RULES FOR
CLASSIFICATION OF

SHIPS / HIGH SPEED, LIGHT CRAFT AND NAVAL SURFACE CRAFT

NEWBUILDINGS

MACHINERY AND SYSTEMS
MAIN CLASS

PART 4 CHAPTER 4

ROTATING MACHINERY, POWER TRANSMISSION

JULY 2008

This booklet includes the relevant amendments and corrections shown in the July 2009 version of Pt.0 Ch.1 Sec.3.

CONTENTS PAGE

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CHANGES IN THE RULES

General

The present edition of the rules includes additions and amendments decided by the Board in June 2008 and supersedes the January 2007 edition of the same chapter.

The rule changes come into force as indicated below.

This chapter is valid until superseded by a revised chapter. Supplements will not be issued except for an updated list of minor amendments and corrections presented in Pt.0 Ch.1 Sec.3. Pt.0 Ch.1 is normally revised in January and July each year.

Revised chapters will be forwarded to all subscribers to the rules. Buyers of reprints are advised to check the updated list of rule chapters printed Pt.0 Ch.1 Sec.1 to ensure that the chapter is current.

Significant editorial changes adopted July 2009

Taking effect immediately

- **Sec.1 Shafting**

— In item E301, the list-item reading “where one interchangeable sensor is fitted one spare sensor shall be stored on board”, has been deleted

— Item F105 concerning rope guard requirements, has been deleted.

Significant editorial changes adopted January 2009

Taking effect immediately

- **Sec.2 Gear Transmissions**

— List item a), sub-item 1) in C206 has been extensively amended in order to introduce an alternative for gear manufacturers who still want to apply small standard test coupons without performing any correlation testing.

Main changes coming into force 1 January 2009

- **Sec.1 Shafting**

— Items A301 and A401 have been amended to include the position and way of electrical grounding in the submitted documentation as a general requirement, and not only a requirement in case of additional class notation **TMON**.

— Item B208 has been modified. To avoid misunderstanding, it has been clarified that the shaft diameter calculated by simplified diameter formula is just an absolute minimum diameter, and that torsional vibration stresses of an actual shafting arrangement has to be considered, i.e. a clean-up of when to use the one or the other has been made.

— In item B306, the Guidance note has been expanded in order to better explain what is meant by ream fitted bolts, with some practical hints how to facilitate easier removal of bolts without damaging bolt holes.

— In item B402, formula for calculating necessary pull-up force for conical connections has been included.

— In item F105, a new requirement has been introduced for installation of rope guard on all vessels in order to protect the stern tube seal from damage and consequently oil leakage from the stern tube.

In addition, a requirement for net/line cutter has been introduced for vessels that are more exposed to waters with ghost lines and nets such as fishing vessels, ferries, tugs etc.

- **Sec.5 Torsionally Elastic Couplings**

— Item B207 has been amended in relation to shear stresses due to centrifugal action for elastic couplings of high speeds.

Corrections and Clarifications

In addition to the above stated rule requirements, a number of detected errors, corrections and clarifications have been made to the existing rule text.

Comments to the rules may be sent by e-mail to *rules@dnv.com*
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SECTION 1
SHAFTING

A. General

A 100 Application

101 *Shafting* is defined as the following elements:

- shafts
- rigid couplings as flange couplings, shrink-fit couplings, keyed connections, clamp couplings, splines, etc. (compliant elements as tooth couplings, universal shafts, rubber couplings, etc. are dealt with in their respective sections)
- shaft bearings
- shaft seals.

Shafts or couplings made of composite materials are subject to special consideration.

Sec.1 also deals with the fitting of the propeller (and impeller for water jet), shaft alignment and whirling.

102 The rules in this section apply to shafting subject to certification for the purposes listed in Ch.2 Sec.1 A200. However, they do not apply for generator shafts, except for single bearing type generators, where documentation may be requested upon request in case of high torsional vibrations. Furthermore, they only apply to shafts made of forged or hot rolled steel. Shafts made of other materials will be considered on the basis of equivalence with these rules.

103 Ch.2 describes all general requirements for rotating machinery, and forms the basis for all sections in Ch.3, Ch.4 and Ch.5.

104 Stern tube oil seals of standard design shall be type approved.

A 200 Documentation of shafts and couplings

201 Drawings of the shafts, liners and couplings shall be submitted. The drawings shall show clearly all details, such as fillets, keyways, radial holes, slots, surface roughness, shrinkage amounts, contact between tapered parts, pull up on taper, bolt pretension, protection against corrosion, welding details etc. as well as material types, mechanical properties, cleanliness (if required, see B203) and NDT specification, see Ch.2 Sec.3 A200. For shafts with a maximum diameter >250 mm (flanges not considered) that shall be quenched and tempered, a drawing of the forging, in its heat treatment shape, shall be submitted upon request.

202 Applicable load data shall be given. The load data or the load limitations shall be sufficient to carry out design calculations as described in B, see also Ch.2 Sec.3 A101. This means as a minimum:

P = maximum continuous power (kW)
or T_0 = maximum continuous torque (Nm)
n_0 = r.p.m. at maximum continuous power.

For plants with gear transmissions the relevant application factors shall be given, otherwise upper limitations (see Ch.3 Sec.1 G for diesel engine drives) will be used:

K_A = application factor for continuous

$$\text{operation} = 1 + \frac{T_v}{T_0} = 1 + \frac{\tau_v}{\tau_0}$$

however, not to be taken less than 1.1, in order to cover for load fluctuations

K_{AP} = application factor for non-frequent peak loads (e.g. clutching-in shock loads or electric motors

$$\text{with star-delta switch)} = \frac{T_{peak}}{T_0} = \frac{\tau_{peak}}{\tau_0}$$

K_{Aice} = application factor due to ice shock loads (applicable for ice classed vessels), see Pt.5 Ch.1 of the Rules for Classification of Ships

ΔK_A = Application factor, torque range (applicable to reversing plants)

$$\Delta K_A = \frac{K_{A(P)(ice)}\ \tau_0 + \tau_{max\ reversed}}{\tau_0}$$

As a safe simplification it may be assumed that $\Delta K_A = 2\ K_A$ or $2\ K_{AP}$ or $2\ K_{Aice}$ whichever is the highest.

Where:

T_v = vibratory torque for continuous operation in the full speed range (~ 90 – 100% of n_0)

τ_v = nominal vibratory torsional stress for continuous operation in the full speed range

τ_0 = nominal mean torsional stress at maximum continuous power

$\tau_{max\ reversed}$ = maximum reversed torsional stress, which is the maximum value of $(\tau + \tau_v)$ in the entire speed range (for astern running), or $\tau_{ice\ rev}$ (for astern running) whichever is the highest.

For direct coupled plants (i.e. plants with no elastic coupling or gearbox) the following data shall be given:

- τ_v = nominal vibratory torsional stress for continuous operation in the entire speed range. See torsional vibration in Ch.3 Sec.1 G300
- τ_{vT} = nominal vibratory torsional stress for transient operation (e.g. passing through a barred speed range) and the corresponding relevant number of cycles N_C. See torsional vibration in Ch.3 Sec.1 G400.
- Reversing torque if limited to a value less than T_0.

For all kinds of plants the necessary parameters for calculation of relevant bending stresses shall be submitted, see F and G.

A 300 Documentation of bearings and seals

301 Drawings of separate thrust bearings, stern tube bearings and oil seals shall be submitted. The drawings shall show all details as dimensions with tolerances, material types, and (for bearings) the lubrication system. (Drawings of ball and roller bearings need not to be submitted.) For main thrust bearings the mechanical properties of the bearing housing and foundation bolts shall be submitted.

If the class notation **TMON** (tailshaft condition monitoring survey arrangement) is applicable, the following additional information is required:

- lubrication oil diagram for the stern tube bearings with identified oil sampling point and a description of the sampling procedure

— the position of aft stern tube bearing temperature sensor(s).

302 For all fluid film bearings the maximum permissible load and maximum permissible operating temperatures with regard to necessary oil film thickness if applicable shall be specified.

303 The maximum permissible lateral movements for shaft oil seals shall be specified.

304 Documentation of the manufacturer's quality control with regard to inspection and testing of materials and parts of bearings and seals shall be submitted upon request.

305 For separate thrust bearings, calculation of smallest hydrodynamic oil film thickness shall be submitted, see B901.

306 Documentation for the control and monitoring system, including set-points and delays, see E, shall be submitted for approval.

For requirements to documentation types, see Ch.9.

A 400 Documentation of shafting system and dynamics

401 Drawings of the complete shafting arrangement shall be submitted. Type designation of prime mover, gear, elastic couplings, driven unit, shaft seals etc. shall be stated on the drawings. The drawings shall show all main dimensions as diameters and bearing spans, bearing supports and any supported elements as e.g. oil distribution boxes.

Position and way of electrical grounding shall be indicated.

402 Shaft alignment calculations are always to be submitted for approval for propulsion plants with:

— intermediate shaft diameters of 400 mm or greater for single screw and 300 mm for multi screw
— gear transmissions with more than one pinion driving the output gear wheel, even if there is only one single input shaft as for dual split paths
— shaft generator or electrical motor as an integral part of the low speed shaft in diesel engine propulsion.

Upon request, shaft alignment calculations may also be required for other plants when these are considered sensitive to alignment.

For required content of the of shaft alignment calculations, see F400.

403 For all propulsion plants other than those listed in 402, only a shaft alignment specification shall be submitted for information. The shaft alignment specification shall include the following items:

— bearing offsets from the defined reference line
— verification data with tolerances (e.g. gap and sag) and jacking loads (including jack correction factors) and conditions (cold or hot, submerged propeller, etc.).

404 Calculations of whirling vibration or lateral rotor vibration may be required upon request. Normally this means determination of natural frequencies.

405 Axial vibration calculations may be required upon request, see also Ch.3 Sec.1 A601 c).

B. Design

B 100 General

101 For design principles see Ch.2 Sec.3 A100. The shafting shall be designed for all relevant load conditions such as rated power, reversing loads, foreseen overloads, transient conditions, etc. including all driving conditions under which the plant may be operated.

102 Determination of loads under the driving conditions specified in 101 is described in F and G as well as in Ch.3 Sec.1 G.

B 200 Criteria for shaft dimensions

201 Shafts shall be designed to prevent fatigue failure and local deformation. Simplified criteria for the most common shaft applications are given in 206, 207 and 208.

Guidance note:

Classification Note 41.4 offers detailed methods on how to assess the safety factor criteria mentioned in Table B1.

Alternative methods may also be considered on the basis of equivalence.

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It is sufficient that either the detailed criteria in Classification Note 41.4 or the simplified criteria are fulfilled. In addition, the shafts shall be designed to prevent rust or detrimental fretting that may cause fatigue failures, see also 402.

202 The major load conditions to be considered are:

— low cycle fatigue (10^3 to 10^4 cycles) due to load variations from zero to full load, clutching-in shock loads, reversing torques, etc. In special cases, such as short range ferries higher number of cycles (~10^5 cycles) may apply
— high cycle fatigue (>>$3 \cdot 10^6$ cycles) due to rotating bending and torsional vibration
— ice shock loads (10^6 to 10^7 cycles), applicable to vessels with ice class notations and ice breakers
— transient vibration as when passing through a barred speed range (10^4 to $3 \cdot 10^6$ cycles).

203 For applications where it may be necessary to take the advantage of tensile strength above 800 MPa and yield strength above 600 MPa, material cleanliness has an increasing importance. Higher cleanliness than specified by material standards may be required (preferably to be specified according to ISO 4947). Furthermore, special protection against corrosion is required. Method of protection shall be approved, see A201.

Table B1 Shaft safety factors

Criteria	*Safety factor, S*
Low cycle ($N_C < 10^4$ stress cycles)	1.25
High cycle ($N_C >> 3 \cdot 10^6$ stress cycles)	1.6
Transient vibration when passing through a barred speed range: ($10^4 < N_C < 3 \cdot 10^6$ stress cycles)	Linear interpolation (log τ-logN diagram) between the low cycle, peak stresses criterion with S = 1.25 and the high cycle criterion with S = 1.5. For propeller shafts in way of and aft of the aft stern tube bearing, the bending influence is covered by an increase of S by 0.05.

204 Stainless steel shafts shall be designed to avoid cavities (pockets) where the sea water may remain un-circulated (e.g. in keyways). For other materials than stainless steel I, II and III as defined in Table B3, special consideration applies to fatigue values and pitting corrosion resistance.

205 The shaft safety factors for the different applications and criteria detailed in Classification Note 41.4 shall be, at least, in accordance with Table B1. See also Guidance Note in 201.

206 *Simplified diameter formulae for plants with low torsional vibration such as geared plants or direct driven plants with elastic coupling.*

The simplified method for direct evaluation of the minimum diameters d for various design features are based on the following assumptions:

— σ_y limited to 0.7 σ_B (for calculation purpose only)
— application factors K_{Aice} and $K_{AP} \le 1.4$
— vibratory torque $T_v \le 0.35\ T_0$ in all driving conditions
— application factor, torque range $\Delta K_A \le 2.7$
— inner diameters $d_i \le 0.5$ d except for the oil distribution shaft with longitudinal slot where $d_i \le 0.77$ d
— protection against corrosion (through oil, oil based coating, material selection or dry atmosphere).

If any of these assumptions are not fulfilled, the detailed method in Classification Note 41.4 may be used, see Guidance Note in 201.

The simplified method results in larger diameters than the detailed method. It distinguishes between:

— low strength steels with $\sigma_B \le 600$ MPa which have a low notch sensitivity, and
— high strength steels with $\sigma_B > 600$ MPa such as alloyed quenched and tempered steels and carbon steels with a high carbon content that all are assumed to have a high notch sensitivity.

A. Low cycle criterion:

$$d_{min} = 28\ k_1 \sqrt[3]{\frac{T_0}{\sigma_y}}$$

k_1 - Factor for different design features, see Table B2.
σ_y - Yield strength or 0.2% proof stress limited to 600 MPa for calculation purposes only

B. High cycle criterion:

$$d_{min} = 17.5\ k_2 \sqrt[3]{\frac{T_0}{0.32\ \sigma_y + 70}} \left(1 + k_3 \left(\frac{M_b}{T_0}\right)^2\right)^{\frac{1}{6}}$$

M_b = Bending moment (Nm), due to hydrodynamic forces on propeller, propeller weight or other relevant sources from the list in F202.

For bending moments due to reactions from T_0 as for gear shafts, M_b shall include the K_A factor of 1.35.

k_2, k_3= Factors for different design features, see Table B2.

The higher value for d_{min} from A and B applies. However, for shafts loaded in torsion only, it is sufficient to calculate d according to A.

Table B2 Factors k_1, k_2 and k_3

Design feature	*Torsion only*		*Combined torsion and bending*		
Specified tensile strength σ_B (Mpa)	≤600 k_1	>600 k_1	≤600 k_2	>600 k_2	k_3
Plain shaft or flange fillet with multi-radii design, see B208, $R_a \le 6.4$	1.00	1.00	1.09	1.13	13
Keyway (semicircular), bottom radius r ≥ 0.015 d, $R_a \le 1.6$	1.16	1.27	1.43	1.46	8
Keyway (semicircular), bottom radius r ≥ 0.005 d, $R_a \le 1.6$	1.28	1.44	1.63	1.66	11
Flange fillet r/d ≥ 0.05, t/d ≥ 0.20, $R_a \le 3.2$	1.05	1.10	1.23	1.26	19
Flange fillet r/d ≥ 0.08 t/d ≥ 0.20, $R_a \le 3.2$	1.04	1.09	1.21	1.24	18
Flange fillet r/d ≥ 0.16 t/d ≥ 0.20, $R_a \le 3.2$	1.00	1.04	1.16	1.18	16
Flange fillet r/d ≥ 0.24 t/d ≥ 0.20, $R_a \le 3.2$	1.00	1.03	1.14	1.17	15
Flange for propeller r/d ≥ 0.10, t/d ≥ 0.25, $R_a \le 3.2$	1.02	1.06	1.17	1.20	17
Radial hole, $d_h \le 0.2$ d, $R_a \le 0.8$	1.10	1.19	1.36	1.38	18
Shrink fit edge, with one keyway	1.00	1.05	1.15	1.22	34
Shrink fit edge, keyless	1.00	1.05	1.13	1.22	28
Splines (involute type) 1)	1.00	1.00	1.05	1.10	15
Shoulder fillet r/d ≥ 0.02, D/d ≤ 1.1, $R_a \le 3.2$	1.05	1.10	1.21	1.25	22
Shoulder fillet r/d ≥ 0.1, D/d ≤ 1.1, $R_a \le 3.2$	1.00	1.03	1.14	1.17	16
Shoulder fillet r/d ≥ 0.2, D/d ≤ 1.1, $R_a \le 3.2$	1.0	1.01	1.12	1.15	13
Relief groove 1), D/d = 1.1, D-d ≤ 2 r, $R_a \le 1.6$	1.00	1.04	1.15	1.17	16
Groove 1) for circlip, D-d ≤ 2 b, D-d ≤ 7.5 r, $R_a \le 1.6$	1.17	1.28	1.38	1.40	27
Longitudinal slot 2) in oil distribution shaft, $d_i \le 0.77$ d, 0.05 d ≤ e ≤ 0.2 d, (1 – e) ≤ 0.5 d, $R_a \le 1.6$	1.49	1.69			

1) applicable to root diameter of notch
2) applicable for slots with outlets each 180° and for outlets each 120°

207 *Simplified diameter formulae for stainless steel shafts subjected to sea water and with low torsional vibration.*

This simplified method for direct evaluation of minimum diameters d_{min} for various design features are based on the same conditions as in 206 except that the protection against corrosion now is protection against crevice corrosion. This means that e.g. keyways shall be sealed in both ends and thus the calculation in 206 applies for such design features. However, for craft where the shaft is stationary for some considerable time, measures should be taken to avoid crevice corrosion in way of the bearings e.g. periodically rotation of shaft or flushing. It is distinguished between 3 material types, see Table B3. The simplified method is only valid for shafts accumulating 10^9 to 10^{10} cycles.

Table B3 Stainless steel types						
Material type	*Main structure*	*Main alloy elements*			*Mechanical properties*	
		% Cr	% Ni	% Mo	σ_B	$\sigma_y = \sigma_{0.2}$
Stainless steel I	Austenitic	16–18	10–14	≥ 2	500–600	$\geq 0.45\ \sigma_B$
Stainless steel II	Martensitic	15–17	4–6	≥ 1	850–1000	$\geq 0.75\ \sigma_B$
Stainless steel III	Ferritic-austenitic (duplex)	25–27	4–7	1-2	600–750	$\geq 0.65\ \sigma_B$

A. The low cycle criterion:

$$d_{min} = 28\ k_1 \sqrt[3]{\frac{T_0}{\sigma_y}}$$

k_1 - Factor for different design features, see Table B4.

For shafts with significant bending moments:

The formula shall be multiplied with:

$$\left(1 + \frac{4}{3}\left(\frac{M_b}{T_0}\right)^2\right)^{\frac{1}{6}}$$

B. The high cycle criterion:

$$d_{min} = 4 \sqrt[3]{T_0} \left(1 + k_3 \left(\frac{M_b}{T_0}\right)^2\right)^{\frac{1}{6}}$$

M_b = Bending moment (Nm), e.g. due to propeller or impeller weight or other relevant sources mentioned in F202. However, the stochastic extreme moment in F301 item 2) shall not be used for either low or high cycle criteria.

k_3 = Factor for different design features, see Table B4.

The highest value for d_{min} from A and B applies.

Table B4 Factors k_1 and k_3			
	A. Low cycle		*B. High cycle*
Design feature 2)*:*	*Stainless Steel* 1)*:*		
	I	II and III	I, II and III
	k_1	k_1	k_3
Plain shaft	1.00	1.00	14
Propeller flange r/d ≥ 0.10 t/d ≥ 0.25	1.04	1.08	19
Shrink fit edge, keyless	The area under the edge is not subject to sea water, thus calculated according to B206		
1) According to Table B3			
2) Surface roughness R_a < 1.6 applies for all design features			

208 *Simplified calculation method for shafts in direct coupled plants.*

1) This method may also be used for other intermediate and propeller shafts that are mainly subjected to torsion. Shafts subjected to considerable bending, such as in gearboxes, thrusters, etc. as well as shafts in prime movers are not included.

   Further, additional strengthening for ships classed for navigation in ice is not covered by this method.

2) The method has following material limitations:

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

   Close to the permissible stresses for transient operation" means more than 70% of permissible value.

   For use in the formulae in this method, σ_B is limited as follows:

   — For C and C-Mn steels up to 600 MPa for use in item 4, and up to 760 MPa for use in item 3.
   — For alloy steels up to 800 MPa.
   — For propeller shafts in general up to 600 MPa (for all steel types).

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

3) Shaft diameters:

   Shaft diameters shall result in acceptable torsional vibration stresses, see item 4) or in any case not to be less than determined from the following formula:

$$d_{min} = F k \sqrt[3]{\frac{P}{n_0} \frac{1}{1 - \frac{d_i^4}{d^4}} \frac{560}{\sigma_B + 160}}$$

   where

   d_{min}= minimum required diameter unless larger diameter is required due to torsional vibration stresses, see item 4)
   d_i = actual diameter of shaft bore (mm)
   d = actual outside diameter of shaft (mm)
   If the shaft bore is ≤ 0.40 d, the expression $1\text{-}d_i^4/d^4$ may be taken as 1.0
   F = factor for type of propulsion installation
   = 95 for intermediate shaft in turbine installation, diesel installation with hydraulic (slip type) couplings, electric propulsion installation
   = 100 for all other diesel installations and propeller shafts
   k = factor for particular shaft design features, see item 5
   n_0 = shaft speed (rpm) at rated power
   P = rated power (kW) transmitted through the shaft (losses in bearings shall be disregarded)
   σ_B = specified minimum tensile strength (MPa) of shaft material, see item 2.

   The diameter of the propeller shaft located forward of the inboard stern tube seal may be gradually reduced to the corresponding diameter for the intermediate shaft using the minimum specified tensile strength of the propeller shaft in the formula and recognising any limitation given in item 2.

4) Permissible torsional vibration stresses:

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

   Torsional vibration calculations shall 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.

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

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

$$\pm\tau_C = \frac{\sigma_B + 160}{18} \cdot c_K \cdot c_D \cdot 1.38 \qquad 0.9 \le \lambda < 1.05$$

where

τ_C = stress amplitude (MPa) due to torsional vibration for continuous operation
σ_B = specified minimum tensile strength (MPa) of shaft material, see item 2
c_K = factor for particular shaft design, see item 5
c_D = size factor, = $0.35 + 0.93 \cdot d_o^{-0.2}$
d = actual shaft outside diameter (mm)
λ = speed ratio = n/n_0
n = speed (rpm) under consideration
n_0 = speed (rpm) of shaft at rated power.

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

In this context, “rapidly” means within just a few seconds, ≈ 4-5 seconds, both upwards and downwards. If this is exceeded, flanged shafts (except propeller flange) shall be designed with a stress concentration factor less than 1.05, see Guidance note below. Alternatively, a calculation method which is taking into account the accumulated number of load cycles and their magnitude during passage of the barred speed range, may be used, see Guidance note to B201.

Guidance note:

This may be obtained by means of a multi-radii design such as e.g. starting with r_1 = 2.5 d tangentially to the shaft over a sector of 5°, followed by r_2 = 0.65 d over the next 20° and finally r_3 = 0.09 d over the next 65°(d = actual shaft outside diameter).

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Restricted speed ranges in normal operating conditions are not acceptable above λ = 0.8. Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships shall enable safe navigation.

The limits of the barred speed range shall be determined as follows:

— The barred speed range shall cover all speeds where τ_C is exceeded. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered.
— The tachometer tolerance (usually $0.01 \cdot n_0$) has to be added in both ends.
— At each end of the barred speed range the engine shall be stable in operation.

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

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

where:

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

5) Table B5 shows k and c_K factors for different design features.

Transitions of diameters shall be designed with either a smooth taper or a blending radius.

Guidance note:

For guidance, a blending radius equal to the change in diameter is recommended.

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Table B5 k and c_K factors for different design features

Intermediate shafts with						*Thrust shafts external to engines*		*Propeller shafts*		
Integral coupling flange [1] and straight sections	Shrink fit coupling [2]	Keyway, tapered connection [3)4)]	Keyway, cylindrical connection [3)4)]	Radial hole [5]	Longitudinal slot [6]	On both sides of thrust collar [1]	In way of bearing when a roller bearing is used	Flange mounted [1] or keyless taper fitted propellers [8]	Key fitted propellers [8]	Between forward end of aft most bearing and forward stern tube seal
k = 1.0	1.0	1.10	1.10	1.10	1.20	1.10	1.10	1.22	1.26	1.15
c_K = 1.0	1.0	0.60	0.45	0.50	0.30 [7]	0.85	0.85	0.55	0.55	0.80

Footnotes

1) Fillet radius shall not be less than 0.08 d.
2) k and c_K refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter shall 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 0.2 d 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 not to exceed 0.3 d.
The intersection between a radial and an eccentric axial bore (see Fig.1) is not covered by this method.
6) Subject to limitations as slot length (l)/outside diameter < 0.8, and inner diameter (d_i)/outside diameter < 0.8 and slot width (e)/outside diameter >0.10. The end rounding of the slot shall not be less than e/2. An edge rounding should preferably be avoided as this increases the stress concentration slightly.
The k and c_K values are valid for 1, 2 and 3 slots, i.e. with slots at 360°, respectively 180° and 120° apart.
7) c_K = 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 c_K is desired, the actual stress concentration factor (scf) shall be documented or determined from the formulae in item 6. In which case:
c_K = 1.45/scf. Note that the scf is defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress (determined for the bored shaft without slots).
8) Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2.5 times the required diameter.

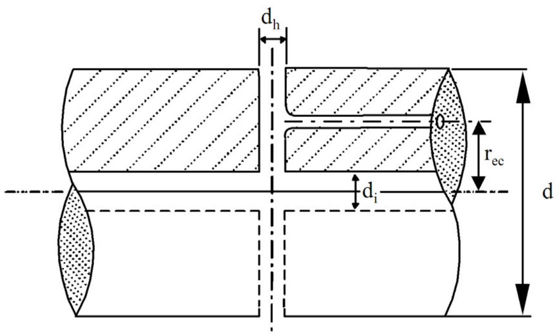


Fig. 1
Intersection between a radial and an eccentric axial bore

6) **Notes:**

A. Shafts complying with this method satisfy the load conditions in 202.

a) Low cycle fatigue criterion (typically < 10^4), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque if applicable. This is addressed by the formula in item 3.

b) High cycle fatigue criterion (typically >10^7), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses. For limits for torsional vibration stresses see item 4.
The influence of reverse bending stresses is addressed by the safety margins inherent in the formula in item 3.

c) The accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation is addressed by the criterion for transient stresses, item 4.

B. Explanation of k and c_K.

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

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

$$c_K \approx \frac{1.45}{scf} \quad \text{and} \quad k \approx \left(\frac{scf}{1.45}\right)^x$$

where the exponent x considers low cycle notch sensitivity.

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

— The size factor c_D being a function of diameter only does not purely represent a statistical size influence, but rather a combination of this statistical influence and the notch sensitivity.

The actual values for k and c_K are rounded off.

C. Stress concentration factor of slots

The stress concentration factor (scf) at the end of slots can be determined by means of the following empirical formulae using the symbols in Footnote 6) in Table B5:

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

This formula applies to:

— slots at 120°, 180° or 360° apart
— slots with semicircular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.
— slots with no edge rounding (except chamfering), as any edge rounding increases the scf slightly.

$\alpha_{t(hole)}$ represents the stress concentration of radial holes (in this context e = hole diameter), and can be determined from:

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

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

B 300 Flange connections

301 In 300 some relevant kinds of flange connections for shafts are described with regard to design criteria. Note that K_A in this context means the highest value of the normal- or misfiring K_A and K_{AP} and K_{Aice}.

In 302 and 303 the parameter d is referred to as the required shaft diameter for a plain shaft without inner bore. This means the necessary diameter for fulfilling whichever shaft dimensioning criteria are used, see 201. For certain stress based criteria the necessary diameter is not directly readable. In those cases the necessary diameter can be found by iteration, but in practice it is better to apply the parameter d as the actual diameter.

302 Flanges (except those with significant bending such as pinion and wheel shafts and propeller- and impeller fitting) shall have a thickness, t at the outside of the transition to the (constant) fillet radius, r, which is not less than:

$$t = \frac{d}{4\left(1 + 2\ \frac{r}{d}\right)^2}$$

d = the required plain, solid shaft diameter, see 301
r = flange fillet radius.

For multi-radii fillets the flange thickness shall not be less than 0.2 d.

In addition, the following applies:

— recesses for bolt holes shall not interfere with the flange fillet, except where the flanges are reinforced correspondingly
— for flanges with shear bolts or shear pins:

$$t \geq \frac{1}{2}\ d_b\ \frac{\sigma_{y,\ bolt}}{\sigma_{y,\ flange}}$$

d_b = diameter of shear bolt or pin
$\sigma_{y,bolt}$ = yield strength of shear bolt or pin
$\sigma_{y,flange}$= yield strength of flange

303 Flanges with significant bending as pinion and wheel shafts, and propeller and impeller fittings shall have a minimum thickness of:

$$t = \frac{d}{3\left(1 + 2\ \frac{r}{d}\right)^2}$$

d = the required plain, solid shaft diameter, see 301
r = flange fillet radius.

For multi-radii fillets the flange thickness shall not be less than 0.25 d. In addition, the following applies:

— recesses for bolt holes shall not interfere with the flange fillet, except where the flanges are reinforced correspondingly
— for flanges with shear bolts or shear pins:

$$t \geq \frac{1}{2}\ d_b\ \frac{\sigma_{y,\ bolt}}{\sigma_{y,\ flange}}$$

d_b = diameter of shear bolt or pin
$\sigma_{y,bolt}$ = yield strength of shear bolt or pin
$\sigma_{y,flange}$ = yield strength of flange

304 Torque transmission based on combinations of shear or guide pins or expansion devices and pre-stressed friction bolts shall fulfil:

A. The friction torque T_F shall be at least twice the repetitive vibratory torque T_v, i.e.:

$$T_F = \frac{\mu\ D\ F_{bolts}}{2000} \geq 2\ T_v \qquad \text{(Nm)}$$

μ = Coefficient of friction, see 307
T_v = $(K_A - 1)\ T_0$ for geared plants (for continuous operation) (Nm)
T_v = $(K_{Aice} - 1)\ T_0$ for ice class notations (Nm)
Highest value of T_v in the entire speed range for continuous operation (i.e. not transient speed range) for direct coupled plants. See torsional vibration in Ch.3 Sec.1 G300 and G400
D = Bolt pitch circle diameter (PCD) (mm)
F_{bolts} = The total bolt pre-stress force of all n bolts (N)

Bolt pre-stress limited as in 308.

B. Twice the peak torque T_{peak} minus the friction torque (see A. above) shall not result in shear stresses beyond the shear yield strength ($\sigma_y/(\sqrt{3})$) of the n ream fitted pins or expansion devices, i.e.:

$$2\ T_{peak} - T_F \leq \frac{\pi\ n\ D\ d_b^2\ \sigma_y}{8 \cdot 10^3\ \sqrt{3}} \qquad \text{(Nm)}$$

T_{peak}= Higher value of (Nm):
- $K_{AP}\ T_0$ or
- $K_{Aice}\ T_0$ or
- $T + T_v$ in the entire speed range considering also normal transient conditions
D = Bolt pitch circle diameter (PCD) (mm)
d_b = Bolt shear diameter (mm)

Guidance note:

T_v in normal transient conditions means with prescribed or programmed way of passing through a barred speed range.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

305 Torque transmission based on n flange coupling bolts mounted with a slight clearance (e.g.< 0.1 mm) and tightened to a specified pre-stress σ_{pre} shall fulfil the following requirements:

— the friction torque shall be at least twice the repetitive vibratory torque (including normal transient conditions), see 304 A.
— bolt pre-stress limited as in 308
— the shear stress τ due to twice the peak torque minus the friction torque combined with the pre-stress σ_{pre} shall not exceed the yield strength σ_y, i.e.:

$$\sqrt{\sigma_{pre}^2 + 3\ \tau^2}\ \leq \sigma_y$$

τ = Shear stress in bolt,

calculated as $\tau = \dfrac{8\ (2\ T_{peak} - T_F)10^3}{D\ \pi\ n\ d_b^2}$

σ_{pre} = Specified bolt pre-stress,

calculated as $\sigma_{pre} = \dfrac{4\ F_{bolts}}{\pi\ n\ d_b^2}$

T_{peak}= Peak torque, see 304 B.

306 Torque transmission based on ream fitted bolts only, shall fulfil the following requirements:

— the bolts shall have a light press fit
— the bolt shear stress due to two times the peak torque T_{peak}, (see 304 B) minus the friction torque T_F, shall not exceed 0.58 σ_y
— the bolt shear stress due to the vibratory torque T_V, for continuous operation shall not exceed σ_y/8.

This means that the diameter of the n fitted bolts shall fulfil the following criteria:

$$d_b \geq 66\sqrt{\frac{2T_{peak} - T_F}{nD\sigma_y}}$$

and

$$d_b \geq 143\sqrt{\frac{T_V}{nD\sigma_y}}$$

Ream fitted bolts may be replaced by expansion devices provided that the bolt holes in the flanges align properly.

Guidance note:

Ream fitted bolts with a light press fit means that the bolts when having a temperature equal to the flange, cannot be mounted by hand. A light pressing force or cooling should be necessary.

In order to facilitate later removal of the bolts it is important that the interference between the bolts and corresponding holes are not excessive. It should only be a few 1/100 mm, i.e. just more than the contraction of the diameter due to the pre-tightening. Therefore, direct contact with liquid nitrogen for cooling the bolts is unnecessary and could lead to cracks in the bolts. It is also beneficial to use bolts which are made from somewhat harder material than the shaft flange is made of (>50HB).

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

307 Torque transmission based on only friction between mating flange surfaces shall fulfil a minimum friction torque of $2T_{peak}$. The coefficient of friction, μ shall be 0.15 for steel against steel and steel against bronze, and 0.12 for steel against nodular cast iron. Other values may be considered for especially treated mating surfaces. The bolt pre-stress is limited as given in 308.

$$2T_{peak} \leq \left(\frac{\mu\ D\ F_{bolts}}{2000}\quad (Nm)\right)$$

D = Bolt pitch diameter (mm)
F_{bolts} = The total bolt pre-stress force of all n bolts
T_{peak} = Peak torque, see 304 B.

308 Bolts may have a pre-stress up to 70% of the yield strength in the smallest section. However, when using 10.9 or 12.9 bolts the thread lubrication procedure has to be especially evaluated, and only tightening by twist angle or better is accepted (e.g. by elongation measurement). If rolled threads, the pre-stress in the threads may be increased up to 90% of the yield strength.

In corrosive environment the upper acceptable material tensile strength is 1350 MPa.

In order to maintain the designed bolt pre-stress under all conditions, these percentages are given on the condition that the peak service stresses combined with the pre-stress do not exceed the yield strength. The bolts shall be designed under consideration of the full thrust and bending moments including reversing. For bending moments on water jet impeller flanges, see F301 item 2.

The length of the female threads shall be at least

0.8 d $\sigma_{ybolt}/\sigma_{yfemale}$

where d is the outside thread diameter and the ratio compensates for the difference in yield strength between the bolt and the female threads.

This requirement is valid when the above mentioned pre-stress is utilised, otherwise a proportional reduction in required thread length may be applied.

B 400 Shrink fit connections

401 *General requirements for all torque transmitting shrink fit connections, including propeller fitting.*

1) The shrink fit connections shall be able to transmit torque and axial forces with safety margins as given in 402 and 403. This shall be obtained by a certain minimum shrinkage amount.

   If the shrunk-on part is subjected to high speeds (e.g. tip speed >50 m/s), the influence of centrifugal expansion shall be considered.

   The following load conditions shall be considered:

   A. In the full speed range (>90%):

   — The rated torque T_0 including any permitted intermittent overload. When combined with the vibratory torque in misfiring condition the rated torque may be reduced proportional with the ratio remaining cylinders/number of cylinders.
   — The highest temporary vibratory torque T_{V0T} in the full speed range. This shall consider the worst relevant operating conditions, e.g. such as sudden misfiring (one cylinder with no injection) and cylinder unbalance (see Ch.3 Sec.1 G301 e). For determination of the vibratory torque in the misfiring condition it is necessary to consider the steady state vibrations in the full speed range regardless of whether the speed range is barred for continuous operation due to torsional vibrations or other operational conditions.
   — For any ice class notation the impact load shall be considered as a temporary vibratory torque: $(K_{Aice}-1)\cdot T_0$.
   — The axial forces such as propeller thrust Th and/or gear forces. The nut force shall be disregarded.
   — For ice class notation the highest axial force (Th_{ice}) in the applicable ice rules.
   — The axial force due to shrinkage pressure at a taper.

   B. At a main resonance (applicable to direct coupled diesel engines):

   — The mean torque T at that resonance.
   — The steady state vibratory torque T_{Vres} regardless if there is a barred speed range.
   — By convention the propeller thrust, any thrust due to

ice impact, the nut force, and the axial force due to shrinkage pressure at the taper shall be disregarded.

Guidance note:

The peak torques when reversing at main resonance are not used in this context and that condition is assumed covered by the required partial safety factors.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

2) The minimum and maximum shrinkage amounts shall be correlated to the measurement that shall be applied for verification. For elements with constant external diameter, diametrical expansion is preferred. Otherwise the pull up length (wet mounting) or the push up force (dry mounting) shall be specified. The clearance of an intermediate sleeve is also to be considered.

3) The taper is normally not to be steeper than 1:20. However, taper of cone as steep as 1:15 is acceptable, provided that a more refined mounting procedure and or a higher safety factor than given in the rules is applied.

4) For tapered connections steeper than 1:30 and all propeller cone mountings where a slippage may cause a relative axial movement between the two members, the axial movement shall be restricted by a nut secured to the shaft with locking arrangement. Alternatively a split fitted ring with locking arrangement may be used.

5) Tapered connections shall be made with accuracy suitable to obtain the required contact between both members. Normally the minimum contact on the taper is 70% when a toolmaker's blue test is specified. Non-contact bands (except oil grooves) extending circumferentially around the hub or over the full length of the hub are not acceptable. At the big end there shall be a full contact band of at least 20% of the taper length.

6) The coefficient of friction μ shall be taken from the table below, unless other values are documented by tests.

Table B6 Static coefficients of friction, μ

Application	*Hub material (shaft material = steel)*		
	Steel	*Cast iron or nodular cast iron*	*Bronze*
Oil injection	0.14	0.12	0.13
Dry fit on taper	0.15	-	0.15
Glycerine injection (parts carefully degreased) [1)]	0.18	0.16	0.17
Heated in oil	0.13	0.10	-
Dry heated/cooled (parts not degreased or protected vs. oil penetration; nor high shrinkage pressure applied)	0.15	0.12	-
Dry heated/cooled (parts degreased and protected vs. oil penetration; or high shrinkage pressure applied)	0.20	0.16	-
Special friction coating	To be specially approved		

1) Marking on coupling/ propeller that glycerine shall be used

402 *Connections other than propeller.*

The following is additional to requirements in 401:

1) The friction capacity shall fulfil:

A. In the full speed range:

Required torque capacity (kNm)
$T_{C1} = 1.8 \cdot T_0 + 1.6 \cdot T_{V0T}$

(If $T_{V0T} < (K_{Aice}-1)\cdot T_0$, replace T_{V0T} by $(K_{Aice}-1)\cdot T_0$)
The minimum value for T_{C1} is $2.5\cdot T_0$.
Tangential force (kN) $F_T = 2 \cdot T_{C1}/D_S$
(D_S is shrinkage diameter (m), mid-length if tapered.)

Axial force (kN):
$F_A = p\cdot\pi\cdot D_S\cdot L\cdot\theta\cdot 10^3 \pm Th$
(replace Th with Th_{ice} if the latter results in a higher F_A)
(in gearboxes, replace Th with the higher value of $K_{AP}\cdot F_{Agear}$ and $K_{Aice}\cdot F_{Agear}$)

Sign convention:

+ for axial forces pulling off the cone such as propellers with pulling action including thrusters and pods with dual direction of rotation and controllable pitch propeller.

– for axial forces pushing up the cone such as propellers with pushing action.

p = surface pressure (MPa)
L = effective length (m) of taper in contact in axial direction disregarding (i.e. not subtracting) oil grooves and any part of the hub having a relief groove
θ = half taper, e.g. taper =1/30, θ= 1/60).

With friction force (kN): $F_{FR} = p\cdot\mu\cdot\pi\cdot D_S\cdot L\cdot 10^3$
the necessary surface pressure p (MPa) can be determined by:

$$p = \frac{\sqrt{F_T^2 \cdot \left(1-\frac{\theta^2}{\mu^2}\right) + Th^2} \pm Th \cdot \frac{\theta}{\mu}}{\mu \cdot \pi \cdot D_S \cdot L \cdot 10^3 \cdot \left(1-\frac{\theta^2}{\mu^2}\right)}$$

Sign convention as above.

B. At a main resonance:

Torque capacity (kNm): $T_{C2} = 1.6\cdot(T + T_{Vres})$

The necessary surface pressure p (MPa) can be determined by:

$$p = \frac{2 \cdot T_{C2}}{\pi \cdot \mu \cdot D_S^2 \cdot L \cdot 10^3}$$

The highest value determined by A and B applies.
Coefficient of friction according to Table B6.

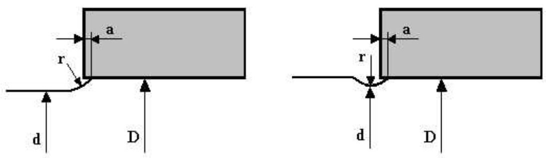
2) Fretting under the ends of shrink fit connections has to be avoided in general. However, very light fretting is accounted for by notch factors see Classification Note 41.4 item 6.5.

In particular for a shrinkage connection with a high length to diameter ratio (>1.5) or if it is subjected to a bending moment, special requirements may apply in order to prevent fretting of the shaft under the edge of the outer member. This may be a relief groove or fillet, higher surface pressure, etc.

Guidance note:

If the surface pressure at the torque end times coefficient of friction is higher than the principal stress variation at the surface, $\sigma < p\,\mu$ (see Fig.2 in Sec.2), fretting is not expected. Other surface pressure criteria may also be considered. If such surface pressure or friction cannot be achieved, it may be necessary to use a relief or a groove.

The groove may be designed as indicated below:



A good choice is D = 1.1 d and r = 2 (D – d) and an axial overshoot at near zero but not less than zero.

Other ways of preventing fretting under the edge of the hub are a relief groove in the hub or a tapered hub outer diameter. However, these alternatives need to be documented by means of detailed analysis as e.g. finite element method calculations.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

3) The permissible stress due to shrinking for the outer member (index "o") depends on the nature of the applied load, coupling design and material. For ductile steels the equivalent stress (von Mises) may be in the range 70% to 80% of the yield strength σ_{yo} for demountable connections and 100% and even some plastic deformation for permanently fitted connections (see below).

   The permissible stress due to shrinking at the outer diameter or at any other critical section (e.g. axial and radial bore intersection) of the inner member (i.e. the shaft, index "i") shall not exceed 50% of the yield strength σ_{yi}.

4) The shrinkage amounts shall be calculated under consideration of the surface roughness as follows:

   ΔD_{min} = minimum shrinkage amount due to tolerances or pull-up distance, minus

   $0.8\ (R_{zi} + R_{zo}) \approx 5\ (R_{ai} + R_{ao})$ (mm)

   ΔD_{max} = maximum shrinkage amount due to tolerances or pull-up distance, minus

   $0.8\ (R_{zi} + R_{zo}) \approx 5\ (R_{ai} + R_{ao})$ (mm).

   R_z = "ten point height" surface roughness (mm) as defined in ISO4287/1 for shaft and hub, respectively.
   R_a = "arithmetical mean" surface roughness (mm) as defined in ISO4287/1 for shaft and hub, respectively.

   The lower value shall be used for calculation of the required friction torque. The upper value shall be used for calculation of stresses in the inner and outer members. For tapered connections the shrinkage amounts shall be converted to pull up lengths
   (Pull-up distance = $\Delta D / 2\theta$, where 2θ is the taper of cone).

5) The following applies for shrinking within the elastic range and both inner and outer member made of steel. The minimum and maximum shrinkage pressures (MPa) are:

   $$p_{min} = (\Delta D_{min}/D_S)\ (E/K)\ 10^{-3}$$
   $$p_{max} = (\Delta D_{max}/D_S)\ (E/K)\ 10^{-3}$$

   The pull-up lengths (mm) are:

   $$\delta_{min} = p_{min} \frac{10^3 D_s}{2\theta} \frac{K}{E}$$

   $$\delta_{max} = p_{max} \frac{10^3 D_s}{2\theta} \frac{K}{E}$$

   The corresponding pull-up force F_{pull} can be estimated as

   $$F_{pull} = p \cdot \pi D_s L\ (\theta + \mu_{pull})\ 10^3 \text{ (kN)}$$

   μ_{pull} = Coefficient of friction during pull-up.

   The diametrical expansions are (mm):

   $$\Delta D_{omax} = p_{max} \frac{10^3 D_s}{E} \frac{2Q_o}{1 - Q_o^2}$$

   $$\Delta D_{omin} = p_{min} \frac{10^3 D_s}{E} \frac{2Q_o}{1 - Q_o^2}$$

   E = $2.05 \cdot 10^5$ MPa
   K = $(1 + Q_i^2)/(1 - Q_i^2) + (1 + Q_o^2)/(1 - Q_o^2)$
   Q_i = inner diameter of inner member/D_S
   Q_o = D_S/outer diameter of outer member

   The minimum shrinkage pressure shall not be less than the necessary pressure p as determined in item 1.

   The equivalent (von Mises) stress in the outer member is (MPa):

   $$\frac{\sqrt{3 + Q_o^4}\ p_{max}}{1 - Q_o^2}$$

   and shall not exceed the permissible stress as given in item 3 above.

   The stress calculation of the inner sleeve shall take any expansion sleeve or compression liner influence into account.

   In the case of several members shrunk on together, and all being within the elastic range, the superposition principle shall used.

6) The following applies to shrinking with a certain amount of plastic deformation in the outer member applicable to parts that are not intended to be disassembled. The simplified approach given here is valid for both members being made of steel and solid inner member, and based on modified Tresca criterion. If these conditions are not fulfilled, a more detailed analysis applies.

   As specified in item 3 above, the stresses in the inner member (shaft) due to shrinking shall not exceed 50% of the yield strength σ_{yi}. Thus the shrinkage pressure is limited to:

   $$p_{i\ lim} = \sigma_{yi} / \sqrt{3}$$

   In order to keep a safety factor of 1.25 versus full plastic deformation of the outer member the shrinkage pressure is limited to:

   $$p_{o\ lim} = 1.6\ \sigma_{yo} / \sqrt{3} \text{ for } Q_o < 0.368$$
   $$p_{o\ lim} = -1.6 \ln(Q_o)\ \sigma_{yo} / \sqrt{3} \text{ for } Q_o > 0.368$$

   The extent of permissible plastic deformation ζ_p (i.e. the ratio between the outer diameter of the plastically deformed zone and D_S) is limited by 2 criteria:

   1) $2 \ln(\zeta_p) - (Q_o\ \zeta_p)^2 + 1 = \sqrt{3}\ p_p/\sigma_{yo}$
      where p_p is the permissible shrinkage pressure and is the smaller value of $p_{o\ lim}$ and $p_{i\ lim}$.

   2) $\zeta_p = (0.7\ Q_o^2 + 0.3)^{1/2} / Q_o$ in order to limit the plastically deformed cross section area to 30% of the full cross section.

   The actual minimum and maximum extents of plastic deformation are calculated as:

   $$\zeta_{min,\ max} = 0.931\ (E/\sigma_{yo})^{1/2}\ (\Delta D_{min,\ max}/D_S)^{1/2}$$

   ζ_{min} is used to calculate the minimum shrinkage pressure as:

   $$p_{min} = \sigma_{yo}\ (1 + 2 \ln(\zeta_{min}) - (Q_o\ \zeta_{min})^2)/\sqrt{3}$$

   ζ_{max} shall not exceed the permissible value ζ_p.

403 Propeller to shaft connections

The following is additional to 401:

1) The friction capacity shall fulfil the following at a temperature of 35°C:

A. In the full speed range:

Required torque capacity (kNm)
$T_{C1} = 2.0 \cdot T_0 + 1.8 \cdot T_{V0T}$
(If $T_{V0T} < (K_{Aice}-1) \cdot T_0$, replace T_{V0T} by $(K_{Aice}-1) \cdot T_0$)
The minimum value for T_{C1} is $2.8 \cdot T_0$.
Tangential force (kN) $F_T = 2 \cdot T_{C1}/D_S$
(D_S is shrinkage diameter (m), mid-length if tapered.)

Axial force (kN) $F_A = p \cdot \pi D_S \cdot L \cdot \theta 10^3 \pm Th$

Sign convention:

\+ for propellers with pulling action including thrusters and pods with dual direction of rotation.
– for propellers with pushing action.

Replace Th with Th_{ice} if this results in a higher F_A.

p = surface pressure (MPa)
L = effective length (m) of taper in contact in axial direction disregarding (i.e. not subtracting) oil grooves and any part of the hub having a relief groove
θ = half taper, e.g. taper =1/30, θ = 1/60.

With friction force (kN) $F_{FR} = p \cdot \mu \cdot \pi \cdot D_S \cdot L \cdot 10^3$
the necessary surface pressure p_{35T} (MPa) at 35°C for safe torque transmission can be determined by:

$$p = \frac{\sqrt{F_T^2 \cdot \left(1 - \frac{\theta^2}{\mu^2}\right) + Th^2} \pm Th \cdot \frac{\theta}{\mu}}{\mu \cdot \pi \cdot D_S \cdot L \cdot 10^3 \cdot \left(1 - \frac{\theta^2}{\mu^2}\right)}$$

Sign convention as above.

B. At a main resonance:

Torque capacity (kNm) $T_{C2} = 1.8 \cdot (T + T_{Vres})$
The necessary surface pressure p (MPa) can be determined by:

$$p_{35T} = \frac{2 \cdot T_{C2}}{\pi \cdot \mu \cdot D_S^2 \cdot L \cdot 10^3}$$

The higher value from A and B shall be used.
Coefficient of friction according to Table B6.

C. Prevention of detrimental fretting under the hub at the top of the shaft cone:

Regardless required surface pressure for torque transmission p_{35T}, the minimum nominal surface pressure at the top of the shaft cone p_{35min} shall not be less than 30 MPa for bronze propellers and 50 MPa for steel propellers. Normally there is no relief groove in the upper end of the hub, and this criterion applies at the same position as the 70% of yield strength criterion. However, special consideration may be given for proven designs (e.g. with relief groove) dealing with the risk of fretting in another, adequate way.

2) For propeller without intermediate sleeve, the corresponding required pull-up length (mm) at 35°C is the greater value of δ_{35T} for torque transmission and δ_{35min} for reducing the risk of fretting:

$$\delta_{35T} = p_{35T} \cdot \frac{D_S \cdot 10^3}{2 \cdot \theta} \cdot \left[\frac{1}{E_h} \cdot \left(\frac{1+Q_o^2}{1-Q_o^2} + \nu_h\right) + \frac{1}{E_S} \cdot \left(\frac{1+Q_i^2}{1-Q_i^2} - \nu_S\right)\right]$$

$$\delta_{35\min} = p_{35\min} \cdot \frac{(D_S + L \cdot \theta) \cdot 10^3}{2 \cdot \theta} \cdot \left[\frac{1}{E_h} \cdot \left(\frac{1+Q_{oB}^2}{1-Q_{oB}^2} + \nu_h\right) + \frac{1}{E_S} \cdot \left(\frac{1+Q_{iB}^2}{1-Q_{iB}^2} - \nu_S\right)\right]$$

where

E_h = the modulus of elasticity of the propeller hub
E_S = the modulus of elasticity of shaft.

Modulus of elasticity to be used:

For Cu1 (Mn-bronze) and Cu2 (Mn-Ni-bronze):
$1.05 \cdot 10^5$ MPa

For Cu3 (Ni-Al-bronze) and Cu4 (Mn-Al-bronze):
$1.15 \cdot 10^5$ MPa

For steel:
$2.05 \cdot 10^5$ MPa

ν_h = the Poisson's ratio for hub
ν_S = the Poisson's ratio for shaft.

Poisson's ratios to be used:

For bronze: 0.33
For steel: 0.29

Q_o = the ratio between D_S and the mean outer diameter of propeller hub at the axial position corresponding to D_S
Q_i = the ratio between the inner diameter of the shaft and D_S.

The additional index "B" refers to the corresponding ratios at the big end of the cone. Note that if the hub has a relief groove at the big end, this is the nearest section that is not relieved.

The minimum pull-up length (mm) at temperature t (t < 35°C) is the greater value of:

$$\delta_{t-T} = \delta_{35T} + \frac{D_S \cdot 10^3}{2 \cdot \theta} \cdot (\alpha_b - \alpha_S) \cdot (35 - t)$$

and

$$\delta_{t-\min} = \delta_{35\min} + \frac{(D_S + L \cdot \theta) \cdot 10^3}{2 \cdot \theta} \cdot (\alpha_b - \alpha_S) \cdot (35 - t)$$

where α is the coefficient of linear expansion

For steel: $\alpha_S = 12.0 \cdot 10^{-6}$ 1/°C
For all copper-based alloys: $\alpha_b = 17.5 \cdot 10^{-6}$ 1/°C

3) For propeller without intermediate sleeve, the maximum equivalent uniaxial stress in the hub (calculated at the big end) at 0°C based on the von Mises criterion shall not exceed 70% of the yield point or 0.2% proof stress (0.2% offset yield strength) for the propeller material based on the specified value for the test piece.

Note that if the hub has a relief groove at the big end, this criterion applies to the nearest section that is not relieved.

Maximum permissible surface pressure (MPa) at 0°C:

$$p_{\max} = \frac{1 - Q_{OB}^2}{\sqrt{3 + Q_{OB}^4}} \cdot (0.7 \cdot \sigma_y)$$

Corresponding maximum permissible pull-up length (mm) at 0°C:

$$\delta_{\max} = \frac{p_{\max}}{p_{35\min}} \cdot \delta_{35\min}$$

Corresponding maximum permissible pull-up length (mm) at temperature t:

$$\delta_{t-\max} = \delta_{\max} - \frac{(D_S + L\cdot\theta)\cdot 10^3}{2\cdot\theta}\left(\alpha_b - \alpha_S\right)\cdot t$$

B 500 Keyed connections

501 Keyed connections are only suitable for unidirectional torque drives with low torque amplitudes and insignificant bending stresses. Conditionally, keyed connections may be used also for dual directional torque drives (see 503).

The following items shall be checked:

- shrinkage pressure to avoid detrimental fretting, see 502
- shear stress in the key, see 503
- surface pressure at shaft keyway side, hub keyway side and key side, see 503
- fatigue strength of the shaft, see 200
- strength of hub, see 504
- intersection with other notches, see 505.

Tapered connections shall not be steeper than 1:12. However, taper of cone as steep as 1:10 is acceptable, provided that a more refined mounting procedure and/or a higher safety factor than given in the rules are applied.

Tapered connections steeper than 1:30 as well as any keyed connection with axial forces, shall be secured against axial movement.

502 In order to avoid detrimental fretting on the shaft under the edge of the hub, there shall be a certain minimum interference fit between shaft and hub. For key connections subjected to bending moments a tight fit is required. The criteria, which also apply to propeller connections, are given in 402 item 2 and Classification Note 41.4 item 6.5. For key connections transmitting torque only, there shall be a minimum interference fit (friction torque) that corresponds to the applicable vibratory torque for continuous operation with a safety factor of 2.0. This means a friction torque (Nm):

$$T_F \geq 2.0\ T_V$$

that may be approximated as the highest value of:

- $2\ (K_A - 1)\ T_0$ for geared plants
- $2\ (K_{Aice} - 1)\ T_0$ for plants with ice class
- $2\ T_v$ for direct coupled plants.

When calculating shrink fit pressures between cylindrical members with one or two keyways, the real pressure is less than the calculated due to relief caused by the keyways. This influence may be approximated by a reduction factor of 0.8. With these assumptions and solid shaft with steel hub the necessary amount of shrinkage Δd (mm) is:

$$\Delta d = T_F/(128\ d\ L\ \mu\ (1 - (d/D)^2))$$

Δd = shrinkage amount (mm) estimated as minimum amount due to specified tolerances or pull-up distance, minus $0.8\ (R_{z\text{-shaft}} + R_{z\text{-hub}}) \approx 5\ (R_{a\text{-shaft}} + R_{a\text{-hub}})$
d = shaft diameter (mm)
D = outer diameter of hub (mm)
L = hub length (mm)
μ = coefficient of friction (0.15 may be used)
R_a, R_z = surface roughness (mm) for shaft and hub, respectively, see 402 4).

However, smaller interference is acceptable when the shaft is dimensioned to sustain some fretting.

For tapered connections the minimum friction torque shall be provided by means of either a specified push up force or a specified pull up length. The latter shall be consistent with Δd above. However, if test pull-up is carried out, the subtraction of the surface roughness term may be omitted.

503 The key shear stress and the surface pressures in the shaft and hub keyways, respectively are calculated on the basis of the applied repetitive peak torque T_{peak} (see 304 B) minus the actual friction torque T_F according to 502. Furthermore, the uneven distribution of the load along a key with a length beyond $L_{eff}/d = 0.5$ is considered empirically. If $L_{eff}/d < 0.5$ then $L_{eff}/d = 0.5$ shall be used in the formulae below.

Shear stress in key (MPa):

$$\tau = (T_{peak} - T_F/S)\ 2\,000\ (1 + 0.25\ (L_{eff}/d - 0.5))/(d\ L_{eff}\ b\ i)$$

Side pressure (for contact with shaft and hub):

$$\sigma = (T_{peak} - T_F/S)\ 2\,000\ (1 + 0.25\ (L_{eff}/d - 0.5))/(d\ L_{eff}\ h_{eff}\ i)$$

L_{eff} = effective bearing length of the key (mm)
b = width of key (mm)
i = number of keys, if 2 keys use i = 1.5
h_{eff} = effective height of key contact with shaft and hub, respectively i.e. key chamfer and keyway edge rounding considered.
S = 2

Permissible shear stress in key: $0.3 \cdot f_d$ times the yield strength of the key material.

Permissible side pressures: $1 \cdot f_S \cdot f_d$ times the respective yield strengths.

f_d = torque direction factor.

For unidirectional torque $f_d = 1$.

For dual directional torque with 10^3 to 10^4 reversals $f_d = 2/3$.

For 10^6 or more reversals $f_d = 1/3$.

f_S = support factor.

$f_S = 1$ for the key

$f_S = 1.2$ for the shaft

$f_S = 1.5$ for the hub

For plants with torque reversals the key shall have a tight sideways fit in both shaft and hub.

504 The tangential stresses in the hub when calculated as an ideal cylindrical member with the maximum amount of shrinkage due to tolerances shall not exceed 35% of the yield strength for steel. For bronze or austenitic steel 45% are permitted.

For tapered connections the dimensions at the upper end shall be used.

505 If a keyway intersects with another notch such as a diameter step, the semicircular part of the end should be placed fully into the shaft part with the larger diameter. If the semicircular end coincides with the fillet in the diameter step, a combination of stress concentrations shall be considered.

506 For propeller fitting the contact between hub and shaft shall be at least 70% with a full contact band at the upper end, when using toolmaker's blue. This full contact band shall be at least 0.2 d wide (excluding the trace of any hub keyway). This means that there has to be a certain distance between the top of cone and the shaft keyway, minimum 0.2 d.

For tapered couplings at least a full contact band at the upper end is required.

B 600 Clamp couplings

601 Clamp couplings shall be fitted with a key that fulfils the requirements in 500. For couplings transmitting thrust, an axial locking device shall be provided.

602 The clamp coupling bolts shall be tightened so that the coupling friction torque T_F as specified in 502 is obtained.

603 The maximum bolt stress when the peak torque (see 304) is applied shall not exceed 2/3 of the bolt yield strength.

604 The hub stress determined in a simplified way as the bolt pre-stress divided by the hub length times minimum hub thickness at the keyway, shall not exceed 40% of the yield strength of the hub material.

B 700 Spline connections

701 Spline connections shall be designed with regard to flank surface duration, shear strength and to avoid fretting (unless life time requirements allow for some). Items 702 and 703 only concern the splines, the shaft strength is dealt with in 200.

702 Spline connections are normally to be "fixed", i.e. having no axial movements in service. "Working" splines (which move axially in service) will be especially considered. Splines for normal applications shall be flank-centred and without backlash (light press fit). Tip centring and backlash is only acceptable for connections which have no reversed torques in any operation mode.

703 The following calculation procedure may be used for spline connections provided:

— Involute "half depth" splines with 30° pressure angle. ("half depth" means common tooth height equal one module).
— Mainly torque transmission, i.e. no significant additional support force. In the case of e.g. an external gear mesh force the outer member shall be supported at each end of the splines and the support shall be a tight fit. Otherwise special considerations shall be taken.
— The length to diameter ratio of the splines shall be so that torsional deflections or bending (due to external forces) deflections corresponding to a misalignment beyond 1 micron per mm spline length are avoided.
— Flank alignment tolerance shall be 0.5 micron per mm spline length for each of the male and female members.

Flank pressure criterion:

$$l\, d^2 > 6\,000\, K_A\, T_0 / HV$$

Shear stress criterion:

$$l\, d^2 > 10^4\, K_A\, T_0 / \sigma_y$$

l = the spline length (mm)
d = the pitch diameter (mm)
HV = the flank hardness of the softer member
σ_y = the yield strength of the core material (minimum of the two members)

B 800 Propeller shaft liners

801 Bronze liners shall be free from porosities and other defects and shall be designed and produced to withstand a hydraulic pressure of 2 bar without showing cracks or leakage.

802 The liner thickness in way of bearings shall not be less than:

$$t = (d + 230)/32 \text{ mm}$$

Between bearings the thickness of a continuous liner shall not be less than 0.75 t.

803 If a continuous liner is made of several lengths, the joining of the pieces shall be made by fusion through the whole thickness of the liner before shrinking. Such liners shall not contain lead.

804 If a liner does not fit the shaft tightly between the bearing portions, the space between the shaft and the liner shall be filled with a plastic insoluble non-corrosive compound.

805 Liners shall be shrunk upon the shaft by heating or hydraulic pressure, and they shall not be secured by pins.

806 Liners shall be designed to avoid water gaining access to the shaft, between the end of the liner and the propeller hub.

B 900 Shaft bearings, dimensions

901 Radial fluid bearings shall be designed with bearing pressures and hydrodynamic oil film thickness suitable for the bearing metals.

For aft stern tube bearings the nominal surface pressure (projected area) shall be below 8 bar for all running conditions including on turning gear.

For other shaft bearings the nominal surface pressure shall be below 12 bar when running in the lower speed range including on turning gear and 18 bar when running in the upper speed range.

For shaft bearings with significant pressure (>12 bar) in plants operating at very low speeds (e.g. electric drives or long term running on turning gear), hydrostatic bearings may be required.

These surface pressures apply to white metal lined bearings. For other lining metals or rubber, reinforced resins, etc., the permissible surface pressures shall be especially considered, but normally not to exceed those for white metal.

For separate thrust bearings the smallest hydrodynamic oil film thickness, taking into consideration the uneven load distribution between the pads, shall be larger than the sum of the average surface roughness of the thrust collar and pad ($R_{a,collar} + R_{a,pad}$).

902 The length of the aft stern tube bearing shall be chosen to provide suitable damping of possible whirling vibration. This means that the length is not only to be chosen with regard to the nominal surface pressure, but also result in a certain length to diameter ratio.

903 Ball and roller bearings shall have a minimum L_{10a} (ISO 281) life time that is suitable with regard to the specified overhaul intervals. The influence of the lubrication oil film may be taken into account for $L_{10a,}$ provided that the necessary conditions, in particular cleanliness, are fulfilled.

B 1000 Bearing design details

1001 Stern tube bearings shall be provided with grooves for oil, air and possible accumulation of dirt. Pipes and cocks for supply and draining of oil and air shall be fitted.

1002 Water lubricated bearings shall be provided with longitudinal grooves for water access.

B 1100 Shaft oil seals

1101 Shaft oil seals are considered on the basis of field experience or alternatively, extrapolation of laboratory tests or previous design.

C. Inspection and Testing

C 100 Certification

101 Regarding certification schemes, short terms, manufacturing survey arrangement (MSA) and important conditions, see Ch.2 Sec.2.

102 All shafts, coupling hubs, bolts, keys and liners shall be tested and documented as specified in Table C1 if not otherwise agreed in a MSA.

Table C1 Requirements for documentation and testing

Part	*Product certificate*	*Chemical composition (ladle analysis)*	*Mechanical properties*	*Ultrasonic testing*	*Crack detection 1)*	*Hydraulic testing*	*Visual and dimensional check 2)*
Shafts 3) 7) for propulsion when torque >100 kNm	NV	W	NV	W	NV	-	NV
Other shafts 3) 7) for propulsion	NV	W	W	W	W	-	NV
Shafts 3) in thrusters 8) and gear transmissions	-	W	W	W	W	-	W
Rigid couplings for propulsion when torque >100 kNm	NV	W	NV	W	W	-	NV
Other rigid couplings and rigid couplings in thrusters and gear transmission	-	W	W	W	W	-	W
Keys, bolts and shear pins	-	TR	TR	-	-	-	W4)
Propeller shaft liners		W	-	-	W5)	W6)	-

1) By means of magnetic particle inspection or dye penetrant. To be carried out in way of all stress raisers (fillets, keyways, radial holes, shrinkage surfaces on propulsion shafts etc.). If especially required due to nominal stress levels, also the plain parts shall be crack detected. No cracks are acceptable, see Ch.2 Sec.3 A202.

2) The visual inspection by the surveyor shall include checking of all stress raisers (see above) with regard to radii and surface roughness, and for plain portions, the surface roughness. It is also to include the shaft's protection against corrosion, if this is provided prior to installation onboard. Dimensional inspection to be done in way of shrinkage surfaces (actual shrinkage amount or individual dimensions shall be documented).

3) Any welds to be NDT checked (ultrasonic testing and surface crack detection) in the presence of the surveyor and documented with NV certificate.

4) Can be omitted for keys, bolts and shear pins in reduction gears and thruster. Can also be omitted for friction bolts of standard type.

5) In way of fusion between pieces.

6) Test pressure 2 bar.

7) However, not applicable for rotor shafts in generators providing electric power for propulsion.

8) Valid for propulsion, dynamic positioning and auxiliary thrusters.

C 200 Assembling in workshop

201 For shafts, hubs and liners that are assembled at the manufacturer's premises, the following shall be verified in the presence of a surveyor:

a) Liners mounted on the shaft with regard to tightness (hammer test) and that any specified space between shaft and liner is filled with a plastic insoluble non-corrosive compound.

b) Shrink fit couplings mounted on the shaft with regard to the approved shrinkage amount (diametrical expansion, pull up length, etc.). For tapered connections the contact between the male and the female part shall be verified as specified and approved.

c) Bolted connections with regard to bolt pretension.

d) Keyed connections with regard to key fit in shaft and hub.

202 Shafts for gas turbine applications, high speed side, shall be dynamically balanced.

D. Workshop Testing

D 100 General

101 Not required.

E. Control and Monitoring

E 100 General

101 The requirements in E is a summary, applicable to shafting. For further details, see Ch.9.

102 Starting interlock shall be provided, whenever shaft brake, if any, is engaged.

E 200 Indications and alarms

201 The shafting shall be fitted with instrumentation and alarms according to Table E1.

E 300 Tailshaft monitoring - TMON

301 When the following design requirements are fulfilled, the class notation **TMON** (tailshaft condition monitoring survey arrangement) may be obtained, see also Pt.7 Ch.1 Sec.6 Q of the Rules for Classification of Ships:

— the stern tube bearings are oil lubricated
— high temperature alarm is fitted on aft stern tube bearing (2 sensors or one easily interchangeable sensor located in the bearing metal near the surface, in way of the area of highest load, which normally will be the bottom area (5 to 7 o'clock) in the aft third of the bearing)
— the setting of the stern tube high temperature alarm is normally not to exceed 65°C. Higher alarm set point may be accepted upon special consideration
— the sealing rings in the stern tube sealing box must be replaceable without shaft withdrawal or removal of propeller
— arrangement for bearing wear down measurement is fitted
— the system must allow representative oil samples to be taken for analysis of oil quality under running conditions. Location where samples shall be taken shall be clearly pointed out on system drawing and test cock to be fitted with signboard. A written procedure for how to take oil samples shall be submitted
— grounding device installed.

302 A test kit for monitoring of possible water content in the stern tube lubricating oil shall be provided on board. The water content is normally not to exceed 2% by volume. If the water content above 2% is detected appropriate action shall be taken.

303 Oil lubricated propeller shafts with roller bearings arranged in the stern tube may be granted TMON see also Pt.7 Ch.1 Sec.6 Q of the Rules for Classification of Ships. Additional requirements for such arrangements are:

a) The bearing temperature shall be monitored. Two sensors (or one sensor easily interchangeable at sea) shall be fitted. Temperature alarm level should normally not exceed 90°C.

b) Vibration monitoring is required for roller bearings. Handheld probes are not accepted; magnetic, glue, screw mountings or equivalent are compulsory.

c) Vibration signal shall be measured as velocity or acceleration. Integration from acceleration to velocity is allowed.

d) The vibration analysis equipment must be able to detect fault signatures in the entire frequency range for the monitored bearing. A reference level under clearly defined operational conditions shall be established. The reference level shall be used as basis for establishing an alarm level.

e) The water contents is normally not to exceed 0.5%.

Table E1 Monitoring of shafting

	Gr 1 Indication Alarm Load reduction	*Gr 2 Automatic start of standby pump with alarm* [1)]	*Gr 3 Shut down with alarm*	*Comments*
1.0 Shafting				
Separate thrust bearings, temperature	IL or IR, HA			To be provided for shaft power > 5 000 kW. Sensor to be placed in the bearing metal or for pads, in the oil outlet. Maximum permissible temperature to be marked on the indicators.
Oil lubricated fluid film bearings, temperature	IL or IR, HA			To be provided for shaft power > 5 000 kW. Sensors to be located near the bearing surface at the area of highest load. Maximum permissible temperature to be marked on the indicators.
Stern tube lubricating oil tank, level	LA			
Stern tube lubricating oil, pressure or flow	LA			Applicable to forced lubrication.
2.0 Additional requirements for TMON				
Aft stern tube bearing, temperature	HA			See 301

Gr 1 Common sensor for indication, alarm, load reduction (common sensor permitted but with different set points and alarm shall be activated before any load reduction)
Gr 2 Sensor for automatic start of standby pump
Gr 3 Sensor for shut down

IL = Local indication (presentation of values), in vicinity of the monitored component
IR = Remote indication (presentation of values), in engine control room or another centralized control station such as the local platform/ manoeuvring console
A = Alarm activated for logical value
LA = Alarm for low value
HA = Alarm for high value
AS = Automatic start of standby pump with corresponding alarm
LR = Load reduction, either manual or automatic, with corresponding alarm, either slow down (r.p.m. reduction) or alternative means of load reduction (e. g. pitch reduction), whichever is relevant.
SH = Shut down with corresponding alarm. May be manually (request for shut down) or automatically executed if not explicitly stated above.

For definitions of LR and SH, see Ch.1 Sec.1 B116 and 117 of the Rules for Classification of Ships.

1) To be provided when standby pump is required, see B900.

F. Arrangement

F 100 Sealing and protection

101 A shaft sealing shall be provided in order to prevent water from gaining access to the internal spaces of the vessel.

102 A sealing shall be provided to prevent water from gaining access to steel shafts, unless approved corrosion resistant material is used.

103 Inboard shafts (inside the inner stern tube seal) shall be protected against corrosion. Depending on the ambient conditions, this may be provided by oil based coating, paint, or similar.

104 Electrical grounding of the propeller shaft lines is mandatory.

F 200 Shafting arrangement

201 The machinery and shafting shall be arranged so that neither external nor internal (self generated) forces can cause harmful effects to the performance of the machinery and shafting.

If shaft brake is fitted, it shall be arranged so that in case of failure in the actuating system, the brake should not be engaged.

202 The shafting system shall be evaluated for the influence of:

— thermal expansion
— shaft alignment forces
— universal joint forces
— tooth coupling reaction forces
— elastic coupling reaction forces (with particular attention to unbalanced forces from segmented elements)
— hydrodynamic forces on propellers
— ice forces on propellers, see Pt.5 Ch.1 of the Rules for Classification of Ships
— hydrodynamic forces on rotating shafts:

  i) outboard inclined propeller shafts or unshielded impeller shafts, see 301 1)

  ii) mean thrust eccentricity caused by inclined water flow to the propeller, see 301 1)

  (Normally applicable to HS, LC and NSC)
— thrust eccentricity in water jet impellers when partially air filled or during cavitation, see 301 2)
— forces due to movements of resiliently mounted machinery (maximum possible movements to be considered)
— forces due to distortion or sink-in of flexible pads.

F 300 Shaft bending moments

301 The shaft bending moments due to forces from sources as listed in 202 are either determined by shaft alignment calculations, see 400, whirling vibration calculations, see G100, or by simple evaluations. However, two of the sources in 202 need further explanations:

1) The hydrodynamic force F on an outboard shaft rotating in a general inclined water flow may be determined as

$$F = 0.87 \cdot 10^{-4}\ \eta\ v\ n\ d^2 \sin\alpha \text{ (N/m shaft length)}$$

d	= shaft diameter (mm)
n	= r.p.m. of the shaft
v	= speed of vessel (knots)
α	= angle (degrees) between shaft and general water flow direction (normally to be taken as parallel to the bottom of the vessel)
η	= “efficiency” of the circulation around the shaft. Unless substantiated by experience, it is not be taken less than 0.6.

In order to determine the bending moments along the shaft line of an outboard shaft (as well as at the front of the hub), the bending moment due to propeller thrust eccentricity shall be determined e.g. as:

$$M_b = 0.074\ \alpha\ D\ T/H \text{ (Nm)}$$

D	= propeller diameter (m)
T	= torque (Nm), which may be taken as the rated torque if low torsional vibration level
H	= propeller pitch (m) at 0.7 radius

The bending moment due to the (horizontal) eccentric thrust should be directed to add to the bending moment due to the hydrodynamic force F in the first bearing span.

2) The stochastic bending moment due to thrust eccentricity in a water jet impeller during air suction or cavitation is based on the worst possible scenario:

50% of the normal impeller thrust (F_{TH} in N) applied at the lower half of the impeller, resulting in a bending moment as:

$$M_b = 0.1\ F_{TH}\ D \text{ (Nm)}$$

D = the impeller diameter (m).

F 400 Shaft alignment

401 The subsequent items under F are only valid for propulsion plants for which approval of alignment calculations are required, see A402.

For geared plants, the calculations are only applicable for the low speed shaft line, which shall include the output gear shaft radial bearings.

402 The shaft alignment calculations shall include the following items for all relevant operating conditions, see 403:

— equipment list, i.e. manufacturer and type designation of prime mover or gear
— input data, including reference to relevant drawings. For direct coupled plants, the crankshaft calculation model shall be according to the engine designer's guidelines. The origin of the applied hydrodynamic propeller loads shall be stated (whether detailed calculations based on measured wake field have been carried out or if they are simply assumed, see Guidance Note to 404)
— list of operating conditions and the respective influence parameters, see 403 and 404
— bearing clearances
— bearing offsets from the defined reference line
— calculated bearing loads and pressures
— bearing reaction influence numbers
— graphical and tabular presentation of the shaft deflections with respect to the defined reference line
— graphical and tabular presentation of the shaft bending stresses as a result of the alignment
— difference in slope between shaft and bearing centrelines in aft sterntube bearing and if applicable, details of proposed slope-bore
— appropriate acceptance criteria, see 404
— verification data with tolerances (e.g. gap and sag and jacking loads including jack correction factors) and condition (cold or hot, submerged propeller, etc.).

403 The shaft alignment calculations have to include the following conditions:

— alignment condition (during erection of shafting)
— cold, static
— hot, static (representative for operation at zero pitch, dead slow or with turning gear)
— hot running (MCR)
— for multi prime mover plants, all relevant combinations of prime mover operation.

404 The shaft alignment calculations shall take into account the influence of:

- buoyancy of propeller
- hydrodynamic propeller loads (horizontal and vertical forces and bending moments, as applicable, see Guidance note 1 below)
- thermal rise of machinery components (including rise caused by heated tanks in double bottom and other possible heat sources)
- gear loads (horizontal and vertical forces and bending moments)
- bearing clearances and angular working position in gear bearings (if necessary to consider the consequential vertical lift), see Guidance note 2 below
- sterntube bearing wear (for bearings with high wear acceptance e.g. with water or grease lubrication)
- load distribution within sterntube bearings (in order to determine representative bearing reaction point(s))
- bearing stiffness (if substantiated by knowledge or evaluation, otherwise infinite)
- hull and structure deflections (caused by e.g. draught changes and aft peak tank filling).

The results have to show margins with respect to acceptance criteria, see 405, depending on the uncertainties related to the above.

Guidance note 1:

If no calculations of the hydrodynamic propeller loads are available, or the basis for such calculations is uncertain, a certain range should be applied. A typical range for upwards bending moment for single screw plants is -5% to +40% of the shaft torque.

If the propeller tip in some operational conditions is in the vicinity of the water line, a downward bending moment will occur. Hence, the lower limit of the range should be extended, e.g. as far as to -25%.

For multi-screw propulsion plants, the alignment calculations have to include the propeller induced horizontal loads. In such plants, the propeller induced loads are influenced by the rotational direction of the propellers, and it is recommended that hydrodynamic propeller loads are calculated based on measured wake field. In the case of no wake field analysis, ranges of +/- 30% horizontally and +/- 20% vertically may be used.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

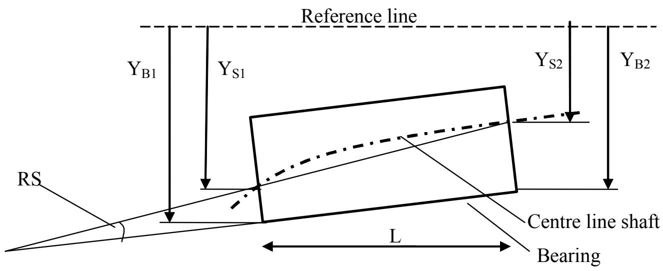
Guidance note 2:

For sensitive systems, e.g. gears with large face width, even small alignment offsets may have large influence on the gear faceload distribution. In such systems, angular position of the shaft has to be found by iteration. Vertical and horizontal offsets may be assessed by means of the vertical and horizontal forces in the previous iteration step. Bearing clearances have to be taken into account, but the oil film thickness can usually be disregarded (except for very light bearing loads). For fluid film bearings the angular working position may be estimated to 20 - 30 degrees off the direction of the force (except for very light bearing loads).

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

405 The shaft alignment has to fulfil the following acceptance criteria for all relevant operating conditions in 403:

- maximum bending stresses in shafts as limited by the shaft dimensions criteria in B200
- acceptance criteria defined by manufacturer of the prime mover, e.g. limits for bearing loads, bending moment and shear force at flange
- acceptance criteria defined by the manufacturer of the reduction gear, e.g. limits for output shaft bearing loads including their maximum difference (i.e. journal positions in gear output shaft bearings, that will influence the tooth faceload distribution)
- bearing load limits defined by bearing manufacturer and B901
- zero or very low bearing loads are acceptable if these have no adverse influence on whirling vibration
- relative nominal slope between shaft and aft sterntube white metal lined bearing should, in general, not exceed $3 \cdot 10^{-4}$ rad (0.3 mm/m) and 50% of min. diametrical bearing clearance divided by the bearing length, whichever is less (otherwise to be compensated with slope bore). For other lining metals or rubber, reinforced resins, etc., the limit for *relative nominal slope* shall be especially considered. For definition of relative nominal slope, see Figure 2
- tolerances for gap and sag less than 5/100 mm are not accepted
- the angular working position of the shaft in bearings with longitudinal oil grooves must not conflict with any running limitations specified by the bearing manufacturer.



$\Delta Y = ABS\,[(Y_{S1} - Y_{B1}) - (Y_{S2} - Y_{B2})]$

$\Delta Y < 0.5 \cdot Cd$

$RS = \Delta Y/L < 0.3$ mm/m

RS: Relative slope between bearing and shaft
Cd: Diametrical clearance between bearing and shaft
L: Length of bearing, or bearing segment if multi-slope

Fig. 2
Relative nominal slope between shaft and aft stern tube bearing

406 The tolerances in the alignment specification have to correlate with the tolerance ranges used in the calculations. The final verification of the alignment shall be carried out afloat in at least one relevant condition as mentioned in 403. In special cases, verification in running condition by means of strain gauges and/or proximity transducers may also be required. In such cases the measurement program shall be submitted for approval.

G. Vibration

G 100 Whirling vibration

101 Calculation of whirling vibration (when required, see A404) is normally restricted to determination of natural frequencies. In special cases also forced whirling may be required.

102 Calculations are normally to be made as parameter studies. Important but uncertain parameters as stiffness of aft stern tube bearing, resulting bearing load position, bearing load distribution over length (if calculating with distributed bearing reaction), entrained water on propeller, etc. shall be varied within their probable range and natural frequencies to be presented as corresponding graphs.

103 Resonance in propeller blade frequencies near the upper operating speed should be avoided. However, exceptions may be made when the vibration mode and the bearing design is so that a heavy damping is expected e.g. high bearing length to diameter ratio combined with a bouncing vibration mode.

104 Resonance with the shaft speed (1st order forward whirl) shall have a separation margin of at least 30% to the operating speed range.

G 200 Rotor vibration

201 Rotor means an assembly of a unit and the couplings and shafts, e.g. a power take off (PTO) driven shaft generator.

202 Resonance of the 1st order is normally to have a separation margin of at least 30% to the operating speed range.

G 300 Axial vibration

301 Axial vibration calculations (when required) shall take the thrust bearing stiffness into consideration, see also Ch.3 Sec.1 A601 c) and G500.

G 400 Vibration measurements

401 If vibration measurements are required, the type of instrumentation, location of pick ups, signal processing method and the measurement program shall be agreed with the Society.

H. Installation Inspection

H 100 Application

101 The requirements in H apply to inspection of installation of shafts, couplings and bearings in propulsion plants. Regarding compliant couplings, see Sec.4 and Sec.5. Unless otherwise stated, a surveyor shall attend the inspections given in H.

H 200 Assembly

201 Flange connections shall be checked with regard to:

— ream fitted bolts, light press fit
— friction bolts, pre-stress by bolt elongation.

202 Clamp couplings shall be checked with regard to tightening of the bolts. Unless otherwise approved, this shall be made by measuring elongation (applicable for through bolts).

203 Keyed connections shall be checked with regard to:

— shrinkage amount between hub and shaft (applicable to cylindrical connections)
— contact between male and female tapered members, (full contact band at upper end required)
— push up force or pull up length of tapered connections
— key tight fit in shaft and hub (applicable to reversing plants).

204 For liners mounted at the yard, see C201.

205 Keyless shrink fit connections shall be checked with regard to:

— circumferential orientation (marking) between the parts (not applicable to sleeve couplings)
— contact[1)] between male and female tapered members (not applicable for couplings certified as hub and sleeve together and contact checked at the manufacturer). As a minimum there shall be a full contact band at the big end
— shrinkage amount, verified by diametrical expansion or pull up length, whichever is approved
— draining and venting (by air).

1) For wet mounting, the contact may be improved by light grinding with a soft disc and emery paper in the hub (not the shaft). A test pull up may also be used to improve the contact.

206 Bearing clearances (for fluid film bearings) shall be recorded.

207 The protection against corrosion of inboard shafts shall be checked, see F103.

208 *Propeller fitting*

a) For flange mounted propellers, the bolt tightening shall be verified.

b) For cone mounted propellers with key, the following shall be verified:

— Contact between propeller and shaft (e.g. by means of toolmaker's blue) to be at least 70% and with full contact band at the upper end, see also 205, footnote [1)].
— Push up force or pull up length, whichever is specified in the approval.
— After final pull-up, the propeller shall be secured by a nut on the propeller shaft. The nut shall be secured to the shaft with the approved locking arrangement. Alternatively, if approved, a split fitted ring with locking arrangement may be used. The ring shall have a tight fit.
— Key fit in both shaft and hub.

c) For keyless cone mounted propellers, the following shall be verified:

— Prior to final pull-up, the contact area [1)] between the mating surfaces shall be not less than 70% of the theoretical contact area (100%). Non-contact bands (except oil grooves) extending circumferentially around the hub or over the full length of the hub are not acceptable. At the big end there shall be a full contact band of at least 20% of the taper length.
— After final pull-up, the propeller shall be secured by a nut on the propeller shaft. The nut shall be secured to the shaft with the approved locking arrangement. Alternatively, if approved, a split fitted ring with locking arrangement may be used. The ring shall have a tight fit.

1) The contact may be improved by light grinding with a soft disc and emery paper in the hub (not the shaft). A test pull up may also be used to improve the contact.

H 300 Shaft alignment

301 The shaft alignment shall be within the tolerances given in the shaft alignment specification.

302 When shaft alignment calculations are required (see A403 and F400) the measured values such as gap and sag, straightness/slope of stern tube bearings, jacking loads with force-displacement diagrams, shall be reported.

I. Shipboard Testing

I 100 Bearings

101 During the sea trial, the temperatures in all fluid film bearings (that are equipped with thermometers) shall be checked.

I 200 Measurements of vibration

201 Measurements of vibration on power take off generators driven from the engine driven reduction gear shall be carried out at 90%, 100% and (at least) 105% of rated (generator) speed with unloaded generator and ship service speed under steady state operation. The measurements shall be made near both bearings in the vertical, horizontal and axial directions. Frequency analyses shall be made in the range of 2 to 100 Hz.

Unless otherwise specified by the generator designer and approved by DNV, the vibration velocities shall not exceed the following:

For long-term continuous operation, i.e. 90% and 100% generator speed:

— 4.5 mm/s r.m.s. per frequency component for vibration caused by internal sources.
— 7.1 mm/s r.m.s. per frequency component for vibration caused by external sources.

For operation in a limited time period, i.e. 105% generator speed:

— 7.1 mm/s r.m.s. per frequency component for vibration caused by internal or external sources.

For definitions, see ISO 10816-3.

Vibration caused by internal sources are defined as those caused by the generator rotor and the shaft couplings between the generator and gearbox. This means the 1st and 2nd order of the generator speed as well as any coupling resonance to torsional and axial vibration.

SECTION 2 GEAR TRANSMISSIONS

A. General

A 100 Application

101 The rules in this section apply to gear transmissions subject to certification, see Ch.2 Sec.1 A200. The rules apply to the gear transmission, it's integrated components, such as coolers and pumps, and the lubrication piping system. Gears for jacking machinery for self elevating offshore units, partially deviate from the rules in this section. This is specified in DNV-OS-D101 Sec.5 D. See Sec.1 regarding shafting and rigid couplings and Sec.3 for clutches.

102 Ch.2 describes all general requirements for rotating machinery and forms the basis for all sections in Ch.3, Ch.4 and Ch.5.

103 The complete gear transmission shall be delivered with a NV certificate that is based on the design approval in B, the component certification in C and the workshop testing in D.

A 200 Documentation

201 *Plans* to be submitted for approval:

a) Arrangement including part list(s)[1]:

— longitudinal section of the unit
— transverse section (applicable to gears with more than 2 shafts).

b) Detail drawings[2]:

— pinion(s) and wheel(s) [3]
— shafts
— hub(s)
— clutch(es) and coupling(s)
— other power transmitting parts

c) Gear casing (unless the wall thickness and bearing supports, including main thrust bearing support, are indicated on the longitudinal section)

d) Gearbox fixation including chocking calculations, if applicable

e) Schematic lubrication oil system diagram including all instruments and control devices.

f) The control and monitoring system, including set-points and delays, see E and Ch.9.

1) The arrangement plans ought to contain as many details as practically possible in order to reduce the total number of plans, e.g. if all details of other items as listed under b) above can be given on the arrangement plans, then separate plans of these items need not be submitted.

2) The plans shall show clearly all details as fillets, keyways and other stress raisers, shrinkage amounts (also for bearings), pull up on taper, surface roughness, bolt pre-tightening, etc. as well as types of material and mechanical properties, cleanliness (if required, see B207) including NDT specification, see Ch.2 Sec.3 A202. "All details" means data that are necessary for evaluation according to the relevant criteria in B.

3) Pinions and wheels can normally be sufficiently described on the longitudinal section and with all particulars listed in 202 b) specified on a data sheet (Form No.: 71.10a). However, if manufacture of the pinion and wheel set is subcontracted, separate sets of plans have also to be submitted.

202 *Particulars* to be submitted for approval:

a) Data according to Table A1 for each gear stage. The various data are explained in the Classification Note 41.2. and a special sheet, *Data Sheet for Gear Calculations, Form No. 71.10a,* has been prepared for this.

Table A1 Gear data

Item	*Particulars*	*Symbol*	*Comments*
Loads [1]	Maximum power (kW) on pinion	P	Alternatively, a load-time spectrum may be used. This is typical for gears designed for relatively short life time (less than for example a million cycles). See also Ch.2 Sec.3 A101.
	R.p.m. of pinion	n_0	
	Rated pinion torque corresponding to maximum power and r.p.m.	T_0	
	Application factors	K_A	Both for normal operation and permissible diesel engine misfiring condition
	Application factor for non-frequent peak loads	K_{AP}	For example start-up of electric motor with star-delta shift or clutching-in shock
	Application factor for ice condition	K_{Aice}	For vessels with ice class (see Pt.5 Ch.1 of the Rules for Classification of Ships)
Faceload distribution	Maximum permissible faceload distribution factor at rated load [2] [3]	$K_{H\beta}$	For bevel gears with ordinary length crowning it is sufficient to specify the minimum permissible face width contact in %.
Dimensions [4]	Number of teeth	z	
	Centre distance	a	For gears with parallel axis only
	Common face width at operating pitch diameter	b	
	Face widths at tooth roots	$b_{1,2}$	
	Total face width including gap	B	For double helical gears only
	Tip diameters	d_a	
	Addenda	h_a	
	Minimum and maximum backlash	j	
	Angle between shafts	Σ	For bevel gears only

Table A1 Gear data (Continued)

Item	*Particulars*	*Symbol*	*Comments*
Tool and gear geometry 4)	Normal module	m_n	In mid section for bevel gears (m_{nm})
	Module of tool	m_0	For bevel gears only
	Transversal module at outer end	m_t	For bevel gears only
	Pressure angle in normal section at reference cylinder	α_n	
	Helix angle at reference cylinder	β	
	Helix angle in the midsection	β_m	For bevel gears only
	Addendum of tool	h_{a0}	Referred to m_n
	Radius at tip of tool	ρ_{a0}	Referred to m_n
	Protuberance	pro	Referred to m_n and excluding grinding amount
	Addendum modification coefficient	x	Referred to m_nIn mid section for bevel gears (x_{hm})
	Number of teeth of cutter	z_c	If pinion type cutter is used
	Addendum modification coefficient of cutter	x_c	If pinion type cutter is used Referred to m_n
	Angle modification	θ_k	For Zyclo Palloid bevel gears only
	Cutter radius	r_{e0}	For Zyclo Palloid and Gleason bevel gears only
	Tooth thickness modification coefficient (mid face)	x_{sm}	For bevel gears only Referred to m_n
Material	Material specification including heat treatment method		See Ch.2 Sec.3, (e.g. EN 10084 18CrNiMo7-6, Case hardened)
	Flank surface hardness, maximum and minimum		
	Mid face tooth root space hardness, minimum 5)		
	Tooth core hardness, minimum 5)		
	Core impact energy (KV) of coupon test at 20°C 5)		If applicable, see C206 b)
	Hardness depths after finishing process, applicable to surface hardened gears	t_{550}, t_{400} and t_{300}	Given as depth to 550HV, 400HV and 300HV as applicable, see Classification Note 41.2
Finishing process	Finishing method of flanks		
	Acceptance level for root grinding notches		Minimum radius and maximum depth
	Shot peening parameters		If applicable
	Surface roughness of flanks	R_z	Mean peak-to-valley roughness
	Surface roughness of tooth root fillet	R_y	Maximum height of the profile
	Tip and root relief	C_a/C_f	Amount and extension Heightwise crowning of tool for bevel gears
	Lead modifications		Amount and extension (end relief, crowning and/or helix correction)
	Grade of accuracy according to ISO 1328, DIN 3962 or ANSI/AGMA 2015-A01	Q	
Lubrication	Type of cooling		Spray, dip, fully submerged, with additional cooling spray, etc.
	Kinematic viscosity (mm^2/s)	ν	At 40°C and 100°C
	FZG damage level (scuffing)		According to ISO 14635-1
	Oil inlet temperature		At normal operation

1) For gears that are subjected to negative torques both the negative torque level as well as the frequency of these occurrences shall be specified unless the following guidance is used:

The negative torque level shall be given in percent of the rated (forward) torque.

The frequency of occurrence may be classed as:

<100 = rare

>100 = frequent

where the numbers refer to torque reversals.

2) For specified faceload distribution factors that are considered as "optimistic" (see Classification Note 41.2) a contact pattern specification at 1 or 2 suitable part loads shall be submitted together with an explanation on how this leads to the specified faceload distribution at rated load, see 204 b).

3) Note that the rated load means normally the maximum rating with the application factor that is decisive for the scantlings. However, if this application factor differs much from the application factor at normal operation, it may be necessary to specify both faceload distribution factors.

4) The data shall be given for both pinion (index 1) and wheel (index 2), and for an idler or planet gear, where applicable.

5) Applicable to case hardened gears only.

203 *Documentation* to be submitted for information only:

a) For power transmitting components of welded construction full details of the joints, welding procedure, filler metal particulars and heat treatment after welding shall be specified.

b) The bearings shall be documented with:

— calculated life time of rolling bearings (L_{10a} according to ISO 281)
— type of material, nominal surface pressure and clearance tolerances for fluid film bearings.

c) Manufacturer's specified overhaul interval, see B702.

d) For propulsion gears, acceptance criteria for shaft alignment where shaft alignment calculations are required according to Sec.1 A402.

e) For welded gears of thin rim design calculations of cyclic stresses in the weld shall be submitted, see B302.

204 *Documentation* to be submitted upon request:

a) If a manufacturer requests approval based on other methods than those described in the rules and Classification Notes (e.g. special calculation methods or tests), additional documentation will be requested. For principles, see Ch.2 Sec.3 A100.

b) For gear stages where the approval is dependent upon obtaining a certain optimistic faceload distribution, tooth contact pattern specifications at some selected part loads will be requested (for approval) together with an explanation on how this leads to the specified faceload distribution at rated load.

c) Balancing specifications for high speed gears (e.g. turbine driven) and for certain medium speed gears with non-machined surfaces of rotating parts (for information only).

d) Calculation of thermal rating for gas turbine driven gears (for approval).

B. Design

B 100 General

101 For design principles see Ch.2 Sec.3 A100. All components in gear transmissions shall be designed for all relevant load conditions such as rated power or overloads, including all driving conditions under which the plant may be operated. Regarding dynamic loads, see Ch.3 Sec.1 G.

102 The gearing may be approved on the basis of calculations (see 201) or tests (see 204) or combination of both.

Guidance note:

The acceptance criteria for calculation assessment are given in 201. They refer to calculation methods as given in Classification Note 41.2, comprising information on calculation of tooth root strength (root fractures), flank surface durability (pitting, spalling, case crushing and tooth fractures starting from the flank), and scuffing.

Alternative methods to the ones given in Classification Note 41.2 may also be considered on the basis of equivalence.

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103 Requirements for shafting and rigid couplings are given in Sec.1.

104 When considered necessary for completion of a type approval process, type testing may be required. Details on this type testing will be especially considered. For special gear designs a type approval may be pending satisfactory service experience, as e.g. after 1 000 to 3 000 hours.

105 For gear transmissions used for vessels with ice class notations the criteria throughout this section apply with the use of the application factor K_{Aice} (see Pt.5 Ch.1 of the Rules for Classification of Ships) replacing K_A and K_{AP} provided that K_{Aice} is greater.

106 Gear transmissions, and in particular power take off (PTO) branches, may be accepted for higher vibratory torques in the low load range than given in Ch.3 Sec.1 G303 e). This may be accepted when gearing, shafts and in particular bearings are designed for such vibratory conditions.

107 For design requirements for components delivered as integral parts of the lubrication, hydraulic operation and cooling systems of the gearbox the following applies as found relevant:

— Electric motors, see Ch.8
— Short lengths of flexible hoses may be used when necessary to admit relative movements between components. The hoses with couplings shall be type approved.

B 200 Gearing

201 The gear teeth shall be designed with the minimum safety factors as given in Table B1, see also Guidance Note to 102.

Table B1 Minimum safety factors

	Auxiliary	*Propulsion* 4)
Tooth root fracture S_F	$1.4^{3)}$	1.55
Pitting and subsurface fatigue S_H $^{1)}$	1.15	1.20
Scuffing S_S $^{1)\,2)}$	1.4	1.5

1) These safety factors apply to medium and high speed gears designed for long lifetimes (e.g. >10^6 load cycles). For slow speed gears with short design life time (e.g. << 10^6 load cycles) and where a certain flank deterioration is acceptable, lower values may be considered.

2) For medium and high speed gears as mentioned above, a minimum difference of 50°C between scuffing temperature and contact temperature applies in addition to the safety factor. However, if an oil inlet temperature alarm is installed, the minimum difference of 30°C between scuffing temperature and actual alarm level applies.

3) If an auxiliary gear stage is arranged as a power take off from a propulsion gearbox, and a tooth fracture of the auxiliary gear stage may cause a consequential damage to the propulsion system, the tooth root safety factor shall be as for propulsion.

4) Safety factors for auxiliary gears may be applied for vessels with class notations **Barge**, **LC Naval**, **Patrol**, **Yacht** and **Crew**.

Due to the scatter of the FZG test results, the FZG level used in the calculations shall be one level lower than the specified. Any gear utilising oils with specified FZG level above 12, the test results for the actual oil shall be documented in a test report from a recognised laboratory, and/or oil supplier.

202 Gear designs may be limited by other criteria as those mentioned in 201. For example if service experience indicates that failure modes other than those in 201 become a problem (such as oblique fractures starting from the active flank, grey staining developing to pitting etc.), a gear design may be rejected even though the criteria in 201 are fulfilled.

203 Gear designs shall take into account all relevant load conditions such as dynamics described in Ch.3 Sec.1 (diesel engines) and in other relevant sections. If vibration or shock loads result in reversed torques, this influence shall be considered.

204 Acceptance of gears may be based on approved tests.

205 For gearing designed to Baltic ice classes and class notations **ICE-05** to **-15** the calculation with K_{Aice} shall assume a number of load cycles equal to 10^6. This is additional to the normal, open sea conditions. The stricter of these criteria are decisive.

206 Quenched and tempered steels and all surface hardened steels shall have a level of cleanliness that is suitable for the permissible stress level for high cycle fatigue. A suitable level in this context is e.g. chart diagram index 2 for all groups A-DS in ISO 4967.

207 Gears classed to "high grade" made of special high grade materials, special high cleanliness will be required, see Ch.2 Sec.3.

208 Pinions and wheels may be made from separate forgings, rolled bars or blanks cut out of a forged bar. Gears made from rolled bars will have tooth root stresses crosswise to the fibre direction of the material. Therefore, normally a 10% reduction of the fatigue strength compared to gears made from separate forgings will be assumed. Correspondingly, gears made from blanks cut out of a forged bar are assumed to have a 20% reduction of the fatigue strength. However, such bars and blanks may be considered equivalent to separate forgings provided that either:

- they are further forged, or
- the steel making process and forging process are specially qualified. see Ch.2 Sec.3 A205.

B 300 Welded gear designs

301 If a pinion or wheel designed for high cycle ($>10^8$) is manufactured by welding, the permissible cyclic stress range (principal stresses) in the welds and HAZ is limited to 2/3 of the threshold value for crack propagation. This depends on the quality (i.e. NDT specification) of the weld with regard to external and internal defects.

As a simplification the following may be used:

- for full penetration welds which are very smooth or machined on all surfaces and 100% tested for surface defects (no linear indication >1.5 mm) and according to ISO 5817 level B for internal defects, the permissible stress range is 50 MPa
- as above, but not very smooth or machined or ground surfaces, 30 MPa
- for welds with inaccessible backside, 15 MPa.

For gears designed for a lower number of stress cycles ($<< 10^8$) higher permissible stress range values may be accepted, alternatively less weld checking.

302 The calculation (usually by FEM) of the actual stress range shall take the full load cycle of the pinion or wheel into account as well as the stress concentration in the weld and HAZ due to fillet radii and or shape of the weld.

303 Welded pinions or wheels shall be stress relieved. If the stress relieving is not the final heat treatment process (as e.g. when followed by a case hardening), the permissible values in 301 shall be reduced by 30%.

B 400 Shrink fitted pinions and wheels

401 Shrunk on pinions or wheels shall be designed to prevent detrimental fretting, macro slippage and micromovements.

402 The criteria for macro slippage and fretting are given in Sec.1 B400. The influence of axial forces and tilting moments shall be considered.

403 Shrink fitted rims shall have a minimum safety of 2.0 against micromovements based on the specified repetitive peak torque. This means that the local shear stress τ between a toothed rim and the hub shall be less than half the local friction $(p + \sigma)\ \mu$:

$$(p + \sigma)\ \mu / \tau > 2 = F_{lim}/F$$

p = nominal shrink fit pressure
σ = local radial stress due to the gear mesh force
μ = coefficient of friction

The following method may be used:

The nominal tangential force per unit face width is:

$$F_t = 2\,000\ T / (b\ d_1)$$

T = the pinion torque (Nm)
b = the face width of the shrink fit surface (mm)
d_1 = the reference diameter of the pinion (mm)

The force per unit face width to be used in the calculation is:

$$F = F_t\ K_A$$

if the movement in the axial tooth force direction is prevented by e.g. a shoulder, or if there is a double helical gear rim made of one body.

$$F = F_t\ K_A / \cos\beta \text{ in all other cases.}$$

The shrinkage pressure p depends on the shrinkage amount, the equivalent rim thickness s_v and the hub flexibility.

$$s_v = s + m_n\ (0.85 - 1.1\ m_n/s)$$

s = the rim thickness from tooth root to shrinkage diameter d_f (mm).

(only valid for $s > 2\ m_n$)

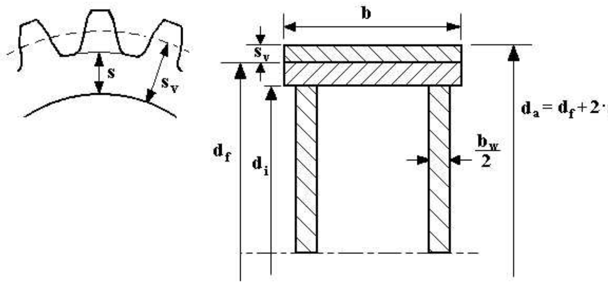

Fig. 1
Shrink fitted rim

The load limit per unit face width F_{lim} when micromovement is expected to start is:

$$F_{lim} = F_{ref}\ F_{corr}\ F_{roll}$$

F_{ref} = the reference load limit calculated as;

$$F_{ref} = 5.65\ p\ \mu\ s\ (0.7 + 2\ \mu)$$

F_{corr} = a correction factor which considers the influence of the hub flexibility (i.e. design and modulus of elasticity E_{hub}). It is unity for a solid steel hub. Otherwise calculated as:

$$F_{corr} = 1.586 - 2.86 \cdot 10^{-6}\ E_{hub} + f(b/b_w),$$

where $f(b/b_w)$ considers the flexibility of a hub with webs. b_w is the total face width of the webs.

$f(b/b_w) = 0.404 \cdot 10^{-3}\ (b/b_w)^3 - 0.01\ (b/b_w)^2 + 0.09\ b/b_w - 0.081$

(equal zero if $b = b_w$)

F_{roll} takes into account the rolling (tangential twist) load of a narrow rim (face width b_{helix}) due to an axial force component. The rolling moment causes a reduced surface pressure at an end of the face width. This is of particular importance for double helical gears with two separate rims. F_{roll} applies even if there is an axial shoulder.

F_{roll} is the minimum value of unity

or $(b_{helix}/d_f + 0.02)\ 4.8/\tan\beta$
or $(b_{helix}/(s + 1.3\ m_n) + 0.4)\ 0.2887/\tan\beta$

The coefficient of friction μ may be taken from Sec.1 Table B6.

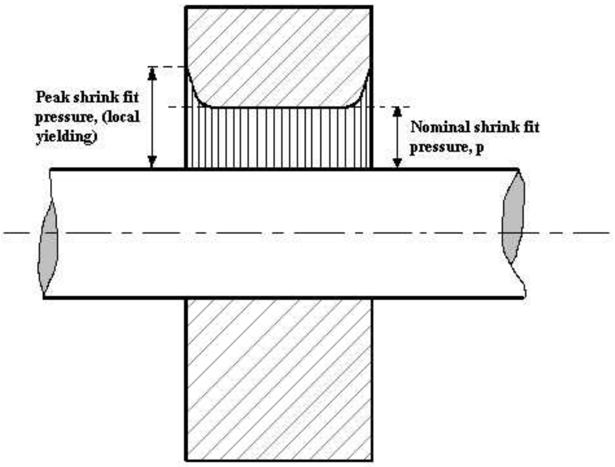

Fig. 2
Shrink fitted body with especially high surface pressure
(only applicable for shafts protruding on each end)

The safety against micromovements is:

$S = (p + \sigma)\ \mu\ /\ \tau = F_{lim}/F.$

B 500 Bolted wheel bodies

501 Bolted wheel bodies (and pinions, if applicable) shall be designed to avoid fatigue failure of the bolts due to pulsating shear stresses when passing the gear mesh zone.

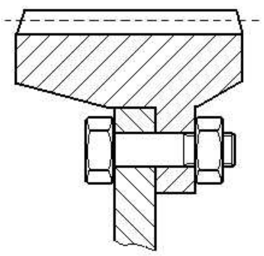

Fig. 3
Bolted wheel body

Guidance note:

The pulsating bolt forces will be reduced if the wheel body is radially supported without radial clearance.

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502 For gear rims that are flexible compared to the hub, the stresses in the bolts shall be calculated upon request (usually by means of FEM) for a mesh force corresponding to $T_0\ K_A$. The shear stress range shall not exceed 0.25 σ_y.

503 Bolts used for flexible rims shall have a tight fit in the holes, i.e. any combination of the tolerances shall not result in a clearance, or the bolts shall be ream fitted with a slight press fit.

B 600 Shafts

601 Shafts shall be designed to prevent fatigue. Detailed criteria are given in Classification Note 41.4. However, simplified criteria for various common gear shaft designs are given in Sec.1 B206.

When gear transmissions are designed for long life time (i.e. >>10^6 cycles), the shafts shall be designed to prevent detrimental fretting that may cause fatigue failures, see also Sec.1 B402. Unless torsional vibration values are defined, the upper permissible values for dynamics as given in Ch.3 Sec.1 G shall be used.

602 Shafts may be divided into 2 groups. These are shafts with:

— significant bending stresses, e.g. pinion and wheel shafts within their bearing spans
— no significant bending stresses, e.g. quill shafts and shafts outside the bearing spans of pinions and wheels.

The major load conditions to be considered are:

— high cycle fatigue (>>10^6 cycles) due to rotating bending and torsional vibration, see Sec.1 B206 B and Classification Note 41.4 item 4
— low cycle fatigue (10^3 to 10^4 cycles) due to load variations from zero to full load, clutching in or starting shock loads, reversing torques, etc., see Sec.1 B206 A and Classification Note 41.4 item 3.

Practically, shafts with significant bending stresses such as pinion and wheel shafts are dimensioned with regard to stiffness (gear mesh considerations) and high cycle fatigue, but hardly ever for low cycle fatigue because the two first will prevail.

603 For short shafts made by blanks cut from forged bars without further forging, see Ch.2 Sec.3 A205.

B 700 Bearings

701 Fluid film bearings shall be designed with bearing pressures that are suitable for the bearing metals. The calculation of bearing pressures shall include the application factor K_A.

702 Ball and roller bearings shall have a minimum L_{10a} (ISO 281) life time that is suitable with regard to the specified overhaul intervals. The influence of the lubrication oil film may be taken into account for $L_{10a,}$ provided that the necessary conditions, in particular cleanliness, are fulfilled.

Guidance note:

If no overhaul intervals are specified, a bearing life time of 40 000 hours may be used for conventional ships and 10 000 hours for yachts or ships and units that are not predominantly used at full load for longer periods.

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B 800 Casing

801 Inspection openings shall be provided in order to enable inspection of all pinions and wheels (measurements of backlash and application of lacquer for contact pattern verification) as well as for access to clutch emergency bolts (if applicable). For special designs (e.g. some epicyclic gears) where inspection openings cannot be provided without severely affecting the strength of the design, holes for boroscope inspections may be accepted as a substitute to openings. Such holes shall be positioned to enable boroscope inspection of all gearing elements.

802 Easy access to all inspection openings shall be provided. This means that no piping or coolers etc. shall be positioned to prevent access.

803 In order to prevent rust, the gear casing shall be provided with proper ventilation.

B 900 Lubrication system

901 The lubrication system shall be designed to provide all bearings, gear meshes and other parts requiring oil with adequate amount of oil for both lubrication and cooling purposes. This shall be obtained under all environmental conditions as stated in Ch.1 Sec.3 B200 of the Rules for Classification of Ships.

902 The lubrication system shall contain at least:

— an oil pump to provide circulation
— a filter of suitable fineness for gearing, hydraulics and bearings (see 702).

Guidance note:

Specification of a pressure filter for maintaining suitable fluid cleanliness may be 16/14/11 according to ISO 4406:1999 and $\beta_{6\text{-}7\,(c)}$ = 200 according to ISO 16889:1999.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

— if necessary, a cooler to keep the oil temperature within the specified maximum temperature when operating under the worst relevant environmental conditions, see 901.

903 For gear transmissions for propulsion where windmilling may be detrimental and considered as a normal working condition, there shall be either:

— a shaft brake designed to hold (statically) twice the highest expected windmilling torque, or
— one pump available in wind-milling condition. This pump shall be additional to any standby pump required by other rules.

The chosen version shall be automatically activated within 30 s after shut down.

904 Gear transmissions for propulsion are normally to have an attached pump. For plants designed to normally operate at such low speeds that the attached pump cannot supply sufficient oil pressure (e.g. plants with frequency controlled electrical motor), the following will be accepted:

— either an extra electric oil pump that is activated at a given pressure, or
— 2 electric main pumps of the same capacity, one of which is arranged as a standby pump with immediate action. These 2 electric pumps shall be supplied from different switchboards.

905 For propulsion gears the lubrication system shall be arranged so that the gear transmission can endure a run out of 5 minutes after a black out without jeopardising any bearings or gear teeth.

This may be provided by e.g.:

— an attached pump with an additional gravity tank (if necessary)
— electric pumps with a gravity tank with sufficient volume and height for 5 minutes supply.

906 Gear transmissions in single propulsion plants shall have a standby pump with immediate activation.

907 For gear transmissions in single propulsion plants the filtering system shall be arranged to make it possible to clean the filters without interrupting the oil supply.

908 Use of flexible hoses in the lubrication system is only permitted where necessary in order to allow for relative movements (e.g. when resiliently mounted). Flexible hoses with their couplings shall be type approved, see Ch.6 Sec.6 D (ship rules).

C. Inspection and Testing

C 100 Certification of parts

101 Regarding certification schemes, short terms, manufacturing survey arrangement (MSA) and important conditions, see Ch.2 Sec.2.

102 The parts in a gear transmission shall be tested and documented according to Table C1 and 200 through 400, if not otherwise agreed in a MSA, see Ch.2 Sec.2 C100. In 200 through 400, the testing and inspection as required in Table C1 are described in more detail.

Table C1 Requirements for documentation and testing

Part	*Product certificate*	*Chemical composition*	*Mechanical properties*	*X-Ray or Ultrasonic testing*	*Crack detection* 1)	*Visual and dimensional check* 2)	*Other* 2)
Pinion and wheels	NV 7)	See 200 - Table C2					
Built-in clutches, bending compliant- and elastic couplings	NV 7)	See Sec.3 C, Sec.4 C and Sec.5 C, respectively					
Shafts, rigid couplings and hubs		See Sec.1 C					
Welded gears		W		W 3)	W 3)	W 3)	W 6)
Casing (welded)		W	W	W 4)	W 5)		
Bolts and keys		TR	TR				
Ancillaries	See C400						

1) By means of magnetic particle inspection or dye penetrant.
2) For details and extent, see relevant paragraph.
3) After final heat treatment (e.g. case hardening).
4) For thickness >100 mm.
5) Of welds near bearings (at least 10% of the welds to be tested).
6) Stress relieving (time-temperature diagram), but not applicable to gears that shall be case hardened.
7) Only applicable if sub-contracted.

C 200 Pinions and wheels

201 The requirements in 200 apply to the toothed parts of pinions and wheels. A flow chart and a summary of these requirements are presented in Fig.4 and Table C2, respectively.

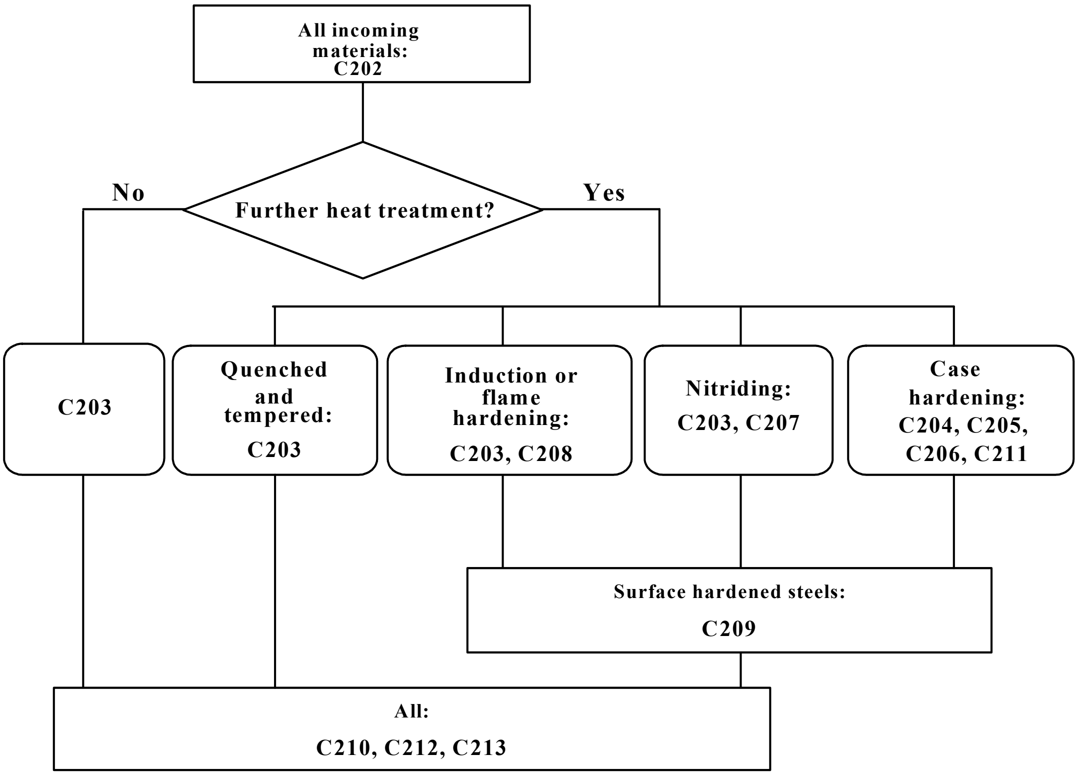

Fig. 4
Flow chart showing the applicable certification rules

202 The incoming material shall be documented with a Work certificate (W) to be within the approved specification as follows:

- chemical composition (ladle analyses)
- ultrasonic testing in suitably machined condition.

For gears classed to "high grade", the requirement for additional documentation as specified in the approval, applies:

- cleanliness according to ISO 4967
- oxygen content
- grain size according to ISO 643
- forging or rolling according to an agreed process.

See also Pt.2 Ch.2 Sec.5 A.

For materials that are delivered in their final core heat treatment condition, such as quenched and tempered steels for later nitriding, induction hardening or no further treatment, the additional requirement for documentation in 203 applies.

203 For quenched and tempered steels (QT) and normalised steels (N) the mechanical testing shall be documented by Work certificate (W) in accordance with Pt.2 Ch.2 Sec.5 E.

Documentation of mechanical properties may be replaced by surface hardness (HB) for QT steels with controlling sections (see ISO 6336-5 Annex A) below 200 mm and for normalised steels of all sizes.

204 For case hardening the gear manufacturer or the heat treatment subcontractor shall have a quality control system[1] acceptable to the Society. This quality control system shall provide that:

- suitable heat treatment is made prior to machining in order to avoid excessive distortions during quenching
- carburising is made in a controlled furnace atmosphere. The furnace shall be equipped with temperature and carbon potential controls and continuously recorded
- the entire case hardening process is checked by means of coupons[2] at regular and agreed intervals with regard to surface microstructure and core microstructure. For details see 205
- the gears are shot cleaned after the heat treatment.

1) If this requirement is not fulfilled or only partly fulfilled, the non fulfilled elements in this paragraph as well as in 205 and 206 shall be inspected in the presence of a surveyor.

2) The coupon shall be representative for the quenching rate of the typical gear sizes. The hardness and microstructure at the centre of the coupon will then be representative for the core of a typical gear. The coupon shall be of the same type of material as the typical gears. The approximate size is minimum diameter 6 modules and length 12 modules. This module shall be either the module of the actual gear to be certified or from the upper range of the production with that material. The coupon shall follow the entire heat treatment and shot cleaning processes and be quenched together with the pinions and wheels in such a way that its quenching rate is as representative as possible.

205 With reference to 204 the requirements for the surface (i.e. polished depth of less than 0.03 mm) microstructure are:

a) Reduction of surface hardness in the outer 0.1 mm of the case shall not be more than 2 HRC.
b) Carbide precipitation at surface and at 0.2 mm depth checked at approximately 400 times magnification. Only

fine dispersed carbides are permitted, see ISO 6336-5.

c) Retained austenite at surface and at 0.2 mm depth not to exceed 25%. To be checked by comparison with reference pictures or by a calibrated magnetoelastic method.

d) Depth of intergranular oxidation (IGO) from unpolished surface shall not exceed 10 + 6 t_{550} (μm). t_{550} should be given in mm.

Requirement for the core in the middle of the coupon:

— to be martensitic or bainitic with no blocky ferrite
— hardness according to approved specification.

If these requirements are not fulfilled, the permissible values for tooth root stresses and contact stresses and contact temperatures (for scuffing) will be reduced according to special consideration.

206 For case hardening the following applies in addition to the requirements in 204 and 205.

Each hardening batch and each material type shall be documented regarding:

a) Hardness profile[1)] (W) with emphasis on depth to 550 HV and 400 HV. For core hardness below 300 HV the depth to 300 HV is also to be checked.

1) The case depths shall be checked on a coupon that follows the entire heat treatment process. The coupons shall be of the same type of material as the actual gears to be certified and may be of a standard size. The correlation between these small coupons and the representative coupons mentioned in 204 shall be documented by means of comparison measurements and included in a MSA.

If small coupons are used, e.g. standard size of ø30-35 mm, and no approved correlation to the actual gear size exists, the following correlation shall be used (applicable for the hardness profile of the flanks with material ISO 683-11 - 18CrNiMo7 and EN 10084 - 18CrNiMo7-6):

For gears with $m_n > 5$ (mm):

i) $t_{550} = (1 - (m_n - 5)/85) \cdot$ measured depth to 550 HV (mm)

ii) $t_{400} = t_{550} \cdot (1.6 - (m_n - 5)/100)$ (mm)

iii) Core hardness = 0.8 · measured core hardness of coupon (HV)

iv) If corrected core hardness < 300 HV then $t_{300} = 1.35 \cdot t_{400}$ (mm)

The grinding amount shall be subtracted from the depths in i), ii) and iv).

For gears with $m_n \leq 5$ (mm):

The grinding amount shall be subtracted from the measured hardness depths.

b) Core impact energy[2)] (KV) (W).

2) The core impact energy has the objective of detecting unacceptable grain growth and shall be verified by means of at least 2 test pieces taken from the centre of a coupon that has followed the entire heat treatment process. The coupon shall have a diameter of at least 2 modules. The coupon may be taken from any of the positions in Fig.5 to Fig.8 in Pt.2 Ch.2 Sec.5 E, as well as from material near the surface.

The impact energy shall be at least 30 J, unless otherwise approved. If the coupon is taken longitudinally in a body with longitudinal grain flow, the minimum value is 40 J.

The core impact energy testing may be waived if all of the following conditions are fulfilled:

— carburising temperature below 940°C
— maximum specified case depth to 550 HV below 3.0 mm
— chemical composition contains grain growth preventing elements (e.g. Al)
— implemented manufacturing survey arrangement for heat treatment.

c) Tooth root hardness[3)] (NV) at mid face of each pinion and wheel.

3) For modules 10 mm and above the surface hardness in the tooth root space in the middle of the face width shall be checked. A small spot shall be polished (No grinding. Polishing depth less than 0.03 mm) and the hardness measured by means of a low force tester. Unless otherwise approved, the minimum hardness shall be 58 HRC. The manufacturer may carry out approved procedure tests in order to establish limit sizes for various material types. Below these limit sizes the hardness will with a high probability turn out to 58 HRC or more, and no such hardness testing is required for the individual gears. The procedure tests shall include various designs and quenching baths.

207 Nitrided gears shall be documented with Work certificates (W) for each heat treatment batch by means of a coupon following the entire nitriding process with regard to:

— case depth (to 400 HV)
— white layer thickness (to be < 0.025 mm).

The coupon shall be of the same material type as the gears. The core properties shall be documented as described in 203.

208 Induction or flame hardened gears shall be documented with a NV certificate with regard to:

— hardness contour
— hardness depth at pitch diameter
— hardness depth at tooth root
— surface structure, random inspection (to be mainly fine acicular martensite).

The hardness pattern shall be checked at a representative test piece with the same geometry (profile and root shape) and type of material as the actual gears (except for face width which may be smaller). For batch production this testing shall be made at least before and after each batch.

The hardness pattern checking applies to both ends and the mid section of the test piece. All three sections shall have values within the approved minimum to maximum range. Each gear to be visually inspected at both ends and the contour shall be consistent with the test piece.

For small gears with spin type hardening (see ISO 6336-5) only the surface structure (random) and external contour need to be checked. The core properties shall be documented as described in 203.

209 For all surface hardened teeth the final flank hardness shall be measured and documented with a Work certificate (W). The hardness shall be measured directly on the flanks near both ends and in the middle and at each 90 degrees. Low force testers are preferred provided suitable surface finish. For batch production a less frequent checking may be approved.

210 All teeth shall be crack detected, no cracks are accepted, see Ch.2 Sec.3 A202. This shall be documented with a Work certificate (W). Normally gears shall be checked by means of the wet fluorescent magnetic particle method. However, nitrided or not surface hardened gears may be checked by the liquid penetrant method. For batch production a reduced extent of crack detection may be approved. The crack detection shall be made prior to any shot peening process.

211 For case hardened gears grind temper inspection shall be carried out randomly and be documented by a test report (TR).

This inspection may be done by:

— nital etching per ISO 14104 or ANSI and or AGMA 2007-B92 (grade B temper permitted on 10% of functional area (FB1))

or

— a calibrated magnetoelastic method (acceptance criteria subject to special consideration)

212 The tooth accuracy of pinions and wheels according to ISO 1328 shall be documented with a Work certificate (W) as follows:

— for specified grade 4 or better, all pinions and wheels shall be measured
— for specified grade 5 [1)] at least 50% shall be measured
— for specified grade 6 [1)] at least 20% shall be measured
— for specified grade 7 or coarser at least 5% shall be measured.

1) When a wheel cannot be measured due to its size or weight, at least every mating pinion shall be measured.

If other standards (e.g. DIN 3962 or ANSI/AGMA 2015-A01) are specified, measurement program equivalent to the above applies (with respect to pitch, profile and lead errors).

Bevel gears (that are not covered by ISO 1328) shall be measured regarding pitch and profile errors if required in connection with the approval. Generally, all bevel gear sets shall be checked for accuracy in a meshing test without load. The unloaded contact pattern shall be consistent with the specified, and documentation thereof shall follow the gear set to the assembly shop.

213 The surveyor shall review all required documentation and carry out visual inspection of the pinions and wheels with special attention to:

— surface roughness of the flanks
— tooth root fillet radius
— surface roughness of tooth root fillet area
— possible grinding notches in the root fillet area. Any grinding (or any other machining) of the root area is not accepted unless this has been especially approved.

Table C2 Summary of certification requirements

Test	*Case hardening*	*Nitriding*	*Induction and flame hardening*	*Through hardening (QT)*	*No hardening (N)*
Chem. composition [1)]	W	W	W	W	W
Ultrasonic [2)]	W	W	W	W	W
Mechanical properties [3)]		W	W	W	W
Case hardness profile [4)]	W	W	NV		
Core impact energy (KV)	W				
Tooth root hardness [5)]	NV				
White layer thickness		W			
Final flank hardness	W	W	W		
Crack detection of all teeth [6)]	W	W	W	W	W
Random grind temper inspection	TR				
Tooth accuracy [7)]	W	W	W	W	W
Visual inspection [8)]	NV	NV	NV	NV	NV

1) Applicable to the incoming material
2) In suitably machined condition, i.e. after forging prior to surface hardening
3) For hardened gears with “controlling section” size ≤ 200 mm (see ISO 6336-5 Annex A) and all not hardened gears (N), surface hardness (HB) is sufficient
4) For relevant test parameters, see 206, 207 or 208
5) For modules ≥ 10 mm, at mid face
6) Prior to any shot peening process
7) See 212 for extent of documentation
8) See 213 for extent of inspection

C 300 Welded gear designs

301 Welded gears shall be documented with Work certificates (W) as follows:

— chemical composition and mechanical properties of all the materials
— stress relieving (time-temperature diagram)
— 100% weld quality control according to ISO 5817. To meet level B for internal defects unless otherwise approved.
— 100% surface crack detection by MPI or dye penetrant. No linear indication >1.5 mm unless approved.
— visual inspection by a surveyor with special emphasis on the shape of the outer weld contour (stress concentrations) at the root.

The 3 last items refer to the gear after the final heat treatment (e.g. after case hardening).

C 400 Ancillaries

401 Pumps, electric motors, coolers, piping, filters, valves, etc. that are delivered as integral parts of the lubrication, hydraulic operation and cooling systems on the gearbox, shall be checked by the gear manufacturer’s quality system as found relevant.

C 500 Assembling

501 Balancing of rotating parts and subassemblies of rotors shall be documented with Work certificates (W) and shall be within the approved specification.

502 Cylindrical shrink fitting of pinions, wheels, hubs, clutches, etc. shall be documented with Work certificates (W) with regard to shrinkage amount. The diameters (and therewith the shrinkage amounts) shall be checked at various positions along the length of the shrinkage surface. If conicity or ovality in a connection with length to diameter ratio >1 result in:

— a shrinkage amount near the minimum tolerance value at the torque transmission end

and

— an amount near the maximum tolerance value at the opposite end,

the shrinkage specification shall be reconsidered with respect to possible fretting near the torque transmission end. (If the non-torque end is subjected to bending stresses, possible fretting must be considered here too.)

503 Tapered shrink fit connections shall be documented with Work certificates (W) with regard to contact area and pull up distance or push up force or diametrical expansion (whichever is the approved specification).

The contact between the male and female parts shall be checked with a thin layer of contact marking compound (e.g. toolmaker's blue). There shall be full contact at the end with torque transmission (which is normally the upper end). If this is not obtained, light correction grinding with a soft disc and emery paper may be done in the female part only (if wet mounting). Alternatively a test pull up may deform small irregularities and result in an improved contact.

504 Keyed connections shall be checked with regard to:

— key fit in shaft and hub (for connections where the torque may be reversed the key shall have a tight fit in both shaft and hub)
— shrinkage amount, see 502
— push up force, see 503.

505 Spline connections shall be checked with regard to:

— tight fit if of the "fixed" type
— lubrication if of "working" type.

506 Bolted connections such as bolted wheel bodies or flange connections shall be checked with regard to:

— tightness of fitted bolts or pins
— pre-stress as specified.

507 Access through inspection openings to gearing and clutch emergency bolts (if applicable) shall be verified, see also B801, B802 and Sec.3 B302.

D. Workshop Testing

D 100 Gear mesh checking

101 The accuracy of the meshing shall be verified for all meshes by means of a thin layer of contact compound (e.g. toolmaker's blue). This shall be done in the workshop in the presence of a surveyor.

When turning through the mesh, the journals shall be in their expected working positions in the bearings. This is particularly important for journals which will assume a position in the upper part of the bearings (and the bearing clearances are different), and when external weights (such as clutches) may cause a pinion to tilt in its bearings.

For small and medium gears with ground or skived flanks on both pinion and wheel it is sufficient to check this at one position of the circumference.

For large gears (wheel diameter >2 m) and for all gears where an inspection after part or full load (in the workshop or onboard) cannot be made, the contact checking shall be made in several (3 or more) positions around the circumference of the wheel.

For bevel gears the contact marking shall be consistent with the documented contact marking from the production, see C212.

For highly loaded gears it may be required to carry out such a mesh contact test under full or high part load by slow turning through a full tooth mesh at 3 or more circumferential positions.

The result of the contact marking shall be consistent with that which would result in the required faceload distribution at rated load.

For propulsion gears connected to shafts in excess of 200 mm diameter, and all multi-pinion gears, the contact marking of the final stage shall be documented by tape on paper or photography and shall be forwarded to the client as a reference for further checking onboard.

The backlash shall be documented for all gear meshes.

102 All gear transmissions shall be spin tested in presence of a surveyor.

Prior to the spin test some teeth at different positions around the circumference of all gear meshes shall be painted with an oil resistant but low wear resistant test lacquer. For multi-mesh gears the lacquer shall be applied to the flanks that mesh with only one other member.

After the spin test the initial contact patterns shall be documented by sketches. The position of the initial contact shall be consistent with that which would result in an acceptable load distribution at rated torque.

103 For gears that are workshop tested with a part load sufficient to verify the load distribution at rated torque, the testing in 101 and 102 may be waived, except for backlash measurements.

Such part load testing will only be representative for the full load condition on board if the in- and out-put shafts are connected to systems that will not impose significant bending moments or forces. Furthermore, the part load shall be so high (normally 40% torque or more) that reliable extrapolation to rated torque can be made. Therefore, this part load testing is subject to approval, see A200. If such part load testing is successfully carried out, the gear transmission certificate may have a remark stating that contact pattern testing onboard may be waived.

104 During the running test the gearbox shall be inspected for leakage.

D 200 Clutch operation

201 For clutches delivered integral with the gear box the clutching-in function shall be tested in the presence of a surveyor. For oil operated clutches the testing shall be made with the oil at normal service temperature.

The pressure - time function shall be within the approved specification and the end pressure at the specified level. No pressure peaks beyond the nominal pressure are allowed. The clutch operation pressure shall be measured as closely as possible to the clutch inlet.

D 300 Ancillary systems

301 The manufacturer shall demonstrate to the surveyor that the lubrication oil intake is placed to be well submerged under all environmental operation conditions for the actual type of vessel. Furthermore, it shall be checked that the oil sprays for lubrication and cooling function properly. After the running test the filters shall be inspected.

302 All equipment delivered with the gearbox regarding indication, alarm and safety systems shall be function tested.

E. Control and Monitoring

E 100 Summary

101 The requirements in E are additional to those given in Ch.9.

102 The gear transmissions shall be fitted with instrumentation and alarms according to Table E1.

103 If individual local pressure indicators are not fitted, quick connectors for a portable instrument shall be provided in order to do local readings and set point verification of switches. The corresponding portable instrument shall be provided on board.

104 Alarms (Gr 1) and start of standby pump shall be without delay, other than those necessary to filter normal parameter fluctuations, if not otherwise approved.

Table E1 Monitoring of gear transmissions

	Gr 1 Indication Alarm Load reduction	*Gr 2 Automatic start of standby pump with alarm 1)*	*Gr 3 shut down with alarm*	*Comments*
1.0 Gear bearing and lubricating oil				
Oil lubricated fluid film bearings (axial and radial), temperature	IR, HA			Applicable to gears with totally transmitted power of 5 MW or more.
Thrust bearing, temperature	IR, HA			Applicable to gears with totally transmitted power of 5 MW or more. Sensors to be placed in the bearing metal or for pads in the oil outlet.
Lubricating oil, pressure	IL, IR, LA	AS		At bearings and spray, if applicable. If equal pressure, one common sensor is sufficient for Gr 1.
Lubricating oil, temperature	IL, or IR, HA			At inlet to bearings, i.e. after cooler.
Lubricating oil temperature	IL or IR			In sump, or before cooler
Sump level 2)	ILor IR			For splash lubricated gears.
2.0 Integrated clutch activating media				
Hydraulic oil	IL, IR, LA	AS	SH	SH means either declutching or engine stop

Gr 1 Common sensor for indication, alarm, load reduction (common sensor permitted but with different set points and alarm shall be activated before any load reduction)
Gr 2 Sensor for automatic start of standby pump
Gr 3 Sensor for shut down

IL = Local indication (presentation of values), in vicinity of the monitored component
IR = Remote indication (presentation of values), in engine control room or another centralized control station such as the local platform/ manoeuvring console
A = Alarm activated for logical value
LA = Alarm for low value
HA = Alarm for high value
AS = Automatic start of standby pump with corresponding alarm
LR = Load reduction, either manual or automatic, with corresponding alarm, either slow down (r.p.m. reduction) or alternative means of load reduction (e. g. pitch reduction), whichever is relevant.
SH = Shut down with corresponding alarm. May be manually (request for shut down) or automatically executed if not explicitly stated above.

For definitions of LR and SH, see Ch.1 Sec.1 B116 and 117 of the Rules for Classification of Ships.

1) To be provided when standby pump is required, see B900 and Ch.1 Sec.3 B300 (Rules for Classification of Ships).

2) For gears with totally transmitted power of 500 kW or less, dipstick inspection is considered adequate.

F. Arrangement

F 100 Installation and fastening

101 The gearbox shall be arranged so that appropriate alignment and running conditions are maintained during all operating conditions. For shaft alignment, see Sec.1 F400.

102 Gearboxes shall be mounted on chocks or epoxy resin. The bolts shall be designed for all relevant operating conditions (in particular when ice classes apply). Flexibly mounted gearboxes will be especially considered.

103 Gearboxes that are or may be subjected to external forces such as thrust, shall have end stoppers. End stoppers may be waived if fitted bolts or equivalent solutions are used.

104 Piping etc. shall not be arranged to obstruct access to inspection openings.

105 All pipe connections shall be screened or otherwise protected as far as practicable in order to avoid oil spray or oil leakage into machinery air intakes or onto potentially hot surfaces.

Any surfaces (with temperatures exceeding 220 degrees if not insulated) that are insulated by means, for which workmanship affects the efficiency of the insulation, are defined as potentially hot surfaces.

G. Vibration

G 100 General

101 Regarding torsional vibration, see the section for the relevant prime mover, e.g. diesel engines Ch.3 Sec.1 G and B105.

102 The vibration of the gearbox foundation (except when flexibly mounted) is normally not to contain gear alien frequency components with amplitudes exceeding 10 mm/s. Alien frequencies are those that are not rotational frequencies of any gear internal parts.

Higher amplitudes may be accepted if considered in the gear design.

H. Installation Inspection

H 100 Application

101 H applies to inspections in connection with installation of complete gearboxes. Regarding external couplings and shafts, and internal clutches, see respective sections.

Unless otherwise stated, a surveyor shall attend the inspections given in H and I.

H 200 Inspections

201 The following inspections shall be carried out:

— shaft alignment, see Sec.1 H300
— fastening of propulsion gearboxes (stoppers and bolt tightening)
— flushing, applicable if the system is opened during installation. Preferably with the foreseen gear oil. If flushing oil is used, residual flushing oil shall be avoided.
— lubrication oil shall be as specified (viscosity and FZG class) on maker's list
— pressure tests to nominal pressure (for leakage) where cooler, filters or piping is mounted onboard
— clutch operation, see Sec.3 H
— tooth contact pattern, see 202.

202 A tooth contact pattern inspection as described in D101 shall be made for gears where the installation on board can alter the initial tooth contact pattern. This means e.g. all gear transmissions with more than one pinion driving the output gear wheel, even if there is only one single input shaft as for dual split paths, and propulsion gears connected to shafts in excess of about 200 mm diameter. The result of the contact pattern check shall be consistent with the result from the workshop.

I. Shipboard Testing

I 100 Gear teeth inspections

101 To prevent initial damage on the tooth flanks (scuffing) and bearings, the gear shall be carefully run in according to the gear manufacturers specification.

102 All inboard gears shall be checked with regard to contact pattern under load.

Exceptions are accepted when:

— this is mentioned in the design approval (due to low stress levels)
— the design makes an inspection impossible without dis-assembling such as certain epicyclic gears (this does not exempt ordinary gears from having suitable inspection openings)
— the contact pattern under load is accepted in the workshop test, see D103.

103 The contact patterns (all gear stages) shall be checked by a suitable lacquer applied to some teeth (normally 2 each 120 degrees) prior to the checking under load. The lacquer shall be applied to flanks that have only one mesh (in order to avoid accumulated patterns). When part load contact pattern checking applies, the lacquer shall be of a kind that quickly shows the final pattern.

104 The gear shall be operated at the specified load level(s) without exceeding that particular level(s). After each specified level the contact patterns shall be checked in the presence of a surveyor. The results, in both height and length directions, shall be within the approved specification.

105 After the full load test, or after the sea trial, all teeth shall be checked for possible failures as scuffing, scratches, grey staining, pits, etc. Shrunk-on rims shall be checked for possible movements relative to the hub.

I 200 Gear noise detection

201 Gears shall be checked for noise in the full speed range (high frequencies as gear mesh frequencies) and in the lower speed range (gear hammer).

202 If the high frequent noise is higher than expected, measurements may be required.

203 Gear hammer shall be detected in the lower speed range and also during diesel engine misfiring tests (see Ch.3 Sec.1 I500). Speed ranges or operating conditions resulting in gear hammer shall be restricted for continuous operation.

I 300 Bearings and lubrication

301 Lubricating oil and bearing temperatures (as far as indication is provided) shall be checked during the full load test. All temperatures shall reach stable values (no slow gradual increase) without exceeding the approved maximum values.

302 After the sea trial all oil filters shall be checked for particles.

SECTION 3
CLUTCHES

A. General

A 100 Application

101 This section applies to clutches, both for use in shaftlines and in gearboxes that are subject to certification, see Ch.2 Sec.1 A200.

102 Clutches of standard design shall be type approved.

103 Ch.2 describes general requirements for rotating machinery. Attention shall be paid to Ch.2 Sec.3 A100 and in particular A102.

104 Clutches shall be delivered with a NV certificate. However, this does not apply to clutches used in gearboxes/thrusters and produced by the gearbox/thruster manufacturer, see Sec.2 Table C1.

A 200 Documentation

201 At least a sectional drawing of the clutch shall be submitted for approval.

The drawing shall show all details such as:

- connection to external shafts
- material specification including NDT specification
- mechanical properties
- heat treatment of splines etc.
- stress raisers
- activation system.

202 The following particulars shall be submitted for each clutch:

- static friction torque (with corresponding working pressure)
- dynamic friction torque (with corresponding working pressure)
- maximum working pressure
- minimum working pressure
- pressure for compressing return springs
- permissible heat development and flash power when clutching-in (upon request when case-by-case approval)
- the control and monitoring system, including set-points and delays, see E, shall be submitted for approval.

  In addition, see Ch.9.

203 For each application the clutching-in characteristics with tolerances (pressure as function of time) including max. engaging speed, shall be submitted for approval.

204 Upon request the documentation (simulation calculation) of the engaging process may be required, see Ch.3 Sec.1 A601 b)3) and Ch.3 Sec.1 G403.

B. Design

B 100 Torque capacities

101 The torque capacities of clutches for auxiliary purposes as well as propulsion shall be:

- static friction torque at least 1.8 T_0 and preferably not above 2.5 T_0[1)]
- dynamic friction torque at least 1.3 T_0

Both requirements referring to nominal operating pressure and no ice class notation.

1) When above 2.5 T_0 the documentation in A204 is obligatory.

102 The torque requirements in 101 may have to be increased for plants with ice class notation, see Pt.5 Ch.1 of the Rules for Classification of Ships. If the ice class application factor $K_{Aice} > 1.4$, the torque capacities shall be increased by the ratio:

$$K_{Aice}/ 1.4.$$

103 For clutches used in plants with high vibratory torques (beyond 0.4 T_0) or intermittent overloads, the torque capacity requirements will be especially considered.

B 200 Strength and wear resistance

201 The relevant parts such as flange connections, shrink fits, splines, key connections, etc. shall meet the requirements given in Sec.1 B300 to B700.

202 If a disc clutch is arranged so that radial movements occur under load, the possible wear of the teeth and splines shall be considered. This may be relevant for clutches in gearboxes where a radial reaction force may act on the discs. Such radial forces may occur due to bearing clearances in either an integrated pinion and clutch design or shafts that are moved off centre due to tooth forces.

203 Trolling clutches are subject to special consideration.

B 300 Emergency operation

301 Clutches for single propulsion plants shall be of a design that enables sufficient torque transmission to be arranged in the event of loss of hydraulic or pneumatic pressure. This means that for plants on board vessels without ice class or reinforcement due to high torsional vibration level at least half of the rated engine torque shall be transmitted.

302 If the requirement in 301 is fulfilled by means of bolts, easy access to all bolts should be provided. For built-in clutches, this means normally that all the bolts shall be on the part of the clutch that is connected to the engine. This in order to gain access to all bolts by using the engine turning gear. Such bolts shall be fitted in place and secured to the clutch. Alternative arrangements are subject to special consideration and in any case it should be possible to carry out the emergency operation within 1 hour. The emergency operation procedure should be given in the operating manual.

B 400 Type testing

401 Type testing in order to verify friction torques as specified in 101 may be required.

B 500 Hydraulic/pneumatic system

501 Clutches in single propulsion plants shall have a standby pump with immediate activation.

C. Inspection and Testing

C 100 Certification

101 Regarding certification schemes, short terms, manufacturing survey arrangement and important conditions, see Ch.2 Sec.2.

C 200 Inspection and testing of parts

201 Power transmitting parts as hubs, flanges and outer parts shall be documented with Work certificates (W) regarding chemical composition of the material and mechanical properties.

202 NDT shall be documented with Work certificate (W).

C 300 Ancillaries

301 Pumps, electric motors, coolers, piping, filters, valves, etc. that are delivered as integral parts of the hydraulic/pneumatic system of the clutch, shall be checked by the manufacturer's quality system as found relevant.

D. Workshop Testing

D 100 Function testing

101 The clutch shall be function tested before certification.

102 If the clutch is delivered with the activation control, the pressure-time function for clutching-in shall be verified in the presence of a surveyor. If the clutch is oil operated this shall be made with a representative oil viscosity.

E. Control, Alarm and Safety Functions and Indication

E 100 Summary

101 The clutches shall be fitted with instrumentation and alarms according to Table E1.

102 If individual local pressure indicators are not fitted, quick connectors for a portable instrument shall be provided in order to do local readings and set point verification of switches. The corresponding portable instrument shall be provided on board.

Table E1 Monitoring of clutches

	Gr 1 Indication Alarm Load reduction	*Gr 2 Automatic start of standby pump with alarm* [1)]	*Gr 3 shut down with alarm*	*Comments*
1.0 Clutch activating media				
Hydraulic/pneumatic air, pressure	IL, IR, LA	AS	SH	SH means either declutching or engine stop.

Gr 1 Common sensor for indication, alarm, load reduction (common sensor permitted but with different set points and alarm shall be activated before any load reduction)
Gr 2 Sensor for automatic start of standby pump
Gr 3 Sensor for shut down

IL = Local indication (presentation of values), in vicinity of the monitored component
IR = Remote indication (presentation of values), in engine control room or another centralized control station such as the local platform/ manoeuvring console
A = Alarm activated for logical value
LA = Alarm for low value
HA = Alarm for high value
AS = Automatic start of standby pump with corresponding alarm
LR = Load reduction, either manual or automatic, with corresponding alarm, either slow down (r.p.m. reduction) or alternative means of load reduction (e. g. pitch reduction), whichever is relevant.
SH = Shut down with corresponding alarm. May be manually (request for shut down) or automatically executed if not explicitly stated above.

For definitions of LR and SH, see Ch.1 Sec.1 B116 and 117.

1) To be provided when standby pump is required, see B501.

F. Arrangement

F 100 Clutch arrangement

101 Clutches shall be arranged to minimise radial support forces, see B202.

102 Easy access to the emergency operation device shall be provided, see B300.

G. Vibration

G 100 Engaging operation

101 The calculation of the engaging process shall be based on the particulars specified in A202 and A203. The calculation shall result in torque, flash power and heat development as functions of time, and shall not exceed the permissible values for the clutch or any other element in the system. See also Ch.3 Sec.1 G403.

H. Installation Inspection

H 100 Alignment

101 Clutches not integrated in a gearbox or thruster, shall be checked for axial and radial alignment in the presence of a surveyor.

I. Shipboard Testing

I 100 Operating of clutches

101 The following shall be checked in the presence of a surveyor:

— when engaged, the operating pressure to be within the approved tolerance
— access to the emergency operation device (see B300), if applicable
— during engaging, the operating pressure as a function of time to be according to the approved characteristics.

102 The clutch engaging as mentioned above, shall be made at the maximum permissible engaging speed. The pressure indication shall be representative for the operating pressure, i.e. measured close to the rotating seal and without throttling between the instrument and operating pressure pipe. No pressure peaks beyond the specified maximum pressure are accepted.

SECTION 4
BENDING COMPLIANT COUPLINGS

A. General

A 100 Application

101 This section applies to couplings used in machinery that is subject to certification, see Ch.2 Sec.1 A200.

Bending compliant couplings are membrane couplings, tooth couplings, link couplings, universal shafts, etc., i.e. all couplings that have a low bending rigidity, but high torsional rigidity. Couplings combining both low bending and low torsional rigidity shall fulfil the requirements in both Sec.4 and Sec.5.

102 Couplings of standard design shall be type approved.

103 Ch.2 describes general requirements for rotating machinery. Attention shall be paid to Ch.2 Sec.3 A100 and in particular A102.

104 Couplings shall be delivered with a NV certificate.

A 200 Documentation

201 Drawings showing the couplings in longitudinal section (for link couplings also transverse section) shall be submitted for approval.

The drawings shall specify:

— material specification including NDT specification
— surface hardening (if applicable)
— shot peening (if applicable)
— design details as keyways, bolt connections, or any other stress concentration.

For power transmitting welds a full NDT specification including acceptance criteria shall be submitted. For tooth couplings the tooth accuracy (ISO 1328) shall be specified.

202 For high speed couplings (for connection to gas turbines) the maximum residual unbalance shall be specified.

203 The following particulars shall be submitted:

— the permissible mean torque
— the permissible maximum torque (impact torque)
— the permissible vibratory torque for continuous operation
— the permissible angular tilt for continuous operation
— the permissible radial misalignment or reaction force (if applicable) for continuous operation
— the permissible axial misalignment for continuous operation
— the angular (tilt), radial and axial stiffness (as far as applicable)
— the maximum permissible r.p.m.

204 Calculations to substantiate the relevant particulars given in 203 shall be submitted upon request.

B. Design

B 100 General

101 For design principles see Ch.2 Sec.3 A100.

102 Couplings for turbine machinery (high speed side) containing high energy rotating parts that may be ejected in the event of a remote failure shall have special guards or design precautions.

B 200 Criteria for dimensioning

201 The couplings shall be designed with suitable safety factors against fatigue (“suitable safety factors” will depend on the method applied, but typically be about 1.5).

202 For connections as flanges, shrink fits, splines, key connections, etc. see the requirements in Sec.1 B300 to B700 respectively.

203 For membrane, link or disc couplings the safety against fatigue shall be documented:

— All relevant combinations of permissible loads (A203) shall be considered
— The calculations may be combined with results from material fatigue tests
— The safety against fatigue may also be documented by fatigue testing of the complete coupling. If so, the load and the kind of loading (or combinations thereof) shall be selected to document the safety when all permissible loads are combined.

204 Tooth couplings shall be designed to prevent tooth fracture, flank pitting and abrasive wear.

The maximum permissible radial reaction force, the permissible mean and vibratory torque, the angular misalignment and the lubrication conditions shall be combined in the calculations.

205 Universal shafts with power transmitting welds shall be designed for a high safety against fatigue in the weld. The calculation shall consider the maximum permissible loads and the specified weld quality.

The stresses in the welds combined with the maximum permissible defects according to the NDT specification shall not cause a stress intensity of more than $2\,\mathrm{MPa}\sqrt{\mathrm{m}}$.

Ball and roller bearings shall have a minimum L_{10a} (ISO 281) life time that is suitable with respect to the specified overhaul intervals.

C. Inspection and Testing

C 100 Certification

101 Regarding certification schemes, short terms, manufacturing survey arrangement and important conditions, see Ch.2 Sec.2.

C 200 Inspection and testing of parts

201 Power transmitting parts as hubs, sleeves, shaft tubes, flanges and flexible elements shall be documented with Work certificates (W) regarding chemical composition of the material, mechanical properties and surface hardness (if surface hardened).

202 NDT shall be documented with Works certificate (W).

203 Welds shall be documented in accordance with the approved specification.

D. Workshop Testing

D 100 Balancing

101 The couplings shall be balanced in accordance with the approved specification.

D 200 Stiffness verification

201 For membrane, link and disc couplings verification of the specified stiffness in angular and axial directions shall be carried out by means of static measurements in the presence of a surveyor.
This applies to:

— one coupling of a series for which type approval is requested
— every case by case approved non-standard coupling.

E. Control, Alarm, Safety Functions and Indication

E 100 General

101 Control, alarm, safety functions and indication are not required.

F. Arrangement

F 100 Coupling arrangement

101 Couplings shall be arranged to avoid the limitations given in A203 to be exceeded. Furthermore, the reaction forces from couplings on the adjacent elements shall be taken into account. All permissible operating conditions shall be considered.

G. Vibration

G 100 General

101 Intentionally left blank.

H. Installation Inspection

H 100 Alignment

101 The coupling alignment (axial, radial and angular) shall be checked in the presence of a surveyor. The alignment shall be within the approved tolerances for the coupling as well as any other limitation specified in the shafting arrangement drawings (in particular for the high speed side of gas turbine plants).

102 The alignment shall be made under consideration of all adjacent machinery such as resiliently mounted engines.

I. Shipboard Testing

I 100 General

101 Intentionally left blank.

SECTION 5
TORSIONALLY ELASTIC COUPLINGS

A. General

A 100 Application

101 This section applies to couplings used in machinery subjected to certification, see Ch.2 Sec.1 A200.

Torsional elastic couplings mean steel, rubber and silicone couplings designed for a low torsional rigidity. Couplings combining both low torsional rigidity and bending flexible elements as membranes or links shall fulfil the requirements in both Sec.4 and Sec.5.

102 Couplings of standard design shall be type approved.

103 Ch.2 describes general requirements for rotating machinery. Attention shall be paid to Ch.2 Sec.3 A100 and in particular A102.

104 Couplings shall have a NV certificate.

A 200 Documentation

201 Drawings showing the couplings in longitudinal section shall be submitted. For elements that are non-symmetrical around the axis of rotation, a transverse section is also needed.

The drawings shall specify:

— type of material and mechanical properties
— surface hardening (if applicable)
— shot peening (if applicable)
— design details as keyways, splines or any other stress concentration

For power transmitting welds a full NDT specification including acceptance criteria shall be submitted.

202 The following particulars shall be submitted:

— rubber shore hardness H (laboratory test on rubber plates)
— permissible mean torque T_{KN} with the corresponding highest nominal shear stress in the elastomer and the bonding stress
— permissible maximum torque T_{Kmax1} for repetitive loads as transient vibration, typically during clutching in etc., see Fig. 1
— permissible maximum torque range ΔT_{max} for repetitive loads as transient vibration, typically as passing through a major resonance during start and stop etc., see Fig. 2
— permissible maximum torque T_{Kmax2} for rare occasional peak loads, e.g. short circuits in generators
— permissible vibratory torque[1)] for continuous operation T_{KV}, see Fig. 3
— permissible power loss[1)] (heat dissipation) P_{KV}
— permissible angular tilt, radial and axial misalignment for continuous operation
— angular (tilt), radial and axial stiffness[1)]
— permissible permanent twist of rubber element (applicable to progressive couplings)
— maximum permissible r.p.m.
— quasi-static torsional stiffness[1)]
— dynamic torsional stiffness[1)] including production tolerance
— damping characteristics[1)] including production tolerance.

1) as a function of the main parameters.

These particulars shall be documented by means of relevant tests and calculations. See B100 and B200.

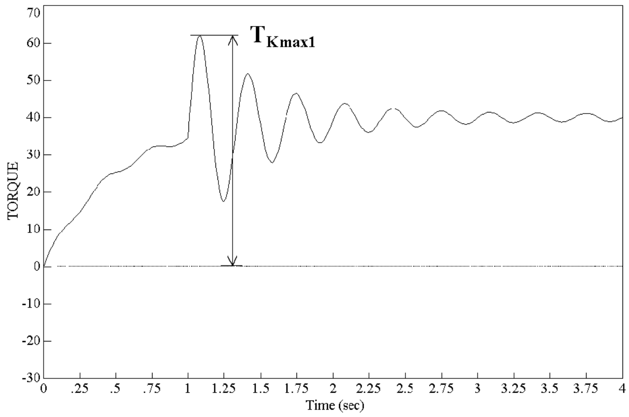

Fig. 1
T_{Kmax1} at transient vibration

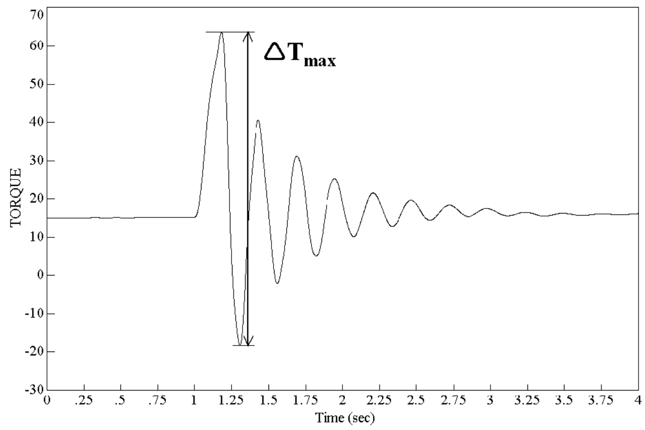

Fig. 2
ΔT_{max} at transient vibration

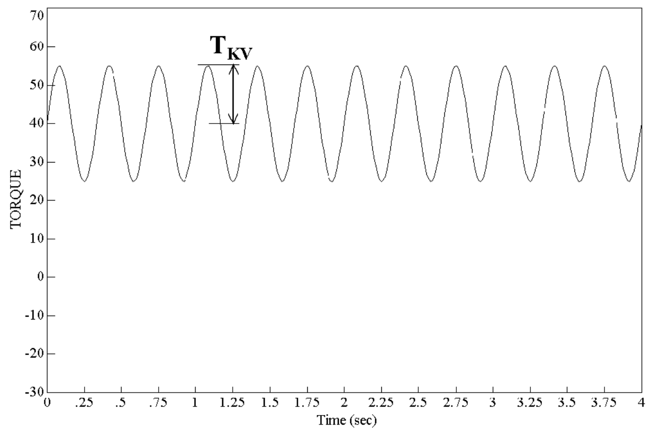

Fig. 3
T_{KV} at continuous operation

203 Definitions of stiffness and damping are:

A) For linear couplings

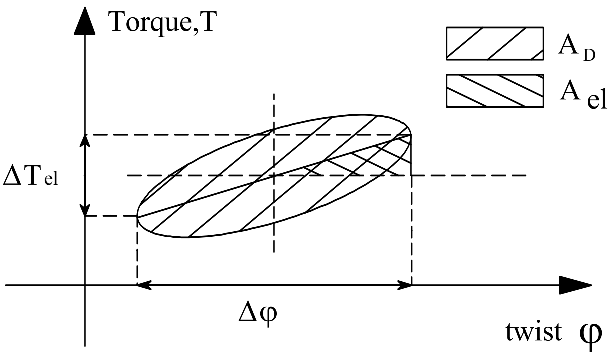

Fig. 4
Linear couplings

The stiffness K ($K = \dfrac{\Delta T_{e1}}{\Delta\varphi}$) is the gradient of a line drawn between the extreme points of the twist as indicated in Fig.4.

For hysteresis plots that deviate from the ellipse (pure viscous damping) the line that determines K shall be drawn through points determined as midpoints between the upper and lower part of the hysteresis curve, see Fig.5.

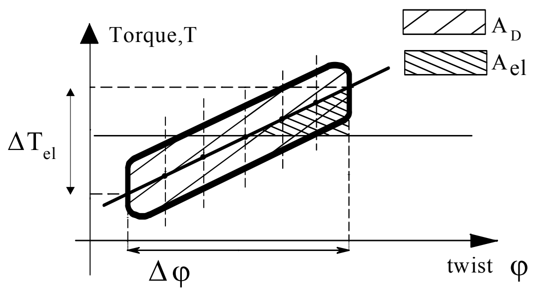

Fig. 5
Hysteresis curve

The damping is the ratio between the area described by the hysteresis loop A_D and the elastic work A_{el},

$$\psi = \frac{A_D}{A_{e1}}$$

For couplings with typical elliptical hysteresis curves, other definitions may be considered.

B) For non-linear couplings

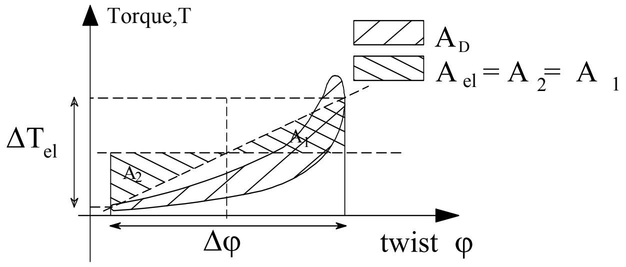

Fig. 6
Non-linear couplings

Plants with non-linear couplings may be calculated by either simulation (numeric time integration) in the time domain or in the frequency domain by linear differential equations.

In the first case the torque – twist plots can be used directly.

In the second case (more common method) representative linearised coupling properties must be used in the calculation. For this purpose the following applies.

The (linearised) stiffness K is the gradient between the extreme points of the twist as indicated above.

For determination of the damping ψ the elastic work A_{el} must be determined so that the above indicated areas of A_{el} are equal ($A_1 = A_2$). Then the same definition as for linear couplings applies.

204 The control and monitoring system, including set-points and delays, if required in E, shall be approved by the Society.

For requirements to documentation, see Ch.9.

B. Design

B 100 General

101 For design principles see Ch.2 Sec.3 A100.

102 See E101 for emergency claw devices.

B 200 Criteria for dimensioning

201 The couplings shall be designed with suitable safety factors (depending on the method applied, see B300) against fatigue and overheating (rubber).

202 For connections as flanges, shrink fits, splines, key connections, etc. see the requirements in Sec.1 B300 to B700 respectively.

203 For steel spring couplings the safety against fatigue shall be documented. All relevant combinations of permissible loads (A202) shall be considered. The calculations may be combined with results from material fatigue tests. The safety against fatigue may also be documented by fatigue testing of the complete coupling. If so, the load and the kind of loading (or combinations thereof) shall be selected to document the safety when all permissible loads are combined.

The design shall be so as to prevent fretting on vital elements.

204 Couplings shall not have rigid torsional deflection limiters (buffers) within the permissible T_{Kmax2}. Furthermore, T_{Kmax2} shall not be less than 1.4 T_{KN}.

205 For ice class notations the couplings shall be designed so that:

1) $T_{Kmax1} \geq T_0\, K_{Aice}$
2) $T_{KN} \geq 0.5\, T_0\, (K_{Aice} + 1)$
3) As long as the natural frequency of the "propeller versus engine"-mode is much lower than the propeller blade passing frequency (ratio < 50%):

   $T_{KV} \geq 0.5\, T_0\, (K_{Aice} - 1)$

   otherwise:

   $T_{KV} > T_0\, (K_{Aice} - 1)$

K_{Aice}= application factor due to ice impact loads (applicable for ice classed vessels), see Pt.5 Ch.1 of the Rules for Classification of Ships.

Alternatively to the above criteria, the ice impact loads on the elastic coupling may be documented by simulation of the transient dynamic response in the time domain. For branched systems, such simulation is in general recommended.

206 For elements that are not designed to avoid local strain concentrations, stricter values for the criteria given in 207 and 208 may apply.

For silicone couplings special considerations apply.

207 For rubber couplings with shear loaded rubber elements the shear stress (MPa) due to T_{KN} shall not exceed the smaller value of:

— 1% of the shore hardness value

or

— 0.65 MPa

The corresponding shear stress in the steel-rubber bonding surfaces is normally not to exceed 0.45 MPa.

For coupling designs where centrifugal action can be of significance, the shear stresses in the rubber element as well as in the bonding surface have to be considered. The evaluation shall take into account the influences of T_{KN} and rpm_{max} separately as well as combined. The permissible stress levels are specially considered.

The shear stress due to the permissible vibratory torque for continuous operation shall not exceed 0.25% of the shore hardness. This shear stress is superimposed to the shear stress due to T_{KN}. The corresponding peak value is not limited by T_{Kmax1} in 208.

208 When not substantiated by means of an approved fatigue testing combined with FE analyses, the following applies: Permissible torque ΔT_{max} and T_{Kmax1} for transient operation (50.000 cycles) are limited to:

a) A nominal shear stress $\Delta\tau_{max}$ not to exceed $\Delta\tau_{max} < 0.24 \cdot 10^{-3}\ H^2$

b) A nominal shear stress τ_{max1} not to exceed in any direction $\tau_{max1} < 0.2 \cdot 10^{-3}\ H^2$ and limited to $T_{Kmax1} \leq 1.5\ T_{KN}$
Note that T_{Kmax1} is not limiting the shear stress due to $T_{KN} + T_{KV}$.

209 For couplings having elements that are loaded in compression, T_{Kmax1} will be specially considered.

210 The strength of the emergency claw device (if required, see 102) shall be documented by calculations. This device shall be designed for a minimum lifetime of 24 hours and combined with all permissible misalignments.

211 Couplings of natural rubber shall not be subjected to ambient temperatures above 70°C. The limit for silicone couplings is 100°C.

B 300 Type testing

301 Type testing applies to all rubber and silicone couplings, but also for special kinds of steel spring couplings.

302 Steel spring couplings that are designed such that the damping properties are essentially non-viscous (e.g. mainly friction damping), shall be dynamically tested in order to establish the dynamic characteristics (stiffness and damping) as functions of their main parameters.

303 Rubber and silicone couplings shall be documented with regard to compatibility with the characteristics and permissible loads given in A202. This shall be made with both calculation and testing:

— As a minimum the dynamic torsional stiffness and the damping shall be verified by testing, see 304. A reduced extent may apply for couplings that are approved for very restricted applications as e.g. in electric motor driven thrusters.
— Couplings used in plants with reciprocating machinery are also to be tested for determination of permissible power loss. Exemptions may only be made if the value for P_{KV} is assessed very much to the safe side.
— The necessity for test documentation of the angular (tilt), radial and axial stiffness depends on the corresponding values for permissible misalignment.
— For case by case approval of a non-standard coupling the documentation (i.e. testing) applies to the necessity for the actual coupling application.
— For type approval of a coupling series where the coupling sizes only differ by a scale factor, the documentation testing for stiffness and damping of only one size (per rubber type) may be sufficient. However, if power loss testing applies, this testing shall be made with at least two different coupling sizes in order to extrapolate for inclusion of the whole series.
— Quasi-static tests such as described in D200 shall be made with the same elements as used for the dynamic testing, and prior to it. The purpose shall establish reference values for certification testing.

304 The purpose for the testing of stiffness and damping shall establish the relations between the quasi-static tests mentioned above and the dynamic behaviour of the coupling. Furthermore, the type testing shall establish the dynamic torsional stiffness and damping (for the relevant rubber qualities of relevant element sizes) as functions of the main parameters such as:

— Mean torque T_M, normally at steps as

T_M/T_{KN} =	0	0.25	0.50	**0.75**	1

— Vibratory torque T_V, normally at steps as

T_V/T_{KV} =	0.50	**1.0**	2.0 *

(* for the purpose of transient vibrations)
— Vibration frequency, normally at steps as

2 Hz	**10 Hz**	20 Hz

and for elements loaded in compression, also at 40 Hz

— Temperature of the element. This is for the purpose of establishing representative stiffness and damping values under various ambient temperatures as well as under high power losses.
Normally at:
reference condition, e.g. 30°C
25% of permissible P_{KV} *
100% of permissible P_{KV} *

For couplings not to be used in diesel engine plants the tests at reference condition will be sufficient.

* P_{KV} as for rotating or non-rotating coupling, whichever is relevant for the laboratory.

It is not required to test all the possible combinations of the conditions mentioned above. Normally reference conditions as e.g. the **bold** values above, are kept constant when one parameter dependency is tested. However, for typically progressive couplings (stiffness increasing with torque) all permissible combinations of mean and vibratory torques shall be tested.

The test results shall be presented as torque-twist plots, together with the details of the evaluation method.

305 The testing of the permissible power loss shall be made by means of at least one temperature sensor in the rubber core at the expected (calculated) position of maximum temperature (position to be approved prior to the testing).

The core temperature during pulsating of the element shall be plotted as a function of time until the end temperature is stabilised. The maximum permissible core temperature is 110°C for natural rubber and 150°C for silicone.

The permissible power loss P_{KV} is defined as the power loss that results in the maximum permissible core temperature. P_{KV} is normally tested at an ambient temperature of 20°C and shall be linearly interpolated to zero at maximum permissible core temperature as a function of operating ambient temperature. For coupling series where the sizes only differ by a scale factor, interpolation and extrapolation may be done by the following formula:

$$P_{KV} = a \ T_{KN}^{b}$$

where the constants a and b can be determined by testing two or more different sizes of couplings in a series.

The power loss is normally to be measured by means of torque-twist plots and applied frequency. Alternative methods may be considered if their relevance can be documented and the results are estimated to the safe side.

When a steady state condition is reached, e.g. not more than 1°C increase per hour, the actual power loss is determined from a torque-twist plot as $P_{KVtest} = A_D \cdot f$ (Hz).

If the core temperature during this test ϑ_{test} is different from the permissible value ϑ_p, the P_{KV} is determined as:

$$P_{KV} = P_{KVtest} \cdot \frac{\vartheta_p - \vartheta_{Aref}}{\vartheta_{test} - \vartheta_A}$$

ϑ_A = ambient temperature during test
ϑ_{Aref} = reference temperature in catalogue.

Alternative methods to torque-twist pulsating may only be accepted if the evaluation of P_{KV} is made conservatively (to the safe side). If rotating with radial or angular misalignment is used, the assessment of the actual power loss in the elements must consider all possible increase of other losses (e.g. in bearings).

Further, the different temperature field versus the real one in torque-twist must be taken into consideration by e.g. finite element analyses or preferably by comparison measurements in order to arrive at a correlation factor between the applied method and the real torque-twist condition.

C. Inspection and Testing

C 100 Certification

101 Regarding certification schemes, short terms, manufacturing Survey Arrangement and important conditions, see Ch.2 Sec.2.

C 200 Inspection and testing of parts

201 Power transmitting parts as hubs, sleeves, shaft tubes, flanges etc. shall be documented with Work certificates (W) regarding chemical composition of the material, mechanical properties and surface hardness.

202 NDT shall be documented with Works certificate (W).

203 Welds shall be documented with a Work certificate (W) in accordance with the approved specification.

D. Workshop Testing

D 100 Stiffness verification

101 Each rubber or silicone coupling or elastic element shall be verified with regard to quasi-static torsional stiffness in the presence of a surveyor. This shall be done by twisting the coupling or by subjecting the elastic elements to a load which is equivalent to the coupling twist. The test torque shall be at least 1.5 T_{KN}. The resulting deflection shall be within the approved tolerance and the deviation shall be specified in the certificate.

102 For couplings that are not approved for use in plants with reciprocating machinery a reduced extent of testing may be accepted.

103 For segmented couplings the assembling of a coupling with segments from different charges (possibly different stiffness) shall be within the approved tolerance range for segment differences.

D 200 Bonding tests

201 For couplings with bonded rubber or silicone elements the bonding shall be checked in the presence of a surveyor. The coupling or elastic element shall be loaded in at least one direction to the 1.5 T_{KN}. At this load the element shall be inspected for any signs of slippage in the bonding surface. Additionally the corresponding torque-deflection curve shall be smooth and show no signs of slippage in the bonding.

202 The bonding may also be documented by alternative tests as e.g. tension where the tensile stress shall be at least as high as the shear stress under 1.5 T_{KN}.

203 For couplings that have a limitation of the permanent twist (all progressive couplings) shall be marked so that the actual permanent twist and the limit twist are legible during service inspections.

D 300 Balancing

301 Couplings for PTO/PTI branches shall be single plane balanced when:

— tip speed >30 m/s
— unmachined surfaces and tip speed >10 m/s

E. Control, Alarm, Safety Functions and Indication

E 100 Summary

101 The elastic couplings for propulsion of single diesel engine plants shall be fitted with instrumentation and alarms according to Table E1.

102 For couplings where twist amplitude alarm is chosen for monitoring of torsional vibration, see Ch.3 Sec.1 G302 j), G303 d) and E103 b), the alarm levels and time delays are subject to special consideration.

Table E1 Monitoring of elastic couplings for single diesel engine propulsion plants

	Gr 1 Indication Alarm Load reduction	*Gr 2 Automatic start of standby pump with alarm*	*Gr 3 shut down with alarm*	*Comments*
1.0 Twist of elastic couplings				
Angular twist amplitudes	IR, HA		SH	Applicable when failure of the elastic element leads to loss of torque transmission [1]
Mean twist angle	IR, HA			

Gr 1 Common sensor for indication, alarm, load reduction (common sensor permitted but with different set points and alarm shall be activated before any load reduction)
Gr 2 Sensor for automatic start of standby pump
Gr 3 Sensor for shut down

IL = Local indication (presentation of values), in vicinity of the monitored component
IR = Remote indication (presentation of values), in engine control room or another centralized control station such as the local platform/ manoeuvring console
A = Alarm activated for logical value
LA = Alarm for low value
HA = Alarm for high value
AS = Automatic start of standby pump with corresponding alarm
LR = Load reduction, either manual or automatic, with corresponding alarm, either slow down (r.p.m. reduction) or alternative means of load reduction (e. g. pitch reduction), whichever is relevant.
SH = Shut down with corresponding alarm. May be manually (request for shut down) or automatically executed if not explicitly stated above.

For definitions of LR and SH, see Ch.1 Sec.1 B116 and 117

[1] May be omitted if the vessel is equipped with a “take me home” device, e.g. a electric motor connected to the gearbox (so-called PTH or PTI). Exemption may also be accepted for couplings that are of a design that enables the full torque to be transmitted in the event of failure of the elastic elements. Such emergency claw devices are not “getting home” devices, but only meant for temporary emergency in order to prevent loss of manoeuvrability in harbours, rivers, etc.

F. Arrangement

F 100 Coupling arrangement

101 Couplings shall be arranged to avoid that the limitations given in A202 are exceeded. Furthermore, the reaction forces from couplings on the adjacent elements shall be taken into account. All permissible operating conditions shall be considered.

Guidance note:
Typical ambient temperature are:

- bell housing (with ventilation openings) 70°C
- free standing at flywheel of diesel engine up to 50°C
- free standing PTO branch from a gearbox 30°C
- outside main engine room, special consideration.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

G. Vibration

G 100 General

101 Torsional vibration is covered by the relevant section for the prime mover, e.g. diesel engines in Ch.3 Sec.1 G. Lateral vibration is covered by Sec.1 G100 and G200.

102 Lateral vibration calculations of arrangements with segmented couplings may be required. The calculations shall consider the rotating forces due to possible unbalanced tangential forces (1.0 order) at full torque as well as corresponding forces due to torsional vibration. Stiffness variations, in accordance with the approved tolerance for the segmented coupling, shall be assumed.

103 The coupling data as stiffness and damping used for torsional vibration analysis shall be representative for the actual ambient temperature as well as the temperature rise due to power loss. Further, the specified production tolerances shall be considered.

H. Installation Inspection

H 100 Alignment

101 The coupling alignment (axial, radial and angular) shall be checked in the presence of a surveyor. The alignment shall be within the approved tolerances for the coupling as well as any other limitation specified in the shafting arrangement drawings.

102 The alignment shall be made under consideration of all adjacent machinery such as resiliently mounted engines, etc.

I. Shipboard Testing

I 100 Elastic elements

101 After the sea trial all rubber elements in propulsion plants and power take off branches shall be visually checked by a surveyor. No cracks or deterioration are acceptable.