PARTE II — PART II RULES FOR THE CONSTRUCTION AND CLASSIFICATION OF VESSELS IDENTIFIED BY THEIR MISSIONS

TÍTULO 11 — TITLE 11 — SHIPS IN GENERAL

SEÇÃO 5 — SECTION 5 — ENGINES AND MECHANICS

CHAPTERS

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CHAPTER A
APPROACH

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A1. SCOPE OF APPLICATION

100. Compliance with the Rules

101. The present Rules apply to propulsion engine installations including auxiliaries and equipment of vessels for Sea Going Navigation.

102. The materials for the manufacture of equipments in general, internal combustion engines, pressure vessels, boilers, pipes and fittings are to meet the requirements stated in these Rules.

103. Designs that are beyond the scope of the Rules and materials with characteristics other than those listed here may be approved, provided that their equivalence and adequacy are recognized by RBNA. For this end, the submission of additional documentation is required as well as additional special tests and trials.

104. At the discretion of RBNA, requirements in addition to those listed here may be requested to all kinds of machinery, wherever deemed indispensable, based on further research or operational experiences.

104. Based upon new researches RBNA may request additional requirements to those in this Section for any machinery where such requirements prove indispensable to the operation.

200. Standards

201. The propulsion systems and all equipment and accessories empoyed on vessels covered by the present Rules are to be designed, constructed and tested according to the latest revisions of the relevant INMETRO standards and, in the absence of those, standards from the following organizations are to be used:

a. ASTM - American RBNA for Testing and Materials;

b. ANSI - American Standard RBNA Institute;

c. ASME - American RBNA of Mechanical Engineers.

CHAPTER B
TECHNICAL DOCUMENTATION

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B1. SCOPE

100. Submission

101. The drawings are to contain all details necessary for full understanding of the design.

102. Wherever necessary, calculation sheets of the component parts, as well as description of the machinery are also to be submitted.

103. Any modification of the design or use of any component part of equipment or arrangement different from those already approved is subject to new submission of documents to RBNA before its manufacture.

104. Drawings and documents to be submitted to RBNA are to have all the dimensions and information required by the international system. Where different widely accepted dimensions from another unit system are employed, the corresponding values in the international system are to be indicated.

B2. DOCUMENTS FOR APPROVAL

100. List of documents to be submitted

101. The following documents are to be submitted for approval of RBNA in at least 3 hard copies or in vitual files:

a. General arrangement of the engine room and every machinery space;

b. General arrangement of the shaft line, with characteristics of the propeller shafts, intermediate shafts and thrust shafts, details of variable pitch

c. system, when appropriate, couplings and clutches, shaft bearings, thrust bearings, telescopic tubes and propellers (including propeller parts, if any) detailing the materials employed, the maximum continuous power of the installation, the number of revolutions per minute and the critical velocity are to be indicated;

102. For internal combustion engines, see list in the Chapter E.

200. Other documents to be provided

201. A copy of the manuals for operation and maintenance of the main engines and of the most important auxiliaries as compressors, pumps and others are also to be presented to RBNA for information.

CHAPTER C
MATERIALS

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C1. MATERIALS FOR MACHINERY

100. Cast iron and forged alloy steel

101. The limit of the minimum tensile strength is 412 N/mm2 (42 kgf/mm2), corresponding to the class 1 of the standard P-EB 392.

102. For the propeller shaft lines using forged alloy steel the tensile strength is not to exceed 800 N/mm^2, with an acceptable range between 800-950 N/mm^2.

103. For applications in other items of machinery which are not the propeller shafts, the tensile strength is not to exceed 1100 N/mm^2, with an acceptable range between 1100 to 1300 N/mm^2. See Part III, Title 62, Section 5, Chapter B.

200. Gray cast iron

201. The limit of the minimum tensile strength is 150 N/mm^2 (15.3 kgf/mm^2), corresponding to the class FC 150 of the NBR 6589. See Part III, Title 62, Section 5, Chapter D.

202. When the use of high-strength cast iron is proposed, its specification is to be submitted for approval along with the project to which the material is intended.

300. Nodular cast iron

301. The limit of the minimum tensile strength is 420 N/mm^2 (42.8 kgf/mm^2), corresponding to the class FE 42012 of standard EB 585- Part 1/79. See Part III, Title 62, Section 5, Chapter E.

400. Forged steels

401. The limit of the minimum tensile strength is 412 N/mm^2 (42 kgf/mm^2), corresponding to the class 1 of the standard EB 391 See Part III, Title 62, Section 5, Chapter C.

500. Roumnd bars of hot-rolled steel

501. Round bars of hot-rolled steel having a diameter up to 230 mm may be used in place of carbon steel forged parts which are to be fully killed. See Part III, Title 62, Section 5, Chapter F.

502. The tensile properties are to meet the requirements of paragraph C1.100. When the round bar diameter exceeds 230 mm, its application is subject to verification.

C2. MATERIALS FOR SHAFT LINES

100. Forged or cast steel

101. The limit of the minimum tensile strength is 402 N/mm^2 (41 kgf/mm^2), but it is not to exceed the maximum values gven in Part III, Título 62, Subchapter C7 Table T.C7.101.1.

200. Cast bronze

201. Cast bronze may be used for shaft diameter up to 80 mm. The limit minimum tensile strength is 206 N/mm^2 (21 kgf/mm^2).

300. Round and square rolled bars

301. For shafts with diameter up to 300 mm forged steel bars may be allowed to replace hot rolled bars, provided that they present the same strength characteristics required for the forged steel (see by Part III, Title 62, Subchapter C7 TabelaT.C7.101.1).

302. The steel is to be killed and be subjected to the following heat treatments pursuant to Part III, Title 62, Subchapter C5, Item 102.

303. The cross-sectional area of the non-machined bar end is not to exceed one-sixth of the cross-sectional area of the ingot.

304. Additionally, rolled bars used in place of forged in propeller shafts are to be subjected to nondestructive surface testing such as magnetic particle, dye penetrant or other method. The irregularities are to be removed at the discretion of the surveyor.

305. Shafts with a diameter exceeding 450 mm are to be subjected to inspection by ultrasound. RBNA, however, may require, at its discretion, ultrasound inspection for shafts with a diameter under 450 mm.

C3. MATERIALS FOR PROPELLERS

100. Manganese bronze and other alloys

101. The chemical composition of the alloys is to be in accordance with Part III, Title 62, Chapter G of the Rules.

102. The minimum traction strength limit is to be: according to the Table T.C3.102.1 below, extracted from Part III, Title 62, Chapter G of the Rules according to the type of alloy (CU1, CU2,CU3,CU4), as defined in the above mentioned Part III of the Rules.

TABLE T.C3.102.1 - MECHANICAL CHARACTERISTICS OF CAST COPPER ALLOYS FOR PROPELLERS (SEPARATELY CAST TEST COUPONS)

Alloy type	Proof stress Rp 0,2 [N/mm^2] min.	Tensile strength R_m [N/mm^2] min.	Elongation A_5 [%] min.
CU1	175	440	20
CU2	175	440	20
CU3	245	590	16
CU4	275	630	18

200. Manganese bronze and other bronze alloys

201. The chemical composition of these alloys is to be subjected to the RBNA approval.

202. The minimum tensile strength limit e is:

a. 451 N/mm2 (46 kgf/mm2), when the specimen comes from separately cast test samples;

b. 412 N/mm^2 (42 kgf/mm^2), when the test specimen comes from an appendix cast from the same part.

300. Cast iron

301. Characteristics complying with C1.200 above.

400. Cast steel

401. Characteristics complying with C1.200 above.

CHAPTER D
PRINCIPLES OF INSTALLATION

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D1. SPECIFIC CONDITIONS

100. Ambient conditions [IACS UR M40]

101. All equipment and systems on board must be designed and cosntructed to withstand the ambient conditions found on board such as temperature, ship’s motions, vibrations, corrosive environment.

102. The ambient condition requirements of the present Chapter D1 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances as to ensure proper operation.

200. Ambient conditions: temperatures [IACS M40.2]

201. The ambient conditions specified under Table T.D1.201.1 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances as to ensure proper operation.

300. Ambient conditions – Inclinations [IACS UR M46]

301. All equipment and systems on board must be designed and cosntructed to resist to the ambient conditions found on board such as temperature, ship’s motions, vibrations, corrosive environment:

302. The ambient conditions specified under the present chapter in Table T.D1.302.1 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances to ensure proper operation.

TABLE T.D1.201.1 – AIR TEMPERATURES

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

TABLE T.D1.201.1 – WATER TEMPERATURES

Coolant	Temperature (°C)
Seawater	+32[2]
Charge air coolant inlet to charge air cooler	See Item D1.400 below

NOTES
1. Electronic appliances are to be suitable for proper operation even with an air temperature of +55°C.
2. The RBNA may approve other temperatures in the case of ships not intended for unrestricted service.

TABLE T.D1.302.1 – ANGLE OF INCLINATION

Installations, components	Angle of inclination [°][2]			
	Athwartships		Fore-and-aft	
	static	dynamic	static	dynamic
Main and auxiliary machinery	15	22,5	5[4]	7,5
Safety equipment, e.g. emergency power installations, emergency fire pump and their devices Switch gear, electrical and electronic appliances[1] and remote control systems	22,5[3]	22,5[3]	10	10

NOTES:
1. Up to an angle of inclination of 45° no undesired switching operations or operational changes may occur.
2. Athwartships and fore-end-aft inclinations may occur simultaneously.
3. In ships for the carriage of liquefied gases and of chemicals the emergency power supply must also remain operable with the ship flooded to a final athwartships inclination up to maximum of 30°.
4. Where the length of the ship exceeds 100m, the fore-and-aft static angle of inclination may be taken as 500/L degrees where L = length of the ship, in metres, as defined in Part II, Title 11, Section 1, Chapter A, Item A2.150.

400. Design reference conditions [IACS UR M28]

401. For the design of the machinery installations and for the purpose of determining the power of main and auxiliary reciprocating internal combustion engines the following ambient reference conditions apply for ships of unrestricted service

a. Total barometric pressure 1000 mbar

b. Air temperature +45°C

c. Relative humidity 60%

d. Sea water temperature 32°C (charge air coolant-inlet)

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

403. The temperature of any surface in the Engine Room, regardless of isolation, is not to exceed 200 ºC under any circumstances. The measurement of the temperature of the Engine Room surfaces may be determined by one of the following processes:

a. Control thermometer on the surface

b. Laser heat trackers

c. Infrared Thermoscan with video

404. The lowest water temperature should be considered as 5^{o} C, except at the Amazon River, where it may be considered to be 10 °C.

405 The ambient Engine Room temperature is not to exceed 45^{o} C.

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

500. Fuels

501. These Rules apply to liquid fuels for operation of machinery and boilers, with flash point above 60 º C. Where fuels of a lower flash point are employed, they will be subjected to special analysis by RBNA. See Part II, Title 11, Section 6 of the Rules.

D2. MACHINERY ARRANGEMENT

100. General arrangement

101. The machinery spaces are to be sufficiently large to allow the easy operation and maintenance of machinery, particularly of the propulsion machinery and shaft line.

102. The instruments and controls of the equipment shall be installed where they may be easily read and arranged so as to makefor an easier operation in the engine room.

103. As far as possible the monitoring equipment and operation shall be centralized and the control of all important parts of the system is to be easily accessible.

104. For machinery and equipment it is to be granted at least:

a. protection against moisture and dust accumulation;

b. easy access, and

c. sufficient ventilation.

105. Equipment should never be installed in escape routes or blocking access to any safety or fire fighting equipment.

200. Ventilation

201. The machinery and cargo pump spaces are to have sufficient ventilation even when the accesses are closed, and any accumulation of toxic, flammable or asphyxiating gases should be avoided as far as practicable. See Part II, Title 11, Section 6, Subchapter F6 of the Rules.

202. Ventilators are to be cut-off from an easily accessible position outside the space ventilated by them.

300. Accesses

301. The access to machinery spaces is to be located so as to allow fast and unobstructed exit of personnel in case of accidents and removal of equipment for repairs.

302. One main entrance and at least one emergency access to the engine room spaces are to be provided.

303. All accesses to the engine room and pump room are to be clearly indicated and labeled so that following these signs anyone can quickly find the exit to the compartment.

400. Lighting

401. All workplaces are to be provided with sufficient lighting as NB-0151 (Calculation of Lighting Levels in Ships).

500. Engine room bottom bilge

501. All bilge sumps are to be accessible and easy to clean. Wastewater is not to flood and / or reach electrical equipment whatever the movements and inclinations of the vessel during operation.

502. The regulations regarding the drainage of oily waters are to be complied with.

600. Thermal insulation

601. Piping containing vapor or hot liquid, exhaust piping from the air compressors and equipments whose operating surface reaches temperature greater than 60 ° C are to be effectively isolated.

602. Exhaust gas piping is to be insulated and located so that no flammable material can trigger its ignition during the installation.

603. Insulating materials are to be non-flammable. Where oil spills or moisture could reach the insulation, it must be adequately protected by metal plates, which may be removed for maintenance or replacement of the insulation.

700. Protective equipments and precautions.

701. The machinery is to be arranged and secured so as to restrict to a minimum the possibility of accidents and fires. In addition to National regulations, the following is to be observed:

a. moving parts, wheels, gears, transmissions belts and pulleys, shaft flanges and other parts that may be sources of accidents to operators are to be fitted with protectyion so that no moving part is exposed ;

b. the discharges of relief valves and safety and drainage devices are to be lead out to safe places,

c. machinery and equipment must be fixed in their bases, as well as parts of large dimensions;

d. machinery foundations are to be of robust construction and properly secured to the hull, so that there is no displacement due to the movements of the vessel; they are to be designed and arranged to withstand the various stresses to which they are subjected and to distribute them to the hull, without transmitting loads to the machinery they support. For guidance, see Part II, Title 11, Section 2, Chapter I, Item I1.201;

e. special attention is to be given to the arrangement of the foundation of the thrust bearing and its securing to the hull;

f. effective means are to be provided to prevent loosening of bolts and nuts of the parts due to vibration;

g. the flooring in places of operation and the engine room ladders are to be of the nonslip type;

h. vertical or inclined ladders and service platforms, side of engines and equipment and passageways are to be protected by railings / handrails / guardrails;

i. normal and emergency passages, service locations and devices are to be illuminated by the main lighting and emergency lighting systems;

j. relief valves, fire fighting system valves, fire fighting piping, emergency bilge pipes and valves, are to be painted so that they can readily be identified by the operator.

D3. TRANSMISSION OF ORDERS

100. Internal communication

101. The engine room and wheelhouse are to be connected by at least one system for transmission of service orders in both directions, and are to have more than one mode of communication with the control stations, one of which shall not use ship's electric energy. See Part II, Title 11, Section 8, subsection E4. Rules.

D4. IDENTIFICATION OF MACHINERY

100. Plates and colours

101. The equipment, pipe fittings and pipe lines shall be easily identified bya colour system, in order to indicate and warn about the actual risks, according to National (ABNT, for Brazilian flag vessels) or International standards.

102. Identification plates are to be posted in each equipment or accessory and piping and fittings.

D5. DIESEL-ELECTRIC PROPULSION INSTALLATION

100. Diesel engine installation

101. The installation of the diesel engines shall follow the above requirements for internal combustion engines.

200. Generator installation

201. The installation of the generators shall follow the requirements of Part II, Title 11, Section 7 of these Rules.

202. A main source of power provided with sufficient capacity is to be installed on board,

203. The main source of electrical power is to consist of at least two generator sets.

CHAPTER E
INTERNAL COMBUSTION ENGINES

CHAPTER CONTENTS

E1. APPROACH

100. Application

101. The Rules in this Section are to apply to internal combustion engines used as main propulsion engines and more important auxiliary machinery such as power generation (e.g. power generating sets).

102. Series production engines, of maximum continuous power up to 140 kW (190 CV) or cylinder diameter up to 300 mm, and of proven performance, may be approved by RBNA by submission or the test bench results for the unit. See Subchapters E8 and E9.

200. Definition of diesel engine type [IACS M32]

201. Engines are of the same type if they do not vary in any detail included in the definition in E1.202. When two engines are to be considered of the same type it is assumed that they do not substantially differ in design and their design details, crankshaft, etc., and the materials used meet Rules requirements and are approved by the RBNA.

202. **Definition:** The type of internal combustion engine expressed by the Engine Builder's designation is defined by:

a. the bore,

b. the stroke,

c. the method of injection (direct or indirect injection),

d. the kind of fuel (liquid, dual-fuel, gaseous),

e. the working cycle (4-stroke, 2-stroke), the gas exchange (naturally aspirated or supercharged), the maximum continuous power per cylinder at maximum continuous speed and/or maximum continuous brake mean effective pressure [(a)] the method of pressure charging (pulsating system, constant pressure system), the charging air cooling system (with or without intercooler, number of stages), cylinder arrangement (in-line, vee). [(b)]

203. Notes :

a. After a large number of engines has been proved successfully by service experience, an increase in power up to maximum 10% may be permitted, without any further type test, provided approval for such power is given.

b. One type test suffices for the whole range of engines having different numbers of cylinders.

E2. PERFORMANCE

100. Power and Capacity of engines and generators.

101. Where the engine is operating at its maximum continuous power it is to support a 10% overload for 30 minutes every 6 hours of operation.

102. The capacity of these generating sets is to be enough so that, when any one of these generator sets is out of action, it is possible to supply the services needed to establish normal operating conditions of propulsion and safety.

103. Minimum comfort conditions are to be ensured for the living quarters, including at least adequate services for cooking, heating, domestic refrigeration, mechanical ventilation, sanitary services and supply of freshwater.

104**.** Moreover, the generator sets should be such that even when one generating set or its primary source of energy stops, the other generator sets are be able to provide the electrical services necessary for starting the main propulsion plant from the dead ship condition.

200. Combustion

201. All new ships are to be equipped with two fuel oil service tanks for each type of fuel used on board necessary for propulsion and essential systems or equivalent arrangements.

202. Such tanks should have a capacity of at least 8 hours of operation at maximum power of the propulsion system maintained, with the electric power generating installation work with the normal load operating at sea.

E3. DOCUMENTS TO APPROVED

100. Documents for the approval of diesel engines [IACS UR M44]

101. For each type of engine that is required to be approved the documents listed in the following table and as far as applicable to the type of engine are to be submitted to the RBNA for approval (A), approval of materials and weld procedure specifications (A*), or for information (R) by each engine manufacturer (see Note 4).

102. After the approval of an engine type has been given by the RBNA 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 RBNA. 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 by the individual RBNA.

TABLE T.E3.102.1 – DOCUMENTS

No.	A/R	Item
1.	R	Engine particulars as per attached sheet
2.	R	Engine transverse cross-section
3.	R	Engine longitudinal section
4.	R/A*	Bedplate and crankcase, cast or welded with welding details and instructions[9]
5.	R	Thrust bearing assembly[3]
6.	R/A*	Thrust bearing bedplate, cast or welded with welding details and instructions[9]
7.	R/A*	Frame/framebox, cast or welding details and instructions[1,9]
8.	R	Tie rod
9.	R	Cylinder head, assembly
10.	R	Cylinder liner
11.	A	Crankshaft, details, each cylinder No.
12.	A	Crankshaft, assembly, each cylinder No.
13.	A	thrust shaft or intermediate shaft (if integral with engine)
14.	A	Shaft coupling bolts
15.	R	Counterweights (if not integral with crankshaft), incluiding fastening
16.	R	Connecting rod
17.	R	Connecting rod, assembly
18.	R	Crosshead, assembly[2]
19.	R	Piston rod, assembly[2]
20.	R	Piston, assembly
21.	R	Camshaft drive, assembly
22.	A	Material specifications of main parts with information on non-destructive material tests and pressure tests[8]
23.	R	Arrangement of foundation (for main engines only)
24.	A	Schematic layout or other equivalent documents of starting air system on the engine[6]
25.	A	Schematic layout or other equivalent documents of fuel oil system on the engine[6]
26.	A	Schematic layout or other equivalent documents of lubrificating oil system on the engine[6]
27.	A	Schematic layout or other equivalent documents of cooling water system on the engine[6]
28.	A	Schematic diagram of engine control and safety system on the engine[6]
29.	R	Schieling and insulation of exhaust pipes, assembly
30.	A	Schieling of high pressure fuel pipes, assembly[4]
31.	A	Arrangement of crankcase explosion relief valve[5]
32.	R	Operational and service manuals[7]
33.	A	Schematic layout or other equivalent documents of hydraulic system(for valve lift) on the engine
34.	A	Type test program and type test report
35.	A	High pressure parts for fuel oil injection system[10]

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. all engines.
5. only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m3 or more.
6. and the system so far as supplied by the engine manufacturer. Where engines incorporate electronic control systems a failure mode and effects analysis (FMEA) is to be submitted to demonstrate that failure of an electronic control system will not result in the loss of essential services for the operation of the engine and that operation of the engine will not be lost or degraded beyond an acceptable performance criteria of the engine.
7. operation and service manuals are to contain maintenance requirements (servicing and repair) including details of any special tools and gauges that are to be used with their fitting/settings together with any test requirements on completion of maintenance.
8. for comparison with RBNA requirements for material, NDT and pressure testing as applicable.
9. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.
10. The documentation to contain specification of pressures, pipe dimensions and materials.

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 RBNA may request further documents to be submitted. This may include details of evidence of existing type approval or proposals for a type testing programme in accordance with Part III, Title 62, Section 5, Subchapter H2.
3. The number of copies to be submitted is left to each RBNA.
4. A Licensee is to submit, for each engine type manufactured, a list of all documents required by the RBNA with the relevant drawing numbers and revision status from both Licensor and Licensee.
4a. 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..
4b. In all cases a complete set of documents will be required by the surveyor(s) attending the Licensee's work.
5. Where the operation and service manuals identify special tools and gauges for maintenance purposes (see footnote 7) refer to Part III, Title 63, Section 6, subchapter B3, item 216..
6. The FMEA reports required by FOOTNOTE 6 will not be explicitly approved by the RBNA

200. Astern power [IACS UR-M25]

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

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

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

204. NOTES:

a. The head revolutions as mentioned above are understood as those corresponding to the maximum continuous ahead power for which the vessel is classed.

b. The reversing characteristics of the propulsion plant are to be demonstrated and recorded during trials.

E4. STARTING SYSTEMS

100. Conditions

101. The equipment for starting the main engines and auxiliary essential services is to operate safely and without risk to operators and should allow the engines to start from the dead ship condition using only the vessel's own resources.

102. When the starting is done by means of compressed air the system is to be in compliance with the Rules of Part II, Title 11, Section 6 for piping, equipment and number of starts.

103. In case the starting systems are driven by electric power they should be in compliance with Part II, Title 11, Section 7.

E5. INTERNAL PIPING SYSTEMS IN ENGINES

100. Cooling systems, lubrication and fuel supply.

101. The recommendations on equipment and piping set out in Part II, Title 11, Section 6 are to be complied with.

102. The exhaust of air cooling engines with radiator will be located so as not to generate excessive heating in locations where the motors are installed.

103. When necessary, the air can be released into the atmosphere by means of pipelines, which are to be properly insulated.

104. If the cooling air is aspirated from the engine room its flow is to be added to that intended for the ventilation of machinery spaces.

200. Exhaust gas system

201. The arrangement of piping and mufflers is to meet the recommendations in Part II, Title 11, Section 6 of these Rules.

E6. ENGINE INSTRUMENTATION FOR SHIPS WITH MANNED ENGINE ROOM

100. Instruments

101. Indicators and tachometers are to be installed at any location where it may be possible to start up the engine, as indicated in what follows.

102. Pressure gauges and tachometers should be marked in red with the allowable pressures and the critical speed range.

103. Where the controls and instruments are part of a programmable electronic system, the requirements of Part II, Title 102, Section 5, Subchapter A6 are to be met.

104. Where the main or auxiliary engines are equipped with automation and control system, the requirements of Part II, Title 102, Section 5,Subchapter A7 are to be complied with.

105. For ships with AB $\geq$ 500, refer to the Code for Alarms and Indicators, IMO resolution A.1021 (26) as amended.

200. For propulsion engines

201. At least the following instruments are required to be mounted on a panel installed on the engine in an easily visible location, or when the engine is controlled remotely, installed in the engine control room.

a. Pressure gauges:

a.1. Lubricating oil

a.2. Fresh water cooling - air starting (if applicable), and a.3. Control air (if applicable).

a.4. Fuel oil

b. Thermometers:

b.1. Lubricating oil

b.2. Fresh water cooling of the cylinder liner, inlet and outlet;

b.3. raw water cooling (when applicable).

c. tachometer;

d. hour meter;

e. ammeter.

202. The propeller speed and direction of rotation of the shaft are to be displayed on the bridge and Engine Control Room (if existent).

203. Visual and audible alarms are to be installed for:

a. Low lube oil pressure, and

b. High temperature fresh water cooling.

204. In case the oil pressure drops below the specified shut-off minimum requiring an immediate shut-off of the engine, an audible and visual alarm different from any other alarms is to be triggered, and the engine is to be stopped automatically.

205. The indicators, alarms and shut-offs are to be in accordance with the tables T.E6.400.1; T.E6.400.2; T.E6.400.3 and T.E6.400.4.

300. For auxilliary engines [IACS UR M2]

301. It is required at least the following instruments, which must be mounted on a panel installed on the engine in an easily visible:

a. Pressure gauges:

a.1. Lubricating oil
a.2. Fresh water cooling
a.3. Starting air (if applicable);
a.4. Air control (if applicable) and
a.5. Fuel oil.

b. Thermometers:

b.1. Lubricating oil

b.2. Fresh water cooling

b.3. Starting air (if applicable);

b.4. Air control (if applicable) and

b.5. Fuel oil.

c. Audible alarms for:

c.1. Lub oil low pressure; and

c.2. High and low cooling fresh water temperature.

d. Tachometer or equivalent instrument;

e. hour meter;

f. ammeter.

Vessels with class notation of automation or remote control from the bridge are to follow the requirements of Part II, Title 102, Section 5 - Automation in addition and / or replacement to those here presented

303. The indicators, alarms and shut-offs are to be in accordance with the tables T.E6.400.1; T.E6.400.2; T.E6.400.3 and T.E6.400.4.

400. Table of alarms and indicators

L: reading
A: high pressure or temperature alarm
B: low pressure or temperature alarm
P: Shut-off, stop
AL: Alarm

TABLE T.E6.400.1 – ALARMS AND INDICATORS 1

Description	**MCP's**	**MCA's**	**Emergência**
Rotation	L		
Over-speed (4)	Al, P	Al, P	Al, P
Operation hours	L	L	L

TABLE T.E6.400.2 – ALARMS AND INCICATORS 2

Description	**MCP's**	**MCA's**	**Emergency**
Lub Oil inlet pressure	L, B, P	L, B, P	L, B
Fuel Oil inlet pressure	L	L	
Fresh water jacket cooling outlet	L, B	L, B	L, B
Fresh water piston cooling outlet (1)	L, B	L, B	L, B (4)
Turbo feed air outlet pressure (2)	L		
Starting air preessure(3)	L, B		
Control air pressure	L, B		

TABLE T.E6.400.3 – ALARMS AND INDICADORS 3

Description	MCP's	MCA's	Emer-gency
OL inlet temperature	L, A	L, A (4)	L, A (4)
FO inlet temperature (5)	L	L	
Jacket fresh water cooling outlet temperature	L, A	L, A	L, A
Piston fresh water cooling outlet temperature (1)	L, A		
Feed air inlet temperature (5)	L		
Feed air outlet temperature (5)	L		
Exhaust gases temperature (6)	L, A (7)		

Other alarms and indicators:

TABLE – ALARMS AND INDICATORS 4

Description	MCP's	MCA's	Emerg ency
Fuel leak in the high pressure piping	Al	Al	Al
Crankcase oil mist detector (8), (9), (10)	L, A	L, A	L, A

(1) For engines fitted with piston cooling system separate from the jacket cooling system

(2) For turbo-powered engines

(3) For engines with compressed air starting

(4) For engines with power greater than or equal to 200 kWh

(5) For engines burning heavy oil

(6) Where the installation at each cylinder outlet and turbo inlet and outlet is feasible.

(7) Turbo outlet bo only

(8) For engines with a power greater than 2250 kW and cylinder diameter greater than 300 mm

(9) Alternative methods must be submitted to RBNA

(10) May be provided with engine shut-off as necessary

E7. CONTROL AND SAFETY DEVICES

100. Speed governor and overspeed protective device for main internal combustion engines [IACS UR M3.1] [UR M3.2] [UR M10] [UR M12 M3.1]

101. 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%.

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

103. When electronic speed governors of main internal combustion engines form part of a remote control system, they are to comply with Part II, Title 11, Section 5, subchapter E9 and namely with the following conditions:

a. 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;

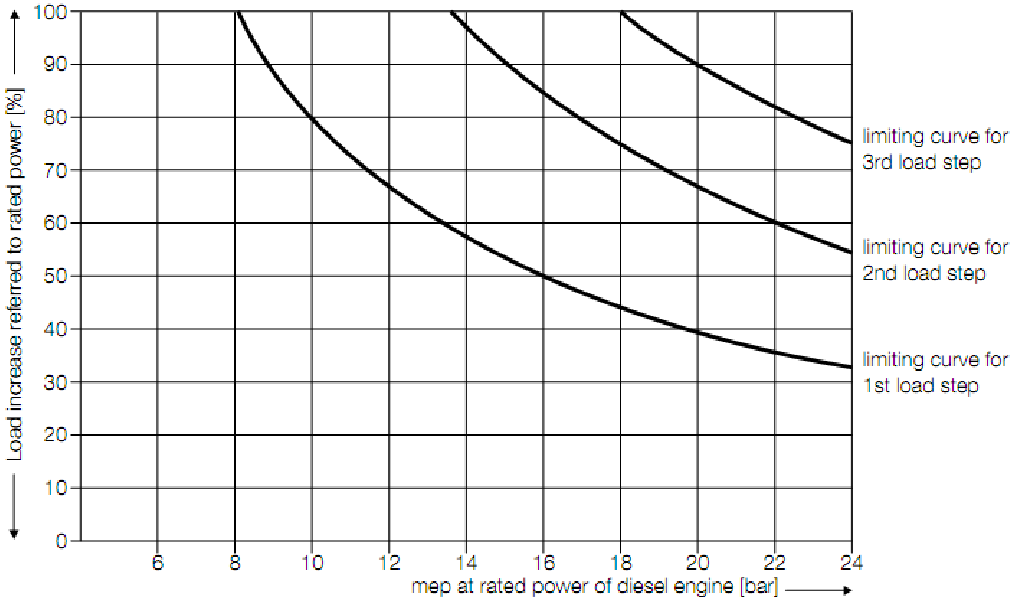
b. local control of the engines is always to be possible, as required by Part II, Title 11, Section 5, subchapter E9, and, to this purpose, from the local control position it is to be possible to disconnect the remote signal, bearing in mind that the speed control according to E7.101. above is not available unless an additional separate governor is provided for such local mode of control.

c. In addition, electronic speed governors and their actuators are to be type tested according to Part III, Title 62, Section 7.

200. Speed governor, overspeed protective and governing characteristics of generator prime movers [IACS URM3.2]

201. Prime movers for driving generators of the main and emergency sources of electrical power are to be fitted with a speed governor which will prevent transient frequency variations in the electrical network in excess of ±10% of the rated frequency with a recovery time to steady state conditions not exceeding 5 seconds, when the maximum electrical step load is switched on or off. In the case when a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10% of the rated speed may be acceptable, provided this does not cause the intervention of the overspeed device as required by Part II, Title 11, Section 5, subchapter E7.100

FIGURE F.E7.205.1 - LIMITING CURVES FOR LOADING 4-STROKE DIESEL ENGINES STEP BY STEP FROM NO-LOAD TO RATED POWER AS FUNCTION OF THE BRAKE MEAN EFFECTIVE PRESSURE



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

203. Prime movers are to be selected in such a way that they will meet the load demand within the ship's mains. Application of electrical load should be possible with 2 load steps and must be such that prime movers – running at no load – can suddenly be loaded to 50% of the rated power of the generator followed by the remaining 50% after an interval sufficient to restore the speed to steady state.

204. Steady state conditions should be achieved in not more than 5 seconds. Steady state conditions are those at which the envelope of speed variation does not exceed +1% of the declared speed at the new power.

205. Application of electrical load in more than 2 load steps can only be permitted, if the conditions within the ship's mains permit the use of such prime movers which can only be loaded in more than 2 load steps (see Figure F.E7.205.1) and provided that this is already allowed for in the designing stage.

206. This is to be verified in the form of system specifications to be approved and to be demonstrated at ship's trials. In this case, due consideration is to be given to the power required for the electrical equipment to be automatically switched on after black-out and to the sequence in which it is connected.

207. This applies analogously also for generators to be operated in parallel and where the power has to be transferred from one generator to another in the event of any one generator has to be switched off.

208. Emergency generator sets must satisfy the governor conditions as per items 1 and 2 even when**:**

a. their total consumer load is applied suddenly, or

b. their total consumer load is applied in steps, subject to:

b.1. the total load is supplied within 45 seconds since power failure on the main switchboard

b.2. the maximum step load is declared and demonstrated

b.3. the power distribution system is designed such that the declared maximum step loading is not exceeded

b.4. the compliance of time delays and loading sequence with the above is to be demonstrated at ship's trials.

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

210. For a.c. generating sets operating in parallel, the governing characteristics of the prime movers shall be such that within the limits of 20% and 100% total load the load on any generating set will not normally differ from its proportionate share of the total load by more than 15% of the rated power of the largest machine or 25% of the rated power of the individual machine in question, whichever is the less.

211. For an a.c. generating set intended to operate in parallel, facilities are to be provided to adjust the governor sufficiently fine to permit an adjustment of load not exceeding 5% of the rated load at normal frequency.

212. Note**:** For guidance, the loading for 4-stroke diesel engines may be limited as given by Figure F.E7.205.1.

300. Protection of internal combustion engines against crankcase explosions [IACS UR M10 (cont)]

301. Crankcase construction and crankcase doors are to be of sufficient strength to withstand anticipated crankcase pressures that may arise during a crankcase explosion taking into account the installation of explosion relief valves required by Part II, Title 11, Section 5, subchapter E7.500. Crankcase doors are to be fastened sufficiently securely for them not be readily displaced by a crankcase explosion.

302. Additional relief valves are to be fitted on separate spaces of crankcase such as gear chain cases for camshaft or similar drives, when the gross volume of such spaces exceeds 0.6 m3.

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

304. Crankcase explosion relief valves are to comply with Part II, Title 11, Section 5, subchapter E7.500

305. Ventilation of crankcase, and any arrangement which could produce a flow of external air within the crankcase, is in principle not permitted except for dual fuel engines where crankcase ventilation is to be provided.

306. Crankcase ventilation pipes, where provided, are to be as small as practicable to minimise the inrush of air after a crankcase explosion.

307. If a forced extraction of the oil mist atmosphere from the crankcase is provided (for mist detection purposes for instance), the vacuum in the crankcase is not to exceed 2.5 *10–4 N/mm2 (2.5 m bar).

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

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

310. A warning notice is to be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. This warning notice is to specify that, whenever overheating is suspected within the crankcase, the crankcase doors or sight holes are not to be opened before a reasonable time, sufficient to permit adequate cooling after stopping the engine.

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

a. for alarm and slow down purposes for low speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore

b. for alarm and automatic shutoff purposes for medium and high speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore

312. Oil mist detection arrangements are to be of a type approved by classification societies and tested in accordance with Part III Title 62 Section 5 subchapter T3 and comply with Part II, Title 11 Section 5, subchapter E7. Engine bearing temperature monitors or equivalent devices used as safety devices have to be of a type approved by classification societies for such purposes.

313. Note: For equivalent devices for high speed engines, refer to IACS Unified Interpretations UI SC 133.

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

a. Schematic layout of engine oil mist detection and alarm system showing location of engine crankcase sample points and piping or cable arrangements together with pipe dimensions to detector.

b. Evidence of study to justify the selected location of sample points and sample extraction rate (if applicable) in consideration of the crankcase arrangements and geometry and the predicted crankcase atmosphere where oil mist can accumulate.

c. The manufacturer's maintenance and test manual.

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

315. A copy of the oil mist detection equipment maintenance and test manual required by Part II, Title 11, Section 5, subchapter E7 is to be provided on board ship.

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

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

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

319. Alarms and shutdowns for the oil mist detection system are to be in accordance with Part II, Title 102,

Section 5, subchapters A8.100 and A8.200 and the system arrangements are to comply with Part II, Title 102, Section 5, subchapters A4.200 and A4.400.

320. 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.

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

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

323. Plans of showing details and arrangements of oil mist detection and alarm arrangements are to be submitted for approval in accordance with Part II, Title 11, Section 5, subchapter E3, item 28 in table T.E3.102.1.

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

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

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

a. Engine particulars – type, power, speed, stroke, bore and crankcase volume.

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

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

d. Operating instructions and the maintenance and test instructions.

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

400. Fire extinguishing systems for scavenge manifolds [IACS UR M12]

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

500. Crankcase explosion relief valves for crankcases of internal combustion engines [IACS UR M9]

501. A safety valve set to operate when the pressure in the cylinder reaches 140% of the maximum pressure of combustion to the maximum service power shall be installed in the main engine cylinders where crankcase diameters exceed 230 mm.

502. For the auxiliary machinery instead of this valve an alarm device of over-pressure in the cylinder of an approved type may be used.

503. Internal combustion engines having a cylinder bore of 200 mm and above or a crankcase volume of 0.6 m3 and above shall be provided with crankcase explosion relief valves as follows:

504. Engines having a cylinder bore not exceeding 250 mm are to have at least one valve near each end, but, over eight crankthrows, an additional valve is to be fitted near the middle of the engine.

505. Engines having a cylinder bore exceeding 250 mm but not exceeding 300 mm are to have at least one valve in way of each alternate crankthrow, with a minimum of two valves.

506. Engines having a cylinder bore exceeding 300 mm are to have at least one valve in way of each main crankthrow.

507. The free area of each relief valve is to be not less than 45 cm2.

508. The combined free area of the valves fitted on an engine must not be less than 115 cm2 per cubic metre of the crankcase gross volume.

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

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

511. Notes

a. The total volume of the stationary parts within the crankcase may be discounted in estimating the crankcase gross volume (rotating and reciprocating components are to be included in the gross volume).

b. Engines are to be fitted with components and arrangements complying with this Chapter, except for E7.513a, E7.514 below and the second bullet point in E7.515, when:

a. an application for certification of an engine is dated on/after 1 January 2006; or

b. installed in new ships for which the date of contract for construction is on or after 1 January 2006.

c. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder.

512. Crankcase explosion relief valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than 0.02 N/mm2 (0.2bar).

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

a. Crankcase explosion relief valves are to type tested in a configuration that represents the installation arrangements that will used on an engine in accordance with Part III, Title 62, Section 5, Chapter J.

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

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

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

b. Copy of type test certification.

c. Installation instructions.

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

e. Actions required after a crankcase explosion.

516. A copy of the installation and maintenance manual required by E7.515 above is to be provided on board ship.

517. Plans of showing details and arrangements of crankcase explosion relief valves are to be submitted for approval.

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

a. Name and address of manufacturer

b. Designation and size

c. Month/Year of manufacture

d. Approved installation orientation

600. Protective devices for starting air mains [IACS UR M11]

601. In order to protect starting air mains against explosion arising from improper functioning of starting valves, an isolation non-return valve or equivalent at the starting air supply connection to each engine

602. For engines having a bore > 230 mm a bursting disc or flame arrester in way of the starting valve of each cylinder for direct reversing engines having a main starting manifold at the supply inlet to the starting air manifold for non-reversing engines.

700. Protection of the engine casing

701. The engine casing is to be of reinforced construction and the inspection ports and fittings, will be dimensioned so as not to undergo permanent deformation due to an over-pressure within the casing.

702. The engine casing should be designed so as to eliminate any large flow of air, and so as to minimize all kinds of lubricating oil leaking through the joints of the inspection doors.

703. The openings for cleaning should be no larger than necessary in order to prevent entry of outside air.

704. Whenever any mechanical suction is installed, it shall not exceed 0.245 N/cm2 (25 mm CA).

705. The installation of alarms indicating the presence of viscosity problems within the engine casing or overheating of moving parts is recommended.

706. A combination of vent pipes for two or more engines on one only may be allowed, always with the outlet in the engine room.

707. In the control room, or preferably next to an inspection door, on each side of the engine, casing, a sign indicating that the inspection doors should not be opened before a sufficient time to allow adequate cooling of the engine is to be installed. This time span should not normally be less than 10 minutes after stopping the engine.

800. Relief Valves

801. A relief valve is to be fitted in closed casings of engines having a cylinder diameter greater than 200 mm or whose casing has a total volume greater than 0.6 m^3.

802. The relief valves are to be of the return type to the piping system, with low inertia, and should readily discharge the over-pressure not greater than 0.2 bar (0.2 kgf/cm^2), closing rapidly after the passage of blast wave so as to avoid sharp entry of air. The arrangement and location of the valves are to be made considering the possibility of minimizing the hazards resulting from flame irruption.

803. The engines are to have at least the following relief valves, where “d” is the cylinder diameter in mm:

a. engines where $200 < d \leq 250$: one valve in the vicinity of each end and, if the engine has more than eight cranks, another near the middle of the engine;

b. engines where $250 < d \leq 300$: one valve corresponding to each alternate crank, plus at least two valves;

c. engines where d> 300, one valve corresponding to each crank.

804. The free area of each relief valve shall be at least 45 cm^2 and the total area of all the relief valves must not be less than 115 cm^2 per each 1 cm^3 of the gross total engine casing volume. When the estimating the gross volume of the casing, the volume of the stationary parts may be reduced.

805. In two-stroke engines with cylinder diameter greater than 230 mm relief valves shall be installed in the air washing caissons, in case they have open connection with the cylinders.

900. Alarm device

901. A machinery alarm system for temperatures and pressures greater than indicated by engine manufacturer must be installed, except for air pressure turbo charger control air and exhaust gas temperature, with indication on the bridge.

902. There is to be indication on the bridge that the alarms have been triggered.

903. In the case of azimuth propeller an alarm system for temperatures and pressures given by the manufacturer must be installed. It must also have an indication on the bridge.

E8. PARTS FOR MAIN AND AUXILIARY DIESEL ENGINES

100. Application [IACS Rec 26 Rec 27]

101. The parts list below is not required for classification, but is provided here as a recommendation for ships with AB> 500 for unlimited service in the open sea. Depending on the design of the engine, other parts than those listed below, such as circuit boards of electronic controls should be considered.

TABLE T.E8.101.1 - RECOMMENDATION TO THE MINIMUM AMOUNT OF MAIN PARTS REQUIRED FOR INTERNAL COMBUSTION ENGINES

Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1 set
Main thrust block	Pads for one face of Michell-type thrust block, or	1 set
	Complete white metal thrust shoe of solid ring type	1 set
	Inner and outer race with rollers, where roller-thrust bearings are fitted	1 set
Cylinder liner	Cylinder liner, complete with joint rings and gaskets	1
Cylinder cover	Cylinder cover, complete with valves, joint rings and gaskets.	1
Cylinder valves	Cylinder cover bolts and nuts, for one cylinder	½ set
	Air inlet valves, complete with casings, seats, springs and other fittings for one cylinder	1 set
	Starting air valve, complete with casting, seat springs and other fittings	1
	Cylinder overpressure sentinel valve, complete	1
	Fuel valves of each size and type fitted, complete with all fittings, for one engine	1 set
Cylinder cooling	Cooling pipes, fittings, and seals or their equivalent for one cylinder unit	1 set
Connecting rod 1 set Bearings	Bottom end bearings or shells of each size and type fitted, complete with shims, bolts and nuts, for one cylinder	1 set
	Top end bearings or shells of each size and type fitted, complete with shims, bolts and nuts, for one cylinder	1 set
Pistons	Crosshead type: piston of each type fitted, complete with piston rod, stuffing box, skirt, rings, studs and nuts	
	Trunk piston type: piston of each type fitted, complete with skirt, rings, studs, nuts, gudgeon pin and connecting rod	
Piston rings	Piston rings, for one cylinder	1 set
Piston cooling	Telescopic cooling pipes and fittings or their equivalent, for one cylinder unit	1 set

E9. BRIDGE CONTROL OF PROPULSION MACHINERY FOR ATTENDED MACHINERY SPACES [IACS UR M47] [UR M43]

100. Bridge control of propulsion machinery for attended machinery spaces [IACS UR M47] [UR M43]

101. Under all sailing conditions, including manoeuvring, the speed, direction of thrust and, if applicable, pitch of the propeller shall be fully controllable from the navigating bridge.

102. In principle the remote control mentioned under E5.201 above is to be performed by a single control device for each independent propeller, with automatic performance of all associated services including, where necessary, means of preventing overload and prolonged running in critical speed ranges of the propelling machinery.

103. The bridge control system is to be independent from the other transmission system; however, one control lever for both systems may be accepted.

104. Operations following any setting of the bridge control d vice including reversing from the maximum ahead service speed in case of emergency are to take place in an automatic sequence and with time intervals acceptable to the machinery.

105. The main propulsion machinery shall be provided with an emergency stopping device on the navigating bridge and independent from the bridge control system.

106. Remote starting of the propulsion machinery is to be automatically inhibited if conditions exist which may hazard the machinery, e.g. shaft turning gear engaged, drop of lubricating oil pressure.

107. For steam turbines a slow-turning device should be provided which operates automatically if the turbine is stopped longer than admissible. Discontinuation of this automatic turning from the bridge must be possible.

108. The design of the bridge control system shall be such that in case of its failure an alarm is given. In this case the speed and direction of the propeller thrust is to be maintained until local control is in operation, unless this is considered impracticable. In particular, lack of power (electric, pneumatic, hydraulic) should not lead to major and sudden change in propulsion power or direction of propeller rotation.

109. The number of automatic consecutive attempts which fail to produce a start shall be limited to maintain sufficient starting air pressure. An alarm shall be provided at an air pressure level, which still permits main engine starting operation.

110. It shall be possible for the propulsion machinery to be controlled from a local position even in the case of failure in any part of the automatic or remote control systems.

111. Remote control of the propulsion machinery shall be possible only from one control location at one time; at such locations interconnected control positions are permitted.

112. The control system shall include means to prevent the propelling thrust from altering significantly when transferring control from one control to another.

113. Each control location is to be provided with means to indicate which of them is in control. Propolusion machinery orders from the navigating bridge shall be indicated in the engine control room or at the manoeuvring platform, as appropriate.

114. The transfer of control between the navigating bridge and machinery spaces shall be possible only in the main machinery space or the main machinery control room.

CHAPTER F
OTHER MACHINERY

CHAPTER CONTENTS

F1. TURBINES AND OTHER DRIVERS

100. Application

101. The installation of turbines and other engines other than internal combustion will be subject to special examination by RBNA.

F2. AUXILIARY MACHINERY

100. Application

101. The auxiliary machinery installations are to be certified under the RBNA supervision of and be in compliance with the relevant Rules requirements.

F3. ALARMS AND SAFEGUARDS FOR EMERGENCY DIESEL ENGINES [IACS UR M63]

100. Field of application

101. These requirements apply to diesel engines required to be immediately available in an emergency and capable of being controlled remotely or automatically operated.

200. Information to be submitted

201. Information demonstrating compliance with these requirements is to be submitted to the relevant RBNA. The information is to include instructions to test the alarm and safety systems.

300. Alarms and safeguards

301. Alarms and safeguards are to be fitted in accordance with Table T.F3.301.1.

302. The safety and alarm systems are to be designed to ‘fail safe’. The charateristics of the ‘fail safe’ operation are to be evaluated on the basis not only of the system and its associated machinery, but also the complete installation, as well as the ship.

303. Regardless of the engine output, if shutdowns additional to those specified in Table T.F3.301.1 are provided except for the overspeed shutdown, they are to be automatically overridden when the engine is in automatic or remote control mode during navigation.

304. The alarm system is to function in accordance with Part II, Title 102, Section 5, subchapter A4.200, with additional requirements that grouped alarms are to be arranged on the bridge.

305. In addition to the fuel oil control from outside the space, a local means of engine shutdown is to be provided.

306. Local indications of at least those parameters listed in Table T.F3.301.1 are to be provided within the same space as the diesel engines and are to remain operational in the event of failure of the alarm and safety system

400. List of minimum recommended spare parts for essential auxiliary machinery of ships for unrestricted service [IACS Rec 30]

401. Auxiliary internal combustion engines and steam turbines driving essential service machinery other than generators. The number of minimum recommended spare parts for auxiliary internal combustion engines and steam turbines driving essential service machinery is to be in accordance with that recommended for internal combustion engines and turbines driving electric generators. When an additional unit for the same purpose and of adequate capacity is fitted, spare parts may be omitted.

TABLE T.F3.301.1 – ALARMS AND SAFEGUARDS

Parameter	≥ 220kW	<220kW
Fuel oil leakage from pressure pipes	❍	❍
Lubricating oil temperature	◓	
Lubricating oil pressure	◒	◒
Oil mist concentration in crankcase[1]	◓	
Pressure or flow of cooling water	◒	
Temperature of cooling water (or cooling air)	◓	◓
Overspeed activated	❍ + ❐	

Note:

1 for engines having a power of more than 2250 kW or a cylinder bore of more than 300mm.

◒ **Alarm for low value**
◓ **Alarm for high value**
❍ **Alarm activated**

TABLE T.F3.401.1 – RECOMMENDED SPARES FOR PUMPS

Spare parts	Number recommended
1. Piston pumps	
1.1 Valve with seats and springs, each size fitted	1 set
1.2 Piston rings, each type and size for one piston	1 set
2. Centrifugal pumps	
2.1 Bearing of each type and size	1
2.2 Rotor sealings of each type and size	1
3. Gear type pumps	
3.1 Bearings of each type and size	1
3.2 Rotor sealings of each type and size	1
NOTES 1. When a sufficiently rated standby pump is available, the spare parts may be dispensed with.	

TABLE T.F3.401.2 – SPARES FOR COMPRESSORS FOR ESSENTIAL SERVICE

Spare parts	Number recommended
1.Suction snd delivery valves complete of each size fitted in one unit	½ set
2.Pison rings of each type and size fitted for one piston	1 set

500. General

501. It is recommended that where, for maintenance or repair work of the essential machinery, special tools or equipment are to be used, these are available on board. When the recommended spares are utilized, it is recommended that new spares are supplied as soon as possible.

CHAPTER G
SHAFT TRANSMISSION LINES

CHAPTER CONTENTS

G1. APPROACH

100. Application

101. The present Subchapter G1 applies to the propulsion shaft lines of the conventional type. When the combination of the components of the shaft line is such that it can not be dimensioned by the criteria set forth herein, sufficient data to verify the design is to be submitted for RBNA approval.

G2. INSTALLATION OF SHAFTS

100. Alignment

101. The shafts alignments are to be supervised by a RBNA surveyor, and a record of the measurements to be submitted to RBNA.

200. Protection against corrosion

201. The application of the coating of resin reinforced with fiberglass is permitted provided that the procedure for application and implementation of the service are approved by RBNA. The fiber is to braided.

FIGURE F.G2.201.1 – BREAIDED FIBERGLASS APPLICATION

202. In propeller shafts with a diameter up to 150 mm, the coating being applied is to be at least two layers of mesh fabric 330 g/ m^2 and / or normal fabric and a mat "roving" of 330 g/ m^2 and / or a "mat "of 450 g/m^2.

203. For propeller shafts with diameter greater than 150 mm, the coating should consist of at least three layers of mesh fabric and / or normal fabric of 330 g/ m^2 and one or more mats "roving" 300 g/ m^2.

204. For protection in the region of the coupling to the propeller see Chapter H, in this Section.

G3. DIMENSIONS OF PROPULSION SHAFTS AND THEIR PERMISSIBLE TORSIONAL VIBRATION STRESSES [IACS UR M68]

100. Scope
[IACS UR M68.1]

101. This subchapter H3 applies to propulsion shafts such as intermediate and propeller shafts of traditional straight forged design and which are driven by rotating machines such as diesel engines, turbines or electric motors.

102. For shafts that are integral to equipment, such as for gear boxes, podded drives, electrical motors and/or generators, thrusters, turbines and which in general incorporate particular design features, additional criteria in relation to acceptable dimensions have to be taken into account. For the shafts in such equipment, the requirements of this subchapter may only be applied for shafts subject mainly to torsion and having traditional design features. Other limitations, such as design for stiffness, high temperature etc. are to be addressed by specific Rules of the RBNA.

103. Explicitly the following applications are not covered by this subchapter:

a. additional strengthening for shafts in ships classed for navigation in ice

b. gearing shafts

c. electric motor shafts

d. generator rotor shafts

e. turbine rotor shafts

f. diesel engine crankshafts (see Chapter J below)

g. unprotected shafts exposed to sea water

200. Alternative calculation methods [IACS UR M68.2]

201. Alternative calculation methods may be considered by the RBNA. Any alternative calculation method is to include all relevant loads on the complete dynamic shafting system under all permissible operating conditions. Consideration is to be given to the dimensions and arrangements of all shaft connections.

202. Moreover, an alternative calculation method is to take into account design criteria for continuous and transient operating loads (dimensioning for fatigue strength) and for peak operating loads (dimensioning for yield strength). The fatigue strength analysis may be carried out separately for different load assumptions, for example as given in

300. Material limitations [IACS UR M68.3]

301. Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the materials are to have a specified minimum ultimate tensile strength (σ_B) of 500 N/mm2. Otherwise materials having a specified minimum ultimate tensile strength (σ_B) of 400 N/mm2 may be used.

302. For use in the following formulae in this subchapter, σ_B is limited as follows:

a. For carbon and carbon manganese steels, a minimum specified tensile strength not exceeding 600 N/mm2 for use in item G3.500 below and not exceeding 760 N/mm2 in G3.400 below.

b. For alloy steels, a minimum specified tensile strength not exceeding 800 N/mm2.

c. For propeller shafts in general a minimum specified tensile strength not exceeding 600 N/mm2 (for carbon, carbon manganese and alloy steels).

303. Where materials with greater specified or actual tensile strengths than the limitations given above are used, reduced shaft dimensions or higher permissible vibration stresses are not acceptable when derived from the formulae in this subchapter.

400. Shaft diameters [IACS UR M68.4]

401. Shaft diameters are not to be less than that determined from the following formula:

$$d = F \bullet k \bullet \sqrt[3]{\frac{P}{n_o} \bullet \frac{1}{1-\frac{d_i^4}{d_o^4}} \bullet \frac{560}{\sigma_B + 160}}$$

where:
d = minimum required diameter in mm
di = actual diameter in mm of shaft bore
do = outside diameter in mm of shaft. If the bore of the shaft is ≤0.40do, the expression

$$1 - d_i^4 / d_o^4$$

may be taken as 1.0
F = factor for type of propulsion installation
= 95 for intermediate shafts in turbine installation, diesel installations with hydraulic (slip type) couplings, electric propulsion installations
= 100 for all other diesel installations and all propeller shafts
k = factor for the particular shaft design features, see G3.700 below

n0 = speed in revolutions per minute of shaft at rated power

p = rated power in kW transmitted through the shaft (losses in gearboxes and bearings are to be disregarded) σ_B= specified minimum tensile strength in N/mm2 of the shaft material, see G3.300 above

402. The diameter of the propeller shaft located forward of the inboard stern tube seal may be gradually reduced to the corresponding diameter required for the intermediate shaft using the minimum specified tensile strength of the propeller shaft in the formula and recognizing any limitations given in G3.300 above.

500. Permissible torsional vibration stresses [IACS UR M68.5]

501. The alternating torsional stress amplitude is understood as $(\tau_{max} - \tau_{min})/2$ as can be measured on a shaft in a relevant condition over a repetitive cycle.

503. Torsional vibration calculations are to include normal operation and operation with any one cylinder misfiring (i.e. no injection but with compression) giving rise to the highest torsional vibration stresses in the shafting.

504. For continuous operation the permissible stresses due to alternating torsional vibration are not to exceed the values given by the following formulae:

$$\pm \tau_C = \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet \left(3 - 2 \bullet \lambda^2\right) \text{ for } \lambda < 0.9$$

$$\pm \tau_C = \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet 1.38 \quad \text{for } 0.9 \le \lambda \le 1.05$$

where:

τ_C = permissible stress amplitude in N/mm2 due to torsional vibration for continuous operation

σ_B = specified minimum ultimate tensile strength in N/mm2 of the shaft material, see also G3.300

c_K = factor for the particular shaft design features, see G3.600
$c_D = 0.35 + 0.93\, d_0^{-0.2}$ - size factor
do = shaft outside diameter in mm
λ = speed ratio = n/n_0
n = speed in revolutions per minute under consideration
n0 = speed in revolutions per minute of shaft at rated power

505. Where the stress amplitudes exceed the limiting values of τ_C for continuous operation, including one cylinder misfiring conditions if intended to be continuously operated under such conditions, restricted speed ranges are to be imposed which are to be passed through rapidly.

506. Restricted speed ranges in normal operating conditions are not acceptable above $\lambda = 0.8$.

507. Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships are to enable safe navigation. The limits of the barred speed range are to be determined as follows:

a. The barred speed range is to cover all speeds where the acceptance limits (τ_C) are exceeded. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered. Additionally the tachometer tolerance has to be added. At each end of the barred speed range the engine is to be stable in operation.

b. In general and subject to (a) the following formula may be applied, provided that the stress amplitudes at the border of the barred speed range are less than τ_C C under normal and stable operating conditions.
where:

$$\frac{16 \bullet n_c}{18 - \lambda_c} \le n \le \frac{(18 - \lambda_c) \bullet n_c}{16}$$

where:
n_c = critical speed in revolutions per minute (resonance speed)
λ_c = speed ratio = n_c / n_o

508. For the passing of the barred speed range the torsional vibrations for steady state condition are not to exceed the value given by the formula:

$$\pm \tau_T = 1.7 \bullet \tau_C / \sqrt{c_K}$$

where

τ_T - permissible stress amplitude in N/mm^2 due to steady state torsional vibration in a barred speed range.

600. Requirements for shafts complying with Part II, Title 11, Section 5, Chapter G [IACS UR M68.7.1]

601. Low cycle fatigue criterion (typically < 104), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque if applicable. This is addressed by the formula in G3.400.

602. High cycle fatigue criterion (typically >> 107), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses. The limits for torsional vibration stresses are given in G3.500. The influence of reverse bending stresses is addressed by the safety margins inherent in the formula in G3.400.

603. The accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation is addressed by the criterion for transient stresses in G3.500.

700. Explanation of k and c_K. [IACS UR M68.7.2]

701. The factors k (for low cycle fatigue) and cK (for high cycle fatigue) take into account the influence of:

a. The stress concentration factors (scf) relative to the stress concentration for a flange with fillet radius of 0.08do (geometric stress concentration of approximately 1.45).

$$c_K \approx \frac{1.45}{scf} \quad \textbf{and} \quad k \approx \left[\frac{scf}{1.45}\right]^x$$

where the exponent *x* considers low cycle notch sensitivity.

b. The notch sensitivity. The chosen values are mainly representative for soft steels (σ_B<600), while the influence of steep stress gradients in combination with high strength steels may be underestimated.

c. The size factor c_D being a function of diameter only does not purely represent a statistical size influence, but rather a combination of this statistical influence and the notch sensitivity. The actual values for k and cK are rounded off.

800. Stress concentration factor of slots [IACS UR M68.7.3]

801. The stress concentration factor (scf) at the end of slots can be determined by means of the following empirical formulae using the symbols in footnote 6):

$$scf = \alpha_{t(hole)} + 0.57 \bullet \frac{(1-e)/d}{\sqrt{\left(1-\frac{d_i}{d}\right) \bullet \frac{e}{d}}}$$

802. This formula applies to:

a. slots at 120 or 180 or 360 degrees apart.
b. slots with semicircular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.

c. slots with no edge rounding (except chamfering), as any edge rounding increases the scf slightly.

803. αt(hole) represents the stress concentration of radial holes (in this context e = hole diameter) and can be determined as :

$$\alpha_{t(hole)} = 2.3 - 3 \cdot \frac{e}{d} + 15 \cdot \left(\frac{e}{d}\right)^2 + 10 \cdot \left(\frac{e}{d}\right)^2 \cdot \left(\frac{d_i}{d}\right)^2$$

or simplified to $\alpha_{t(hole)} = 2.3$

[UR M68.6] TABLE T.G3.508.1 - K AND CK FACTORS FOR DIFFERENT DESIGN FEATURES (see G3.700 below)

intermediate shafts with						thrust shafts external to engines		propeller shafts		
integral coupling flange1) and straight sections	shrink fit coupling2)	Keyway, tapered connection 3)4)	Keyway, cylindrical connection 3)4)	radial hole5)	longitudinal slot6)	on both sides of thrust collar1)	in way of bearing when a roller bearing is used	Flange mounted or keyless taper fitted propellers8)	Key fitted propellers8)	Between forward end of aft most bearing and forward stern tube seal
k=1.0	1.0	1.10	1.10	1.10	1.20	1.10	1.10	1.22	1.26	1.15
c_K=1.0	1.0	0.60	0.45	0.45	0.30[7)]	0.85	0.85	0.55	0.55	0.80

Note: Transitions of diameters are to be designed with either a smooth taper or a blending radius. For guidance, a blending radius equal to the change in diameter is recommended.

Footnotes

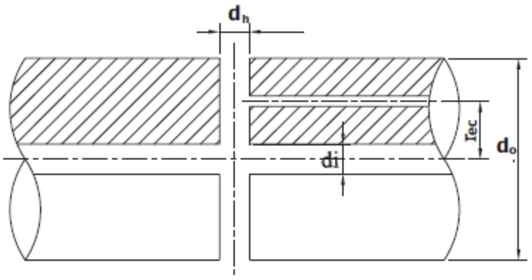
1) Fillet radius is not to be less than 0.08d.

2) k and ck refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter is to be provided, e.g. a diameter increase of 1 to 2 % and a blending radius as described in the table note.

3) At a distance of not less than 0.2do from the end of the keyway the shaft diameter may be reduced to the diameter calculated with k=1.0.

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

5) Diameter of radial bore (dh) not to exceed 0.3do. The intersection between a radial and an eccentric (rec) axial bore (see below) is not covered by this subchapter



6) Subject to limitations as slot length (l)/outside diameter < 0.8 and inner diameter (di)/outside diameter < 0.8 and slot width (e)/outside diameter > 0.10. The end rounding of the slot is not to be less than e/2. An edge rounding should preferably be avoided as this increases the stress concentration slightly. The k and cK values are valid for 1, 2 and 3 slots, i.e. with slots at 360 respectively 180 and respectively 120 degrees apart.

7) cK = 0.3 is a safe approximation within the limitations in 6). If the slot dimensions are outside of the above limitations, or if the use of another cK is desired, the actual stress concentration factor (scf) is to be documented or determined fromG3.800. In which case:

$$c_K = 1.45 / scf$$

Note that the scf is defined as the ratio between the maximum local principal stress and √3 times the nominal torsional stress (determined for the bored shaft without slots).

8) Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2.5 times the required diameter.

G4. LENGTH OF AFT STERN BUSH BEARING BUSH BEARING [IACS UR M52]

100. Oil lubricated bearings of white metal [IACS UR M52.1]

101. 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.

102. 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.

200. Oil lubricated bearings of synthetic rubber, reinforced resin or plastic materials [IACS UR M52.2]

201. 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.

202. 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.

300. Water lubricated bearings of lignum vitae [IACS UR M52.3]

301. 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.

302. Note: Lignum vitae is the generic name for several dense, resinous hardwoods with good lubricating properties. The original high quality Lignum Vitae is almost unobtainable and other types of wood such as Bulnesia Sarmiento (or Palo Santo or Bulnesia Arabia) are commonly used now.

400. Water lubricated bearings of synthetic material [IACS UR M52.4]

401. 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.

402. For a bearing design substantiated by experiments to the satisfaction of the RBNA 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.

Guidance

Guide values for the maximum permissible distance between bearings l_{max} [mm] can be determined using the formula (1):

$$(1) \qquad l_{max} = K1\sqrt{d}$$

d = diameter of shaft between bearings [mm]

K_1 = 450 for oil-lubricated white metal bearings

= 280 for grey cast iron, grease-lubricated stern tube bearings

= 280 – 350 for water-lubricated rubber bearings in stern tubes and shaft brackets (upper values for special designs only).

Where the shaft speed exceeds 350 min^{-1} it is recommended that the maximum bearing spacing is determined in accordance with formula (2) in order to avoid excessive loads due to bending vibrations. In limiting cases a bending vibration analysis for the shafting system is recommended.

$$(2) \qquad l_{max} = K2.\sqrt{(d/n)}$$

n = shaft speed [min^{-1}]

K_2 = 8400 for oil-lubricated white metal bearings

= 5200 for grease-lubricated, grey cast iron bearings and for rubber bearings inside stern tubes and tail shaft brackets.

In general, the distance between bearings should not be less than 60 % of the maximum permissible distance as calculated using formula (1) or (2) respectively.

End of guidance

500. Liners

501. The thickness *e* (in mm) of the bronze liners fitted on the propeller shaft or the taishaft shaft in the region of bearings, shall not be less than that calculated by the formula:

e = 0,04 (dp + 130)

502. Liners of other materials will be subject to special consideration of the RBNA.

503. Outside the region of the liners continuous bronze bearing may have thickness reduced to 75% *e*.

504. All the liners are to be shrunk-fitted by contraction or forced onto the shaft, under pressure, and locked by pins or other similar devices should not be fitted.

505. Where the liner in the region between the bearings is fitted leaving a cleatance, such clearance is to be filled up under pressure with a water-insoluble non-corrosive material.

600. Bushings

601. The bushing thickness depends on the bushing material to be used and will be examined by RBNA.

700. Thrust bearings

701. The location and design of the thrust bearing not built into the motor or gearbox are to be presented for approval.

G5. COUPLINGS

100. Flanges

101. The thickness of the coupling flanges cast integrally with shafts, for conventional design of the shaft line, is to be at least equal to 25% of the diameter calculated for the corresponding shaft.

102. When the couplings are not fitted through flanges cast integrally with the shafts, they are to be fitted and sized so to withstand the tangential forces and the thrust force in reverse.

200. Coupling bolts [IACS UR M34]

201. For intermediate, thrust and propeller shaft couplings having all fitted coupling bolts, the coupling bolt diameter is not less than that given by the following formula:

$$d_b = 0.65\sqrt{\frac{d^3(T+160)}{iDT_b}}$$

where

d_b = diameter (mm) of fitted coupling bolt

d = Rule diameter (mm), i.e., minimum required diameter of intermediate shaft made of material with tensile strength T, taking into account ice strengthening requirements where applicable

i = number of fitted coupling bolts

D = pitch circle diameter (mm) of coupling bolts

T = tensile strength (N/mm2) of the intermediate shaft material taken for calculation

T_b = tensile strength (N/mm2) of the fitted coupling bolts material taken for calculation while: $T \leq T_b \leq 1{,}7T$, but not higher than 1000 N/mm2.

202. The design of coupling bolts in the shaftline other than that covered by G5.201 above are to be considered and approved by the RBNA individually.

203. For intermediate shafts, thrust shafts and inboard end of propeller shafts the flange is to have a minimum thickness of 0,20 times the rule diameter d of the intermediate shaft or the thickness of the coupling bolt diameter calculated for the material having the same tensile strength as the corresponding shaft, whichever is greater.

204. Special consideration will be given by the RBNA for flanges having non-parallel faces, but in no case is the thickness of the flange to be less than the coupling bolt diameter.

205. Fillet radii at the base of the flange should in each case be not less than 0,08 times the actual shaft diameter. Fillets are to have a smooth finish and should not be recessed in way of nuts and bolt heads. The fillet may be formed of multiradii in such a way that the stress concentration factor will not be greater than that for a circular fillet with radius 0,08 times the actual shaft diameter.

Guidance

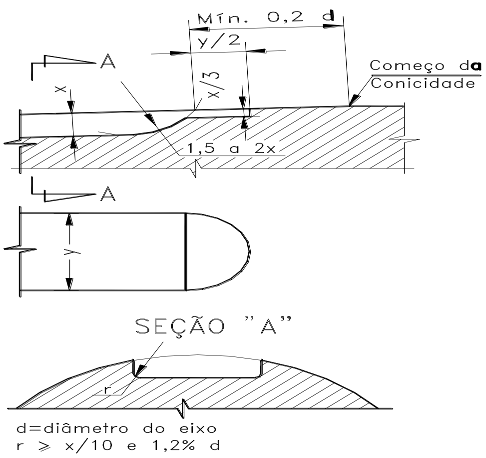
Note: "Circle of pitch" *is the circumference passing through the center of the holes of the coupling bolts.*

Eng of guidance

300. Keys

301. In the keyed connections, the stress concentrations are to be reduced by rounding the corners of its slot (see Figure F.G5.301.1 - Keys, for example). The radii of the corners of the keyway should not be less than 1.2% of the diameter of the shaft.

.FIGURE F.G5.301.1 - KEYS



Começo da conicitade = taper start
Diâmetro do eixo d = shaft diameter

302. The threaded holes for fixing the keys are not to be located at a distance less than 1.5 times the width of the key, from the forward end of the slot.

303. The key is sized to transmit the maximum torque shaft "T", with areas obeying the following values:

a. Tangencial area:

$$at = \frac{1{,}4 \times T}{r} \times \frac{2\sqrt{3}}{\sigma y} \times 10^3$$

b. Lateral area (half height of the key):

$$al = \frac{1{,}4 \times T}{R} \times \frac{2}{\sigma y} \times 10^3$$

where

σy: yield strength (Kgf/mm²)

R shaft radius at the slot location, in mm.

T: Torque in kgf×m (daN×m), which shall be calculated as follows:

$$T = 716{,}2 \times \frac{P}{R} \quad RPM$$

where:

P is the power transmitted in CV.

400. Taper and threaded end

401. The taper of the coupling is to be in accordance with Table T.G5.401.1 as follows:

TABLE TG5.401.1 – TAPER OF COUPLINGS

flange of coupling and shatft	between 1:10 e 1:20
propeller and propulsion shaft	between 1:10 e 1:15
propeller and propulsion shaft (assembly with oil)	between 1:15 e 1:20

402. The outer diameter of screw thread of shaft end should not be less than 60% of the largest diameter of the cone.

G6. TORSIONAL VIBRATIONS

100. Application

101. Torsional loads for the purposes of these Rules are additional loads due to torsional vibrations in main or auxiliary engines.

102. The calculation of torsional vibrations, covering the entire range of speeds and expected conditions will be required for installations where the power transmitted by shaft exceeds 373 kW (500 BHP).

103. Such calculations are to include:

a. Basic data used to establish such calculations and more particularly the dynamic characteristics of the equivalent system of installation, ie, motors, shafts, propellers, gearboxes, etc..;

b. Tables of natural frequencies;

c. Vectorial sum of the amplitudes due to motor impulses for each vibration mode and the various harmonics that can produce dangerous critical speeds;

d. Order of combustion;

e. Characteristics of torsional vibration dampers, when used, and data to allow verification of their efficiency.

200. Measurement of torsional vibration

201. During the sea trials the torsional vibration of the propulsion system is to be measured, covering the entire range of service speeds. The values obtained will be submitted to RBNA.

202. The RBNA may waive the measurement of torsional vibrations during the sea trials in the case an identical propulsion installation report is presented, or when the shaft power transmitted by the propulsion installation is less than 149 kW (200 BHP) per shaft .of being presented .

300. Prohibited speed ranges

301. When the results of calculations or measurements of torsional vibrations show critical speeds for which additional loads are excessive, operaton in such speed ranges is to be prohibited for continuous service.

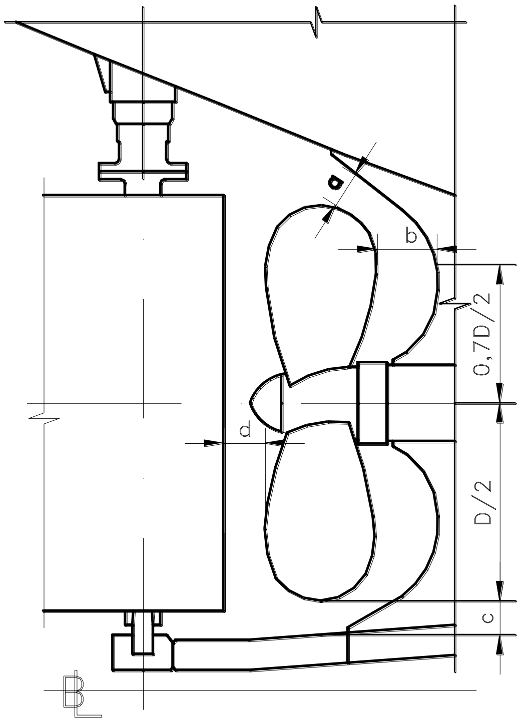
302. These ranges of critical speeds are to be marked in red on the speed dial and a plate is to be fixed near the location of the motor control with instructions indicating the prohibited speeds.

303. Speed ranges are also prohibited for continuous service where the torsional vibration, while not causing an increase in torsional loads, may cause damage to some parts of the installation, such as gear teeths, couplings etc

400. Propeller clearances

401. Propeller clearances recommended to avoid vibrations are indicated in Figure F.G6.401.1.

FIGURE F.G6.401.1 – PROPELLES CLEARANCES



Where:

D: propeller diameter
a: 0,1 D
b: 0,2 D
c: 1,5 a
d: 0,1 D

402. Propeller without a sole piece: it is recommended to leave a clearance between the propeller and the lowest point of the hull of the order of 0.15 D to 0.2 D.

CHAPTER H
[IACS M56]
GEAR REDUCTION/REVERSING BOXES AND COUPLINGS

CHAPTER CONTENTS

H1. APPROACH

100. Application

101. The present Rules apply to propulsion main engine gearboxes and couplings and, maneuvering or safety of the vessel whose power output is greater than 373 kW (500 BHP). Reduction and reversing boxes with less than 373 kW power need not be classified, but are to be installed and tested to the satisfaction of the surveyor.

102. Gearboxes and couplings of the most important auxiliary machinery shall be type approved by RBNA.

103. Gearboxes and reversion gears of series manufacturing for maximum continuous power of up to 140 kW (190 hp), with proven performance, may be approved upon submission of general features and designs to RBNA for approval and monitoring.

104. The Rules of Part II, Title 11, Section 5, Chapter H3 apply to straight or helical tooth gears, internal or external with parallel shafts and bevel gears as well'.

105. The Rules of this Part II, Title 11, Section 5, Chapter H3 are not applicable when the following conditions are present:

a. helical or straight tooth gears with transverse contact ratio ε_a less than 1.0;

b. helical or straight tooth gears with transverse contact ratio ε_a greater than 2.5;

c. the existence of interference between the top tooth and curvatures of the roots of tooth;

d. sharp teeth;

e. the existence of zero clearance between teeth.

200. Standards

201. This Part II, Title 11, Section 5, Chapter H is based on the following standards and requirements:

a. For bevel gears:

ISO / DIS 10300
NBR 7262

The nomenclature is in accordance with:

NBR 6174,
NBR 6884.

b. To spur gears:

ISO 6336-1,
ISO 6336-2,
ISO 6336-5,
Requirements of IACS M56.

The nomenclature of gears is shown in Figure F.H1.301.1 - Basic nomenclature straight tooth gears.

300. Definitions

301. The symbols used in Part II, Title 11, Title 11, Section 5, Chapter H are defined below. Other symbols introduced in connection with the definition of influence factors are described in the appropriate sections.

302. **Definitions of Symbols:**

The definition of key parameters is given below. The complete list of symbols is shown in Table T.H2.100.1

See illustrations of basic the nomenclature ob the basic parameters of spur gears in figure F.H1.301.1

P maximum continuous power transmitted by the gear set in kWh

$\mathbf{n}_{1,2}$ Rotational speed of the pinion or wheel, rpm

Guidance

Wheel also designated as crown

End of guidance

u Gear ratio

a Centre distance, mm, defined as the shortest distance between centres that do not intersect

b Common face width, in mm, which for double helical gear is defined as the width of a helix

$\mathbf{b}_{1,2}$ facewidth of pinion, wheel

B Total face width of double helical gear including the gap width in mm

d Reference diametre

$\mathbf{d}_{1,2}$ Reference diameter of pinion, wheel

$\mathbf{d}_{a1,2}$ Tip diameter of pinion, wheel

$\mathbf{d}_{f}$ Root diameter in mm, defined as the circle passing through the bottom of the gaps between the teeth

$\mathbf{d}_{w}$ Working diameter in mm, the height that defines the distance that the conjugate tooth enters into the of the tooth’s backlash.

h tooth depth h, in mm, measured between the root circle and of the top circle

x addendum modification coefficient

$\mathbf{x}_{1,2}$ addendum modification coefficient of pinion, wheel

z Number of teeth

$\mathbf{z}_{n}$ Virtual number of teeth of a helical gear or the equivalent number of teeth: the virtual amount or equivalent of teeth is the amount of teeth that would have a spur gear equivalent to a worm gear. This virtual quantity is used in determining the resistance of the teeth of a helical gear

m Module m represents the size of the tooth and is the ratio between the diameter of the primitive circle e mm and number of teeth of a gear. When two gears are coupled, their modules are to be equal.

$\mathbf{m}_{n}$ Normal module, used in the helical gears, where: $\mathbf{m_n = m*\cos\beta}$ where $\boldsymbol{\beta}$ is the angle of inclination of the propeller.

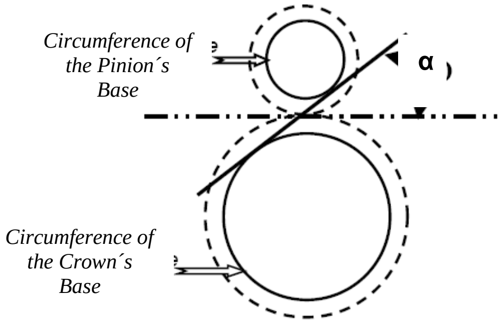
$\mathbf{m}_{t}$ Transverse module

$\mathbf{T}_{1,2}$ torque in way of pinion, wheel

$\boldsymbol{\alpha}$ Pressure angle of the cylinder of reference, in degrees, that is defined as the angle that the path of contact interacts with the common tangent to the primitive circles at the main point.

Line of action: is the locus of points of contact of the teeth during the gearing. The usual pressure angles are 14 ½ ° and 20 °, the latter being the most common value.

FIGURE H1.301.1 ANGLE OF NORMAL PRESSURE FOR HELICAL GEARS



α_n normal pressure angle at reference cylinder for helical gears, mesured on the perpendicular direction to the teeth, it is different from the action angle, which is measured in the direction of rotation

FIGURA H1.301.2 – NORMAL PRESSURE ANGLE

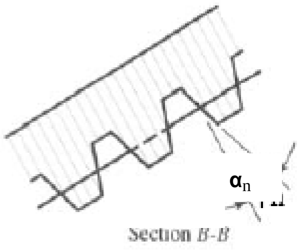


FIGURE H1.301.3 – POINT OF CONTACT

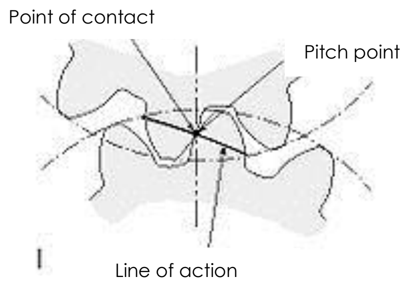
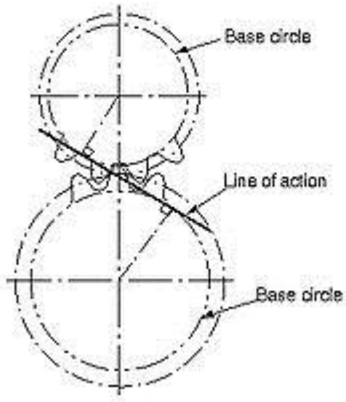


FIGURE H1.301.4 –BASE CIRCLE



g_α length of the path of contact (line of action).

α_t Transverse pressure angle, in degrees, in the cylinder of reference.

α_{tw} transverse pressure angle at working pitch cylinder.

β Angle or inclination of the propeller, in degrees, in the cylinder of reference

β_b Helix angle, in degrees, at base cylinder.

v linear speed at pitch diameter, in m/s.

p Circular pitch or simply pitch of a spur gear is the distance taken, on an arc, on the pitch circle between of two corresponding points of consecutive teeth

p_d Pitch diameter valid for the English system measures is the ratio of the number of teeth and the pitch circle diameter, the diameter measured in inches.

FIGURE H1.301.5 – PITCH NOMENCLATURE

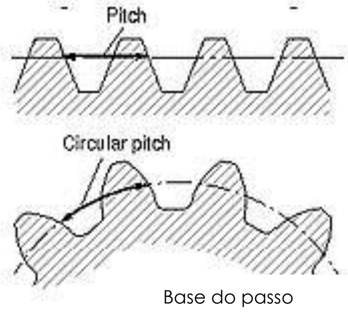
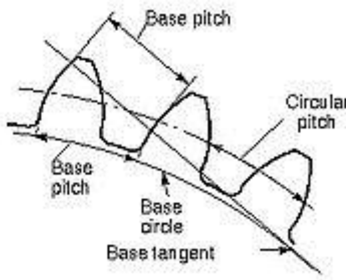


FIGURE H1.301.6 – PITCH NOMENCLATURE



p_n Normal pitch: the spur gears have only one circular and one diametral pitch. Helical gears have the normal circular pitch p_n which is the distance between corresponding points of two adjacent teeth measured in the plane B-B that is perpendicular to the helical, and the cross pitch **p_t** defined below.

p_t The helical gears are provided with the circular transverse pitch p_t which is the distance between corresponding points of two adjacent teeth measured in the plane A-A that is perpendicular to the shaft.

FIGURE F.H1.301.7 – ANGLE OF ACTION

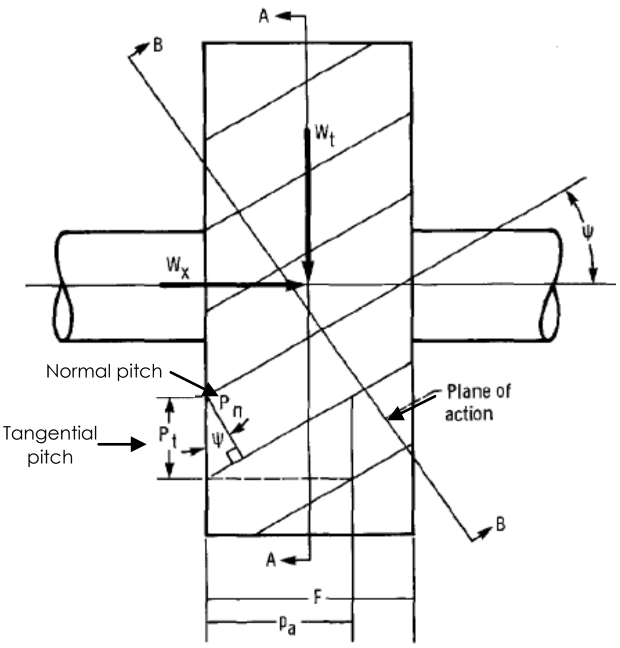
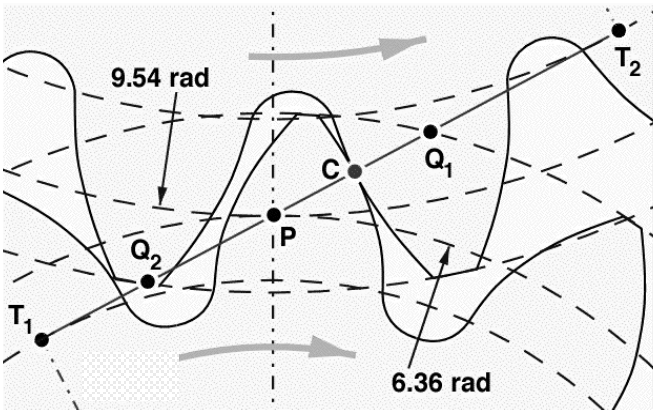


FIGURE F.H1.301.8 - LINE OF ACTION



Path of contact: the tooth of the pinion touches a tooth of the crown in the ***path of contact*** C ("knot") which moves itself along the ***path of contact*** and along the faces of the teeth while the rotation proceeds. Since the contact can not be made outside the teeth, the contact has to take place along the path of contact only between the points Q2 and Q1 of it and within the two circles of addendum. The segment Q2-Q1 is named ***path of contact,*** being, therefore the segment of the path of contact determined by the interception of the circle of addendum of the pinion with the circle of addendum of the crown.

Thus, the line of action pictured above is the locus of points of contact of the teeth during the gearing. Notes:

a. The contact point moves along the path of contact;

b. The path of contact is limited by the two addenda.

c. The perpendicular between the path of contact and the tooth flanks in the point of contact;

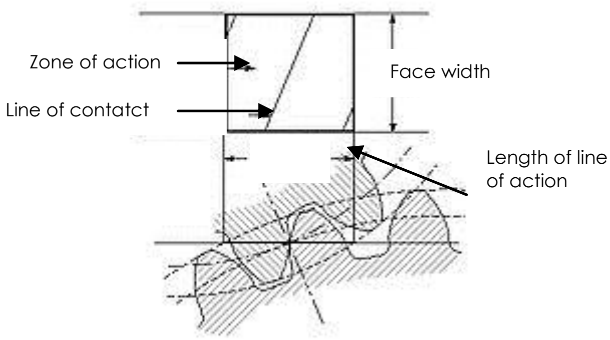
d. There is a relative sliding between the teeth, especially at the beginning and end of the contact

e. Ensuring of the backlash at the tip of the tooth is given due to the fact that the dedendum exceeds the addendum;

f. The difference between ttooth thickness and tooth space as measured along the pitcy circle is called ***backlash***

Interference: The portion of the sspur gear below the base circle is sometimes cut as a straight radial line and not as an involute cure. Hence, if contact should occur below the base circle, nonconjugate action (interference) will occur.

FIGURE F.H1.301.9 – FACE CPMTATCT



ε Contact ratio is generally the number of angular pitches through which the face of one tooth rotates from beginning to end of the contact

$\boldsymbol{\varepsilon_\alpha}$ Transverse contact ratio, being defined as the ratio between the angle of action and the angular pitch. Angle of action is the angle that the gear moves while a certain pair of teeth keeps geared, i.e., from the first to the last point of contact.

$\boldsymbol{\varepsilon_\beta}$ Overlap ratio is the ratio of face contact in an axial plane, or the ratio of the width of the face with the axial pitch. For bevel and hypoid gears it is the ratio of advancing of the face in relation to the circular pitch.

$\boldsymbol{\varepsilon_\gamma}$ Total contact ratio is the sum of the ratio of transver and overlap ratios.

$\mathbf{F_t}$ Nominal transverse tangential load at reference cylinder per mesh

$\mathbf{F_{bt}}$ nominal tangential load on base cylinder in the transverse section

Bending stress of the tooth, N/mm^2

$\sigma_{F\,lim}$ Nominal plain-bar stress for bin “i”, N/mm^2

σ_{FP} Permissible tooth root stress, N/mm^2

σ_{H} Contact stress (Herziana), N/mm^2

$\sigma_{H\,lim}$ **C**ontact limit stress (Herziana), N/mm^2

σ_{HP} Allowable stress number (contact) (Herziana), N/mm^2

R_z Mean peak to valey roughness, in µm

Q Quality ISO 1328 – 1975

HV Vickers Hardness

303. **Geometrical definitions for cylindrical gears**

a. For internal gearing z2, a, d2, da2, db2 and dw2 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 = z2 / z1

= dw2 / dw1

= d2 / d1

For external gears u is positive, for internal gears u is negative.

b. Length of the line of action:

b.1. For external gears, in mm:

$$g_\alpha = 0{,}5 * \sqrt{(\mathbf{d}_{a1}^2 - \mathbf{d}_{b1}^2)} + 0{,}5 * \sqrt{(\mathbf{d}_{a1}^2 - \mathbf{d}_{b1}^2)} - a \sin \alpha_{tw}$$

b.2. For internal gears, in mm:

$$g_\alpha = 0{,}5 * \sqrt{(\mathbf{d}_{a1}^2 - \mathbf{d}_{b1}^2)} - 0{,}5 * \sqrt{(\mathbf{d}_{a1}^2 - \mathbf{d}_{b1}^2)} - a \sin \alpha_{tw}$$

b.3. Transverse contact ratio:

$$\varepsilon_\alpha = \frac{g_\alpha}{\pi * m_n * \cos_{\alpha t} / \cos \beta}$$

b.4. Contact ratio of the face:

$$\varepsilon_\beta = \frac{b * \sin \beta}{\pi * m_n}$$

b.5. Total contact ratio:

$$\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$$

b.6. Helical angle of the cylinder base:

$$\tan \beta_b = \frac{\tan_{\alpha n}}{\cos \beta}$$

b.7. Diameter of reference:

$$d = \frac{z * m_n}{\cos \beta}$$

b.8. Diameter of base:

$$d_b = d \cos \alpha_t = d_w \cos \alpha_{tw}$$

b.9. Distance between centers:

$$a = 0{,}5 * (dw_1 - dw_2)$$

b.10. Virtual number of teeth:

$$z_n = \frac{z}{\cos^2 \beta_b * \cos \beta}$$

b.11. Transverse module:

$$m_t = \frac{m_n}{\cos \beta}$$

$$\text{inv } \alpha = \frac{\tan \alpha - \pi * \alpha}{180} \quad \text{(ângulos em °)}$$

b.12. Linear speed at pitch diameter:

$$v = d_{1,2} * n_{1,2} / 19099$$

304. **Specific nomenclature for bevel gears:**

a. **Primitive Cone:** the teeth of a bevel gear are made related to a primitive cone, instead of a primitive cylinder. The elements of the primitive cone cone intersect at the **vertex of the cone**

b. **Cone of the tops or external:** the cone made of the tops of the teeth.

c. **Cone of the root or internal:** cone made of the roots of the teeth.

d. **Angle of the primitive cone:** angle between the geratrix of the primitive cone and the shaft.

The conversion from a bevel gear to a cylindrical equivalent (virtual) is based on the middle section of the bevel gear.

The index *v* refers to the virtual cylindrical gear (equivalent).

The index *m* refers to the middle section of the bevel gear.

δ_1, δ_2 = pitch angle of the pinion (crown)

δa_1, δa_2 = face angle of the pinion (crown)

Σ = angles between the shafts

β_m = Average spiral angle

de_1, de_2 = external diameter of pitch of the pinion (crown)

dm_1, dm_2 = average diameter of pitch of the pinion (crown)

dv_1, dv_2 = virtual reference diameter of cylindrical gear of the pinion (crown)

$R_{e1,2}$ = external distance of the cone of the pinion (crown)

R_m= average distance of the cone of the pinion (crown)

305**. Bevel gears conversion formulae and specific formulae**

Note: determining the proportions of bevel gears is done by converting them to equivalent cylindrical gears and are then using the formulas for spur gears

Number virtual of teeth of a bevel gear:

$z_{v1} = z_1 / \cos \delta_1$
$z_{v2} = z_2 / \cos \delta_2$

For $\Sigma = 90°$

$$z_{v1} = \frac{\sqrt{u^2 + 1}}{u}$$

$$z_{v2} = z_2 \sqrt{u^2 + 1}$$

Gear ratio of the virtual cylindrical gear:
$u_v = z_{v2} / z_{v1}$

For $\Sigma = 90°$
$u_v = u^2$

Geometrical definitions:

$\delta_1 + \delta_2 = \Sigma$

$\tan \alpha_{vt} = (\tan \alpha_n) / (\cos \beta_m)$

$\tan \beta_{bm} = (\tan \beta_m) / (\cos \alpha_{vt})$

$\beta_{vb} = \arcsin (\text{sen}\ \beta_m * \cos \alpha_n)$

$R_e = d_{e1,2} / (2 * \sin \delta_{1,2})$

$R_m = R_e – (b/2)$, para $b \leq R/3$

Diameter of reference of the pinion, crown, referring to the medium section of the bevel gear:

$d_{m1} = d_{e1} – b * \sin \delta_1$
$d_{m2} = d_{e2} – b * \sin \delta_2$

Modules:
External transversal module:
$m_{et} = d_{e2} / z_2 = de_1 / z_1$

External normal module:

$m_{na} = m_t * \cos \beta_m$

Medium normal module:
$m_{mn} = m_{mt} * \cos \beta_m$
$m_{mn} = m_{et} * (R_m/R_e) * \cos \beta_m$

Reference diameter of the virtual cylindrical gear:

$d_{v1} = d_{m1} / \cos_{\delta 1}$
$d_{v2} = d_{m2} / \cos_{\delta 2}$

Virtual diameter of the base of the pinion (crown):

$d_{vb1} = d_{v1} * \cos_{\alpha vt}$
$d_{vb2} = d_{v2} * \cos_{\alpha vt}$
$d_{vb2} = d_{v2} * \cos_{\alpha vt}$

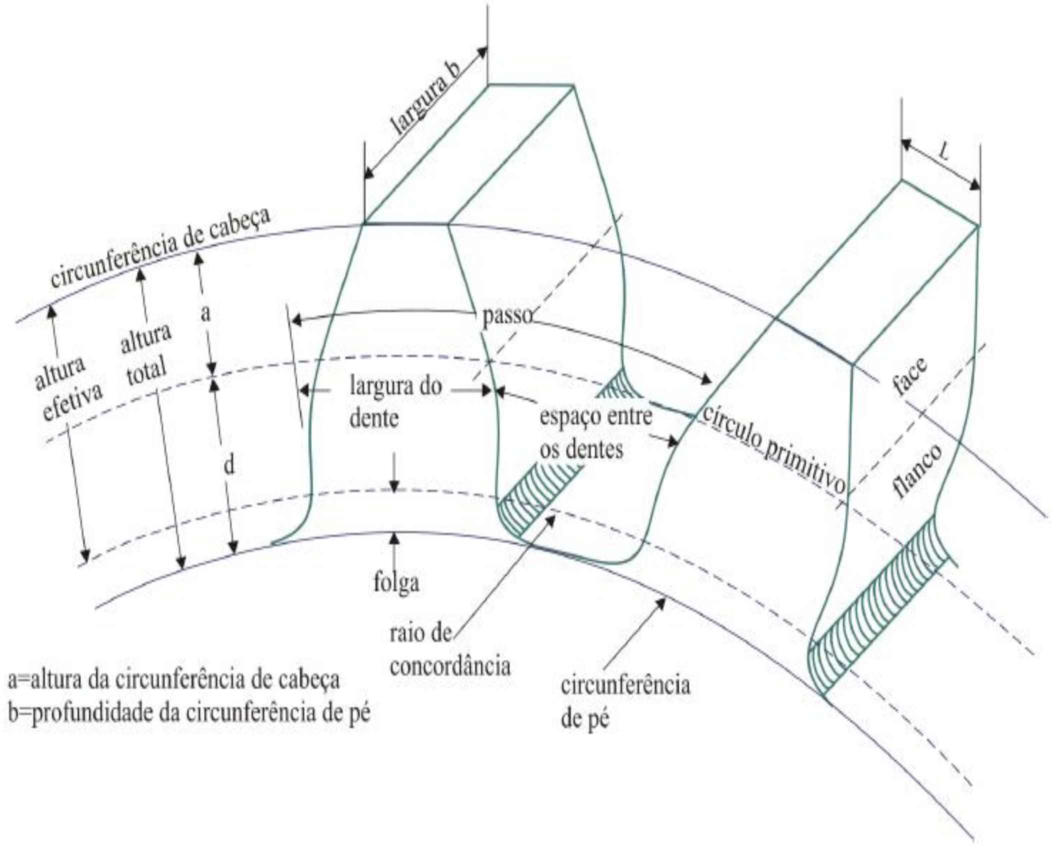
Where α_{vt} is the transverse pressure angle of the virtual cylindrical gear given by:

$\alpha_{vt} = \arccos [(d_{vb1} + d_{vb2})/2 * \alpha_v]$, α_{vt} in degrees

and a_v is the distance of Center of the virtual gear:

$\alpha_v = 0{,}5 * (d_{v1} + d_{v1})$

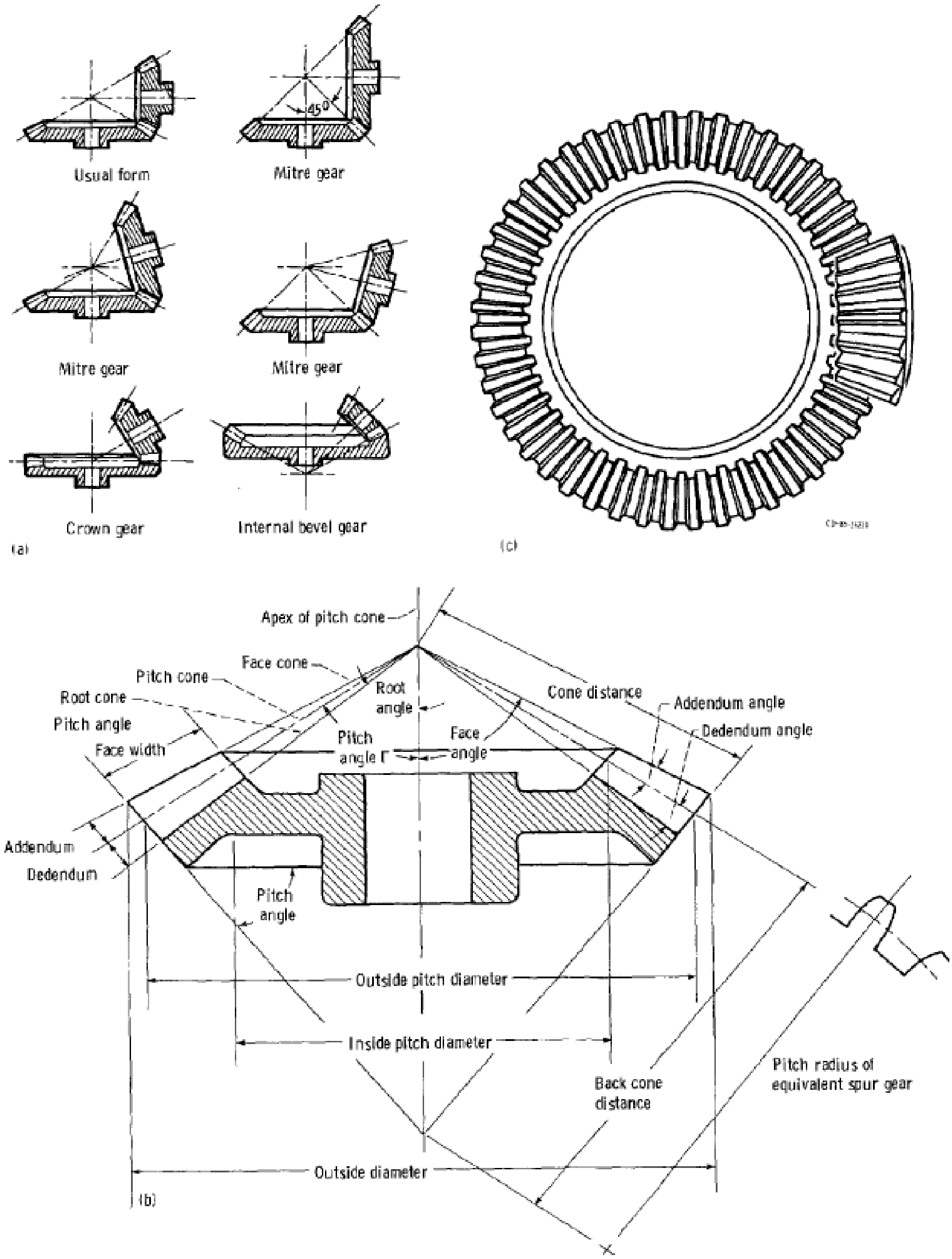
FIGURE F.H1.300.1 – BASIC NOMENCLATURE OF SPUR GEARS.



PORTUGUESE	ENGLISH
a= altura da circunferencia de cabeça	Height of the head circumference
altura efetiva	Effective height
altura total	Total height
círculo primitivo	Primitive circle
circunferência da cabeça	Head circumference
circunferência de pé	Foot circumference
d = profundidade de conferencia de pé	d = depth of the foot circumference
espaço entre dentes	Gap between teeth
face	Face
flanco	Flank
folga	Gap
largura b	b Width
largura do dente	Width of the tooth
passo	Pitch
raio de concordancia	Radius of concordance

FIGURE F.H1.303.1 – BASIC NOMENCLATURE OF BEVEL GEARS

See table T.H1.301.1 – Basic nomenclature of bevel gears

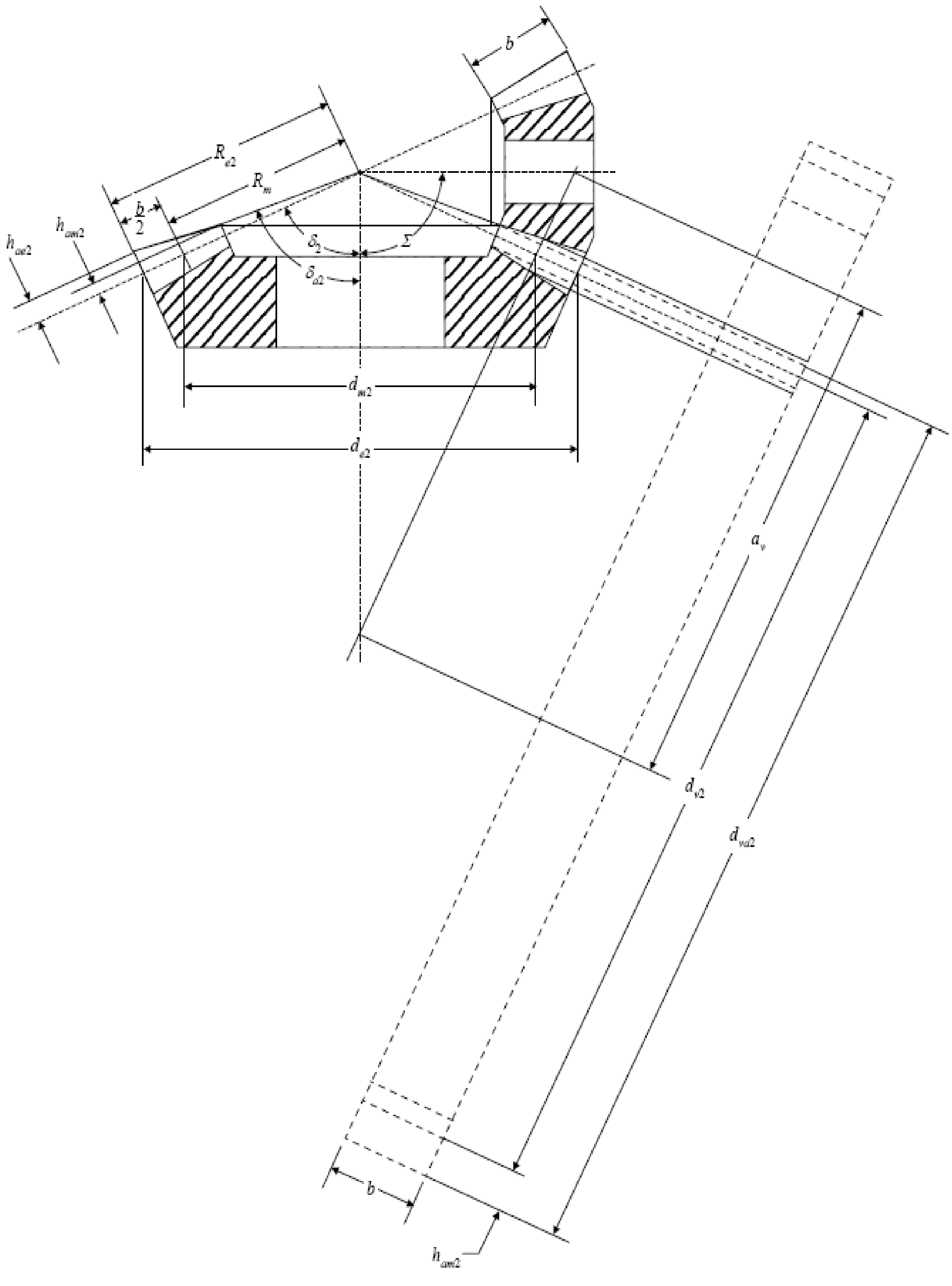


(a) Gear arrangements.
(b) Gear terminology.
(c) Gear pair.

TABLE T.H1.303.1 – BASIC NOMENCLATURE OF BEVEL GEARS REFERRING TO FIG. F.H1.303.1

1. Forma usual	**1. Usual form**
2. Engrenagem de 45°	**2. Mitre gear**
3. Engrenagem de cruz	**3. Crown gear**
4. Engrenagem cônica interna	**4. Internal bevel gear**
5. Vértice do cone de passo	**5. Apex of pitch cone**
6. Cone facial	**6. Face cone**
7. Cone de passo	**7. Pitch cone**
8. Cone da raiz	**8. Root cone**
9. Ângulo de passo	**9. Pitch angle**
10. Contato de face	**10. Face width**
11. Adendum	**11. Addendum**
12. Dedendum	**12. Dedendum**
13. Ângulo do passo	**13. Pitch angle**
14. Diâmetro do passo interno	**14. Outside pitch diameter**
15. Diâmetro do passo externo	**15. Inside pitch diameter**
16. Diâmetro externo	**16. Outside diameter**
17. Distância do cone de trás	**17. Back cone distance**
18. Raio do passo da engrenagem reta equivalente	**18. Pitch redius of equivalent spur gear**
19. Ângulo dedendum	**19. Dedendum angle**
20. Ângulo Addendum	**20. Addendum angle**
21. Distância do cone	**21. Cone distance**
22. Ângulo facial	**22. Face angle**
23. Ângulo de passo	**23. Pitch angle**
24. Ângulo da raiz	**24. Root angle**
a. Organização das engrenagens	**a. Gear arrangements**
b. Organização da engrenagens	**b. Gear terminology**
c. Engrenagem em par	**c. Gear pair**

FIGURE F.H1.303.2 – CONVERSION OF BEVEL GEAR TO CYLINDRICAL GEAR EQUIVALENT



306. **Subscripts to symbols:**

A — Application

a — Addendum
Tooth tip

α — Transverse contact
Profile

b — Base circle
Face width

β — Helix
Face width
Crowning

C — Pitch point

cal — Calculated parameter

co — Contact pattern

γ — Total (total value)

D — Speed transformation
Reducing or increasing

dyn — Dynamic

Δ — Rogh specimen

E — Elasticity of material
Resonance

e — Outer limit of single pair tooth contact

eff — Effective value, real stress

t — Transverse plane

0 — Tool
Basic value

ε — Contact ratio

F — Tooth root stress

f — Tooth roote, dedendum

G — Geometry

H — Hertzian stress (contact stress)

i — internal
bin number

k — values related to the notched test piece

L — Lubrication

lim — Value of reference strength

M — Mean stress influence

ma — manufacturing

max — maximum value

min — minimum value

N

n — Normal plane
Virtual spur gear of a helical gear
Number of revolutions

oil — Oil

P — Permissible value
Rack profile

p — Pitch
Values related to the smooth polished test piece

pla — Planet gear

R — Roughness

r — Radial

red — Reduced

rel — Relative

s — Tooth thickness
notch effect

sh — Shaft

stat — Static (load)

sun — Sun pinion, sun gear

T — Test gear

t — Transverse plane

th — Theoretical

v — Velocity
Losses

W — Pairing of materials

w — Working Ithis subscript may replace the prime)

X — Dimension (absolute)

y — Running-in
Any point on the tooth flank

z — Sun

0 — Basic value

1 Referring to gear pinion

2 Referring to the wheel (crown)

Note: as $n_{1,2}$ means: rotation of the pinion (1) and crown (2).

γ Total value

307. **Table of symbols**

TABLE T.H1.307.1 – SYMBOLS USED IN THIS CHAPTER H

Symbols	Description	Unit
A,B,C,D,E	Points in path of contact (pinion root to pinion tip, independent of the driver, for geometrical considerations only)	
a	Centre distance	mm
α	Pressure angle of (without subscript, at the reference cylinder)	degree
B	Total face width of double helix gear including gap width	mm
b	Face width	mm
B	Helix angle (without subscript, at reference cylinder)	degree
C	constant, coefficient	
	Relief of the tooth flank	µm
c	Constant	
γ	Auxiliary angle	degree
D	Design diameter	mm
d	Diameter (without subscript, reference diameter)	mm
δ	Deflection	µm
E	Modulus of elasticity	N/mm^2
Eh	Material designation of case-hardened wrought steel	
Eht	Case detpth, see ISO 6336-5	mm
e	Auxiliary quantity	
ε	Contact ratio, overlap ratio, relative eccentricity	
ς	Roll angle	degree
F	Composite and cumulative deviations	µm
	Force or load	N
f	Deviation, tooth deformation	µm
G	Shear modulus	N/mm^2
GG	Material designation for gray cast iron	
GGG	Material designation for nodular cast iron (pearlitic, bainitic ferrite structure)	
GTS	Material designation for black malleable iron (pearlitic structure)	
g	Path of contact	mm
θ	Temperature	°C
HB	Brinel hardness l	
HRC	Rockwell hardness (C scale)	
HR 30N	Rockwell hardness (30 N scale) see ISO 6336-5	
HV	Vickers hardness	
HV 1	Vickers hardness at the load *F* = 9,.81 N – see ISO 6336-5	
HV 10	Vickers hardness at the load *F* = 98,10 N - see ISO 6336-5	
h	Tooth depth (without subscript, root circle to tip circle)	mm
H	Effective dynamic viscosity of oil wedge at the mean temperature of wedge	m Pas
IF	Material designation for flame or induction hardened wrought special steel	
i	Transmision ratio	
	Bin	
J	Jominy hardenability - see ISO 6336-5	
K	Constant, factors concerning tooth load	
L	Design lengths	mm
l	Bearing span	mm

Symbols	Description	Unit
Γ	Parameter on the line of action	
M	Moment of a force	Nm
	Mean stress ratio	
ME	Symbols identifying material and heat treatment requirements (see ISO 6336-5).	
MQ		
ML		
m	Module	mm
	Mass	kg
μ	Coefficient of friction	
N	Number, exponent, resonance ratio	
NT	Material designation for nitirided wrought steel, nitriding steel	
NV	Material designation for through-hardened wrought steel, nitride, nitrocarburized	
n	Rotational speed	rpm
	Number of load cycles	1/min
ν	Poisson's ratio	
	Kinematic viscosity of the oil	mm^2/s
P	Transmitted power	kW
p	Pitch	mm
	Number of planet gears	
	Slope of the Woehler damage line	
q	Aauxiliary factor	
	Flexibility of pair of meshing teeth (mm*μm)/N	
	Material allowance for finish machining - see ISO 6336-3	
r	radius (without subscript, reference radius)	
ρ	radius of curvature	
	density (for steel, $\rho = 7{,}83 \times 10^{-6)}$	$kgmm^3$
S_t	Material designation for normalized base steel (σ_B<800 N/mm^2)	
S	Safety factor	
s	Tooth thickness, distance between mid-plane of pinion and the middle of the bearing span	mm
Σ	Angle between shafts of bevel gears	
σ	Normal stress	
R	Distance from the cone	
T	Torque	Nm
	Tolerance	μm
τ	Shear stress	N/mm^2
	Angular pitch	mm
u	Gear ratio $(z_2/z_1) \geq 1^a$	
V	Material designation for through-hardened wrought special steel, alloy or carbon (σ_B<800 N/mm^2)	
v	Tangential speed (without subscript, at reference circle = tangential sped at pitch circle)	m/s
w	Specific load (per unit of facewidth F_t/b)	N/mm
ψ	auxiliary angle	degree
x	Profile shift coeficient	
χ	factor of operation (running-in)	
Y	factor related to tooth root stress (Lewis factor)	
y	Running-in allowance (with subscript α or β only)	μm
Z	factor related to contact stress	
z	Number of teeth	
ω	Angular speed	rad/s

Symbol	Description	Unit
α_{en}	Form-factor pressure angle, pressure angle at the outer point of single pair tooth contact of virtual spur gears	degree
α_n	Normal pressure angle	degree
α_t	Transverse pressure angle	degree
α_{wt}	Pressure angle at the pitch cylinder	degree
α_{Fen}	Load direction angle, relevant to direction of application of load at the outer point of single pair tooth contact of virtual spur gears	degree
α_{Pn}	Normal pressure angle of the basic rack for cylindrical gears	degree
B^*	Constant	
b_{cal}	Calculated facewidth	mm
b_{c0}	Length of tooth bearing pattern at no load (contact marking)	
b_{eH}	Effective facewidth	mm
b_{red}	Reduced facewidth (facewidth minus end relief)	mm
b_s	Web thickness	mm
b_B	Facewidth of onea helix on a double helix gear	mm
$b_{I(II)}$	Length of tendrelief	mm
β_b	Base helix angle	degree
β_e	Form-factro helix angle, helix angle at the outer point of single tooth contact	degree
β_m	Mean angle of the spiral	degree
C_a	Top relief	µm
C_{ay}	Top relief by running-in	µm
C_B	Basic rack factor (same rack for pinion and wheel)	
C_{B1}	Basic rack factor (pinion)	
C_{B2}	Basic rack (wheel)	
C_M	Correction factor	
C_R	Gear blank factor	
$C_{ZL,ZR,ZV}$	Factors for determining the lubricant film factors	
C_β	Crowning height	µm
$C_{I(II)}$	Top relief	µm
c_γ	Mean value of mesh stiffness per unit facewidth	N/(mm*µm)
$c_{\gamma\alpha}$	Mean value of mesh stiffness per unit facewidth (used for K_v, $KH_{h\alpha}$ and $K_{F\alpha}$)	N/(mm*µm)
$c_{\gamma\beta}$	Mean value of the stiffness of the gearing per unity of facewidth used for $K_{H\beta}$ and $K_{F\beta}$)	N/(mm*µm)
c'	Maximum tooth stiffness per unit of facewidth (simple stiffness) of a tooth pair	
c'_{th}	Theoretical single stiffess	N/(mm*µm)
D_{be}	Bearing bore diameter (plain bearing)	mm
D_{sh}	Journal diameter (plain bearings)	mm
$d_{a1,2}$	Tip diameter (pinion, wheel)	mm
d_{be}	Base diameter	mm
d_e	diameter of circle passing through the outer point of a single pair tooth in contact	mm
$d_{f1,2}$	diameter of root of pinion (crown)	mm
d_{Nf}	diameter of root form	mm
d_{sh}	outer shaft diameter, nominal for deflection	mm
d_{shi}	inner diameter of a hollow shaft	mm

Symbol	Description	Unit
d_{soi}	diameter in the onset of evolute	mm
d_w	Diameter pitch	mm
$d_{1,2}$	diameter of reference of pinion (crown)	mm
$\delta_{1,2}$	Angle of cone of reference of pinion (crown)	degree
$\delta_{a1,a2}$	Angle of pinion top (crown)	degree
$\delta_{b\ th}$	Combined deflection of geared teeth assuming uniformly distributed load along facewidth	µm
δ_q	Difference of thickness in the measurement by the probe of misalignment of gearing	µm
δ_S	Elogation of fracture	%
ε_α	Cross contact ratio	
$\varepsilon_{\alpha n}$	ratio of virtual contact, contact ratio	
	Cross virtual spur gear	
ε_β	Overlapping ratio	
ε_γ	Total contact ratio $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$	
ε_1	Addendum contact ratio of the pinion ε_1 = CE / p_{bt}	
ε_2	contact ratio of the addendum of the crown ε_2 = AC / p_{bt}	
ζ_{aw}	roll angle of the point of pitch of working up to the top diameter	degree
ζ_{fw}	roll angle of the diameter of the root for the point of the pitches' run	degree
$F_{be\ r}$	Radial force in the bearing	N
F_{bn}	Rated load, normal to the line of contact	N
F_{bt}	Cross rated load in the plane of action (base tangent plane)	N
F_m	Mean cross tangential load in the circle of reference	N
	Relevant to gearing calculations, $F_m = F_t*K_A*K_v$	
$F_{m\ T}$	Partial cross tangential load in the circle of reference	N
F_{max}	Maximum tangential load in the tooth for the calculated gearing	N
F_t	Rated cross tangential load in the cylinder of reference per gearing	N
F_{th}	Tangential load determinant in a transverse plane for $K_{H\alpha}$ and $K_{F\alpha}$ $Fth = F_t*K_A*K_v*K_{H\beta}$	N
F_α	Total deviation of profile	µm
F_β	Total deviation of helix	µm
$F_{\beta 6}$	Tolerance in the total deviation of helix for ISO grade 6	µm
$F_{\beta x}$	Equivalent initial misalignment (after the running-in)	µm
$F_{\beta x\ cv}$	Equivalent initial misalignment for determining the crown height (estimated)	µm
$F_{\beta x\ T}$	Equivalent misalignment measured submitted to partial load	µm
$F_{\beta y}$	Equivalent effective misalignment (after the running-in)	µm
f_{be}	Component of the equivalent misalignment due to bearing deformation	µm
f_{ca}	Component of equivalent misalignment due to block deformation	
$f_{f\alpha}$	Deviation of the profile form	µm
F_m	Misalignment of the gearing due to manufacture deviations	µm
f_{pt}	Cross deviation of single pitch	µm
f_{par}	Non-parallelism of pinion and crown shafts (manufacture)	µm
f_{pb}	Cross deviation of base pitch	µm
f_{sh}	Component of misalignment due to deformations of pinion and crown shafts	µm
f_{shT}	Component of misalignment due to deformations of shaft and pinion measured in partial load	µm
f_{sh0}	Deformation of the shaft submitted to specific load	µm*mm/N

Symbol	Description	Unit
$f_{H\beta}$	Deviation of the inclination of the helix	µm
$f_{H\beta 6}$	Tolerance of the deviation of helix for ISO grade 6	µm
g_α	Length of the line of action	mm
h_{aP}	Adendum of basic rack of cylindrical gears	mm
$h_{01,2}$	Adendum of pinion tool (crown)	mm
h_{fP}	Dedendum of basic rack of cylindrical gears	mm
h_{f2}	Dedendum of tooth of inner gear	mm
h_{min}	Minimum thickness of the lubricant film	mm
h_{Fe}	Arm of bending moment for stress of tooth root	mm
	Relevant to application of load in the outer point of one single pair of teeth in contact	
h_t	Height of the head of the tooth	mm
J^*	Moment of inertia of facewidth	$kg*mm^2$/ mm
K'	Constant of forging of pinion	
K_v	Dynamic factor	
K_A	Application factor	
$K_{F\alpha}$	Cross load factor of (root stress)	
$K_{F\beta}$	Face load fator of (root stress)	
$K_{H\alpha}$	Transverse load factor (contact stress)	
$K_{H\beta}$	Face load factor (contact stress)	
K_γ	Mesh load factor (takes into account the uneven distribution of the load between meshes for effective length of roller (roller bearings)	
l_a	Effective length of bearing (roller bearing)	mm
l_b	see g_α	
m^*	Individual relative mass of gear per unit of face referred to the line of action	kg/mm
m_n	Normal module	
m_{red}	Reduced mass gear pair mass per face unit face width referred to the line of action	kg/mm
m_t	Transverse module	
N_F	Exponent	
N_i	Number of cycles up to the failure of the bin i	mm
N_L	Number of cycles of load	
N_S	ratio of resonance in the main band of resonance	
$n_{1,2}$	Rotational speed of the pinion (crown)	
n_i	Number of cycles for bin i	
n_E	Ressonance speed	min^{-1} / s^{-1}
p_{bn}	Normal base pitch	
p_{bt}	Transverse base pitch	min^{-1}
p_d	Outer diametral pitch	mm
p_{r01}, p_{r02}	Protuberance of the tool for pinion, crown	mm
q'	Minimum value for the flexibility of a pair of meshing teeth	mm
q_{pr}	Protuberance of the tool - see ISO 6336-3	mm
q_s	Notch parameter $q_s = s_{Fn} / 2\ \rho_F$	(mm*m) / N
q_{sk}	Notch parameter of notched test specimen	
q_{sT}	Notch parameter of the standard reference test gear q_{sT} = 2,5	
q_α	Auxiliary factor	
R_a	Arithmetic mean roughness value, R_a = 1/6 R_z	µm

Symbol	Description	Unit
R_m	Mean distance of the cone	
R_z	Mean peak-to-valley roughness (ISO 4287 and ISO 4288)	μm
R_{zk}	Mean peak-to-valley roughness of the notched, rough test specimen	μm
R_{zT}	Mean peak-to-valley roughness o the standard reference test gear, RzT = 10	μm
r_b	Base radius	
ρ_{fP}	Root fillet radius of the of the basic rack for cylindrical gears	mm
ρ_g	Radius of grinding notch	
ρ_{red}	Radius of relative curvature	
ρ_C	Radius of the relative curvature at the pitch surface	mm
ρF	Tooth root radius of the at the critical section	mm
ρ'	Slip layer thickness	mm
S_F	Safety factor for tooth breakage	mm
S_H	Safety factor for pitting	mm
s_c	Film thickness of marking compound used in contact pattern determination	μm
s_{pr}	Residual fillet undercut, $s_pr = q_{pr} - q$	
s_{Fn}	Tooth root chord in the critical section	
s_R	Rim thickness	mm
$\sigma_{k\ lim}$	Nominal notched-bar stress number (bending)	N / mm^2
$\sigma_{p\ lim}$	Nominal plain-bar stress number (bending)	mm
σ_B	Tensile strength	mm
σ_F	Tooth root stress	
σ_{Fi}	Tooth root stress for bin i	
$\sigma_{F\ lim}$	Normal stress number (bending)	N / mm^2
σ_{FE}	Allowable stress number (bending) $\sigma_{FE} = \sigma_{F\ lim}\ Y_{ST}$	N / mm^2
σ_{FG}	Tooth root stress limit	N / mm^2
σ_{FP}	Permissible tooth root stress	N / mm^2
σ_{F0}	Nominal tooth root stress	
σ_H	Contact stress	N / mm^2
σ_{Hi}	Contact stress for bin i	N / mm^2
$\sigma_{H\ lim}$	Allowable stress number (contact)	N / mm^2
σ_{HG}	Pitting stress limit	N / mm^2
σ_{HP}	Allowable contact stress	N / mm^2
σ_{H0}	Rated contact stress	N / mm^2
σ_S	Yield stress	N / mm^2
σ_i	Stress for bin i	N / mm^2
$\sigma_{0,2}$	Test stress (0,2% permanent set)	N / mm^2
$T_{1,2}$	Nominal torque at the pinion (or wheel)	N / mm^2
T_{eq}	Equivalent torque	N / mm^2
T_i	Torque for the bin i	N / mm^2
T_n	Nominal torque	Nm
t_g	Maximum depth of grinding notch	Nm
U	Sum of the individual damage parts	Nm

Symbol	Description	Unit
w_m	Mean specific load (per unit face width)	N / mm
w_t	Tangential load per unit tooth width, including overload factors	N / mm
$x_{1,2}$	Profile shift coefficient of pinion (or wheel)	
χ^*	Relative stress gradient in the root of a crown	mm^{-1}
χ_β	Factor characterizing equivalent misalignment after running-in	
χ^*_p	Relative stress gradient in a smooth polished test piece	mm^{-1}
Y_{DT}	Deep tooth factor	
Y_F	Tooth form factor for the influence on nominal tooth root stress with load applied at the outer point of a single pair tooth contact	
Y_M	Mean stress influence factor	
Y_{Nk}	Life fator for tooth root stress, relevant to the notched test piece	
Y_{NT}	Life factor for tooth root stress, relevant to theplian polished test piece	
Y_R	Life factor for tooth root stress for reference test conditions	
$Y_{R\ rel\ k}$	Relative roughness factor, the quotient of the gear tooth root surface factor of interest divided by the notch test piece factor, $Y_{R\ rel\ k} = Y_R / Y_{Rk}$	
Y_S	Stress correction factor, for the conversion of the nominal tooth root stress, determined for application of load at the outer point of single pair tooth contact, to the local tooth root stress	
Y_{Sg}	Stress correction factors for teeth with grinding notches	
Y_{Sk}	Correction factor relevant to the nothced test piece	
Y_{ST}	Stress correction factor, relevant to the dimensions of the reference test gears	
Y_X	Size factor (tooth root)	
Y_β	helix angle factor (tooth root)	
Y_δ	Notch sensitivity factor of the actual gear (relative to a polished test piece)	
$Y_{\delta k}$	Sensitivity factor of a notched specimen (relative to a polished test piece)	
$Y_{\delta T}$	Sensitivity factor of the standard reference test gear, relative to the smooth polished test piece	
$Y_{\delta\ rel\ k}$	Relative notch sinesitivity fator, the quotient of the gear notch sensitivity fator of interest divided by the notched test piece fator, $Y_{\delta\ rel\ k} = Y_\delta / Y_{\delta k}$.	
$Y_{\delta\ rel\ T}$	Relative notch sensitivity factor, the quotient of the gear notch sensitivity factor of interest divieded by the standard test gear factor, $Y_{\delta\ rel\ T} = Y_\delta / Y_{\delta ?T}$.	
y_α	Running-in allowance for a gear pair	Mm
y_β	Running-in allowance (equivalent misalignment)	µm
Z_v	Velocity factor	
Z_B, Z_d	Single pair tooth contact factors for pinion (wheel)	
Z_E	Elasticity factor	
Z_H	Zone factor	
Z_L	Lubrication factor	
Z_N	Life factor for contact stress	
Z_{NT}	Life factor for contact stress for reference test conditions	
Z_R	Roughness factor affecting surface durability	
Z_W	Work hardening factor	$\sqrt{N/mm^2}$

Symbol	Description	Unit
Z_X	Size factor (pitting)	
Z_β	Helix angle factor (pitting)	
Z_ε	factor of the ratio of the contact (pite)	
z_n	Virtual number of teeth in a helical gear	
$z_{1,2}$	Number of teeth of pinion (or crown)	
$\omega_{1,2}$	Angular velocity of pinion (or crown)	

H2. DOCUMENTS

100. Documents for approval

101. Assembly and sectional drawings together eith the necessary detail drawings and parts list are to be submitted to RBNA in three hard copies or in virtual files for approval. They shall contain all the data required to enable the load calculations to be checked.

102. The data listed below, as applicable, is to be submitted along with the required documents:

TABLE T.H2.102.1 - LIST OF INPUT DATA FOR EVALUATING LOAD-BEARING CAPACITY - GENERAL

General			
Main gear propulsion	Diesel with hydraulic coupling		
	Diesel with other type of coupling		
	Diesel with elastic coupling		
	Turbine		
	Diesel electric		
Type of gerar	Cylindrical gears		
	Bevel gears		
Nominal rated power		P	kW
Rotation per minute of each gear at the rated power	Pinion Wheel	n_1 n_2	1/min

TABLE T.H2.102.2 -DATA TO BE SUBMITTED FOR CYLINDRICAL GEARS

Number of teeth	Pinion Wheel	z_1 z_2	-
Tip diameter	Pinion Wheel	da_1 da_2	mm
Base diameter	Pinion Wheel	db_1 db_2	mm
Transverse pressure angle		α_t	deg
Normal pressure angle		α_n	deg
Face width		b	mm
Helix angle		β	deg
Normal module		m_n	mm
Operating centre distance		a	mm
Web thickness		b_s	Mm
Rim thickness		s_r	mm
External shaft diameter		d_{sh}	mm
Internal shaft diameter		d_{shi}	mm
Face width of one helix on a double helical gear		b_B	mm
Basic rack dedendum		h_{fP}	mm
Mean peak-to-valley roughness of the gear pair		R_z	µm
Mean flank peak-to-valley roughness of the gear pair		R_{zf}	µm
Initial equivalent misalignment		$F_{\beta x}$	µm
Material allowance for finish machining		q	mm
Elasticity factor		Z_E	N/mm^2
Rocwell hardness		HRC	-
Alowable stress number (contact)		σ_{Hlim}	N/mm^2
Nominal stress number (bending)		Σ_{Flim}	N/mm^2
Tip radius of the tool		ρ_{a0}	mm
Tool protuberance		P_{r0}	mm
Profile (addendum) shift coefficient (tool)		x	-

TABLE T.H2.103.1 -DATA TO BE SUBMITTED FOR BEVEL GEARS

Number of virtual teeth		z_{vn}	-
Transverse pressure angle		α_t	deg
Angle of pressure at the highest point of the gearing of a single pari of teeth		α_{an}	deg
Angle of application of the load at the highest point of the gearing of a single pair of teeth		α_{Fan}	
Common face width (for double gear helix gear width of one helix)		b	mm
Mean helix angle		β_m	deg
Mean normal module		m_n	mm
Pitch angle		δ	deg
Web thickness		b_s	Mm
Rim thickness		s_r	mm
External shaft diameter		d_{sh}	mm
Internal shaft diameter		d_{shi}	mm
Face width of one helix on a double helical gear		b_B	mm
Addendum of the basic rack profile		H_{aP}	mm
Dedendum of the basic rack profile		h_{fP}	mm
Mean peak-to-valley roughness of the gear pair		R_z	µm
Mean flank peak-to-valley roughness of the gear pair		R_{zf}	µm
Initial equivalent misalignment		$F_{\beta x}$	µm
Material allowance for finish machining		q	mm
Elasticity factor		Z_E	N/mm^2
Rocwell hardness		HRC	-
Alowable stress number (contact)		σ_{Hlim}	N/mm^2
Nominal stress number (bending)		Σ_{Flim}	N/mm^2
Addendum of the tool		h_{a0}	mm
Tip radius of the tool		ρ_{a0}	mm
Tool protuberance		P_{r0}	mm
Profile (addendum) shift coefficient (tool)		x_{hm}	-
Thickness modification coefficient		x_{sm}	-
Residual cam left by the protuberance		S_{pr}	mm

103. **Documents to be submitted**

Plans of construction of shafts and flanges:

a. General arrangement

b. Transversal assembly

c. Details of the casing

d. Diagram of the load on bearings

e. Plans of the shafts with the material specification

f. Plans of construction of pinion and crown, including specification and details of the procedure of hardening:

f.1. Mechanical properties of the core and the surface;

f.2. Diagram of the depth of the hardened layer as a function of the values of hardening

g. Specification and details of the procedure of the finishing processes:

g.1. Method of finishing of the teeth flanks;

g.2. Surface roughness for flank and rooth of the teeth;

g.3. Correction of the teeth flanks, if any;

g.4. Degree of accuracy with PR-245 (based in DIN 3963 / 3967) or ISO 1328;

g.5. Calculation of interference of pinions, wheels of crowns and / or center indicating the tolerances for minimal and maximum interference

g.6. Calculation of the capacity of load of the gears; and

g.7. The casing construction drawings

h. Operational diagram lubrication system, indicating:

h.1. grade oil specified

h.2. oil temperature expected during the operation

h.3. kinematic viscosity oil

i. Operational diagram control systems, monitoring and safety

i.1. Cross-sectional and longitudinal mounting of gears, indicating the type of clutch, specifying the materials used including chemical composition, heat treatment and mechanical properties, indicating also the process of welding and stress relief where applicable

Form calculation of the gears

H3. DESIGN OF GEARS DETERMINATION OF LOAD CAPACITY

100. Nominal tangential load F_t

Guidance

The normal tangelatial load (tangencial to the reference cylinder with diameter d and perpendicular to an axial palne) is calculated from the nominal torque T transmitted by the gear setr.

End of guidance

101. Calculation of nominal tangential load:

a. Pinion (crown) torque:

$$T_{1,2} = \frac{9549 * P}{n_{1,2}}$$

where:

P = power (kW)
n = rotation(rpm)

b. Tangential load:

$$F_t = \frac{2000 * T_{1,2}}{d_{1,2}}$$

where:

F_t = tangential load (N)
d = reference diameter (mm)

c. For **bevel gears**:

$$F_{mt} = \frac{2000 * T_{1,2}}{d_{m1,2}}$$

where:

F_{mt} = tangential load for bevel gear(N)
$T_{1,2}$ = determined above
$d_{m1,2}$ = mean diameter of the pinion (crown) pitch

200. General factors of influence

201. **Application factor K_A**

The application factor, KA, accounts for dynamic overloads from sources external to the gearing.

KA, for gears designed for infinite life is defined as the ratio between the maximum repetitive cyclictorque applied to the gear set and the nominal rated torque.

The nominal rated torque is defined by the rated power and speed and is the torque used in the ratingcalculations

The application factor **K_A** is given in the table T.H3.201.1 below:

TABLE T.H3.201.1 – FACTOR K_A

Main Propulsion	K_A
Diesel engine with hydraulic or electromagnetic slip coupling	1.00
Diesel engine with high elasticity coupling	1.30
Diesel engine with other types of coupling	1.50
Turbines	1.05
Electric motor	1.05
Auxiliary Gears	**K_A**
Diesel engine with hydraulic or electromagnetic slip coupoling	1.00
Diesel engine with high elasticity coupling	1.20
Diesel engine with other types of coupling	1.40
Electric motor	1.00

202. **Load sharing factor $K\gamma$**

The load sharing factor, $K\gamma$ accounts for the maldistribution of load in multiple path transmissions (dual tandem, epicyclic, double helix, etc.)

$K\gamma$ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor depends on the accuracy and flexibility of the branches.

The values are given in the Table T.H3.202.1 below:

a. Dual gears in tandem:

a.1. Without hollow shaft **$K\gamma$** = 1.15

a.2. With hollow shaft **$K\gamma$ =** 1.30

b. Epicyclical gears:

d.1. With 3 or less planetary gears **$K\gamma$ =** 1.02

d.2. With 4 or less planetary gears **$K\gamma$ =** 1.20

d.3. With 5 or less planetary gears **$K\gamma$ =** 1.30

d.4. With 6 or less planetary gears **$K\gamma$ =** 1.40

Note: a hollow shaft is a shaft having torsional flexibility to improve the distribution of load between the gears

203. **Dynamic factor K_v**

The dynamic factor **$\underline{K_v}$** accounts for internally generated dynamic loads due to vibrations of pinion and wheel against each other.

Kv is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load (**$Ft * KA * K\gamma$**)

The method may be applied only to cases where all the following conditions are satisfied:

a. Steel gears with heavy rim sections

b. $\dfrac{F_t}{b} > 150$ **N / mm**

c. $z_1 < 50$

d. running speed in the subcritical range:

For helical gears $(v^* z_1)/ 100 < 14$

For spur gears $(v^* z_1)/ < 10$

d.5. This method may be applied by all types of gears if:

$(v^* z_1)/ 100 < 3$

For gears other than the above, reference can be made to method B (calculation resonance) of ISO 6336-1 is to be applied

e. For helical gears with $\varepsilon_\beta \geq 1$:

$K_v = K_{v1}$ where $K_{v1} = 1 + K1 (v_{z1}/100)$

where K1 assumes the values specified in the Table T.H3.203.1.

f. For helical gears with $\varepsilon_\beta < 1$

$K_v = K_{v2} - \varepsilon_\beta *(K_{v2} - K_{v1})$

where K_{v2} is the K_v for spur gears

g. For bevel gears:

Regarding the conditions b, c. and d, use the formulas above with the following parameters:

g.1. The actual z_1 is to be used instead of the virtual z_{v1};

g.2. v is to be replaced by v_{mt} (tangential speed in tue mean section); and

g.3. F_t is to be replaced by F_{mt}

TABLE T.H3.203.1 – VALUES OF K1 FOR THE CALCULATION OF K_V

K1 FOR SPUR GEARS					
3*	4	5	6	7	8
0.022	0.030	0.043	0.062	0.092	0.125
K1 FOR HELICAL GEARS					
3*	4	5	6	7	8
0.0125	0.0165	0.0230	0.0330	0.0480	0.0700

* degree of accuracy complying with ISO. Where gears are coupled with different degrees of accuracy, the degree corresponding to the lower accuracy is to be employed.

FIGURE F.H3.203.1 - DYNAMIC FACTOR FOR HELICAL GEAR. ISO GRADES OF ACCURACY 3 - 8 (2)

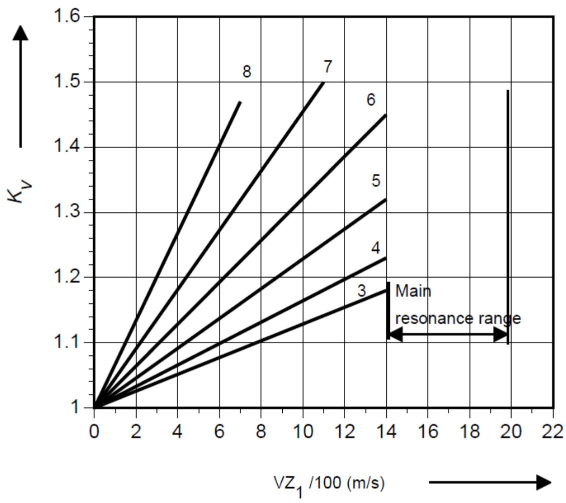
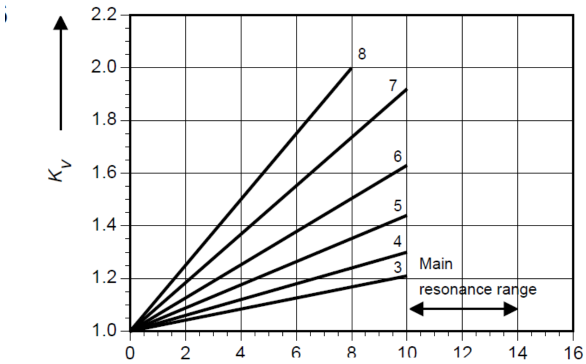


FIGURE F.H3.203.2 - DYNAMIC FACTOR FOR SPUR GEAR. ISO GRADES OF ACCURACY 3 - 8 2



Y- Z1 /100 (m/s)

(2) ISO grades of accuracy according to ISO 1328. In case of mating gears with different grades of accuracy the grade corresponding to the lower accuracy should be used.

204. **Face load distribution factors $K_{H\beta}$ and $K_{F\beta}$**

The face load distribution factors, KHβ for contact stress, KFβ for tooth root bending stress, account for the effects of non-uniform distribution of load across the facewidth.

Factor KHβ and factor KFβ for bending stress at the root of the tooth, and may be defined as:

a. Factor of load on face for the contact stress $K_{H\beta}$

$$K_{H\beta} \frac{\text{Maximum load per unit of face width}}{\text{Mean load per unit of face width}}$$

That is, $K_{H\beta} = (F/b)_{max} \ / \ (F_m/b)$

a.1. Factor of load of face width for stress in the tooth root $K_{F\beta}$

$$KF\beta = \frac{\text{maximum bending stress at tooth root per unit facewidth}}{\text{mean bending stress at tooth root per unit facewidth}}$$

KFβ can be expressed as a function of the factor KHβ.

The factors KHβ and KFβ mainly depend on:

i. gear tooth manufacturing accuracy;

ii. errors in mounting due to bore errors;

iii. bearing clearances;

iv. wheel and pinion shaft alignment errors;

v. elastic deflections of gear elements, shafts, bearings, housing and foundations which support the gear elements;

vi. thermal expansion and distortion due to operating temperature;

vii. compensating design elements (tooth crowning, end relief, etc.).

The face load distribution factors, KHβ for contact stress, and KFβ for tooth root bending stress, can be determined according to the method C2 outlined in the ISO 6336/1 standard.

The tangential loads on the reference cylinder are used for an approximate calculation, i.e., using the transverse load specific:

$$F_m / b = (F_t * K_A * K_v)/b$$

In the reference cylinder and the corresponding maximum loading location.

The $K_{H\beta}$ values are applicable only for gears as below:

i. Equipped with crown, casing, crown shaft and bearings of robust construction;

ii. Pinion fitted on solid or hollow shaft with a ratio of inside diameter to outside diameter not exceeding 0.5 and located symmetrically between the bearings, and

iii. No external load acting on the pinion shaft.

205. The face load factor $K_{H\beta}$

It is calculated by the average load across the face (F_m/b) the factor of rigidity of gearing ($c_{\gamma\beta}$) and misalignment total effective ($F_{\beta y}$).

The effective value of misalignment to be used is to be obtained by combining two factors:

a. The effect of fabrication errors of all relevant factors are considered by the coefficient f_{ma}

b. the effect of elastic deflections of the pinion and pinion shaft is considered by f_{sh}

The manner in which the two elements are combined depends on the modification of the propeller (bulging, correction of the helix, relief from the top or any).

One of the two equations below may be employed depending upon whether the touch point is calculated over the full width of the face.

b_{cal} / b ≤1 corresponding to:

$$(F_{\beta y} * c_{\gamma\beta}) / (2F_m/b) \geq 1$$

$$K_{H\beta} = \sqrt{(2 * F_{\beta y} * c_{\gamma\beta}) / (F_m/b)} \geq 2$$

$$b_{cal} / b = \sqrt{(2 * F_m/b) / (F_{\beta y} * c_{\gamma\beta})}$$

b_{cal} / b >1 corresponding to:

$$(F_{\beta y} * c_{\gamma\beta}) / (2F_m/b) < 1$$

$$K_{H\beta} = 1 + [(F_{\beta y} * c_{\gamma\beta}) / (2 * F_m/b)]$$

$$b_{cal} / b = 0,5 + [(F_m/b) / (F_{\beta y} * c_{\gamma\beta})]$$

where:
$F_m = (F_t * K_A * K_v)$
F_t calculated in H3.101
K_A calculated in H3.201
K_v calculated in H3.202

$K_{H\beta}$ for bevel gears:

$$K_{H\beta} = 1,5 * (0,85/B_b) * K_{H\beta be}$$

The bearing factor representing the influence of the bearings on the load distribution on the face is given by:

FACTOR $K_{H\beta be}$

Both members assembled on dual support	: 1.10
One member assembled on dual support	: 1.25
No member assembled on dual support:	: 1.50

206. **$F_{\beta y}$ initial equivanlent misaligment** is the after the running-in and is given by:

$$F_{\beta y} = F_{\beta x} - y_\beta = F_{\beta x} * \chi_\beta$$

where:

$F_{\beta x}$: verify item 207
y_β: e χ_β: verify item 210

207. **$F_{\beta x}$ initial equivalent misaligment**, i.e., the abslute value of the sum of the deformations, displacements and manufacturing deviations of the pinion and crown measured in the plan of action and that can be determined as follows.

$$F_{\beta x} = 1,33 * B1 * f_{Sh} + B2 * y_\beta$$

where:

f_{Sh}: see item 208

Case a: Gear pair in which the dimensions and compliance with the standard of contact were not tested and which behavior of the bearing submitted to load is defective:
Gear pairs of which the size and suitability of the contact pattern are not proven and the bearing pattern under load is imperfect:

$$F_{\beta x} = 1,33 * B1 * f_{Sh} + B2 * f_{ma} \text{ com } F_{\beta x} \geq F_{\beta x\ min}$$

where:

f_{ma}: see item 209

B1 and B2 are obtained from table T.H3.207.1

Case b: Gear pairs with verification of the favourable position of the contact pattern (e.g. by modification of the teeth or adjustment of bearings)

$$F_{\beta x} = |\, 1.33 * B1 * f_{Sh} - f_{H\beta 6} \,| \text{ com } F_{\beta x} \geq F_{\beta x\ min}$$

where

B1 obtained from table T.H3.207.1

$f_{H\beta 6}$ defined as the total tolerance of deviation of the helix to an accuracy grade ISO 6

TABLE T.H3.207.1 – CONSTANTS B1 AND B2

Modification of the helix			Constants	
N°	**Type**	**Value**	**B1**	**B2**
1	None	-	1	1
2	Central bulging	$C_\beta = 0,5\, f_{ma}$ [(a)]	1	1
3	Central bulging	$C_\beta = 0.5(f_{ma} + f_{sh})$ [(a)]	0.5	0.5
4	Correction of helix [(b)]	Commercial form calculated to match the torque being analyzed	0,1[(c)]	0,1
5	Correction of the helix plus central bulging	Case 2 + case 4	0,1[(c)]	0,5
6	Relief of tip	Appropriate amount of $C_{I(II)}$ [(d)]	0,7	0,7

[(a)] C_β is the height of rounding (see Appendix D of the standard in reference)
[(b)] Predominantly applied under constant load conditions
[(c)] Valid only for best manufacturing practices, otherwise use the appropriate values (higher)
[(d)] $C_{I(II)}$ Relief of tip, (see Annex E of the reference standard)

Case c: For gears having ideal contact pattern, full helix modification, under load (for both helices of double helical gears):

$$F_{\beta x} = F_{\beta x\ min}$$

The definition of $F_{\beta x\ min}$ in the above equations is the following:

where $F_{\beta x\ min}$ is the greater of the two values:

$$F_{\beta x\ min} = (0.005\ \text{mm*µm / N}) * (F_m / b)$$

or:

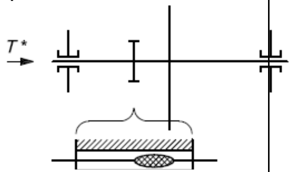
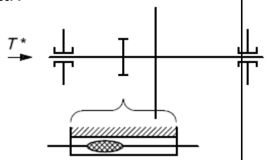
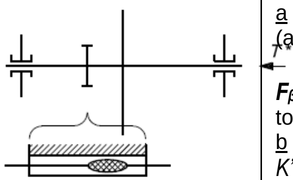
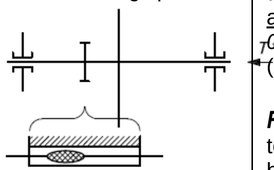
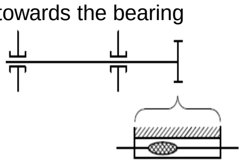
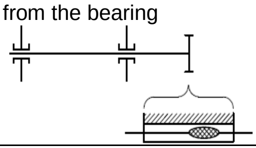
$$F_{\beta x\ min} = 0.005 * f_{H\beta}$$

where:

$f_{H\beta}$ = deviation of the inclination of the helix

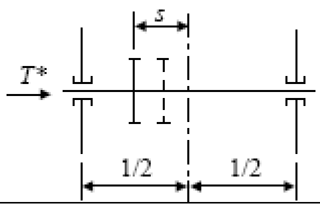
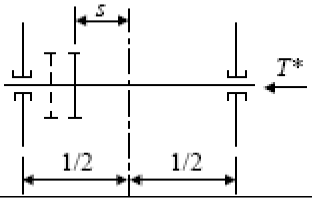
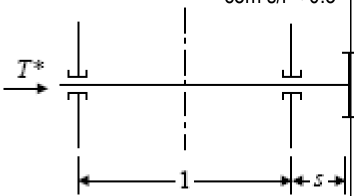
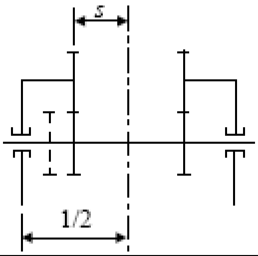
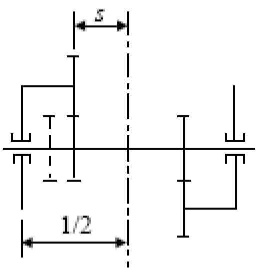
The rules for the use of the above equations are shown in table T.H3.207.2

TABLE T.H3.207.2 – RULES FOR DETERMINING OF $F_{\beta x}$ IN RELATION TO THE STANDARD OF CONTACT

Fig	*Standard of contact*	*Determination of $F_{\beta x}$*
a	Contact pattern lies towards mid bearing span 	$F_{\beta x}$ calculated according to the equation of the case b (compensatory)
b	Contact pattern lies away from mid bearing span 	$F_{\beta x}$ calculated according to the equation of the case a (additive)
c	Contact pattern lies towards mid bearing span 	$F_{\beta x}$ calculated according to the equation of the case a $K'{*}l{*}sl{*}d_a^2(d_1 / d_{sh})^4 \leq B$ (additive) $F_{\beta x}$ calculated according to the equation of the case b $K'{*}l{*}sl{*}d_a^2(d_1 / d_{sh})^4 > B$ (compensatory)
d	Contact pattern lies away from mid bearing span 	$F_{\beta x}$ calculated according to the equation of the case a $(K'{*}l{*}sl{*}d_a^2(d_1 / d_{sh})^4 \leq B–0,3$ (additive) $F_{\beta x}$ calculated according to the equation of the case b $K'{*}l{*}sl{*}d_a^2(d_1 / d_{sh})^4 > B\text{-}0,3$ (compensatory)
e	Contact pattern lies towards the bearing 	$F_{\beta x}$ calculated according to the equation of the case a (additive)
f	Contact pattern lies away from the bearing 	$F_{\beta x}$ calculated according to the equation of the case b (compensatory)

The value f_{sh} takes into account the components of the equivalent misalignment due to bending and torsion of the pinion and pinion shaft.

TABLE T.H3.208.1 – FACTOR *K'* FOR CALCULATION OF THE PINION DECENTRALIZATION

Factor K'		Fig	Arrangement
Without	With		
stiffening			
0.48	0.80	a	with s/l < 0.3 
-0.48	-0.80	b	with s/l < 0.3 
1.33	1.33	c	com s/l < 0.3 
-0.36	-0.60	d	com s/l < 0.3 
-0.6	-1.0	e	com s/l < 0.3 

Notes:

1. In hollow shafts when d1/dsh ≥ 1,15, it is considered as hardened. If d1/dsh < 1,15 is considered without hardening . When the pinion slides on the shaft, on a pin or similar accessory, the shaft is considered as hardened.
2. ***T*** is the tip at which the torque is applied to input or output regardless of the direction of rotation.
3. Dashed lines indicate the helix less deformed or a double helix.

Note: for hollow shafts, the equations below provide sufficiently accurate results provided that the hole diameter does not exceed 0.5 times the diameter of the shaft.

208. **Equivalent misalignment f_{sh} :**

The value fsh takes into account the components of equivalent misalignment resulting from bending and twisting of the pinion and pinion shaft, and its value may be determined as follows.

a. **For both spur gears and single helical gear**:

$$f_{sh} = (F_m/b) * 0{,}046*\{[B + K'(l_s/d_1^2)*(d_1/d_{sh})^4 - 0{,}3] + 03\}*(b/d_1)^2$$

For ***F**$_m$* / ***b*** see Part II, Title 11, Section 5, chapter H3, item 204 above.

with *B** equal to 1 if the total power is transmitted through a single engagement.

b. **For double helical gears**:

$$f_{sh} = (F_m/b) * 0{,}046*\{[B + K'(l_s/d_1^2)*(d_1/d_{sh})^4 - 0{,}3] + 03\}*(b_B/d_1)^2$$

with *B** equal to 1,5, if the total power is transferred by a single engagement.

$\boldsymbol{b_B} = \boldsymbol{b}/2$ is the width of one helix

209. **f_{ma} Mesh misalignment - Misalignment factor of gearing due to deviations in the manufacturing**.

f_{ma} is the maximum separation between the tooth flanks of the meshing teeth of mating gears, when the teeth are held in contact without significant load, the shaft journals being in their working attitudes.

f_{ma} depends on the way in which the deviations of individual components in the plane of action combine, i.e.
whether the helix slope deviation $f_{H\beta}$ of each gear and the alignment deviation of the shafts are additive or compensatory, or whether the alignment of the shafts is adjustable (e.g. by means of adjustable bearings).

Simplified method of calculating f_{ma}

In the design stage: $f_{ma} = 1{,}0 * F_{\beta}$.

For gear pairs with adjustment devices or appropriate modification of the helix:

$f_{ma} = 0{,}5*F_{\beta}$

For helix deviations due to manufacturing inaccuracy:

$f_{ma} = 0{,}5*F_{\beta}$ (F_{β} = total deviation of the helix)

210. **Running-in allowance y_β and running-in factor χ_β**

y_β is the factor that characterizes the equivalent misalignment after the running-in.

It is convenient touse χ_β in calculations, but only as long as y_β is proportional to $F_{\beta x}$

a. For materials:

a.1. **St** (normalized basic steel σ_B < 800 N/mm2) ;

a.2. **St** (cast);

a.3. **V** special forged steel hardened in the core;

a.4. Alloy or carbon with $\sigma_B \geq 800$ N/mm2);

a.5. **Vfund** cast steel hardened in the core;

a.6. **GGG** (pearlitic nodular cast iron;

a.7. Bainitic;

a.8. Ferritic structure.); and

a.9. **GTS** (black malleable cast steel, pearlitic structure).

Note: See ISO 6336-5 in reference to the specifications and procedures of various heat treatments

The Constant K 'allows a tolerance for the position of the pinion from the edge of torque, and is to be taken from the table T.H3.208.1

$y\beta = 320 * F\beta x. / \sigma H lim)$

$\chi_\beta = 1 - (320 / \sigma_{H\,lim})$
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Where:

$y_\beta \leq F_{\beta x}$ e $\chi_\beta \geq 0$;

$v \leq 5$ m/s – no restrictions;

5 m/s $\leq v \leq$ 10 m/s – the upper limit of y_β is 25 600/ $\sigma_{H\,lim}$ corresponding to $F_{\beta x}$ = 80 µm;

v > 10 m/s – the upper limit of y_β is 12 800/ $\sigma_{H\,lim}$ corresponding to $F_{\beta x}$ = 40 µm;

b. For materials:

b.1. **GGG** (pearlitic nodular cast iron, bainitic, ferritic structure.);

b.2. **GG** (Gray cast iron and nodular cast iron, ferritic structure).

$y_\beta = 0{,}55 * F_{\beta x.}$

$\chi_\beta = 0{,}45$

Where:

$v \leq 5$ m/s – no restricions;

5 m/s $\leq v \leq$ 10 m/s – the upper limit of y_β is 0,45 µm corresponding to $F_{\beta x}$ = 80 µm;

v > 10 m/s – the upper limit of y_β is 22 µm corresponding to $F_{\beta x}$ = 40 µm;

c. For materials:

Eh hardened forged steel ingot:

IF special forged steel induction or flame hardened:

NV cast steel hardened in the core, nitride and nitrocarburated.

$y_\beta = 0{,}15 * F_{\beta x.}$

$\chi_\beta = 0{,}85$

For all speeds, being the upper limit of µm, corresponding to $F_{\beta x}$ = 40 µm

When the material of pinion differs from the material of the crown, $y_{\beta 1}$ e $\chi_{\beta 1}$ for the pinion, and $y_{\beta 2}$ and $\chi_{\beta 2}$ for a crown are to be determined separately.

The mean values for each:

$y_\beta = 0.5 \cdot (y_{\beta 1} + y_{\beta 2})$

$\chi_\beta = 0.5 \cdot (\chi_{\beta 1} + \chi_{\beta 2})$

are to be used in the calculation

211. **Determination of σ_{Hlim} and σ_{Flim}**

Guidance

The allowable stress number, σH lim, is derived from a contact pressure that may be sustained for a specified number of cycles without the occurrence of progressive pitting. For some materials, 5 × 107 stress cycles are considered to be the beginning of the long-life strength range

End of guidance

The valuesof allowable stress limit for σ_{Hlim} and σ_{Flim} can be calculated by the following equation:

$\sigma_{Hlim} = A*x + B$

$\sigma_{Flim} = A*x + B$

where

A and B are constants defined below.

The hardness range is restricted by the minimum and maximum values given in the table T.H3.210.1below:

TABLE T.H3.210.1 – CALCULATION OF σ_{Hlim} AND σ_{Fli}

MATERIAL	TYPE	INITIALS	FIGURE*	QUALITY **	CONTACT σ_{Hlim}	BENDING σ_{Flim}	HARDNESS H V	
							MINIMUM HARDNESS	MAXIMUM HARDNESS
Forged steels hardened in the core	Carbon steels	V	5 6	MQ	0,925 HV+360	0,240HV10+163	135 115	210 215
	Alloy steels	V	5 6	MQ	1,313 HV+373	0,425 HV10+187	200	360
Cast steels hardened in the core	Carbon steels	V fund	7 8	ML-MQ	0,831HV+300	0,224HV+117	130	215
	Alloy steels	V fund	7 8	ML-MQ	1,276HV+298	0,364HV+161	200	360
Forged steels hardened in the surface (case hardened)	Normal grade	Eh	9 10	MQ	1500	425	660	800
	High quality special	Eh	9 10	ME	1650	525	660	800
Cast and forged steels hardened by flame or induction		IF	11 12	MQ	0,541HV+882	0,138HV+290 369	500 500 570	615 570 615
Steel nitrided by gas hardened, tempered and nitrided by gas	Nitriding	NT nit	13 a 14 a	MQ	1250	420	650 650	900 900
	Thorough hardened	NV nit	13b 14b	MQ	998	363	450 450	650 650

* As standard ISO 6336-5 item 5.3.5

** Quality: The classification for quality as standard ISO 6336-5:

ML – appropriate to more modest demands as much as the thermal treatment as on the quality of material;
MQ – appropriate to the manufacture by manufacturers with experience at moderate cost
ME – appropriate to requirements asociated with a high degree of operational reliability

σ_{Hlim} = maximum stress allowed for contact

σ_{Flim} = rated bending stress

σ_{FE} = bending stress allowed for elastic regime, being $\sigma_{FE} = \sigma_{Flim} * Y_{ST}$

Y_{ST} is a correction factor related to the dimensions of the test gears in reference and will be defined forward.

212. **Face load factor for tooth root stress $K_{F\beta}$**

KFβ takes into account the effect of the load distribution over the face width on the stresses at the tooth root. It depends on the variables which are determined for KHβ and also on the face width to tooth depth ratio, b/h.

$K_{F\beta}$ is to be determined by the following formula:

$$K_{F\beta} = K_{H\beta}^{1/1 + (h/b) + (h/b)^2}$$

Where b/h is the least between b_1/h_1 e b_2/h_2 but should not be lesser than 3.

a. For double helix gears, b_B is to be used in place of b.

b. In cases in which the tooth tip be under light load or without load, i. e., with relief of tooth tip, we have:

$$K_{F\beta} = K_{H\beta}$$

c. For bevel gears:

$$K_{F\beta} = K_{H\beta} / K_{FO}$$

$K_{FO} = 0{,}211^*(r_{eo} / R_m)^q + 0{,}789$ (For spiral bevel gears)

$q = (0{,}279 / \log(\sin \beta_m)$

where:

K_{F0} = 1 for spur gears or with inclination 0 (zero bevel)

r_{eo} = cuttting radius, mm

R_m = mean cone distance

Restrictions for K_{F0}

If K_{F0} <1, use K_{F0} = 1

If K_{F0} >1,15 use K_{F0} =1,15

213. **Factors of transverse load distribution $K_{H\alpha}$ and $K_{F\alpha}$**

The factors of distribution of transversal load $K_{H\alpha}$ for contact stresses and $K_{F\alpha}$ for bending stresses in the tooth's root take into accout the effects of the failure of pitch and the profie in the distribution of transversal load between two or more pairs of geared teeth.

The transverse load factors, $K_{H\alpha}$ for surface stress and $K_{F\alpha}$ for tooth root stress, account for the effect of the non-uniform distribution of transverse load between several pairs of simultaneously contacting gear teeth as follows.

The transverse load factors are defined as the ratio of the maximum tooth load occurring in the mesh of a gear pair at near zero min−1 to the corresponding maximum tooth load of a similar gear pair which is free from inaccuracies. The main influences are

The main influences are:

a. deflection under load

b. Profile modifications

c. Tooth manufacturing accuracy

d. Running-in effects

Values of $K_{H\alpha}$ and $K_{F\alpha}$ for contactt $\varepsilon_\gamma \leq 2$:

$$K_{H\alpha} = K_{F\alpha} = (\varepsilon_\gamma / 2) \{0{,}9 + 0{,}4[c_{\gamma\alpha} (f_{pb} - y_a) / F_{tH}/b]\}$$

Values of $K_{H\alpha}$ and $K_{F\alpha}$ for contact ratio $\varepsilon_\gamma > 2$

$$K_{H\alpha} = K_{F\alpha} = \{0{,}9 + 0{,}4\sqrt{2(\varepsilon_\gamma - 1)} / \varepsilon_\gamma * [c_{\gamma\alpha} (f_{pb} - y_a) / F_{tH} / b]\}$$

$c_{\gamma\alpha}$ = mesh stiffness in accordance

f_{pb} = the larger of the base pitch deviations of pinion or wheel should be used; 50 % of this tolerance may be used, when profile modifications compensate for the deflections of the teeth at the actual load level

y_a = running-in allowance;

F_{tH} = determinant tangential load in a transverse plane, being $F_{tH} = F_t{*}K_A{*}K_V{*}K_{H\beta}$

For bevel gears:

$$F_{mtH} = F_{mt}{*}K_A{*}K_V{*}K_{H\beta}$$

For bevel gears, f_{pt}, εv_γ, F_{mtH}, F_{mt} e α_{vt} (equivalent) to be employed in place of f_{pb}, ε_V, F_{tH}, F_t e α_t

Limiting conditions for $K_{H\alpha}$

$$K_{H\alpha} > \varepsilon_V / (\varepsilon_\alpha * Z_\varepsilon^2)$$

Then replace $\underline{K_{H\alpha}}$ by $\varepsilon_V / (\varepsilon_\alpha * Z_\varepsilon^2)$ and if $\underline{K_{H\alpha}}$ < 1, use 1,0 for $\underline{K_{H\alpha}}$

$Z_\varepsilon = \sqrt{[(4- \varepsilon_\alpha) / 3] * (1- \varepsilon_\beta) + (\varepsilon_\beta / \varepsilon_\alpha)}$, for the contact factor (pitting) for helical gears $\varepsilon_\beta < 1$

$Z_\varepsilon = \sqrt{1 / \varepsilon_\alpha}$, for the contact factor (pitting) for helical gears $\varepsilon_\beta \geq 1$

$Z_\varepsilon = \sqrt{(4 - \varepsilon_\alpha) / 3}$, , for the contact factor (pitting) for spur gears

For bevel gears:

When $K_{H\alpha} > \varepsilon_{v\gamma} / (_{\varepsilon v\alpha} * Z_{LS}^2)$ usar $K_{H\alpha} > \varepsilon_{v\gamma} / (\varepsilon_{v\alpha} * Z_{LS}^2)$

For calculation of the load distribution factor see H3.309 below.

Restrictions to $K_{F\alpha}$

$K_{F\alpha} > (\varepsilon_v/0,25 * \varepsilon_\alpha + 0,75)$

300. Surface durability

Note: The present item H3.300 is based on ISO 6336 - Part 2 and Recommendation M56 of the International Association of Classification Societies. For bevel gears, it is based on ISO DIS 10300.

301. The criterion for surface durability is based on the Hertz pressure on the operating pitch point or at theinner point of single pair contact. The contact stress σ_H must be equal to or less than the permissible
contact stress σ_{HP}.

302. **Contact stress**

The contact stress σ_H is to be determined as follows:

$$\sigma_H = \sigma_{HO} \sqrt{K_A\, K_\gamma\, K_v\, K_H\alpha\, K_{H\beta}} \leq \sigma_{HP}$$

where:

σ_{HO} = basic value of contact stress for pinion and wheel

For spur gears with ε_β = 0

For the pinion:

$$\sigma_{HO} = Z_B Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t\,(u+1)}{d_1 b\, u}}$$

For the wheel:

$$\sigma_{HO} = Z_D Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t\,(u+1)}{d_1 b\, u}}$$

where:

Z_B : single pair mesh factor for pinion – see H3.304

Z_D : single pair mesh factor for wheel – see H3.304

Z_H : zone factor – see H3.305

Z_E : elasticity factor - see H3.306

Z_ε : contact ratio factor – see H3.307

Z_β : helix angle factor – see H3.308

F_t = nominal tangential load at reference cylinder in the transverse section

b: common face width

d_1 : reference diameter of pinion

u : gear ratio (positive for external gears and negative for inrernal gears)

For factors K_A, K_γ, K_v, $K_H\alpha$ e $K_{H\beta}$ see H3.200 above.

For bevel gears:

σ_{HO1} : basic factor of contact stress for the pinion

$$\sigma_{HO1} = Z_{M-B} Z_H Z_E Z_{LS} Z_\beta Z_K \sqrt{\frac{F_{mt}\,(u+1)}{d_{v1} * l_{bm}} \frac{u_v + 1}{u_v}}$$

For angle $\Sigma = \delta_1 + \delta_2 = 90°$

$$\sigma_{HO1} = Z_{M-B} Z_H Z_E Z_{LS} Z_\beta Z_K \sqrt{\frac{F_{mt}\,(u+1)}{d_{m1} * l_{bm}} \frac{u^2 + 1}{u}}$$

where:

Z_{M-B} : factor of half zone

Z_H : factor of zone (see H3.305)

Z_E : factor of elasticity (see H3.306)

Z_{LS} : factor of load distribution (see H3.309)

Z_β : factor of helix angle (see H3.308)

Z_K: factor of bevel gear “flank” (see H3.310)

F_{mt} : rated tangential load

b : Common face width

d_{m1} : conical pinion mean diameter of the pitch

d_{v1} : virtual diameter of pinion reference

u : reduction ratio (positive for outer gears and negative for inner gears)

u_v : virtual reduction ratio (equivalent) of the cylindrical gear

303. Permissible contact stress

The permissible contact stress σ_{HP} is to be evaluated separately for pinion and crown;

$$\sigma_{HP} = (\sigma_{Hlim} * Z_N / S_H) * Z_L * Z_v * Z_r * Z_w * Z_X$$

where:

σ_{Hlim} : endurance limit for contact stress

Z_N : life factor for contact stress (see H3.312)

S_H : safety factor for contact stress (see H3.318)

Z_L : lubrication factor (see H3.313)

Z_v = speed factor (see H3.314)

Z_r = roughness factor (see H3.315)

Z_w = hardness ratio factor (see H3.316)

Z_X = size factor for contact stress (see H3.317)

304. **Single pair mesh factors, ZB and ZD**

The single pair mesh factors, ZB for pinion and ZD 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 ZH.

The factors transform the contact stresses determined at the pitch point to contact stresses considering the flank curvature at the inner point of single pair contact.

The single pair mesh factors, ZB for pinions and ZD for wheels, can be determined as follows:

a. **For spur gears when $\varepsilon_\beta = 0$:**

Z_B = M1 or 1, whichever os the larger value;

Z_D = M1 ou 1, whichever is the larger value

$$M_1 = \frac{\tan \alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a1}/d_{b1})^2 - 1} - (2\pi/z_1)\right]*\left[\sqrt{(d_{a2}/d_{b2})^2 - 1} - (\varepsilon_a - 1)*(2\pi/z_2)\right]}}$$

$$M_2 = \frac{\tan \alpha_{wt}}{\sqrt{\left[\sqrt{(d_{a2}/d_{b2})^2 - 1} - (2\pi/z_2)\right]*\left[\sqrt{(d_{a1}/d_{b1})^2 - 1} - (\varepsilon_a - 1)*(2\pi/z_1)\right]}}$$

i.2. **To bevel gears when $\varepsilon_\beta = 0$**:

In the formulas of M1 and M2 of the item above:

- α_{wt} is to be replaced by α_{vt}
- d_a is to be replaced by d_{va}
- d_b is to be replaced by d_{vb}
- ε_a is to be replaced by ε_{va}
- z is to be replaced by z_v

a. **To cylindrical helical gears when $\varepsilon_\beta \geq 1$**:

$Z_B = Z_D = 1$

b. For bevel gears when $\varepsilon_\beta \geq 1$

Z_{M-B} = M or 1, whichever be greater

$$M = \frac{\tan \alpha_{vt}}{\sqrt{\left[\sqrt{(d_{va1}/d_{vb1})^2 - 1} - \varepsilon_a(\pi/z_{v1})\right]*\left[\sqrt{(d_{va2}/dv_{b2})^2 - 1} - (\varepsilon_a)*(2\pi/z_{v2})\right]}}$$

c. **To cylindrical helical gears when $0 < \varepsilon_\beta < 1$** values of Z_B and Z_D are determined by linear interpolation between the values Z_B and Z_D for spur gears, and between the values Z_B and Z_D for helical gears with $\varepsilon_\beta < 1$.

Therefore:

$Z_B = M_1 - \varepsilon_\beta(M_1 - 1)$ e $Z_B \geq 1$

$Z_D = M_2 - \varepsilon_\beta(M_2 - 1)$ e $Z_D \geq 1$

d. For bevel gears when $0 < \varepsilon_\beta < 1$

Z_{M-B} = M or 1, whichever is greater, where:

$$M = \frac{\tan \alpha_{vt}}{\sqrt{\left[\sqrt{(d_{va1}/d_{vb1})^2 - 1} - (2+(\varepsilon_a - 2) * \varepsilon_\beta) * (\pi / z_{v1})\right] * \left[\sqrt{(d_{va2}/d_{vb2})^2 - 1} - (2*(\varepsilon_a - 1) + (2 - \varepsilon_a) * \varepsilon_\beta) * (\pi / z_{v2})\right]}}$$

305. **Zone factor Z_H**

The zone factor, ZH, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and relates the tangential force at the reference cylinder to the normal force at the pitch cylinder.

The zone factor, ZH, can be calculated as follows :

g.4. **For spur gears**:

$$Z_H = \sqrt{(2*\cos \beta_b * \cos \alpha_{wt}) / (\cos^2 \alpha_t * \text{sen } \alpha_{wt})}$$

g.5. For bevel gears:

$$Z_H = 2*\sqrt{\cos \beta_{vb} / \text{sen } (2*\alpha_{vt})}$$

306. **Elasticity factor Z_E**

The elasticity factor, ZE, accounts for the influence of the material properties E (modulus of elasticity) and υ (Poisson's ratio) on the Hertz pressure.

The elasticity factor Z_E for steel gears (E = 206000 N/mm^2, υ = 0,3) is equal to:

$$Z_E = 189{,}8 \text{ N}^{1/2} / \text{mm}$$

For other materials, the values of E and υ should be consulted in specialized tables.

307. **Contact ratio factor Zε**

The contact ratio factor, Zε, accounts for the influence of the transverse contact ratio and the overlap ratio on the specific surface load of gears.

The factor Zε can be calculated as follows:

To spur gears:

$$Z\varepsilon = \sqrt{(4-\varepsilon_\alpha)/3}$$

To helical gears with $\varepsilon_\beta < 1$

$$Z\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3} * (1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}$$

To helical gears with $\varepsilon_\beta \geq 1$

$$Z\varepsilon = \sqrt{(1/\varepsilon_\alpha)}$$

308. **Factor of helix angle Z_β**

The helix angle factor, Zβ, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Zβ is dependent only on the helix angle.

The factor of helix angle, Z_β, can be calculated as follows:

$$Z_\beta = \sqrt{\cos \beta}$$

where β is the reference angle of the helix.

309. **Factor of load distributuion Z_{LS} to bevel gears**

The factor of load distribution Z_{LS} takes into account the load distribution between one or more conjugate pairs in contact.

For:

$\varepsilon_{v\gamma} \leq 2$: $Z_{LS} = 1$

For:

$\varepsilon_{v\gamma} > 2$ e $\varepsilon_{v\beta} > 1$

$$Z_{LS} = [\, 1 + 2 * \{1-(2/\varepsilon_{v\gamma})^{1,5}\} * \sqrt{1-(4/\varepsilon_{v\gamma}^2)}\,]^{-0,5}$$

For other cases consult the reference standard (ISO/DIS 10300)

310. **Factor of flank for bevel gears, Z_K**

The factor of flank for **bevel gears**, Z_K, takes into account the diference between the loading for the cylindrical gears and for bevel gears and adjust the contact stresses so that the allowable stresses be applied.

$Z_K = 0{,}8$

311. **Endurance limit for contact stress, σ_{Hlim}**

For a given material, σHlim is the limit of repeated contact stress which can be permanently endured.

The value of σHlim can be regarded as the level of contact stress which the material will endure without pitting for at least 50 x 106 load cycles.

For this purpose, the pitting is defined as:

a. For gears without superficial hardening:

b. Area with pitting > 2% of the total active area of the flank

For gears with superficial hardening:

c. Área of pitting > 0,5% of the total active área of flank; or

d. 4% of an area of a tooth flank in particular

The values of σ_{Hlim} are to correspond to a failure probability of 1% or less.

The contact stress limit σ_{Hlim} can be determined according to the values given in ISO 6336/5, reproduced in H3.211 item above.

312. **Life factor Z_N**

The life factor, ZN, 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 in table T.H3.312.1, extracted from the reference ISO 6336-2.

TABLE T.H3.312.1 – LIFE FACTOR Z_N

Material	***Number of load cycles***	***Life factor Z_N***
St, V GGG (perlitic, bainitic) GTS (perlitic) Eh, If Only when some degree of pitting be tolerated	$N_L < 6 \times 10^5$ estático	1,6
	$N_L = 10^7$	1,3
	$N_L = 10^9$	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufacture, lubrification and experience	1,0
GGG (perlitic, bainitic) GTS (perlitic) Eh, If	$N_L = 10^5$	1,6
	$N_L = 5 \times 10^7$	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufacture, lubrification and experience	1,0
GG, GGG (perlitic) NT (nitrided) NV (nitrided	$N_L = 10^5$	1,3
	$N_L = 25 \times 10^6$	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufacture, lubrification and experience	1,0
NV (nitrocarburetado)	$N_L = 10^5$	1,1
	$N_L = 25 \times 10^6$	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufacture, lubrification and experience	1,0

St normalized basic steel $\sigma_B < 800$ N/mm2
V special forged steel with core hardening, alloy or carbon with $\sigma_B \geq 800$ N/mm2
GGG nodular cast iron (perlitico., bainitic, ferritic structure),
GTS (Black malleable cast iron, pearlitic structure).
GG Gray cast iron
Eh Forged steels with case hardened
IF Steel castings and forgings hardened by flame or induction
NT Gas nitrided steel
nitrided) nitrided steel, hardened, tempered and gas nitrided
NV (nitrided) nitrided steel, hardened, tempered and gas nitrided
NV (nitro carburated) steel hardened, tempered, nitro carburated

Nota: See standard ISO 6336-5 in reference to the specifications and procedures of various heat treatments.

313. **Lubricant factor Z_L**

The lubricant factor, ZL, accounts for the influence of the type of lubricant and its viscosity.

The factor Z_L can be calculated from the following equation:

$$Z_L = C_{ZL} + \frac{4*(1{,}0-C_{ZL})}{(1{,}2 + 134/\upsilon\ 40)^2}$$

In the range of 850 N / mm^2

$C_{ZL} = [(\sigma_{Hlim} – 850) * 0{,}08/350)] + 0{,}83$

If σ_{Hlim} < 850 N/mm^2 consider C_{ZL} = 0,83

If σ_{Hlim} > 1200 N/mm^2 consider C_{ZL} = 0,91

and:

υ_{40} = rated kinematic viscosity of the oil at 40 ° C in mm^2/s regarded as viscosity ISO classification, represented in the table T.H3.313.1 below:

TABLE T.H3.3132.1 – GRADE OF LUBRICANT VISCOSITY ACCORDING TO ISO

Grade of viscosity of lubricant According to ISO	VG 32	VG 46	VG 68	VG 100	VG 150	VG 220	VG 320
Mean viscosity υ_{40} N/mm2	32	46	68	100	150	220	320
Mean viscosity υ_{50} N/mm2	21	30	43	61	89	125	180

314. **Speed factor Zv**

The speed factor Zv takes into account the influence of the pitch line velocity in the capacity of the surface resistance, and can be calculated as follows:

$$Zv = C_Zv + \frac{2*(1{,}0-C_Zv)}{\sqrt{0{,}8 + 32/v}}$$

In the range 850 N/mm^2 ≤ σ_{Hlim} ≤ 1200 N/mm^2 C_Zv can be calculated as follows:

$C_{ZL} = [(\sigma_{Hlim} – 850) * 0{,}08/350)] + 0{,}85$

315. **Roughness factor Z_R**

The roughness factor Z_R accounts for the influence of the surface roughness on the surface endurance capacity.

The roughness factor Z_R can be calculated by the following equations:

$Z_R = (3/R_{z10}) * CZR$

where:

$R_z = (R_{z1} + R_{z2}) / 2$

The peak-to-valley roughness determined for the pinion R_{Z1} and for the wheel R_{Z2} are mean values for the peak-to-valley roughness R_Z measured on several tooth flanks (R_Z as defined in the reference standard).

$$R_{z10} = R_z \sqrt[3]{\frac{10}{\rho_{red}}}$$

where the relative radius of curvature is given by:

$\rho_{red} = \rho_1 * \rho_2 / (\rho_1 + \rho_2)$

wherein:
$\rho_{1,2} = 0{,}5\ d_{b1,2} * \tan \alpha_{tw}$

(for internal gears, d_b has negative sign).

If the roughness stated is an Ra value (= CLA value) (= AA value) the following approximate relationship can be applied :

$Ra = C_{LA} = AA = Rz/6$

In the range 850 N/mm^2 ≤ σ_{Hlim} ≤ 1200 N/mm^2 CZR can be calculated as follows:

$C_{ZR} = 0{,}32 – 0{,}0002 * \sigma_{Hlim}$

If σ_{Hlim} < 850 N/mm^2 consider C_{ZR} = 0,150

If σ_{Hlim} > 1200 N/mm^2 consider C_{ZR} = 0,080

316. **Hardness factor Z_W**

The hardness ratio factor, ZW, accounts for the increase of surface durability of a soft steel gear meshing with a significantly harder gear with a smooth surface.

The factor of hardness ratio Z_W can be calculated by the following equations:

ZW apply to the soft gear only.
The factor mainly depends on:

a. hardness of the soft gear;

b. alloying elements of the soft gear;

c. tooth flank roughness of the harder gear.

$Z_W = 1{,}2 – (HB-130)/1700$

where:

HB = Brinell hardness of the softer material

For HB < 130, Z_W = 1,2 will be adopted.

For HB > 470, Z_W = 1,0 will be adopted.

317. **Size factor Z_x**

The size factor, ZX, accounts for the influence of tooth dimensions on permissible contact stress and reflects the non-uniformity of material properties.

The factor mainly depends on:

a. material and heat treatment;

b. tooth and gear dimensions;

c. ratio of case depth to tooth size.

d. ratio of case depth to equivalent radius of curvature.

For through-hardened gears and for surface-hardened gears with adequate casedepth relative to tooth size and radius of relative curvature ZX = 1. When the case depth is relatively shallow then a smaller value of ZX should be chosen.

The value Z_x is given in Table T.H3.317.1 below:

TABLE T.H3.317.1 – VALUES OF Z_x

PINION HEAT TREATMENT		Z_X
Hardening by carbiding and induction	$m_n \leq 10$ $10 < m_n < 30$ $30 \leq m_n$	1,00 $1{,}05 - 0{,}005m_n$ 0,9
Nitrided	$m_n < 7{,}5$ $7{,}5 < m_n < 30$ $30 \leq m_n$	1,00 $1{,}08 - 0{,}011m_n$ 0,75
Hardening of the core	All modules	1,00
For **bevel gears**, the *mn* (normal module) is to be replaced by *mmn* (normal module in the middle of the face width).		

318. **Safety factor for contact stress SH**

The safety factor for contact stress, SH, can be assumed by RBNA taking into account the type of application.

The following guidance values can be adopted :

a. Main propulsion gears : 1.20 4 1.40

b. Auxiliary gears : 1.15 4 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 RBNA.

TABLE T.H3.318.1 – SAFETY FACTORS FOR CONTACT STRESS $S_{H\ min}$ / SAFETY FATORS FOR BENDING STRESS OF THE TOOTH $SF_{H\ min}$

Application	S_H *min*	*SF min*
Propulsion gears	1,40	1,80
Propulsion gears for yachts and small vessels with one propeller	1.25	1,50
Propulsion gears for yachts and small vessels with more than one propeller	1,20	1,45
Auxiliary gears	1,15	1,40

400. Calculation of tooth root bending strength

401. The criterion for tooth root bending strength is the permissible limit of local tensile strength in the root fillet.

The root stress σF and the permissible root stress σFP shall be calculated separately for the pinion and the wheel.

σF must not exceed σFP.

The root stress σ_F and the allowable root stress σ_{FP} will be calculated separately to pinion and crown.

σ_F is not to exceed σ_{FP}.

The following formulae and definitions apply to gears having rim thickness greater than 3.5 mn.

The result of rating calculations made by following this method are acceptable for normal pressure angles up to 25° and reference helix angles up to 30°.

For larger pressure angles and large helix angles, the calculated results should be confirmed by experience as by method A of the reference standard.

402. **Tooth root bending stress for pinion and wheel**

For spur gears

$$\sigma_{F1,2} = \frac{\boldsymbol{F_t}}{b * m_n} * \boldsymbol{Y_F} * \boldsymbol{Y_S} * \boldsymbol{Y_\beta} * \boldsymbol{K_A} * \boldsymbol{K_\gamma} * \boldsymbol{K_{F\alpha}} * \boldsymbol{K_{F\beta}} \leq \sigma_{FP1,2} \text{ N/mm}^2$$

For bevel gears

$$\sigma_{F1,2} = \frac{\boldsymbol{F_{mt}}}{b * m_n} * \boldsymbol{Y_{F\alpha}} * \boldsymbol{Y_{S\alpha}} * \boldsymbol{Y\varepsilon} * \boldsymbol{Y_K} * \boldsymbol{Y_{LS}} * \boldsymbol{K_A} * \boldsymbol{K_\gamma} * \boldsymbol{K_{F\alpha}} * \boldsymbol{K_{F\beta}} \leq \sigma_{FP1,2} \text{ N/mm}^2$$

where:

Y_F , Y_{Fa}: tooth form factor (see H3.404)

Y_S , Y_{Sa}: Stress correction factor (see H3.405)

Y_β : Helix angle factor (see H3. 406)

$Y\varepsilon$: Contact ratio factor (see H3.105)

Y_K : Bevel gear factor (see H3.407)

Y_{LS} : Load sharing factor (see H3.406)

F_t, : tangential load (see H3.101)

F_{mt},: tangential load for bevel gears (see H3.101)

K_A: Application factor (see H3.201)

K_γ : Load distribution factor (see H3.202)

K_v : Dynamic factor (see H3.203)

$K_{F\alpha}$: Transversal load distribution factors (see H3.211)

$K_{F\beta}$: Face load distribution factor (root stress) (see H3.204)

b : face width (see H1.301)

m_n : normal module (see H1.301)
m_{mn} : mean normal module (see H1.304)

403. **Permissible tooth root bending stress for pinion and wheel σ_{mt}**

$$\sigma_{FP} = (\sigma_{FE} * Y_d * Y_N/S_F) * Y_{\delta \, rel \, T} * Y_{R rel T} * Y_X$$

where:

Y_d : design factor (see H3.408)

Y_N : life factor for tooth root stress for reference test conditions (see H3.409)

$Y_{\delta relT}$: relative notch sensitive factor (see H3.410)

Y_{RrelT} : relative surface factor, the quotient of the gear tooth root surface factor of interest divided by the tooth root surface factor of the reference test gear,
$Y_{R\,rel\,T} = Y_R / Y_{RT}$ (see H3.411)

Y_X = size factor (tooth root) (see H3.412)

σ_{FE} : allowable stress number (bending)

S_F : safety factor for tooth root bending stress

404. **Tooth form factor Y_F , Y_{Fa}**

The tooth form factor, Y_F, represents the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact.

Y_F shall be determined separately for the pinion and the wheel.

In the case of helical gears, the form factors for gearing shall be determined in the normal section, i.e. for the virtual spur gear with virtual number of teeth zn.

To spur gears:

$$Y_F = \frac{6 * \frac{h_F}{m_n} \cos \alpha_{Fen}}{(s_{Fn} / m_n)^2 \cos \alpha_n}$$

$$Y_F = \frac{6 * \frac{h_{Fa}}{m_{mn}} \cos \alpha_{Fan}}{(s_{Fn} / m_{mn})^2 \cos \alpha_n}$$

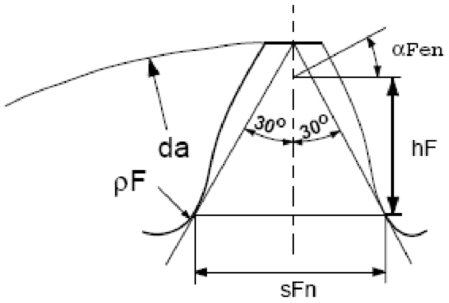
Where:

h_f, h_{fa}: bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact

s_{Fn} : tooth chord at the critical section

α_{Fen}, α_{Fan} : pressure angle at the outer point of single tooth pair contact in the normal section

FIGURE F.H3.404.1 - SECTION NORMAL OF A TOOTH (TO CALCULATE H_F, S_{FN}, A_{FEN})



405. **External gears**

For spur gears:

Tooth root chord at the critical section, s_{Fn}

$$\frac{s_{Fn}}{m_n} = z_n * \sin\left(\frac{\pi}{3} - \upsilon\right) + \sqrt{3} * \left(\frac{G}{\cos\upsilon} - \frac{\rho_{a0}}{m_n}\right)$$

$$\upsilon = 2 * \frac{G}{z_n} * \tan \upsilon - H$$ in degrees to be calculated by interaction

G: Parameter defined by:

$$G = \frac{\rho_{a0}}{m_n} - \frac{h_{a0}}{m_n} + x$$

H: Parameter defined by:

$$H = \frac{2}{\mathbf{z_n}} * \left(\frac{\pi}{\mathbf{2}} - \frac{E}{\mathbf{m_n}} \right) - \frac{\pi}{\mathbf{3}}$$

where:

$$\mathbf{z_n} = \frac{\mathbf{z}}{\cos^2\beta_b * \cos\beta}$$

and:

$$\beta_b = \arccos \sqrt{1-(\sin\beta * \cos\alpha_n)^2}$$

E: parameter defined by:

$$E = \frac{\pi}{4} * m_n - h_{a0} * \tan\alpha_n + \frac{S_{pr}}{\cos\alpha_n} * (1-\sin\alpha_n) \quad \frac{\rho_{a0}}{\cos\alpha_n}$$

S_{pr}: Residual fillet undercut, in mm

$S_{pr} = p_{r0} - q$

$S_{pr} = 0$ (gear without the protuberance)

FIGURE F.H3.404.2 - TOOTH BASIC PROFILE

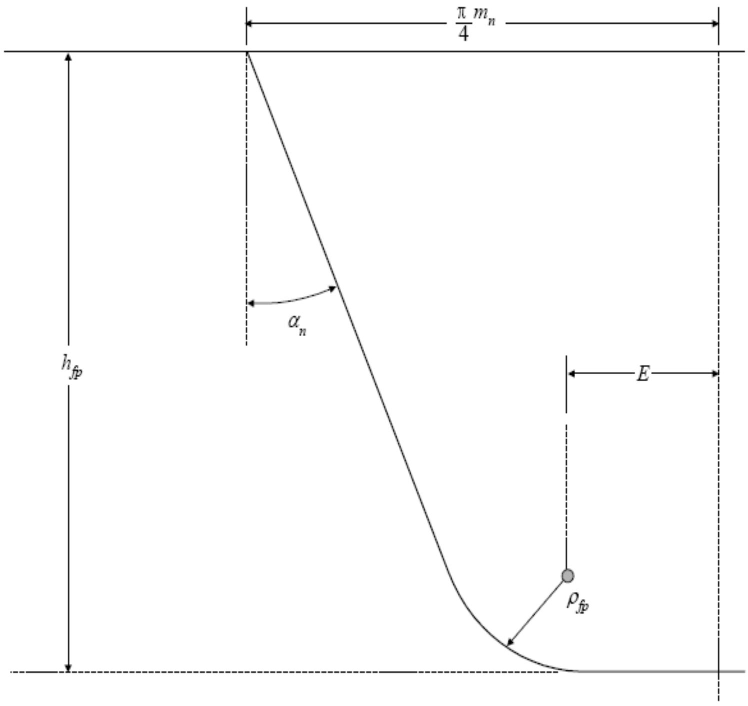
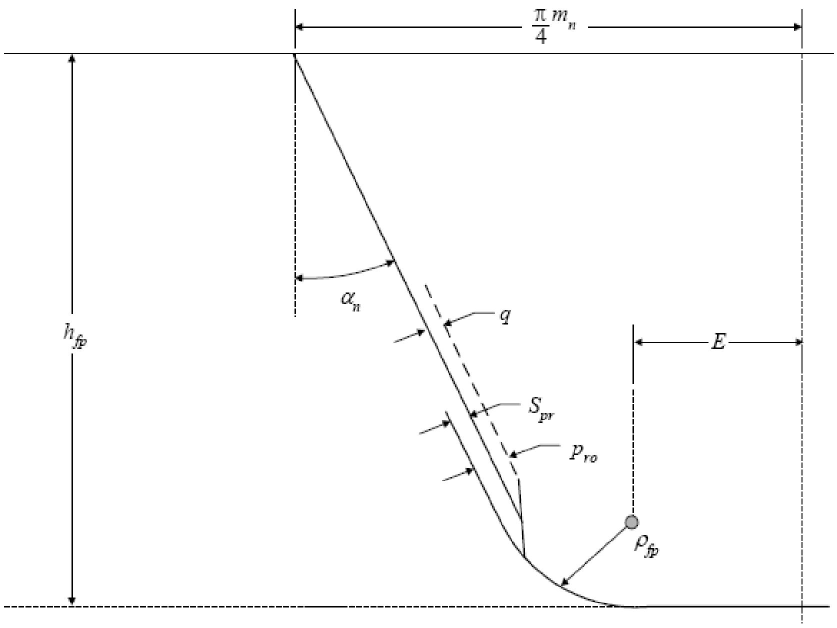


FIGURA F.H3.404.3 - TOOTH BASIC PROFILE



E , h_{a0} , α_n , Spr , p_{r0}, q e ρ_{a0} are shown in figure F.H3.404.2.

h_{a0} = adendum to the tool; mm

S_{pr} = fillet undercut; mm

p_{r0} = tool protuberance; mm

q = material tolerances for finish machining; mm

ρ_{a0} = tool tip radius; mm

z_n = virtual number of teeth

x = adendum modification coefficient
α_{Fen} = load direction angle relevant todirection application of the load at the highest point of the gearing of a single pair tooth contact of virtual spur gears

$\alpha_{Fen} = \alpha_{en} - \gamma_e$ in degrees

α_{en} = for factor pressure angle, pressur angle at the outer point of single pair tooth contat of virtual spur gears

$\alpha_{en} = \arccos(d_{bn} / d_{en})$ in degrees

Bending moment arm h_F, in mm:

$$\frac{h_F}{m_n} = \frac{1}{2} * \left\{ \left(\cos\gamma_e - \sin\gamma_e * \tan\alpha_{Fem}\right) * \frac{d_{em}}{m_n} - z_n * \cos\left[\frac{\pi}{3} - \upsilon\right] - \frac{G}{\cos\upsilon} + \frac{\rho_{a0}}{m_n} \right\}$$

ρ_F : Fillet radius at contact point of 30° tangent, in mm

$$\frac{\rho_F}{m_n} = \frac{\rho_{a0}}{m_n} + \frac{2 * G^2}{\cos\upsilon * (z_n * \cos^2\upsilon - 2 * G)}$$

γ_e: Parameter defined by:

$$\gamma_e = \left[\frac{0{,}5 * \pi + 2 * x * \tan\alpha_n}{z_n} + \mathrm{inv}\alpha_n - \mathrm{inv}\alpha_{en} \right] * \frac{180}{\pi} \quad \text{degrees}$$

Determination of bending moment arm h_F

$d_{an1} = d_{n1} + d_{a1} - d_1$ mm
$d_{an2} = d_{n2} + d_{a2} - d_2$ mm
$d_{n1,2} = z_{n1,2} *+ m_n$ mm
$d_{bn1,2} = d_{n1,2} * \cos\alpha_n$ mm

$$d_{en1} = \frac{2 * z_1}{|z_1|} * \sqrt{ \left\{ \left[\frac{d_{an1}}{2}\right]^2 - \left[\frac{d_{bn1}}{2}\right]^2 - \frac{\pi * d_1 * \cos\beta * \cos\alpha_n}{|z_1|} * (\varepsilon_{\alpha n} - 1) + \frac{d_{bn1}}{2} \right\}^2 + \left[\frac{d_{bn1}}{2}\right]^2 } \quad \text{mm}$$

$$d_{en2} = \frac{2 * z_2}{|z_2|} * \sqrt{ \left\{ \left[\frac{d_{an2}}{2}\right]^2 - \left[\frac{d_{bn2}}{2}\right]^2 - \frac{\pi * d_2 * \cos\beta * \cos\alpha_n}{|z_2|} * (\varepsilon_{\alpha n} - 1) + \frac{d_{bn1}}{2} \right\}^2 + \left[\frac{d_{bn1}}{2}\right]^2 } \quad \text{mm}$$

$\varepsilon_{\alpha n} = (\varepsilon_\alpha / \cos^2\beta_b)$

For bevel gears:

Tooth root chord at the critical section, s_{Fn}

$$\frac{s_{Fn}}{M_{mn}} = z_{vn} * \sin\left(\frac{\pi}{3} - \upsilon\right) + \sqrt{3} * \left(\frac{G}{\cos\upsilon} - \frac{\rho_{a0}}{m_{mn}}\right)$$

$$\upsilon = 2 * \frac{G}{z_{vn}} * \tan\upsilon - H$$ in degrees to be calculated by interaction

G: Parameter defined by:

$$G = \frac{\rho_{a0}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm}$$

H: Parameter defined by:

$$H = \frac{2}{z_{vn}} * \left[\frac{\pi}{2} - \frac{E}{m_{mn}}\right] - \frac{\pi}{3}$$

E: Parameter defined by:

$$E = \frac{\pi}{4} * m_{mn} - x_{mn} * m_{mn} - h_{a0} * \tan\alpha_n \frac{S_{pr}}{\cos\alpha_n} * (1 - \sin\alpha_n) * \frac{\rho_{a0}}{\cos\alpha_n}$$

S_{pr}: Residual fillet undercut, in mm

$S_{pr} = p_{r0} - q$

***E* , h_{a0} , α_n , *Spr* , p_{r0}, *q* e ρ_{a0}** are shown in the figure F.H3.404.2.

h_{a0} = adendum of the tool; mm

S_{pr} = residual cam left by the protuberance; mm

p_{r0} = tool protuberance; mm

q = tolerances of material to the finishing of machine; mm

ρ_{a0} = tool tip radius; mm

z_{vn} = numero virtual of teeth

x_{hm} = coeficiente of modificação perfil

x_{sm} = coeficiente of modificação of the espessura of the tooth (no meio of the face)

α_{Fan} = angle to application of the load at the point highest of the gearing of a single pair of teeth

α_{an} = angle of pressure at the point highest of the gearing of a single pair of teeth

Bending moment arm h_{FA}:

$$\frac{h_{Fa}}{m_{mn}} = \frac{1}{2} * \left\{ \left(\cos\gamma_a - \sin\gamma a_e * \tan\alpha_{Fan} \right) * \frac{d_{van}}{m_{mn}} - z_{vn} * \cos\left[\frac{\pi}{3} - \upsilon \right] - \frac{G}{\cos\upsilon} + \frac{\rho_{a0}}{m_{mn}} \right\}$$

ρ_F : **Fillet radius at contact point of 30° tangent, in mm** (see figure F.H3.404.1).

$$\frac{\rho_F}{m_{mn}} = \frac{\rho_{a0}}{m_{mn}} + \frac{2 * G^2}{\cos \upsilon * (z_{vn} * \cos^2 \upsilon - 2 * G)}$$

Normal load pressure at the tooth tip

$\alpha_{Fan} = \alpha_{an} - \gamma_a$

$\alpha_{an} = \arccos(a_{vbn}/d_{van})$ degrees

γ_a: Parameter defined by:

$$\gamma_a = \left[\frac{0{,}5 * \pi * 2 * (xhm * \tan\alpha_n + x_{sm})}{z_{vn}} + \text{inv}\,\alpha_n - \text{inv}\,\alpha_{an} \quad * \right] * \frac{180}{\pi} \text{degrees}$$

$\beta_{bm} = \arccos \sqrt{1-(\sin_{\beta m} * \cos \alpha_n)^2}$ degrees

$d_{van1} = d_{vn1} + d_{va1} - d_{v1}$ mm
$d_{van2} = d_{vn2} + d_{va2} - d_{v2}$ mm
$d_{an2} = d_{n2} + d_{a2} - d_2$ mm
$d_{vn1,2} = z_{vn1,2} * m_n$ mm
$d_{vbn1,2} = d_{vn1,2} * \cos\alpha_n$ mm

405. Stress correction factor Y_S

The stress correction factor, YS, 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.

YS shall be determined separately for the pinion and for the wheel.

The stress correction factor, YS, can be determined with the following equation (having range of validity: $1 \leq qs < 8$):

For spur gears:

$Y_S = (1{,}2 + 0{,}13)\ qs^{\ (1/(1{,}1 + 2{,}3/L)}$

where:

$qs = s_{Fn}/2*\rho_F$

qs = notch parameter

$\rho_{F\ =}$ root fillet radius in the critical section

$L = s_{Fn}/h_f$

For bevel gears:

$Y_{Sa} = (1{,}2 + 0{,}13)\ qs^{\ (1/(1{,}1 + 2{,}3/La)}$

$qs = s_{Fn}/2*\rho_F$

where:

qs = notch parameter

$\boldsymbol{\rho_F}$ = Fillet radius at contact point of 30° tangent, in mm (see figure F.H3.404.1).

$L = s_{Fn}\ /\ h_F$ for spur gears

$L_a = s_{Fn}\ /\ h_{Fa}$ for bevel gears

h_{Fm} h_{Fa}, s_{Fn}, ρ_F see figure F.H3.404.1 - Section normal of a tooth (to calculate $h_{f,}$, s_{Fn}, α_{Fen})

406. Helix angle factor Y_β

The helix angle factor, Y_β, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth.

$Y_\beta = 1 - \varepsilon_\beta*(\beta/120)$

where:

β is the helix reference angle in degrees.

When $\varepsilon_\beta > 1{,}0$, adopt = 1,0, and adopt na angle of 30° when $\beta > 30°$.

407. Bevel gear factor Y_K

$Y_K = (1/2) + (b/4*l'_b) + (l'_b/4*b)$

$l'_b = l_b*\cos\beta_{bm}$

407. Load distribution factor Y_{LS}

The load distribution factor Y_{LS} takes into account the difference between two or more pairs of teeth.

$Y_{LS} = Z_{LS}^2 \geq 0{,}7$

For Z_{LS} see item H3.309 above.

408. Limit of bending strength σ_{FE}

For a given material, σ_{FE} is the local tooth root stress which can be permanently endured.

According to the reference standard the number of 3 x 106 cycles is regarded as the beginning of the endurance limit.

σ_{FE} is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment).

Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d.

Other conditions such as pre-stress or alternating stress will be covered by factor Y_d.

For the values of σ_{FE} see notes of the table T.H3.210.1.

The sFE values are to correspond to a failure probability 1% or less.

The endurance limit mainly depends on:

a. material composition, cleanliness and defects;

b. mechanical properties;

c. residual stresses;

d. hardening process, depth of hardened zone, hardness gradient;

e. material structure (forged, rolled bar, cast). The bending endurance limit, σ_{FE} can be determined, in general, making reference to values indicated in ISO 6336/5, quality MQ.

409. Design factor Y_d

The design factor, Yd, takes into account the influence of load reversing and shrinkfit prestressing on the tooth root strength, relative to the tooth root strength with unidirectional load as defined for σ_{FE}.

Y_d = 1,00 in general;

Y_d = 0,90 for gears with occasional part load in reversed direction, such as main wheel in reversing gearboxes;

Y_d = 0,70 for idler gears.

410. **Life factor Y_N**

The life factor takes into account the higher tooth root bending stress permissible in case a limited life (number of cycles) is required.

Y_N é calculated by the table T. H3.410.1 below.

TABLE T. H3.410.1 – LIFE FACTOR *Y_N*

Material	*Number of load cycles*	*Life factor Y_N*
V GGG (perlitic, bainitic) GTS (perlitic)	$N_L \leq 10^3$ estático	2,5
	N_L = 3 x 106	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufatcuring, lubrification and experience	1,0
Eh, If	$N_L \leq 10^3$ estático	2,5
	N_L = 3 x 106	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufatcuring, lubrification and experience	1,0
GG, GGG (perlitic) NT (nitritided) NV (nitritided) St	$N_L \leq 10^3$ estático	1,6
	N_L = 3 x 106	1,0
	$N_L = 10^{10}$	0,85
	Excelent material, manufatcuring, lubrification and experience	1,0
NV (nitrocarbided)	$N_L \leq 10^3$ static	1,0
	N_L = 3 x 106	1,0
	$N_L = 10^{10}$	0,85
	Excellent material, manufacturing, lubrication and experience	1,0

1) Abbreviations

St normalized basic steel σ_B < 800 N/mm2
V steel forged special with core hardening, carbon or alloy with σ_B ≥ 800 N/mm2
GGG nodular cast iron (perlitic., bainitic, ferrític struture),
GTS (black malleable cast iron pearlitic structure).
GG Gray cast iron
Eh Forged steels with surface hardening (case hardened)
NT Gas nitrited steel
NV (nitrited) Steel nitrited, hardened, temperated and gas nitrited
NV (nitro carbided) Steel hardened, tempered, nitro carbided

2) NL = n*60*HPD*DPY*YRS
N = rotation (rpm)
HPD = hours of operation by day
DPY = days per year
YRS = years (normal service life of the ship = 25 years)

Note: *Above table according to Standard ISO 6336-3;see Standard ISO 6336-5 in reference to the specifications and procedures of various heat treatments*

411. **Relative notch sensitivity factor, $Y_{\delta relT}$**

The relative notch sensitivity factor, $Y_{\delta re1T}$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit.

The test relative sensitivity factor $Y_{\delta relT}$ can be determined as follows:

a. For notch parameters included in the range 1.5 < qs < 4, it may be assumed:

$Y_{\delta relT} = 1{,}0$

b. For notch parameters included in the range qs < 1.5, it may be assumed:

$Y_{\delta relT} = 0{,}95$

c. For notch parameters included outside the above ranges, $Y_{\delta relT}$ may be determined according to the Standard ISO 6336-3.

412. **Relative surface finishing factor**

The surface finishing relative factor of takes into account the dependence of the tooth root strength and the surface condition in the tooth corner, especially the dependence of the surface roughness peak to valley.

The factor Y_{RrelT} is to be calculated by table T.H3.412.1 below.

TABLE T.H3.412.1 – FACTOR Y_{RrelT}

	1 ≤ Rz ≤ 40	Rz < 1
Case hardened steels through - hardened steels (sB 3 800 N/mm2)	$Y_{R\,rel\,T} = 1{,}675 - 0{,}53\,(Rz + 1)^{0,1}$	1,120
For nitrided steels:	$Y_{R\,rel\,T} = 4{,}3 - 3{,}26\,(Rz + 1)^{0,005}$	1,025
For normalized steels (σ_b < 800 N/mm^2)	$Y_{R\,rel\,T} = 5{,}3 - 4{,}2\,(Rz + 1)^{0,001}$	1,070

where:

R_z = mean peak to-valley roughness of tooth root fillets

σ_b = tensile stress in N/mm^2

The method applied here is valid only when scratches or similar defects that have depth less than 2*Rz.

If the roughness stated is an Ra value (= CLA value) (= AA value) the following approximate relationship can be applied:

Ra = CLA = AA = Rzr /6

413. **Size factor Y_x**

The size factor, YX, takes into account the decrease of the strength with increasing size.

The factor Y_x can be determined by table H3.413.1 below

TABLE T.H3.413.1 – FACTOR Y_x

$Y_x = 1{,}00$	For $m_n < 5$	Generally
Y_x = 1,03-0,06 mm	For 5 < m_n < 30	Normalized and through-hardened steels
$Y_x = 0{,}85$	FOR $m_n \geq 30$	
Y_x = 1,05 – 0,010 mm	For 5 < m_n < 25	Surface hardened steels
$Y_x = 0{,}80$	For $m_n \geq 25$	

414. **Safety factor for tooth root bending stress, SF_{min}**

The safety factor for tooth root bending stress, S_F, can be assumed by the RBNA taking into account the type of application.

The guidance values in Table T.H3.318.1 above can be adopted.

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 RBNA.

H4. GEARS

100. Shafts

101. The shaft diameter of the gears in the region of the bearings should not be smaller than the diameter required for the intermediate shaft plus:

a. 10% where the wheel is driven by two pinions about the 180°, and

b. 15% where the wheel is driven by only one or two pinion gears about the 120°.

102. The shaft materials are to comply with the Part III, Title 62, chapter C1 of these Rules.

200. Teeth

201. The teeth should be designed to withstand a load corresponding to the maximum linear torque transmitted by the gearbox when in continuous service.

See Subchapter H3 for the design of the gears.

202. The hardness of the pinion teeth is to be at least 20% higher than the hardness of the corresponding teeth of the gear.

See Part III, Title 62, Chapter C14 of these Rules concerning to materials to the gears.

203. In general the finish of the surface of the teeth is not to present roughness greater than 5.1 μm, in arithmetic average or center.

However, gears with power less than 3728 kW (5000 hp) and roughness greater than 1.5 μm may be specially considered taking into account the lubricant recommended by the manufacturer.

300. Gear casings

301. The gearbox casings are to be of robust construction in order to minimize elastic deflections and maintain the accuracy of assembly of the gears. They are to be designed to withstand operating conditions without presenting harmful deflections ruling:

a. elastic loads;

b. forces generated by the energy transmitted, and

c. inertial effects of gears inside the casings due to dynamic forces of the vessel, due to horizontal accelerations of 1 g and vertical 2g. Where g = acceleration of gravity.

302. The construction of the casing is to be made so to consider a sufficient number of points for suitable inspection of the gears for checking the contact of the teeth and measuring the clearance of thrust bearings. Alternative methods for special inspection openings can be analyzed by RBNA.

400. Balancing

401. The gearboxes shafts, gears and pinions are to be statically and dynamically balanced.

402. For gearboxes where the speed does not exceed 150 RPM only static balancing is required .

500. Fittings

501. The gearboxes are to be provided of suitable instruments to check:

a. level of oil;

b. temperature of the oil, and

c. pressure of the oil.

502. Easy access to the lube oil pumps, when coupled to the gearbox, is to be provided for operation and maintenance.

503. For roller bearings the maximum allowable pressure, minimum thickness of the layer of oil, estimated maximum internal temperature and maximum static load are to be according to the applicable industry standards.

504. For ball bearings, the service life is not to be less than 20,000 hours for AV drivers and 5000 hours for AR drivers. The calculations are to be according to the applicable industrial standards.

H5. COUPLINGS

100. Gear couplings

101. The teeth are to be effectively lubricated. Small couplings may be lubricated by splashing.

102. For large couplings or propulsion main engine couplings forced lubrication are to be used.

200. Flexible couplings

201. The flexible couplings are to be properly dimensioned, so that their static moment of rupture be equal to or greater than eight times the moment at the coupled elements,

202. If, when in operation, the flexible is coupling causing axial thrust on the coupled elements, it is to be designed means to absorb this thrust.

203. Flexible couplings groups to Diesel generators are to be sized to absorb surges of torque caused by short circuit.

CHAPTER I
PROPELLERS

CHAPTER CONTENTS

I1. APPROACH

100. Application

101. The present Chapter I apply for propulsion systems fitted with propellers.

102. Other types of propulsion systems will be subject to special design analysis for approval.

103. The data and details required to verify of the design of propellers, as well as the characteristics of the material used in its manufacture, are to be submitted to RBNA for approva.

104. The manufacture of the propeller shall be supervised by RBNA. See Part III of the Rules.

I2. COMMON PROPELLER DIMENSIONING AND CONSTRUCTION

100. Blade thickness

101. The blade thickness will be assessed by RBNA.

102. Manganese bronze propellers dimensioned by systematic series Troost, Kaplan and Schaffran, generally have thicknesses that meet requirements of these Rules

103. The blades, boss and all the external surfaces of the propeller are to be well finished and polished.

200. Propeller key

201. The key is to have an accurate fit in the boss. Where keyless propellers are employed, the instructions for the fitting and the detailed calculations of the stresses should be submitted to the verification of RBNA.

300. Fitting to shaft

301. See Part I, Títle 02, Section 2, Subchapter C6, item 600 of the present Rules.

I3. REMOVABLE BLADES PROPELLER

100. Assembly

101. The face of the flange should be supported entirely by the boss and gaps should be minimized.

102. The dimensioning of the bolt attachment or other means is to be compatible with the blade root's strength at the setting to the boss

I4. CONTROLLABLE PITCH PROPELLER

100. Particulars

101. The particulars of variable pitch control systems are to be submitted to RBNA for approval.

200. Hydraulic pitch control

201. Where the pitch adjustment mechanism is hydraulically operated two hydraulic pumps driven by independent systems are to be installed.

202. For power installations up to 149 kW (200 bhp), one of the pumps can be manually operated, provided that the time to move the blades in the fore to aft position is within the range of ten (10) seconds.

300. Indicators

301. The controllable pitch system is to be provided with an indicator of the position of the blades fitted in the engine room, bridge and control station.

302. The dimensioning of the attachment by bolts or by other means is to be compatible with the blade root's strength at the boss.

400. Pitch control in emergency

401. A device shall be provided for pitch control in emergencies.

I5. BALANCING

100. Control

101. The propellers are to be statically balanced. The residual unbalance is to be such that the centrifugal force resulting in the service rotation does not exceed 2% of the weight of the propeller.

I6. PROTECTION AGAINST CORROSION

100. Contact propeller x shaft

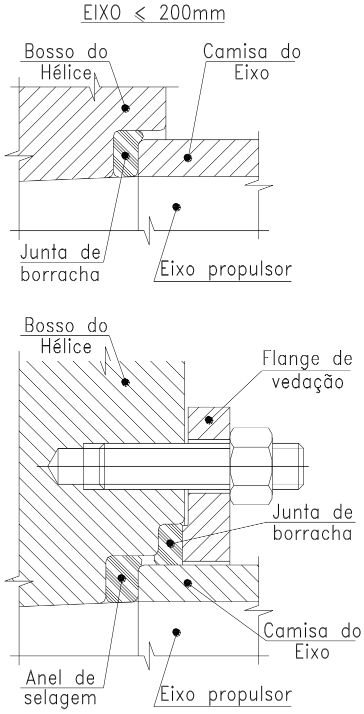
101. The steel parts of the shaft, unprotected, should have all the spaces between the protection of nut, boss, the propeller hub and shaft filled with sebum or mass of red lead or other suitable corrosion-resistant material to prevent water ingress.

102. The contact of bronze with steel in the presence of water is to be avoided.

103. A well fitted ring of soft rubber is to be provided in the forward end of the propeller. When the rubber ring is mounted externally, the recess of the boss is to be filled with material insoluble in water and non-corrosive and the gaps are to be the minimum possible.

104. When the rubber ring is mounted internally, an adequate gap between the liner and boss and the ring is to be maintained and the ring to have its dimenions enlarged in order to be mounted with a tight fit in the empty space, when the propeller is tightened against the shaft. See Figure F.I6.104.1 – Propeller

FIGURE F.I6.104.1 – PROPELLER



I7. KEYLESS FITTING OF PROPELLERS WITHOUT ICE STRENGTHENING [IACS UR K3]

100. Requirements to be satisfied [IACS UR K3.1]

101. The formulae, etc., given herein are not applicable for propellers where a sleeve is introduced between shaft and boss.

102. The taper of the propeller shaft cone should not exceed 1/15.

103. Prior to final pull-up, the contact area between the mating surfaces is to be checked and should not be less than 70% of the theoretical contact area (100%). Non-contact bands extending circumferentially around the boss or over the full length of the boss are not acceptable.

104. After final pull-up, the propeller is to be secured by a nut on the propeller shaft. The nut should be secured to the shaft.

105. The factor of safety against friction slip at 35°C is not to be less than 2,8 under the action of rated torque (based on rated power r.p.m.) plus torque due to torsionals as defined in I7.400 below.

106. For the oil injection method the coefficient of friction should be 0,13 for bosses made in copper-based alloy and steel.

107. The maximum equivalent uniaxial stress in the boss at 0°C based on the Mises-Hencky criterion(σE) should not exceed 70% of the yield point or 0.2% proof-stress (0,2%

offset yield strength) for the propeller material based on the test piece value. For cast iron the value should not exceed 30% of the nominal tensile strength.

200. Material constants [IACS UR K3.2]

201. Modulus of elasticity

a. Cast and forged steel 2,1 x 104 kgf/mm2

b. Cast iron 1,0 x 104 kgf/mm2

c. Copper based alloys,

d. Cu 1 and Cu 2 1,1 x 104 kgf/mm2

e. Copper based alloys, Cu 3 and Cu 4 1,2 x 104 kgf/mm2

202. Poisson's ratio

a. Cast and forged steel 0,29

b. Cast iron 0,26

c. All copper-based alloys 0,33

203. Coefficient of linear expansion

a. Cast and forged steel and cast iron 12,0 x 10-6 mm/mm°C

b. All copper-based alloys 17,5 x 10-6 mm/mm°C

300. Formulae used [IACS K3.3]

301. The formulae given below, for the ahead condition, will also give sufficient safety in the astern condition.

302. The formulae are applicable for soild shafts only.

303. Minimum required surface pressure at 35°:

$$P_{35} = \frac{ST}{AB}\left[-S\theta + \sqrt{\mu^2 + B\left(\frac{F_v}{T}\right)^2}\right]$$

where $B = \mu^2 - S^2\,\theta^2$

a. Corresponding minimum pull-up length at 35°C

$$\delta_{35} = P_{35}\frac{D_s}{2\theta}\left[\frac{1}{E_b}\left(\frac{K^2+1}{K2-1}+\nu_b\right)+\frac{1}{E_s}(1-\nu_s)\right]$$

304. Minimum pull-up length at temperature t (t < 35°C)

$$\delta_1 = \delta_{35} + \frac{D_s}{2\theta}(\alpha_b - \alpha_s)(35-t)$$

a. Corresponding minimum surface pressure at temperature t

$$P_t = P_{35}\frac{\delta_t}{\delta_{35}}$$

b. Minimum push-up load at temperature t

$$W_t = AP_t(\mu+\theta)$$

305. Maximum permissible surface pressure at 0°C

$$P_{max} = \frac{0{,}7\sigma_y(K^2-1)}{\sqrt{3K^4+1}}$$

a. Corresponding maximum permissible pull-up length at 0°C

$$\delta_{max} = \frac{P_{max}}{P_{35}}\delta_{35}$$

306. Shear force at interface

$$F_v = \frac{2cQ}{D_s}$$

307. Rated thrust developed for free running vessels (if not given)

$$T = 132\frac{H}{V_s} \quad or \quad T = 4{,}3\bullet 10^6\,\frac{H}{\mathrm{PN}}$$

400. Nomenclature [IACS K3.4]

401. Nomenclature for the formulas in this Chapter

A = 100% theoretical contact area (mm2) between boss and shaft, as read from drawings and disregarding oil grooves

Ds = diameter (mm) of propeller shaft at the midpoint of the taper in the axial direction

Db = mean outer diameter (mm) of propeller boss at the axial position corresponding to Ds

K =Db/Ds

Fv = shear force at interface = 2cQ/Ds (kgf)

Q = rated torque (kgf•mm) transmitted according to rated horsepower, H, and speed of propeller shaft

T = rated thrust (kgf)

c = constant,

c = 1,0 for turbines, geared diesel drives, electric drives and for direct diesel drives with a hydraulic or an electromagnetic or high elasticity coupling

c = 1,2 for a direct diesel drive.
The RBNA reserves the right to increase the c constant if the shrinkage has to absorb an extreme high pulsating torque.

H = rated brake horsepower (PS)

P = mean propeller pitch (mm)

N = propeller speed (r.p.m.) at rated brake horsepower

Vs = ship speed (knots) at rated horsepower

S = factor of safety against friction slip at 35°C

θ = half taper of propeller shaft, e.g. taper = 1/15. θ = 1/30

μ = coefficient of friction between mating surfaces

P35 = surface pressure (kgf/mm2) between mating surfaces at 35°C

Pt= surface pressure (kgf/mm2) between mating surfaces at temperature t°C

P0 = surface pressure (kgf/mm2) between mating surfaces at temperature 0°C

Pmax = maximum allowable surface pressure (kgf/mm2) at 0°C

δ35 = pull-up length (mm) at temperature 35°C

δt= pull-up length (mm) at temperature t°C

δmax = maximum allowable pull-up length (mm) at temperature 0°C

Wt = push-up load (kgf) at temperature t°C

σE = equivalent uniaxial stress (kgf/mm2) in the boss according to the Mises-Hencky criterion

αs = coefficient of linear expansion (mm/mm°C) of shaft material

αb = coefficient of linear expansion (mm/mm°C) of boss material

Es = modulus of elasticity (kgf/mm2) of shaft material

Eb = modulus of elasticity (kgf/mm2) of boss material

vs = Poisson’s ratio for shaft material

vb = Poisson’s ratio for boss material

σy = yield point or 0,2% proof stress (0,2% offset yield strength) of propeller material (kgf/mm2)

CHAPTER J
CALCULATION OF CRANKSHAFTS FOR IC ENGINES
[IACS UR M53]

CHAPTER CONTENTS

J1. APPROACH [IACS UR M53.1]

100. Scope

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

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

200. Field of application

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

300. Principles of calculation

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

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

a. fillet transitions between the crankpin and web as well as between the journal and web,

b. outlets of crankpin oil bores.

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

304. 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, 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 show whether or not the crankshaft concerned is dimensioned adequately.

400. Drawings and particulars to be submitted

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

a. crankshaft drawing (which must contain all data in respect of the geometrical configurations of the crankshaft)

b. 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)

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

d. number of cylinders

e. rated power [kW]

f. rated engine speed [r/min]

g. direction of rotation (see. Figure F.J1.401.1)

firing order with the respective ignition intervals and, where necessary,

h. V-angle αv [°] (see Figure F.J1.401.1)

i. cylinder diameter [mm]

j. stroke [mm]

k. maximum net cylinder pressure Pmax [bar]

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

m. connecting-rod length LH [mm]

n. all individual reciprocating masses acting on one crank [kg]

o. digitized gas pressure curve presented at equidistant intervals [bar versus Crank Angle] (at least every 5° CA) • for engines with articulated-type connecting-rod (see F.J1.401.2)

p. distance to link point LA [mm]

q. link angle αN [°]

r. connecting-rod length LN [mm]

s. cylinder diameter [mm]

t. stroke [mm]

u. maximum net cylinder pressure Pmax [bar]

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

w. connecting-rod length LH [mm]

x. all individual reciprocating masses acting on one crank [kg] intervals [bar versus Crank Angle] (at least every 5° CA)

y. for engines with articulated-type connecting-rod (see fig. FJ1.401.2)

z. distance to link point LA [mm]

aa. link angle αN [°]

bb. connecting-rod length LN [mm]

cc. details of crankshaft material

dd. material designation (according to ISO,EN,DIN, AISI, etc..)

ee. mechanical properties of material (minimum values obtained from longitudinal test specimens)

ff. tensile strength [N/mm²]

gg. yield strength [N/mm²]

hh. reduction in area at break [%]

ii. elongation A5 [%]

jj. impact energy – KV [J]

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

ll. Every surface treatment affecting fillets or oil holes shall be subject to special consideration

mm. Particulars of alternating torsional stress calculations, see item J3.200 below..

FIGURE F.J1.401.1 – DESIGNATION OF THE CYLINDERS

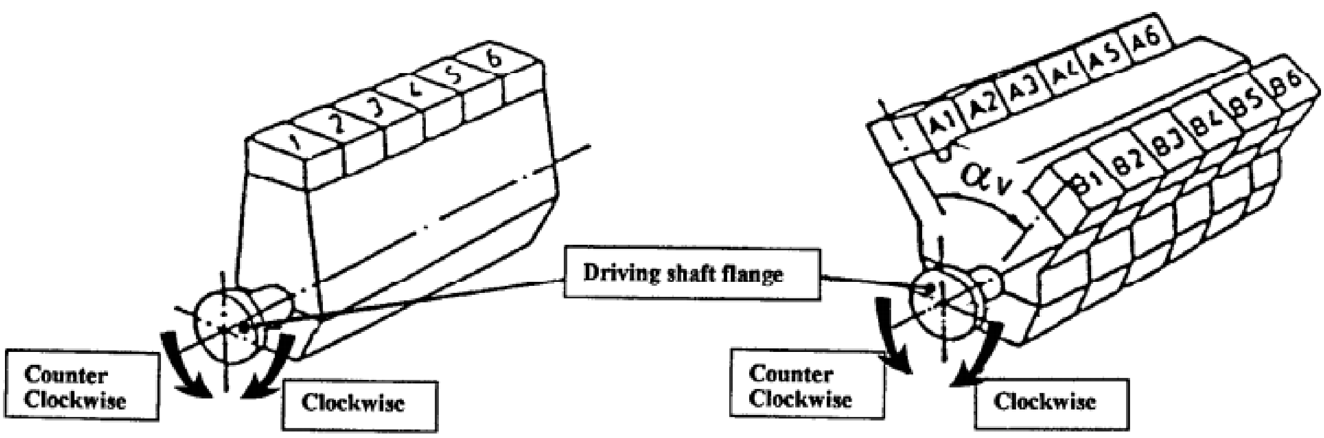


FIGURE F.J1.401.2 – ARTICULATED-TYPE CONNECTING-ROD

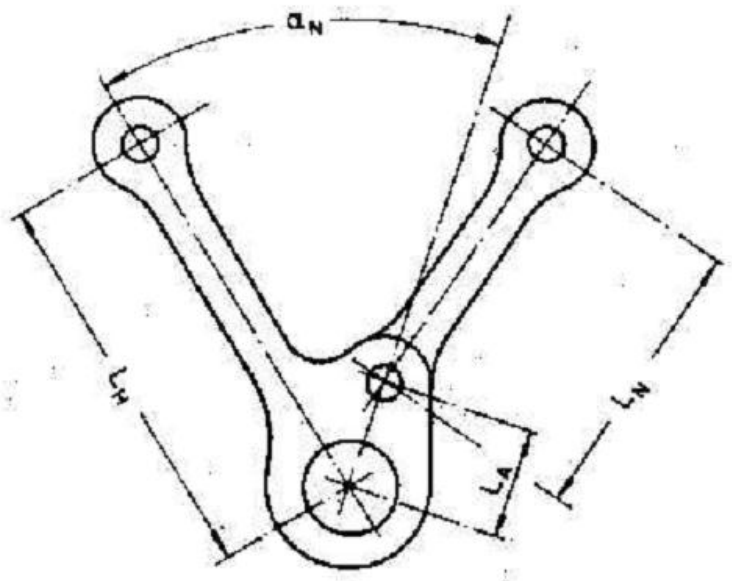


FIGURE F.J1.401.3 – CRANKTHROW FOR LINE ENGINE

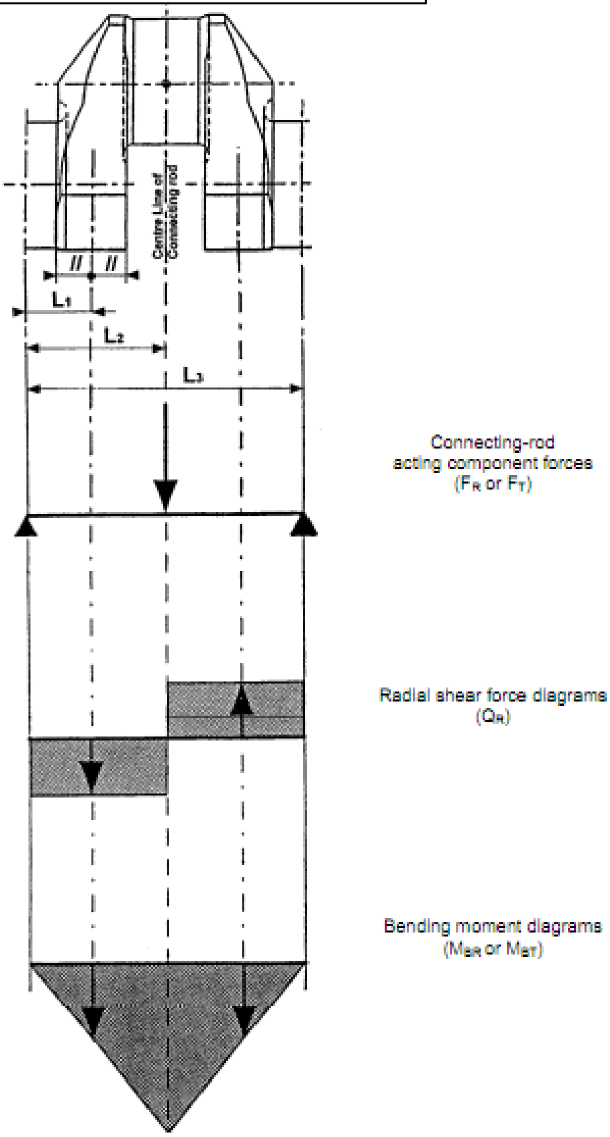


Fig. 3 Crankthrow for in line engine

FIGURE F.J2.103.1 – CRANKTHROW FOR V ENGINE

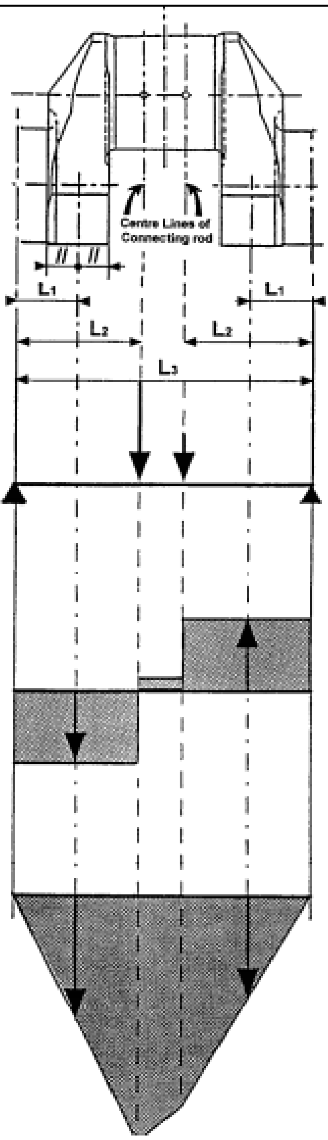


Fig. 4 Crankthrow for Vee engine

L1 = Distance between main journal centre line and crankweb centre (see also F.J2.202. for crankshaft without overlap)

L2 = Distance between main journal centre line and connecting-rod centre

L3 = Distance between two adjacent main journal centre lines

J2. CALCULATION OF ALTERNATING STRESSES DUE TO BENDING MOMENTS AND RADIAL FORCES [IACS UR M53.2 .1]

100. Assumptions [IACS UR M53.2 .1.1]

101. The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints (distance L3, see fig. F.J1.501.3 and F.J1.501.4).

102. The bending moments MBR, MBT are calculated in the relevant section based on triangular bending moment diagrams due to the radial component FR and tangential component FT of the connecting-rod force, respectively (see fig. F.J1.501.3).

103. For crankthrows with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see fig. F.J2.103.1).

200. Bending moments and radial forces acting in web [IACS UR M53.2 .1.1.1]

201. The bending moment MBRF and the radial force QRF are taken as acting in the centre of the solid web (distance L1) and are derived from the radial component of the connecting-rod force.

202. The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crank web. This reference section results from the web thickness W and the web width B (see fig. F.J2.202.1).

203. Mean stresses are neglected.

FIGURE F.J2.301.1.– CRANKPIN SECTION THROUGH THE OIL BORE

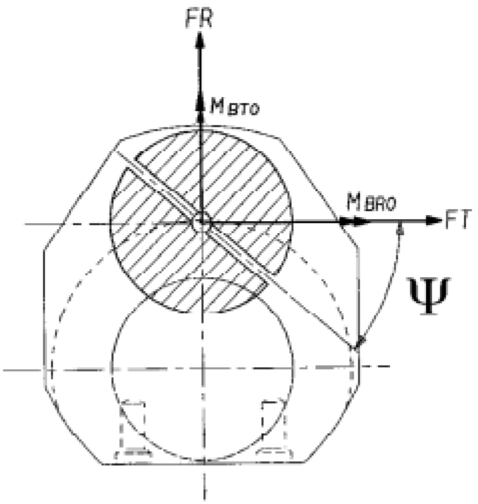
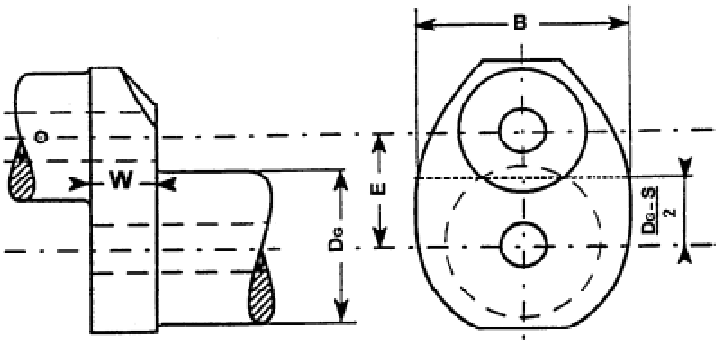
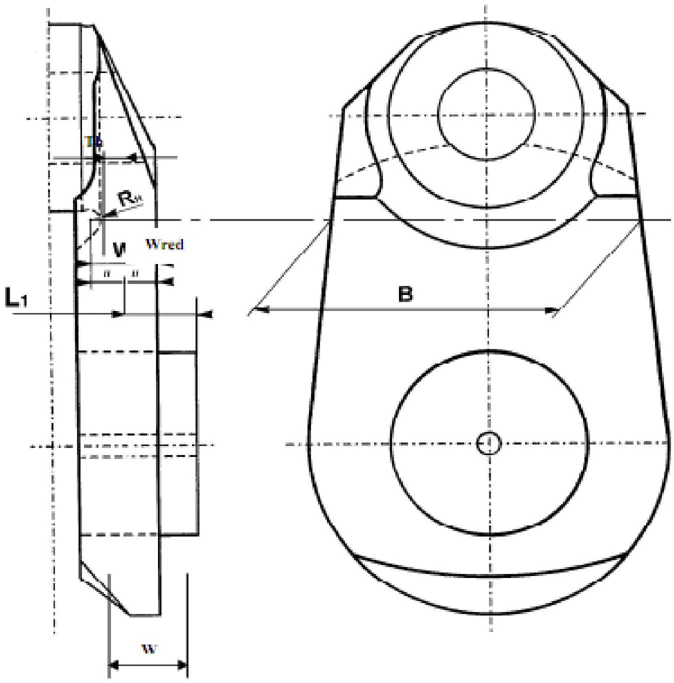


FIGURE F.J2.202.1 CRANKSHAFT OVERLAPPED AND WITHOUT OVERLAP



Overlapped crankshaft



Crankshaft without overlap

300. Bending acting in outlet of crankpin oil bore [IACS UR M53.2 .1.1.2]

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

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

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

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

303. Mean bending stresses are neglected.

400. Calculation of nominal alternating bending and compressive stresses in web [IACS UR M53.2 .1.2]

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

402. Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments M_{BRF}, M_{BRO}, M_{BTO} and radial forces Q_{RF} - as defined in J2.200 and J2.300 - will then be calculated.

403. In case of V-type engines, the bending moments - progressively calculated from the gas and inertia forces - of the two cylinders acting on one crankthrow are superposed according to phase. Different designs (forked connecting-rod, articulated-type connecting-rod or adjacent connecting-rods) shall be taken into account.

404. Where there are cranks of different geometrical configurations in one crankshaft, the calculation is to cover all crank variants.

405. The decisive alternating values will then be calculated according to:

$$Xn = \pm \tfrac{1}{2}[X_{max} - X_{min}]$$

where:

X_N is considered as alternating force, moment or stress

X_{max} is maximum value within one working cycle

X_{min} is minimum value within one working cycle

500. Nominal alternating bending and compressive stresses in web cross section [IACS UR M53.2 .1.2.1]

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

$$\sigma_{BFN} = \pm(M_{BRFN}/W_{eqw})*10^3*Ke$$

$$\sigma_{QFN} = \pm(Q_{RFN}/F) * K_e$$

where:

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

M_{BRFN} [Nm] = alternating bending moment related to the centre of the web (see figures F.J1.501.3 and F.J2.103.1)

$$M_{BRFN} = \pm \tfrac{1}{2} * (M_{BRFmax} - M_{BRFmin})$$

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

Ke = 0.8 for 2-stroke engines

Ke = 1.0 for 4-stroke engines

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

Q_{RFN} [N] = alternating radial force related to the web (see figures F.J1.501.2 and F.J1.501.3)

$$Q_{RFN} = \pm \tfrac{1}{2} * (QRFN_{max} - Q_{RFNmin}$$

F [mm²] area related to cross-section of web

$$F = B \bullet W$$

600. Nominal alternating bending stress in outlet of crankpin oil bore [IACS UR M53.2 .1.2.2]

601. The calculation of nominal alternating bending stress is as follows:

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

where:

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

M_{BON} [Nm] alternating bending moment calculated at the outlet of crankpin oil bore

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

with

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

and

ψ ° angular position (see figure F.J2.301.1)

We [mm^3] section modulus related to cross-section of axially bored crankpin

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

700. Calculation of alternating bending stresses in fillets [IACS UR M53.2 .1.3]

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

a. **For the crankpin fillet**:

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

where:

σ_{BH} [N/mm^2] alternating bending stress in crankpin fillet

α_B [-] stress concentration factor for bending in crankpin fillet (determination - see subchapter J4 below)

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

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

where:

σ_{BG} [N/mm^2] alternating bending stress in journal fillet

β_B [-] stress concentration factor for bending in journal fillet (determination - see subchapter J4 below)

β_Q [-] stress concentration factor for compression due to radial force in journal fillet (determination - see subchapter J4).

603. **Calculation of alternating bending stresses in outlet of crankpin oil bore**

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

where:

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

γ_B [-] stress concentration factor for bending in crankpin oil bore (determination - see subchapter J4)

J3. CALCULATION OF ALTERNATING TORSIONAL STRESSES [IACS UR M53.2.2]

100. General [IACS UR M53.2.2.1]

101. The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in items in subchapter J4.200.

102. The manufacturer shall specify the maximum nominal alternating torsional stress.

200. Calculation of nominal alternating torsional stresses [IACS UR M53.2.2.2]

201. The maximum and minimum torques are to be ascertained for every mass point of the complete dynamic system and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines.

202. Whilst doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring [*] in one of the cylinders). The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected.

(*) Misfiring is defined as cylinder condition when no combustion occurs but only compression cycle.

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

204. The values received from such calculation are to be submitted to RBNA.

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

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

$$M_{TN} = \pm\frac{1}{2}\left[M_{Tmax} - M_{Tmin}\right]$$

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

where:

τN [N/mm²] nominal alternating torsional stress referred to crankpin or journal

M_{TN} [Nm] maximum alternating torque

WP [mm³] polar section modulus related to cross-section of axially bored crankpin or bored journal

MT_{max} [Nm] maximum value of the torque

MT_{min} [Nm] minimum value of the torque

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

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

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

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

300. Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore [IACS UR M53.2.2.3]

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

a. For the crankpin fillet:

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

where:

τ_H [N/mm²] alternating torsional stress in crankpin fillet

α_T [-] stress concentration factor for torsion in crankpin fillet (determination - see subchapter J4)

τ_N [N/mm²] nominal alternating torsional stress related to crankpin diameter

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

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

where:

τ_G [N/mm²] alternating torsional stress in journal fillet

β_T [-] stress concentration factor for torsion in journal fillet (determination - subchapter J4)

τ_N [N/mm²] nominal alternating torsional stress related to journal diameter

c. For the outlet of crankpin oil bore:

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

where:

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

γ_T [-] stress concentration factor for torsion in outlet of crankpin oil bore (determination- see subchapter J4)

τ_N [N/mm²] nominal alternating torsional stress related to crankpin diameter

J4. EVALUATION OF STRESS CONCENTRATION FACTORS [IACS M 53. 3]

100. Approach [IACS M 53. 3.1]

101. The stress concentration factors are evaluated by means of the formulae according to items J4.200, J4.200 and J4.400 applicable to the fillets and crankpin oil bore of solid forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. It must be noticed that stress concentration factor formulae concerning the oil bore are only applicable to a radially drilled oil hole. All formulae are based on investigations of FVV (Forschungsvereinigung Verbrennungskraftmaschinen) for fillets and on investigations of ESDU (Engineering Science Data Unit) for oil holes.

103. Where the geometry of the crankshaft is outside the boundaries of the analytical stress concentration factors (SCF) the calculation method detailed in Appendix III, subchapter J13 below may be undertaken.

104. All crank dimensions necessary for the calculation of stress concentration factors are shown in Figure F.J4.104.1 7.

105. The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (VON MISES) – occurring in the fillets under bending load – to the nominal bending stress related to the web cross-section (see J11 below).

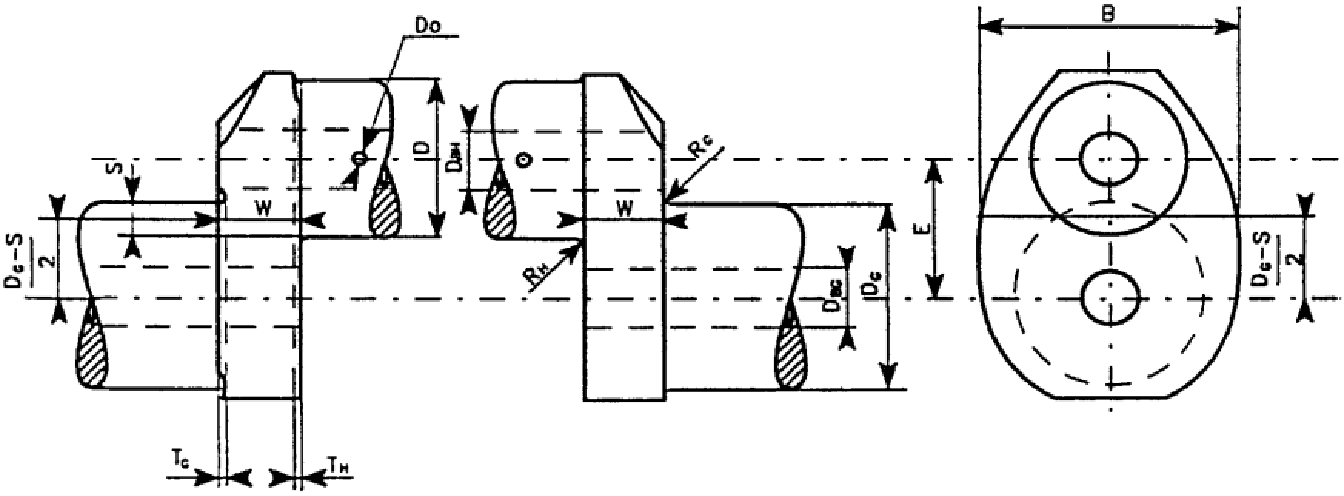
106. The stress concentration factor for compression (β_Q) in the journal fillet is defined as the ratio of the maximum equivalent stress (VON MISES) – occurring in the fillet due to the radial forc – to the nominal compressive stress related to the web cross-section.

107. The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum equivalent shear stress – occurring in the fillets under torsional load – to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see subchapter J11 - Appendix I).

108. The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress – occurring at the outlet of the crankpin oil-hole under bending and torsional loads – to the corresponding nominal stress related to the axially bored crankpin cross section (see subchapter J12 - Appendix II).

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

FIGURE F.J4.104.1 – CRANK DIMENSIONS



Actual dimensions:

D [mm] crankpin diameter

D_{BH} [mm] diameter of axial bore in crankpin

Do[mm] diameter of oil bore in crankpin

R_H [mm] fillet radius of crankpin

T_H [mm] recess of crankpin fillet

D_G [mm] journal diameter

D_{BG} [mm] diameter of axial bore in journal

R_G [mm] fillet radius of journal

T_G [mm] recess of journal fillet

E [mm] pin eccentricity

S [mm] pin overlap

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

W (*) [mm] web thickness

B (*) [mm] web width

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

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

$$Wred = W - (T_H - R_H)$$ refer to figure F.J2.202.1

web width B must be taken in way of crankpin fillet radius centre according to fig. F.J2.202.1

110. The following related dimensions will be applied for the calculation of stress concentration factors in Table T.J4.110.1:

TABLE T.J4.110.1 – RELATED DIMENSIONS FOR CALCULATION OF STRESS CONCENTRATION FACTORS

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

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

$$s \leq 0.5$$

$$0.2 \leq w \leq 0.8$$

$$1.1 \leq b \leq 2.2$$

$$0.03 \leq r \leq 0.13$$

$$0 \leq d_G \leq 0.8$$

$$0 \leq d_H \leq 0.8$$

$$0 \leq d_O \leq 0.2$$

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

If calculated f (recess) < 1 then the factor f (recess) is not to be considered (f (recess) = 1)

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

200. Crankpin fillet [IACS M 53. 3.2]

201. The stress concentration factor for bending (α_B) is:

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

where:

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

$f(w) = 2.1790 \cdot w^{0.7171}$
$f(b) = 0.6840 - 0.0077 \cdot b + 0.1473 \cdot b^2$

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

The stress concentration factor for torsion (α_T) is:

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

where:

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

300. Journal fillet (not applicable to semi-built crankshaft) [IACS M 53. 3.3]

301. The stress concentration factor for bending (β) is:
$\beta_B = 2.7146 \cdot f_B(s, w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})$

where:

$f_B(s,w) = -1.7625 + 2.9821 \cdot w - 1.5276 \cdot w^2 + (1-s) \cdot (5.1169 - 58.8089 \cdot w + 3.1391 \cdot w^2) + (1-s)^2 \cdot (-2.1567 + 2.3297 \cdot w - 1.2952 \cdot w^2)$

$f_B(w) = 2.2422 \cdot w^{0.7548}$
$f_B(b) = 0.5616 + 0.1197 \cdot b + 0.1176 \cdot b^2$

$f_B(r) = 0.1908 \cdot r^{(-0.5568)}$
$f_B(d_G) = 1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$
$f_B(d_H) = 1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$
$f(\text{recess}) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$

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

$\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$

where:

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

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

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

303. if the diameters and fillet radii of crankpin and journal are the same, the stress concentration factor for torsion (βT) is:

$$\beta_T = \alpha_T$$

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

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

where:

f (r,s), f (b) and f (w) are to be determined in accordance with item J4.200 (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter:

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

400. Outlet of crankpin oil bore [IACS M 53. 3.4]

401. The stress concentration factor for bending (γ_B) is:

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

402. The stress concentration factor for torsion (γ_T) is:

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

J5. ADDITIONAL BENDING STRESSES [IACS M 53.4]

100. Additional bending stresses [IACS M 53.4]

101. In addition to the alternating bending stresses in fillets (see item J2.700) 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 T.J5.101.1:

TABLE T.J5.101.1 – ADDITIONAL BENDING STRESS σ_{add}

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

(*) The additional stress of ± 30 N/mm² is composed of two components

a. an additional stress of ± 20 N/mm² resulting from axial vibration

b. an additional stress of ± 10 N/mm² resulting from misalignment / bedplate deformation

102. It is recommended that a value of ± 20 N/mm2 be used for the axial vibration component for assessment purposes where axial vibration calculation results of the complete dynamic system (engine/shafting/gearing/propeller) are not available. Where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

J6. CALCULATION OF EQUIVALENT ALTERNATING STRESS [IACS UR M 53.5]

100. Approach [IACS UR M 53.5.1]

101. In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a Von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (see subchapter J11 - Appendix I).

102. As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the Von Mises criterion.

Related to the crankpin diameter

103. At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see subchapter J12 - Appendix II).

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

J7. EQUIVALENT ALTERNATING STRESS [IACS UR M 53.5.2]

100. Equivalent bending stress calculation [IACS UR M 53.5.2]

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

102. For the crankpin fillet:

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

103. For the journal fillet:

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

104. For the outlet of crankpin oil bore:

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

where:

σ_v [N/mm^2] equivalent alternating stress

For other parameters see items J2.700, J3.300 and J4.400.

J8. CALCULATION OF FATIGUE STRENGTH [IACS M 53.6]

100. Calculation of fatigue stress [IACS M 53.6]

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

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

with:
$R_X = R_H$ in the fillet area
$R_X = D_O/2$ in the oil bore area
Related to the journal diamenter:

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

where:

σ_{DW} [N/mm^2] allowable fatigue strength of crankshaft
K [-] factor for different types of crankshafts without surface treatment. Values greater than 1 are only applicable to fatigue strength in fillet area.
= 1.05 for continuous grain flow forged or drop-forged crankshafts

= 1.0 for free form forged crankshafts (without continuous grain flow)

- factor for cast steel crankshafts with cold rolling treatment in fillet area

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

σB [N/mm] minimum tensile strength of crankshaft material

For other parameters see item J4.300.

102. When a surface treatment process is applied, it must be approved by RBNA.

103. These formulae are subject to the following conditions:

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

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

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

J9. ACCEPTABILITY CRITERIA [IACS M 53.7]

100. Acceptability criteria [IACS M 53.7]

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

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

where:

Q [-] acceptability factor

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

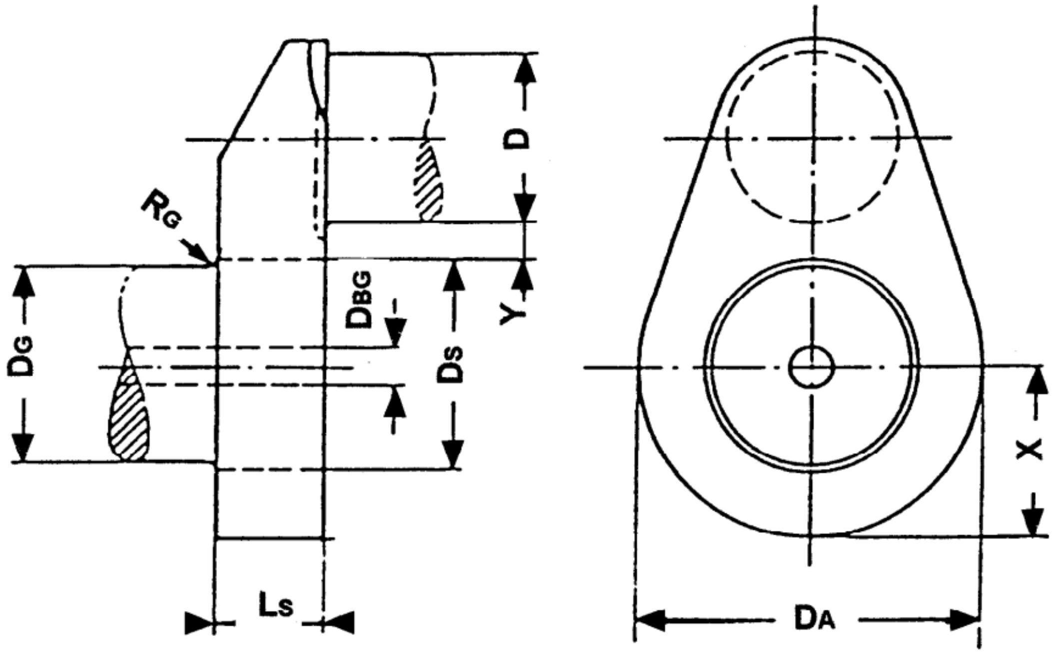
$$Q \geq 1.15$$

J10. CALCULATION OF SHRINK-FITS OF SEMI-BUILT CRANKSHAFT [IACS M 53.8]

100. Approach [M 53.8.1]

101. All crank dimensions necessary for the calculation of the shrink-fit are shown in figure J.J10.101.1.

FIGURA F.J10.101.1 CRANKTHROW OF SEMI-BUILT CRANKSHAFT



where:

D_A [mm] outside diameter of web
or twice the minimum distance x between centre-line of journals and outer contour of web, whichever is less

D_S [mm] shrink diameter
D_G [mm] journal diameter
D_{BG} [mm] diameter of axial bore in journal
L_S [mm] length of shrink-fit
R_G [mm] fillet radius of journal
y [mm] distance between the adjacent generating lines of journal and pin
$y \geq 0.05 \cdot D_S$
Where y is less than 0.1·DS special consideration is to be given to the effect of the stress due to the shrink-fit on the fatigue strength at the crankpin fillet.

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

$$R_G \geq 0.015 \cdot D_G$$

and

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

where the greater value is to be considered.

103. The actual oversize Z of the shrink-fit must be within the limits Zmin and Zmax calculated in accordance with items J10.300 and J10.400.

105. In the case where J10.200 condition cannot be fulfilled then J10.300 and J10.400 calculation methods of Zmin and Zmax are not applicable due to multizone-plasticity problems.

106. In such case Zmin and Zmax have to be established based on FEM calculations.

200. Maximum permissible hole in the journal pin [M 53.8.2]

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

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

where:

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

M_{max} [Nm] absolute maximum value of the torque MT_{max} in accordance with J3.300

μ [-] coefficient for static friction, however a value not greater than 0.2 is to be taken unless documented by experiments.

σ_{SP} [N/mm^2] minimum yield strength of material for journal pin

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

300. Necessary minimum oversize of shrink-fit [M 53.8.3]

301. The necessary minimum oversize is determined by the greater value calculated according to:

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

and

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

where:

Zmin [mm] minimum oversize
Em [N/mm²] Young's modulus
σ_{SW} [N/mm²] minimum yield strength of material for crank web

Q_A [-] web ratio, $Q_A = \frac{D_S}{D_A}$

QS [-] shaft ratio, $Q_S = \frac{D_{BG}}{D_S}$

400. Maximum permissible oversize of shrink-fit [M 53.8.4]

401. The maximum permissible oversize is calculated according to:

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

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

J11. APPENDIX 1: DEFINITION OF STRESS CONCENTRATION FACTORS IN CRANKSHAFT FILLETS

100. Definition of stress concentration factors in crankshaft fillets

101. See table T.J11.101.1 below

J12. APPENDIX II: STRESS CONCENTRATION FACTORS AND STRESS DISTRIBUTION AT THE EDGE OF OIL DRILLINGS

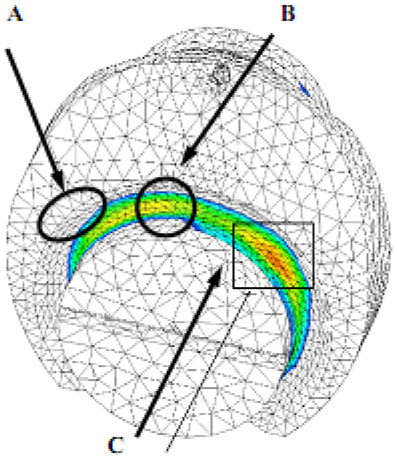
100. Stress Concentration Factors and stress distribution at the edge of oil drillings

101. See table T.J12.101.1below

TABLE T.J11.101.1

M53
(cont)

Definition of Stress Concentration Factors in crankshaft fillets **Appendix I**



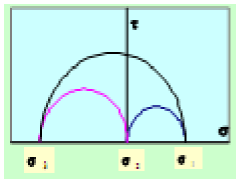
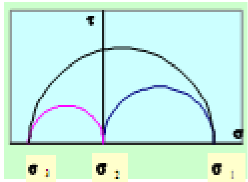
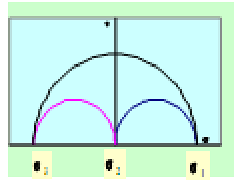
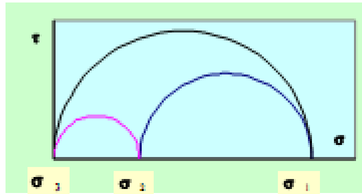
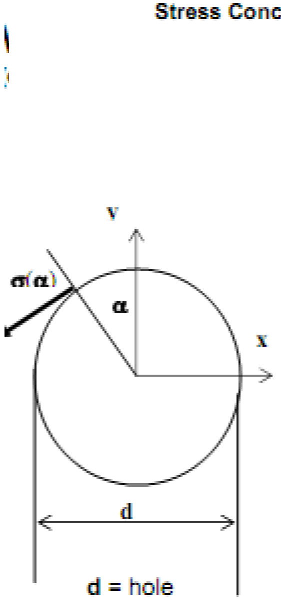
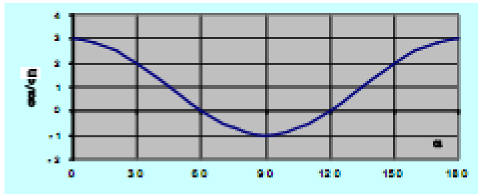
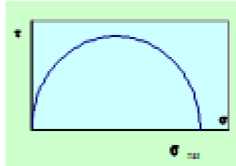
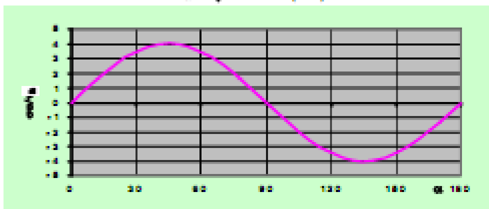
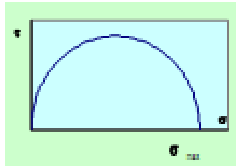
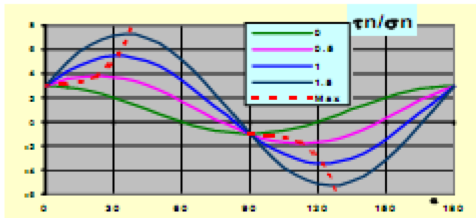
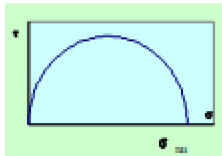
	Stress	Max $\|\|\sigma_3\|\|$	Max σ_1	
Torsional loading	Location of maximal stresses	A	C	B
	Typical principal stress system Mohr's circle diagram with $\sigma_2 = 0$	 $\|\|\sigma_3\|\| > \sigma_1$	 $\sigma_1 > \|\|\sigma_3\|\|$	 $\sigma_1 \approx \|\|\sigma_3\|\|$
	Equivalent stress and **S.C.F.**	$\tau_{equiv} = \frac{\sigma_1 - \sigma_3}{2}$ S.C.F. $= \frac{\tau_{equiv}}{\tau_n}$ for α_T, β_T		
Bending loading	Location of maximal stresses	B	B	B
	Typical principal stress system Mohr's circle diagram with $\sigma_3 = 0$	 $\sigma_2 \neq 0$		
	Equivalent stress and **S.C.F.**	$\sigma\,equiv = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 . \sigma_2}$ S.C.F. $= \frac{\sigma\,equiv}{\sigma_n}$ for $\alpha_B, \beta_B, \beta_Q$		

TABLE T.J12.101.1

Stress Concentration Factors and stress distribution at the edge of oil drillings **Appendix II**



Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha = \sigma_n \gamma_B / 3\ [1+2\cos(2\alpha)]$ 	 $\gamma_B = \sigma_{max}/\sigma_n$ for $\alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_T\ \tau_n \sin(2\alpha)$ 	 $\gamma_T = \sigma_{max}/\tau_n$ for $\alpha = \frac{\pi}{4} + k\,\frac{\pi}{2}$
Tension + shear	$\begin{bmatrix} \sigma n & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3}\sigma_n\left\{1+2\left[\cos(2\alpha)+\frac{3}{2}\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\sin(2\alpha)\right]\right\}$ 	 $\sigma_{max} = \frac{\gamma_B}{3}\sigma_n\left[1+2\sqrt{1+\frac{9}{4}\left(\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\right)^2}\right]$ for $\alpha = \frac{1}{2}tg^{-1}\left(\frac{3\gamma_T\tau_n}{2\gamma_B\sigma_n}\right)$

J13. APPENDIX 3: ALTERNATIVE METHOD FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD

100. Approach

101. The objective of the analysis is to develop Finite Element Method (FEM) calculated figures as an alternative to the analytically calculated Stress Concentration Factors (SCF) at the crankshaft fillets. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries and accordingly the application of these formulae is limited to those geometries.

102 The SCF's calculated according to the rules of this document are defined as the ratio of stresses calculated by FEM to nominal stresses in both journal and pin fillets.

103. When used in connection with the present method in present Chapter J or the alternative methods, von Mises stresses shall be calculated for bending and principal stresses for torsion.

104. The procedure as well as evaluation guidelines are valid for both solid cranks and semibuilt cranks (except journal fillets).

105. The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.

106. The calculation of SCF at the oil bores is not covered by this document.

107. It is advised to check the element accuracy of the FE solver in use, e.g. by modeling a simple geometry and comparing the stresses obtained by FEM with the analytical solution for purê bending and torsion. Boundary Element Method (BEM) may be used instead of FEM.

200. Model requirements

201. The basic recommendations and perceptions for building the FE-model are presented in J13.203 below.

202. It is obligatory for the final FE-model to fulfill the requirement in J13.400 below.

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

a. ¾ The model consists of one complete crank, from the main bearing centerline to the opposite side main bearing centerline

b. ¾ Element types used in the vicinity of the fillets:

a. 10 node tetrahedral elements

b. 8 node hexahedral elements

c. 20 node hexahedral elements

c. ¾ Mesh properties in fillet radii. The following applies to ±90 degrees in circumferential direction from the crank plane:

d. ¾ Maximum element size a=r/4 through the entire fillet as well as in the circumferential direction. When using 20 node hexahedral elements, the element size in the circumferential direction may be extended up to 5a. In the case of multi-radii fillet r is the local fillet radius. (If 8 node hexahedral elements are used even smaller element size is required to meet the quality criteria.)

e. ¾ Recommended manner for element size in fillet depth direction

e.1. First layer thickness equal to element size of a

e.2. Second layer thickness equal to element to size of 2a

e.3. Third layer thickness equal to element to size of 3a

f. ¾ Minimum 6 elements across web thickness.

g. ¾ Generally the rest of the crank should be suitable for numeric stability of the solver.

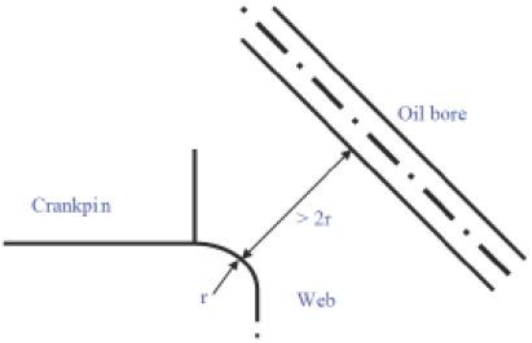
h. ¾ Counterweights only have to be modeled only when influencing the global stiffness of the crank significantly.

i. ¾ Modeling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than 2r, see figure T.J13.203.1.

j. ¾ Drillings and holes for weight reduction have to be modeled.

k. ¾ Sub-modeling may be used as far as the software requirements are fulfilled.

FIGURE F.J13.203.1 - .Oil bore proximity to fillet.



300. Material

301. The present Chapter does not consider material properties such as Young's Modulus (E) and Poisson's ratio (ν). In FE analysis those material parameters are required, as strain is primarily calculated and stress is derived from strain using the Young's Modulus and Poisson's ratio.

302. Reliable values for material parameters have to be used, either as quoted in literature or as measured on representative material samples.

303. For steel the following is advised:

$$E = 2.05 \bullet 10^5 \text{ MPa} \quad \text{and} \quad \nu = 0.3$$

400. Element mesh quality criteria

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

402**.** **Principal stresses criterion** The quality of the mesh should be assured by checking the stress component normal to the surface of the fillet radius. Ideally, this stress should be zero. With principal stresses 1 σ , 2 σ and 3 σ the following criterion is required:

403. **Averaged/unaveraged stresses criterion**: The criterion is based on observing the discontinuity of stress results over elements at the fillet for the calculation of SCF:

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

500. Load cases

501. To substitute the analytically determined SCF in the present Chapter the following load cases have to be calculated.

502. Torsion

a. In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded pure torsion. In the model surface warp at the end faces is suppressed.

b. Torque is applied to the central node located at the crankshaft axis. This node acts as the master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

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

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

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

504. The maximum value taken for the subsequent calculation of the SCF

$$\alpha_T = \frac{\tau_{equiv,\alpha}}{\tau_N}$$

$$\beta_T = \frac{\tau_{equiv,\beta}}{\tau_N}$$

where N τ is nominal torsional stress referred to the crankpin and respectively journal as per J13.300 above with the torsional torque T:

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

600. Pure bending (4 point bending)

601. In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded in pure bending. In the model surface warp at the end faces is suppressed.

602. The bending moment is applied to the central node located at the crankshaft axis. This node acts as the master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

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

FIGURE F.J13.501.1 - BOUNDARY AND LOAD CONDITIONS FOR THE TORSION LOAD CASE.

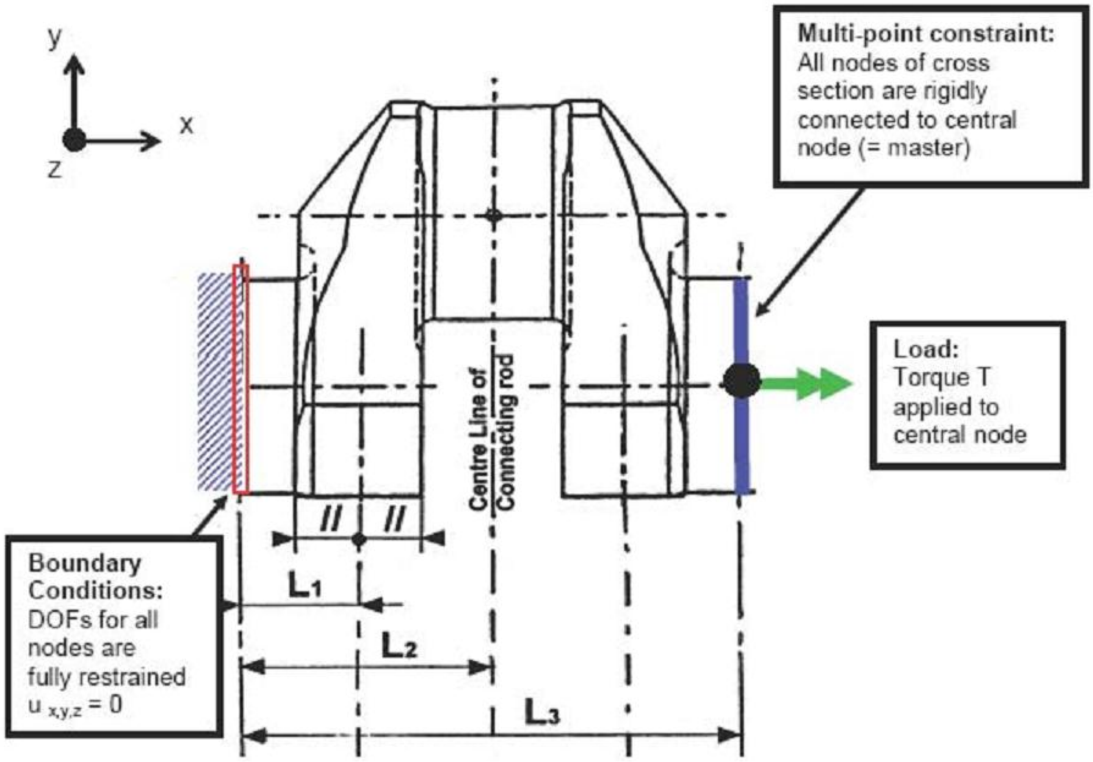
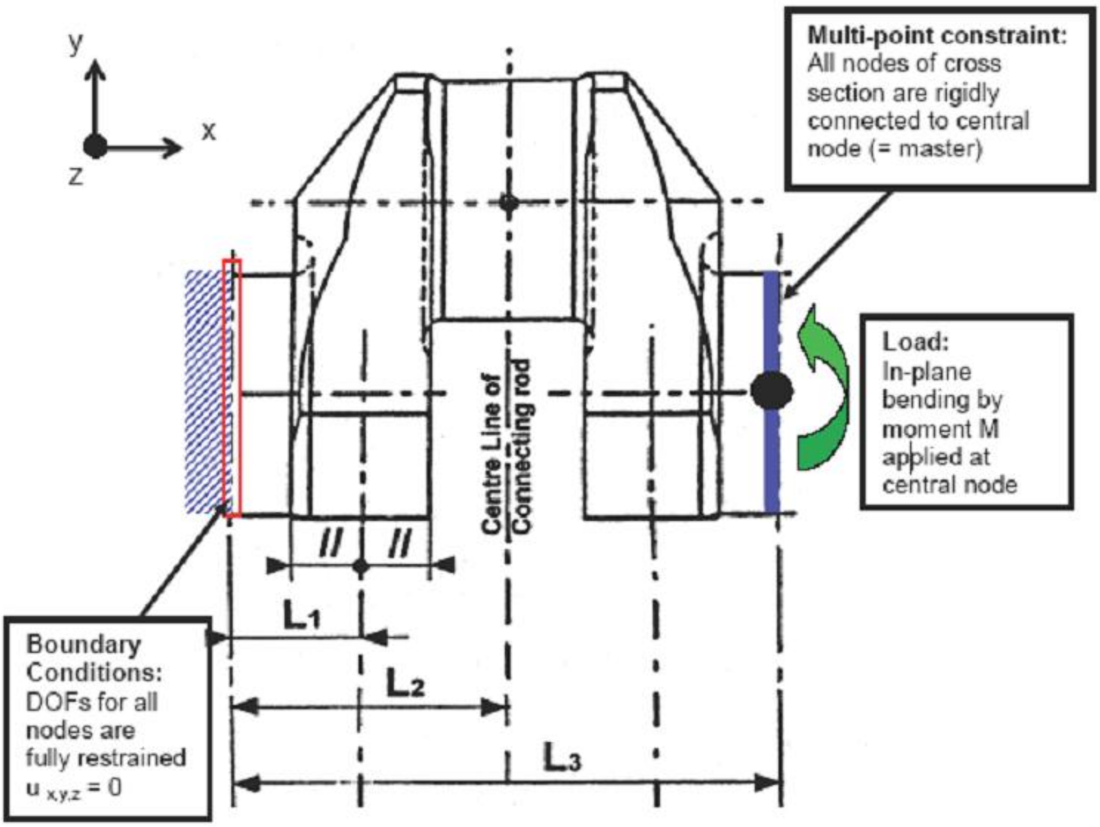


FIGURE F.J13.603.1 - BOUNDARY AND LOAD CONDITIONS FOR THE PURE BENDING LOAD CASE.



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

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

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

605. Nominal stress N σ is calculated as per the present Chapter a. with the bending moment M:

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

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

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

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

703. The force is applied to the central node located at the pin centre-line of the connecting rod.

704. This node is connected to all nodes of the pin cross sectional area. Warping of the sectional area is not suppressed.

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

FIGURE F.J13.701.1 - BOUNDARY AND LOAD CONDITIONS FOR THE 3-POINT BENDING LOAD CASE OF AN INLINE ENGINE.

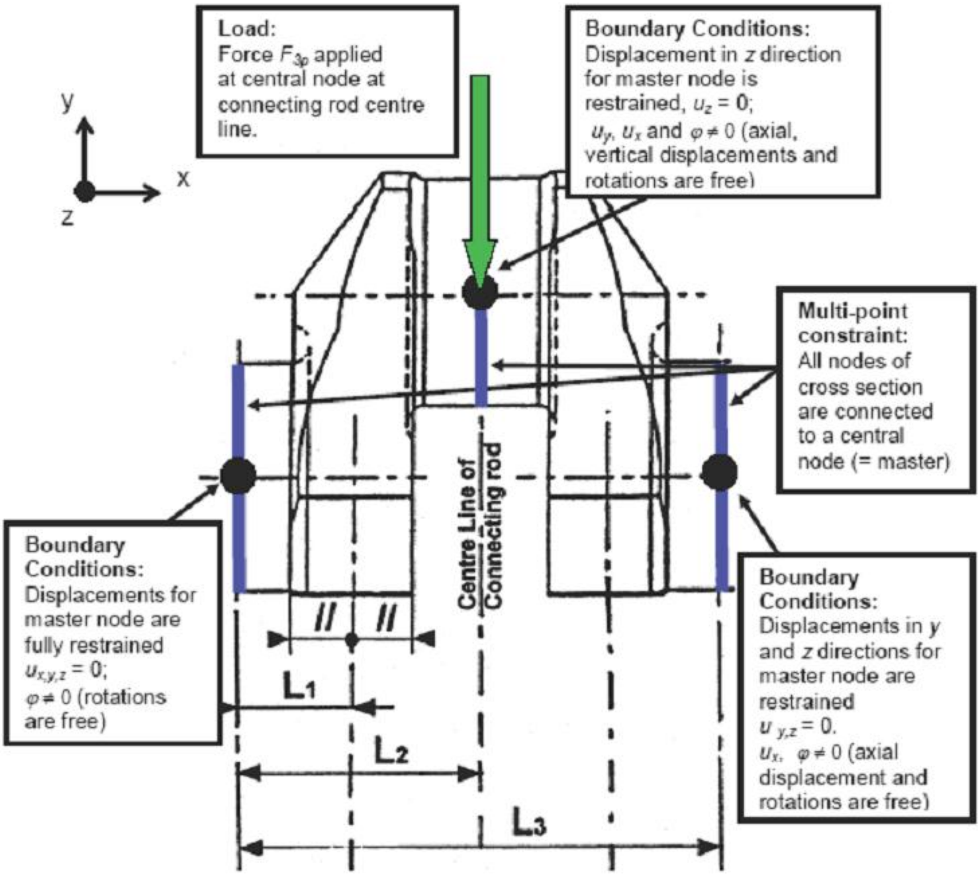
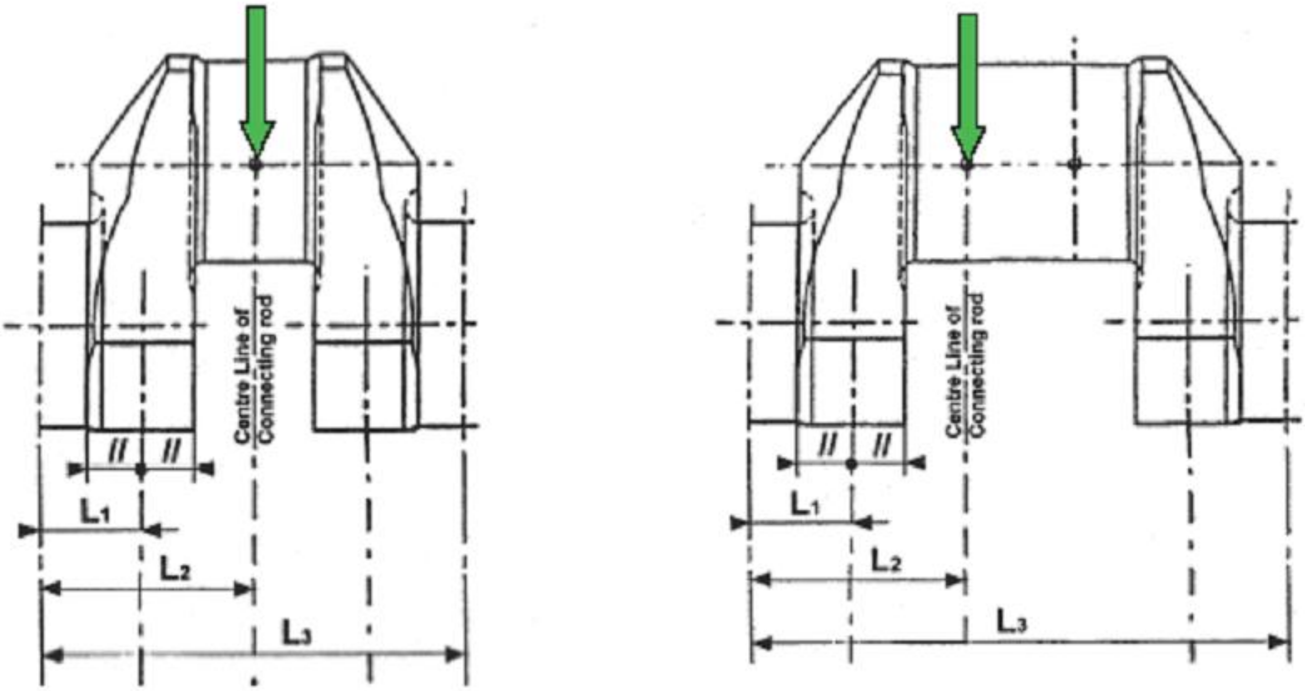


FIGURE F.J13.705.1 - LOAD APPLICATIONS FOR IN-LINE AND V-TYPE ENGINES.



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

800. Method 1

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

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

where:

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

σ_{N3P} Nominal bending stress in the web centre due to the force $F3_{3P}$ [N] applied to the centre-line of the actual connecting rod, see figure F.J13.705.1.

β_B as determined in J13.600 above.

$\sigma_{3QP} = Q_{3P}/B*W$ where Q_{3P} is the radial (shear) force in the web due to the force F_{3P} [N] applied to the centre-line of the actual connecting rod, see also figure F.J1.501.3 F.J2.103.1 in J2 .

900. Method 2

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

902. The SCF is then calculated according to

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

903. For symbols see J13.800 above.

904. When using this method the radial force and stress determination in J2 becomes superfluous. The alternating bending stress in the journal fillet as per 2.700 is then evaluated:

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

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

CHAPTER K
STEERING GEAR
[IACS UR M42]

K1. APPROACH

100. Application

In addition to the requirements contained in the Amendments to the 1974 SOLAS Convention, Chapter II–I Reg. 29 and 30, and related Guidelines (see Annex 2 of IMCO document MSC XLV/4) the following requirements apply to new ocean-going vessels of 500 GRT and upwards. These requirements may be applied to other vessels at the discretion of the RBNA.

K2. DEFINITIONS

100. Definitions

101. **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.

102. **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.

103. **Steering gear power unit means**:

a. in the case of electric steering gear, and electric motor and its associated electrical equipment,

b. in the case of electrohydraulic steering gear, an electric motor and its associated electrical equipment and connected pump,

c. in the case of other hydraulic steering gear, a driving engine and connected pump.

104. Auxiliary steering gear means the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not

including the tiller, quadrant or components serving the same purpose.

105. **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.

106. **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.

107**. Rudder actuator** means the component which converts directly hydraulic pressure into mechanical action to move the rudder.

108. **Maximum working pressure** means the maximum expected pressure in the system when the steering gear is operated to comply with Regulation 29.3.2.

K3. PLANS AND SPECIFICATIONS

100. Plans and specifications to be submitted

101. Before starting construction, all relevant plans and specifications are to be submitted to the RBNA for approval.

K4. DESIGN PRINCIPLES

100. Design

101. The construction should be such as to minimize local concentrations of stress.

200. Dynamic loads for fatigue and fracture mechanic analysis

201. The dynamic loading to be assumed in the fatigue and fracture mechanics analysis considering SOLAS Regulation 29.2.2 and 29.17.1 and relating Guidelines, will be established at the discretion of the RBNA.

202. Both the case of high cycle and cumulative fatigue are to be considered.

K5. CONSTRUCTION PRINCIPLES

100. Power piping arrangements

101. The power piping for hydraulic steering gears is to be arranged so that transfer between units can be readily effected.

102. 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.

103. For all vessels with non-duplicated actuators, isolating valves are to be fitted at the connection of pipes to the actuator, and are to be directly fitted on the actuator.

104. Arrangements for bleeding air from the hydraulic system are to be provided where necessary.

105. Piping, joints, valves, flanges and other fittings are to comply with RBNA requirements for Class 1 components. The design pressure is to be in accordance with K5.506 below.

200. Rudder angle limiters

201. Power-operated steering gears are to be provided with positive arrangements, such as limit switches, for stopping the gear before the rudder stops are reached.

202. These arrangements are to be synchronized with the gear itself and not with the steering gear control.

300. Materials

301. Ram cylinders; pressure housings of rotary vane type actuators; hydraulic power piping valves, flanges and fittings; and all steering gear components transmitting mechanical forces to the rudder stock (such as tillers, quadrants or similar components) should be of steel or other approved ductile material, duly tested in accordance with the requirements of the RBNA. In general, such material should not have an elongation of less than 12 per cent nor a tensile strength in excess of 650 N/mm2

302. Grey cast iron may be accepted for redundant parts with low stress level, excluding cylinders, upon special consideration.

400. Welds

401. The welding details and welding procedures should be approved.

402. All welded joints within the pressure boundary of a rudder actuator or connecting parts transmitting mechanical loads should be full penetration type or of equivalent strength.

500. Oil seals

501. Oil seals between non-moving parts, forming part of the external pressure boundary, should be of the metal upon metal type or of an equivalent type.

502. Oil seals between moving parts, forming part of the external pressure boundary, should be duplicated, so that the failure of one seal does not render the actuator inoperative. Alternative arrangements providing equivalent protection against leakage may be accepted at the discretion of the RBNA.

503. All steering gear components transmitting mechanical forces to the rudder stock, which are not protected against overload by structural rudder stops or mechanical buffers, are to have a strength at least equivalent to that of the rudder stock in way of the tiller. For piping, joints, valves, flanges and other fittings see K5.104 above.

504. Rudder actuators other than those covered by SOLAS Regulation 29.17 and relating Guidelines should be designed in accordance with Class 1 pressure vessels (notwithstanding any exemptions for hydraulic cylinders).

505. In application of such rules the permissible primary general membrance stress is not to exceed the lower of the following values:

$$\frac{\sigma_B}{A} \text{ or } \frac{\sigma_y}{B}$$

where:

σB = specified minimum tensile strength of material at ambient temperature

σy = specified minimum yield stress or 2 per cent proof stress of the material, at ambient temperature

A and B are given by the Table T.K5.505.1.

TABLE T.K5.505.1

	Steel	Cast Steel	Nodular Cast Iron
A	3.5	4	5
B	1.7	2	3

506. The design pressure is to be at least equal to the greater of the following:

a. 1.25 times the maximum working pressure,

b. the relief valve setting.

507. Accumulators, if any, are to comply with RBNA requirements for pressure vessels.

600. Hoses

601. Hose assemblies of type approved by the RBNA may be installed between two points where flexibility is required but should not be subjected to torsional deflection (twisting) under normal operating conditions. In general, the hose should be limited to the length necessary to provide for flexibility and for proper operation of machinery.

602. Hoses should be high pressure hydraulic hoses according to recognized standards and suitable for the fluids, pressures, temperatures and ambient conditions in question.

603. Burst pressure of hoses should not be less than four times the design pressure.

700. Relief valves

701. Relief valves for protecting any part of the hydraulic system which can be isolated, as required by Regulation 29.2.3 should comply with the following:

a. The setting pressure should not be less than 1.25 times the maximum working pressure.

b. The minimum discharge capacity of the relief valve(s) should not be less than the total capacity of the pumps, which can deliver through it (them), increased by 10 per cent.

702. Under such conditions the rise in pressure should not exceed 10 per cent of the setting pressure. In this regard, due consideration should be given to extreme foreseen ambient conditions in respect of oil viscosity.

703. The RBNA may require, for the relief valves, discharge capacity tests and/or shock tests.

K6. ELECTRICAL INSTALLATIONS

100. Electrical installations and monitoring systems

101. Electrical installations should comply with the requirements of the RBNA.

200. Alternative source of power

201. Where the alternative power source required by SOLAS Regulation 29.14 is a generator, or an engine driven pump, automatic starting arrangements are to comply with the requirements relating to the automatic starting arrangements of emergency generators.

300. Monitoring and alarm systems

301. Monitoring and alarm systems, including the rudder angle indicators, should be designed, built and tested to the satisfaction of the RBNA.

302. Where hydraulic locking, caused by a single failure, may lead to loss of steering, an audible and visual alarm, which identifies the failed system, shall be provided on the navigating bridge.

303. Note: This alarm should be activated whenever:

a. position of the variable displacement pump control system does not correspond with given order; or

b. incorrect position of 3-way full flow valve or similar in constant delivery pump system is detected.

K7. OPERATING INSTRUCTIONS

100. Operating instructions

101. Where applicable, following standard signboard should be fitted at a suitable place on steering control post on the bridge or incorporated into operating instruction on board:

CAUTION

IN SOME CIRCUMSTANCES WHEN 2 POWER UNITS ARE RUNNING SIMULTANEOUSLY THE RUDDER MAY NOT RESPOND TO HELM. IF THIS HAPPENS STOP EACH PUMP IN TURN UNTIL CONTROL IS REGAINED.

102. The above signboard is related to steering gears provided with 2 identical power units intended for simultaneous operation, and normally provided with either their own control systems or two separate (partly or mutually) control systems which are/may be operated simultaneously.

103. Note: Existing vessels according to SOLAS 1986 shall have minimum the above signboard, when applicable.

CHAPTER T
TESTS

CHAPTER CONTENTS

T1. ENGINES AND OTHER MACHINERY EQUIPMENTS

100. Quay and sea trials

101. A Program of Surveys and Testing is to be submitted to RBNA for approval, with quay trials, for calibration of operation before shipping, and sea trials, from which will result in a Report of Survey and Testing, where the indexes and performance of engines and machinery equipment are recorded.

102. The assessment of performance of propulsion engines consists of four hours of uninterrupted operation at nominal power.

200. Temperature of the machinery spaces

201. Ambient temperatures at various locations of the machinery spaces are to be measured with the engine in normal working operation, after a minimum of one (1) hour, and openings closed. The temperatures are not to exceed 45^{o} C.

202. During the sea trial temperatures are to be measured at the exposed as well as isolated surfaces of combustion engines, boilers and their discharge pipelines to detect spots with temperatures above 220 ° C.

The most critical points to be examined are:

a. Engine casing;

b. Cylinder head lubricating valves;

c. cylinder head covers;

d. connections to the exhaust manifold;

e. exhaust pipes, specially overlap between metal plates and insulation;

f. bedplates and supports for exhaust manifold;

g. turbochargers, specially at their flanges;

h. output for temperature and pressure sensors;

i. surface of lighting reflectors; and

j. Most critical points that statistically have been the cause of fires.

300. Clearances and tolerances

301. The clearances of bearings and couplings are to be measured during sea trials.

T2. TRANSMISSION ELEMENTS

100. Bearings, liners and bearing bushings

101. In addition to the requirements of Part III of the Rules, liners and bronze bearings or other approved materials should have their composition identified and traced, free from porosity and / or harmful defects and tightness hydrostatically tested at a pressure of 20 N/mm² (2 Kgf/mm²).

102. The temperatures of bearings are to be taken during sea trials.

T3. STEERING GEAR TESTS

100. Testing at the manufacturer

101. See Part III, Title 62, Section5, Chapter I.

200. Trials

201. The steering gear should be tried out on the trial trip in order to demonstrate to the Surveyor's satisfaction that the requirements of the Rules have been met. The trial is to include the operation of the following:

a. the steering gear, including demonstration of the performances required by Regulation 29.3.2 and 29.4.2. For controllable pitch propellers, the propeller pitch is to be at the maximum design pitch approved for the maximum continuous ahead R.P.M. at the main steering gear trial.

a.1. If the vessel cannot be tested at the deepest draught, steering gear trials shall be conducted at a displacement as close as reasonably possible to full-load displacement as required by Section 6.1.2 of ISO 19019:2005 on the conditions that either the rudder is fully submerged (zero speed waterline) and the vessel is in an acceptable trim condition, or the rudder load and torque at the specified trial loading condition have been predicted and extrapolated to the full load condition. In any case for the main steering gear trial, the speed of ship corresponding to the number of maximum continuous revolution of main engine and maximum design pitch applies.

b. the steering gear power units, including transfer between steering gear power units.

c. the isolation of one power actuating system, checking the time for regaining steering capability.

d. the hydraulic fluid recharging system.

e. the emergency power supply required by Regulation 29.14.

f. the steering gear controls, including transfer of control and local control.

g. the means of communication between the wheelhouse, engine room, and the steering gear compartment.

h. the alarms and indicators required by regulations 29, 30 and K5.300 above, these tests may be effected at dockside.

i. where steering gear is designed to avoid hydraulic locking this feature shall be demonstrated.

T4. TESTING OF MASS PRODUCED I.C. ENGINES AT MANUFACTURER

See Part III, Title 62, Section 5 , Chapter H.

T5. ADDITIONAL TESTS COMMON FOR ALL NOTATIONS OF AUTOMATION SYSTEMS

100. Qualification of the components.

101. Manufacturers of control systems must certify that the mechanical, electrical and the solid state components, made by them, were individually tested and have satisfactory results or that were passed through the batch tests of samples, in order to establish its suitability for the requested service, including compliance with conditions described in chapter C. above. See Part III,Title 63, Section 8 for the main tests of equipment.

200. Quay and sea trials

201. After the installation, the automation system is submitted to the quay and sea trials to demonstrate that the entire system operates satisfactorily during conditions of standby, maneuvers, continuous operation and transfer between stations.

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