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CR Classification Society

FOUNDED 1951

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

April 2018

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PART I – CLASSIFICATION AND SURVEY

April 2018

List of major changes in Part I from 2017 edition

Nil.

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART I CLASSIFICATION AND SURVEY

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Chapter 1

Classification of Aluminum Vessels

1.1 General

1.1.1 Vessels of aluminum alloys which have been built and surveyed in accordance with the Rules for the Construction and Classification of Aluminum Vessels (hereinafter referred to as the Rules) published by CR Classification Society (hereinafter referred to as the Society) or alternatives found to represent an overall safety standard equivalent to that of the Rules will be assigned a class in the Register of Ships (hereinafter referred to as the Register) and will continue to be classed so long as they are found, upon examination at the prescribed surveys, to be maintained in a fit and efficient condition and in accordance with the requirements of the Rules. The Rules apply to vessels under 90m in length and 100 GT and over intended for unrestricted ocean service. Vessels different from above may be specially considered.

(a) For HSC ships: In addition to the applicable requirements, they are also to follow the requirements of the Rules for the Construction and Classification of High-Speed Craft (hereinafter referred to as the Rules for High-Speed Craft) and the requirements relevant to the Rules for the Construction and Classification of Steel Ships (hereinafter referred to as the Rules for Steel Ships), if concerned.

(b) For Non-HSC ships: In addition to the applicable requirements, they are also to follow the requirements relevant to the Rules for Steel Ships, if concerned.

1.1.2 Classification will be conditional upon compliance with the Rules in respect of both hull and machinery (i.e., main and auxiliary engines, boilers, essential appliances, pumping arrangements and electrical equipment).

1.1.3 The Rules are framed on the understanding that ships will be properly loaded and handled; they do not, unless stated in the class notation, provide for special distributions or concentrations of loading. The Society may also require additional strengthening to be fitted in any ship which, in their opinion, may be subjected to severe stresses due to particular features in her design, or when it is desired to make provisions for exceptionally loaded or ballasted conditions. In these cases particulars are to be submitted for consideration.

1.1.4 The Rules do not cover certain technical characteristics, such as trim, hull vibration, etc., but the Society is willing to advise on such matters although it cannot assume responsibility for them.

1.1.5 Alternatives are to be accepted, provided that they are considered by the Society to be equivalent to the Rules.

1.1.6 Ships the construction of which involves novel features of design in respect of hull, machinery or equipment and to which the provisions of the Rules are not directly applicable may be classed, when approved by the Society on the basis that the Rules insofar as applicable have been complied with and that special consideration based on the best information available at the time has been given to the novel features. The Rules are framed on the understanding that ships are not to be operated in environmental conditions more than those agreed for the design basis and approval, without the prior agreement of the Society.

1.1.7 Register

Aluminum vessels with their class approved by the Society are to be recorded in the Register. The Register is to be printed annually and is to contain the names of ships and other useful items of information such as class notations, owners, shipbuilders, dimensions, machinery particulars, the date of build, etc.

1.1.8 Date of build

(a) The date of build is normally to be the date of completion of the classification initial survey during construction of ships built under the inspection.

(b) If the period between launching and completion or putting a ship to use is unduly made longer than usual, the date of launching may be additionally indicated in the Register.

(c) If a ship is not immediately put into service after completion, but is laid up for a period, the ship is to be dry-docked (or on slipway) for examination by the Surveyor of the Society (hereinafter referred to as the Surveyor) before proceeding to sea, and the subsequent special survey is based on the date of such an examination provided that the result of such survey is satisfactory in all respects.

1.1.9 Loading conditions and any other preparations required to permit a ship with a notation specifying some service limitation to undertake a service voyage, either from port of loading to service area or from one service area to another, are to be in accordance with arrangement agreed by the Society prior to the voyage.

1.1.10 Damage, repairs and alternations

Any damage, defect, breakdown or grounding, which could invalidate the conditions for which a class has been assigned, is to be reported to the Society without delay.

1.1.11 For ships, the arrangements and equipment of which are required to comply with the requirements of the International Convention Regulations and applicable Protocols and Amendments relating thereto, such compliance is to be demonstrated by possession of applicable Convention Certificates issued by the Government of the State whose flag the ship is flying (hereinafter refer to as the Administration) or by any organization authorized by the Administration.

1.1.12 Where an on-board computer system having either a longitudinal strength or a stability computation capability or both, is provided on new ships, or newly installed on existing ships, then the system is to be certified for such use in accordance with the Society's procedure for approval of on-board computer systems for stability calculation.

1.2 Application

1.2.1 Except in the case of a special directive by the Society, no new Regulation or alteration to any existing Regulation relating to character of classification or to class notation is to be applied to existing ships.

1.2.2 Except in the case of a special directive by the Society, no new Rule or alteration to any existing Rule materially affecting classification is to be applied compulsorily within six months of its adoption, nor after the approval of the original midship section or equivalent structural plans. Where it is desired to use existing previously approved plans for a new contract, written application is to be made to the Society.

1.3 Classification Symbols and Notations

1.3.1 Vessels of aluminum alloys which have been built/surveyed to the satisfaction of the Society and approved by the Committee will be additionally assigned with **Aluminum Alloy Hull** notaion indicating compliance with the hull requirements of the Rules.

1.3.2 In addition to the above notation, they are also to follow the requirements of the Rules for Steel Ships and/or the requirements relevant to the Rules for High-Speed Craft, if concerned.

1.4 Application for Classification and Surveys

1.4.1 An application for the classification of ships is to be submitted in writing or typing by the builder for a ship to be built under the Society's survey or by the owner or owner's representative for a ship not built under the Society's survey.

1.4.2 An application for surveys for maintenance of class is to be submitted in writing or typing by e-mail or fax by the owner or owner's representative.

1.5 Surveys of Aluminum Vessels

1.5.1 General

(a) All ships classed with the Society are subjected to the periodical surveys followed the requirements of the Rules for Steel Ships and/or the requirements relevant to the Rules for High-Speed Craft.

(b) The contents of surveys: In addition to the applicable requirements, they are also to follow the requirements of the Rules for Steel Ships and/or the requirements relevant to the Rules for High-Speed Craft.

1.5.2 Classification initial survey during construction

(a) New ships are to be built in accordance with the Rules. The constructional plans and particulars of the hull, equipment and machinery are to be submitted for approval of the Society as required in the Rules for Steel Ships and/or in the Rules for High-Speed Craft before the work is commenced. In case these plans need to be modified or altered, a re-approval is indispensable.

(b) The new machinery including boilers, pressure vessels and electrical equipment for ships classed or intended to be classed is to be manufactured under and surveyed according to the applicable requirements in the Rules for Steel Ships.

(c) From the commencement of the work until the completion of the ship and final test of the machinery under working condition, the Surveyors are to be satisfied that the materials, workmanship and arrangements are satisfactory and in accordance with the Rules or the approved plans, or any material, workmanship or arrangement found to be unsatisfactory, it is to be rectified.

(d) The materials used in the construction of hulls and machinery intended for classification are to be of good quality and free defects and are to be tested in accordance with the requirements of the Rules. The aluminum alloys is to be manufactured by an approved process at works recognized by the Society. Alternatively, tests to the satisfaction of the Society will be required to demonstrate the suitability of the material.

(e) Copies of finish plans (showing the ship as built), essential Certificates and records, required loading and other instruction manuals are to be readily available for use when required by the Surveyor, and may be required to be kept on board.

1.5.3 Classification survey of ships not built under survey

(a) Ships which have not been built under survey to the Society, but which are submitted for classification, are to be subjected to a classification survey of ships not built under survey.

1.6 Approval

1.6.1 Survey reports

(a) Upon completion of a survey of a classed ship, the Surveyor is to send one original and one copy of his/her reports with recommendations, if set up, to the applicant, and at the same time, two copies to the Society in which one copy to the administration. The Society reserves the right for final decision on the Surveyor's recommendations.

1.6.2 Decision of classification

(a) Any member of the Committee or the staff of the Society having either direct or indirect interest in a ship to be classed, is not permitted to be present at or to participate in the meeting for the decision of the classification.

1.7 Certificates of Classification

See the relevant sections in Part I of the Rules for Steel Ships.

1.8 Notice of Surveys

1.8.1 It is the responsibility of the Owners to ensure that all surveys necessary for the maintenance of class are carried out at the proper time and in accordance with the instruction of the Society.

1.8.2 The Society will give timely notice to an Owner about forthcoming surveys by means of a letter or e-mail. The omission of such notice, however, does not absolve the Owner from his responsibility to comply with CR's survey requirements for maintenance of class.

1.9 Withdrawal of Class

See the relevant sections in Part I of the Rules for Steel Ships.

1.10 Reclassification

See the relevant sections in Part I of the Rules for Steel Ships.

1.11 Survey Fees and Expenses

See the relevant sections in Part I of the Rules for Steel Ships.

1.12 International Conventions and Codes

Where authorized by the government of the country in which a ship is registered or intended to be registered and upon request by builders or owners of the ship, the Society is to survey a new or existing ship for compliance with the provisions of International Conventions and Codes.

1.13 Governmental Regulations

Where authorized by a government agency and upon request of the Owners of the ships, the Society will survey and certify a new or existing ship for compliance with particular regulations of that government.

1.14 Sea Trial

In the classification survey of all ships, sea trials specified in the Rules for Steel Ships and/or Rules for High-Speed Craft are to be complied with.

1.15 Stability Experiment

1.15.1 In the classification survey, stability experiment by inclining test of a ship are to be carried out upon completion of the ship. A stability information booklet, which is to be prepared on the basis of the particulars of stability determined by the results of stability experimental and to be approved by the Society, is to be provided on board.

1.15.2 In the classification survey of the ships not built under the Society‘s survey, inclining test may be dispensed with, provided that sufficient information based on previous inclining tests is available and neither alteration nor repair affecting the stability has been made after the previous tests.

1.15.3 The inclining test of an individual ship may be dispensed with, provided that available stability data are obtained from the inclining tests of a sister ship or other adequate means and a special approval is given by the Society.

1.16 Liability and Compensation

All services conducted by the Society shall follow the clauses of liability and compensation in the Rules for Steel Ships.

Chapter 2

Survey Requirements of Aluminum Vessels

2.1 General

2.1.1 General

(a) The Surveyor is to have free access at any time in order to examine a classed ship and to make sure of her good condition.

(b) When a survey becomes due or any damage or alterations which may affect the technical fitness or the class to the hull or machinery of the ship occurred, the owner or his representative is to apply in time for a survey to be made without waiting for notice from the Society.

(c) In the case of any disagreement or dispute between the owner and the Surveyor or other officers regarding the inspection, examination and survey work, an appeal in writing or typing from owner or owner's representative which is submitted by e-mail or fax for re-survey or explanation may be made to the Society.

(d) Though the survey of a certain part of the ship being surveyed is not included in this chapter, the Surveyor may, if deemed necessary, make an additional survey of such a part. The Head Office of the Society also reserves the rights to perform an occasional survey whenever reasonable necessity exists.

(e) Modification of requirements

(i) At the periodical surveys, the Surveyor may modify the requirements for periodical surveys specified in this chapter having regard to the size, service engaged, age, construction, results of last surveys and actual condition of the ship.

(ii) For spaces where effective coatings are found to be in a GOOD condition, the extent of internal examination or gauging requirements specified in this chapter may be specially considered at the discretion of the Surveyor.

(f) Additional requirements to prevent from the detention by Port State Control and to ensure the safety for bulk carriers, general dry cargo ships and tankers over 15 years old and for non general dry cargo ships over 20 years old when carry out the periodical survey:

(i) For spaces where coatings are found to be in a POOR condition, the spaces are to be de-rusted/de-scaled, thickness measured and examined. If the measured areas were found in substantial corrosion condition, the areas are to be cropped and renewed before the periodical survey was completed. Otherwise, the spaces are to be de-rusted/de-scaled, thickness measured and examined annually. If the measured spaces were not in substantial corrosion condition, the spaces are to be re-coated to be at least in FAIR condition before the periodical survey was completed. Otherwise, the spaces are to be de-rusted/de-scaled, thickness measured and examined annually.

(ii) For spaces where the substantial areas are found, although the coatings are in a FAIR or GOOD condition, the substantial corrosion areas are to be cropped and renewed before the periodical survey was completed. Otherwise, the substantial corrosion areas are to be de-rusted/de-scaled, thickness measured and examined annually.

2.1.2 Definitions

(a) An overall survey is a survey intended to report on the overall condition of the hull structure and determine the extent of additional close-up surveys.

(b) A close-up survey is a survey where the details of structural components are within the close visual inspection range of the Surveyor, i.e. normally within the reach of hand.

2.1.3 Procedures for class related services

See the relevant requirements in Part I paragraph 2.1.3 of the Rules for Steel Ships.

2.2 Bottom Surveys

See relevant requirements in Part I of the Rules for Steel Ships. Special attention is to be given to the connections of dissimilar metals and to all parts liable to rapid deterioration.

2.3 Propeller Shaft and Tube Shaft Surveys

See relevant requirements in Part I of the Rules for Steel Ships.

2.4 Boiler and Thermal Oil Heater Surveys

See relevant requirements in Part I of the Rules for Steel Ships.

2.5 Annual Surveys

See relevant requirements in Part I of the Rules for Steel Ships. Special attention is to be given to the connections of dissimilar metals and to all parts liable to rapid deterioration.

2.6 Intermediate Surveys

See relevant requirements in Part I of the Rules for Steel Ships. Special attention is to be given to the connections of dissimilar metals and to all parts liable to rapid deterioration.

2.7 Special Surveys

See relevant requirements in Part I of the Rules for Steel Ships. Special attention is to be given to the connections of dissimilar metals and to all parts liable to rapid deterioration.

2.8 Surveys of Refrigerated Cargo Installations

See relevant requirements in Part I of the Rules for Steel Ships.

2.9 Surveys of Special Type Vessels

See relevant requirements in Part I of the Rules for Steel Ships.

2.10 Classification Survey of Ships not Built under Survey

See relevant requirements in Part I of the Rules for Steel Ships.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART II – HULL CONSTRUCTION AND EQUIPMENT

April 2018

List of major changes in Part II from 2017 edition

Nil.

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

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Chapter 1

General

1.1 General

1.1.1 This Part is framed for aluminum high-speed craft of speed greater or equal to that defined in 1.1.3. Special consideration is to be given to crafts of novel design or for special purpose.

1.1.2 The structural design which is not in direct compliance with the requirements of this Part may be accepted by the Society provided that such design is considered satisfactory and equivalent to the Rules.

1.1.3 "High-speed craft" is a craft capable of maximum speed, in meters per second (m/s), equal to or exceeding:

$3.7\nabla^{0.1667}$

where:

∇ = Volume of displacement corresponding to the design waterline (m^3).

but excluding craft the hull of which is supported completely clear above the water surface in non-displacement mode by aerodynamic forces generated by ground effect.

1.1.4 For aluminum craft of maximum speed less than that defined in 1.1.3, structural scantlings are to comply with Chapter 2 of this Part, but rational reduction in scantlings may be made at the discretion of this Society and the equipment may be subjected to the special consideration based on Chapter 4 of this Part or the Rules for the Construction and Classification of Steel Ships. These measures are to be approved by this Society.

1.2 Stability

1.2.1 Intact stability

(a) For all aluminum cargo craft ≥ 500 GT making voyages that are no more than 8 hours at 90% maximum speed from a place of refuge and having maximum speeds equal to or exceeding $3.7\nabla^{0.1667}$ (m/s), the intact stability is to comply with the Chapter 2 of the Rules for the Construction and Classification of High-Speed Craft of this Society.

(b) For all aluminum passenger craft making voyages that are no more than 4 hours at 90% maximum speed from a place of refuge and having maximum speeds equal to or exceeding $3.7\nabla^{0.1667}$ (m/s), the intact stability is to comply with the Chapter 2 of the Rules for the Construction and Classification of High-Speed Craft of this Society.

(c) For other craft, the intact stability is to comply with the IMO International Code on Intact Stability.

(d) In case the above criteria are not applicable to a particular craft, the intact stability will be reviewed by this Society in accordance with other recognized criteria appropriate to the craft's type, size, and intended service.

1.2.2 Damage stability

(a) For all aluminum cargo craft ≥ 500 GT making voyages that are no more than 8 hours at 90% maximum speed from a place of refuge and having maximum speeds equal to or exceeding $3.7\nabla^{0.1667}$ (m/s), the damage stability is to comply with the Chapter 2 of the Rules for the Construction and Classification of High-Speed Craft of this Society.

(b) For all aluminum passenger craft making voyages that are no more than 4 hours at 90% maximum speed from a place of refuge and having maximum speeds equal to or exceeding $3.7\nabla^{0.1667}$ (m/s), the damage stability is to comply with the Chapter 2 of the Rules for the Construction and Classification of High-Speed Craft of this Society.

(c) For other aluminum cargo craft ≥ 80m and aluminum passenger craft, the damage stability is to comply with the SOLAS Convention Chapter II-1 Part B.

(d) In case the above criteria are not applicable to a particular craft, the damage stability will be reviewed by this Society in accordance with other recognized criteria appropriate to the craft's type, size, and intended service.

1.3 Principles for Hull Structure Arrangement

1.3.1 Framing, webs, girders, and non-tight structural bulkheads

(a) The shell, main weather or freeboard deck, and the sides and tops of long superstructures are in general to be longitudinally framed; depending on craft length, speed and structural stability, craft may also be transversely framed.

(b) Bulkheads, partial bulkheads or web frames are to be arranged in the main hull and in long superstructures or deckhouses to provide effective transverse rigidity. They are to be provided also in the main hull under the ends of superstructures or deckhouses.

(c) Longitudinal frames are to be supported by transverse web frames, transverse bulkheads or other transverse structure. Longitudinals are in general to be continuous in way of transverse supporting members except at transverse bulkheads where they may be intercostal provided continuity of strength and end fixity are maintained. Depending on craft length and details, special consideration will be given to longitudinals being intercostal at transverse webs. With transverse framing, deck and bottom girders are to be provided. Girders may be intercostal at transverse bulkheads provided continuity of strength is maintained and end fixity is provided.

(d) Transverses are to be arranged as continuous web rings and girders are to be aligned with stiffeners at bulkheads. Alternative arrangements that provide fixity at the ends of transverses and girders will be specially considered.

(e) Engines are to be supported and secured by substantial girders, suitably stiffened, supported against tripping and supported at bulkheads. Foundations for auxiliary machinery are to provide for secure attachment of the equipment and are to be effectively attached to the hull structure. A substantial foundation and seating is to be provided for the anchor winch or windlass.

(f) At supporting members, the attachment of all internal structural members is to provide end fixity, and effective load transmission. Special consideration will be given to reduced end fixity where the alternative structure has equivalent strength.

(g) The webs of all members are to be effectively attached to the shell, deck or bulkhead plating, to their supporting members and to face bars.

1.3.2 Watertight Bulkheads

(a) Collision bulkhead

Crafts are to be provided with a collision bulkhead fitted within the range in 0.05L and 0.05L+3 m abaft the stem at the design load waterline, where L is defined in 2.1.8. The bulkheads are to be intact except for approved pipe penetrations, and are to extend to the main weather deck preferably in one plane. In craft having long superstructures at the forward end, the bulkheads are to be extended weathertight to the superstructure deck. Provided the extensions are not less than 0.05L abaft the stem at the design load waterline, they need not be fitted directly over the collision bulkhead; in such cases, the part of the deck forming the step is to be weathertight. Special consideration will be given to the arrangements of collision bulkheads for governmental service craft such as patrol boats, search and rescue craft etc.

(b) Engine Room

The engine room is to be enclosed by watertight bulkheads at both ends extending to the main weather deck.

(c) Chain Locker

Chain lockers which located abaft of collision bulkheads and extending into forepeak tanks are to be watertight.

1.3.3 Tanks

(a) The arrangements of all integral tanks, their intended service, and the heights of the overflow pipes are to be indicated clearly on the drawings submitted for approval.

(b) All tanks and void spaces are to be accessible for inspection and repair.

1.3.4 Shell Plating

The bottom shell plating is to extend to the chine or upper turn of bilge. In general, the side shell is to be of the same thickness from its lower limit to the gunwale. Increases in thickness are required in way of skegs, shaft struts, hawse pipes etc. Bow thruster tube is to be equivalent to the surrounding shell thickness.

1.3.5 Decks

(a) Where a deck is stepped or has a break, suitable brackets are to be provided at the side shell.

(b) Decks passing into superstructures within the 0.5L amidships are to be increased in thickness in way of the break.

1.3.6 Means of Escape

At least two means of escape to the main weather deck are to be provided from the main hull spaces. They are to be as far apart as practicable, and are to be operable from both sides.

1.3.7 Double Bottoms

(a) Passenger crafts that are on international voyages that operate more than four hours at 90% maximum speed from a port of refuge are to be fitted with double bottoms.

(b) Cargo crafts that are on international voyages that are more than eight hours at 90% maximum speed from a port of refuge are to be fitted with double bottoms.

(c) The inner bottoms are to be fitted fore and aft between the peaks or as near thereto as practicable. Where, for special reasons in design, it may be desired to omit the double bottom, the arrangements are to be clearly indicated on the plans when first submitted for approval. A double bottom need not be fitted in way of deep tanks provided the safety of the craft in the event of bottom damage is not thereby impaired. It is recommended that the inner bottom be arranged to protect the bilges as much as possible and that it be extended to the sides of the craft.

1.4 Tank and Compartment Testing

1.4.1 Hydrostatic and other accepted watertight tests for watertight compartments are to be carried out after all work in connection with watertightness are completed, but before the painting, ceiling, cementing, etc., are applied. However, the tests may be conducted after the application of special coatings applied to the internal structure, provided that all welded connections are surveyed prior to application of the coatings and found to be satisfactory by the Surveyor.

1.4.2 Watertight and weathertight boundaries are to be tested in compliance with the provisions given in the Part II Chapter 1 of the Rules for the Construction and Classification of Steel Ships of this Society.

Chapter 2

Structure

2.1 General

2.1.1 This Chapter covers those elements of hull and superstructure which provide longitudinal and other primary and local strength of the craft as a whole and also other important components such as foils and skirts which are directly associated with the hull and superstructure.

2.1.2 Materials used for the hull and superstructure and the other features referred to in 2.1.1 should be adequate for the intended use of the craft.

2.1.3 The structure should be capable of withstanding the static and dynamic loads which can act on the craft under all operating conditions in which the craft is permitted to operate, without such loading resulting in inadmissible deformation and loss of watertightness or interfering with the safe operation of the craft.

2.1.4 Cyclic loads, including those from vibrations which can occur on the craft should not:

(a) impair the integrity of structure during the anticipated service life of the craft or the service life agreed with the Administration;

(b) hinder normal functioning of machinery and equipment; and,

(c) impair the ability of the crew to carry out its duties.

2.1.5 The vibration check shall be performed during the sea trials of the craft. Where deemed necessary, the Society may require vibration measurements to be carried out using suitable instruments; where appropriate, remedial measures may be required to adequately eliminate situations deemed unacceptable.

2.1.6 The Administration should be satisfied that the choice of design conditions, design loads and accepted safety factors corresponds to the intended operating conditions for which certification is sought.

2.1.7 This chapter is applied to the type of mono-hull craft, catamaran, surface-effect ship, air-cushion vehicle and hydrofoil craft etc.

2.1.8 The definitions of the following symbols and terms are applicable for this chapter:

FP = Forward perpendicular
AP = After perpendicular
L = Length (m) of craft, the distance on the summer load line, or if applicable, the design load waterline in the displacement mode, from the fore side of the stem to the centerline of the rudder stock. L is not to be less than 96% and need not be greater than 97% of the length on the summer load line, The forward end of L is to coincide with the foreside of the stem on the waterline on which L is measured
B = Maximum moulded breadth (m)
D = Moulded depth (m), measured from moulded base line to mouldeddeckline at the uppermost continuous deck amidship
d = Moulded draft (m) of full loaded condition
Δ = Moulded displacement (tonne) at d, in seawater (specific weight = 1.025)
C_b = Block coefficient
= $\Delta/(1.025 \cdot L \cdot B_w \cdot d)$
B_w = Maximum waterline beam (m) at the full loaded waterline
V = Maximum speed, in knots
g = Acceleration of gravity, 9.81 m/s^2
LCG = Longitudinal center of gravity of the craft (m)

Chine – for craft without clearly identified chine, the chine is the hull point at which the tangent to the hull is inclined 50° to the horizontal.

Bottom – the lower part of hull between the keel and the chine.

Side – the part of hull between the chine and the main deck.

Main deck – the uppermost complete deck of the hull.

Cross structure – the structure connecting two hulls.

Deadrise angle – for craft without clearly identified deadrise angle, the deadrise angle is to be taken from horizontal line to the line joining the baseline at center and the chine.

2.1.9 The section modulus and moment of inertia of stiffening members are provided by the member and a portion of the plating to which it is attached. For primary supporting members, the effective width of plating is to be equal to the lesser of either one half the sum of spacing on each side of the member, 0.33 times the unsupported span, or 750 mm. For stiffeners, the effective width of plating is to be equal to either one-half the sum of spacing on each side of the member or the width obtained from the following equation, whichever is less.

$w = 60t$

where:

w = Effective width of plating, in mm

t = Thickness of attached plating, in mm

2.1.10 The primary supporting members are members of the beam, girder or stringer type which provide the overall structural integrity of the hull envelope and tank boundaries, e.g. double bottom floors and girders, transverse side structure, deck transverses, bulkhead stringers and vertical webs on longitudinal bulkheads.

2.1.11 The stiffeners is a collective term for secondary supporting structural members.

2.2 Documentation

2.2.1 The following plans are to be submitted for approval:

(a) midship section

(b) construction profile and deck plan

(c) shell expansion

(d) watertight bulkhead

(e) tank structure

(f) engine room construction

(g) after peak construction

(h) fore peak construction

(i) superstructure and deck house

(j) hatchways, hatch covers and shell ports

(k) shaft brackets

(l) flaps or foils

(m) rudder and rudder stock

(n) mooring equipment

(o) operation manual

2.2.2 The following plans are to be submitted for information:

(a) general arrangement

(b) engine room arrangement

(c) capacity plan

(d) hydrostatic curves or tables

2.2.3 Additional documentation as follows are to be submitted for reference.

(a) calculation sheet for longitudinal strength

(b) scantling calculation

(c) information of strength analysis

(d) lines or offset table

2.3 Design Vertical Acceleration

2.3.1 Design vertical acceleration at gravity centre of craft, a_{cg}, is to be determined by the designer according to associated design practice. The design vertical acceleration is the average of the 1/100 highest accelerations in sea conditions expected.

2.3.2 The vertical accelerations a_{cg} of the craft are related to the significant wave height $H_{1/3}$ and craft maximum speed V as follows:

$$a_{cg} = \frac{7.6\tau\times10^{-6}}{d\,C_b}(12H_{1/3} + B_W)(50 - \beta_{cg})\left(\frac{V}{\sqrt{L_W}}\right)^2 \quad g$$

where:

C_b = Block coefficient
d = Draft(m)
$H_{1/3}$ = Significant wave height(m) is the average crest-to-trough height of the highest one third of the zero-upcrossing waves in a specified period.
β_{cg} = Deadrise angle at LCG (degree), 10° to 30° to be taken
τ = Trim angle at speed V (degree), more than 4° to be taken
V = Maximum speed (knots)
B_W = Maximum waterline breadth (m), breadth of one hull for mutihull craft
L_W = Waterline length at draft d (m)
a_{cg} = Average 1/100 highest accelerations of the craft at LCG, expressed in g (where g = 9.81m/s^2)

2.3.3 The relationships between allowable speed and significant wave height shall be stated in the "Craft Operation Manuals" and be shown on navigation bridge by signal board.

2.3.4 The designer is to assume a wave height which may be encountered according to the craft's service restriction as shown in Table II 2-1.

Table II 2-1
Service Restriction

Area of operation	Significant wave height	Factor of service restriction, *F_S
Unrestricted service area	$H_{1/3} > 4.0$m	1.0
Restricted service area	$H_{1/3} \le 4.0$m	0.7
	$H_{1/3} \le 2.0$m	0.5
Smooth sea service area	$H_{1/3} \le 0.5$m	0.3

Notes:
* F_s = Factor of service restriction
$H_{1/3}$ = Significant wave height

2.3.5 The design vertical acceleration at longitudinal position other than LCG is to be in accordance with follows:

$$a_x = k_v \cdot a_{cg}$$

where:
a_{cg} = Vertical acceleration at LCG, as specified in 2.3.2

k_v = Longitudinal distribution factor of vertical acceleration given in Fig. II 2-1

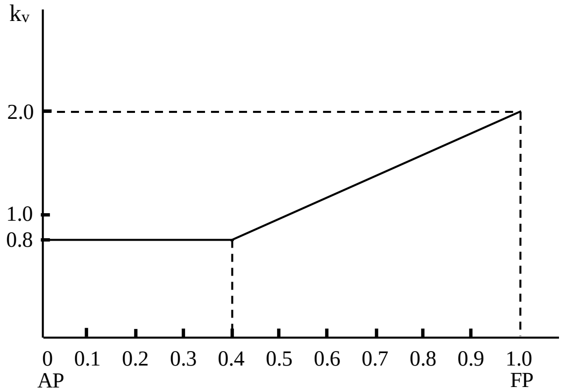

Fig. II 2-1
Acceleration Distribution k_v

2.4 Design Pressure

2.4.1 The design slamming pressure at LCG on bottom of craft is to be taken as:

$$P_{cg} = \frac{100\Delta}{L_W \cdot B}(1 + a_{cg})\, K_a \qquad \text{kN/m}^2$$

where:

a_{cg} = Vertical acceleration at LCG (g)

K_a = Design factor for impact area

$$= 0.62 - 0.47\frac{r^{0.75} - 10}{r^{0.75} + 10}$$

r = $1000\dfrac{A_D}{A_R}$

A_D = Design area (cm^2). For plating it is the shell plate panel but not to take as more than $2S^2$. For longitudinals, transverses, stiffeners and girders, it is the shell area supported by them, but need not be taken less than $0.33l^2$, where S is spacing of longitudinals or stiffeners (cm) and l is length (cm) of unsupported span of internals, see 2.7.6

A_R = Reference area (cm^2)

$$= 7000\frac{\Delta}{d}$$

2.4.2 The design slamming pressure at longitudinal position other than LCG is to be as follows:

$$P_x = P_{cg} \cdot \left(\frac{1 + a_x}{1 + a_{cg}}\right)\left(\frac{70 - \beta_x}{70 - \beta_{cg}}\right) \qquad \text{kN/m}^2$$

where:

a_{cg}, β_{cg} = As specified in 2.3.2

β_x = Deadrise angle at any longitudinal position (degree), 10°~50° to be taken

a_x = Vertical acceleration at any longitudinal position (g), as specified in 2.3.5

P_{cg} = Slamming pressure at LCG as specified in 2.4.1

2.4.3 The pressure acting on weather deck is to be calculated as follows:

$P_d = 0.2L + 7.6$ kN/m^2

2.4.4 The pressure acting on unexposed deck is to be calculated as follows:

$P_d = 0.1L + 6.1$ kN/m^2

2.4.5 The pressure acting on enclosed accommodation decks is to be as follows:

$P_d = 5.0$ kN/m^2

2.4.6 Where the deck is designed to carry deck cargo, the pressure acting on the deck is to be calculated.

$P_d = W (1 + 0.5a_x)$ kN/m^2

where:
W = Deck cargo load (kN)
a_x = Vertical acceleration of the craft at the longitudinal position considered (g)

2.4.7 The pressure acting on the front wall of superstructure and deck house is to be as follows:

$P_h = 24.0$ kN/m^2 to plating and stiffeners

2.4.8 The pressure acting on the side and aft end wall of superstructure and deck house is to be as follows:

$P_h = 13.0$ kN/m^2 to plating
$P_h = 10.0$ kN/m^2 to stiffeners

2.4.9 The pressure acting on the house top, to plating and to stiffeners are to be as follows:

$P_h = 7.0$ kN/m^2

2.4.10 The pressure acting on collision bulkhead and watertight bulkhead is to be calculated as follows:

$P_h = 10h$ kN/m^2

where:
h = Height from considered point up to bulkhead deck at center (m)

2.4.11 The pressure acting on tank boundaries is to be calculated as follows:

$P_h = 10h$ kN/m^2

where:
h = The greatest height from the considered point to the greatest of following:

(a) 2/3 air pipe height from tank top

(b) 2/3 distance to weather deck

(c) 0.01L + 0.15 (m)

(d) 0.46m

2.4.12 The slamming pressure acting on cross structures is to be calculated as follows:

$$P = K_1 K_2 V V_R \left(1 - \frac{G_A}{H_{1/3}}\right) \quad \text{kN/m}^2$$

where:

G_A = Air gap, height of the underside of cross deck above lightest draft waterline (m)
K_1 = Longitudinal distribution factor as shown in Fig. II 2-2
K_2 = Cross deck impact factor
= 0.17 for protected structure
= 0.33 for unprotected structure
V = Maximum speed (knots)
V_R = $\frac{8\,H_{1/3}}{\sqrt{L_W}} + 2$ (knots)
$H_{1/3}$ = As defined in 2.3.2

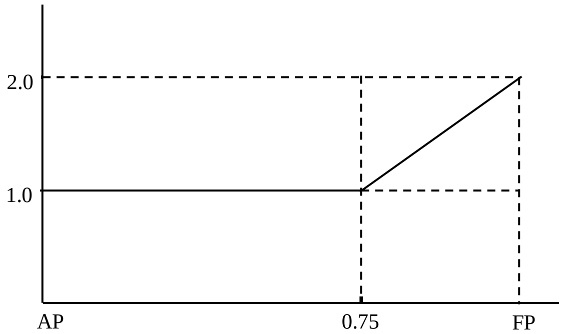

Fig. II 2-2
Longitudinal Distribution Factor K_1

2.4.13 Sea pressure acting on bottom shell is to be calculated as follows:

$$P_h = 10(F_s H + d) \quad \text{kN/m}^2$$

where:

F_s = Factor of service restriction as shown on Table II 2-1
H = Wave parameter
= 0.0172L + 3.653 (m)
d = Moulded draft (m)

2.4.14 Sea pressure acting on side shell is to be calculated as follows:

$P_h = 10(F_sH + h)$ kN/m^2

where:

F_s, H = As specified in 2.4.13

h = Height from the load point to the design waterline for load point below the design waterline, and 0 for load point on or above the design waterline (m)

2.5 Hull Girder Strength

2.5.1 For crafts having L > 50 m or with L/D > 12, the longitudinal strength of hull girder in high speed navigation condition, specified in 2.5.4, and in displacement condition, specified in 2.5.5, are to be checked.

2.5.2 For twin-hull craft and SES, the transverse strength, specified in 2.5.7, and torsional strength of cross-deck, specified in 2.5.8, are also to be checked.

2.5.3 For hydrofoil craft, the calculation of longitudinal strength is to be effected for the most severe condition from hull-borne, take-off mode to foil-borne.

2.5.4 The longitudinal bending moments in high speed navigation condition are to be assumed as follow:

$M_{BH} = M_{BS} = 0.55\Delta L(1+a_{cg})$ kN-m

where:

M_{BH} = Longitudinal hogging bending moment

M_{BS} = Longitudinal sagging bending moment

2.5.5 The longitudinal bending moments in displacement condition are to be assumed as follows:

$M_{BH} = M_{SW} + 0.21C_WL^2BC_bC_R$ kN-m

$M_{BS} = M_{SW} + 0.12C_WL^2B(C_b + 0.7)C_R$ kN-m

where:

L, B, C_b are as defined in 2.1.8

M_{SW} = Still water bending moment in most severe loading condition (kN-m)

C_W = 0.02L+6

C_R = 1.0 for unrestricted service

= 0.9 for $H_{1/3} \le 4.0$ m service restriction

= 0.75 for $H_{1/3} \le 2.0$ m service restriction

= 0.6 for $H_{1/3} \le 0.5$ m service restriction

2.5.6 The shear force is to be assumed as follows:

$$T_B = \frac{3.2M_B}{L} \quad \text{kN}$$

where:

M_B = The greater of M_{BH} and M_{BS} as specified in 2.5.4 and 2.5.5 as applicable
L = As defined in 2.1.8

2.5.7 The transverse bending moment for twin-hull craft of L＜50 m is to be assumed as follows:

$$M_B = \frac{\Delta b \cdot a_{cg}}{5} \cdot g \qquad \text{kN-m}$$

where:
b = Transverse distance, in m, between the center line of two hulls
a_{cg} = As defined in 2.3.2
g, Δ = As defined in 2.1.8

2.5.8 The transverse bending moment for twin-hull craft of L ≥ 50 m is to be assumed as the greater of follows:

$$M_B = M_S (1 + a_{cg}) \qquad \text{kN-m}$$
$$M_B = M_S + F_y (z - 0.75d) \qquad \text{kN-m}$$

where:
M_S = Still water transverse bending moment (kN-m)
F_y = Horizontal split force on immersed hull

$$= \frac{7L^2}{(L/d)^{1.5}}\left(1+\frac{V}{10\sqrt{L}}\right)\left(1.6-\frac{6}{\sqrt{L}}\right)\left(53-\frac{2L}{B_W}\right) \qquad \text{kN}$$

z = Height from base line to neutral axis of cross structure (m)
d = Full load draft (m)
$\frac{V}{\sqrt{L}}$ needs not be taken greater than 3.0

2.5.9 The vertical shear force in centerline between twin-hull is to be assumed as follows:

$$T_B = \frac{\Delta\, a_{cg}}{4} \cdot g \qquad \text{kN}$$

2.5.10 The twin-hull pitch connection moment may be assumed as follows:

$$M_p = \frac{\Delta L \cdot a_{cg}}{8} \cdot g \qquad \text{kN-m}$$

2.5.11 The twin-hull torsional moment, along the longitudinal axis is to be assumed as follows:

$$Mt = \frac{\Delta b \cdot a_{cg}}{4} \cdot g \qquad \text{kN-m}$$

where:
b = Transverse distance between the center line of the two hulls (m)

2.5.12 The required section modulus, respectively at bottom and main deck are to be calculated as follows:

$$SM = \frac{M}{17.5} \cdot Q \cdot 0.9 \times 10^2 \qquad cm^3$$

where:

M = Longitudinal bending moment assumed in 2.5.4 or 2.5.5

Q = $0.9 + 115/\sigma_y$, but not less than $635/(\sigma_y+\sigma_u)$

σ_y = Minimum yield stress of welded aluminum(N/mm^2)

σ_u = Minimum ultimate strength of welded aluminum (N/mm^2)

2.6 Direct Calculations

2.6.1 General

(a) Direct calculations generally require to be carried out, in the opinion of the Society, to check primary structures for craft of length L > 90 m or speed V > 45 knots.

(b) In addition, direct calculations are to be carried out to check scantlings of primary structures of craft whenever, in the opinion of the Society, hull shapes and structural dimensions are such that scantling formulas in 2.7 are no longer deemed to be effective.

2.6.2 Loads

(a) In general, the loading conditions specified in 2.6.3 are to be considered.

(b) The slamming pressure is to be calculated as stipulated in 2.4.

(c) In three-dimensional analyses, special attention is to be paid to the distribution of weights and buoyancy and to the dynamic equilibrium of the craft.

(d) In the case of three-dimensional analysis, the longitudinal distribution of impact pressure is to be considered individually, in the opinion of the Society. In general, the impact pressure is to be considered as acting separately on each transverse section of the model, the remaining sections being subject to the hydrostatic pressure.

2.6.3 Loading conditions

(a) Still-water condition:

(i) Forces caused by weights which are expected to be carried in the full load condition. The weight is distributed according to the weight booklet of the craft.

(ii) Outer hydrostatic load in still water.

(b) Vertical acceleration condition:

(i) Forces caused by weights which are expected to be carried in the full load condition. The weight is distributed according to the weight booklet of the craft.

(ii) Forces of inertia due to the vertical acceleration a_x of the craft, considered in a downward direction.

(c) Slamming pressure condition: The slamming pressure on the bottom and the side shell of the craft is to be considered.

(d) Horizontal split force condition for catamaran: The horizontal split force on immersed hull of catamaran is to be considered.

(e) Pitch connecting moment condition for catamaran: The pitch connecting moment on the cross deck of a catamaran is to be considered.

2.6.4 Structural model

(a) In general, the extent of the model is to be such as to allow analysis of the behavior of the main structural elements and their mutual effects.

(b) In general, primary structures of crafts are to be modeled with finite element adopting a medium size mesh. In the opinion of the Society, detailed analysis with finite mesh are required for areas where stresses, calculated with medium-mesh, exceed allowable limits and the type of structure gives reason to suspect the presence of high stress concentrations.

2.6.5 Boundary conditions

The boundary conditions depend on the model extent and the loading conditions considered.

2.6.6 Checking criteria

The stresses given by the above calculations are not to be greater than the following allowable values (N/mm^2):

(a) normal stress: $\sigma = \dfrac{0.65\ \sigma_y}{K_m K_s}$

(b) shear stress: $\tau = \dfrac{0.35\ \sigma_y}{K_m K_s}$

(c) Von Mises equivalent bending stress: $\sigma_{all} = \dfrac{0.80\ \sigma_y}{K_m K_s}$

where:

σ_y = Yielding strength of material (N/mm^2)

K_m = 2.15

K_s = Safety coefficient, to be assumed:

= 1.00 for combined loading conditions

= 1.25 for loading condition in still water

2.7 Structural Scantlings

This section stipulates requirements for the scantlings of hull structures (plating, stiffeners, primary supporting members). The loads acting on such structures are to be calculated in accordance with the provisions of 2.4.

2.7.1 The thickness of the shell, deck or bulkhead plating is not to be less than obtained from 2.7.2 to 2.7.4, whichever is greater.

2.7.2 Thickness due to lateral loads

$$t = s\,k_1 \sqrt{\frac{P\,k_2}{1000 \cdot \sigma_a}} \qquad \text{mm}$$

where:

s = Shorter edge of plate panel (mm)

k_1 = Correction factor for curved panels, given in Table II 2-2

h = As shown in Fig II 2-3 (mm)

P = Design pressure given in 2.4 (kN/m^2)

k_2 = $\dfrac{0.5}{1 + 0.623(\frac{s}{l})^6}$

σ_a = Allowable stress given in Table II 2-3 (N/mm^2)

l = Longer edge of plate panel (mm)

Table II 2-2
Correction Factor for Curved Panel

h/s	k_1
0 ~ 0.03	1.0
0.03 ~ 0.1	1.1－3·h/s
≥ 0.1	0.8

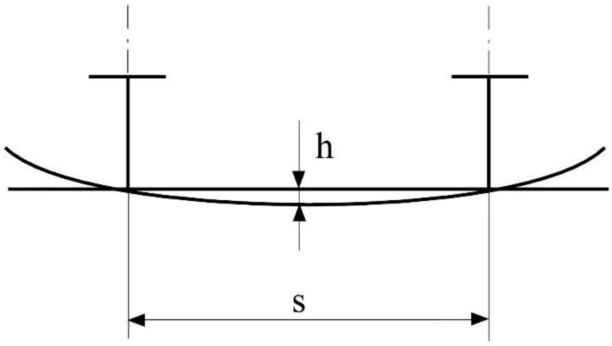

Fig. II 2-3
Curvature of a Curved Panel

Table II 2-3
Allowable Stress of Plating

Structural Members	Allowable stress, σ_a
Bottom and side shell plating－slamming pressure	0.90 σ_y
Bottom and side shell plating－sea pressure	0.55 σ_y
Deck plating－strength deck	0.60 σ_y
Deck plating－lower deck	0.60 σ_y
Bulkheads－tank boundary	0.60 σ_y
Bulkheads－watertight	0.95 σ_y
Superstructures and deck houses－front, sides, ends, tops	0.60 σ_y

Note: σ_y=Yielding strength (N/mm^2) of welded aluminum

2.7.3 Buckling strength of plate panel
These requirements apply to plate panels subject to compressive load.

(a) Elastic buckling stress

$$\sigma_E = 0.9\,m\,E\left(\frac{t}{s}\right)^2 \qquad \text{N/mm}^2$$

where:

σ_E = Elastic buckling stress (N/mm^2)

m = 4.0 for longitudinally framed plate panel

$= C\left[1+\left(\frac{s}{l}\right)^2\right]^2$ for transversely framed plate panel

E = 6.90 x 10^4 N/mm^2

t = Plate thickness (mm)

s = Shorter edge of plate panel (mm)

l = Longer edge of plate panel (mm)

C = 1.21where stiffeners are T or angle sections

= 1.10where stiffeners are bulb plates

= 1.05where stiffeners are flat bars

(b) Critical buckling stress

$\sigma_c = \sigma_E$ when $\sigma_E \leq 0.5\sigma_y$

$\sigma_c = \sigma_y\left(1-\frac{\sigma_y}{4\sigma_E}\right)$ when $\sigma_E > 0.5\sigma_y$

where:

σ_E = Elastic buckling stress calculated in2.7.3(a) (N/mm^2)

σ_y = Yielding strength (N/mm^2) of welded aluminum

(c) Calculated compressive stress

$$\sigma_W = \frac{M\,y}{I}\times 10^5 \qquad \text{N/mm}^2$$

where:

σ_W = Working compressive stress in panel being considered (N/mm^2)

M = Total bending moment given in 2.5 (kN-m)

y = Vertical distance from the neutral axis to the considered location (m)

I = Moment of inertia of the hull girder (cm^4)

(d) Buckling strength criteria

$\sigma_c \geq \sigma_W$

2.7.4 Minimum thickness of plate

(a) The thickness of shell, decks, and bulkheads is not to be less than those obtained from Table II 2-4.

(b) The minimum thickness of the shell plating is to be increased 50% in way of skegs, shaft struts, hawse pipes etc. Bow thruster tube thickness is to be equivalent to the surrounding shell thickness.

Table II 2-4
Minimum Thickness of Plate

Structural members	Minimum thickness
Bottom shell	$0.7\sqrt{Lq_a}+1.0$ (minimum 4.0 mm)
Side shell	$0.62\sqrt{Lq_a}+1.0$ (minimum 3.5 mm)
Strength deck	$0.62\sqrt{Lq_a}+1.0$ (minimum 3.5 mm)
Lower decks, W.T. bulkheads, deep tank bulkheads	$0.52\sqrt{Lq_a}+1.0$ (minimum 3.5 mm)

Note:
q_a = $115/\sigma_{ya}$
σ_{ya} = Minimum unwelded yield strength for aluminum alloys, in N/mm^2, but not to be taken as more than 0.7 of the ultimate tensile strength in the as-welded condition

2.7.5 Stiffeners and primary supporting members

The scantling of each longitudinal, stiffener, transverse web, stringer and girder is not to be less than obtained from 2.7.6 to 2.7.8.

2.7.6 Section modulus

The ends of members are to be effectively attached to the supporting structures. The section modulus of each longitudinal, stiffener, transverse web, stringer and girder is not to be less than given by the following equation:

$$SM=\frac{P s l^2}{12\sigma_a}\times 10^3 \qquad cm^3$$

where:
P = Design pressure given in 2.4 (kN/m^2)
s = Spacing of longitudinal, stiffener, transverse web or girder (m)
l = Span of longitudinal, stiffener, transverse in web or girder between supports, where bracket end connections are supported by bulkheads, l may be measured onto the bracket (m)
σ_a = Allowable stress given in Table II 2-5 (N/mm^2)

Table II 2-5
Allowable Stress of Stiffeners and Primary Supporting Members

Structural Members	Allowable stress, σ_a
Bottom longitudinals	0.50 σ_y
Side longitudinals	0.50 σ_y
Deck longitudinals－strength deck	0.33 σ_y
Deck longitudinals– other deck	0.40 σ_y
Bottom transverses	0.60 σ_y
Side transverses	0.60 σ_y
Deck transverses – strength decks	0.75 σ_y
Deck transverses – other decks	0.75 σ_y
Watertight bulkheads	0.85 σ_y
Deep tank bulkheads	0.60 σ_y
Superstructure and deckhouse	0.70 σ_y

Note: σ_y=Yielding strength (N/mm^2) of welded aluminum

2.7.7 Buckling strength of stiffener

These requirements apply to longitudinals, stiffeners, transverse webs, stringers and girders subject to compressive load.

(a) Elastic buckling stress

$$\sigma_E = \frac{E\,I_a}{A\,l^2} \times 10^{-3} \qquad \text{N/mm}^2$$

where:

σ_E = Elastic buckling stress (N/mm^2)

E = 6.90 x 10^4 N/mm^2

I_a = Moment of inertia of the member being considered together with attached plating (cm^4)

A = Cross-sectional area of the member being considered together with attached plating (cm^2)

l = Span of longitudinal (m)

(b) Critical buckling stress

$$\sigma_c = \sigma_E \qquad \text{when } \sigma_E \leq 0.5\sigma_y$$

$$\sigma_c = \sigma_y\left(1 - \frac{\sigma_y}{4\,\sigma_E}\right) \qquad \text{when } \sigma_E > 0.5\sigma_y$$

where:

σ_E = Elastic buckling stress calculated in2.7.7(a) (N/mm^2)

σ_y = Yielding stress (N/mm^2)of welded aluminum

(c) Calculated compressive stress

$$\sigma_W = \frac{M\,y}{I} \times 10^5 \qquad \text{N/mm}^2$$

where:

σ_W = Working compressive stress in panel being considered (N/mm^2)

M = Total bending moment given in 2.5 (KN-m)

y = Vertical distance from the neutral axis to the considered location (m)

I = Moment of inertia of the hull girder (cm^4)

(d) Buckling strength criteria

$$\sigma_c \geq \sigma_W$$

2.7.8 Minimum thickness of stiffener

The thickness of webs and face plates are not to be less than obtained from the following equations:

(a) Webs

$$t = \frac{d_W}{C}\sqrt{\frac{\sigma_y}{\sigma_d}} \quad \text{mm}$$

where:

t = Required minimum thickness (mm)

d_W = Depth of the web (mm)

C = 35

σ_y = Yielding strength (N/mm^2)of welded aluminum

σ_d = 127.6 N/mm^2

(b) Face bars

$$t = \frac{b_f}{C}\sqrt{\frac{\sigma_y}{\sigma_d}} \quad \text{mm}$$

where:

σ_y, σ_d = As defined in 2.7.8(a)

t = Required minimum thickness (mm)

b_f = Outstanding width of the face bar (mm), see Fig. II 2-4

C = 9

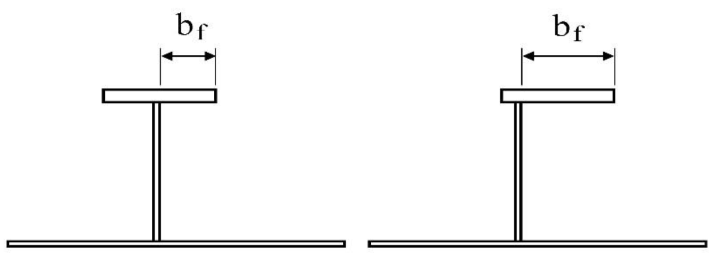

Fig. II 2-4
Definition of b_f

2.8 Rudder

2.8.1 All the requirements concerning about the material, loads, scantlings of the rudders are to comply with the requirements in Part II Chapter 24 of the Rules for the Construction and Classification of Steel Ships. Rudders of special types will be individually considered by the Society.

2.8.2 Rudders which are intended to be operated at the maximum angle of helm during high-speed navigation are to be designed on the basis of direct calculations from the designer. The acceptability of calculated results will be individually considered by the Society in each separate case.

Chapter 3

Keels, Stems and Shaft Struts

3.1 Keels

3.1.1 Bar keels

(a) Where bar keels are fitted the required section modulus (Z) and moment of inertia (I),about the minor axis, are not to be less than those obtained from the following equations.

$$Z = Q\cdot(0.244L^3+38.3L^2+1815L+23000) \qquad mm^3$$

$$I = \frac{2.06\times10^5}{E}\cdot(0.162L^4+36.5L^3+2950L^2+97700L+1040000) \qquad mm^4$$

where:

Q = 0.9 + 115/σ_y, but not less than635/(σ_y+σ_u)

σ_y = Minimum yield stress of welded parent material (N/mm^2)

σ_u = Minimum ultimate strength of welded aluminum (N/mm^2)

L = Ship length (m), as defined in 2.1.8

E = Young's modulus of the aluminum alloy being considered (N/mm^2)

(b) The ratio of depth to thickness is not to be more than 4.5.

3.1.2 Plate keels

The thickness of the plate keel throughout the length of the craft is to be not less than the bottom shell required in 2.7.

3.2 Stems

3.2.1 Bar stems

(a) Where bar stems are fitted the required section modulus (Z) and moment of inertia (I), about the minor axis, are not to be less than those obtained from the following equations.

$$Z = Q\cdot(0.179L^3+27.6L^2+1190L+9430) \qquad mm^3$$

$$I = \frac{2.06\times10^5}{E}\cdot(0.102\,L^4+23\,L^3+1810\,L^2+54100\,L+386000\,) \quad mm^4$$

where:

Q, L and E =As defined in 3.1.1(a)

(b) This thickness and width is to be maintained between the keel and design load waterline. Above the designed load waterline they may be gradually reduced until the section area at the head is 70% of that obtained from the equations.

(c) The ratio of width to thickness is not to be more than 5.5. The thickness of the bar stem in general should also not be less than twice the shell thickness.

3.2.2 Plate stems

Where plate stems are used, they are not to be less in thickness than the bottom shell plating required in 2.7, where s is the frame spacing, or 610 mm if greater. Plate stems are to be suitably stiffened.

3.3 Stern Frames

Craft that are fitted with stern frames, shoe pieces, rudder horns, and rudder gudgeons are to meet the applicable requirements in Part II Chapter 2 of the Rules for the Construction and Classification of Steel Ships.

3.4 Shaft Struts

3.4.1 General

Tail-shaft (propeller-shaft) struts where provided may be of the V or I type. The thickness of the strut barrelor boss is to be at least 1/5 the diameter of the tail shaft. The length of the strut barrel or boss is to be adequate to accommodate the required length of propeller-end bearings. The following equations are for solid struts having streamline cross-sectional shapes. For hollow section and non-streamlined struts, the equivalent cross sectional area, inertia, and section modulus (about the major axis) are not to be less than required by 3.4.2 and 3.4.3.

3.4.2 V Strut

(a) The required section modulus (Z) and moment of inertia (I) of each streamlined section strut arm, about the minor axis, are not to be less than those obtained from the following equations.

$Z = 0.0266d^3 \cdot Q$ mm^3
$I = 0.00442d^4 \cdot Q$ mm^4

where:

d = Required diameterof tail shaft(mm), see Part III

Q= As defined in 3.1.1(a)

(b) Where the included angle is less than 45 degrees, the foregoing scantlings are to be specially considered.

3.4.3 I Strut

The required section modulus (Z) and moment of inertia (I)of each streamlined section strut arm, about the minor axis, are not to be less than those obtained from the following equations.

$Z = 0.0752d^3 \cdot Q$ mm^3
$I = 0.0176d^4 \cdot Q$ mm^4

where:

d = Required diameterof tail shaft(mm), see Part III

Q = As defined in 3.1.1(a)

3.4.4 The length of the longer leg of a V strut or the leg of an I strut, measured from the outside perimeter of the strut barrel or boss to the outside of the shell plating, is not to exceed 10.6 times the diameter of the tail shaft. Where this length is exceeded, the width and thickness of the strut are to be increased, and the strut design will be given special consideration. Where strut length is less than 10.6 x required tail shaft diameter, the section modulus of the strut may be reduced in proportion to the reduced length, provided the section modulus is not less than 0.85 x required section modulus.

Chapter 4
Equipment

4.1 General

4.1.1 A primary assumption made in this chapter is that aluminum high speed crafts will only need an anchor for emergency purposes.

4.1.2 The arrangements for anchoring, towing and berthing and the local craft structure, the design of the anchor, towing and berthing arrangements and the local craft structure shall be such that risks to persons carrying out anchoring, towing or berthing procedures are kept to a minimum.

4.1.3 All anchoring equipment, towing bitts, mooring bollards, fairleads, cleats and eyebolts shall be so constructed and attached to the hull that in use up to design loads, the watertight integrity of the craft will not be impaired. Design loads and any directional limitations assumed shall be listed in the craft operating manual.

4.1.4 Under any operating load up to the breaking strength of the anchor cable or mooring lines, the loads on the bitts, bollards, etc., shall not result in damage to the hull structure that will impair its watertight integrity. A strength margin of at least 20% above the resultant load based on the minimum specified breaking strength of the relevant cable or warp shall be required.

4.1.5 Only anchoring equipment is considered for the purpose of classification.

4.1.6 Documents to be submitted

A detailed drawing, showing all the elements necessary for the evaluation of the equipment number of the craft, is to be submitted together with the calculations of the EN number. The anchoring equipment to be fitted on the concerned craft is to be specified.

4.2 Anchoring

4.2.1 Aluminum high speed crafts shall be provided with at least one anchor with its associated cable or cable and warp and means of recovery. Every craft shall be provided with adequate and safe means for releasing the anchor, its cable and warp.

4.2.2 Good engineering practice shall be followed in the design of any enclosed space containing the anchor recovery equipment to ensure that persons using the equipment are not put at risk. Particular care shall be taken with the means of access to such spaces, the walkways, the illumination and protection from the cable and the recovery machinery.

4.2.3 Adequate arrangements shall be provided for two-way voice communication between the operating compartment and persons engaged in dropping, weighing or releasing the anchor.

4.2.4 The anchoring arrangements shall be such that any surfaces against which the cable may chafe (for example, hawse pipes and hull obstructions) are designed to prevent the cable from being damaged and fouled. Adequate arrangements shall be provided to secure the anchor under all operational conditions.

4.2.5 The craft shall be protected so as to minimize the possibility of the anchor and cable damaging the structure during normal operation.

4.3 Towing

4.3.1 Adequate arrangements shall be provided to enable the craft to be towed in the worst intended conditions. Where towage is to be from more than one point a suitable bridle shall be provided.

4.3.2 The towing arrangements shall be such that any surfaces against which the towing cable may chafe (for example, fairleads), is of sufficient radius to prevent the cable being damaged when under load.

4.3.3 The maximum permissible speed at which the craft may be towed shall be included in the operating manual.

4.4 Berthing

4.4.1 Where necessary, suitable fairleads, bitts and mooring ropes shall be provided.

4.4.2 Adequate storage space for mooring lines shall be provided such that they are readily available and secured against the high relative wind speeds and accelerations which may be experienced.

4.5 Equipment

4.5.1 General

(a) The anchoring equipment required in 4.5.2 is intended for temporary, occasional mooring of a craft within a harbour or sheltered area when the craft is awaiting berth, tide, etc.

(b) The equipment is therefore not designed to hold a craft off fully exposed coasts in rough weather or to stop a craft which is moving or drifting. In this condition the loads on the anchoring equipment increase to such a degree that its components may be damaged or lost owing to the high energy forces generated, particularly in large craft.

(c) For craft where frequent anchoring in open sea is expected, the owner's and shipyard's attention is drawn to the fact that anchoring equipment should be provided in excess of the requirements of these Rules.

(d) The anchoring equipment required in 4.5.2 is designed to hold a ship in good holding ground in conditions such as to avoid dragging of the anchor. In poor holding ground the holding power of the anchors will be significantly reduced.

(e) For small craft, with a length $L \leq 25$ m, some partial exemption from these Rules may be accepted especially for what concerns anchor operation; in particular, where proper and safe anchor operation is assured, hand-operated machinery and/or absence of hawse pipe may be accepted.

4.5.2 Equipment number

(a) General

(i) The equipment is in general to be in accordance with the requirements given in Table II 4-1, reduced as per Table II 4-2 in accordance with Service Restriction Notation.

(ii) When two bow anchors are fitted, the mass of each anchor, the diameter and the length of each chain cable are to comply with the requirements of the above-mentioned table.

(iii) The equipment number EN is to be calculated as follows:

$$EN = \Delta^{2/3} + 2BH + 0.1A$$

where:

Δ = The maximum displacement (tonne)

H = Effective height (m) from the summer load waterline to the top of the uppermost deckhouse, to be measured as follows:

$H = a + \Sigma h_i \sin\theta_i$

a = The distance (m) from summer load water line amidships to the upper deck at side

h_i = The height (m) on the centerline of each tier of deck houses having an actual breadth greater than B/4 where B is the breadth (m) as defined in 2.1.8 of Chapter 2. Sheer and trim are to be ignored.

θ_i = The angle of inclination aft of each front bulkhead as shown on Fig. II 4-1.

A = The area (m^2) in profile view of the hull superstructures and deck houses above the summer load waterline which is within the rule length of the craft defined in 2.1.8 of Chapter 2 and with a breadth greater than B/4.

(iv) If a deck house broader than B/4 is placed on top of another deck house equal to or less than B/4 in breadth, only the widest is to be considered and the narrowest may be ignored.

(v) Windscreens or bulwarks and hatch coamings more than 1.5 m in height above the deck at side are to be regarded as parts of superstructures and houses when determining H and A.

(vi) In the calculation of A, when a bulwark is more than 1.5 m in height, the crosshatched area of Fig. II 4-1 is to be considered.

(vii) For catamarans, the cross-sectional area of the tunnel above the waterline may be deducted from BH in the formula.

4.5.3 Anchors

(a) Mass of anchors

(i) Table II 4-1 indicates the mass of a "high holding power anchor" (HHP) i.e. anchor having a holding power greater than that of an ordinary anchor.

(ii) "Super high holding power anchors" (SHHP), i.e. anchors having a holding power equal to, at least, four times that of an ordinary anchor, may be usedas indicated in Table II 4-1.

(iii) The actual mass of each anchor may vary within (+7, -7) percent of the value shown in the table, provided that the total mass of anchors is not less than would have been required for anchors of equal mass.

(iv) Normally HHP or SHHP anchors are to be used. Possible use of ordinary anchors would be specially considered by the Society.

(b) Anchor design

(i) Anchors are to have appropriate shape and scantlings in compliance with Society requirements and are to be constructed in compliance with Society requirements.

(ii) A high or super high holding power anchor is to be suitable for use on board without any prior adjustment or special placement on the ground.

(iii) For approval and/or acceptance as a high or super high holding power anchor, the anchor is to have a holding power equal, respectively, to at least twice or four times that of an ordinary stockless anchor of the same mass.

(iv) Comparative tests on ordinary stockless anchors are to be carried out at sea and must provide satisfactory results on various types of seabeds.

(v) Alternatively sea trials by comparison with a previously approved HHP anchor may be accepted as a basis for approval.

(vi) Such tests are to be carried out on anchors whose masses are, as far as possible, representative of the full range of sizes proposed for the approval.

(vii) At least two anchors of different sizes are to be tested. The mass of the greatest anchor to be approved is not to be in excess of 10 times that of the maximum size tested and the mass of the smallest is to be not less than 0.1 times that of the minimum size tested.

(viii) Tests are normally to be carried out by means of a tug, but, alternatively, shore-based tests may be accepted.

(ix) The length of the chain cable connected to the tested anchor, having a diameter appropriate to its mass, is to be such that the pull acting on the shank remains practically horizontal. For this purpose a scope of chain cable equal to 10 is deemed normal; however lower values may be accepted.

(x) Three tests are to be carried out for each anchor and type of ground.

(xi) The pull is to be measured by means of a dynamometer; measurements based on the bollard pull against propeller's revolutions per minute curve may be accepted instead of dynamometer readings.

(xii) Anchor stability and its ease of dragging are to be noted down, whenever possible.

(xiii) Upon satisfactory outcome of the above tests, the Society will issue a certificate declaring the compliance of "high or super high holding power" anchors with its relevant Rules.

4.5.4 Chain cables

(a) Bow anchors are to be used in connection with stud link chain cables whose scantlings and steel grades are to be in accordance with the requirements of the Society.

(b) Normally Grade 2 or Grade 3 stud link chain cables are to be used with HHP anchors. In case of SHHP anchors Grade 3 chain cables are to be used.

(c) Proposal for use of Grade 1 chain cables connected to ordinary anchors will be specially considered by the Society.

(d) For craft with an Equipment Number EN≤205 studless short link chain cables may be used provided that:

(i) steel grade of the studless chain is to be equivalent to the steel grade of the stud chains it replaces, i.e. referring to ISO standard 1834:

- Class M (4) [grade 400] in lieu of grade 2,
- Class P (5) [grade 500] in lieu of grade 3.

(ii) equivalence in strength is to be based on breaking load.

(iii) the studless chain cable meets the requirements of the Society.

(e) The proof loads PL and breaking loads BL, in kN, required for the studless link chain cables are given by the following formulae, where d, in mm, is the required diameter of Grade 2 and Grade 3 stud chain cables taken from Table II 4-1:

Grade 2: $PL_2 = 9.807\, d^2 (44 - 0.08\, d)\, 10^{-3}$

$BL_2 = 13.73 d^2 (44 - 0.08\, d)\, 10^{-3}$

Grade 3: $PL_3 = 13.73\, d^2 (44 - 0.08\, d)\, 10^{-3}$

$BL_3 = 19.61\, d^2 (44 - 0.08\, d)\, 10^{-3}$

(f) The method of manufacture of chain cables and the characteristics of the steel used are to be approved by the Society for each manufacturer. The material from which chain cables are manufactured and the completed chain cables themselves are to be tested in accordance with the appropriate requirements.

(g) Chain cables are to be made of unit lengths (shots) of 27.5 m minimum joined together by Dee or lugless shackles.

4.5.5 Steel wire ropes for anchors

(a) Steel wire ropes may be used as an alternative to stud link chain cables required in Table II 4-1 when EN≤500, provided that the following requirements are complied with.

(b) The length L_{swr} of steel wire rope is to be not less than

$L_{swr} = L_{ch}$ when $EN \leq 130$

$L_{swr} = L_{ch} (EN + 850) / 900$ when $130 < EN \leq 500$

where L_{ch} is the length of stud link chain cable required by the Table II 4-1.

(c) The effective breaking load of the steel wire rope is to be not less than the required breaking load of the chain cable it replaces.

(d) A short length of chain cable having scantlings complying with 4.5.4 is to be fitted between the steel wire rope and the bow anchor. The length of this chain part is to be not less than 12.5 m or the distance from the anchor in its stowed position to the windlass, whichever is the lesser.

4.5.6 Synthetic fibre ropes for anchors

(a) Synthetic fibre ropes may be used as an alternative to stud link chain cables required in Table II 4-1 when EN≤130, provided that the following requirements are complied with.

(b) Fibre ropes are to be made of polyamide or other equivalent synthetic fibres, excluding polypropylene.

(c) The length L_{sfr} of the synthetic fibre rope is to be not less than

$L_{sfr} = L_{ch}$ when $EN \leq 60$

$L_{sfr} = L_{ch} (EN + 170) / 200$ when $60 < EN \leq 130$

where L_{ch} is the length of stud link chain cable required by the Table II 4-1.

(d) The effective breaking load P_s in kN, of the synthetic fibre rope is to be not less than the following value:

$$P_s = 2.2\ BL^{8/9}$$

where BL, in kN, is the required breaking load of the chain cable replaced by the synthetic fibre rope (BL can be determined by the formulae given in 4.5.5).

(e) A short length of chain cable complying with 4.5.5 is to be fitted between the synthetic fibre rope and the bow anchor.

4.5.7 Attachment pieces

Both attachment pieces and connection fittings for chain cables are to be designed and constructed in such a way as to offer the same strength as the chain cable and are to be tested in accordance with the appropriate requirements.

4.5.8 Arrangement of anchors and chain cables

(a) The bow anchors, connected to their own chain cables, are to be so stowed as to always be ready for use.

(b) Hawse pipes are to be of a suitable size and so arranged as to create, as far as possible, an easy lead for the chain cables and efficient housing for the anchors.

(c) For this purpose chafing lips of suitable form with ample lay-up and radius adequate for the size of the chain cable are to be provided at the shell and deck. The shell plating at the hawse pipes is to be reinforced as necessary.

4.5.9 Chain stopper

(a) A chain stopper is normally to be fitted between the windlass and the hawse pipe in order to relieve the windlass of the pull of the chain cable when the ship is at anchor.

(b) A chain stopper is to be capable of withstanding a pull of 80% of the breaking load of the chain cable; the deck at the chain stopper is to be suitably reinforced. However, fitting of a chain stopper is not compulsory.

(c) Chain tensioners or lashing devices supporting the weight of the anchor when housed in the anchor pocket are not to be considered as chain stoppers.

(d) Where the windlass is at a distance from the hawse pipe and no chain stopper is fitted, suitable arrangements are to be provided to lead the chain cable to the windlass.

4.5.10 Chain locker

(a) The chain locker is to be of a capacity adequate to stow all chain cable equipment and provide an easy direct lead to the windlass.

(b) Where two anchor lines are fitted, the port and starboard chain cables are to be separated by a steel bulkhead in the locker.

(c) The inboard ends of chain cables are to be secured to the structure by a fastening able to withstand a force not less than 15% nor more than 30% of the breaking load of the chain cable.

(d) In an emergency, the attachments are to be easily released from outside the chain locker.

(e) Where the chain locker is arranged aft of the collision bulkhead, its boundary bulkheads are to be watertight and a drainage system provided.

4.5.11 Anchoring sea trails

(a) The anchoring sea trials are to be carried out on board in the presence of a Society surveyor.

(b) The test is to demonstrate that the windlass complies with the requirements given in 4.5.9(e).

(c) The brake is to be tested during lowering operations.

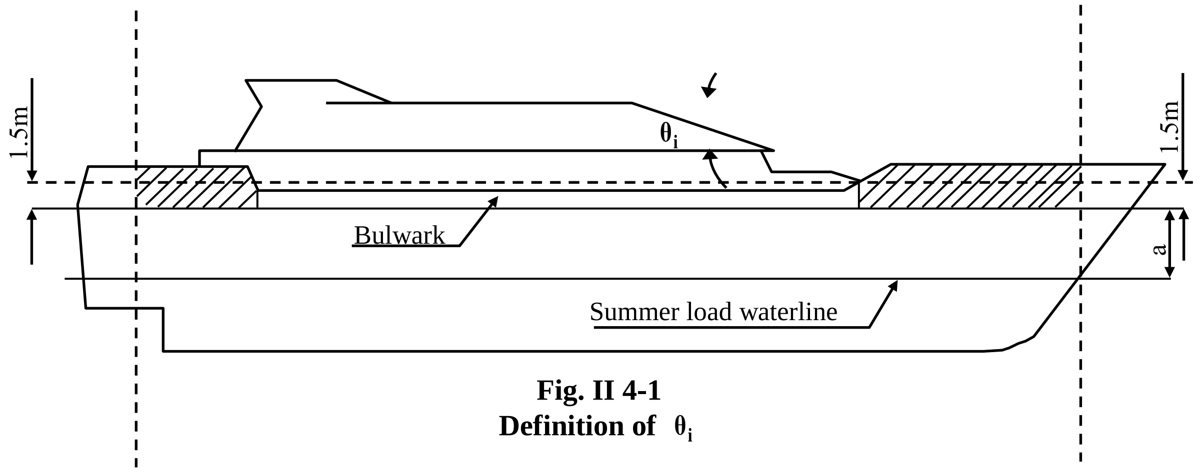

Fig. II 4-1
Definition of θ_i

Table II 4-1
Equipment

Equipment No.		Number	Anchors		Stud-link chain cables				Mooring lines	
			Mass per anchor		Total length (m)	Diameter and steel grade			Steel or fibre ropes	
			HHP	SHHP		Grade 1	Grade 2	Grade 3	Minimum number and length	Minimum breaking strength
Over	Up to		(kg)	(kg)		(mm)	(mm)	(mm)	(m)	(kN)
30	39	1	93	62	115	12.5			2 X 40	32
40	49	1	119	79	115	12.5			2 X 40	32
50	59	1	146	97	130	14	12.5		3 X 40	34
60	69	1	171	114	130	14	12.5		3 X 40	34
70	79	1	198	138	130	16	14		3 X 50	37
80	89	1	224	149	130	16	14		3 X 50	37
90	99	1	251	167	150	17.5	16		3 X 55	39
100	109	1	276	184	150	17.5	16		3 X 55	39
110	119	1	303	202	150	19	17.5		3 X 55	44
120	129	1	329	219	150	19	17.5		3 X 55	44
130	139	1	356	237	165	20.5	17.5		3 X 60	49
140	149	1	383	255	165	20.5	17.5		3 X 60	49
150	159	1	408	272	165	22	19		3 X 60	54
160	174	1	441	294	165	22	19		3 X 60	54
175	189	1	480	320	180	24.5	20.5		3 X 60	59
190	204	1	521	347	180	24.5	20.5		3 X 60	59
205	219	1	560	373	180	26	22	20.5	4 X 60	64
220	239	1	606	404	180	26	22	20.5	4 X 60	64
240	259	1	659	439	200	28	24	22	4 X 60	69
260	279	1	711	474	200	28	24	22	4 X 60	69
288	299	1	764	509	215	30	26	24	4 X 70	74
300	319	1	816	544	215	30	26	24	4 X 70	74
320	339	1	869	579	215	32	28	24	4 X 70	78
340	359	1	926	617	215	32	28	24	4 X 70	78
360	379	1	974	649	230	34	30	26	4 X 70	88
380	399	1	1028	685	230	34	30	26	4 X 70	88
400	424	1	1086	724	230	36	32	28	4 X 70	98
425	449	1	1152	768	230	36	32	28	4 X 70	98
450	474	1	1226	817	230	36	32	28	4 X 70	108
475	499	1	1284	856	230	38	34	30	4 X 70	108
500	549	2	1403	935	248	40	34	30	4 X 80	123
550	599	2	1535	1024	264	42	36	32	4 X 80	132
600	659	2	1694	1129	264	44	38	34	4 X 80	147
660	719	2	1853	1235	264	46	40	36	4 X 80	157
720	779	2	2012	1341	281	48	42	36	4 X 85	172
780	839	2	2171	1447	281	50	44	38	4 X 85	186
840	909	2	2329	1553	281	52	46	40	4 X 85	201
910	979	2	2515	1676	297	54	48	42	4 X 85	216
980	1059	2	2700	1800	297	56	50	44	4 X 90	230
1060	1139	2	2912	1941	297	58	50	46	4 X 90	250
1140	1219	2	3124	2082	314	60	52	46	4 X 90	270
1220	1299	2	3335	2224	314	62	54	48	4 X 90	284
1300	1389	2	3574	2382	314	64	56	50	4 X 90	309
1390	1479	2	3812	2541	330	66	58	50	5 X 90	324
1480	1569	2	4050	2700	330	68	60	52	5 X 95	324
1570	1669	2	4315	2876	330	70	62	54	5 X 95	333
1670	1789	2	4632	3088	347	73	64	56	5 X 95	353
1790	1930	2	4950	3300	347	76	66	58	5 X 95	378

Table II 4-2
Equipment Reductions for Service Restriction Notations (see Table II 4-1)

Service restriction notation	Number of bow anchors	Mass change per anchor	Length change of stud-link chain cables
Greater Coastal Service	1	No reduction	No reduction
Coastal Service	1	-30%	No reduction
Greater Coastal Service	2	-30%	+60%
Coastal Service	2	-50%	+60%

Notes:

(1) Other specified restricted service notation may be specially considered.

(2) Please refer to CR Steel Ship Rules Part I/1.4.4 for the definition of service restriction notation.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART III – MACHINERY INSTALLATIONS – CONSTRUCTION AND SHAFTING

April 2018

List of major changes in Part III from 2017 edition

Nil.

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART III MACHINERY INSTALLATIONS-CONSTRUCTION AND SHAFTING

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Chapter 1

General

1.1 General

1.1.1 The requirements of this Part are applicable to the machinery intended for ships without special service limitations or restrictions. The Society may, however, modify the requirements in certain particular cases for their application to ships with service limitations or restrictions.

1.1.2 The Society will be prepared to give special consideration to the novel features of design in respect of the machinery based on the best information available at the time.

1.1.3 Passenger ships intended for classification are to be constructed in accordance with the requirements of the Society as well as those of Governmental and International Convention Regulations.

1.1.4 The formulae for scantlings of parts of the machinery given in the present Part do not take into consideration the possibility of additional stresses due to the presence of dangerous vibrations in the installation at speeds within the operating range, and the manufacturer of the machinery is required to take responsibility in the application of these formulae.

1.2 Units and Formulae

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

1.2.2 Pressure gauges may be calibrated in bar.
where: 1 bar = 0.1 MPa

1.2.3 Ambient reference conditions

(a) The rating for classification purposes of main and essential service auxiliary machineries intended for installation in seagoing ships to be classed for unrestricted service is to be based on a total barometric pressure of 1,000 mbar, an engine room ambient temperature or suction air temperature of 0°C to 45°C, a relative humidity of 60% and a sea water temperature of 32°C or, where applicable, the temperature of the charge air coolant at the inlet of 32°C. In the case of open deck location, the temperature range is to be –25°C to 45°C. The engine manufacturer is not expected to provide simulated ambient reference conditions at a test bed.

(b) In the case of ships to be classed for restricted service, the rating is to be suitable for the temperature conditions associated with the geographical limits of the restricted service.

1.2.4 Power rating
Where requirements to dimensions in the Rules are based on output and revolutions, the values to be applied are to be derived from the following:

(a) For main propelling machinery, the maximum continuous output and corresponding revolutions per minute give the maximum torque for which the machinery is to be classed.

(b) For essential auxiliary machinery, the maximum continuous output and corresponding revolutions per minute which will be used in services.

1.3 Essential Service Auxiliaries

1.3.1 When applying the requirements of the Rules, the so called essential service auxiliaries may generally be assumed as specified hereinafter.

1.3.2 Auxiliaries include their prime movers and controllers which are necessary for the propulsion of the ship:
Cooling pumps.
General service pumps.
Fuel pumps.
Lubricating oil pumps.
Purifiers.
Hydraulic pumps (for control use).
Air compressors (for starting and control use).
Scavenging pumps, blowers and exhaust gas turbochargers.
Condensate pumps.
Drain pumps.
Circulating cooling water pumps.
Condenser ejector pumps.
Gland exhaust fans.
Boiler feed water pumps.
Boiler water circulating pumps.
Boiler fans.
Electric Generators.
Evaporators (for main propelling machinery and boiler use).

1.3.3 Auxiliaries include their prime movers and controllers which are necessary for the safety of life and for the safety or operation of the ship at sea:
Bilge pumps (including pumps for oily bilge separator use).
Ballast pumps.
Fire pumps.
Steering gear.
Athwartship thrust units.
Windlasses.
Mooring winches or capstans.
Hydraulic appliances (for windlasses and mooring machineries).
Ventilating fans (for machinery spaces).
Electric generators (for emergency source of power).
Machineries and equipment of venting, purging, gas freeing and ventilation systems (for tankers).
Machinery and boiler control equipment.
Incinerator.
Other auxiliaries deemed as an essential by the Society.

1.4 Materials

1.4.1 The materials used for the construction of main parts of machinery are to comply with the requirements in Part XI of "Rules for the construction and classification of steel ships", or such other appropriate material specifications as may be approved by the Society in connection with a particular design.

1.4.2 The materials for component parts of machinery are to be subjected to those test set out in the relevant Chapters of this Part. The society may, however, require material tests conducted on other important component parts and test methods not included in this Part.

1.4.3 The materials for component parts of machinery exposed to high temperature or chemical attack are to be of suitable quality.

1.4.4 Metal spraying and locking of cracks by special clamps etc. on shafting and other dynamically stressed parts of machinery are not permitted.

1.4.5 Steam reciprocating engines for main propelling and essential auxiliary services, where installed, are to comply with the applicable parts dealing with steam reciprocating engines in the 1976 edition of the Rules.

1.5 Drawings and Data

1.5.1 For machineries built under special survey during construction, drawings showing the proposed arrangements of machinery compartments and such drawings of the machineries as stated in the subsequent Chapters of this Part are to be submitted for approval before proceeding with the work.

1.5.2 The proposed dimensions and quality of materials as well as all important arrangements and details are to be made clear in the drawings.

1.5.3 Drawings and descriptions of centralized and automatic control systems for the main propelling installation and essential service auxiliaries are to be submitted for consideration in accordance with the applicable requirements in Part VIII of "Rules for the construction and classification of steel ships."

1.5.4 For any novel design of machinery, detailed drawings of parts and necessary data are to be submitted for consideration.

1.6 General Construction

1.6.1 Inclination of ships

The designs and constructions of machinery installations are to be in compliance with accepted marine engineering practices and the machineries are to be operable with complete reliability in all positions and motions with the ship under the conditions as shown in Table III 1-1.

Table III 1-1
Inclination of Ships

Type of machinery installations	Athwartships[1]		Bow-and-stern[1]	
	Static inclination (list)	Dynamic inclination (rolling)	Static inclination (trim)	Dynamic inclination (pitching)
Main propulsion machinery Main boilers and important auxiliary boilers Prime movers driv- ing generators (ex- cluding those for emergency) auxilia- ry machinery (ex- cluding auxiliary machinery for spe- cific use etc.) and their driving units	15°	22.5°	5°	7.5°
Emergency installa- tions (emergency generators, emer- gency fire pumps and prime movers to drive them)Switchgears[2] (Circuit breakers, etc.) Equipment for auto- matic and remote controls	22.5°[3]	22.5°[3]	10°	10°

Notes:

(1) Athwartships and bow-and-stern inclinations may occur simultaneously.

(2) Up to an angle of inclination of 45°, undesired switching operations or operational changes are not to be caused.

(3) In ships carrying liquefied gases in bulk and ships carrying dangerous chemicals in bulk, the arrangement is to be such that the emergency power supply must also remain operable with the ship flooded to a final athwart-ships inclination up to a maximum of 30°.

1.6.2 Availability for operation

Ship's machinery is to be so arranged that it can be brought into operation from the "dead ship" condition using only the facilities available on board. "Dead ship" condition is understood to mean that the entire machinery installation, including the power supply, is out of operation and that auxiliary services such as compressed air, starting current from batteries, etc., for bringing the main propulsion into operation and for the restoration of the main power supply are not available. In order to restore operation from the "dead ship" condition, an emergency generator may be used provided that it is ensured that the emergency power supply from it is available at all times. It is assumed that means are available to start the emergency generator at all times.

1.6.3 Astern powers

The machinery built for propulsion purposes are to be of sufficient power for going astern in order to secure proper control of the ship in all normal circumstances. The astern power is to be of sufficient for maintaining in free route astern 70% of the ahead rated shaft revolutions for a period of at least 30 minutes. The output astern which may be developed in transient conditions is to be such as to enable the braking of the ship within a reasonable time. For main propulsion systems with reversing gear, controllable pitch propellers or electric propulsion equipment, the running astern is not to lead to the overload of the propulsion machinery.

1.6.4 Turning gear

Provision is to be made so that it is possible to turn the main propelling and auxiliary machineries. For the propelling turbine, the turning gear is to be power driven, and if electric, is to be continuously rated. An interlocking warning device is to be provided to ensure that the machinery cannot be started while the turning gear is engaged.

1.6.5 All component parts of machinery subject to stresses are to be made of approved sound materials, and have clearances and proper fits consistent with the best marine engineering practices.

1.6.6 Welded construction

The welded constructions are to be in compliance with the requirements specified in Part X.

1.6.7 Safety devices on moving parts

(a) Efficient means are to be provided to prevent the loosening of nuts and screws of moving parts.

(b) The moving parts of machinery and shafting are to be efficiently protected by means of handrails, screens, etc.

1.6.8 Seating and fixing

(a) The machinery seating is to be of rigid construction and adequately attached to the hull. The effects of hull structure deformation on the machinery and excessive stressing due to shock and vibration as well as the thermal expansion of machinery are to be taken into account. Provision is to be made, as far as practicable, to ensure continuity between the longitudinal and transverse elements of the seating and the corresponding elements of the adjacent hull.

(b) The machinery is to be so securely bolted to the seating as to prevent any displacement due to the movements of the ship. The bedplate chock is to be of uniform fit before bolts are tightened.

(c) The accessories of machinery as well as spare parts of large dimensions are to be strongly secured so that they can not move or become loose under the movements of the ship.

1.6.9 Arrangements and ventilations

(a) The arrangement of machinery in the machinery compartment and tunnel is to be such that sufficient space is allowed for easy operation, maintenance and overhauling of machinery.

(b) Necessary instruments for a safe and speedy maneuvering and indicating the working condition of the machinery are to be clearly arranged at suitable positions.

(c) The machinery compartment is to be well ventilated and the ventilator is to be so arranged that any accumulation of inflammable gases is prevented as far as practicable.

(d) Flange joints and special joints (threaded pipe joints, compression fitting joints, etc.) in fuel oil, lubricating oil and other flammable oil pipings, and apparatuses contain fuel oil, lubricating oil or other flammable oil are not to be located right above hot surface on boilers, stream pipe lines, thermal oil pipe lines, exhaust gas pipe lines, silencers or exhaust gas driven turbochargers, etc. and to be arranged far apart therefrom, as far as practicable. An approved means to prevent oil from spraying is to be provided at these joints and apparatuses except where Society deems unnecessary to provide such means.

1.6.10 Communications

(a) The ship is to be fitted with 2 independent means of communicating orders between the bridge and engine room control station from which the engines are normally controlled. One of these means is to be an engine room telegraph which visually indicates the order and response, both at the engine room control station and on the bridge.

(b) At least one means of communication is to be provided between the bridge and any other control station from which the main propelling machinery may be controlled.

(c) An engineers' alarm is to be provided to be operated from the engine control room or at the maneuvering platform as appropriate, and to be clearly audible in the engineers' accommodation.

1.6.11 Protection against noise

Measures are to be taken to reduce machinery noise in machinery spaces to acceptable levels as determined by the National Regulations of the country in which the ship is registered. If this noise cannot be sufficiently reduced the source of excessive noise is to be suitably insulated or isolated or a refuge from noise is to be provided if the space is required to be manned. Ear protectors are to be provided for personnel required to enter such spaces, if necessary.

1.6.12 Control and safety

Main and auxiliary machinery essential for the propulsion, control and safety of the ship are to be provided with effective means for its operation and control. All control systems essential for the propulsion, control and safety of the ship are to be independent or designed such that failure of one system does not degrade the performance of another system.

1.6.13 Operating and maintenance instructions for ship machinery and equipment

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

1.7 Tests and Inspections

1.7.1 For the machinery built under the special survey during construction, the extent of supervision by the Society is to be as follows:

(a) Approval of constructional drawings and their calculations.

(b) Approval of proper materials used and their testing.

(c) Supervising proper execution of work and safe installation of machinery.

(d) Trial testing of machinery.

1.7.2 The machinery for ships classed or intended to be classed is normally to be tested and inspected in the presence of the Surveyor at the following occasions in accordance with the requirements of the Rules and the approved drawings:

(a) Material tests for component parts of machinery set out in the relevant Chapters of this Part.

(b) Confirmations of materials and components of machinery delivered from other manufacturers.

(c) Workmanship for machining from the commencement of work until the finish inspections for component parts of machinery.

(d) Tightness, balancing and non-destructive tests etc. for component parts of machinery set out in the relevant chapters of this Part.

(e) Shop trials of machinery.

(f) Installations of main propelling machinery, shafting, gearing and essential service auxiliaries, etc. on-board.

(g) On-board trials of machinery and essential systems.
When, however, the shop trials have been carried out and the trial requirements are fulfilled, the on-board trial may be modified suitably at the discretion of the Surveyor.

1.7.3 The Society will be prepared on application to adopt the alternative methods of inspection for the production line machinery and component parts subject to approval of manufacturer's production procedure and quality control.

1.7.4 Other tests and inspections not included in this Part may be required if deemed necessary by the Society.

1.7.5 Where the machinery or component parts of machinery have appropriate certificates, the tests and inspections may be wholly or partially dispensed with subject to further considerations and special approval by the Society.

1.8 Certification on Basis of Approved Quality Assurance Scheme for Machinery

1.8.1 Application
For machinery intended to be used for essential auxiliary services having an output of less than 750 kW, consideration will be given on application to the acceptance of standardized, batch and line produced machinery without tests and inspections of individual units subject to approval of the proposed designs and the manufacturer's quality control program.

1.8.2 Approval of works
A certificate will issued to each approved works, indicating the products for which approval has been granted.

1.8.3 Maintenance of approval
The Surveyors are to visit the works at intervals determined by the nature of the product, rate of production and standard of quality control arrangements. Maintenance of approval is to be subject to carrying out survey by regular and systematic auditing for the continuance of those details of supplies, organization, procedures and methods which form the basis for granting approval.

1.8.4 Certification of products
Nominated personnel of the manufacturer will be authorized by the Society to dispatch the products with certificates signed on behalf of the manufacturer to certify that the products conform to their design specifications and to the requirements of the Society. These certificates are to be countersigned by the Surveyor to certify that the approved arrangements are being kept under review by regular and systematic auditing of the manufacturing and quality control procedures.

1.8.5 Suspension or withdrawal of approval
When attention to significant faults or deficiencies in the manufacturing or quality control procedures have been drawn by the Surveyors, and these have not been rectified within a reasonable time, approval of the works will be suspended with notice in writing by the Society. When approval has been suspended and the manufacturer has proved unable or unwilling to take corrective action, approval will be withdrawn.

Chapter 2
Gas Turbines

2.1 General

2.1.1 Gas turbines intended for main propelling and essential auxiliary service are to be constructed and installed in accordance with the following requirements under the supervision and to the satisfaction of the Surveyor.

2.1.2 These requirements are also to be complied with as far as they are applicable to exhaust gas turbine driving scavenge air blowers, turbo-compressors and turbo-chargers. Features of gas turbine machinery which are not included in this Chapter will be specially considered.

2.1.3 Gas turbines driving electric propulsion generators are to be constructed and installed in accordance with the following requirements as well as the requirements stated in Chapter 13 of Part VII of "Rules for the construction and classification of steel ships."

2.2 Drawings and Data

2.2.1 The turbine manufacturer is to submit the following drawings for approval:
Sectional assembly, casing, combustion chambers, gasifiers, regenerators or recuperators, turbine rotors together with compressor rotors including turbine discs and blading details, maneuvering and control arrangement, turbine installation details and main condensers as well as starting arrangement, fuel oil, lubricating, air inlet and exhaust systems for gas turbines.

2.2.2 The turbine manufacturer is to submit the following data together with drawings for approval: Shaft horsepower, revolutions, working pressure and working temperature corresponding to the maximum continuous output, mass and velocity of rotating element, critical speed of rotors, material specification, etc. for the purpose of checking calculated stresses, welding details and rotor stress calculations as well as calculations regarding the torsional vibration as stated in 6.4 of this Part.

2.2.3 Inlet and exhaust ducting and securing arrangement (including details of resilient mounts where applicable) plans are to be submitted.

2.2.4 Components fabricated by means of welding will be considered for acceptance if constructed by firms whose works are properly equipped to undertake welding of the standards appropriate to the components. Details are to be submitted for consideration.

2.2.5 Before work is commenced, manufacturers are to submit for consideration details of proposed welding procedures and their proposals for routine examination of joints by non-destructive means.

2.2.6 The manufacturer's proposals for testing the gas turbine are to be submitted for consideration.

2.3 Materials

2.3.1 Materials intended for the following component parts of turbines are to be tested and inspected in the presence of the Surveyor in accordance with the requirements of Part IX or to the requirements of the specifications approved in connection with the design: Rotors, discs, blades, diaphragms, casings and Group-I and group-II pipes as classified in Part VI of “Rules for the construction and classification of steel ships.”

2.3.2 Turbine rotors and discs are generally to be of forged steel. Rotors for small turbines, however, may be made by a special casting process or by welding. Rotors and discs for gas turbines are to be made from materials non-corrodible in exhaust gases.

2.3.3 Turbine blades, shroud rings and binding wires are to be of non-corrodible materials.

2.3.4 Turbine casings and other castings subject to high pressure and temperature are to be of materials suitable for the stress and heat to which they may be exposed and are to be properly heat treated to remove internal stresses. Grey cast iron may be used for working parts with temperature up to 220°C.

2.3.5 For components of novel design special consideration will be given to the material test and non-destructive testing requirements.

2.4 Main Turbine Arrangement

2.4.1 Astern power

(a) The main propelling turbine installation is to be provided with the astern turbine which is constructed to provide the sufficient power to secure proper control of the ship in all normal circumstances. Accordingly, in the multi-screw installation, at least 2 astern turbines are to be provided.

(b) Astern turbines are to be able to develop, when operating astern under transient conditions, 80% of the rated ahead torque at 50% of the ahead revolutions and to maintain in free route astern 70% of the ahead revolutions for a period of at least 30 minutes without undue heating of the ahead turbines.

2.4.2 Emergency operation

The installation is to be so arranged that navigation with reduced power is possible in case of emergency.

2.5 Construction

2.5.1 Turbine casings

(a) Turbine casings are to be so designed that bearings are not adversely affected by heat flow from ad- jacent hot parts of the turbine gases.

(b) Guide vanes, diaphragms, labyrinth seals, etc. are to be arranged into the casing in such a way that they can expand freely without changing the position of their axes.

(c) The flange mating surface of turbine casing halves is to be metallically jointed. A thin layer of heat resistant flange cement may, however, be applied to the flange mating surface.

(d) Turbine casings are to be fitted with drains in places where water may collect.

2.5.2 Diaphragms

Diaphragms are to be preheated before guide vane connections are welded, and annealed subsequent to the welding.

2.5.3 Rotors, discs and blades

(a) Turbine and compressor rotors and discs are to be so designed that excessive vibration does not occur within the operating range of speeds. It is recommended that overhung rotors operating with variable speeds are of under-critical design. Over-critical rotors may, however, be approved if running and trial tests give satisfactory results.

(b) Blades are to be so designed as to avoid abrupt change in section and to provide an ample amount of stiffness to minimize deflection and vibration, and to have sufficiently large axial and radial clearances so that no interference with the static member can occur including creep in the material.

(c) Smooth and sufficiently large radius fillets are to be provided at abrupt changes of section of rotors, discs and blades. Edges of holes in turbine discs and blades are to be finely machined and preferably polished at the highly stressed areas.

2.5.4 Bearings and lubrication

(a) Small auxiliary turbines and exhaust gas blowers may, however, be approved with anti-friction bearings.

(b) An effective means is to be provided to prevent gas or condensate from entering the bearings. The bearings are to be so designed that the lubricant does not mix with the gas.

(c) Turbines intended for main propelling and driving generators are to be fitted with emergency arrangement by which lubricating oil is supplied to maintain adequate lubrication when lubrication oil supply is stopped in the event of the failure of lubricating pumps. Gravity head tanks or equivalent means may be accepted for this purpose. For lubrication system, See 2.7.2 and 2.7.3 of this Part and Part VI of “Rules for the construction and classification of steel ships”.

2.5.5 Expansion indicators

Indicators for determining the axial position of rotors relative to their casings and for showing the longitudinal expansion of casings at the sliding feet, if fitted, are to be provided for main turbines. The latter indicators are to be fitted at both sides and be readily visible.

2.5.6 Salt removal

Gas turbines intended for operation in salt atmospheres are to be fitted with an arrangement for removing or preventing the accumulation of salt deposits in compressors and turbines.

2.5.7 Major joints are to be designed as full-strength welds and for complete fusion of the joint.

2.5.8 Stress relief heat treatment is to be applied to all cylinders, rotors and associated components on completion of the welding of all joints.

2.6 Submitted Materials

The following information and calculations are to be submitted:

(a) Details of the acoustic enclosure fire detection and extinguishing system, where applicable.

(b) Power/speed operational envelope. Calculations and information for short term high power operation, where applicable. Operation and Maintenance Manuals.

(c) Calculations of the critical speeds of blade and rotor vibration, giving full details of the basic assumptions. An analysis of the effect of a rotor blade failure and any details of service experience.

(d) High temperature characteristics of the materials, where applicable, including (at the working temperatures) the associated creep rate and rupture strength for the designed service life, fatigue strength, corrosion resistance and scaling properties. Particulars of heat treatment, including stress relief, where applicable. Material specifications covering the listed components together with details of any surface treatments, non-destructive testing and hydraulic tests.

2.6.1 The most onerous pressures and temperatures to which each component may be subjected are to be indicated on plans or provided as part of the design specification.

2.6.2 Calculations of the steady state stresses, including the effect of stress raisers, etc., in the turbine and compressor rotors and blading at the maximum speed and temperature in service are to be submitted. Such calculations should indicate the designed service life and be accompanied, where possible, by test results substantiating the limiting criteria.

2.6.3 Details of calculations and tests to establish the service life of other stressed parts, including gearing (where applicable), bearings, seals, etc., are also to be submitted. All calculations and tests should take account of all relevant environmental factors including particular type of service and fuel intended to be used.

2.7 Safety Devices of Gas Turbines

2.7.1 Governors and overspeed protective devices

(a) All main and auxiliary gas turbines are to be provided with speed control governors (or overspeed governors) and separate overspeed protective devices. The overspeed protective devices are to be independent from the governors and so adjusted that the output shaft speed may not be exceeded by more than 15% of the maximum continuous speed, and are to have functions as specified in 2.8.2.

(b) The governors of gas turbines to drive generators are to conform to the requirements in 3.2 of Part VII of “Rules for the construction and classification of steel ships.”

2.7.2 Emergency stopping devices

(a) Gas turbines are to be provided with emergency stopping devices operated by suitable hand trip gears installed at the control station.

(b) Gas turbines are to be provided with automatic fuel oil shut-off devices that operate in the following cases. In addition, means are to be provided so that alarms will be issued at the control station when the automatic fuel oil shut-off devices come into action.
- (i) Overspeed.
- (ii) Low lubricating oil pressure.
- (iii) Failure in automatic starting.
- (iv) Flame-out.
- (v) Excessive vibrations.

2.7.3 Alarms
Gas turbines are to be provided with alarm devices that operate in the following cases:

(a) High temperature of turbine inlet or outlet gas.

(b) Low lubricating oil pressure (working before the function of the emergency stopping device specified in preceding 2.7.2).

(c) Low fuel oil supply pressure.

2.7.4 Equipment for starting main and auxiliary turbines is to be provided and arranged such that the necessary initial charge of starting air or initial electric power can be developed on board the craft without external aid. If for this purpose an emergency air compressor or electric generator is required, these units are to be power driven by manually started oil engines except in the case of small installations where a hand operated compressor of approved capacity may be accepted. Alternatively, other devices of approved type may be accepted as a means of providing the initial start.

2.8 Tests and Inspections

2.8.1 Hydraulic pressure tests

Hydraulic pressure tests on turbine parts after machining are to be carried out in the presence of the Surveyor under the conditions specified in Table III 2-1.

Table III 2-1
Hydraulic Pressure Tests on Turbine Parts

Parts to be Tested	Test Pressure	Remarks
Turbine casings(1)	1.5 W	At least 0.2
Cooling water spaces	1.5 W	At least 0.4
heat exchangers	See Part V	
Group-I and -II pipes and fittings	See Part VI	
Where: W = Maximum working pressure for the respective parts, in MPa.		

Note:
(1) Test pressure for exhaust gas turbine casings, see Table III 3-3.

2.8.2 Balancing tests

Finished rotors, complete with balanced and all the rotating parts, are to be dynamically balanced in the presence of the Surveyor.

2.8.3 Non-destructive tests

The following turbine parts are to be tested in the presence of the Surveyor by approved non-destructive testing methods:

(a) Turbine casing – at abrupt changes of sections and at points liable to casting faults.

(b) Diaphragm – guide vanes and welded connections.

(c) Turbine rotor, disc, blading, welded connections and shrink surfaces – over the complete surfaces and fillets.

2.8.4 Shop trials

(a) The warm running tests on the bladed exhaust gas turbine are to extend over a period of 20 minutes at the working speed and 10 minutes at a speed 10% above the maximum working speed.

(b) The overspeed protective device is to be set not more than 15% above the maximum design speed.

(c) On completion of the above shop trials, turbines are to be overhauled and inspected for fineness, wear, clearances on major working parts, securities of locking devices, etc. in the presence of the Surveyor.

2.8.5 On-board trials

(a) After installation on-board, the main propelling and essential auxiliary turbines are to be trial tested under working conditions in the presence of the Surveyor to demonstrate that the entire installation is working satisfactorily without undue vibrations of hull or turbines within the service range. For this purpose, the following trials on-board are normally to be carried out as far as practicable.

(b) For main propelling turbines, rated maximum continuous revolutions for at least 2 hours, corresponding revolutions of partial loads of the turbine rated output in ahead running and astern running for suitable durations, minimum speed and safety device tests are normally to be carried out.

(c) For turbines intended for auxiliary purposes, they are to be run at the maximum continuous output for at least 3 hours, and partial and overloads for suitable durations if deemed necessary by the Surveyor. These load tests may be based on the power of driven auxiliary.

(d) On completion of the above on-board trials, the turbine, to the discretion of the Surveyor, may be opened up for inspection to a certain extent as deemed necessary.

Chapter 3

Diesel Engines

3.1 General

3.1.1 The construction and installation of diesel engines for main propelling and essential service are to be carried out in accordance with the following requirements under the supervision and to the satisfaction of the Surveyor.

3.1.2 Diesel engines which drive electric propulsion generators are to be constructed and installed in accordance with the following requirements as well as the requirements stated in 3.2 of Part VII of "Rules for the construction and classification of steel ships."

3.1.3 The diesel engine intended for emergency electric source is also to comply with the requirements stated in Chapter 11 of Part VII of "Rules for the construction and classification of steel ships."

3.1.4 These Rules are applicable to machinery systems burning distillate fuels which do not require to be heated.

3.1.5 It is the responsibility of the Shipbuilder as main contractor to ensure that the information required is prepared and submitted.

3.1.6 The main propulsion machinery will be approved for the maximum continuous power, and associated shaft speed, required to achieve the maximum craft velocity at the certified maximum operational weight in smooth water.

3.1.7 Main propulsion machinery will be considered for operation at a higher power rating than the classification rating for short time intervals (referred to as short term high power operation) in conjunction with the intended operation service profile.

3.1.8 Roller element bearings are to have a design life of at least 30,000 hours, based upon the design operating conditions, including short term high power operation. A design life of less than 30,000 hours would be accepted, provided it is proposed in conjunction with the manufacturer's design/maintenance manual.

3.2 Drawings and Data

3.2.1 For each type of engine to be approved the documents listed in the Table III 3-1 and as far as applicable to the type of engine are to be submitted to the Society for approval (A) or for information (R) by each engine manufacturer (see Note 3). After the approval of an engine type has been given by the Society 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 Society. In cases where 2 identifications (R/A) are given, the first refers to cast design and the second to welded design. The assignment of the letter R does not preclude possible comments.

3.2.2 At least four copies of plans, information and specifications as listed are to be submitted before commencement of manufacture.

3.2.3 Plans showing the arrangement of resiliently mounted machinery are to indicate the number, position, type, and design of mounts.

3.2.4 The plans of arrangement of resin chocks for machinery requiring accurate alignment are to be submitted.

3.2.5 Where it is proposed to use alloy castings, micro alloyed or alloy steel forgings or iron castings, details of the chemical composition, heat treatment and mechanical properties are to be submitted.

3.2.6 Plans and details for dead craft condition starting arrangements are to be submitted for appraisal.

3.3 Materials

3.3.1 Materials intended for the component parts of diesel engines are to be tested and inspected in the presence of the Surveyor in accordance with the requirements of Part IX or with the requirements of the specification approved in connection with the design as listed in 3.10.2 of this Chapter.

3.3.2 Engine parts subject to stress are to be made of sound materials, and cylinders, cylinder liners, cylinder covers, pistons, etc. under high temperature or pressure are to be made of materials suitable for the stress and temperature to which they are exposed.

3.3.3 Materials used in the construction of machinery and its installation are not to contain asbestos.

3.3.4 The specified minimum tensile strength of castings and forgings for crankshafts is to be selected within the following general limits:

(a) Carbon-manganese steel castings –
400 to 550 N/mm^2.

(b) Carbon-manganese steel forgings (normalised and tempered) –
400 to 600 N/mm^2.

(c) Carbon-manganese steel forgings (quenched and tempered) –
not exceeding 700 N/mm^2.

(d) Alloy steel castings –
not exceeding 700 N/mm^2.

(e) Alloy steel forgings –
not exceeding 1000 N/mm^2.

(f) Spheroidal or nodular graphite iron castings –
370 to 800 N/mm^2.

3.4 Construction

3.4.1 General

(a) Frames and bedplates are to be of rigid and oil-tight construction. Crank cases are to be strongly built and doors or covers securely fastened and made air and oil-tight so that they can withstand a considerable excessive pressure within the crank cases without any risk of damage.

Table III 3-1
Drawings and Data to be Submitted

No.	A/R	Item
1	A	Engine particulars
2	R	Engine transverse cross-section
3	R	Engine longitudinal section
4	R/A	Bedplate or crankcase, cast or welded with welding details and instructions
5	A	Thrust bearing assembly[3]
6	R/A	Thrust bearing bedplate, cast or welded with welding details and instructions[1]
7	R/A	Frame/column, cast or welded with welding details and instructions[1]
8	R	Tie rod
9	R	Cylinder cover, assembly
10	R	Cylinder jacket or engine block[1,2]
11	R	Cylinder liner[2]
12	A	Crankshaft, details, each cylinder No.
13	A	Crankshaft, assembly, each cylinder No.
14	A	Thrust shaft or intermediate shaft (if integral with engine)
15	A	Coupling bolts
16	A	Counterweights (if not integral with crankshaft)
17	A	Connecting rod
18	A	Connecting rod, assembly[2]
19	R	Crosshead, assembly[2]
20	R	Piston rod, assembly[2]
21	R	Piston, assembly
22	R	Camshaft drive, assembly
23	A	Material specifications of main parts with information on non-destructive material tests and pressure tests
24	A	Arrangement of foundation bolts (for main engines only)
25	A	Schematic layout or other equivalent documents of starting air system on the engine[6]
26	A	Schematic layout or other equivalent documents of fuel oil system on the engine[6]
27	A	Schematic layout or other equivalent documents of lubricating oil system on the engine[6]
28	A	Schematic layout or other equivalent documents of cooling water system on the engine[6]
29	A	Schematic diagram of engine control and safety system on the engine[6]
30	R	Shielding and insulation of exhaust pipes, assembly
31	A	Shielding of high pressure fuel pipes, assembly[4]
32	A	Arrangement of crankcase explosion relief valve[5]
33	R	Operation and service manuals

Footnotes:

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. For attended engine room: only engines with a bore of 250 mm and above.
   For unattended engine room: all engines.
5. Only for engines with bore exceeding 200 mm.
6. The entire system. If this is part of the goods to be supplied by the engine manufacturer.

Notes:

1. The approval of exhaust gas turbochargers, charge air coolers, etc. is to be obtained by the respective manufacturer.
2. Where considered necessary, the Society may request further documents to be submitted.
3. A Licensee is to submit, for each engine type manufactured, a list of all documents required with the relevant drawing numbers and revision status from both Licensor and Licensee. Where the Licensee proposes design modifications to components, the associated documents are to be submitted by the Licensee for approval or for information. In case of significant modifications a statement is to be made confirming the Licensor's acceptance of the changes.
   In all cases a complete set of documents will be required by the Surveyor(s) attending the Licensee's work.

(b) Passages for cooling water and lubricating oil are to be carefully cleaned of sand and scale.

(c) Clutches or reversing gear built in engines are to be in accordance with the requirements stated in Chapter 5 of this Part.

3.4.2 Cylinder relief valves

A relief valve set to relieve at not more than 140% of the designed maximum firing pressure is to be fitted on each cylinder when the cylinder bore exceeds 230 mm. For auxiliary engines, an approved device capable of giving warning of excessive cylinder pressure will be specially considered as an alternative. Outlet ports of cylinder relief valves are to be so placed as to avoid endangering personnel.

3.4.3 Crankcase explosion relief valves

(a) Crankcases are to be provided with lightweight spring loaded relief valves or other quick-acting and self-closing devices able to lift rapidly in order to prevent any considerable excessive pressure within the crankcase in the event of internal explosion and to close automatically after the passage of the explosion wave in order to prevent any inrush of external air. Relief valves are to be so designed that the discharge from valves is shielded by flame trap to minimize the possibility of danger and damage arising from the emission of flame, and the operation of the valves is to be actuated by an excessive pressure as low as possible with 0.02 MPa to be the maximum.

(b) Generally, in large engines at least one relief valve is to be fitted to each cylinder crankcase and separate gear or chain case for camshaft or similar drive, when the gross volume of such space is 0.6 m^3 and above. In small engines where there is free communication between all cylinder crankcases, relief valves are to be provided in numbers not less than those given in Table III 3-2. Where only 2 relief valves are required, they are to be located at or near each end of the crankcase. In V-type cylinder arrangement engines, half the number of cylinders may generally be taken in using this table. In engines having a cylinder of less than 200 mm bore or having a crankcase volume of less than 0.6 m^3, relief valves may be omitted.

Table III 3-2
Crankcase Explosion Relief Valves

Cylinder Bore D (mm)	No. of Cylinders per Engine	Min. Number of Explosion Relief Valves per Engine
$200 \le D < 250$	≤ 8 > 8	2 3
$250 \le D < 300$	Any number	One for each alternate cylinder, See Note
$300 \le D$	Any number	One per cylinder

Note:

(1) For engines having 3, 5, 7, 9 etc. cylinders, the number of explosion relief valves is not to be less than 2, 3, 4, 5 etc. ... respectively.

(c) The minimum free area of crankcase explosion relief valves is to be determined as follows:

$$A = \frac{CV}{Z}$$

where:

A = Minimum free area of each explosion relief valve, in cm^2, but not less than 45 cm^2.

V = Total gross volume of crankcase compartment, in m^3. The volume of stationary parts within the crankcase may be deducted.

Z = Number of explosion relief valves fitted to each engine, which is to be not less than given in Table III 3-2.

C = 115 for engines with a free communication crankcase, and
= 50 for crosshead type engines with a diaphragm fitted between the cylinder and a crankcase.

3.4.4 Protection against scavenging spaces

(a) Crosshead type engine scavenging spaces in open connection to the cylinders are to be provided with explosion relief valves. These devices are to be so arranged to discharge that no damage for operators can occur.

(b) Crosshead type engine scavenging spaces in open connection to the cylinders are to be connected to an approved fire extinguishing system, which is to be entirely separate from the fire extinguishing system of the engine room.

3.4.5 Crankcase Oil Mist Detection Arrangements

(a) Crankcase oil mist detection arrangements are required for diesel engines of 2250 kW maximum continuous power and above or having cylinders of more than 300mm bore, and in case of engine failure, the following means are automatically employed. However, in cases where alternative devices (e.g. engine bearing temperature monitors or equivalent devices) deemed appropriate by the Society are provided, such devices may be used instead of crankcase oil mist detection arrangements.
   (i) In the case of low speed (crosshead) engines, alarms are to activate, and speed be reduced automatically or manually.
   (ii) In the case of medium/high speed (trunk piston) engines, alarms are to activate and diesel engines are to be stopped or have their fuel supply shut off automatically.

(b) The crankcase oil mist detection arrangements required in 3.4.5(a) above are to be of an approved type and in accordance with the following requirements:
   (i) The oil mist detection arrangements are to provide an alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangements.
   (ii) The oil mist detection arrangements are to provide an indication that any lenses fitted in the equipment and used in determination of the oil mist level have been partially obscured to a degree that will affect the reliability of the information and alarm indication.
   (iii) The oil mist detection arrangements are to be capable of being tested on the test bed and board under engine at standstill and running at normal operating conditions.
   (iv) Oil mist detection and alarm information is to be capable of being read from a safe location away from engine.
   (v) In the case of ships which apply the rules for automatic or remote control and monitoring system, the density of crankcase oil mist is also to be capable of being monitored, as specified in Table VIII 4-5A, VIII 4-5B and VIII 4-7 of Part VIII of “Rules for the construction and classification of steel ships.”
   (vi) Each engine is to be provided with its own independent oil mist detection arrangement and a dedicated alarm.
   (vii) The layout of the arrangements, pipes and cables, pipe dimensions, the location of engine crankcase sample points, sample extraction rate and the way of maintenance and test are to be in accordance with the engine designer’s and oil mist manufacturer’s instructions.
   (viii) Where sequential oil mist detection arrangements are provided, the sampling frequency and time is to be as short as reasonably practicable.
   (ix) A copy of the maintenance and test manual is to be provided on board ship.

3.4.6 Crankcase ventilation

(a) If provisions are made for ventilation of enclosed crankcases by means of breathers or mechanical suctions, the corresponding suction pressure is not to exceed 25 mm of water. Crankcases are not to be ventilated by a blast of air.

(b) Where a breather is provided connecting any part of the crankcase, this breather is to have a dimension as small as practicable to minimize the inrush of air after an explosion and is to be led to a safe position on deck or other approved position.

(c) Crankcase lubricating oil pipes from engine sumps to their respective drain tanks are to be submerged at their outlet ends. Where 2 or more engines are installed, the pipes leading to crankcases as well as breathers and lubricating oil pipes are to be laid independently of one another to avoid intercommunication between the crank-cases.

3.4.7 Warning notice

A notice of warning is always to be fitted either on the control stand or, preferably, on a discernible place near the crankcase door on both sides of the engine specifying that the crankcase doors or sight holes are not to be opened within a reasonable period which is normally not less than 10 minutes after stopping the engine in case overheat is suspected within the crankcase.

3.4.8 Speed governor and overspeed protective device of main diesel engines

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

(b) In addition to this speed governor each main engine having a rated power of 220 kW or over which can be declutched or which drives a controllable pitch propeller, is to be fitted with a separate overspeed protective device, which including its driving mechanism, has to be independent from the governor required in 3.4.8 (a) above, and so adjusted that the speed may not be exceeded by more than 20% of the maximum continuous revolutions.

(c) When electronic speed governors of main diesel engines form part of a remote control system, they are to comply with the following condition:

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

(ii) local control of the engines is always to be possible, and, to this purpose, from the local control position it is to be possible to disconnect the remote signal, bearing in mind that the speed control, is not available unless an additional separate governor is provided for such local mode of control.

(d) In addition, electronic speed governors and their actuators are to be type tested.

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

(a) For overspeed protective device, see 3.4.8(b) above, and so adjusted that the speed may not be exceeded by more than 15% of the maximum continuous revolutions.

(b) For speed governor characteristics of generator prime movers, see Chapter 3 of Part VII of “Rules for the construction and classification of steel ships.”

3.4.10 The junction of the oil hole with the crankpin or main journal surface is to be formed with an adequate radius and smooth surface finish down to a minimum depth equal to 1.5 times the oil bore diameter.

3.4.11 Care is to be taken to avoid stress concentrations such as sharp corners and abrupt changes in section.

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

(a) Description of valve with details of function and design limits.

(b) Copy of type test certification.

(c) Installation instructions.

(d) Maintenance and in-service instructions to include testing and renewal of any sealing arrangements.

(e) Actions required after a crankcase explosion.

3.4.13 A copy of the installation and maintenance manual required by 3.4.12 is to be provided on board the ship.

3.4.14 Means are to be provided to ensure that machinery can be brought into operation from the dead craft condition without external aid.

3.4.15 Dead craft condition for the purpose of 3.4.14 is to be understood to mean a condition under which the main propulsion plant and auxiliaries are not in operation. In restoring propulsion, no stored energy for starting and operating the propulsion plant is assumed to be available. Additionally, neither the main source of electrical power nor other essential auxiliaries is assumed to be available for starting and operating the propulsion plant.

3.4.16 Where there is no emergency generator installed, the arrangements for bringing main and auxiliary machinery into operation are to be such that the initial charge of starting air or initial electrical power and any power supplies for engine operation can be developed on board the craft without external aid. If, for this purpose, an emergency air compressor or an electric generator is required, these units are to be powered by a hand-starting oil engine or a hand-operated compressor. The arrangements for bringing main and auxiliary machinery into operation are to have capacity such that the starting energy and any power supplies for engine operation are available within 30 minutes of a dead craft condition.

3.4.17 Two or more filters are to be fitted in the oil fuel supply lines to the main and auxiliary engines, and the arrangements are to be such that any filter can be cleaned without interrupting the supply of filtered oil fuel to the engines.

3.4.18 Drip trays are to be fitted under oil fuel filters and other fittings which are required to be opened up frequently for cleaning or adjustment or where there is the possibility of leakage. Alternative arrangements may be acceptable and full details should be submitted for consideration.

3.4.19 At least two independent means of stopping the engines quickly from the control station under any conditions are to be available.

3.5 Crankshafts

3.5.1 General

(a) Scope

These Rules for the scantling of crankshafts are to be applied to diesel engines for main 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.

(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.

(c) Principles of calculation

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

The calculation is also based on the assumption that the fillet transitions between the crankpin and web as well as between the journal and web are the areas exposed to the highest stresses. The outlets of oil bores into crankpins and journals are to be formed in such a way that the safety margin against fatigue at the oil bores is not less than that acceptable in the fillets. The engine manufacturer, if requested by the Society, is to submit a documentation supporting his oil bore design. Calculation of crankshaft strength consists initially in determining the nominal alternating bending and nominal alternating torsional stresses which, multiplied by the appropriate stress concentration factors using the theory of constant energy of distortion (von Mises' Criterion), result in an equivalent alternating stress (uni-axial stress). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material. This comparison will then show whether or not the crankshaft concerned is dimensioned adequately.

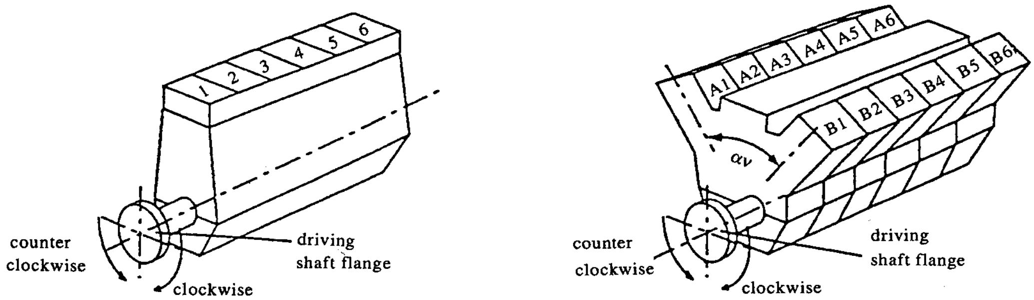


Fig. III 3-1
Designation of the Cylinders

(d) Drawings and particulars to be submitted

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

(i) crankshaft drawing which must contain all data in respect of the geometrical configuration of the crankshaft.

(ii) type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod).

(iii) operating and combustion method (2-stroke or 4-stroke cycle/direct injection, precombustion chamber, etc.).

(iv) number of cylinders.

(v) rated power (kW).

(vi) rated engine speed (1/min).

(vii) sense of rotation (see Fig. III 3-1).

(viii) firing order with the respective ignition intervals and, where necessary, V-angle α_v (see Fig. III 3-1).

(ix) cylinder diameter (mm).

(x) stroke (mm).

(xi) maximum cylinder pressure P_{max} (bar).

(xii) charge air pressure (bar) (before inlet valves or scavenge ports, whichever applies).

(xiii) nominal compression ratio.

(xiv) connecting rod length L_H (mm).

(xv) oscillating weight of one crank gear (kg) (in case of V-type engines, where necessary, also for the cylinder unit with master and articulated-type connecting rod or forked and inner connecting rod.).

(xvi) digitalized gas pressure curve presented at equidistant intervals (bar versus crank angle)

intervals equidistant and integrally divisible by the V-angle, but not more than 5° CA.

(xvii) for engines with articulated-type connecting rod (see Fig. III 3-2)

(1) distance to link point L_A (mm).

(2) link angle α_N(°).

(3) connecting rod length L_N (mm).

(xviii) for the cylinder with articulated-type connecting rod

(1) maximum cylinder pressure P_{max} (bar).

(2) charge air pressure (bar) (before inlet valves or scavenge ports, whichever applies.).

(3) nominal compression ratio.

(4) digitalized gas pressure curve presented at equidistant intervals (bar versus crank angle);

(xix) details of crankshaft material;

(xx) material designation;

(xxi) mechanical properties of material;

The minimum requirements of Rule's materials must have been complied with:

(1) tensile strength (N/mm^2);

(2) yield strength (N/mm^2);

(3) reduction in area at break (%);

(4) elongation A_5 (%);

(5) impact energy – KV (J);

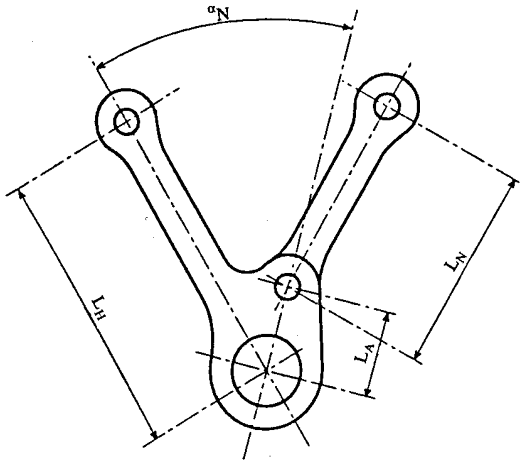


Fig. III 3-2
Articulated-type Conncting Rod

(xxii) method of material melting process(open-hearth furnace, electric furnace, etc.);

(xxiii) type of forging (free form forged, continuous grain flow forged, drop-forged, etc.; with description of the forging process);

(xxiv) heat treatment;

(xxv) surface treatment of fillets, journals and pins (induction hardened, flame hardened, nitrided, rolled, shot peened, etc. with full details concerning hardening);

(1) hardness at surface (HV);

(2) hardness as a function of depth (mm);

(3) extension of surface hardening;

(xxvi) particulars for alternating torsional stresses, see item 3.5.2(b).

3.5.2 Calculation of stresses

(a) Calculation of alternating stresses due to bending moments and shearing forces

(i) Assumptions

The calculation is based on a statically determined system, so that only one single crank throw is considered of which the journals are supported in the center of adjacent bearings and which is subject to gas and inertia forces. The bending length is taken as the length between the two main bearings (distance L_3), see Figs III 3-3 and III 3-4. The nominal bending moment is taken as the bending moment in the crank web cross-section in the center of the solid web (distance L_1) basing on a triangular bending moment load due to the radial components of the connecting rod force. For crank throws with two connecting rods acting upon one crankpin the nominal bending moment is taken as a bending moment obtained by superposition of the two triangular bending moment loads according to phase. The nominal alternating stresses due to bending moments and shearing forces are to be related to the cross-sectional area of the crank web. This reference area of cross-section results from the web thickness W and the web width B in the center of the overlap of the pins or, if appropriate, at the center of the adjacent generating lines of the two pins if they do not overlap, see Fig. III 3-5. Nominal mean bending stresses are neglected.

(ii) Calculation of nominal alternating bending and shearing stresses

As a rule the calculation is carried out in such a way that the individual radial forces acting upon the crank pin owing to gas and inertia forces will be calculated for all crank positions within one working cycle. A simplified calculation of the radial forces may be used as the discretion of the Society. By means of these radial forces variable in time within one working cycle and taking into account the distance of acting position on the pin, the time curve of the bending moment M_B in the web center as defined in 3.5.2(a)(i) will then be calculated. The decisive nominal alternating bending moment will then be calculated

$$M_{BN} = \pm \frac{1}{2}(M_{B,max} - M_{B,min})$$

and, from the latter, the nominal alternating bending stress which will be modified by the empirical factor K_e which considers to some extent the influence of adjacent cranks and bearing restraint.

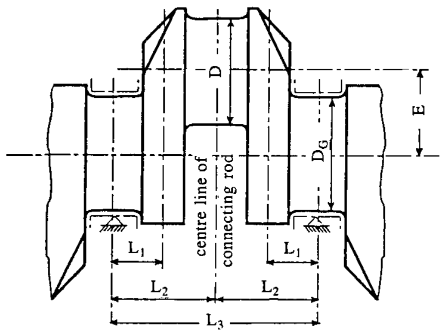


Fig. III 3-3
Crank Throw for In-line Engine

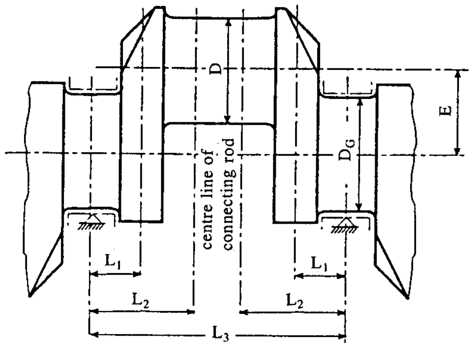


Fig. III 3-4
Crank Throw for Engine with 2 Adjacent Connecting Rods

$$\sigma_{BN} = \pm \frac{M_{BN}}{W_{eq}} \cdot 10^3 \cdot K_e$$

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

K_e = 0,8 for 2-stroke engines

= 1.0 for 4-stroke engines

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, the differing designs (forked connecting rod, articulated-type connecting rod or adjacent connecting rods) being taken into account. Where there are cranks of different geometrical configuration (e.g., asymmetric cranks) in one crankshaft, the calculation is to cover all crank variants. The calculation of the nominal alternating shearing stress is as follows:

$$\sigma_{QN} = \pm \frac{Q_N}{F} \bullet K_e$$

$$Q_N = \pm \frac{1}{2} (Q_{max} - Q_{min})$$

$$F = B \bullet W$$

where:

M_{BN} = Nominal alternating bending moment (Nm);

σ_{BN} = Nominal alternating bending stress (N/mm^2);

W_{eq} = Equatorial moment of resistance related to cross-sectional area of web (mm^3);

Q_N = Nominal alternating shearing force (N);

σ_{QN} = Nominal alternating stress due to shearing force (N/mm^2);

F = Area related to cross-section of web (mm^2).

(iii) 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.

(1) For the crankpin fillet

$$\sigma_{BH} = \pm(\alpha_B \bullet \sigma_{BN})$$

where:

σ_{BH} = alternating bending stress in crankpin fillet (mm^2);

α_B = stress concentration factor for bending in crankpin fillet (determination – see 3.5.3).

(2) For the journal fillet

$$\sigma_{BG} = \pm(\beta_B \bullet \sigma_{BN} + \beta_Q \bullet \sigma_{QN})$$

where:

σ_{BG} = alternating stresses in journal fillet (N/mm^2);

β_B = stress concentration factor for bending in journal fillet (determination – see 3.5.3);

β_Q = stress concentration factor for shearing (determination – see 3.5.3).

(b) Calculation of alternating torsional stresses

(i) General

The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in 3.5.2(b)(ii). The maximum value obtained from such calculations will be used by the Society when determining the equivalent alternating stress, according to 3.5.5. In the absence of such a maximum value it will be necessary for Society to incorporate a fixed value in the calculation for the crankshaft dimensions on the basis of an estimation.

(ii) Calculation of nominal alternating torsional stresses

The max. and min. alternating torques are to be ascertained for every mass point of the system and for the entire speed range by means of a harmonic synthesis of the forced vibration from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines. Whilst doing so, allowance must be made for the dampings that exist in the system and for unfavorable conditions (misfiring in one of the cylinders). The speed stages are to be selected in such a way that the transient response can be recorded with sufficient accuracy. The values received for such calculations are to be submitted. The nominal alternating torsional stress in every mass point, which is essential to the assessment, results from the following equation:

$$\tau_N = \pm \frac{M_T}{W_P} \bullet 10^3$$

$$M_T = \pm \frac{1}{2}(M_{T\max} - M_{T\min})$$

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

Where:

τ_N = Nominal alternating torsional stress referred to crankpin or journal (N/mm^2);

M_T = Nominal alternating torque (Nm);

W_P = Polar moment of resistance related to cross-sectional area of bored crankpin or bored journal (mm^3);

M_{Tmax}, M_{Tmin} = Extreme values of the torque with consideration of the mean torque.

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

Barred speed ranges are to be so arranged that satisfactory operation is possible despite of their existence. These are to be no barred speed ranges above a speed ration of $\lambda \geq 0.8$ of the rated speed. The approval of crankshafts is to be based on the installation having the lowest acceptability factor. Thus, for each installation, it is to be ensured by suitable calculation that the approved nominal alternating torsional stress is not exceeded. This calculation is to be submitted for assessment.

(iii) Calculation of alternating torsional stresses in fillets

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

(1) For the crankpin fillet:

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

where:

τ_H = Alternating torsional stress in crankpin fillet (N/mm^2);

α_T = Stress concentration factor for torsion in crankpin fillet (determination – see 3.5.3).

(2) For the journal fillet:

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

Where:

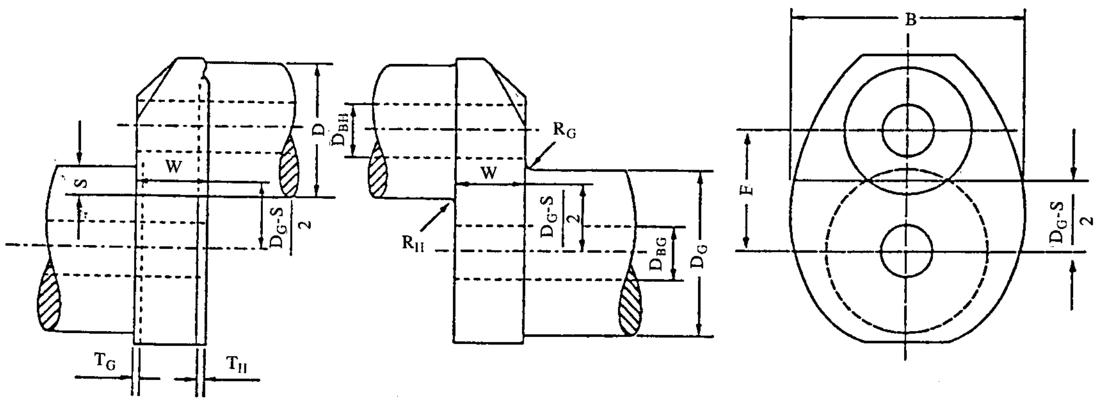
τ_G = Alternating torsional stress in journal fillet (N/mm^2).

β_T = Stress concentration factor for torsion in journal fillet (determination – see 3.5.3).

3.5.3 Calculation of stress concentration factors

(a) General

The stress concentration factors for bending (α_B, β_B) is defined as the ratio of the maximum bending stress-occurring in the fillets under bending load acting in the central cross-section of a crank-to the nominal stress related to the web cross-section. The nominal stress has to be determined under the bending moment in the middle of the solid web. The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum torsional stress-occurring under torsional load in the fillets-to the nominal stress related to the bored crankpin or journal cross-section. The stress concentration factor for shearing (β_Q) is defined as the ratio of the maximum shear stress-occurring in the journal fillet under bending load-to the nominal shear stress related to the web cross-section. Where the stress concentration factors cannot be furnished by reliable measurements the values may be evaluated by means of the formulae according to 3.5.3(b) and 3.5.3(c), applicable to the fillets of solid-forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. All crank dimensions necessary for the calculation of stress concentration factors are shown in Fig. III 3-5.



Actual dimensions:

D = Crankpin diameter (mm);

D_{BH} = Diameter of bore in crankpin (mm);
R_H = Fillet radius of crankpin (mm);
T_H = Recess of crankpin (mm);
D_G = Journal diameter (mm);
D_{BG} = Diameter of bore in journal (mm);
R_G = Fillet radius of journal (mm);
T_G = Recess of journal (mm);
E = Pin eccentricity (mm);
S = Pin overlap (mm);

$$S = \frac{D + D_G}{2} - E$$

W = Web thickness (mm).
B = Web width (mm).

Fig. III 3-5
Crank Dimensions Necessary for the Calculation of Stress Concentration Factors

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

crankpin fillets	journal fillets
$r = R_H/D$	$r = R_G/D$
$s = S/D$ $w = W/D$ $b = B/D$ $d_G = D_{BG}/D$ $d_H = D_{BH}/D$ $t_H = T_H/D$ $t_G = T_G/D$	

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

$-0.5 \leq s \leq 0.7$
$0.2 \leq w \leq 0.8$
$1.2 \leq b \leq 2.2$
$0.03 \leq r \leq 0.13$
$0 \leq d_G \leq 0.8$
$0 \leq d_H \leq 0.8$

The factor f (recess) which accounts for the influence of a recess in the fillets is valid if

$t_H \leq R_H/D$
$t_G \leq R_G/D$

and is to be applied within the range
$-0.3 \leq s \leq 0.5$

(b) Crankpin fillet

The stress concentration factor for bending (α_B) is:

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

where:

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

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

$f(b) = 0.6840 - 0.0077 \cdot b + 0.1473 \cdot b^2$
$f(r) = 0.2081 \cdot r^{(-0.5231)}$
$f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3$
$f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241\ d_H^2 + 2.4147 \cdot d_H^3$
$f(recess) = 1 + (t_H+t_G) \cdot (1.8+3.2 \cdot s)$

The stress concentration factor for torsion (α_T) is:

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

where:

$f(r,s) = r[-0.322+0.1015\bullet(1-s)]$
$f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 - 0.857 \cdot b^3$
$f(w) = w^{(-0.145)}$

(c) Journal fillet

The stress concentration factor for bending (β_B) is:

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

where:

$f_B(s,w) = -1.7625 + 2.9821 \cdot w - 1.5276\ w^2 + (1-s) \cdot (5.1169-5.8089 \cdot w+3.1391 \cdot w^2)$
$+ (1-s)^2 \cdot (-2.1567+2.3297 \cdot w-1.2952 \cdot w^2)$
$f_B(w) = 2.2422 \cdot w^{0.7548}$
$f_B(b) = 0.5616 + 0.1197 \cdot b + 0.1176 \cdot b^2$
$f_B(r) = 0.1908 \cdot r^{(-0.5568)}$
$f_B(d_G) = 1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$
$f_B(d_H) = 1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$
$f(recess) = 1 + (t_h+t_G) \cdot (1.8+3.2 \cdot s)$

The stress concentration factor for shearing (β_Q) is:

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

where:

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

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

$f_Q(b) = -0.5 + b$
$f_Q(r) = 0.5331 \cdot r^{(-0.2038)}$
$f_Q(d_H) = 0.9937 - 1.1949 \cdot d_H + 1.7373 \cdot d_H^2$
$f(recess) = 1 + (t_H+t_G) \cdot (1.8+3.2 \cdot s)$

The stress concentration factor for torsion (β_T) is:

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

where:

f(r,s), f(b) and f(w) are to be determined in accordance with 3.5.3(b) (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter:

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

3.5.4 Additional bending stresses

In addition to the alternating bending stresses in fillets (see 3.5.2(a)(iii)) 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 table:

Type of engine	σ_{add}(N/mm^2)
crosshead engines	± 30
trunk piston engines	± 10

3.5.5 Calculation of equivalent alternating stress

(a) General

The equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet. For this calculation the theory of constant energy of distortion (von Mises' Criterion) is to be used.

In this it is assumed that the maximum alternating bending stresses and maximum alternating torsional stresses within a crankshaft occur simultaneously and at the same point.

(b) Equivalent alternating stress

(i) For the crankpin fillet:

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

(ii) For the journal fillet:

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

where:

σ_v = Equivalent alternating stress (N/mm^2);

for other parameters see 3.5.2(a)(iii), 3.5.2(b)(iii) and 3.5.4.

3.5.6 Calculation of fatigue strength

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

(a) Related to the crankpin diameter:

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

(b) Related to the journal diameter:

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

where:

σ_{DW} = Allowable fatigue strength of crankshaft (N/mm^2);

K = Factor for different types of forged and cast crankshafts without surface treatment;

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

= 1.0 for free form forged crankshafts;

= 0.93 for cast steel crankshafts;

σ_B = Minimum tensile strength of crankshaft material (N/mm^2).

It is to be considered that for calculation purposes R_H and R_G are not to be taken less than 2 mm.

Where no results of the fatigue tests conducted on full size crank throws or crankshafts which have been subjected to surface treatment are available, the K-factors for crankshafts without surface treatment are to be used. In each case the experimental values of fatigue strength carried out with full size crank throws or crankshafts are subject to special consideration of the Society. The survival probability for fatigue strength values derived from testing is to be to the satisfaction of the Society and in principle not less than 80%.

3.5.7 Calculation of shrink-fits of semi-built crankshafts

(a) General

All crank dimensions necessary for the calculation of the shrink-fit are shown in Fig. III 3-6.

where:

D_S = Shrink diameter (mm).

L_S = Length of shrink-fit (mm).

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

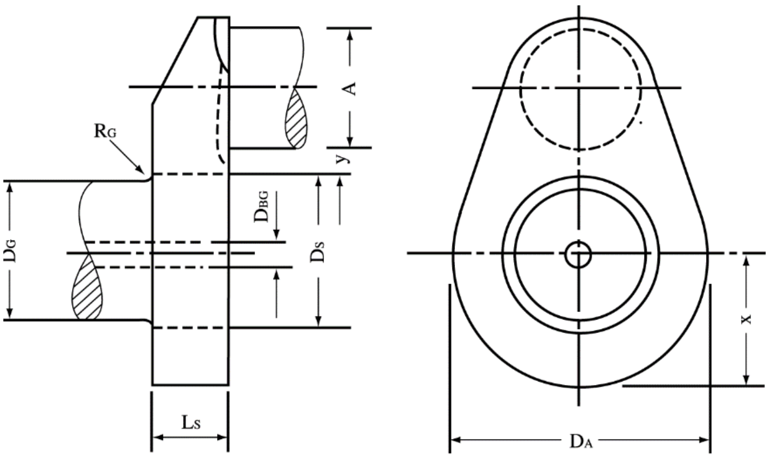


Fig. III 3-6
Crank Throw of Semi-built Crankshaft

y = distance between the adjacent generating lines of journal and pin (mm).

$y \geq 0.05 \cdot D_S$

Where y is less than $0.1 \cdot D_S$ special consideration is to be given to the effect of the stress due to the shrink on the fatigue strength at the crankpin fillet.

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

$R_G \geq 0.015\ D_G$ and $R_G \geq 0.5\ (D_S - D_G)$, where the greater value is to be considered.

The actual oversize Z of the shrink-fit must be within the limits Z_{min} and Z_{max} calculated in accordance with 3.5.7(b) and 3.5.7(c).

(b) Necessary minimum oversize of shrink-fit

The necessary minimum oversize is determined by the greater value calculated in accordance with items (i) and (ii) stated bellows:

(i) The calculation of the minimum oversize is to be carried out for the crank throw with the absolute maximum torque M_{max}. The torque M_{max} corresponds to the maximum value of the torque M_{Tmax} ascertained as per item 3.5.2(b) for the various mass points of the crankshaft.

$$Z_{min} \geq \frac{4 \bullet 10^3}{\pi \bullet \mu} \bullet \frac{S_R \bullet M_{max}}{E_m \bullet D_S \bullet L_S} \bullet \frac{1 - Q_A^2 \bullet Q_S^2}{(1 - Q_A^2) \bullet (1 - Q_S^2)}$$

with

$$Q_A = \frac{D_S}{D_A}, \quad Q_S = \frac{D_{BG}}{D_S}, \mu = 0{,}20 \quad \text{for} \quad \frac{L_S}{D_S} \geq 0.40$$

where:

Z_{min} = Minimum oversize (mm);

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

Q_A, Q_S= Ratio of different diameters;

μ = Coefficient for static friction;

E_m = Young's modulus (N/mm^2).

(ii) In addition to item (i) the minimum oversize is also to be calculated according to the following formula:

$$Z_{min} \geq \frac{\sigma_S \bullet D_S}{E_m}$$

where:

σ_S = Minimum yield strength of material for crank web (N/mm^2).

(c) Maximum permissible oversize of shrink-fit

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

$$Z_{max} \leq \frac{\sigma_S \bullet D_S}{E_m} + \frac{0.8 \bullet D_S}{1000}$$

where:

Z_{max} = Maximum oversize (mm).

This condition serves to restrict the shrinkage in the fillet.

3.5.8 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 both for the crankpin fillet and the journal fillet and is based on the formula:

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

where:

Q = Acceptability factor.

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

$Q \geq 1.15$

3.5.9 Fillets and oil holes

(a) Fillets at the junctions of crank webs with crank journals or pins, where they are formed as solid forgings or castings, are to be machined to a radius not less than 5% of the actual diameter of the journal or the pin and are to have a smooth finish.

(b) Oil holes at the surfaces of crank journals and pins are to be rounded to an even contour with a smooth finish.

3.5.10 Reference marks are to be provided on the outer junction of the crank webs with the crank pins and journals.

3.5.11 The part of the journal outside the end bearing may be gradually tapered to the same diameter as the adjacent shaft.

3.5.12 The diameter in way of flywheel or eccentric sheave for pump fitted on the crankshaft or additional shaft between the aftermost journal bearing and the thrust shaft is not to be less than that required for the crankshaft.

3.5.13 Coupling flanges

(a) The thickness of coupling flanges at the pitch circle of bolt holes is not to be less than the required diameter of the coupling bolts.

(b) The fillet between coupling flange and crankshaft is to have a radius not less than 8% of the diameter of the corresponding crankshaft provided that recesses are to be avoided in way of bolt heads and nuts.

(c) Where a coupling flange is separate from the crankshaft, the arrangement is to be submitted for consideration and provision is to be made for the coupling to resist any twisting force and astern pull.

3.6 Starting Arrangements

3.6.1 Air starting

Compressed air starting arrangements are to be in compliance with the requirements specified in 4.6 of Part VI of "Rules for the construction and classification of steel ships."

3.6.2 Electric starting

(a) Where main propelling and auxiliary engines are fitted with electric starters, at least 2 starting batteries are to be installed sufficient in their combined capacity without recharging to provide the consecutive starts, as required in 4.6.2 of Part VI of "Rules for the construction and classification of steel ships" for air starting, within 30 minutes.

(b) The connections to the starting batteries are to be such that the batteries can be used alternately. Two charging facilities are required for the starting batteries, one automatic device supplied from a charging dynamo on the engine, and another device, may be of manually, supplied from the ship's electric system. Each of the charging devices is to be able to recharge one battery completely within 6 hours.

(c) The starting battery is not to be used for any purpose other than starting and running the engine. If it is also used for other purposes, the battery capacity is to be increased accordingly and the circuits are to be completely separated from the starting system.

3.6.3 Starting for emergency generator sets (as the emergency source of power)

(a) Where emergency generator sets are fitted, they are to be capable of being started readily when cold.

(b) A hand starting is acceptable if it is demonstrated to be practicable. Where other means are provided for starting, they are, in general, to provide for not less than 3 consecutive starts in a period of 30 minutes without recourse to other source of power within the machinery space.

(c) Starting for emergency generator sets, see 11.5 of Part VII of "Rules for the construction and classification of steel ships."

3.7 Air Intake and Exhaust Arrangements

3.7.1 Scavenging air arrangements

(a) The 2-stroke cycle engine is to be provided with at least one scavenging blower, either of the reciprocating or the rotary type.

(b) Where one independent scavenging blower is arranged for a 2-stroke cycle engine, 2 prime movers driving alternately or 2 prime movers of same power arranging duplicately are to be provided. Each of the prime movers of the duplicate means is to have a sufficient capacity to operate the main propelling engine at the revolutions developing more than 1/2 of the maximum continuous output without trouble in case of failure of one of the prime movers.

3.7.2 Exhaust gas turbochargers

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

(b) Where the main propulsion engine can not be operated only with the exhaust driven turbochargers in case of starting or low speed range, an auxiliary scavenging air system is to be provided.

3.7.3 Exhaust arrangements

(a) Exhaust lines and silencers are to be effectively lagged or cooled where the surface temperature may exceed 220°C. Where the exhaust line or silencer is water cooled and the cooling water outlet can be closed, special arrangements are to be provided to prevent an excessive rise in pressure occurring within the cooling spaces.

(b) Silencers are to be fitted with easily accessible cleaning openings and with means to drain water.

(c) If several engine exhaust lines are conducted to a common silencer, special arrangement is to be made to prevent exhaust gas from entering the engine which is not being used.

(d) Where the engine exhaust line is conducted near the water line, efficient means are to be provided to prevent the water from entering the engine.

(e) Where necessary, the exhaust line is to be fitted with suitable draining arrangement and means to allow for expansion.

(f) The engine exhaust line and the boiler uptake are not to be connected except where the boiler is arranged to utilize waste heat from the engine. Where the engine exhaust gases pass through the oil-fired boiler, the exhaust line leading to the boiler is to be so arranged that boiler unburnt oil cannot enter the engine exhaust line. For this purpose, the changeover valve in the exhaust line is to be interlocked with the burner so that oil cannot be supplied to the burner while the valve is open.

(g) Oil carriers are to be provided with silencers or special spark arresters which will prevent soot particles from being discharged to the atmosphere as sparks.

(h) In 2-stroke cycle main engines fitted with exhaust gas turbo-chargers which operate on the impulse systems, provision is to be made to prevent broken piston rings entering the turbine casing and causing damage to blades and nozzle rings.

3.8 Fuel, Lubrication and Cooling Arrangements

3.8.1 Fuel oil arrangements

(a) For the arrangement of fuel oil, see 4.4 of Part VI of "Rules for the construction and classification of steel ships."

(b) All external high pressure fuel delivery lines between the high pressure fuel pumps and fuel injectors are to be protected with a jacketed piping system capable of containing fuel from a high pressure line failure. A jacketed pipe incorporates an outer pipe into which the high pressure fuel pipe is placed forming a permanent assembly. Metallic hose of approved design may be accepted as the outer pipe. The jacketed piping system is to include a means for collection of leakages and arrangements are to be provided for an alarm to be given of a fuel line failure. For existing ships the keel of which were laid on or before June 30, 1998 are to comply with the above requirements by 1st July 2003, except that a suitable enclosure recognized by the Society on engines having an maximum continuous output of 375 kW or less having fuel injection pumps serving more than one injector may be used as alternative to the jacketed piping system.

(c) All surfaces with temperatures above 220°C which may be impinged as a result of a fuel system failure are to be properly insulated.

(d) Oil fuel lines are to be screened or otherwise suitably protected to avoid as far as practicable oil spray or oil leakages onto hot surfaces, into machinery air intakes, or other sources of ignition. The number of joints in such piping systems is to be kept to a minimum.

(e) Suitable arrangements are to be made for draining any oil fuel leakage and for preventing contamination of lubricating oil by fuel oil. If flexible hoses are used for shielding purpose, these are to be of an approved type.

(f) On engines intended for service in unattended machinery spaces, the high pressure oil fuel injection piping is to be shielded irrespective of the bore of the cylinders.

(g) When in return piping the pulsation of pressure with peak to peak values exceed 2 N/mm^2, shielding of this piping is also required.

3.8.2 Lubricating oil arrangements

(a) Lubricating oil arrangements are to be in accordance with the requirements of 4.5 of Part VI of “Rules for the construction and classification of steel ships” in addition to the following.

(b) If enclosed crankcases are used as lubricating oil sumps, they are to be so arranged that the contained oil can be drained at any time and that purifiers or suitable filters for lubricating oil are provided.

(c) Lubricating oil lines are to be provided with pressure gauges or other adequate means at suitable positions to indicate that proper circulation is being maintained.

(d) Precautions are to be taken to ensure satisfactory lubrication of the scavenging blower and turbo-charger when starting and when running at reduced speeds. Lubricating oil from the drive and the bearing is not to be allowed to mix with the scavenging and turbocharging air.

(e) Main engines and auxiliary engines with maximum continuous output exceeding 375 kW are to be provided with alarm devices which give visible and audible alarming in the event of failure of supply of lubricating oil or appreciable reduction in lubricating oil pressure, and also with devices to stop the operation of the engine automatically by lower pressure after the function of alarms.

3.8.3 Cooling arrangements

(a) Cooling arrangements are to be in accordance with the requirements of 4.3 of Part VI of “Rules for the construction and classification of steel ships” in addition to the following.

(b) Discharge pipes for cooling water or cooling oil are to be provided with thermometers and preferably be fitted with adequate means to indicate the proper circulation.

(c) In engines having 2 or more cylinders, adequate means are to be provided to adjust the quantity of cooling water or cooling oil to make cooling uniformly for each cylinder and piston.

(d) Drain arrangements are to be provided on water jackets and cooling water lines at their lowest positions. Relief valves are to be fitted in the main lines to the jackets to release excessive pressure.

(e) Cooling water or cooling oil is to be discharged from the cooling spaces, where practicable, at its highest position.

3.9 Mass Produced Engines

3.9.1 Definition

(a) Mass produced engines, for main and auxiliary purposes, are defined as those which are produced under the following criteria:

(i) In quantity under strict quality control of material and parts, according to a quality assurance scheme acceptable to the Society.

(ii) By the use of jigs and automatic machine tools designed to machine parts to specified tolerances for interchangeability, and which are verified on a regular inspection basis.

(iii) By assembly with parts taken from stock and requiring little or no fitting.

(iv) With bench tests carried out on individual assembled engines according to a specified programme.

(v) With appraisal by final examination of engines selected at random after workshop testing.

(b) Castings, forgings and other parts for use in mass produced engines are also to be produced by methods similar to those given in 3.9.1(a)(i), (ii) and (iii), with appropriate inspection.

(c) Hydraulic testing of components is to comply with Table III 3-3.

(d) The specification of a mass produced engine is to define the limits of manufacture of all component parts. The total production output is to be certified by the manufacturer and verified as may be required, by the Society in accordance with the agreed manufacturer's quality assurance scheme, see 3.9.1(a)(i).

3.9.2 Procedure for approval of mass produced engines

(a) The procedure outlined in 3.9.2(b) to 3.9.2(e) applies to the inspection and certification of mass produced oil engines having a bore not exceeding 300 mm.

(b) For the approval of a mass produced engine type, the manufacturer is to submit a list of subcontractors for main parts.

(c) The manufacturer is to supply full information regarding the manufacturing processes and quality control procedures applied in the workshops. The information is to address the following:

(i) Organization of quality control systems.

(ii) Recording of quality control operations.

(iii) Qualification and independence of personnel in charge of quality control.

(d) A running type test of at least 100 hours duration is to be carried out on an engine chosen from the production line. The type testing is to comply with 3.9.5.

(e) Reserves the right to limit the duration of validity of approval of a mass produced engine. The Society is to be informed, without delay, of any change in the design of the engine, in the manufacturing or control processes, in the selection of materials or in the list of subcontractors for main parts.

3.9.3 Continuous review of production

(a) The Society’s Surveyors are to be provided free access to the manufacturer's workshops and to the quality control files.

(b) The control of production, which is subject to survey, is to include the following:

(i) Inspection and testing records are to be maintained to the satisfaction of the Surveyor.

(ii) The system for identification of parts is to be in accordance with recognized practice, and acceptable to the Society.

(iii) The manufacturer is to provide full information about the quality control of the parts supplied by subcontractors for which certification may be required. The Society reserves the right to apply direct and individual inspection procedures for parts supplied by subcontractors when deemed necessary.

(iv) At the request of an attending Surveyor, a workshop test may be required for an individual engine.

3.9.4 Compliance and inspection certificate

(a) Each engine which is to be installed on a ship classed by the Society is to be supplied with a statement certifying that the engine is identical to the one which underwent the tests specified in 3.9.2(d), and state the test and inspection results. The statement is to be made on a form agreed with the Society. Each statement is to include the identification number which appears on the engine. A copy of this statement is to be submitted to the Society.

3.9.5 Type test conditions

(a) The requirements in this section are applicable to the type testing of mass produced internal combustion engines where the manufacturer has requested approval. Omission or simplification of the type test requirements will be considered by the Society for engines of an established type on application by the manufacturer.

(b) The engine to be tested is to be selected from the production line and agreed by the Society.

(c) The duration and programme of type tests is to include the following:

(i) 80 h at rated output.

(ii) 8 h at 110 per cent overload.

(iii) 10 h at varying partial loads (25 per cent, 50 per cent, 75 per cent and 90 per cent of rated output).

(iv) 2 h at maximum intermittent loads.

(v) Starting tests.

(vi) Reverse running of direct reversing engines.

(vii) Testing of speed governor.

(viii) Testing of over-speed device.

(ix) Testing of lubricating oil system failure alarm device.

(x) Testing of the engine with turbocharger out of action, when applicable.

(xi) Testing of minimum speed for main propulsion engines and the idling speed for auxiliary engines.

(d) The type tests in 3.9.5(c) at the required outputs are to be combined together in working cycles for the whole duration within the limits indicated. See also 3.9.5(j) and 3.9.5(k).

(e) The overload testing required by 3.9.5(c) is to be carried out with the following conditions:

(i) 110 per cent of rated power at 103 per cent revolutions per minute for engines directly driving propellers.

(ii) 110 per cent of rated power at 100 per cent revolutions per minute for engines driving electrical generators or for other auxiliary purposes.

(f) For prototype engines, the duration and programme of tests are to be specially agreed between the manufacturer and the Society.

(g) As far as practicable during type testing the following particulars are to be continuously recorded:

(i) Ambient air temperature.

(ii) Ambient air pressure.

(iii) Atmospheric humidity.

(iv) External cooling water temperature.

(v) Fuel and lubrication oil characteristics.

(h) In addition to the particulars stated in 3.9.5(g) and as far as practicable, the following are also to be continuously measured and recorded:

(i) Engine revolutions per minute.

(ii) Brake power.

(iii) Torque.

(iv) Maximum combustion pressure.

(v) Indicator pressure diagrams where practicable.

(vi) Exhaust smoke (with an approved smoke meter).

(vii) Lubricating oil pressure and temperature.

(viii) Exhaust gas temperature in exhaust manifold, and, where facilities are available, from each cylinder.

(ix) For turbocharged engines:

(1) Turbocharger revolutions per minute.

(2) Air temperature and pressures before and after turbo-blower and charge cooler.

(3) Exhaust gas temperature and pressures before and after the turbine.

(4) The cooling water inlet temperature to the charge air cooler.

(i) After the type test, the main parts and especially those subject to wear are to be dismantled for examination by the Society's Surveyors.

(j) For engines that are required to be approved for different purposes (multi-purpose engines), and that have different performances for each purpose, the programme and duration of test is to be modified to cover the whole range of the engine performance, taking into account the most severe conditions and intended purpose(s).

(k) The rated output for which the engine is to be tested is the output corresponding to that declared by the manufacturer and agreed by the Society, i.e. actual maximum power which the engine is capable of delivering continuously between the normal maintenance intervals stated by the manufacturer at the rated speed and under the stated ambient conditions.

3.10 Tests and Inspections

3.10.1 Hydraulic pressure tests on diesel engine parts after machining are to be carried out in the presence of the Surveyor under the conditions specified in Table III 3-3.

3.10.2 Material and non-destructive tests required on diesel engine parts.

(a) Materials intended for the principal components of diesel engines and their non-destructive test are to conform to the requirements given in Table III 3-4.

(b) All required material tests are to be witnessed in the presence of the Society's Surveyor.

(c) For important structural parts of engines, examination of welded seams by approved methods of inspection may be required.

(d) In addition to tests mentioned above, where there is evidence to doubt the soundness of any engine component, non-destructive test by approved detecting methods may be required.

3.10.3 Engine alignment

The crankshaft alignment is to be checked each time the engine is lined up. This is at least to include measurement of the crank web deflections at each crank.

Table III 3-3
Hydraulic Tests and Test Pressures Required on Diesel Engine Parts

Parts to be Tested		Test Pressure, MPa
Cylinder cover, cooling space. Cylinder liner, over the whole of cooling space. Piston crown, cooling space, after assembling with piston rod.		0.7

Cylinder jacket, cooling space.
Exhaust system:
  Exhaust valve, cooling space.
  Exhaust line, cooling space.
Scavenging and turbo-charging system:
  Pump cylinder,
  Blower, cooling space.
  Exhaust gas turbine casing, cooling space.
Cooler, each side.
		0.4 but not less than 1.5W

Fuel injection system:
  Pump body, pressure side.
  Valve.
  Pipe.
		1.5W or W + 29.5
		whichever is smaller and see Note.

Hydraulic system: High pressure piping for hydraulic drive of exhaust gas valve		1.5W
Engine driven air compressor (cylinders, covers, intercoolers and after-coolers)	Air side	1.5W
	Water side	0.4 but not less than 1.5W
Engine driven pumps (oil, water, fuel, bilge)		0.4 but not less than 1.5W
Piping system		Apply the requirements in Chapter 7 of Part VI of Rules for the construction and classification of steel ships
Where: W=Maximum working pressure for the respective parts, in MPa.		

Note: Where fuel injection pumps or valve bodies are made of forged steel, hydraulic pressure tests may be omitted.

Table III 3-4
Application of Materials and Non-destructive Tests to Principal Components of Diesel Engines

Principal components			Cylinder bore D (mm)								
			D ≤ 300			300 < D ≤ 400			400 < D		
			①	②	③	①	②	③	①	②	③
1	Chankshaft	Solid forged type	O	O	O	O	O	O	O	O	O
		Web, pin and journal of built-up or semi-built-up type	O	O	O	O	O	O	O	O	O
		Others (for example welded type)	O	O	O	O	O	O	O	O	O
2	Coupling flanges on crankshaft (if not integral)								O		
3	Coupling bolts for crankshaft								O		
4	Steel piston crowns				O			O	O	O	O
5	Piston rods		O	O		O	O		O	O	O
6	Connecting rods together with connecting rod bearing caps		O	O		O	O		O	O	O
7	Steel parts of cylinder liners					O			O		
8	Steel cylinder covers				O	O		O	O	O	O
9	Bed-plates of welded construction	Plates and transverse bearing girders made of forged or cast steel	O			O			O		
		Cast steel parts including welded joints		O	O		O	O		O	O
10	Thrust blocks of welded construction, plates and transverse bearing girders made of forged or cast steel		O			O			O		
11	Frames and crankcases of welded construction		O			O			O		
12	Entablatures of welded construction		O			O			O		
13	Tie rods		O	O		O	O		O	O	
14	Steel gear wheels for camshaft drive								O	O	
15	Bolts and studs (for cylinder covers, crossheads, connecting rod bearings, main bearings)					O			O	O	
16	Turbine discs, blades, blower impellers and rotor shafts of exhaust driven turboblower		O	O	O	O	O	O	O	O	O
17	Crossheads								O		
18	Pipes, valves and fittings attached to engine classified in Group I or Group II in Chapter 1 of Part VI of Rules for the construction and classification of steel ships		O			O			O		

Notes:

1. Materials intended for the components marked with circlets in Column ① are to comply with the requirements in Part XI of Rules for the construction and classification of steel ships.
2. Materials intended for the components marked with circlets in Column ② are to be tested by magnetic particle test or liquid penetrant test.
3. Materials intended for the components marked with circlets in Column ③ are to be tested by ultrasonic test.

3.10.4 Programme for works trials

Engines, which are to be subjected to trials on the test bed at the manufacturer's works and the Society's supervision, are to be tested in accordance with the scope as specified below.

(a) Scope of works trials

For all stages, the engine is going to be tested, the pertaining operation values are to be measured and recorded by the engine manufacturer. In each case all measurements conducted at the various load points are to 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.

(i) Main engines driving propellers

(1) 100% power (rated power) at rated engine speed (n_o):
at least 60 min – after having reached steady conditions.

(2) 110% power at engine speed n = 1.032 n_o:
30-45 min. – after having reached steady conditions.

(3) 90% (or normal continuous cruise power), 75%, 50% and 25% power in accordance with the nominal propeller curve.

(4) Starting and reversing maneuvers.

(5) Testing of governor and independent overspeed protective device.

(6) Shut down device.

(ii) Main engines driving generators for propulsion

The test is to be performed at rated speed with a constant governor setting under conditions of:

(1) 100% power (rated power) at rated engine speed:
at least 50 min – after having reached steady conditions.

(2) 110% power:
30 min – after having reached steady conditions.

(3) 75%, 50% and 25% power and idle run.

(4) Start-up tests.

(5) Testing of governor and independent overspeed protective device.

(6) Shut-down device.

(iii) Engines driving auxiliaries

Test to be performed in accordance with 3.10.4(a)(ii).

(b) Inspection of components

Random checks of components to be presented for inspection after the works trials.

(c) Parameters to be measured

The data to be measured and recorded, when testing the engine at various load points, are to include all necessary parameters for the engine operation. The crankshaft deflection is to be checked when this check is required by the manufacturer during the operating life of the engine.

(d) In addition the scope of the trials may be expanded depending on the engine application.

3.10.5 Programme for Shipboard trials (dock and sea trials)

After the conclusion of the running-in programme, prescribed by the engine manufacturer, engines are to undergo the trials as specified below:

(a) Scope of sea trials

(i) Main propulsion engines driving fixed propellers

(1) At rated engine speed n_o : at least 2 hours.

(2) At engine speed n = 1.032•n_o: 30 minutes.

(3) At minimum on-load speed.

(4) Starting and reversing maneuvers.

(5) In reverse direction of propeller rotation during the dock or sea trials at a minimum engine speed of n = 0.7•n_o: 10 minutes.

(6) Monitoring, alarm and safety systems.

(ii) Main propulsion engines driving controllable pitch propellers or reversing gears

Controllable pitch propellers are to be tested with various propeller pitches.

(iii) Main engines driving generators for propulsion

The tests to be performed at rated speed with a constant governor setting under conditions of:

(1) 100% power (rated power): at least 2 hours.

(2) 110% power: 30 minutes.

(3) In reverse direction of propeller rotation during the dock or sea trials at a minimum speed of 70% of the nominal propeller speed: 10 minutes

(4) Starting maneuvers.

(5) Monitoring, alarm and safety systems.

NOTE:

Tests are to be based on the rated electrical powers of the driven generators.

(iv) Engines driving auxiliaries

Engines driving generators or important auxiliaries are to be subjected to an operational test for at least 2 hours. During the test, the set concerned is required to operate at its rated power for an extended period. It is to be demonstrated that the engine is capable of supplying 100% of its rated power, and in the case of shipboard generating sets account is to be taken of the times needed to actuate the generator's overload protection system.

(v) The suitability of engine burn residual or other special fuels is to be demonstrated, if machinery installation is arranged to burn such fuels.

(b) In addition the scope of the trials may be expanded in consideration of the special operating conditions, such as towing, trawling etc.

Chapter 4

Deck Machinery and Essential Auxiliaries

4.1 General

4.1.1 The requirements of this chapter are applicable to the steering gear, athwartship thruster, windlass, mooring winch, capstan, reciprocating compressor and essential service pumps, etc.

4.1.2 Athwartship thrust units, not including azimuth thrusters intended for main propulsion, are regarded as part of the steering function. The requirements dealing with steering gear, shafting, gearing and couplings, propeller, remote control systems as well as electric installations in the relevant parts of the Rules, as far as they are applicable, are to be complied with for the athwartship thrust units.

4.1.3 The requirements in 4.3 of this Part are to be complied with, as far as they are applicable, for the mooring winch and capstan.

4.1.4 Drawings and data

The manufacturers are to submit the following drawings together with data for approval:

(a) For steering gear:

(i) Drawings

(1) General arrangements of steering gear

(2) Details of tiller, etc.

(3) Assembly and details of power units

(4) Assembly and details of rudder actuators

(5) Piping diagram of hydraulic pipes; Arrangements of control systems

(6) Diagram of hydraulic and electrical systems
(including alarm devices and automatic steering gear)

(7) Arrangements and diagram of an alternative source of power

(8) Diagram of a rudder angle indicator

(9) Other drawing considered necessary by the Society

(10) At least four copies of the plans and information are to be submitted.

(ii) Data

(1) Particulars

(2) Operating instructions (including drawing showing the change-over procedure for power units and control systems, drawings showing the sequence of automatic supply of power from an alternative source of power, data showing the kind, particulars and an assembly of the power source in the case that the alternative source of power is an independent source of power and information about hydraulic fluid quality)

(3) Manuals for countermeasures to be taken at the time of a single failure of the power actuating system;

(4) Calculation sheet of the strength of essential parts.

(5) Other data considered necessary by the society.

(b) For windlass, mooring winch and capstan

General arrangement, detail drawings of the shaft, cable lifting wildcat for windlass and brake, necessary particulars for capacity, speed, driving power, and material specifications, etc.

(c) For reciprocating compressor

(i) The compressor manufacturers are to submit the following drawings for approval:

Sectional assembly and crank shafts.

(ii) The compressor manufacturers are to submit the following data together with drawings for approval:

Design pressures and temperatures of air in different stages, capacity, revolutions, particulars, power required and material specifications, etc.

(d) For pumps

General arrangement including capacity, head, revolutions and driving power, description of pump including overload protection and other safety devices, and material specifications, etc.

4.1.5 Materials

(a) For steering gear

(i) All the steering gear components and the rudder stock are to be of sound reliable construction to the Surveyor’s satisfaction.

(ii) All components transmitting mechanical forces to the rudder stock are to be tested according to the requirements of Part XI of “Rules for the construction and classification of steel ships.”

(iii) Ram cylinders, pressure housings of rotary vane type actuators, hydraulic power piping, valves, flanges and fittings; and all steering gear components transmitting mechanical forces to the rudder stock (such as tillers, quadrants, or similar components) are to be of steel or other approved ductile material, duly tested in accordance with the requirements of Part XI of “Rules for the construction and classification of steel ships”. In general, such material is to have an elongation of not less than 12 per cent nor a tensile strength in excess of 650 N/mm^2. Special consideration will be given to the acceptance of grey cast iron for valve bodies and redundant parts with low stress levels.

(b) For windlasses, mooring winches and capstans, the material used is generally to comply with 4.1.5(a) above. The cable lifting wildcats generally to be of cast steel.

(c) For reciprocating compressor

(i) Crankshaft and connecting rods are normally to be of forged steel, cast steel or approved spheroidal or nodular graphite cast iron. The use of other material is to be specially approved in each case by the Society.

(ii) Materials intended for crankshafts and the spheroidal or nodular graphite cast iron connecting rods are to be tested and inspected in the presence of the Surveyor in accordance with the requirements of Part XI of “Rules for the construction and classification of steel ships” or to the requirements of the specifications approved in connection with the design.

(d) For pumps which handle corrosive fluids, noncorrodible materials are to be used. Cast iron pump bodies are not to be used for circulating pumps of forced circulation boilers for which the design pressure exceeds 1.0 MPa.

(e) Materials intended for the following component parts are to be tested and inspected in the presence of the Surveyor in accordance with the requirements of Part XI of “Rules for the construction and classification of steel ships” or to the requirements of the specifications approved in connection with the design.

(i) For athwartship thrusters, windlass, mooring winches, capstans and pumps requiring a driving power of 375 kW and more: shafts.

(ii) For piping: group-I and -II pipes and valves as classified in Part VI of "Rules for the construction and classification of steel ships."

4.2 Steering Gears

4.2.1 General

(a) Each ship is to be provided with a main steering gear and an auxiliary steering gear in accordance with the requirements of the Rules. The main steering gear and the auxiliary steering gear are to be so arranged that the failure of one of them will not render the other one inoperative.

(b) Whilst the requirements satisfy the relevant regulations of the International Convention for the Safety of Life at Sea 1974 (SOLAS 1974) as amended, attention is to be given to any relevant statutory requirements of the National Authority of the country in which the ship is registered.

(c) Consideration will be given to other cases, or to arrangements which are equivalent to those required by the Rules.

(d) Where the steering gear is so arranged that more than one system (either power or control) can be simultaneously operated, the risk of hydraulic locking caused by single failure is to be considered.

(e) Definitions relating to steering gear

(i) Steering gear control system means the equipment by which orders are transmitted from the navigating bridge to the steering gear power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables.

(ii) Main steering gear means the machinery, rudder actuator(s), the steering gear power units, if any, and ancillary equipment and the means of applying torque to the rudder stock (e.g. tiller or quadrant) necessary for effecting movement of the rudder for the purpose of steering the ship under normal service conditions.

(iii) Steering gear power unit means:

(1) in the case of electric steering gear, an electric motor and its associated electrical equipment,

(2) in the case of electrohydraulic steering gear, an electric motor and its associated electrical equipment and connected pump,

(3) in the case of other hydraulic steering gear, a driving engine and connected pump.

(iv) Auxiliary steering gear means the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not including the tiller, quadrant of components serving the same purpose.

(v) Power actuating system means the hydraulic equipment provided for supplying power to turn the rudder stock, comprising a steering gear power unit or units, together with the associated pipes and fittings, and a rudder actuator. The power actuating systems may share common mechanical components, i.e. tiller, quadrant and rudder stock, or components serving the same purpose.

(vi) Maximum ahead service speed means the greatest speed which the ship is designed to maintain in service at sea at her deepest sea going draught at maximum propeller RPM and corresponding engine MCR.

(vii) Rudder actuator means the component which converts directly hydraulic pressure into mechanical action to move the rudder.

(viii) Maximum working pressure means the maximum expected pressure in the system when the steering gear is operated to comply with 4.2.2(b).

4.2.2. Main steering gear

(a) The main steering gear and rudder stock are to be of adequate strength and capable of steering the ship at maximum ahead service speed, and are also to be designed that they will not be damaged at maximum astern speed. However, this design requirement need not be proved by trials at maximum astern speed and maximum rudder angle.

(b) The main steering gear is to be capable of putting the rudder over from 35° on one side to 35° on the other side with the ship at its deepest seagoing draught and running ahead at maximum ahead service speed and, under the same conditions, from 35° on either side to 30° on the other side in not more than 28 seconds.

(c) The main steering gear is to be operated by power where necessary to meet the requirements of 4.2.2(b) above and in any case when the Rules required a rudder stock over 120 mm in diameter in way of the tiller, excluding strengthening for navigation in ice.

4.2.3 Auxiliary steering gear

(a) The auxiliary steering gear is to be of adequate strength and capable of steering the ship at navigable speed and of being brought speedily into action in an emergency.

(b) The auxiliary steering gear is to be capable of putting the rudder over from 15° on one side to 15° on the other side in not more than 60 seconds with the ship at its deepest seagoing draught and running ahead at 1/2 of the maximum ahead service speed or 7 knots, whichever is the greater.

(c) The auxiliary steering gear is to be operated by power where necessary to meet the requirements of 4.2.3(b) above and in any case when the Rules require a rudder stock of over 230 mm in diameter in way of the tiller, excluding strengthening for navigation in ice.

4.2.4 Where the main steering gear comprises 2 or more identical power units, an auxiliary steering gear need not be fitted, provided that:

(a) In a passenger ship, the main steering gear is capable of operating the rudder as required by 4.2.2(b) of this Part while any one of the power units is out of operation.

(b) In a cargo ship, the main steering gear is capable of operating the rudder as required by 4.2.2(b) of this Part while operating with all power units.

(c) The main steering gear is so arranged that after a single failure in its piping system or in one of the power units the defect can be isolated so that steering capability can be maintained or speedily regained.

4.2.5 Power units

Main and auxiliary steering gear power units are to be arranged to re-start automatically when power is restored after a power failure and capable of being brought into operation from a position on the navigating bridge. In the event of a power failure to any one of the steering gear power units, an audible and visual alarm is to be given on the navigating bridge. These power units are also to be arranged so that transfer between units can be readily effected.

4.2.6 Hand steering gear

Hand steering gears are only acceptable when the force required to operate the gear does not exceed 155N under normal condition, and are to be so designed that the rudder can be moved from hardover to hardover with a maximum of 25 turns of the hand steering wheel.

4.2.7 Rudder stoppers

The steering gear is to be arranged with the adequate stoppers within the gear itself to limit the movement of the rudder in normal service, otherwise the movements of quadrants and tillers are to be limited by structural stoppers arranged on the deck. Power operated steering gears are to be provided with positive arrangements for stopping the gear before the rudder stops are reached. These arrangements are to be synchronized with the gear itself and not with the steering gear control.

4.2.8 Steering gear control systems

(a) A steering gear control system is to be provided:
- (i) for main steering gear, both on the navigating bridge and in the steering gear compartment;
- (ii) where the main steering gear is arranged in accordance with 4.2.4 of this Part, by 2 independent control systems, both operable from the navigating bridge. This does not require duplication of the steering wheel or steering lever. Where the control system consists of a hydraulic telemotor, a second independent system need not be fitted, except in a tanker, chemical tanker or gas carrier of 10,000 tons gross and upwards, and
- (iii) for the auxiliary steering gear, in the steering gear compartment and, if power operated, it is also to be operable from the navigating bridge and is to be independent of the control system for the main steering gear.

(b) Any main and auxiliary steering gear control system operable from the navigating bridge is to comply with the following requirements:
- (i) if electric, it is to be served by its own separate circuit supplied from the associated steering gear power circuit from a point within the steering gear compartment, or directly from the same section of switchboard busbars, main or emergency, to which the associated steering gear power circuit is connected;
- (ii) means are to be provided in the steering compartment for disconnecting any control system operable from the navigating bridge from the steering gear it serves;
- (iii) the system is to be capable of being brought into operation from a position on the navigating bridge;
- (iv) in the event of a failure of electrical power supply to the steering gear control system, an audible and visible alarm is to be given on the navigating bridge; and
- (v) each separate supply circuit of steering gear control is to be provided with short circuit protection only.

(c) Where the arrangement is such that a single failure may cause hydraulic lock and loss of steering, an audible and visual alarm which identifies the failed system or component is to be provided on the navigating bridge. The alarm is to be activated upon steering gear failure if:
- (i) position of the variable displacement pump control system does not correspond to the given order, or
- (ii) incorrect position of 3-way full follow valve or similar in constant delivery pump system is detected.

4.2.9 Rudder angle indicators
The angular position of the rudder is to be indicated in the navigating bridge, if the main steering gear is power operated. The rudder angle indication is to be independent of the steering gear control system, and readily visible from the control position. In addition, the angular position of the rudder is to be recognizable in the steering gear compartment.

4.2.10 Electrical equipment

(a) Electric power circuits and protective devices for steering gear is to comply with the requirements in 2.3 of Part VII of “Rules for the construction and classification of steel ships.”

(b) Indicators for running indication of each main and auxiliary motor of steering gear power units are to be installed on the navigating bridge and at a suitable main machinery control position.

(c) Electric control systems are to be independent and separated as far as is practicable throughout their length.

4.2.11 Components and piping

(a) Special consideration is to be given to the suitability of any essential component which is not duplicated. Where appropriate, any such essential component is to utilize anti-friction bearings which are permanently lubricated or provided with lubrication fittings.

(b) All steering gear components transmitting mechanical forces to the rudder stock, which are not protected against overload by structural rudder stoppers or mechanical buffers, are to have a strength at least equivalent to that of the rudder stock in way of the tiller.

(c) Piping, joints, valves, flanges and other fittings are to comply with the requirements of Part VI of "Rules for the construction and classification of steel ships". The design pressure for calculations to determine the scantlings of piping and other steering gear components subjected to internal hydraulic pressure is to be at least 1.25 times the maximum working pressure to be expected under the operational condition specified in 4.2.2(b) of this Part, taking into account any pressure which may exist in the low pressure side of the system. Fatigue criteria may be applied for the design of piping and components, taking into account pulsating pressures due to dynamic loads.

(d) Relief valves are to be fitted to any part of the hydraulic system which can be isolated and in which pressure can be generated from the power source of external forces. The setting of the relief valves is not to exceed the design pressure. The valves are to be of adequate size and so arranged as to avoid an undue rise in pressure above the design pressure.

(e) Relief valves for protecting any part of the hydraulic system which can be isolated, as required by 4.2.11(d) above are to comply with the following:

(i) The setting pressure is not to be less than 1.25 times the maximum working pressure.

(ii) The minimum discharge capacity of the relief valves(s) is not to be less than 110% of the total capacity of the pumps which can deliver through it (them). Under such conditions the rise in pressure is not to exceed 10% of the setting pressure. In this regard, due consideration is to be given to extreme foreseen ambient conditions in respect of oil viscosity.

(f) Flexible hoses

Hose assemblies approved by the Society may be installed between 2 points where flexibility is required but are not to be subjected to torsional deflection (twisting) under normal operating conditions. In general, the hose is to be limited to the length necessary to provide for flexibility and for proper operation of machinery.

(g) Hydraulic power operated steering gear are to be provided with:

(i) arrangement to maintain the cleanliness of the hydraulic fluid taking into consideration the type and design of the hydraulic system;

(ii) a low level alarm for each hydraulic fluid tank to give the earliest practicable indication of hydraulic fluid leakage. Audible and visual alarm are to be given on the navigating bridge and in the machinery space where they can be readily observed, and

(iii) a fixed storage tank having sufficient capacity to recharge at least one power actuating system including the hydraulic fluid tank, where the main steering gear is required to be power operated. The storage tank is to be permanently connected by piping in such a manner that the hydraulic system can be readily recharged from a position within the steering gear compartment and provided with a contents gauge.

(h) For the steering gears which are so arranged that more than one system (either power or control) can be simultaneously operated, where hydraulic locking, caused by a single failure, may lead to loss of steering, an audible and visual alarm, which identifies the failed system, is to be provided on navigation bridge.

4.2.12 Display of operating instructions

Appropriate operating instructions with a block diagram showing the change-over procedures for steering gear control systems and steering gear acting systems are to be permanently displayed in the navigating bridge and in the steering gear compartment.

4.2.13 Additional requirements for tankers, chemical tankers or gas carriers of 10,000 tons gross and upwards and every other ship of 70,000 tons gross and upwards are as follows:

The main steering gear is to comprise 2 or more identical power units complying with the requirements in 4.2.4 of this Part.

4.2.14 Additional requirements for tankers, chemical tankers or gas carriers of 10,000 tons gross and upwards are as follows:

Subject to the requirements in 4.2.15 of this Part the following are to be complied with:

(a) The main steering gear is to be so arranged that in the event of loss of steering capability due to a single failure in any part of one of the power actuating systems of the main steering gear, excluding 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 after the loss of one power actuating system.

(b) The main steering gear is to comprise either:

(i) 2 independent and separate power actuating systems, each capable of meeting the requirements in 4.2.2(b) of this Part, or

(ii) at least 2 identical power actuating systems which, acting simultaneously in normal operation, are capable of meeting the requirements in 4.2.2(b) of this Part. Where necessary to comply with these requirements, inter-connection of hydraulic power actuating systems is to 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 remain fully operational.

(c) Steering gears other than of the hydraulic type are to achieve equivalent standards.

4.2.15 Additional requirements for tankers, chemical tankers or gas carriers of 10,000 gross tonnage and upwards but of less than 100,000 tons deadweight are as follows:

(a) Solutions other than those set out in 4.2.14 of this Part which need not apply the single failure criterion to the rudder actuator or actuators, may be permitted provided that an equivalent safety standard is achieved and that:

(i) following loss of steering capability due to a single failure of any part of the piping system or in one for the power units, steering capability is regained within 45 seconds, and

(ii) where the steering gear includes only a single rudder actuator, special consideration is given to stress analysis for the design including fatigue analysis and fracture mechanics analysis, as appropriate, the material used, the installation of sealing arrangements and the testing and inspection and provision of effective maintenance.

4.2.16 The steering unit is to be secured to the seating by fitted bolts, and suitable chocking arrangements are to be provided. The seating is to be of substantial construction.

4.2.17 Arrangements for bleeding air from the hydraulic system are to be provided, where necessary.

4.2.18 If steering systems can also be operated from other positions, then two-way communication is to be arranged between the control station and these other positions.

4.3 Windlass

4.3.1 General

A windlass of sufficient power and suitable for the size of chain cable is to be fitted to the ship to operate the anchors.

4.3.2 Definitions

(a) "Working Load": The working load, derived from the nominal diameter and the grade of anchor chain cables, is the tensile force exerted upon the cable lifter in the tangential direction when the anchor and anchor chain cables are being hoisted.

(b) "Overload Pull": The necessary temporary overload capacity of the windlass.

(c) "Holding Load": The maximum static load on the anchor chain cables which the cable lifter brake is to withstand.

(d) "Nominal Speed": The average speed of recovery of 55 m (two lengths) of anchor chain cables when 82.5m (three lengths) of the cables are submerged and freely suspended at commencement of lifting.

4.3.3 Performance

The windlass is to have the following performance:

(a) The windlass is to be capable of continuous operation for a period of 30 minute under the working load and also be capable of operating under the overload for a period of 2 minutes at reduced speed.

(b) The working load of the windlass is to be based on the following values:
- (i) Grade E1 anchor chain cable: 37.5 d^2 (N).
- (ii) Grade E2 anchor chain cable: 42.5 d^2 (N).
- (iii) Grade E3 anchor chain cable: 47.5 d^2 (N).

where

d = Nominal diameter of anchor chain cable(mm).

(c) The overload is to be 150% of the working load.

(d) The holding load is to be as follows:
- (i) with cable stopper: 0.45 $\times$ breaking test load of cable;
- (ii) without cable stopper: 0.8 $\times$ breaking test load of cable.

(e) The holding load of control brake is to be 130% of the working load when equipped with the control brake.

(f) The rated hoisting speed is to be 0.15 m/s or more. The conditions in this case are such that the anchor and the anchor chain cables are used where the efficiency of the hawse pipe is 70%.

4.3.4 Construction

(a) The windlass is to be so designed as to endure impact of waves and to ensure smooth operation of the components. The closed portions of the windlass installed on exposed decks are to have suitable watertight construction.

(b) The cable lifters for ordinary and couple type windlasses are to be so constructed that they can be driven independently and simultaneously.

(c) The cable lifter is to be as follows:
- (i) The cable lifter is to be provided with five teeth at minimum, and the angle of contact of the anchor chain cable to the cable lifter is to be at least 110°.
- (ii) The cable lifter is to be de-clutchable from the drive. Power operated clutches are also to be de-clutchable by hand.

(d) The cable lifter brake gear is to be as follows:
- (i) Electric windlass is to be provided with an automatic control brake system which operates when the control handle is in the "off" position or when the power supply is cut off. The automatic control brake system is to be capable of sustaining the holding load of control brake.
- (ii) Each cable lifter is to be fitted with a hand-brake which may be remotely controlled, capable of sustaining the holding load.

(e) Each remotely controlled windlass is to be fitted with a quick acting local emergency stop mechanism, which, when came into action, cuts off power for the windlass and applies the control brake system.

(f) Prime movers and gearing are to be provided with protective devices and safety devices against excessive torque and shock, as follows:
- (i) Overpressure preventive device for hydraulic equipment.
- (ii) Slipping clutch between electric motor and reduction gear.
- (iii) Protective device for the electric motor against overload.
- (iv) Covers for open gears.
- (v) Cover for preventing the operator from burning by touching the steam cylinder at excessively high temperature.
- (vi) Cover for crank disc.

(g) The speed of the rotation of the cable lifter is to be controllable.

4.3.5 Strength

(a) The parts for the windlass such as the bedplate, cable lifter, cable lifter shaft, bearing frame, brake gear, holding-down bolt, etc. are to have such strength that the stress on these parts is below the yield points of the materials when sustaining the holding load on the cable lifter.

(b) The driving section is to have such strength that the stress on each part is below 40% of the yield points of the materials used when the working load is applied.

4.3.6 Tests and Inspection

The windlass and cable lifter unit are to be tested as follows and the test results are to be recorded.

(a) "No-load Test": The windlass is to be run without load once in normal and once in reverse directions, for a sum of 30 minutes, under the rated voltage at the speed of rotation equivalent to the rated speed. When the windlass is provided with a gear change, an additional 5 min similar test is to be carried out for each additional gear change.

During the test, the following items are to be checked or measured:

(i) tightness against oil leakage;

(ii) temperature of bearings;

(iii) pressure of abnormal noise.

(b) "Load Test": The windlass is to, as a rule, be checked to verify that the working load, rated speed and overload pull are attainable as specified.

(c) "Cable Lifter Brake Test": The holding power of the cable lifter brake is to be verified.

The cable lifter brake is also to be tested with the anchor dropping, operated onboard with the holding load controlled and sustained by applying the brake at each half length of the chain.

(d) Performance Tests

(i) When provided with the remote control or other special device, their performances are to be verified.

(ii) The function of the automatic control brake system for electric windlass is to be tested at the manufacturer's shop of the electric motor.

(iii) The clutch and slipping clutch (for electric windlass) are to be tested to verify their performance.

4.4 Reciprocating Compressors

4.4.1 General

The following requirements apply to the compressors for supplying starting air of main propelling and auxiliary engines and the compressors for cargo refrigerating machinery. They apply mainly to reciprocating compressors of the types normally used aboard ships. For the capacity required for starting air compressors, see 4.6 of Part VI of "Rules for the construction and classification of steel ships" and the refrigerant compressors are also to comply with Part X of "Rules for the construction and classification of steel ships."

4.4.2 Construction

(a) The temperature of compressed air at the discharge of each stage is not to exceed 160°C for multistage compressors or 200°C for single-stage compressors; and for discharge pressure of less than 1.0 MPa, an increase of 20°C may be permitted. The cooling of air compressors is to be so designed that the temperature of the air discharged to the starting air vessel is substantially not to exceed 95°C in service. The design of the cooler for compressors is to be based on a sea water temperature of not less than 32°C for water cooling and an air temperature of not less than 45°C for air cooling.

(b) Safety devices

(i) Each stage of the air compressor is to be fitted with a suitable relief valve which cannot be isolated and which is so dimensioned and adjusted that in the event of the discharge line being shut-off, the accumulation will not exceed 10% of the approved working pressure, which are not to be greater than the approved design pressure in each stage of the air compressor. The casings of the cooling water spaces are to be fitted with a relief valve or safety disc so that sample relief will be provided in the vent of the bursting of an air cooler tube.

(ii) The refrigerant compressor is to be protected against over-pressure in such a manner that an equalizing of the pressure difference between the discharge and suction sides is possible by a relief valve and/or safety disc. Where one compressor stage consists of several cylinders, each capable of being isolated, each cylinder is to be provided with a relief valve. The relief valve and safety disc are to be set at a pressure not greater than the design pressure as specified in Table X 2-2 of Part X of "Rules for the construction and classification of steel ships."

(c) Every compressor stage and the refrigerant suction line are to be fitted with a suitable pressure gauge. In the intermediate stage of the air compressor requiring a driving power of less than 18 kW, the above pressure gauge may be omitted. It is recommended that the pressure gauge marked with the corresponding refrigerant temperatures be provided for the refrigerant compressor.

(d) The air compressor is to be provided with a separator in the final stage. It may also be advisable to fit a cooler. The separator or the cooler which serves as a separator is to be fitted at its lowest point with a drain where condensate may be observed as it runs off.

4.5 Pumps

4.5.1 Reciprocating pumps and other displacement pumps are to be fitted with sufficient large relief valves which cannot be isolated and which protect the pump casing from excessive pressure when the delivery valve is shut. For pumps handling inflammable fluids, the discharge from the relief valves is to be fed back to the suction side of the pumps.

4.5.2 Rotary pumps are to be so designed that they can be operated without damage occurring when the delivery valves are shut.

4.5.3 The size, capacity and numbers required for the pump are to be in accordance with the requirements in relevant Parts.

4.6 Tests and Inspections

4.6.1 Hydraulic pressure tests

Hydraulic pressure tests on deck machinery and pump parts after machining are to be carried out in the presence of the Surveyor under the conditions specified in Table III 4-1.

Table III 4-1
Hydraulic Test Pressure on Deck Machinery and Pump Parts

Parts to be Tested	Test Pressure, MPa
Steering gear:	
Steam reciprocating steering engine.	See 2.9.1 of this Part*.
Hydraulic steering gear, pump case, cylinder etc.	1.5 W or W + 7, whichever is smaller.
Windlass:	
Steam reciprocating windlass engine.	See 2.9.1 of this Part*.
Diesel windlass engine.	See 3.10.1 of this Part*.
Hydraulic pump and motor.	1.5 W or W + 7, whichever is smaller.
Reciprocating compressors:	
Air Compressor:	
Cylinder, liner, cover, inter- and after-coolers.	
Compressed air side.	1.5 W
Cooling water space.	0.4 but not less than 1.5 W
Refriagerant compressor.	See Part X*
Pump:	
Pump prime mover, steam or diesel engine.	See 2.9.1 and 3.10.1 of this Part*.
Pump casing.	0.4 but not less than 1.5 W.
Piping:	
Group-I and -II pipes and fittings.	See Part VI*.
Where: W = Design pressure and/or maximum working pressure for therespective parts, in MPa.	

Note: * refer to Rules for the Construction and Classification of Steel Ships.

4.6.2 Shop trials

(a) The following operational tests are to be carried out at the manufacturer's workshop in the presence of the Surveyor:

(i) For the steering gear: Characteristic tests of hydraulic pump units, if used. Running tests of steering gear. Adjustments and tests of safety devices and brake arrangements.

Each new design power unit pump for steering gear is to be type tested before come into the market.

The type test is to be for a duration of not less than 100 hours, the test arrangements are to be such that the pump may run in idling conditions, and at maximum delivery capacity at maximum working pressure. During the test, idling periods are to be alternated with periods at maximum delivery capacity at maximum working pressure. The passage from one condition to another is to occur at least as quickly as on board. During the whole test no abnormal heating, excessive vibration or other irregularities are permitted. After the test, the pump is to be disassembled and inspected. Type tests may be waived for a power unit which has been proven to be reliable in marine service.

(ii) For the athwartship thruster: Running tests of thruster. Adjustment and tests of the control and monitoring systems.

(iii) For the windlass, see 4.3.6 of this Part.

(iv) For the reciprocating compressor:

Running test for 2 hours and safety device test. Charging test for air compressor. Performance test for refrigerant compressor if deemed necessary by the Surveyor.

(v) For the pump: Characteristic tests with the pump running at designed condition.

(b) The overhaul inspection after shop trial is to be carried out in the presence of the Surveyor. The range and extent of inspection are subject to the discretion of the Surveyor.

4.6.3 On-board trials

(a) For the steering gear : 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 the Rules have been met. The trial is to include the operation of the following:

(i) The steering gear, including demonstration of the performances required by 4.2.2(b) and 4.2.3 (b). For the main steering gear trial, the propeller pitch of controllable pitch propellers is to be at the maximum continuous ahead RPM.

If the vessel cannot be tested at the deepest draught, alternative trial conditions may be specially considered. In this case, for the main steering gear trial, the speed of the ship corresponding to the maximum continuous revolutions of the main engine is to apply;

(ii) The steering gear power units, including transfer between steering gear power units;

(iii) The isolation of one power actuating system, checking the time for regaining steering capability;

(iv) The hydraulic fluid recharging system;

(v) The emergency power supply required by 2.3.10 of Part VII of "Rules for the construction and classification of steel ships";

(vi) The steering gear controls, including transfer of control and local control;

(vii) The means of communication between the steering gear compartment and the wheelhouse, also the engine room, if applicable;

(viii) The alarms and indicators;

(ix) Where the steering gear is designed to avoid hydraulic locking this feature is to be demonstrated.

Test items (iv), (vii), (viii) & (ix) may be effected at the dockside.

(b) For the windlass, an anchoring test is to be carried out in the presence of the Surveyor to demonstrate that the windlass with brakes, etc., functions satisfactorily and that the lifting power specified by the Rules can be developed. See 4.3.6 of this Part.

(c) For the athwartship thruster, mooring winch and capstan, an on-board running under working condition, if available, is to be carried out in the presence of the Surveyor.

(d) For reciprocating compressor, the following on-board trials are normally to be carried out in the presence of the Surveyor:

(i) Running and charging.

(ii) Safety device adjusting and setting.

(e) For pumps, an on-board running under working condition is to be carried out in the presence of the Surveyor.

4.6.4 Alternative proposals will be specially considered where any of the tests required by 4.6.2 and 4.6.3 above are considered impracticable.

Chapter 5
Gearing and Couplings

5.1 General

5.1.1 The requirements of this Chapter are applicable to reduction gearing and couplings for main propelling purpose and for driving essential service auxiliaries.

5.1.2 Drawings and data

(a) The design and arrangement of gearing components such as shafting, coupling, clutches and gears in propulsion, manoeuvring and lifting devices are subject to approval and certification.

(b) The following data are to be submitted together with drawings for approval:

Material specifications, maximum power transmitted by each pinion in continuous running and corresponding revolutions, number of teeth and modules of teeth in each pinion and gear wheel, pitch circle diameters, pressure angle and helix angle, addendum and dedendum, face width tooth profiles together with base and pitch circle diameters and fillet radii, minimum backlash, addendum modifications and tooth bearing corrections to the teeth, finishing operations for the teeth, final accuracy foreseen of the teeth and their meshing, data for rim shrunk on gear wheel and/or hub shrunk on shaft with the shrinkage allowances, and welding details including sequences and stress relieving where the casing and/or the gear wheel are of welded construction.

(c) Documentation and strength calculations on the basis of Society Rules are to be submitted. Alternatively, recognized calculation procedures may be accepted.

5.1.3 Materials

(a) All components of gearing and couplings which transmit part of the turning moment for the main propelling purpose and for driving the essential service auxiliaries are to be tested and inspected in the presence of the Surveyor to comply with the requirements in Part XI of the Rules for the Construction and Classification of Steel Ships or to the requirements of the specifications approved in connection with the design.

(b) The components described above are normally to be made of forged or cast steel. The flangeless plain gear shaft may also be made of hot rolled carbon steel bars.

(c) The use of cast iron for gearing and coupling is to be approved by the Society in each particular case.

(d) In the selection of materials for pinion and gear wheel teeth, consideration is to be given to their combined ability to resist wear, pitting and scuffing. In general, for gears of through-hardened steel, except in the case of low reduction ratios, provision is also to be made for a hardness differential between pinion teeth and gear wheel teeth for which purposes the specified minimum tensile strength of the gear wheel rim material is not to be more than 85% of that of the pinion. For unhardened gears, the tensile strength of the material of pinion teeth is generally to be at least 100 N/mm^2 above that of the gear wheel teeth.

(e) Gear wheel and rim forgings with a specified minimum tensile strength not in excess of 760 N/mm^2 may be made in carbon-manganese steel. Gear wheel or rim forgings where the specified minimum tensile strength is in excess of 760 N/mm^2, and all pinion sleeve forgings are to be made in a suitable alloy steel.

5.2 Construction

5.2.1 Gear wheels and pinions

(a) In the case of gear wheels with shrunk-fitted rim, the rim is to be of sufficient thickness to ensure a good shrink fit without undue distortion and the shrinkage is to be sufficient for transmitting the load with adequate safety. The shrinking is preferably to be completed prior to cutting the teeth and is to be carried out to the satisfaction of the Surveyor.

(b) In the case of welded construction, the full details of welding procedure are to be approved in the first instance by the Surveyor before work is commenced. The welded gear wheel is to be stress relieved prior to cutting the teeth and particulars of which are to be submitted. All welds in the completed gear wheel are to have satisfactory smooth appearance and even contour, and are to be proved by means of radiographic or magnetic particle detection.

(c) In general, arrangements are to be made so that the interior structure of the gear wheel may be examined. An alternative proposal will be specially considered in each case.

5.2.2 Gear casings

(a) The gear casing is to be of rigid construction in order that it is free from distortion when chocked and secured to its seating to maintain the alignment of gear elements at sea. For this purpose the casing is to be constructed with heavy ribbing with adequate flanges on both end walls and side walls.

(b) Inspection openings are to be provided at the peripheries of the gear casing to enable the teeth of pinions and gear wheels to be readily examined. Where the construction of the gear casing is such that sections of structure cannot readily be moved for inspection purposes, access openings of adequate size are also to be provided at the ends of the gear casing to permit examination of the structure of gear wheels. Their attachment to the shafts is to be capable of being examined by removal of bearing caps or by equivalent means.

(c) The gear casing of welded construction is to be stress relieved on completion and particulars of which are to be submitted.

(d) For gear casing fabricated by fusion welding the carbon content of the steels should generally not exceed 0.23 per cent. Steel with higher carbon content may be approved subject to satisfactory results from weld procedure tests.

5.2.3 Couplings

(a) The coupling is to be so constructed that it has sufficient strength to transmit the load and is capable of being operating safely and efficiently.

(b) When hydraulic or compressed air is employed in the operation of the clutch for main propelling purpose, an hydraulic pump or an air compressor of reserve use, or other suitable means are to be provided to ensure that the ship can proceed the safe voyage in case of failure of the hydraulic or compressed air main supply system. The Society may, however, modify the above mentioned requirements for small ships.

5.3 Design – Load Capacity of Involute Parallel Axis Spur and Helical Gears

5.3.1 General influence factors

(a) Symbols and units

a	=	Centre distance, in mm;
b	=	Common facewidth, in mm;
$b_{1,2}$	=	Facewidth of pinion, wheel, in mm;
d	=	Reference diameter, in mm;
$d_{1,2}$	=	Reference diameter of pinion, wheel, in mm;
$d_{a1,a2}$	=	Tip diameter of pinion, wheel, in mm;
$d_{b1,b2}$	=	Based diameter of pinion, wheel, in mm;
$d_{f1,f2}$	=	Root diameter of pinion, wheel, in mm;
$d_{w1,w2}$	=	Working diameter of pinion, wheel, in mm;
F_t	=	Nominal tangential load, in N;
F_{bt}	=	Nominal tangential load on base cylinder in the transverse section, in N;
h	=	Tooth depth, in mm;
m_n	=	Normal module, in mm;
m_t	=	Transverse module, in mm;
$n_{1,2}$	=	Rotational speed of pinion, wheel in revs/min;
P	=	Maximum continuous power transmitted by the gear set, in kW;
$T_{1,2}$	=	Torque in way of pinion, wheel, in Nm;
U	=	Gear ratio;
v	=	Linear speed at pitch diameter, in m/s;
$x_{1,2}$	=	Addendum modification coefficient of pinion, wheel;
z	=	Number of teeth;
$z_{1,2}$	=	Number of teeth of pinion, wheel;
z_n	=	Virtual number of teeth;
α_n	=	Normal pressure angle at reference cylinder, in deg,;
α_t	=	Transverse pressure angle at ref. Cylinder, in deg.;
α_{tw}	=	Transverse pressure angle at working pitch cylinder, in deg.;
β	=	Helix angle at reference, in deg.;
β_b	=	Helix angle at base cylinder, in deg.;
ε_α	=	Transverse contact ratio;
ε_β	=	Overlap ratio;
ε_γ	=	Total contact ratio;

(b) Geometrical definitions

For internal gearing z_2, a, d_2, d_{a2}, d_{b2} and d_{w2} are negative. The pinion is defined as the gear with the smaller number of teeth, therefore the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity:

$$u = z_2/z_1 \quad = d_{w2}/d_{w1} \quad = d_2/d_1$$

For external gears u is positive, for internal gears u is negative.

In the equation of surface durability b is the common 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 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 adopted.

$\tan \alpha_t$ = $\tan \alpha_n / \cos \beta$

$\tan \beta_b$ = $\tan \beta \cos \alpha_t$

d = $z\, m_n / \cos \beta$

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

a = $0.5\,(d_{w1}+d_{w2})$

z_n = $z/(\cos^2 \beta_b \cos\beta)$

m_t = $m_n/\cos \beta$

inv α = $tg\alpha - \pi\alpha/180$; α : angle in degrees

inv α_{tw} = inv $\alpha_t + 2\, tg\alpha_n\, (x_1+x_2)/(z_1+z_2)$

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

the positive sign is used for external gears, the negative sign for internal gears;

ε_β = $b \sin \beta/\pi$, in (mm); for double helix, b is to be taken as the width of one helix;

ε_γ = $\varepsilon_\alpha + \varepsilon_\beta$

v = $d_{1,2} \bullet n_{1,2}/19099$

(c) Nominal tangential load, F_t

The nominal tangential load, F_t, tangential to the reference cylinder and perpendicular to the relevant axial plane, is calculated directly from the maximum continuous power transmitted by the gear set by means of the following equations:

$T_{1,2}$ = $9549\, P/n_{1,2}$

F_t = $2000\, T_{1,2}/d_{1,2}$

(d) General influence factors

(i) Application factor, K_A

The application factor, K_A, accounts for dynamic overloads from sources external to the gearing.

K_A, for gears designed for infinite life is defined as the ratio between the maximum repetitive cyclic torque applied to the gear set and the nominal rated torque. When operating near a critical speed of the drive system, a careful analysis of conditions must be made. The application factor, K_A, is to be determined by measurements or by system analysis acceptable to the Society. Where a value determined in such a way cannot be supplied, the following values can be considered:

(1) Main propulsion

(A) diesel engine with hydraulic or electro- Magnetic slip coupling : 1.00

(B) diesel engine with high elasticity coupling: 1.30

(C) diesel engine with other couplings : 1.50

(2) Auxiliary gears

(A) electric motor, diesel engine with hydraulic or electromagnetic slip coupling: 1.00

(B) diesel engine with high elasticity coupling: 1.20

(C) diesel engine with other couplings : 1.40

(ii) Load sharing factor, K_γ

The load sharing factor, K_γ accounts for the maldistribution of load in multiple path transmissions (dual tandem, epicyclical, double helix, etc.). K_γ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor mainly depends on accuracy and flexibility of the branches. The load sharing factor, K_γ, is to be determined by measurements or by system analysis. Where a value determined in such a way cannot be supplied, the following values can be considered for epicyclical gears:

(1) up to 3 planetary gears: 1.00

(2) 4 planetary gears: 1.20

(3) 5 planetary gears: 1.30

(4) 6 planetary gears and over: 1.40

(iii) 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. K_v is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load (F_t K_A K_γ). The dynamic factor, K_v, can be calculated as follows: The method may be applied only to cases where all the following conditions are satisfied:

(1) steel gears of heavy rims sections

(2) $F_t/b > 150$ N/mm

(3) $z_1 < 50$

(4) running speed in the subcritical range:

(A) for helical gears: $(v_{z1})/100<14$

(B) for spur gears: $(v_{z1})/100<10$

This method may be applied to all types of gears if

$(v_{z1})/100<3$

For helical gears of overlap ratio > unity K_v is obtained from Fig. III 5-1.

For spur gears K_v is obtained from Fig. III 5-2.

For helical gears of overlap ratio < unity K_v is obtained by means of linear interpolation between the values obtained from Fig. III 5-1 and III 5-2.

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

Where:

K_{v1} is the K_v value for helical gears, given by Fig. III 5-1.

K_{v2} is the K_v value for spur gears, given by Fig. III 5-2.

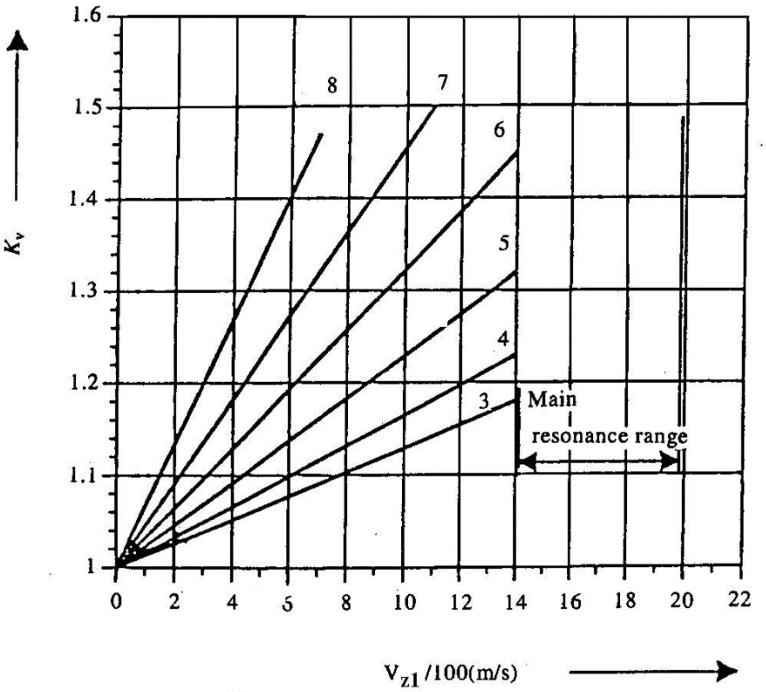


Fig. III 5-1
Dynamic Factor for Helical Gear. ISO Grades of Accuracy 3-8

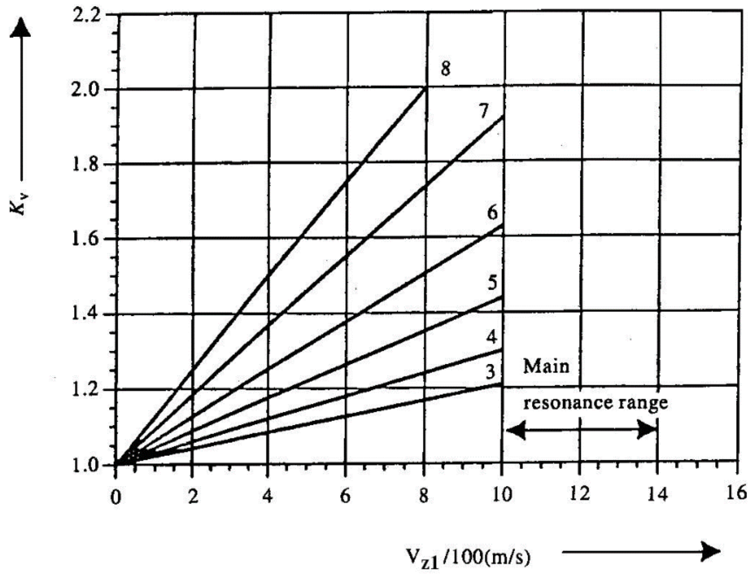


Fig. III 5-2
Dynamic Factor for Spur Gear. ISO Grades of Accuracy 3-8

Note: 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 is to be used.

Table III 5-1
Values of the Factor K_1 for the Calculation of K_V

	K_1 ISO GRADES OF ACCURACY					
	3	4	5	6	7	8
Spur gears	0.022	0.030	0.043	0.0620	0.092	0.125
Helical gears	0.0125	0.0165	0.0230	0.0330	0.0480	0.0700

K_v can also be determined as follows:

$K_v = 1 + K_1 (v_{z1})/100$

K_1 values are specified in the following Table III 5-1.

(iv) Face load distribution factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress, $K_{F\beta}$ for tooth root bending stress, account for the effects of non-uniform distribution of load across the facewidth.

$K_{H\beta}$ 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:

$$K_{F\beta} = \frac{\text{(maximum bending stress at tooth root per unit facewidth)}}{\text{(mean bending stress at tooth root per unit facewidth)}}$$

The mean bending stress at tooth root relates to the considered facewidth b_1 resp. b_2.

$K_{F\beta}$ can be expressed as a function of the factor $K_{H\beta}$.

(1) In case the hardest contact is at the end of the facewidth $K_{F\beta}$ is given by the following equations:

$$K_{F\beta} = K_{H\beta}^{N}$$

$$N = \frac{(b/h)^2}{1 + (b/h) + (b/h)^2}$$

(b/h)= facewidth/tooth height ratio, the minimum of b_1/h_1 or b_2/h_2. For double helical gears, the facewidth of only one helix is to be used.

(2) In case of gears where the end of the face width are lightly loaded or unloaded (end relief or crowning):

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

(v) 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:

(1) total mesh stiffness;

(2) total tangential load F_t, K_A, K_γ, K_v, $K_{H\beta}$;

(3) base pitch error;

(4) tip relief;

(5) running-in allowances.

5.3.2 Surface durability (pitting)

(a) Scope and general remarks

The criterion for surface durability is based on the Hertz pressure on the operating pitch point or at the inner point of single pair contact. The contact stress σ_H must be equal to or less than the permissible contact stress σ_{HP}.

(b) Basic equations

(i) Contact stress

$$\sigma_H = \sigma_{HO} \sqrt{K_A K_\gamma K_v K_{H\alpha} K_{H\beta}} \le \sigma_{HP}$$

where:

σ_{HO}= Basic value of contact stress for pinion and wheel

$$\sigma_{HO} = Z_B Z_H Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{u+1}{u}} \qquad \text{for pinion}$$

$$\sigma_{HO} = Z_D Z_H Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{u+1}{u}} \qquad \text{for wheel}$$

where:

Z_B = Single pair mesh factor for pinion; 5.3.2(b)(iii)

Z_D = Single pair mesh factor for wheel ; 5.3.2(b)(iii)

Z_H = Zone factor; 5.3.2(b)(iv)

Z_E = Elasticity factor; 5.3.2(b)(v)

Z_ε = Contact ratio factor; 5.3.2(b)(vi)

Z_β = Helix angle factor; 5.3.2(b)(vii)

F_t = Nominal tangential load at reference cylinder in the transverse section; 5.3.1(a)

b = Common facewidth;

d_1 = Reference diameter of pinion;

u = Gear ratio (for external gears u is positive, for internal gears u is negative);

K_A, K_γ, K_v, $K_{H\alpha}$ and $K_{H\beta}$, see 5.3.1.

(ii) Permissible contact stress

The permissible contact stress σ_{HP} is to be evaluated separately for pinion and wheel:

$$\sigma_{HP} = (\sigma_{Hlim} Z_N/S_H) \times Z_L Z_v Z_R Z_W Z_X$$

where:

σ_{Hlim} = Endurance limit for contact stress 5.3.2(b)(viii);

Z_N = Life factor for contact stress; 5.3.2(b)(ix)

Z_L = Lubrication factor; 5.3.2(b)(x)

Z_v = Speed factor; 5.3.2(b)(x)

Z_R = Roughness factor; 5.3.2(b)(x)

Z_W = Hardness ratio factor; 5.3.2(b)(xi)

Z_X = Size factor for contact stress; 5.3.2(b)(xii)

S_H = Safety factor for contact stress; 5.3.2(b)(xiii)

(iii) Single pair mesh factors, Z_B and Z_D

The single pair mesh factors, Z_B for pinion and Z_D for wheel, account for the influence on contact stresses of the tooth flank curvature at the inner point of single pair contact in relation to Z_H.

The single pair mesh factors, Z_B for pinions and Z_D for wheels, can be determined as follows:

(1) For spur gears, $\varepsilon_\beta = 0$

$Z_B = M_1$ or 1 whichever is the larger value.

$Z_D = M_2$ or 1 whichever is the larger value.

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

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

For helical gears when $\varepsilon_\beta \geq 1$

$Z_B = Z_D = 1$

For helical gears when $\varepsilon_\beta < 1$ the values of Z_B, Z_D are determined by linear interpolation between Z_B, Z_D for spur gears and Z_B, Z_D for helical gears having $\varepsilon_\beta \geq 1$.

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 $Z_D \geq 1$

(iv) Zone factor, Z_H

The zone factor, Z_H, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and relates the tangential force at the reference cylinder.

The zone factor, Z_H, can be calculated as follows:

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

(v) 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 Hertz pressure.

The elasticity factor, Z_E, for steel gears ($E = 206000$ N/mm^2, $\nu = 0.3$) is equal to:

$Z_E = 189.8$ (N$^{1/2}$/mm)

(vi) Contact ratio factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of the transverse contact ratio and the overlap ratio on the specific surface load of gears. The contact ratio factor, Z_ε, can be calculated as follows:

(1) Spur gears:

$$Z_\varepsilon = \sqrt{\frac{4 - \varepsilon_\alpha}{3}}$$

(2) Helical gears:

(A) for $\varepsilon_\beta < 1$ $\quad Z_\varepsilon = \sqrt{\frac{4 - \varepsilon_\alpha}{3}(1 - \varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}$

(B) for $\varepsilon_\beta \geq 1$ $\quad Z_\varepsilon = \sqrt{\dfrac{1}{\varepsilon_\alpha}}$

(vii) Helix angle factor, Z_β

The helix angle factor, Z_β, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Z_β is dependent only on the helix angle. The helix angle factor, Z_β, can be calculated as follows:

$$Z_\beta = \sqrt{\cos\beta}$$

Where β is the reference helix angle.

(viii) Endurance limit for contact stress, σ_{Hlim}

For a given material, σ_{Hlim} is the limit of repeated contact stress which can be permanently endured. The value of σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least 50×10^6 load cycles. For this purpose, pitting is defined by:

(1) for not surface hardened gears:
pitted area > 2% of total active flank area;

(2) for surface hardened gears:
pitted area > 0.5% of total active flank area, or > 4% of one particular tooth flank area.
The σ_{Hlim} values are to correspond to a failure probability of 1% or less.
The endurance limit for contact stress σ_{Hlim}, can be determined, in general, making reference to values indicated in ISO 6336/5, quality MQ.

(ix) Life factor, Z_N

The life factor, Z_N, accounts for the higher permissible contact stress in case a limited life (number of cycles) is required.

The life factor, Z_N, can be determined according to method B outlined in the ISO 6336/2 standard.

(x) 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 and the roughness factor, Z_R, accounts for the influence of the surface roughness on the surface endurance capacity. The factors may be determined for the softer material where gear pairs are of different hardness.

(1) Lubricant factor, Z_L

$$Z_L = C_{ZL} + \frac{4(10.0 - C_{ZL})}{(1.2 + 134 / \nu_{40})^2}$$

In the range 850 N/mm^2,

$$C_{ZL} = \left(\frac{\sigma_{H\lim} - 850}{350} 0.08\right) + 0.83$$

If $\sigma_{Hlim} < 850$ N/mm^2, take $C_{LZ} = 0.83$.
If $\sigma_{Hlim} > 1{,}200$ N/mm^2, take $C_{LZ} = 0.91$.
Where:
ν_{40} = Nominal kinematic viscosity of the oil at 40°C,

(2) Speed factor, Z_v
In the range 850 N/mm^2 $\leq \sigma_{Hlim} \leq$ 1,200 N/mm^2, C_{Zv} can be calculated as follows:

$$Z_v = C_{Zv} + \frac{2(1.0 - C_{Zv})}{\sqrt{0.8 + 32 / v}}$$

$$C_{Zv} = \left(\frac{\sigma_{H\lim} - 850}{350} 0.08\right) + 0.85$$

(3) Roughness factor, Z_R

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

Where:

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

The peak-to-valley roughness determined for the pinion R_{Z1} and for the wheel R_{Z2} are mean values for the peak-to-valley roughness R_Z measured on several tooth flanks R_Z.

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

relative radius of curvature:

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

Wherein:
$\rho_{1,2} = 0.5 \bullet d_{b1,2} \bullet \tan\alpha_{tw}$
(also for internal gears, d_b negative sign)
If the roughness stated is an R_a value (= C_{LA} value) (= A_A value) the following approximate relationship can be applied:
$R_a = C_{LA} = A_A = R_Z/6$
In the range 850 N/mm^2 $\leq \sigma_{Hlim} \leq$ 1,200 N/mm^2, C_{ZR} can be calculated as follows:
$C_{ZR} = 0.32 - 0.0002\ \sigma_{Hlim}$
If $\sigma_{Hlim} <$ 850 N/mm^2, take $C_{ZR} = 0.150$.
If $\sigma_{Hlim} >$ 1,200 N/mm^2, take $C_{ZR} = 0.080$.

(xi) Hardness ratio factor, Z_W

The hardness ratio factor, Z_W, account for the increase of surface durability of a soft steel gear meshing with a significantly harder gear with a smooth surface.

Z_W apply to the soft gear only.

$$Z_W = 1.2 - \frac{H_B - 130}{1700}$$

Where:

H_B = Brinell hardness of the softer material.

For $H_B < 130$, $Z_W = 1.2$ will be used.

For $H_B > 470$, $Z_W = 1.0$ will be used.

(xii) 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.

For through-hardened gears and for surface-hardened gears with adequate case depth relative to tooth size and radius of relative curvature $Z_X = 1$. When the case depth is relatively shallow then a smaller value of Z_X is to be chosen.

(xiii) Safety factor for contact stress, S_H

The following guidance values can be adopted:

(1) Main propulsion gears: 1.20 ~ 1.40.

(2) Auxiliary gears: 1.15 ~ 1.20.
For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of the Society.

5.3.3 Tooth root bending strength

(a) Scope and general remarks

The criterion for tooth σ_F root bending strength is the permissible limit of local tensile strength in the root fillet. The root stress σ_F and the permissible root stress σ_{FP} are to be calculated separately for the pinion and the wheel.

σ_F must not exceed σ_{FP}. The following definitions apply to gears having rim thickness greater than 3.5 mm.

(b) Basic equations

(i) Tooth root bending stress for pinion and wheel

$$\sigma_F = (F_t/bm_n) Y_F\, Y_S\, Y_\beta\, K_A\, K_\gamma\, K_v\, K_{F\alpha}\, K_{F\beta} \le \sigma_{FP}$$

where:

Y_F	=	Tooth form factor;	5.3.3 (c)
Y_S	=	Stress correction factor;	5.3.3 (d)
Y_β	=	Helix angle factor;	5.3.3 (e)
$F_t, K_A, K_\gamma, K_v, K_{F\alpha}, K_{F\beta}$			5.3.1
b			5.3.1 (d)

(ii) Permissible tooth root bending stress for pinion and wheel

$$\sigma_{FP} = (\sigma_{FE}\, Y_d/S_F)\, Y_{\sigma reIT}\, Y_{RreIT}\, Y_X$$

where:

σ_{FE}	=	Bending endurance limit;
Y_d	=	Design factor;
Y_N	=	Life factor;
$Y_{\sigma reIT}$	=	Relative notch sensitivity factor;
Y_{ReIT}	=	Relative surface factor;
Y_X	=	Size factor;
S_F	=	Safety factor for tooth root bending stress.

(c) Tooth form factor, Y_F

The tooth form factor, Y_F, represents the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F is to be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing are to be determined in the normal section, i.e. for the virtual spur gear with virtual number of teeth z_n.

$$Y_F = \frac{6 \dfrac{h_F}{m_n} \cos \alpha_{Fen}}{\left(\dfrac{s_{Fn}}{m_n}\right)^2 \cos \alpha_n}$$

Where:

h_F = Bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact, in mm;

s_{Fn} = Tooth root chord in the critical section, in mm;

α_{Fen} = Pressure angle at the outer point of single tooth pair contact in the normal section.

The result of rating calculations made by following this method are acceptable for normal pressure angles up to 25° and reference helix angles up to 30°.

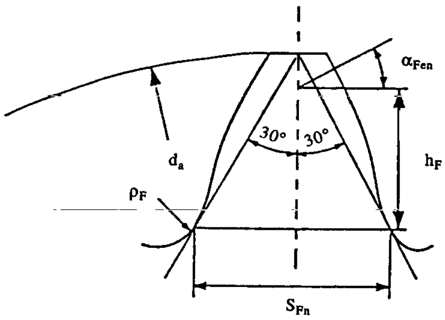


Fig. III 5-3
For the Calculation of h_F, s_{Fn} and α_{Fen}

(d) Stress correction factor, Y_S

The stress correction factor, Y_S, is used to convert the nominal bending stress to the local tooth root stress, taking into account that not only bending stresses arise at the root.

Y_S applies to the load application at the outer point of single tooth pair contact.

Y_S is to be determined separately for the pinion and for the wheel.

The stress correction factor, Y_S, can be determined with the following equation
(having range of validity: $1 \le q_s < 8$):

$$Y_S = (1.2 + 0.13L)\, q_s^{\left(\frac{1}{1.12 + 2.3/L}\right)}$$

Where:

$$q_s = \frac{s_{Fn}}{2\rho_F}$$

q_s = Notch parameter;

ρ_F = Root fillet radius in the critical section;

L = s_{Fn}/h_f

For h_F and S_{Fn} see 5.3.3 (c).

(e) Helix angle factor, Y_β

The helix angle factor, Y_β, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth. The helix angle factor, Y_β can be calculated as follows:

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120}$$

where: β = Reference helix angle in degrees.

One (1.0) is substituted for ε_β when $\varepsilon_\beta > 1.0$, and 30° is substituted for $\beta > 30°$.

(f) Bending endurance limit, σ_{FE}

For a given material, σ_{FE} is the local tooth root stress which can be permanently endured. The number of 3×10^6 cycles is regarded as the beginning of the endurance limit. σ_{FE} is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment). Other conditions such as alternating stress or pre-stressing etc. are covered by the design factor Y_d. The σ_{FE} values are to correspond to a failure probability 1% or less. The bending endurance limit, σ_{FE} can be determined, in general, making reference to values indicated in ISO 6336/5, quality MQ.

(g) 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} .

The design factor, Y_d, for load reversing, can be determined as follows:

Y_d = 1.00 in general;

Y_d = 0.9 for gears with occasional part load in reversed direction, such as main wheel in reversing gearboxes;

Y_d = 0.7 for idler gears.

(h) Life factor, Y_N

The life factor, Y_N, accounts for the higher tooth root bending stress permissible in case a limited life (number of cycles) is required.

The life factor, Y_N, can be determined according to method B outlined in ISO 6336/3 standard.

(i) Relative notch sensitivity factor, $Y_{\delta relT}$

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit.

The factor mainly depends on material and relative stress gradient.

The relative notch sensitivity factor, $Y_{\delta relT}$, can be determined as follows:

(i) for notch parameter values (see 5.3.3 (d)) included in the range $1.5 < q_s < 4$, it can be assumed:

$Y_{\delta relT} = 1.0$

(ii) for notch parameter outside said range $Y_{\delta relT}$ can be calculated as outlined in the reference standard.

(j) Relative surface factor, Y_{RrelT}

The relative surface factor, Y_{Rre1T}, takes into account the dependence of the root strength on the surface condition in the tooth root fillet, mainly the dependence on the peak to valley surface roughness.

The relative surface factor, Y_{RrelT} can be determined as follows:

$R_z < 1$	$1 \le R_z \le 40$	
1.120	$1.675 - 0.53(R_z+1)^{0.1}$	case hardened steels through - hardened steels ($\sigma_\beta \ge 800$ N/mm^2)
1.070	$5.3 - 4.2\ (R_z+1)^{0.01}$	normalised steels ($\sigma_\beta < 800$ N/mm^2)
1.025	$4.3\text{-}3.26\ (R_z+1)^{0.005}$	nitrided steels

Where: R_z = Mean peak-to-valley roughness of tooth root fillets, µm.

If the roughness stated is an R_a value (= C_{LA} value) (= A_A value) the following approximate relationship can be applied:

$R_a = C_{LA} = A_A = R_z/6$

(k) Size factor, Y_X

The size factor, Y_X, takes into account the decrease of the strength with increasing size.

The size factor, Y_X, can be determined as follows:

$Y_X = 1.00$	for $m_n \le 5$	generally
$Y_X = 1.03 - 0.06\ m_n$	for $5 < m_n$, 30	normalised and through –
$Y_X = 0.85$	for $m_n \ge 30$	hardened steels
$Y_X = 1.05 - 0.01\ m_n$	for $5 < m_n < 25$	surface
$Y_X = 0.80$	for $m_n \ge 25$	hardened steels

(l) Safety factor for tooth root bending stress, S_F

The safety factor for tooth root bending stress, S_F, can be assumed by the Society taking into account the type of application.

(i) Main propulsion gears:1.55 ~ 2.00.

(ii) Auxiliary gears: 1.40 ~ 1.45.

5.3.4 Torsional vibrations

The calculations for torsional vibration behavior are to be submitted for review to the Society in due time. The calculations must include the equivalent mass-elastic system, natural frequencies and corresponding vibration modes, as well as the forced response for gears, couplings shaftings. Calculations are to be checked in accordance with Society Rules. Torsional vibration measurements may be required by the Society. Such measurements will be necessary to detect dynamic torques, especially for plants working under unclearly defined conditions. As far as such measurements aim to provide special knowledge concerning load and response characteristics, adequate measuring techniques are to be applied, by agreement with the Society.

5.3.5 Whirling vibrations

The calculations of natural frequencies for whirling are to be submitted for information.. This will regularly be the case for thin, long shaftings supported by few bearings. The calculations may be required in a simplified form i.e. natural frequencies, or as complete forced vibrations using methods or programmes approved by the Society.

5.3.6 Lateral vibrations

The vibration calculations for resilently mounted reciprocating main engines are to be submitted for information. The calculations may be carried out in a simplified form, i.e. natural frequencies for the six degrees of freedom and corresponding modes, provided that no resonances excited by major engine orders are within the operational speed range.

5.4 Workmanship

5.4.1 Gear cutting

(a) The constructional precision of the complete gear cutting machinery is to comply with satisfactory, recognized standards for high quality machine tools.

(b) All gears are to be cut on machines which are maintained at a high standard of accuracy, and the gear cutting machinery used is to operate preferably under conditions of temperature control.

(c) Application of hand tools, filing or scraping of the active tooth surfaces is usually not permitted. A limited application of honing when tip-relieving may be permitted to remove local high spots appearing after the last surface finish. All sharp edges on the tip and ends of gear teeth after cutting or finishing are to be removed.

5.4.2 Tooth hardening

(a) The surface treatments of the teeth are to ensure the continuity of the hardened zone on the whole height of the tooth flank and more particularly on the root fillet. The thickness of the hardened zone remaining after finishing is to be at least equal to twice the depth corresponding to the maximum shear stress.

(b) Where the pinion and the toothed portion of the gear wheel are case-hardened and tempered, the teeth flanks are to be ground while the bottom lands of the teeth remain only case-hardened. Where the teeth are nitrided, the thickness of the hardened zone is not to be less than 0.5 mm and the grinding of nitrided teeth is normally not required. The use of any other hardening of the teeth such as induction or flame hardening, is to be submitted for special consideration.

5.4.3 Shrinkage

The appropriate shrinkage conditions of the rim and pinion or gear wheel body are to be submitted for consideration. The use of other means of fixation such as pins is not permissible.

5.5 Tests and Inspections

5.5.1 Hardening

Where the teeth of gears are made of through-hardened steels, the test specimens representative of the hardened teeth are normally to be provided for examination and to be demonstrated in advance that the thickness of the hardened zone is satisfactory. The hardened zone of finished teeth is to be hardness tested and inspected with suitable non-destructive test methods.

5.5.2 Welding

When welding is employed in the construction of gearing and couplings, the test specimens representative of the welded joints used in the construction are to be provided for examination and mechanical test where deemed necessary by the Surveyor.

5.5.3 Dynamic balancing

Finished pinions and wheels are to be dynamically balanced in two planes where their pitch line velocity exceeds 25 m/s. Where their pitch line velocity does not exceed 25 m/s 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:

$$B = 24 \cdot W/N$$

where:

B = Maximum allowable residual unbalance, in N · mm;

W = Weight of rotating part, in N;
N = rpm at rated speed.

5.5.4 Accuracy

(a) The surface finish of teeth and the accuracy of gear cutting in all gears are to be inspected and demonstrated at the workshop to the satisfaction of the Surveyor.

(b) Undulations, i.e. the periodic or semi-periodic departure of the actual tooth surface from the design surface, are to be measured at approximately mid-depth of the working surface of the tooth. The undulation records are to be submitted for inspection.

(c) Cumulative pitch errors are to be measured by taking readings in the transverse plane in equal spans around the total circumference of the gear wheel or pinion. The results are to be plotted in the usual manner and submitted for inspection.

5.5.5 Non-destructive examination

(a) Magnetic particle or liquid penetrant examination is to be carried out on the teeth of all surface hardened forgings. This examination may also be requested on the finished machined teeth of through hardened gear forgings.

(b) The ultrasonic examination is to be carried out by the manufacturer for all forgings where the finished diameter of the surfaces, where teeth will be cut, is in excess of 200 mm, and is to provide the Surveyor with a signed statement that such inspection has not revealed any significant internal defects.

(c) On gear forgings where the teeth have been surface hardened, additional test pieces may be required to be processed with the forgings and subsequently sectioned to determine the depth of the hardened zone. These examinations are to be carried out at the discretion of the Surveyor, and for induction or carburised gearing the depth of the hardened zone is to be in accordance with the approved specification. For nitrided gearing, the full depth of the hardened zone, i.e. depth to core hardness, is to be not less than 0.5 mm and the hardness at a depth of 0.25 mm is to be not less than 500 Hv.

5.5.6 Meshing tests and trial

(a) The accuracy of gear meshing is to be verified in the presence of the Surveyor by coating pinion and gear wheel in turn with a thin film of copper sulphate, an approved spirit lacquer, or other equivalent prior to the commencement of the trial, and turning the gears with sufficient pressure to ensure contact between teeth. The thickness of the coating to determine the contact marking is not to exceed 0.005 mm. In general, the meshing area is to be consistent with that which will result in evenly distributed tooth contact at full load. For through-hardened gears, it is recommended that the contact is at least 40% of the working depth for 35% of the length, and at least 20% of the working depth for a further 35% of the length on each helix, without hard bearing on the tooth flank area.

(b) A permanent record is to be made of the meshing contact for the purpose of checking the alignment when installed on board the craft.

(c) Where, due to the compactness of the gear unit, meshing tests of individual units cannot be verified visually, consideration may be given to the gear manufacturer providing suitable evidence that the design meshing condition has been attained on units of the same design.

(d) The gear manufacturer is required to verify by means of measurements in the presence of the Surveyor that the gear case is free from distortion when erected on-board and secured to its seating.

(e) The on-board trial of gearing is to be carried out for a sufficient duration during sea trial to prove that the gears operate satisfactorily under service conditions. After sea trial, the teeth of all gear elements are to be inspected in the presence of the Surveyor. The contact marking after run under service load is also to be verified by using a suitable lacquer applied prior to the commencement of the trial.

(f) The couplings are to be subjected to test at the workshop. Where a test cannot be carried out at the workshop, the operational trial is to be carried out after installation on-board the ship.

Chapter 6

Shafting, Propellers and Propulsion Shafting System

6.1 Shaftings

6.1.1 Scope

The requirements of this section apply to propulsion shafting and power transmission system which transmit power from prime mover driving generators and essential service auxiliaries. The torsional vibration of shaftings are to comply with the requirements specified in 6.4 of this Chapter.

6.1.2 Drawings and data

Drawings and data to be submitted are generally as follows:

(a) Drawings for approval (including specifications of material)
- (i) shafting arrangement
- (ii) Thrust shaft
- (iii) Intermediate shaft
- (iv) Stern tube shaft
- (v) Propeller shaft
- (vi) Stern tube and stern tube bearing
- (vii) Stern tube sealing device
- (viii) Shaft bracket bearing
- (ix) Shaft couplings and coupling bolts
- (x) Shafts which transmit power to generator or essential service auxiliaries.

(b) Data for reference
- (i) Data necessary for the calculations of shafting strength specified in this section.
- (ii) Data deemed necessary by this Society.

6.1.3 Materials

(a) Material intended for the principal components of shafting are to be confirmed to the requirements specified in Part XI of the Rules for the Construction and Classification of Steel Ships in addition to the following:

(b) The materials intended for the thrust-, intermediate and propeller-shafts, coupling flanges and coupling bolts are to be tested and inspected in the presence of the Surveyor in accordance with the requirements in Part XI of the Rules for the Construction and Classification of Steel Ships: or the specifications approved in connection with the design.

(c) Main shafting and accessories are to be of forged steel. Cast steel couplings may be accepted under certain circumstances. The hot rolled carbon steel bars complying with Part XI of the Rules for the Construction and Classification of Steel Ships may be used for the plain, flangeless shaft. Where parts of the shafting are made of material other than steel, these are to be specially approved by the Society.

(d) The specified minimum tensile strength of forgings for propeller shaft and other shafts is to be selected within the following general limits:

(i) Carbon and carbon mangenese steel: 400 to 600 N/mm^2

(ii) Alloy steel (Age-hardened martensitic stainless steels or other high strength alloy materials): not exceeding 800 N/mm^2 and for other forgings is not to exceed 1100 N/mm^2

(e) Where it is proposed to use alloy steel and the other composite material, details of the chemical composition, heat treatment and mechanical properties are to be submitted for approval.

(f) Ultrasonic tests are required on shaft forging and /or alloy steel materials before machining.

(g) Hollow shafts

(i) The outer diameters of the hollow shafts are not to be less than that required by the formulae for the corresponding solid shafts multiplying by the following coefficient:

$$k = \sqrt[3]{\frac{1}{1-(\frac{d_i}{d_o})^4}}$$

(ii) However, where the inner bore of the hollow shaft does not exceed 40% of the outer diameter, no increase over diameter required for the corresponding solid shaft need be provided.

(iii) The notations used in 6.1.3(g)(i) above are defined as follows:

k = Multiplying coefficient for hollow shafts.

d_i= Inner bore of hollow shafts, in mm.

d_o= Outer diameter of hollow shafts, in mm.

6.1.4 Strength calculations and construction.

(a) The diameter of the intermediate shaft is not to be less than that obtained from the following formula:

$$d_1 = fFC_1K_1\sqrt[3]{\frac{H}{N}} \quad , \quad f = \sqrt[3]{\frac{560}{S+160}}$$

(b) The notations used in (a) above are defined as follows:

f	=	Material factor for steel main shafting and quill shafts.
S	=	Specified minimum tensile strength of steel of which the shafting is designed to be made, in N/mm^2
d_1	=	Diameter required for intermediate shaft, in mm.
H	=	Maximum continuous power transmitted by intermediate shafts, in kW.
N	=	Corresponding revolutions per minute of shafts at maximum continuous power, in rpm.
F	=	95 for turbine installations, electric propulsion installations and diesel engine installations through reduction gears or with slip type couplings, and 100 for other diesel engine installations.
C_1	=	1.00 for ocean going and greater coasting service ships, and
	=	0.96 for other ships.
K_1	=	Constant as given in Table III 6-1.

Table III 6-1
Constant K_1 for Intermediate Shafts

K_1	Description of Intermediate Shafts Designed
1.0	Plain shafts with integral coupling flanges or shrink fit couplings.
1.1	Shafts with keyways where the fillet radii in the transverse section of the bottom of the keyway are to be not less than 0.0125 d_1,, and Shafts with transverse or radial holes where the diameter of the hole is not greater than 0.3 d_1 (d_1 is determined with K_1 = 1.0).
1.2	Shafts with longitudinal slots having a length of not more than 1.4 d_1 and a width of not more than 0.2 d_1 (d_1 is determined with K_1 = 1.0)

(c) The diameter of the thrust shaft is not to be less than that obtained from the following formula:

$$d_2 = 1.1 \cdot f \cdot F \cdot C_1 \sqrt{\frac{H}{N}}$$

(d) Outside a length equal to the thrust shaft diameter from the collars, the diameter may be gradually reduced to that required for the intermediate shaft with K_1 = 1.0.

(e) The notations used in (c) above are defined as follows:

d_2 = Diameter required at the collars of the thrust shaft transmitting torque or in way of the axial bearing where a roller bearing is used as a thrust bearing, in mm.

f, F, C_1, H and N are as defined in 6.1.4(b) of this Part.

(f) The diameter of the propeller shaft and tube shaft is not be less than that obtained from the following formula:

$$d_3 = f\,K_2 \sqrt[3]{\frac{H}{N}}$$

(g) Propeller shafts which run in stern tubes and tube shafts may have the diameter forward of the length to the forward stern tube seal gradually reduced to the diameter of the intermediate shaft. Abrupt changes in shaft section at the propeller shaft or tube shaft to intermediate shaft couplings are to be avoided.

(h) The notations used in (f) above are defined as follows:

d_3 = Diameter required for propeller shaft and tube shaft, in mm.

f = Material factor as defined in 6.1.4(a) of this Part

K_2 = Constant as given in Table III 6-2 for propeller shaft, and 115 for tube shaft.

Table III 6-2
Constant K_2 for Propeller Shafts

<table>
<tr><th rowspan="2">Zone</th><th rowspan="2">Design</th><th colspan="3">Category</th></tr>
<tr><th>A</th><th>B</th><th>C</th></tr>
<tr><td rowspan="2">I</td><td>(1)</td><td>122</td><td rowspan="2">126</td><td rowspan="2">132</td></tr>
<tr><td>(2)</td><td>126</td></tr>
<tr><td>II</td><td>–</td><td rowspan="2">115</td><td rowspan="2">115</td><td rowspan="2">121</td></tr>
<tr><td>III</td><td>–</td></tr>
</table>

Where:

The zone, design and category to be applied for the calculation of the propeller shaft diameter are defined as follows:

1. Zone

I. For the shaft diameter extending over the portion from immediatley forward of the forward face of the propeller hub (or the forward face of the propeller shaft flange, if applicable) to a length not less than that to the forward edge of the bearing immediately forward the propeller or 2.5 d_3, whichever is the greater.

II. For the shaft diameter extending over the portion from the forward end of the length required by Zone I to the forward end of the stern bearing seal.

III. For the shaft diameter extending over the portion from the forward end of the forward stern tube seal to the coupling flange joining to the intermediate shaft.

2. Design

(1). For a shaft carrying a keyless propeller, or where the propeller is attached to an integral flange.

(2). For a shaft carrying a keyed propeller.

3. Category

A. Where a shaft is fitted with a continuous liner or is oil lubricated and provided with an approved type of oil sealing gland.

B. For a shaft of corrosion resistant material which is exposed to sea water.

C. For a shaft of non-corrosion resistant material which is exposed to sea water.

(i) Where propeller is force fitted on the propeller shaft, the fixing part is to be of sufficient strength against torque to be transmitted.

(j) Where a key is provided to fixing part, ample fillets are to be provided at the corners of the keyway and key is to have a true fit in the keyway. The fore end of keyway on the propeller shaft is to be rounded smoothly for avoiding an excessive stress concentration.

(k) The diameter of shafts transmitting power to generators or essential auxiliary machinery is to, in principle, conform to the requirements in 6.1.4(a)(b).

6.1.5 Shafting Accessories

6.1.5.1 Coupling bolts

(a) The diameter of fitted coupling bolts at the joining face of the coupling is not to be less than that obtained from the following formula:

$$d_4 = 0.65 \sqrt{\frac{d_1^3 (S_s + 160)}{Z D S_b}}$$

(b) If the transmission of the torque is based on friction between the mating surfaces of flange couplings, special consideration will be given to the design.

6.1.5.2 Coupling flanges

(a) The thickness of coupling flanges at the pitch circle of the bolt holes is not to be less than the required diameter of the coupling bolts as obtained from 6.1.5.1 (a) of this Part paying due regard to the minimum tensile strength of the steel of which the coupling flanges are designed to make. However, the thickness of the coupling flange is in no case to be less than 20% of the required diameter of the corresponding shaft.

(b) The fillet between coupling flange and shaft is to have a radius not less than 8% of the diameter of the corresponding shaft provided that recesses are to be avoided in way of bolt heads and nuts.

(c) Where a coupling flange is separate from the shafts, the arrangement is to be submitted for consideration. In this case, the coupling flange is to be of cast or forged steel and provision is to be made for the coupling to resist any twisting force and astern pull.

6.1.5.3 Shaft liners

(a) The thickness of shaft liner to be fitted to the tube shaft or propeller shaft is not to be less than that obtained from the following formula:

$t_1 = 0.03d_3 + 7.5$

$t_2 = 0.75\ t_1$

(b) The continuous liner is to be cast in one length. If it is made of 2 or more lengths, the jointing of the separate pieces is to be made with an approved method of fusion through the whole thickness of the liner or by an approved rubber lining. The joint between the separate liners is not to be located in way of the bushing or stern gland.

(c) The liner is to be securely shrunk on or forced upon the shaft by pressure. Locking device is not to be used for securing purpose. 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 force with a compound insoluble in water and non-corrosive.

(d) The bronze liner is to be made of a high grade composition, free from porosity and other defects, and is to prove tight under hydraulic pressure test as stated in 6.9.1 of this Part.

6.1.5.4 Aft. stern tube bearings

(a) The stern tube gland for propeller shaft and the shaft tunnel bearings for intermediate shafts are to be accessible at all times without shifting of cargo and to be provided with efficient means of lubrication.

(b) The length of bearing in the stern tube next to and supporting the propeller is to be as follows:

(i) Oil lubricated bearings of white metal

(1) The length of white metal lined bearings is to be not less than 2.0 times the rule diameter of the shaft in way of the bearing.

(2) The length of the bearing may be less provided the normal bearing pressure is not more than 8 bar as determined by static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft. However, the minimum length is to be not less than 1.5 times the actual diameter.

(3) Where the stern bearing is white metal lined and lubricated by oil or grease, the following are to be complied with:

(A) The oil sealing gland is to be of an approved type and capable of operating under the various sea water temperatures it may be subject to in service.

(B) The space between propeller shaft and stern tube is to be kept filled with oil or grease of good quality. The stern bearing is to be provided with proper grooves for oil, air and possible accumulation of dirt. Pipes and cocks for supplying and drawing off oil as well as for air are to be fitted.

(C) Where lubrication is made by means of gravity, the lubricating oil tank is to be located above the load water line and provided with a low level alarm.

(D) Means are to be provided, where necessary, for cooling the lubricating oil of the stern bearing.

(ii) Oil lubricated bearings of synthetic rubber, reinforced resin or plastic materials

(1) For bearings of synthetic rubber, reinforced resin or plastics materials which are approved for use as oil lubricated stern bush bearings, the length of the bearing is to be not less than 2.0 times the rule diameter of the shaft in way of the bearing.

(2) The length of bearing may be less provided the nominal bearing pressure is not more than 6 bar as determined by static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft. However, the minimum length is to be not less than 1.5 times the actual

diameter.Where the material has proven satisfactory testing and operating experience, consideration may be given to an increased bearing pressure.

(3) Means of lubrication are to be submitted for special consideration by the Society.

(iii) Water lubricated bearings of lignum vitae

(1) Where the bearing comprises staves of wood (known as lignum vitae), the length of the bearing is to be not less than 4.0 times the rule diameter of the shaft in way of the bearing.

(2) Adequate means are to be provided to supply ample amount of clean water for lubrication and cooling. Forced water lubrication by a suitable independent pump or from other pressure source is recommended to be provided when the required shaft diameter exceeds 350 mm.

(iv) Water lubricated bearings of synthetic material

(1) Where the bearing is constructed of synthetic materials which are approved for use as water lubricated stern bush bearings such as rubber or plastics the length of the bearing is to be not less than 4.0 times the rule diameter of the shaft in way of the bearing.

(2) For a bearing design substantiated by experiments to the satisfaction of the Society consideration may be given to a bearing length not less than 2.0 times the rule diameter of the shaft in way of the bearing.

(3) Means of lubrication are to be submitted for special consideration by the Society.
A reduction of the bearing length may be approved if the bearing is shown by means of bench tests to have sufficient load bearing capacity.

6.1.5.5 The notations used in 6.1.5.1 and 6.1.5.3 above are defined as follows:

d_4 = Diameter required for coupling bolt, in mm.

d_1 = Diameter of intermediate shaft calculated with

K_1 = 1 in 6.1.4(b), in mm.

S_s = Specified tensile strength of intermediate shaft material taken for the calculation in 6.1.4(a) N/mm^2.

S_b = Specified tensile strength of bolt material N/mm^2, while in general $S_s \le S_b \le 1.7S_s$, and the upper limit of the value of S_b used for the calculation is to be 1,000 N/mm^2.

Z = Number of bolts in one coupling.

D = Pitch circle diameter of the coupling bolt holes, in mm.

t_1 = Thickness required for shaft liner in way of stern bearing or gland, in mm.

t_2 = Thickness required for shaft liner outside portion of stern bearing and gland, in mm.

d_3 = As defined in 6.1.4(f) of this Part.

6.1.5.6 Protection for propeller shaft against corrosion

(a) The propeller shaft made of material which is not resistible to sea water is to be properly protected.

(b) The exposed steel of the propeller shaft is to be protected from the action of sea water by filling all spaces between propeller cap, hub and shaft with a suitable material.

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

6.2 Propellers

6.2.1 Scope

The requirements of this Chapter are applicable to the screw propellers. Where a design is proposed to which the following cannot be applied, special strength calculations are to be submitted for consideration.

6.2.2 Drawings and data

(a) The drawings of the propeller including spare one, if supplied, are to be submitted for approval. These drawings are to contain all the details necessary for examination in accordance with the Rules for design and materials as follows:

Type of the propelling machinery and power transmitted to the propeller speed consistent with the above mentioned power, maximum thrust, geometrical data of the propeller including number of blades, diameter, pitch, thickness and width at the various radii and rake, etc., material specifications and strength calculations.

(b) For the built-up propeller, the drawings submitted are to include, in addition, the number, position, diameter and threading characteristics of the studs as well as their material used. The securing arrangements of the studs are also to be given.

(c) For ships such as trawlers or tugs, for which several working ranges are provided, the above mentioned characteristics are to be indicated for these various ranges.

(d) In addition to the drawings and data required in 6.1.2(a) above, the following are to be submitted for controllable pitch propeller:

A diagram showing the variation of the maximum thrust of the propeller and of the corresponding pitch for the maximum power and number of revolutions of the main shafting, as a function of the ship's speed from zero to the maximum. The detail drawings of the pitch selection mechanism of the blades and corresponding control gear. The hydraulic piping control system, instrumentation and alarm system, and the strength calculations for internal mechanism.

6.2.3 Materials

(a) Casting for propellers and propeller blades are to comply with the requirements specified in 7.1.3, Part IV of the Rules for the Construction and Classification of Steel Ships.

(b) The specified minimum tensile strength of the castings, is to be not less than stated in Table III 6-3, the details of the chemical composition, mechanical properties and density are to be provided, together with results of fatigue tests in sea water in order to assign a value for allowable stress. Where propellers of carbon and low alloy steels are provided with an approved method of cathodic protection, special consideration will be given to the value of allowable stress.

Table III 6-3
Material for Propellers

Material	Specified minimum tensile strength N/mm^2	Density g/cm^3	Allowable Stress N/mm^2
Carbon steels	400	7.9	20.6
Low alloy steels	440	7.9	20.6
13% chromium stainless steels	540	7.7	41
Chromium-nickel Austenitic stainless steels	450	7.9	41
Duplex stainless steels	590	7.8	41
Grade Cu 1 Manganese bronze (high tensile brass)	440	8.3	39
Grade Cu 2 Ni-Manganese bronze (high tensile brass)	440	8.3	39
Grade Cu 3 Ni-Aluminium bronze (high tensile brass)	590	7.6	56
Grade Cu 4 Mn-Aluminium bronze (high tensile brass)	630	7.5	46

(c) Where propeller materials are proposed for which details of service experience is not available, their suitability are to be especially demonstrated to the Society.

(d) The materials for the propellers, the blade attaching studs, and the components and pitch selection mechanism for controllable pitch propellers are to be tested and inspected in the presence of the Surveyor to comply with the requirements in Part XI of the Rules for the Construction and Classification of Steel Ships or to the requirements of the specifications approved in connection with the design.

6.2.4 Strength calculations

(a) For propeller blade of conventional design, the required blade thickness is to comply with the following formula:

$$t = C_1 K_m \sqrt{k_1 k_2 \frac{H \times 10^6}{NBZS}}$$

Where

t = Required propeller blade thickness measured parallel to the axis of propeller shaft rotation, in mm. For solid propeller, thickness at 0.25R and 0.6R are to be calculated, for detachable propeller thickness at 0.35R and 0.6R are to be calculated, the calculated thickness is not including the rounded fillet.

C_1 = $1 + \dfrac{E}{D} + \dfrac{N}{10,000}$

E = Blade rake of aft, measured as the distance between the tip of the blade and a perpendicular where the line of the blade face intersects with the axis of the propeller, in mm.

D = Propeller diameter, in mm.
N = Revolutions of the propeller, in rpm.
K_m = Material constant.
= 16.92 for copper alloy.
= 18.54 for cast steel.
= 21.40 for cast iron.

$k_1 = 1 + 4\left(\frac{E}{D}\right)^2$, rake factor

k2 = Pitch factor

$= 2.78\frac{D}{P} + 1.72$ for 0.25R section
$= 2.42\frac{D}{P} + 1.51$ for 0.35R section
$= 0.82\frac{D}{P} + 0.51$ for 0.6R section

P = Propeller pitch measured on the face. For the propeller having a non-uniform pitch the mean effective pitch $\frac{\Sigma\,(RBP)}{\Sigma\,(RB)}$ is to apply. Where R, B and P are the dimensions at the radius considered, in mm.
B = Developed blade width measured at the blade face at radius considered, in mm.
Z = Number of blades:
S = Minimum tensile strength of propeller material, in N/mm^2.

(b) Skewed propeller

(i) The maximum skew angle of a propeller blade is defined as the angle, in projected view of the blade, between a line drawn through the blade tip and the shaft centreline and a second line through the shaft centreline which acts as a tangent to the locus of the mid-points of the helical blade sections, see Fig. III 6-1.

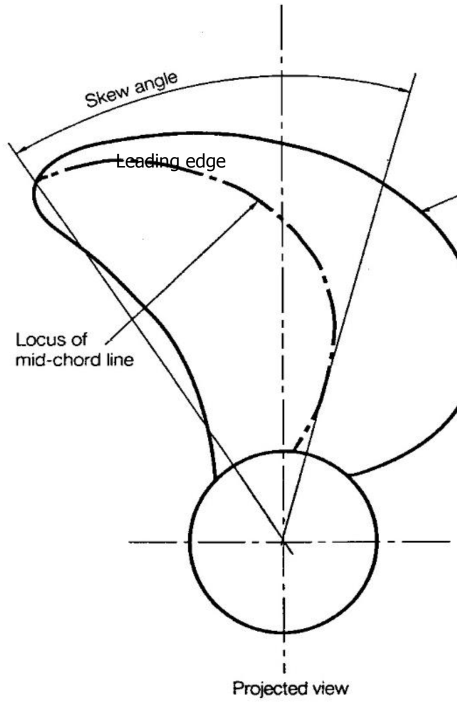


Fig. III 6-1
Definition of Skew Angle

(ii) For propellers having a skew angle in excess of 25°or greater, but less than 50°, the blade thickness at the radius of 0.25 R and 0.6 R are not to be less than the values obtained by multiplying the value [t] calculated by the formula in 6.2.4(a) above, by the coefficients $K_{0.25}$ and $K_{0.6}$ respectively given in the formulas below:

$$K_{0.25} = 0.75(1 + 0.1\theta_S)^{0.25}$$

$$K_{0.6} = 0.54(1 + 0.1\theta_S)^{0.5}$$

Where θ_S = proposed skew angle in degrees as defined in 6.2.4 (b)(i)

The thickness of the remaining radii are to be joined by a fair curve and the sections are to be of suitable aerofoil section.

(iii) Results of detailed calculations where carried out, are to be submitted.

(iv) For propellers having a skew angle in excess of 50, a detailed blade stress computation is to be submitted for special approval.

(c) A reduction in the blade root thickness below that required in 6.2.4 (a) above may be permitted on the blade of special form where the blade rigidity, flow characteristics, distribution of the pressure over the blade, etc., are such that a sufficient margin of safety based on the tensile strength is maintained.

(d) For controllable pitch propellers, the pitch P entered in the formula of 6.2.4 (a) above for calculating the blade root thickness required for tugs and similar ships in the pitch corresponding to the maximum propeller thrust measured at the bollard pull test. When this pitch is not communicated to the Society, a root thickness based on a pitch ratio $\frac{P}{D}$ = 0.5 will be assumed for the calculations.

6.2.5 Blade attaching studs

(a) The thread bottom diameter of the blade attaching studs for the built-up propeller is not to be less than obtained by the following formula:

$$d = 1.17K_2\sqrt{\frac{10^6 C_2 H}{N n_f d_p Z} + \frac{WDN^2}{10^6 n_t}}$$

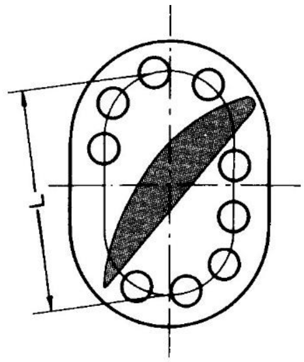
Where:

d = Diameter of the blade attaching studs, measured to the bottom of the thread, in mm.

d_p = Pitch circle diameter of the studs or where the studs are not arranged in a circle, 85% of the greater distance L (mm) between the studs on the face side and the back of the propeller blade, in mm, see Fig. III 6-2.

W = Mass of the blade including the flange, in kg.

n_t = Total number of studs attaching each blade.



d_p = 0.85L

Fig. III 6-2
Blade Attaching Studs

n_f = Number of studs on the face side of the blade.

C_2 = Factor as given in the following Table

$\frac{P}{D}$	C_2
0.5	10.74
0.6	9.11
0.7	7.88
0.8	6.93
0.9	6.25
1.0	5.71
1.1	5.30
1.2	5.03
1.3	4.89

K_2 = $0.26\sqrt{\frac{590}{S}}$

S = Minimum tensile strength of material of which the stud is designed to be made, in N/mm^2.

H, N, Z are the same as defined in 6.2.4 (a)

(b) The plain machined portion of the propeller blade attaching stud may be reduced to 90% of the stud diameter measured to the bottom of the thread.

(c) The attaching studs are to be prevented from accidentally becoming loose.

6.2.6 Construction

6.2.6.1 Fitting of propeller

(a) Where bronze propeller is force fitted on the steel propeller shaft without use of key, the minimum and maximum limits of pull-up length are to be given by the following formulae. For the taper of more than 1/15, these limits of pull-up length are to be subject to special consideration.

$$L_{min} = 206 \times 10^6 K_1 \frac{H}{NA \tan\theta} + 2.75 \times 10^{-6} \frac{d_m}{\tan\theta}(35 - t)$$

$$L_{max} = 0.35 \frac{K_1}{K_2} \bullet \frac{d_m Y}{\tan\theta} - 2.75 \times 10^{-6} \frac{d_m}{\tan\theta} \bullet t$$

Where

L_{min}	=	Minimum pull-up length at temperature t, in mm.
L_{max}	=	Maximum allowable pull-up length at temperature t, in mm.
H	=	Maximum continuous output of the engine driving the propeller, in kW.
N	=	Revolutions of propeller, in rpm.
A	=	100% theoretical contact area between boss and shaft, as read from drawings and disregarding oil grooves, in mm^2.
d_m	=	Diameter of propeller shaft at the midpoint of the contact taper in axial direction, in mm.
D_m	=	Mean outer diameter of propeller boss at axial position corresponding d_m, in mm.
d_o	=	Diameter of hole if the propeller shaft is hollowed, in mm.
a	=	D_m/d_m, diameter ratio.
b	=	d_o/d_m, diameter ratio.
θ	=	Half taper of propeller shaft, e.g. taper = 1/15, $\tan\theta$ = 1/30.
t	=	Temperature of propeller shaft and boss at time of fitting propeller, in °C.
Y	=	Propeller material's yield point, or 0.2% proof stress, in N/mm^2.

$$K_1 = 8.47 \times 10^{-6} \frac{a^2+1}{a^2-1} + 4.83 \times 10^{-6} \frac{1+b^2}{1-b^2} + 1.39 \times 10^{-6}$$

$$K_2 = \frac{\sqrt{3a^4+1}}{a^2-1}$$

(b) The propeller hub is to be adequately fitted on the propeller shaft cone or securely bolted on the propeller shaft.

(c) Normally, the taper in diameter of the propeller shaft cone is not to be steeper than 1/10. The whole surface of the propeller shaft cone and the propeller hub bore are to be uniformly fitted.

(d) suitable pulling force is to be applied for the fitting of the propeller to the shaft cone so that the contact friction developed between the surfaces is sufficient to resist the full torque of the shaft. For keyless bore propeller, the fitting of hub in accordance with the calculations specified in 6.2.6.1 of this Part is to be submitted for approval. A copy of the fitting curve relative, to temperature and means for determining any subsequent movement is to be placed on board.

(e) The forward taper edge of the propeller hub is to be rounded.

(f) Where the propeller hub is bolted to the shaft, the appropriate design and calculations are to be submitted to demonstrate that their attaching bolts and pins are in sufficient strength and suitable coupled.

(g) The propeller hub is not to be fitted or removed by means of local heating.

(h) Prior to final pull-up, the contact area between the mating surfaces is to be checked and is not to be less than 70% of the theoretical contact area. Non-contact bands extending circumferentially around the boss or over the full length of the boss are not acceptable.

(i) After final pull-up, the propeller is to be secured by a nut on the propeller shaft. The nut is to be secured to the shaft.

(j) The external diameter of the thread for propeller retaining nut is not to be less than 60% of the required diameter of the propeller shaft.

6.2.6.2 Controllable pitch propeller

(a) The controllable pitch propeller is to be arranged so that the blades can be reliably held in any pitch position.

(b) The hydraulically operated pitch selection mechanisms are to be provided with 2 independent, mechanically driven pump units. The propulsion installations up to 150 kW need only be fitted with one mechanically driven pump unit, provided an additional hand operated pump is available by means of which the blades can be moved from the ahead to the astern positions in a sufficiently short period of time.

(c) The controllable pitch propeller system is to be provided with an indicator in the engine room showing the actual position of the blades. Where the controllable pitch propeller system is controlled from the bridge, a similar indicator is also to be provided therein and the system is to be capable of being operated from the engine room in emergency.

(d) Visual and audible alarms are to be provided in the engine room control station to indicate the low hydraulic oil pressure, high hydraulic oil pressure and high hydraulic oil temperature.

6.3 Waterjet Propulsion Systems

6.3.1 Scope

Water-jet propulsion systems are to conform to requirements in this section, according to their design, in additional to the applicable requirements in this Chapter.

6.3.2 Drawings and data

Drawings and data to be submitted are generally as follows:

(a) Drawings and data for approval

- General arrangement plans showing details of shafting assembly indicating bearing positions, steering assembly, reversing assembly, shaft sealing arrangement assembly, longitudinal section of the complete waterjet unit.

- Detailed and dimensioned plans indicating scanting, materials of construction and, where applicable, surface finish of the following:

(i) The arrangement of the system, including the intended method of attachment to the hull and building, tunnel geometry, shell openings, method of stiffening, reinforcement, etc.

(ii) The torque transmitting components, including the shafting system, impeller and stator if fitted.

(iii) The steering components, together with a description and line diagram of the control circuit. This is to include steerable exit water-jet nozzles where fitted.

(iv) The components of the retractable buckets where these are used for providing astern thrust.

(v) The bearing or bearings absorbing the thrust and supporting the impeller, together with the method of lubrication.

(vi) The details of any shafting support or guide vanes used in the water-jet system.

- Shafting arrangement (showing the arrangements, shapes and constructions of the main propulsion machineries, reduction gears, clutches, couplings, main shafts, bearings. thrust bearings, sealing devices and impellers).

- Details of water intake duct.

- Construction of impeller (showing the detailed blade profiles, the maximum radius of the impeller from the center of the main shaft, number of blades and material specifications).

- Details of bearings, thrust bearings and forward sealing devices of the main shaft.

- Details of deflectors.

- Details of reversers.

- Diagram of hydraulic piping system.

- Calculation sheets of torsional vibration of main shaft.

(b) Drawings and data for reference

- Calculation sheets of bending natural frequency when bending vibration due to self-weight is expected.
- Strength calculation sheets for deflectors and reversors.
- Strength calculations based on fatigue considerations incorporating the maximum continuous torque rating and the most onerous operating condition.
- Calculations supporting the connection method of the impeller to the shaft including details of the fit, push-up, securing, bolting arrangements, etc. In addition, where lengths of shafts are joined using couplings of the shrunk element type, full particulars of the method of achieving the grip force.
- Others deemed necessary by this Society.

6.3.3 Materials

The materials of parts of the water-jet propulsion system are suitable for respective uses intended, and the following essential components are to comply with the requirements in Part XI of the Rules for the Construction and Classification of Steel Ships:

(a) Main shaft

(b) Shaft coupling and coupling bolts

(c) Impeller

(d) Water intake duct, nozzle and impeller casing which are composing a part of shell plating

6.3.4 Construction

(a) The following design load conditions are to be considered:
maximum thrust force ahead.
maximum side force and moment,
maximum reversing force and moment.

(b) The supporting area of the stern is to be adequately strengthened to withstand the above design load.

(c) Support for shaft bearing in way of duct penetration is to be adequately strengthened against primary structure.

(d) Others deemed necessary by this society.

6.3.5 Monitoring and alarms

In addition to the requirements of Rules for Steel Ships Part VIII Ch.3, alarms and monitoring requirements are indicated as follwed:

(a) An indication of the angular position of the nozzle is to be provided at each station from which it is possible to control the direction of thrust from the units.

(b) An indication of both the required and actual reversing bucket position is to be provided at each station from which it is possible to control the reversal of thrust.

(c) All alarms associated with water jet unit faults are to be indicated individually at the control stations and in accordance with Table III 6-4 and the alarm system specified by Rules for Steel Ships Part VIII Ch.3.

Table III 6-4
Alarms

Item	Alarm	Note
Hydraulic system pressure	Low	
Hydraulic oil supply tank level	Low	
Hydraulic oil temperature	High	Where an oil cooler is fitted
Lubricating oil temperature	High	
Lubricating oil pressure	Low	In forced lubrication systems
Lubricating oil tank level	Low	Where a tank is provided
Ratio of jet rpm/vessel speed	High	Only if installed power per jet > 4 MW
Control system failure	Fault	Includes follow-up failure of steering or reversing system
Control system power supply	Failure	

6.4 Torsional Vibration of Shafting

6.4.1 Scope

(a) The requirements of this Chapter are applicable to the torsional vibration of main propulsion shafting system and auxiliary diesel engines for essential service.

(b) The torsional vibration of novel designed engine or some parts of the installation such as gear, chain, cam mechanism or elastic coupling etc. is to be submitted for special approval.

6.4.2 Torsional vibration calculations

(a) The torsional vibration calculations are to be submitted for approval are to include the following:
   (i) The basic data for establishing such calculation and more particularly the dynamic characteristics of the equivalent system of the whole installation, i.e. machinery, shafting and propeller etc.
   (ii) The table of natural frequencies for 1-, 2- and possibly more than 2-node modes of vibration.
   (iii) The vector summations for critical speeds of all significant orders up to 115% of rated speed.
   (iv) The arrangement of crank and particulars of fir-ing order of the engine.
   (v) The stress estimates for critical speeds whose severity approaches or exceeds the allowable limit specified.

(b) The similar calculation is to be made for the spare propeller, if different from the working propeller.

(c) The various working ranges of the propelling machinery when running continuously are to be specified. In particular, for the trawler, the trawling speed is to be stated. The idling speed of the machinery is also to be mentioned.

(d) For the installation including torsional vibration damper, the characteristics of this damper and the data permissible to check its efficiency is to be submitted together with the torsional vibration calculations for approval.

(e) When the additional stress determined by the above calculations indicates significantly higher, the stress value is to be presumed and investigated by ser-vice experience of the similar previous installation.

6.4.3 Torsiograph test

(a) The torsiograph test is required to verify the above calculations and to assist in determining the barred range of continuous running in the case of the new construction or conversion where the torsional critical speed arrangement differs significantly from the previous installation.

(b) The torsiograph test is to be carried out in the presence of the Surveyor.

(c) In the following cases, the omission of torsiograph test may be considered:
   (i) The main propelling machinery having no vibration damper together with shafting and propeller is similar to the installation for which torsiograph record has previously been taken in the presence of the Surveyor and verified that the noteworthy torsional critical speeds do not exist between the interval from 80% to 105% of the rated speed.
   (ii) For installations having some parts of the shafting and propeller slightly different from the one for which torsiograph record has previously been taken and proved to be satisfactory the additional stress due to torsional vibration can easily be presumed and verified that the noteworthy torsional critical speeds do not exist between 80% to 105% of the rated speed.

6.4.4 Allowable stress

(a) For continuous operation the permissible stresses due to alternating torsional vibrations are not to exceed the values given by the following formulae:

(i) for $\lambda < 0.9$

$$\tau_1 = \pm \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet (3 - 2\lambda^2)$$

(ii) for $0.9 \leq \lambda < 1.05$

$$\tau_1 = \pm \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet 1.38$$

where:

τ_1 = Permissible stress due to torsional vibrations for continuous operation, in N/mm^2;

σ_B = Tensile strength of the shaft material, in N/mm^2;

c_K = Factor for different shaft design features, see Table III 6-5;

Table III 6-5
c_K – Factors

for intermediate shafts with	integral coupling flanges	1.0
	shrink fit couplings	1.0
	keyways	0.6
for thrust shafts external to engines	on both sides of thrust collar	0.85
	in way of axial bearing where a roller bearing is used as a thrust bearing	0.85
for propeller and crankshafts	for K_2 = 122 and 126 see Table IV 6-2	0.55

c_D = Size factor;

= $0.35 + 0.93\, d^{-0.2}$

d = Shaft diameter, in mm;

λ = Speed ratio;

= n/n_1

n = Speed in question, in min^{-1};

n_1 = Rated speed, in min^{-1}.

(b) For transient running the permissible stresses due to the alternating torsional vibrations are not, in any case, to exceed the values given by the following formula:

$$\tau_2 = 1.7\,\tau_1 / \sqrt{c_k} \quad \text{for} \quad \lambda \leq 0.8$$

where:

τ_2 = Permissible stress due to torsional vibrations for transient running, in N/mm^2.

(c) In general, the tensile strength of the steel used is to be between 400 and 800 N/mm. For the calculation of the permissible limits of stresses due to torsional vibrations, σ_B is not to be taken as more than 600 N/mm.

(d) The allowable limit of additional stresses in generator engine crankshafts, where the critical occurs between 0.95 to 1.05 times the rated speed, the additional stresses are not to exceed the values obtained from the following

$\tau_1 = 31\ N/mm^2$

(e) The values given in the above formulae are to be applied for the shafting and crankshaft of ordinary design without having excessive stress concentration, otherwise special consideration is to be made. Stresses are nominal values based on diameter of crank-pins, or on the minimum propeller shaft diameter between the big end of the taper and the forward stern gland.

(f) Where shafting is made of materials other than steel, the limit of additional stress will be fixed by the Society after examination of the result of the fatigue test carried out on the said materials.

(g) For propeller shafts of non-corrosion resistant material which are exposed to sea water, the allowable stress limit is to be of 70% of the value specified in the formulae in (a).

6.4.5 Barred range

(a) When the result of calculation or torsiograph record shows the critical speed for which the additional stress due to torsional vibration exceeds the value specified in (a) and (b) of 6.4.4, the corresponding speed range is to be barred out for continuous running. This barred range is to be crossed out in red on the tachometer and a warning notice is to be fitted near the engine control stand.

(b) The torsional critical speed resulting from additional stress exceeds the allowable limit under intermittent operating condition is not to be crossed over even in intermittent running except if proof is given that the maximum value of such stress cannot be reached during operation in which the engine speed rapidly passes through the critical range.

(c) The details of barred range together with mode of vibration, critical speed, maximum additional stress and the part of shafting where this maximum additional stress occurs are to be registered in the Machinery Classification Survey Report.

(d) When the propeller is driven through reduction gears, or when the auxiliary equipment such as a blower is driven through gears, a barred range is to be provided at the critical speed if the gear tooth chatter occurs during continuous operation at the critical range.

(e) The width of barred range is to take into consideration the breadth and severity of the critical, but to be extended at least as follows:

(i) For 1-node vibration

$$\frac{16 \bullet \eta_c}{18-\lambda} \leq Z \leq \frac{(18-\lambda) \bullet \eta_c}{16}$$

(ii) For 2-node vibration

$$\frac{1}{1.05}\ \eta_c \leq Z \leq 1.05\ \eta_c$$

where:

Z = Barred range, in $\min^{-1}$.

η_c = Critical speed, in $\min^{-1}$.

(f) When the additional stress due to torsional vibration exceeds the allowable limit specified in 6.4.4 and a barred range is not acceptable, the dynamic system is to be redesigned, or dampering or detuning arrangements provided to remove the critical speed from the operating range or to reduce the magnitude of the vibration stress. Where vibration dampers or flexible couplings are fitted, it may be required that torsiograph tests be carried out to verify their efficient.

6.5 Lateral (Whirling) Vibrations

The designer or the builder is to evaluate all main propulsion shafting systems to ensure that the amplitudes of lateral (whirling) vibration are of acceptable magnitude throughout the engine operating speed range, unless experience with similar propulsion system installations makes it unnecessary. When on basis of lateral vibration calculations it is proposed by the designer or builder to provide barred speed ranges within the engine operating speed range, the calculations are to be submitted for information. The barried speed ranges due to lateral vibration are to be verified and established by measurement. Calculation of the lateral vibration characteristics of shafting systems having supports outboard of the hull or incorporating cardan shafts are to be submitted. The calculations of lateral vibration, taking account of bearing, oil-film (where applicable) and structural dynamic stiffnesses, are to investigate the excitation frequencies which may result in significant amplitudes within the speed range, and are to indicate relative deflections and bending moments throughout the shafting system.

6.6 Axial Vibrations

(a) The designer or the builder is to evaluate all main propulsion systems to ensure that the amplitudes of axial vibration are of acceptable magnitude 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 propulsion system installations makes it unnecessary. The axial vibrations may be controlled by axial vibration de-tuners 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 basis of axial vibration calculations it is proposed by the designer or builder 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.

(b) Calculation of axial vibration natural frequency are to be carried out using appropriate techniques, taking into account the effects of flexibility of the thrust bearing, for shaft systems where the propeller is:

(i) Driven directly by a reciprocating internal combustion engine.

(ii) Driven via gears, or directly by an electric motor, and where the total length of shaft between propeller and thrust bearing is in excess of 60 times the intermediate shaft diameter.

(c) Where an axial vibration damper is fitted, the calculations are to consider the effect of a malfunction of the damper.

6.7 Propulsion Shafting Alignment

6.7.1 General

Propulsion shafting is to be aligned with the location and spacing of the shaft bearings being such as to give acceptable bearing reactions and shaft bending moments for all conditions of ship loading and operation. The designer or the builder is to evaluate the propulsion shafting system taking into consideration any forces or factors which may affect the reliability of the propulsion shafting system including weight of the propeller and shafts, hydrodynamic forces acting on the propeller, number of propeller blades in relation to diesel engine cylinders, misalignment forces, thermal expansion, flexibility of engine and thrust bearing foundations, engine induced vibrations, gear tooth loadings, flexible couplings, effect of power take-off arrangements from the propulsion shafting system driving auxiliaries, etc., as applicable, as well as any limits for vibration and loadings specified by the equipment manufacturers.

6.7.2 Shafting alignment calculation is to be submitted for approval for the following shafting systems where the tailshaft has a diameter of 250 mm or grater in way of the aftermost sterntube bearing.

(a) The all geared installations.

(b) The installations with one shaftline bearing, or less, inboard of the forward sterntube bearing.

(c) The prime movers or shaftline bearings are installed on resilient mountings.

6.7.3 The shafting alignment calculations are to take into account.

(a) The thermal displacements of the bearings between cold static and hot dynamic machinery conditions.

(b) The buoyancy effect of the propeller immersion due to the craft's operating draughts.

(c) The effect of predicted hull deformations over the range of the craft's operating draughts, where known.

(d) The gear forces, where appropriate.

(e) For multi-engined installations, possible contributions in the mode of operation;

(f) The propeller offset thrust effects, where applicable.

(g) The bearing loading in the horizontal plane, where appropriate.

(h) The bearing weardown, where applicable, and its effect on the bearing loads.

6.7.4 The shafting alignment calculations are to state:

(a) The expected bearing loads at light and normal ballast, fully load and any other draughts deemed to be part of the craft's operating profile, in cold and hot, static and dynamic conditions for the machinery.

(b) The bearing influence coefficients and the deflection, slope, bending moment and shear force along the shaftline.

(c) The details of propeller offset thrust effects, where employed in calculation.

(d) The details of proposed slope-bore of the aftermost sterntube bearing, where applicable.

(e) The manufacturer's specified limits for bending moment and shear force at the shaft couplings of the gearbox and prime movers.

(f) The estimated bearing weardown rates for water or grease-lubricated sterntube bearings;

(g) The origin of findings where the effect of hull deformation has been considered by finite element calculations or measured results from sister or similar craft have been used;

(h) The anticipated thermal rise of prime movers and gearing units between cold static and hot running conditions.

(i) The manufacturer’s allowable bearing loads.

6.7.5 A shafting alignment procedure is to be submitted for all main propulsion installations.

(a) The expected bearing loads at light and normal ballast, fully load and any other draughts deemed to be part of the craft’s operating profile, in cold and hot, static and dynamic conditions for the machinery.

(b) The maximum permissible loads for the proposed bearing deigns.

(c) The design bearing offsets from the straight line.

(d) The design gaps and sags;

(e) The location and loads for the temporary shaft supports.

(f) The expected relative slope of the shaft and the bearing in the aftermost sterntube bearing;

(g) The details of slope-bore of the aftermost sterntube bearing, where applied.

(h) The expected shear forces and bending moments at the forward end flange of the shafting system connecting to the gear output shaft or for direct-drive installations, to the prime mover output flange.

(i) The proposed bearing load measurement technique and its estimated accuracy.

(j) The jack correction factors for each bearing where the bearing load is measured using a specified jacking technique.

(k) The proposed shaft alignment acceptance criteria, including the tolerances.

(l) The flexible coupling alignment criteria.

6.8 Thrusters

6.8.1 Scope

The requirements of this Chapter for fixed or azimuth thruster units which are used for propulsion and steering, and also applies to transverse propulsion thrusters which are an aid to manoeuvring.

6.8.2 Drawings and data

The following plans are to be submitted.

(a) The general arrangement sectional assembly plan showing all the connections of the torque transmitting components from the prime mover to the propeller, including the nozzle ring structure and nozzle support struts if a nozzle is provided.

(b) The detailed and dimensional plans of the individual torque transmitting components.

(c) The schematic plans of lubricating and hydraulic systems with pipe material, relief valves and working pressure.

6.8.3 Materials

The materials used in the construction are to be manufactured and test in accordance with Part XI of the “Rules for the Construction and Classification of Steel Ships”.

6.9 Tests and Inspections

6.9.1 Tightness tests

(a) Before being fitted, the propeller shaft liner, preferably in the finished state, is to be hydraulically tested for tightness to a pressure of 0.2 N/mm^2 in the presence of the Surveyor.

(b) Before being fitted, the cast stern tube, preferably in the completed state, is to be hydraulically tested for tightness to a pressure of 0.2 N/mm^2 in the presence of the Surveyor. Stern tubes made from other than casting are to be subjected to the tightness test on the hull spaces through which the stern tube passes.

(c) For oil sealing glands as stated in 6.1.5.4(b)(iii) of this Part, the leak test by oil pressure is to be made in the presence of the Surveyor after being installed on-board.

6.9.2 The following inspections for main shafting and accessories are to be carried out in the presence of the Surveyor complying with the relevant requirements of this Chapter:

(a) Material test, before shrinkage and final inspection for shafts and shafting accessories.

(b) Contacting test of the propeller hub to the cone of the propeller shaft prior to final pull-up.

(c) Assembling of propeller to the shaft on-board.

(d) Alignment of shafting on-board.

(e) Checking of fitness for coupling bolts.

(f) Measurement of vibrations on board.

6.9.3 All finished propellers are to be examined for material defects, measured for dimensional accuracy of diameter and pitch and static balancing at the manufacturer’s workshop.

6.9.4 The contact test of the propeller hub to the cone of the propeller shaft prior to final pull-up and their assembling on-board are subject to the Surveyor’s examination.

6.9.5 The controllable pitch propeller system is subject to pressure, tightness and operational tests at the manufacturer's workshop in the presence of the Surveyor.

6.9.6 Shop tests of keyless propellers

(a) It shall be demonstrated the bedding of the propeller with the shaft. Sufficient time is to be allowed for the temperature of the components to equalize before bedding. Alternative means for demonstrating the bedding of the propeller will be considered.

(b) Means are to be provided to indicate the relative axial position of the propeller boss on the shaft taper

6.9.7 Final fitting of keyless propellers

(a) After verifying that the propeller and shaft are at the same temperature and the mating surfaces are clean and free from oil or grease, the propeller is to be fitted on the shaft under survey. The propeller nut is to be securely locked to the shaft.

(b) The permanent reference marks are to be made on the propeller boss nut and shaft to indicate angular and axial positioning of the propeller

(c) The outside of the propeller boss is difficult to stamp with the following details:
   (i) For oil injection method of fitting, the start point load, in Newtons, and the axial pull-up at 0°C and 35°C, in mm.
   (ii) For the dry fitting method, the push-up load at 0°C and35°C, in Newtons.

(d) A copy of the fitting curve relative to temperature and means for determining any subsequent movement of the propeller are to be placed on board.

6.9.8 Final fitting of keyed propellers
The fit of the screwshaft cone to both the working and any spare propeller is to be carried out under survey. The satisfactory fit for keyed type propellers should show a light, overall marking of the cone surface with a tendency towards heavier marking in way of the larger diameter of the cone face. The final fit to cone should be made with the key in place.

6.9.9 Shop test of controllable pitch propellers

(a) The components of controllable pitch propellers are also subject to material tests, as in the case of solid propellers.

(b) Examination of all the major components including dimensional checks, hydraulic pressure testing of the hub and cone assembly and the oil distribution box, where fitted, together with a full shop trial of the completed controllable pitch propeller assembly, is to be carried out.

6.9.10 Shop tests and installation of waterjet systems

(a) The completed waterjet unit is to carry out a tightness test in which an internal hydrostatic pressure of 1,5 bar above the maximum working pressure of the unit is to be applied.

(b) As the impeller is fitted to the shaft using an interference fit, the bedding of the impeller with the shaft is to be demonstrated in the shop to the satisfaction of the Surveyor. Sufficient time is to be allowed for the temperature of the components to equalise before bedding. A contact marking between the bore of the impeller boss and the shaft surface of better than 80 per cent is to be demonstrated when the contact marking ink is spread thinly on the surface of the shaft. Alternative means for demonstrating the bedding of the impeller will be considered.

(c) Means are to be provided to indicate the relative axial position of the impeller boss on the shaft. Permanent reference marks are to be made on the impeller boss, shaft and any nut to indicate angular and axial positioning of the impeller.

(d) A copy of the fitting curve relative to temperature and means for determining any subsequent movement are to be placed on board.

(e) The impeller running clearances are to be checked following the installation of the unit in the ship.

(f) The thrust bearing clearances in the water-jet system are to be verified against the required design values. This is to be done following the installation of the unit in the ship.

6.9.11 The following tests and inspections of water-jets are to be carried out:

(a) The balancing of the impeller or the blades.

(b) Non-destructive examination of impeller blades and the principal component parts of the propulsion system.

(c) The quality of the fit of the impeller boss on the shaft taper.

(d) The fitting of the impeller to the shaft and its subsequent functional testing.

(e) The finished surfaces of the impeller boss, hub, conical bores, fillets, cones and blade surfaces are to be shown to conform to the tolerances specified on the impeller drawing.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART IV – BOILERS, PRESSURE VESSELS, THERMAL OIL HEATERS AND INCINERATORS

April 2018

List of major changes in Part IV from 2017 edition

Nil.

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART IV BOILERS, PRESSURE VESSELS, THERMAL OIL HEATERS AND INCINERATORS

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1.1 General 1

Chapter 1
General

1.1 General

1.1.1 The boilers, pressure vessels, thermal oil heaters and incinerators are to be in accordance with the Survey requirements of Part V of the "Rules for the construction and classification of steel ships."

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART V – PIPING AND PUMPING SYSTEMS

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Nil.

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2018

PART V
PIPING AND PUMPING SYSTEMS

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Chapter 1 General **1**

1.1 General 1

Chapter 1

General

1.1 General

1.1.1 Piping and pumping system are to be in accordance with Part VI of the “Rules for the construction and classification of steel ships”. The use of steel, copper or other non-aluminum pipes, valves and fittings will require special attention to avoid galvanic corrosion with dissimilar metals as specified in 1.1.2 below. Aluminum piping, valves and fittings will be subject to special consideration.

1.1.2 Piping systems are to consist of pipes and fittings of the same or comparable material. Piping runs of material not comparable with aluminum are to be isolated from the hull by suitable isolating brackets or insulation material. Where non-aluminum pipes pass through decks, bulkheads, tank tops, and shell plating, they are to be isolated from vessel’s structure with suitable insulations.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART VI – ELECTRICAL INSTALLATIONS

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PART VI
ELECTRICAL INSTALLATIONS

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1.4 Shore Power... 1
1.5 Cathodic Protection Installations 1

Chapter 1
General

1.1 General

1.1.1 In general, the electrical installations are to be in accordance with Part VII of the “Rules for the construction and classification of steel ships.”

1.1.2 The Society is prepared to give special consideration to the novel features of design in respect of the electrical installation based on the best information available at the time.

1.1.3 The electrical systems are to be isolated, from the hull at all times. Hull return systems are not to be used. Floating ground systems between the engine and related machinery components may be installed where required. In addition to power supply systems, attention for maintaining electrical isolation is to be given to communication devices, instrumentation and shore-power systems where used. See also 1.4.

1.2 DC Systems

1.2.1 Batteries generally are not to be grounded to propulsion engines or related machinery components. Where it is necessary for batteries to be grounded to the hull, the negative poles are to be connected to the hull. Batteries for engine starting may be grounded to the engine.

1.3 AC Systems

1.3.1 AC power supplies are to be isolated from the hull at all times. A high resistance continuity tester (such as a 90 volt DC battery, NE2 neon through 100K ohms) is to be carried on board in order that the electrical installation may be checked at the time of installation and at regular intervals to insure isolation of AC circuits.

1.4 Shore Power

1.4.1 The shore electrical power is to enter the ship through a 1:1 isolation transformer. Additional precaution to prevent electrolysis of the hull when docking are recommended.

1.5 Cathodic Protection Installations

1.5.1 Sacrificial Anode Systems

Sacrificial Anodes for use in sea water on aluminum hulls are to be effective for the hull material being protected. For proposed systems, the calculations, types, numbers, sizes and placement of anodes are to be submitted for review.

1.5.2 Impressed Current System

(a) Where impressed current cathodic protection systems are proposed, complete details, including types of anodes, voltages, arrangements and schematic of the wiring system, are to be submitted for review.

(b) Cables for cathodic protection systems are not to be run though oil tanks. Where passing through cofferdams, pump room and similar hazardous space, cables are to be enclosed in extra-heavy pipe, and are to be shielded from damage in cargo spaces and other areas where they may be exposed to mechanical damage. If piping used is not aluminum, it is to be isolated from the hull. It is recommended that impressed equipped with alarm devices to indicate inadequate or excessive current and reversed polarity.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART VII – AUTOMATIC OR REMOTE CONTROL AND MONITORING SYSTEMS

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PART VII
AUTOMATIC OR REMOTE CONTROL AND MONITORING SYSTEMS

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Chapter 1 General **1**

1.1 General 1

Chapter 1

General

1.1 General

1.1.1 In general, the automatic and remote control systems are to be in accordance with Part VIII of the “Rules for the construction and classification of steel ships.”

1.1.2 The Society is prepared to give special consideration to the novel features of design in respect of the electrical installation based on the best information available at the times.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART VIII – FIRE PROTECTION, DETECTION AND EXTINCTION

April 2018

List of major changes in Part VIII from 2017 edition

1.1.1	Amend No.1

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS
2018

PART VIII
FIRE PROTECTION, DETECTION AND EXTINCTION

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Chapter 1 General **1**

1.1 General 1

Chapter 1

General

1.1 General

1.1.1 In general, the aluminum light structure crafts which are capable of maximum speed as specified in 1.1.3 of Part II of the Rules and not to proceed in the course of their voyage more than the time as specified in 1.2.1(a) or 1.2.1(b) of Part II from a place of refuge, are to comply with Chapter 4, Chapter 7 and Chapter 8 of the "Rules for the Construction and Classification of High-Speed Craft", the aluminum passenger ships and the aluminum cargo ships are to be in accordance with Part IX of the "Rules for the Construction and Classification of Steel Ships."

1.1.2 The aluminum cargo ships below 500 gross tonnage or ships of restricted or special services or ships not propelled by mechanical means, the requirements in this Part may be modified to the satisfaction of the Society.

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART IX – MATERIALS

April 2018

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Nil.

RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART IX MATERIALS

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Chapter 1

General

1.1 General

1.1.1 In general, the materials of aluminum alloys for use in the construction of hull, machinery, pressure vessel, piping system together with their surrounding fittings in ships are to be in accordance with Part XI of the “Rules for the construction and classification of steel ships.”

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RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF ALUMINUM VESSELS 2018

PART X – WELDING

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PART X WELDING

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Chapter 1
General

1.1 General

1.1.1 The welding procedures, welder qualifications, operation controls, welding materials and welding constructions to be used in the welding of ship constructions, equipment, machinery, pressure vessels, piping, etc. of a ship classed or intended to be classed with the Society are to comply with the requirements of this Part, unless otherwise specially approved.

1.1.2 Requirements given in this Part are applicable to the welding of aluminum alloys made by manual, semi-automatic or automatic "electric arc fusion weld" process.

1.1.3 For "semi-automatic" and "automatic" welding, the welding gun provides continuous wire feed; for semi-automatic welding, the welding gun is held manually; and for automatic welding the welding gun is machine held with various degrees of controlled motion provided by the machine.

1.2 Tests, Inspections and Examinations

1.2.1 Prior to the operation of welding, the welding procedure, welder qualifications and welding materials are to be approved and subjected to a satisfactory test by the Society in accordance with the requirements given in this Part.

1.2.2 All welding work is to be carried out in accordance with the approved welding specifications and normal good welding practices under a satisfactory qualification control, and to be subjected to the satisfaction of the Surveyor.

1.2.3 All finished welds are to be tested, inspected and examined preferably by visual, non-destructive detection or hydrostatic methods, and, if deemed necessary, or as required in Parts IV and V of the Rules, by workmanship tests to ensure the welds until the satisfaction of the Surveyor.

1.3 Test Specimens and Mechanical Tests

1.3.1 Test specimens and mechanical tests of welds are to be prepared and performed in accordance with the requirements of this Part and the applicable requirements given in Chapter 2 of Part X.

1.3.2 Tensile tests

(a) Deposited metal tensile and longitudinal tensile test specimens are to be of type T1 as given in Table XI 2-1 of Part XI of the Rules for the construction and classification of steel ships, where, the diameter, d, is generally to be 10 mm. Except otherwise required, longitudinal axis of the specimens is to coincide with the center of the weld approximately in way of the half thickness of the plate.

(b) Transverse tensile test specimens for butt weld tests are to be machined to the dimensions shown in Fig. X 1-1. The upper and lower surfaces of the weld are to be filed, ground or machined flush with the surface of the plate. The thickness of test specimens is to be of full thickness of the test assemblies.

1.3.3 Bending tests

(a) Face and root bending test specimens
   (i) Test specimens, except for guided bend tests, are to be 30 mm in width and of the full plate thickness.
   (ii) Test specimens for guided bend tests are to be 38 mm in width and of the full plate thickness in case the thickness of the specimen is 9 mm or less. Where the thickness exceeds 9 mm, the specimen is to be machined on the compression side to reduce the thickness to 9 mm.

(b) Test specimens for side bending tests are to be 9 mm in width and of the full plate thickness.

(c) Bending test specimens are generally to be 250 mm in length. The weld deposite metal is to be located in way of mid-length of the specimen to subject to maximum tension and compression in test.

(d) Weld reinforcements and back straps are to be removed, filed, ground or machined flush with the surfaces of test specimens. Edges of test specimens are to be rounded to a radius of 1.5 mm.

(e) Guided bend test are to be carried out by jigs shown in Fig. X 1-2.

(f) The test is considered to be satisfactory if, after bending, the specimen is not to show any crack or other open defects exceeding 3 mm in any direction on the outer surface.

1.3.4 Fracture tests

(a) A fillet welded test assembly or specimen in form of T joint is to have the fillet weld on one side gouged or machined to facilitate breaking the another side fillet weld by closing the two abutting plates together, subjecting the root of weld to tension (see Fig. X 1-3).

(b) The fractured surface of the fillet weld is to be examined and there is to be no evidence of incomplete penetration, internal cracking or lack of root fusion and to be reasonably free from porosity. The incomplete fusion at the root corner of fillet may be acceptable provided the total length of the incompletely fused areas is not more than 10% of the total welded length.

(c) The fracture test specimen may be cut into short sections to facilitate breaking open.

1.3.5 Retests and additional tests

(a) Where the result of a tensile or bend test does not comply with the requirements, duplicate test specimens of the same type are to be prepared from either the same test assembly of the first test or the assembly newly welded with same welding condition as the first test assembly and are to be satisfactorily tested.

(b) Where the retest fails to meet the requirements in (a), further retests may be made on a newly welded assembly of different welding conditions, and these new test specimens are to include all tests required for the original assembly, even those which were previously satisfactory.

1.4 Other Inspections and Examinations

1.4.1 Surface inspections

(a) All welded joints of welding structures or test assemblies prior to the preparation of test specimens are to be subjected to a visual inspection after removing of slag and complete cooling, and if deemed necessary, examination by the methods of magnifying glass, liquid penetrant or magnetic particle for detection of surface defects may be required.

(b) The surface of the finished weld is to be sound, uniform and substantially free from crack, slag inclusion, porosity, undercut, overlap or other injurious defects. Care is to be taken to ensure adequate penetration and fusion. Each cutting section of the welded test assembly, is also to be examined to ensure that complete fusion has taken place.

(c) Fillet welds are to be of the size as required by the Rules, but not to be too excessive. Butt welds are to have uniform width and reasonable reinforcement, the reinforcement is not to be less than the minimum requirements nor too excessive.

1.4.2 Macro-etching examinations

(a) The transverse or longitudinal section of the welding joint is to be polished and etched for examination.

(b) The section of welding joint is to be shown that the weld is free from crack, poor penetration, lack of fusion or any other injurious defects.

1.4.3 Non-destructive examinations

(a) Non-destructive examinations for welding structures or welded test assemblies are to be carried out in the positions required in the Rules preferably by the methods of X-Ray or ultrasonic detection to ascertain the overall soundness of the weld.

(b) Acceptable recognized standards are to be used in evaluating the non-destructive detections of the welds.

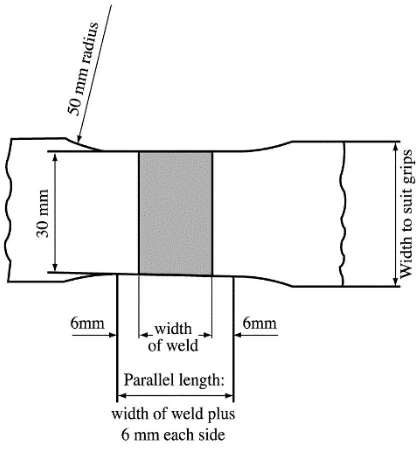


Fig. X 1-1
Dimensions of Transverse Tensile Test Specimens for Butt Weld Test

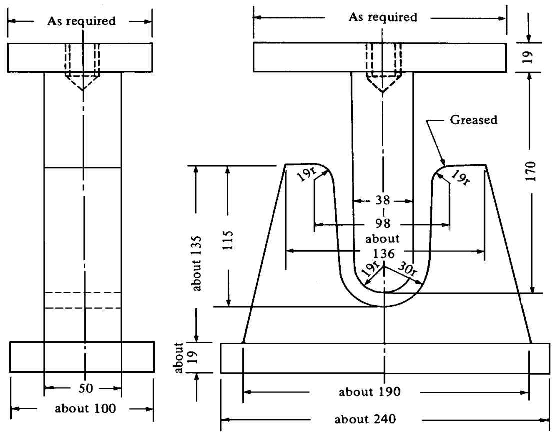


Fig. X 1-2
Guided Bend Test Jig (Dimensions in mm)

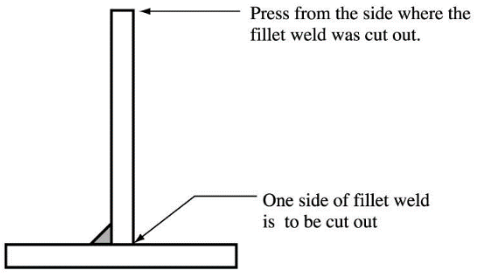


Fig. X 1-3
Fracture Test of Fillet Weld

Chapter 2
Welding Procedures

2.1 General

2.1.1 Where it is intended to use automatic or semi-automatic welding in shipyard or works, to build welded constructions for special services, or to use new materials in which there is no previous experience, or to use new welding processes, the procedure to be adopted is to be approved by the Society in advance.

2.1.2 Welding procedure specification subjected to approval is to contain the following information:

(a) applicable extent of the material grades, thickness or dimensions,

(b) edge preparation of joints,

(c) specification of welding materials,

(d) welding position, process and technique,

(e) welding sequence, speed, electric polarity, voltage and current,

(f) preheat, temperature control and post-weld heat treatment, and

(g) non-destructive examinations.

2.1.3 Procedures for the welding of all kinds of joints, with respect to types of welding materials, edge preparation, welding techniques and positions proposed to be adopted are to be established to the satisfaction of the Surveyor. The size of welding materials, current, voltage, rate of deposit and number of runs actually employed, are not to deviate from the established procedures as far as practicable.

2.2 Approval of Welding Procedures

2.2.1 The welding procedure intended to be accepted by the Society is to be subjected to a satisfactory demonstration welding test, unless otherwise specified.

2.2.2 Test assemblies are to be prepared with the same or equivalent material and welded by same procedure as indicated in the welding procedure specification. Generally, the thickness of test assemblies for tests is to be equal to the maximum thickness of the materials capably applied by the welding procedure for single run welding. The qualified thickness range for muti-run welding may be up to double thickness of the test piece. In case the welding procedure is applicable for welding various grades of materials, test assemblies prepared by representative grades of materials among them may be accepted.

2.3 Butt Welding Tests

2.3.1 Test assemblies

(a) The test assembly is to be made by welding together 2 plates of minimum 150 mm in width and of sufficient length for preparing all necessary test specimens required. As far as possible, plates are to have a size which can simulate the heat transfer during the production of welding. Plates are to be so cut that the final rolling direction is parallel to the weld, unless otherwise specified.

(b) For butt welding of pipes, the test assembly is to be made by welding together two full sections of pipes having a minimum length of 150 mm. In the case of large diameter pipes, the test assembly may be made as those for the plate given in 2.3.1(a) above, except the direction of weld joint is to be same as that of actual application.

2.3.2 Welding joints of the test assembly are to be subjected to a visual inspection and X-Ray examination or other non-destructive detection methods for whole length to ascertain if there are any defects in the weld, prior to the preparation of test specimens.

2.3.3 Test specimens

(a) The following test specimens are to be prepared from each test assembly as shown in Fig. X 2-1:
   (i) 2 transverse tensile,
   (ii) one face and one root guided bend when thickness is not more than 20 mm,
   (iii) one Macro-etching.

2.3.4 Test requirements

(a) The tensile strength of the transverse tensile test is not to be less than the specified requirements for the base material of the test assembly.

(b) Guided bend tests are to be in compliance with the requirements given in 1.3.3 of this Part.

(c) The transverse section of welding joints is to be subjected to a satisfactory macro-etching examination in compliance with the requirements given in 1.4.2 of this Part.

2.3.5 Where the welding procedures for butt welded joints are approved, the kinds of welding joints can include fillet weld joints corresponding to the welding position applied for the butt welded joints.

2.4 Fillet Welding Tests

2.4.1 Test assemblies and specimens

(a) A test assembly as shown in Fig. X 2-2 is to be prepared, the plate thickness, the size of the fillet weld and welding conditions are to be the same as those for actual applications except that only one side of the fillet is to be welded.

(b) Abutting members of the test assembly are to be straight and in intimate contact and securely tacked at ends before the fillet weld is made.

2.4.2 Test requirements

(a) Surface inspections

The test assembly, upon completion of welding, is to be subjected to a satisfactory surface inspection in compliance with the requirements given in 1.4.1 of this Part.

(b) Macro-etching examinations

Two transverse sections cut within 50 mm from both ends of the test assembly are to be subjected to a satisfactory macro-etching examination in compliance with the requirements given in 1.4.2 of this Part.

(c) Fracture tests

The remaining section of the test assembly is to be subjected to a satisfactory fracture test in compliance with the requirements given in 1.3.4 of this Part.

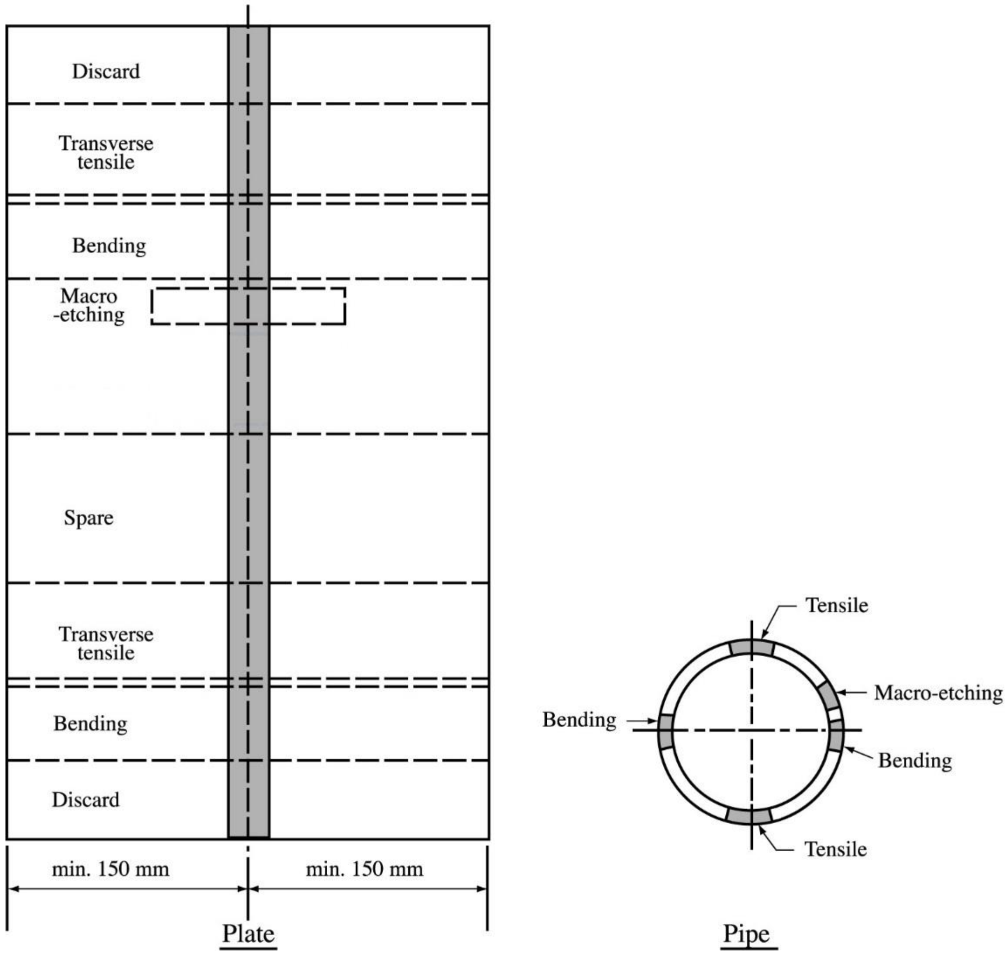


Fig. X 2-1
Butt Weld Test Assembly for Welding Procedure Approval

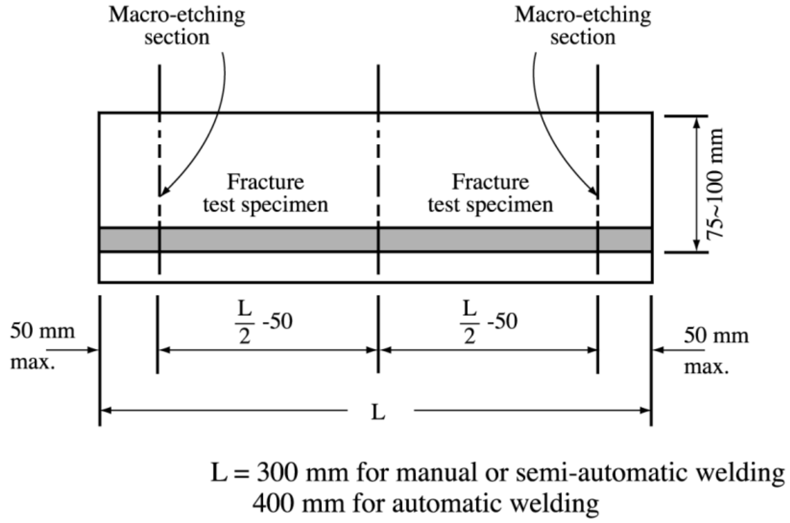


Fig. X 2-2
Fillet Weld Test Assembly for Welding Procedure Approval

Chapter 3

Welder Qualifications and Control of Welding Operations

3.1 Control of Welding Operations

3.1.1 The welding is to be carried out in shipyard or other works recognized by the Society. Such welding shops are to have suitable workshop, welding facilities, adequate protection against weather and other necessary equipment and to prove their qualification for the welding operation in a satisfactory manner.

3.1.2 The Surveyor is to satisfy himself that all welders and welding operators are properly qualified and are experienced in the work proposed and in the proper use of the welding processes and procedures to be followed. The Surveyor is to satisfy himself as to the employment of a sufficient number of skilled supervisors to ensure a thorough supervision and control of all welding operations.

3.1.3 The shipyards or other works are to have a proper management and control system for the employed welders. Sufficient information containing the training history, experience, dates and results of qualification tests, renewal tests, requalification tests, etc. of the welders is to be prepared for the Surveyor’s examination at any time.

3.2 Welder Qualifications

3.2.1 Each welder who intends to engage in the welding work specified in this Part is to pass the qualification tests required according to the applicable welding procedures and kinds of material to be welded and to be furnished with the Qualification Certificate issued by the Society.

3.2.2 The requirements of welder qualification given in this chapter are applicable to welders who intend to weld the weldable materials as specified in Chapters 10 (aluminum alloy) of Part XI of “Rules for the construction and classification of steel ships” or their equivalents in manual or semi-automatic welding practice.

3.2.3 The welder who intends to engage in the automatic welding operations is to be of well being experienced for the specific welding work concerned.

3.2.4 For welders who have been approved by other Classifications Societies or appropriate Organizations with the qualification equivalent to the requirements of this chapter may be accepted by the Society without further testing for the relevant qualification.

3.2.5 A Welder’s Qualification Certificate (W.Q. Cert.) valid for three years is to be issued to the welder who has passed the required qualification tests or certificate renewal tests by the Society. Where the welder is qualified for welding aluminum alloy, a special notation “For Aluminum Alloy” is to be noted in the W.Q. Cert.

3.2.6 Welder qualifications are approved in various welding positions and material thicknesses as given in Table X 3-1.

3.3 Welder’s Qualification Tests

3.3.1 The application for Welder’s Qualification tests is generally to be made to the Society by the shipyard or works where welders are employed. The applicant is to prepare the necessary welding equipment and test or examination appliances for the qualification tests in a satisfactory manner. The preparations of test assemblies and test specimens and the mechanical tests or other examinations are to be carried out in the presence of the Surveyor.

3.3.2 The Welder's qualification is defined by "Grades" (considering the allowable thickness and pipe welding – a, A, C, D) and "Classes" (considering the welding position – I, II and III) as shown in Table X 3-1.

3.3.3 Any welder who intends to be qualified for class higher than the lowest class of the corresponding grade is to have the qualification of the lowest class of the corresponding grade previously provided or to pass the test simultaneously.

3.3.4 Test assemblies for plate welding

(a) Test assemblies are to be prepared in accordance with the requirements given in Table X 3-1 and shown in Fig. X 3-1. Plates for preparation of the test assembly are to be of materials as specified in Part XI of "Rules for the construction and classification of steel ships" or their equivalents.

(b) Welding procedure.

(i) Plates are to be fixed corresponding to the welding position as shown in Fig. X 3-3 without change of the plate direction throughout the welding operation.

(ii) The backing strap is to be contiguous with plates. The welding is to be done from one side only.

(iii) For the vertical position test assembly, the welding is to be done in upward procedure.

(iv) Plates are to be so welded that they will be approximately plane after welding; however, the warping angle is not to be more than 5°.

3.3.5 Test assemblies for pipe welding

(a) Test assemblies are to be prepared in accordance with the requirements given in Table X 3-1 and shown in Fig. X 3-2. Pipes for preparation of the test assembly are to be of the materials as specified in Part XI of "Rules for the construction and classification of steel ships" or their equivalents.

(b) Welding procedure

(i) For rotating pipe welding, the pipe is to be rotated in horizontal axis so that the welding will be done in downhand position throughout the welding of whole circumference.

(ii) For fixed pipe welding, pipes are to be fixed horizontally and vertically so that the welding will be done in downhand, vertical, overhead and horizontal positions.

(iii) The welding is to be done from outside of the pipe only. For type C test assemblies back chipping or gouging and back welding are not to be carried out. For type D test assemblies, the backing ring is to be continuous with the inner surface of pipes.

3.3.6 Welding materials

(a) Welding materials used for preparing the test assembly are to be of the electrodes or wires as specified in Chapter 4 of this Part corresponding to the aluminum alloy grade of the test assembly base material and the welding position suitably.

(b) Each pass of the weld is to be made by the size of welding material which is the same as that used in the actual working practice.

3.3.7 Test requirements

(a) Test assemblies are not to be subjected to preheating, peening and post-weld heat treatment.

(b) Welding joints of the test assembly are to be subjected to a visual inspection prior to the preparation of test specimens. The surface of welding joints is to have uniform width and height of reinforcement, to be free from crack, undercut, significant sharp, icicle, overlap or other injurious defects.

(c) Guided bend test specimens as required in Table X 3-1 are to be taken from each test assembly except type C, each specimen is to be prepared and to be subjected to guided bend test in complying with the requirements given in 1.3.3 of this Part.

(d) Guided bend tests required in 3.3.7(c) above may be substituted by X-Ray examinations or other acceptable non-destructive examination methods subject to the approval of the Society.

(e) For Qualification tests of C-I and C-II, type C test assemblies are to be sealed on both ends by welded plates or other suitable means and to be subjected to a satisfactory hydraulic test by a pressure specified in 5.5.1 and 5.5.2 of Part XI of "Rules for the construction and classification of steel ships."

3.3.8 Retests and additional tests

(a) Where any test specimen fails to fulfill the requirements, the retesting in compliance with the requirements, given in 1.3.5 of this Part may be made.

(b) If the test fails, the welder is to be retrained for at least three months before undergoing a retest.

3.4 W.Q. Certificate Renewal Tests

3.4.1 Qualified welders who intend to maintain their qualifications are to pass the W.Q. Certificate Renewal Tests given in Table X 3-2 before the expiration date of the W.Q. Certificate.

3.4.2 Test assemblies

(a) One test assembly is to be prepared for each certified qualification in compliance with the requirements given in Table X 3-2.

(b) For plate welding qualification, test assemblies are to be made by the double-welded butt joint welding practice given in 5.3.1(a) and (b) of "Rules for the construction and classification of steel ships."

(c) For pipe welding qualification, test assemblies are to be made by socket joint fillet welding.

3.4.3 The test is considered satisfactory provided that the examinations for test assemblies given in Table X 3-2 have been considered satisfactory by the Surveyor.

3.4.4 Upon request of shipyards or works, examinations may be made on the actual work productions welded by the welder in lieu of test assemblies, provided the productions for examination are welded in similar condition to the required test assemblies and accepted by the Surveyor.

3.4.5 Where the test fails to fulfill the Surveyor's satisfaction, the retesting in compliance with the requirements given in 3.3.8 of this Part may be applied.

3.5 Requalifications

3.5.1 In case the validity of W.Q. Certificate expires, the welder may be re-qualified by fulfilling the following requirements.

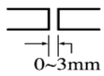
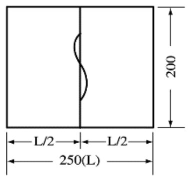
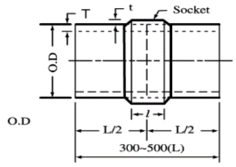
(a) For the validity which has expired not more than 12 months, the requirements of W.Q. Certificate renewal test are to be fulfilled.

(b) For the validity which has expired more than 12 months, the full requirements of Welder’s qualification test are to be fulfilled.

Table X 3-1
Welder Qualifications

Welder's Qualification			Applicable Welding Works		Requirements of Welder's Qualification Tests			
					Test Assemblies			No. of Test Specimens to be taken from each Test Assembly for Tests (Guided Bend Test)
Kind	Grades	Classes	Allowable Thickness (mm)	Welding Positions	Type	No. and Welding Position	Thickness (mm)	
Plate Welding	a-	I	≤ 9	All Positions	A	1 – Vertical and 1 – Overhead	3.2 ~ 5	1 – Face bending and 1–Root bending
		II		Downhand, Horizontal and Vertical		1 – Vertical		
		III		Downhand		1 – Downhand		
	A-	I	≤ 19	All Positions		1 – Vertical and 1 – Overhead	9	
		II		Downhand, Horizontal and Vertical		1 – Vertical		
		III		Downhand		1 – Downhand		
Pipe Welding	C-	I	≤ 5	All Positions (Fixed Pipe)	C	1 – Horizontally fixed and 1 – Vertically fixed	4 ~ 5 (nominal dia. 80 ~ 150)	(The test assemblies are to be subjected to Hydraulic test by a pressure specified in 5.5.1 & 5.5.2 of Part XI of "Rules for the construction and classification of steel ships")
		II		Downhand (Rotated Pipe)		1 – Horizontally rotated		
	D-	I	≤ 19	All Positions (Fixed Pipe)	D	1 – Horizontally fixed and 1 – Vertically fixed	9 ~ 11 (nominal dia. 125 ~ 300)	2 – Face bending and 2 – Root bending
		II		Downhand (Rotated Pipe)		1 – Horizontally rotated		

Table X 3-2
Welder's Qualification Certificate Renewal Test Requirements

<table>
<tr><th colspan="2" rowspan="3">Welder's Qualification</th><th colspan="5">Test Assemblies</th><th rowspan="3">Examination Items</th></tr>
<tr><th colspan="2">Welding Positions</th><th rowspan="2">Plate thickness or Wall Thickness (mm)</th><th rowspan="2">Edge Preparation of Butt Joint</th><th rowspan="2">Dimensions (mm)</th></tr>
<tr><th>1st Side</th><th>2nd Side</th></tr>
<tr><td rowspan="6">Plate Welding</td><td>a-I</td><td>Overhead</td><td>Any</td><td rowspan="3">3.2 ~ 5</td><td rowspan="3"></td><td rowspan="6"></td><td rowspan="6">Visual and X-Ray</td></tr>
<tr><td>a-II</td><td>Vertical</td><td>Any</td></tr>
<tr><td>a-III</td><td>Downhand</td><td>Downhand</td></tr>
<tr><td>A-I</td><td>Overhead</td><td>Any</td><td rowspan="3">≥ 9</td><td rowspan="3">V, U, X or H shapes</td></tr>
<tr><td>A-II</td><td>Vertical</td><td>Any</td></tr>
<tr><td>A-III</td><td>Downhand</td><td>Downhand</td></tr>
<tr><td rowspan="6">Pipe Welding</td><td>C-I</td><td colspan="2">Horizontally fixed</td><td rowspan="2">≥ 4 (nominal dia. ≥ 80)</td><td colspan="2" rowspan="6">1.25T ≥ t ≥T
l =100 or O.D. whichever is lesser</td><td rowspan="6">Visual and Hydraulic test W.T.P. 20 MPa</td></tr>
<tr><td>C-II</td><td colspan="2">Horizontally rotated</td></tr>
<tr><td>D-I</td><td colspan="2">Horizontally fixed</td><td rowspan="2">≥ 9 (nominal dia. ≥ 125)</td></tr>
<tr><td>D-II</td><td colspan="2">Horizontally rotated</td></tr>
<tr><td></td><td colspan="2"></td><td rowspan="2"></td></tr>
<tr><td></td><td colspan="2"></td></tr>
</table>

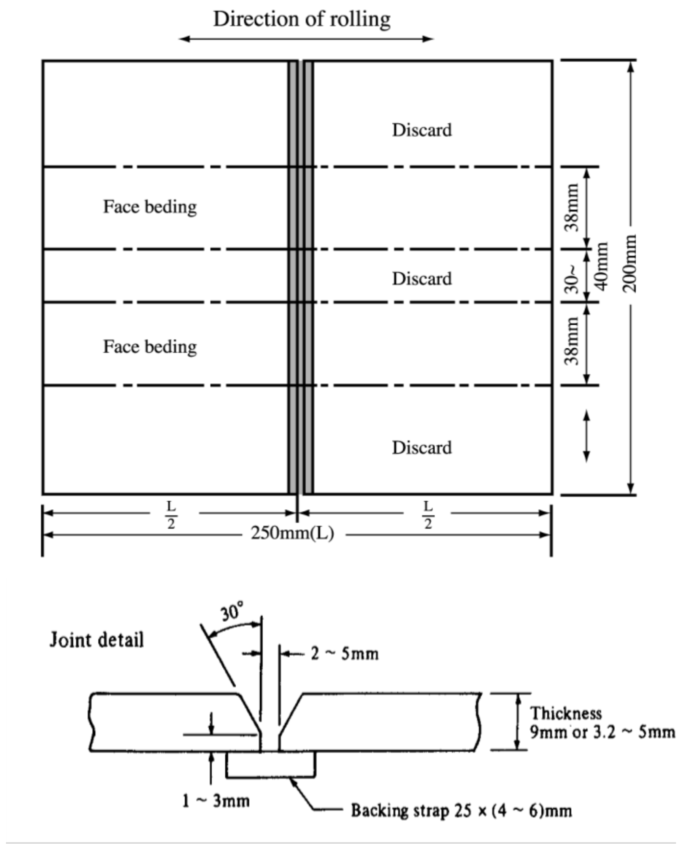


Fig. X 3-1
Type A Test Assembly for Welder's Qualification Test (Plate)

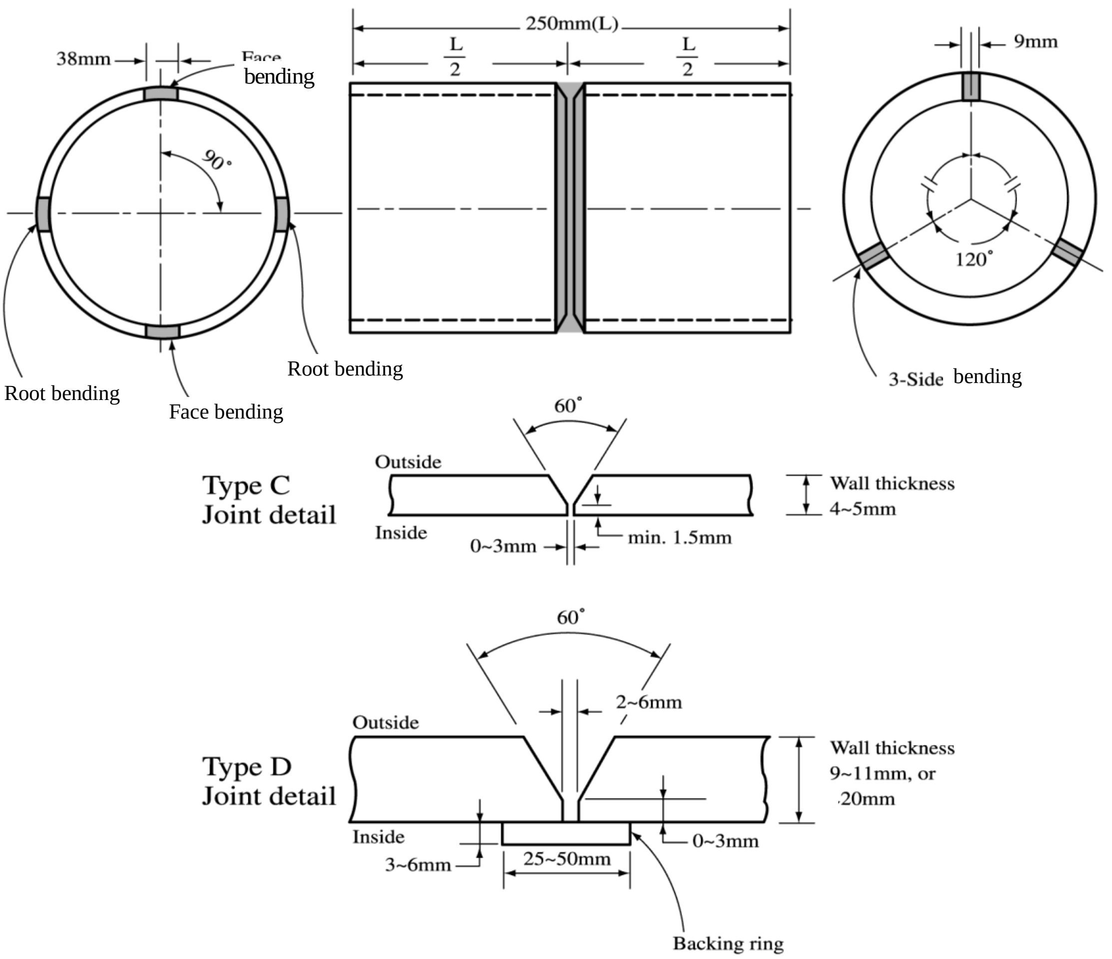


Fig. X 3-2
Type C and Type D Test Assemblies for Welder's Qualification Test (Pipe)

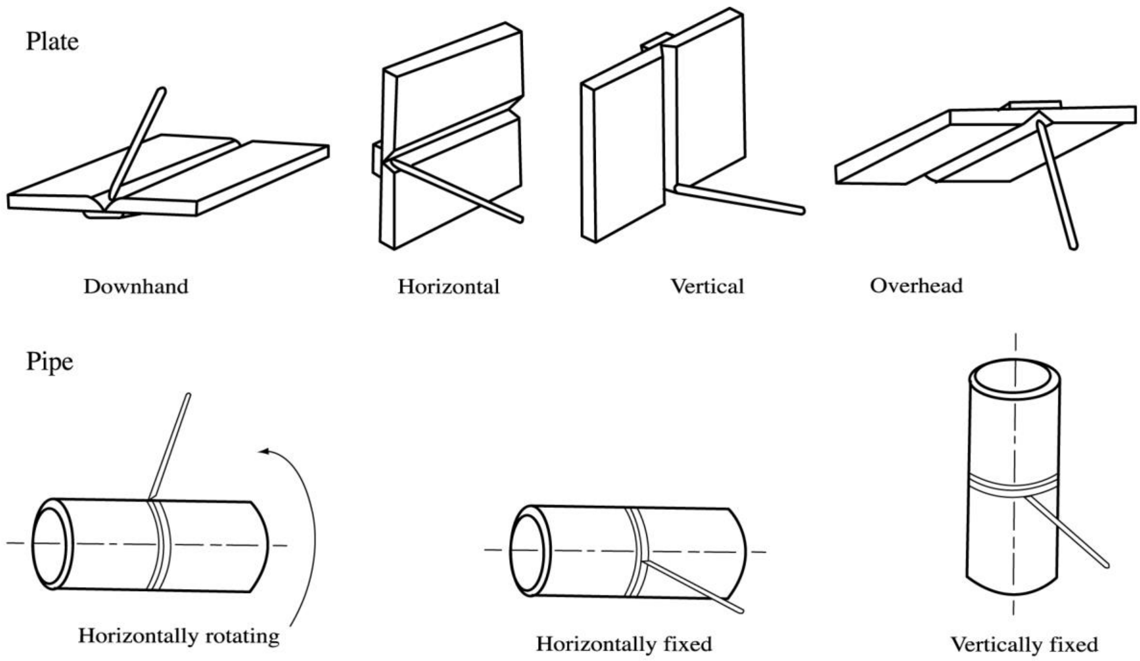


Fig. X 3-3
Welding Positions of Test Assembly

Chapter 4

Welding Materials and Constructions

4.1 General

Aluminum alloy welding materials and constructions are to be in according with the relevant requirements in Chapter 4 and 5 of Part XII of Rules for the construction and classification of steel ships.

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