence of tooth dimensions on permissible contact stress and reflects the non-uniformity of material properties.

The factor mainly depends on:

- Material and heat treatment
- Tooth and gear dimensions
- Ratio of case depth to tooth size
- Ratio of case depth to equivalent radius of curvature

For through-hardened gears and for surface-hardened gears with minimum required effective case depth including root of 1.14 mm (0.045 in.) relative to tooth size and radius curvature $Z_X = 1$. When the case depth is relatively shallow, then a smaller value of Z_X should be chosen.

The size factors, Z_X, are to be obtained from 4-3-1A1/Table 2.

21.29 Safety Factor for Contact Stress, S_H

Based on the application, the following safety factors for contact stress, S_H , are to be applied:

Main propulsion gears (including PTO):	1.40
Duplicated (or more) independent main propulsion gears (including azimuthing thrusters):	1.25
Main propulsion gears for yachts, single screw:	1.25
Main propulsion gears for yachts, multiple screw:	1.20
Auxiliary gears:	1.15

Note: For the above purposes, yachts are considered pleasure craft not engaged in trade or carrying passengers, and not intended for charter-service.

23 Tooth Root Bending Strength

The criterion for tooth root bending strength is the permissible limit of local tensile strength in the root fillet. The tooth root stress, σ_F, and the permissible tooth root stress, σ_{FP}, are to be calculated separately for the pinion and the wheel, whereby σ_F is not to exceed the permissible tooth root stress σ_{FP}.

The following formulas apply to gears having a rim thickness greater than 3.5 m and further for all involute basic rack profiles, with or without protuberance, however, with the following restrictions:

- The 30° tangents contact the tooth-root curve generated by the basic rack of the tool
- The basic rack of the tool has a root radius $\rho_{fp} > 0$
- The gear teeth are generated using a rack type tool.

23.1 Tooth Root Bending Stress for Pinion and Wheel

Cylindrical gears:

$$\sigma_{F1,2} = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta} \leq \sigma_{FP1,2} \quad \text{N/mm}^2, \text{kgf/mm}^2, \text{psi}$$

Bevel gears:

$$\sigma_{F1,2} = \frac{F_{mt}}{b \cdot m_{mn}} \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\varepsilon \cdot Y_K \cdot Y_{LS} \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta} \leq \sigma_{FP1,2} \quad \text{N/mm}^2, \text{kgf/mm}^2, \text{psi}$$

where

Y_F, Y_{Fa}	=	tooth form factor, see 4-3-1A1/23.5 below
Y_S, Y_{Sa}	=	stress correction factor, see 4-3-1A1/23.7 below
Y_β	=	helix angle factor, see 4-3-1A1/23.9 below
Y_ε	=	contact ratio factor, see 4-3-1A1/23.11 below
Y_K	=	bevel gear factor, see 4-3-1A1/23.13 below
Y_{LS}	=	load sharing factor, see 4-3-1A1/23.15 below

F_t, F_{mt}, K_A, K_γ, K_v, $K_{F\alpha}$, $K_{F\beta}$, b, m_n, m_{mn}, see 4-3-1A1/9, 4-3-1A1/11, 4-3-1A1/13, 4-3-1A1/15, 4-3-1A1/19, 4-3-1A1/17, 4-3-1A1/5, and 4-3-1A1/7 of this Appendix, respectively.

23.3 Permissible Tooth Root Bending Stress

$$\sigma_{FP1,2} = \frac{\sigma_{FE}}{S_F} \cdot Y_d \cdot Y_N \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X \quad \text{N/mm}^2, \text{kgf/mm}^2, \text{psi}$$

where

σ_{FE}	=	bending endurance limit, see 4-3-1A1/23.17 below
Y_d	=	design factor, see 4-3-1A1/23.19 below
Y_N	=	life factor, see 4-3-1A1/23.21 below
$Y_{\delta relT}$	=	relative notch sensitivity factor, see 4-3-1A1/23.23 below
Y_{RrelT}	=	relative surface factor, see 4-3-1A1/23.25 below
Y_X	=	size factor, see 4-3-1A1/23.27 below
S_F	=	safety factor for tooth root bending stress, see 4-3-1A1/23.29 below

23.5 Tooth Form Factor, Y_F, Y_{Fa}

The tooth form factors, Y_F and Y_{Fa}, represent the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact.

The tooth form factors, Y_F and Y_{Fa}, are to be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing are to be determined in the normal section, i.e., for the virtual spur gear with virtual number of teeth, z.

Cylindrical gears:

$$Y_F = \frac{6 \cdot (h_F / m_n) \cdot \cos \alpha_{Fen}}{(s_{Fn}/m_n)^2 \cdot \cos \alpha_n}$$

Bevel gears:

$$Y_{Fa} = \frac{6 \cdot (h_{Fa} / m_{mn}) \cdot \cos \alpha_{Fan}}{(s_{Fn}/m_{mn})^2 \cdot \cos \alpha_n}$$

where

h_F, h_{Fa} = bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact; mm (in.)

s_{Fn} = width of tooth at highest stressed section; mm (in.)

α_{Fen}, α_{Fan} = normal load pressure angle at tip of tooth; degrees

For determination of h_F, h_{Fa}, s_{Fn} and α_{Fen}, α_{Fan}, see 4-3-1A1/23.5.1, 4-3-1A1/23.5.2, 4-3-1A1/23.5.3 below and 4-3-1A1/Figure 1.

23.5.1 External Gears

Width of tooth, s_{Fn}, at tooth-root normal chord:

$$\frac{s_{Fn}}{m_n} = z_n \cdot \sin\left(\frac{\pi}{3} - \vartheta\right) + \sqrt{3} \cdot \left(\frac{G}{\cos \vartheta} - \frac{\rho_{a0}}{m_n}\right)$$

$$\vartheta = 2 \cdot \frac{G}{z_n} \cdot \tan \vartheta - H \quad \text{degrees; to be solved iteratively}$$

$$G = \frac{\rho_{a0}}{m_n} - \frac{h_{a0}}{m_n} + x$$

$$H = \frac{2}{z_n} \cdot \left(\frac{\pi}{2} - \frac{E}{m_n}\right) - \frac{\pi}{3} \quad \text{[SI and MKS units]}$$

$$H = \frac{2}{z_n} \cdot \left(\frac{\pi}{2} - \frac{E}{25.4 \cdot m_n}\right) - \frac{\pi}{3} \quad \text{[US units]}$$

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

$$\beta_b = \arccos \sqrt{1 - (\sin \beta \cdot \cos \alpha_n)^2} \quad \text{degrees}$$

$$E = \frac{\pi}{4} \cdot m_n - h_{a0} \cdot \tan \alpha_n + \frac{S_{pr}}{\cos \alpha_n} - (1 - \sin \alpha_n) \cdot \frac{\rho_{a0}}{\cos \alpha_n} \quad \text{[SI and MKS units]}$$

$$E = 25.4 \cdot \left(\frac{\pi}{4} \cdot m_n - h_{a0} \cdot \tan \alpha_n + \frac{S_{pr}}{\cos \alpha_n} - (1 - \sin \alpha_n) \cdot \frac{\rho_{a0}}{\cos \alpha_n}\right) \quad \text{[US units]}$$

$S_{pr} = p_{r0} - q$

$S_{pr} = 0$ when gears are not undercut (non-protuberance hob)

where:

E , h_{a0} , α_n , S_{pr} , p_{r0}, q and ρ_{a0} are shown in 4-3-1A1/Figure 2.

h_{a0} = addendum of tool; mm (in.)

S_{pr} = residual undercut left by protuberance; mm (in.)

p_{r0} = protuberance of tool; mm (in.)

q = material allowances for finish machining; mm (in.)

ρ_{a0} = tip radius of tool; mm (in.)

z_n = virtual number of teeth

x = addendum modification coefficient

α_{Fen} = angle for application of load at the highest point of single tooth contact

α_{en} = pressure angle at the highest point of single tooth contact

Bending moment arm h_F:

$$\frac{h_F}{m_n} = \frac{1}{2} \cdot \left[\left(\cos\gamma_e - \sin\gamma_e \cdot \tan\alpha_{Fen}\right) \cdot \frac{d_{en}}{m_n} - z_n \cdot \cos\left(\frac{\pi}{3} - \vartheta\right) - \frac{G}{\cos\vartheta} + \frac{\rho_{a0}}{m_n} \right]$$

$$\frac{\rho_F}{m_n} = \frac{\rho_{a0}}{m_n} + \frac{2 \cdot G^2}{\cos\vartheta \cdot (z_n \cos^2\vartheta - 2G)}$$

where

ρ_F = root fillet radius in the critical section at 30° tangent; mm (in.); see 4-3-1A1/Figure 1.

Normal load pressure angle at tip of tooth, α_{Fen} , α_{Fan}:

$\alpha_{Fen} = \alpha_{en} - \gamma_e$ degrees

$$\alpha_{en} = \arccos\left(\frac{d_{bn}}{d_{en}}\right) \quad \text{degrees}$$

$$\gamma_e = \left(\frac{0.5 \cdot \pi + 2 \cdot x \cdot \tan\alpha_n}{z_n} + \operatorname{inv}\alpha_n - \operatorname{inv}\alpha_{en}\right) \cdot \frac{180}{\pi} \quad \text{degrees}$$

$d_{an1} = d_{n1} + d_{a1} - d_1$ mm (in.)

$d_{an2} = d_{n2} + d_{a2} - d_2$ mm (in.)

$d_{n1,2} = z_{n1,2} \cdot m_n$ mm (in.)

$d_{bn1,2} = d_{n1,2} \cdot \cos\alpha_n$ mm (in.)

$$d_{en1} = \frac{2 \cdot z_1}{|z_1|} \cdot \sqrt{\left[\sqrt{\left(\frac{d_{an1}}{2}\right)^2 - \left(\frac{d_{bn1}}{2}\right)^2} - \frac{\pi \cdot d_1 \cdot \cos\beta \cdot \cos\alpha_n}{|z_1|} \cdot (\varepsilon_{\alpha n} - 1)\right]^2 + \left(\frac{d_{bn1}}{2}\right)^2} \quad \text{mm (in.)}$$

$$d_{en2} = \frac{2 \cdot z_2}{|z_2|} \cdot \sqrt{\left[\sqrt{\left(\frac{d_{an2}}{2}\right)^2 - \left(\frac{d_{bn2}}{2}\right)^2} - \frac{\pi \cdot d_2 \cdot \cos\beta \cdot \cos\alpha_n}{|z_2|} \cdot (\varepsilon_{\alpha n} - 1)\right]^2 + \left(\frac{d_{bn2}}{2}\right)^2} \quad \text{mm (in.)}$$

$$\varepsilon_{an} = \frac{\varepsilon_\alpha}{\cos^2 \beta_b} \quad \text{degrees}$$

Note: z_1, z_2 are positive for external gears and negative for internal gears.

23.5.2 Internal Gears *(2002)*

The tooth form factor of a special rack can be substituted as an approximate value of the form of an internal gear. The profile of such a rack should be a version of the basic rack profile, so modified that it would generate the normal profile, including tip and root circles, of an exact counterpart of the internal gear. The tip load angle is $\alpha_{Fen} = \alpha_n$.

Width of tooth, s_{Fn2}, at tooth-root normal chord:

$$\frac{s_{Fn2}}{m_n} = 2 \cdot \left[\frac{\pi}{4} + \tan\alpha_n \cdot \left(\frac{h_{fp2} - \rho_{fp2}}{m_n}\right) + \frac{\rho_{fp2} - S_{pr2}}{m_n \cdot \cos\alpha_n} - \frac{\rho_{fp2}}{m_n} \cdot \cos\frac{\pi}{6}\right]$$

Bending moment arm h_{F2}:

$$\frac{h_{F2}}{m_n} = \frac{d_{en2} - d_{fn2}}{2 \cdot m_n} - \left[\frac{\pi}{4} + \left(\frac{h_{fp2}}{m_n} - \frac{d_{en2} - d_{fn2}}{2 \cdot m_n}\right) \cdot \tan\alpha_n\right] \cdot \tan\alpha_n - \frac{\rho_{fp2}}{m_n} \cdot \left(1 - \sin\frac{\pi}{6}\right)$$

$$d_{fn2} = d_{n2} + d_{f2} - d_2 \quad \text{mm (in.)}$$

$$h_{fp2} = \frac{d_{n2} - d_{fn2}}{2} \quad \text{mm (in.)}$$

$$\rho_{fp2} = \rho_{F2} = \frac{\rho_{a02}}{2} \quad \text{mm (in.)}$$

Note: In the case of a full root fillet $\rho_{F2} = \rho_{a02}$ is to be used, or as an approximation:

$$\rho_{a02} = 0.30 \cdot m_n \quad \text{mm (in.)}$$

$$\rho_{fp2} = \rho_{F2} = 0.15 \cdot m_n \quad \text{mm (in.)}$$

$$h_{f2} = (1.25\ldots1.30) \cdot m_n \quad \text{mm (in.)}$$

$$\rho_{fp2} = \frac{c_p}{1 - \sin\alpha_n} = \frac{h_{f2} - h_{Nf2}}{1 - \sin\alpha_n} = \frac{d_{Nf2} - d_{f2}}{2 \cdot (1 - \sin\alpha_n)} \quad \text{mm (in.)}$$

where

d_{f2} = root diameter of wheel; mm (in.); see 4-3-1A1/Figure 1.

d_{Nf2}, h_{Nf2} = is the diameter, dedendum of basic rack at which the usable flank and root fillet of the annulus gear meet; mm (in.)

c_p = is the bottom clearance between basic rack and mating profile; mm (in.)

Note: The diameters d_{n2} and d_{en2} are to be calculated with the same formulas as for external gears.

23.5.3 Bevel Gears *(2006)*

Width of tooth, s_{Fn}, at tooth-root normal chord:

$$\frac{s_{Fn}}{m_{mn}} = z_{vn} \cdot \sin\left(\frac{\pi}{3} - \vartheta\right) + \sqrt{3} \cdot \left(\frac{G}{\cos\vartheta} - \frac{\rho_{a0}}{m_{mn}}\right)$$

$$\vartheta = 2 \cdot \frac{G}{z_{vn}} \cdot \tan\vartheta - H \qquad \text{degrees; to be solved iteratively}$$

$$G = \frac{\rho_{a0}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm}$$

$$H = \frac{2}{z_{vn}} \cdot \left(\frac{\pi}{2} - \frac{E}{m_{mn}}\right) - \frac{\pi}{3} \qquad \text{[SI and MKS units]}$$

$$H = \frac{2}{z_{vn}} \cdot \left(\frac{\pi}{2} - \frac{E}{25.4 \cdot m_{mn}}\right) - \frac{\pi}{3} \qquad \text{[US units]}$$

$$E = \frac{\pi}{4} \cdot m_{mn} - x_{sm} \cdot m_{mn} - h_{a0} \cdot \tan\alpha_n + \frac{S_{pr}}{\cos\alpha_n} - (1 - \sin\alpha_n) \cdot \frac{\rho_{a0}}{\cos\alpha_n} \qquad \text{[SI and MKS units]}$$

$$E = 25.4 \cdot \left(\frac{\pi}{4} \cdot m_{mn} - x_{sm} \cdot m_{mn} - h_{a0} \cdot \tan\alpha_n + \frac{S_{pr}}{\cos\alpha_n} - (1 - \sin\alpha_n) \cdot \frac{\rho_{a0}}{\cos\alpha_n}\right) \qquad \text{[US units]}$$

$$S_{pr} = p_{r0} - q$$

$S_{pr} = 0$ when gears are not undercut (non-protuberance hob)

where:

E, h_{a0}, α_n, S_{pr}, p_{r0}, q and ρ_{a0} are shown in 4-3-1A1/Figure 2.

h_{a0} = addendum of tool; mm (in.)

S_{pr} = residual undercut left by protuberance; mm (in.)

p_{r0} = protuberance of tool; mm (in.)

q = machining allowances; mm (in.)

ρ_{a0} = tip radius of tool; mm (in.)

z_{vn} = virtual number of teeth

x_{hm} = profile shift coefficient

x_{sm} = tooth thickness modification coefficient (midface)

α_{Fan} = angle for application of load at the highest point of single tooth contact

α_{an} = pressure angle at the highest point of single tooth contact

Bending moment arm h_{Fa}:

$$\frac{h_{Fa}}{m_{mn}} = \frac{1}{2} \cdot \left[(\cos\gamma_a - \sin\gamma_a \cdot \tan\alpha_{Fan}) \cdot \frac{d_{van}}{m_{mn}} - z_{vn} \cdot \cos\left(\frac{\pi}{3} - \vartheta\right) - \frac{G}{\cos\vartheta} + \frac{\rho_{a0}}{m_{mn}}\right]$$

$$\frac{\rho_F}{m_{mn}} = \frac{\rho_{a0}}{m_{mn}} + \frac{2 \cdot G^2}{\cos\vartheta \cdot (z_{vn} \cdot \cos^2\vartheta - 2 \cdot G)}$$

where

ρ_F = root fillet radius in the critical section at 30° tangent; mm (mm, in.); see 4-3-1A1/Figure 1.

Normal load pressure angle at tip of tooth α_{Fan}, α_{an}:

$$\alpha_{Fan} = \alpha_{an} - \gamma_a \quad \text{degrees}$$

$$\alpha_{an} = \arccos\left(\frac{d_{vbn}}{d_{van}}\right) \quad \text{degrees}$$

$$\gamma_a = \left(\frac{0.5 \cdot \pi + 2 \cdot (x_{hm} \cdot \tan\alpha_n + x_{sm})}{z_{vn}} + inv\alpha_n - inv\alpha_{an}\right) \cdot \frac{180}{\pi} \quad \text{degrees}$$

$$\beta_{bm} = \arccos\sqrt{1 - (\sin\beta_m \cdot \cos\alpha_n)^2} \quad \text{degrees}$$

(See also 4-3-1A1/7 of this Appendix.)

$$d_{van1} = d_{vn1} + d_{va1} - d_{v1} \quad \text{mm (in.)}$$

$$d_{van2} = d_{vn2} + d_{va2} - d_{v2} \quad \text{mm (in.)}$$

$$d_{vn1} = z_{vn1} \cdot m_{mn} \quad \text{mm (in.)}$$

$$d_{vn2} = z_{vn2} \cdot m_{mn} \quad \text{mm (in.)}$$

$$d_{vbn1} = d_{vn1} \cdot \cos\alpha_n \quad \text{mm (in.)}$$

$$d_{vbn2} = d_{vn2} \cdot \cos\alpha_n \quad \text{mm (in.)}$$

23.7 Stress Correction Factor, Y_S, Y_{Sa}

The stress correction factors, Y_S and Y_{Sa}, are used to convert the nominal bending stress to the local tooth root stress.

Y_S applies to the load application at the outer point of single tooth pair contact. Y_S is to be determined for pinion and wheel, separately.

For notch parameter q_s within a range of ($1 \le q_s < 8$):

$$q_s = \frac{s_{Fn}}{2 \cdot \rho_F}$$

Cylindrical gears:

$$Y_S = (1.2 + 0.13 \cdot L) \cdot q_s^{\left(\frac{1}{1.21 + (2.3/L)}\right)}$$

Bevel gears:

$$Y_{Sa} = (1.2 + 0.13 \cdot L_a) \cdot q_s^{\left(\frac{1}{1.21 + (2.3/L_a)}\right)}$$

where:

ρ_F = root fillet radius in the critical section at 30° tangent mm (in.)

L = s_{Fn}/h_F for cylindrical gears

L_a = s_{Fn}/h_{Fa} for bevel gears

h_F , h_{Fa}, s_{Fn}, ρ_F see 4-3-1A1/Figure 3.

23.9 Helix Angle Factor, Y_β

The helix angle factor, Y_β, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth.

$$Y_\beta = 1 - \varepsilon_\beta \cdot \frac{\beta}{120}$$

where:

β = reference helix angle in degrees for cylindrical gears.

$\varepsilon_\beta > 1.0$ a value of 1.0 is to be substituted for ε_β

$\beta > 30°$ an angle of 30° is to be substituted for β

23.11 Contact Ratio Factor, Y_ε

The contact factor, Y_ε, covers the conversion from load application at the tooth tip to the load application for bevel gears.

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha}\text{; for } \varepsilon_\beta = 0$$

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha} - \left(\frac{0.75}{\varepsilon_\alpha} - 0.375\right) \cdot \varepsilon_\beta \text{ ; for } 0 < \varepsilon_\beta < 1$$

$Y_\varepsilon = 0.625$; for $\varepsilon_\beta \geq 1$

23.13 Bevel Gear Factor, Y_K

The bevel gear factor, Y_K, accounts for the differences between bevel and cylindrical gears.

$$Y_K = \frac{1}{2} + \frac{b}{4 \cdot \ell'_{bm}} + \frac{\ell'_{bm}}{4 \cdot b}$$

$$\ell'_{bm} = \ell_{bm} \cdot \cos\beta_{vb}$$

23.15 Load Sharing Factor, Y_{LS}

The load sharing factor, Y_{LS}, accounts for the differences between two or more pair of teeth for $\varepsilon_\gamma > 2$.

$Y_{LS} = Z_{LS}^2 \geq 0.7$

for Z_{LS}, see 4-3-1A1/21.13 of this appendix.

23.17 Allowable Stress Number (Bending), σ_{FE}

For a given material, σ_{FE} is the limit of repeated tooth root stress that can be sustained. For most materials, their stress cycles may be taken at 3×10^6 as the beginning of the endurance limit, unless otherwise specified.

The endurance limit, σ_{FE}, is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment). Other conditions such as e.g., alternating stress or prestressing are covered by the design factor Y_d.

The endurance limit mainly depends on:

- Material composition, cleanliness and defects
- Mechanical properties
- Residual stress

- Hardening process, depth of hardened zone, hardness gradient
- Material structure (forged, rolled bar, cast)

The σ_{FE} values are to correspond to a failure probability of 1% or less. The values of σ_{FE} are to be determined from 4-3-1A1/Table 3 or to be advised by the manufacturer, together with technical justification for the proposed values. For gears treated with controlled shot peening process, the value, σ_{FE}, may be increased by 20%.

23.19 Design Factor, Y_d

The design factor, Y_d, takes into account the influence of load reversing and shrink fit prestressing on the tooth root strength, relative to the tooth root strength with unidirectional load as defined for σ_{FE}.

Y_d = 1.0 in general;

Y_d = 0.9 for gears with part load in reversed direction, such as main wheel in reversing gearboxes;

Y_d = 0.7 for idler gears.

23.21 Life Factor, Y_N

The life factor, Y_N, accounts for the higher permissible tooth root bending stress in case a limited life (number of load cycles) is specified.

The factor mainly depends on:

- Material and hardening
- Number of cycles
- Influence factors ($Y_{\delta relT}$, Y_{RrelT}, Y_X)

The life factor, Y_N, can be determined from 4-3-1A1/Table 5.

23.23 Relative Notch Sensitivity Factor, $Y_{\delta relT}$ *(2002)*

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit.

The factor mainly depends on the material and relative stress gradient.

For notch parameter values within the range of $1.5 \le q_s < 4$, $Y_{\delta relT} = 1.0$

For $q_s < 1.5$, $Y_{\delta relT} = 0.95$

For notch parameter $q_s \ge 4$, $Y_{\delta relT}$ can be determined by the methods outlined in ISO 6336-3, Section 11 Sensitivity factors, Y_δ, $Y_{\delta T}$, $Y_{\delta k}$ and relative notch sensitivity factors, $Y_{\delta relT}$, $Y_{\delta relk}$.

23.25 Relative Surface Factor, Y_{RrelT}

The relative surface factor, Y_{RrelT}, as given in the following table, takes into account the dependence of the tooth root bending strength on the surface condition in the tooth root fillet, but mainly the dependence on the peak to valley surface roughness.

	$R_{zr} < 1$ µm	SI & MKS units	US units
	$R_{zr} < 39$ µin	$1\ \mu m \le R_{zr} \le 40 \mu m$	$39\ \mu in \le R_{zr} \le 1575\ \mu in$
Case hardened steels, through - hardened steels $U \ge 800$ N/mm² (82 kgf/mm², 1.16 × 10⁵ psi)	1.120	$1.675 - 0.53 \cdot (R_{zr} + 1)^{0.1}$	$1.675 - 0.53 \cdot (0.0254 \cdot R_{zr} + 1)^{0.1}$
Normalized steels $U < 800$ N/mm² (82 kgf/mm², 1.16 × 10⁵ psi)	1.070	$5.3 - 4.2 \cdot (R_{zr} + 1)^{0.01}$	$5.3 - 4.2\ (0.0254\ R_{zr} + 1)^{0.01}$
Nitrided steels	1.025	$4.3 - 3.26 \cdot (R_{zr} + 1)^{0.005}$	$4.3 - 3.26 \cdot (0.0254 \cdot R_{zr} + 1)^{0.005}$
R_{zr} = mean peak to-valley roughness of tooth root fillets; µm (µm, µin)			

This method is only applicable where scratches or similar defects deeper than $2R_{zr}$ are not present.

If the stated roughness is an R_a value, also known as arithmetic average (*AA*) and centerline average (*CLA*), the following approximate relationship may be applied:

$$R_a = CLA = AA = R_{zr} / 6$$

23.27 Size Factor (Root), Y_X

The size factor (root), Y_X, takes into account the decrease of the strength with increasing size.

The factor mainly depends on:

- Material and heat treatment
- Tooth and gear dimensions
- Ratio of case depth to tooth size

SI and MKS units		
$Y_X = 1.00$	for $m_n \le 5$	Generally
$Y_X = 1.03 - 0.006 \cdot m_n$	for $5 < m_n < 30$	Normalized and through tempered- hardened steels
$Y_X = 0.85$	for $m_n \ge 30$	
$Y_X = 1.05 - 0.010 \cdot m_n$	for $5 < m_n < 25$	Surface hardened steels
$Y_X = 0.80$	for $m_n \ge 25$	
US units		
$Y_X = 1.00$	for $m_n \le 0.1968$	Generally
$Y_X = 1.03 - 0.1524 \cdot m_n$	for $0.1968 < m_n < 1.181$	Normalized and through tempered- hardened steels
$Y_X = 0.85$	for $m_n \ge 1.181$	
$Y_X = 1.05 - 0.254 \cdot m_n$	for $0.1968 < m_n < 0.9842$	Surface hardened steels
$Y_X = 0.80$	for $m_n \ge 0.9842$	

Note: For *Bevel gears*, the m_n (normal module) is to be substituted by m_{mn} (normal module at mid-facewidth).

23.29 Safety Factor for Tooth Root Bending Stress, S_F

Based on the application, the following safety factors for tooth root bending stress, S_F, are to be applied:

Main propulsion gears (including PTO):	1.80
Duplicated (or more) independent main propulsion gears (including azimuthing thrusters):	1.60
Main propulsion gears for yachts, single screw:	1.50
Main propulsion gears for yachts, multiple screw:	1.45
Auxiliary gears:	1.40

Note: For the above purposes, yachts are considered pleasure craft not engaged in trade or carrying passengers, and not intended for charter-service.

TABLE 1
Values of the Elasticity Factor Z_E and Young's Modulus of Elasticity E

(Ref. 4-3-1A1/21.9)

The value of E for combination of different materials for pinion and wheel is to be calculated by:

$$E = \frac{2 \cdot E_1 \cdot E_2}{E_1 + E_2}$$

SI units						
Pinion			Wheel			
Material	Young's Modulus of elasticity E_1 N/mm²	Poisson's ratio ν	Material	Young's Modulus of elasticity E_2 N/mm²	Poisson's ratio ν	Elasticity Factor Z_E $N^{1/2}$/mm
Steel	206000	0.3	Steel	206000	0.3	189.8
			Cast steel	202000		188.9
			Nodular cast iron	173000		181.4
			Cast tin bronze	103000		155.0
			Tin bronze	113000		159.8
			Lamellar graphite cast iron (gray cast iron)	126000 to 118000		165.4 to 162.0
Cast steel	202000		Cast steel	202000		188.0
			Nodular cast iron	173000		180.5
			Lamellar graphite cast iron (gray cast iron)	118000		161.4
Nodular cast iron	173000	0.3	Nodular cast iron	173000	0.3	173.9
			Lamellar graphite cast iron (gray cast iron)	118000		156.6
Lamellar graphite cast iron (gray cast iron)	126000 to 118000		Lamellar graphite cast iron (gray cast iron)	118000		146.0 to 143.7
Steel	206000	0.3	Nylon	7850 (mean value)	0.5	56.4

continued....

TABLE 1 (continued)
Values of the Elasticity Factor Z_E and Young's Modulus of Elasticity E

MKS units						
Pinion			Wheel			
Material	Young's Modulus of elasticity E_1 kgf/mm²	Poisson's ratio ν	Material	Young's Modulus of elasticity E_2 kgf/mm²	Poisson's ratio ν	Elasticity Factor Z_E kgf$^{1/2}$/mm
Steel	2.101×10^4	0.3	Steel	2.101×10^4	0.3	60.609
			Cast steel	2.060×10^4		60.321
			Nodular cast iron	1.764×10^4		57.926
			Cast tin bronze	1.050×10^4		49.496
			Tin bronze	1.152×10^4		51.029
			Lamellar graphite cast iron (gray cast iron)	1.285×10^4 to 1.203×10^4		52.817 to 51.731
Cast steel	2.060×10^4		Cast steel	2.060×10^4		60.034
			Nodular cast iron	1.764×10^4		57.639
			Lamellar graphite cast iron (gray cast iron)	1.203×10^4		51.540
Nodular cast iron	1.764×10^4	0.3	Nodular cast iron	1.764×10^4	0.3	55.531
			Lamellar graphite cast iron (gray cast iron)	1.203×10^4		50.007
Lamellar graphite cast iron (gray cast iron)	1.285×10^4 to 1.203×10^4		Lamellar graphite cast iron (gray cast iron)	1.203×10^4		46.622 to 45.888
Steel	2.101×10^4	0.3	Nylon	800.477 (mean value)	0.5	18.010

continued......

TABLE 1 (continued)
Values of the Elasticity Factor Z_E and Young's Modulus of Elasticity E

US units						
Pinion			Wheel			
Material	Young's Modulus of elasticity E_1 psi	Poisson's ratio ν	Material	Young's Modulus of elasticity E_2 psi	Poisson's ratio ν	Elasticity factor Z_E lbf$^{1/2}$/in
Steel	2.988×10^7	0.3	Steel	2.988×10^7	0.3	2.286×10^3
			Cast steel	2.930×10^7		2.275×10^3
			Nodular cast iron	2.509×10^7		2.185×10^3
			Cast tin bronze	1.494×10^7		1.867×10^3
			Tin bronze	1.639×10^7		1.924×10^3
			Lamellar graphite cast iron (gray cast iron)	1.827×10^7 to 1.711×10^7		1.992×10^3 to 1.951×10^3
Cast steel	2.930×10^7		Cast steel	2.930×10^7		2.264×10^3
			Nodular cast iron	2.509×10^7		2.174×10^3
			Lamellar graphite cast iron (gray cast iron)	1.711×10^7		1.944×10^3
Nodular cast iron	2.509×10^7	0.3	Nodular cast iron	2.509×10^7	0.3	2.094×10^3
			Lamellar graphite cast iron (gray cast iron)	1.711×10^7		1.886×10^3
Lamellar graphite cast iron (gray cast iron)	1.827×10^7 to 1.711×10^7		Lamellar graphite cast iron (gray cast iron)	1.711×10^7		1.758×10^3 to 1.731×10^3
Steel	2.988×10^7	0.3	Nylon	1.139×10^6 (mean value)	0.5	679.234

TABLE 2
Size Factor Z_X for Contact Stress
(Ref. 4-3-1A1/21.27)

SI and MKS units

Z_X, size factor for contact stress	Material
	For through-hardened pinion treatment
1.0	All modules (m_n)
	For carburized and induction-hardened pinion heat treatment
1.0	$m_n \leq 10$
$1.05 - 0.005\, m_n$	$m_n < 30$
0.9	$m_n \geq 30$
	For nitrided pinion treatment
1.0	$m_n < 7.5$
$1.08 - 0.011\, m_n$	$m_n < 30$
0.75	$m_n \geq 30$
For ***Bevel gears***, the m_n (normal module) is to be substituted by m_{mn} (normal module at mid-facewidth).	

US units

Z_X, size factor for contact stress	Material
	For through-hardened pinion treatment
1.0	All modules (m_n)
	For carburized and induction-hardened pinion heat treatment
1.0	$m_n \leq 0.394$
$1.05 - 0.127\, m_n$	$m_n < 1.181$
0.9	$m_n \geq 1.181$
	For nitrided pinion treatment
1.0	$m_n < 0.295$
$1.08 - 0.279\, m_n$	$m_n < 1.181$
0.75	$m_n \geq 1.181$
For ***Bevel gears***, the m_n (normal module) is to be substituted by m_{mn} (normal module at mid-facewidth).	

TABLE 3
Allowable Stress Number (contact) $\sigma_{H\text{lim}}$ and Allowable Stress Number (bending) σ_{FE} *(2002)*

(Ref. 4-3-1A1/21.19, 4-3-1A1/23.17)

SI units	$\sigma_{H\text{lim}}$ *N/mm²*	σ_{FE} *N/mm²*	*Reference Standard ISO 6336-5:1996(E) ISO Figure and Material Quality*
Case hardened (carburized) CrNiMo steels:			Fig. 9, MQ
of ordinary grade;	1500	920	Fig. 11, MQ (1)
of specially approved high quality grade (to be based on review and verification of established testing procedure).	1650	1050	Fig. 9, ME Fig. 11, ME (2)
Other case hardened (carburized) steels	1500	840	Fig. 9, MQ Fig. 11, MQ (3)
Gas nitrided steels: hardened, tempered and gas nitrided, Surface hardness: 700-850 HV10	1250	920	Fig. 13a, MQ Fig. 14a, MQ
Through hardened steels: hardened, tempered and gas nitrided, Surface hardness: 500-650 HV10	1000	740	Fig. 13b, MQ Fig. 14b, MQ
Through hardened steels: hardened, tempered or normalized and nitro-carburized, Surface hardness: 450-650 HV10	950	780	Fig. 13c, ME-MQ Fig. 14c, ME-MQ
Flame or induction hardened steels, Surface hardness: 520-620 HV10	0.65·HV10 + 830	0.25·HV10 + 580	Fig. 10, MQ Fig. 12, MQ
Alloyed through hardening steels, Surface hardness: 195-360 HV10	1.32·HV10 + 372	0.78 HV10 + 400	Fig. 5, MQ Fig. 7, MQ
Through hardened carbon steels, Surface hardness: 135-210 HV10	1.05·HV10 + 335	0.50·HV10 + 320	Fig. 5, Carbon steel, MQ Fig. 7, Carbon steel, MQ
Alloyed cast steels, Surface hardness: 198-358 HV10	1.30 HV10 + 295	0.68 HV10 + 325	Fig. 6, MQ-ML Fig. 8, MQ-ML
Cast carbon steels, Surface hardness: 135-210 HV10	0.87·HV10 + 290	0.50·HV10 + 225	Fig. 6, Carbon steel, MQ-ML Fig. 8, Carbon steel, MQ-ML

TABLE 3 (continued) Allowable Stress Number (contact) $\sigma_{H\lim}$ and Allowable Stress Number (bending) σ_{FE} *(2002)*

MKS units	$\sigma_{H\lim}$ *kgf/mm²*	σ_{FE} *kgf/mm²*	*Reference Standard ISO 6336-5:1996(E) ISO Figure and Material Quality*
Case hardened (carburized) CrNiMo steels: of ordinary grade;	153.0	93.8	Fig. 9, MQ Fig. 11, MQ (1)
of specially approved high quality grade (to be based on review and verification of established testing procedure).	168.3	107.1	Fig. 9, ME Fig. 11, ME (2)
Other case hardened (carburized) steels	153.0	85.7	Fig. 9, MQ Fig. 11, MQ (3)
Gas nitrided steels: hardened, tempered and gas nitrided, Surface hardness: 700-850 HV10	127.5	93.8	Fig. 13a, MQ Fig. 14a, MQ
Through hardened steels: hardened, tempered and gas nitrided, Surface hardness: 500-650 HV10	102.0	75.5	Fig. 13b, MQ Fig. 14b, MQ
Through hardened steels: hardened, tempered or normalized and nitro-carburized, Surface hardness: 450-650 HV10	96.9	79.5	Fig. 13c, ME-MQ Fig. 14c, ME-MQ
Flame or induction hardened steels, Surface hardness: 520-620 HV10	0.0663·HV10 + 84.6	0.0255·HV10 + 59.1	Fig. 10, MQ Fig. 12, MQ
Alloyed through hardening steels, Surface hardness: 195-360 HV10	0.1346·HV10 + 37.9	0.0795 HV10 + 40.8	Fig. 5, MQ Fig. 7, MQ
Through hardened carbon steels, Surface hardness: 135-210 HV10	0.1071·HV10 + 34.2	0.0510·HV10 + 32.6	Fig. 5, Carbon steel, MQ Fig. 7, Carbon steel, MQ
Alloyed cast steels, Surface hardness: 198-358 HV10	0.1326 HV10 + 30.1	0.0693 HV10 + 33.1	Fig. 6, MQ-ML Fig. 8, MQ-ML
Cast carbon steels, Surface hardness: 135-210 HV10	0.0887·HV10 + 29.6	0.0510·HV10 + 22.9	Fig. 6, Carbon steel, MQ-ML Fig. 8, Carbon steel, MQ-ML

TABLE 3 (continued)
Allowable Stress Number (contact) $\sigma_{H\lim}$ and Allowable Stress Number (bending) σ_{FE} *(2002)*

US units	$\sigma_{H\lim}$ *psi*	σ_{FE} *psi*	*Reference Standard ISO 6336-5:1996(E) ISO Figure and Material Quality*
Case hardened (carburized) CrNiMo steels: of ordinary grade; of specially approved high quality grade (to be based on review and verification of established testing procedure).	 217557 239312	 133435 152290	Fig. 9, MQ Fig. 11, MQ (1) Fig. 9, ME Fig. 11, ME (2)
Other case hardened (carburized) steels	217557	121832	Fig. 9, MQ Fig. 11, MQ (3)
Gas nitrided steels: hardened, tempered and gas nitrided, Surface hardness: 700-850 HV10	181297	133435	Fig. 13a, MQ Fig. 14a, MQ
Through hardened steels: hardened, tempered and gas nitrided, Surface hardness: 500-650 HV10	145038	107328	Fig. 13b, MQ Fig. 14b, MQ
Through hardened steels: hardened, tempered or normalized and nitro-carburized, Surface hardness: 450-650 HV10	137786	113129	Fig. 13c, ME-MQ Fig. 14c, ME-MQ
Flame or induction hardened steels, Surface hardness: 520-620 HV10	94.3·HV10 + 120381	36.3·HV10 + 84122	Fig. 10, MQ Fig. 12, MQ
Alloyed through hardening steels, Surface hardness: 195-360 HV10	191.5·HV10 + 53954	113.1 HV10 + 58015	Fig. 5, MQ Fig. 7, MQ
Through hardened carbon steels, Surface hardness: 135-210 HV10	152.3·HV10 + 48588	72.5·HV10 + 46412	Fig. 5, Carbon steel, MQ Fig. 7, Carbon steel, MQ
Alloyed cast steels, Surface hardness: 198-358 HV10	188.6 HV10 + 42786	98.6 HV10 + 47137	Fig. 6, MQ-ML Fig. 8, MQ-ML
Cast carbon steels, Surface hardness: 135-210 HV10	126.2·HV10 + 42061	72.5·HV10 + 32633	Fig. 6, Carbon steel, MQ-ML Fig. 8, Carbon steel, MQ-ML

Notes

HV10: Vickers hardness at load F = 98.10 N, see ISO 6336-5

1 Core hardness ≥ 25 HRC, Jominy hardenability at J = 12 mm ≥ HRC 28 and Surface hardness: 640-780 HV10

2 Core hardness ≥ 30 HRC, Surface hardness: 660-780 HV10

3 Core hardness ≥ 25 HRC, Jominy hardenability at J = 12 mm < HRC 28 and Surface hardness: 640-780 HV10

TABLE 4
Determination of Life Factor for Contact Stress, Z_N

(Ref. 4-3-1A1/21.21)

Material	*Number of load cycles*	*Life factor Z_N*
St, V, GGG (perl., bain.), GTS (perl.), Eh, IF; Only when a certain degree of pitting is permissible	$N_L \leq 6 \times 10^5$, static	1.6
	$N_L = 10^7$	1.3
	$N_L = 10^9$	1.0
	$N_L = 10^{10}$ Optimum lubrication, material, manufacturing, and experience	0.85 1.0
St, V, GGG (perl., bain.), GTS (perl.), Eh, IF	$N_L \leq 10^5$, static	1.6
	$N_L = 5 \times 10^7$	1.0
	$N_L = 10^{10}$ Optimum lubrication, material, manufacturing, and experience	0.85 1.0
GG, GGG (ferr.), NT (nitr.), NV (nitr.)	$N_L \leq 10^5$, static	1.3
	$N_L = 2 \times 10^6$	1.0
	$N_L = 10^{10}$ Optimum lubrication, material, manufacturing, and experience	0.85 1.0
NV (nitrocar.)	$N_L \leq 10^5$, static	1.1
	$N_L = 2 \times 10^6$	1.0
	$N_L = 10^{10}$ Optimum lubrication, material, manufacturing, and experience	0.85 1.0

St:	steel (U < 800 N/mm^2, 82 kgf/mm^2, 1.16×10^5 psi)
V:	through-hardening steel, through-hardened (U ≥ 800 N/mm^2)
GG:	gray cast iron
GGG (perl., bain., ferr.):	nodular cast iron (perlitic, bainitic, ferritic structure)
GTS (perl.):	black malleable cast iron (perlitic structure)
Eh:	case-hardening steel, case hardening
IF:	steel and GGG, flame or induction hardened
NT (nitr.):	nitriding steel, nitrided
NV (nitr.):	through-hardening and case-hardening steel, nitrided
NV (nitrocar.):	through-hardening and case-hardening steel, nitrocarburized

TABLE 5
Determination of Life Factor for Tooth Root Bending Stress, Y_N

(Ref. 4-3-1A1/23.21)

Material	*Number of load cycles*	*Life factor Y_N*
V, GGG (perl., bain.), GTS (perl.)	$N_L \leq 10^4$, static	2.5
	$N_L = 3 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	Optimum lubrication, material, manufacturing, and experience	1.0
Eh, IF (root)	$N_L \leq 10^3$, static	2.5
	$N_L = 3 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	Optimum lubrication, material, manufacturing, and experience	1.0
St, NT, NV (nitr.), GG, GGG (ferr.)	$N_L \leq 10^3$, static	1.6
	$N_L = 3 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	Optimum lubrication, material, manufacturing, and experience	1.0
NV (nitrocar.)	$N_L \leq 10^3$, static	1.0
	$N_L = 3 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	Optimum lubrication, material, manufacturing, and experience	1.0

Notes:

1) Abbreviations of materials are as explained in 4-3-1A1/Table 4 and 4-3-1A1/21.21 of this Appendix.

2) $N_L = n \cdot 60 \cdot HPD \cdot DPY \cdot YRS$

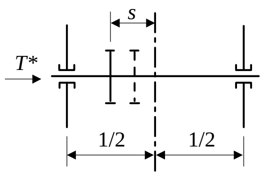
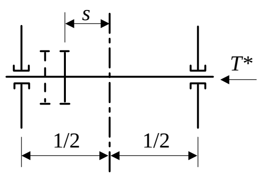
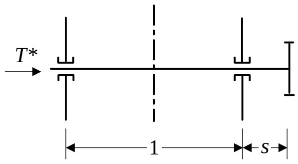
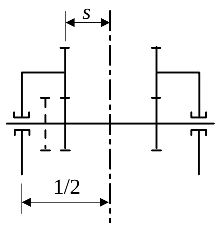
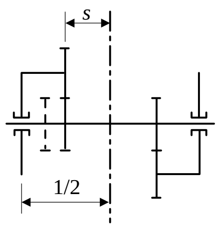
n = rotational speed, rpm.

HPD = operation hours per day

DPY = days per year

YRS = years (normal life of vessel = 25 *years*

TABLE 6
Constant K' for the Calculation of the Pinion Offset Factor γ

Factor K′		*Figure*	*Arrangement*
with stiffening (1)	*without stffening* (1)		
0.48	0.8	a)	 with $s/\ell < 0.3$
-0.48	-0.8	b)	 with $s/\ell < 0.3$
1.33	1.33	c)	 with $s/\ell < 0.3$
-0.36	-0.6	d)	 with $s/\ell < 0.3$
-0.6	-1.0	e)	 with $s/\ell < 0.3$

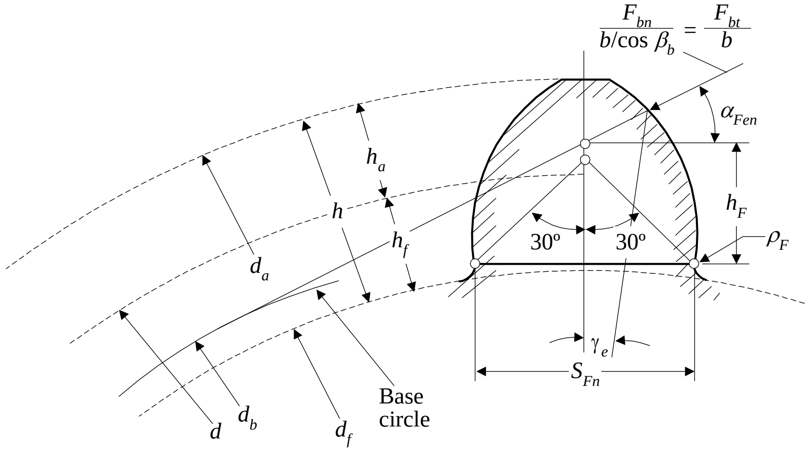
1. When $d_t/d_{sh} \geq 1.15$, stiffening is assumed; when $d_t/d_{sh} < 1.15$, there is no stiffening. Furthermore, scarcely any or no stiffening at all is to be expected when a pinion slides on a shaft and feather key or a similar fitting, nor when normally shrink fitted.

T^* is the input or output torqued end, not dependent on direction of rotation.

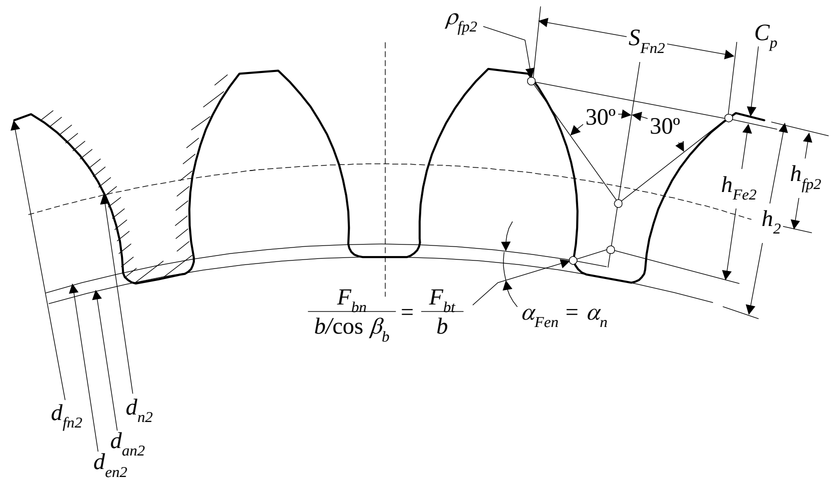
Dashed line indicates the less deformed helix of a double helical gear.

Determine t_{sh} from the diameter in the gaps of double helical gearing mounted centrally between bearings.

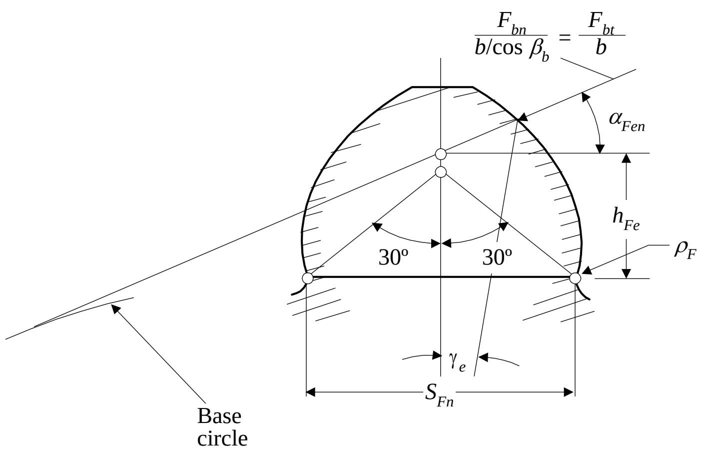
FIGURE 1
Tooth in Normal Section



External cylindrical gears

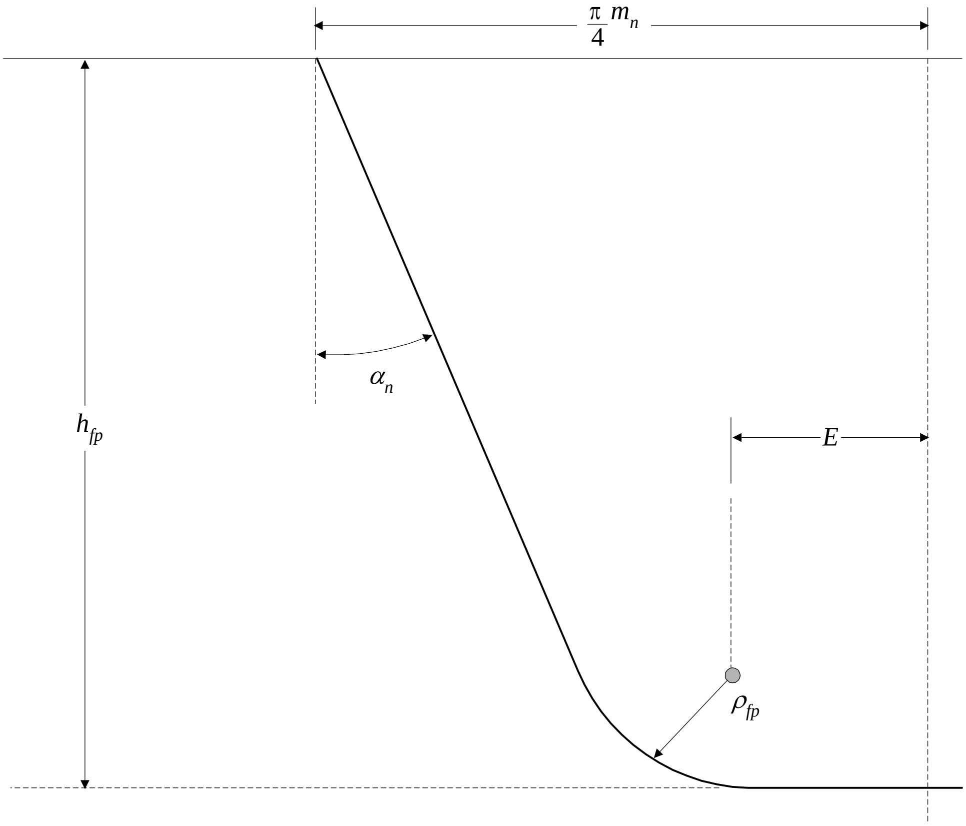


Internal cylindrical gears

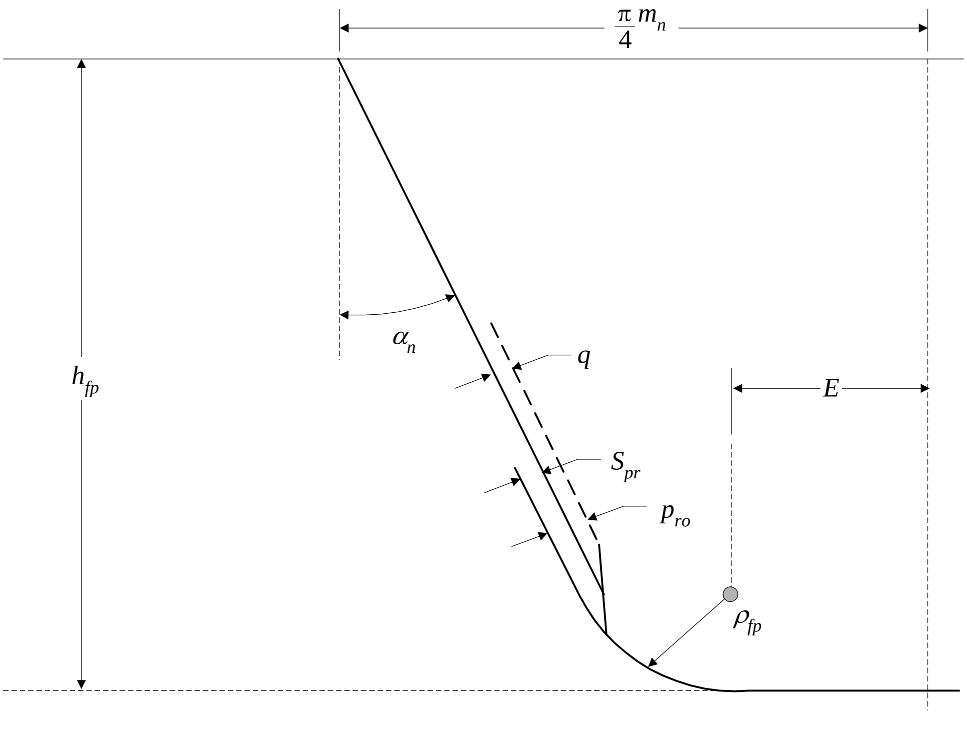


Bevel gears

FIGURE 2
Dimensions and Basic Rack Profile of the Tooth (Finished Profile)



Without undercut



With undercut

FIGURE 3
Wheel Blank Factor C_R, Mean Values for Mating Gears of Similar or Stiffer Wheel Blank Design

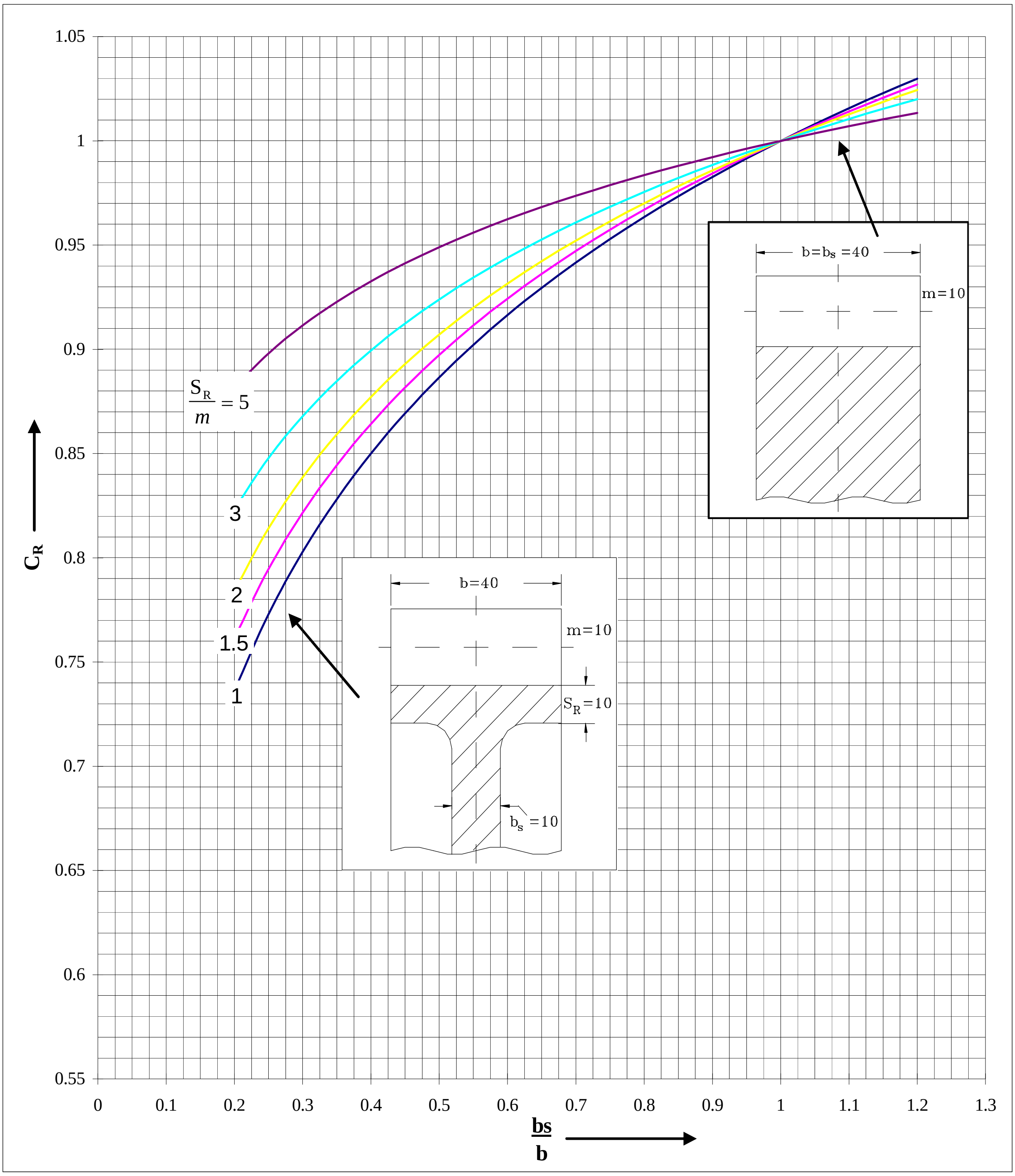


FIGURE 4
Bevel Gear Conversion to Equivalent Cylindrical Gear

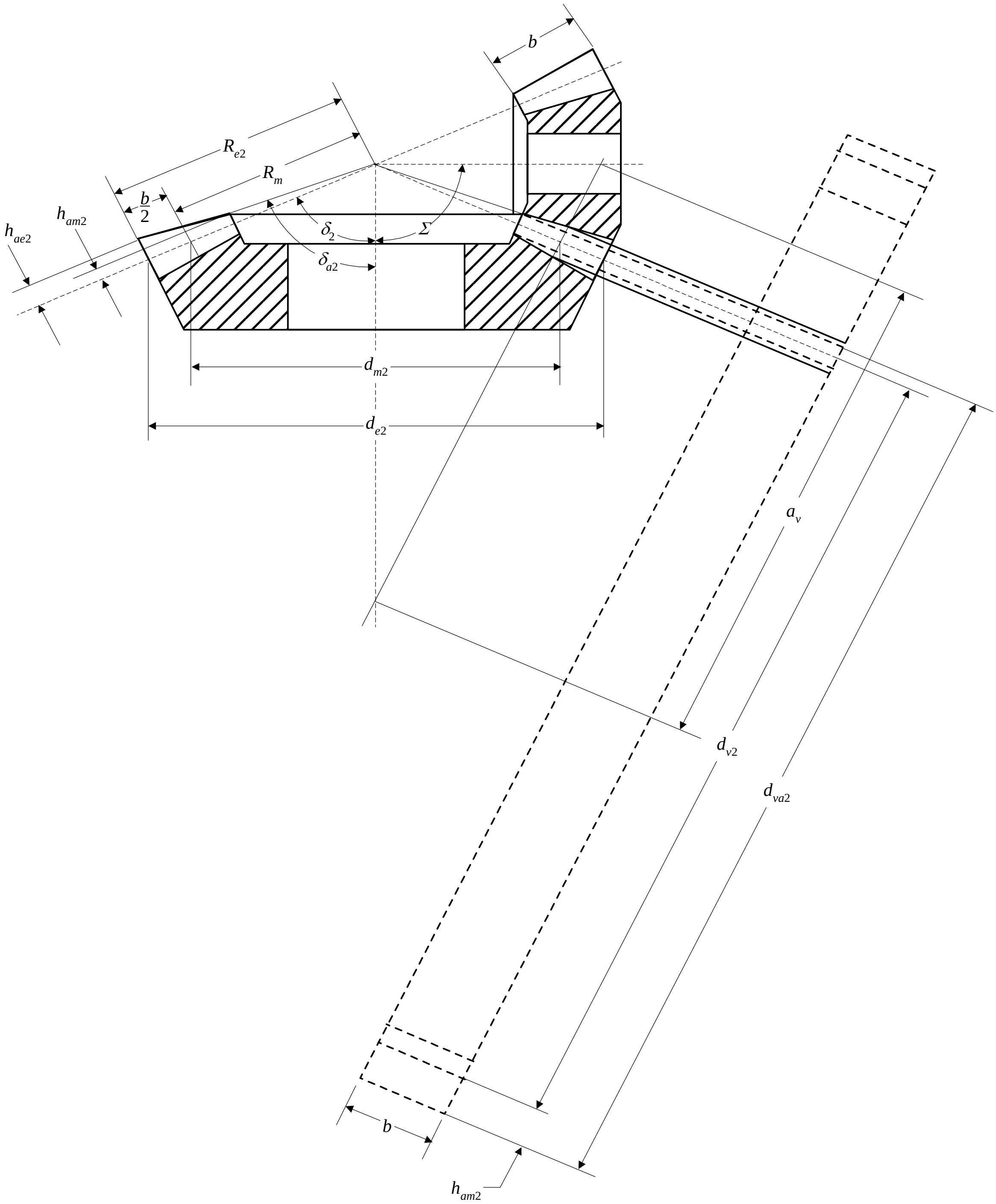
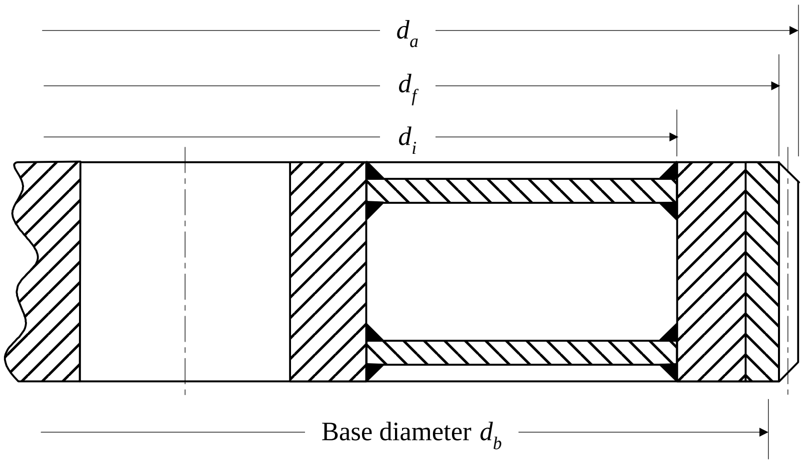


FIGURE 5
Definitions of the Various Diameters



PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 1 Appendix 2 – Guidance for Spare Parts

1 General

While spare parts are not required for purposes of classification, the spare parts listed below are provided as a guidance for vessels intended for unrestricted service. The maintenance of spare parts aboard each vessel is the responsibility of the owner.

3 Spare Parts for Gears

a) Sufficient packing rings with springs to repack one gland of each kind and size.

b) One (1) set of thrust pads or rings, also springs where fitted, for each size.

c) Bearing bushings sufficient to replace all of the bushings on every pinion, and gear for main propulsion; spare bearing bushings sufficient to replace all of the bushings on each non-identical pinion and gear having sleeve-type bearings or complete assemblies consisting of outer and inner races and cages complete with rollers or balls where these types of bearings are used.

d) One (1) set of bearing shoes for one face, for one single-collar type main thrust bearing where fitted. Where the ahead and astern pads differ, pads for both faces are to be provided.

e) One (1) set of strainer baskets or inserts for oil filters of special design of each type and size.

f) Necessary special tools.

PART
4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 1 Appendix 3 – Gear Parameters

For purposes of submitting gear design for review, the following data and parameters may be used as a guide.

Gear manufacturer .... Year of build..
Shipyard.. Hull number..
Ship's name... Shipowner...

GENERAL DATA [1]
Gearing type...
.... (non reversible, single reduction, double reduction, epicyclic, etc.)
Total gear ratio.. ....
Manufacturer and type of the main propulsion plant (or of the auxiliary machinery).............................
....
Power [2] ....kW, (PS, hp) Rotational speed [2].............................. RPM
Maximum input torque for continuous service..N-m, (kgf-m, lbf-ft)
Maximum input rotational speed for continuous service.. RPM
Type of coupling: (stiff coupling, hydraulic or equivalent coupling, high-elasticity coupling, other couplings, quill shafts, etc.).. ....
Specified grade of lubricating oil....................... ....
Expected oil temperature when operating at the classification power (mean values of temperature at inlet and outlet of reverse and/or reduction gearing).......... ....
Value of nominal kinematic viscosity, ν, at 40°C or 50°C of oil temperature.............................. (mm^2/s)

CHARACTERISTIC ELEMENTS OF PINIONS AND WHEELS Gear drive designation		First gear Drive	Second gear drive	Third gear drive	Fourth gear drive
Transmitted rated power, input in the gear drive – kW (mhp, hp) (3)	P				
Rotational speed, input in the gear drive (RPM) (2)	n				
Nominal transverse tangential load at reference cylinder – N, (kgf, lbf) (4)	F_t				
Number of teeth of pinion	z_1				
Number of teeth of wheel	z_2				
Gear ratio	u				
Center distance – mm (in.)	a				
Facewidth of pinion – mm (in.)	b_1				
Facewidth of wheel – mm (in.)	b_2				
Common facewidth – mm (in.)	b				
Overall facewidth, including the gap for double helical gears – mm (in.)	B				

1 The manufacturer can supply values, if available, supported by documents for stress numbers σ_{Hlim} and σ_{FE} involved in the formulas for gear strength with respect to the contact stress and with respect to the tooth root bending stress. See 4-3-1A1/21 and 4-3-1A1/23.

2 Maximum continuous performance of the machinery for which classification is requested.

3 It is intended the power mentioned under note (2) or fraction of it in case of divided power.

4 The nominal transverse tangential load is calculated on the basis of the above mentioned maximum continuous performance or on the basis of astern power when it gives a higher torque. In the case of navigation in ice, the nominal transverse tangential load is to be increased as required by 4-3-1A1/9.

CHARACTERISTIC ELEMENTS OF PINIONS AND WHEELS Gear drive designation		First gear drive	Second gear Drive	Third gear drive	Fourth gear drive
Is the wheel an external or an internal teeth gear?	-				
Number of pinions	-				
Number of wheels	-				
Is the wheel an external or an internal teeth gear	-				
Is the pinion an intermediate gear?	-				
Is the wheel an intermediate gear?	-				
Is the carrier stationary or revolving (star or planet-carrier)? (5)	-				
Reference diameter of pinion – mm (in.)	d_1				
Reference diameter of wheel – mm (in.)	d_2				
Working pitch diameter of pinion – mm (in.)	d_{w1}				
Working pitch diameter of wheel – mm (in.)	d_{w2}				
Tip diameter of pinion – mm (in.)	d_{a1}				
Tip diameter of wheel – mm (in.)	d_{a2}				
Base diameter of pinion – mm (in.)	d_{b1}				
Base diameter of wheel – mm (in.)	d_{b2}				
Tooth depth of pinion – mm (in.) (6)	h_1				
Tooth depth of wheel – mm (in.) (6)	h_2				
Addendum of pinion – mm (in.)	h_{a1}				
Addendum of wheel – mm (in.)	h_{a2}				
Addendum of tool referred to normal module for pinion	h_{a01}				
Addendum of tool referred to normal module for wheel	h_{a02}				
Dedendum of pinion – mm (in.)	h_{f1}				
Dedendum of wheel – mm (in.)	h_{f2}				
Dedendum of basic rack [referred to m_n (=h_{a01})] for pinion	h_{fp1}				
Dedendum of basic rack [referred to m_n (=h_{a02})] for wheel	h_{fp2}				
Bending moment arm of tooth – mm (in.) (7)	h_F				
Bending moment arm of tooth – mm (in.) (8)	h_{F2}				
Bending moment arm of tooth – mm (in.) (9)	h_{Fa}				
Width of tooth at tooth-root normal chord – mm (in.) (7, 9)	S_{Fn}				
Width of tooth at tooth-root normal chord – mm (in.) (8)	S_{Fn2}				
Angle of application of load at the highest point of single tooth contact (degrees)	α_{Fen}				
Pressure angle at the highest point of single tooth contact (degrees)	α_{en}				

5 Only for epicyclic gears.

6 Measured from the tip circle (or circle passing through the point of beginning of the fillet at the tooth tip) to the beginning of the root fillet.

7 Only for external gears.

8 Only for internal gears.

9 Only for bevel gears.

CHARACTERISTIC ELEMENTS OF PINIONS AND WHEELS Gear drive designation		First gear drive	Second gear drive	Third gear Drive	Fourth gear drive
Normal module – mm (in.)	m_n				
Outer normal module – mm (in.)	m_{na}				
Transverse module	m_t				
Addendum modification coefficient of pinion	x_1				
Addendum modification coefficient of wheel	x_2				
Addendum modification coefficient refers to the midsection (9)	x_{mn}				
Tooth thickness modification coefficient (midface) (9)	x_{sm}				
Addendum of tool – mm (in.)	h_{a0}				
Protuberance of tool – mm (in.)	P_{r0}				
Machining allowances – mm (in.)	q				
Tip radius of tool – mm (in.)	ρ_{a0}				
Profile form deviation – mm (in.)	f_{fa}				
Transverse base pitch deviation – mm (in.)	f_{pb}				
Root fillet radius at the critical section – mm (in.)	ρ_F				
Root radius of basic rack [referred to m_n (= ρ_{a0})] – mm (in.)	ρ_{fp}				
Radius of curvature at pitch surface – mm (in.)	ρ_c				
Normal pressure angle at reference cylinder (degrees)	α_n				
Transverse pressure angle at reference cylinder (degrees)	α_t				
Transverse pressure angle at working pitch cylinder (degrees)	α_{tw}				
Helix angle at reference cylinder (degrees)	β				
Helix angle at base cylinder (degrees)	β_b				
Transverse contact ratio	ε_α				
Overlap ratio	ε_β				
Total contact ratio	ε_γ				
Angle of application of load at the highest point of single tooth contact (degrees) (9)	α_{Fan}				
Reference cone angle of pinion (degrees) (9)	δ_1				
Reference cone angle of wheel (degrees) (9)	δ_2				
Shaft angle (degrees) (9)	Σ				
Tip angle of pinion (degrees) (9)	δ_{a1}				
Tip angle of wheel (degrees) (9)	δ_{a2}				
Helix angle at reference cylinder (degrees) (9)	β_m				
Cone distance pinion. wheel – mm (in.) (9)	R				
Middle cone distance – mm (in.) (9)	R_m				

9 Only for bevel gears.

MATERIALS

First gear drive

Pinion: Material grade or specification ....
Complete chemical analysis...
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10).. N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) .... % Hardness (HB, HV10 or HRC)......................................
Heat treatment...
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
Specified grade of accuracy (according to ISO 1328)
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

Wheel: Material grade or specification ....
Complete chemical analysis...
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10) .... N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) .... % Hardness (HB, HV10 or HRC)......................................
Heat treatment ....
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

10 Relevant to core of material

11 In case of shrink-fitted pinions, wheel rims or hubs

Second gear drive

Pinion: Material grade or specification ....
Complete chemical analysis...
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10) .... N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) ....% Hardness (HB, HV10 or HRC) ....
Heat treatment..
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

Wheel: Material grade or specification ....
Complete chemical analysis...
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10) .... N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) ....%; Hardness (HB, HV10 or HRC) ....
Heat treatment..
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

10 Relevant to core of material

11 In case of shrink-fitted pinions, wheel rims or hubs

Third gear drive

Pinion: Material grade or specification ....
Complete chemical analysis..
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10).. N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) ....%; Hardness (HB, HV10 or HRC) ....
Heat treatment..
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)...
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

Wheel: Material grade or specification ....
Complete chemical analysis..
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10).. N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) ....%; Hardness (HB, HV10 or HRC) ....
Heat treatment..
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)...
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

10 Relevant to core of material

11 In case of shrink-fitted pinions, wheel rims or hubs

Fourth gear drive

<u>Pinion:</u> Material grade or specification ....
Complete chemical analysis...
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10) .... N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) .... %; Hardness (HB, HV10 or HRC) ....
Heat treatment...
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

<u>Wheel:</u> Material grade or specification ....
Complete chemical analysis...
....
....
Minimum ultimate tensile strength (10) .... N/mm², (kgf/mm², lbf/in²)
Minimum yield strength (10) .... N/mm², (kgf/mm², lbf/in²)
Elongation (A_5) .... %; Hardness (HB, HV10 or HRC) ....
Heat treatment...
Description of teeth surface-hardening ....
....
....
Specified surface hardness (HB, HV10 or HRC)..
Depth of hardened layer versus hardness values (if possible in diagram) ....
....
....
Finishing method of tooth flanks (hobbed, shaved, lapped, ground or shot-peened teeth).......................
....
....
Specified surface roughness R_Z or R_a relevant to tooth flank and root fillet ....
Amount of tooth flank corrections (tip-relief, end-relief, crowning and helix correction) if any............
....
Specified grade of accuracy (according to ISO 1328) ....
....
Amount of shrinkage with tolerances specifying the procedure foreseen for shrinking and measures proposed to ensure the securing of rims. (11) ....

10 Relevant to core of material

11 In case of shrink-fitted pinions, wheel rims or hubs

VARIOUS ITEMS (indicate other elements, if any, in case of particular types of gears)

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PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 2 Propulsion Shafting

1 General

1.1 Application *(1 July 2006)*

This section applies to shafts, couplings, clutches and other power transmitting components for propulsion purposes.

Shafts and associated components used for transmission of power, essential for the propulsion of the vessel, are to be so designed and constructed to withstand the maximum working stresses to which they may be subjected in all service conditions.

Consideration may be given to designs based on engineering analyses, including fatigue considerations, as an alternative to the provisions of this section. Alternative calculation methods are 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).

Additional requirements for shafting intended for vessels strengthened for navigation in ice are provided in Part 6.

1.3 Definitions

For the purposes of using shaft diameter formulas in this section, the following definitions apply.

1.3.1 Tail Shaft

Tail Shaft is the part of the propulsion shaft aft of the forward end of the propeller end bearing.

1.3.2 Stern Tube Shaft

Stern Tube Shaft or *Tube Shaft* is the part of the propulsion shaft passing through the stern tube from the forward end of the propeller end bearing to the in-board shaft seal.

1.3.3 Line Shaft

Line Shaft is the part of the propulsion shaft in-board of the vessel.

1.3.4 Thrust Shaft

Thrust Shaft is that part of the propulsion shaft which transmits thrust to the thrust bearing.

1.3.5 Oil Distribution Shaft

Oil Distribution Shaft is a hollow propulsion shaft where the bore and radial holes are used for distribution of hydraulic oil in controllable pitch propeller installations.

1.5 Plans and Particulars to be Submitted

The following plans and particulars are to be submitted for review:

1.5.1 For Propulsion Shafting *(2008)*

Shafting arrangement

Rated power of main engine and shaft rpm

Thrust, line, tube and tail shafts, as applicable

Couplings – integral, demountable, keyed, or shrink-fit, coupling bolts* and keys

Engineering analyses and fitting instructions for shrink-fit couplings

Shaft bearings

Stern tube

Shaft seals

Shaft lubricating system

Power take-off to shaft generators, propulsion boosters, or similar equipment, rated 100 kW (135 hp) and over, as applicable

Materials

* *Note:* Specific details regarding the interference fit of the coupling bolts are to be submitted. In addition, calculations and detail design basis for the sizing of the fitted bolts are to be submitted if the sizing of the bolts as per 4-3-2/5.19.1 of the Rules is not based on as-built line shaft diameter "*D*".

1.5.2 For Clutches

Construction details of torque transmitting components, housing along with their materials and dimensions.

Rated power and rpm

Engineering analyses

Clutch operating data

1.5.3 For Flexible Couplings

Construction details of torque transmitting components, housing, along with their dimensions and materials

Static and dynamic torsional stiffness and damping characteristics

Rated power, torque, and rpm.

Engineering analyses

Allowable vibratory torque for continuous and transient operation.

Allowable power loss (overheating)

Allowable misalignment for continuous operation

1.5.4 For Cardan Shafts

Dimensions of all torque transmitting components and their materials

Rated power of main engine and shaft rpm

Engineering analyses

Clutch operating data

1.5.5 Calculations

Propulsion shaft alignment calculations where propulsion shaft is sensitive to alignment (see 4-3-2/7.3).

Torsional vibration analyses

Axial and lateral (whirling) vibration calculations where there are barred speed ranges within engine operating speed range

3 Materials

3.1 General

Materials for propulsion shafts, couplings and coupling bolts, keys and clutches are to be of forged steel or rolled bars, as appropriate, in accordance with Section 2-3-7 and Section 2-3-8 or other specifications as may be specially approved with a specific design. Where materials other than those specified in the Rules are proposed, full details of chemical composition, heat treatment and mechanical properties, as appropriate, are to be submitted for approval.

3.1.1 Ultimate Tensile Strength

In general, the minimum specified ultimate tensile strength of steel used for propulsion shafting is to be between 400 N/mm^2 (40.7 kgf/mm^2, 58,000 psi) and 800 N/mm^2 (81.5 kgf/mm^2, 116,000 psi).

3.1.2 Elongation

Material with elongation (L_o/d = 4) of less than 16% is not to be used for any shafting component, with the exception that material for non-fitted alloy steel coupling bolts manufactured to a recognized standard may have elongation of not less than 10%.

3.3 Weldability *(2008)*

Where repair by welding or where cladding by welding is contemplated, steel used for propulsion shafts is to have carbon content in accordance with 2-3-7/1.1.2. For approval of welding of the shaft, refer to Appendix 7-A-11 “Guide for Repair and Cladding of Shafts” of the ABS *Rules for Survey After Construction (Part 7)*.

3.5 Shaft Liners

Liners may be of bronze, stainless steel or other approved alloys and are to be free from porosity and other defects. Continuous liners are to be in one piece or, if made of two or more lengths, the joining of the separate pieces is to be done by an approved method of welding through not less than two-thirds the thickness of the liner or by an approved rubber seal arrangement.

3.7 Material Tests

3.7.1 General

Materials for all torque-transmitting parts, including shafts, clutches, couplings, coupling bolts and keys are to be tested in the presence of the Surveyor. The materials are to meet the specifications of 2-3-7/5. 2-3-7/7 and 2-3-8/1 or other specifications approved in connection with the design.

3.7.2 Alternative Test Requirements

3.7.2(a) 375 kW (500 hp) or less. Materials for parts transmitting 375 kW (500 hp) or less may be accepted by the Surveyor based on verification of manufacturer’s certification and witnessed hardness check.

3.7.2(b) Coupling bolts. Coupling bolts manufactured and marked to a recognized standard will not require material testing.

3.7.3 Inspections and Nondestructive Tests

Shafting and couplings are to be surface examined by the Surveyor.

Forgings for tail shafts 455 mm (18 in.) and over in finished diameter are to be ultrasonically examined in accordance with 2-3-7/1.13.2. Tail shafts in the finished machine condition are to be subjected to magnetic particle, dye penetrant or other nondestructive examinations. They are to be free of linear discontinuities greater than 3.2 mm ($^1/_8$ in.), except that in the following locations the shafts are to be free of all linear discontinuities:

3.7.3(a) Tapered tail shafts: the forward one-third length of the taper, including the forward end of any keyway and an equal length of the parallel part of the shaft immediately forward of the taper.

3.7.3(b) Flanged tail shafts: the flange fillet area.

5 Design and Construction

5.1 Shaft Diameters

The minimum diameter of propulsion shafting is to be determined by the following equation:

$$D = 100K \cdot \sqrt[3]{\frac{H}{R}\left(\frac{c_1}{U + c_2}\right)}$$

where

D = required solid shaft diameter, except hollow shaft; mm (in.)

H = power at rated speed; kW (PS, hp) (1 PS = 735 W; 1 hp = 746 W)

K = shaft design factor, see 4-3-2/Table 1 or 4-3-2/Table 2

R = rated speed rpm

U = minimum specified ultimate tensile strength of shaft material (regardless of the actual minimum specified tensile strength of the material, the value of U used in these calculations is not to exceed that indicated in 4-3-2/Table 3; N/mm^2 (kgf/mm^2, psi)

c_1 and c_2 are given below:

	SI units	*MKS units*	*US units*
c_1	560	41.95	3.695
c_2	160	16.3	23180

TABLE 1
Shaft Design Factors K and C_K for Line Shafts and Thrust Shafts *(2006)*

Factor	*Propulsion drives*	*Design features (1)*							
		Integral flange	*Shrink fit coupling*	*Key-ways (2)*	*Radial holes, transverse holes (3)*	*Longitud-inal slots (4)*	*On both sides of thrust collars*	*In way of axial bearings used as thrust bearings*	*Straight sections*
K	Type A	0.95	0.95	1.045	1.045	1.14	1.045	1.045	0.95
	Type B	1.0	1.0	1.1	1.1	1.2	1.1	1.1	1.0
C_K		1.0	1.0	0.6	0.5	0.3	0.85	0.85	1.0

Type A: Turbine drives; electric drives; diesel drive through slip couplings (electric or hydraulic).

Type B: All other diesel drives.

Notes

1 Geometric features other than those listed will be specially considered.

2 After a length of not less than 0.2D from the end of the keyway, the shaft diameter may be reduced to the diameter calculated for straight sections.

Fillet radii in the transverse section of the keyway are not to be less than 0.0125D.

3 Diameter of bore not more than 0.3D.

4 Length of the slot not more than 1.4D, width of slot not more than 0.2D, whereby D is calculated with K = 1.0.

TABLE 2
Shaft Design Factors K and C_K for Tail Shafts and Stern Tube Shafts [1] *(2006)*

Factor	*Propulsion drive*	*Stern tube configuration*	*Tail shafts: propeller attachment method [2]*			*Stern tube shafts [7, 8]*
			Keyed [3]	*Keyless attachment by shrink fit [4]*	*Flanged [5]*	
K	All	Oil lubricated bearings	1.26	1.22	1.22	1.15
	All	Water lubricated bearings: continuous shaft liners or equivalent (see 4-3-2/5.17.6)	1.26	1.22	1.22	1.15
	All	Water lubricated bearings: non-continuous shaft liners [6]	1.29	1.25	1.25	1.18
C_K			0.55	0.55	0.55	0.8

Notes

1. Tail shaft may be reduced to stern tube shaft diameter forward of the bearing supporting the propeller, and the stern tube shaft reduced to line shaft diameter inboard of the forward stern tube seal.
2. Other attachments are subject to special consideration.
3. Fillet radii in the transverse section at the bottom of the keyway are not to be less than 0.0125D.
4. See also 4-3-2/5.11 and 4-3-3/5.15.2.
5. For flange fillet radii and flange thickness, see 4-3-2/5.19.3.
6. For Great Lakes Service, K factor corresponding to continuous liner configuration may be used.
7. K factor applies to shafting between the forward edge of the propeller-end bearing and the inboard stern tube seal.
8. Where keyed couplings are fitted on stern tube shaft, the shaft diameters are to be increased by 10% in way of the coupling. See Note 2 of 4-3-2/Table 1.

TABLE 3
Maximum Values of U to be Used in Shaft Calculations *(1 July 2006)*

	SI units N/mm^2	*MKS units* kgf/mm^2	*US units psi*
1. For all alloy steel shafts except tail shafts and tube shafts stated in 3 and 4 below.	800	81.5	116,000
2. For all carbon and carbon-manganese shafts except tail shafts and tube shafts stated in 3 and 4 below.	760	77.5	110,200
3. For tail shafts and tube shafts in oil lubricated bearings or in saltwater lubricated bearings but fitted with continuous liner or equivalent (see 4-3-2/5.17.6).	600	61.2	87,000
4. For tail shafts and tube shafts in saltwater lubricated bearings fitted with non-continuous liners.	415	42.2	60,000

5.3 Hollow Shafts

For hollow shafts where the bore exceeds 40% of the outside diameter, the minimum outside shaft diameter is not to be less than that determined through successive approximation utilizing the following equation:

$$D_o = D\sqrt[3]{\frac{1}{[1-(D_i/D_o)^4]}}$$

where

D_o = required outer diameter of shaft; mm (in.)

D = solid shaft diameter required by 4-3-2/5.1; mm (in.)

D_i = actual inner diameter of shaft; mm (in.)

5.5 Alternative Criteria

As an alternative to the design equations shown in 4-3-2/5.1 and 4-3-2/5.3, shafting design may be considered for approval on the basis of axial and torsional loads to be transmitted, bending moment and resistance against fatigue. A detailed stress analysis showing a factor of safety of at least 2.0 for fatigue failure is to be submitted for approval with all supporting data.

5.7 Key *(2006)*

In general, the key material is to be of equal or higher strength than the shaft material. The effective area of the key in shear is to be not less than *A*, given below. The effective area is to be the gross area subtracted by materials removed by saw cuts, set screw holes, chamfer, etc., and is to exclude the portion of the key in way of spooning of the key way.

Note: Keyways are, in general, not to be used in installations with slow speed, crosshead or two-stroke engines with a barred speed range.

$$A = \frac{D^3}{5.1 r_m} \cdot \frac{Y_S}{Y_K}$$

where

A = shear area of key; mm^2 (in^2)

D = line shaft diameter; mm (in.); as determined by 4-3-2/5.1

r_m = shaft radius at mid-length of the key; mm (in.)

Y_S = specified yield strength of shaft material; N/mm^2 (kgf/mm^2, psi)

Y_K = specified yield strength of key material; N/mm^2 (kgf/mm^2, psi)

5.9 Strengthening for Navigation in Ice

For vessels to be assigned with **Ice Class** notations, shafting is to be designed in accordance with 6-1-1/55 or 6-1-2/29.

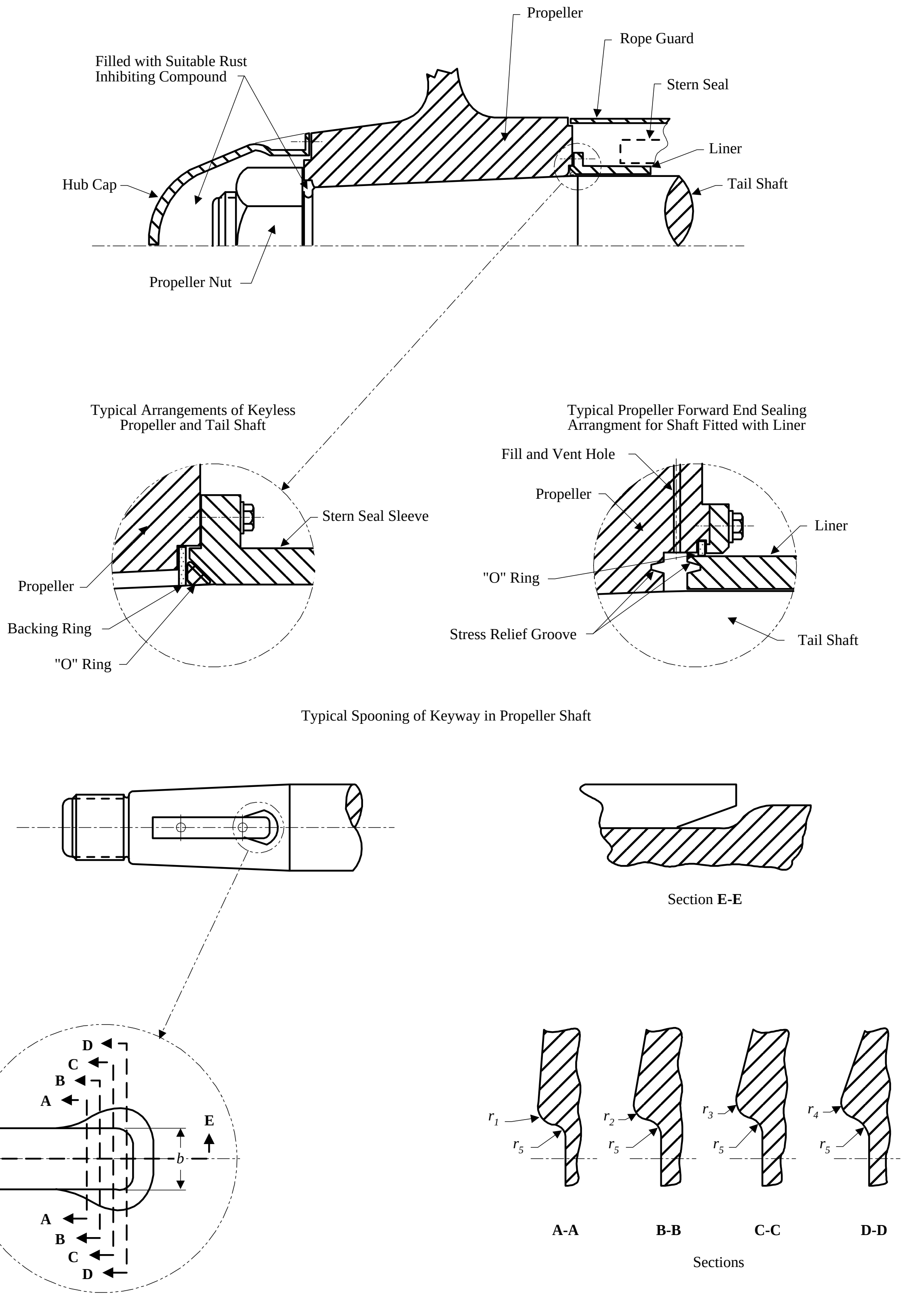
5.11 Tail Shaft Propeller-end Design

Tail shafts are to be provided with an accurate taper fit in the propeller hub, particular attention being given to the fit at the large end of the taper. In general, the actual contact area is to be at least 70% of the theoretical contact area. The key is to fit tightly in the keyway and be of sufficient size (see 4-3-2/5.7) to transmit the full torque of the shaft, but it is not to extend into the propeller hub counterbore (to accommodate the liner) on the forward side of the propeller hub. The forward end of the keyway is to be so cut in the shaft as to give a gradual rise from the bottom of the keyway to the surface of the shaft (see 4-3-2/Figure 1). Ample fillets (see Note 2 of 4-3-2/Table 1) are to be provided in the corners of the keyway and, in general, stress concentrations are to be reduced as far as practicable.

5.13 Propeller-end Seal

Effective means are to be provided to prevent water having access to the shaft at the part between the after end of the liner and the propeller hub and between the shaft and the propeller. See typical sealing arrangements in 4-3-2/Figure 1. See also 4-3-3/9.5.

FIGURE 1
Typical Arrangements and Details of Fitting of Tail Shaft and Propeller



5.15 Tail Shaft Bearings

5.15.1 Water-lubricated Bearings

The length of the bearing, next to and supporting the propeller, is to be not less than four times the required tail-shaft diameter. However, for bearings of rubber, reinforced resins, or plastic materials, the length of the bearing, next to and supporting the propeller, may be less than four times, but not less than two times the required tail shaft diameter, provided the bearing design is being substantiated by experimental tests to the satisfaction of the Bureau.

5.15.2 Oil-lubricated Bearings

5.15.2(a) White metal. 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 (0.0815 kgf/mm^2, 116 psi), 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.

5.15.2(b) Synthetic material. 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 (0.0611 kgf/mm^2, 87 psi), 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.

5.15.2(c) Cast iron or bronze. 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.

5.15.2(d) Stern tube bearing oil lubricating system sampling arrangement (2001). 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.

5.15.3 Grease-lubricated Bearings

The length of grease-lubricated bearings is to be not less than four times the diameter of the required tail shaft diameter.

5.17 Tail Shaft Liners

5.17.1 Thickness at Bearings

5.17.1(a) Bronze liner. The thickness of bronze liners to be fitted to tail shafts or tube shafts is not to be less than that given by the following equation.

$$t = \frac{T}{25} + 5.1 \text{ mm} \quad \text{or} \quad t = \frac{T}{25} + 0.2 \text{ in.}$$

where

t = thickness of liner; mm (in.)

T = required diameter of tail shaft; mm (in.)

5.17.1(b) Stainless steel liner. The thickness of stainless steel liners to be fitted to tail shafts or tube shafts is not to be less than one-half that required for bronze liners or 6.5 mm (0.25 in.), whichever is greater.

5.17.2 Thickness Between Bearings

The thickness of a continuous bronze liner between bearings is to be not less than three-fourths of the thickness required in way of bearings.

5.17.3 Liner Fitting

All liners are to be carefully shrunk or forced upon the shaft by pressure and they are not to be secured by pins. If the liner does not fit the shaft tightly between the bearing portions, the space between the shaft and liner is to be filled by pressure with an insoluble non-corrosive compound.

5.17.4 Glass Reinforced Plastic Coating

Glass reinforced plastic coatings may be fitted on propulsion shafting when applied by an approved procedure to the satisfaction of the Surveyor. Such coatings are to consist of at least four plies of cross-woven glass tape impregnated with resin, or an equivalent process. Prior to coating, the shaft is to be cleaned with a suitable solvent and grit-blasted. The shaft is to be examined prior to coating and the first layer is to be applied in the presence of the Surveyor. Subsequent to coating, the finished shaft is to be subjected to a spark test or equivalent to verify freedom from porosity to the satisfaction of the Surveyor. In all cases where reinforced plastic coatings are employed, effective means are to be provided to prevent water from gaining access to the metal of the shaft. Provisions are to be made for overlapping and adequately bonding the coating to fitted or clad liners. The end of the liner is to be stepped and tapered as required to protect the end of the wrapping.

5.17.5 Stainless Steel Cladding

Stainless steel cladding of shafts is to be carried out in accordance with Appendix 7-A-11 "Guide for Repair and Cladding of Shafts" of the ABS *Rules for Survey After Construction (Part 7).*

5.17.6 Continuous Liner or Equivalent

Stainless steel cladding in 4-3-2/5.17.5 and metallic liners in 4-3-2/5.17.1, if of non-continuous construction but if the exposed shaft is protected with fiber glass reinforced plastic coating in accordance with 4-3-2/5.17.4, may be credited as "continuous" liners for purposes of:

- Determining required tail shaft and tube shaft diameters (see 4-3-2/5.1 and 4-3-2/5.3), and
- Periodical tail shaft survey [see 7-2-1/13.1.2(c)].

5.19 Couplings and Coupling Bolts

5.19.1 Fitted Bolts *(2008)*

The minimum diameter of fitted shaft coupling bolts is to be determined by the following equation. The bolts are to be assembled with an interference fit.

$$d_b = 0.65\sqrt{\frac{D^3(U+c)}{NBU_b}}$$

where

B = bolt circle diameter; mm (in.)

c = constant, as given below

	SI unit	*MKS unit*	*US unit*
c	160	16.3	23,180

d_b = diameter of bolt at joints; mm (in.)

D = minimum required shaft diameter designed considering the largest combined torque (static and dynamic), acting at the shaft in vicinity of the respective coupling flanges; mm (in), see 4-3-2/7.5 but not less than the minimum required line shaft diameter (see 4-3-2/5.1); mm (in.)

N = number of bolts fitted in one coupling

U = minimum specified tensile strength of shaft material, as defined in 4-3-2/5.1; N/mm^2 (kgf/mm^2, psi)

U_b = minimum specified tensile strength of bolt material; N/mm^2 (kgf/mm^2, psi), subject to the following conditions:

a) Selected bolt material is to have minimum specified tensile strength U_b at least equal to U.

b) Regardless of the actual minimum tensile strength, the value of U_b used in these calculations is not to exceed $1.7U$ nor 1000 N/mm^2 (102 kgf/mm^2, 145,000 psi).

5.19.2 Non-fitted Bolts

The diameter of pre-stressed non-fitted coupling bolts will be considered upon the submittal of detailed preloading and stress calculations and fitting instructions. The tensile stress of the bolt due to prestressing and astern pull is not to exceed 90% of the minimum specified yield strength of the bolt material. In addition, the bearing stress on any member such as the flange, bolt head, threads or nut is not to exceed 90% of the minimum specified yield strength of the material of that member.

For calculation purpose, to take account of torsional vibratory torque, the following factors may be applied to the transmitted main torque, unless the actual measured vibratory torque is higher, in which case, the actual vibratory torque is to be used:

- For direct diesel engine drives: 1.2
- For all other drives and for diesel engine drives with elastic coupling: 1.0

5.19.2(a) Torque transmission by friction. Where torque is to be transmitted by friction provided by prestressed non-fitted bolts only and the bolts are under pure tension, the factor of safety against slip under the worst operating conditions, including mean transmitted torque plus torque due to torsional vibration, is to be at least as follows:

- Inaccessible couplings (external to the hull or not readily accessible): 2.8
- Accessible couplings (internal to the hull): 2.0

5.19.2(b) Torque transmission by combined friction and shear. Where torque is to be transmitted by combination of fitted bolts and prestressed non-fitted bolts, the components are to meet the following criteria:

- *Fitted bolts.* The shear stress under the maximum torque corresponding to the worst loaded condition is to be not more than 50% of the minimum specified tensile yield strength of the bolt material.
- *Non-fitted bolts.* The factor of safety against slip, under the maximum torque corresponding to the worst loaded condition and the specified bolt tension, is to be at least 1.6 for inaccessible couplings and 1.1 for accessible couplings.

5.19.2(c) Torque transmission by dowels. Dowels connecting the tail shaft flange to the controllable pitch propeller hub, utilized with prestressed non-fitted bolts to transmit torque, are considered equivalent to fitted bolts and are to comply with 4-3-2/5.19.1 and, if applicable, 4-3-2/5.19.2(b). The dowels are to be accurately fitted and effectively secured against axial movement.

5.19.3 Flanges

5.19.3(a) Flange thickness. The thickness of coupling flanges integral to the shaft is not to be less than the minimum required diameter of the coupling bolts or $0.2D$, where D is as defined in 4-3-2/5.1, whichever is greater.

The fillet radius at the base of a coupling flange is not to be less than 0.08 times the actual shaft diameter. Consideration will be given to fillets of multiple radii design; such fillet is normally to have a cross-sectional area not less than that of a required single-radius fillet. In general, the surface finish for fillet radii is not to be rougher than 1.6 μm (63 μin) RMS. Alternatively, 1.6 μm CLA (center line average) may be accepted.

5.19.3(b) Flange thickness – connection to controllable pitch propeller. The thickness of the coupling flange integral to the tail shaft for connection to the forward face of the controllable pitch propeller hub is to be not less than 0.25*D*, where *D* is as defined in 4-3-2/5.1.

For the tail shaft flange supporting the propeller, the fillet radius at the base of the flange is to be at least 0.125*D*. Special consideration will be given to fillets of multiple-radius design; see 4-3-2/5.19.3(a). The fillet radius is to be accessible for nondestructive examination during tail shaft surveys. See 7-5-1/3.5.

5.19.4 Demountable Couplings

The strength of demountable couplings and keys is to be equivalent to that of the shaft. Couplings are to be accurately fitted to the shaft. Where necessary, provisions for resisting thrust loading are to be provided.

Hydraulic and other shrink fit couplings will be specially considered upon submittal of detailed preload and stress calculations and fitting instructions. In general, the torsional holding capacity under nominal working conditions and based on the minimum available interference fit (or minimum pull-up length) is to be at least 2.8 times the transmitted mean torque plus torque due to torsional vibration (see 4-3-2/5.19.2) for inaccessible couplings (external to the hull or not readily accessible). This factor may be reduced to 2.0 times for accessible couplings (internal to the hull). The preload stress under nominal working conditions and based on the maximum available interference fit (or maximum pull-up length) is not to exceed 70% of the minimum specified yield strength.

The following friction coefficients are to be used:

Oil injection method of fit:	0.13
Dry method of fit:	0.18

5.19.5 Flexible Couplings

5.19.5(a) Design. Flexible couplings intended for use in propulsion shafting are to be of approved designs. Couplings are to be designed for the rated torque, fatigue and avoidance of overheating. Where elastomeric material is used as a torque-transmitting component, it is to withstand environmental and service conditions over the design life of the coupling, taking into consideration the full range of maximum to minimum vibratory torque. Flexible coupling design will be evaluated, based on submitted engineering analyses.

5.19.5(b) Torsional displacement limiter. Flexible couplings with elastomer or spring type flexible members, whose failure will lead to total loss of propulsion capability of the vessel, such as that used in the line shaft of a single propeller vessel, are to be provided with a torsional displacement limiter. The device is to lock the coupling or prevent excessive torsional displacement when a pre-determined torsional displacement limit is exceeded. Operation of the vessel under such circumstances may be at reduced power. Warning notices for such reduced power are to be posted at all propulsion control stations.

5.19.5(c) Barred range. Conditions where the allowable vibratory torque or the allowable dissipated power may be exceeded under the normal operating range of the engine are to be identified and are to be marked as a barred range in order to avoid continuous operation within this range.

5.19.5(d) Diesel generators. Flexible couplings for diesel generator sets are to be capable of absorbing short time impact torque due to electrical short-circuit conditions up to 6 (six) times the nominal torque.

5.19.6 Clutches *(2002)*

5.19.6(a) Design. Clutches intended for use in propulsion shafting are to be of approved design. They are to be designed to transmit the maximum power at rated speed. The minimum service factor, determined by the ratio of the clutch static holding capacity to the rated torque, is to be as follows:

Clutch Design Type	*Minimum Service Factor*
Drum-type clutch or Disc type, air-actuated, air cooled clutches	
Shafting system fitted with fixed pitch propeller	1.7
Shafting system fitted with fixed pitch propeller and shaft brake	1.5
Shafting system fitted with controllable pitch propeller	1.5
Hydraulically-actuated, oil cooled multiple-plate clutches	1.7

The minimum service factor will be required to be increased if the shafting vibratory torque is excessive, clutch thermal capacity is exceeded because of frequent clutch engagements during vessel operations, the clutch shoe material used has limited service experience or the clutch will be allowed to slip during vessel operations. Calculations are to be submitted for review.

5.19.6(b) System arrangements. Arrangements are to be made such that, in the event of failure of the clutch actuating system, each clutch remains capable of being engaged and transmitting an adequate power considered necessary for propulsion and maneuvering of the vessel.

5.19.6(c) Coupling bolts. Coupling bolts are to comply with 4-3-2/5.19.1 and 4-3-2/5.19.2 and are to be of sufficient strength to support the weight of the elements, as well as to transmit all necessary forces.

5.19.7 Locking Arrangement

After assembly, all coupling bolts and associated nuts are to be fitted with locking arrangement.

5.21 Cardan Shaft

Cardan shafts are to be designed in accordance with the equation for propulsion shaft in 4-3-2/5.1, and flanges and bolts are to be in accordance with 4-3-2/5.19.1, 4-3-2/5.19.2 and 4-3-2/5.19.3. The design of splines, yokes and cross-members are to be evaluated based on engineering analyses which are to be submitted for review. Where applicable, the cardan shaft assembly is to contain provisions for bearing thrust or pull from the propeller.

7 Propulsion Shaft Alignment and Vibrations

7.1 General

In addition to the design requirements addressed above, considerations are to be given to additional stresses in the shafting system given rise to by shaft alignment in relation to location and spacing of the shaft bearings, and by axial, lateral and torsional vibrations.

7.3 Shaft Alignment Calculations *(2010)*

In general, shaft alignment calculations, as well as a shaft alignment procedure, are to be submitted for reference. However, calculations for the following alignment-sensitive types of installations are to be submitted for review:

i) Propulsion shafting of diameter greater than 300 mm (11.81 in.).

ii) Propulsion shafting with reduction gears where the bull gear is driven by two or more ahead pinions.

iii) Propulsion shafting with power take-off or with booster power arrangements.

iv) Propulsion shafting for which the tail shaft bearings are to be slope-bored.

The alignment calculations are to include bearing reactions, shear forces and bending moments along the shafting, slope boring details (if applicable) and detailed description of alignment procedure.

The alignment calculations are to be performed for theoretically aligned cold and hot conditions of the shaft with specified alignment tolerances.

Calculations are to be performed for the maximum allowable alignment tolerances and are to show that:

- Bearing loads under all operating conditions are within the acceptable limits specified by the bearing manufacturer.
- Bearing reactions are always positive (i.e., supporting the shaft).
- Shear forces and bending moments on the shaft are within acceptable limits in association with other stresses in the shaft.
- Forces and moments on propulsion equipment are within the limits specified by the machinery manufacturers.
- In general, if the calculated relative misalignment slope between the shaft and the tail shaft bearing is greater than $0.3*10^{-3}$ [rad], then consideration is to be given to reducing the relative misalignment slope by means of slope-boring or bearing inclination. The slope boring angle calculation is to be based on the static afloat conditions with hot engine.

7.5 Torsional Vibrations *(1 July 2006)*

7.5.1 Allowable Torsional Vibration Stress *(1 July 2006)*

The torsional vibration stress in the propulsion shafting system is not to exceed the allowable vibratory stress, *S*, given in 4-3-2/Table 4. The analysis of torsional vibrations shall account for stresses resulting from vector summation of responses (synthesis) of all relevant excitation harmonics.

The stress limit *S* is applicable for propulsion shafting systems, including types of couplings, dampers, clutches, etc., where torsional vibratory torque is the only load of significance.

For propulsion shafts, and equipment integral to the shaft, where vibratory torque is not the only significant source of load, the stress limit *S* does not apply. Design criteria of such shafts are contained in the following applicable sections:

i) Crankshafts: see 4-2-1/5.9,

ii) Turbine rotor shafts: see 4-2-3/5.1, and 4-2-4/5.3

iii) Gear shafts: see 4-3-1/5.9

iv) Electric motor shafts: see 4-8-3/3.11

v) Generator shafts: see 4-8-3/3.11

vi) Other shafts and equipment that falls under the subject criteria need to be designed considering maximum combined load acting within operating speed range of the propulsion system.

TABLE 4
Allowable Torsional Vibratory Stress *(1 July 2006)*

	SI units	*MKS units*	*US units*
S = allowable vibratory stress	$\frac{U+160}{18} C_K C_D C_r$ N/mm^2	$\frac{U+16.3}{18} C_K C_D C_r$ kgf/mm^2	$\frac{U+23180}{18} C_K C_D C_r$ psi
U = minimum tensile strength of shaft material	To be taken as not more than 600 N/mm^2 (see Note)	To be taken as not more than 61.2 kgf/mm^2 (see Note)	To be taken as not more than 87,000 psi (see Note)
C_K = shaft design factor	See 4-3-2/Tables 1 and 2.		
C_D = size factor	$0.35+\frac{0.93}{\sqrt[5]{d}}$	$0.35+\frac{0.93}{\sqrt[5]{d}}$	$0.35+\frac{0.487}{\sqrt[5]{d}}$
d = actual shaft diameter	mm	mm	in.
C_r = speed ratio factor	$3-2\lambda^2$ for $\lambda < 0.9$; 1.38 for $0.9 \le \lambda \le 1.05$		
λ	$\lambda = \frac{\text{Critical Speed (RPM) at which vibratory stress is calculated}}{\text{rated speed (RPM)}}$		

Note: *(1 July 2006)* Regardless of the actual minimum specified tensile strength of the shaft (tail shaft, tube shaft, line shaft and crankshaft, as applicable) material, the value of *U* used in these calculations is not to exceed the values indicated. Higher values of *U*, but not exceeding 800 N/mm^2 (81.5 kgf/mm^2, 116,000 psi), may be specially considered for the line shaft.

7.5.2 Diesel Engine Installations *(1 July 2006)*

For diesel engine installations, vibratory stresses are to be calculated with any one cylinder not firing and the calculations are to be submitted for information.

7.5.3 Barred Speed Ranges *(1 July 2006)*

When torsional vibratory stresses exceed the foregoing limits at an rpm within the operating range but less than 80% of rated speed, a barred range is to be provided. The allowable vibratory stress in a barred range due to the alternating torsional vibrations is not to exceed the values given by the following:

$$S_2 = \frac{1.7S}{\sqrt{C_k}} \quad \text{for } \lambda \le 0.8$$

where

S_2 = allowable vibratory stress within a barred range, N/mm^2 (kgf/mm^2, psi)

λ, S, C_k are as defined in 4-3-2/7.5.1.

Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the shaft material is to have a specified minimum ultimate tensile strength of not less than 500 N/mm^2 (50.9 kgf/mm^2, 72,500 psi). Otherwise materials having a specified minimum ultimate tensile strength of not less than 400 N/mm^2 (40.8 kgf/mm^2, 58,000 psi) may be used.

Barred ranges are not acceptable in the speed range between 0.8 and 1.05 of the rated speed. The existence of a barred range at speeds less than 0.8 of the rated speed is to be considered in establishing standard operating speeds for the vessel. The width of the barred range is to take into consideration the breadth and severity of the critical speed but is not to be less than the following limits:

$$\frac{16n_c}{18-\lambda} \ge n_\ell \qquad \text{and} \qquad \frac{(18-\lambda)n_c}{16} \le n_u$$

where

n_c = critical speed

n_ℓ = lower limit

n_u = upper limit

λ is as defined in 4-3-2/Table 4.

7.5.4 Marking of Tachometer and Alarms

Where a barred speed range is identified as in 4-3-2/7.5.3, the tachometer is to be marked and a warning notice is to be displayed at all propulsion control stations (local and remote) to caution that operation in the barred range is to be avoided except for passing through. Where remote propulsion control is fitted on the navigation bridge or where a centralized control station is fitted, means are to be provided at these remote propulsion control stations to alert the operator of any operation of the propulsion drive within the barred range. This may be achieved by a visual display or alarm.

7.5.5 Other Effects

Because critical torsional vibration has deleterious effects other than shafting fatigue, the limits in 4-3-2/7.5.1 are not intended for direct application as design factors, and it is desirable that the service range above 90% of rated speed be kept clear of torsional critical speeds insofar as practicable.

7.5.6 Torsiograph Tests *(2006)*

When the calculation indicates that criticals occur within the operating range, whose severity approaches or exceeds the limits in 4-3-2/7.5.1, torsiograph tests may be required to verify the calculations and to assist in determining ranges of restricted operation.

7.5.7 Vibration Dampers

When torsional vibratory stresses exceed the limits in 4-3-2/7.5.1 and a barred range is not acceptable, the propulsion system is to be redesigned or vibration dampers are to be fitted to reduce the stresses.

7.5.8 Gears

When the propeller is driven through reduction gear, or when geared booster power or power take-off is provided, a barred range is to be provided at the acceptable critical speed if gear tooth chatter occurs during continuous operation at this speed.

7.7 Axial Vibrations

The designer or the builder is to evaluate the shafting system to ensure that axial vibration characteristics in association with diesel engine or propeller blade-rate frequency forces will not result in deleterious effects throughout the engine operating speed range, with consideration also given to the possibility of the coupling of torsional and axial vibration, unless experience with similar shafting system installations makes it unnecessary. The axial vibrations may be controlled by axial vibration detuners to change the natural frequency of the system or by axial vibration dampers to limit the amplitude of axial vibrations to an acceptable level.

When on the basis of axial vibration calculations the designer or builder proposed to provide barred speed ranges within the engine operating speed range, the calculations are to be submitted for information. The barred speed ranges due to axial vibrations are to be verified and established by measurement.

7.9 Lateral (Whirling) Vibrations

The designer or the builder is to evaluate the shafting system to ensure that the amplitudes of lateral (whirling) vibration are of acceptable magnitude throughout the engine operating speed range, unless experience with similar shafting system installations makes it unnecessary.

When on the basis of lateral vibration calculations, the designer or builder proposed to provide barred speed ranges within the engine operating speed range, the calculations are to be submitted for information. The barred speed ranges due to lateral vibration are to be verified and established by measurement.

9 Inspection, Testing and Certification

9.1 General

Shafting components are to be inspected, tested and certified by a Surveyor at the plant of the manufacturer in accordance with the following requirements.

9.3 Material Testing

For testing of shafting component materials, see 4-3-2/3.7.

9.5 Propulsion Shafts and Associated Parts

9.5.1 Power Transmitting Parts

All propulsion shafts and associated parts, such as coupling bolts, are to be visually examined for surface flaws, out of roundness, straightness, and dimensional tolerances. The Surveyor, in case of doubts, may require additional nondestructive testing. See 4-3-2/3.7.3 for tail shaft requirements.

9.5.2 Liners

Shaft liners are to prove tight under hydrostatic test of 1.0 bar (1 kgf/cm^2, 15 psi). After assembly, the fit of the liner to the shaft is to be checked for freedom from voids. Any void in way of bearings is to be dealt with as in 4-3-2/5.17.3.

9.7 Flexible Couplings, Clutches, Cardan Shafts, etc.

Manufactured torque transmitting parts, such as flexible couplings, clutches (independent of the gear assembly), cardan shafts, etc. are to be inspected, tested, and certified by a Surveyor at the plant of manufacture. Alternatively, these parts may be certified under Type Approval Program (see 1-1-4/7.7).

11 Installation and Trials

11.1 Shaft Alignment

11.1.1 Alignment *(2010)*

Shaft alignment is to be carried out in the presence of a Surveyor. Final alignment is to be verified in the afloat condition with superstructure in place and major welding work completed. When alignment calculations are required to be submitted in accordance with 4-3-2/7.3, the alignment calculated data are to be verified and recorded by appropriate measurement procedures in the presence and to the satisfaction of a Surveyor, in accordance with 4-3-2/11.1.1(a), and 4-3-2/11.1.1(b) and 4-3-2/11.1.1(c) as applicable.

11.1.1(a) Bearing Reaction. Bearing reactions are required to be verified by such means as hydraulic jack method and / or strain gauge method on all accessible shafting bearings namely:

i) Forward stern tube bearing

ii) Intermediate shaft bearing(s)

iii) Minimum three aftmost main engine bearings (for directly coupled propulsion systems only)

iv) Main-gear shaft bearings

11.1.1(b) Vessels with Hull Deflections not Accounted. For vessels where hull deflections are not explicitly accounted for in alignment design, and the shaft alignment analysis is based on a ship structure not affected by deflection, additional alignment verification is required, and is to consist of:

i) A verification of the bearing reactions for at least one service-draft condition of the vessel.

11.1.1(c) Sighting Through and Bearing Positioning Conducted in Block Stage. In cases where sighting through and bearing positioning are conducted in block stage of the vessel construction, and a monitoring system to verify shaft-bearing alignment is not installed, the verification of the following procedures is required:

i) Slope boring angle (as applicable)

ii) Bearing vertical offset positioning

iii) Engine vertical offset positioning

iv) Sag and gap procedure

11.1.2 Cast Resin Chocks

Resin chocks, intended for chocking of the shaft bearing foundation or stern tube, are to be of an approved type (see 1-1-A3/5 for type approval). Resin chocks are not to be relied upon to maintain watertight integrity of the hull or the oiltight integrity of the lubricating oil system. Accordingly, direct contact of resin chocks with water or oil is to be avoided. Where used, the arrangements and installation procedures are to be in accordance with the manufacturer's recommendations. Arrangements of the proposed installation, along with installation parameters such as deadweight, holding-down bolt tightening torque, etc., and calculations showing that the manufacturer's specified allowable pressure is not exceeded, are to be submitted for review in each case.

11.3 Vibration Measurement

11.3.1 Torsional Vibration

Where torsiograph measurement is required as per 4-3-2/7.5.6, the measurement is to be taken in the presence of a Surveyor.

When a barred speed range is provided in accordance with 4-3-2/7.5.3, tachometer marking, warning notice, and alarms at remote control stations (where fitted), as described in 4-3-2/7.5.4, are to be fitted.

Electronic speed regulating devices may be preset to step-pass the barred range in addition to the warning notice.

When the propeller is driven through reduction gears, the Surveyor is to ascertain that no gear-tooth chatter occurs throughout the operating range. Otherwise, a barred speed range as per 4-3-2/7.5.3 is to be provided; see 4-3-2/7.5.8.

11.3.2 Axial and Lateral Vibrations

When calculations indicate that barred speed ranges are present as per 4-3-2/7.7 and 4-3-2/7.9, these barred speed ranges are to be verified and recorded by appropriate measurement procedures in the presence and to the satisfaction of a Surveyor.

11.5 Circulating Currents

Where means are provided to prevent circulating currents from passing between the propeller, shaft and the hull, a warning notice plate is to be provided in a visible place cautioning against the removal of such protection.

11.7 Sea Trial

The shafting installation is to be tested during sea trials under various maneuvering conditions. It is to be free from dangerous vibration and to the satisfaction of the Surveyor.

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 3 Propellers

1 General

1.1 Application

This section applies to propellers intended for propulsion. It covers fixed pitch and controllable pitch propellers. Propellers for thrusters used for maneuvering and dynamic positioning are covered in Section 4-3-5. Performance of propellers, in respect to developing the designed output, is to be demonstrated during sea trials.

Additional requirements for propellers intended for vessels strengthened for navigation in ice are provided in Part 6.

1.3 Definitions

For purpose of this section, the following definitions apply.

1.3.1 Skew Angle

Skew Angle (θ) of a propeller is the angle measured from ray 'A' passing through the tip of blade at mid-chord line to ray 'B' tangent to the mid-chord line on the projected blade outline. See 4-3-3/Figure 1.

1.3.2 Highly Skewed Propeller

A *Highly Skewed Propeller* is one whose skew angle is more than 25°.

1.3.3 Propeller Rake

1.3.3(a) Rake. *Rake* is the distance at the blade tip between the generating line and the line perpendicular to the propeller axis that meets the generating line at the propeller axis. See 4-3-3/Figure 2.

1.3.3(b) Rake angle (ϕ). *Rake Angle* of a propeller is the angle measured from the plane perpendicular to shaft centerline to the tangent to the generating line at a specified radius (0.6 × radius for the purpose of this section). See 4-3-3/Figure 2.

FIGURE 1
Maximum Skew Angle

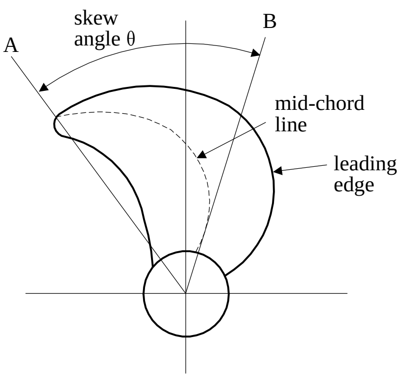
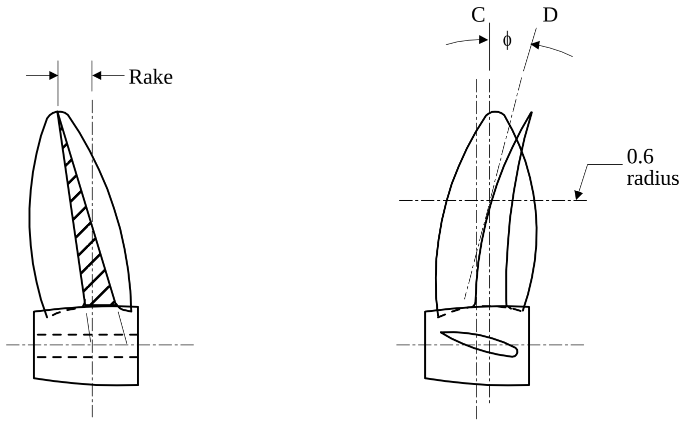


FIGURE 2
Rake and Rake Angle



1.5 Plans and Particulars to be Submitted

1.5.1 Fixed Pitch Propeller of Conventional Design

Material

Design characteristics of propeller

Dimensions and tolerances

Propeller plan

Blade thickness calculations

1.5.2 Controllable Pitch Propeller of Conventional Design

As per 4-3-3/1.5.1

Hub and hub to tail shaft flange attachment bolts

Propeller blade flange and bolts

Internal mechanism

Hydraulic piping control system

Instrumentation and alarm system

Strength calculations for internal mechanism

1.5.3 Highly Skewed Propeller and Other Unconventional Designs

In addition to the foregoing, where propeller blade designs are of the types for which the Rules do not provide simplified blade thickness calculations, such as

highly skewed propellers with $\theta > 50°$;

high skewed propellers made of other than Type 4 materials with $50° \geq \theta > 25°$;

controllable pitch propellers with $\theta > 25°$;

cycloidal propellers;

propeller load and stress analyses demonstrating adequacy of blade strength are to be submitted.

1.5.4 Keyless Propeller

Where propellers are to be fitted to the shaft without keys, stress calculations for hub stresses and holding capacity, along with fitting instructions, are to be submitted.

3 Materials

3.1 Normally Used Propeller Materials

4-3-3/Table 1 shows the properties of materials normally used for propellers. See 2-3-14/3 and Section 2-3-15 for full details of the materials.

Where an alternative material specification is proposed, detailed chemical composition and mechanical properties are to be submitted for approval (for example, see Section 2-3-14 and Section 2-3-15). The *f* and *w* values of such materials to be used in the equations hereunder will be specially considered upon submittal of complete material specifications including corrosion fatigue data to 10^8 cycles.

TABLE 1
Propeller Materials *(2008)*

		Tensile strength			*Yield strength*			*Elongation, % Gauge Length*	
Type	*Material*	*N/mm²*	*kgf/mm²*	*lb/in²*	*N/mm²*	*kgf/mm²*	*lb/in²*	*4d*	*5d*
2	Manganese bronze	450	46	65,000	175	18	25,000	20	18
3	Nickel-manganese bronze	515	53	75,000	220	22.5	32,000	18	16
4	Nickel-aluminum bronze	590	60	86,000	245	25	36,000	16	15
5	Manganese-nickel-aluminum bronze	630	64	91,000	275	28	40,000	20	18
CF-3	Stainless steel	485	49	70,000	205	21	30,000	35	32

3.3 Stud Materials

The material of the studs securing detachable blades to the hub is to be of at least Grade 2 forged steel or equally satisfactory material; see 2-3-7/7 for specifications of Grade 2 forged steel.

3.5 Material Testing

Materials of propellers cast in one piece and materials of blades, hub, studs and other load-bearing parts of controllable pitch propellers are to be tested in the presence of a Surveyor. For requirements of material testing, see 2-3-14/3 and Section 2-3-15 and 2-3-7/7.

5 Design

5.1 Blade Thickness – Fixed Pitch Propeller

Propeller blades of thrusters (as defined in 4-3-5/1.5) and wide-tip blades of ducted propellers are to be in accordance with the provisions of Section 4-3-5. The thickness of the propeller blades of conventional design ($\theta \le 25°$) is not to be less than that determined by the following equations:

$$t_{0.25} = S\left[K_1\sqrt{\frac{AH}{C_n CRN}} \pm \left(\frac{C_s}{C_n}\right)\left(\frac{BK}{4C}\right)\right]$$

$$A = 1.0 + \frac{6.0}{P_{0.70}} + 4.3P_{0.25}$$

$$B = \left(\frac{4300wa}{N}\right)\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

$$C = (1 + 1.5P_{0.25})(Wf - B)$$

where (units of measures are given in SI (MKS, and US) units respectively):

a = expanded blade area divided by disc area

a_s = area of expanded cylindrical section at 0.25 radius; mm^2 (in^2)

C_n = section modulus coefficient at the 0.25 radius. C_n is to be determined by the following equation:

$$C_n = \frac{I_0}{U_f W T^2}$$

If the calculated C_n value exceeds 0.10, the required thickness is to be computed with C_n = 0.10.

C_s = section area coefficient at 0.25 radius and is to be determined by the following equation:

$$C_s = \frac{a_s}{WT}$$

The values of C_s and C_n, computed as stipulated above, are to be indicated on the propeller drawing.

D = propeller diameter; m (ft)

f, w = material constants from the following table:

Material type (see 4-3-3/3.1)	*SI and MKS units*			*US units*	
	f	*w*		*f*	*w*
2	2.10	8.3		68	0.30
3	2.13	8.0		69	0.29
4	2.62	7.5		85	0.27
5	2.37	7.5		77	0.27
CF-3	2.10	7.75		68	0.28

Note:

The *f* and *w* values of materials not covered will be specially considered upon submittal of complete material specifications including corrosion fatigue data to 10^8 cycles.

H = power at rated speed; kW (PS, hp)

I_0 = moment of inertia of expanded cylindrical section at 0.25 radius about a straight line through the center of gravity parallel to the pitch line or to the nose-tail line; mm^4 (in^4)

K = rake of propeller blade, in mm (in.) (positive for aft rake and negative for forward rake)

K_1 = coefficient as given below

	SI	*MKS*	*US*
K_1	337	289	13

N = number of blades

$P_{0.25}$ = pitch at one-quarter radius divided by propeller diameter, corresponding to the design ahead condition

$P_{0.70}$ = pitch at seven-tenths radius divided by propeller diameter, corresponding to the design ahead condition

R = rpm at rated speed

S = factor, as given below. If greater than 1.025, equate to 1.025.

SI & MKS units	*US units*
1.0 for $D \le 6.1$ m	1.0 for $D \le 20$ ft
$\sqrt{\dfrac{(D+24)}{30.1}}$ for $D > 6.1$ m	$\sqrt{\dfrac{(D+79)}{99}}$ for $D > 20$ ft

$t_{0.25}$ = minimum required thickness at the thickest part of the blade section at one quarter radius; mm (in.)

T = maximum designed thickness of blade section at 0.25 radius from propeller drawing; mm (in.)

U_f = maximum nominal distance from the moment of inertia axis to points of the face boundary (tension side) of the section; mm (in.)

W = expanded width of a cylindrical section at 0.25 radius; mm (in.)

5.3 Blade Thickness – Controllable-pitch Propellers

The thickness of the controllable pitch propeller blade of conventional design ($\theta \le 25°$) is not to be less than determined by the following equation:

$$t_{0.35} = K_2\sqrt{\frac{AH}{C_n CRN}} \pm \left(\frac{C_s}{C_n}\right)\left(\frac{BK}{6.3C}\right)$$

$$A = 1.0 + \frac{6.0}{P_{0.70}} + 3P_{0.35}$$

$$B = \left(\frac{4900wa}{N}\right)\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

$$C = (1 + 0.6P_{0.35})(Wf - B)$$

where the symbols used in these formulas are the same as those in 4-3-3/5.1, except as modified below:

a_s = area of expanded cylindrical section at 0.35 radius; mm^2 (in^2)

C_n = section modulus coefficient at the 0.35 radius and is to be determined by the following equation:

$$C_n = \frac{I_0}{U_f WT^2}$$

If the calculated C_n value exceeds 0.10, the required thickness is to be computed with $C_n = 0.10$.

C_s = section area coefficient at 0.35 radius and is to be determined by the following equation:

$$C_s = \frac{a_s}{WT}$$

The values of C_s and C_n, computed as stipulated above, are to be indicated on the propeller drawing.

I_0 = moment of inertia of expanded cylindrical section at 0.35 radius about a straight line through the center of gravity parallel to the pitch line or to the nose-tail line; mm^4 (in^4)

K_2 = coefficient as given below

	SI	*MKS*	*US*
K_2	271	232	10.4

$P_{0.35}$ = pitch at 0.35 radius divided by D

T = maximum designed thickness of blade section at 0.35 radius from propeller drawing; mm (in.)

$t_{0.35}$ = required minimum thickness of the thickest part of the blade section at 0.35 radius; mm (in.)

W = expanded width of a cylindrical section at 0.35 radius; mm (in.)

5.5 Blade Thickness – Highly Skewed Fixed-pitch Propellers

5.5.1 Propeller Blades with Skew Angle θ; where 25°< θ ≤ 50°

The provisions of 4-3-3/5.5.1 are applicable to fixed pitch propellers having a skew angle over 25° but not exceeding 50°, and made of Type 4 material only. For propellers of other materials, or where skew angle is greater than 50°, see 4-3-3/5.5.2.

5.5.1(a) Blade thickness at 0.25 radius. The maximum thickness at 0.25 radius is to be not less than the thickness required in 4-3-3/5.1 for fixed pitch-propellers multiplied by the factor *m* as given below:

$$m = \sqrt{1 + 0.0065(\theta - 25)}$$

5.5.1(b) Blade thickness at 0.6 radius. The maximum thickness of the blade section at 0.6 radius is to be not less than that obtained from the following equations:

$$t_{0.6} = K_3 \cdot \sqrt{(1 + C_{0.9})\left(1 + \frac{2C_{0.9}}{C_{0.6}}\right)\left(\frac{HD\Gamma}{RP_{0.6}Y}\right)^{0.5}}$$

$$\Gamma = \left(1 + \frac{\theta - 25}{\theta}\right)\left(\phi^2 + 0.16\phi \cdot \theta \cdot P_{0.9} + 100\right)$$

where

$C_{0.6}$ = expanded chord length at the 0.6 radius divided by propeller diameter

$C_{0.9}$ = expanded chord length at the 0.9 radius divided by propeller diameter

K_3 = coefficient as given below:

	SI	*MKS*	*US*
K_3	12.6	6.58	1.19

$P_{0.6}$ = pitch at the 0.6 radius divided by propeller diameter

$P_{0.9}$ = pitch at the 0.9 radius divided by propeller diameter

$t_{0.6}$ = required thickness of the blade section at 0.6 radius; mm (in.)

Y = minimum specified yield strength of type 4 propeller material; N/mm^2 (kgf/mm^2, psi). See 4-3-3/Table 1.

θ = skew angle in degrees (see 4-3-3/1.3.1)

ϕ = rake angle in degrees [see 4-3-3/1.3.3(b)] at 0.6 radius, positive for aft rake

H, *D*, and *R* are as defined in 4-3-3/5.1.

5.5.1(c) Blade thickness between 0.6 and 0.9 radii. The maximum thickness at any radius between 0.6 and 0.9 radii is to be not less than that obtained from the following equation:

$$t_x = 3.3D + 2.5(1 - x)(t_{0.6} - 3.3D) \quad \text{mm; or}$$

$$t_x = 0.04D + 2.5(1 - x)(t_{0.6} - 0.04D) \quad \text{in.}$$

where:

t_x = required minimum thickness of the thickest part of the blade section at radius ratio x.

$t_{0.6}$ = thickness of blade section at the 0.6 radius, as required by 4-3-3/5.5.1(b)

x = ratio of the radius under consideration to $D/2$; $0.6 < x \leq 0.9$

5.5.1(d) Trailing edge thickness at 0.9 radius. The edge thickness at 0.9 radius measured at 5% of chord length from the trailing edge is to be not less than 30% of the maximum blade thickness required by 4-3-3/5.5.1(c) above at that radius.

5.5.2 Propeller of Other Than Type 4 Materials with Skew Angle θ; where 25°< θ ≤ 50°

Propellers made of materials other than Type 4 and with skew angle 25°< θ ≤ 50° are subject to special consideration. Design analyses, as indicated in 4-3-3/5.7, are to be submitted.

5.5.3 Propeller Blades with Skew Angle θ > 50°

Propellers with the maximum skew angle exceeding 50° will be subject to special consideration. Design analyses, as indicated in 4-3-3/5.7, are to be submitted.

5.7 Blades of Unusual Design

Propellers of unusual design, such as those indicated in 4-3-3/5.5.2 and 4-3-3/5.5.3, controllable pitch propeller of skewed design (θ > 25°), skewed propeller (θ > 25°) with wide-tip blades, cycloidal propellers, etc., are subject to special consideration based on submittal of propeller load and stress analyses. The analyses are to include, but be not limited to the following:

- Description of method to determine blade loading
- Description of method selected for stress analysis
- Ahead condition is to be based on propulsion machinery's maximum rating and full ahead speed
- Astern condition is to be based on the maximum available astern power of the propulsion machinery (the astern power of the main propelling machinery is to be capable of 70% of the ahead rpm corresponding to the maximum continuous ahead power, as required in 4-1-1/7.5); and is to include crash astern operation
- Fatigue assessment
- Allowable stress and fatigue criteria

5.9 Blade-root Fillets

Fillets at the root of the blades are not to be considered in the determination of blade thickness.

5.11 Strengthening for Navigation in Ice

For vessels to be assigned with **Ice Class** notations, propellers are to be designed in accordance with 6-1-1/53 and 6-1-2/29.

5.13 Controllable Pitch Propellers – Pitch Actuation System

5.13.1 Blade Flange and Mechanisms

The strength of the propeller blade flange and pitch changing mechanism of controllable-pitch propellers subjected to the forces from propulsion torque is to be at least 1.5 times that of the blade at design pitch conditions.

5.13.2 Stud Bolt Area

The sectional area of the stud bolts at the bottom of the thread, *s*, is to be determined by the following equations:

	SI units	*MKS units*	*US units*
s	$\frac{0.056Wkft_{0.35}^2}{rn}$ mm^2		$\frac{0.0018Wkft_{0.35}^2}{rn}$ in^2
k	$\frac{621}{U+207}$	$\frac{63.3}{U+21.1}$	$\frac{90{,}000}{U+30{,}000}$

where

s = area of one stud at bottom of thread

n = number of studs on driving side of blade

r = radius of pitch circle of the studs; mm (in.)

k = material correction factor for stud materials better than ABS Gr. 2 forged steel

U = ultimate tensile strength of the stud material; N/mm^2 (kgf/mm^2, psi)

See 4-3-3/5.1 for *f* and 4-3-3/5.3 for *W* and $t_{0.35}$.

5.13.3 Blade Pitch Control *(2002)*

5.13.3(a) Bridge control. Where the navigation bridge is provided with direct control of propulsion machinery, it is to be fitted with means to control the pitch of the propeller.

5.13.3(b) Duplication of power unit. At least two hydraulic power pump units for the pitch actuating system are to be provided and arranged so that the transfer between pump units can be readily effected. For propulsion machinery spaces intended for unattended operation (**ACCU** notation), automatic start of the standby pump unit is to be provided.

The emergency pitch actuating system [as required by 4-3-3/5.13.3(c)iii)] may be accepted as one of the required hydraulic power pump units, provided it is no less effective.

5.13.3(c) Emergency provisions. To safeguard the propulsion and maneuvering capability of the vessel in the event of any single failure in either the remote pitch control system or the pitch actuating system external to the propeller shaft and oil transfer device (also known as oil distribution box), the following are to be provided:

i) Manual control of pitch at or near the pitch-actuating control valve (usually the directional valve or similar).

ii) The pitch is to remain in the last ordered position until the emergency pitch actuating system is brought into operation.

iii) An emergency pitch actuating system. This system is to be independent of the normal system up to the oil transfer device, provided with its own oil reservoir and able to change the pitch from full ahead to full astern.

5.13.3(d) Integral oil systems. Where the pitch actuating hydraulic system is integral with the reduction gear lubricating oil system and/or clutch hydraulic system, the piping is to be arranged such that any failure in the pitch actuating system will not leave the other system(s) non-operational.

5.13.3(e) Provisions for testing. Means are to be provided in the pitch actuating system to simulate system behavior in the event of loss of system pressure. Hydraulic pump units driven by main propulsion machinery are to be fitted with a suitable by-pass for this purpose.

5.13.3(f) Multiple propellers. For vessels fitted with more than one controllable pitch propeller, each of which is independent of the other, only one emergency pitch actuating system [as required by 4-3-3/5.13.3(c)iii)] need be fitted, provided it is arranged such that it can be used to provide emergency pitch-changing for all propellers.

5.13.3(g) Hydraulic piping. Hydraulic piping is to meet the requirements of 4-6-7/3.

5.13.4 Instrumentation

All controllable pitch propeller systems are to be provided with instrumentation as provided below:

5.13.4(a) Pitch indicators. A pitch indicator is to be fitted on the navigation bridge. In addition, each station capable of controlling the propeller pitch is to be fitted with a pitch indicator.

5.13.4(b) Monitoring. Individual visual and audible alarms are to be provided at the engine room control station to indicate hydraulic oil low pressure and high temperature and hydraulic tank low level. A high hydraulic oil pressure alarm is to be fitted, if required by the proposed system design and, if fitted, is to be set below the relief valve setting.

For vessels assigned with **ACC** or **ACCU** notations, see 4-9-2/Table 1 and 4-9-3/Table 2 for monitoring on the navigation bridge and in the centralized control station, respectively.

5.15 Propeller Fitting

5.15.1 Keyed Fitting

For shape of the keyway in the shaft and size of the key, see 4-3-2/5.7, 4-3-2/Figure 1 and 4-3-2/5.11.

5.15.2 Keyless Fitting

5.15.2(a) Design criteria. The factor of safety against slip of the propeller hub on the tail shaft taper at 35°C (95°F) is to be at least 2.8 under the action of maximum continuous ahead rated torque plus torque due to torsional vibrations. See 6-1-1/53.7 for propellers requiring ice strengthening. For oil injection method of fit, the coefficient of friction is to be taken no greater than 0.13 for bronze/steel propeller hubs on steel shafts. The maximum equivalent uniaxial stress (von Mises-Hencky criteria) in the hub at 0°C (32°F) is not to exceed 70% of the minimum specified yield stress or 0.2% proof stress of the propeller material.

Stress calculations and fitting instructions are to be submitted (see 4-3-3/1.5.4) and are to include at least the following:

- Theoretical contact surface area
- The maximum permissible pull-up length at 0°C (32°F) as limited by the maximum permissible uniaxial stress specified above
- The minimum pull-up length and contact pressure at 35°C (95°F) to attain a safety factor against slip of 2.8
- The proposed pull-up length and contact pressure at fitting temperature
- The rated propeller ahead thrust

5.15.2(b) Nomenclature. The symbols used are defined as follows.

A	=	100% of contact surface area between propeller hub and shaft taper (i.e., $A = \pi D_s L$); mm^2 (in^2). Oil grooves may be ignored. The propeller hub forward and aft counterbore lengths (ℓ_1 and ℓ_2 in 4-3-3/Figure 3) and the forward and aft inner edge radii (r_1 and r_2 in 4-3-3/Figure 3), if any, are to be excluded.
B	=	dimensionless constant based on μ, θ and S

c = coefficient, dependent on the type of propulsion drive: 1.0 for drives such as turbine, geared diesel, electric, and direct diesel with elastic coupling; and 1.2 for direct diesel drive. This value may have to be increased for cases where extremely high pulsating torque is expected in service.

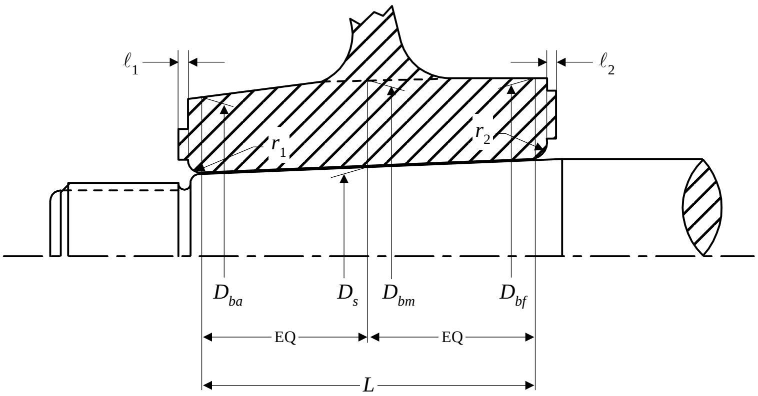
D_b = mean outer diameter of propeller hub corresponding to D_s; mm (in.) D_b is to be calculated as the mean of D_{bm}, D_{bf} and D_{ba}, outer diameters of hub corresponding to D_s, the forward point of contact and the aft point of contact, respectively, see 4-3-3/Figure 3.

$$D_b = \frac{D_{ba} + D_{bm} + D_{bf}}{3}$$

D_{bm} = mean outer diameter of propeller boss, in mm (in.), at the axial position corresponding to D_s, see 4-3-3/Figure 3.

D_s = diameter of shaft at mid-point of the taper in axial direction; mm (in.), taking into account the exclusion of forward and aft counterbore length and the forward and aft edge radii, see 4-3-3/Figure 3.

FIGURE 3
Theoretical Contact Surface Between Hub and Shaft



E_b = modulus of elasticity of hub material, see 4-3-3/Table 2

E_s = modulus of elasticity of shaft material, see 4-3-3/Table 2

F_v = shear force at propeller/shaft interface; N (kgf, lbf)

H = power at rated speed; kW (PS, hp)

K = ratio of D_b to D_s, see 4-3-3/Figure 3.

L = contact length, in mm (in.), see 4-3-3/Figure 3

P = mean propeller pitch; mm, (in.)

P_{min} = minimum required mating surface pressure at 35°C (95°F); N/mm^2 (kgf/mm^2, psi)

P_t = minimum required mating surface pressure at temperature t; N/mm^2 (kgf/mm^2, psi)

P_{max} = maximum permissible mating surface pressure at 0°C; N/mm^2 (kgf/mm^2, psi)

Q = rated torque corresponding to H and R; N-mm (kgf-mm, lbf-in)

R = rpm at rated speed

S = factor of safety against slippage at 35°C (95°F)

T = rated propeller thrust; N (kgf, lbf)

t_{ref} = 35°C (95°F)

v = vessel speed at rated power; knots (knots)

α_b = coefficient of linear expansion of propeller hub material; mm/mm°C (in/in°F); see 4-3-3/Table 2

α_s = coefficient of linear expansion of shaft material; mm/mm°C (in/in°F); see 4-3-3/Table 2

δ_{min} = minimum pull-up length at 35°C (95°F); mm (in.)

δ_t = minimum pull-up length at temperature t; mm (in.)

δ_{max} = maximum permissible pull-up length at 0°C (32°F); mm (in.)

θ = half taper of shaft; e.g. if taper =1/15, θ = 1/30

σ_y = yield stress or 0.2% proof stress of propeller material; N/mm^2 (kgf/mm^2, psi)

μ = coefficient of friction between mating surfaces; to be taken as 0.13 for fitting methods using oil injection and hubs of bronze of steel

ν_b = Poisson's ration of hub material, see 4-3-3/Table 2

ν_s = Poisson's ratio of shaft material, see 4-3-3/Table 2

TABLE 2
Material Constants

Material	*Modulus of Elasticity*				*Poisson's*		*Coefficient of Expansion*	
	N/mm²	*kgf/mm²*	*psi*		*Ratio*		*mm/mm°C*	*in/in°F*
Cast and forged steel	20.6×10^4	2.1×10^4	29.8×10^6		0.29		12.0×10^{-6}	6.67×10^{-6}
Bronzes, Types 2 & 3	10.8×10^4	1.1×10^4	15.6×10^6		0.33		17.5×10^{-6}	9.72×10^{-6}
Bronzes, Types 4 & 5	11.8×10^4	1.2×10^4	17.1×10^6		0.33		17.5×10^{-6}	9.72×10^{-6}

5.15.2(c) Equations. The taper on the tail shaft cone is not to exceed 1/15. Although the equations given below are for ahead operation, they may be considered to provide an adequate safety margin for astern operation also.

The minimum mating surface pressure at 35°C (95°F), P_{min}, is to be:

$$P_{min} = \frac{ST}{AB}\left[-S\theta + \sqrt{\mu^2 + B\left(\frac{F_v}{T}\right)^2}\right] \quad \text{N/mm}^2 \text{ (kgf/mm}^2\text{, psi)}$$

The rated propeller thrust, T, submitted by the designer is to be used in these calculations. In the event that this is not submitted, one of the equations in 4-3-3/Table 3 may be used, subject to whichever yields the larger value of P_{min}.

TABLE 3
Estimated Propeller Thrust, T

SI units (N)	*MKS units (kgf)*	*US units (lbf)*
$1762\dfrac{H}{v}$ or $57.4\times10^6\cdot\dfrac{H}{PR}$	$132\dfrac{H}{v}$ or $4.3\times10^6\cdot\dfrac{H}{PR}$	$295\dfrac{H}{v}$ or $0.38\times10^6\cdot\dfrac{H}{PR}$

The shear force at interface, F_v, is given by

$$F_v = \frac{2cQ}{D_s} \quad \text{N (kgf, lbf);}$$

Constant B is given by:

$$B = \mu^2 - S^2\theta^2$$

The corresponding [i.e., at 35°C (95°F)] minimum pull-up length, δ_{min}, is:

$$\delta_{min} = P_{min}\,\frac{D_s}{2\theta}\left[\frac{1}{E_b}\left(\frac{K^2+1}{K^2-1}+\nu_b\right)+\frac{1}{E_s}(1-\nu_s)\right] \quad \text{mm (in.);}$$

$$K = \frac{D_b}{D_s}$$

The minimum pull-up length, δ_t, at temperature, t, where t < 35°C (95°F), is:

$$\delta_t = \delta_{min} + \frac{D_s}{2\theta}\,(\alpha_b - \alpha_s)(t_{ref} - t) \quad \text{mm (in.)}$$

The corresponding minimum surface pressure, P_t, is:

$$P_t = P_{min}\frac{\delta_t}{\delta_{min}} \quad \text{N/mm}^2\ \text{(kgf/mm}^2\text{, psi)}$$

The maximum permissible mating surface pressure, P_{max}, at 0°C (32°F) is:

$$P_{max} = \frac{0.7\sigma_y(K^2-1)}{\sqrt{3K^4+1}} \quad \text{N/mm}^2\ \text{(kgf/mm}^2\text{, psi)}$$

and the corresponding maximum permissible pull-up length, δ_{max}, is:

$$\delta_{max} = \frac{P_{max}}{P_{min}}\,\delta_{min} \quad \text{mm (in.)}$$

7 Certification

7.1 Material Tests

Propeller materials are to be tested in the presence of a Surveyor. See 4-3-3/3.5.

7.3 Inspection and Certification

Finished propellers are to be inspected and certified at the manufacturer's plant by a Surveyor. The blade forms, pitch, blade thickness, diameters, etc. are to be checked for conformance with approved plans. The entire surface of the finished propeller is to be examined visually and by liquid penetrant method. See 2-3-14/3.21. All finished propellers are to be statically balanced in the presence of the Surveyor. As far as practicable, reference is to be made to the provisions of ISO 484 for these purposes.

The surfaces of stainless steel propellers are to be suitably protected from the corrosive effect of the industrial environment until fitted on the vessel. See 2-3-15/3.

9 Installation, Tests and Trial

9.1 Keyed Propellers

The sides of the key are to have a true fit in the keyways of the propeller hub and the shaft. See also 4-3-2/5.11 for tail shaft propeller-end design.

9.3 Controllable Pitch Propellers – Fit of Studs and Nuts

Studs, nuts and bolts are to have tight-fitting threads and are to be provided with effective means of locking. Effective sealing arrangements are to be provided in way of the bolt or stud holes against sea water ingress or oil leakage. Bolts, nuts and studs are to be of corrosion resistant materials or adequately protected from corrosion.

9.5 Protection Against Corrosion

The exposed steel of the shaft is to be protected from the action of the water by filling all spaces between cap, hub and shaft with a suitable material. The propeller assembly is to be sealed at the forward end with a well-fitted soft-rubber packing ring. When the rubber ring is fitted in an external gland, the hub counterbore is to be filled with suitable material, and clearances between shaft liner and hub counterbore are to be kept to a minimum. When the rubber ring is fitted internally, ample clearance is to be provided between liner and hub. The rubber ring is to be sufficiently oversized to squeeze into the clearance space provided; and, where necessary, a filler piece is to be fitted in way of the propeller-hub keyway to provide a flat, unbroken seating for the rubber ring. The recess formed at the small end of the taper by the overhanging propeller hub is also to be packed with rust-preventive compound. See 4-3-2/5.13 for sealing requirements and 4-3-2/Figure 1 for typical arrangements.

9.7 Circulating Currents

Where means are provided to prevent circulating currents from passing between the propeller, shaft and the hull, a warning notice plate is to be provided in a visible place cautioning against the removal of such protection.

9.9 Keyed and Keyless propellers – Contact Area Check and Securing

The propeller hub to tail shaft taper contact area is to be checked in the presence of a Surveyor. In general, the actual contact area is to be not less than 70% of the theoretical contact area. Non-contact bands extending circumferentially around the propeller hub or over the full length of the hub are not acceptable. Installation is to be in accordance with the procedure referred to in 4-3-3/5.15.2(a) and final pull-up travel is to be recorded. After final pull-up, propellers are to be secured by a nut on the after end of the tail shaft. The nut is to be secured to the tail shaft against loosening. See also 4-3-2/5.11.

9.11 Controllable Pitch Propellers – Hydrostatic Tests

The completed piping system of the controllable pitch propeller hydraulic system is to be hydrostatically tested at a pressure equal to 1.5 times the design pressure in the presence of a Surveyor. Relief-valve operation is to be verified.

9.13 Sea Trial

The designed performance of the propeller at rated speed is to be demonstrated during sea trial. For controllable pitch propellers, the blade pitch control functions, from full ahead through full astern, are to be demonstrated. The emergency provisions in 4-3-3/5.13.3(c) are also to be demonstrated.

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 4 Steering Gears

1 General

1.1 Application *(2007)*

This section is applicable to vessels 90 meters in length or over for which steering is effected by means of a rudder or rudders and an electric, hydraulic or electro-hydraulic steering gear.

Additional requirements for azimuthal thrusters are given in 4-3-5/5.11.

Steering gears intended for vessels strengthened for navigation in ice are to comply also with additional requirements in Part 6.

For convenience, additional requirements specific to passenger vessels and to vessels intended to carry oil, chemical or liquefied gases in bulk are provided in 4-3-4/23 and 4-3-4/25 hereunder.

1.3 Basic Principles *(2002)*

All vessels are to be provided with power-operated means of steering. Such means, as a minimum, are to be supported by duplication of power units, and by redundancy in piping, electrical power supply, and control circuitry. Steering is to be capable of being readily regained in the event of the failure of a power unit, a piping component, a power supply circuit or a control circuit. In addition, duplication of rudder actuators is to be provided for oil and fuel oil carriers, chemical carriers and gas carriers in accordance with the requirements in 4-3-4/5.3 and 4-3-4/25, as applicable.

1.5 Definitions

For the purpose of this section the following definitions apply:

1.5.1 Steering Gear

Steering Gear is the machinery, rudder actuators, steering gear power units and ancillary equipment and the means of applying torque to the rudder stock (e.g., tiller or quadrant) necessary for effecting movement of the rudder for the purpose of steering the vessel under normal service conditions.

1.5.2 Steering Gear Power Unit

Steering Gear Power Unit is:

i) In the case of electro-hydraulic steering gears, an electric motor and its associated electrical equipment and connected pump.

ii) In the case of other hydraulic steering gears, a driving engine and connected pump.

iii) In the case of electric steering gears, an electric motor and its associated electrical equipment.

1.5.3 Power Actuating System

Power Actuating System of hydraulic and electro-hydraulic steering gears is the hydraulic equipment provided for supplying power to turn the rudder stock, comprising a steering gear power unit or units, together with the associated pipes and fittings and a rudder actuator. Where duplicated power actuating systems are required by the Rules, they may share common mechanical components, i.e., tiller, quadrant and rudder stock, or components serving the same purpose.

1.5.4 Rudder Actuator

Rudder Actuator is the component which directly converts hydraulic pressure into mechanical action to move the rudder. This may be a hydraulic cylinder or a hydraulic motor.

1.5.5 Maximum Working Pressure

Maximum Working Pressure is the pressure needed to satisfy the operational conditions specified in 4-3-4/1.9.

1.5.6 Steering Gear Control System

Steering Gear Control System is the equipment by which orders are transmitted from the navigation bridge to the steering gear power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables. For the purpose of the Rules, steering wheels, steering levers, and rudder angle feedback linkages are not considered to be part of the control system

1.5.7 Maximum Ahead Service Speed

Maximum Ahead Service Speed is the greatest speed which the vessel is designed to maintain in service at sea at the deepest seagoing draft.

1.5.8 Rule Required Upper Rudder Stock Diameter

The *Rule Required Upper Rudder Stock Diameter* is the rudder stock diameter in way of the tiller, calculated as given in 3-2-14/7.1. This required diameter excludes strengthening for navigation in ice.

1.7 Steering Gear Compartment

The steering gear is to be protected from the weather. Steering gear compartments are to be readily accessible and, as far as practicable, separated from the machinery spaces. Working access is to be provided to the steering gear machinery and controls with handrails, gratings or other non-slip surfaces to ensure suitable working conditions in the event of hydraulic fluid leakage.

The steering gear compartment is to be provided with visual compass readings.

1.9 Performance

The steering gear is to be capable of:

i) Putting the rudder from 35° on one side to 35° on the other side with the vessel running ahead at the maximum continuous rated shaft rpm and at the summer load waterline and, under the same conditions, from 35° on either side to 30° on the other side in not more than 28 seconds; and

ii) With one of the power units inoperative, putting the rudder from 15° on one side to 15° on the other side in no more than 60 seconds with the vessel running ahead at the summer load waterline at one half of the maximum ahead service speed or 7 knots, whichever is the greater.

For passenger vessels, see 4-3-4/23.

1.11 Plans and Particulars to be Submitted

The following plans and particulars are to be submitted for review:

Arrangement of steering gear machinery

Hydraulic piping system diagram

Power supply system diagrams

Motor control system diagrams

Steering control system diagrams

Instrumentation and alarm system diagrams

Drawings and details for rudder actuators

Drawings and details for torque transmitting parts and parts subjected to internal hydraulic pressure

Weld details and welding procedure specifications

Rated torque

3 Materials

3.1 General

All parts of the steering gear transmitting forces to the rudder and pressure retaining components of hydraulic rudder actuators are to be of steel or other approved ductile material. In general materials are not to have a tensile strength in excess of 650 N/mm^2 (66 kgf/mm^2, 94,300 psi).

Gray cast iron or other material having an elongation ($L_0 /d = 4$) less than 12% in 50 mm (2 in.) is not to be used for these parts.

3.3 Material Testing

Except as modified below, materials for the parts and components mentioned in 4-3-4/3.1 are to be tested in the presence of the Surveyor in accordance with the requirements of Part 2, Chapter 3 or such other appropriate material specifications as may be approved in connection with a particular design.

3.3.1 Coupling Bolts and Keys

Material tests for steering gear coupling bolts and torque transmitting keys need not be witnessed by the Surveyor, but manufacturer's test certificates traceable to these components are to be presented upon request.

3.3.2 Small Parts of Rudder Actuators *(2010)*

Material tests for forged, welded or seamless steel parts (including the internal components) of rudder actuators that are under 150 mm (6 in.) in internal diameter need not be carried out in the presence of the Surveyor. Such parts may be accepted on the basis of a review of mill certificates by the Surveyor.

3.3.3 Tie Rod Nuts

Material tests for commercially supplied tie-rod nuts need not be witnessed by the Surveyor provided the nuts are in compliance with the approved steering gear drawings and are appropriately marked and identified in accordance with a recognized industry standard. Mill test reports for the tie-rod nuts are to be made available to the Surveyor upon request. For all non-standard tie-rod nuts, material testing is required to be performed in the presence of the Surveyor.

3.3.4 Piping Material

Piping materials need not be tested in the presence of the Surveyor. Pipes may be accepted based on certification by the mill, and on physical inspection and review of mill certificate by the Surveyor.

5 System Arrangements

5.1 Power Units

The steering gear is to be composed of two or more identical power units and is to be capable of operating the rudder as required by 4-3-4/1.9i) and 4-3-4/1.9ii). The power units are to be served by at least two power circuits (see 4-3-4/11). Power units are required to be type tested, see 4-3-4/19.5.

5.3 Rudder Actuators

Steering gears may be composed of a single rudder actuator for all vessels except the following:

- For oil carriers, fuel oil carriers, chemical carriers and gas carriers of 100,000 tonnes deadweight and above, the steering gear is to be comprised of two or more identical rudder actuators.
- For oil carriers, fuel oil carriers, chemical carriers and gas carriers of 10,000 gross tonnage and above but less than 100,000 tonnes deadweight, the steering gear may be comprised of a single, non-duplicated rudder actuator, provided it complies with 4-3-4/25.5.

5.5 Single Failure Criterion

The hydraulic system is to be designed so that after a single failure in the piping system or one of the power units, the defect can be isolated so that the integrity of the remaining part of the system will not be impaired and the steering capability can be maintained or speedily regained. See also 4-3-4/9.

5.7 Independent Control Systems

Two independent steering gear control systems are to be provided, each of which can be operated from the navigation bridge. These control systems are to allow rapid transfer of steering power units and of control between the units. See 4-3-4/13.

5.9 Non-duplicated Components

Essential components which are not required to be duplicated are to utilize, where appropriate, anti-friction bearings, such as ball bearings, roller bearings or sleeve bearings which are to be permanently lubricated or provided with lubrication fittings.

5.11 Power Gear Stops *(2010)*

The steering gear is to be fitted with arrangements, such as limit switches, for stopping the gear before the structural rudder stops (see 3-2-14/1.7) or positive mechanical stops within the steering gear are reached. These arrangements are to be synchronized with the rudder stock or the position of the gear itself and may be an integral part of the rudder actuator. Arrangements to satisfy this requirement through the steering gear control system are not permitted.

5.13 Steering Gear Torques *(2003)*

5.13.1 Minimum Required Rated Torque

The rated torque of the steering gear is not to be less than the expected torque, as defined in 3-2-14/1.5.

5.13.2 Maximum Allowable Torque

The transmitted torque, T_{max}, of the steering gear is not to be greater than the maximum allowable torque, T_{ar}, based on the actual rudder stock diameter.

5.13.2(a) Transmitted torque. The transmitted torque, T_{max}, is to be based on the relief valve setting and to be determined in accordance with the following equations:

- *For ram type actuator:*

  $T_{max} = P \cdot N \cdot A \cdot L_2/(C \cdot \cos^2\theta)$ kN-m (tf-m, Ltf-ft)

- *For rotary vane type actuator:*

  $T_{max} = P \cdot N \cdot A \cdot L_2/C$ kN-m (tf-m, Ltf-ft)

- *For linked cylinder type actuator:*

  $T_{max} = P \cdot N \cdot A \cdot L_2 \cos\theta/C$ kN-m (tf-m, Ltf-ft)

where

P = steering gear relief valve setting pressure, bar (kgf/cm^2, psi)

N = number of active pistons or vanes

A = area of piston or vane, mm^2 (cm^2, in^2)

L_2 = torque arm, equal the distance from the point of application of the force on the arm to the center of the rudder stock at zero (0) degrees of rudder angle, m (ft)

C = factor, 10000 (1000, 2240)

θ = maximum permissible rudder angle (normally 35 degrees)

5.13.2(b) Maximum allowable torque for rudder stock. The maximum allowable torque, T_{ar}, for the actual rudder stock diameter is to be determined in accordance with the following equation:

$$T_{ar} = 2.0(D_r/N_u)^3/K_s \quad \text{kN-m (tf-m, Ltf-ft)}$$

where

K_s = material factor for rudder stock (see 3-2-14/1.3)

D_r = actual rudder stock diameter at minimum point below the tiller or the rotor, mm (in.)

N_u = factor, 42.0 (89.9, 2.39)

7 Mechanical Component Design

7.1 Mechanical Strength

All mechanical components which transmit force to or from the rudder are to have strength equivalent to that of the Rule required upper rudder stock (see 4-3-4/1.5.8).

7.3 Rudder Actuators

7.3.1 Design

Rudder actuators are to be designed in accordance with the requirements of pressure vessels in Section 4-4-1, except that the maximum allowable stress S is not to exceed the lower of the following:

$$\frac{U}{A} \text{ or } \frac{Y}{B}$$

where

U = minimum specified tensile strength of material at room temperature

Y = minimum specified yield point or yield strength

A and B are factors given below:

	Rolled or forged steel	*Cast steel*	*Nodular cast iron*
A	3.5	4	5
B	1.7	2	3

For requirements relative to vessels intended to carry oil, chemicals, or liquefied gases in bulk of 10,000 gross tonnage and over, but less than 100,000 tonnes deadweight, fitted with non-duplicated rudder actuators, see 4-3-4/25.5.

7.3.2 Oil Seals

Oil seals between non-moving parts forming part of the exterior pressure boundary are to be of the metal upon metal type or of an equivalent type. Oil seals between moving parts forming part of the external pressure boundary are to be fitted in duplicate so that the failure of one seal does not render the actuator inoperative. Alternative seal arrangements may be acceptable provided equivalent protection against leakage can be assured.

7.5 Tillers, Quadrants and Other Mechanical Parts

7.5.1 General

All steering gear parts, such as tillers, quadrants, rams, pins, tie rods and keys, which transmit force to or from the rudder, are to be proportioned so as to have strength equivalent to that of the Rule required upper rudder stock, taking into consideration the difference in materials between the rudder stock and the component.

7.5.2 Tillers and Quadrants

7.5.2(a) Tiller or quadrant hub. Dimensions of the hub are to be as follows (use consistent system of units):

i) Depth of the hub is not to be less than *S*.

ii) Mean thickness of the hub is not to be less than *S*/3.

iii) Notwithstanding 4-3-4/7.5.2(a)ii) above, the polar section modulus of the hub is not to be less than that given below:

$$0.196S^3\frac{K_h}{K_s}$$

where

S = Rule-required upper rudder stock diameter

K_s = material factor of rudder stock (see 3-2-14/1.3)

K_h = material factor of hub (see 3-2-14/1.3)

7.5.2(b) Tiller or quadrant arm. The section modulus of the tiller or quadrant arm anywhere along its length is not to be less than that given below (use consistent system of units):

$$\frac{0.167S^3(L_2 - L_1)}{L_2}\cdot\frac{K_t}{K_s}$$

where

L_2 = distance from the point of application of the force on the arm to the center of the rudder stock

L_1 = distance between the section of the arm under consideration and the center of the rudder stock

K_t = material factor of tiller or quadrant arm (see 3-2-14/1.3)

Other symbols are as defined above.

7.5.2(c) Bolted hub. Split or semi-circular tiller or quadrant hubs assembled by bolting are to have bolts on any side having a total cross-sectional area not less than that given below (use a consistent system of units):

$$\frac{0.196S^3}{L_3}\cdot\frac{K_b}{K_s}$$

where

L_3 = distance between the center of the bolts and the center of the rudder stock

K_b = material factor of bolt (see 3-2-14/1.3)

Other symbols are as defined above.

The thickness of the bolting flange is not to be less than the minimum required diameter of the bolt.

7.5.2(d) Tiller pin. The total effective shear area of the tiller pin is not to be less than that given below. (Use consistent system of units):

$$\frac{0.196S^3}{L_2}\cdot\frac{K_p}{K_s}$$

where

K_p = material factor of the pin (see 3-2-14/1.3)

Other symbols are defined above.

7.5.3 Tie Rod

For multiple rudder installations or similar, where tie rod (or jockey bar) is fitted between tillers to synchronize them, the buckling strength of the tie rod is not to be less than that given below (use a consistent system of units):

$$\frac{0.113S^3U_R}{L_2}$$

where

U_R = ultimate tensile strength of the rudder stock

Other symbols are defined above.

7.7 Rudder Stock to Tiller/Quadrant Connection

7.7.1 Key *(2006)*

The effective area of the key in shear is not to be less than that given below (use a consistent system of units):

$$\frac{0.196S^3}{r}\frac{K_k}{K_s}$$

where

S = Rule-required upper rudder stock diameter

r = actual rudder stock radius at mid length of key

K_s = material factor of rudder stock (see 3-2-14/1.3)

K_k = material factor of key (see 3-2-14/1.3)

Bearing stresses of the tiller and rudder stock keyways are not to be more than 90% of the applicable material yield stress.

7.7.2 Keyless Coupling

Hydraulic or shrink fitted keyless coupling is to be based on preload stress calculations and fitting procedures. The calculated torsional holding capacity is to be at least 2.0 times the transmitted torque based on the steering gear relief valve setting. The coefficient of friction for the oil injection method of fit is to be taken as no greater than 0.13 and that for dry method is to be taken as no greater than 0.18. Preload stress is not to exceed 70% of the minimum yield strength.

7.9 Welding

All welded joints within the pressure boundary of a rudder actuator or connecting parts transmitting mechanical loads are to be full penetration type or to be of other approved design.

9 Hydraulic System

9.1 System Design

9.1.1 General

The hydraulic system is to be fitted with two or more power units, see 4-3-4/5.1. It may be fitted with a single hydraulic rudder actuator unless required otherwise by 4-3-4/5.3.

9.1.2 Piping Arrangements

Piping is to be arranged such that:

- Single failure criteria in 4-3-4/5.5 are met.
- Transfer between power units can be readily effected.
- Air may be bled from the system.

9.1.3 Hydraulic Lock *(2006)*

Hydraulic lock may occur where a piping system is arranged such that malfunctions (for example, in directional valves or in the valve control) can cause power units to work in a closed circuit against each other rather than in parallel delivering fluid to the rudder actuator, thus resulting in loss of steering. Where a single failure can lead to hydraulic lock and loss of steering, an audible and visual hydraulic lock alarm, which identifies the failed system, is to be provided on the navigation bridge. See also Note 4 of 4-3-4/Table 1.

Alternatively, an independent steering failure alarm for follow-up control systems complying with the following requirements may be provided in lieu of a hydraulic lock alarm.

Where an independent steering failure alarm is installed for follow-up control systems, it is to comply with the following:

9.1.3(a) The steering failure alarm system is to actuate an audible and visible alarm in the wheelhouse when the actual position of the rudder differs by more than 5 degrees from the rudder position ordered by the follow-up control systems for more than:

- 30 seconds for ordered rudder position changes of 70 degrees;
- 6.5 seconds for ordered rudder position changes of 5 degrees; and

The time period calculated by the following formula for ordered rudder positions changes between 5 degrees and 70 degrees:

$$t = (R/2.76) + 4.64$$

where:

t = maximum time delay in seconds

R = ordered rudder change in degrees

9.1.3(b) The steering failure alarm system must be separate from, and independent of, each steering gear control system, except for input received from the steering wheel shaft.

9.1.3(c) Each steering failure alarm system is to be supplied by a circuit that:

i) Is independent of other steering gear system and steering alarm circuits.

ii) Is fed from the emergency power source through the emergency distribution panel in the wheelhouse, if installed; and

iii) Has no overcurrent protection except short circuit protection

9.1.4 Isolation Valves

Isolating valves are to be fitted on the pipe connections to the rudder actuators. For vessels with non-duplicated rudder actuators, the isolating valves are to be directly mounted on the actuator.

9.1.5 Filtration

A means is to be provided to maintain the cleanliness of the hydraulic fluid.

9.1.6 System Overpressure Protection

Relief valves are to be provided for the protection of the hydraulic system at any part which can be isolated and in which pressure can be generated from the power source or from external forces. Each relief valve is to be capable of relieving not less than 110% of the full flow of the pump(s) which can discharge through it. With this flow condition, the maximum pressure rise is not to exceed 10% of the relief valve setting, taking into consideration increase in oil viscosity for extreme ambient conditions.

The relief valve setting is to be at least 1.25 times the maximum working pressure (see 4-3-4/1.5.5), but is not to exceed the maximum design pressure (see 4-3-4/9.5.1).

9.1.7 Fire Precautions

Where applicable, the provisions of 4-6-7/3.7.1 are to be met.

9.3 Hydraulic Oil Reservoir and Storage Tank

In addition to the power unit reservoir, a fixed hydraulic oil storage tank independent of the reservoir is to be provided. The storage tank is to have sufficient capacity to recharge at least one power actuating system, including the power unit reservoir. The tank is to be permanently connected by piping in such a manner that the system can be readily recharged from a position within the steering gear compartment. The storage tank is to be provided with an approved level indicating system.

See also 4-6-7/3.3 for arrangements of the power unit reservoir and the storage tank.

9.5 Piping Design

9.5.1 System Pressure

Hydraulic system piping is to be designed to at least 1.25 times the maximum working pressure (4-3-4/1.5.5), taking into account any pressure which may exist in the low-pressure side of the system.

9.5.2 Pipes and Pipe Fittings

Pipes and pipe branches are to meet the design requirements of 4-6-2/5.1 and 4-6-2/5.3. Pipe joints are to be in accordance with 4-6-2/5.5 in general, and 4-6-7/3.5.1 in particular. Particular attention is to be paid to footnotes 1 and 2 in 4-6-7/Table 1 where additional limitations on pipe joints are specified for steering gear hydraulic piping. See also 4-3-4/9.7.3.

9.7 Piping Components

9.7.1 Power Units

Power units are to be certified by the Bureau. See 4-3-4/5.1 and 4-3-4/19.3.

9.7.2 Rudder Actuators

Rudder actuators are to be design approved and are to be certified by the Bureau. See 4-3-4/5.3, 4-3-4/7.3 and 4-3-4/19.7.

9.7.3 Pipes and Pipe Fittings

For pipes and pipe fittings, refer to 4-3-4/9.5.2. Piping materials for hydraulic service are to be traceable to manufacturers' certificates, but need not be certified by the Bureau. See also 4-6-1/7 for certification of piping system components.

9.7.4 Other Piping Components

For valves, hoses and accumulators, refer to 4-6-7/3.5. For relief valve, see also 4-3-4/9.1.6.

9.7.5 Relief Valves

In addition to 4-3-4/9.7.4, discharge capacity test reports verifying the capacity required in 4-3-4/9.1.6 for all relief valves are to be submitted for review.

11 Electrical Systems

11.1 Power Supply Feeders

Each electric or electro-hydraulic steering gear is to be served by at least two exclusive circuits, fed directly from the main switchboard; however, one of the circuits may be supplied through the emergency switchboard. Each of duplicated power units required by 4-3-4/5.1 is to be served by one of these circuits. The circuits supplying an electric or electro-hydraulic steering gear are to have adequate rating for supplying all motors, control systems and instrumentation which are normally connected to them and operated simultaneously. The circuits are to be separated throughout their length as widely as is practicable. See also 4-8-2/7.11 for the steering gear power supply.

11.3 Electrical Protection

11.3.1 General

Each steering gear feeder is to be provided with short-circuit protection which is to be located at the main or the emergency switchboard, as applicable. Power unit motor overload protection is normally not to be provided, except as indicated in 4-3-4/11.3.3. Other means of protection, namely, motor overload alarm and motor phase failure alarm, as applicable, are to be provided as indicated in 4-3-4/Table 1 item d. See also 4-8-2/9.17.5 for protection of the steering gear feeder circuit.

11.3.2 Direct Current Motors

The feeder circuit breaker is to be set to trip instantaneously at not less than 300% and not more than 375% of the rated full-load current of the steering gear motor, except that the feeder circuit breaker on the emergency switchboard may be set to trip at not less than 200%.

11.3.3 Alternating Current Motors

In addition to short circuit protection, overload protection may be permitted if it is set at a value not less than 200% of the full load current of the motor (or of all the loads on the feeder), and is to be arranged to permit the passage of the starting current.

11.3.4 Fuses

The use of fuses instead of circuit breakers for steering gear motor feeder short circuit protection is not permitted.

11.5 Undervoltage Release

Power unit motor controllers and other automatic motor controllers are to be fitted with undervoltage release (capable of restarting automatically when power is restored after a power failure).

11.7 Motor Rating

11.7.1 Steering Gears with Intermittent Working Duty

Electric motors of and converters associated with electro-hydraulic steering gears with intermittent working duty are to be at least of 25% non-periodic duty rating (corresponding to S6 of IEC Publication 60034-1), as per 4-8-3/3.3.3 and 4-8-3/Table 4. Electric motors of electro-mechanical steering gears are, however, to be at least of 40% non-periodic duty rating (corresponding to S3 of IEC Publication 60034-1).

11.7.2 Steering Gears with Continuous Working Duty

Electric motors of and converters associated with steering gears with continuous working duty are to be of continuous rating (corresponding to S1 of IEC Publication 60034-1), as per 4-8-3/3.3.4 and 4-8-3/Table 4.

11.9 Emergency Power Supply

Where the required rudder stock diameter (see 4-3-4/1.5.8) is over 230 mm (9 inches), an alternative power supply, sufficient at least to supply one steering gear power unit and its associated control system and the rudder angle indicator, is to be provided automatically within 45 seconds either from the emergency source of electrical power or from an independent source of power located in the steering gear compartment. This independent source of power is to be used only for this purpose.

The steering gear power unit under alternative power supply is to be capable of moving the rudder from 15° on one side to 15° on the other side in not more than 60 seconds with the vessel at the summer draft while running at one half the maximum ahead service speed or 7 knots, whichever is the greater [see 4-3-4/1.9ii)].

In every vessel of 10,000 gross tonnage and upwards, the alternative power supply is to have a capacity for at least 30 minutes of continuous operation and in any other vessel for at least 10 minutes.

13 Control Systems

13.1 Redundancy

There are to be two independent control systems (see definition in 4-3-4/1.5.6) provided, each of which can be operated from the navigation bridge. These control systems are to be independent in all respects and are to provide on the navigation bridge all necessary apparatus and arrangements for the starting and stopping of steering gear motors and the rapid transfer of steering power and control between units.

Control cables and piping for the independent control systems are to be separated throughout their length. This does not require duplication of a steering wheel or steering lever.

In addition, local steering gear control is to be provided in the steering gear compartment.

13.3 Power Supply

If the control systems operable from the navigation bridge are electric, then each system is to be served by its own separate circuits supplied from a steering gear power circuit in the steering gear compartment, or directly from the switchboard bus bars supplying that steering gear power circuit at a point on the switchboard adjacent to the supply to the steering gear power circuit.

Circuits supplying power to steering gear controls are to be provided with short-circuit protection only.

13.5 Control System Override

13.5.1 Steering Gear Compartment

Means are to be provided in the steering gear compartment to disconnect the steering gear control system from the power circuit when local control is to be used. Additionally, if more than one steering station is provided, a selector switch is to disconnect completely all stations, except the one in use.

13.5.2 Autopilot *(2003)*

13.5.2(a) Steering gear systems provided with an autopilot system are to have a device at the primary steering station to completely disconnect the autopilot control to permit change over to manual operation of the steering gear control system. A display is to be provided at the steering station to ensure that the helmsman can readily and clearly recognize which mode of steering control (autopilot or manual) is in operation.

13.5.2(b) In addition to the changeover device as in 4-3-4/13.5.2(a), for primary steering stations, where fitted with an automatic autopilot override to change over from autopilot control to manual operation, the following are to be provided.

i) The automatic override of the autopilot is to occur when the manual helm order is 5 degrees of rudder angle or greater.

ii) An audible and visual alarm is to be provided at the primary steering station in the event that the automatic autopilot override fails to respond when the manual helm order is 5 degrees of rudder angle or greater. The alarm is to be separate and distinct from other bridge alarms, and is to continue to sound until it is acknowledged.

iii) An audible and visual alarm that is immediately activated upon automatic autopilot override actuation is to be provided at the primary steering station. The alarm is to be distinct from other bridge alarms, and is to continue to sound until it is acknowledged.

13.7 Hydraulic Telemotor

Where the control system consists of a hydraulic telemotor, a second independent system need not be fitted, except in oil or fuel oil carriers, chemical carriers, or gas carriers of 10,000 gross tonnage and above (see also 4-3-4/25).

15 Instrumentation

Instruments for monitoring the steering gear system are to be provided, as indicated in 4-3-4/Table 1. All alarms are to be audible and visual and are to be of the self-monitoring type so that a circuit failure will cause an alarm condition. There are to be provisions for testing alarms.

TABLE 1
Steering Gear Instrumentation *(2003)*

Monitored Parameters		*Display/Alarm*	*Location*
a)	Rudder angle indicator [1]	Display	• Navigation bridge • Steering gear compartment
b)	Power unit motor running	Display	• Navigation bridge • Engine room control station
c)	Power unit power supply failure	Alarm	• Navigation bridge • Engine room control station
d)	Power unit motor overload [2]	Alarm	• Navigation bridge • Engine room control station
e)	Power unit motor phase failure [2], [3]	Alarm	• Navigation bridge • Engine room control station
f)	Control power failure	Alarm	• Navigation bridge • Engine room control station
g)	Hydraulic oil reservoir low level	Alarm	• Navigation bridge • Engine room control station
h)	Hydraulic lock [4]	Alarm	• Navigation bridge
i)	Auto-pilot running [5]	Display	• Navigation bridge
j)	Auto-pilot failure [5]	Alarm	• Navigation bridge
k)	Steering mode (autopilot/manual) indication	Display	• Navigation bridge
l)	Automatic autopilot [5] override failure	Alarm	• Navigation bridge
m)	Automatic autopilot [5] override activated	Alarm	• Navigation bridge

Notes

1 The rudder angle indication is to be independent of the steering gear control system, and readily visible from the control position.

2 The operation of this alarm is not to interrupt the circuit.

3 For three phase AC supply only.

4 The alarm is to be activated when the position of the variable displacement pump control system does not correspond to the given order; or when the incorrect position of the 3-way full flow valve or similar in the constant delivery pump system is detected.

5 If provided.

17 Communications

A means of communication is to be provided between the navigation bridge and the steering gear compartment. Additionally, communication is to be provided between these spaces and the main propulsion control station, in accordance with 4-8-2/11.5.

19 Certification

19.1 General

Steering gear components are to be inspected, tested and certified by a Surveyor at the plant of manufacture in accordance with the following requirements. Hydraulic oil pumps are to be certified, see 4-6-1/7.3.1i).

19.3 Material Testing

For testing of steering gear component materials, see 4-3-4/3.3.

19.5 Prototype Tests of Power Units

A prototype of each new design power unit pump is to be shop tested for a duration of not less than 100 hours. The testing is to be carried out in accordance with an approved program and is to include the following as a minimum:

i) The pump and stroke control (or directional control valve) is to be operated continuously from full flow and relief valve pressure in one direction through idle to full flow and relief valve pressure in the opposite direction.

ii) Pump suction conditions are to simulate lowest anticipated suction head. The power unit is to be checked for abnormal heating, excessive vibration or other irregularities. Following the test, the power unit pump is to be disassembled and inspected in the presence of a Surveyor.

19.7 Components Shop Tests *(2008)*

Each component of the steering gear piping system, including the power units, rudder actuators and piping, is to be inspected by a Surveyor during fabrication, and hydrostatically tested to 1.5 times the relief valve setting (or system design pressure) in the presence of a Surveyor.

21 Installation, Tests and Trials

21.1 Steering Gear Seating

Steering gears are to be bolted to a substantial foundation effectively attached to the hull structure. Suitable chocking arrangements are to be provided to the satisfaction of the Surveyor.

21.3 Operating Instructions

Appropriate operating instructions with a block diagram showing changeover procedures for steering gear control systems and steering gear power units are to be permanently displayed on the navigation bridge and in the steering gear compartment. Where failure alarms are provided to indicate hydraulic locking, instructions are to be permanently posted on the navigation bridge and in the steering gear compartment for the operator to shut down the failed system.

21.5 Installation Tests

After installation on board the vessel, the complete piping system, including power units, rudder actuators and piping, is to be subjected to a hydrostatic test equal to 110% of the relief valve setting, including a check of the relief valve operation in the presence of the Surveyor.

21.7 Sea Trials

The steering gear is to be tried out on the trial trip in order to demonstrate to the Surveyor’s satisfaction that the requirements of this section have been met. The trials are to be performed with the rudder fully submerged. Where full rudder submergence cannot be obtained in ballast conditions, alternative procedures for trials with less than full rudder submergence are to be submitted for consideration.

21.7.1 Full Speed Trial

Satisfactory performance is to be demonstrated under the following conditions:

i) Changing the rudder position from 35° on either side to 30°on the other side in not more than 28 seconds with the vessel running ahead at the maximum continuous rated shaft rpm. For controllable pitch propellers, the propeller pitch is to be at the maximum design pitch approved for the above maximum continuous ahead rated rpm.

ii) Unless 4-3-4/21.7.2iii), 4-3-4/23.3 or 4-3-4/25.7 is applicable, this test is to be carried out with all power units intended for simultaneous operation for this condition under actual operating conditions.

21.7.2 Half Speed Trial

Satisfactory performance is to be demonstrated under the following conditions.

i) Changing the rudder position from 15° on either side to 15° on the other side in not more than 60 seconds while running at one-half of the maximum ahead speed or 7 knots whichever is the greater.

ii) This test is to be conducted with either one of the power units used in 4-3-4/21.7.1ii) in reserve.

iii) This test may be waived where the steering gear consists of two identical power units with each capable of meeting the requirements in 4-3-4/21.7.1i).

21.7.3 Steering Gears with More than Two Power Units

Where three or more power units are provided, the test procedures are to be specially considered on the basis of the specifically approved operating arrangements of the steering gear system.

21.7.4 Additional Items

The trial is also to include the operation and verification of the following:

i) The power units, including transfer between power units.

ii) The emergency power supply, if applicable.

iii) The steering gear controls, including transfer of control and local control.

iv) The means of communication between the navigation bridge, engine room and the steering gear compartment.

v) The alarms and indicators required by 4-3-4/15 above (test may be done at dockside).

vi) The storage and recharging system in 4-3-4/9.3 above (test may be done at dockside).

vii) The isolation of one power actuating system and time for regaining steering capability (test may be done at dockside).

viii) Where the steering gear is designed to avoid hydraulic locking (4-3-4/9.1.3 above), this feature is to be demonstrated.

ix) Where practicable, simulation of a single failure in the hydraulic system, and demonstration of the means provided to isolate it and the regaining of steering capability, as in 4-3-4/5.5 and 4-3-4/9.1.3 above.

x) The stopping of the steering gear before the rudder stop is reached, as in 4-3-4/5.11 above.

23 Additional Requirements for Passenger Vessels

23.1 Performance

The steering gear is to be designed to be capable of operating the rudder, as required by 4-3-4/1.9i), with any one of the power units inoperative.

23.3 Sea Trials

The performance test criteria in 4-3-4/21.7.1i) is to be demonstrated during sea trial with any one of the power units in reserve.

25 Additional Requirements for Oil or Fuel Oil Carriers, Chemical Carriers and Gas Carriers

25.1 Vessels of 10,000 Gross Tonnage and Upwards

25.1.1 Single Failure Criterion

The steering gear is to be so arranged that in the event of the loss of steering capability due to a single failure in any part of one of the power actuating systems (see 4-3-4/1.5.3), excluding the tiller, quadrant or components serving the same purpose, or seizure of the rudder actuators, steering capability is to be regained in not more than 45 seconds.

25.1.2 Power Actuating System

The steering gear is to comprise either:

i) Two independent and separate power actuating systems, each capable of meeting the requirements of 4-3-4/1.9i); or

ii) At least two identical power actuating systems which, acting simultaneously in normal operation, is to be capable of meeting the requirements of 4-3-4/1.9i). Where necessary to comply with this requirement, interconnection of hydraulic power actuating systems may be provided. Loss of hydraulic fluid from one system is to be capable of being detected and the defective system automatically isolated so that the other actuating system or systems is to remain fully operational.

25.1.3 Non-hydraulic Steering Gears

Steering gears other than of the hydraulic type is to achieve equivalent standards.

25.3 Alternative for Vessels 10,000 Gross Tonnage and Upwards but Less than 100,000 Tonnes Deadweight

Vessels within this size range, in lieu of completely meeting the requirements in 4-3-4/25.1, may, as an alternative, exclude the application of single failure criterion to rudder actuator, provided that an equivalent safety standard is achieved and that:

i) Following the loss of steering capability due to a single failure of any part of the piping system or in any one of the power units, steering capability is to be regained within 45 seconds; and

ii) The single rudder actuator meets the requirements of 4-3-4/25.5.

25.5 Non-duplicated Rudder Actuators for Vessels of 10,000 Gross Tonnage and Upwards but Less than 100,000 Tonnes Deadweight

For oil or fuel oil carriers, chemical carriers or gas carriers of 10,000 gross tonnage and upwards but of less than 100,000 tonnes deadweight, a single rudder actuator may be accepted, provided the following additional requirements are complied with.

25.5.1 Analysis

Detailed calculations are to be submitted for the rudder actuator to show the suitability of the design for the intended service. This is to include a stress analysis of the pressure retaining parts of the actuator to determine the stresses at the design pressure.

Where considered necessary due to the design complexity or manufacturing procedures, a fatigue analysis and fracture mechanic analysis may be required. In connection with these analyses, all foreseen dynamic loads are to be taken into account. Experimental stress analysis may be required in addition to, or in lieu of, theoretical calculations depending on the complexity of the design.

25.5.2 Allowable Stresses

For the purpose of determining the general scantlings of parts of rudder actuators subject to internal hydraulic pressure, the allowable stresses are not to exceed:

$$\sigma_m \leq f$$

$$\sigma_\ell \leq 1.5f$$

$$\sigma_b \leq 1.5f$$

$$\sigma_\ell + \sigma_b \leq 1.5f$$

$$\sigma_m + \sigma_b \leq 1.5f$$

where

σ_m = equivalent primary general membrane stress

σ_ℓ = equivalent primary local membrane stress

σ_b = equivalent primary bending stress

σ_B = specified minimum tensile strength of material at ambient temperature

σ_Y = specified minimum yield stress or 0.2 percent proof stress of material at ambient temperature

f = the lesser of σ_B/A or σ_Y/B, where A and B are as follows:

	Steel	*Cast steel*	*Nodular cast iron*
A	4	4.6	5.8
B	2	2.3	3.5

25.5.3 Burst Test

Pressure retaining parts not requiring fatigue analysis and fracture mechanic analysis may be accepted on the basis of a certified burst test and the detailed stress analysis required by 4-3-4/25.5.1 need not be submitted.

The minimum bursting pressure is to be calculated as follows:

$$P_b = PA\frac{\sigma_{Ba}}{\sigma_B}$$

where

P_b = minimum bursting pressure

P = design pressure as defined in 4-3-4/9.5.1

A = from table in 4-3-4/25.5.2

σ_{Ba} = actual tensile strength

σ_B = tensile strength as defined in 4-3-4/25.5.2

25.5.4 Nondestructive Testing

The rudder actuator is to be subjected to complete nondestructive testing to detect both surface flaws and volumetric flaws. The procedure and acceptance criteria are to be in accordance with the requirements of recognized standards or as may be determined by approved fracture mechanic analysis.

25.7 Sea Trials

For vessels having two independent and separate power actuating systems as per 4-3-4/25.1.2i), the performance test criteria in 4-3-4/21.7.1i) are to be demonstrated during sea trial with any one of the power units in reserve. The capabilities of the steering gear to function as required in 4-3-4/25.1.1 and 4-3-4/25.3, as applicable, are also to be demonstrated; this may be conducted at dockside.

PART
4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 5 Thrusters and Dynamic Positioning Systems

1 General

1.1 Application

The provisions of this section apply to maneuvering thrusters not intended to assist in propulsion, and to azimuthal and non-azimuthal thrusters intended for propulsion, maneuvering or dynamic positioning, or a combination of these duties.

Maneuvering thrusters intended to assist maneuvering and dynamic positioning thrusters, where fitted, may, at the request of the owners, be certified in accordance with the provisions of this section. In such cases, appropriate class notations, as indicated in 4-3-5/1.3, will be assigned upon verification of compliance with corresponding provisions of this section.

Thrusters intended for propulsion with or without combined duties for assisting in maneuvering or dynamic positioning are to comply with appropriate provisions of this section in association with other relevant provisions of Part 4, Chapter 3.

Thruster types not provided for in this section, such as cycloidal propellers, pump or water-jet type thrusters, will be considered, based on the manufacturer's submittal on design and engineering analyses.

1.3 Class Notations

1.3.1 **APS** Notation

Self-propelled vessels, where fitted with thrusters capable of producing thrusts primarily in the athwartship direction and intended to assist in maneuvering the vessel, at the discretion of the owners, may comply with the provisions of 4-3-5/1 through 4-3-5/13 of this section. And upon verification of compliance, the class notation **APS** (athwartship thruster) may be assigned.

1.3.2 **PAS** Notation

Non-self-propelled vessels, where fitted with thrusters to assist in the maneuvering or propelling while under tow, at the discretion of the owners, may comply with the provisions of 4-3-5/1 through 4-3-5/13 of this section; and upon verification of compliance, the class notation **PAS** (propulsion assist) may be assigned.

1.3.3 **DPS-0**, **-1**, **-2** & **-3** Notations

Self-propelled or non-self-propelled vessels, where fitted with a system of thrusters, positioning instruments and control systems to enable the vessel to maintain position at sea without external aid, at the discretion of the owners, may comply with 4-3-5/1 through 4-3-5/15 of this section. Upon request by the owner and upon verification of compliance with the applicable requirements, the class notation **DPS** (dynamic positioning system) followed by a numeral of **0**, **1**, **2** or **3**, to indicate the degree of redundancy of the system, will be assigned.

1.5 Definitions

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

1.5.1 Thruster

1.5.1(a) General. Thrusters are devices capable of delivering side thrust or thrusts through 360° to improve the vessel's maneuverability, particularly in confined waters. There are three generic types of thrust-producing devices: the lateral or tunnel thruster, commonly known as 'bow-thruster', which consists of a propeller installed in a athwartship tunnel; jet type thruster, which consists of a pump taking suction from the keel and discharge to either side; and azimuthal thruster, which can be rotated through 360° so that thrust can be developed in any direction. Cycloidal propellers can be considered a type of azimuthal thruster.

1.5.1(b) Propeller-type thruster. Regardless of whether they are normally used for propulsion, propellers intended to be operated for an extended period of time during service in a condition where the vessel is not free running approximately along the direction of the thrust are to be considered thrusters for the purposes of this section.

1.5.2 Continuous Duty Thruster

A continuous duty thruster is a thruster designed for continuous operation, such as dynamic positioning thrusters, propulsion assist, or main propulsion units.

1.5.3 Intermittent Duty Thruster

An intermittent duty thruster is a thruster which is designed for operation at peak power or rpm levels, or both, for periods not exceeding one (1) hour followed by periods at the continuous rating or less, with total running time not exceeding eight (8) hours in twenty (20) hours. Generally, such thrusters are not meant to operate more than 1000 hours per year.

1.7 Plans and Particulars to be Submitted

The general arrangements of the thruster installation, its location of installation, along with its supporting auxiliary machinery and systems, fuel oil tanks, foundations, watertight boundary fittings, etc., are to be submitted. The rated power/rpm and the rated thrust are to be indicated. For azimuthal thrusters, the mechanical and control systems for rotating the thruster assembly or for positioning the direction of thrust are to be submitted. In addition, plans of each component and of the systems associated with the thruster are to be submitted as detailed in the applicable sections of these Rules. Typically, the following are applicable:

Supporting structures:	Section 3-1-2
Diesel engine prime mover:	4-2-1/1.9
Electric motor and controller:	4-8-1/5.5.1 and 4-8-1/5.5.4
Gearing:	4-3-1/1.5
Shafting:	4-3-2/1.5
Propellers:	4-3-3/1.5
Piping system:	4-6-1/9
Control and instrumentation:	4-9-1/7

3 Materials

3.1 General *(2008)*

Materials entered into the construction of the torque-transmitting components of the thruster are to be in accordance with the applicable requirements of Part 4 of the Rules. For instance, material requirements for propellers are to be in accordance with 4-3-3/3; materials for gears, 4-3-1/3; materials for shafting, 4-3-2/3; materials for steering systems, 4-3-4/3.1, etc. All material specifications are contained in Part 2, Chapter 3.

Where alternative material specifications are proposed, complete chemical composition and mechanical properties similar to the material required by these Rules are to be submitted for approval.

3.3 Material Testing

3.3.1 Testing by a Surveyor

The materials of the following components are to be tested in the presence of a Surveyor for verification of their compliance with the applicable requirements of Part 2, Chapter 3, or such other appropriate material specifications as may be approved in connection with a particular design.

- Shafts, shaft flanges, keys
- Gears (propulsion and steering)
- Propellers
- Impellers
- Couplings
- Coupling bolts

Bolts manufactured to a recognized standard and used as coupling bolts need not to be tested in the presence of a Surveyor.

3.3.2 Thruster Rated 375 kW (500 hp) or Less

Materials for thrusters of 375 kW (500 hp) or less, including shafting, gears, pinions, couplings and coupling bolts may be accepted on the basis of the manufacturer's certified mill test reports and a satisfactory surface inspection and hardness check witnessed by a Surveyor.

5 Design

5.1 Prime Movers

5.1.1 Internal Combustion Engines

Internal combustion engines used for driving thrusters are to comply with the design, construction, testing and certification requirements of Part 4, Chapter 2. Engine support systems are to be in accordance with Section 4-6-5; except that standby pumps and similar redundancy specified for propulsion engines are not required for thruster engines.

5.1.2 Electric Motors

Electric motors driving thrusters are to comply with the design, construction, testing and certification requirements of Section 4-8-3. Power for thruster motors may be derived from ship service generators; except that precautions, such as interlock arrangements, are to be fitted to prevent starting except when there are enough generators on-line to support the starting and running of the thruster motor. All ship service generators may be put on line for this purpose, see 4-8-2/3.1.2.

5.3 Propellers

5.3.1 General

In general, the thruster propellers are to comply with the requirements of Section 4-3-3, except as modified below.

5.3.2 Propeller Blades of Conventional Design

Where the propeller blades are of conventional design with skew angle not exceeding 25°, the thickness of the propeller blade is not to be less than determined by the following equations. Fillets at the root of the blades are not to be considered in the determination of the blade thickness.

5.3.2(a) Fixed pitch propellers. The minimum required blade thickness at 0.25 radius, $t_{0.25}$, is to be determined by the following equations:

$$t_{0.25} = K_1\sqrt{\frac{AH}{C_n CRN}} \pm \left(\frac{C_s}{C_n}\right)\left(\frac{BK}{4C}\right) \quad \text{mm (in.)}$$

$$A = 1.0 + \frac{6.0}{P_{0.70}} + 4.3P_{0.25}$$ for free running propellers

$$A = 7.2 + \frac{2.0}{P_{0.70}} + 4.3P_{0.25}$$ For propellers performing bollard pull, athwartship thrusting, dynamic positioning and similar duties;

$$B = \frac{4300wa}{N}\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

$$C = (1.0 + 1.5P_{0.25})(Wf - B)$$

Other symbols are defined in 4-3-5/5.3.2(d).

5.3.2(b) Controllable pitch propellers. The minimum required blade thickness at 0.35 radius, $t_{0.35}$, is to be determined by the following equations:

$$t_{0.35} = K_2\sqrt{\frac{AH}{C_n CRN}} \pm \left(\frac{C_s}{C_n}\right)\left(\frac{BK}{6.3C}\right) \quad \text{mm (in.)}$$

$$A = 1.0 + \frac{6.0}{P_{0.70}} + 3P_{0.35}$$ for free running propellers

$$A = 7.2 + \frac{2.0}{P_{0.70}} + 3P_{0.35}$$ for non-free running propellers [see 4-3-5/5.3.2(a)]

$$B = \frac{4900wa}{N}\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

$$C = (1.0 + 0.6P_{0.35})(Wf - B)$$

Other symbols are defined in 4-3-5/5.3.2(d).

5.3.2(c) Nozzle propellers (wide-tip blades). The minimum required blade thickness at 0.35 radius, $t_{0.35}$, is to be determined by the following equations:

$$t_{0.35} = K_3\sqrt{\frac{AH}{C_n CRN}} \pm \left(\frac{C_s}{C_n}\right)\left(\frac{BK}{5.6C}\right) \quad \text{mm (in.)}$$

$$A = 1.0 + \frac{6.0}{P_{0.70}} + 2.8P_{0.35}$$ for free running propellers

$$A = 7.2 + \frac{2.0}{P_{0.70}} + 2.8P_{0.35}$$ for non-free running propellers [see 4-3-5/5.3.2(a)]

$$B = \frac{4625wa}{N}\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

$$C = (1.0 + 0.6P_{0.35})(Wf - B)$$

Other symbols are defined in 4-3-5/5.3.2(d).

5.3.2(d) Symbols. The symbols used in the above formulas are defined, in alphabetical order, as follows (the units of measure are in SI (MKS and US) systems, respectively):

a = expanded blade area divided by the disc area

a_s = area of expanded cylindrical section at 0.25 or 0.35 radius, as applicable; mm^2 (in^2)

C_n = section modulus coefficient at 0.25 or 0.35 radius, as applicable; to be determined by the following equation:

$$C_n = \frac{I_0}{U_f WT^2}$$

If the value of C_n exceeds 0.1, the required thickness is to be computed with $C_n = 0.1$.

C_s = section area coefficient at 0.25 or 0.35 radius, as applicable, to be determined by the following equation:

$$C_s = \frac{a_s}{WT}$$

The values of C_s and C_n computed as stipulated above are to be indicated on the propeller drawing.

D = propeller diameter; m (ft)

f, w = material constants, see table below:

Material type	*SI & MKS units*		*US units*	
	f	*w*	*f*	*w*
2	2.10	8.3	68	0.30
3	2.13	8.0	69	0.29
4	2.62	7.5	85	0.27
5	2.37	7.5	77	0.27
CF-3	2.10	7.75	68	0.28

H = power at rated speed; kW (PS, hp)

I_0 = moment of inertia of the expanded cylindrical section at 0.25 or 0.35 radius about a straight line through the center of gravity parallel to the pitch line or to the nose-tail line; mm^4 (in^4)

K = rake of propeller blade, in mm (in.) (positive for aft rake and negative for forward rake)

K_1, K_2, and K_3 are constants and are to be of values as specified below:

	SI unit	*MKS unit*	*US unit*
K_1	337	289	13
K_2	271	232	10.4
K_3	288	247	11.1

N = number of blades

$P_{0.25}$ = pitch at 0.25 radius divided by propeller diameter

$P_{0.35}$ = pitch at 0.35 radius divided by propeller diameter, corresponding to the design ahead condition

$P_{0.7}$ = pitch at 0.7 radius divided by propeller diameter, corresponding to the design ahead condition

R = rpm at rated speed

T = maximum design thickness at 0.25 or 0.35 radius from propeller drawing mm (in.)

$t_{0.25}$ = required thickness of blade section at 0.25 of propeller radius; mm (in.)

$t_{0.35}$ = required thickness of blade section at 0.35 of propeller radius; mm (in.)

U_f = maximum normal distance from the moment of inertia axis to points in the face boundary (tension side) of the section; mm (in.)

W = expanded width of a cylindrical section at the 0.25 or 0.35 radius

5.3.3 Blades of Unusual Design

Propellers of unusual design for thruster duties, such as:

- Propellers with the skew angle $\theta > 25°$
- Controllable pitch propellers with skew angle $\theta > 25°$
- Propellers with wide-tip blades and skew angle $\theta > 25°$
- Cycloidal propellers, etc.

are subject to special consideration based on submittal of propeller load and stress analyses. See 4-3-3/5.7.

5.3.4 Propeller Blade Studs and Bolts

5.3.4(a) Area. Studs used to secure propeller blades are to have a cross-sectional area at the minor diameter of the thread of not less than that determined by the equations in 4-3-3/5.13.2.

5.3.4(b) Fit of studs and nuts. Studs are to be fitted tightly into the hub and provided with an effective means for locking. The nuts are also to have a tight-fitting thread and be secured by stop screws or other effective locking devices.

5.3.5 Blade Flange and Mechanism

The strength of the propeller blade flange and internal mechanisms of controllable-pitch propellers subjected to the forces from propulsion torque is to be determined as follows:

- For intermittent duty thrusters, be at least equal to that of the blade design pitch condition.
- For continuous duty thrusters, be at least 1.5 times that of the blade at design pitch condition.

5.5 Gears

5.5.1 Continuous Duty Gears

Gears for continuous duty thrusters are to meet the provisions of Section 4-3-1.

5.5.2 Intermittent Duty Gears

Gears for intermittent duty thrusters, as defined in 4-3-5/1.5.3, are to be in accordance with a recognized standard and are to be submitted for consideration. See e.g., Appendix 4-3-1A1.

5.7 Shafts

5.7.1 Gear Shafts

Gear and pinion shaft diameters are to be determined by the equations in 4-3-1/5.9.

5.7.2 Propeller and Line Shafts

Shafting is to be in accordance with the provisions of 4-3-2/5.1 through 4-3-2/5.17, and cardan shafts, 4-3-2/5.21.

5.7.3 Couplings and Clutches

Shaft couplings, clutches, etc. are to be in accordance with the provisions of 4-3-2/5.19.

5.9 Anti-friction Bearings

Full bearing identification and life calculations are to be submitted. Calculations are to include all gear forces, thrust vibratory loads at maximum continuous rating, etc. The minimum L10 life is not to be less than the following:

i) Continuous duty thrusters (propulsion and **DPS-0**, **1**, **2**, and **3**): 20,000 hours

ii) Intermittent duty thrusters: 5,000 hours

Shorter life may be considered in conjunction with an approved bearing inspection/replacement program reflecting calculated life.

5.11 Steering Systems *(2007)*

Steering systems for azimuthal thrusters are to meet the requirements of Section 4-3-4, as applicable, and the following requirements.

5.11.1 Vessels with Only One Azimuthal Thruster

For vessels that are arranged with only one azimuthal thruster as the only means of propulsion and steering, the thruster is to be provided with steering systems of a redundant design such that a single failure in one system does not effect the other system.

5.11.2 Cargo Vessels with Two Azimuthal Thrusters

For cargo vessels that are arranged with two azimuthal thrusters as the only means of propulsion and steering, each thruster is to be provided with at least one steering system. The steering system for each thruster is to be independent of the steering system for the other thruster.

5.11.3 Passenger Vessels with Two Azimuthal Thrusters

For passenger vessels that are arranged with two azimuthal thrusters as the only means of propulsion and steering, each thruster is to be provided with steering systems of a redundant design such that a single failure in one system does not effect any other system.

5.11.4 Performance

Each azimuthal thruster is to be capable of rotating at a speed of not less than 0.4 rpm (from 35 degrees on either side to 30 degrees on the other side in not more than 28 seconds) while steering the vessel with the vessel running ahead at the maximum continuous rated shaft rpm and at the summer load waterline. Where the azimuthal thruster is arranged to rotate for the crash stop or astern maneuver, the azimuthal thruster is to be capable of rotating at the speed of not less than 2.0 rpm (180 degrees in not more than 15 seconds) to account for the crash stop or astern maneuver.

5.13 Access for Inspection *(2007)*

Adequate access covers are to be provided to permit inspection of gear train without disassembling thruster units.

7 Controls and Instrumentation

7.1 Control System

An effective means of controlling the thruster from the navigation bridge is to be provided. Control power is to be from the thruster motor controller or directly from the main switchboard. Propulsion thrusters are also to be fitted with local means of control.

For specific requirements related to class notation **DPS-0**, **1**, **2** or **3**, see 4-3-5/15.9.

7.3 Instrumentation *(2008)*

Alarms and instrumentation are to be provided in accordance with 4-3-5/Table 1, as applicable.

TABLE 1
Instrumentation for Thrusters

Monitored Parameter	*Navigation Bridge*	*Main Control Station* [1, 2]
Engine low lubricating oil pressure alarm	x [1]	x
Engine coolant high temperature alarm	x [1]	x
Motor overload alarm	x [1]	x
Thruster RPM	x	x
Thrust direction (azimuthing type)	x	x
Thruster power supply failure alarm	x	x
Controllable pitch propellers hydraulic oil low pressure alarm	x [1]	x
Controllable pitch propellers hydraulic oil high pressure alarm	x [1]	x
Controllable pitch propellers hydraulic oil high temperature alarm	x [1]	x
Fire detection	x	x

Notes:

1. Either an individual indication or a common trouble alarm may be fitted at this location, provided individual indication is installed at the equipment (or main control station)

2. For vessels not fitted with a main control station, the indication is to be installed at the equipment or other suitable location

9 Communications

A means of voice communication is to be provided between the navigation bridge, main propulsion control station and the thruster room.

For specific requirements related to class notation **DPS-0**, **1**, **2** or **3**, see 4-3-5/15.11.

11 Miscellaneous Requirements for Thruster Rooms

11.1 Ventilation

Thruster rooms are to be provided with suitable ventilation so as to allow simultaneously for crew attendance and for thruster machinery to operate at rated power in all weather conditions.

11.3 Bilge System for Thruster Compartments

Thrusters installed in normally unattended spaces are to be arranged such that bilge pumping can be effected from outside the space. Alternatively, where bilge pumping can only be effected from within the space, a bilge alarm to warn of high bilge water level is to be fitted in a centralized control station, the navigation bridge or other normally manned control station. For bilge systems in general, see 4-6-4/5.5.11.

Thrusters in enclosed modules (capsules) are to be provided with a high water level alarm. At least one pump capable of bilging the module is to be operable from outside the module.

11.5 Fire Fighting Systems

In general, spaces where thrusters are located, including enclosed modules, are to be protected with fire fighting system in accordance with 4-7-2/1.

13 Certification and Trial

Thrusters and associated equipment are to be inspected, tested and certified by the Bureau in accordance with the following requirements, as applicable:

Diesel engines:	Section 4-2-1
Gas turbines:	Section 4-2-3
Electric motors:	Section 4-8-3
Gears:	Section 4-3-1
Shafting:	Section 4-3-2
Propellers:	Section 4-3-3

Upon completion of the installation, performance tests are to be carried out in the presence of a Surveyor in a sea trial. This is to include but not limited to running tests at intermittent or continuous rating, variation through design range of the magnitude and/or direction of thrust, vessel turning tests and vessel maneuvering tests.

15 Dynamic Positioning Systems

15.1 General

15.1.1 Class Notations and Degree of Redundancy

Dynamic positioning systems may be assigned with different class notations depending on the degree of redundancy built into the system, as defined below. These notations are not a requirement for class and are to be assigned only at specific request.

DPS-0 For vessels which are fitted with a dynamic positioning system with centralized manual position control and automatic heading control to maintain the position and heading under the specified maximum environmental conditions.

DPS-1 For vessels which are fitted with a dynamic positioning system which is capable of automatically maintaining the position and heading of the vessel under specified maximum environmental conditions having an independent centralized manual position control with automatic heading control.

DPS-2 For vessels which are fitted with a dynamic positioning system which is capable of automatically maintaining the position and heading of the vessel within a specified operating envelope under specified maximum environmental conditions during and following any single fault, excluding a loss of compartment or compartments.

DPS-3 For vessels which are fitted with a dynamic positioning system which is capable of automatically maintaining the position and heading of the vessel within a specified operating envelope under specified maximum environmental conditions during and following any single fault, including complete loss of a compartment due to fire or flood.

15.1.2 Definitions

15.1.2(a) Dynamic positioning (DP) system. The *dynamic positioning system* is a hydro-dynamic system which controls or maintains the position and heading of the vessel by centralized manual control or by automatic response to the variations of the environmental conditions within the specified limits.

15.1.2(b) Specified maximum environmental conditions. The *specified maximum environmental conditions* are the specified wind speed, current and wave height under which the vessel is designed to carry out intended operations.

15.1.2(c) Specified operating envelope. The *specified operating envelope* is the area within which the vessel is required to stay in order to satisfactorily perform the intended operations under the specified maximum environmental conditions.

15.1.2(d) Single fault. The *single fault* is an occurrence of the termination of the ability to perform a required function of a component or a subsystem in any part of the DP system. For vessels with **DPS-3** notation, the loss of any single compartment is also to be considered a single fault.

15.1.3 Plans and Data to be Submitted *(2008)*

Where one of the class notations described in 4-3-5/15.1.1 is requested, the following plans and data are to be submitted for review as applicable.

- System description including a block diagram showing how the various components are functionally related
- Details of the position reference system and environmental monitoring systems
- Location of thrusters and control system components
- Details of the DP alarm system and any interconnection with the main alarm system
- Electrical power generation and distribution system and its interconnections with the control system
- Details of the consequence analyzer (see 4-3-5/15.9.6)
- Thruster remote control system
- Automatic DP control and monitoring system
- Certification of suitability of control equipment for the marine atmosphere
- Environmental force calculations and design safe operating envelope
- Thruster design
- Thruster force calculations and predicted polar plots
- Failure modes and effects analysis (FMEA) (see 4-3-5/15.1.4) (**DPS-2** and **DPS-3**)
- DP operations manual (see 4-3-5/15.1.5)
- Test schedule

15.1.4 Failure Modes and Effect Analysis *(2008)*

A failure modes and effect analysis (FMEA) is to be carried out and is to be sufficiently detailed to cover all the systems associated with the dynamic positioning of the vessel and is to include but not be limited to the following information:

- A description of all the systems associated with the dynamic positioning of the vessel and a functional block diagram showing their interaction with each other. Such systems would include the DP electrical or computer control systems, electrical power distribution system, power generation, fuel systems, lubricating oil systems, cooling systems, backup control systems, etc.
- All significant failure modes
- The most predictable cause associated with each failure mode
- The transient effect of each failure on the vessels position
- The method of detecting that the failure has occurred
- The effect of the failure upon the rest of the system's ability to maintain station
- An analysis of possible common failure mode

Where parts of the system are identified as non-redundant and where redundancy is not possible, these parts are to be further studied with consideration given to their reliability and mechanical protection. The results of this further study are to be submitted for review.

15.1.5 DP Operations Manual *(2008)*

For each vessel, a dynamic positioning operations manual is to be prepared and submitted solely for verification that the information in the manual, relative to the dynamic positioning system, is consistent with the design and information considered in the review of the system. One copy of the operations manual is to be kept onboard.

The DP operations manual is intended to provide guidance for the DP operator about the specific dynamic positioning installations and arrangements of the specific vessel. The DP operations manual is to include but not be limited to the following information.

- A description of all the systems associated with the dynamic positioning of the vessel, including backup systems and communication systems
- The block diagram showing how the components are functional related, as described in 4-3-5/15.1.3
- A description of the different operational modes and transition between modes.
- Definitions of the terms, symbols and abbreviations
- A functional description of each system, including backup systems and communication systems
- Operating instructions for the normal operational mode (and the operational modes after a failure) of the DP electrical or computer control systems, manual position control system, manual thruster control system, DP equipment (thrusters, electric motors, electric drives or converters, electric generators, etc.)
- Operating instructions for the systems and equipment, indicated in the above paragraph, during failure conditions
- References to where more specific information can be found onboard the vessel, such as the detailed specific operation instructions provided by the manufacturer of the DP electrical or computer control systems, manufacturer's troubleshooting procedures for vendor-supplied equipment, etc.

15.3 Thruster System

15.3.1 General

In general, the thrusters are to comply with the requirements of 4-3-5/3 through 4-3-5/13, as applicable.

15.3.2 Thruster Capacity

15.3.2(a) Vessels with ***DPS-0*** *or* ***DPS-1*** *notation***.** These vessels are to have thrusters in number and of capacity sufficient to maintain position and heading under the specified maximum environmental conditions.

15.3.2(b) Vessels with ***DPS-2*** *or* ***DPS-3*** *notation*. These vessels are to have thrusters in number and of capacity sufficient to maintain position and heading, in the event of any single fault, under the specified maximum environmental conditions. This includes the failure of any one thruster.

15.3.3 Thruster Configuration

When determining the location of thrusters, the effects due to the interference with other thrusters, hull or other surfaces are to be considered.

15.5 Power Generation and Distribution System

15.5.1 General

The following requirements are in addition to the applicable requirements in Part 4, Chapter 8.

15.5.2 Power Generation System

15.5.2(a) Vessels with ***DPS-2*** *notation*. Generators and their distribution systems are to be sized and arranged such that, in the event of any section of bus bar being lost for any reason, sufficient power is to remain available to supply the essential ship service loads, the critical operational loads and to maintain the vessel position within the specified operating envelope under the specified maximum environmental conditions.

Essential services for generators and their prime movers, such as cooling water and fuel oil systems, are to be arranged such that, with any single fault, sufficient power remains available to supply the essential loads and to maintain position within the specified operating envelope under the specified maximum environmental conditions.

15.5.2(b) Vessels with ***DPS-3*** *notation*. Generators and their distribution systems are to be sized and arranged in at least two compartments so that, if any compartment is lost due to fire or flood, sufficient power is available to maintain position within the specified operating envelope, and to start any non running load without the associated voltage dip causing any running motor to stall or control equipment to drop out.

Essential services for generators and their prime movers, such as cooling water and fuel oil systems, are to be arranged so that, with any single fault in the systems or the loss of any single compartment, sufficient power remains available to supply the essential loads, the critical operational loads and to maintain position within the specified operating envelope under the specified maximum environmental conditions.

15.5.3 Power Management System *(2008)*

For **DPS-2** and **DPS-3** notations, power management systems are is to be provided to ensure that sufficient power is available for essential operations, and to prevent loads from starting while there is insufficient generator capacity. At least two power management systems are to be provided, to account for the possible failure of either power management system. Consideration will be given to techniques such as shedding of non essential loads or interfacing with control system to provide temporary thrust reduction to ensure availability of power.

Each power management system is to be supplied with power from an uninterruptible power system (UPS) and arranged such that a loss of power to one power management system will not result in the loss of power to the other power management systems.

For **DPS-3** notation, the power management systems are to be located and arranged such that no single fault, including fire or flood in one compartment, will render all the power management systems inoperable.

15.5.4 Uninterruptible Power Systems (UPS) *(2008)*

For **DPS-1**, **DPS-2** and **DPS-3** notations, an independent uninterruptible power system is to be provided for each independent control system and its associated monitoring and reference system. Each uninterruptible power system is to be capable of supplying power for a minimum of 30 minutes after failure of the main power supply.

For **DPS-3** notation, the back-up control system required by 4-3-5/15.9.3(c) and its associated reference system is to be provided with a dedicated independent UPS. The uninterruptible power systems are to be located and arranged such that no single fault, including fire or flood in one compartment, will interrupt the power supplied to back up control system required by 4-3-5/15.9.3(c) and its associated reference system.

15.7 Environment Sensor and Position Reference System

15.7.1 Vessels with **DPS-0** or **DPS-1** Notation

For **DPS-0** notation, a position reference system, a wind sensor and a gyro-compass are to be fitted. For **DPS-1** notation, they are to be provided in duplicate.

15.7.2 Vessels with **DPS-2** Notation *(2008)*

In addition to the systems in 4-3-5/15.7.1 for **DPS-1** notation, a third independent position reference system, a third wind sensor and a third gyro-compass are to be provided. Two of the position reference systems may operate on the same principle. A single failure is not to affect simultaneously more than one position reference system (i.e., no common mode failures).

15.7.3 Vessels with **DPS-3** Notation *(2008)*

In addition to the requirements in 4-3-5/15.7.2 for **DPS-2** notation, the third wind sensor, third gyro-compass and the third independent position reference system are to be directly connected to the back-up control station with their signals repeated to the main control station.

15.7.4 Signal Processing

Where three position reference systems are required, the control computers are to use signal processing techniques to validate the data received. When out of range data occurs, an alarm is to be given.

15.9 Control System

15.9.1 Control and Monitoring System Components

In general, control and monitoring (alarms and instrumentation) system components for dynamic positioning systems of vessels intended to be assigned with **DPS** notations are to comply with the provisions of Section 4-9-7.

15.9.2 Control Stations

15.9.2(a) Control station arrangement. The main dynamic positioning control station is to be so arranged that the operator is aware of the external environmental conditions and any activities relevant to the DP operation.

15.9.2(b) Emergency shut-down (2008). An emergency shut-down facility for each thruster is to be provided at the main dynamic positioning control station. The emergency shut-down facility is to be independent of the automatic control systems (4-3-5/15.9.3), manual position control system (4-3-5/15.9.4) and manual thruster control system (4-3-5/15.9.5). The emergency shut-down facility is to be arranged to shut-down each thruster individually.

*15.9.2(c) Vessels with **DPS-3** notation*. For **DPS-3** notation, an emergency back-up control station is to be provided in a separate compartment located and arranged such that no single fault, including a fire or flood in one compartment, will render both the main and back-up control system inoperable.

15.9.3 Position Keeping Control System Redundancy *(2008)*

*15.9.3(a) Vessels with **DPS-1** notation*. An automatic control system and a manual position control system with automatic heading control are to be fitted. Transfer of control between the two systems is to be initiated manually.

*15.9.3(b) Vessels with **DPS-2** notation*. Two automatic control systems and a manual position control system with automatic heading control are to be fitted. The two automatic control systems are to be independent, self-monitoring and arranged such that, should one fail, control is automatically transferred to the other. The cabling for the control systems and the thrusters is to be arranged such that under single fault conditions it will remain possible to control sufficient thrusters to stay within the specified operating envelope.

15.9.3(c) Vessels with ***DPS-3*** *notation.* Three automatic control systems and a manual position control system with automatic heading control are to be fitted. The two automatic control systems located at the dynamic positioning control station are to be independent, self-monitoring and arranged such that, should one fail, control is automatically transferred to the other. The third automatic control system is to be located in the emergency back-up control station and transfer of control to it is to be initiated manually. The cabling for the control systems and the thrusters is to be arranged such that under single fault conditions, including loss of a compartment due to fire or flood, it will remain possible to control sufficient thrusters to stay within the specified operating envelope.

15.9.4 Manual Position Control System *(2008)*

The manual position control system described in 4-3-5/15.9.3 is to be independent of the automatic control systems so that it will be operational if the automatic control systems fail. The system is to provide one joystick for manual control of the vessel position and is to be provided with the arrangements for automatic heading control.

15.9.5 Manual Thruster Control System *(2008)*

In addition to 4-3-5/15.9.3 and 4-3-5/15.9.4, a manual thruster control system is required as per 4-3-5/7.1. The manual thruster control system is to be independent of the automatic control systems so that it will be operational if the automatic control systems fail. The system is to provide an effective means of individually controlling each thruster from the navigation bridge. The system is to provide an individual joystick for each thruster.

Any failure in the manual position control system is not to affect the capabilities of the manual thruster control system to individually control each thruster.

15.9.6 Consequence Analysis and DP Alert System – Vessels with **DPS-2** or **DPS-3** Only

For vessels with **DPS-2** or **DPS-3** notation, the DP control system is to incorporate a consequence analyzer that monitors the vectorial thrust necessary to maintain position under the prevailing environmental conditions and perform calculations to verify that in the event of a single failure there will be sufficient thrust available to maintain position in steady state and during transients.

15.9.7 Alarms and Instrumentation

The displays, alarms and indicators as specified in 4-3-5/Table 2 are to be provided at each control station, as applicable.

TABLE 2
Instrumentation at DPS Control Station *(2008)*

System	*Monitored Parameters*	*Alarm*	*Display*
Thruster Power System	Engine lubricating oil pressure – low	x	
	Engine coolant temperature – high	x	
	CPP hydraulic oil pressure – low and high	x	
	CPP hydraulic oil temperature – high	x	
	CPP pitch		x
	Thruster RPM		x
	Thrust direction		x
	Thruster motor/semiconductor converter coolant leakage	x	
	Thruster motor semiconductor converter temperature		x
	Thruster motor short circuit		x
	Thruster motor exciter power available		x
	Thruster motor supply power available		x
	Thruster motor overload	x	
	Thruster motor high temperature	x	
Power Distribution System	Status of automatically controlled circuit breakers		x
	Bus bar current and power levels		x
	High power consumers – current levels		x
System Performance	Excursion outside operating envelope	x	
	Control system fault	x	
	Position sensor fault	x	
	Vessels target and present position and heading		x
	Wind speed and direction		x
	Selected reference system		x
Specific Requirements for **DPS-2** & **DPS-3**	Thruster location (pictorial)		x
	Percentage thrust		x
	Available thrusters on stand-by		x
	DP alert through consequence analyzer	x	
	Position information of individual position reference systems connected		x

15.11 Communications for Vessels with Dynamic Positioning *(2008)*

One means of voice communication is to be provided between each DP control position and the navigation bridge, the engine control position, any other relevant operation control centers associated with DP and any location required by 4-3-5/9.

15.13 Certification and Trials

15.13.1 Control and Monitoring System Equipment

Control and monitoring (alarms and instrumentation) system equipment used in a dynamic positioning system to be assigned with a **DPS** notation are to be certified for suitability for marine atmospheres.

Hydraulic and pneumatic piping systems associated with the dynamic positioning system are to be subjected to pressure tests at 1.5 times the relief-device setting using the service fluid in the hydraulic system and dry air or dry inert gas for pneumatic systems as testing media. The tests are to be carried out in the presence of a Surveyor.

15.13.2 Trials *(2010)*

Upon completion and installation of the dynamic positioning system, complete performance tests are to be carried out to the Surveyor's satisfaction at the sea trials. The schedule of these tests is to be designed to demonstrate the level of redundancy established in the FMEA (Failure Modes and Effects Analysis, see 4-3-5/15.1.4). Where practicable, the test environment is to reflect the limiting design operating conditions.

PART

4

CHAPTER 3 Propulsion and Maneuvering Machinery

SECTION 6 Propulsion Redundancy *(2005)*

1 General

1.1 Application

The requirements in this section apply to vessels equipped with propulsion and steering systems designed to provide enhanced reliability and availability through functional redundancy. Application of the requirements of this section is optional. When a vessel is designed, built and surveyed in accordance with this section, and when found satisfactory, a classification notation, as specified in 4-3-6/3, as appropriate, will be granted.

It is a prerequisite that the vessels are also to be classed to ✠ **ACCU** notation, in accordance with Part 4, Chapter 9.

1.3 Objective

The objective of this section is to provide requirements which reduce the risk to personnel, the vessel, other vessels or structures, the environment and the economic consequences due to a single failure causing loss of propulsion or steering capability. This is achieved through varying degrees of redundancy based upon the vessel's Classification Notations, as described in 4-3-6/3.

The requirements in this section aim to ensure that following a single failure, the vessel is capable of either:

i) Maintaining course and maneuverability at reduced speeds without intervention by other vessels, or

ii) Maintaining position under adverse weather conditions, as described in 4-3-6/7.3, to avoid uncontrolled drift and navigating back to safe harbor when weather conditions are suitable.

In addition, this section addresses aspects which would reduce the detrimental effects to the propulsion systems due to a localized fire in the machinery spaces.

1.5 Definitions

For the purpose of this section, the following definitions are applicable:

1.5.1 Auxiliary Services System

All support systems (e.g., fuel oil system, lubricating oil system, cooling water system, compressed air and hydraulic systems, etc.) which are required to run propulsion machinery and propulsors.

1.5.2 Propulsion Machinery Space

Any space containing machinery or equipment forming part of the propulsion systems.

1.5.3 Propulsion Machine

A device (e.g., diesel engine, turbine, electrical motor, etc.) which develops mechanical energy to drive a propulsor.

1.5.4 Propulsion System

A system designed to provide thrust to a vessel, consisting of one or more propulsion machines, one or more propulsors, all necessary auxiliaries and associated control, alarm and safety systems.

1.5.5 Propulsor

A device (e.g., propeller, waterjet) which imparts force to a column of water in order to propel a vessel, together with any equipment necessary to transmit the power from the propulsion machinery to the device (e.g., shafting, gearing, etc.).

1.5.6 Steering System

A system designed to control the direction of movement of a vessel, including the rudder, steering gear, etc.

1.7 Plans and Data to be Submitted

In addition to the plans and data required by the Rules, the following are to be submitted:

i) Results of computations showing that, upon any single failure in the propulsion and steering systems, the vessel is able to meet the capability requirements of 4-3-6/7.1, if applicable, with details of the computational methods used. Alternatively, the results of model testing are acceptable as evidence.

ii) A Failure Mode and Effect Analysis (FMEA) or equivalent. The integrity of the propulsion systems, steering systems and auxiliary service systems is to be verified by means of a Failure Mode and Effect Analysis (FMEA) or equivalent method and is to show that a single failure will not compromise the criteria as specified in 4-3-6/7.

iii) A Testing Plan to cover the means whereby verification of the redundancy arrangements will be accomplished.

iv) A general arrangement detailing locations of all machinery and equipment necessary for the correct functioning of the propulsion and steering systems, including the routing of all associated power, control and communication cables. (Required for **R1-S** and **R2-S** only).

v) Operating Manual, as required in 4-3-6/13.

3 Classification Notations

Where requested by the Owner, propulsion and steering installations which are found to comply with the requirements specified in this section and which have been constructed and installed under survey by the Surveyor will be assigned with the following class notations, as appropriate.

i) **R1** A vessel fitted with multiple propulsion machines but only a single propulsor and steering system will be assigned the class notation **R1**.

ii) **R2** A vessel fitted with multiple propulsion machines and also multiple propulsors and steering systems (hence, multiple propulsion systems) will be assigned the class notation **R2**.

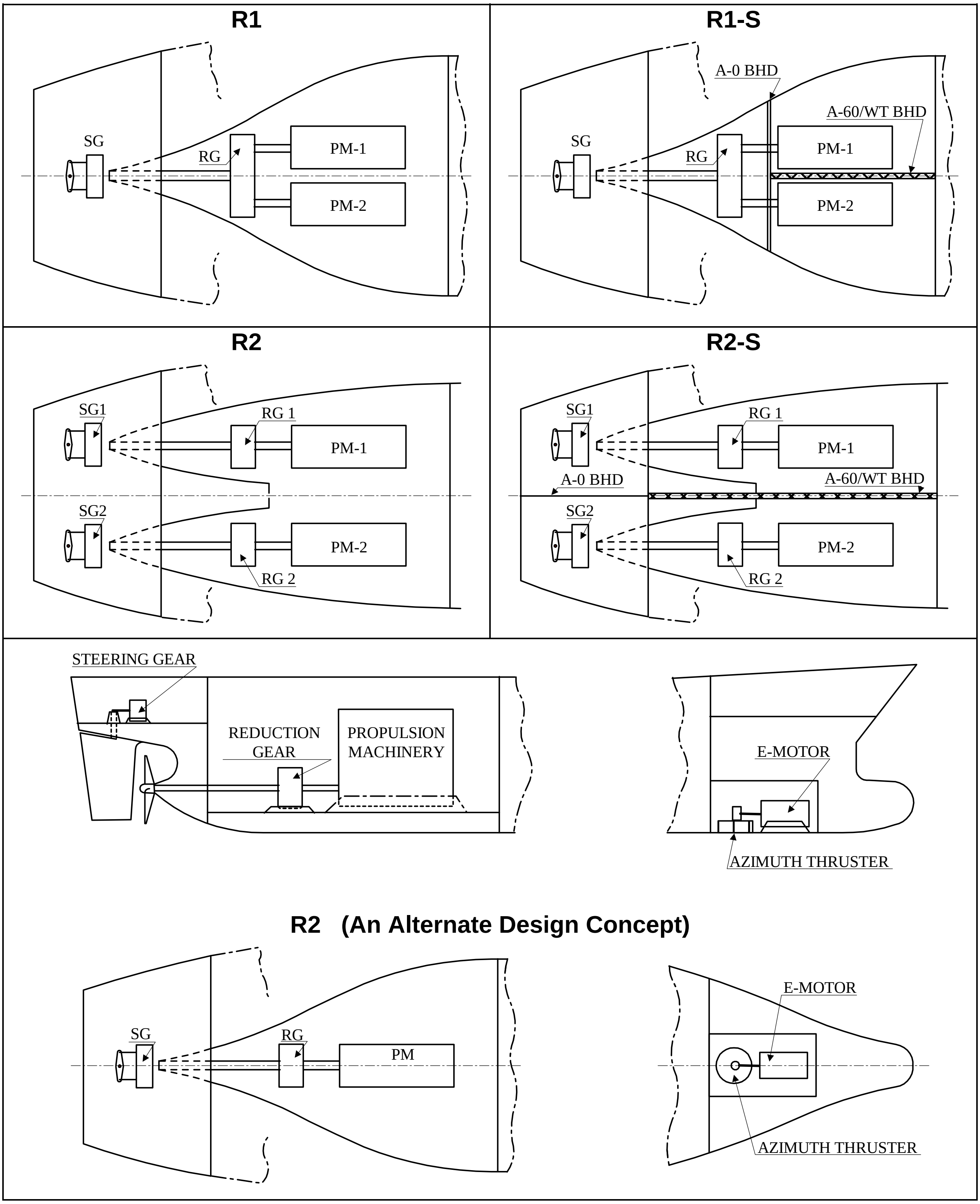
iii) **R1-S** A vessel fitted with only a single propulsor but having the propulsion machines arranged in separate spaces such that a fire or flood in one space would not affect the propulsion machine(s) in the other space(s) will be assigned the class notation **R1-S**.

iv) **R2-S** A vessel fitted with multiple propulsors (hence, multiple propulsion systems) which has the propulsion machines and propulsors, and associated steering systems arranged in separate spaces (propulsion machinery space and steering gear flat) such that a fire or flood in one space would not affect the propulsion machine(s) and propulsor(s), and associated steering systems in the other space(s) will be assigned the class notation **R2-S**.

Example arrangements for each of the above notations are shown in 4-3-6/Figure 1.

v) **+** (Plus Symbol) The mark **+** will be affixed to the end of any of the above class notations (e.g., **R1+**, **R2-S+**) to denote that the vessel's propulsion capability is such that, upon a single failure, propulsive power can be maintained or immediately restored to the extent necessary to withstand adverse weather conditions without drifting, in accordance with 4-3-6/7.3. The lack of the mark **+** after the class notation indicates that the vessel is not intended to withstand the adverse weather conditions in 4-3-6/7.3, but can maintain course and maneuverability at a reduced speed under normal expected weather conditions, in accordance with 4-3-6/7.1.

FIGURE 1
Arrangements of Propulsion Redundancy



5 Single Failure Concept

The degree of redundancy required to meet the objectives of this section is based upon a single failure concept. The concept accepts that failures may occur but that only one such failure is likely at any time. The final consequence of any single failure is not to compromise the propulsion and steering capability required in 4-3-6/7, unless otherwise specified.

5.1 Single Failure Criteria

5.1.1 **R1** Notation

For **R1**, the single failure criterion is applied to the propulsion machines, its auxiliary service systems and its control systems. This notation does not consider failure of the propulsor or rudder, or total loss of the propulsion machinery space or steering gear flat due to fire or flood.

5.1.2 **R2** Notation

For **R2**, the single failure criterion is applied to the propulsion machines, propulsors, auxiliary service systems, control systems and steering systems. This notation does not consider total loss of the propulsion machinery space or steering gear flat due to fire or flood.

5.1.3 **R1-S** Notation

For **R1-S**, the single failure criterion is applied as for **R1,** but a fire or flood in one of the propulsion machinery spaces is also considered.

5.1.4 **R2-S** Notation

For **R2-S**, the single failure criterion is applied as for **R2**, but a fire or flood in one of the propulsion machinery spaces or steering gear flats is also considered.

7 Propulsion and Steering Capability

7.1 Vessels Without + in Class Notation

Upon a single failure, the propulsion system is to be continuously maintained or restored within two (2) minutes (for alternate standby propulsion per 4-3-6/7.3 below) such that the vessel is capable of advancing at a speed of at least one-half its design speed or seven knots, whichever is less, for at least 36 hours when the vessel is fully loaded. Adequate steering capability is also to be maintained at this speed.

7.3 Vessels with + in Class Notation

In addition to 4-3-6/7.1 above, upon a single failure, the propulsion and steering system is to be continuously maintained or immediately restored within two (2) minutes, as in the case when an alternate standby type of propulsion is provided, (e.g., electric motor, diesel engine, waterjet propulsion, etc.) such that the vessel is capable of maneuvering into an orientation of least resistance to the weather, and once in that orientation, maintaining position such that the vessel will not drift for at least 36 hours. This may be achieved by using all available propulsion and steering systems including thrusters, if provided. This is to be possible in all weather conditions up to a wind speed of 17 m/s (33 knots) and significant wave height of 4.5 m (15 ft) with 7.3 seconds mean period, both of which are acting concurrently in the same direction. The severest loading condition for vessel's maneuverability is also to be considered for compliance with this weather criterion. Compliance with these capability requirements is to be verified by computational simulations, and the detailed results are to be submitted for approval. The estimated optimum capability is to be documented in the operating manual, as required in 4-3-6/13.

9 System Design

9.1 Propulsion Machinery and Propulsors

At least two independent propulsion machines are to be provided. As appropriate, a single failure in any one propulsion machine or auxiliary service system is not to result in propulsion performance inferior to that required by 4-3-6/7.1 or 4-3-6/7.3, as applicable.

9.1.1. **R1** Notation

For **R1** notation, the propulsion machines and auxiliary service systems may be located in the same propulsion machinery space and the propulsion machines may drive a single propulsor.

9.1.2 **R2** Notation

For **R2** notation, at least two propulsors are to be provided such that a single failure of one will not result in propulsion performance inferior to that required by 4-3-6/7.1 or 4-3-6/7.3, as applicable. The propulsion machines and auxiliary service systems may, however, be located in the same propulsion machinery space.

9.1.3 **R1-S** Notation

For **R1-S** notation, the propulsion machines and auxiliary service systems are to be separated in such a way that total loss of any one propulsion machinery space (due to fire or flood) will not result in propulsion performance inferior to that required by 4-3-6/7.1 or 4-3-6/7.3, as applicable. The propulsion machines may, however, drive a single propulsor, and the main propulsion gear or main power transmitting gear is to be located outside the propulsion machinery spaces separated by a bulkhead meeting the criteria per 4-3-6/9.3.

9.1.4 **R2-S** Notation

For **R2-S** notation, at least two propulsors are to be provided, and the propulsion systems are to be installed in separate spaces such that a single failure in one propulsor or a total loss of any one propulsion machinery space (due to fire or flood) will not result in propulsion performance inferior to that required by 4-3-6/7.1 or 4-3-6/7.3, as applicable.

9.3 System Segregation

Where failure is deemed to include loss of a complete propulsion machinery space due to fire or flooding (**R1-S** and **R2-S** notations), redundant components and systems are to be separated by watertight bulkheads with an A-60 fire classification.

Service access doors which comply with 3-2-9/9.1 may be provided between the segregated propulsion machinery spaces. A means of clear indication of open/closed status of the doors is to be provided in the bridge and at the centralized control station. Unless specially approved by the flag Administration, these service access doors are not to be accounted for as the means of escape from the machinery space Category A required by the requirements of Regulation II-2/13 of SOLAS 1974, as amended.

9.5 Steering Systems

An independent steering system is to be provided for each propulsor. Regardless of the type and the size of vessel, each steering system is to meet the requirements of Regulation II-1/29.16 of SOLAS 1974, as amended.

The rudder design is to be such that the vessel can turn in either direction with one propulsion machine or one steering system inoperable.

For **R2-S** notation, the steering systems are to be separated such that a fire or flood in one steering compartment will not affect the steering system(s) in the other compartment(s), and performance in accordance with 4-3-6/7.1 or 4-3-6/7.3, as applicable, is maintained.

For **R2** and **R2-S** notations, in the event of steering system failure, means are to be provided to secure rudders in the amidships position.

9.7 Auxiliary Service Systems

At least two independent auxiliary service systems, including fuel oil service tanks, are to be provided and arranged such that a single failure will not result in propulsion performance inferior to that required by 4-3-6/7.1 or 4-3-6/7.3, as applicable. However, a single failure in the vital auxiliary machinery (e.g., pumps, heaters, etc.), excluding failure of fixed piping, is not to result in reduction of the full propulsion capability. In order to meet this requirement, it will be necessary to either cross-connect the auxiliary service systems and size the components (pumps, heaters, etc.) to be capable of supplying two or more propulsion machines simultaneously, or provide duplicate components (pumps, heaters, etc.) in each auxiliary system in case one fails.

With the exception of the fuel oil service tank venting system, interconnections between auxiliary service systems will be considered, provided that the same are fitted with means (i.e., valves) to disconnect or isolate the systems from each other.

For **R1-S** and **R2-S** notations, the above-mentioned independent auxiliary service systems are to be segregated in the separate propulsion machinery spaces. With the exception of fuel oil service tank venting systems, interconnections of auxiliary service systems will be acceptable, provided that the required disconnection or isolation means are fitted at both sides of the bulkhead separating the propulsion machinery spaces. Position status of the disconnection or isolation means is to be provided at the navigation bridge and the centralized control station. Penetrations in the bulkhead separating the propulsion machinery spaces and steering gear flats (as in the case of **R2-S** notation) are not to compromise the fire and watertight integrity of the bulkhead.

9.9 Electrical Distribution Systems

Electrical power generation and distribution systems are to be arranged such that following a single failure in the systems, the electrical power supply is maintained or immediately restored to the extent that the requirements in 4-3-6/7 are met.

Where the vessel’s essential equipment is fed from one main switchboard, the bus bars are to be divided into at least two sections. Where the sections are normally connected, detection of a short circuit on the bus bars is to result in automatic separation. The circuits supplying equipment essential to the operation of the propulsion and steering systems are to be divided between the sections such that a loss of one section will not result in performance inferior to that defined in 4-3-6/7. A fully redundant power management system is to be provided so that each section of the switchboard can function independently.

For **R1-S** and **R2-S** notations, the ship service power generators, their auxiliary systems, the switchboard sections and the power management systems are to be located in at least two machinery spaces separated by watertight bulkheads with an A-60 fire classification. The power distribution is to be so arranged that a fire or flooding of one machinery space is not to result in propulsion capability inferior to that defined in 4-3-6/7. Where an interconnection is provided between the separate propulsion machinery spaces, a disconnection or isolation means are to be provided at both sides of the bulkhead separating the propulsion machinery spaces. Position status of the disconnection or isolation means is to be provided at the navigation bridge and the centralized control station. Fire or flooding of one machinery space is not to result in propulsion capability inferior to that defined in 4-3-6/7. The power cables from the service generator(s) in one propulsion machinery space are not to pass through the other propulsion machinery space containing the remaining service generator(s).

Additionally, for **R1-S** and **R2-S** notations, subject to approval by the Administration, the requirements for self-contained emergency source of power may be considered satisfied without an additional emergency source of electrical power, provided that:

i) All generating sets and other required sources of emergency source of power are designed to function at full rated power when upright and when inclined up to a maximum angle of heel in the intact and damaged condition, as determined in accordance with Part 3, Chapter 3. In no case need the equipment be designed to operate when inclined more than 22.5° about the longitudinal axis and/or when inclined 10° about the transverse axis of the vessel.

ii) The generator set(s) installed in each machinery space is of sufficient capacity to meet the requirements of 4-8-2/3 and 4-8-2/5.

iii) The arrangements required in each machinery space are equivalent to those required by 4-8-2/5.9.1, 4-8-2/5.13 and 4-8-2/5.15, so that a source of electrical power is available at all times for the services required by 4-8-2/5.

9.11 Control and Monitoring Systems

The control systems are to be operable both independently and in combination from the bridge or the centralized control station. The mode of operation is to be clearly indicated at each position from which the propulsion machinery may be controlled.

It is to be possible to locally control the propulsion machinery and the propulsor.

For **R1-S** and **R2-S** notations, the control and monitoring system for the propulsor (e.g., controllable pitch propeller control), including all associated cabling, is to be duplicated in each space, and fire or flooding of one space is not to adversely affect operation of the propulsor from the other space.

9.13 Communication Systems

The requirements of 4-8-2/11.5 are to be complied with for all installed propulsion control positions.

For **R1-S** and **R2-S** notations, the communications cables to each control position are not to be routed through the same machinery space.

11 Fire Precautions

The requirements of this section apply to Category A machinery spaces only.

Pumps for oil services are to be fitted with shaft sealing devices, which do not require frequent maintenance to prevent oil leakage, such as mechanical seals.

For **R1** and **R2** notations, the following requirements are to be complied with in order to minimize the risk of common damage due to a localized fire in the machinery space.

i) Each auxiliary services system is to be grouped and separated as far as practicable.

ii) Electrical cables supplying power to redundant equipment are to exit the switchboard and be routed to the equipment, as far apart as practicable.

13 Operating Manual

An operating manual, which is consistent with the information and criteria upon which the classification is based, is to be placed aboard the vessel for the guidance of the operating personnel. The operating manual is to give clear guidance to the vessel's crew about the vessel's redundancy features and how they may be effectively and speedily put into service in the event that the vessel's normal propulsion capability is lost. The operating manual is to include the following, as a minimum:

i) Vessel's name and ABS ID number

ii) Simplified diagram and descriptions of the propulsion systems in normal condition

iii) Simplified diagram and descriptions of the propulsion redundancy features

iv) Reduced propulsion capability in terms of estimated worst sea-states which the vessel may withstand without drifting (for vessels with **+** in the Class Notation)

v) Test results for the vessel's maneuverability at reduced speed (for vessels without **+** in the Class Notation).

vi) Step-by-step instructions for the use of the redundancy features

vii) Description of the communication systems

viii) Detailed instructions for local propulsion machinery control

The operating manual is to be submitted for review by the American Bureau of Shipping solely to ensure the presence of the above information, which is to be consistent with the design information and limitations considered in the vessel's classification. The American Bureau of Shipping is not responsible for the operation of the vessel.

Any modifications made to the existing propulsion systems are to be approved by the Bureau. The operating manual is to be updated accordingly and submitted to the Bureau for review.

15 Test and Trial

During the sea trial, the propulsion and steering capability are to be tested in accordance with an approved test program to verify compliance with this section.

15.1 Fault Simulation Test

Simulation tests for the redundancy arrangements are to be carried out to verify that, upon any single failure, the propulsion and steering systems remain operational, or the back-up propulsion and steering systems may be speedily brought into service.

15.3 Communication System Test

The effectiveness of the communication systems, as required in 4-3-6/9.13 above, is to be tested to verify that local control of the propulsion systems may be carried out satisfactorily.

17 Survey After Construction

The surveys after construction are to be in accordance with the applicable requirements as contained in the ABS *Rules for Survey After Construction (Part 7).*