Crankshafts

Amended Rules and Guidance

Rules for the Survey and Construction of Steel Ships Part D
Guidance for the Survey and Construction of Steel Ships Part D

Reasons for Amendment

The Society's provisions for the dimensions of crankshafts of reciprocating internal combustion engines are specified in Part D of its Rules for the Survey and Construction of Steel Ships. In addition, provisions for the methods to be used to evaluate crankshaft designs based on fatigue strength are specified in IACS Unified Requirement(UR) M53. The Society has already incorporated the UR's requirements into its Rules, but this was done back in 1986 when the evaluation methods specified by IACS were new and had yet to establish a track record of successful application at the time; for this reason, these provisions were instead incorporated as a new annex into Part D of the Guidance for the Survey and Construction of Steel Ships as an alternative method and this annex has never been undergone a formal review since that time.

Accordingly, as a part of a comprehensive review of the NK Rules, relevant provisions for crankshafts are transferred from Part D of the Guidance to Part D of the Rules in consideration of their successful application of the years. In addition, older provisions for crankshafts currently in Part D of the Rules are correspondingly transferred to Part D of the Guidance as part of the same comprehensive review.

Outline of Amendment

The main details of the amendment are as follows:

(1) Transfers Annex D2.3.1-2(2) and Appendices D1 to D4, Part D of the Guidance to Annex 2.3.1 and Appendices 1 to 4, Part D of the Rules respectively.

(2) Transfers the contents of 2.3.1 and 2.3.2, Part D of the Rules to D2, Part D of the Guidance.

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

Part D MACHINERY INSTALLATIONS

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;~~

$$\cancel{K_m = 1.0}$$

~~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~~ $\cancel{(d_c/d_a)^3}$.

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

~~where~~

$\cancel{b}$: ~~Breadth of crank web (*mm*)~~

$\cancel{d_a}$: ~~Actual diameter of crankpin or journal (*mm*)~~

$\cancel{t}$: ~~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* · *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:~~

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

~~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)~~

<u>Built-up crankshaft approval is to be as deemed appropriate by the Society.</u>

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.

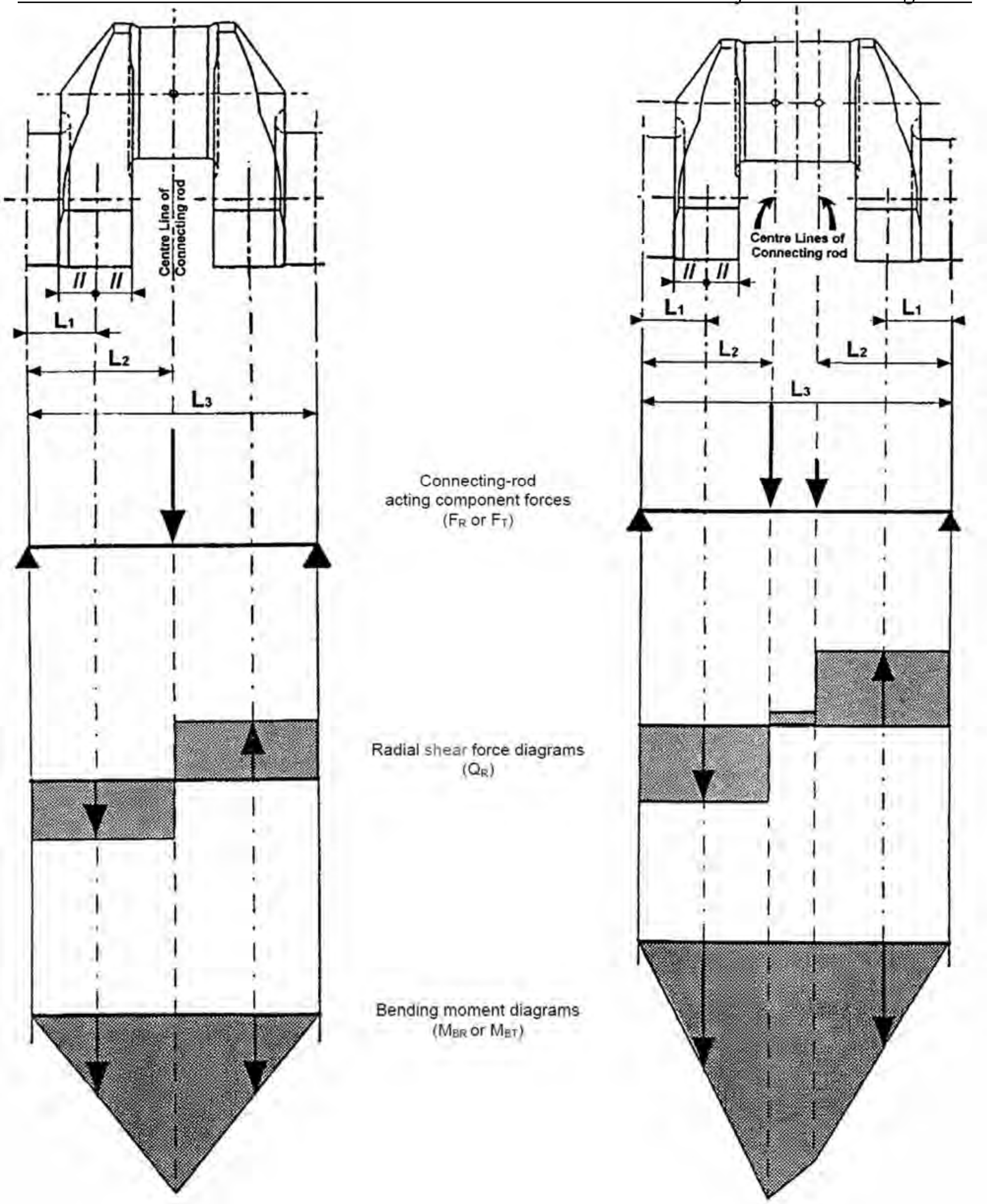
(6) Bending moments acting in outlets of crankpin oil bores

 (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

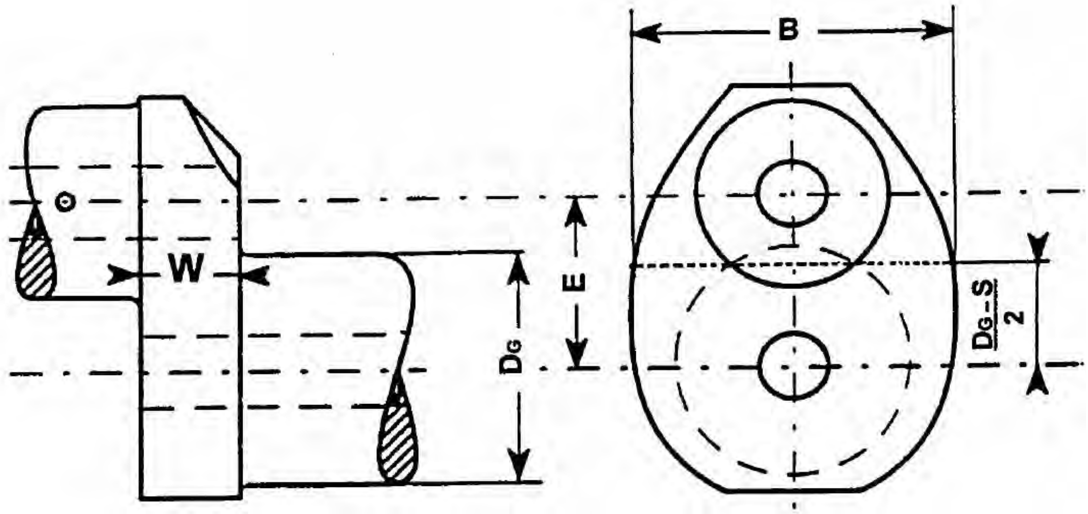


L_1 = Distance between main journal centre line and crank web centre (*See* also **Fig. 3** for crankshafts without overlaps)

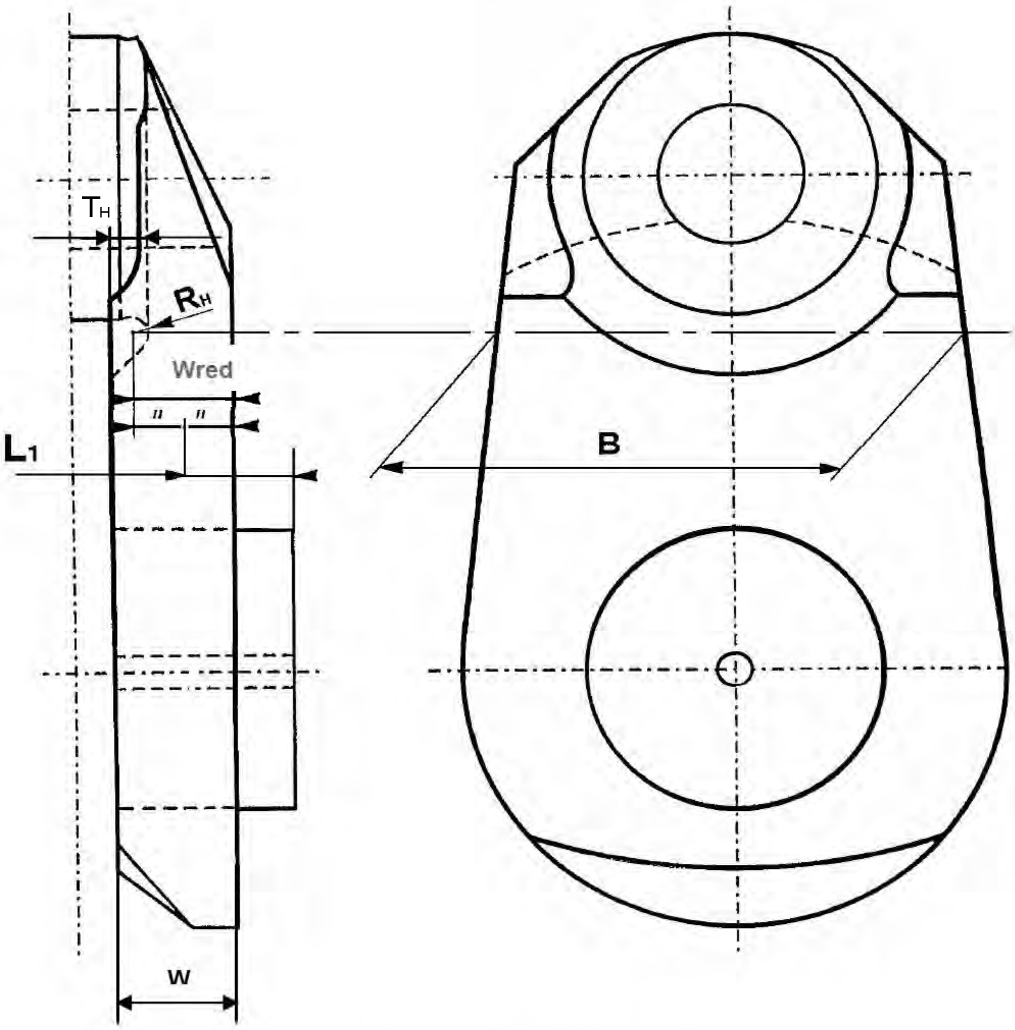
L_2 = Distance between main journal centre line and connecting rod centre

L_3 = Distance between two adjacent main journal centre lines

Fig. 3 Reference Areas of Crank Web Cross Sections

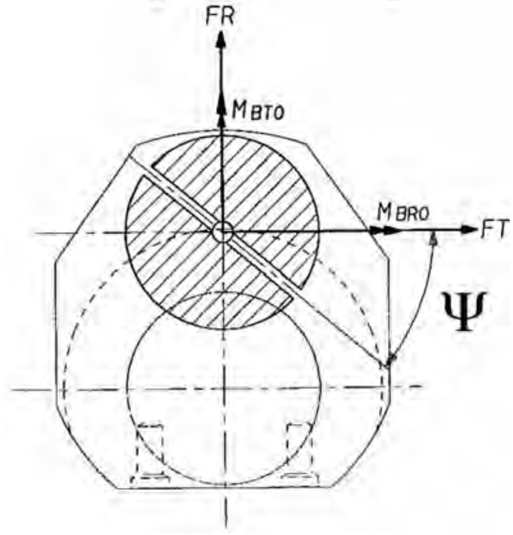


Overlapped crankshaft



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) $(0.03 \le r \le 0.13)$

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

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

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

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

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

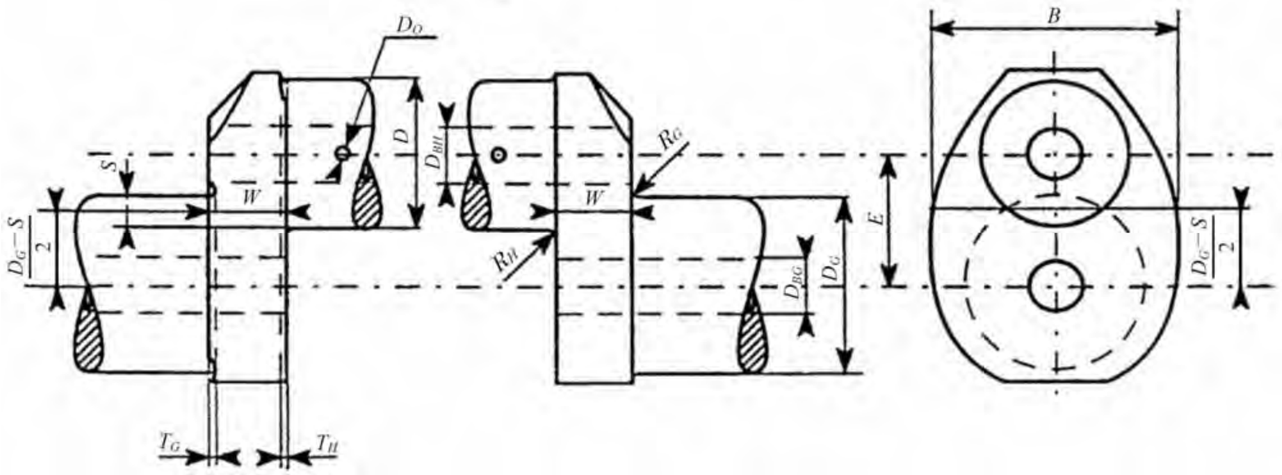
$d_H = D_{BH}/D$ $(0 \le d_H \le 0.8)$

$t_H = T_H/D$

$t_G = T_G/D$

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

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

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

$$f_Q(b) = -0.5 + b$$

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

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

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

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*.)

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

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

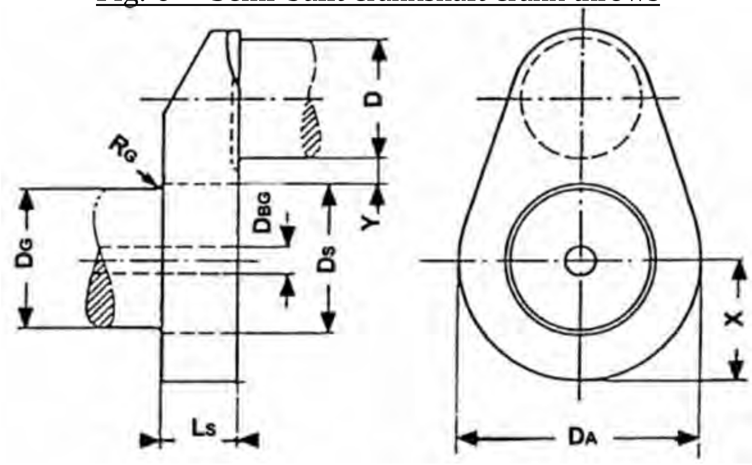
$$= \frac{\sigma_{DW}}{\sigma_V}$$

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·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 $5a$. 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

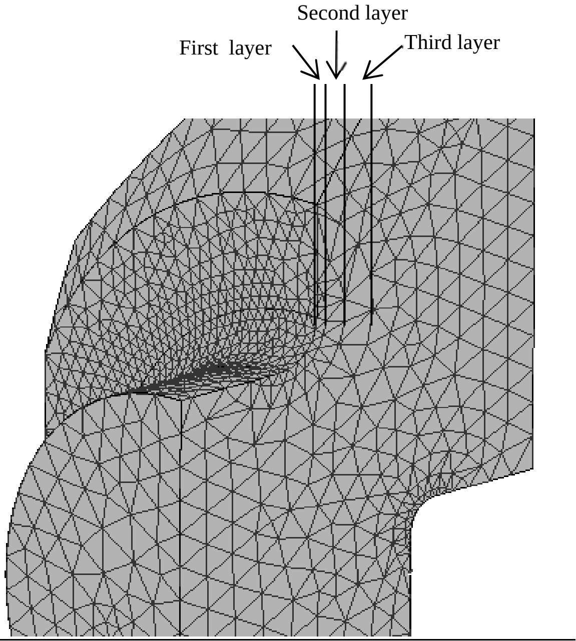
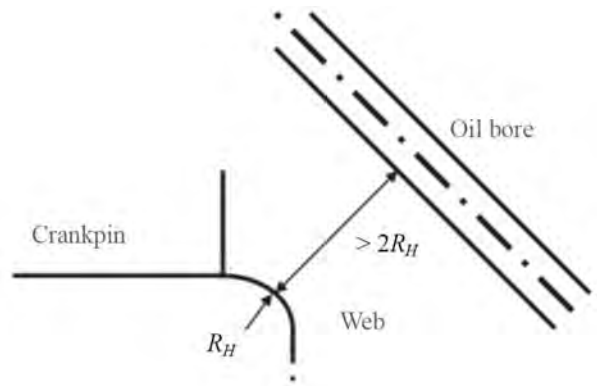


Fig. 2 Oil Bore Proximity to Fillet



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.

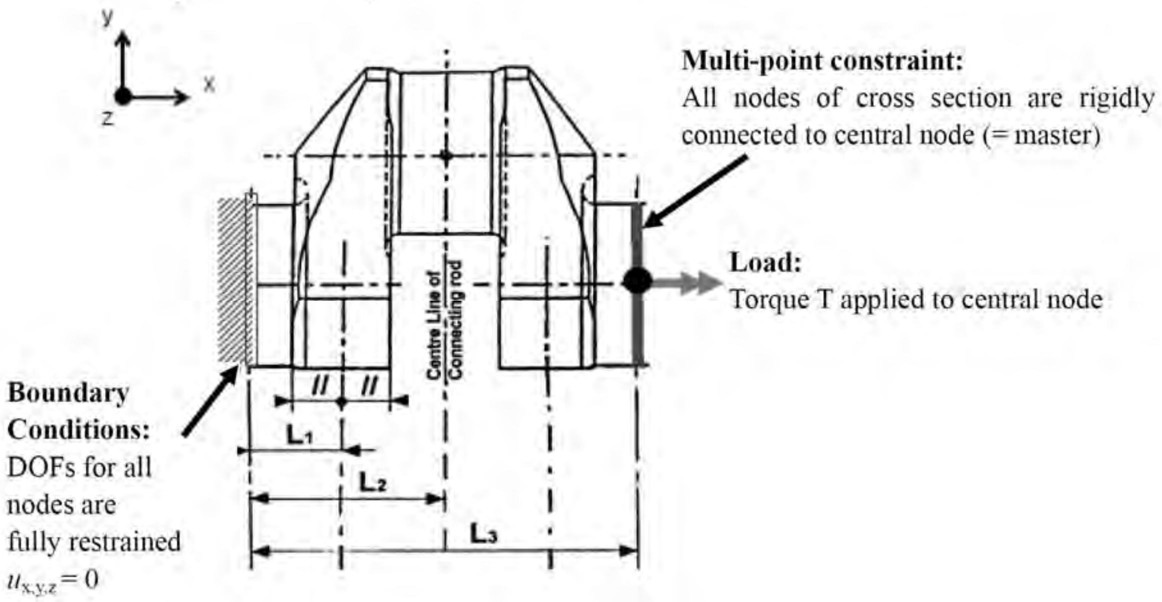
$$\alpha_T = \frac{\tau_{equiv,\alpha}}{\tau_N}$$

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

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:

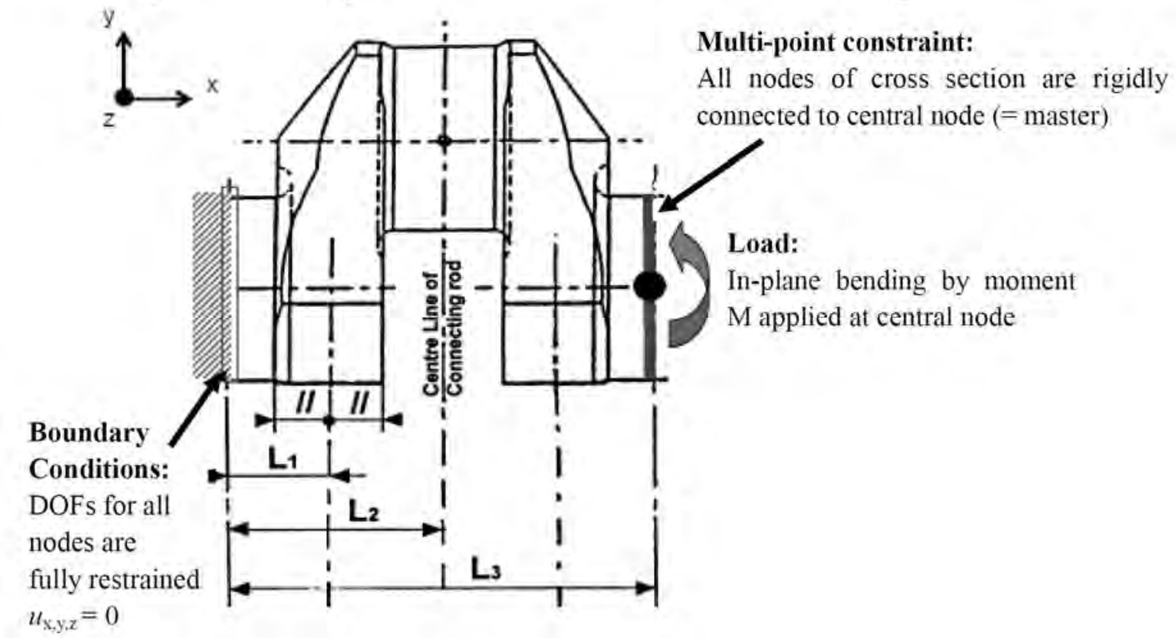
$$\alpha_B = \frac{\sigma_{equiv,\alpha}}{\sigma_N}$$

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

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.

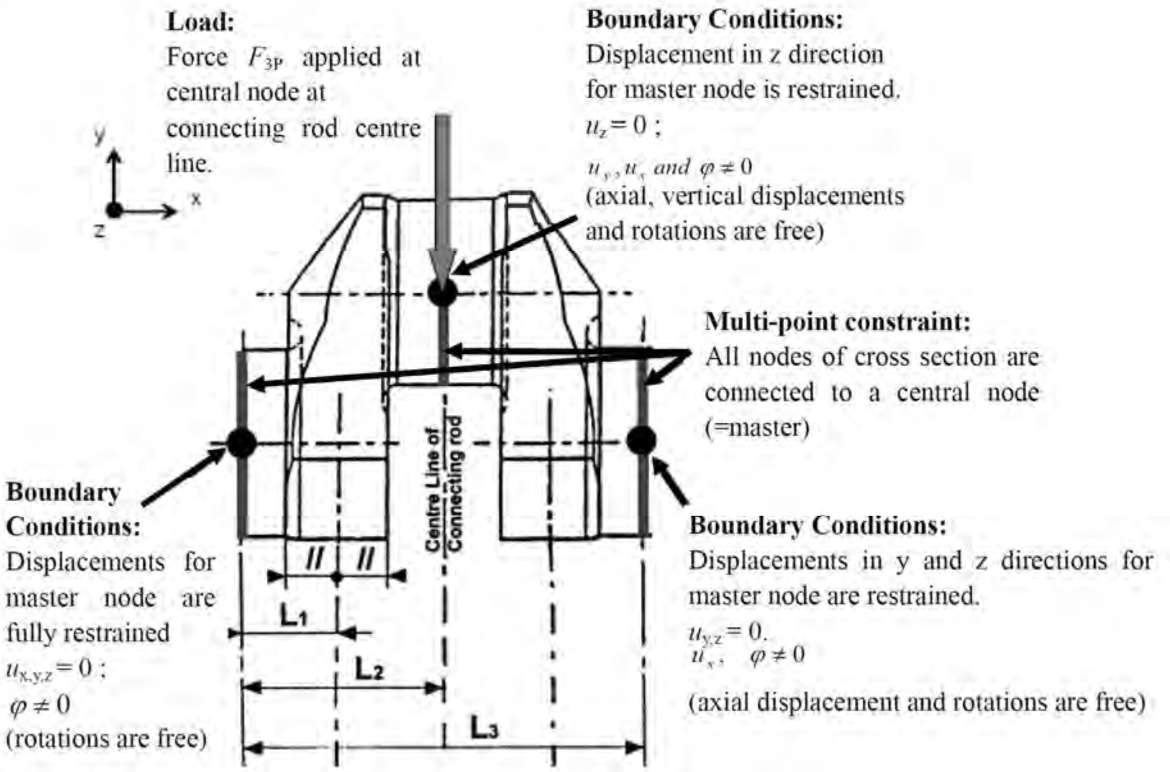
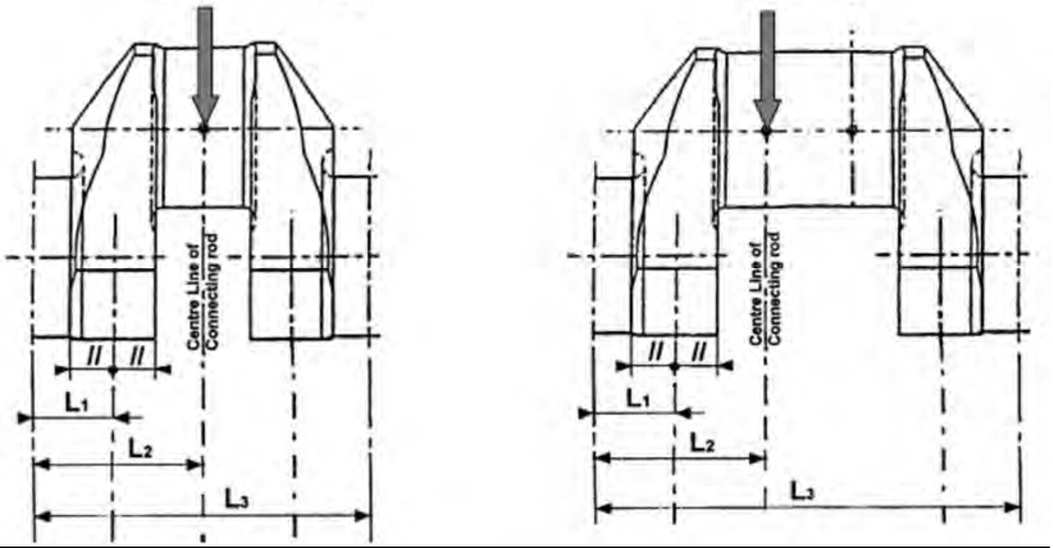


Fig. 6 Load Applications for In-line and Vee Type Engines



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.

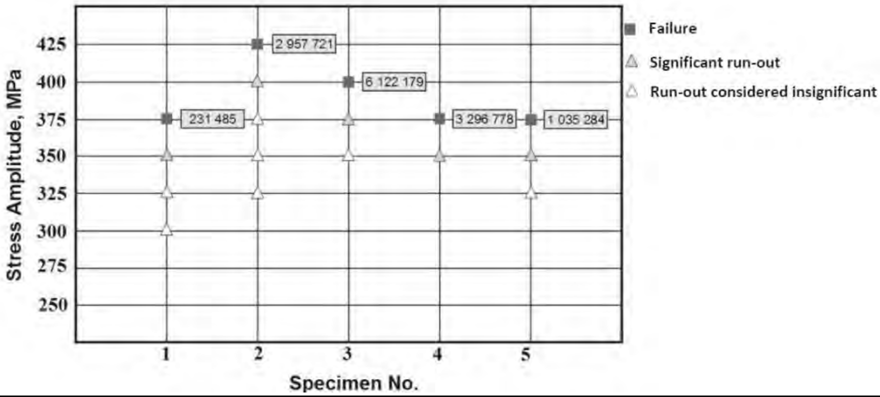
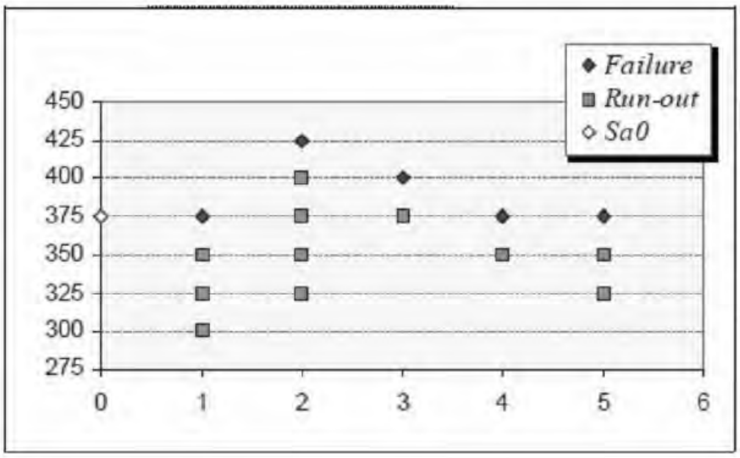


Fig. 2 Processing of a Modified Staircase Test Results.



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

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

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

(4) Calculation of sample mean is as follows:

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

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

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

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

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

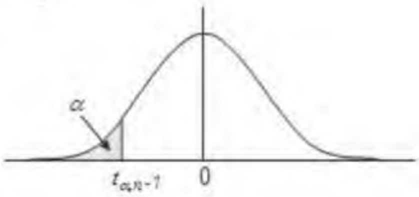
****

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:

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

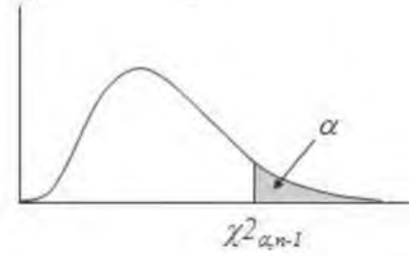
To be conservative, some authors would consider *n* to be 5, as the physical number of used specimens, then $t_{\alpha,n-1} = 1.533$.

****

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

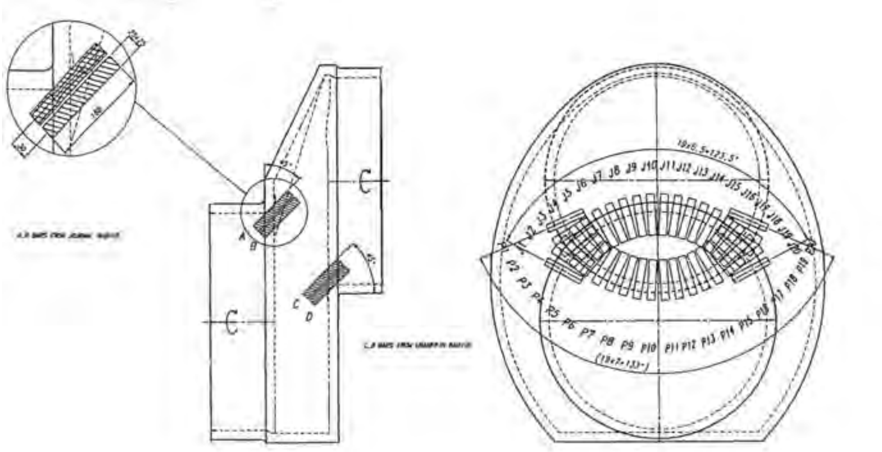
1 In this appendix, a small specimen is considered to be one of the specimens taken from a crank throw.

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

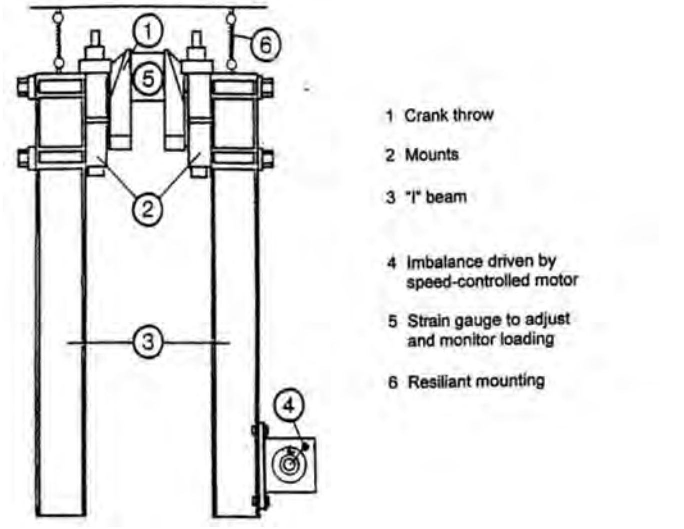
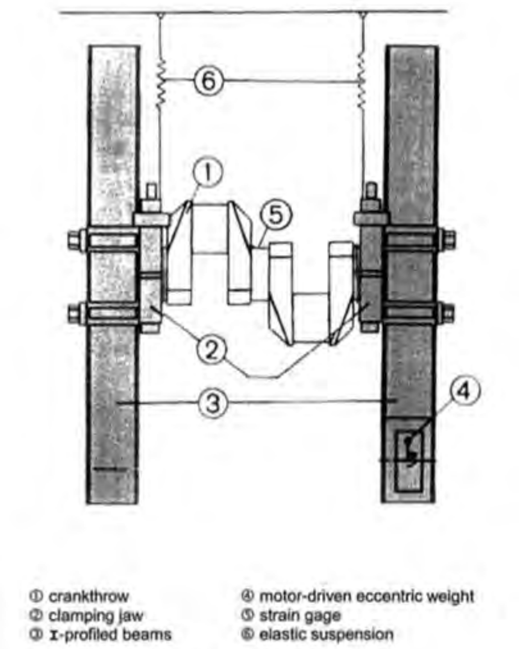


Fig. 7 An Example of Testing Arrangement of the Resonance Tester for Torsion Loading with Double Crank Throw Section



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 $\pm 30\ \%$ 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.

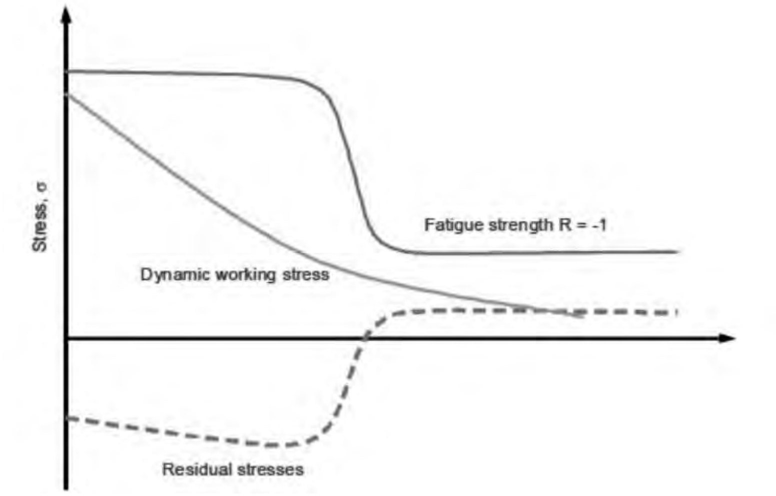
3.1 Calculation Principles

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**)

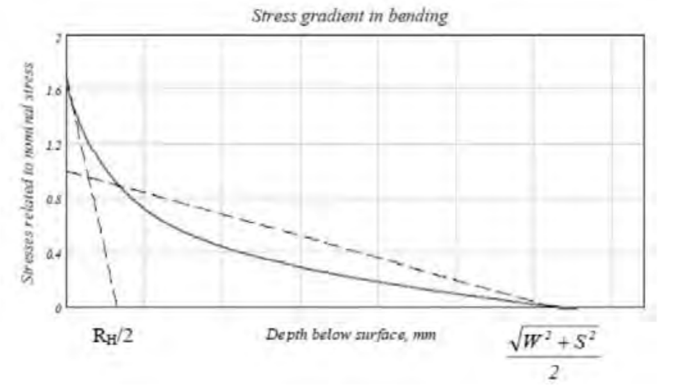
$$\alpha_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_H}} + 1 - \left(\frac{2 \cdot t}{D}\right)^{\frac{0.6}{\sqrt{\alpha_T}}}$$

$$\beta_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_G}} + 1 - \left(\frac{2 \cdot t}{D_G}\right)^{\frac{0.6}{\sqrt{\beta_T}}}$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

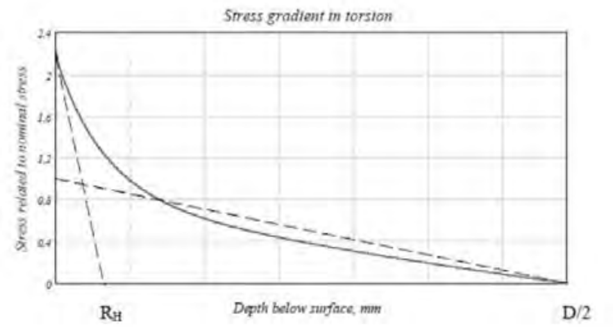
3 If the pin is hardened only and the end of the hardened zone is closer to the fillet than three times the maximum hardness depth, FEA is to be used to determine the actual stresses in the transition zone.

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

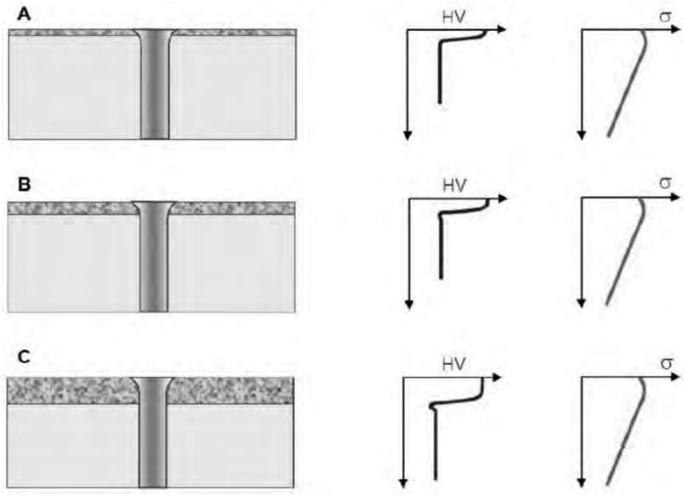
1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from **1.3** of **Annex 2.3.1** if within its applicability range.

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

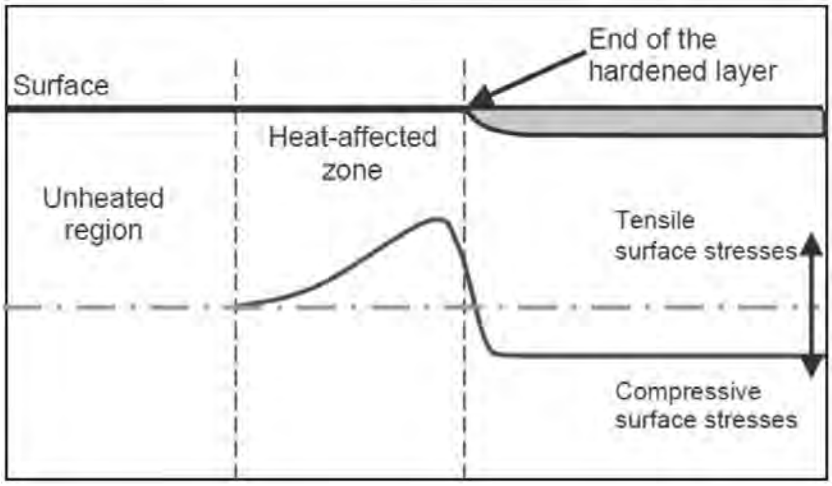
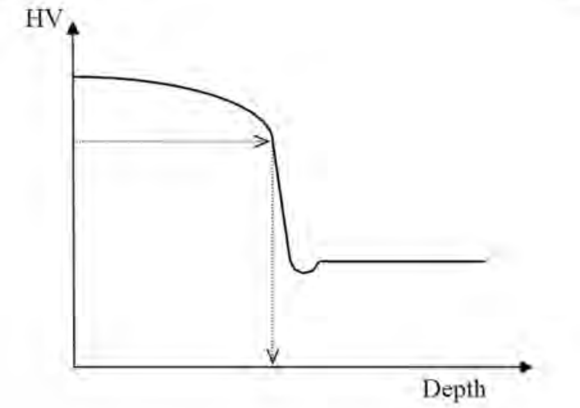


Fig. 6 Typical Hardness as a Function of Depth.



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:

$$\cancel{\sigma_{Ftransition,cpin}}$$

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

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.

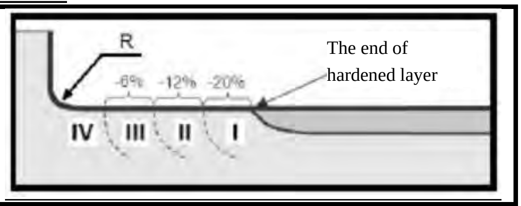
3 For the purpose of considering subsurface fatigue, below the hard layer, the disadvantage of tensile residual stresses has to be considered by subtracting 20 % from the value determined above. This 20 % is based on the mean stress influence of alloyed quenched and tempered steel having a residual tensile 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:

$$HV(t) = HV_{core} + (HV_{surface} - HV_{core}) \cdot \left(\frac{50}{HV_{surface} - HV_{core}}\right)^{\left(\frac{t}{t_N}\right)^2}$$

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.

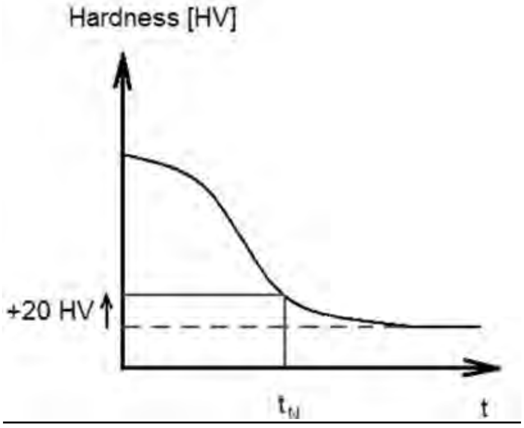
3 In contrast to induction-hardening the nitrited components have no such distinct transition to the core. Although the compressive residual stresses at the surface are high, the balancing tensile stresses in the core are moderate because of the shallow depth.

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.

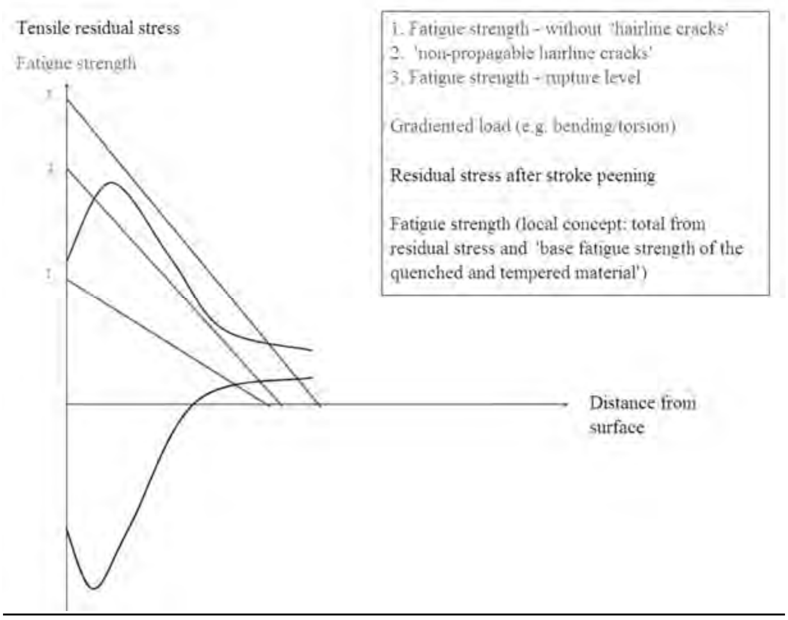
6.1.2 Stroke Peening by Means of a Ball

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 $\pm 10\ \%$ 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 $\pm 15\ \%$ 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 ± 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.

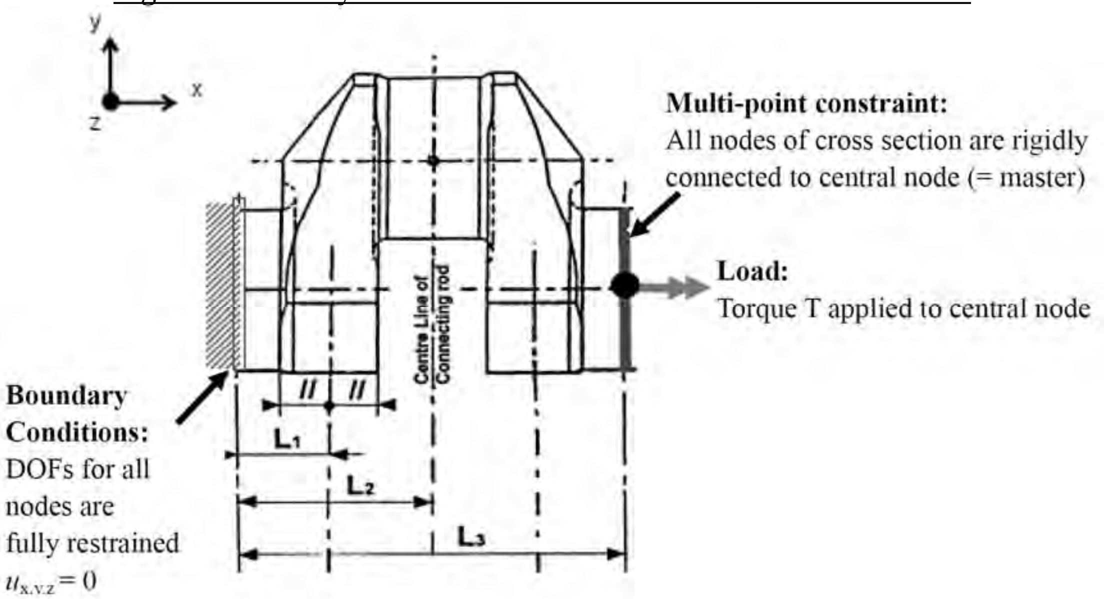
3.1 Load Cases and Assessment of Stress

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:

$$\gamma_T = \frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\tau_N}$$

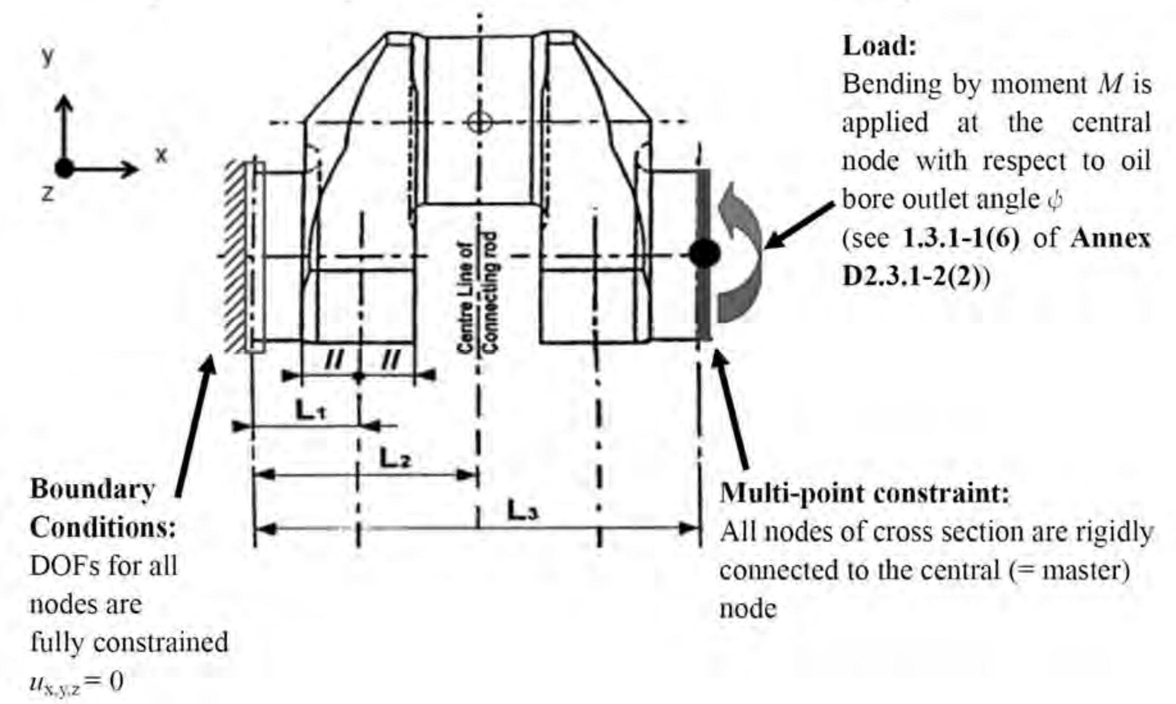
where the nominal torsion stress τ_N referred to the crankpin is calculated as follows (for W_P *see* **1.3.2** of **Annex 2.3.1**) :

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

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**):

$$\sigma_N = \frac{M}{W_e}$$

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

Part D MACHINERY INSTALLATIONS

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 : $10^{-2} A L P_{max}$

T : $10^{-2} B S P_{mi}$

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	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 D2.3.1-2(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

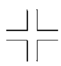
Number of cylinders	Minimum firing interval between two cylinders on one crankpin											
	45°		60°		90°		270°		300°		315°	
	A	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*
6	1.60	4.1	1.47	4.0	1.40	4.0	1.40	4.0	1.30	4.4	1.20	4.3
8		5.5		5.5		5.5		5.5		5.3		5.2
10		6.7		7.0		6.5		6.5		6.1		5.9
12		7.5		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
				15.0
16				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:~~

~~$Q \geq 1.15$~~

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.**

$\sigma = \sigma_a \cdot f_m \cdot f_s + \alpha$ (*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="4">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></td><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>

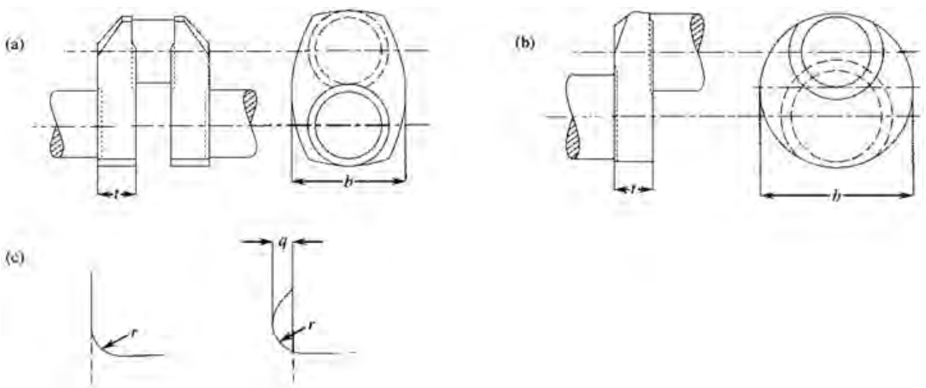
Notes:

(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・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 $(1 - R^2)$

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

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:

$$d_h^2 t P_m \geq C T D^2$$

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**.~~

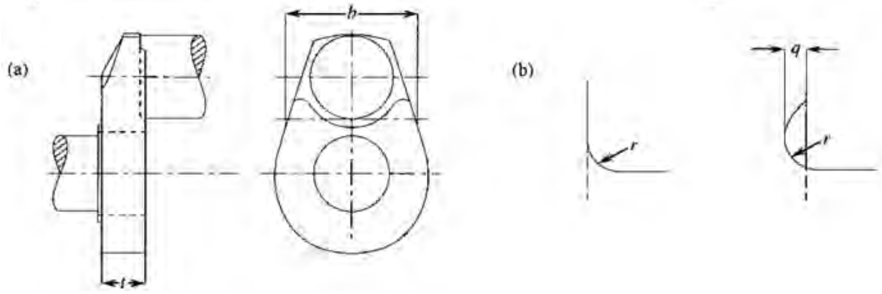
~~3~~7 In cases where ~~**2.3.2-2(1), Part D of the Rules**~~-**4(1)** is applied and where ~~**2.3.1-2, Part D of the Rules**~~**D2.3.1-4- and -5** ~~including D2.3.1-3 above~~ is applied in accordance with ~~**2.3.2-2(2), Part D of the Rules**~~-**4(2)**, the dimensions of the crankweb breadth *b*, 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~~ I~~

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:

$$\sigma_x = 1.08\alpha_{KB}\frac{M_W}{Z} \quad (1)$$

$$\sigma_y = 0.285\alpha_{KB}\frac{M_W}{Z} \quad (2)$$

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:

$$\tau_f = \alpha_{KT}\frac{T}{Z_p} \quad (5)$$

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)