Shafting Systems

Amended Rules and Guidance

Rules for the Survey and Construction of Steel Ships Part D
Rules for the Survey and Construction of Inland Waterway Ships
Guidance for the Survey and Construction of Steel Ships Part D
Guidance for the Survey and Construction of Inland Waterway Ships
Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use

Reason for Amendment

IACS Unified Requirement (UR) M68 specifies requirements for the materials, construction, strength, etc. of shafting systems, and these requirements have already been incorporated into the NK Rules. Requirements related to intermediate shafts made of high strength steels (i.e. minimum specified tensile strengths exceeding 800 N/mm^2) have been incorporated into annex of the NK Guidance.

In addition, requirements related to shaft alignment calculations were incorporated into annexes of the NK Guidance and the requirements has become common practice with established track records without problems.

Accordingly, as a part of a comprehensive review of the NK Rules, these requirements have been moved from the annexes of the Guidance to annexes of the Rules, and relevant requirements are amended to clarify their correspondence with the IACS UR M68.

Outline of Amendment

The main contents of this amendment are as follows:

(1) Clarifies that the power transmission systems and shafting systems of auxiliary machinery for cargo handling and their prime movers fall within the scope of application of modified requirements.

(2) Changes the term “tallow” to “grease” with respect to the acceptable means for preventing shaft corrosion by water.

(3) Transfers requirements for coefficient C_K concerning the type and shape of shafts in the case of longitudinal slots from the Guidance to the Rules.

(4) Transfers requirements for the use of high strength materials for intermediate shafts from the annex of the Guidance to the annex of the Rules.

(5) Transfers requirements for the calculation of shaft alignment from the annex of the Guidance to the annex of the Rules.

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

Part D MACHINERY INSTALLATIONS

Chapter 1 GENERAL

1.1 General

Paragraph 1.1.4 has been amended as follows.

1.1.4 Modification of Requirements*

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 $d_i \leq 0.4 d_a$, 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:

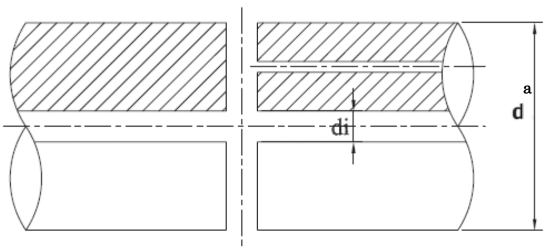
(1) The fillet radius at the base of the flange is not to be less 0.08 *times* the diameter of the shaft.

(~~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
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 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~~ in accordance with provisions specified elsewhere, 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 in accordance with **Annex 6.2.13** including bending moments, bearing loads and deflection curve of the shafting is to be ~~carried out for approval~~ submitted to the Society for approval.

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:

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

$d =$ 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

Paragraph 8.3.1 has been amended as follows.

8.3.1 Barred Speed Range for Avoiding Continuous Operation*

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.

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

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.

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. Test procedures are to be in accordance with *Section* 10 of *ISO* 1352.

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 *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

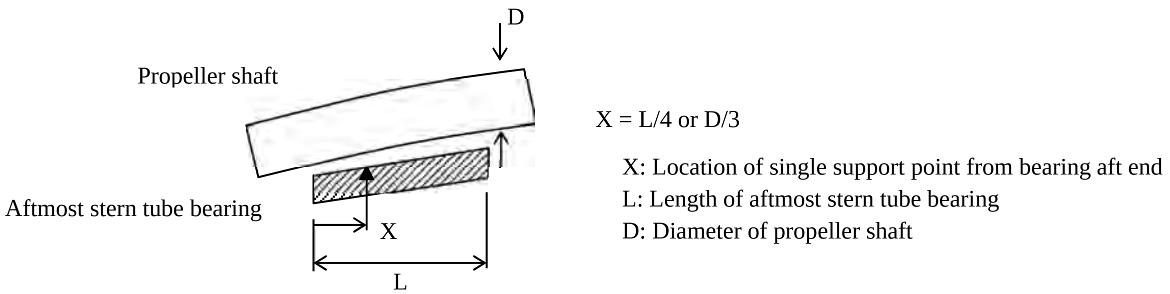
1 Static loads are to be used in shaft alignment calculations.

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.

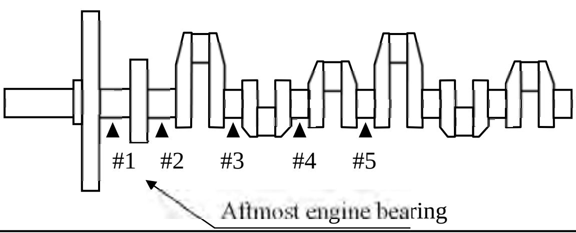
Fig. 1.2.2-1 Location of Single Support Point in Aftmost Stern Tube Bearings



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	3×10^{-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

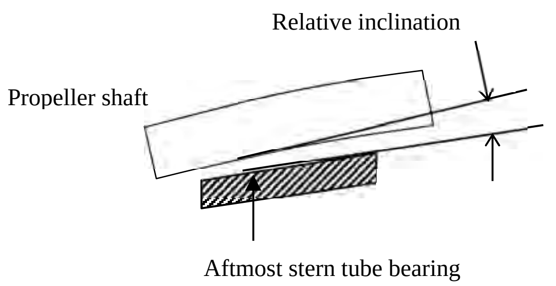
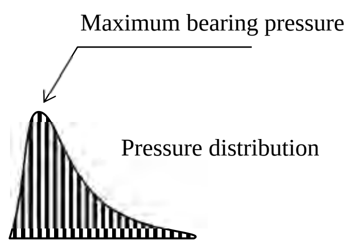


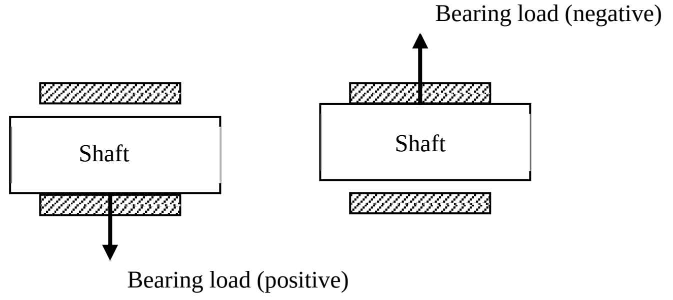
Fig. 1.3.1-2(b) Maximum Bearing Pressure



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.

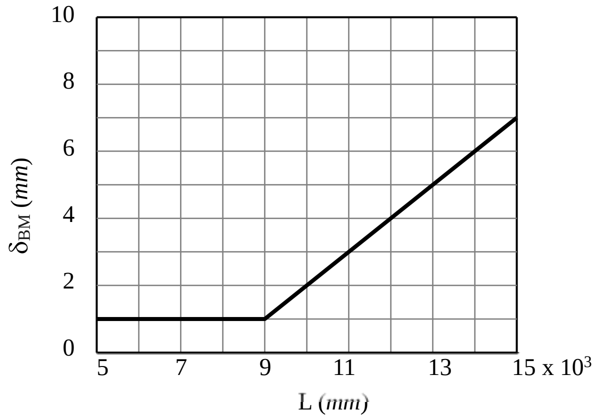
5 In the case of shafting with reduction gears, any differences in bearing loads between the fore and aft bearings of wheel gears in hot conditions are to be within those allowable limits specified by manufacturers.

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)

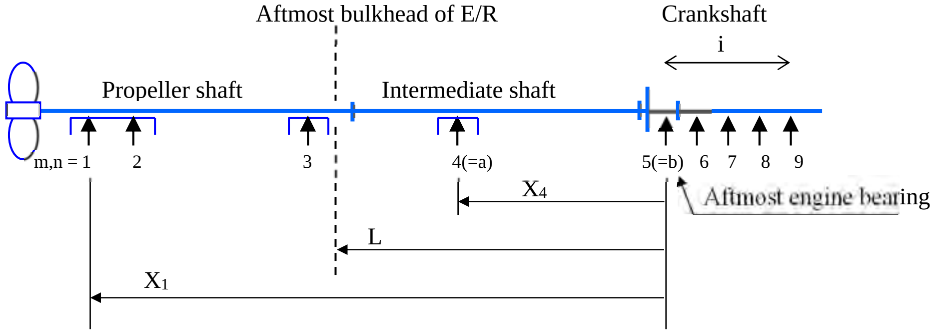
b : Support point numbers of aftmost engine bearings (counted from the aft of such shafting)

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:

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

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

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.

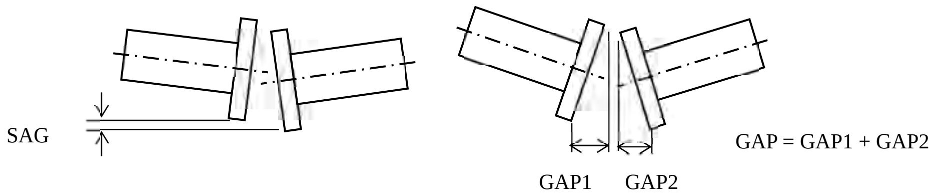
3 Other documents such as those showing results structural analysis evaluating the extent of hull deflection may be required by the Society in cases where stern hull construction is considered to be unconventional.

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.

“Rules for the survey and construction of inland waterway ships” has been partly amended as follows:

Part 7 MACHINERY INSTALLATIONS

Chapter 1 GENERAL

1.1 General

1.1.4 Modification of Requirements*

Sub-paragraph -1 has been amended as follows.

1 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 4 SHAFTINGS

4.2 Materials, Construction and Strength

4.2.2 Intermediate Shafts*

1 The diameter of the intermediate shafts 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 7.4.1**
k_1: Factor given in **Table 7.4.2**
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.
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 $K = 1$

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

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 7.4.1 is omitted.)

Table 7.4.2 has been amended as follows.

Table 7.4.2 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:

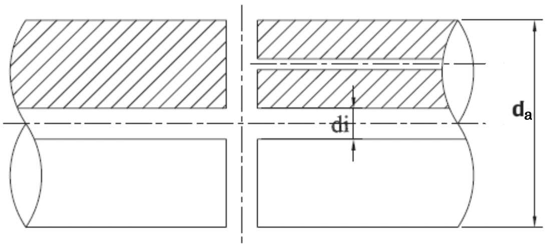
(1) The fillet radius at the base of the flange is not to be less 0.08 *times* the diameter of the shaft.

(~~1~~2) In cases where shafts, during continuous operation, experience torsional vibration stress exceeding 85 % of τ_1 given in **6.2.2-1(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 **6.3.**

(~~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 ~~=~~ $\geqq$ $e / 2$

~~W~~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.

4.2.3 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.

4.2.4 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 **4.2.2-1** at the portion 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 7.4.3 has been amended as follows.

Table 7.4.3 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 4.2.2.~~

Table 7.4.4 has been amended as follows.

Table 7.4.4 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 4.2.2-1 considering T_S=400.~~

4.2.7 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.

4.2.10 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 **4.2.4-1** 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 designed maximum load 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 designed maximum load 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 **4.2.4-1** or **-2**. However, where the nominal bearing pressure is not more than 0.6 *MPa* and bearings having a construction and arrangement ~~specially approved by the Society~~ in accordance with provisions specified elsewhere, 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)** above, 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 **4.2.4-1** 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 **4.2.4-1** 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 **4.2.4-1** or **-2**.

Paragraph 4.2.12 has been amended as follows.

4.2.12 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 in accordance with **Annex 6.2.13, Part D of the Rules for the Survey and Construction of Steel Ships** including bending moments, bearing loads and deflection curve of the shafting is to be ~~carried out for approval~~ submitted to the Society for approval.

Chapter 6 TORSIONAL VIBRATION OF SHAFTINGS

6.2 Allowable Limit

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

1 For ships in which reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), the torsional vibration stresses 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:

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

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

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. 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 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 7.6.1**.

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)

(-2 and -3 are omitted.)

Table 7.6.1 has been amended as follows.

Table 7.6.1 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 7.4.2.~~

(~~2~~1) ~~To be in accordance with note (4) of Table 7.4.2.~~ 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 7.6.2 is omitted.)

6.3 Barred Speed Range

Paragraph 6.3.1 has been amended as follows.

6.3.1 Barred Speed Range for Avoiding Continuous Operation*

1 In cases where the torsional vibration stresses exceed the allowable limit τ_1 specified in **6.2**, 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.

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

~~W~~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 engine is to be stable in operation at each end of the 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 **6.2** 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 **-1** and **-2** above is not readily available, transferring of the resonant points of torsional vibrations and other necessary measures are to be taken.

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

Part D MACHINERY INSTALLATIONS

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 *N/mm²* 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~~.

Paragraph D6.2.10 has been amended as follows.

D6.2.10 Stern Tube Bearings and Shaft Bracket Bearings

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~~ $\cancel{N/mm^2}$ ~~and are in accordance with the requirments of **Annex D6.2.2 “GUIDANCE FOR USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS”**.~~

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~~ $\cancel{C_K}$ ~~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~~

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

$\cancel{l}$ ~~: Slot length~~

$\cancel{e}$ ~~: Slot width~~

$\cancel{d_i}$ ~~: Inside diameter of the hollow shaft at the slot~~

$\cancel{d_a}$ ~~: Outside diameter of the hollow shaft~~

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

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

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)**~~

“Guidance for the survey and construction of inland waterway ships” has been partly amended as follows:

Part 7 MACHINERY INSTALLATIONS

Chapter 4 SHAFTINGS

4.2 Materials, Construction and Strength

4.2.4 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” specified in **4.2.4-1, Part 7 of the Rules** 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 7.4.2.4-1**

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 7.4.4** which is for k_3 specified in **4.2.4-2, Part 7 of the Rules**, is to be in accordance with **Table 7.4.2.4-2**. Furthermore, this requirement may be applied to propeller shafts Kind 2 and stern tube shafts Kind 2.

Note of Table 7.4.2.4-1 has been amended as follow.

Table 7.4.2.4-1 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 $2.5d_s$, 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 **4.2.~~2~~4-~~1~~3, Part 7 of the Rules** ~~where it is assumed that T_s = 400 N/mm²~~.

Note of Table 7.4.2.4-2 has been amended as follow.

Table 7.4.2.4-2 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 $2.5d_s$, 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 **4.2.~~2~~4-~~1~~3, Part 7 of the Rules**~~ where it is assumed that T_s = 400 N/mm^2~~.

Paragraph 4.2.10 has been amended as follows.

4.2.10 Stern Tube Bearings and Shaft Bracket Bearings

1 The wording "provisions specified elsewhere" in **4.2.10-1(1)(a)i), Part 7 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 **4.2.10-1(1)(a)i), Part 7 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 ~~Guidance~~Rules for the Survey and Construction of Steel Ships**.

(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 **4.2.10-1(1)(b)ii), Part 7 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 **4.2.10-1(1)(b)ii), Part 7 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 ~~Guidance~~Rules for the Survey and Construction of Steel Ships** 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 **4.2.10-1(2)(b), Part 7 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 **4.2.10-1(2)(b), Part 7 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 non-flow alarm is to be provided at the lubricating water inlet.

Paragraph 4.2.12 has been deleted.

~~**4.2.12 Shaft Alignment**~~

~~For the approval of the shaft alignment calculation required in **4.2.12, Part 7 of the Rules**, a calculation sheet in accordance with **Annex D6.2.13 "GUIDANCE FOR CALCULATION OF SHAFT ALIGNMENT", Part D of the Guidance for the Survey and Construction of Steel Ships** is to be submitted.~~

Chapter 6 TORSIONAL VIBRATION OF SHAFTINGS

6.2 Allowable Limit

6.2.6 Detailed Evaluation for Strength

Sub-paragraph -3 has been deleted.

~~**3** In cases where intermediate shafts with longitudinal slots given in **Table 7.6.1, Chapter 6, Part 7 of the Rules** are equipped, the value of~~ $\cancel{C_K}$ ~~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:~~

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

$\cancel{l}$ ~~: Slot length~~

$\cancel{e}$ ~~: Slot width~~

$\cancel{d_i}$ ~~: Inside diameter of the hollow shaft at the slot~~

$\cancel{d_a}$ ~~: Outside diameter of the hollow shaft~~

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

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

"Guidance for the approval and type approval of materials and equipment for marine use" has been partly amended as follows:

Part 6 MACHINERY

Chapter 2 TYPE APPROVAL OF USE OF MACHINERY AND EQUIPMENT

2.1 General

Paragraph 2.1.1 has been amended as follows.

2.1.1 Scope

The requirements of this chapter deal with the tests and inspection relating to the approval of the machinery and equipment listed for which approval of the Society is to be obtained in advance before they are used in ships as required by **the Rules for the Survey and Construction of Steel Ships** (hereinafter referred to as "the Rules").

((1) to (4) are omitted.)

(5) Stern tube bearings (**6.2.10-1~~(3)~~(1)(b)i) and (2)(a), Part D of the Rules**)

((6) to (13) are omitted.)