RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D Machinery Installations

Rules for the Survey and Construction of Steel Ships
Part D 2022 AMENDMENT NO.1
Guidance for the Survey and Construction of Steel Ships
Part D 2022 AMENDMENT NO.1

Rule No.45 / Notice No.31 30 June 2022
Resolved by Technical Committee on 26 January 2022

ClassNK
NIPPON KAIJI KYOKAI

An asterisk (*) after the title of a requirement indicates that there is also relevant information in the corresponding Guidance.

RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

RULES

Part D Machinery Installations

2022 AMENDMENT NO.1

Rule No.45 30 June 2022
Resolved by Technical Committee on 26 January 2022

An asterisk (*) after the title of a requirement indicates that there is also relevant information in the corresponding Guidance.

Rule No.45 30 June 2022
AMENDMENT TO THE RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

"Rules for the survey and construction of steel ships" has been partly amended as follows:

Part D MACHINERY INSTALLATIONS

Amendment 1-1

Chapter 21 SELECTIVE CATALYTIC REDUCTION SYSTEMS AND ASSOCIATED EQUIPMENT

21.1 General

21.1.1 Application

Sub-paragraph -2 has been amended as follows.

2 Urea based ammonia (e.g. AUS 40 (a 40% urea – 60% water aqueous urea solution specified in *ISO* 18611-1:2014) is to be used as reductant agent in SCR systems. In cases where another reductant agent is used, however, special consideration is to be given to such systems in accordance with their respective designs as well as the following **(1)** and **(2)**:
((1) and (2) are omitted)

21.4 Requirements for Construction and Arrangements, etc.

21.4.1 Construction and Arrangement

Sub-paragraph -2 has been amended as follows.

2 Reductant agent storage tanks are to be protected from excessively high or low temperatures applicable to the particular concentration of the solution. Depending on the operational area of the ship, this may necessitate the fitting of heating and/or cooling systems. The physical conditions recommended by applicable recognized standards (such as *ISO* 18611-3:2014) are to be taken into account to ensure that the contents of the reductant agent tank are maintained to avoid any impairment of the reductant agent during storage.

EFFECTIVE DATE AND APPLICATION (Amendment 1-1)

1. The effective date of the amendments is 30 June 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to SCR systems other than those which fall under the following:

(1) SCR systems for which the application for approval of use is submitted to the Society on or after 1 January 2022; or

(2) SCR systems used on ships for which the date of contract for construction* is on or after 1 January 2022.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.

For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:

(1) such alterations do not affect matters related to classification, or

(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.

The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

Amendment 1-2

Chapter 1 GENERAL

1.1 General

1.1.4 Modification of Requirements*

Sub-paragraphs (1) and (2) have been amended as follows.

For the following machinery installations, piping systems and all their respective control systems, some requirements of this Part may be modified appropriately provided that the Society considers such modifications acceptable:

(1) Small prime movers (including power transmission systems and shafting systems) for either driving generators or auxiliary machinery ~~(including power transmission systems and shafting systems)~~

(2) Auxiliary machineries for cargo handling and their prime movers (including power transmission systems and shafting systems)

(3) Machinery installations as deemed appropriate by the Society after considering their capacity, purpose and conditions of service

Chapter 6 SHAFTINGS

6.1 General

Paragraph 6.1.2 has been amended as follows.

6.1.2 Drawings and Data*

Drawings and data to be submitted are generally as follows:

(1) Drawings for approval (including specifications of material)

((a) to (k) are omitted.)

(l) In the case of propeller shafts Kind 1*C*, four sets of drawings and data of the following **i)** to **viii)**:

(i) to vii) are omitted.)

viii) Shaft alignment calculation sheets in accordance with **Annex 6.2.13**.

(2) (Omitted)

6.2 Materials, Construction and Strength

6.2.2 Intermediate Shafts*

Sub-paragraph -1 has been amended as follows.

1 The diameter of the intermediate shafts made of steel forgings (excluding stainless steel forgings, etc.) is not to be less than the value given by the following formula:

$$d_0 = F_1 k_1 \cdot \sqrt[3]{\frac{H}{N_0}\left(\frac{560}{T_s + 160}\right)K}$$

where

d_0 : Required diameter of intermediate shaft (*mm*)

H : Maximum continuous output of engine (*kW*)

N_0 : Number of revolutions of intermediate shaft at maximum continuous output (*rpm*)

F_1 : Factor given in **Table D6.1**

k_1 : Factor given in **Table D6.2**

T_s : Specified tensile strength of intermediate shaft material (N/mm^2)

The upper limit of the value of T_S used for the calculation is to be 760 N/mm^2 for carbon steel forgings and 800 N/mm^2 for low alloy steel forgings. The upper limit of the value of T_s used for the calculation may be increased to 950 N/mm^2 ~~where deemed appropriate by the Society~~ when intermediate shafts are manufactured using steel forgings (excluding stainless steel forgings) which have specified minimum tensile strengths greater than 800 N/mm^2 and are in accordance with **Annex 6.2.2 "Use of High-Strength Materials for Intermediate Shafts"**.

K : Factor for hollow shaft and given by the following formula. In cases where $d_i \leq 0.4d_a$, it may be considered that $K = 1$

$$K = \frac{1}{1 - \left(\frac{d_i}{d_a}\right)^4}$$

where

d_i : Inside diameter of hollow shaft (*mm*)

d_a : Outside diameter of hollow shaft (*mm*)

2 The diameter of the intermediate shaft of material other than specified in **-1** above is to be deemed appropriate by the Society.

(Table D6.1 are omitted.)

Table D6.2 has been amended as follows.

Table D6.2 Values of k_1

Shaft with integral flange coupling (1)	Shaft with flange coupling either shrink fit, push fit or cold fit (~~1~~2)	Shaft with keyway (~~2~~3)(4)	Shaft with transverse hole (~~3~~5)	Shaft with longitudinal slot (~~4~~6)	Shaft with splines (~~5~~7)
1.0	1.0	1.1	1.1	1.2	1.15

Notes:

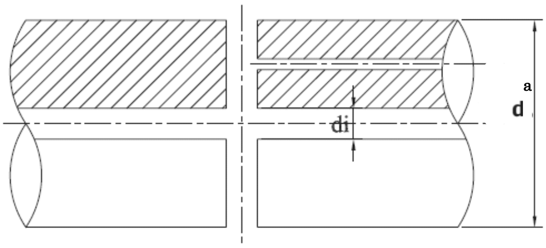
(1) The fillet radius at the base of the flange is not to be less 0.08 *times* the diameter of the shaft.

(~~1~~2) In cases where shafts, during continuous operation, experience torsional vibration stress exceeding 85 % of τ_1 given in **8.2.2-1(1)**, an increase of 1 to 2 % in diameter to the fit diameter and a blending radius nearly equal to the change in diameter are to be provided.

(~~2~~3) After a length of not less than 0.2 d_0 from the end of the keyway, the diameter of a shaft may be reduced progressively to the diameter calculated with k_1=1.0.
The fillet radius in the transverse section of keyway bottom is to be 0.0125 d_0 or more.

(4) Keyways are in general not to be used in installations with a barred speed range in accordance with **8.3**.

(~~3~~5) The diameter of the hole is not to be more than 0.3 d_0. When a transverse hole intersects an eccentric axial hole (see below), the value is to be determined by the Society based on the submitted data in each case.



(~~4~~6) The shape of the slot is to be in accordance with the following: any edge rounding other than by chamfering is to be avoided in principle; the number of slots is to be 1, 2 or 3 and they are to be arranged 360, 180 or 120 *degrees* apart from each other respectively.

(a) $l < 0.8d_a$
(b) $d_i < 0.7d_a$
(c) $0.15d_a < e \leqq 0.2d_a$
(d) r ~~=~~ $\geqq e / 2$

where

l : slot length
d_a : outside diameter of the hollow shaft
d_i : inside diameter of the hollow shaft
e : slot width
r : end rounding of the slot

(~~5~~7) The shape of the spline is to conform to *JIS B* 1601 or the equivalent thereof.

6.2.3 Thrust Shafts

Sub-paragraph -3 has been renumbered to sub-paragraph -4, and sub-paragraph -3 has been added as follows.

(-1 and -2 are omitted.)

3 The fillet radius at the base of the thrust collar on both sides is not to be less 0.08 *times* the diameter of the shaft.

~~3~~4 The diameter of the thrust shaft of material other than specified in **-1** above is to be deemed appropriate by the Society.

6.2.4 Propeller Shafts and Stern Tube Shafts*

Sub-paragraph -3 has been renumbered to sub-paragraph -4, and sub-paragraph -3 has been added as follows.

(-1 and -2 are omitted.)

3 The shaft diameter may be reduced by either a smooth taper or a blending radius nearly equal to the change in diameter to the diameter calculated by the formula given in **6.2.2-1** at the portions located forward of the fore end of the fwd stern tube seal. In cases where shafts are manufactured using stainless steel, shaft diameters calculated as $T_s = 400$ are to be used.

~~3~~4 The diameters of propeller shafts and stern tube shafts other than those prescribed in **-1** and **-2** are to be deemed appropriate by the Society.

Table D6.3 has been amended as follows.

Table D6.3 Values of k_2

	Application		k_2
1	The portion between the big end of the tapered part of propeller shaft (in cases where the propeller is fitted with a flange, the fore face of the flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_S, whichever is greater	For a shaft carrying a keyless propeller, or where the propeller is attached to an integral flange	1.22
		For a shaft carrying a keyed propeller	1.26
2	Excluding the portion given in 1 above, the portion up to the fore end of the fwd stern tube seal in the direction of the bow		1.15$^{(1)}$
3	Stern tube shaft		1.15$^{(1)}$
4	The portion located forward of the fore end of the fwd stern tube seal		1.15~~(2)~~

Notes:

(1) At the boundary, the shaft diameter is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

~~(2) The shaft diameter may be reduced by either a smooth taper or a blending radius nearly equal to the change in diameter to the diameter calculated by the formula given in **6.2.2**.~~

Table D6.4 has been amended as follows.

Table D6.4 Values of k_3

	Application	*KSUSF* 316 *KSUS*316-*SU*	*KSUSF* 316*L* *KSUS*316*L*-*SU*
1	The portion between the big end of the tapered part of propeller shaft (in cases where the propeller is fitted with a flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_S, whichever is greater	1.28	1.34
2	Excluding the portion given in 1 above, the portion up to the fore end of the fwd stern tube seal in the direction of the bow	1.16(1)	1.22(1)
3	The portion located forward of the fore end of the fwd stern tube seal	1.16~~(2)~~	1.22~~(2)~~

Notes:

(1) At the boundary, the shaft diameter is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

~~(2) The shaft diameter may be reduced by either a smooth taper or a blending radius nearly equal to the change in diameter to the diameter calculated by the formula given in **6.2.2-1** considering T_S = 400.~~

6.2.7 Corrosion Protection of Propeller Shafts and Stern Tube Shafts*

Sub-paragraph -3 has been amended as follows.

3 Spaces between the propeller cap or propeller boss and the propeller shaft are to be filled up with ~~tallow,~~grease or provided with other effective means to protect the shaft against corrosion by water.

6.2.10 Stern Tube Bearings and Shaft Bracket Bearings*

Sub-paragraph -1 has been amended as follows.

1 The aftermost stern tube bearing or shaft bracket bearing which supports the weight of propeller is to comply with the following requirements **(1)** to **(3)**:

(1) In the case of oil lubricated bearings.

(a) In the case of white metal~~,~~

i) The length of the bearing is not to be less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**. However, where the nominal bearing pressure (determined by the static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft in way of the bearing, hereinafter defined the same way in this chapter) is not more than 0.8 *MPa* and special consideration is given on the construction and arrangement in accordance with provisions specified elsewhere ~~and specially approved by the Society~~, the length of the bearing may be fairly shorter than that specified above. However, the minimum length is to be not less than 1.5 *times* the actual diameter of the propeller shaft.

ii) The stern tube is to be always filled with oil. Adequate means are to be provided to measure the temperature of oil in the stern tube.

iii) In cases where a gravity tank supplying lubricating oil to the stern tube bearing is fitted, it is to be located above the load water line and provided with a low level alarm device. However, in cases where the lubricating system is designed to be used under the condition that the static oil pressure of the gravity tank is lower than the water pressure, the tank is not required to be above the load water line.

iv) The lubricating oil is to be cooled by submerging the stern tube in the water of the after peak tank or by some other suitable means.

(b) In the case of materials other than white metal~~.~~

i) The materials, construction and arrangement are to be approved by the Society.

ii) For bearings of synthetic rubber, reinforced resin or plastics materials which are approved for use as oil lubricated stern tube bearings, the length of the bearing is to be not less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**. However, where nominal bearing pressure is not more than 0.6 *MPa* and bearings have a construction and arrangement ~~specially approved by the Society~~ <u>in accordance with provisions specified elsewhere</u>, the length of the bearing may be fairly shorter than that specified above. However, the minimum length is to be not less than 1.5 *times* the actual diameter of the propeller shaft.

iii) Notwithstanding the requirement given in **ii)**, the Society may allow use of bearings whose nominal bearing pressure is more than 0.6 *MPa* where the material has proven satisfactory testing and operating ~~experience~~histories.

(2) In the case of water lubricated bearings~~.~~

(a) The materials, construction and arrangement are to be approved by the Society.

(b) The length of the bearing is to be not less than 4 *times* the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**, or 3 *times* the actual diameter, whichever is greater. However, for bearings of synthetic materials, such as rubber or plastics, that are approved for use as water lubricated stern tube bearings and where special consideration is given to their construction and arrangement in accordance with provisions specified elsewhere, the length of the bearing may be fairly shorter than that specified above. However, minimum length is to be not less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**, or 1.5 *times* the actual diameter, whichever is greater.

(3) In the case of grease lubricated bearings~~.~~

In cases where the actual diameter of the propeller shaft is not more than 100 *mm*, grease lubricated bearings may be used. The length of the bearing is to be not less than 4 *times* the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**.

Paragraph 6.2.13 has been amended as follows.

6.2.13 Shaft Alignment~~*~~

For the main propulsion shafting having an oil-lubricated propeller shaft of which diameter is not less than 400 *mm*, the shaft alignment calculation <u>in accordance with **Annex 6.2.13**</u> including bending moments, bearing loads and deflection curve of the shafting is to be ~~carried out for approval~~ <u>submitted to the Society for approval</u>.

Chapter 8 TORSIONAL VIBRATION OF SHAFTINGS

8.2 Allowable Limit

8.2.2 Intermediate Shafts, Thrust Shafts, Propeller Shafts and Stern Tube Shafts*

Sub-paragraph -1 has been amended as follows.

1 For ships in which the reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), the torsional vibration stresses acting on the intermediate shafts, thrust shaft, propeller shafts and stern tube shafts made of steel forgings (excluding stainless steel, etc.) are to be in accordance with the following requirements **(1)** and **(2)**. However, those shafts classified as either propeller shafts Kind 2 or stern tube shafts Kind 2 are to be deemed appropriate by the Society.

(1) For continuous operation, when the number of revolutions is within the range of 80 % to 105 % of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in the following formulae:

$$\tau_1 = \frac{T_s + 160}{18} C_K C_D (3 - 2\lambda^2)(\lambda \leq 0.9)$$

$$\tau_1 = 1.38 \frac{T_s + 160}{18} C_K C_D (0.9 < \lambda)$$

where

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \leq 1.05$ (*N/mm*^{2})

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

T_s : Specified tensile strength of shaft material (*N/mm*^{2})

However, the value of T_s for using in the formulae is not to exceed 800 *N/mm*^{2} (600*N/mm*^{2} for carbon steels in general) in intermediate shafts and thrust shafts, and 600 *N/mm*^{2} in propeller shafts and stern tube shafts. The upper limit of the value of T_s used for the calculation may be increased to 950 *N/mm*^{2} ~~in intermediate shafts~~ where ~~deemed appropriate by the Society~~ the intermediate shafts are manufactured using steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800 *N/mm*^{2} and are in accordance with the requirements of **Annex 6.2.2 "Use of High-Strength Materials for Intermediate Shafts"**. Where propeller shafts and stern tube shafts are made of the approved corrosion resistant materials or other materials having no effective means against corrosion by sea water, the value of T_s for use in the formulae is to be as deemed appropriate by the Society.

C_K : Coefficient concerning to the type and shape of the shaft, given in **Table D8.1**.

C_D : Coefficient concerning to the shaft size and determined by the following formula:

$$C_D = 0.35 + 0.93 d^{-0.2}$$

$d =$ Diameter of the shaft(mm)

(2) (Omitted)

Table D8.1 has been amended as follows.

Table D8.1 Values of C_K (~~5~~4)

Intermediate shaft with						Thrust shaft		Propeller shaft and stern tube shaft	
integral flange coupling	flange couplings either shrink fit, push fit or cold fit	keyway, tapered connection	Keyway, cylindrical connection	transverse hole ~~(1)~~	longitudinal slot (~~2~~1)	on both sides of thrust collar	in way of part subjected to axial load of roller bearing	near the big end of the tapered part of propeller shaft (~~3~~2)	excluding the portion given in the left column (~~4~~3)
1.0	1.0	0.6	0.45	0.50	0.30	0.85	0.85	0.55	0.80

Notes:

~~(1) To be in accordance with **note (3) of Table D6.2**.~~

(~~2~~1) ~~To be in accordance with **note (4) of Table D6.2**.~~ For intermediate shafts with longitudinal slots, values of C_K may be determined using the following formulae:

$$C_K = 1.45/scf$$

$$scf = \alpha_{t(hole)} + 0.80\frac{(l-e)/d_a}{\sqrt{\left(1-\frac{d_i}{d_a}\right)\frac{e}{d_a}}}$$

where

scf : Stress concentration factor at the end of slots defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress determined for the hollow shafts without slots (values obtained through Finite Element Calculation may be used as well)

l : Slot length

e : Slot width

d_i : Inside diameter of the hollow shaft at the slot

d_a : Outside diameter of the hollow shaft

$\alpha_{t(hole)}$: Stress concentration factor of radial holes (in this context, e = hole diameter) determined by the following formula (an approximate value of 2.3 may be used as well)

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

(~~3~~2) The portion between the big end of the tapered part of the propeller shaft (in cases where the propeller is fitted with a flange, the fore face of the flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_s, whichever is greater. In this case d_s is the required diameter of the propeller shaft or stern tube shaft.

(~~4~~3) The portion in the direction of the bow up to the fore end of the fwd stern tube seal.

(~~5~~4) Any value of C_K other than those above is to be determined by the Society based on the submitted data in each case.

(Table D8.2 is omitted.)

8.3 Barred Speed Range

8.3.1 Barred Speed Range for Avoiding Continuous Operation*

Sub-paragraphs -1 to -3 have been amended as follows.

1 In cases where the torsional vibration stresses exceed the allowable limit τ_1 specified in **8.2**, barred speed ranges are to be marked with red zones on the engine tachometers and these ranges are to be passed through as quickly as possible. In this case, barred speed ranges are to be imposed in accordance with the following:

(1) The barred speed ranges are to be imposed between the following speed limits.

$$\frac{16N_c}{18-\lambda} \leq N_0 \leq \frac{(18-\lambda)N_c}{16}$$

where

N_0 : The number of revolutions to be barred (*rpm*)

N_c : The number of revolutions at the resonant critical (*rpm*)

λ : Ratio of the number of revolutions at the resonant critical to the number of maximum continuous revolutions

(2) For controllable pitch propellers, both full and zero pitch conditions are to be considered.

(3) The tachometer tolerance is to be considered.

(4) The engines are to be stable in operation at each end of barred speed ranges.

(~~3~~5) Restricted speed ranges in one cylinder misfiring conditions are to enable safe navigation even where the ship is provided with only one propulsion engine.

2 In cases where the range in which the stresses exceed the allowable limit τ_1 specified in **8.2** is verified by measurements, such range may be taken as the barred speed range for avoiding continuous operation, notwithstanding the required range specified in **-1**~~, having regard to the tachometer accuracy~~.

3 For engines ~~where~~for which clearing the barred speed range for avoiding continuous operation specified in **8.3.1-1** and **-2** above is not readily available, transferring of the resonant points of torsional vibrations and other necessary measures are to be taken.

Chapter 12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

12.1 General

12.1.2 Terminology

Sub-paragraph -4 has been amended as follows.

4 Flexible Hose Assemblies

Flexible hose assemblies are short length metallic or non-metallic hoses that are normally ~~those~~ flexible ~~hoses~~ with end fittings. Flexible hose assemblies for essential services or containing either flammable or toxic media are not to exceed 1.5 *m* in length.

Chapter 13 PIPING SYSTEMS

13.2 Piping

13.2.1 General*

Sub-paragraph -5(1) has been amended as follows.

5 Protection of piping systems and fittings

(1) All pipes, including seawater pipes, valves, cocks, pipe fittings, valve operating rods, handles, etc. in cargo holds for dry cargoes (including cargo spaces of container ships and ro-ro ships) are to be protected from ~~mechanical damage~~ ~~in cases where they are located in cargo holds or other spaces where they may be subject to impacts (e.g. fish holds, chain lockers)~~ impact of cargo where they are liable to be damaged. Where a casing is provided for protection, the casing is to be constructed so as to facilitate easy removal for inspection.

Annex 6.2.2 has been added as follows.

Annex 6.2.2 USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS

1.1 Application

This annex applies to low alloy steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800 N/mm^2, but less than 950 N/mm^2 and which are intended for use as intermediate shaft material.

1.2 Torsional Fatigue Test

1.2.1 General Requirements

A torsional fatigue test is to be performed to verify that the material exhibits similar fatigue life as conventional steels. The torsional fatigue strength of said material is to be equal to or greater than the allowable limit of the torsional vibration stresses τ_1 given by the formulae in **8.2.2-1(1), Part D of the Rules**. The test is to be carried out with notched and unnotched specimens respectively. For calculation of the stress concentration factor of the notched specimen, the notch factor is to be evaluated in consideration of the severest torsional stress concentration in the design criteria.

1.2.2 Test Conditions

Test conditions are to be in accordance with **Table 1.1**. Mean surface roughness is to be less than 0.2 μm for R_a and the absence of localised machining marks is to be verified by visual examination at low magnification (x20) as required by *Section* 8.4 of *ISO* 1352:2011. Test procedures are to be in accordance with *Section* 10 of *ISO* 1352:2011.

Table 1.1 Test conditions

Loading type	Torsion
Stress ratio	R = −1
Load waveform	Constant-amplitude sinusoidal
Evaluation	S-N curve
Number of cycles for test termination	1 x 10^7 cycles

1.2.3 Acceptance criteria

Measured high-cycle torsional fatigue strength τ_{C1} and low-cycle torsional fatigue strength τ_{C2} are to be equal to or greater than the values given by the following formulae:

$$\tau_{C1} \geq \tau_{1,\lambda=0} = \frac{\sigma_B + 160}{6} \cdot C_K \cdot C_D$$

$$\tau_{C2} \geq 1.7\tau_{C1}/\sqrt{C_K}$$

where

C_K: Coefficient related to the type and shape of the shaft. To be determined using the formulae (modified as needed) specified in **Note (1) of Table D8,1, Part D of the Rules**. However, the stress concentration factor for computing C_K can be determined in consideration of actual design conditions. For unnotched specimens, the stress concentration factor is 1.0.

C_D: Coefficient related to shaft size. To be determined using the formula (modified as needed) specified **8.2.2-1(1), Part D of the Rules**.

σ_B: Specified tensile strength of the shaft material (N/mm^2)

1.3 Cleanliness Requirements

Low alloy steel forgings are to have a degree of cleanliness shown in **Table 1.2** when tested according to *ISO* 4967:2013 *method A*. Representative samples are to be obtained from each heat of forged or rolled products. In addition, the forgings are also to comply with the minimum requirements of **Table K6.2, Part K of the Rules**, with particular attention given to minimising the concentrations of sulphur, phosphorus and oxygen in order to achieve the cleanliness requirements. The specific steel composition is required to be approved by the Society.

Table1.2 Cleanliness requirements

Inclusion group	Series	Limiting chart diagram index I
Type A	Fine	1
	Thick	1
Type B	Fine	1.5
	Thick	1
Type C	Fine	1
	Thick	1
Type D	Fine	1
	Thick	1
Type DS	-	1

1.4 Inspection

Low alloy steel forging are to be subjected to the ultrasonic testing specified in **6.1.10-1(1), Part K of the Rules**.

Annex 6.2.13 has been added as follows.

Annex 6.2.13 CALCULATION OF SHAFT ALIGNMENT

1.1 General

1.1.1 Application

1 This annex applies to the shaft alignment calculations required by **6.2.10**, **6.2.11** and **6.2.13, Part D of the Rules**. With regard to the paragraphs in **1.3** of this annex, application is to be in accordance with **Table 1.1.1-1**.

Table 1.1.1-1 Application of conditions for calculation etc.

Type of main propulsion machinery	Paragraphs 1)2)		
	1.3.1	**1.3.2**	**1.3.3** 3)
Two-stroke cycle engines	●	●	●
Four-stroke cycle engines	●	●	-
Steam turbines	●	●	-

Notes:

1) ●: Applicable -: Not applicable
2) **1.3.1**: Light draught condition (cold condition)
   1.3.2: Light draught condition (hot condition)
   1.3.3: Full draught condition (hot condition)
3) Only applicable to oil tankers, ships carrying dangerous chemicals in bulk, bulk carriers and general dry cargo ships in the following cases:
   - Oil tankers are those ships defined in **1.3.1(11)**, **Part B of the Rules**;
   - Ships carrying dangerous chemicals in bulk are those ships defined in **2.1.43**, **Part A of the Rules**;
   - Bulk carriers are those ships defined in **1.3.1(13)**, **Part B of the Rules**; and
   - General dry cargo ships are those ships defined in **1.3.1(15)**, **Part B of the Rules**.

2 Notwithstanding **-1** above, **1.1.2**, **1.2.1** and **1.3.1** (excluding **1.3.1-4**) below are to apply to those shaft alignment calculations required by **6.2.10** and **6.2.11, Part D of the Rules** in cases where main propulsion shafting is comprised of oil-lubricated propeller shafts with diameters less than 400 *mm*.

3 Alternative methods of calculation different from those described in this annex may be employed subject to the prior approval of the Society.

1.1.2 Calculation Sheets for Shaft Alignment

Calculation sheets for shaft alignment that include the following data are to be submitted for approval:

(1) Diameters (outer and inner) and lengths of shafts
(2) Length of bearings
(3) Concentrated loads and loading points
(4) Support points
(5) Bearing offsets from reference lines
(6) Reaction influence numbers
(7) Bending moments and bending stresses
(8) Bearing loads and nominal bearing pressures
(9) Relative inclination of propeller shafts and aftmost stern tube bearings or the maximum bearing pressure in aftmost stern tube bearings
(10) Deflection curves for any shafting

(11) Sags and gaps between shaft coupling flanges
(12) Procedures for measuring bearing loads (in cases where such measurements are required)

1.2 Models of Shafting

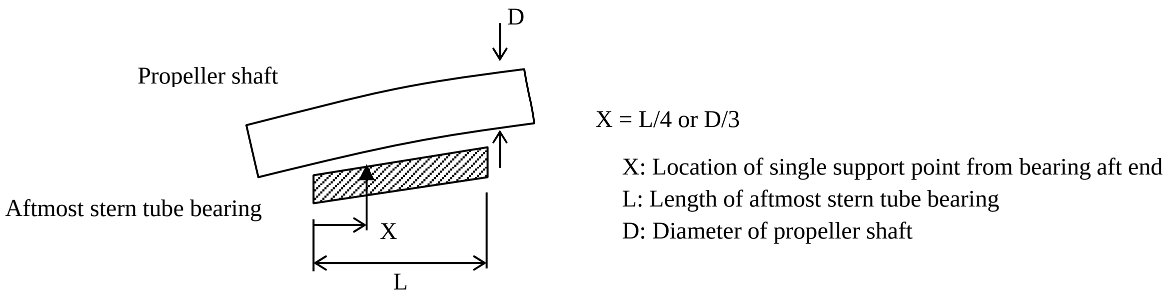
1.2.1 Loads

1 Static loads are to be used in shaft alignment calculations.
2 Any buoyancy forces working on shafting are to be considered as loads. Tensile forces due to cam shaft drive chains specified by engine manufacturers are also to be considered as loads for engines.

1.2.2 Bearings

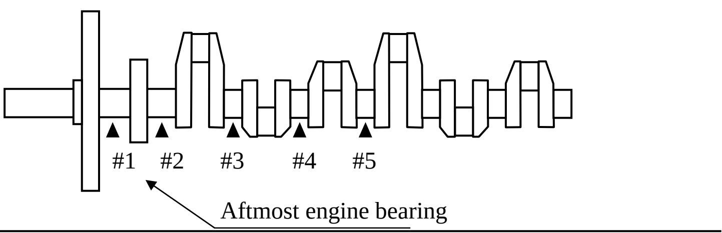
1 In cases where only one support point is assumed in aftmost stern tube bearings, its location is to be at L/4 or D/3 from the aft end of such bearings. In cases where two support points are assumed, their locations are to be at each end of those aftmost stern tube bearings. In cases where three or more support points are assumed, their locations may be decided by the designer. The location of support points in each bearing, other than those aftmost stern tube bearings, is to be at the centre of such bearings.

Fig. 1.2.2-1 Location of Single Support Point in Aftmost Stern Tube Bearings



2 Either rigid supports or elastic supports may be acceptable as the type of supports used.
3 In cases where thrust shafts are integrated with crankshafts, not less than five main bearings of such engines are to be considered in shaft alignment calculations.

Fig. 1.2.2-3 Number of Main Engine Bearings to be Considered



1.2.3 Equivalent Diameter of Crankshafts

When evaluating the shafting of two-stroke cycle engines used as main propulsion machinery, the equivalent diameters of crankshafts specified by engine manufacturers are to be used in shaft alignment calculations in order to give due consideration to any lesser bending stiffness that exists in actual crankshafts compared with simply using those diameters of crank journals in models.

1.2.4 Shafting with Reduction Gears

In the case of shafting with reduction gears such as those found in main steam turbines or geared reciprocating internal combustion engines, shafting from propellers to wheel gears is to be considered in shaft alignment calculations.

1.3 Load Condition and Evaluation of Calculation Results

1.3.1 Light Draught Condition (Cold Condition)

1 Shaft alignment calculations are to be performed under the assumption that ships are in light draught conditions and main propulsion machinery are in cold conditions. In cases where shafts are coupled before launching, shaft alignment calculations are to be performed for such coupled conditions instead of for light draught conditions without taking any buoyancy forces on propellers into account.

2 In cases where aftmost stern tube bearings consist of oil-lubricated white metal, evaluations are to be made of nominal bearing pressure together with either the relative inclination between propeller shafts and aftmost stern tube bearings or the maximum bearing pressures in such aftmost stern tube bearings, either of which is to be determined in order to prevent any edge loading on bearings. Calculated values are to be within those allowable limits shown in **Table 1.3.1-2**.

Table 1.3.1-2 Allowable Limits for Aftmost Stern Tube Bearings (Oil-Lubricated White Metal)

	Allowable Limit	Notes
Nominal bearing pressure	0.8 *MPa*	
Relative inclination between the propeller shaft and the aftmost stern tube bearing	3 x 10^{-4} *rad*	Applicable in cases where the number of support points is one or two. In the case of two support points, the relative inclination is to be calculated at each end of bearings. (see **Fig. 1.3.1-2(a)**)
Maximum bearing pressure	40 *MPa*	Applicable in cases where the maximum bearing pressure is calculated. (see **Fig. 1.3.1-2(b)**)

Fig. 1.3.1-2(a) Relative Inclination

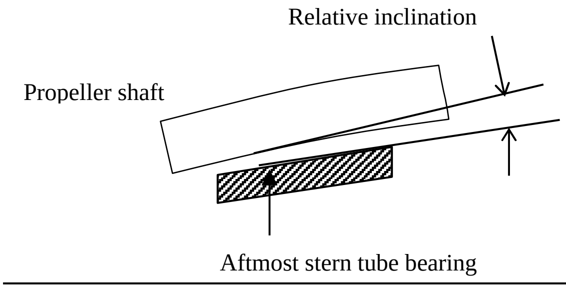
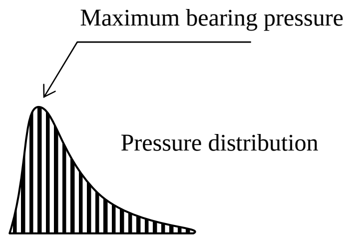


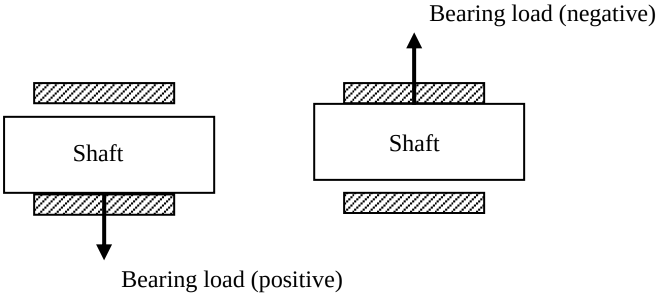
Fig. 1.3.1-2(b) Maximum Bearing Pressure



3 Bending moments (absolute values) calculated for any bearing are not to be more than the value determined for the aftmost stern tube bearings.

4 In principle, bearing loads calculated at each bearing are to be positive values. However, in the case of aftmost bearings of two-stroke cycle engines used as main propulsion machinery, bearing loads of zero may be accepted as zero (negative values are not acceptable.) subject to the agreement of the engine manufacturer. Directions of bearing loads are shown in **Fig. 1.3.1-4**.

Fig. 1.3.1-4 Direction of Bearing Loads



1.3.2 Light Draught Condition (Hot Condition)

1 Shaft alignment calculations are to be performed under the assumption that ships are in light draught conditions and reciprocating internal combustion engines used as main propulsion machinery is in hot conditions. In such cases, any increases in offset specified by manufacturers for engine bearings and those bearings in reduction gears are to be considered under hot conditions.

2 In the calculations **-1** above, full immersion condition of propellers may also be taken into account in such calculations.

3 In cases where shafts are coupled before launching, the calculations in **-1** above are to be performed under the assumption that there is no change in bearing offsets from reference lines between those conditions before and after launching.

4 Bearing loads calculated at each bearing are to be positive values.

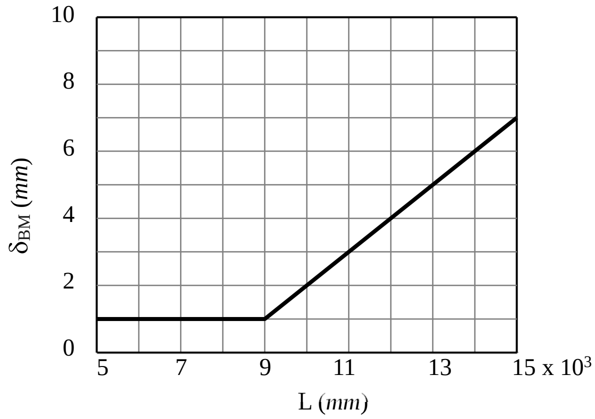
5 In the case of shafting with reduction gears, any differences in bearing loads between the fore and aft bearings of wheel gears in hot conditions are to be within those allowable limits specified by manufacturers.

6 Bending moments due to propeller eccentric thrusts may be taken into account in such calculations.

1.3.3 Full Draught Condition (Hot Condition)

1 Shaft alignment for oil tankers, ships carrying dangerous chemicals in bulk, bulk carriers, and general dry cargo ships is to be designed so as to satisfy the following criteria in order that all engine bearings are fairly evenly loaded, even under any hull deflection that occurs in cases where ships are in full draught conditions. The extent of any relative displacement due to differences between hull deflection that occurs in light draught conditions and hull deflection that occurs in full draught conditions which result in second or third aftmost engine bearings becoming unloaded, as measured at aftmost bulkheads of engine rooms (calculated as δ_{B2} and δ_{B3}, respectively), is to be greater than those allowable lower limits δ_{BM} shown in **Fig. 1.3.3-1 (a)**.

Fig. 1.3.3-1(a) Allowable Lower Limit δ_{BM} for δ_{B2} and δ_{B3}.



Distance from support points of aftmost engine bearings to aftmost bulkheads of engine rooms (see Fig. 1.3.3-1(b))

The relative displacements δ_{B2} and δ_{B3} given above are to be calculated using the formulae in **(1)** or **(2)** below, which are used to calculate the reaction influence numbers in alignment calculations, depending on the type of bearing supports adopted (elastic or rigid supports).

(1) In the case of elastic supports, δ_{B2} and δ_{B3} can be obtained with i = 2 or 3, respectively as follows:

$$\delta_{Bi} = -R_i/S_i$$

where

i : Engine bearing numbers as counted from the aft of engines

R_i : Reaction forces at the i-th number engine bearing as determined by those calculations in **1.3.2** (*kN*)

S_i : Influence numbers for the i-th number engine bearing in cases where hull deflection at aftmost bulkheads of engine rooms becomes -1 *mm*; obtained from the following equation (*kN/mm*):

$$S_i = \sum_{n=1}^{a-1} C_{b+i-1,n}(1.5x_n - 0.5) + \sum_{n=a}^{b-1} C_{b+i-1,n}{x_n}^{1.5}$$

where

$$x_n = X_n/L$$

n : Support point numbers of shafting (counted from the aft of such shafting)

a : Number of nearest support points forward of aftmost bulkheads of engine rooms (counted from the aft of such shafting)

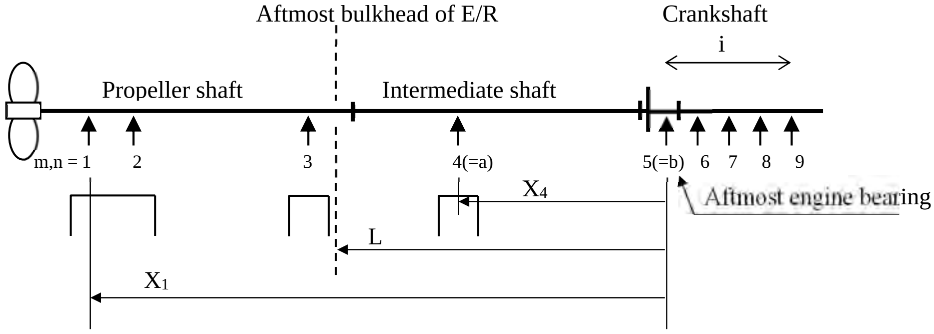
b : Support point numbers of aftmost engine bearings (counted from the aft of such shafting)

X_n : Distance from support point b to support point n (*mm*)

L : Distance from the support point b to aftmost bulkheads of engine rooms (*mm*)

$C_{m,n}$: Influence number at support point m in cases where the relative displacement at support point n becomes -1 *mm* (*kN/mm*) (see **Fig. 1.3.3-1(b)**)

Fig. 1.3.3-1(b) Engine Bearing Numbers and Support Point Numbers



(2) In the case of rigid supports, δ_{B2} or δ_{B3} can be obtained by solving the following simultaneous equations **(1)** and **(2)**, respectively as follows:

$$\left.\begin{array}{l} S_1\delta_{B2} + (C_{1,1} - K)\delta_1 + C_{1,3}\delta_3 + C_{1,4}\delta_4 + C_{1,5}\delta_5 = C_{1,2}R_2/K \\ S_2\delta_{B2} + C_{2,1}\delta_1 + C_{2,3}\delta_3 + C_{2,4}\delta_4 + C_{2,5}\delta_5 = (C_{2,2} - K)R_2/K \\ S_3\delta_{B2} + C_{3,1}\delta_1 + (C_{3,3} - K)\delta_3 + C_{3,4}\delta_4 + C_{3,5}\delta_5 = C_{3,2}R_2/K \\ S_4\delta_{B2} + C_{4,1}\delta_1 + C_{4,3}\delta_3 + (C_{4,4} - K)\delta_4 + C_{4,5}\delta_5 = C_{4,2}R_2/K \\ S_5\delta_{B2} + C_{5,1}\delta_1 + C_{5,3}\delta_3 + C_{5,4}\delta_4 + (C_{5,5} - K)\delta_5 = C_{5,2}R_2/K \end{array}\right\} \quad (1)$$

$$\left.\begin{array}{l} S_1\delta_{B3} + (C_{1,1} - K)\delta_1 + C_{1,2}\delta_2 + C_{1,4}\delta_4 + C_{1,5}\delta_5 = C_{1,3}R_3/K \\ S_2\delta_{B3} + C_{2,1}\delta_1 + (C_{2,2} - K)\delta_2 + C_{2,4}\delta_4 + C_{2,5}\delta_5 = C_{2,3}R_3/K \\ S_3\delta_{B3} + C_{3,1}\delta_1 + C_{3,2}\delta_2 + C_{3,4}\delta_4 + C_{3,5}\delta_5 = (C_{3,3} - K)R_3/K \\ S_4\delta_{B3} + C_{4,1}\delta_1 + C_{4,2}\delta_2 + (C_{4,4} - K)\delta_4 + C_{4,5}\delta_5 = C_{4,3}R_3/K \\ S_5\delta_{B3} + C_{5,1}\delta_1 + C_{5,2}\delta_2 + C_{5,4}\delta_4 + (C_{5,5} - K)\delta_5 = C_{5,3}R_3/K \end{array}\right\} \quad (2)$$

where

K : Stiffness of bearing supports, given as $K = 5000$ (*kN/mm*)

S_i : Influence number for i-th number engine bearing (see **(1)** above)

$C_{i,j}$: Influence number for the i-th number engine bearing in cases where the relative displacement at the j-th number engine bearing becomes -1 *mm* (*kN/mm*) (However, the i-th and j-th numbers are counted from the aft of engines.)

$\delta_i (i = 1,2,3,4,5)$: Elastic relative displacement at each engine bearing resulting from the relative displacement δ_{B2} or δ_{B3}. (δ_i is unknown.)

2 Notwithstanding **-1** above, the Society may examine and accept alternative criteria, provided that documentation is submitted that makes it possible to evaluate the condition of engine bearings in cases where ships are in full draught conditions.

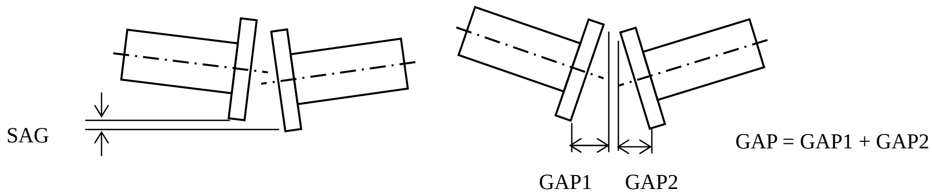
3 Other documents such as those showing results structural analysis evaluating the extent of hull deflection may be required by the Society in cases where stern hull construction is considered to be unconventional.

1.4 Matters Relating to Shaft Alignment Procedures

1.4.1 Sags and Gaps between Shaft Coupling Flanges

Sags and gaps between shaft coupling flanges in an uncoupled condition are to be calculated under the condition that bearing offsets from reference lines are those used in those calculation described in **1.3.1** above.

Fig. 1.4.1 Sag and Gap between Shaft Coupling Flanges



1.4.2 Procedure for Measuring Bearing Loads

In cases where bearing loads are measured using the jack-up technique, documentation describing the measurement procedures followed (including jack-up positions, load correction factors and expected jack-up loads) is to be prepared. The immersion of propellers at the time of such measurements is also to be considered in the bearing loads measured.

EFFECTIVE DATE AND APPLICATION (Amendment 1-2)

1. The effective date of the amendments is 1 July 2022.
2. Notwithstanding the amendments to the Rules, the current requirements apply to ships for which the date of contract for construction* is before the effective date.
   * "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.
2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.
   For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:
   (1) such alterations do not affect matters related to classification, or
   (2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.
   The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.
3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.
4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

Amendment 1-3

Chapter 2 RECIPROCATING INTERNAL COMBUSTION ENGINES

2.1 General

2.1.2 Terminology*

Sub-paragraph -2 has been amended as follows.

2 The terminology used in the application of ~~**-1(3)** and **-2** of~~ **2.1.3** ~~as well as~~and **2.1.4** is as specified in the following **(1)** to **(36)**:

((1) to (25) are omitted)

(26) "Quality assurance" means all the planned and systematic activities implemented within the quality system, and demonstrated as needed to provide adequate confidence that an entity will fulfil requirements for quality. Refer to *ISO* ~~9000 series~~9001:2015.

((27) to (36) are omitted)

EFFECTIVE DATE AND APPLICATION (Amendment 1-3)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements may apply to reciprocating internal combustion engines whose type is the same type of those for which the application for approval of use is submitted to the Society before the effective date.

Amendment 1-4

Chapter 2 RECIPROCATING INTERNAL COMBUSTION ENGINES

2.1 General

Paragraph 2.1.3 has been amended as follows.

2.1.3 Drawings and Data*

1 Drawings and data to be submitted are generally as follows:

(1) Drawings and data for approval

<u>Drawings and data specified in **Table D2.1(a)**</u>

~~(a) Connecting rod bearings (including bolts details) of 4-stroke cycle engines~~

~~(b) High pressure oil pipes for driving exhaust valves with its shielding~~

~~(c) High pressure fuel oil pipes with its shielding and clamping~~

~~(d) Piping arrangements fitted to engines (including fuel oil, lubricating oil, cooling oil, cooling water, pneumatic and hydraulic systems, and information regarding the size, materials and working pressure of pipes)~~

~~(e) The drawings and data as specified in **(3)(d)** to **(f)**~~

~~(f) The drawings and data, etc. as required by the requirements of **2.1.4** (excluding those specified in **2.1.3-1(3)**)~~

~~(g) The following drawings and data for exhaust driven turbochargers:~~

~~i) Category *A* turbochargers (upon request)~~

~~1) Sectional assembly (including principal dimensions and names of components)~~

~~2) Containment test report~~

~~3) Test procedures~~

~~ii) Category *B* turbochargers~~

~~1) Sectional assembly (including principal dimensions and materials of housing components for containment evaluation.)~~

~~2) Documentation of containment in the event of the disc fracture specified in **2.5.1-6**~~

~~3) Documentation of following operational data and limitations~~

~~・Maximum permissible operating speed (*rpm*)~~

~~・Maximum permissible exhaust gas temperature at the turbine inlet~~

~~・Minimum lubrication oil inlet pressure~~

~~・Maximum lubrication oil outlet temperature~~

~~・Maximum permissible vibration levels (self- and externally generated vibrations)~~

~~・Alarm level for overspeed (levels are also to be indicated on engine control system diagrams)~~

~~・Alarm level for exhaust gas temperature at the turbine inlet (levels are also to be indicated on engine control system diagrams)~~

~~・Lubrication oil inlet pressure low alarm set point (levels are also to be indicated on engine control system diagrams)~~

~~・Lubrication oil outlet temperature high alarm set point (levels are also to be indicated on engine control system diagrams)~~

~~4) Diagram of lubrication oil systems (diagrams included in piping arrangements fitted to engines may be accepted instead)~~

~~5) Test report of type test (only for type tests)~~
~~6) Test procedure (only for type tests)~~

~~iii) Category *C* turbochargers~~
~~1) Drawings listed in **ii)** above~~
~~2) Drawings of the housing and rotating parts (including details of blade fixing)~~
~~3) Material specifications (including mechanical properties and chemical composition) of the parts mentioned in **2)** above~~
~~4) Welding details and welding procedures for the parts mentioned in **2)** above, if made of welded construction~~

(2) Drawings and data for reference

<u>Drawings and data specified in **Table D2.1(b)**</u>

~~(a) A list containing all drawings and data submitted (with relevant drawing numbers and revision status)~~
~~(b) Gudgeon pins~~
~~(c) Connecting rod bearings (including bolts details) of 2-stroke cycle engines~~
~~(d) Rocker valve gears~~
~~(e) Cylinder cover fixing bolts and valve box fixing bolts~~
~~(f) Engine control system diagram (including the monitoring, safety and alarm systems)~~
~~(g) Construction and arrangement of dampers, detuners, balancers or compensators, bracings as well as all calculation sheets related to engine balancing and engine vibration prevention~~
~~(h) Location of measures preventing oil from spraying out from joints in flammable oil piping (if fitted)~~
~~(i) The following drawings and data for exhaust driven turbochargers (only for category *C* turbochargers):~~
~~i) Documentation of the safe torque transmission specified in **2.5.1-6** when the disc is connected to the shaft by an interference fit~~
~~ii) Information on expected lifespan (Creep, low cycle fatigue and high cycle fatigue are to be considered.)~~
~~iii) Operation and maintenance manuals~~
~~(j) Other drawings and data deemed necessary by the Society~~

~~(3) Drawings and data for the purpose of inspection and testing of reciprocating internal combustion engines~~
~~(a) A list containing all drawings and data submitted (including relevant drawing numbers and revision status)~~
~~(b) Engine particulars to be in the form designated by the Society~~
~~(c) Material specifications of main parts with information on non-destructive testing and pressure testing as applicable to the material~~
~~(d) Bedplate and crankcase of welded design, with welding details and welding instructions for approval of materials and weld procedure specifications. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.~~
~~(e) Thrust bearing bedplate of welded design, with welding details and welding instructions for approval of materials and weld procedure specifications. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.~~
~~(f) Frame/framebox/gearbox of welded design, with welding details and instructions for approval of materials and weld procedure specifications. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables~~

~~and fit-up conditions.~~

~~(g) Crankshaft, assembly and details~~

~~(h) Thrust shaft or intermediate shaft (if integral with engine)~~

~~(i) Shaft coupling bolts~~

~~(j) Bolts and studs for main bearings~~

~~(k) Bolts and studs for cylinder heads and exhaust valve (two stroke design)~~

~~(l) Bolts and studs for connecting rods~~

~~(m) Tie rods~~

~~(n) Schematic layout or other equivalent drawings and data on the reciprocating internal combustion engine of the following **i)** to **vii)** (Details of the system so far as supplied by the licensee such as: main dimensions, operating media and maximum working pressures).~~

~~i) Starting air system~~

~~ii) Fuel oil system~~

~~iii) Lubricating oil system~~

~~iv) Cooling water system~~

~~v) Hydraulic system~~

~~vi) Hydraulic system (for valve lift)~~

~~vii) Engine control and safety system~~

~~(o) Shielding of high pressure fuel pipes, assembly~~

~~(All engines)~~

~~(p) Construction of accumulators for hydraulic oil and fuel oil~~

~~(q) High pressure parts for fuel oil injection system~~

~~The documentation to contain specifications for pressures, pipe dimensions and materials.~~

~~(r) Arrangement and details of the crankcase explosion relief valve (only for engines of a cylinder diameter of 200 *mm* or more or a crankcase volume of 0.6 m^3 or more)~~

~~(s) Oil mist detection and/or alternative alarm arrangements~~

~~(t) Cylinder head~~

~~(u) Cylinder block, engine block~~

~~(v) Cylinder liner~~

~~(w) Counterweights (if not integral with crankshaft), including fastening~~

~~(x) Connecting rod with cap~~

~~(y) Crosshead~~

~~(z) Piston rod~~

~~(aa) Piston, assembly, including identification (e.g. drawing number) of components~~

~~(ab) Piston head~~

~~(ac) Camshaft drive, assembly, including identification (e.g. drawing number) of components~~

~~(ad) Flywheel~~

~~(ae) Arrangement of foundation (for main engines only)~~

~~(af) Fuel oil injection pump~~

~~(ag) Shielding and insulation of exhaust pipes and other parts of high temperature which may be impinged as a result of a fuel system failure, assembly~~

~~(ah) Construction and arrangement of dampers~~

~~(ai) For electronically controlled engines, assembly drawings or arrangements of the following **i)** to **iv)**:~~

~~i) Control valves~~

~~ii) High-pressure pumps~~

~~iii) Drive for high pressure pumps~~

~~iv) Valve bodies, if applicable~~

~~(aj) Operation and service manuals~~

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

~~(ak) Test program resulting from FMEA (for engine control system) in cases of engines that rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves~~

~~(al) Production specifications for castings and welding (sequence)~~

~~(am) Certification of an approval of use for environmental tests, control components~~

~~Drawings and data modified for a specific application are to be submitted to the Society for information or approval, as applicable.~~

~~(an) Quality requirements for engine production~~

~~(ao) Other drawings and data deemed necessary by the Society~~

2 The drawings and data for the inspection and testing specified in **-1~~(3)~~** (the items represented by the mark ○ in **Table D2.1(a)** and **Table D2.1(b)**, hereinafter indicated the same way throughout this Chapter)~~above~~ are to be submitted in accordance with **2.1.4-1** by the engine manufacturer producing engines with the drawings and data whose approval of use has been obtained in accordance with **2.1.1-3** (hereinafter referred to as “licensee” in this Chapter). Such drawings and data, however,~~but~~ may be submitted by the licensor in accordance with **2.1.4-2**.

Table D2.1(a) has been added as follows.

Table D2.1(a) Drawings and Data for Approval

	Items	For inspection and testing
(1)	Engine particulars (in the format designated by the Society)	○
(2)	Material specifications of main parts with information on non-destructive testing and pressure testing as applicable to the material	○
(3)	Bedplate and crankcase of welded design, with welding details and welding instructions(1)	○
(4)	Thrust bearing bedplate of welded design, with welding details and welding instructions(1)	○
(5)	Frame/framebox/gearbox of welded design, with welding details and instructions(1)	○
(6)	Crankshaft, assembly and details	○
(7)	Thrust shaft or intermediate shaft (if integral with engine)	○
(8)	Shaft coupling bolts	○
(9)	Connecting rod bearings (four-stroke design)	－
(10)	Bolts and studs for connecting rods (four-stroke design)	○
(11)	Schematic layout or other equivalent drawings and data on the reciprocating internal combustion engine of the following **(a)** to **(g)** (details of the system so far as supplied by the licensee such as: main dimensions, operating media and maximum working pressures). (a) Starting air system (b) Fuel oil system (c) Lubricating oil system (d) Cooling water system (e) Hydraulic system (f) Hydraulic system (for valve lift) (g) Engine control and safety system	○
(12)	High pressure oil pipes for driving exhaust valves with its shielding	－
(13)	Shielding of high pressure fuel pipes, assembly (all engines)	○
(14)	High pressure parts for fuel oil injection system The documentation to contain specifications for pressures, pipe dimensions and materials.	○
(15)	Arrangement and details of the crankcase explosion relief valve (only for engines of a cylinder diameter of 200 *mm* or more or a crankcase volume of 0.6 m^3 or more)	○
(16)	Oil mist detection and/or alternative alarm arrangements	○
(17)	Connecting rod with cap (four-stroke design)	○
(18)	Arrangement of foundation (for main engines only)	○
(19)	The drawings, data, etc. required by **2.1.4**.	○

Table D2.1(a) Drawings and Data for Approval (continued)

	Items	For inspection and testing
(20)	The following drawings and data for exhaust driven turbochargers: (a) Category *A* turbochargers (upon request) i) Sectional assembly (including principal dimensions and names of components) ii) Containment test report iii) Test procedures (b) Category *B* turbochargers i) Sectional assembly (including principal dimensions and materials of housing components for containment evaluation.) ii) Documentation of containment in the event of the disc fracture specified in **2.5.1-6** iii) Documentation of following operational data and limitations ・Maximum permissible operating speed (*rpm*) ・Maximum permissible exhaust gas temperature at the turbine inlet ・Minimum lubrication oil inlet pressure ・Maximum permissible vibration levels (self- and externally generated vibrations) ・Alarm level for exhaust gas temperature at the turbine inlet (levels are also to be indicated on engine control system diagrams) ・Lubrication oil inlet pressure low alarm set point (levels are also to be indicated on engine control system diagrams) ・Lubrication oil outlet temperature high alarm set point (levels are also to be indicated on engine control system diagrams) iv) Diagram of lubrication oil systems (diagrams included in piping arrangements fitted to engines may be accepted instead) v) Test report of type test (only for type tests) vi) Test procedure (only for type tests) (c) Category *C* turbochargers i) Drawings listed in **(b)** above ii) Drawings of the housing and rotating parts (including details of blade fixing) iii) Material specifications (including mechanical properties and chemical composition) of the parts mentioned in **ii)** above iv) Welding details and welding procedures for the parts mentioned in **ii)** above, if made of welded construction	－
(21)	Other drawings and data deemed necessary by the Society	○

Notes:

(1) For approval of materials and weld procedure specifications, the weld procedure specification is to include details of pre- and post-weld heat treatments, weld consumables and fit-up conditions.

Table D2.1(b) has been added as follows.

Table D2.1(b) Drawings and Data for Reference

	Items	For inspection and testing
(1)	A list containing all drawings and data submitted (including relevant drawing numbers and revision status)	○
(2)	Bolts and studs for main bearings	○
(3)	Connecting rod bearings (two-stroke design)	−
(4)	Bolts and studs for cylinder heads and exhaust valve (two-stroke design)	○
(5)	Bolts and studs for connecting rods (two-stroke design)	○
(6)	Tie rods	○
(7)	Piston pins	−
(8)	Construction of accumulators for hydraulic oil and fuel oil	○
(9)	Cylinder head fixing bolts and valve box fixing bolts	−
(10)	Rocker valve gears	−
(11)	Cylinder head	○
(12)	Cylinder block, engine block	○
(13)	Cylinder liner	○
(14)	Counterweights (if not integral with crankshaft), including fastening	○
(15)	Connecting rod with cap (two-stroke design)	○
(16)	Crosshead	○
(17)	Piston rod	○
(18)	Piston, assembly, including identification (e.g. drawing number) of components	○
(19)	Piston head	○
(20)	Camshaft drive, assembly, including identification (e.g. drawing number) of components	○
(21)	Flywheel	○
(22)	Fuel oil injection pump	○
(23)	Shielding and insulation of exhaust pipes and other parts of high temperature which may be impinged as a result of a fuel system failure, assembly	○
(24)	Construction and arrangement of dampers	○
(25)	Construction and arrangement of detuners, balancers or compensators, bracings as well as all calculation sheets related to engine balancing and engine vibration prevention	−
(26)	For electronically controlled engines, assembly drawings or arrangements of the following **(a)** to **(d)**: (a) Control valves (b) High-pressure pumps (c) Drive for high pressure pumps (d) Valve bodies, if applicable	○
(27)	Operation and service manuals[(1)]	○
(28)	Engine control system diagram (including the monitoring, safety and alarm systems)	−
(29)	Test program resulting from FMEA (for engine control system) in cases of engines that rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves	○
(30)	Production specifications for castings and welding (sequence)	○
(31)	Certification of an approval of use for environmental tests, control components[(2)]	○
(32)	Quality requirements for engine production	○
(33)	Location of measures preventing oil from spraying out from joints in flammable oil piping (if fitted)	−

Table D2.1(b) Drawings and Data for Reference (continued)

	Items	For inspection and testing
(34)	The following drawings and data for exhaust driven turbochargers (only for category *C* turbochargers): (a) Documentation of the safe torque transmission specified in **2.5.1-6** when the disc is connected to the shaft by an interference fit (b) Information on expected lifespan (creep, low cycle fatigue and high cycle fatigue are to be considered) (c) Operation and maintenance manuals	－
(35)	Other drawings and data deemed necessary by the Society	○

Notes:

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

(2) Drawings and data modified for a specific application are to be submitted to the Society for reference or approval, as applicable.

2.1.4 Approval of Reciprocating Internal Combustion Engines*

Sub-paragraphs -1(1) to (3) have been amended as follows.

1 Reciprocating internal combustion engines are to be approved in accordance with the following **(1)** to **(6)**:

(1) Development of documents and data for engine production

(a) Prior to the start of the reciprocating internal combustion engine approval process in accordance with the following **(c)**~~**(3)**~~ and subsequent sub-paragraphs of this paragraph, a design approval is to be obtained as specified separately by the Society.

(b) Each type of reciprocating internal combustion engine is to be provided with a certificate of approval of use obtained by the licensor in accordance with **2.1.1-3**. For the first engine of a type or for those with no service records, the process of an approval of use and the approval process for production by the licensee may be performed simultaneously.

(c) The licensor is to review the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained for the application and develop, if necessary, application specific drawings and data for production of reciprocating internal combustion engines for the use of the licensee in developing the reciprocating internal combustion engine specific production drawings and data for the inspection and testing specified~~listed~~ in **2.1.3-1**~~**(3)**~~.

(d) If substantive modifications to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained have been made in the drawings and data of reciprocating internal combustion engines to be produced, the affected drawings and data are to be resubmitted to the Society as specified separately by the Society.

(2) Drawings and data for the ~~purpose of~~ inspection and testing of reciprocating internal combustion engines

(a) The licensee is to develop the drawings and data for the inspection and testing specified~~listed~~ in **2.1.3-1**~~**(3)**~~ and a comparison list of these drawings and data to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained by the licensor and submit these drawings and the comparison list to the Society.

(b) ~~In applying~~As for the drawings and data for the inspection and testing specified in

2.1.3-1~~(3)~~, if there are differences in the technical content on the licensee's production drawings and data of the reciprocating internal combustion engine compared to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained by the licensor, the licensee is to submit "Confirmation of the licensor's acceptance of licensee's modifications" approved by the licensor and signed by the licensee and licensor. If the licensor acceptance is not confirmed, the reciprocating internal combustion engine manufactured by the licensee is to be regarded as a different engine type and is **2.1.1-3** is to apply to the reciprocating internal combustion engine.

(c) In applying **(b)** above, modifications applied by the licensee are to be provided with appropriate quality requirements.

(d) The Society returns the drawings and data specified in **(a)** and **(b)** above to the licensee with confirmation that the design has been approved.

(e) The licensee or its subcontractors are to prepare to be able to provide the drawings and data specified in **(a)** and **(b)** above so that the Surveyor can use the information for inspection purposes during manufacture and testing of the reciprocating internal combustion engine and its components.

(3) Additional drawings and data

In addition to the drawings and data for the inspection and testing specified~~listed~~ in **2.1.3-1~~(3)~~**, the licensee is to be able to provide to the Surveyor performing the test specified in **2.6.1** upon request the relevant detail drawings, production quality control specifications and acceptance criteria. These drawings and data are for supplemental purposes to the survey only.

((4) to (6) are omitted.)

2.2 Materials, Construction and Strength

2.2.1 Materials

Sub-paragraph -1 has been amended as follows.

1 Materials intended for the principal components of reciprocating internal combustion engines and their non-destructive tests as well as surface inspections and dimension inspections are to conform to the requirements given in **Table D2.~~1~~2**. However, with respect to ultrasonic testing as well as surface inspections and dimension inspections, submission or presentation of test results to the Surveyor may be considered sufficient. In cases where deemed necessary by the Society, tests or inspections may also be required for any parts not specified in **Table D2.~~1~~2**.

Table D2.1 has been renumbered to Table D2.2.

Table D2.~~1~~2 Application of Materials and Non-destructive Tests as well as Surface Inspections and Dimension Inspections to Principal Components of Reciprocating Internal Combustion Engines
(Omitted)

2.3 Crankshafts

2.3.1 Solid Crankshafts*

Sub-paragraph -1 has been amended as follows.

1 The diameters of crankpins and journals are to be not less than the value given by the following formula:

$$d_c = \left\{\left(M + \sqrt{M^2 + T^2}\right)D^2\right\}^{\frac{1}{3}} K_m K_s K_h$$

where

d_c : Required diameter of crankshaft (*mm*)

M : $10^{-2}ALP_{max}$

T : $10^{-2}BSP_{mi}$

S : Length of stroke (*mm*)

L : Span of bearings adjacent to crank measured from centre to centre (*mm*)

P_{max} : Maximum combustion pressure in cylinder (*MPa*)

P_{mi} : Indicated mean effective pressure (*MPa*)

A and B: Coefficients given in **Table D2.~~2~~3** and **Table D2.~~3~~4** for engines having equal firing intervals (in the case of Vee engines, those with equal firing intervals on each bank.). Special consideration will be given to values *A* and *B* for reciprocating internal combustion engines having unequal firing intervals or for those not covered by the Tables.

D : (Omitted)

K_m : (Omitted)

K_s : (Omitted)

K_h : (Omitted)

Table D2.2 and Table D2.3 have been renumbered to Table D2.3 and Table D2.4.

Table D2.~~2~~3 Value of Coefficients *A* and *B* for Single Acting In-line Engines
(Omitted)

Table D2.~~3~~4(a) Value of Coefficients *A* and *B* for Single Acting 2-stroke cycle Vee Engines with Parallel Connecting Rods
(Omitted)

Table D2.~~3~~4(b) Value of Coefficients *A* and *B* for Single Acting 4-stroke cycle Vee Engines with Parallel Connecting Rods
(Omitted)

2.4 Safety Devices

2.4.3 Protection against Crankcase Explosion*

Sub-paragraph -2 has been amended as follows.

2 The number and locations of the explosion relief valves specified in **-1** are to be in accordance with **Table D2.~~4~~5**.

Table D2.4 has been renumbered to Table D2.5.

Table D2.~~4~~5 Number and Location of Explosion Relief Valves
(Omitted)

2.5 Associated Installations

2.5.1 Exhaust Driven Turbochargers*

Sub-paragraph -8 has been amended as follows.

8 For categories *B* and *C* turbochargers, the indications and alarms listed in the **Table D2.6~~5~~** are to be provided. Indications may be provided at local locations, monitoring stations or control stations. Alarm levels may be equal to permissible limits~~,~~ but are not to be reached when operating the engine at 110 % power, or at any approved intermittent overload beyond 110 % in cases where the turbochargers are fitted to engines for which intermittent overload power is approved.

Table D2.5 has been renumbered to Table D2.6.

Table D2.~~5~~6 Alarms and Indications of Turbochargers
(Omitted)

2.5.8 Accumulators and Common Accumulators for Electronically-controlled Engines which are used as the Main Propulsion Machinery

Sub-paragraph -1 has been amended as follows.

1 Accumulators and common accumulators are to comply with the requirements in **Chapter 10**. However, notwithstanding this requirement, material~~s~~ and non-destructive tests as well as surface inspections and dimension inspections are to be in accordance with **Table D2.~~1~~2** and hydrostatic tests are to be in accordance with **Table D2.~~6~~7**.

2.5.11 Failure Mode Effect Analysis for Electronically-controlled Engines which are used as the Main Propulsion Machinery

Sub-paragraph (3) has been amended as follows.

Failure Mode Effect Analysis (FMEA) is to be carried out, for electronic control systems, in order to confirm that any one equipment or circuits in such systems which lose function may not cause any malfunction or deterioration in other equipment or circuits, in accordance with the following:

((1) and (2) are omitted.)

(3) FMEA results are to be created in table form as shown in **Table D2.~~6~~7** or be of equivalent forms thereto.

((4) and (5) are omitted.)

Table D2.6 has been renumbered to Table D2.7.

Table D2.~~6~~7 Failure Mode Effect Analysis Table for Electronically-controlled Engines which are used as the Main Propulsion Machinery

(Omitted)

2.6 Tests

2.6.1 Shop Tests*

Sub-paragraph -1 has been amended as follows.

1 For components or accessories specified in **Table D2.~~7~~8**, hydrostatic tests are to be carried out on the water or oil side of the component at the pressures shown in the Table. In cases deemed necessary by the Society, tests may also be required for any components not specified in **Table D2.~~7~~8**.

Table D2.7 has been renumbered to Table D2.8.

Table D2.~~7~~8 Hydrostatic Test Pressure

(Omitted)

Chapter 11 WELDING FOR MACHINERY INSTALLATIONS

11.2 Welding Procedure and Related Specifications

11.2.1 Approval of Welding Procedure and Related Specifications*

Sub-paragraph -1(1) has been amended as follows.

1 The manufacturer is to obtain the approval of the welding procedures in the following cases:

(1) Where the welding procedures are first adopted for the welding work specified below.

- (a) (Omitted)
- (b) (Omitted)
- (c) Welding work for the principal components of prime movers, etc. (these principal components are specified in **Table D2.~~1~~2**, **3.2.1-1**, **4.1.2(5)** and **5.2.1-1**; hereinafter, this definition applies throughout this Chapter)
- (d) (Omitted)
- (e) (Omitted)
- (f) (Omitted)

((2) and (3) are omitted.)

EFFECTIVE DATE AND APPLICATION (Amendment 1-4)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements may apply to reciprocating internal combustion engines for which the application for approval is submitted to the Society before the effective date.

Amendment 1-5

Chapter 2 RECIPROCATING INTERNAL COMBUSTION ENGINES

Section 2.3 has been amended as follows.

2.3 Crankshafts

2.3.1 Solid Crankshafts <u>and Semi-Built Crankshafts</u>*

~~1 The diameters of crankpins and journals are to be not less than the value given by the following formula:~~

$$\cancel{d_c = \left\{\left(M + \sqrt{M^2 + T^2}\right) D^2\right\}^{\frac{1}{3}} K_m K_s K_h}$$

~~where~~

~~d_c : Required diameter of crankshaft (mm)~~

~~M : $10^{-2} ALP_{max}$~~

~~T : $10^{-2} BSP_{mi}$~~

~~S : Length of stroke (mm)~~

~~L : Span of bearings adjacent to crank measured from centre to centre (mm)~~

~~P_{max} : Maximum combustion pressure in cylinder (MPa)~~

~~P_{mi} : Indicated mean effective pressure (MPa)~~

~~A and B : Coefficients given in **Table D2.2** and **Table D2.3** for engines having equal firing intervals (in the case of Vee engines, those with equal firing intervals on each bank.). Special consideration will be given to values A and B for reciprocating internal combustion engines having unequal firing intervals or for those not covered by the Tables.~~

~~D : Cylinder bore (mm)~~

~~K_m : Value given by the following **(1)** or **(2)** in accordance with the specified tensile strength of the crankshaft material. However, the value of K_m for materials other than steel forgings and steel castings is to be determined by the Society in each case.~~

~~(1) In cases where the specified tensile strength of material exceeds 440 N/mm^2;~~

$$\cancel{K_m = \sqrt[3]{\frac{440}{440 + \frac{2}{3}(T_s - 440)}}}$$

~~where~~

~~T_s : Specified tensile strength of material (N/mm^2)~~

~~The value of T_s is not to exceed 760 N/mm^2 for carbon steel forgings and 1080 N/mm^2 for low alloy steel forgings.~~

~~(2) In cases where the specified tensile strength of material is not more than 440 N/mm^2 but not less than 400 N/mm^2;~~

~~$K_m = 1.0$~~

~~K_s : Value given by the following **(1)**, **(2)**, or **(3)** in accordance with the manufacturing method of crankshafts.~~

~~(1) In cases where the crankshafts are manufactured by a special forging process approved by the Society as well as where the product quality is stable and the fatigue strength is considered to be improved by 20% or more in comparison with that of the free forging process;~~

$$K_s = \sqrt[3]{\frac{1}{1.15}}$$

~~(2) In cases where the crankshafts are manufactured by a manufacturing process using a surface treatment approved by the Society as well as where the product quality is stable and the fatigue strength is recognized as being superior;~~

$$K_s = \sqrt[3]{\frac{1}{1 + \rho/100}}$$

~~where~~

~~ρ : Degree of improvement in strength approved by the Society relative to the surface hardening (%)~~

~~(3) In cases other than **(1)** and **(2)** above;~~

$K_s = 1.0$

~~K_h : Value given by the following **(1)** or **(2)** in accordance with the inside diameter of the crankpins or journals.~~

~~(1) In cases where the inside diameter is one-third or more than that of the outside diameter;~~

$$K_h = \sqrt[3]{\frac{1}{1 - R^4}}$$

~~where~~

~~R : Quotient obtained by dividing the inside diameter of a hollow shaft by its outside diameter~~

~~(2) In cases where the inside diameter is less than one-third of the outside diameter;~~

$K_h = 1.0$

~~Table D2.2 Value of Coefficients A and B for Single Acting In-line Engines~~

Number of cylinders	2-stroke cycle		4-stroke cycle	
	A	B	A	B
1	1.00	8.8	1.25	4.7
2		8.8		4.7
3		10.0		4.7
4		11.1		4.7
5		11.4		5.4
6		11.7		5.4
7		12.0		6.1
8		12.3		6.1
9		12.6		6.8
10		13.4		6.8
11		14.2		7.4
12		15.0		7.4

~~Table D2.3(a) Value of Coefficients *A* and *B* for Single Acting 2-stroke cycle Vee Engines with Parallel Connecting Rods~~

~~Number of cylinders~~	~~Minimum firing interval between two cylinders on one crankpin~~					
	~~45°~~		~~60°~~		~~90°~~	
	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~
~~6~~	~~1.05~~	~~17.0~~	~~1.00~~	~~12.6~~	~~1.00~~	~~17.0~~
~~8~~		~~17.0~~		~~15.7~~		~~20.5~~
~~10~~		~~19.0~~		~~18.7~~		~~20.5~~
~~12~~		~~20.5~~		~~21.6~~		~~20.5~~
~~14~~		~~22.0~~		~~21.6~~		~~20.5~~
~~16~~		~~23.5~~		~~21.6~~		~~23.0~~
~~18~~		~~24.0~~		~~21.6~~		~~23.0~~
~~20~~		~~24.5~~		~~24.2~~		~~23.0~~

~~Table D2.3(b) Value of Coefficients *A* and *B* for Single Acting 4-stroke cycle Vee Engines with Parallel Connecting Rods~~

~~Number of cylinders~~	~~Minimum firing interval between two cylinders on one crankpin~~											
	~~45°~~		~~60°~~		~~90°~~		~~270°~~		~~300°~~		~~315°~~	
	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~	~~*A*~~	~~*B*~~
~~6~~	~~1.60~~	~~4.1~~	~~1.47~~	~~4.0~~	~~1.40~~	~~4.0~~	~~1.40~~	~~4.0~~	~~1.30~~	~~4.4~~	~~1.20~~	~~4.3~~
~~8~~		~~5.5~~		~~5.5~~		~~5.5~~		~~5.5~~		~~5.3~~		~~5.2~~
~~10~~		~~6.7~~		~~7.0~~		~~6.5~~		~~6.5~~		~~6.1~~		~~5.9~~
~~12~~		~~7.5~~		~~8.2~~		~~7.5~~		~~7.5~~		~~6.9~~		~~6.6~~
~~14~~		~~8.4~~		~~9.2~~		~~8.5~~		~~8.5~~		~~7.5~~		~~7.3~~
~~16~~		~~9.3~~		~~10.1~~		~~9.5~~		~~9.5~~		~~8.2~~		~~7.9~~
~~18~~		~~10.1~~		~~11.1~~		~~10.5~~		~~10.5~~		~~8.8~~		~~8.5~~
~~20~~		~~11.5~~		~~14.0~~		~~11.5~~		~~11.5~~		~~9.5~~		~~9.2~~

~~**2** The dimensions of crank webs are to comply with the following requirements:~~

~~(1) The thickness and breadth of crank webs, the diameters of the crankpins and journals, are to comply with the conditions of the following formula. However, the thickness of crank webs is to be not less than 0.36 *times* the diameter of crankpins and journals. When the actual diameters of the crankpin and journal are larger than the required diameter of the crankshaft as determined by the formula in -**1**, the left side of the following formula may be multiplied by $(d_c/d_a)^3$.~~

~~$\{0.122(2.20 - b/d_a)^2 + 0.337\}(d_a/t)^{1.4} \leq 1$~~

~~where~~

~~b : Breadth of crank web (*mm*)~~

~~d_a : Actual diameter of crankpin or journal (*mm*)~~

~~t : Thickness of crank web (*mm*)~~

~~(2) The radius in fillets at the junctions of crank webs with crankpins or journals is to be not less than 0.05 *times* the actual diameter of crankpins or journals, respectively.~~

1 The requirements in this paragraph apply to solid-forged and semi-built crankshafts made of forged or cast steel, with one crank throw between main bearings that are used for reciprocating internal combustion engines for propulsion and auxiliary purposes in cases where such engines are capable of continuous operation at their rated power when running at their rated speed.

2 The torsional stress in crankpins and journals is to be evaluated by carrying out forced vibration calculations including the stern shafting and the values of the acceptability factor *Q* calculated by **Annex 2.3.1 "CALCULATION METHOD OF CRANKSHAFT STRESS"** are to comply with the following formula:

$Q \geq 1.15$

3 In cases where a crankshaft design involves the use of surface treated fillets, where fatigue parameter influences are tested, or where working stresses are measured, relevant documents for such calculations and analyses are to be submitted to the Society in order to demonstrate equivalence to **-2 above**.

4 Approval of crankshafts other than those specified in the requirements of this paragraph is to be as deemed appropriate by the Society.

2.3.2 Built-up Crankshafts*

~~**1** The dimensions of crankpins and journals of built-up crankshafts are to comply with the following requirements in **(1)** and **(2)**:~~

~~(1) The diameters of crankpins and journals are to comply with the requirements in **2.3.1-1**.~~

~~(2) The diameters of axial bores in journals are to comply with the following formula:~~

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

~~D_{BG} : Diameter of axial bore in journal (*mm*)~~

~~D_S : Journal diameter at the shrinkage fit (*mm*)~~

~~S_R : Safety factor against slipping (a value not less than 2 is to be taken)~~

~~M_{max} : Absolute maximum torque at the shrinkage fit ($N \cdot m$)~~

~~μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)~~

~~L_S : Length of shrinkage fit (*mm*)~~

~~σ_{SP} : Minimum yield strength of material used for journal (N/mm^2)~~

~~**2** The dimensions of crank webs are to comply with the following requirements in **(1)** and **(2)**:~~

~~(1) The thickness of crank webs in way of the shrinkage fit is to comply with the following formula:~~

$$\cancel{t \geq \frac{C_1 T D^2}{C_2 d_h{}^2} \frac{1}{\left(1 - \frac{1}{r_s{}^2}\right)}}$$

$$\cancel{t \geq 0.525 d_c}$$

~~where~~

~~t : Thickness of crank web measured parallel to the axis (*mm*)~~

~~C_1 : 10 for 2-stroke cycle in-line engines / 16 for 4-stroke cycle in-line engines~~

~~T : Same as given in **2.3.1-1**~~

~~D : Cylinder bore (*mm*)~~

~~C_2 : $12.8\alpha - 2.4\alpha^2$, but in the case of a hollow shaft, C_2 is to be multiplied by $(1 - R^2)$~~

$$\cancel{\alpha = \frac{Shrinkage\ allowance(mm)}{d_h} \times 10^3}$$

~~R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter~~

~~d_h :Diameter of the hole at shrinkage fit (*mm*)~~

$$\cancel{r_s = \frac{External\ diameter\ of\ web(mm)}{d_h}}$$

~~d_c :Required diameter of crankshaft determined by the formula in **2.3.1-1** (*mm*)~~

~~(2) The dimensions in fillets at the junctions of crank webs with crankpins of semi-built-up crankshafts are to comply with the requirements in **2.3.1-2**.~~

~~**3** In cases of built-up crankshafts, the value of α used in **-2 (1)** is to be within the following~~

~~range:~~

$$\cancel{\frac{1.1Y}{225} \leq \alpha \leq \left(\frac{1.1Y}{225} + 0.8\right)\frac{1}{1-R^2}}$$

~~where~~

~~Y : Specified yield point of crank web material (N/mm^2)~~

~~R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter~~

~~However, when the specified yield point of the crank web exceeds 390 N/mm^2 or the value obtained by the following formula is less than 0.1, the value used for α is to be approved by the Society.~~

~~where~~

$$\cancel{\frac{S - d_p - d_J}{2d_p}}$$

~~S : Length of stroke (mm)~~

~~d_p : Diameter of the crankpin (mm)~~

~~d_J : Diameter of the journal (mm)~~

Built-up crankshaft approval is to be as deemed appropriate by the Society.

2.3.3 Shaft Couplings and Coupling Bolts*

(Omitted)

~~2.3.4 Detailed Evaluation for Strength~~

~~In cases where the crankshafts do not satisfy the requirements given in **2.3.1** and **2.3.2**, special considerations will be made provided that detailed data and calculations regarding the strength of crankshafts are submitted to the Society and are considered appropriate.~~

Annex 2.3.1 has been added as follows.

Annex 2.3.1 CALCULATION METHOD OF CRANKSHAFT STRESS

1.1 Scope

This annex applies to solid-forged and semi-built crankshafts of reciprocating internal combustion engines made of forged or cast steel, with one crank throw between main bearings.

1.2 Principles of Calculation

1 The principles of calculation in this Guidance are as follows:

(1) The design of crankshafts is based on an evaluation of safety against fatigue in highly stressed areas.

(2) These calculations are also based on the assumption that areas exposed to highest stresses are those that are listed below. In addition, attention is to be paid to prevent any excessive stress concentrations in outlets of journal oil bores.
   (a) Fillet transitions between crankpins and webs
   (b) Fillet transitions between journals and webs
   (c) Outlets of crankpin oil bores

(3) Calculations of crankshaft strength require that nominal alternating bending (*See* **1.3.1**) and nominal alternating torsional stresses (*See* **1.3.2**) are determined first. Then, these values are multiplied by appropriate stress concentration factors (*See* **1.4**) which results in equivalent alternating stresses (uni-axial stresses) (*See* **1.6**).

(4) Equivalent alternating stresses are evaluated in accordance with the following:
   (a) In fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a Von Mises equivalent stress under additional assumptions that bending and torsion stresses are time phased and that corresponding peak values occur at the same locations.
   (b) At oil hole outlets, bending and torsion lead to two different stress fields which can be represented by equivalent principal stresses equal to the maximum of principal stresses resulting from combinations of these two stress fields under the assumption that bending and torsion are time phased.

(5) Equivalent alternating stresses are then compared with the fatigue strengths of selected crankshaft materials (*See* **1.7**). These comparisons are to show whether or not those crankshafts concerned are dimensioned adequately (*See* **1.8**).

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

1.3 Calculation of Stresses

1.3.1 Alternating Bending Stress

1 Assumptions

Calculations of alternating bending stresses are based on the following assumptions:

(1) Calculations are based on statically determined systems, composed of a single crank throw supported in the centre of adjacent main journals and subject to gas and inertia forces.

(2) Bending lengths are taken as the length between the two main bearing midpoints (distance L_3, *See* **Fig. 1** and **Fig. 2**).

(3) The bending moments M_{BR} and M_{BT} are calculated based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting rod force, respectively (*See* **Fig. 1**).

(4) For those crank throws with two connecting rods acting upon one crankpin, the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams in accordance with phase (*See* **Fig. 2**).

(5) Bending moments and radial forces acting in webs

(a) The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of solid webs (distance L_1) and are derived from the radial components of connecting rod forces.

(b) Alternating bending and compressive stresses due to bending moments and radial forces are to be related to cross-sections of crank webs. These reference sections result from the web thickness W and the web width B (*See* **Fig. 3**).

(c) Mean stresses are neglected.

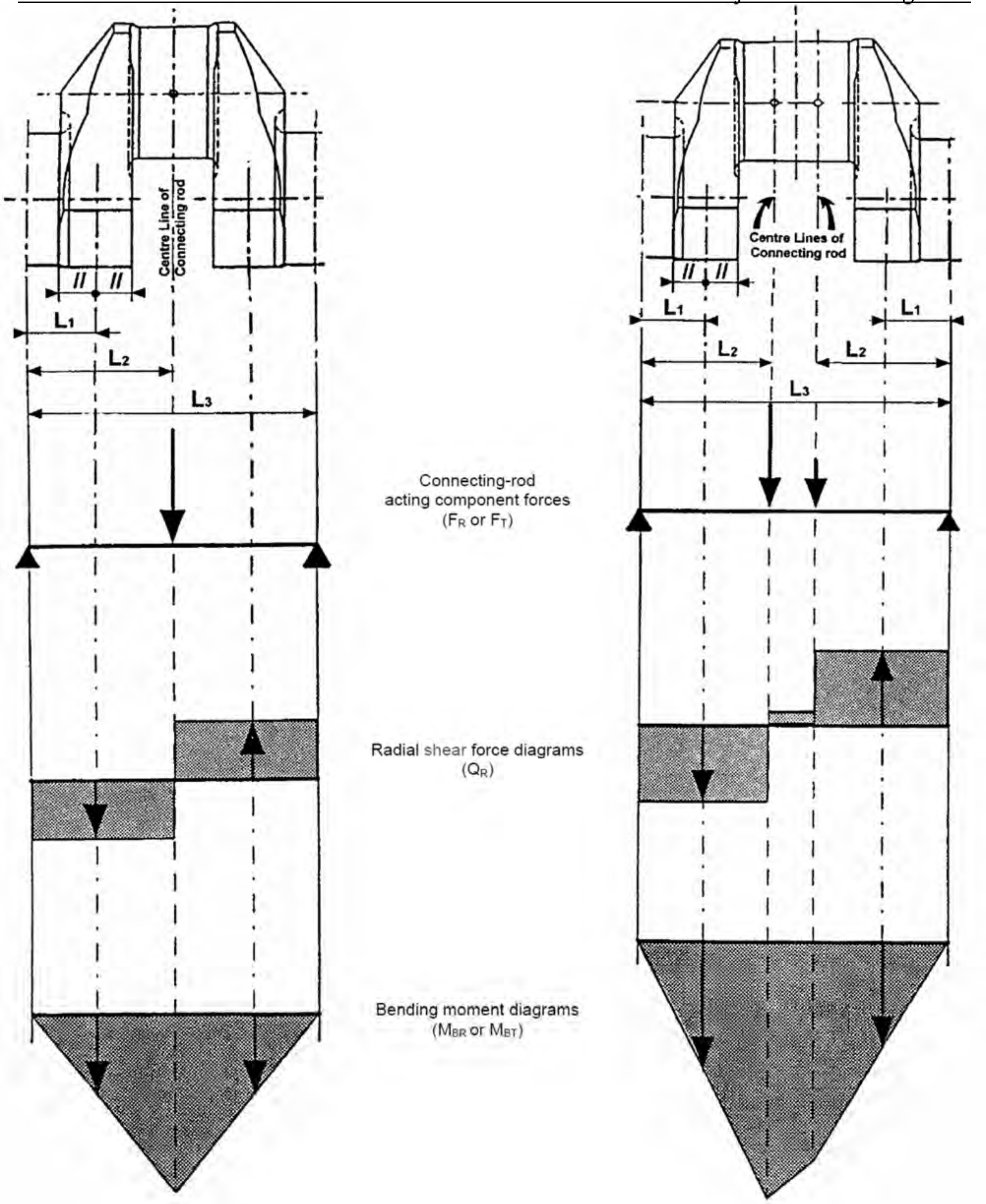
(6) Bending moments acting in outlets of crankpin oil bores

(a) Two relevant bending moments are taken in crankpin cross-sections through oil bores and are derived from the radial and tangential components of connecting rod forces (*See* **Fig. 4**).

(b) Any alternating stresses due to these bending moments are to be related to the cross-sections of axially bored crankpins.

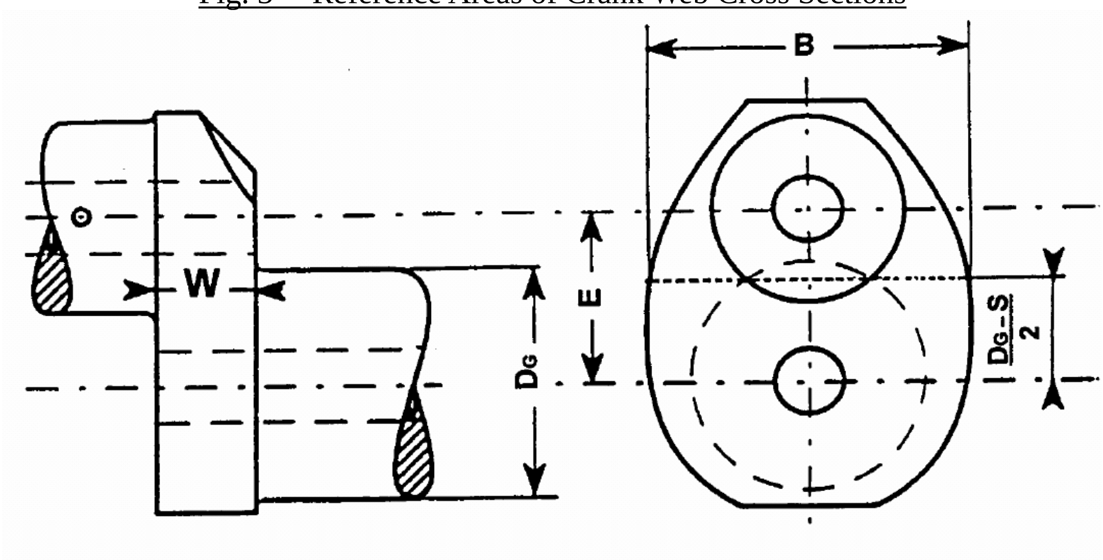
(c) Mean bending stresses are neglected.

Fig. 1 Crank Throw for In-line Engines Fig. 2 Crank Throw for Vee type Engines with Two Adjacent Connecting Rods

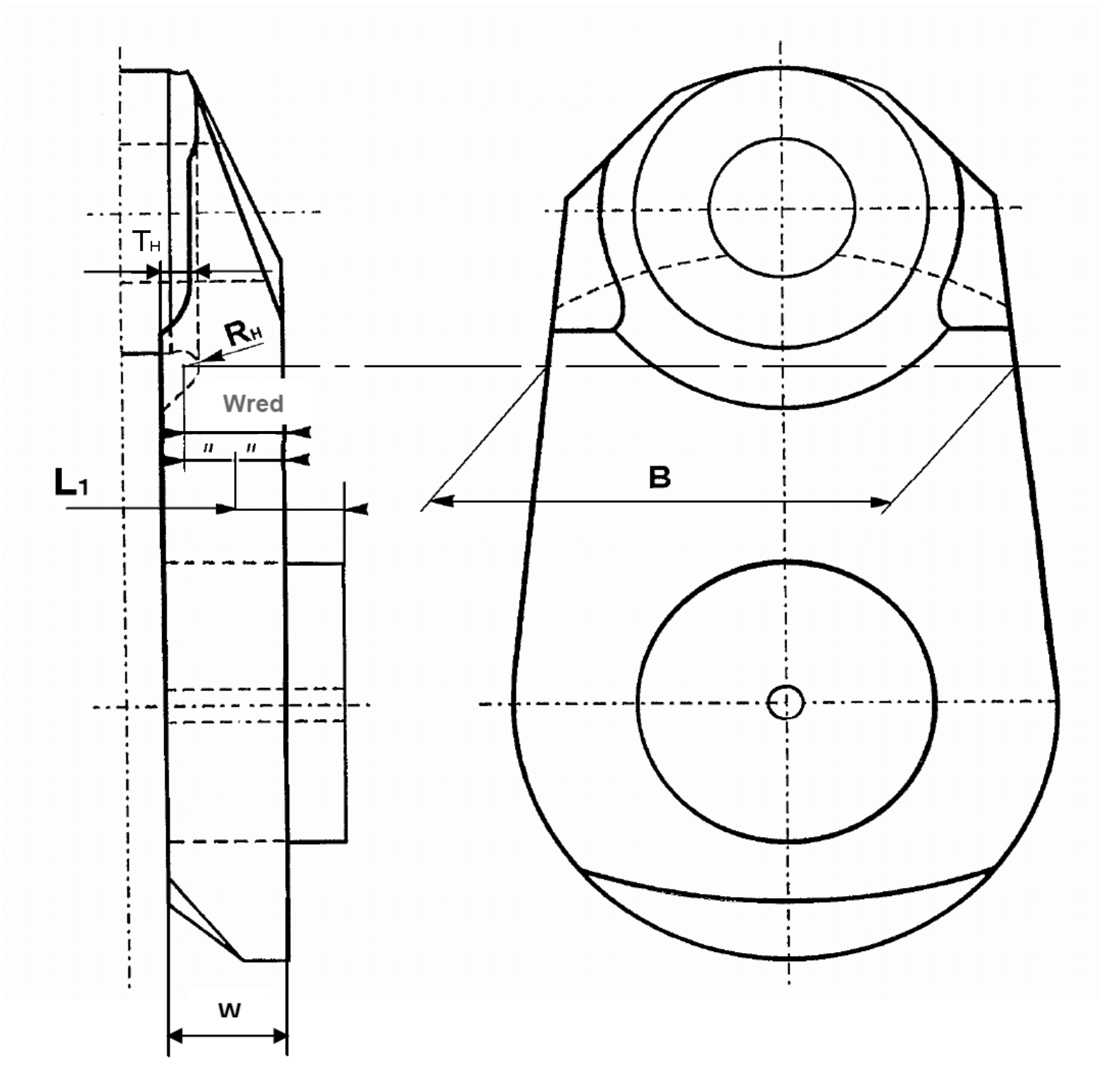


L_1 = Distance between main journal centre line and crank web centre (*See* also **Fig. 3** for crankshafts without overlaps)
L_2 = Distance between main journal centre line and connecting rod centre
L_3 = Distance between two adjacent main journal centre lines

Fig. 3 Reference Areas of Crank Web Cross Sections

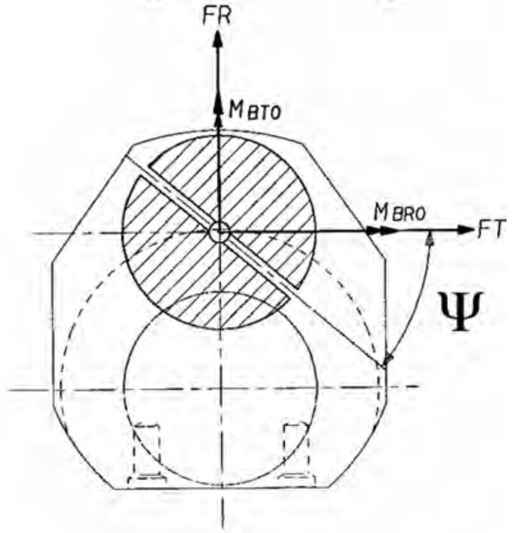


Overlapped crankshaft



Crankshaft without overlap

Fig. 4 Crankpin Sections through Oil Bores



M_{BRO} is the bending moment of the radial component of the connecting rod force.
M_{BTO} is the bending moment of the tangential component of the connecting rod force.

2 Nominal Alternating Bending and Compressive Stresses

(1) Calculation procedures are as follows:

(a) Radial and tangential forces, due to gas and inertia loads, acting upon crankpins at connecting-rod positions will be calculated over one working cycle.

(b) Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments M_{BRF}, M_{BRO}, M_{BTO} and radial forces Q_{RF}, as defined in **-1(5)** and **(6)**, will then be calculated.

(c) In case of Vee type engines, bending moments, progressively calculated from gas and inertia forces, of the two cylinders, acting on one crank throw, are superposed in accordance with phase. Different designs (forked connecting rods, articulated-type connecting rods or adjacent connecting rods) are to be taken into account.

(d) In cases where there are cranks of different geometrical configurations in one crankshaft, calculations are to cover all crank variants.

(2) Nominal alternating bending and compressive stresses in web cross-sections

(a) Calculation of nominal alternating bending stresses is as follows:

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

$$M_{BRFN} = \pm\frac{1}{2}(M_{BRF\max} - M_{BRF\min})$$

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

where

σ_{BFN} : Nominal alternating bending stress related to the web (N/mm^2)

W_{eqw} : Section modulus related to cross-section of web (mm^3)

Ke : Empirical factor considering to some extent the influence of adjacent cranks and bearing restraint with:
Ke = 0.8 for 2-stroke engines
Ke = 1.0 for 4-stroke engines

M_{BRFN} : Alternating bending moment related to the centre of the web ($N \cdot m$) (*See*

Fig. 1 and **Fig. 2**)

$M_{BRF\max}$: Maximum bending moment related to the centre of the web within one working cycle ($N\cdot m$)

$M_{BRF\min}$: Minimum bending moment related to the centre of the web within one working cycle ($N\cdot m$)

(b) Calculation of nominal alternating compressive stresses is as follows:

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

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

$$F = BW$$

where

σ_{QFN} : Nominal alternating compressive stress due to radial force related to the web (N/mm^2)

Q_{RFN} : Alternating radial force related to the web (N) (*See* **Fig. 1** and **Fig. 2**)

$Q_{RF\max}$: Maximum radial force related to the web within one working cycle (N)

$Q_{RF\min}$: Minimum radial force related to the web within one working cycle (N)

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

(3) Nominal alternating bending stress in outlets of crankpin oil bores

Calculation of nominal alternating bending stresses is as follows:

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

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

$$We = \frac{\pi}{32}\left(\frac{D^4 - D_{BH}^4}{D}\right)$$

where

σ_{BON} : Nominal alternating bending stress related to the crankpin diameter (N/mm^2)

M_{BON} : Alternating bending moment calculated at the outlet of crankpin oil bore ($N\cdot m$)

$M_{BO\max}$: Maximum value of bending moment M_{BO} within one working cycle ($N\cdot m$)

$M_{BO\min}$: Minimum value of bending moment M_{BO} within one working cycle ($N\cdot m$)

M_{BO} : Bending moment acting in outlet of crankpin oil bore ($N\cdot m$)

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

ψ: Angular position (*See* **Fig. 4**)

We : Section modulus related to cross-section of axially bored crankpin (mm^3)

D, D_{BH} : *see* **1.4.1**

3 Alternating Bending Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating bending stresses in crankpin fillets is as follows:

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

where

σ_{BH} : Alternating bending stress in crankpin fillet (N/mm^2)

α_B : Stress concentration factor for bending in crankpin fillet (*See* **1.4.2** and **3.1.2-2** of **Appendix 1**)

(2) Calculation of alternating bending stresses in journal fillets (not applicable to semi-built crankshafts) is as the following formulae in **(a)** or **(b)**:

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

where

σ_{BG} : Alternating bending stress in journal fillet (*N/mm*^{2})

β_B : Stress concentration factor for bending in journal fillet (*See* **1.4.3** and **3.1.2-2** of **Appendix 1**)

β_Q : Stress concentration factor for compression due to radial force in journal fillet (*See* **1.4.3** and **3.1.3-2(1)** of **Appendix 1**)

(b) $\sigma_{BG} = \pm(\beta_{BQ} \cdot \sigma_{BFN})$

β_{BQ} : Stress concentration factor for bending and compression due to radial force in journal fillet (*See* **3.1.3-2(2)** of **Appendix 1**)

(3) The calculation of the alternating bending stress in the outlet of crankpin oil bore (only applicable to radially drilled oil hole) is as follows:

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

where

σ_{BO} : Alternating bending stress in outlet of crankpin oil bore (*N/mm*^{2})

γ_B : Stress concentration factor for bending in crankpin oil bore (*See* **1.4.4** and **3.1.2-2** of **Appendix 4**)

1.3.2 Alternating Torsional Stresses

1 Nominal Alternating Torsional Stresses

Calculations for nominal alternating torsional stresses are to be carried out in accordance with the following in order to specify maximum nominal alternating torsional stresses. In addition, maximum nominal alternating torsional stress is to be specified, and the values obtained from such calculations are to be submitted to the Society.

(1) The maximum and minimum torques are to be ascertained for all of the mass points of complete dynamic systems and for entire speed ranges by means of harmonic synthesis of the forced vibrations from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines.

(2) Whilst doing so, allowances must be made for any damping that exists in such systems and for any unfavourable conditions (misfiring, which is defined as the cylinder condition when only compression cycle without any combustion occurs in one of the cylinders).

(3) Speed step calculations are to be selected in such ways that any resonance found in operational speed ranges of engines is detected.

Nominal alternating torsional stresses in mass points calculated results from the following equations:

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

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

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

where

τ_N : Nominal alternating torsional stress related to crankpin or journal (*N/mm*^{2})

W_P : Polar section modulus related to the cross-section of an axially bored crankpin or a bored journal (*mm*^{3})

M_{TN} : Maximum alternating torque ($N \cdot m$)

$M_{T\max}$: Maximum torque ($N \cdot m$)

$M_{T\min}$: Minimum torque ($N \cdot m$)

D, D_{BH}, D_{BG}, D_G: *see* **1.4.1**

In cases where barred speed ranges are necessary, they are to be so arranged that satisfactory operation is possible despite their existence in accordance with **8.2.5** and **8.3.1, Part D of the Rules**. In addition, there are to be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for the normal firing condition.

For crankshaft assessments, the nominal alternating torsional stress considered in **-2** below is the highest calculated value, in accordance with the above method, occurring at the most torsionally loaded mass point of the crankshaft system. Where barred speed ranges exist, the torsional stresses within such ranges are not to be considered in assessment calculations. Crankshaft approval is to instead be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by engine manufacturer). Thus, for each installation, it is to be ensured through suitable calculation that the approved nominal alternating torsional stress is not exceeded. Such calculations are to be submitted to the Society for assessment.

2 Alternating Torsional Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating torsional stresses in crankpin fillets is as follows:

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

where

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

α_T : Stress concentration factor for torsion in crankpin fillet (*See* **1.4.2** and **3.1.1-3** of **Appendix 1**)

τ_N : Nominal alternating torsional stress related to crankpin diameter (N/mm^2)

(2) Calculation of alternating torsional stresses in journal fillets (not applicable to semi-built crankshafts) is as follows:

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

where

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

β_T : Stress concentration factor for torsion in journal fillet (*See* **1.4.3** and **3.1.1-3** of **Appendix 1**)

τ_N : Nominal alternating torsional stress related to journal diameter (N/mm^2)

(3) Calculation of alternating stresses in outlets of crankpin oil bores due to torsion (only applicable to radially drilled oil holes) is as follows:

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

where

σ_{TO} : Alternating stress in outlet of crankpin oil bore due to torsion (N/mm^2)

γ_T : Stress concentration factor for torsion in outlet of crankpin oil bore (*See* **1.4.4** and **3.1.1-2** of **Appendix 4**)

τ_N : Nominal alternating torsional stress related to crankpin diameter (N/mm^2)

1.4 Stress Concentration Factors

1.4.1 Explanation of Terms and Symbols

1 The terms used in this **1.4** are defined as follows:

(1) The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (Von Mises), occurring in fillets under bending loads, to the nominal bending stress related to web cross-sections.

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

(3) The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum

equivalent shear stress, occurring in fillets under torsional loads, to the nominal torsional stress related to axially bored crankpins or journal cross-sections.

(4) The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress, occurring at outlets of crankpin oil bores under bending and torsional loads, to the corresponding nominal stress related to axially bored crankpin cross-sections.

2 The symbols used in this **1.4** mean as follows (*See* **Fig. 5**):

D : Crankpin diameter (*mm*)

D_{BH} : Diameter of axial bore in crankpin (*mm*)

D_O : Diameter of oil bore in crankpin (*mm*)

R_H : Fillet radius of crankpin (*mm*)

T_H : Recess of crankpin fillet (*mm*)

D_G : Journal diameter (*mm*)

D_{BG} : Diameter of axial bore in journal (*mm*)

R_G : Fillet radius of journal (*mm*)

T_G : Recess of journal fillet (*mm*)

E : Pin eccentricity (*mm*)

S : Pin overlap (*mm*)

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

W : Web thickness (*mm*)

In the case of 2-stroke semi-built crankshafts with $T_H > R_H$, the web thickness is to be considered as equal to:

$W_{red} = W - (T_H - R_H)$ (*See* **Fig. 3**)

B : Web width (*mm*)

In the case of 2-stroke semi-built crankshafts, the web width is to be taken in way of crankpin fillet radius centre in accordance with **Fig. 3**.

$r = R_H/D$ (in crankpin fillets), R_G/D(in journal fillets) ($0.03 \le r \le 0.13$)

$s = S/D$ ($s \le 0.5$)

$w = W/D$ ($0.2 \le w \le 0.8$)

$b = B/D$ ($1.1 \le b \le 2.2$)

$d_O = D_O/D$ ($0 \le d_O \le 0.2$)

$d_G = D_{BG}/D$ ($0 \le d_G \le 0.8$)

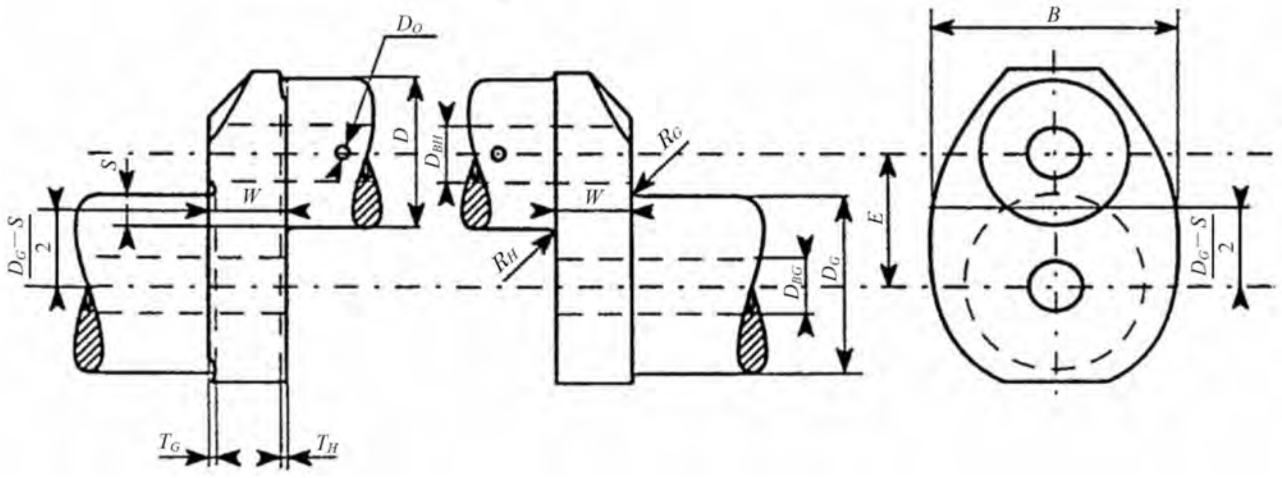
$d_H = D_{BH}/D$ ($0 \le d_H \le 0.8$)

$t_H = T_H/D$

$t_G = T_G/D$

Where the geometry of crankshaft is outside the above ranges, stress concentration factors in crankpin fillets, journal fillets and outlets of crankpin oil bores are to be calculated by utilizing the Finite Element Method (FEM) given in **Appendix 1** and **Appendix 4**. In such cases, care is to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses. In cases where stress concentration factors are evaluated by methods other than theses, relevant documents and the analysis method adopted are to be submitted to the Society in order to demonstrate their equivalence to the methods specified in this paragraph.

Fig. 5 Crank Dimensions



1.4.2 Stress Concentration Factors in Crankpin Fillets

1 The stress concentration factor for bending (α_B) is as follows:

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

where

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

If $s < -0.5$, then $f(s,w)$ is to be calculated after replacing the actual value of s by -0.5.

$$f(w) = 2.1790 \cdot w^{0.7171}$$
$$f(b) = 0.6840 - 0.0077 \cdot b + 0.1473 \cdot b^2$$
$$f(r) = 0.2081 \cdot r^{(-0.5231)}$$
$$f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3$$
$$f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241 \cdot d_H^2 + 2.4147 \cdot d_H^3$$
$$f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$$

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

2 The stress concentration factor for torsion (α_T) is as follows:

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

where

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

If $s < -0.5$, then $f(r,s)$ is to be calculated by replacing the actual value of s by -0.5.

$$f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 + 0.857 \cdot b^3$$
$$f(w) = w^{(-0.145)}$$

1.4.3 Stress Concentration Factors in Journal Fillets

1 The stress concentration factor for bending (β_B) is as follows:

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

where

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

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

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

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

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

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

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

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

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

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

where

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

$f_Q(w) = \dfrac{w}{0.0637 + 0.9369 \cdot w}$

$f_Q(b) = -0.5 + b$

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

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

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

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

3 The stress concentration factor for torsion (β_T) is as follows:

$\beta_T = \alpha_T$ if diameters and fillet radii of crankpins and journals are the same.

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

where

$f(r,s)$, $f(b)$ and $f(w)$ are to be determined in accordance with **1.4.2** (*See* calculation of α_T). However, the radius of the journal fillet is to be related to the journal diameter:

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

1.4.4 Stress Concentration Factors in Outlet of Crankpin Oil Bore

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

$\gamma_B = 3 - 5.88 \cdot d_O + 34.6 \cdot {d_O}^2$

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

$\gamma_T = 4 - 6 \cdot d_O + 30 \cdot {d_O}^2$

1.5 Additional Bending Stresses

In addition to the alternating bending stresses in fillets (σ_{BH} and σ_{BG}) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying σ_{add} as follows:

σ_{add} = ±30 *N/mm*^{2} for crosshead engines

= ±10 N/mm^2 for trunk piston engines
(*) The additional stress of ±30 N/mm^2 is composed of the following two components:
(1) an additional stress of ±20 N/mm^2 resulting from axial vibration
(2) an additional stress of ±10 N/mm^2 resulting from misalignment or bedplate deformation

It is recommended that a value of ±20 N/mm^2 be used for the axial vibration component for assessment purposes in cases where axial vibration calculation results of the complete dynamic system (engine, shafting, gears and propellers) are not available. However, in cases where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

1.6 Equivalent Alternating Stress

1.6.1 Equivalent Alternating Stress in Crankpin Fillets

Equivalent alternating stress in crankpin fillets is calculated in accordance with the following:

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

where
σ_V : Equivalent alternating stress (*N/mm*^{2})
for other parameters *see* **1.3.1-3**, **1.3.2-2** and **1.5**.

1.6.2 Equivalent Alternating Stress in Journal Fillets

Equivalent alternating stress in journal fillets is calculated according to the following:

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

for parameters *see* **1.6.1**.

1.6.3 Equivalent Alternating Stress in Outlets of Crankpin Oil Bores

Equivalent alternating stress in outlets of crankpin oil bores is calculated according to the following:

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

for parameters *see* **1.6.1**.

1.7 Fatigue Strength

1.7.1 Fatigue Strength in Crankpin Fillets

1 The fatigue strength in crankpin fillets is evaluated according to the following: (For calculation purposes, R_H is to be taken as not less than 2 *mm*.)

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

where
σ_{DW} : Allowable fatigue strength of crankshaft (*N/mm*^{2}) in cases where the surfaces of fillets, the outlets of oil bores and the insides of oil bores (down to a minimum depth equal to 1.5 *times* the oil bore diameter) are all smoothly finished
K : Factor for the different types of crankshafts without surface treatment

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

= 1.0 for free form forged crankshafts (without continuous grain flow)

Factor for cast steel crankshafts with cold rolling treatment in fillet areas

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

As an alternative, the value of K can be determined by experiments based either on full-size crank throws (or crankshafts) or on specimens taken from full-size crank throws.

σ_B : Minimum tensile strength of crankshaft material (*N/mm*^{2})

for other parameters *see* **1.4**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full-size crank throw (or crankshaft) or on specimens taken from a full-size crank throw, evaluation of test results is to be carried out in accordance with **Appendix 2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the fillets, every surface-treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix 3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

1.7.2 Fatigue Strength in Journal Fillets

The fatigue strength in journal fillets is evaluated according to the following: (For calculation purposes, R_G is to be taken as not less than 2 *mm*.)

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

for parameters *see* **1.7.1**

1.7.3 Fatigue Strength in Outlets of Crankpin Oil Bores

1 The fatigue strength in outlets of crankpin oil bores is evaluated according to the following: (For calculation purposes, $D_O/2$ is to be taken as not less than 2 *mm*.)

$$\sigma_{DW} = \pm K[0.42\sigma_B + 39.3] \times \left[0.264 + 1.073D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{2}{D_O}}\right]$$

K : Factor for forged crankshafts without surface treatment

= 1.0

Factor for cast steel crankshafts with cold rolling treatment in fillet areas

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

As an alternative, the value of K can be determined by experiments based either on full-size crank throws (or crankshafts) or on specimens taken from full-size crank throws.

for other parameters *see* **1.7.1**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full-size crank throw (or crankshaft), evaluation of test results is to be carried out in accordance with **Appendix 2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the outlets of oil bores, every surface treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix 3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

1.8 Acceptability Criteria

In order to determine whether the dimensions of crankshafts are sufficient, comparisons between equivalent alternating stresses and fatigue strength are to be made. The acceptability factors of crankpin fillets, journal fillets and the outlets of crankpin oil bores are to comply with the following criteria:

$Q \geq 1.15$

where

Q : Acceptability factor

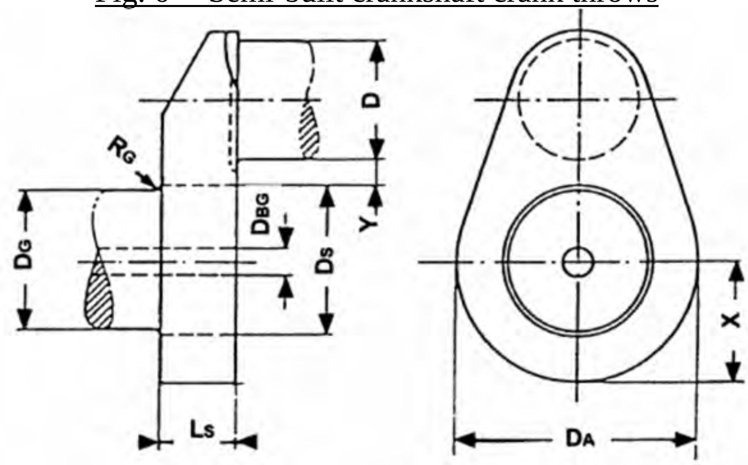
$$= \frac{\sigma_{DW}}{\sigma_V}$$

1.9 Semi-Built Crankshaft Shrink-Fit Calculations

1.9.1 General

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

Fig. 6 Semi-built crankshaft crank throws



D_A : Outside diameter of web or twice the minimum distance x between centre line of journals and outer contour of web, whichever is less (*mm*)

D_S : Shrink diameter (*mm*)

D_G : Journal diameter (*mm*)

D_{BG} : Journal axial bore diameter (*mm*)

L_S : Shrink-fit length (*mm*)

R_G : Journal fillet radius (*mm*)

y : Distance between the adjacent generating lines of journal and pin (*mm*)

$y \geq 0.05 \cdot D_S$

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

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

$$R_G \geq 0.015 \cdot D_G$$

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

where the greater value is to be considered.

3 The actual oversize Z of the shrink-fit is to be within the limits *Zmin* and *Zmax* calculated in accordance with **1.9.3** and **1.9.4**. In cases where the conditions given in **1.9.2** cannot be fulfilled, the above *Zmin* and *Zmax* are not applicable due to multizone-plasticity problems. In such cases, *Zmin* and *Zmax* are to be obtained through FEM calculations.

1.9.2 Journal Axial Bore Diameters

Journal axial bore diameters are to comply with the following formula:

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

S_R : Safety factor against slipping (a value not less than 2 is to be taken)

M_{max} : Absolute maximum torque at the shrinkage fit ($N \cdot m$)

μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)

σ_{SP} : Minimum yield strength of material used for journal (N/mm^2)

1.9.3 Necessary Minimum Shrink-Fit Oversize

The necessary minimum oversize is determined by the greater value calculated according to the following formula:

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

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

Z_{min} : Minimum oversize (mm)

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

σ_{SW} : Minimum yield strength of material for crank web (N/mm^2)

Q_A : Web ratio, $Q_A = D_S / D_A$

Q_S : Shaft ratio, $Q_S = D_{BG} / D_S$

1.9.4 Maximum Permissible Shrink-Fit Oversize

The maximum permissible oversize is calculated according to the following formula:

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

Appendix 1 has been added as follows.

Appendix 1 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD

1.1 General

The objective of the analysis is to develop Finite Element Method (FEM) calculated figures as an alternative to the analytically calculated Stress Concentration Factors (SCF) at the crankshaft fillets. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries and accordingly the application of these formulae is limited to those geometries.
The SCF's calculated in accordance with the rules of this appendix are defined as the ratio of stresses calculated by FEM to nominal stresses in both journal and pin fillets. When used in connection with the present method in **Annex 2.3.1**, von Mises stresses is to be calculated for bending and principal stresses for torsion.
The procedure as well as evaluation guidelines are valid for both solid cranks and semi-built cranks (except journal fillets).
The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.
The calculation of SCF at the oil bores is covered by **Appendix 4**.
It is advised to check the element accuracy of the FE solver in use, e.g. by modeling a simple geometry and comparing the stresses obtained by FEM with the analytical solution for pure bending and torsion.
Boundary Element Method (BEM) may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and perceptions for building the FE-model are presented in **2.1.1**. It is obligatory for the final FE-model to fulfill the requirement in **2.2**.

2.1.1 Element Mesh Recommendations

1 In order to fulfil the mesh quality criteria, it is advised to construct the FE model for the evaluation of Stress Concentration Factors in accordance with the following recommendations:

(1) The model consists of one complete crank, from the main bearing centre line to the opposite side main bearing centre line.

(2) Element type used in the vicinity of the fillets:

- (a) 10-node tetrahedral elements
- (b) 8-node hexahedral elements
- (c) 20-node hexahedral elements

(3) Mesh properties in fillet radii applied to ±90 degrees in a circumferential direction from the crank plane are as follows:

- (a) Maximum element size *a* through the entire fillet as well as in the circumferential direction is to be $a=R_H/4$ in crankpin fillets and $a=R_G/4$ in journal fillets. When using 20-node hexahedral elements, the element size in the circumferential direction may be extended up to 5*a*. In the case of multi-radii fillet, the local fillet radius is to be applied.
- (b) Element size in fillet depth direction (*See* **Fig. 1**):
  - i) First layer thickness equal to element size of *a*

ii) Second layer thickness equal to element to size of 2*a*
iii) Third layer thickness equal to element to size of 3*a*

(4) A minimum of 6 elements are to be set across the web thickness.
(5) The rest of the crank is to be suitable for numeric stability of the solver.
(6) Counterweights have to be modelled only when influencing the global stiffness of the crank significantly.
(7) Modelling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than $2R_H$ or $2R_G$ (*See* **Fig. 2**)
(8) Drillings and holes for weight reduction have to be modelled.
(9) Sub-modelling may be used as far as the software requirements are fulfilled.

Fig. 1 Element Size in Fillet Depth Direction

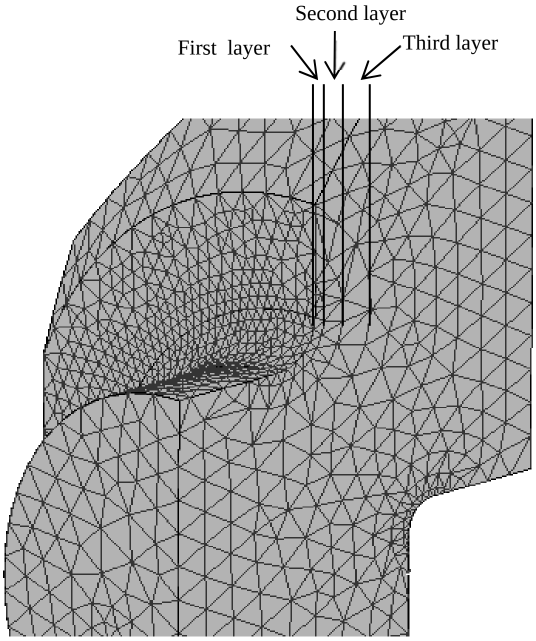
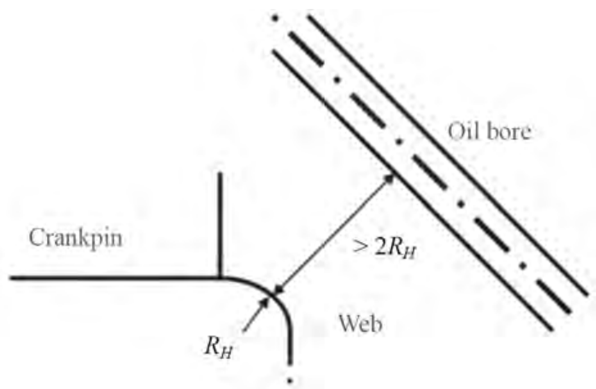


Fig. 2 Oil Bore Proximity to Fillet



2.1.2 Material

1 Material properties applied to steels are as follows:

Young's Modulus : $E = 2.05 \cdot 10^5 MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters are to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

If the actual element mesh does not fulfill any of the following criteria at the area examined for SCF evaluation, then a second calculation with a refined mesh is to be performed.

2.2.1 Principal Stresses Criterion

The quality of the mesh is to be assured by checking the stress component normal to the surface of the fillet radius. Ideally, this stress is to be zero. With principal stresses σ_1, σ_2 and σ_3, the following criterion is required:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0.03 \cdot \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

2.2.2 Averaged/Unaveraged Stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than by 5 % from the 100 % averaged nodal stress results at this node at the examined location.

3.1 Load Cases

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 3** where the torque is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, principal stresses are extracted and the equivalent torsional stress is calculated as follows:

$$\tau_{equiv} = \max\left(\frac{|\sigma_1 - \sigma_2|}{2}, \frac{|\sigma_2 - \sigma_3|}{2}, \frac{|\sigma_1 - \sigma_3|}{2}\right)$$

3 The maximum value taken for the subsequent calculation of the stress concentration factors for torsion in crankpin and journal fillet.

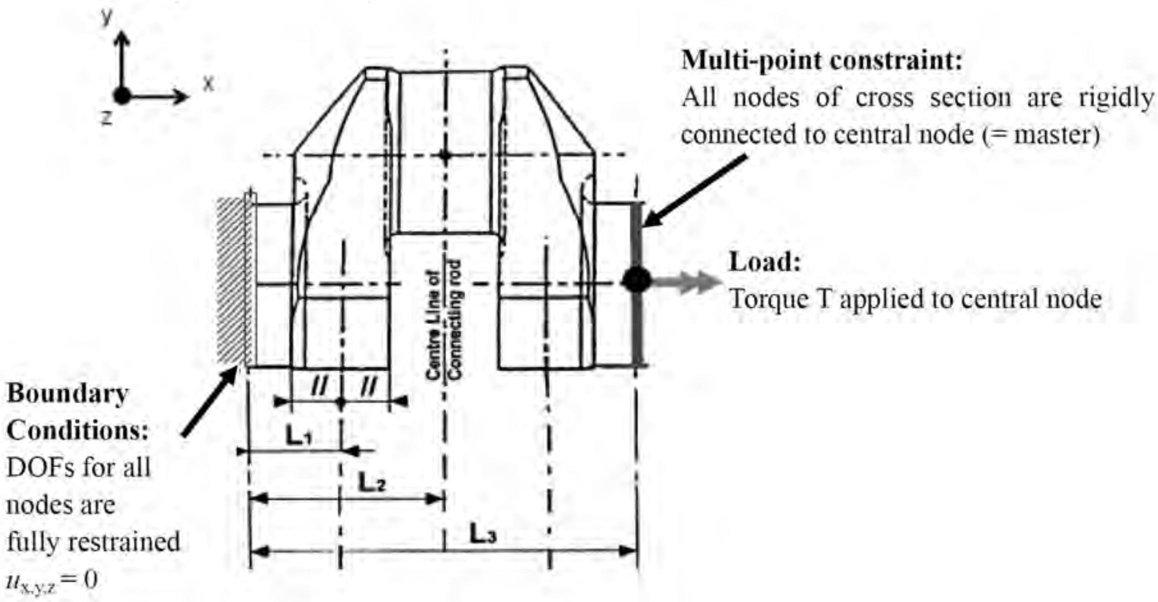
$$\alpha_T = \frac{\tau_{equiv,\alpha}}{\tau_N}$$

$$\beta_T = \frac{\tau_{equiv,\beta}}{\tau_N}$$

where τ_N is nominal torsional stress for the crankpin and journal respectively and is calculated as follows (for W_P *see* **1.3.2** of **Annex 2.3.1**):

$$\tau_N = \frac{T}{W_P}$$

Fig. 3 Boundary and Load Conditions for the Torsion Load Case



3.1.2 Pure Bending (4-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 4** where the bending moment is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the stress concentration factors for bending in crankpin and journal fillet according to the following formulae:

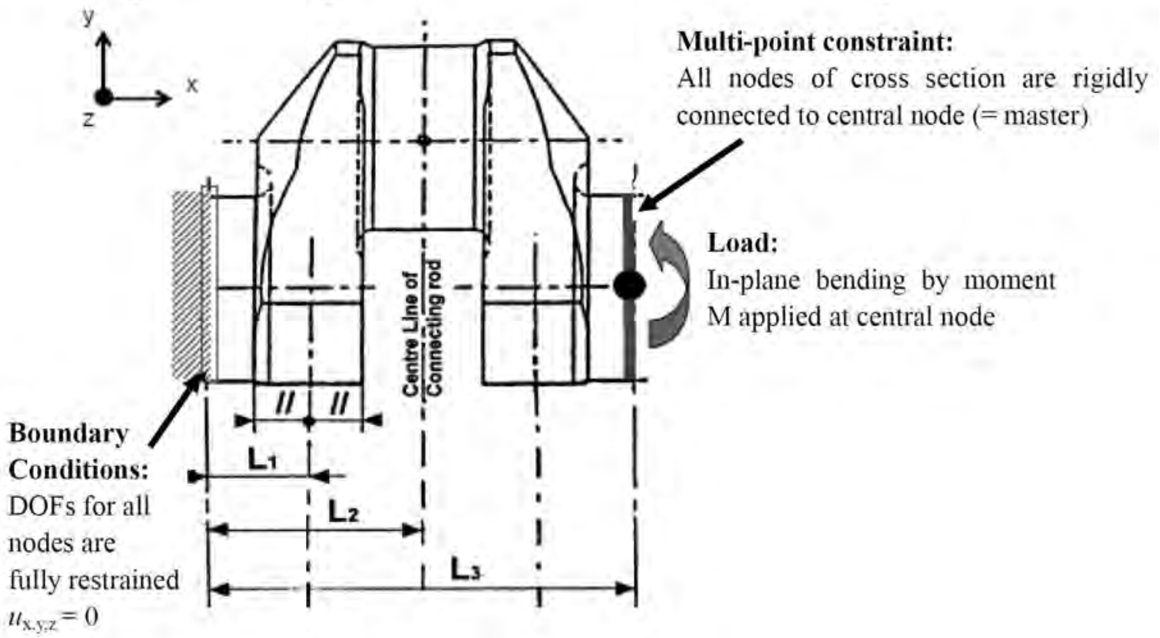
$$\alpha_B = \frac{\sigma_{equiv,\alpha}}{\sigma_N}$$

$$\beta_B = \frac{\sigma_{equiv,\beta}}{\sigma_N}$$

where σ_N is nominal bending stress for the crankpin and journal respectively and is calculated as follows (for W_{eqw} *see* **1.3.1-2(2)** of **Annex 2.3.1**):

$$\sigma_N = \frac{M}{W_{eqw}}$$

Fig. 4 Boundary and Load Conditions for the Pure Bending Load Case



3.1.3 Bending with Shear Force (3-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 5** where the force is applied to the central node located at the pin centre line of the connecting rod.

Fig. 5 Boundary and Load Conditions for the 3-point Bending Load Case of an Inline Engine.

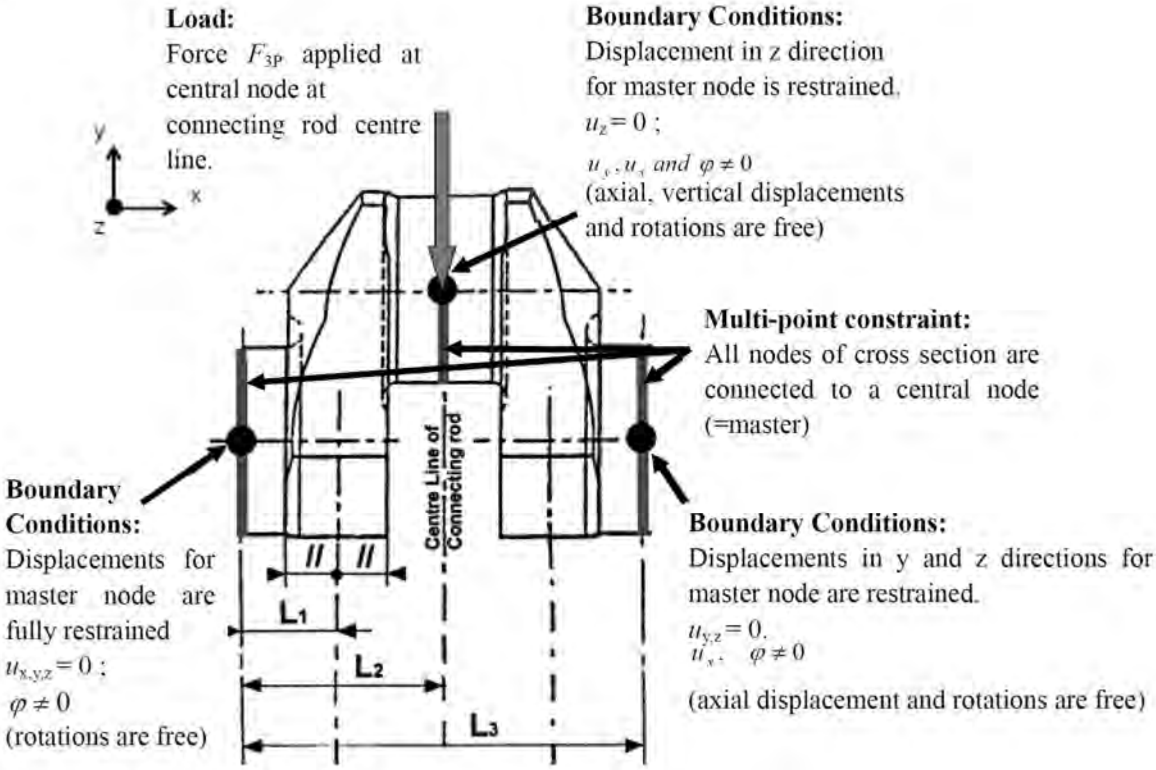
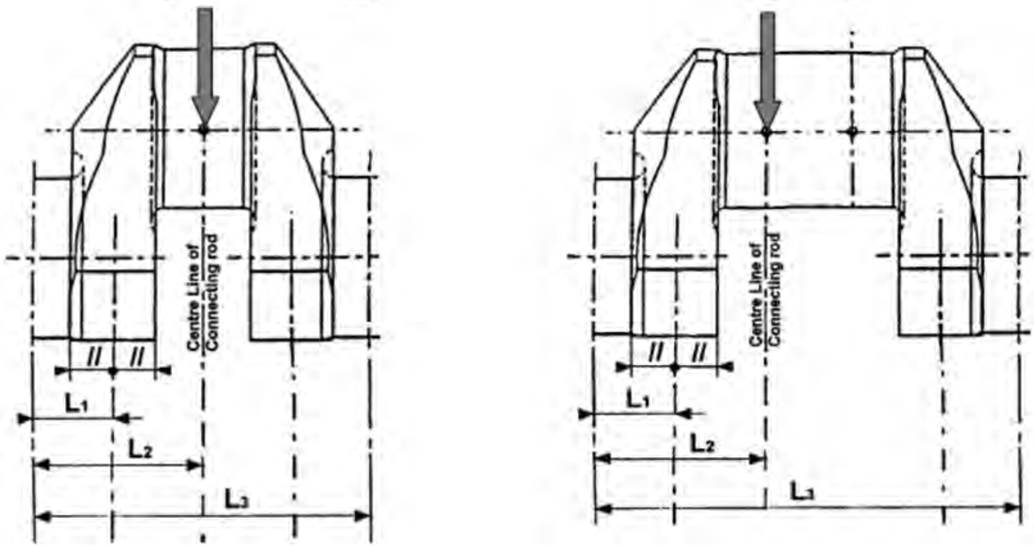


Fig. 6 Load Applications for In-line and Vee Type Engines



2 The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The stress concentration factors in the journal fillet can be determined as shown **i)** or **ii)**.

(1) Stress concentration factor for compression due to radial force in journal fillet β_Q is calculated according to the following:

$\sigma_{3P} = \sigma_{N3P} \cdot \beta_B + \sigma_{Q3P} \cdot \beta_Q$

where

σ_{3P} : as found by the Finite Element Calculation

σ_{N3P}: Nominal bending stress in the web centre due to force F_{3P} applied to the centre line of the actual connecting rod (*See* **Fig. 6**)

β_B : as determined in **3.1.2-2**

$$\sigma_{Q3P} = Q_{3P}/(B.W)$$

Q_{3P} : the radial (shear) force in the web due to the force F_{3P} applied to the centre line of the actual connecting rod (*See* **Fig. 1** and **Fig. 2** of **Annex 2.3.1**)

(2) The stress concentration factor for bending and compression due to radial force in journal fillet β_{BQ} is calculated according to the following:

$$\beta_{BQ} = \frac{\sigma_{3P}}{\sigma_{N3P}}$$

for the relevant parameters *See* **(1)**.

Appendix 2 has been added as follows.

Appendix 2 GUIDANCE FOR EVALUATION OF FATIGUE TESTS

1.1 Introduction

Fatigue testing can be divided into two main groups; testing of small specimens and full-size crank throws. Testing can be made using the staircase method or a modified version thereof which is presented in this appendix. Other statistical evaluation methods may also be applied.

1.2 Small Specimen Testing

1 For crankshafts without any fillet surface treatment, the fatigue strength can be determined by testing small specimens taken from a full-size crank throw.
2 When other areas in the vicinity of the fillets are surface treated introducing residual stresses in the fillets, this approach cannot be applied.
3 One advantage of this approach is the rather high number of specimens which can be then manufactured. Another advantage is that the tests can be made with different stress ratios (R-ratios) and/or different modes e.g. axial, bending and torsion, with or without a notch. This is required for evaluation of the material data to be used with critical plane criteria.

1.3 Full-size Crank Throw Testing

1 For crankshafts with surface treatment the fatigue strength can only be determined through testing of full-size crank throws.
2 The load can be applied by hydraulic actuators in a 3- or 4-point bending arrangement, or by an exciter in a resonance test rig. The latter is frequently used, although it usually limits the stress ratio to $R = -1$.

2.1 Evaluation of Test Results

2.1.1 Principles

1 Prior to fatigue testing the crankshaft is to be tested as required by quality control procedures, e.g. for chemical composition, mechanical properties, surface hardness, hardness depth and extension, fillet surface finish, etc.
2 The test samples are to be prepared so as to represent the "lower end" of the acceptance range e.g. for induction hardened crankshafts this means the lower range of acceptable hardness depth, the shortest extension through a fillet, etc. Otherwise, the mean value test results is to be corrected with a confidence interval: a 90 % confidence interval may be used both for the sample mean and the standard deviation.
3 The test results are to be evaluated to represent the mean fatigue strength, with or without taking into consideration the 90 % confidence interval as mentioned above. The standard deviation is to be considered by taking the 90 % confidence into account. Subsequently the result to be used as the fatigue strength is then the mean fatigue strength minus one standard deviation.
4 If the evaluation aims to find a relationship between (static) mechanical properties and the fatigue strength, the relation is to be based on the real (measured) mechanical properties, not on the specified minimum properties.
5 The calculation technique in **2.1.4** was developed for the original staircase method. However,

since there is no similar method dedicated to the modified staircase method the same is applied for both.

2.1.2 Staircase Method

1 In the original staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress corresponding to the expected average fatigue strength.

(2) If the specimen specified in **(1)** survives 10^7 cycles, it is discarded and the next specimen is subjected to a stress that is one increment above the previous.

(3) A survivor is always followed by the next using a stress one increment above the previous, as specified in **(2)**. The increment is to be selected to correspond to the expected level of the standard deviation.

(4) When a specimen fails prior to reaching 10^7 cycles, the obtained number of cycles is noted and the next specimen is subjected to a stress that is one increment below the previous.

2 This original staircase method is only suitable when a high number of specimens are available.

3 The minimum number of test specimens is to be 25.

2.1.3 Modified Staircase Method

1 When a limited number of specimens are available, it is advisable to apply the modified staircase method.

2 In the modified staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress level that is most likely well below the average fatigue strength.

(2) When this specimen specified in **(1)** has survived 10^7 cycles, this same specimen is subjected to a stress level one increment above the previous. The increment is to be selected to correspond to the expected level of the standard deviation. This is continued with the same specimen until failure.

(3) Then the number of cycles is recorded and the next specimen is subjected to a stress that is at least 2 increments below the level where the previous specimen failed.

3 The acquired result of a modified staircase method is to be used with care, since some results available indicate that testing a runout on a higher test level, especially at high mean stresses, tends to increase the fatigue limit. However, this “training effect” is less pronounced for high strength steels (e.g. $UTS > 800\ MPa$).

4 The minimum number of test specimens is to be 3.

2.1.4 Calculation of Sample Mean and Standard Deviation

A hypothetical example of tests for 5 crank throws is presented further in the subsequent text.

(1) When using the modified staircase method and the evaluation method of Dixon and Mood, the number of samples will be 10, meaning 5 run-outs and 5 failures, i.e.:

Number of samples, $n = 10$

(2) Furthermore, the method distinguishes between:

(a) Less frequent event is failures: $C = 1$

(b) Less frequent event is run-outs: $C = 2$

The method uses only the less frequent occurrence in the test results, i.e. if there are more failures than run-outs, then the number of run-outs is used.

(3) In the modified staircase method, the number of run-outs and failures are usually equal. However, the testing can be unsuccessful, e.g. the number of run-outs can be less than the number of failures if a specimen with 2 increments below the previous failure level goes directly to failure. On the other hand, if this unexpected premature failure occurs after a rather high number of cycles, it is possible to define the level below this as a run-out.

(4) Dixon and Mood's approach, derived from the maximum likelihood theory, which also may be applied here, especially on tests with few samples, presented some simple approximate equations for calculating the sample mean and the standard deviation from the outcome of the staircase test.

(a) The sample mean can be calculated as follows:

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2}\right) \text{ when } C = 1$$

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} + \frac{1}{2}\right) \text{ when } C = 2$$

(b) The standard deviation can be found by

$$s = 1.62 \cdot d \cdot \left(\frac{F \cdot B - A^2}{F^2} + 0.029\right)$$

where:

S_{a0} is the lowest stress level for the less frequent occurrence

d is the stress increment

$$F = \sum f_i$$

$$A = \sum i \cdot f_i$$

$$B = \sum i^2 \cdot f_i$$

i is the stress level numbering

f_i is the number of samples at stress level *i*

The formula for the standard deviation is an approximation and can be used when

$$\frac{B \cdot F - A^2}{F^2} > 0.3$$

and

$$0.5 \cdot s < d < 1.5 \cdot s$$

If any of these two conditions are not fulfilled, a new staircase test is to be considered or the standard deviation is to be taken quite large in order to be on the safe side.

(5) If increment *d* is greatly higher than the standard deviation *s*, the procedure leads to a lower standard deviation and a slightly higher sample mean, both compared to values calculated when the difference between the increment and the standard deviation is relatively small. Respectively, if increment *d* is much less than the standard deviation *s*, the procedure leads to a higher standard deviation and a slightly lower sample mean.

****

Example

Hypothetical test results are shown in **Fig. 1**. The processing of the results and the evaluation of the sample mean and the standard deviation are shown in **Fig. 2**.

Fig. 1 Log Sheet of a Modified Staircase Test.

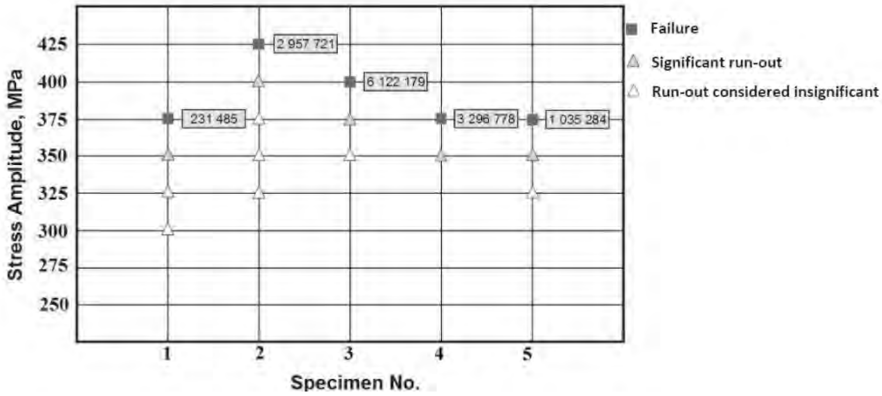
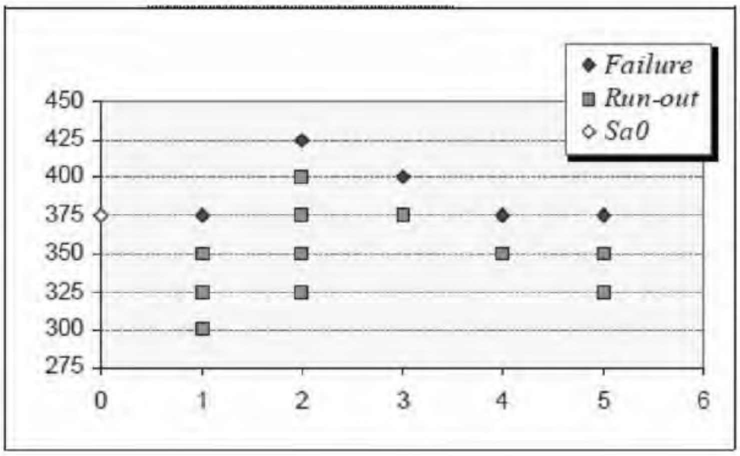


Fig. 2 Processing of a Modified Staircase Test Results.



i	fi	i*fi	i^2*fi
2	1	2	4
1	1	1	1
0	3	0	0
Σ	5	3	5
	F	*A*	*B*

$$F = \sum f_i$$

$$A = \sum i \cdot f_i$$

$$B = \sum i^2 \cdot f_i$$

Notes:
i = 0, 1, 2… is the stress level numbering, the numbering usually starts from zero
f_i is number of test specimen at stress level, i

Sample mean and standard deviation are evaluated as follows based upon **Fig. 2**.

(1) Stress level 0, $S_{a0} := 375MPa$
Level 0 is the lowest value of the less frequent occurrence in the test results.

(2) Stress increment, $d := 25MPa$

(3) $F := 5,\ \ A := 3,\ \ B := 5$

(4) Calculation of sample mean is as follows:

$$S_a := S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2}\right) \ \ C = 1 \ \ S_a = 375.5MPa$$

(5) Calculation of sample standard deviation is as follows:

$$s := 1.62 \cdot d \cdot \left(\frac{B \cdot F - A^2}{F^2} + 0.029\right) \ \ S = 27.09MPa$$

(6) Calculation of standard deviation ratio is as follows:

$$S_r := \frac{s}{S_a} \ \ S_r = 0.072$$

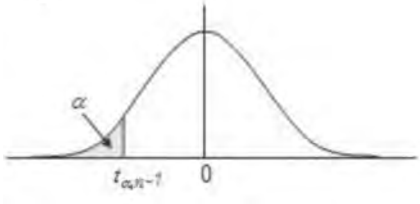
****

2.1.5 Confidence Interval for Mean Fatigue Limit

1 If the staircase fatigue test is repeated, the sample mean and the standard deviation will most likely be different from the previous test. Therefore, it is necessary to assure with a given confidence that the repeated test values will be above the chosen fatigue limit by using a confidence interval for the sample mean.

2 The confidence interval for the sample mean value with unknown variance is known to be distributed in accordance with the *t*-distribution (also called student's *t*-distribution) which is a distribution symmetric around the average. (*See* **Fig. 3**)

Fig. 3 Student's *t*-distribution



Note:
The confidence level normally used for the sample mean is 90 %, meaning that 90 % of sample means from repeated tests will be above the value calculated with the chosen confidence level. The **Fig. 3** shows the t-value for (1-α) · 100 % confidence interval for the sample mean.

3 If S_a is the empirical mean and s is the empirical standard deviation over a series of n samples, in which the variable values are normally distributed with an unknown sample mean and unknown variance, the $(1-\alpha)\cdot 100\ \%$ confidence interval for the mean is:

$$P\left(S_a - t_{\alpha,n-1}\cdot\frac{s}{\sqrt{n}} < S_{aX\%}\right) = 1-\alpha$$

4 The resulting confidence interval is symmetric around the empirical mean of the sample values, and the lower endpoint can be found as:

$$S_{aX\%} = S_a - t_{\alpha,n-1}\cdot\frac{s}{\sqrt{n}}$$

which is the mean fatigue limit (population value) to be used to obtain the reduced fatigue limit where the limits for the probability of failure are taken into consideration.

****

Example

Applying a 90 % confidence interval ($\alpha = 0.1$) and n = 10 (5 failures and 5 run-outs) leads to $t_{\alpha,n-1} = 1.383$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990. Note that $v = n-1$ in the tables.). Hence:

$$S_{a90\%} = S_a - 1.383\cdot d\cdot\frac{s}{\sqrt{n}} = S_a - 0.4373\cdot s$$

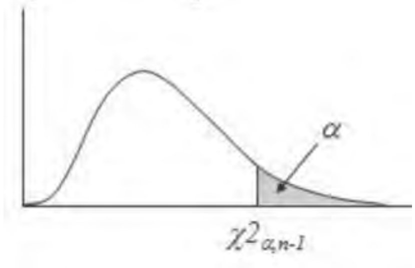
To be conservative, some authors would consider n to be 5, as the physical number of used specimens, then $t_{\alpha,n-1} = 1.533$.

****

2.1.6 Confidence Interval for Standard Deviation

1 The confidence interval for the variance of a normal random variable is known to possess a chi-square distribution with $n-1$ degrees of freedom (*See* **Fig. 4**).

Fig. 4 Chi-square Distribution



Note:
The confidence level on the standard deviation is used to ensure that the standard deviations for repeated tests are below an upper limit obtained from the fatigue test standard deviation with a confidence level. **Figure 4** shows the chi-square for (1-α)· 100 % confidence interval for the variance.

2 An assumed fatigue test value from *n* samples is a normal random variable with a variance of σ^2 and has an empirical variance s^2. Then a $(1-\alpha)\cdot 100\ \%$ confidence interval for the variance is obtained according to the following formulae:

$$P\left(\frac{(n-1)s^2}{\sigma^2} < \chi^2_{\alpha,n-1}\right) = 1-\alpha$$

3 A $(1-\alpha)\cdot 100\ \%$ confidence interval for the standard deviation is obtained by the square root of the upper limit of the confidence interval for the variance and can be obtained according to the following formula:

$$S_{X\%} = \sqrt{\frac{n-1}{\chi^2_{\alpha,n-1}}}\cdot s$$

This standard deviation (population value) is to be used to obtain the fatigue limit, where the limits for the probability of failure are taken into consideration.

****

Example

Applying a 90 % confidence interval ($\alpha = 0.1$) and *n* = 10 (5 failures and 5 run-outs) leads to $\chi^2_{\alpha,n-1} = 4.168$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990).

Hence:

$$S_{90\%} = \sqrt{\frac{n-1}{4.168}}\cdot s = 1.47\cdot s$$

To be conservative, some authors would consider *n* to be 5, as the physical number of the used specimens, then $\chi^2_{\alpha,n-1} = 1.064$.

****

3.1 Small Specimen Testing

3.1.1 General

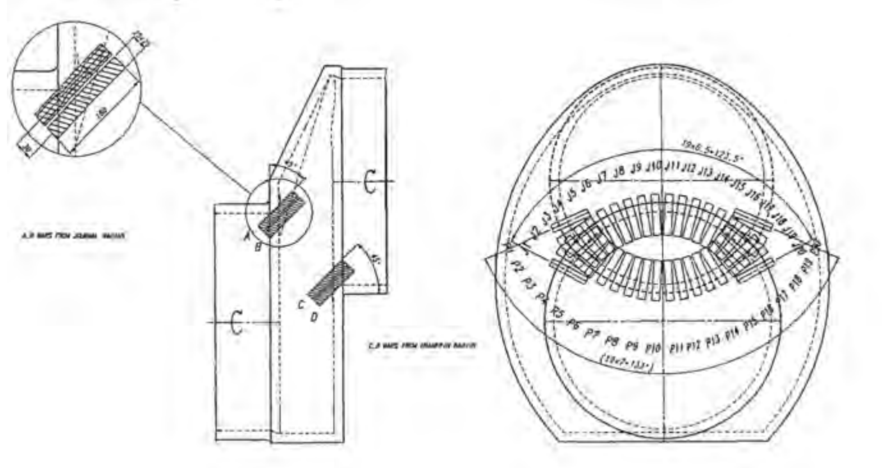
1 In this appendix, a small specimen is considered to be one of the specimens taken from a crank throw.

2 Since the specimens are to be representative for the fillet fatigue strength, they are to be taken out close to the fillets. (*See* **Fig. 5**)

3 It is to be made certain that the principal stress direction in the specimen testing is equivalent to the full-size crank throw. The verification is recommended to be done by utilizing the finite element method.

4 The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

Fig. 5 Specimen Locations in a Crank Throw



3.1.2 Determination of Bending Fatigue Strength

1 It is advisable to use un-notched specimens in order to avoid uncertainties related to the stress gradient influence. Push-pull testing method (stress ratio $R = -1$) is preferred, but especially for the purpose of critical plane criteria other stress ratios and methods may be added.

2 In order to ensure principal stress direction in push-pull testing to represent the full-size crank throw principal stress direction and when no further information is available, the specimen is to be taken at a 45-degree angle as shown in **Fig. 5**. If the objective of the testing is to document the influence of high cleanliness, test samples taken from positions approximately 120 degrees in a circumferential direction may be used. (*See* **Fig. 5**) If the objective of the testing is to document the influence of continuous grain flow (cgf) forging, the specimens are to be restricted to the vicinity of the crank plane.

3.1.3 Determination of Torsional Fatigue Strength

1 If the specimens are subjected to torsional testing, the selection of samples is to follow the same guidelines as for bending above. The stress gradient influence has to be considered in the evaluation.

2 If the specimens are tested in push-pull and no further information is available, the samples are to be taken out at a 45-degree angle to the crank plane in order to ensure collinearity of the principal stress direction between the specimen and the full-size crank throw. When taking the specimen at a distance from the (crank) middle plane of the crankshaft along the fillet, this plane rotates around the pin centre point making it possible to resample the fracture direction due to torsion (the results are to be converted into the pertinent torsional values).

3.1.4 Other Test Positions

1 If the test purpose is to find fatigue properties and the crankshaft is forged in a manner likely to lead to cgf, the specimens may also be taken longitudinally from a prolonged shaft piece where specimens for mechanical testing are usually taken. The condition is that this prolonged shaft piece is heat treated as a part of the crankshaft and that the size is so as to result in a similar quenching rate as the crank throw.

2 When using test results from a prolonged shaft piece, it has to be considered how well the grain flow in that shaft piece is representative for the crank fillets.

3.1.5 Correlation of Test Results

1 The fatigue strength achieved by specimen testing is to be converted to correspond to the full-size crankshaft fatigue strength with an appropriate method (size effect).

2 When using the bending fatigue properties from tests mentioned in **3.1**, it is to be kept in mind that successful continuous grain flow (cgf) forging leading to elevated values compared to other (non cgf) forging, will normally not lead to a torsional fatigue strength improvement of the same magnitude. In such cases it is advised to either carry out also torsional testing or to make a conservative assessment of the torsional fatigue strength. This approach is applicable when using the Gough Pollard criterion. However, this approach is not recognised when using the von Mises or a multi-axial criterion such as Findley.

3 If the found ratio between bending and torsion fatigue differs significantly from $\sqrt{3}$, one is to consider replacing the use of the von Mises criterion with the Gough Pollard criterion. Also, if critical plane criteria are used, it has to be kept in mind that cgf makes the material inhomogeneous in terms of fatigue strength, meaning that the material parameters differ with the directions of the planes.

4 Any addition of influence factors is to be made with caution. If for example a certain addition for clean steel is documented, it may not necessarily be fully combined with a *K*-factor for cgf. Direct testing of samples from a clean and cgf forged crank is preferred.

4.1 Full-Size Testing

4.1.1 Hydraulic Pulsation

1 A hydraulic test rig can be arranged for testing a crankshaft in 3-point or 4-point bending as well as in torsion. This allows for testing with any *R*-ratio.

2 Although the applied load is to be verified by strain gauge measurements on plain shaft sections for the initiation of the test, it is not necessarily used during the test for controlling load. It is also pertinent to check fillet stresses with strain gauge chains.

3 Furthermore, it is important that the test rig provides boundary conditions as defined in **3.1** of **Appendix 3**.

4 The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

4.1.2 Resonance Tester

1 A rig for bending fatigue normally works with an *R*-ratio of -1. **Fig. 6** shows a layout of the testing arrangement.

2 The applied load is to be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains.

3 Clamping around the journals is to be arranged in a way that prevents severe fretting which could lead to a failure under the edges of the clamps. If some distance between the clamps and the journal fillets is provided, the loading is consistent with 4-point bending and thus representative for the journal fillets also.

4 In an engine, the crankpin fillets normally operate with an *R*-ratio slightly above -1 and the journal fillets slightly below -1. If found necessary, it is possible to introduce a mean load (deviate from $R = -1$) by means of a spring preload.

5 A rig for torsion fatigue can also be arranged as shown in **Fig. 7**. When a crank throw is subjected to torsion, the twist of the crankpin makes the journals move sideways. If one single crank throw is tested in a torsion resonance test rig, the journals with their clamped-on weights will vibrate heavily sideways. This sideway movement of the clamped-on weights can be reduced by having two crank throws, especially if the cranks are almost in the same direction. However, the journal in the middle will move more.

6 Since sideway movements can cause some bending stresses, the plain portions of the crankpins are to also be provided with strain gauges arranged to measure any possible bending that could have an influence on the test results.

7 Similarly, to the bending case the applied load is to be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains as well.

Fig. 6 An Example of Testing Arrangement of the Resonance Tester for Bending Loading

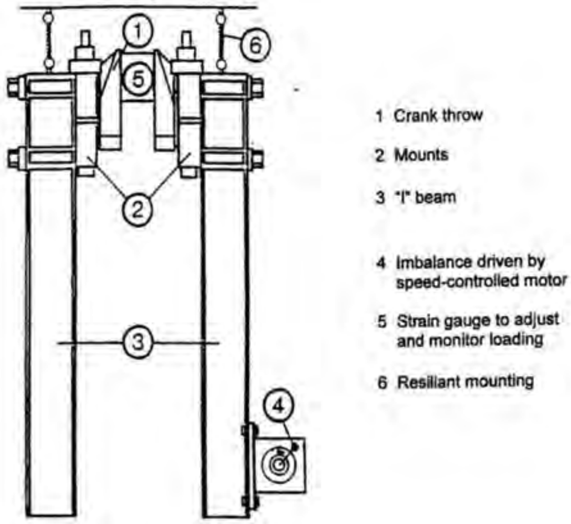
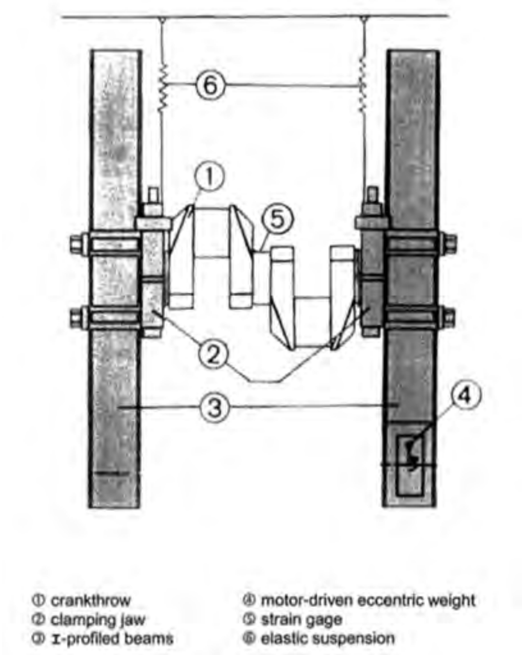


Fig. 7 An Example of Testing Arrangement of the Resonance Tester for Torsion Loading with Double Crank Throw Section



4.1.3 Use of Results and Crankshaft Acceptability

1 In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability (*See* **1.8** of **Annex 2.3.1**), the Gough-Pollard approach and the maximum principal equivalent stress formulation can be applied for the following cases:

(1) Related to the crankpin diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BH}}{\sigma_{DWCT}}\right)^2 + \left(\frac{\tau_{BH}}{\tau_{DWCT}}\right)^2}\right)^{-1}$$

where:

σ_{DWCT} : fatigue strength by bending testing

τ_{DWCT} : fatigue strength by torsion testing

(2) Related to crankpin oil bore:

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

where:

σ_{DWOT} : fatigue strength by means of maximum principal stress from torsion testing

(3) Related to the journal diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BG}}{\sigma_{DWJT}}\right)^2 + \left(\frac{\tau_G}{\tau_{DWJT}}\right)^2}\right)^{-1}$$

where:

σ_{DWJT} : fatigue strength by bending testing

τ_{DWJT} : fatigue strength by torsion testing.

2 In case increase in fatigue strength due to the surface treatment is considered to be similar between the above cases, it is sufficient to test only the most critical location in accordance with the calculation where the surface treatment had not been taken into account.

5.1 Use of Existing Results for Similar Crankshafts

5.1.1 Use of Existing Results

1 For fillets or oil bores without surface treatment, the fatigue properties found by testing may be used for similar crankshaft designs providing:

(1) Material:
  - (a) Similar material type
  - (b) Cleanliness on the same or better level
  - (c) The same mechanical properties can be granted (size versus hardenability)

(2) Geometry:
  - (a) Difference in the size effect of stress gradient is insignificant or it is considered
  - (b) Principal stress direction is equivalent. (*See* **3.1**)

(3) Manufacturing:
  - (a) Similar manufacturing process

2 Induction hardened or gas nitrited crankshafts will suffer fatigue either at the surface or at the transition to the core. The surface fatigue strength as determined by fatigue tests of full-size cranks, may be used on an equal or similar design as the tested crankshaft when the fatigue initiation occurred at the surface. With the similar design, it is meant that a similar material type and surface hardness are used and the fillet radius and hardening depth are within approximately ± 30 % of the tested crankshaft.

3 Fatigue initiation in the transition zone can be either subsurface, i.e. below the hard layer, or at the surface where the hardening ends. The fatigue strength at the transition to the core can be determined by fatigue tests as described above, provided that the fatigue initiation occurred at the transition to the core. Tests made with the core material only will not be representative since the tension residual stresses at the transition are lacking.

4 It has to be noted also what some recent research has shown:

The fatigue limit can decrease in the very high cycle domain with subsurface crack initiation due to trapped hydrogen that accumulates through diffusion around some internal defect functioning as an initiation point. In these cases, it would be appropriate to reduce the fatigue limit by some percent per decade of cycles beyond 10^7. Based on a publication by Yukitaka Murakami “Metal Fatigue: Effects of Small Defects and Non-metallic Inclusions” the reduction is suggested to be 5 % per decade especially when the hydrogen content is considered to be high.

Appendix 3 has been added as follows.

Appendix 3 GUIDANCE FOR CALCULATION OF SURFACE TREATED FILLETS AND OIL BORE OUTLETS

1.1 Introduction

This appendix deals with surface treated fillets and oil bore outlets. The various treatments are explained and some empirical formulae are given for calculation purposes.
Please note that measurements or more specific knowledge is to be used if available. However, in the case of a wide scatter (e.g. for residual stresses) the values are to be chosen from the end of the range that would be on the safe side for calculation purposes.

2.1 Definition of Surface Treatment

"Surface treatment" is a term covering treatments such as thermal, chemical or mechanical operations, leading to inhomogeneous material properties - such as hardness, chemistry or residual stresses - from the surface to the core.

2.2 Surface Treatment Methods

The following list given in **Table 1** covers possible treatment methods and how they influence the properties that are decisive for the fatigue strength.

Table 1 Surface Treatment Methods and the Characteristics They Affect.

Treatment method	Affecting
Induction hardening	Hardness and residual stresses
Nitriding	Chemistry, hardness and residual stresses
Case hardening	Chemistry, hardness and residual stresses
Die quenching (no temper)	Hardness and residual stresses
Cold rolling	Residual stresses
Stroke peening	Residual stresses
Shot peening	Residual stresses
Laser peening	Residual stresses
Ball coining	Residual stresses

Note:
It is important to note that since only induction hardening, nitriding, cold rolling and stroke peening are considered relevant for marine engines, other methods as well as combination of two or more of the above are not dealt with in this appendix. In addition, die quenching can be considered in the same way as induction hardening.

3.1 Calculation Principles

3.1.1 General

1 The basic principle is that the alternating working stresses is to be below the local fatigue strength (including the effect of surface treatment) wherein non-propagating cracks may occur. (*See* also **6.1.2** for details) This is then divided by a certain safety factor.
2 This applies through the entire fillet or oil bore contour as well as below the surface to a depth below the treatment - affected zone - i.e. to cover the depth all the way to the core.
3 Consideration of the local fatigue strength is to include the influence of the local hardness,

residual stress and mean working stress.

4 The influence of the “giga-cycle effect”, especially for initiation of subsurface cracks, is to be covered by the choice of safety margin.

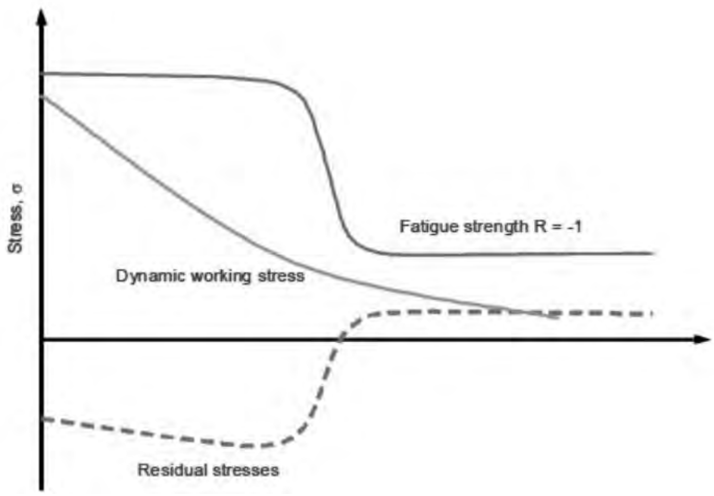
5 It is of vital importance that the extension of hardening/peening in an area with concentrated stresses be duly considered.

6 Any transition where the hardening/peening is ended is likely to have considerable tensile residual stresses. This forms a “weak spot” and is important if it coincides with an area of high stresses.

7 Alternating and mean working stresses are to be known for the entire area of the stress concentration as well as to a depth of about 1.2 times the depth of the treatment. (*See* **Fig. 1**)

8 The acceptability criterion is to be applied stepwise from the surface to the core as well as from the point of maximum stress concentration along the fillet surface contour to the web.

Fig. 1 Stresses as Functions of Depth, General Principles (In case of Induction Hardening)



Note:
The base axis is either the depth (perpendicular to the surface) or along the fillet contour.

3.2 Evaluation of Local Fillet Stresses

3.2.1 Evaluation Based upon FEM

It is necessary to have knowledge of the stresses along the fillet contour as well as in the subsurface to a depth somewhat beyond the hardened layer. Normally this will be found via FEA as described in **Appendix 3**. However, the element size in the subsurface range will have to be the same size as at the surface. For crankpin hardening only the small element size will have to be continued along the surface to the hard layer.

3.2.2 Evaluation Based upon a Simplified Approach

1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined stress concentration factors (SCFs), as in **1.4** of **Annex 2.3.1** if within its validity range, and a relative stress gradient inversely proportional to the fillet radius. Bending and torsional stresses are to be addressed separately. The combination of these is addressed by the acceptability criterion.

2 The subsurface transition-zone stresses, with the minimum hardening depth, can be determined by means of local stress concentration factors along an axis perpendicular to the fillet surface.

(1) Calculation of the local SCFs $\alpha_{B\text{-local}}$ and $\beta_{B\text{-local}}$ for bending in crankpin and journal fillets is

as follows: (*See* **Fig. 2**)

$$\alpha_{B-local} = (\alpha_B - 1) \cdot e^{\frac{-2 \cdot t}{R_H}} + 1 - \left(\frac{2 \cdot t}{\sqrt{W^2 + S^2}}\right)^{\frac{0.6}{\sqrt{\alpha_B}}}$$

$$\beta_{B-local} = (\beta_B - 1) \cdot e^{\frac{-2 \cdot t}{R_G}} + 1 - \left(\frac{2 \cdot t}{\sqrt{W^2 + S^2}}\right)^{\frac{0.6}{\sqrt{\beta_B}}}$$

For parameters see **1.3.1-3** and **1.4** of **Annex 2.3.1**

(2) Calculation of the local SCFs $\alpha_{T\text{-local}}$ and $\beta_{T\text{-local}}$ for torsion in crankpin and journal fillets is as follows: (*See* **Fig. 3**)

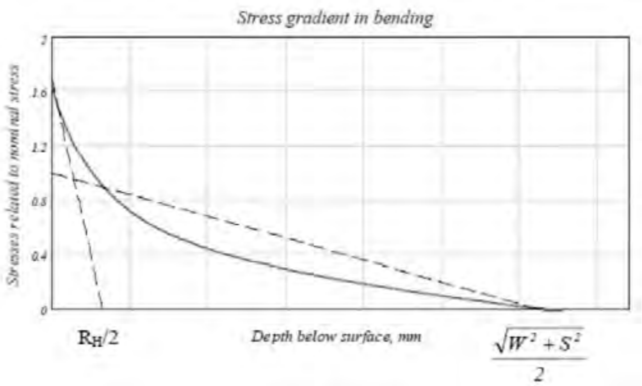
$$\alpha_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_H}} + 1 - \left(\frac{2 \cdot t}{D}\right)^{\frac{0.6}{\sqrt{\alpha_T}}}$$

$$\beta_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_G}} + 1 - \left(\frac{2 \cdot t}{D_G}\right)^{\frac{0.6}{\sqrt{\beta_T}}}$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

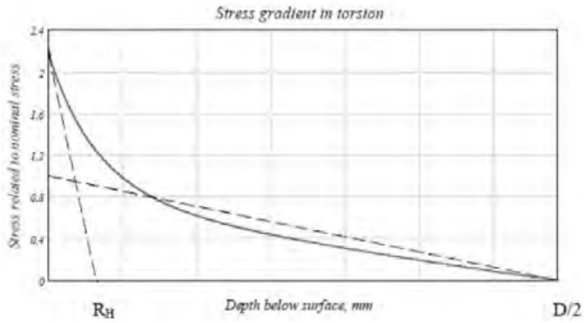
3 If the pin is hardened only and the end of the hardened zone is closer to the fillet than three times the maximum hardness depth, FEA is to be used to determine the actual stresses in the transition zone.

Fig. 2 Bending SCF in the Crankpin Fillet as a Function of Depth.



Note:
The corresponding SCF for the journal fillet can be found by replacing R_H with R_G

Fig. 3 Torsional SCF in the Crankpin Fillet as a Function of Depth.



Note:
The corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G

3.3 Evaluation of Oil Bore Stresses

3.3.1 Evaluation Based upon FEM

1 Stresses in the oil bores can be determined also by FEA.

2 The element size is to be less than 1/8 of the oil bore diameter D_O and the element mesh quality criteria are to be followed as prescribed in **Appendix 1**.

3 The fine element mesh is to continue well beyond a radial depth corresponding to the hardening depth.

4 The loads to be applied in the FEA are the torque and the bending moment, with four-point bending. (*See* **3.1.1** and **3.1.2** of **Appendix 1**)

3.3.2 Evaluation Based upon a Simplified Approach

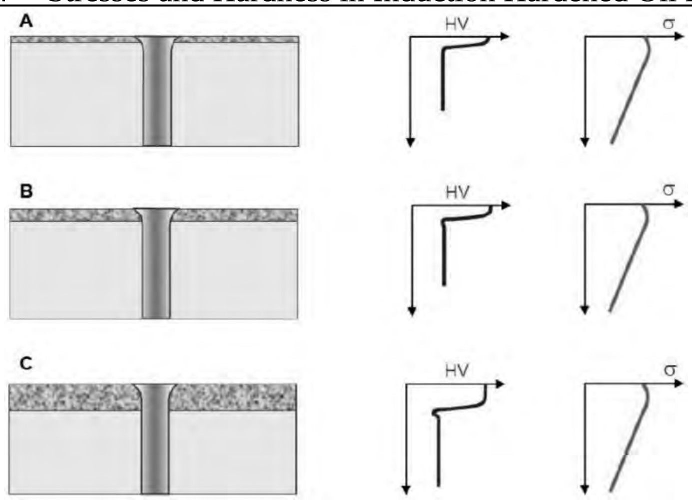
1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from **1.3** of **Annex 2.3.1** if within its applicability range.

2 Bending and torsional stresses at the point of peak stresses are combined as in **1.6** of **Annex 2.3.1**.

3 **Figure 4** indicates a local drop of the hardness in the transition zone between a hard and soft material. Whether this drop occurs depends also on the tempering temperature after quenching in the QT process.

4 The peak stress in the bore occurs at the end of the edge rounding. Within this zone the stress drops almost linearly to the centre of the pin. As can be seen from **Fig. 4**, for shallow (A) and intermediate (B) hardening, the transition point practically coincides with the point of maximal stresses. For deep (C) hardening the transition point comes outside of the point of peak stress and the local stress can be assessed as a portion $(1-2tH/D)$ of the peak stresses where tH is the hardening depth.

Fig. 4 Stresses and Hardness in Induction Hardened Oil Holes



5 The subsurface transition-zone stresses (using the minimum hardening depth) can be determined by means of local stress concentration factors along an axis perpendicular to the oil bore surface.

(1) Calculation of the local SCF $\gamma_{B-local}$ for bending in crankpin oil bores is as follows:

$$\gamma_{B-local} = (\gamma_B - 1) \cdot e^{\frac{-4t}{D_O}} + 1$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

(2) Calculation of the local SCF $\gamma_{T-local}$ for torsion in crankpin oil bores is as follows:

$$\gamma_{T-local} = (\gamma_T - 1) \cdot e^{\frac{-2t}{D_o}} + 1$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

3.4 Acceptability Criteria

The acceptability factors of crankpin fillets, journal fillets and the outlets of crankpin oil bores are to comply with the following criteria, which is specified in **1.8** of **Annex 2.3.1**:

$$Q \geq 1.15$$

This is to be extended to cover also surface treated areas independent of whether surface or transition zone is examined.

4.1 Induction Hardening

4.1.1 General

1 Generally, the hardness specification is to specify the surface hardness range i.e. minimum and maximum values, the minimum and maximum extension in or through the fillet and also the minimum and maximum depth along the fillet contour. The referenced Vickers hardness is considered to be *HV*0.5...*HV*5.

2 The induction hardening depth is defined as the depth where the hardness is 80 % of the minimum specified surface hardness.

3 In the case of crankpin or journal hardening only, the minimum distance to the fillet is to be specified due to the tensile stress at the heat-affected zone as shown in **Fig. 5**.

4 If the hardness-versus-depth profile and residual stresses are not known or specified, one may assume the following:

(1) The hardness profile consists of two layers (*See* **Fig. 6**):

  (a) Constant hardness from the surface to the transition zone

  (b) Constant hardness from the transition zone to the core material

(2) Residual stresses in the hard zone of 200 *MPa* (compression)

(3) Transition-zone hardness as 90 % of the core hardness unless the local hardness drop is avoided

(4) Transition-zone maximum residual stresses (von Mises) of 300 *MPa* (tension)

5 If the crankpin or journal hardening ends close to the fillet, the influence of tensile residual stresses has to be considered. If the minimum distance between the end of the hardening and the beginning of the fillet is more than 3 *times* the maximum hardening depth, the influence may be disregarded.

Fig. 5 Residual Stresses Along the Surface of a Pin and Fillet

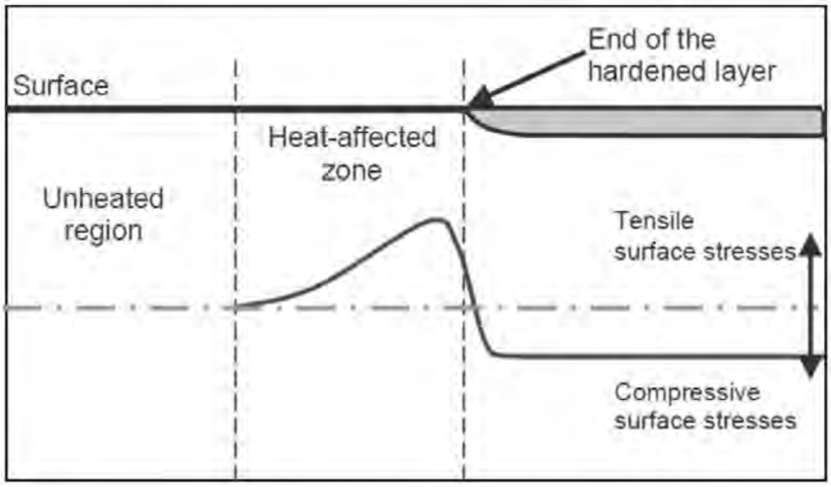
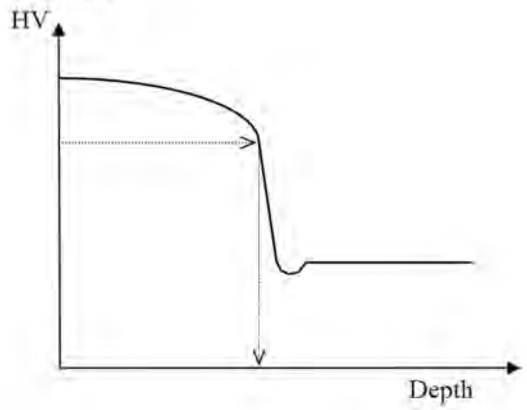


Fig. 6 Typical Hardness as a Function of Depth.



Note:
The arrows indicate the defined hardening depth. Note the indicated potential hardness drop at the transition to the core. This can be a weak point as local strength may be reduced and tensile residual stresses may occur.

4.2 Local Fatigue Strength

4.2.1 General

Induction-hardened crankshafts will suffer fatigue either at the surface or at the transition to the core.

4.2.2 Evaluation Based upon Fatigue Testing

1 The fatigue strengths, for both the surface and the transition zone, can be determined by fatigue testing of full-size cranks as described in **Appendix 2**.

2 In the case of a transition zone, the initiation of the fatigue can be either subsurface (i.e. below the hard layer) or at the surface where the hardening ends.

3 Tests made with the core material only will not be representative since the tensile residual stresses at the transition are lacking.

4.2.3 Evaluation Based upon Calculations

1 The surface fatigue strength can be determined empirically as follows:

$$\sigma_{Fsurface} = 400 + 0.5 \cdot (HV - 400) \ [MPa]$$

where

HV : surface Vickers hardness

The equation provides a conservative value, with which the fatigue strength is assumed to include the influence of the residual stress. The resulting value is valid for a working stress ratio of $R = -1$. It has to be noted also that the mean stress influence of induction-hardened steels may be significantly higher than that for QT steels.

2 The fatigue strength in the transition zone, without taking into account any possible local hardness drop, is to be determined by the following:

$$\sigma_{Ftransition,cpin} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \cdot \left[0.264 + 1.073 \cdot Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{X}}\right]$$

where

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O/2$ for oil bore outlet

For parameters *see* **1.4** of **Annex 2.3.1**

The influence of the residual stress is not included in the equation.

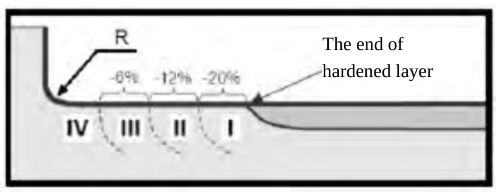
3 For the purpose of considering subsurface fatigue, below the hard layer, the disadvantage of tensile residual stresses has to be considered by subtracting 20 % from the value determined above. This 20 % is based on the mean stress influence of alloyed quenched and tempered steel having a residual tensile stress of 300 *MPa*.

4 When the residual stresses in **-3** are known to be lower, also smaller value of subtraction is to be used. For low-strength steels the percentage chosen is to be higher.

5 For the purpose of considering surface fatigue near the end of the hardened zone - i.e. in the heat-affected zone shown in the **Fig. 5** - the influence of the tensile residual stresses can be considered by subtracting a certain percentage, in accordance with **Table 2**, from the value determined by the above formula.

Table 2 The Influence of Tensile Residual Stresses at a Given Distance from the End of the Hardening towards the Fillet

Area	Distance from the end of the hardening towards the fillet	Ratio
I	0 to 1.0 of the max. hardening depth	20%
II	1.0 to 2.0 of the max. hardening depth	12%
III	2.0 to 3.0 of the max. hardening depth	6%
IV	3.0 or more of the max. hardening depth	0%



5.1 Nitriding

5.1.1 General

1 The hardness specification is to include the surface hardness range (min and max) and the minimum and maximum depth.

2 Only gas nitriding is considered.

3 The referenced Vickers hardness is considered to be HV 0.5.

4 The nitriding depth tN is defined as the depth to a hardness of 50 HV above the core hardness.

5 The hardening profile is to be specified all the way to the core.

6 If this is not known, it may be determined empirically via the following formula:

$$HV(t) = HV_{core} + \left(HV_{surface} - HV_{core}\right) \cdot \left(\frac{50}{HV_{surface} - HV_{core}}\right)^{\left(\frac{t}{t_N}\right)^2}$$

where:

t : The local depth

$HV(t)$: Hardness at depth t

HV_{core} : Core hardness (minimum)

$HV_{surface}$: Surface hardness (minimum)

tN : Nitriding depth as defined above (minimum)

5.2 Local Fatigue Strength

5.2.1 General

It is important to note that in nitrided crankshaft cases, fatigue is found either at the surface or at the transition to the core.

5.2.2 Evaluation Based on Fatigue Testing

The fatigue strength can be determined by tests as described in **Appendix 2**.

5.2.3 Evaluation Based on Calculations

1 Alternatively, the surface fatigue strength (principal stress) can be determined empirically and conservatively as follows:

$\sigma_{Fsurface} = 450MPa$

This is valid for a surface hardness of 600 *HV* or greater.

Note that this fatigue strength is assumed to include the influence of the surface residual stress and applies for a working stress ratio of $R = -1$.

2 The fatigue strength in the transition zone can be determined via the following formula:

$$\sigma_{Ftransition,cpin} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \cdot \left[0.264 + 1.073 \cdot Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{X}}\right]$$

where:

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O/2$ for oil bore outlet

Note that this fatigue strength is not assumed to include the influence of the residual stresses.

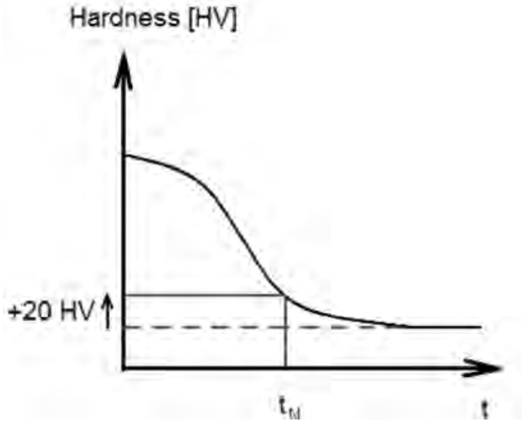
3 In contrast to induction-hardening the nitrited components have no such distinct transition to the core. Although the compressive residual stresses at the surface are high, the balancing tensile stresses in the core are moderate because of the shallow depth.

4 For the purpose of analysis of subsurface fatigue the disadvantage of tensile residual stresses in and below the transition zone may be even disregarded in view of this smooth contour of a nitriding hardness profile.

5 Although in principle the calculation is to be carried out along the entire hardness profile, it can be limited to a simplified approach of examining the surface and an artificial transition point. (*See* **Fig. 7**)

6 This artificial transition point can be taken at the depth where the local hardness is approximately 20 *HV* above the core hardness. In such a case, the properties of the core material are to be used. This means that the stresses at the transition to the core can be found by using the local SCF formulae mentioned in **3.2.2** or **3.3.2** when inserting $t = 1.2tN$.

Fig. 7 Sketch of the Location for the Artificial Transition Point in the Depth Direction



6.1 Cold Forming

6.1.1 General

1 The advantage of stroke peening or cold rolling of fillets is the compressive residual stresses introduced in the high-loaded area.

2 The fatigue strength has to be determined by fatigue testing (*See* also **Appendix 2**). Such testing is normally carried out as four-point bending, with a working stress ratio of $R = -1$.

3 From these results, the bending fatigue strength - surface - or subsurface-initiated depending on the manner of failure - can be determined and expressed as the representative fatigue strength for applied bending in the fillet.

4 In comparison to bending, the torsion fatigue strength in the fillet may differ considerably from the ratio $\sqrt{3}$ (utilized by the von Mises criterion). The forming-affected depth that is sufficient to prevent subsurface fatigue in bending, may still allow subsurface fatigue in torsion. Another possible reason for the difference in bending and torsion could be the extension of the highly stressed area.

5 The results obtained in a full-size crank test can be applied for another crank size provided that the base material (alloyed Q+T) is of the similar type and that the forming is done so as to obtain the similar level of compressive residual stresses at the surface as well as through the depth. This means that both the extension and the depth of the cold forming are to be proportional to the fillet radius.

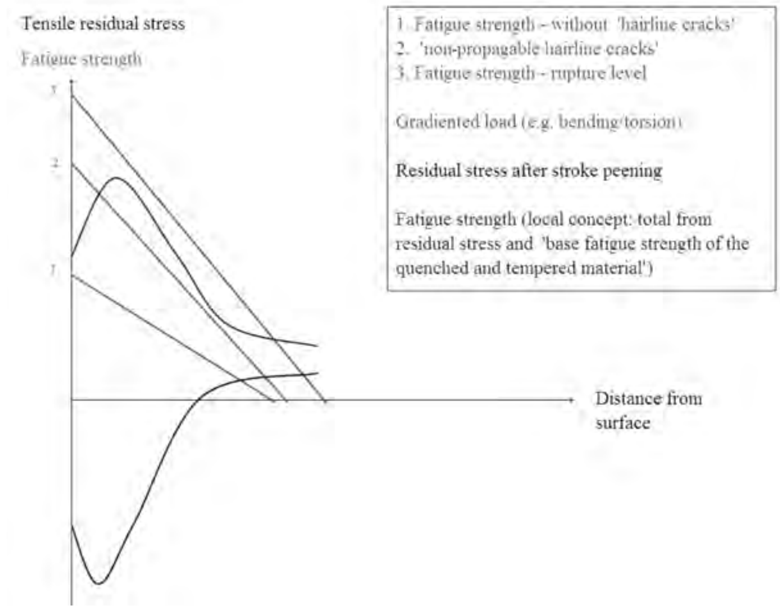
6.1.2 Stroke Peening by Means of a Ball

1 If both bending and torsion fatigue strengths have been investigated and differ from the ratio $\sqrt{3}$, the von Mises criterion is to be excluded.

2 If only bending fatigue strength has been investigated, the torsional fatigue strength is to be assessed conservatively. If the bending fatigue strength is concluded to be x % above the fatigue strength of the non-peened material, the torsional fatigue strength is not to be assumed to be more than 2/3 of x % above that of the non-peened material.

3 As a result of the stroke peening process the maximum of the compressive residual stress is found in the subsurface area. Therefore, depending on the fatigue testing load and the stress gradient, it is possible to have higher working stresses at the surface in comparison to the local fatigue strength of the surface. Because of this phenomenon small cracks may appear during the fatigue testing, which will not be able to propagate in further load cycles and/or with further slight increases of the testing load because of the profile of the compressive residual stress. Put simply, the high compressive residual stresses below the surface "arrest" small surface cracks. (*See* 2. in **Fig. 8**)

Fig. 8 Working and Residual Stresses below the Stroke-peened Surface.



Note:
Straight lines 1…3 represent different possible load stress gradients.

4 In fatigue testing with full-size crankshafts these small "hairline cracks" is to not be considered to be the failure crack. The crack that is technically the fatigue crack leading to failure, and that therefore shuts off the test-bench, is to be considered for determination of the failure load level. This also applies if induction-hardened fillets are stroke-peened.
5 In order to improve the fatigue strength of induction-hardened fillets it is possible to apply the stroke peening process in the crankshafts' fillets after they have been induction-hardened and tempered to the required surface hardness. If this is done, it might be necessary to adapt the stroke peening force to the hardness of the surface layer and not to the tensile strength of the base material.
6 The effect on the fatigue strength of induction hardening and stroke peening the fillets is to be determined by a full-size crankshaft test.

6.1.3 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved by applying stroke peening, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:
(1) Ball size relative to fillet radius within ± 10 % in comparison to the tested crankshaft
(2) At least the same circumferential extension of the stroke peening
(3) Angular extension of the fillet contour relative to fillet radius within ± 15 % in comparison to the tested crankshaft and located to cover the stress concentration during engine operation
(4) Similar base material, e.g. alloyed quenched and tempered
(5) Forward feed of ball of the same proportion of the radius
(6) Force applied to ball proportional to base material hardness (if different)
(7) Force applied to ball proportional to square of ball radius

6.1.4 Cold Rolling

1 The fatigue strength can be obtained by means of full-size crank tests or by empirical methods, if these are applied so as to be on the safe side.

2 If both, bending and torsion fatigue strengths have been investigated, and differ from the ratio $\sqrt{3}$, the von Mises criterion is to be excluded.

3 If only bending fatigue strength has been investigated, the torsional fatigue strength is to be assessed conservatively. If the bending fatigue strength is concluded to be $x\ \%$ above the fatigue strength of the non-rolled material, the torsional fatigue strength is to not be assumed to be more than 2/3 of $x\ \%$ above that of the non-rolled material.

6.1.5 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved applying cold rolling, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:

(1) At least the same circumferential extension of cold rolling

(2) Angular extension of the fillet contour relative to fillet radius within $\pm 15\ \%$ in comparison to the tested crankshaft and located to cover the stress concentration during engine operation

(3) Similar base material, e.g. alloyed quenched and tempered

(4) Roller force to be calculated so as to achieve at least the same relative (to fillet radius) depth of treatment

Appendix 4 has been added as follows.

Appendix 4 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE OIL BORE OUTLETS OF CRANKSHAFTS THROUGH UTILISATION OF THE FINITE ELEMENT METHOD

1.1 General

The objective of the analysis described in this appendix is to substitute the analytical calculation of the stress concentration factor (SCF) at the oil bore outlet with suitable finite element method (FEM) calculated figures. The former method is based on empirical formulae developed from strain gauge readings or photo-elasticity measurements of various round bars. In cases where these formulae are outside their applicable scope, the FEM-based method is to be used.

The SCF calculated in accordance with the rules set forth in this appendix is defined as the ratio of FEM-calculated stresses to nominal stresses calculated analytically. In use in connection with the present method in **Annex 2.3.1**, principal stresses are to be calculated.

The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.

It is advisable to check the element accuracy of the FE solver in use, e.g. by modelling a simple geometry and comparing the FEM-obtained stresses with the analytical solution.

A boundary element method (BEM) approach may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and assumptions for building of the FE-model are presented in **2.1.1**. The final FE-model is to meet one of the criteria in **2.2**.

2.1.1 Element Mesh Recommendations

For the mesh quality criteria to be met, construction of the FE model for the evaluation of stress concentration factors in accordance with the following recommendations is advised:

(1) The model consists of one complete crank, from the main bearing centre line to the opposite side's main bearing centre line.

(2) The following element types are used in the vicinity of the outlets:
- (a) 10-node tetrahedral elements
- (b) 8-node hexahedral elements
- (c) 20-node hexahedral elements

(3) The following mesh properties for the oil bore outlet are used:
- (a) Maximum element size $a = r/4$ through the entire outlet fillet as well as in the bore direction (if 8-node hexahedral elements are used, even smaller elements are required for meeting of the quality criterion)
- (b) Recommended manner for element size in the fillet depth direction:
  - i) First layer's thickness equal to element size of a
  - ii) Second layer's thickness equal to element size of $2a$
  - iii) Third-layer thickness equal to element size of $3a$

(4) The rest of the crank is to be suitable for numeric stability of the solver

(5) Drillings and holes for weight reduction have to be modelled

(6) Submodeling may be used as long as the software requirements are fulfilled.

2.1.2 Material

1 Material properties applied to steels as follows.

Young's Modulus : $E = 2.05 \cdot 10^5 MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters have to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

If the actual element mesh does not fulfil any of the following criteria in the area examined for SCF evaluation, a second calculation, with a finer mesh is to be performed.

2.2.1 Principal-stresses Criterion

The quality of the mesh is to be assured through checking of the stress component normal to the surface of the oil bore outlet radius. With principal stresses σ_1, σ_2 and σ_3 the following criterion is to be met:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0.03 \cdot \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

2.2.2 Averaged/Unaveraged-stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than 5 % from the 100 % averaged nodal stress results at this node at the location examined.

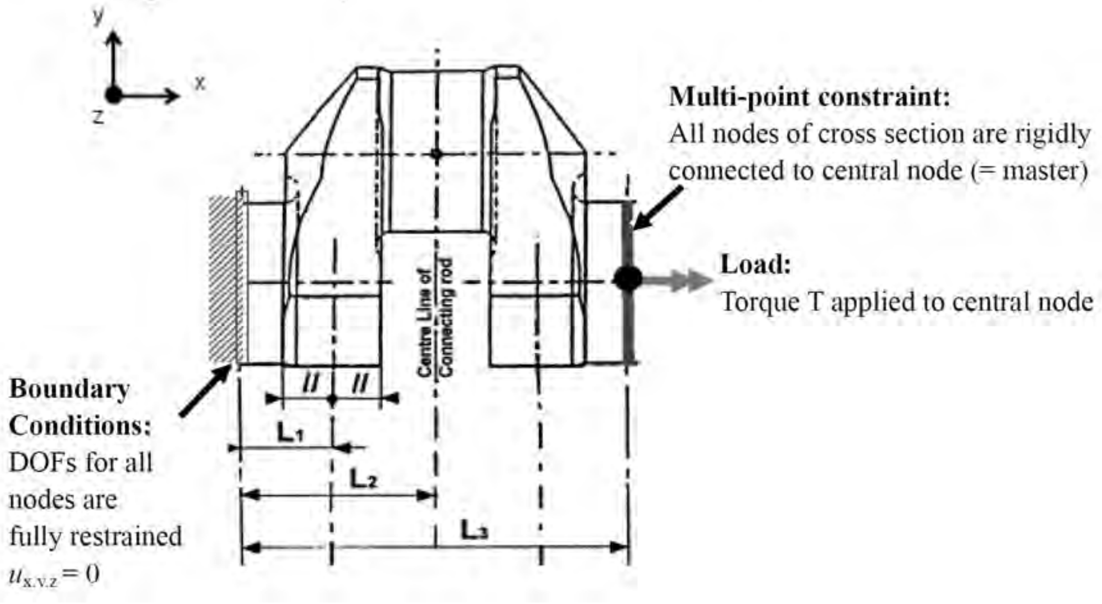
3.1 Load Cases and Assessment of Stress

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 1** where the torque is applied to the central node located at the crankshaft axis.

Fig. 1 Boundary and Load Conditions for the Torsion Load Case



2 For all nodes in an oil bore outlet, the principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_T = \frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\tau_N}$$

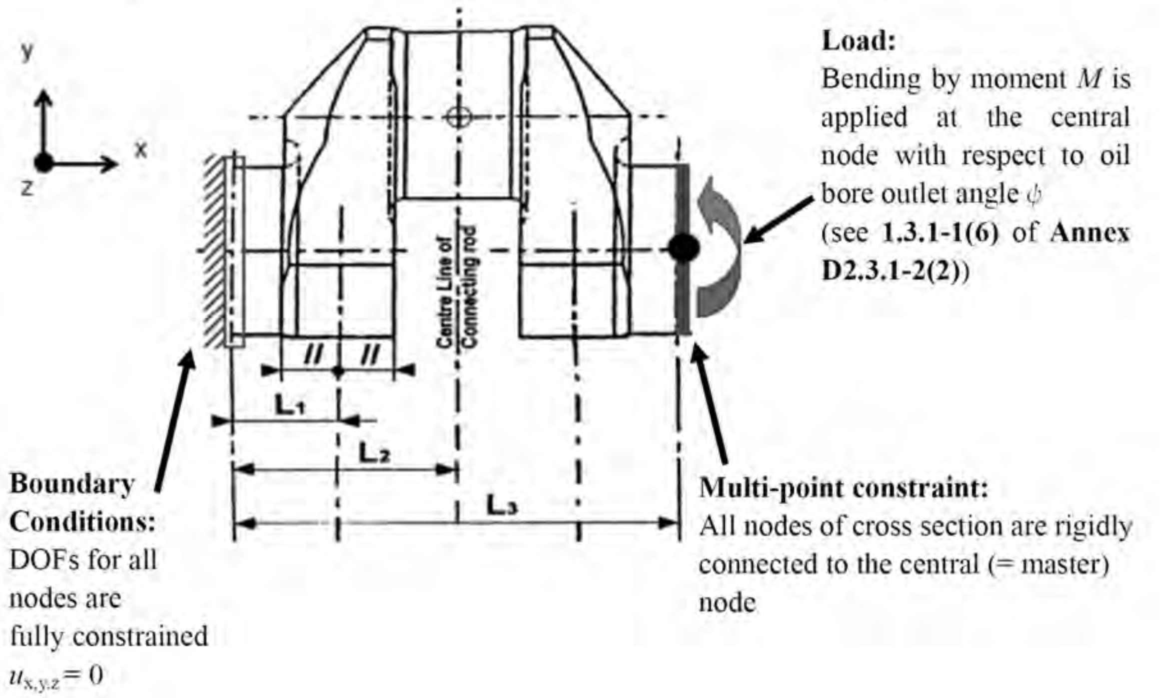
where the nominal torsion stress τ_N referred to the crankpin is calculated as follows (for W_P *see* **1.3.2** of **Annex 2.3.1**) :

$$\tau_N = \frac{T}{W_P}$$

3.1.2 Bending

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 2** where the bending moment is applied to the central node located at the crankshaft axis.

Fig. 2 Boundary and Load Conditions for the Pure Bending Load Case



2 For all nodes in the oil bore outlet, principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_B = \frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\sigma_N}$$

where the nominal bending stress σ_N referred to the crankpin is calculated as follows (for W_e *see* **1.3.2** of **Annex 2.3.1**):

$$\sigma_N = \frac{M}{W_e}$$

EFFECTIVE DATE AND APPLICATION (Amendment 1-5)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to crankshafts for which the application for approval is submitted to the Society before the effective date.

Amendment 1-6

Chapter 5 POWER TRANSMISSION SYSTEMS

5.2 Materials and Construction

5.2.1 Materials

Sub-paragraph -1(1) has been amended as follows.

1 Materials used for the following components (hereinafter referred to as "the principal components of the power transmission system") are to comply with the requirements in **Part K**.

(1) Power transmission shafts (including power take-off (PTO) shafts) and gears
(2) Power transmission parts of couplings
(3) Power transmission parts of clutches
(4) Coupling bolts

2 (Omitted)

5.3 Strength of Gears

Paragraph 5.3.1 has been amended as follows.

5.3.1 Application*

The requirements in **5.3** apply to external tooth cylindrical gears having an involute tooth profile. All other gears are to be as deemed appropriate by the Society. In addition, enclosed gear strength calculations are to be in accordance with **Annex 5.3.1 "Calculation of Strength of Enclosed Gears"**.

Paragraph 5.3.5 has been amended as follows.

5.3.5 Detailed Evaluation for Strength*

Special consideration will be given to the gearing devices, notwithstanding the requirements in **5.3.3** and **5.3.4**, provided that detailed data and calculations on their strength are submitted to the Society and considered appropriate. In addition, the wording "detailed data and calculations on their strength" means calculations based on **Annex 5.3.1 "Calculation of Strength of Enclosed Gears"**.

Annex 5.3.1 has been added as follows.

Annex 5.3.1 CALCULATION OF STRENGTH OF ENCLOSED GEARS

1.1 Application and Basic Principles

1.1.1 Application

This annex applies to enclosed gears used for transmission systems which transmit power from main propulsion machinery and prime movers driving generators and essential auxiliaries (excluding auxiliary machinery for specific use, etc., hereinafter the same in this annex).

1.1.2 Basic Principles

The gear strength calculation methods specified in this annex deal with surface durability (pitting) and tooth root bending strength. All influence factors related to strength are defined regarding their physical interpretation. Some of these factors are to be determined either by gear geometry etc,. Other factors are to be approximated according to methods deemed acceptable by the Society.

1.2 Symbols and Units

The main symbols introduced in this annex are listed below.

a : center distance (mm)
b : common facewidth (mm)
$b_{1,2}$: facewidth of pinion , wheel (mm)
d : reference diameter (mm)
$d_{1,2}$: reference diameter of pinion, wheel (mm)
$d_{a1,2}$: tip diameter of pinion, wheel (mm)
$d_{b1,2}$: base diameter of pinion, wheel (mm)
$d_{f1,2}$: root diameter of pinion, wheel (mm)
$d_{w1,2}$: working diameter of pinion, wheel (mm)
F_t : nominal tangential load (N)
h : tooth depth (mm)
m_n : normal module (mm)
m_t : transverse module (mm)
$n_{1,2}$: rotational speed of pinion, wheel (rpm)
P : maximum continuous power transmitted by the gear set (kW)
$T_{1,2}$: torque in way of pinion, wheel (Nm)
u : gear ratio
v : linear speed at pitch diameter (m/s)
$x_{1,2}$: addendum modification coefficient of pinion, wheel
z : number of teeth
$z_{1,2}$: number of teeth of pinion, wheel
z_n : virtual number of teeth
α_n : normal pressure angle at reference cylinder (°)
α_t : transverse pressure angle at reference cylinder (°)
α_{tw} : transverse pressure angle at working pitch cylinder (°)

β : helix angle at reference cylinder (°)

β_b : helix angle at base cylinder (°)

ε_α : transverse contact ratio

ε_β : overlap contact ratio

ε_γ : total contact ratio

σ_H : contact stress at the operating pitch point or at the inner point of single pair contact (N/mm^2)

σ_{HO} : basic value of contact stress (N/mm^2)

K_A : application factor

K_γ : load sharing factor

K_V : dynamic factor

$K_{H\alpha}$: transverse load distribution factor for contact stress

α_{Pn} : normal pressure angle of basic rack for cylindrical gear (°)

h_{fp} : dedendum of basic rack for cylindrical gear (mm)

$K_{H\beta}$: face load distribution factor for contact stress

σ_{HP} : permissible contact stress (N/mm^2)

σ_{Hlim} : endurance limit for contact stress (N/mm^2)

Z_N : life factor for contact stress

Z_L : lubricant factor

Z_V : speed factor

Z_R : roughness factor

Z_W : hardness ratio factor

Z_X : size factor for contact stress

S_H : safety factor for contact stress

Z_B : single pair mesh factor for pinion

Z_D : single pair mesh factor for wheel

Z_H : zone factor

Z_E : elasticity factor ($\sqrt{N/mm^2}$)

Z_ε : contact ratio factor

Z_β : helix angle factor for contact stress

σ_F : tooth root bending stress (N/mm^2)

Y_F : tooth form factor

h_F : bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact (mm)

S_{FN} : tooth root chord in the critical section (mm)

α_{SFN} : pressure angle at the outer point of single tooth pair contact in the normal section (°)

Y_S : stress correction factor

ρ_F : root fillet radius in the critical section (mm)

ρ_{fp} : root fillet radius of the basic rack for cylindrical gears (mm)

S_{pr} : residual fillet undercut (mm)

Y_β : helix angle factor for tooth root bending stress

Y_B : rim thickness factor

s_R : rim thickness of gears (*mm*)
h : tooth height (*mm*)
Y_{DT} : deep tooth factor
$K_{F\alpha}$: transverse load distribution factor for tooth root bending stress
$K_{F\beta}$: face load distribution factor for tooth root bending stress
σ_{FP} : permissible tooth root bending stress (N/mm^2)
σ_{FE} : bending endurance limit (N/mm^2)
Y_N : life factor for tooth root bending stress
Y_d : design factor
$Y_{\delta\, relT}$: relative notch sensitivity factor
q_s : notch parameter
ρ' : slip-layer thickness (*mm*)
$Y_{R\, relT}$: relative surface factor
Y_X : size factor for tooth root bending stress
S_F : safety factor for tooth root bending stress

1.3 Geometrical Definitions

In the case of internal gearing z_2, a, d_2, d_{a2}, d_{b2} and d_{w2} are negative. The pinion is defined as the gear with the smaller number of teeth; therefore, the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity.

$$u = \frac{z_2}{z_1} = \frac{d_{w2}}{d_{w1}} = \frac{d_2}{d_1}$$

In the case of external gears, u is positive. In the case of internal gears, u is negative. In the equation of surface durability, b is the common facewidth on the pitch diameter. In the equation of the tooth root, bending stress b_1 or b_2 are the facewidths at their respective tooth roots. In any case, b_1 and b_2 are not to be taken as greater than b by more than one module (m_n) on either side. The common facewidth b may be used also in the equation of teeth root bending stress if either significant crowning or end relief has been adopted.

$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta}$$

$$\tan\beta_b = \tan\beta\cos\alpha_t$$

$$d_{1,2} = \frac{z_{1,2}m_n}{\cos\beta}$$

$$d_{b1,2} = d_{1,2}\cos\alpha_t$$

$$\left.\begin{aligned} d_{w1} &= \frac{2a}{u+1} \\ d_{w2} &= \frac{2au}{u+1} \end{aligned}\right\}\text{where } a = 0.5(d_{w1} + d_{w2})$$

$$z_{n1,2} = \frac{z_{1,2}}{\cos^2\beta_b \cdot \cos\beta}$$

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

$$\text{inv}\alpha = \tan\alpha - \frac{\pi\alpha}{180};\ \alpha(°)$$

$$\mathrm{inv}\alpha_{tw} = \mathrm{inv}\alpha_t + 2\tan\alpha_n \frac{x_1 + x_2}{z_1 + z_2}$$

or

$$\cos\alpha_{tw} = \frac{m_t(z_1 + z_2)}{2a}\cos\alpha_t$$

$$\varepsilon_\alpha = \frac{0.5\sqrt{d_{a1}^2 - d_{b1}^2} \pm 0.5\sqrt{d_{a2}^2 - d_{b2}^2} - a\sin\alpha_{tw}}{\pi m_t \cos\alpha_t}$$

A positive sign is used for external gears, a negative sign for internal gears.

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

In the case of double helix gears, *b* is to be taken as the width of one helix.

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

$$v = \frac{\pi d_{1,2} n_{1,2}}{60 \cdot 10^3}$$

1.4 Nominal Tangential Load, F_t

Nominal tangential loads, F_t, which are tangential to cylinders and perpendicular to planes are to be calculated directly from the maximum continuous power transmitted by gear sets using the following equations:

$$T_{1,2} = \frac{30 \cdot 10^3 P}{\pi n_{1,2}}$$

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

1.5 Loading Factors

1.5.1 Application Factor, K_A

1 The application factor, K_A, accounts for dynamic overloads from source external to the gearing. K_A for gears designed for infinite lifespans is defined as the ratio between maximum repetitive cyclic torques applied to gear sets and nominal rated torques. Nominal rated torque is defined by rated power and speed and is the torque used in rating calculations. This factor mainly depends on the following:

(1) The characteristics of driving and driven machines;
(2) The ratio of masses;
(3) The type of couplings;
(4) Operating conditions (over speed, changes in propeller load conditions, etc.)

2 In cases where drive systems are operating at level near their critical speed, a careful analysis of conditions is to be made. K_A is to be determined either by direct measurements or by a system analysis that is acceptable to the Society. In cases where values determined in such ways cannot be provided, the following values may be used:

(1) Main propulsion

K_A = 1.00 (reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)

= 1.30 (reciprocating internal combustion engines with high elasticity couplings)

= 1.50 (reciprocating internal combustion engines with other couplings)

However, in cases where vessels using reduction gears are affixed with **Ice Class** Notation, as required in **8.6, Part I of the Rules**.

(2) Auxiliary gears

K_A = 1.00 (electric motors, reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)

= 1.20 (reciprocating internal combustion engines with high elasticity couplings)

= 1.40 (reciprocating internal combustion engines with other couplings)

1.5.2 Load Sharing Factor, K_γ

The load sharing factor, K_γ, accounts for the maldistribution of loads in multiple path transmissions (dual tandems, epicyclics, double helixes, etc.). K_γ is defined as the ratio between those maximum loads through actual paths and those evenly distributed loads. This factor mainly depends on the accuracy and the flexibility of the branches. K_γ is to be determined by measurements or by system analysis. In cases where values determined in such ways cannot be provided, the following values can be used with respect to epicyclic gears:

K_γ = 1.00 (up to 3 planetary gears)

= 1.20 (4 planetary gears)

= 1.30 (5 planetary gears)

= 1.40 (6 planetary gears and over)

1.5.3 Internal Dynamic Factor, K_V

1 The internal dynamic factor, K_V, accounts for those internally generated dynamic loads due to vibrations of pinions and wheels against each other. K_V is defined as the ratio between those maximum loads which dynamically act on tooth flanks and maximum externally applied loads ($F_t K_A K_V$). This factor mainly depends on the following:

(1) Transmission errors depending on pitch and profile errors;

(2) Masses of pinions and wheels;

(3) Gear mesh stiffness variations as gear teeth pass through meshing cycles;

(4) Transmitted loads including application factors;

(5) Pitch line velocities;

(6) Dynamic unbalance of gears and shafts;

(7) Shaft and bearing stiffness;

(8) Damping characteristics of gear systems.

2 The internal dynamic factor, K_V, is to be calculated as follows; however, this method is to be applied only to cases where all of the following conditions **(1)** to **(4)** are satisfied:

(1) Running speeds in the following subcritical ranges:

$$\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 10\ (m/s)$$

(2) $\beta = 0°$ (In the case of spur gears)

$\beta \leq 30°$ (In the case of helical gears)

(3) pinion with relatively low number of teeth:

$z_1 < 50$

(4) solid disc wheels or heavy steel gear rim

This method may be applied to all types of gears, if $\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3\ (m/s)$, as well as to helical gears where $\beta > 30°$. For gears other than the above, reference is to be made to Method B outlined in the reference standard *ISO* 6336-1:2019.

(a) For those helical gears with an overlap ratio $\geq$ unity and spur gears, the value of K_V is to be determined as follows:

$$K_V = 1 + \left(\frac{K_1}{K_A \frac{F_t}{b}} + K_2 \right) \cdot \frac{v \cdot z_1}{100} K_3 \sqrt{\frac{u^2}{1+u^2}}$$

K_1 : Factor specified in **Table 5.3-1.**

K_2 : Factors for all *ISO* accuracy grades. Values are as follows:

= 0.0193 (In the case of spur gears)

= 0.0087 (In the case of helical gears)

K_3 : Values are to be calculated as follows:

$$= 2.0 \left(\frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} \leq 0.2 \right)$$

$$= 2.071 - 0.357 \cdot \frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} \left(\frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} > 0.2 \right)$$

If $K_A F_t / b$ is less than 100 *N/mm*, this value is assumed to be equal to 100 *N/mm*.

(b) In the case of helical gears with an overlap ratio < unity, the value of K_V is to be obtained by means of linear interpolation as follows:

$$K_V = K_{V2} - \varepsilon_\beta (K_{V2} - K_{V1})$$

K_{V1} : Values for helical gears specified in accordance with **(a)**

K_{V2} : Values for spur gears specified in accordance with **(a)**

In the case of mating gears with different grades of accuracy, the grade corresponding to the lower accuracy is to be used.

Table 5.3-1 Values of K_1

Type of gears	*ISO* grades of accuracy					
	3	4	5	6	7	8
Spur gears	2.1	3.9	7.5	14.9	26.8	39.1
Helical gears	1.9	3.5	6.7	13.3	23.9	34.8

Notes:

ISO accuracy grades according to *ISO* 1328-2:2020.

1.5.4 Face Load Distribution Factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress, and $K_{F\beta}$ for tooth root bending stress account for the effects of the non-uniform distribution of loads across facewidths. $K_{H\beta}$ is defined as the ratio between the maximum load per unit facewidth and the mean load per unit facewidth, and $K_{F\beta}$ is defined as the ratio between the maximum bending stress at tooth root per unit facewidth and the mean bending stress at tooth root per unit facewidth.

The mean bending stress at tooth root relates to the considered facewidth b_1 and b_2 respectively. $K_{F\beta}$ can be expressed as a function of the factor, $K_{H\beta}$. $K_{H\beta}$ and $K_{F\beta}$ mainly depend on the following:

(1) Gear tooth manufacturing accuracy;
(2) Errors in mounting due to bore errors;
(3) Bearing clearances;
(4) Wheel and pinion shaft alignment errors;

(5) Elastic deflections of gear elements, shafts, bearings, housing, and foundations which support the gear elements;
(6) Thermal expansion and distortion due to operating temperature;
(7) Compensating design elements (tooth crowning, end relief, etc.).

The value for $K_{H\beta}$ is to be determined as follows:

if $\frac{F_{\beta y}C_\gamma b}{2F_t K_A K_\gamma K_V} \geq 1$

then $K_{H\beta} = \sqrt{\frac{2F_{\beta y}C_\gamma b}{F_t K_A K_\gamma K_V}}$

if $\frac{F_{\beta y}C_\gamma b}{2F_t K_A K_\gamma K_V} < 1$

then $K_{H\beta} = 1 + \frac{F_{\beta y}C_\gamma b}{2F_t K_A K_\gamma K_V}$

C_γ : tooth stiffness parameter ($N/(mm \cdot \mu m)$)

The value for C_γ is to be calculated as follows:

$$C_\gamma = 0.8 C_{th} C_R C_B \cos\beta(0.75\varepsilon_\alpha + 0.25)$$

$$C_{th} = \frac{1}{q}$$

$$q = 0.04723 + \frac{0.15551}{z_{n1}} - \frac{0.25791}{z_{n2}} - 0.00635x_1 - \frac{0.11654}{z_{n1}}x_1$$
$$-0.00193x_2 - \frac{0.24188}{z_{n2}}x_2 + 0.00529x_1^2 + 0.00182x_2^2$$

$C_R = 1$ (In the case of solid disc gears)

$C_R = 1 + \frac{\ln\left(\frac{b_s}{b}\right)}{5e^{\left(\frac{S_R}{5m_n}\right)}}$ (In the case of non-solid disc gears)

b_s : thickness of central web (mm)

S_R : average thickness of rim (mm)

However, in cases where for non-solid disc gears of $\frac{b_s}{b} < 0.2$ or $\frac{S_R}{m_n} < 1.0$, the value for C_R is to be determined by the Society on a case by case basis.

C_B is a basic rack coefficient that accounts for the deviations between the actual basic rack profile of gears and their standard basic rack profile. The value for C_B is to be calculated as follows:

$$C_B = \left[1 + 0.5\left(1.2 - \frac{h_{fp}}{m_n}\right)\right][1\text{-}0.02(20^\circ - \alpha_{Pn})]$$

$F_{\beta y}$ =effective equivalent misalignment (μm). The value for $F_{\beta y}$ is to be calculated as follows:

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

In the case of gears that are not surface hardened

$$y_\beta = \frac{320}{\sigma_{Hlim}} F_{\beta x}$$

However, the following conditions are to be satisfied.

$y_\beta \le F_{\beta x}$

$y_\beta \le 25600/\sigma_{Hlim}$ (5<v<10 m/sec)

$y_\beta \le 12800/\sigma_{Hlim}$ (10 m/sec<v)

In the case of surface hardened gears

$y_\beta = 0.15F_{\beta x}$

However, $y_\beta \le 6.0$ (μm) is to be satisfied.

$F_{\beta x}$: original effective equivalent misalignment (μm), $F_{\beta x}$ is to be calculated as follows:

$F_{\beta x} = 1.33 f_{sh} + f_{ma}$

f_{sh}: takes into account the components of equivalent misalignment resulting from bending and twisting of pinion and pinion shaft, f_{sh} is to be calculated as follows (μm):
In the case of gears without crowning or end relief

$$f_{sh} = 0.023\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with end relief

$$f_{sh} = 0.016\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with crowning

$$f_{sh} = 0.012\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with helix angle modification

$f_{sh} = 0$

However, in all cases f_{sh} is not to be taken as value less than that calculated by the following expressions:

$$0.005\frac{F_t K_A K_\gamma K_V}{b} \text{ (In the case of spur gears)}$$

or

$$0.010\frac{F_t K_A K_\gamma K_V}{b} \text{ (In the case of helical gears)}$$

γ = pinion ratio factor. The value for γ is to be calculated as follows:

$$\gamma = \left[\left|0.7 + K'\frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \text{ (In the case of spur and helical gears)}$$

$$\gamma = 2\left[\left|1.2 + K'\frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \text{ (In the case of double helical gears)}$$

K', ℓ and S are constant factors used for the calculation of the pinion ratio factor, γ, the bearing span and the distance between mid-plane of pinion and middle of such bearing spans, respectively. Values for K' are given in **Table 5.5-1**.

f_{ma} = the misalignment resulting from manufacturing errors (μm). The value for f_{ma} is to be calculated as follows:

$f_{ma} = 1.0F_\beta$ (In the case of the assembly of gears without any modification or adjustment)

$= 0.7F_\beta$ (In the case of gear pairs with well-designed end relief)

$= 0.5F_\beta$ (In the case of gear pairs with means for adjustment or with helix modifications or suitably crowned)

F_β : tolerance on total helix deviation (μm)

The value for $K_{F\beta}$ is to be determined as follows:

(1) In cases where the hardest contact is at the end of the facewidth, $K_{F\beta}$ is to be calculated as follows:

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

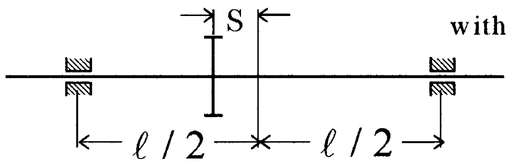
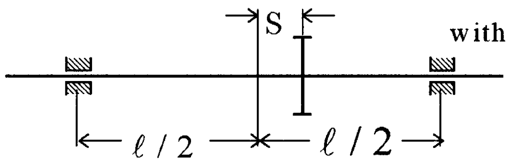
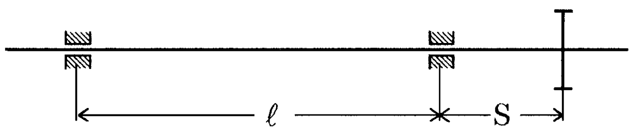
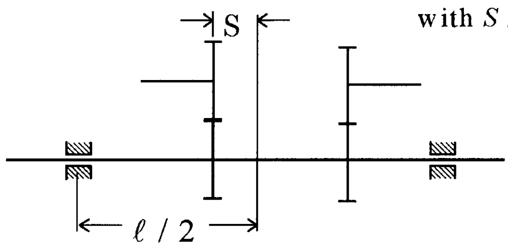
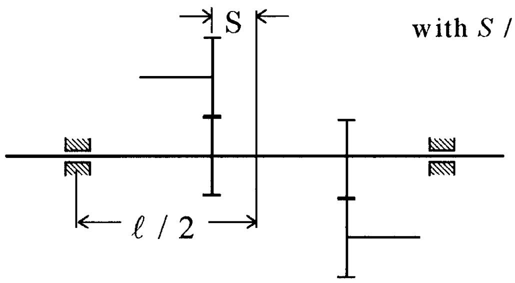
$$N = \frac{\left(\frac{b}{h}\right)^2}{1 + \frac{b}{h} + \left(\frac{b}{h}\right)^2}$$

b/h= facewidth/tooth height ratio, the smaller of b_1/h_1 or b_2/h_2. In the case of double helical gears, the facewidth of only one is to be used. However, in cases where b/h<3.0, b/h is to be taken as 3.0.

(2) In cases of gears where the ends of the facewidth are lightly loaded or unloaded (end relief or crowning), the value for $K_{F\beta}$ is to be calculated as follows:

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

Table 5.5-1 Values of K'

	K'	
	with stiffening[1)]	without stiffening[2)]
Input  with $S / \ell < 0.3$	0.48	0.80
Input  with $S / \ell < 0.3$	-0.48	-0.80
Input  with $S / \ell < 0.5$	1.33	1.33
 with $S / \ell < 0.3$	-0.36	-0.60
 with $S / \ell < 0.3$	-0.6	-1.0

Notes:

1) In cases where $d_1/d_{sh} \geq 1.15$, stiffening is assumed.

2) In cases where $d_1/d_{sh} < 1.15$, or where the pinion slides on a shaft or is shrink fitted, no stiffening is assumed.

However, d_{sh} is the external diameter of a solid shaft equivalent to the actual one in bending deflection (mm)

1.5.5 Transverse Load Distribution Factors, $K_{H\alpha}$ and $K_{F\alpha}$

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, account for the effects of pitch and profile errors on the transversal load distribution between two or more pairs of teeth in mesh. $K_{H\alpha}$ and $K_{F\alpha}$ mainly depend on the following:

(1) Total mesh stiffness;

(2) Total tangential loads, $F_t\, K_A\, K_\gamma\, K_V\, K_{H\beta}$

(3) Base pitch errors

(4) Tip relief

(5) Running-in allowances

$K_{H\alpha}$ and $K_{F\alpha}$ are to be determined as follows:

$$K_{H\alpha} = K_{F\alpha} = \frac{\varepsilon_\gamma}{2}\left(0.9 + 0.4\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_t K_A K_\gamma K_V K_{H\beta}}\right)\ (\varepsilon_\gamma \le 2)$$

$$K_{H\alpha} = K_{F\alpha} = 0.9 + 0.4\sqrt{\frac{2(\varepsilon_\gamma-1)}{\varepsilon_\gamma}}\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_t K_A K_\gamma K_V K_{H\beta}}\ (\varepsilon_\gamma > 2)$$

however

$$1.0 \le K_{H\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_\varepsilon^2}$$

$$1.0 \le K_{F\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha Y_\varepsilon}$$

where

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_{\alpha n}}$$

The value for $\varepsilon_{\alpha n}$ is the same as that specified in **1.7.2**.

All symbols used in the equation to determine $K_{H\alpha}$ and $K_{F\alpha}$, except for y_α and f_{pb}, are the same as those used in the equation for determining $K_{H\beta}$.

y_α and f_{pb} are to be calculated as follows:

$$y_\alpha = \frac{160}{\sigma_{Hlim}} f_{pb} \text{ (In the case of through hardened gears)}$$

$$= 0.075 f_{pb} \qquad \text{(In the case of surface hardened gears)}$$

f_{pb} is to be taken as the larger value of base pitch deviation of pinions or wheels (μm)

1.6 Surface Strength

1.6.1 Equation

The criterion for surface strength is based on the Hertz pressure on operating pitch points or at inner points of single pair contacts. This criterion, as given by the following equation, is that contact stress σ_H is to be equal to or less than permissible contact stress σ_{HP}.

$$\sigma_H = \sigma_{HO}\sqrt{K_A K_\gamma K_V K_{H\alpha} K_{H\beta}} \le \sigma_{HP}$$

where

σ_{HO} is basic value of contact stress for pinions and wheels (N/mm^2).

1.6.2 Equations for Basic Contact Stress

1 Basic contact stresses for pinions and wheels are to be calculated as follows:

$$\sigma_{HO} = Z_B Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} \text{ (In the case of pinions)}$$

$$= Z_D Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} \text{ (In the case of wheels)}$$

2 Single Pair Mesh Factors Z_B and Z_D

The single-pair mesh factors, Z_B for pinions and Z_D for wheels, account for the influence on contact stress of tooth flank curvatures at inner points of single pair contacts. These factors transform those contact stresses determined at pitch points to contact stresses considering flank curvatures at inner points of single pair contacts. Z_B and Z_D are to be determined as follows:

(1) In the case of spur gears, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater.

$$M_1 = \frac{\tan\alpha_{tw}}{\left\{\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - \frac{2\pi}{z_1}\right]\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_2}\right]\right\}^{\frac{1}{2}}}$$

(2) In the case of helical gears with $\varepsilon_\beta \geq 1$, the value for Z_B is to be taken as 1.0;

(3) In the case of helical gears with $\varepsilon_\beta > 1$, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater:

$$Z_B = M_1 - \varepsilon_\beta(M_1 - 1)$$

(4) In the case of spur gears, the value for Z_D is to be taken as 1.0 or as follows, whichever is greater.

$$M_2 = \frac{\tan\alpha_{tw}}{\left\{\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - \frac{2\pi}{z_2}\right]\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_1}\right]\right\}^{\frac{1}{2}}}$$

(5) In the case of helical gears with $\varepsilon_\beta \geq 1$, Z_D is to be taken as 1.0.

(6) In the case of helical gears with $\varepsilon_\beta < 1$, Z_D is to be taken as 1.0 or as follows, whichever is greater.

$$Z_D = M_2 - \varepsilon_\beta(M_2 - 1)$$

(7) In the case of internal gears, Z_D is to be taken as 1.0.

3 Zone Factor, Z_H

The zone factor, Z_H, accounts for the influence on Hertzian pressure of tooth flank curvatures at pitch points and relates those tangential forces at reference cylinders to those normal forces at pitch cylinders. Z_H is to be calculated as follows:

$$Z_H = \sqrt{\frac{2\cos\beta_b}{\cos^2\alpha_t \tan\alpha_{tw}}}$$

4 Elasticity Factor, Z_E

The elasticity factor, Z_E, accounts for the influence of the material properties E (modulus of elasticity) and υ (Poisson's ratio) on the Hertzian pressure. In the case of steel gears, Z_E is to be calculated as follows:

$$Z_E = 189.8(\sqrt{N/mm^2}$$

In other cases, reference is to be made to the reference standard *ISO* 6336-2:2019.

5 Contact Ratio Factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of transverse contact ratios and overlap ratios on the specific surface loads of gears.

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \qquad \text{(In the case of spur gears)}$$

$$= \sqrt{\frac{4-\varepsilon_\alpha}{3}(1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}} \qquad \text{(In the case of helical gears with } \varepsilon_\beta < 1\text{)}$$

$$= \sqrt{\frac{1}{\varepsilon_\alpha}} \qquad \text{(In the case of helical gears with } \varepsilon_\beta \geq 1\text{)}$$

6 Helix Angle Factor, Z_β

The helix angle factor, Z_β, accounts for the influence of helix angles on surface durability, allowing for such variables as distribution of loads along lines of contact. Z_β is dependent only on helix angles and its value can be obtained by the following formula:

$$Z_\beta = \sqrt{\frac{1}{\cos\beta}}$$

where β is the reference helix angle.

1.6.3 Permissible Contact Stress

1 Permissible contact stress, σ_{HP} is to be calculated as follows:

$$\sigma_{HP} = \sigma_{Hlim}\frac{Z_N Z_L Z_V Z_R Z_W Z_X}{S_H}$$

2 Endurance Limit for Contact Stress, σ_{Hlim}

For a given material, σ_{Hlim} is the limit of repeated contact stress which can be permanently endured. σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least 5×10^7 load cycles. "Pitting" is defined in the case of non-surface hardened gears, the pitted area > 2 % of total active flank area; in the case of surface hardened gears, the pitted area > 0.5 % of total active flank area, or > 4 % of one particular tooth flank area. The σ_{Hlim} values are to correspond to a failure probability of 1 % or less.

The endurance limit mainly depends on the following:

(1) Material composition, cleanliness and defects;

(2) Mechanical properties;

(3) Residual stresses;

(4) Hardening process, depth of hardened zone, hardness gradient;

(5) Material structure (forged, rolled bar, cast).

Endurance limit for contact stress σ_{Hlim} is as given in **Table 6.3-1**. However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of factors **(1)** through **(5)** mentioned above.

Table 6.3-1 Value of σ_{Hlim} (N/mm^2)

Steel type	σ_{Hlim}
Normalized structural steels	HB+190
Through hardening carbon steels	HB+350
Through hardening alloy steels	1.33HB+367
Induction hardened alloys	0.6HV+850
Nitrided alloys	1000
Soft nitrided alloys	1.14HV+437; however, 950 for HV>450
Nitrided steels	1250
Carburized hardened alloys	1500

Note:
HB: Brinell Hardness; *HV*: Vickers Hardness

3 Life Factor for Contact Stress, Z_N

The life factor for contact stress, Z_N, accounts for the higher permissible contact stress in cases where a limited life (number of cycles) is required. Values larger than 1.0 are to be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;
(2) Number of cycles;
(3) Influence factors (Z_R, Z_V, Z_L, Z_W, Z_X).

The life factor, Z_N, is to be determined according to Method B outlined in the reference standard *ISO* 6336-2:2019.

4 Lubricant Factor, Z_L

The lubricant factor, Z_L, like the speed factor, Z_V, and roughness factor, Z_R accounts for the influence of the type of lubricant and its viscosity on surface endurance. These factors are to be determined for softer materials in cases where gear pairs are of different hardness. These factors mainly depend on the following:

(1) Viscosity of lubricant in contact zones;
(2) The sum of the instantaneous velocity of tooth surfaces;
(3) Loads;
(4) Relative radius of curvature at pitch points;
(5) Surface roughness of teeth flanks;
(6) Hardness of pinions and wheels.

The value for Z_L is to be calculated as follows:

$$Z_L = C_{ZL} + \frac{4(1.0 - C_{ZL})}{(1.2 + 134/v_{40})^2}$$

where

$$C_{ZL} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.83$$

(In cases where 850 $N/mm^2 \le \sigma_{Hlim} \le$ 1200 N/mm^2)
= 0.83 (In cases where σ_{Hlim} < 850 N/mm^2)
= 0.91 (In cases where σ_{Hlim} >1200 N/mm^2)

v_{40} : Nominal kinematic viscosity of the oil at 40℃ (mm^2/s)

5 Speed Factor, Z_V

The speed factor, Z_V, accounts for the influence of pitch line velocities on surface endurance. The value for Z_V is to be calculated as follows:

$$Z_V = C_{Z_V} + \frac{2(1.0 - C_{Z_V})}{\sqrt{0.8 + 32/v}}$$

where

$$C_{Z_V} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.85$$

(In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq 1200$ N/mm^2)

= 0.85 (In cases where σ_{Hlim} < 850 N/mm^2)

= 0.93 (In cases where σ_{Hlim} >1200 N/mm^2)

6 Roughness Factor, Z_R

The roughness factor, Z_R, accounts for the influence of surface roughness on surface endurance. The value for Z_R is to be calculated as follows:

$$Z_R = \left(\frac{3}{R_{Z10}}\right)^{C_{ZR}}$$

$$R_{Z10} = R_Z \cdot \sqrt[3]{\frac{10}{\rho_{red}}}$$

$$R_Z = \frac{R_{Z1} + R_{Z2}}{2}$$

where

R_{Z1}, R_{Z2} : Respective mean peak to valley roughness for pinions and wheels.

R_Z : Refer to the reference standard *ISO* 6336-2:2019.

ρ_{red} : Relative radius of curvature. The value for ρ_{red} is to be calculated as follows:

$$\rho_{red} = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$\rho_{1.2} = 0.5 \cdot d_{b1\cdot 2} \cdot \tan\alpha_{tw}$ (In the case of internal gears the value d_b is negative)

In cases where the roughness stated is an arithmetic mean roughness, i.e. R_a value, and the conversion $R_Z = 6R_a$ can be applied.

$C_{ZR} = 0.32 - 0.0002\sigma_{Hlim}$ (In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq 1200$ N/mm^2)

= 0.15 (In cases where σ_{Hlim} < 850 N/mm^2)

= 0.08 (In cases where σ_{Hlim} >1200 N/mm^2)

7 Hardness Ratio Factor, Z_W

The hardness ratio factor, Z_W, accounts for the increase in surface durability of soft steel gears meshing with significantly harder gears with smooth surfaces in the following cases:

(1) Surface-hardened pinion with through-hardened wheel

$$Z_W = 1.2 \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB < 130)$$

$$= \left(1.2 - \frac{HB - 130}{1700}\right) \cdot \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (130 \leq HB \leq 470)$$

$$= \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB > 470)$$

where

HB : Brinell hardness of the tooth flanks of the softer gear of the pair

R_{ZH} : equivalent roughness (μm)

$$R_{zH} = \frac{R_{Z1}(10/\rho_{red})^{0.33}(R_{Z1}/R_{Z2})^{0.66}}{(v \cdot v_{40}/1500)^{0.33}}$$

(2) Through-hardened pinion and wheel

When the pinion is substantially harder than the wheel, the work hardening effect increases the load capacity of the wheel flanks. Z_W applies to the wheel only, not to the pinion.

$Z_W = 1$ ($HB_1/HB_2 < 1.2$)

$= 1 + \left(0.00898\frac{HB_1}{HB_2} - 0.00829\right) \cdot (u-1)$ ($1.2 \leq HB_1/HB_2 \leq 1.7$)

$= 1 + 0.00698 \cdot (u-1)$ ($HB_1/HB_2 > 1.7$)

$HB_{1,2}$: Brinell hardness of the pinion and the wheel respectively.

If gear ratio u>20, then the value u=20 is to be used. In any case, if the calculated Z_W <1, then the value Z_W = 1 is to be used.

(3) In cases other than **(1)** and **(2)** above;

$Z_W = 1$

8 Size Factor for Contact Stress, Z_X

The size factor for contact stress, Z_X, accounts for the influence of tooth dimensions on permissible contact stress and reflects the inhomogeneity of material properties. This factor mainly depends on the following:

(1) Materials and heat treatments;
(2) Tooth and gear dimensions;
(3) Ratio of case depth to tooth size;
(4) Ratio of case depth to equivalent radius of curvature.

For through hardened gears and for surface hardened gears with adequate case depth relative to tooth size and radius of relative curvature Z_X = 1.0, in cases where the case depth is relatively shallow then a smaller value of Z_X is to be taken.

9 Safety Factor for Contact Stress, S_H

The safety factor for contact stress, S_H, is to be taken as follows:

(1) In the case of main propulsion gears: 1.20
(2) In the case of auxiliary gears: 1.15

In cases where the gears of duplicated independent propulsion or auxiliary machinery, has been duplicated beyond that what is required for its respective class, a reduced value may be taken at the discretion of the Society.

1.7 Bending Strength

1.7.1 Equation

The tooth root bending stress σ_F and the permissible tooth root bending stress σ_{FP} are to be calculated separately for the pinion and the wheel. The criterion for tooth root bending strength, as given by the following equation, is that the tooth root bending stress in the tooth root fillet σ_F is to be equal to or less than the permissible tooth root bending stress σ_{FP}.

$$\sigma_F = \frac{F_t}{bm_n} Y_F Y_S Y_\beta Y_B Y_{DT} K_A K_\gamma K_V K_{F\alpha} K_{F\beta} \leq \sigma_{FP}$$

However, the following definitions and equations apply only to those gears having a rim thickness greater than 3.5 m_n. The results of calculations using the following method are acceptable for normal pressure angles up to 25 degrees and reference helix angles up to 30 degrees. In the case of larger pressure angles and larger helix angles, the calculated results are to be confirmed by experience as by Method A of the reference standard *ISO* 6336-3:2019.

1.7.2 Tooth Root Bending Stress for Pinion and Wheel

1 Tooth Form Factor, Y_F

The tooth form factor, Y_F, accounts for the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F is to be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing are to be determined in normal sections (i.e. for virtual spur gears with virtual numbers of teeth, Z_n).The value for Y_F is to be calculated as follows:

$$Y_F = \frac{6\dfrac{h_F}{m_n}\cos\alpha_{Fen}}{\left(\dfrac{S_{Fn}}{m_n}\right)^2\cos\alpha_n}$$

S_{Fn}, h_F and α_{Fen} are to be calculated as follows:

$$S_{Fn} = m_n z_n \sin\left(\frac{\pi}{3}-\theta\right) + \sqrt{3}m_n\left(\frac{G}{\cos\theta}-\frac{\rho_{fp}}{m_n}\right)$$

$$G = \frac{\rho_{fp}}{m_n} - \frac{h_{fp}}{m_n} + x$$

$$\theta = \frac{2G}{z_n}\tan\theta - \frac{2}{z_n}\left(\frac{\pi}{2}-\frac{E}{m_n}\right)+\frac{\pi}{3}$$

$$E = \frac{\pi}{4}m_n - h_{fp}\tan\alpha_n + \frac{S_{Pr}}{\cos\alpha_n} - (1-\sin\alpha_n)\frac{\rho_{fp}}{\cos\alpha_n}$$

S_{pr} is illustrated in **Fig. 7.2-1.** However, in cases where racks are without undercuts, S_{pr} is to be taken as zero.

$$h_F = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e\tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n\cos\left(\frac{\pi}{3}-\theta\right) - \frac{G}{\cos\theta} + \frac{\rho_{fp}}{m_n}\right]$$

$$\alpha_{Fen} = \alpha_{en} - \gamma_e$$

$$\gamma_e = \frac{0.5\pi + 2x\tan\alpha_n}{z_n} + \mathrm{inv}\alpha_n - \mathrm{inv}\alpha_{en}$$

$$\alpha_{en} = \arccos\left(\frac{d_{bn}}{d_{en}}\right)$$

$$d_{en} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2 - \left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d\cos\beta\cos\alpha_n}{|z|}(\varepsilon_{\alpha n}-1)\right]^2 + \left(\frac{d_{bn}}{2}\right)^2\right\}^{\frac{1}{2}}$$

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$

$$\beta_b = \arccos\sqrt{1-\sin^2\beta\cos^2\alpha_n}$$

$$d_{b_n} = d_n\cos\alpha_n$$

$$d_n = m_n z_n$$

$$d_{a_n} = d_n + d_a - d$$

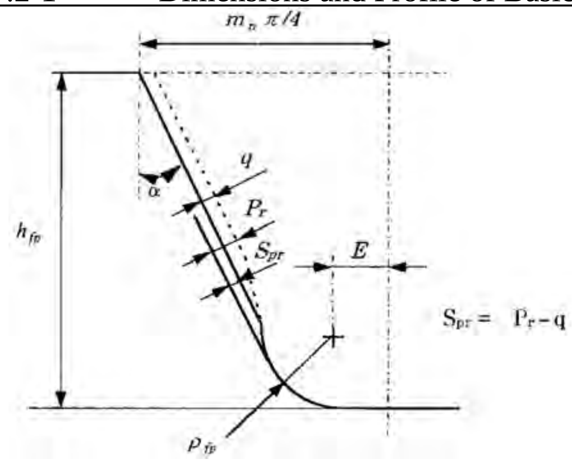
In the case of internal gears, the following coefficients used for determining the form factor are to be calculated as follows:

$$S_{Fn2} = m_n z_n \sin\left(\frac{\pi}{6} - \theta\right) + m_n\left(\frac{G}{\cos\theta} - \frac{\rho_{fp}}{m_n}\right)$$

$$h_{Fn2} = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e \tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n \cos\left(\frac{\pi}{6} - \theta\right) - \sqrt{3}\left(\frac{G}{\cos\theta} - \frac{\rho_{fp2}}{m_n}\right)\right]$$

$$d_{en2} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2 - \left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d_2 \cos\beta \cos\alpha_n}{|z|}(\varepsilon_{\alpha n} - 1)\right]^2 + \left(\frac{d_{bn2}}{2}\right)^2\right\}^{\frac{1}{2}}$$

Fig. 7.2-1 Dimensions and Profile of Basic Rack



2 Stress Correction Factor, Y_S

The stress correction factor, Y_S, is used to convert nominal bending stress into local tooth root stress, taking into account that not only bending stresses arise at roots. Y_S applies to those load applications at the outer points of single tooth pair contacts. Y_S is to be determined separately for pinions and for wheels. The value for Y_S is to be determined as follows (in the effective range: $1 \leq q_S < 8$).

$$Y_s = \left(1.2 + 0.13\frac{S_{Fn}}{h_F}\right) q_S^L$$

$$L = \frac{1}{1.21 + 2.3\dfrac{h_F}{S_{Fn}}}$$

where

However, since q_S is a notch parameter, it is to be calculated as follows:

$$q_S = \frac{S_{Fn}}{2\rho_F}$$

However, since ρ_F is the root fillet radius in the critical section (mm), it is to be calculated as follows:

$$\rho_F = \rho_{fp} + \frac{2 m_n G^2}{\cos\theta(z_n \cos^2\theta - 2G)}$$

For the calculation of ρ_F, the procedure outlined in the reference standard *ISO* 6336-3:2019 is to be used.

3 Helix Angle Factor for Bending Stress, Y_β

The helix angle factor for bending stress, Y_β, is used to convert the stress calculated for point loaded cantilever beams representing gear teeth to the stress induced by loads along oblique load lines into cantilever plates which represent helical gear teeth. The value for Y_β can be calculated as follows:

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120}$$

where β is the reference helix angle in degree.

However, in cases where $\beta > 30°$ then $\beta = 30°$; and,

Also, in cases where $\varepsilon_\beta > 1$ then ε_β is to be taken as 1.0.

4 Rim thickness factor, Y_B

The rim thickness factor, Y_B, is a simplified factor used to de-rate thin rimmed gears. For critically loaded applications, this method is to be replaced by a more comprehensive analysis. Factor Y_B is to be determined as follows:

(1) For external gears:

$$Y_B = 1.6 \cdot \ln\left(2.242\frac{h}{s_R}\right) \qquad (0.5 < s_R/h < 1.2)$$

$$= 1 \qquad (s_R/h \geq 1.2)$$

s_R : rim thickness of external gears (*mm*)

h : tooth height (*mm*)

The case $s_R/h \leq 0.5$ is to be avoided.

(2) For internal gears:

$$Y_B = 1.15 \cdot \ln\left(8.324\frac{m_n}{s_R}\right) \qquad (1.75 < s_R/m_n < 3.5)$$

$$= 1 \qquad (s_R/m_n \geq 3.5)$$

s_R : rim thickness of internal gears (*mm*)

The case $s_R/m_n \leq 1.75$ is to be avoided.

5 Deep tooth factor, Y_{DT}

The deep tooth factor, Y_{DT}, adjusts the tooth root stress to take into account high precision gears and contact ratios within the range of virtual contact ratio $2.05 \leq \varepsilon_{\alpha n} \leq 2.5$.

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$

Factor Y_{DT} is to be determined as follows:

$Y_{DT} = 0.7$(*ISO* accuracy grade ≤ 4 and $\varepsilon_{\alpha n} > 2.5$)

$Y_{DT} = 2.366 - 0.666\,\varepsilon_{\alpha n}$(*ISO* accuracy grade ≤ 4 and $2.05 < \varepsilon_{\alpha n} \leq 2.5$)

$Y_{DT} = 1.0$(In all other cases)

1.7.3 Permissible Tooth Root Bending Stress, σ_{FP}

1 Permissible tooth root bending stress σ_{FP} is to be calculated as follows:

$$\sigma_{FP} = \frac{\sigma_{FE} Y_d Y_N Y_{\delta relT} Y_{RrelT} Y_X}{S_F}$$

2 Bending Endurance Limit, σ_{FE}

For a given material, σ_{FE} is the local tooth root stress which can be permanently endured. According to the reference standard *ISO* 6336-5:2016, the number of 3×10^6 cycles may be regarded as the beginning of the endurance limit. σ_{FE} is defined as unidirectional pulsating stress with a minimum stress zero (neglecting any residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d. σ_{FE} values are to correspond to a failure probability of 1 % or less. These endurance limits mainly depends on the following:

(1) Material composition, cleanliness and defects;

(2) Mechanical properties;

(3) Residual stresses;

(4) Hardening process, depth of hardened zone, hardness gradient

(5) Material structure (forged, rolled bar, cast)

The value for σ_{FE} is to be calculated as follows:

$$\sigma_{FE} = 2\sigma_{Flim}$$

Values for σ_{Flim} are given in **Table 7.3-1.** However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of the factors **(1)** through **(5)** mentioned above.

3 Design Factor, Y_d

The design factor, Y_d, takes into account the influence of load reversing and shrink fit prestressing on tooth root strength, relative to tooth root strength with unidirectional loads as defined for σ_{FE}. Y_d for load reversing is to be determined as follows:

$Y_d = 1.00$ (In general cases)

$= 0.90$ (In the case of gears with occasional part loads in reversed directions, such as the main gears in reversing gearboxes)

$= 0.70$ (In the case of idler gears)

4 Life Factor for Bending Stress, Y_N

The life factor for bending stress, Y_N, accounts for the higher tooth root bending stress permissible in cases where limited life is required. Values greater than 1.0 will be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;

(2) Number of load cycles (service life);

(3) Influence factors($Y_{\delta relT}$, Y_{RrelT}, Y_X).

Y_N is to be determined according to Method B outlined in the reference standard *ISO* 6336-3:2019.

5 Relative Notch Sensitivity Factor, $Y_{\delta relT}$

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent of the influence of concentrated stress on fatigue endurance limits. This factor mainly depends on materials and relative stress gradients. This factor is to be calculated as follows:

$$Y_{\delta relT} = \frac{1 + \sqrt{0.2\rho'(1 + 2q_s)}}{1 + \sqrt{1.2\rho'}}$$

q_s : notch parameter

ρ' : slip-layer thickness (mm)

However, the values to be used for ρ' are those given in **Table 7.3-2**.

6 Relative Surface Factor, Y_{RrelT}

The relative surface factor, Y_{RrelT}, takes into account the influence of surface conditions in tooth root fillets on root strength and mainly depends on peak to valley surface roughness. The value for Y_{RrelT} is to be determined as shown in **Table 7.3-3**.

7 Size Factor for Bending Stress, Y_X

The size factor for bending stress, Y_X, takes into account decreases of strength with increasing size. This factor mainly depends on the following:

(1) Material and heat treatment;

(2) Tooth and gear dimensions;

(3) Ratio of case depth to tooth size.

The value for Y_X is to be determined as shown in **Table 7.3-4**.

8 Safety Factor for Tooth Root Bending Stress, S_F

The safety factor for tooth root bending stress, S_F, is to be taken as follows:

(1) 1.55 for main propulsion gears;

(2) 1.40 for auxiliary gears;

In addition, in cases where the gears of duplicated independent propulsion or auxiliary machinery which are duplicated beyond that required for their respective classes, a reduced value may be used at the discretion of the Society.

Table 7.3-1 Values of σ_{Flim} (N/mm^2)

Steel type	σ_{Flim}
Normalized structural steels	0.45HB+70
Through hardening carbon steels	0.25HB+160
Through hardening alloy steels	0.45HB+180
Induction hardened alloys	0.14HV+285; however, 365 for HV>570
Nitrided alloys	365
Soft nitrided alloys	0.66HV+88; however, 385 for HV>450
Nitrided steels	420
Carburized hardened alloys	465; however, 500 for HRC>30 in core

HB: Brinell Hardness; HV: Vickers Hardness; HRC: C Scale Rockwell Hardness

Table 7.3-2 Values of ρ' (mm)

Materials	ρ'
Nitriding steels, surface or through hardened	0.1005
Steels having yield strength about 300 N/mm^2	0.0833
Steels having yield strength about 400 N/mm^2	0.0445
Through hardened steels having yield strength about 500 N/mm^2	0.0281
Through hardened steels having yield strength 600 N/mm^2	0.0194
Through hardened steels having $\sigma_{0.2}$ about 800 N/mm^2	0.0064
Through hardened steels having $\sigma_{0.2}$ about 1000 N/mm^2	0.0014
Surface hardening steels, surface hardened	0.0030

Table 7.3-3 Values of Relative Surface Factor, Y_{RrelT}

Materials	Y_{RrelT}
Case hardened steels, Through hardened steels ($\sigma_B \geq 800$ N/mm^2)	1.120 (for R_Z <1) $1.674 - 0.529(R_Z+1)^{0.1}$ (for 1≤ $R_Z \leq 40$)
Normalized steels ($\sigma_B < 800$ N/mm^2)	1.070 (for R_Z <1) $5.306 - 4.203(R_Z+1)^{0.01}$ (for 1≤ $R_Z \leq 40$)
Nitriding steels	1.025 (for R_Z <1) $4.299 - 3.259(R_Z+1)^{0.0058}$(for1≤ $R_Z \leq 40$)

Notes

1: R_Z mean peak to valley roughness of tooth root fillets (μm).

2: This method is only valid when scratches or similar defects deeper than 2R_Z do not exist.

3: If the roughness stated is an arithmetic mean roughness, i.e. R_a value, the approximation R_Z=6R_a may be applied.

Table 7.3-4 Size Factor for Bending Stress, Y_X

m_n	Materials	Y_X
$m_n \leq 5$	General	1.00
$5 < m_n < 30$	Normalized and through hardened steels	$1.03 - 0.006m_n$
$m_n \geq 30$		0.85
$5 < m_n < 25$	Surface hardened steels	$1.05 - 0.010m_n$
$m_n \geq 25$		0.80

EFFECTIVE DATE AND APPLICATION (Amendment 1-6)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to all gears previously approved by the Society prior to the effective date for which no failure has occurred, and no changes related to strength, such as the scantlings of the gear meshes, materials, etc. have been made.

Amendment 1-7

Chapter 12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

Table D12.8 has been amended as follows.

Table D12.8 Application Classifications of Mechanical Joints(1)

Application Purpose	System	Kind of Connections~~(2)~~(3)			Classification of pipe system	Fire endurance test condition(12)
		Pipe Union	Compression Coupling	Slip-on Joint~~(10)~~(11)		
Flammable fluids~~(9)~~(10) (Flash point ≦60 Material °C)	Cargo oil lines~~(6)~~(4)	+	+	+	dry	30 min dry(2)
	Crude oil washing lines~~(6)~~(4)	+	+	+	dry	
	Vent lines~~(5)~~(6)	+	+	+	dry	
Inert gases	Water seal effluent lines	+	+	+	wet	30 min wet(2)
	Scrubber effluent lines	+	+	+	wet	30 min wet(2)
	Main lines(4)~~(6)~~(5)	+	+	+	dry	30 min dry(2)
	Distributions lines~~(6)~~(4)	+	+	+	dry	30 min dry(2)
Flammable fluids~~(9)~~(10) (Flash point > 60 °C)	Cargo oil lines~~(6)~~(4)	+	+	+	dry	30 min dry(2)
	Fuel oil lines~~(4)~~(5)(6)	+	+	+	wet	30 min wet(2)
	Lubricating oil lines~~(4)~~(5)(6)	+	+	+	wet	
	Hydraulic oil~~(4)~~(5)(6)	+	+	+	wet	
	Thermal oil~~(4)~~(5)(6)	+	+	+	wet	
Sea Water	Bilge lines~~(3)~~(7)	+	+	+	dry/wet	8 min dry + 22 min wet(2)
	Water filled fire extinguishing systems, e.g. fire main, sprinkler systems~~(5)~~(6)	+	+	+	wet	30 min wet(2)
	Non water filled fire extinguishing systems, e.g. foam, drencher systems and fire main~~(5)~~(6)	+	+	+	dry/wet	8 min dry + 22 min wet(2) (comply with Chapter 26, Part R)
	~~Fire main(5)~~	~~+~~	~~+~~	~~+~~		
	Ballast systems~~(3)~~(7)	+	+	+	wet	30 min wet(2)
	Cooling water systems~~(3)~~(7)	+	+	+	wet	30 min wet(2)
	Tank cleaning services	+	+	+	dry	Fire endurance test not required

Table D12.8 Application Classifications of Mechanical Joints[1](Continued)

Application Purpose	System	Kind of Connections~~[2]~~[3]			Classification of pipe system	Fire endurance test condition[12]
		Pipe Union	Compression Coupling	Slip-on Joint~~[10]~~[11]		
Sea Water	Non-essential systems	+	+	+	dry dry/wet wet	Fire endurance test not required
Fresh water	Cooling water systems~~[3]~~[7]	+	+	+	wet	30 min wet[2]
	Condensate returns~~[3]~~[7]	+	+	+	wet	30 min wet[2]
	Non-essential systems	+	+	+	dry dry/wet wet	Fire endurance test not required
Sanitary/ Drains/ Scuppers	Deck drains (internal)~~.~~~~[7]~~[8]	+	+	+	dry	Fire endurance test not required
	Sanitary drains	+	+	+	dry	
	Scuppers and discharges (overboard)	+	+	-	dry	
Sounding/Vents	Water tanks/Dry spaces	+	+	+	dry, wet	Fire endurance test not required
	Oil tanks (f.p.> 60 °C)~~.~~~~[4]~~[5][6]	+	+	+	dry	
Miscellaneous	Starting/Control air~~[3]~~[7]	+	+	-	dry	30 min dry[2]
	Service air (non-essential)	+	+	+	dry	Fire endurance test not required
	Brine	+	+	+	wet	
	CO_2 systems~~[3]~~ (outside protected space)	+	+	-	dry	30 min dry[2]
	CO_2 systems (inside protected space)	+	+	-	dry	Mechanical joints are to be constructed of materials with melting point above 925°C. (refer to Chapter 25, Part R)
	Steam	+	+	~~-~~+ ~~[8]~~[9]	wet	Fire endurance test not required

Notes:

(1) +: Application is allowed; -: Application is not allowed

(2) Fire endurance test in accordance with **9.3.2(6), Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use.**

(~~2~~3) If mechanical joints include any components which readily deteriorate in case of fire, the following **(~~3~~4)** to **(~~6~~7)** apply.

~~(3) Inside machinery spaces of category *A*, fire resistant types approved by the Society.~~

(4) Fire endurance test is to be applied when mechanical joints are installed in pump rooms and open decks.

(~~4~~5) Slip-on joints are not accepted inside machinery spaces of category *A* or accommodation spaces. May be accepted in machinery spaces other than those of category *A* provided that the joints are located in easily visible and accessible positions (refer to *MSC/Circ*.734).

(~~5~~6) Fire resistant types approved by the Society except in cases where such mechanical joints are installed on open decks as defined in **9.2.3-2(10), Part R of the Rules**; this excludes spaces in the cargo areas of tankers, ships carrying liquefied gases in bulk and ships carrying dangerous chemicals in bulk (as defined in **3.2.6, Part R**, **1.1.4(6), Part N** and **1.3.1(4), Part S**), but not used for fuel oil lines, fire extinguishing systems and fire mains.

(~~6~~7) ~~In pump rooms and open decks, fire resistant types approved by the Society.~~ Fire endurance test is to be applied when mechanical joints are installed inside machinery spaces of category *A*

(~~7~~8) Only above the freeboard deck.

(89) Slip type slip-on joints as shown in **Fig. D12.1** may be used for pipes on deck with a design pressure of 1.0 *MPa* or less**.**

(~~9~~10) Piping where mechanical joints are used is also to comply with the requirements specified in **13.2.4-4**.

(~~10~~11) Piping where slip joints are used is also to comply with the requirements specified in **13.2.4-6**.

(12) If a connection has passed the “30 min dry” test, it is considered suitable also for applications for which the “8 min dry + 22 min wet” and/or “30 min wet” tests are required. If a connection has passed the “8 min dry+22 min wet” test, it is considered suitable also for applications for which the “30 min wet” test is required.

Chapter 13 PIPING SYSTEMS

13.2 Piping

13.2.4 Mechanical Joints*

Sub-paragraph -6 has been amended as follows.

6 The following **(1)** ~~to~~ and **(~~3~~2)** limitations apply to use of slip-on joints, in addition to **-2** to **-5** above.

(1) Slip-on joints are not to be used on pipelines in cargo holds, tanks and other spaces which are not easily accessible (refer to *MSC/Circ.*734), ~~unless approved by the Society~~ except that these joints may be permitted in tanks that contain the same media.

~~(2) Application of slip-on joints inside tanks may be permitted only for the same media that is in the tanks; this includes those tanks specified in **(1)** above.~~

(~~3~~2) Usage of slip type slip-on joints as the main means of pipe connection is not permitted except in cases where compensation of axial pipe deformation is necessary.

EFFECTIVE DATE AND APPLICATION (Amendment 1-7)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to mechanical joints other than those that fall under the following:

(1) mechanical joints for which the application for approval is submitted to the Society on or after 1 July 2022.

(2) mechanical joints for which the date of renewal of approval of use is on or after 1 July 2022.

Amendment 1-8

Chapter 14 PIPING SYSTEMS FOR TANKERS

14.2 Cargo Oil Pumps, Cargo Oil Piping Systems, Piping in Cargo Oil Tanks, etc.

Paragraph 14.2.2 has been amended as follows.

14.2.2 Arrangement of Cargo Oil Piping Systems~~*~~

(-1 to -6 are omitted.)

~~**7** All cargo oil tanks and cargo piping systems are to be electrically bonded to hull structures by suitable methods such as metal-to-metal contact using welding or bolts, or bonding straps, etc. The following tanks and piping systems which are not permanently connected to the hull of the ship are to be connected to the hull of the ship by bonding straps:~~

~~(1) Cargo tanks which are electrically separated from the hull of the ship (*e.g.*, independent cargo oil tanks);~~

~~(2) Pipe connections which can be removed (*e.g.*, spool pieces); and~~

~~(3) Wafer-style valves with non-conductive (*e.g.*, PTFE) gaskets or seals.~~

~~**8** The bonding straps specified in **-7** above are to comply with the following requirements:~~

~~(1) Clearly visible so that any shortcomings can be clearly detected;~~

~~(2) Designed and sited so that they are protected against mechanical damage and that they are not affected by high resistivity contamination (*e.g.*, corrosive products or paint); and~~

~~(3) Easy to install and replace.~~

7 Earthing and bonding of cargo tanks, piping systems, etc. for the control of static electricity are to comply with following requirements:

(1) The hazard of an incentive discharge due to the build-up of static electricity resulting from the flow of liquids/gases/vapours can be avoided if the resistance between the cargo tanks, piping systems, etc. and the hull of the ship is not greater than 1 $M\Omega$.

(2) This value of resistance will be readily achieved without the use of bonding straps where cargo tanks, piping systems, etc. are directly or via their supports, either welded or bolted to the hull of the ship.

(3) Bonding straps are required for cargo tanks, piping systems, etc. which are not permanently connected to the hull of the ship, e.g.
   (a) Independent cargo tanks;
   (b) Cargo tanks/piping systems which are electrically separated from the hull of the ship;
   (c) Pipe connections arranged for the removal of spool pieces; and
   (d) Wafer-style valves with non-conductive (e.g PTFE) gaskets or seals.

(4) Where bonding straps are required, they are to be:
   (a) Clearly visible so that any shortcomings can be clearly detected;
   (b) Designed and sited so that they are protected against mechanical damage and that they are not affected by high resistivity contamination e.g. corrosive products or paint; and
   (c) Easy to install and replace.

EFFECTIVE DATE AND APPLICATION (Amendment 1-8)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to ships for which the date of contract for construction is before the effective date.

Amendment 1-9

Chapter 21 SELECTIVE CATALYTIC REDUCTION SYSTEMS AND ASSOCIATED EQUIPMENT

21.2 Design

21.2.2 Material

Sub-paragraph -1 has been amended as follows.

1 Reductant tanks are to be of steel or other equivalent material with a melting point above 925 °C.

（Note）

The wording “to be of steel or other equivalent material” is not applicable for integral tanks on FRP vessels such as those listed below, provided that the integral tanks are coated and/or insulated with a self-extinguishing material.

(1) FRP vessels complying with Regulation 17 of *SOLAS* Chapter II-2 based upon its associated IMO guidelines (*MSC*.1/*Circ*.1574), and

(2) FRP vessels exempted from the application of *SOLAS* e.g., yachts, fast patrol, navy vessels, etc., generally of less than 500 gross tonnage, subject to yacht codes or flag regulations.

(-2 to -4 are omitted.)

21.4 Requirements for Construction and Arrangements, etc.

21.4.5 Safety Devices and Alarm Devices

Sub-paragraphs -2 and -3 have been amended as follows.

1 (Omitted)

2 Alarm devices, to be activated in the event of any of the abnormal conditions given in **Table D21.1**, are to be provided at control stations of SCR systems.

3 SCR systems are to be fitted with monitoring devices at control stations of SCR systems, and these devices are to be capable of indicating the information listed in the following **(1)** to **(4)**:

(1) Liquid levels in tanks for reductant agent

(2) Temperatures in tanks for reductant agent

(3) Exhaust gas temperatures at inlets

(4) Pressures at inlets or differential pressures across catalyst block

4 (Omitted)

Table D21.1 has been amended as follows.

Table D21.1 Alarm points for SCR system[1]

Monitored Variables	
Liquid levels in tank for reductant agent	H L
Temperature in tank for reductant agent	H L
Exhaust gas pressure at inlet [2]	H
Exhaust gas temperature at inlet	H ~~L~~
Exhaust gas temperature at outlet[3]	H
Power loss of control, alarm, monitoring or safety devices	○

Notes:

(1) "H" and "L" mean "high" and "low". "○" means abnormal condition occurred.

(2) Differential pressure across catalyst block may be accepted in lieu.

(3) Alarms may be omitted in cases where means are provided to prevent damage by soot fire.

21.8 Tests

21.8.1 Tests at Facilities (Shop tests)

Sub-paragraph -4 has been amended as follows.

4 For reductant agent supply pumps, shop trials are to be carried out according to test procedures deemed appropriate by the Society. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

EFFECTIVE DATE AND APPLICATION (Amendment 1-9)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Rules, the current requirements apply to SCR whose applications for approval are submitted to the Society before the effective date installed on ships for which the date of contract for construction* is before the effective date.

3. Notwithstanding the provision of preceding 2., the amendments to the Rules may apply to SCR whose applications for approval are submitted to the Society before the effective date installed on ships for which the date of contract for construction* is before the effective date upon request of the owner.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.

For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:

(1) such alterations do not affect matters related to classification, or

(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.

The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

Amendment 1-10

Chapter 22 EXHAUST GAS CLEANING SYSTEMS AND ASSOCIATED EQUIPMENT

22.1 General

22.1.1 Application

Sub-paragraphs -1 to -3 have been amended as follows.

1 The requirements in this chapter apply to exhaust gas cleaning systems and associated equipment installed to reduce sulphur oxides and particular matter emitted from fuel oil combustion units such as reciprocating internal combustion engines and boilers, and which use sodium hydroxide solutions or calcium hydroxide (hereinafter referred to "chemical treatment fluid" in this chapter) that has corrosive properties or which are otherwise considered to represent a hazard to personnel.

2 In cases where exhaust gas cleaning systems which use chemical agents other than those specified in **-1** above are used, ~~special consideration is to be given to such systems in accordance with their respective designs~~safety measures are to be taken according to the result of a risk assessment to be conducted to analyse the risks, in order to eliminate or mitigate the hazards to personnel brought by the use of such exhaust gas cleaning systems, to an extent equivalent to systems complying with this chapter.

3 In cases where exhaust gas cleaning systems which do not use chemical agents are used, the term "liquids containing ~~sodium hydroxide solutions~~chemical treatment fluid" is to be read as "liquids which have passed through scrubber chambers"; this, however, does not apply to **-4**, **-9** and **-10** of **22.4.1**, **22.7.1-2** and **22.7.2-2(1)**.

4 (Omitted)

5 (Omitted)

22.1.2 Terminology

Sub-paragraphs (1) and (3) have been amended as follows.

The terms used in this chapter are defined as follows:

(1) "Exhaust gas cleaning system" means a system which consists of storage tanks for residues, etc., washwater supply pumps, ~~sodium hydroxide solution~~chemical treatment fluid supply pumps, washwater injection systems and scrubber chambers.

(2) (Omitted)

(3) "Washwater" means freshwater or sea water (including cases where sodium hydroxide or calcium hydroxide is added) which is injected into scrubber chambers or exhaust gas inlets, and includes liquids which have passed through scrubber chambers.

(4) (Omitted)

(5) (Omitted)

22.1.3 Drawings and Data to be Submitted

Sub-paragraphs (1) and (2) have been amended as follows.

(1) Plans and documents for approval
((a) to (e) are omitted.)
(f) Construction of storage tanks for ~~sodium hydroxide solution~~chemical treatment fluid or liquid containing ~~sodium hydroxide solution~~chemical treatment fluid and their arrangements.
(g) Ventilation systems for compartments installed with equipment for using or handling ~~sodium hydroxide solutions~~chemical treatment fluid, such as storage tanks, or for the compartments specified in **22.4.2-3**
(h) Piping diagrams (including details of watertight bulkheads and penetrations of fire-resisting divisions)
((i) to (l) are omitted.)
(2) Plans and documents for reference
((a) to (e) are omitted.)
(f) The results of risk assessments conducted to analyse the risks specified in **22.1.1-2**
(~~f~~g) Other drawings considered necessary by the Society

22.2 Design

22.2.1 General Requirements

Sub-paragraphs -1 and -2 have been amended as follows.

1 In addition to the requirements in this Chapter, pipes, valves, pipe fittings and auxiliaries are to satisfy the requirements in **Chapter 12**. In such cases, the term "sea water" is to be read as "liquids containing ~~sodium hydroxide solutions~~chemical treatment fluid". However, ~~pipes containing sodium hydroxide solutions only are to be classified as Group I~~regardless of design pressure and temperature, piping systems containing chemical treatment fluids only are to comply with the requirements applicable to Class I piping systems specified in **Chapter 12**. As far as practicable, e.g. except for the flange connections that connect to tank valves, the piping systems are to be joined by welding.
2 In addition to the requirements in this Chapter, air pipes and sounding pipes are to satisfy the requirements in **13.6** and **13.8** (excluding **13.6.1-5** and **13.6.2-3**). In such cases, the term "fuel oil" is to be read as "liquids containing ~~sodium hydroxide solutions~~chemical treatment fluid".
(-3 and -4 are omitted.)

Paragraph 22.2.2 has been amended as follows.

22.2.2 Material

1 Materials used for exhaust gas cleaning systems are to be selected in consideration of notch ductility at operating temperatures and pressures, their corrosive effects and the possibility of hazardous reactions.
2 Storage tanks and pipes/piping systems for chemical treatment fluids which transfer undiluted chemical treatment fluids are to be of steel or other equivalent material with a melting point above 925 °C.
3 Storage tanks and pipes/piping systems for chemical treatment fluids are to be made with a

material compatible with chemical treatment fluids, or coated with appropriate anti-corrosion coating.

Note:

Several metals are incompatible with the chemical treatment fluids, e.g. NaOH is incompatible with zinc, aluminum, etc.

22.4 Requirements for Construction and Arrangements, etc.

Paragraph 22.4.1 has been amended as follows.

22.4.1 Construction and Arrangement

1 ~~Sodium hydroxide solution~~Chemical treatment fluids storage tanks may be located within the engine room.

2 ~~Sodium hydroxide solution~~Chemical treatment fluids storage tanks are to be protected from excessively high or low temperatures applicable to the particular concentration of the ~~solution~~fluids. Depending on the operational area of the ship, this may necessitate the fitting of heating and/or cooling systems.

3 Drip trays of a sufficient size are to be provided under storage tanks for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids as well as any equipment using or handling such liquids, such as pumps, to prevent the spread of any spillage in the compartments where they are installed.

4 (Omitted)

5 The storage tank for chemical treatment fluids is to be arranged so that any leakage will be contained and prevented from making contact with heated surfaces. All pipes or other tank penetrations are to be provided with manual closing valves attached to the tank. In cases where such valves are provided below top of tank, they are to be arranged with quick acting shutoff valves which are to be capable of being remotely operated from a position accessible even in the event of chemical treatment fluid leakages.

6 The storage tanks are to have sufficient strength to withstand a pressure corresponding to the maximum height of a fluid column in the overflow pipe, with a minimum of 2.4 *m* above the top plate taking into consideration the specific density of the treatment fluid.

~~5~~7 Where ~~sodium hydroxide solution~~chemical treatment fluids is stored in tanks which form part of the ship's hull, the following **(1)** to **(~~6~~4)** are to be considered during the design and construction:

(1) These tanks may be designed and constructed as integral part of the hull, (e.g. double bottom, wing tanks).

(2) These tanks are to be coated with appropriate anti-corrosion coating and are to be segregated by cofferdams, void spaces, pump rooms, empty tanks or other similar spaces so as to not be located adjacent to accommodation, cargo spaces containing cargoes which react with chemical treatment fluids in a hazardous manner as well as any food stores, oil tanks and fresh water tanks.

(3) These tanks are to be designed and constructed as per the structural requirements applicable to hull and primary support members for deep tank construction after taking into account the specific gravity ~~of sodium hydroxide solution~~.

~~(4) These tanks are to be fitted with but not limited to level gauge, temperature gauge, high temperature alarm, high and low level alarm, etc.~~

~~(5) These tanks are to be segregated by cofferdams, void spaces, pump rooms, empty tanks or other similar spaces so as to not be located adjacent to accommodation or service spaces, cargo spaces containing cargoes which react with sodium hydroxide solutions in a~~

~~hazardous manner as well as any food stores, oil tanks and fresh water tanks.~~

(~~6~~4) These tanks are to be included in the ship's stability calculation.

~~6~~8 The chemical treatment fluid ~~P~~piping ~~for liquids containing sodium hydroxide solutions~~ and venting systems are to be independent of other ship service piping and/or systems.

~~7~~9 The chemical treatment fluid ~~P~~piping systems ~~for liquids containing sodium hydroxide solutions~~ are not to pass through or to extend into accommodation, service spaces, or control stations.

~~8~~10 Piping systems for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids are not to pass through or to extend into any storage tanks for other liquids, except where deemed appropriate by the Society.

~~9~~11 Piping systems for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids, excluding those near nozzles spraying washwater, are to be so arranged to prevent any outflows or leakage from the piping system from coming into contact with any high temperature equipment surfaces. Such piping systems are especially not to be located immediately above or near equipment such as boilers, steam pipes or exhaust gas pipes.

1~~0~~2 Storage tanks for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids are to satisfy the following requirements:

(1) The tanks are to be so arranged to prevent liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids escaping or leaked from the tanks from coming into contact with high temperature equipment surfaces. Such tanks are especially not to be located immediately above or near equipment such as boilers, steam pipes or exhaust gas pipes.

(2) In cases where shore connections with standard couplings are fitted onto filling-up pipe lines, proper protection against any spraying of ~~sodium hydroxide solutions~~chemical treatment fluids, such as effective enclosures, is to be provided in consideration of the sodium hydroxide solution spraying out during filling-up operations.

1~~1~~3 Discharge pipes from storage tanks for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids are to be fitted with stop valves directly on the tank.

~~**12** Piping systems for sodium hydroxide solutions which, if damaged, would allow the solution to escape from storage tanks are to be fitted with cocks or valves directly onto the tank. Such cocks or valves are to be capable of being closed from accessible positions even in the event of solution leakages.~~

1~~3~~4 ~~Residue t~~Tanks for residues generated from the exhaust gas cleaning process are to satisfy the following requirements:

(1) ~~Residues removed from washwater used in scrubber chambers are to be stored in tanks independent of the oil residue (sludge) tanks fitted in accordance with **Chapter 2, Part 3 of the Rules for Marine Pollution Prevention Systems**. In addition, such residues are to be discharged to appropriate reception facilities.~~The tanks are to be independent from other tanks, except in cases where these tanks are also used as the over flow tanks for chemical treatment fluids storage tank.

(2) Manholes or access holes in a sufficient size are to be provided at such locations that each part of the tank can be cleaned without difficulties.

(3) Tank capacities are to be decided in consideration of the number and kinds of installed exhaust gas cleaning systems as well as the maximum number of days between ports where residue can be discharged ashore. In the absence of precise data, a figure of 30 *days* is to be used.

(4) Where residue tanks used in closed loop chemical treatment systems are also used as the overflow tanks for chemical treatment fluids storage tank, the requirements for storage tanks apply.

1~~4~~5 (Omitted)

1~~5~~6 For distance pieces fitted onto the piping systems specified in **-1~~4~~5** above, where materials other than hull construction materials are used and where two or more kinds of different metallic materials are arranged adjacent to each other, appropriate measures are to be taken to prevent bimetallic corrosion.

17 The following connections are to be screened or provided with other appropriate means, and fitted with drip trays to prevent the spread of any spillage where they are installed:

(1) Detachable connections between pipes (flanged connections and mechanical joints, etc.);

(2) Detachable connections between pipes and equipment such as pumps, strainers, heaters, valves; and

(3) Detachable connections between equipment mentioned in **(1)** and **(2)** above.

18 The drip trays specified in **-17** above are to be fitted with drain pipes which lead to appropriate tanks, such as residue tanks specified in **-14** above, which are fitted with high level alarm, or are to be fitted with alarms for leak detection. In cases where such tank is an integral tank, **-7(1)** and **(2)** above are to be applied to the tank (the term “these tanks” specified in **-7(1)** and **(2)** is to be read as “appropriate tanks, such as residue tanks”).

Paragraph 22.4.2 has been amended as follows.

22.4.2 Ventilation Systems

1 If storage tanks for ~~sodium hydroxide solutions~~chemical treatment fluids ~~or equipment for using or handling sodium hydroxide solutions, such as solution supply pumps,~~ is installed in a closed compartment, the area is to be served by an effective mechanical ventilation system of extraction type providing not less than 6 air changes per hour which is independent from the ventilation system of accommodation, service spaces, or control stations. The ventilation system is to be capable of being controlled from outside the compartment. ~~If the ventilation stops, an audible and visual alarm shall be provided outside the compartment adjacent to each point of entry and inside the compartment, together with a warning notice requiring the use of such ventilation~~A warning notice requiring the use of such ventilation before entering the compartment is to be provided outside the compartment adjacent to each point of entry.

2 Notwithstanding the requirements specified in **-1** above, where storage tanks for ~~sodium hydroxide solutions~~chemical treatment fluids ~~or equipment for using or handling sodium hydroxide solutions, such as the solution supply pump~~ are located within an engine room a separate ventilation system is not required when the general ventilation system for the space is arranged so as to provide an effective movement of air in the vicinity of the storage tank and equipment and is to be maintained in operation continuously except when the storage tank is empty and has been thoroughly air purged.

3 ~~In cases where sodium hydroxide solutions are stored within tanks which form part of the ship’s hull, ventilation systems for enclosed compartments normally entered by ship personnel which are located adjacent to such tanks are to be capable of giving at least 20 air changes per hour and of being operated from outside the compartment in accordance with the following **(1)** or **(2)**~~ The requirements specified in **-1** also apply to the following closed compartments normally entered by persons:

(1) ~~In cases where the tanks are adjacent to the engine room, the requirements of **-2** above apply.~~when they are adjacent to the integral storage tank for chemical treatment fluids and there are possible leak points (e.g. manhole, fittings) from these tanks; or

(2) ~~In cases where the tanks are adjacent to enclosed compartments normally entered by ship personnel, the requirements of **-1** above apply~~when the treatment fluid piping systems pass through these compartments, unless the piping system is made of steel or other equivalent material with melting point above 925 °C and with fully welded joints.

Paragraph 22.4.3 has been added as follows.

22.4.3 Venting Systems of Storage Tanks for Chemical Treatment Fluids

1 The vent pipes of the storage tank are to terminate in a safe location on the weather deck and the tank venting system is to be arranged to prevent entrance of water into the tank for chemical treatment fluids.

2 Storage tanks for chemical treatment fluids are to be arranged so that they can be emptied of the fluids and ventilated by means of portable or permanent systems.

Paragraph 22.4.3 has been renumbered to 22.4.4, and Paragraph 22.4.4 has been amended as follows.

22.4.~~3~~4 Safety Devices and Alarm Devices

1 Exhaust gas cleaning systems are to be fitted with safety devices which are capable of automatically stopping exhaust gas washwater supply pumps and ~~sodium hydroxide solution~~ chemical treatment fluids pumps in the event of any of the following failures:

(1) Abnormal increase of the liquid level in the scrubber

(2) Abnormal increase of the pressure at the inlet or the differential pressure across the scrubber chamber (in cases where changeover devices for exhaust gas pipes are not fitted)

2 (Omitted)

3 Alarm devices, to be activated in the event of any of the abnormal conditions given in **Table D22.1**, are to be provided at control stations of exhaust gas cleaning systems.

4 Exhaust gas cleaning systems are to be fitted with monitoring devices at control stations for exhaust gas cleaning systems, and these devices are to indicate the information listed in **(1)** to **(5)**:

(1) Liquid levels in scrubber chambers

(2) Liquid levels in tanks for ~~sodium hydroxide solutions~~chemical treatment fluids

(3) Temperatures in tanks for ~~sodium hydroxide solutions~~chemical treatment fluids **(**where the heating and/or cooling systems specified in **-6** are provided)

(4) Exhaust gas temperatures at outlets

(5) Pressures at inlets or differential pressures across scrubber chambers

5 (Omitted)

6 Each storage tank for chemical treatment fluids is to be provided with level monitoring arrangements and high/low level alarms. In cases where heating and/or cooling systems are provided, high and/or low temperature alarms or temperature monitoring are also to be provided accordingly.

Table D22.1 Alarm points for exhaust gas cleaning system(1)

Monitored Variables	
Liquid level in scrubber chamber	H
Temperature of washwater supply (in cases where the washwater includes ~~sodium hydroxide solutions~~chemical treatment fluids) (2)	H
Liquid levels in tank for ~~sodium hydroxide solution~~chemical treatment fluids	H L
Temperature in tank for ~~sodium hydroxide solution~~chemical treatment fluids (3)	H L
Exhaust gas pressure at the inlet (~~2~~4)	H
Exhaust gas temperature at the outlet	H
Power loss of control, alarm, monitoring or safety devices	○

Notes:

(1) “H” and “L” mean “high” and “low”. “ ” means abnormal condition occurred.

(2) To detect high washwater temperature due to abnormal conditions of heat exchangers; however, alarms need not be fitted in cases where heat exchangers are not used.

(3) This alarm is not required when heating and/or cooling systems are not provided.

(~~2~~4) Differential pressure across scrubber chamber may be accepted in lieu.

22.6 Safety and Protective Equipment

Paragraph 22.6.1 has been amended as follows.

22.6.1 General

1 For the protection of crew members, ~~the safety and protective equipment specified in **(1)** to **(4)** is to be stored at locations outside the compartment containing the exhaust gas cleaning system and easily accessible in the event of any leakages of liquids containing sodium hydroxide solutions. The safety and protective equipment is to cover all skin so that no part of the body is unprotected. The locations at which the equipment is stored are to be clearly marked so as to be easily identifiable~~the ship is to have on board suitable personnel protective equipment. The number of personnel protective equipment carried onboard is to be appropriate for the number of personnel engaged in regular handling operations or that may be exposed in the event of a failure; but in no case is there to be less than two sets available onboard.

2 Personnel protective equipment is to consist of the following.

(1) Large apron of chemical-resistant material

(2) Special gloves with long sleeves

(3) Suitable footwear

(4) Suitable protective equipment consisting of coveralls and tight-fitting goggles or face shields or both

~~**2**~~**3** Eyewash and safety showers are to be ~~located in the vicinity of sodium hydroxide solution filling stations and sodium hydroxide solution supply pumps.~~provided, the location and number of eyewash stations and safety showers are to be derived from the detailed installation arrangements. As a minimum, the following stations are to be provided:

(1) In the vicinity of transfer or treatment pump locations for chemical treatment fluids. If there are multiple transfer or treatment pump locations on the same deck then one eyewash and safety shower station may be considered for acceptance provided that the station is easily accessible from all such pump locations on the same deck.

(2) An eyewash station and safety shower is to be provided in the vicinity of a chemical bunkering station on-deck. If the bunkering connections are located on both port and starboard sides, then consideration is to be given to providing two eyewash stations and safety showers, one for each side.

(3) An eyewash station and safety shower is to be provided in the vicinity of any part of the system where a spillage/drainage of chemical treatment fluids may occur and in the vicinity of system connections/components of the fluids that require periodic maintenance.

22.7 Tests

Paragraph 22.7.1 has been amended as follows.

22.7.1 Tests at Facilities (Shop tests)

1 ~~Sodium hydroxide solution~~Chemical treatment fluids independent storage tanks are to be subjected to hydrostatic tests at ~~a pressure corresponding to a water head of 2.5 *m* above the top~~

~~plate~~pressures corresponding to the maximum heights of fluid columns in overflow pipes, with a minimum of 2.4 *m* above the top plate taking into consideration the specific density of the treatment fluid.

2 After completion of the fabrication process, piping, valves and pipe fittings, for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids, design pressure of which exceeds 0.35 *MPa* are to be subjected to hydrostatic tests together with the welded fittings at a pressure equal to 1.5 times the design pressure.

3 The pressure parts of ~~sodium hydroxide solution~~chemical treatment fluids supply pumps and washwater supply pumps are to be subjected to hydrostatic tests at a pressure equal to 1.5 times the design pressure or 0.2 *MPa*, whichever is greater. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

4 For ~~sodium hydroxide solution~~chemical treatment fluids supply pumps and washwater supply pumps, shop trials are to be carried out according to test procedures deemed appropriate by the Society. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

5 Electrical motors and their corresponding control gears used for ~~sodium hydroxide solution~~chemical treatment fluids supply pumps and washwater supply pumps are to be tested in accordance with relevant requirements in **Part H**. Shop tests for electrical motors whose continuous rated capacities are less than 100 *kW* and their corresponding control gears may be replaced by manufacturer tests. In such cases, submission or presentation of test records may be required by the Society.

Paragraph 22.7.2 has been amended as follows.

22.7.2 Tests after Installation On Board

1 In cases where ~~sodium hydroxide solutions~~chemical treatment fluids are carried in tanks which form part of the ship's hull, the tanks are to be subjected to hydrostatic tests in accordance with **2.1.5(1), Part B**. Where the specific gravities of the liquids used for the tests are less than those of the ~~sodium hydroxide solution~~chemical treatment fluids, an appropriate additional head is to be considered.

2 After installation on board, exhaust gas cleaning systems are to be tested in accordance with the following:

(1) Piping systems for liquids containing ~~sodium hydroxide solutions~~chemical treatment fluids (except overboard discharge pipes) are to be subjected to leak tests at pressures equal to 1.5 times the design pressure or 0.4 *MPa*, whichever is greater.

((2) to (4) are omitted.)

EFFECTIVE DATE AND APPLICATION (Amendment 1-10)

1. The effective date of the amendments is 1 July 2022.
2. Notwithstanding the amendments to the Rules, the current requirements apply to EGCS whose applications for approval are submitted to the Society before the effective date installed on ships for which the date of contract for construction* is before the effective date.
3. Notwithstanding the provision of preceding 2., the amendments to the Rules may apply to EGCS whose applications for approval are submitted to the Society before the effective date installed on ships for which the date of contract for construction* is before the effective date upon request of the owner.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.
2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.
For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:
(1) such alterations do not affect matters related to classification, or
(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.
The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.
3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.
4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D **Machinery Installations**

GUIDANCE

2022 AMENDMENT NO.1

Notice No.31 30 June 2022

Resolved by Technical Committee on 26 January 2022

Notice No.31 30 June 2022
AMENDMENT TO THE GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

"Guidance for the survey and construction of steel ships" has been partly amended as follows:

Part D MACHINERY INSTALLATIONS

Amendment 1-1

D1 GENERAL

D1.1 General

Paragraph D1.1.4 has been amended as follows.

D1.1.4 Modification of Requirements

For those machinery installations specified in **1.1.4, Part D of the Rules** (excluding those specified in other ~~P~~parts of the Rules), some requirements of **Part D of the Rules** may be modified as follows:

(1) Prime movers (including power transmission systems and shafting systems~~; hereinafter the same~~) driving generators, auxiliary machinery essential for main propulsion and auxiliary machinery for manoeuvring and the safety~~:~~.
  - (a) Prime movers with an output less than 100 *kW*
    - i) Submission of drawings may be omitted.
    - ii) Materials which comply with the requirements of any national standard may be accepted for the principal components. In this case, materials (excluding valves and pipe fittings) are to be manufactured by a manufacturer approved by the Society.
    - iii) Shop tests in the presence of the Surveyor may be substituted for manufacturer's tests. In this case, submission or presentation of test records may be required by the ~~S~~surveyor.
  - (b) Prime ~~M~~movers with an output not less than 100 *kW* but less than 375 *kW*~~:~~
    - i) Materials used for principal components may be dealt with under the requirements specified in **(a)ii)**.
    - ii) Hydrostatic tests as well as dynamic balancing tests, overspeed tests and trial runs of turboblowers at the manufacturer may be dealt with under the requirements specified in **(a)iii)**.

(2) Prime movers (including power transmission systems and shafting systems) for auxiliary machinery for cargo handling~~:~~.
  - (a) Prime movers with an output less than 375 *kW* may be dealt with under the requirements of **(1)(a)**.
  - (b) Prime movers with an output 375 *kW* or over may be dealt with under the requirements of **(1)(b)**.

((3) to (8) are omitted.)

D6 SHAFTINGS

D6.1 General

Paragraph D6.1.2 has been deleted.

~~**D6.1.2 Drawings and Data**~~

~~The “Shaft alignment calculation sheets” referred to in **6.2.1(1)(l)viii), Part D of the Rules** mean those in accordance with **Annex D6.2.13**.~~

D6.2 Materials, Construction and Strength

Paragraph D6.2.2 has been deleted.

~~**D6.2.2 Intermediate Shafts**~~

~~The wording “where deemed appropriate by the Society” in **6.2.2-1, Part D of the Rules** means cases where the intermediate shaft is manufactured using steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800~~ $\cancel{N/mm^2}$ ~~and are in accordance with the requirments of **Annex D6.2.2 “GUIDANCE FOR USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS”**.~~

D6.2.4 Propeller Shafts and Stern Tube Shafts

1 As for the diameter of propeller shaft Kind 2 or stern tube shafts Kind 2 made of carbon steel or low alloy steel, the wording “to be deemed appropriate by the Society” means to calculate the required diameter by the following formula:

$$d_s = 100k_3 \cdot \sqrt[3]{\frac{H}{N_0}}$$

d_s : Required diameter of propeller shaft (*mm*)
H : Maximum continuous output of main propulsion machinery (*kW*)
N_0 : Number of revolutions of shaft at maximum continuous output (*rpm*)
k_3 : Factor concerning shaft design, given in **Table D6.2.4-1**

2 The value of k_3 for propeller shafts and stern tube shafts made of stainless steel forgings, etc. other than those indicated in the **Table D6.4** which is for k_3 specified in **6.2.4-2, Part D of the Rules**, is to be in accordance with **Table D6.2.4-2**. Furthermore, this requirement may be applied to propeller shafts Kind 2 and stern tube shafts Kind 2.

Note of Table D6.2.4-1 has been amended as follow.

Table D6.2.4-1 Values of k_3

	Application	k_3
1	The portion from the big end of the tapered part of a propeller shaft (in the case of a flange connected propeller, the forward end of the of the flange) to the forward end of the after most stern tube bearing or to 2.5 d_s, whichever is larger	1.33
2	Excluding any portion specified in 1 above, the portion in the direction toward the bow side up to the forward end of the forward stern tube sealing assembly	1.21(1)
3	The portion between the forward end of the forward stern tube sealing assembly and the intermediate shaft coupling	1.21(2)

Notes:

(1) The diameter of the boundary portion ~~should~~is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

(2) The diameter may be reduced~~, by either a smooth taper or a blending radius nearly equal to the change in diameter, up to the diameter calculated by the formula given~~ in accordance with **6.2.~~2~~4.-~~1~~3, Part D of the Rules** ~~where it is assumed that T_s = 400 N/mm^2~~.

Note of Table D6.2.4-2 has been amended as follow.

Table D6.2.4-2 Values of k_3

Application		Shaft material	
		Austenitic stainless steel with 0.2 % proof stress not less than 205 N/mm^2	Precipitation hardened martensite stainless steel with 0.2 % proof stress not less than 400 N/mm^2
1	The portion from the big end of the tapered part of a propeller shaft (in the case of a flange connected propeller, the forward end of the flange) to the forward end of the after most stern tube bearing or to 2.5 d_s, whichever is larger	1.28	1.05
2	Excluding the portion shown in 1 above, the portion in the direction toward the bow side up to the forward end of the forward stern tube sealing assembly	1.16(1)	0.94(1)
3	The portion between the forward end of the forward stern tube sealing assembly and the intermediate shaft coupling	1.16(2)	0.94(2)

Notes:

(1) The diameter of the boundary portion ~~should~~is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

(2) The diameter may be reduced~~, by either a smooth taper or a blending radius nearly equal to the change in diameter, up to the diameter calculated by the formula given~~ in accordance with **6.2.~~2~~4-~~1~~3, Part D of the Rules** ~~where it is assumed that T_s = 400 N/mm^2~~.

D6.2.10 Stern Tube Bearings and Shaft Bracket Bearings

Sub-paragraphs -1 to -3 have been amended as follows.

1 The wording "provisions specified elsewhere" in **6.2.10-1(1)(a)i), Part D of the Rules** means the following **(1)** and **(2)** in principle:

~~When the length of a bearing is less than twice the required diameter in accordance with **6.2.10-1(1)(a)i), Part D of the Rules**, the following **(1)** and **(2)** are, in principle, to be satisfied.~~

(1) Shaft alignment calculations are to be carried out in accordance with the requirements in **Annex ~~D~~6.2.13 "~~GUIDANCE FOR~~ CALCULATION OF SHAFT ALIGNMENT", Part D of the Rules**.

(2) For improving the lubricating condition of the bearing, the following measures are to be taken:

  (a) A lubricating oil inlet is to be provided at the aft end of the bearing to ensure the forced circulation of the lubricating oil.

  (b) Either of the following devices to measure stern tube bearing metal temperature at the aft end bottom along with high temperature alarms (with a preset value of 60 °C or below) is to be provided:

    i) Two or more temperature sensors embedded in the metal; or

    ii) An embedded temperature sensor, replaceable from inboard the ship, and a spare temperature sensor.
    In this case, the replacement of such sensors according to procedures submitted beforehand is to be demonstrated.

  (c) Low level alarms are to be provided for lubricating oil sump tanks.

2 The wording "~~construction and arrangement specially approved by the Society~~provisions specified elsewhere" in **6.2.10-1(1)(b)ii), Part D of the Rules** means the following **(1)** and **(2)** in principle:

~~When the length of a bearing is less than twice the required diameter in accordance with **6.2.10-1(1)(b)ii), Part D of the Rules**, the following **(1)** and **(2)** are, in principle, to be satisfied.~~

(1) Nominal bearing pressure, etc. calculated in accordance with **Annex ~~D~~6.2.13 "~~GUIDANCE FOR~~ CALCULATION OF SHAFT ALIGNMENT", Part D of the Rules** are to be within the allowable limits specified in the Type Approval Certificate.

(2) The measures for lubricating condition specified in **-1(2)** are to be taken.

3 The wording "provisions specified elsewhere" in **6.2.10-1(2)(b), Part D of the Rules** means the following **(1)** and **(2)** in principle:

~~When the length of a bearing is less than 4 *times* the required diameter of the propeller shaft or less than 3 *times* the actual diameter, whichever is greater, in accordance with **6.2.10-1(2)(b), Part D of the Rules**, the following **(1)** and **(2)** are, in principle, to be satisfied.~~

(1) Nominal bearing pressure is to be within the allowable limit specified in the Type Approval Certificate.

(2) Forced lubrication using water pumps is to be adopted and a low flow alarm is to be provided at the lubricating water inlet.

Paragraph D6.2.13 has been deleted.

~~**D6.2.13 Shaft Alignment**~~

~~For the approval of the shaft alignment calculation required in **6.2.13, Part D of the Rules**, a calculation sheet in accordance with **Annex D6.2.13 "GUIDANCE FOR CALCULATION OF SHAFT ALIGNMENT"** is to be submitted.~~

D8 TORSIONAL VIBRATION OF SHAFTINGS

D8.2 Allowable Limit

D8.2.2 Intermediate Shafts, Thrust Shafts, Propeller Shafts and Stern Tube Shafts

Sub-paragraph -3 has been deleted.

~~**3** The wording “where deemed appropriate by the Society” in **8.2.2-1(1), Part D of the Rules** means cases where the intermediate shaft is manufactured using steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800 *N/mm*² and are in accordance with the requirments of **Annex D6.2.2 “GUIDANCE FOR USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS”**.~~

D8.2.6 Detailed Evaluation for Strength

Sub-paragraph -3 has been deleted.

~~**3** In cases where intermediate shafts with longitudinal slots given in **Table D8.1, Part D of the Rules** are equipped, the value of C_K may be determined by using the following formulae:~~

$$\cancel{C_K = 1.45/scf}$$

$$\cancel{scf = \alpha_{t(hole)} + 0.80\frac{(l-e)/d_a}{\sqrt{\left(1-\frac{d_i}{d_a}\right)\frac{e}{d_a}}}}$$

~~where~~

~~scf : Stress concentration factor at the end of slots defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress determined for the hollow shafts without slots (Values obtained through Finite Element Calculation may be used as well)~~

~~l : Slot length~~

~~e : Slot width~~

~~d_i : Inside diameter of the hollow shaft at the slot~~

~~d_a : Outside diameter of the hollow shaft~~

~~$\alpha_{t(hole)}$: Stress concentration factor of radial holes (in this context, e = hole diameter) determined by the following formula (an approximate value of 2.3 may be used as well)~~

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

D13 PIPING SYSTEMS

D13.5 Bilge and Ballast Piping

D13.5.10 Dewatering Arrangements for Bulk Carriers, etc.

Sub-paragraph -2(4) has been amended as follows.

2 With respect to the provisions of **13.5.10, Part D of the Rules**, bilge and ballast systems for the dewatering arrangements (hereinafter, referred to as "the dewatering systems") are to comply with the following requirements:

((1) to (3) are omitted.)

(4) Enclosures of electrical equipment for the dewatering systems installed in spaces where the systems are to be installed, are to provide protection to the IP68 standard as defined in *IEC* 60529:1989/AMD2:2013/COR1:2019 for a water head equal to the height of the space in which the electrical equipment is installed for a time duration of at least 24 *hours*.

((5) and (6) are omitted.)

D13.8 Sounding Pipes

D13.8.5 Water Level Detection and Alarm Systems for Bulk Carriers, etc.

Sub-paragraph -3(3) has been amended as follows.

3 The wording "the systems to have constructions and functions deemed appropriate by the Society" in **13.8.5-1(4), Part D of the Rules** means those systems complying with the following requirements and being of a type approved by the Society in accordance with the provisions of **Chapter 5, Part 7 of the Approval and Type Approval of Materials and Equipment for Marine Use** or those systems approved by an organization deemed appropriate by the Society in accordance with the Resolution *MSC*.188(79).

((1) and (2) are omitted.)

(3) Electrical installations for the systems installed in the following areas are to be of an intrinsically safe type of *Exib* complying at least with *IEC* 60079-11:2011 and the maximum surface temperature of the installations is not to exceed 85 ºC, except electrical installations installed in ships designed only to carry cargo which are not combustible or explosive atmosphere. In addition, in cases where a ship is designed to carry only limited kinds of cargo, the maximum surface temperature may be appropriately relaxed depending on the kind of cargo. In this case, such limitations relating to cargo are to be documented in booklets for cargo operations. Finally, those electric installations installed at the edges of the following areas are to be approved at the discretion of the Society with due consideration being given to their design with respect to gas-tightness, etc.
    (a) Cargo holds
    (b) Enclosed spaces adjacent to cargo holds having openings without a gas-tight or watertight door/hatch and the like into a hold
    (c) Areas within 3 *m* of any cargo hold mechanical exhaust ventilation outlet

((4) to (8) are omitted.)

Annex D6.2.2 has been deleted.

~~Annex D6.2.2 GUIDANCE FOR USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS~~

~~(Omitted)~~

Annex D6.2.13 has been deleted.

~~Annex D6.2.13 GUIDANCE FOR CALCULATION OF SHAFT ALIGNMENT~~

~~(Omitted)~~

EFFECTIVE DATE AND APPLICATION (Amendment 1-1)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to ships for which the date of contract for construction* is before the effective date.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.

For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:

(1) such alterations do not affect matters related to classification, or

(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.

The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

Amendment 1-2

D2 RECIPROCATING INTERNAL CONBUSTION ENGINES

D2.1 General

D2.1.3 Drawings and Data

Sub-paragraphs (1) and (2) have been amended as follows.

For the following data, those represented by two sizes in generic range of turbochargers (i.e. the same components, materials, etc., with the only difference being the size) are acceptable.

(1) The documentation for safe torque transmission specified in **(34)(a), Table D2.1(b)** **~~2.1.3-1(2)(i)i)~~, Part D of the Rules**.

(2) The operation and maintenance manuals listed in **(34)(c), Table D2.1(b)~~2.1.3-1(2)(i)iii)~~, Part D of the Rules**.

D10 PRESSURE VESSELS

D10.9 Tests

D10.9.1 Shop Tests

Sub-paragraph -2 has been amended as follows.

2 Notwithstanding the requirements in **10.9.1, Part D of the Rules**, hydrostatic tests of heat exchangers fitted to engines having cylinder bores of 300 *mm* or less may be omitted~~,~~ (see **Table D2.7~~6~~ of the Rules**).

EFFECTIVE DATE AND APPLICATION (Amendment 1-2)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements may apply to reciprocating internal combustion engines for which the application for approval is submitted to the Society before the effective date.

Amendment 1-3

D2 RECIPROCATING INTERNAL CONBUSTION ENGINES

Section D2.3 has been amended as follows.

D2.3 Crankshafts

D2.3.1 Solid Crankshafts and Semi-Build Crankshafts

1 In applying **2.3.1-4, Part D of the Rules**, solid crankshaft and semi-built crankshaft approvals are to be according to the following.

2 The diameters of crankpins and journals are to be not less than the value given by the following formula:

$$d_c = \left\{ \left(M + \sqrt{M^2 + T^2} \right) D^2 \right\}^{\frac{1}{3}} K_m K_s K_h$$

where

d_c : Required diameter of crankshaft (*mm*)

M : $10^{-2} ALP_{max}$

T : $10^{-2} BSP_{mi}$

S : Length of stroke (*mm*)

L : Span of bearings adjacent to crank measured from centre to centre (*mm*)

P_{max} : Maximum combustion pressure in cylinder (*MPa*)

P_{mi} : Indicated mean effective pressure (*MPa*)

A and B : Coefficients given in **Table D2.3.1-2** for engines having equal firing intervals (in the case of Vee type engines, those with equal firing intervals on each bank.). Special consideration will be given to values A and B for reciprocating internal combustion engines having unequal firing intervals or for those not covered by the Tables.

D : Cylinder bore (*mm*)

K_m : Value given by the following **(1)** or **(2)** in accordance with the specified tensile strength of the crankshaft material. However, the value of K_m for materials other than steel forgings and steel castings is to be determined by the Society in each case.

(1) In cases where the specified tensile strength of material exceeds 440 N/mm^2

$$K_m = \sqrt[3]{\frac{440}{440 + \frac{2}{3}(T_s - 440)}}$$

where

T_s : Specified tensile strength of material (N/mm^2)

The value of T_s is not to exceed 760 N/mm^2 for carbon steel forgings and 1080 N/mm^2 for low alloy steel forgings.

(2) In cases where the specified tensile strength of material is not more than 440 N/mm^2 but not less than 400 N/mm^2

$K_m = 1.0$

K_s : Value given by the following **(1)**, **(2)**, or **(3)** in accordance with the manufacturing method of crankshafts.

(1) In cases where the crankshafts are manufactured by a special forging process approved by the Society as well as where the product quality is stable and the fatigue strength is

considered to be improved by 20 % or more in comparison with that of the free forging process

$$K_s = \sqrt[3]{\frac{1}{1.15}}$$

(2) In cases where the crankshafts are manufactured by a manufacturing process using a surface treatment approved by the Society as well as where the product quality is stable and the fatigue strength is recognized as being superior

$$K_s = \sqrt[3]{\frac{1}{1+\rho/100}}$$

where

ρ : Degree of improvement in strength approved by the Society relative to the surface hardening (%)

(3) In cases other than **(1)** and **(2)** above

$K_s = 1.0$

K_h : Value given by the following **(1)** or **(2)** in accordance with the inside diameter of the crankpins or journals.

(1) In cases where the inside diameter is one-third or more than that of the outside diameter

$$K_h = \sqrt[3]{\frac{1}{1-R^4}}$$

where

R : Quotient obtained by dividing the inside diameter of a hollow shaft by its outside diameter

(2) In cases where the inside diameter is less than one-third of the outside diameter

$K_h = 1.0$

Table D2.3.1-2(1) Value of Coefficients A and B for Single Acting In-line Engines

Number of cylinders	2-stroke cycle		4-stroke cycle	
	A	B	A	B
1		8.8		4.7
2		8.8		4.7
3		10.0		4.7
4		11.1		4.7
5		11.4		5.4
6	1.00	11.7	1.25	5.4
7		12.0		6.1
8		12.3		6.1
9		12.6		6.8
10		13.4		6.8
11		14.2		7.4
12		15.0		7.4

Table D2.3.1-2(2) Value of Coefficients *A* and *B* for Single Acting 2-stroke cycle Vee Type Engines with Parallel Connecting Rods

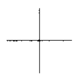
Number of cylinders	Minimum firing interval between two cylinders on one crankpin					
	45°		60°		90°	
	A	*B*	*A*	*B*	*A*	*B*
6	1.05	17.0	1.00	12.6	1.00	17.0
8		17.0		15.7		20.5
10		19.0		18.7		20.5
12		20.5		21.6		20.5
14		22.0		21.6		20.5
16		23.5		21.6		23.0
18		24.0		21.6		23.0
20		24.5		24.2		23.0

Table D2.3.1-2(3) Value of Coefficients *A* and *B* for Single Acting 4-stroke cycle Vee Type Engines with Parallel Connecting Rods

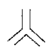
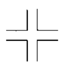
Number of cylinders	Minimum firing interval between two cylinders on one crankpin											
	45°		60°		90°		270°		300°		315°	
	A	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*
6	1.60	4.1	1.47	4.0	1.40	4.0	1.40	4.0	1.30	4.4	1.20	4.3
8		5.5		5.5		5.5		5.5		5.3		5.2
10		6.7		7.0		6.5		6.5		6.1		5.9
12		7.5		8.2		7.5		7.5		6.9		6.6
14		8.4		9.2		8.5		8.5		7.5		7.3
16		9.3		10.1		9.5		9.5		8.2		7.9
18		10.1		11.1		10.5		10.5		8.8		8.5
20		11.5		14.0		11.5		11.5		9.5		9.2

Table D2.3.1-~~1~~2(4) Values of Coefficients *A* and *B* (In cases of Unequal Firing Intervals)

(1) 4-stroke cycle in-line engines

Number of cylinders	Arrangement of crank	*A*	*B*
4		1.25	4.7

(2) 2-stroke cycle vee engines

Number of cylinders	Minimum firing interval between two clyinders on one crankpin	Arrangement of crank	*A*	*B*
12	60°		1.00	21.6
12	60°		1.00	15.0
16	60°		1.00	26.3

~~**1** Coefficients *A* and *B* for engines having unequal firing intervals are to be in accordance with **Table D2.3.1-1**.~~

~~**2**~~**3** In cases where the diameter of crankpins or journals is less than the required diameter d_c given in ~~**2.3.1-1, Part D of the Rules**~~**-2** above, consideration will be given in each case on the basis of the stress levels in fillets, the torsional stress levels in crankpins and journals and the material of the crankshaft. In this connection, the stress levels in fillets are to be in accordance with the following ~~**(1) or (2)**~~:

~~(1)~~ In cases where the torsional stress in crankpins and journals are evaluated without carrying out a forced vibration calculation including the stern shaftings:

The diameter may be acceptable where the value of equivalent stress amplitude σ_e calculated by the **Annex D2.3.1**~~**-2(1)**~~ "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS** ~~**I**~~" is not more than the allowable stress σ obtained from the formula below with the coefficient shown in **Table D2.3.1-**~~**2**~~**3.**

$\sigma = \sigma_a \cdot f_m \cdot f_s + \alpha$ (N/mm^2)

However, where deemed appropriate by the Society, the diameter in consideration of the allowable stress of crankshafts, including fillet parts, that have been hardened by surface treatment and the resultant stress distribution may be acceptable.

~~(2) In cases where the torsional stress in crankpins and journals are evaluated by carrying out a forced vibration calculation including the stern shaftings: The diameter may be acceptable where the value of the acceptability factor *Q* calculated by the **Annex D2.3.1-2(2)** "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS II**" complies with the following formula:~~

$\cancel{Q \geq 1.15}$

4 The dimensions of crank webs are to comply with the following requirements:

(1) The thickness and breadth of crank webs, the diameters of the crankpins and journals, are to comply with the conditions of the following formula. However, the thickness of crank webs is to be not less than 0.36 *times* the diameter of crankpins and journals. When the actual diameters of the crankpin and journal are larger than the required diameter of the crankshaft as determined by the formula in **-2**, the left side of the following formula may be multiplied by $(d_c/d_a)^3$.

$\{0.122(2.20 - b/d_a)^2 + 0.337\}(d_a/t)^{1.4} \leq 1$

where

b : Breadth of crank web (*mm*)

d_a : Actual diameter of crankpin or journal (*mm*)

t : Thickness of crank web (*mm*)

(2) The radius in fillets at the junctions of crank webs with crankpins or journals is to be not less than 0.05 *times* the actual diameter of crankpins or journals, respectively.

~~**3**~~**5** In cases where the dimensions of crankwebs fail to meet the requirements specified in ~~**2.3.1-2(1), Part D of the Rules**~~**-4(1)** above, consideration will be given in accordance with the following:

(1) The dimensions of the crankwebs may be acceptable in cases where the actual diameters of crankpins and journals are not less than the required diameter d_c calculated by ~~**2.3.1-1, Part D of the Rules**~~**-2** by replacing *M* and *T* with those specified below.

In this case, the dimensions are to be within the following ranges;

$0 \leq q/r \leq 1,\ -0.3 \leq h/d \leq 0.4,\ 8 \leq d/r \leq 27$

$1.1 \leq b/d \leq 2.1,\ 0.2 \leq t/d \leq 0.56$

$M = 10^{-2} A P_{\max} L \alpha_{KB}/5$

$T = 10^{-2} B P_{mi} S \alpha_{KT}/1.8$

where

α_{KB} : Stress concentration factor for bending, as specified below;

$$\alpha_{KB} = 4.84 f_1 f_2 f_3 f_4 f_5$$

$$f_1 = 0.420 + 0.160\sqrt{d/r - 6.864}$$

$$f_5 = 1 + 81[0.769 - (0.407 - h/d)^2]$$
$$\times (q/r)\ (r/d)^2$$

$$f_3 = 0.285(2.2 - b/d)^2 + 0.785$$

$$f_4 = 0.444(d/t)^{1.4}$$

$$f_5 = 1 - [(h/d + 0.1)^2/(4t/d - 0.7)]$$
$$\cdots (t/d \geq 0.36)$$
$$= 1 - 1.35(h/d + 0.1)^2$$
$$\cdots (t/d < 0.36 \text{ and } h/d > -0.1)$$
$$= 1 \quad \cdots (t/d < 0.36 \text{ and } h/d \leq -0.1)$$

α_{KT} : Stress concentration factor for torsion, as specified below;

$$\alpha_{KT} = 1.75 g_1 g_2 g_3$$

$$g_1 = 31.6(0.152 - r/d)^2 + 0.67$$

$$g_2 = 1.04 + 0.317h/d$$

$$g_3 = 1.31 - 0.233b/d$$

d : actual diameter of crankpin or journal (*mm*)

r : radius in fillet (*mm*)

q : recess (*mm*)

h : overlap between crankpin and journal (*mm*)

$$h = (d_p + d_j - S)/2$$

~~Other symbols are the same as those used in **2.3.1, Part D of the Rules**.~~

(2) In cases where the dimensions of the crankwebs fail to meet the requirements even after applying **(1)** above, the acceptance criteria specified below may be used:

~~(a)~~ In cases where the torsional stresses in crankpins and journals are evaluated without carrying out a forced vibration calculation including the stern shaftings:

The dimensions may be acceptable in cases where the value of the equivalent stress amplitude σ_e calculated by the **Annex D2.3.1~~-2(1)~~** "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS~~ I~~**" is not more than the allowable stress σ obtained from the formula below with the coefficient shown in **Table D2.3.1-~~2~~3.**

$$\sigma = \sigma_a \cdot f_m \cdot f_s + \alpha \quad (N/mm^2)$$

However, where deemed appropriate by the Society, the dimensions in consideration of the allowable stress of crankshafts, including fillet parts, that have been hardened by surface treatments and the resultant stress distribution may be acceptable.

~~(b) In cases where the torsional stresses in crankpins and journals are evaluated by carrying out a forced vibration calculation including the stern shaftings:~~

~~The dimensions may be acceptable where the value of the acceptability factor Q calculated by the **Annex D2.3.1-2(2) "GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS II"** complies with the following formula:~~

~~$Q \geq 1.15$~~

~~4~~6 The dimensions of the crankweb breadth *b*, crankweb thickness *t*, radius in fillet *r* and recess *q* used for ~~**2.3.1-2, Part D of the Rules** and -3~~ **-4** and **-5** above are to be in accordance with the following (*See* **Fig. D2.3.1 -1**):

(1) As for "*b*", the breadth on the perpendicular bisector of the line between the crankpin centre

and journal centre is to be used.

(2) As for "t", the thickness at the same section specified in **(1)** is to be used. In this case, the recess q need not be accounted in the thickness even when it is provided.

(3) As for "r", the radius connecting to the crankpin or journal is to be used when a composite radius is provided.

7 Semi-built crankshafts are to be in accordance with **D2.3.2**.

Table D2.3.1-~~2~~3 Coefficient of Allowable Stress at Fillet

<table>
<tr><td rowspan="5">σ_a
(N/mm^2)</td><td rowspan="2">Stroke cycle of engine</td><td rowspan="2">Type of crankshaft</td><td colspan="3">Shaft diameter ϕ (1) (mm)</td></tr>
<tr><td>$d \geq 200$</td><td>$200 > d \geq 100$</td><td>$100 > d$</td></tr>
<tr><td rowspan="2">2-cycle</td><td>Semi-built-up</td><td>54</td><td>——</td><td>——</td></tr>
<tr><td>Solid</td><td>74</td><td>142-0.34d</td><td rowspan="2">108</td></tr>
<tr><td>4-cycle</td><td>Solid</td><td>83</td><td>133-0.25d</td></tr>
<tr><td>f_m</td><td colspan="5">$1+\frac{2}{3}\left(\frac{T_s^{(2)}}{440}-1\right)$</td></tr>
<tr><td rowspan="3">f_s</td><td colspan="5">Manufacturing method</td></tr>
<tr><td>Ordinary method</td><td colspan="2">Method (1) for K_s specified in ~~2.3.1-1,~~-2 ~~Part D of the Rules~~</td><td colspan="2">Method (2) for K_s specified in ~~2.3.1-1,~~-2 ~~Part D of the Rules~~</td></tr>
<tr><td>1</td><td colspan="2">1.15</td><td colspan="2">1+ ρ (3)/100</td></tr>
<tr><td rowspan="3">α
(N/mm^2)</td><td colspan="5">Main bearing material</td></tr>
<tr><td colspan="3">White metal</td><td colspan="2">Aluminum or kelmet</td></tr>
<tr><td colspan="3">0</td><td colspan="2">10</td></tr>
</table>

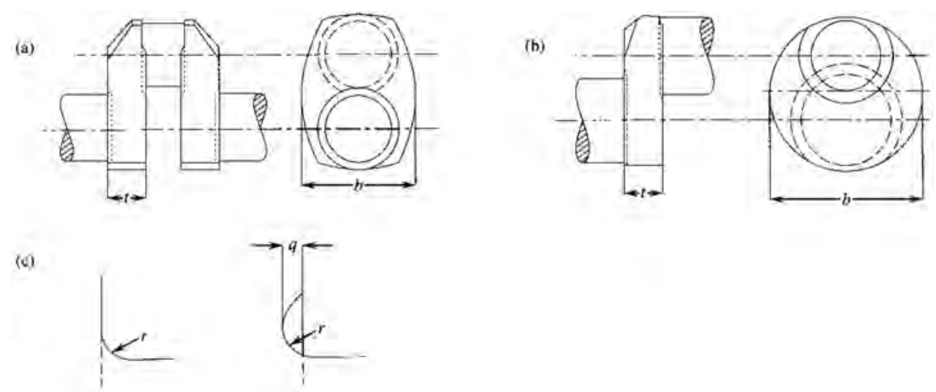
Notes:

(1) d is to be the actual diameter of crankpin or journal, whichever is larger.

(2) T_s signifies the minimum specified tensile strength (N/mm^2) of the crankshaft materials. The limit of T_s for computing f_m is to be in accordance with the requirements in ~~2.3.1-1, Part D of the Rules~~-2.

(3) ρ signifies the degree of strength improvement (%) approved by the Society relative to surface hardening.

Fig. D2.3.1-1 Dimensions for Webs of Solid Crankshafts



D2.3.2 Built-up Crankshafts

1 In applying **2.3.2, Part D of the Rules**, built-up crankshaft approval is to be in accordance with the followings.

2 The dimensions of crankpins and journals of built-up crankshafts are to comply with the following requirements in **(1)** and **(2)**:

(1) The diameters of crankpins and journals are to comply with the requirements in **D2.3.1-2**.

(2) The diameters of axial bores in journals are to comply with the following formula:

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

D_{BG} : Diameter of axial bore in journal (*mm*)
D_S : Journal diameter at the shrinkage fit (*mm*)
S_R : Safety factor against slipping (a value not less than 2 is to be taken)
M_{max} : Absolute maximum torque at the shrinkage fit ($N \cdot m$)
μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)
L_S : Length of shrinkage fit (*mm*)
σ_{SP} : Minimum yield strength of material used for journal (N/mm^2)

~~1~~**3** The wording "maximum torque at the shrinkage fit" in ~~**2.3.2-1(2), Part D of the Rules**~~**-2(2)** above means, in principle, $M_{T\max}$ shown in **1.3.2-1 of the Annex** ~~**D**~~**2.3.1**~~**-2(2)**~~ "~~**GUIDANCE FOR**~~ **CALCULATON METHOD OF CRANKSHAFT STRESS** ~~**II**~~".

4 The dimensions of crank webs are to comply with the following requirements in **(1)** and **(2)**:

(1) The thickness of crank webs in way of the shrinkage fit is to comply with the following formula:

$$t \geq \frac{C_1 T D^2}{C_2 {d_h}^2} \frac{1}{\left(1 - \frac{1}{{r_s}^2}\right)}$$

$$t \geq 0.525 d_c$$

where

t : Thickness of crank web measured parallel to the axis (*mm*)
C_1 : 10 for 2-stroke cycle in-line engines / 16 for 4-stroke cycle in-line engines
T : Same as given in **D2.3.1-2**
D : Cylinder bore (*mm*)
C_2 : $12.8\alpha - 2.4\alpha^2$, but in the case of a hollow shaft, C_2 is to be multiplied by $(1 - R^2)$

$$\alpha = \frac{Shrinkage \quad allowance(mm)}{d_h} \times 10^3$$

R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter
d_h :Diameter of the hole at shrinkage fit (*mm*)

$$r_s = \frac{External\ diameter\ of\ web(mm)}{d_h}$$

d_c :Required diameter of crankshaft determined by the formula in **D2.3.1-2** (*mm*)

(2) The dimensions in fillets at the junctions of crank webs with crankpins of semi-built-up crankshafts are to comply with the requirements in **D2.3.1-4**.

5 In cases of built-up crankshafts, the value of α used in **-4(1)** is to be within the following range:

$$\frac{1.1Y}{225} \leq \alpha \leq \left(\frac{1.1Y}{225} + 0.8\right) \frac{1}{1 - R^2}$$

where

Y : Specified yield point of crank web material (N/mm^2)
R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter

However, when the specified yield point of the crank web exceeds 390 *N/mm*^{2} or the value obtained by the following formula is less than 0.1, the value used for α is to be approved by the Society.

where

$$\frac{S - d_p - d_j}{2d_p}$$

S : Length of stroke (*mm*)

d_p : Diameter of the crankpin (*mm*)

d_j : Diameter of the journal (*mm*)

~~2~~6 In cases where the dimensions of crankwebs fail to meet the requirements in **~~2.3.2-2(1), Part D of the Rules~~-4(1)**, they may be acceptable provided that either the following **(1)** or **(2)** is satisfied.

(1) In cases where the maximum torque at the shrinkage fit is evaluated without carrying out a forced vibration calculation including the stern shaftings:

$$d_h^2 t P_m \geq CTD^2$$

where

C : 103 for 2-stroke cycle in-line engines

165 for 4-stroke cycle in-line engines

P_m : Surface pressure at shrinkage fit, as given by the following formula

$$P_m = Y\left\{\log_e K + \frac{1}{2}\left(1 - \frac{K^2}{r_s^2}\right)\right\}(1 - R^2)$$

$$K = 0.9\sqrt{\frac{206\alpha}{Y} + 0.25}$$

~~Other symbols are the same as those used in **2.3, Part D of the Rules**.~~

(2) In cases where the maximum torque at the shrinkage fit is evaluated by carrying out a forced vibration calculation including the stern shaftings:

$$\alpha \geq \frac{4 \times 10^3 S_R M_{T\max}\left(1 - \frac{R^2}{r_s^2}\right)}{\pi \mu E d_h^2 t\left(1 - \frac{1}{r_s^2}\right)(1 - R^2)}$$

where

$M_{T\max}$: Maximum torque at shrinkage fit, as shown in **1.3.2-1 of the Annex ~~D~~2.3.1~~-2(2)~~** "**~~GUIDANCE FOR~~ CALCULATION METHOD OF CRANKSHAFT STRESS~~-II~~**"($N \cdot m$)

E : Modulus of longitudinal elasticity (*N/mm*^{2})

~~Other symbols are the same as those used in **2.3, Part D of the Rules**.~~

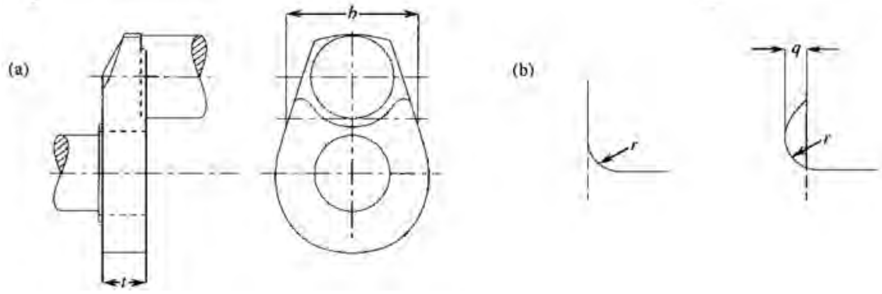
~~3~~7 In cases where **~~2.3.2-2(1), Part D of the Rules~~-4(1)** is applied and where **~~2.3.1-2, Part D of the Rules~~D2.3.1-4-** and **-5** ~~including **D2.3.1-3** above~~ is applied in accordance with **~~2.3.2-2(2), Part D of the Rules~~-4(2)**, the dimensions of the crankweb breadth *b*, crankweb thickness *t*, radius in fillet *r* and recess *q* above are to be in accordance with the following (*See* **Fig. D2.3.2-1**):

(1) As for "*b*", the breadth on the line perpendicularly intersected to the line between the crankpin centre and journal centre and tangent to the crankpin is to be used.

(2) As for "*t*", the thickness at the same section specified in **(1)** is to be used. In this case, the recess *q* need not be accounted in the thickness even when it is provided, and the ring around the shrinkage hole is not to be included in the thickness.

(3) As for “*r*”, the radius connecting to the crankpin or journal is to be used when a composite radius is provided.

Fig. D2.3.2-1 Dimensions for Webs of Semi-built-up Crankshafts



D2.3.3 Shaft Couplings and Coupling Bolts

The wording “to be of sufficient strength” in **2.3.3-2, Part D of the Rules** means to be in accordance with the following **(1)** or **(2)**:

((1) is omitted.)

(2) Detailed calculation sheets for the strength of couplings (for the procedures and contents of these calculations, the following **(a)** to **(f)** are to be considered as standards) are to be submitted to the Society for approval. In this case, it is to be verified that the thickness of the coupling flange is larger than the diameter of the bolts determined by the formula in **2.3.3-1, Part D of the Rules** using the tensile strength of the bolt material assumed to be equivalent to the tensile strength of the crankshaft material.

(a) With the procedures specified in the following **(b)** to **(f)**, it is to be verified that the stress at the coupling is less than the allowable value. As the stress value in this case, comparisons are to be made by applying appropriate safety factors for yield points for bending stress, bending fatigue limits, yield points for torsional stress and torsional fatigue limits of the crankshaft material considering four types of stress, such as the maximum bending stress, fluctuating bending stress, the maximum torsional stress and fluctuating torsional stress.

(b) The maximum bending moment and fluctuating bending moment of this portion are to be determined in accordance with the requirements specified in the **Annex D2.3.1~~-2(1)~~** “**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS~~-I~~**” or **Annex ~~D~~2.3.1~~-2(2)~~** “**~~GUIDANCE FOR~~ CALCULATION METHOD OF CRANKSHAFT STRESS~~-II~~**.” Mean torque of this portion is to be determined.

((c) to (f) are omitted.)

Annex D2.3.1-2(1) has been renumbered to Annex D2.3.1, and Title of Annex D2.3.1 has been amended as follows.

Annex D2.3.1~~-2(1)~~ GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS~~-1~~

1.2 Calculation of Stresses

The direct calculation method of local stress at crank-pin fillets or crack-journal fillets of crankshafts is as follows:

Paragraph 1.2.1 has been amended as follows.

1.2.1 Stress at Fillets Due to Bending Moments

Stress at fillets due to bending moments is to be obtained by the following formulae:

$\sigma_x = 1.08\alpha_{KB}\frac{M_W}{Z}$ (1)

$\sigma_y = 0.285\alpha_{KB}\frac{M_W}{Z}$ (2)

where

σ_x : Axial stress due to bending moment at fillet

σ_y : Circumferential stress due to bending moment at fillet

α_{KB} : Stress concentration factor for bending, as shown in **D2.3.1-~~3~~5(1)**

Z : Section modulus of crankpin or journal

M_W : Bending moment at the centre of the web thickness, parallel to the crankplane

((1) to (3) are omitted.)

Paragraph 1.2.2 has been amended as follows.

1.2.2 The Torsional Stress at Fillets Due to Twisting Moments

The torsional stress at fillets due to twisting moments is to be obtained by the following formula:

$\tau_f = \alpha_{KT}\frac{T}{Z_p}$ (5)

where

τ_f : Torsional stress in fillet at the root of webs

α_{KT} : Stress concentration factor for torsion, as specified in **D2.3.1-~~3~~5(1)**

Z_p : Polar section modulus of crankpin or journal

T : Twisting moment acting on crankpin or journal, which is to be determined by sequentially summing up the moments from the free end side. External forces to be considered are the same as the external forces for bending moments

Annex D2.3.1-2(2) has been deleted.

~~Annex D2.3.1-2(2) GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS II~~

(Omitted)

Appendix D1 has been deleted.

~~Appendix D1 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD~~

(Omitted)

Appendix D2 has been deleted.

~~Appendix D2 GUIDANCE FOR EVALUATION OF FATIGUE TESTS~~

(Omitted)

Appendix D3 has been deleted.

~~Appendix D3 GUIDANCE FOR CALCULATION OF SURFACE TREATED FILLETS AND OIL BORE OUTLETS~~

(Omitted)

Appendix D4 has been deleted.

~~Appendix D4 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE OIL BORE OUTLETS OF CRANKSHAFTS THROUGH UTILISATION OF THE FINITE ELEMENT METHOD~~

(Omitted)

EFFECTIVE DATE AND APPLICATION (Amendment 1-3)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to crankshafts for which the application for approval is submitted to the Society before the effective date.

Amendment 1-4

D5 POWER TRANSMISSION SYSTEMS

D5.2 Materials and Construction

Paragraph D5.2.3 has been added as follows.

D5.2.3 General Construction of Gearings

The words "enough" and "sufficient" referred to in **5.2.3, Part D of the Rules** mean being designed in accordance with national or international standards such as *JIS.*

D5.3 Strength of Gears

D5.3.1 Application

Sub-paragraph (2) has been amended as follows.

In the case of bevel gears, the wording"deemed appropriate by the Society"in **5.3.1, Part D of the Rules** means as follows:

(1) (Omitted)

(2) Strength of the interior of gear teeth

The Vickers hardness (*HV*) of the interior of gear teeth is not to be less than the value calculated by the following formula. However, this requirement does not apply to bevel gears for which the tip diameter (outer end) is smaller than 1,100 *mm*:

If $\frac{\mathrm{z}}{\mathrm{w}} < 0.79$ then $\frac{\mathrm{z}}{\mathrm{w}}$ is to be taken as 0.79.

$$HV = 1.11 S_H p \left[\frac{z}{w} - \frac{\left(\frac{z}{w}\right)^2}{\sqrt{1+\left(\frac{z}{w}\right)^2}} \right]$$

HV : Vickers hardness

S_H: Safety factor for contact stress, is to comply with the requirements in **Annex ~~D5.3.5~~5.3.1 "~~GUIDANCE FOR~~ CALCULATION OF STRENGTH OF ENCLOSED GEARS" 1.6.3-9, Part D of the Rules.**

p : Real hertzian stress (*MPa*). The upper limit of the value of p used in this calculation is to be 1,500 *MPa.*

$p = AS_c$

S_c: Contact stress (*MPa*), to be calculated according to *ISO* 10300 standards.

A: If S_c is calculated according to *ISO* 10300 standards, then the coefficients are to be determined, in consideration of analysis results, by the Society on a case by case basis. In addition, if S_c is calculated according to *ISO* 10300 standards, A is to be taken as 1.32

w : Half the hertzian contact width (*mm*), to be calculated by the following formula:

$$w = \frac{p\rho_C}{56300}$$

$$\rho_C = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$$\rho_1 = 0.5 d_{vn1} \sin\alpha_n$$

$$\rho_2 = 0.5 d_{vn2} \sin\alpha_n$$

$$d_{vn1} = d_{m1} \frac{\sqrt{1+u^2}}{u} \frac{1}{\cos^2\beta_{vb}}$$

d_{m1} : Mean pitch diameter of pinion（mm）

u : Gear ratio

$\beta_{vb} = \arcsin(\sin\beta_m \cos\alpha_n)$

β_m : Mean spiral angle

α_n : Normal pressure angle

$$d_{vn2} = u^2 d_{vn1}$$

z : Depth from teeth surface to evaluation point (mm)

Paragraph D5.3.5 has been deleted.

~~**D5.3.5 Detailed Evaluation for Strength**~~

~~It is acceptable that the bending and surface strength of gears are calculated based on **Annex D5.3.5 "GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS"**.~~

Section D5.4 has been added as follows.

D5.4 Gear Shafts and Flexible Shafts

D5.4.1 Gear Shafts

The word "sufficient" in **5.4.1-1(2), Part D of the Rules** means being designed in accordance with national or international standards such as *JIS.*

Paragraph D5.4.3 has been added as follows.

D5.4.3 Couplings and Coupling Bolts

The word "sufficient" in **5.4.3, Part D of the Rules** means being designed in accordance with national or international standards such as *JIS.*

Annex D5.3.5 has been deleted.

~~Annex D5.3.5 GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS~~

~~(Omitted)~~

EFFECTIVE DATE AND APPLICATION (Amendment 1-4)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to all gears previously approved by the Society prior to the effective date for which no failure has occurred, and no changes related to strength, such as the scantlings of the gear meshes, materials, etc. have been made.

Amendment 1-5

D11 WELDING FOR MACHINERY INSTALLATIONS

D11.2 Welding Procedure and Related Specifications

D11.2.2 Execution of Tests

Sub-paragraphs -1 and -2 have been amended as follows.

1 Approval tests for welding procedures and related specifications that fall under **11.2.2(1), Part D of the Rules** are to comply with the following requirements. For items not specified in the following requirements, **4.1.3** and **4.2** to **4.6, Part M of the Rules** are to be applied correspondingly. In cases where it is difficult to meet the above requirements, approval tests are to be as deemed appropriate by the Society.

(1) Selection of welding consumables
In general, a welding consumable for which the requirements related to strength (i.e. yield point or proof stress and tensile strength) of deposited weld metal is higher than strength of base metals and which resemble to base metals in the chemical composition is to be selected.

(2) Tests for butt welded joints
- (a) The kinds of tests, the areas subjected to tests and the number of specimens is to be in accordance with the requirements specified in **Table D11.2.2-1**.
- (b) The ~~testing temperatures during impact tests and the~~ values of minimum mean absorbed energy are to comply with the requirements of base metals. <u>In addition, testing temperatures are to be lower than the testing temperatures required for base metals.</u>
- (c) The Vickers hardness measured by hardness tests is, as a standard, to comply with the values specified in **Table D11.2.2-2** depending on the requirements related to the yield point or proof stress of base metals.

(3) Tests for fillet weld joints, T-joints with full penetration and T-joints with partial penetration
- (a) For the number of specimens for hardness tests, the requirements specified in **Table D11.2.2-1** are to be applied correspondingly.
- (b) The Vickers hardness measured by hardness tests is to be in accordance with **(2)(c)**.

2 Approval tests for welding procedures and related specifications that fall under **11.2.2(2), Part D of the Rules** are to be comply with the following requirements. ~~For the approval tests for welding procedures and related specifications applied to the welding work for materials used at high temperatures, the Society may require a creep test or a high temperature tensile test where deemed necessary.~~ For items not specified in the following requirements, **4.1.3** and **4.2** to **4.6, Part M of the Rules** are to be applied correspondingly.

(1) Test assemblies
Test assemblies are to be of the same or equivalent material used in the actual welding work. Additionally, the thickness of the test assemblies is, in principle, to be equal to the maximum thickness of the materials to be used in the actual welding work.

(2) Tests for butt welded joints
- (a) The kinds of tests, areas subjected to tests and the number of specimens is to be in accordance with the requirements specified in **Table D11.2.2-3**.
- (b) Test specimens are to be collected in accordance with **Fig. D11.2.2-1**.
- <u>(c) Minimum mean absorbed energy values for impact tests are to be in accordance with **-1 (2)(b)**.</u>
- <u>(d) The Vickers hardness measured by hardness tests is to be in accordance with **-1 (2)(c)**.</u>

(e) Notwithstanding **(a)** above, in principle, creep tests or high temperature tensile tests are to be added as a reference in cases where deemed necessary by the Society for the welding work of components used at temperatures higher than 1/2 of the melting points (absolute temperature) of base metals or the welding consumables, whichever is lower. Creep tests are to be performed in accordance with *ISO* 204, *JIS Z* 2271 or equivalent standards, and high temperature tensile tests are to be performed in accordance with *ISO* 6892-2, *JIS Z* 0567 or equivalent standards.

(3) Tests for fillet weld joints
The kinds of tests to be conducted are finished inspections, macro-structure inspections, hardness tests and fracture tests.

3 (Omitted)

EFFECTIVE DATE AND APPLICATION (Amendment 1-5)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to welding procedure and related specifications for which the application for approval is submitted to the Society before the effective date.

Amendment 1-6

D14 PIPING SYSTEMS FOR TANKERS

D14.2 Cargo Oil Pumps, Cargo Oil Piping Systems, Piping in Cargo Oil Tanks, etc.

Paragraph D14.2.2 has been amended as follows.

D14.2.2 Arrangement of Cargo Oil Piping Systems

~~**1**~~ "Cargo piping systems" in **14.2.2-7, Part D of the Rules** includes cargo oil pipes, vent pipes, tank washing pipes, etc.

~~**2** For the purpose of the requirements in **14.2.2-7, Part D of the Rules**, earthing is to conform to the requirements of **2.1.4, Part H of the Rules** and the resistance between cargo oil tanks/cargo piping systems (cargo oil pipes, vent pipes, tank washing pipelines, etc.) and the hull is to be not greater than 1 *MΩ*.~~

EFFECTIVE DATE AND APPLICATION (Amendment 1-6)

1. The effective date of the amendments is 1 July 2022.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to ships for which the date of contract for construction is before the effective date.