

CLASS GUIDELINE

DNVGL-CG-0037

Edition July 2019

Calculation of crankshafts for reciprocating internal combustion engines

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FOREWORD

DNV GL class guidelines contain methods, technical requirements, principles and acceptance criteria related to classed objects as referred to from the rules.

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CHANGES – CURRENT

This document supersedes the November 2015 edition of DNVGL-CG-0037.

Changes in this document are highlighted in red colour. However, if the changes involve a whole chapter, section or subsection, normally only the title will be in red colour.

Changes July 2019

<i>Topic</i>	<i>Reference</i>	<i>Description</i>
Restructure of the guideline	General	The guideline is totally restructured to correlate with IACS UR M53 in its chapter and subchapter structure.
Delete IACS UR M53 from the guideline	General	IACS UR M53 method is deleted from the guideline.
Update and correction of the DNV GL method	All appendixes	The appendixes have been updated to correlate to the DNV GL method, described in this guideline, in its structure and formulas.
	App.D [2.1] and App.D [4.3]	Appendix D has been updated with last part of App.D [2.1] related to principles and limitations and new formulas in App.D [4.3] .

Editorial corrections

In addition to the above stated changes, editorial corrections may have been made.

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SECTION 1 CALCULATION OF STRESSES

1 General

1.1 Scope

This class guideline contains an alternative method to the method given in IACS UR M53 for calculation of safety against fatigue failure in pin and journal fillets and in oil bores.

In principle, any relevant calculation method may be applied, e.g. the fatigue criteria assume that maximum respectively minimum bending stresses occur simultaneously with the maximum respectively minimum torsional stresses.

If properly documented, an evaluation of the fatigue capacity by using multi-axial fatigue methods may replace the criteria applied in this class guideline.

A crankshaft will be approved when the calculation methods, as presented in this class guideline, is fulfilled. Crankshafts that cannot satisfy these rules may be subject to special consideration when detailed calculations and/or measurements are made to demonstrate equivalence to these rules.

1.2 Field of application

This method apply to engines for both main propulsion and auxiliary purpose, being designed for continuous operation at the specified (nominal) rating. If frequent use of overload (in excess of nominal) rating is foreseen, i.e. accumulating several millions of stress cycles, this overload should form the basis for the calculation. Further, the methods apply to solid forged or semibuilt crankshafts of forged or cast steel, and with one crank throw between main bearings. Designs with more than one crank throw between main bearings, fillets between offset crankpins, fully built crankshafts, or welded crankshafts will be specially considered on basis of equivalence with the "normal" requirements.

This method is also valid for:

- surface treated fillets
- when fatigue parameter influences are tested
- when working stresses are measured.

Note that surface treated fillets also include crankpin hardening that may influence the fillet fatigue strength.

1.3 Principles of calculations

The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas. The calculation is also based on the assumption that the areas exposed to highest stresses are:

- fillet transitions between the crankpin and web as well as between the journal and web.
- outlets of crankpin oil bores
- outlets of journal oil bores (only considered in the DNV GL method described in [3.5]).

The basic difference between the alternative DNV GL method described in this guideline and IACS UR M53 method is that IACS UR M53 considers Von Mises stresses, while the alternative DNV GL method consider principal stresses.

The methods of crankshaft fatigue are based on:

- A combination of bending and torsional stresses where bending fatigue strength and torsional fatigue strength are separately assessed, i.e. also valid for anisotropic material behaviour.
- Bending stress measurements, or continuous beam analysis, or 3-point bending, depending on manufacturer's documentation.
- Measured stress concentration factors, or analysed with FE as described in App.C, or empirical formulae with one standard deviation above 50% reliability.

- Fatigue strength assessment based on full scale testing or small scale (see [App.D](#)), combined with a calculation procedure considering the influence of material strength, forging method, mean stress, and when applicable also fillet surface hardening, shot peening or cold rolling including the influence of the depth of this surface treatment (see [App.E](#)).

For semibuilt crankshafts, additional criteria cover the safety against slippage of shrink fits. In case of relatively small distance between the pin and the journal shrinkage diameter, the influence of the shrinkage stresses on the fatigue strength is considered.

1.4 Drawings and particulars to be submitted

For the calculation of crankshafts, the documents and particulars listed below shall be submitted. DNV GL form [ENG 401](#) shall be used for this purpose:

- crankshaft drawing (which shall contain all data in respect of the geometrical configurations of the crankshaft)
- type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod)
- operating and combustion method (2-stroke or 4-stroke cycle/direct injection, pre-combustion chamber, etc.)
- number of cylinders
- rated power [kW]
- rated engine speed [r/min]
- direction of rotation, see [Figure 1](#)
- firing order with the respective ignition intervals and, where necessary,
- V-angle α_v [°], see [Figure 1](#)
- cylinder diameter [mm]
- stroke [mm]
- maximum net cylinder pressure p_{\max} [bar]
- charge air pressure [bar] (before inlet valves or scavenge ports, whichever applies)
- connecting rod length L_H [mm], see [Figure 2](#).

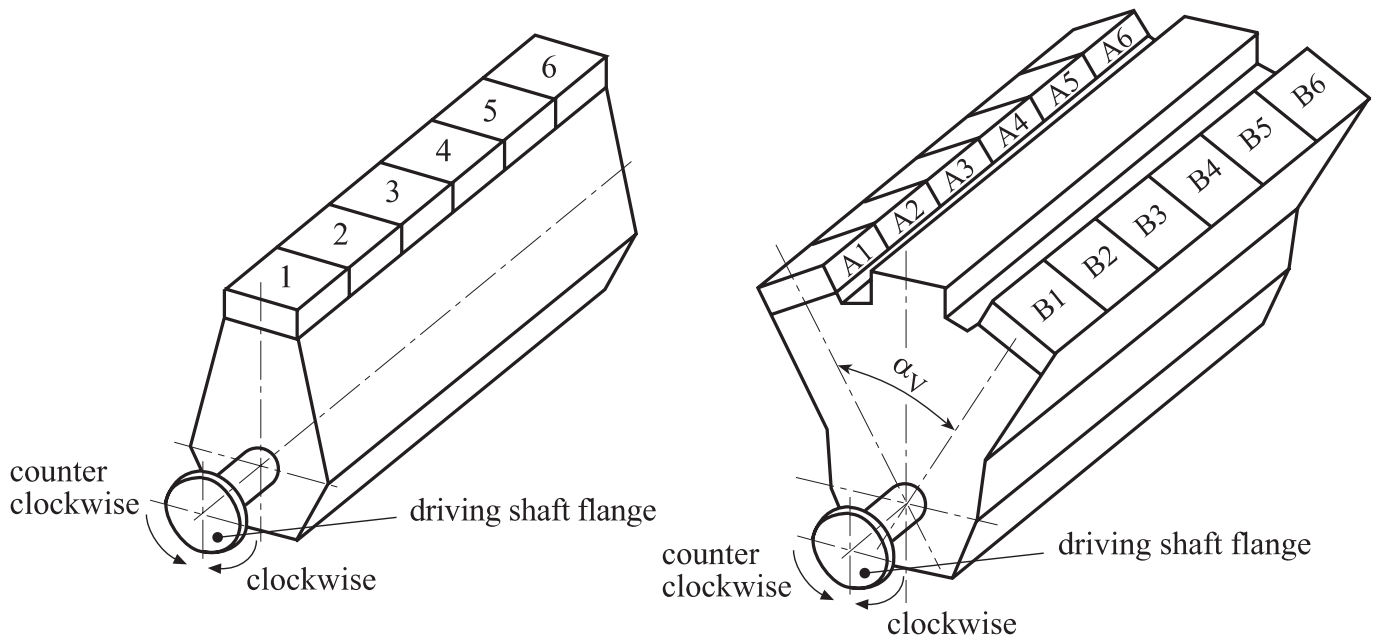


Figure 1 Designation of the cylinders

Addition to IACS UR M53:

- nominal compression ratio
- reciprocating mass per cylinder [kg]
- rotating mass (see crank radius) per connecting rod
- rotating mass (see crank radius) of cranks incl. counter weights [kg] and positions (lengthwise and angular)

---end of IACS UR M53 addition---

- digitized gas pressure curve presented at equidistant intervals [bar versus crank angle] (at least every 5° CA)
- for engines with articulated-type connecting rod, see [Figure 2](#)
 - distance to link point L_A [mm]
 - link angle α_N [°]
 - connecting rod length L_N [mm]

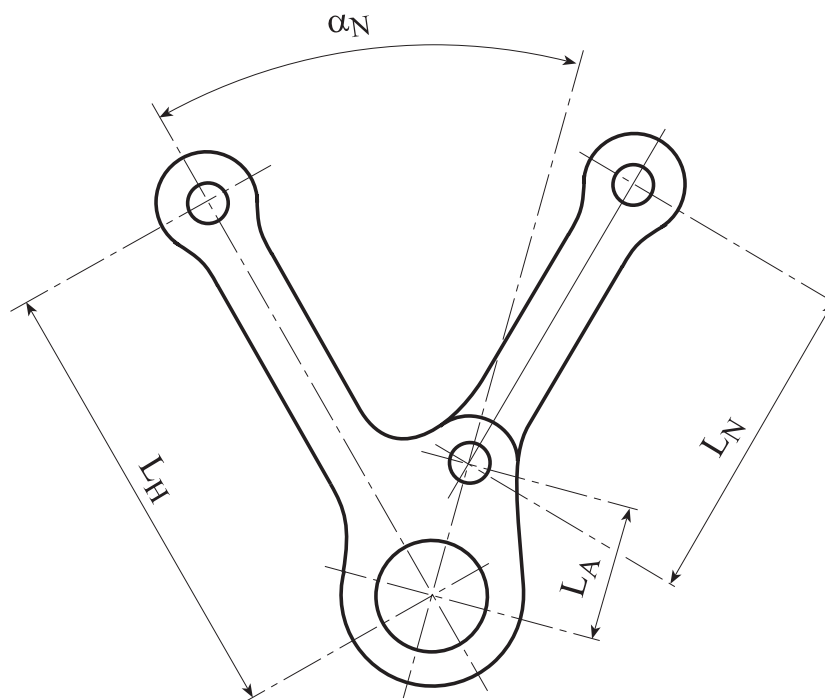


Figure 2 Articulated-type connecting rod

Details of crankshaft material:

- material designation (according to ISO, EN, DIN, AISI, etc.)
- mechanical properties of material (minimum values obtained from longitudinal test specimens):
 - tensile strength [N/mm²]
 - yield strength [N/mm²]
 - reduction in area at break [%]
 - elongation A₅ [%]
 - impact energy KV [J]
- type of forging (free form forged, continuous grain flow forged, drop-forged, etc. with description of the forging process).

The following is in addition to what the IACS UR M53 method requires:

- finishing method of fillets and oil holes
- surface hardening of fillets, if applicable (induction hardened, nitrided etc.):
 - extension of surface hardening (max and min)
 - hardness at surface (HV) (max and min)
 - hardness (HV) as a function of depth [mm]
- cold deformation of fillets, if applicable (shot peening, stroke peening, cold rolling):
 - extension of treatment
 - intensity (shot peening)
 - depth of treatment (residual stresses) for stroke peening and cold rolling, as determined for the applied force and roller or ball geometry.

---end of IACS UR M53 addition---

— Particulars of alternating torsional stress calculations, see [2.2].

2 Calculation of stresses

This method deals with nominal stresses that later are multiplied with stress concentration factors from [3].

In the case that FE calculations are made of a global model that also includes local stresses, the directly determined stresses shall be used in the acceptance criteria in [7].

In addition to the alternating stresses, the mean stresses shall be determined in order to assess the mean stress influence on the fatigue strength.

The maximum and minimum bending moments and radial forces during a complete working cycle, may be determined by thorough analysis, e.g. FEA.

Continuous beam model analysis may be applied if its relevance can be documented.

If no such analyses are available, the 3-point bending may be used as described in the following.

2.1 Calculation of stresses due to bending moments and radial forces

2.1.1 Assumptions

The calculation is based on a statically determined system, composed of a single crankthrow supported in the center of adjacent main journals and subjected to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints, distance L_3 , see Figure 3 and Figure 4.

The bending moment M_{BR} , M_{BT} are calculated in the relevant sub-section based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting rod force, see Figure 3.

For crankthrows with two connecting rods acting upon one crankpin the relevant bending moments are obtained by super position of the two triangular bending moment diagrams according to phase, see Figure 4.

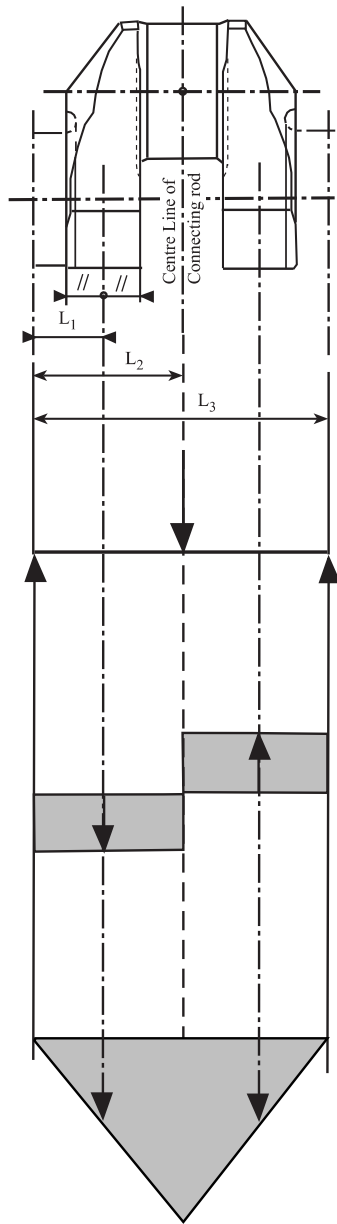


Figure 3 Crankthrow for in line engine

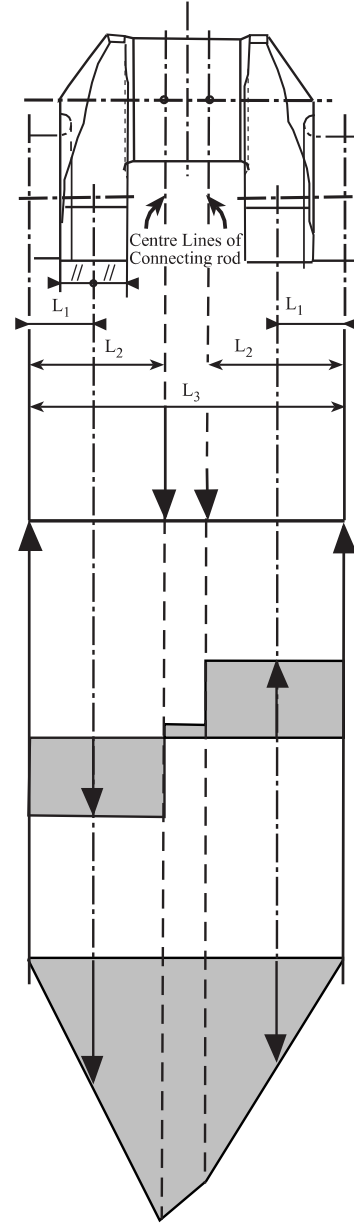


Figure 4 Crankthrow for V- engine with 2 adjacent connecting rods

where:

- L_1 = distance between main journal center line and crank web center, see [Figure 5](#) for crankshaft without overlap
- L_2 = distance between main journal center line and connecting rod center
- L_3 = distance between two adjacent main journal center lines

2.1.1.1 Bending moments and radial forces acting in web

The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the center of the solid web (distance L_1) and are derived from the radial component of the connecting rod force.

The alternating bending and compressive stresses due to bending moments and radial forces shall be related to the cross-section of the crank web. This reference section results from the web thickness W and the web width B , see Figure 5.

Mean stresses shall be determined.

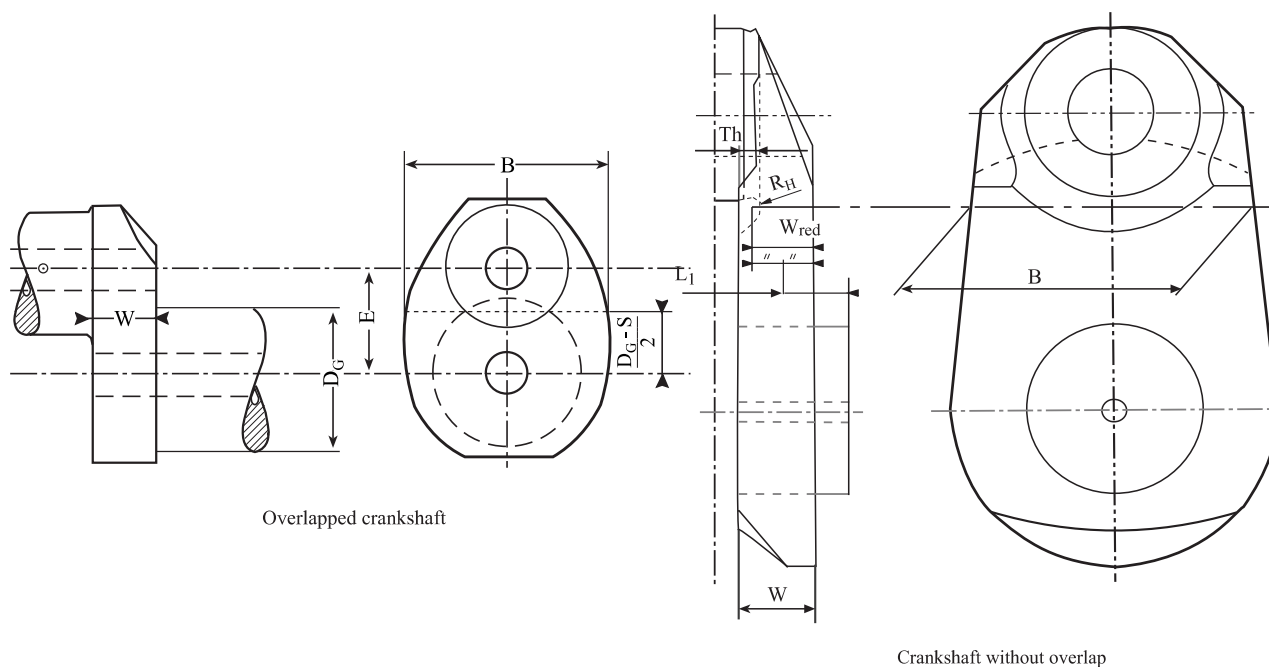


Figure 5 Reference area of crankweb cross section

2.1.1.2 Bending acting in outlet of crankpin oil bore

The two relevant bending moments are taken in the crankpin cross-section through the oil bore.

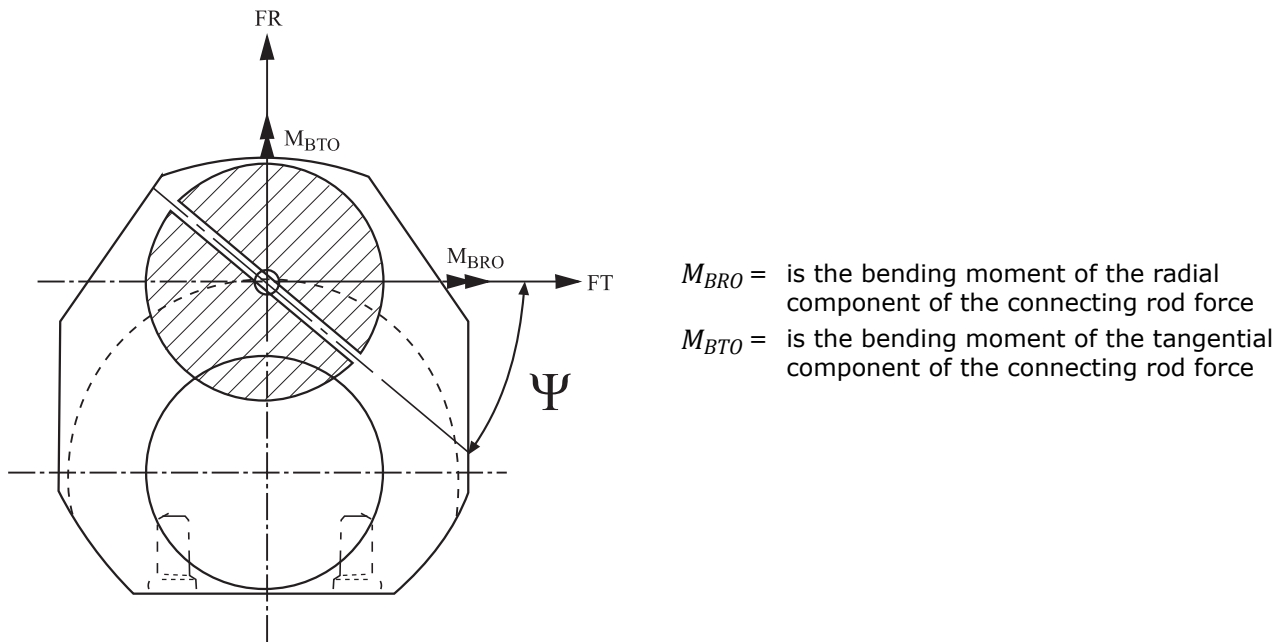


Figure 6 Crankpin section through the oil bore (clockwise rotation)

The alternating and mean stresses due to these bending moments shall be related to the cross-sectional area of the axially bored crankpin.

Mean stresses shall be determined.

2.1.2 Calculation of nominal alternating bending and compressive stresses in web

The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting rod position will be calculated over one working cycle.

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 the radial force Q_{RF} , as defined in [2.1.1.1] and [2.1.1.2], will then be calculated.

In the case of V-type engines, the bending moments, progressively calculated from the gas and inertia forces, of two cylinders acting on one crank throw are superimposed according to phase. Different designs (forked connecting rod, articulated-type connecting rod or adjacent connecting rods) shall be taken into account.

Where there are cranks of different geometrical configurations in one crankshaft, the calculation shall cover all crank variants. The decisive alternating values will then be calculated according to:

$$X_N = \pm \frac{1}{2} \times (X_{\max} - X_{\min})$$

where:

X_N = is considered as alternating force, moment or stress

X_{\max} = is maximum value within one working cycle

X_{\min} = is minimum value within one working cycle

The mean values X_{NM} shall be calculated according to:

$$X_{NM} = \frac{1}{2} \times (X_{\max} + X_{\min})$$

2.1.2.1 Nominal alternating bending and compressive stresses in web cross section

A. Alternating Stress

The calculation of the nominal alternating bending and compressive stresses is as follows:

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

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \times K_e$$

where:

σ_{BFN} = nominal alternating bending stress related to the web [N/mm²]

M_{BRFN} = alternating bending moment related to the center of the web, see [Figure 3](#) and [Figure 4](#) [Nm]

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

W_{eqw} = first moment of area (sectional modulus) related to cross-section of web [mm³]

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

K_e = empirical factor considering to some extent the influence of adjacent crank and bearing restraint

$K_e = 0.8$ for 2-stroke engines

$K_e = 1.0$ for 4-stroke engines

σ_{QFN} = nominal alternating compressive stress due to radial force related to the web [N/mm²]

Q_{RFN} = alternating radial force related to the web, see [Figure 3](#) and [Figure 4](#)

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

F = area related to cross-section of web [mm²]

$$F = B \times W$$

B. Mean stresses

The mean values shall be calculated according to:

$$M_{BRFNM} = \frac{1}{2} \times (M_{BRF_{\max}} + M_{BRF_{\min}})$$

$$\sigma_{BFNM} = \frac{M_{BRFNM}}{W_{eqw}} \times 10^3 \times K_e$$

$$Q_{RFNM} = \frac{1}{2} \times (Q_{RF_{\max}} + Q_{RF_{\min}})$$

$$\sigma_{QFNM} = \frac{Q_{RFNM}}{F} \times K_e$$

2.1.2.2 Nominal bending stress in outlet of crankpin oil bore

A. Alternating stresses

The calculation of nominal alternating bending stress is as follows:

$$\sigma_{BON} = \pm \frac{M_{BON}}{W_e} \times 10^3$$

where:

σ_{BON} = nominal alternating bending stress related to the crankpin diameter [N/mm²]

M_{BON} = alternating bending moment calculated at the outlet of crankpin oil bore [Nm]

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

with

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

ψ = angular position, see Figure 6 [°]

W_e = first moment of area (sectional modulus) related to cross-section of axially bored crankpin [mm³]

$$W_e = \frac{\pi}{32} \left[\frac{D^4 - D_{BH}^4}{D} \right]$$

B. Mean stresses

The mean values shall be calculated according to:

$$M_{BONM} = \frac{1}{2} \times (M_{BO_{\max}} + M_{BO_{\min}})$$

$$\sigma_{BONM} = \frac{M_{BONM}}{W_e} \times 10^3$$

2.1.2.3 Nominal bending stresses in outlet of journal oil bore

A. Alternating stress

If the 3-point bending method is applied (resulting in zero bending moment), the alternating bending moment in the journal M_{BONG} shall be taken at least as $0.5 \times M_{BON}$.

The nominal alternating bending stress is then:

$$\sigma_{BONG} = \pm \frac{M_{BONG}}{\frac{\pi}{32} \times \left(\frac{D_G^4 - D_{BG}^4}{D_G} \right)} \times 10^3$$

B. Mean stress

If the 3-point bending method is applied, the mean bending moment in the journal M_{BONGM} may be disregarded

2.1.3 Calculation of bending stresses in fillets

A. Alternating stresses

The calculation of stresses shall be carried out for the crankpin fillet as well as for the journal fillet. For the crankpin fillet:

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

where

σ_{BH} = alternating bending stress in crankpin fillet [N/mm²]

α_B = stress concentration factor for bending in crankpin fillet (determination, see [3]) [-]

For the journal fillet (not applicable to semi-built crankshaft):

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

where:

σ_{BG} = alternating bending stress in journal fillet [N/mm²]

β_B = stress concentration factor for bending in journal fillet (determination, see [3]) [-]

β_Q = stress concentration factor for compression due to radial force in journal fillet determination, see [3] [-]

B. Mean Stresses

The mean stresses shall be calculated according to:

The crankpin fillet:

$$\sigma_{BHM} = \alpha_B \times \sigma_{BFNM}$$

The journal fillet:

$$\sigma_{BGM} = - \left| \beta_B \times \sigma_{BFNM} + \beta_Q \times \sigma_{QFN} \right|$$

Note:

The signs above are only correct when the bending moment due to max gas pressure is more than twice the bending moment due to mass forces in TDC.

---e-n-d---o-f---n-o-t-e---

2.1.4 Calculation of bending stresses in outlet of crankpin oil bore

A. Alternating stress

The alternating stresses shall be calculated according to:

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

where:

σ_{BO} = alternating bending stress in outlet of crankpin oil bore [N/mm²]

γ_B = stress concentration factor for bending in crankpin oil bore (determination, see [3]) [-]

B. Mean stress

The mean stress shall be calculated according to:

$$\sigma_{BOM} = \gamma_B \times \sigma_{BONM}$$

2.1.5 Calculation of bending stresses in outlet of journal oil bore

A. Alternating stress

$$\sigma_{BOG} = \pm (\gamma_{BG} \times \sigma_{BONG})$$

where:

σ_{BONG} = alternating bending stress in outlet of journal oil bore [N/mm²]

γ_{BG} = stress concentration factor for bending in journal oil bore (determination, see [3]) [-]

B. Mean stresses

$$\sigma_{BOGM} = \gamma_{BG} \times \sigma_{BONGM}$$

2.2 Calculation of alternating torsional stresses

2.2.1 General

The calculation for nominal alternating torsional stresses shall be undertaken by the engine manufacturer according to the information contained in [2.2.2].

The manufacturer shall specify the maximum nominal alternating torsional stress.

2.2.2 Calculation of nominal alternating torsional stresses

The max. and min. torques shall be ascertained for every mass point of the complete dynamic system and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines. Whilst doing so, allowance shall be made for the damping that exists in the system and for unfavorable conditions (misfiring*) in one of the cylinders. The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected.

Note:

*) Misfiring is defined as a cylinder condition when no combustion occurs but only compression cycle.

---e-n-d---o-f---n-o-t-e---

Where barred speed ranges are necessary, they shall be arranged so that satisfactory operation is possible despite their existence. There shall be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for normal firing conditions.

The values received from such calculation shall be submitted.

Alternatively the crankshaft assessment may be based on the specified maximum alternating torsional stress.

For type approvals it is advised to include a variation of the various standard dampers respectively flywheels etc.

Directly coupled engines should take typical installations into account. Flexibly coupled engines may be considered as cut in the elastic coupling.

Where calculations are not available as e.g. for some type approvals, the manufacturer shall state a limiting vibratory torque to be used in the calculation.

The nominal alternating torsional stress in every mass point, which is essential to the assessment, results from the following equation:

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

$$M_{TN} = \pm \frac{1}{2} \times (M_{T_{\max}} - M_{T_{\min}})$$

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

where:

τ_N = nominal alternating torsional stress referred to crankpin or journal [N/mm²]

M_{TN} = maximum alternating torque [Nm]

- W_P = polar first u of area (sectional modulus) related to cross-section of axially bored crankpin or bored journal [mm³]
 M_{Tmax} = maximum value of the torque [Nm]
 M_{Tmin} = minimum value of the torque [Nm]

For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in further calculations is the highest calculated value, according to above method, occurring at the most torsionally loaded mass point of the crankshaft system.

Where barred speed ranges exist, the torsional stresses within these ranges shall not be considered for assessment calculations. The approval of crankshaft will be based on the installation having the largest nominal alternating torsional stress, but not exceeding the maximum figure specified by the engine manufacturer.

Thus, for each installation, it shall be ensured by suitable calculation that this approved nominal alternating torsional stress is not exceeded. This calculation shall be submitted for assessment.

The nominal mean torsional stress is:

$$\tau_{NM} = \frac{M_{TNM}}{W_P} \times 10^3$$

$$M_{TNM} = \frac{1}{2} \times (M_{T_{max}} + M_{T_{min}})$$

This may be determined in every mass point which is essential to the crankshaft assessment, but a more practical approach shall use the mean stress at the crank that has the highest alternating torque. This can be determined from a quasistatic torque evaluation through the crankshaft (vectorial torque summation assuming clamped end and no dynamics).

2.2.3 Calculation of torsional stresses in fillets and outlet of oil bores

The calculation of stresses shall be carried out for the crankpin fillet, the journal fillet and the outlet of the oil bores.

2.2.3.1 For the crankpin fillet

A. Alternating stress

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

where:

- τ_H = alternating torsional stress in crankpin fillet [N/mm²]
 α_T = stress concentration factor for torsion in crankpin fillet (determination, [3]) [-]
 τ_N = nominal alternating torsional stress related to journal diameter [N/mm²]

B. Mean stress

$$\tau_{HM} = \alpha_T \times \tau_{NM}$$

where:

- τ_{NM} = nominal mean torsional stress related to journal diameter [N/mm²]

2.2.3.2 For the journal fillet (not applicable to semi-built crankshafts)

A. Alternating stress

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

where:

τ_G = alternating torsional stress in journal fillet [N/mm²]

β_T = stress concentration factor for torsion in journal fillet (determination, [3]) [-]

τ_N = nominal alternating torsional stress related to journal diameter [N/mm²]

B. Mean stress

$$\tau_{GM} = \pm (\beta_T \times \tau_{NM})$$

where:

τ_{NM} = nominal mean torsional stress related to journal diameter [N/mm²]

2.2.3.3 For the outlet of crankpin oil bore

A. Alternating stress

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

where:

σ_{TO} = alternating stress in outlet of crankpin oil bore due to torsion [N/mm²]

γ_T = stress concentration factor for torsion in outlet of crankpin oil bore (determination, see [3]) [-]

τ_N = nominal alternating torsional stress related to crankpin diameter [N/mm²]

B. Mean stress

$$\sigma_{TOM} = \gamma_T \times \tau_{NM}$$

where:

τ_{NM} = nominal mean torsional stress related to crankpin diameter [N/mm²]

2.2.3.4 For the outlet of journal oil bore

A. Alternating stress

$$\sigma_{TOG} = \pm (\gamma_{TG} \times \tau_N)$$

where:

σ_{TOG} = alternating stress in outlet of journal oil bore due to torsion [N/mm²]

$\gamma_{TG}[-]$ = stress concentration factor for torsion in outlet of journal oil bore, see [3] [-]

τ_N = nominal alternating torsional stress related to journal diameter [N/mm²]

B. Mean stress

$$\sigma_{TOGM} = \gamma_{TG} \times \tau_{NM}$$

where:

τ_{NM} = nominal mean torsional stress related to journal diameter [N/mm²]

3 Evaluation of stress concentration factors

3.1 General

The stress concentration factors are evaluated by means of the formulae according to [3.2], [3.3] and [3.4] applicable to the fillets and crankpin oil bore of solid forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. It shall be noticed that stress concentration factor formulae concerning the oil bore are only applicable to a radially drilled oil hole. All formulae are based on investigations of FVV (Forschungsvereinigung Verbrennungskraftmaschinen) for fillets and on investigations of ESDU (Engineering Science Data Unit) for oil holes.

Note:

Where the geometry of the crankshaft is outside the boundaries of the analytical stress concentration factors (SCF) the calculation method detailed in App.C may be undertaken.

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All crank dimensions necessary for the calculation of stress concentration factors are shown in Figure 3 to Figure 7. Table 1 defines the stress concentration factors used.

Table 1 Stress concentration factors

α_B, β_B	=	Stress concentration factors in bending. It is the ratio between the maximum absolute value of the principal stresses in the fillets and the absolute value of the nominal bending stress referred to the web cross section.
β_Q	=	Stress concentration factor for compressive stresses in the journal fillet. It is the ratio between the maximum fillet stress (absolute value) and the nominal compressive stress (absolute value) related to the web cross section.
α_T, β_T	=	Stress concentration factor for torsion. It is the ratio of the maximum equivalent shear stress which is occurring in the fillets under torsional load to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see App.A).
γ_B	=	Stress concentration factors for bending. Is the ratio of the maximum principal stress occurring at the outlet of the crankpin oil hole under bending - to the corresponding nominal stress related to the axially bored crankpin cross section (see App.B).
γ_T	=	Stress concentration factors for torsion. Is the ratio of the maximum principal stress occurring at the outlet of the crankpin oil hole under torsional loads - to the corresponding nominal stress related to the axially bored crankpin cross section (see App.B).

For outlet bores in journals the same definitions apply.

When reliable measurements and/or calculations are available, which can allow direct assessment of stress concentration factors, the relevant documents and their analysis method shall be submitted in order to demonstrate their equivalence to present rules evaluation. This shall always be performed when dimensions are outside of any of the validity ranges for the empirical formulae presented in [3.2] to [3.4].

App.C and App.F describe how FE analyses may be used for the calculation of the stress concentration factors. Care should be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

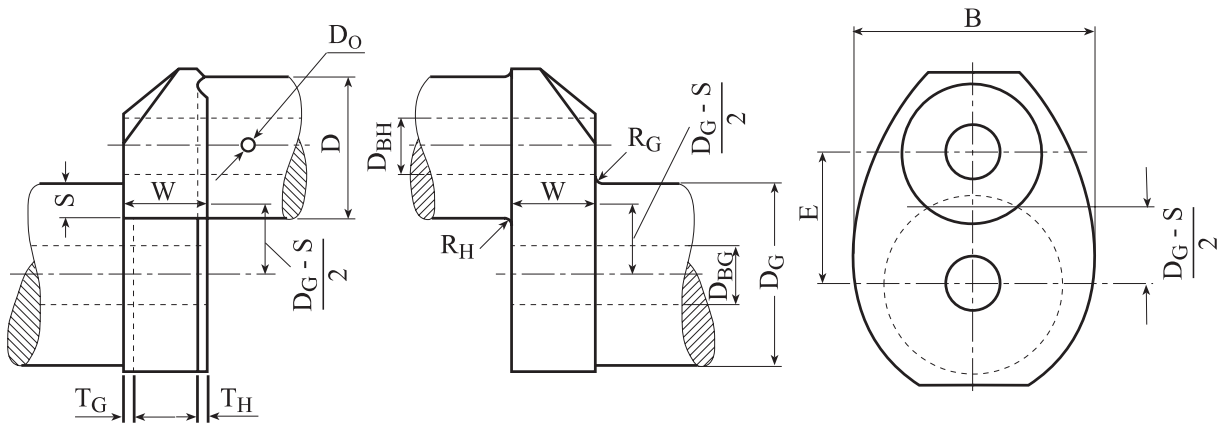


Figure 7 Crank dimensions

Where:

- D = crankpin diameter [mm]
- D_{BH} = diameter of axial bore in crankpin [mm]
- D_O = diameter of oil bore in crankpin [mm]
- D_{OG} = diameter of oil bore in journal [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]
- B* = web width [mm]

* In the case of 2 stroke semi-built crankshafts:

— when $T_H > R_H$, the web thickness shall be considered as equal to:

$$W_{red} = W - (T_H - R_H) \text{ see Figure 5}$$

— web width B shall be taken in way of crankpin fillet radius centre according to Figure 5.

The following related dimensions will be applied for the calculation of stress concentration factors in:

Table 2 Crank parameters

<i>Crankpin fillet</i>	<i>Journal fillet</i>
$r = \frac{R_H}{D}$	$r_G = \frac{R_G}{D_G}$
$s = \frac{S}{D}$	$s_G = \frac{S}{D_G}$
$w = \frac{W}{D}$ (with overlap)	$w_G = \frac{W}{D_G}$ (with overlap)
$w = \frac{W_{red}}{D}$ (without overlap)	$w_G = \frac{W_{red}}{D_G}$ (without overlap)
$b = \frac{B}{D}$	$b_G = \frac{B}{D_G}$
$d_o = \frac{D_o}{D}$	$d_{oG} = \frac{D_{oG}}{D_G}$
$d_H = \frac{D_{BH}}{D}$	$d_G = \frac{D_{BH}}{D_G}$
$t_H = \frac{T_H}{D}$	$t_G = \frac{T_G}{D}$

Stress concentration factors are valid for the ranges of related dimensions for which the investigations have been carried out.

Note:

FVV investigations were made with models having $D=D_G$. Using the empirical formulae on crankshafts with $D_G>D$ may lead to journal fillet stress concentration factors that are more uncertain than when $D=D_G$.

---e-n-d---o-f---n-o-t-e---

Ranges are as follows:

Table 3 Ranges of crank parameters

$$\begin{aligned}
 s &\leq 0.5 \\
 0.2 &\leq w \leq 0.8 \\
 1.1 &\leq b \leq 2.2 \\
 0.03 &\leq r \leq 0.13 \\
 0 &\leq d_G \leq 0.8 \\
 0 &\leq d_H \leq 0.8 \\
 0 &\leq d_o \leq 0.2
 \end{aligned}$$

Low range of crank parameter 's' may be extended down to large negative values provided that:

- if calculated f (recess) < 1 then the factor f (recess) shall not be considered (f (recess) = 1)
- if $s < -0.5$ then f (s,w) and f (r,s) shall be evaluated replacing actual value of s by -0.5 .

The same ranges of validity apply when the related dimensions are referred to the journal diameter.

As the original (FVV) empirical formulae for the crankpin and the journal fillets were established on the basis of 50% reliability, and have standard deviations of approximately 10% (approximately 7% of torsion), the coefficients have been increased by a factor for bending (presently 1,10) and a factor for torsion (presently 1,07) for the purpose of higher reliability.

3.2 Crankpin fillet

The stress concentration factor for bending (α_B) is:

$$\alpha_B = 3.30 \times f(s, w) \times f(w) \times f(b) \times f(r) \times f(d_G) \times f(d_H) \times f(recess)$$

The formula is applicable to principal stresses and one standard deviation above 50% reliability where:

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

$$f(w) = 2.1790 \times w^{0.7171}$$

$$f(b) = 0.6840 - 0.0077 \times b + 0.1473 \times b^2$$

$$f(r) = 0.2081 \times r^{(-0.5231)}$$

$$f(d_G) = 0.9993 + 0.27 \times d_G - 1.0211 \times d_G^2 + 0.5306 \times d_G^3$$

$$f(d_H) = 0.9978 + 0.3145 \times d_H - 1.5241 \times d_H^2 + 2.4147 \times d_H^3$$

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

The stress concentration factor for torsion (α_T) is:

$$\alpha_T = 0.856 \times f(r, s) \times f(b) \times f(w)$$

The formula is applicable to one standard deviation above 50% reliability where:

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

$$f(b) = 7.8955 - 10.654 \times b + 5.3482 \times b^2 - 0.857 \times b^3$$

$$f(w) = w^{(-0.145)}$$

3.3 Journal fillet, not applicable to semi-built crankshafts

The stress concentration factor for bending (β_B) is:

$$\beta_B = 3.33 \times f_B(s_G, w_G) \times f_B(w) \times f_B(b) \times f_B(r) \times f_B(d_G) \times f_B(d_H) \times f(recess)$$

The formula is applicable to principal stresses and one standard deviation above 50% reliability where:

$$f_B(s_G, w_G) = -1.7625 + 2.9821 \times w_G - 1.5276 \times w_G^2 + (1 - s_G) \times (5.1169 - 5.8089 \times w_G + 3.1391 \times w_G^2) + (1 - s_G)^2 \times (-2.1567 + 2.3297 \times w_G - 1.2952 \times w_G^2)$$

$$f_B(w_G) = 2.2422 \times w_G^{0.7548}$$

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

$$f_B(r_G) = 0.1908 \times r_G^{(-0.5568)}$$

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

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

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

The stress concentration factor for compression (β_Q)

$$\beta_Q = 3.69 \times f_Q(s) \times f_Q(w) \times f_Q(b) \times f_Q(r) \times f_Q(d_H) \times f(recess)$$

The formula is applicable to principal stresses and one standard deviation above 50% reliability. where:

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

$$f_Q(w_G) = \frac{w_G}{0.0637 + 0.9369 \times w_G}$$

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

$$f_Q(r_G) = 0.5331 \times r_G^{(-0.2038)}$$

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

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

The stress concentration factor for torsion (β_T) is:

$$\beta_T = 0.856 \times f(r_G, s_G) \times f(b_G) \times f(w_G)$$

The stress concentration is applicable to one standard deviation above 50% reliability where:

$$f(r_G, s_G) = r_G^{[-0.322 + 0.1015 \times (1 - s)]}$$

$$f(b_G) = 7.8955 - 10.654 \times b_G + 5.3482 \times b_G^2 - 0.857 \times b_G^3$$

$$f(w_G) = w_G^{(-0.145)}$$

3.4 Outlet of crankpin oil bore

The stress concentration factor for bending (γ_B) is:

$$\gamma_B = 3 - 5.88 \times d_o + 34.6 \times d_o^2$$

The stress concentration factor for torsion (γ_T) is:

$$\gamma_T = 4 - 6 \times d_o + 30 \times d_o^2$$

3.5 Outlet of journal oil bore

the stress concentration factor for bending (γ_{BG}) is:

$$\gamma_{BG} = 3 - 5.88 \times d_{OG} + 34.6 \times d_{OG}^2$$

The stress concentration factor for torsion (γ_{TG}) is:

$$\gamma_{TG} = 4 - 6 \times d_{OG} + 30 \times d_{OG}^2$$

3.6 Inclined oil bores

Inclined oil bores have higher stress concentrations than oil bores perpendicular to the surface.

For small deviations from perpendicular oil bore exit the stress concentration factors in [3.4] and [3.5] shall be multiplied with:

For bending: $1 + \frac{\varphi}{80}$

For torsion: $1 + \frac{\varphi}{100} + \frac{\psi}{200}$

Range for validity:

$$\varphi < 25^\circ$$

$$\psi < 40^\circ$$

where

φ = is the angular deviation (degrees) from perpendicular oil bore direction as projected in the transverse (crosswise) plane

ψ = is the angular deviation (degrees) from perpendicular oil bore direction as projected longitudinally in the crankpin plane.

4 Additional bending stress

In addition to the alternating bending stresses in fillets (see [2.1.3]) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations shall be considered by applying σ_{add} as given by Table 4:

Table 4 Additional bending stress

Type of engine	σ_{add} [N/mm ²]
Crosshead engines	± 30 (*)
Trunk piston engines	± 10

* The additional stress of ± 30 [N/mm²] is composed of two components:

- 1) an additional stress of ± 20 [N/mm²] resulting from axial vibrations.
- 2) an additional stress of ± 10 [N/mm²] resulting from misalignment/bedplate deformation.

It is recommended that a value of ± 20 [N/mm²] is used for the axial vibration component for assessment purposes where axial vibration results of the complete dynamic system (engine/shafting/gearing/propeller) are not available. Where axial vibration calculation results of the complete system are available, the calculated figures may be used instead.

Other values may be used if substantiated.

5 Calculations of equivalent alternating stress

This chapter is not relevant for this method, but is mentioned here to retain the numbering of chapters.

6 Calculation of fatigue strength

6.1 General

The fatigue strength is defined as the permanently endurable (i.e. $>>10^7$ cycles) alternating principal stress considering influences as mean stress, mechanical strength, size effect, stress gradient, production method, surface roughness, and surface treatment as hardening and/or cold deformation if applicable.

The fatigue strength may be determined by tests or by empirical formulae as given below. Test results may also be combined with pertinent elements of the empirical formulae, e.g. push-pull test results can be combined with mean stress influence and so on.

When the fatigue strength is determined by means of testing (full size or small specimen), the number of test and the statistical evaluation method shall be selected so as to enable determination of the fatigue strength as the average value minus one standard deviation. For more details, see App.D.

It is also essential that the tested material is as representative as possible for the future production, i.e. material cleanliness, production method, surface roughness, and mechanical strength shall not be superior to the minimum requirements for the production.

When full size tests are used to determine fatigue strength of crankshafts with surface treated fillets (and/or oil holes), the acceptance shall be combined with an approved procedure for quality control of the surface treatment.

6.2 Empirical formulae for non-surface treated fillets

The fatigue strengths in bending and torsion to be determined as:

For web fillets:

$$\sigma_F = K_b \times f(\sigma_B; \sigma_{0.2}) \times f(R_a) \times f(\chi) \times f(\sigma_M)$$

$$\tau_f = K_t \times \frac{1}{\sqrt{3}} \times f(\sigma_B; \sigma_{0.2}) \times f(R_a) \times f(\chi) \times f(\tau_M)$$

For oil holes:

$$\sigma_{fh} = K_h \times f(\sigma_B; \sigma_{0.2}) \times f(R_a) \times f(\chi) \times f(\sigma_M)$$

where:

- K_b , K_t and K_h , see [Table 5](#)
- $f(\sigma_B; \sigma_{0.2})$, see [\[6.2.2\]](#)
- $f(R_a)$, see [\[6.2.3\]](#)
- $f(\chi)$, see [\[6.2.4\]](#)
- $f(\sigma_M)$, see [\[6.2.5\]](#)
- $f(\tau_M)$, see [\[6.2.5\]](#)

When crankpins and/or journals are surface hardened, special attention shall be given to the transition zones between surface hardened areas and the soft core material, see [\[6.3\]](#) and [App.E](#).

6.2.1 Influence of production methods

K_b and K_t represent the influence of various production methods on fatigue in web fillet, and K_h in oil holes.

Table 5 K factors

Groups	K_b	K_t	K_h
Group 1: continuous grain flow forged ¹⁾	1.1	1.05	1
Group 2 free form forged ²⁾	1	1	1
Group 3: slab forged ³⁾ , form cut and twisted	0.9	0.9	1
Group 4: cast steel	0.7	.085	0.7
Group 5: Clean ⁴⁾ steel, cgf and free form forging	1.1	1.1	1.1
Clean ⁴⁾ steel, slab forging	1.0	1.0	1.1
Group 6: "Super" clean forged steel	see below ⁵⁾		

- 1) Continuous grain flow (cgf) forged means documented forging that results in a flow of segregations tangential to the main principal stress direction in the web fillets. It does not apply to oil holes (where

$K_h = 1$). The documentation shall be made on a crankthrow that is cut in the crank plane and etched in order to illustrate the grain flow. It is essential that this is made with fillets in finished condition. I.e. for recessed fillets, cgf may be difficult to obtain as the recess normally cuts the grain flow.

- 2) Free form forging means all kinds of forging processes that are neither cgf nor slab forging (see footnote 3).
- 3) Slab forging means processes where a forged slab is oxycut or machined to form crankthrows, and then twisted into positions. All similar processes where material from the center of the slab ends up in the web fillets fall into this category. This does not apply to oil holes provided that they are not in a part that originates from the slab center i.e. normally $K_h = 1$ for oil holes.
- 4) For steels with a especially high cleanliness (see [DNVGL-RU-SHIP Pt.2 Ch.2 Sec.5](#))
- 5) Factors will be decided on case by case basis.

6.2.2 Mechanical strength influence

$f(\sigma_B; \sigma_{0.2})$ = the influence of the mechanical strength on the push-pull fatigue strength.

$$f(\sigma_B; \sigma_{0.2}) = 0.218 \times \sigma_B + 293 \times \left(\frac{\sigma_{0.2}}{\sigma_B} \right) - 72$$

$$(\text{from VBFeh 1983 II: } \sigma_F = 0.242 \times \sigma_B + 325 \times \frac{\sigma_{0.2}}{\sigma_B} - 80)$$

Note:

Use of $\sigma_B > 800 \text{ MPa}$ or $\sigma_{0.2} > 600 \text{ MPa}$ may require stricter requirements for cleanliness than those given in some material specifications.

---e-n-d---o-f---n-o-t-e---

Note:

The above factors in [Table 5](#) shall not be used when realistic values are used in the formulae.

---e-n-d---o-f---n-o-t-e---

6.2.3 Surface roughness influence

The surface roughness condition of all web fillets and oil holes (to a depth of at least 1,5 times the oil bore diameter) is assumed to be smooth grinding or better, preferably polishing. Web fillets should be polished along the fillet contour. Hence the $f(R_a)$ is unity. However, for oil bores that are not smoothly finished to the above mentioned depth, a factor of $f(R_a) = 0.9$ applies.

$f(R_a)$ = the influences of surfaces roughness.

6.2.4 Stress gradient influence

The other size influences such as statistical and metallurgical are not included because:

- the statistical is covered by the use of one standard deviation below average for test results, respectively covered by the empirical formulae for push-pull.
- the metallurgical is covered by use of representative test specimens taken from one (or both) extended end(s) of the forged crankshafts or equivalent for single throws.

$f(\chi)$ = the size influence due to the stress gradient.

$f(\chi)$ is different for bending and may be taken as:

$$f(\chi) = 1 + \left(0.5 \times 10^{\left(\frac{-\sigma_{0.2}}{712} \right)} \right) \times \sqrt{\frac{2}{R_X}}$$

$f(\chi)$ is different for torsion and may be taken as:

$$f(\chi) = 1 + \left(0.5 \times 10^{\left(\frac{-\sigma_{0.2}}{712} \right)} \right) \times \sqrt{\frac{1}{R_X}}$$

where:

R_X is R_H for crankpin fillets and R_G for journals fillets and $\frac{D_O}{2}$ respectively $\frac{D_{OG}}{2}$ for oil bores.

However, R_X is not to be taken <2mm for calculation purposes.

6.2.5 Mean stress influence

The mean stress influence is considered by means of a parabolic relation as:

$$\sqrt{1 - \frac{\sigma_{mean}}{\sigma_B}} \text{ respectively } \sqrt{1 - \frac{|\tau_{mean}|}{\tau_B}}$$

where:

$$\tau_B = \frac{\sigma_B}{\sqrt{3}}$$

For crankpin fillets:

$$\sigma_{Mean} = \sigma_{BHM}, \text{ see [2.1.3], B.}$$

$$\tau_{Mean} = \tau_{THM}, \text{ see [2.2.3.1] b.}$$

For journal fillets:

$$\sigma_{Mean} = \sigma_{BGM}, \text{ see [2.1.3] B.}$$

$$\tau_{Mean} = \tau_{GM}, \text{ see [2.2.3.2] B.}$$

It should be noted that crankpin fillet mean bending stresses normally are positive (tension) and hence reduce the fatigue strength. Journal fillet mean bending stresses are normally negative (compression) and hence increase the fatigue strength. For torsion, any mean stress regardless direction reduces the fatigue strength.

For oil holes in crankpin:

$$\sigma_{Mean} = \frac{1}{3} \times \sigma_{BOM} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \left(\frac{\sigma_{TOM}}{\sigma_{BOM}} \right)^2} \right]$$

For σ_{BOM} , see [2.1.4] B.

For σ_{TOM} , see [2.2.3.3] B.

For oil holes in journals:

$$\sigma_{Mean} = \frac{1}{3} \times \sigma_{BOGM} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \left(\frac{\sigma_{TOGM}}{\sigma_{BOGM}} \right)^2} \right]$$

For σ_{BOGM} , see [2.1.5] B.

For σ_{TOGM} , see [2.2.3.4] B.

6.3 Surface treated fillets and / or oil holes

Surface treatment of fillets and oil holes means any method aiming at increased fatigue strength in local areas. It is divided into surface hardening and cold deformation.

Surface hardening means all kinds of thermal treatment such as induction hardening and nitriding. These processes result in increased surface hardness, and if properly made, also compressive residual stresses to a certain depth. However, especially for induction hardening, the transition zones depthwise and at the end of the layer have tensile residual stresses. It is therefore essential to check also transition zones for fatigue.

Cold deformation means all kinds of "cold" treatments such as shot peening, stroke peening, cold rolling and ball coining (oil holes). These processes result in negligible hardness increase but in considerable compressive residual stresses to a certain depth. Also with these processes the transition zones have to be checked for fatigue.

See App.D for fatigue testing and App.E for evaluation of safety against fatigue in surface treated fillets and/or oil bore outlets.

7 Acceptance criteria

7.1 General

For the fatigue criteria mentioned in [7.2], [7.3] and [7.4] a common minimum safety factor applies, as:

$$S = 1.15$$

All stresses are in N/mm².

7.2 the crankpin fillet criterion

The stresses in the crankpin fillet are to fulfil the following criterion:

$$\left(\frac{\sigma_H}{\sigma_{fH}} \right)^2 + \left(\frac{\tau_H}{\tau_{fH}} \right)^2 \leq \frac{1}{S^2}$$

where

$\sigma_H = \sigma_{BH} + \sigma_{add}$, see [2.1.3], respectively [4]

$\tau_H =$ see [2.2.3.1] A.

$\sigma_{fH} = \sigma_f$ as determined in [6] for the crankpin fillet

$\tau_{fH} = \tau_f$ as determined in [6] for the crankpin fillet.

For semibuilt crankshafts with a relatively low distance between the pin and the shrunk in journal ($s = \frac{S}{D} > -0.1$), it is necessary to consider the influence of the shrinkage stresses on the crankpin fillet fatigue strength. This applies in particular to the approximately 45 degrees crankpin plane.

The above mentioned criterion may be used, however, with some modifications as:

- Nominal bending stresses as well as stress concentration factors to be calculated in the 45 degree plane.
- Shrinkage stresses in the fillet of the 45° plane to be added to the mean bending stresses when determining the fatigue strength.

7.3 The journal fillet criterion

(Not applicable to semibuilt crankshafts.)

The stresses in the journal fillet are to fulfil the following criterion:

$$\left(\frac{\sigma_G}{\sigma_{fG}}\right)^2 + \left(\frac{\tau_G}{\tau_{fG}}\right)^2 \leq \frac{1}{s^2}$$

where:

$\sigma_G = \sigma_{BG} + \sigma_{add}$, see [2.1.3], respectively [4]

$\tau_G =$ see [2.2.3.3]

$\sigma_{fG} = \sigma_f$ as determined in [6] for the journal fillet

$\tau_{fG} = \tau_f$ as determined in [6] for the journal fillet.

7.4 The oil hole criterion

The stresses in the oil holes shall fulfil the following criteria:

Outlet of crankpin bores:

$$\frac{1}{3} \times \sigma_{BO} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \times \left(\frac{\sigma_{TO}}{\sigma_{BO}} \right)^2} \right] \leq \frac{\sigma_{fO}}{s}$$

where:

$\sigma_{BO} =$ see [2.1.4] A.

$\sigma_{TO} =$ see [2.2.3.4] A.

$\sigma_{fO} = \sigma_{fh}$ as determined in [6] for the crankpin oil hole, with attention to [6.3] if the crankpin is surface hardened.

Outlet of journal bores

$$\frac{1}{3} \times \sigma_{BOG} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \times \left(\frac{\sigma_{TOG}}{\sigma_{BOG}} \right)^2} \right] \leq \frac{\sigma_{fOG}}{s}$$

where:

$\sigma_{BOG} =$ see [2.1.5] A.

$\sigma_{TOG} = \text{see [2.2.3.4] A.}$

$\sigma_{fOG} = \sigma_{fh}$ as determined in [6] for the journal hole, with attention to [6.3] if the journal is surface hardened.

8 Calculation of shrink-fits of semi-built crankshafts

See IACS UR M53.8.

9 Appendix information

The appendixes in this guideline are based on IACS UR M53 appendixes and shall be used when applicable with the following comments:

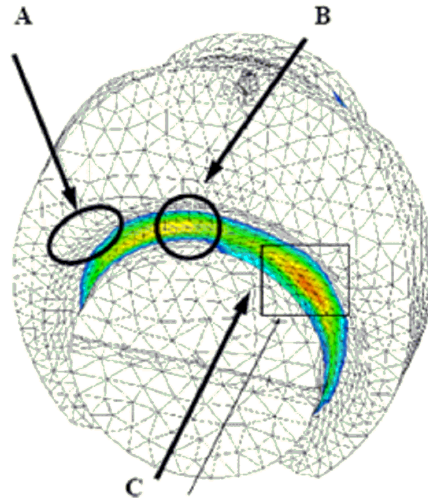
- the appendixes are numbered from A to F which is corresponding to IACS UR M53 Appendix I to VI
- text or content added or modified compared to the IACS appendixes is written in bold-italic
- text deleted from the original is not highlighted.

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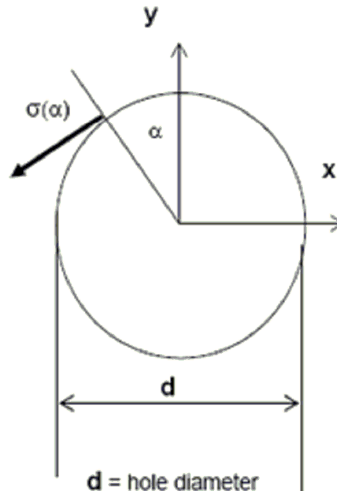
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APPENDIX A DEFINITION OF STRESS CONCENTRATION FACTORS IN CRANKSHAFT FILLETS



	Stress	Max $ \sigma_3 $	Max σ_1	
Torsional loading	Location of maximal stresses	A	C	B
	Typical principal stress system Mohr's circle diagram with $\sigma_2 = 0$			
	Equivalent stress and S.C.F.	$\tau_{equiv} = \frac{\sigma_1 - \sigma_3}{2}$ $S.C.F. = \frac{\tau_{equiv}}{\tau_e} \text{ for } \alpha_T, \beta_T$		
Bending loading	Location of maximal stresses	B	B	B
	Typical principal stress system Mohr's circle diagram with $\sigma_3 = 0$			
	Equivalent stress and S.C.F.	$\sigma_{equiv} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2}$ $S.C.F. = \frac{\sigma_{equiv}}{\sigma_e} \text{ for } \alpha_B, \beta_B, \beta_Q$		

APPENDIX B STRESS CONCENTRATION FACTORS AND STRESS DISTRIBUTION AT THE EDGE OF OIL DRILLINGS



Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma_n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha = \sigma_n \gamma_B / 3 [1 + 2 \cos(2\alpha)]$	$\gamma_B = \sigma_{\max} / \sigma_n \text{ for } \alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_T \tau_n \sin(2\alpha)$	$\gamma_T = \sigma_{\max} / \tau_n \text{ for } \alpha = \frac{\pi}{4} + k \frac{\pi}{2}$
Tension + shear	$\begin{bmatrix} \sigma_n & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3} \sigma_n \left\{ 1 + 2 \left[\cos(2\alpha) + \frac{3}{2} \frac{\gamma_T}{\gamma_B} \frac{\tau_n}{\sigma_n} \sin(2\alpha) \right] \right\}$	$\sigma_{\max} = \frac{\gamma_B}{3} \sigma_n \left[1 + 2 \sqrt{1 + \frac{9}{4} \left(\frac{\gamma_T}{\gamma_B} \frac{\tau_n}{\sigma_n} \right)^2} \right]$ $\text{for } \alpha = \frac{1}{2} \text{tg}^{-1} \left(\frac{3\gamma_T \tau_n}{2\gamma_B \sigma_n} \right)$

APPENDIX C ALTERNATIVE METHOD FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILISING FINITE ELEMENT METHOD

Contents

1. General
2. Model requirements
 - 2.1. Element mesh recommendations
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 - 2.3. Element mesh quality criteria
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3. Load cases
 - 3.1. Torsion
 - 3.2. Pure bending (4 point bending)
 - 3.3. Bending with shear force (3 point bending)
 - 3.3.1. Method 1
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1 General

The objective of the analysis is to substitute the analytically calculated stress concentration factors (SCF) at the crankshaft fillets by suitable FEM calculated figures. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries. Use of these formulae beyond any of the various validity ranges can lead to erroneous results in either direction, i.e. results that more inaccurate than indicated by the mentioned standard deviations. Therefore the FEM-based method is highly recommended.

The SCF's calculated according to 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 [Sec.1](#) or the alternative methods, von Mises stresses shall 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 shall be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude applied for all load cases.

The calculation of SCF at the oil bores is at present not covered by this document. It is advised to check the element accuracy of the FE solver in use, e.g. by modelling of a simple geometry and compare 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 Model requirements

The basic recommendations and perceptions for building the FE-model are presented in [Sec.1 \[2.1\]](#). It is mandatory for the final FE-model to fulfil the requirement in [\[2.3\]](#).

2.1 Element mesh recommendations

In order to fulfil the mesh quality criteria it is advised to construct the FE-model for the evaluation of stress concentration factors according to the following recommendations:

- The model consists of one complete crank, from main bearing centre-line to opposite side main bearing centre-line.
- Element types used in the vicinity of the fillets:
 - 10 node tetrahedral elements
 - 8 node hexahedral elements
 - 20 node hexahedral elements.
- Mesh properties in fillet radii. The following applies to ± 90 degrees in circumferential direction from the crank plane:
 - Maximum element size $a = \frac{r}{4}$ through the entire fillet as well as the in circumferential direction. When using 20 node hexahedral elements, the element size in the circumferential direction may be extended up to $5 \times a$. In the case of multi-radii fillet, r is the local fillet radius (if 8 node hexahedral elements are used even smaller element size is required to meet the quality criteria).
 - Recommended manner for element size in fillet depth direction:
 - first layer thickness equal to element size of a
 - second layer thickness equal element to size of $2 \times a$
 - third layer thickness equal element to size of $3 \times a$.
- Minimum 6 elements across web thickness.
- Generally the rest of the crank should be suitable for numeric stability of the solver.
- Counterweights only shall be modelled when influencing the global stiffness of the crank significantly.
- Modelling of oil drillings not necessary as long as the influence on global stiffness is negligible or the proximity to the fillet is more than $2r$, [Figure 1](#) below.
- Drillings and holes for weight reduction shall be modelled.
- Sub-modeling may be used as far as the software requirements are fulfilled.

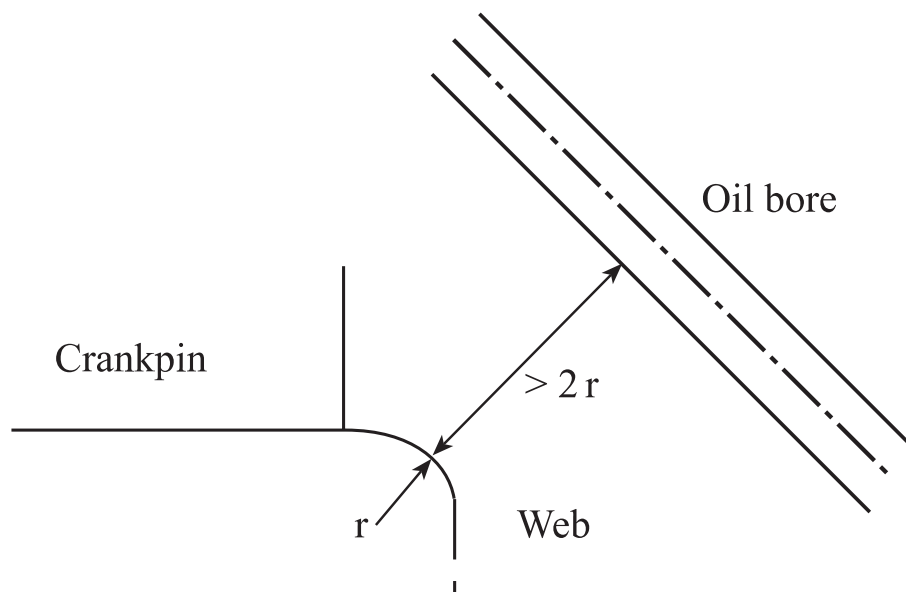


Figure 1 Proximity to fillet

2.2 Material

This guideline does not consider material properties such as Young's modulus (E) and Poisson's ratio (ν). In FE analysis those material parameters are required, as strain is primarily calculated and stress is derived from strain using the Young's modulus and Poisson's ratio. Reliable values for material parameters shall be used, either as quoted in literature or measured on representative material samples.

For steel the following is advised:

$$E = 2.05 \times 10^5 \text{ MPa} \text{ and } \nu = 0.3$$

2.3 Element mesh quality criteria

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

2.3.1 Principal stresses criterion

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

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

2.3.2 Averaged/un-averaged stresses criterion

The criterion is based on observing the discontinuity of stress results over elements at the fillet for the calculated SCF:

- Un-averaged nodal stress results calculated from each element connected to a node_i should differ less than 5% from the 100% averaged nodal stress results at this node_i at the examined location.

3 Load cases

To substitute the analytically determined SCF in [Sec.1](#) the following load cases shall be calculated.

3.1 Torsion

In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded in pure torsion. In the model surface warp at the end faces is suppressed.

Torque is applied to the central node located at the crankshaft axis. This node acts as master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

Boundary and load conditions are valid for both in-line and V-type engines.

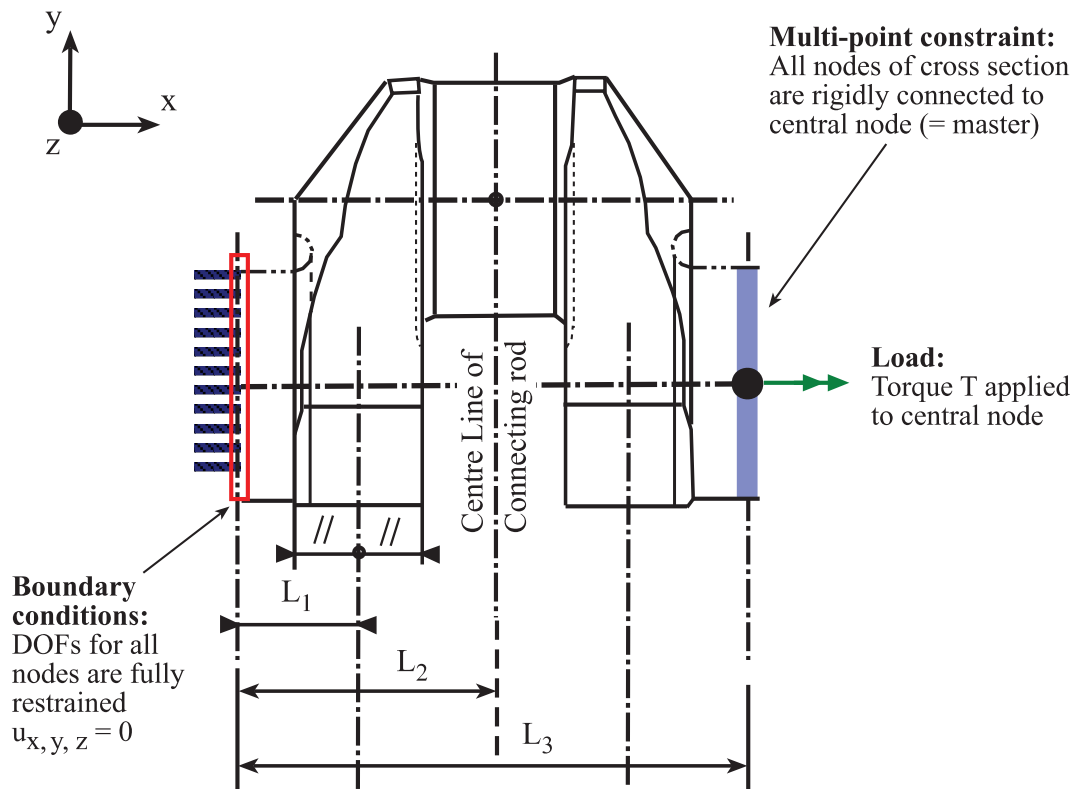


Figure 2 Boundary and load conditions for the torsion case

For all nodes in both journal and crank pin fillet principal stresses are extracted and the equivalent torsional stress is calculated:

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

The maximum value taken for the subsequent calculation of the SCF:

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

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

Where τ_N is nominal torsion stress referred to the crankpin respectively journal as per [Sec.1 \[2.2.2\]](#) with the torsional torque τ :

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

3.2 Pure bending (4 point bending)

In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded in pure bending. In the model surface warp at the end faces is suppressed.

The bending moment is applied to the central node located at the crankshaft axis. This node acts as master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

Boundary and load conditions are valid for both in-line and V-type engines.

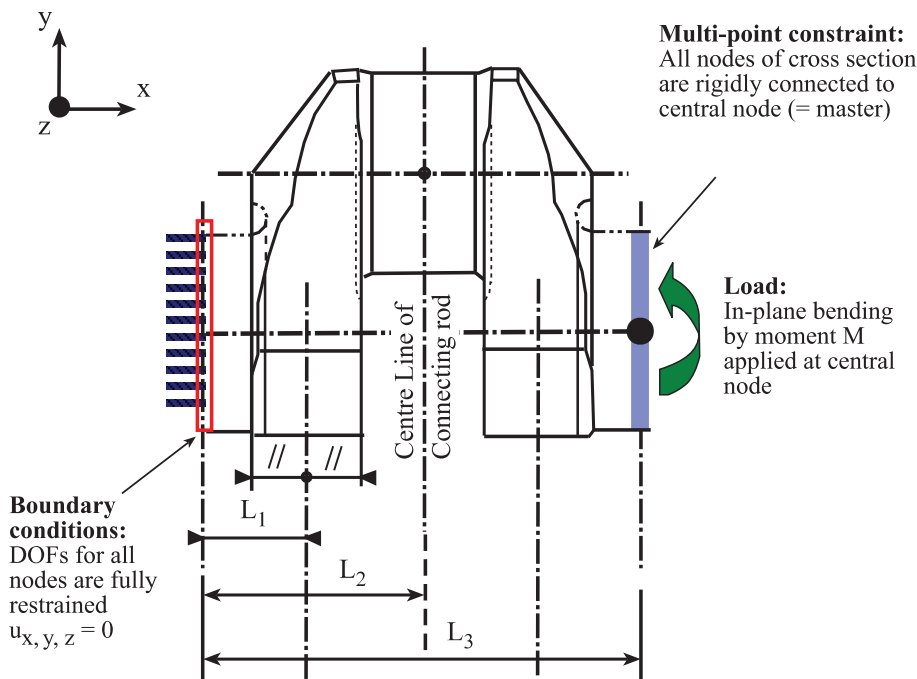


Figure 3 Boundary and load conditions for the pure bending load case

For all nodes in both journal and pin fillet von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the SCF according to:

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

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

Nominal stress σ_N is calculated as per [Sec.1 \[2.1.2.1\]](#) with the bending moment M :

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

3.3 Bending with shear force (3 point bending)

This load case is calculated to determine the SCF for pure transverse force (radial force, β_Q) for the journal fillet. In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded in 3-point bending. In the model, surface warp at the both end faces is suppressed. All nodes are connected rigidly to the centre node, boundary conditions are applied to the centre nodes. These nodes act as master nodes with 6 degrees of freedom.

The force is applied to the central node located at pin centre-line of the connecting rod. This node is connected to all nodes of the pin cross sectional area. Warping of the sectional area is not suppressed.

Boundary and load conditions are valid for in-line and V-type engines. V-type engines can be modelled with one connecting rod force only. Using two connecting rod forces will make no significant change to the SCF.

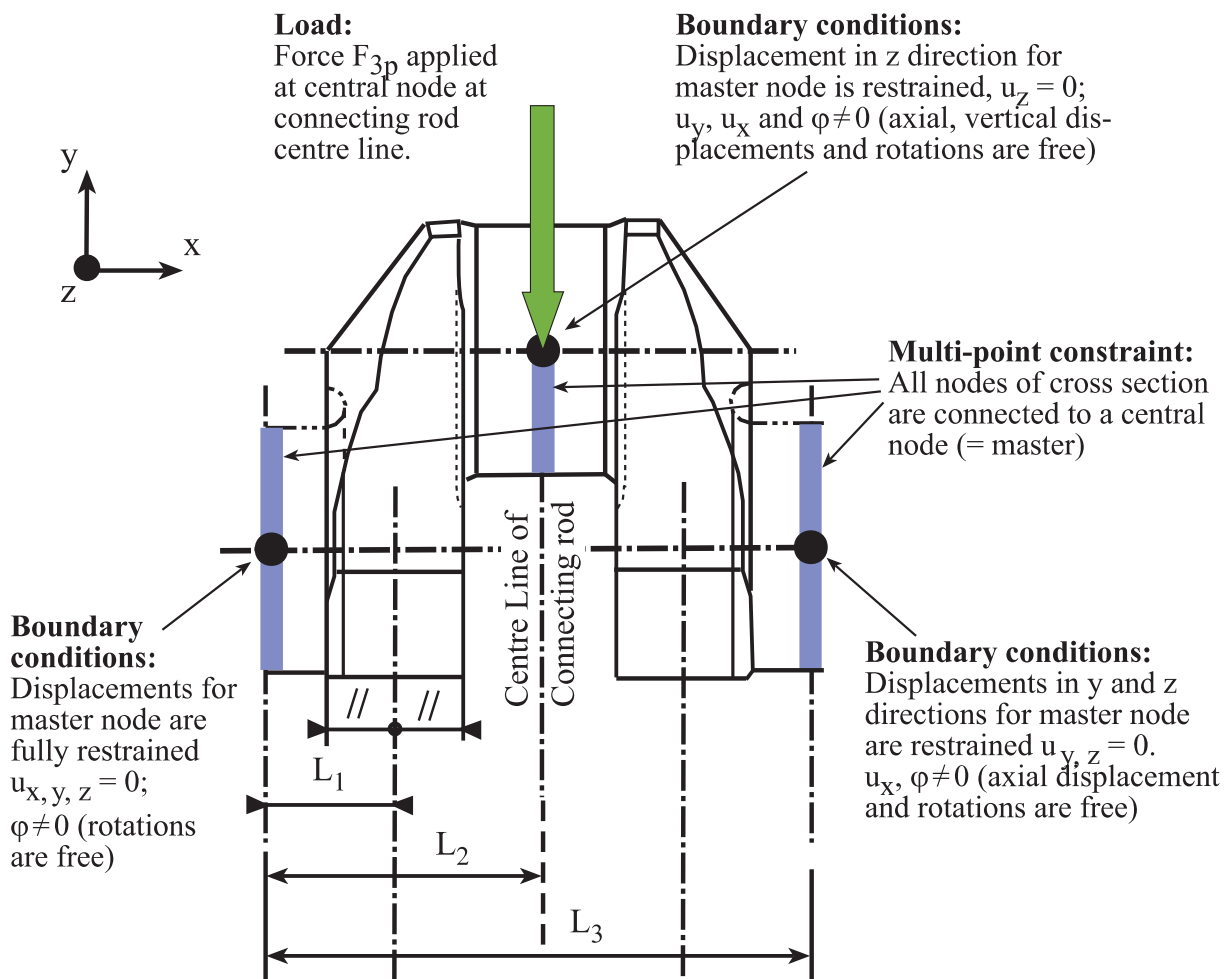


Figure 4 Boundary and load conditions for the 3-point bending load case for an in-line engine

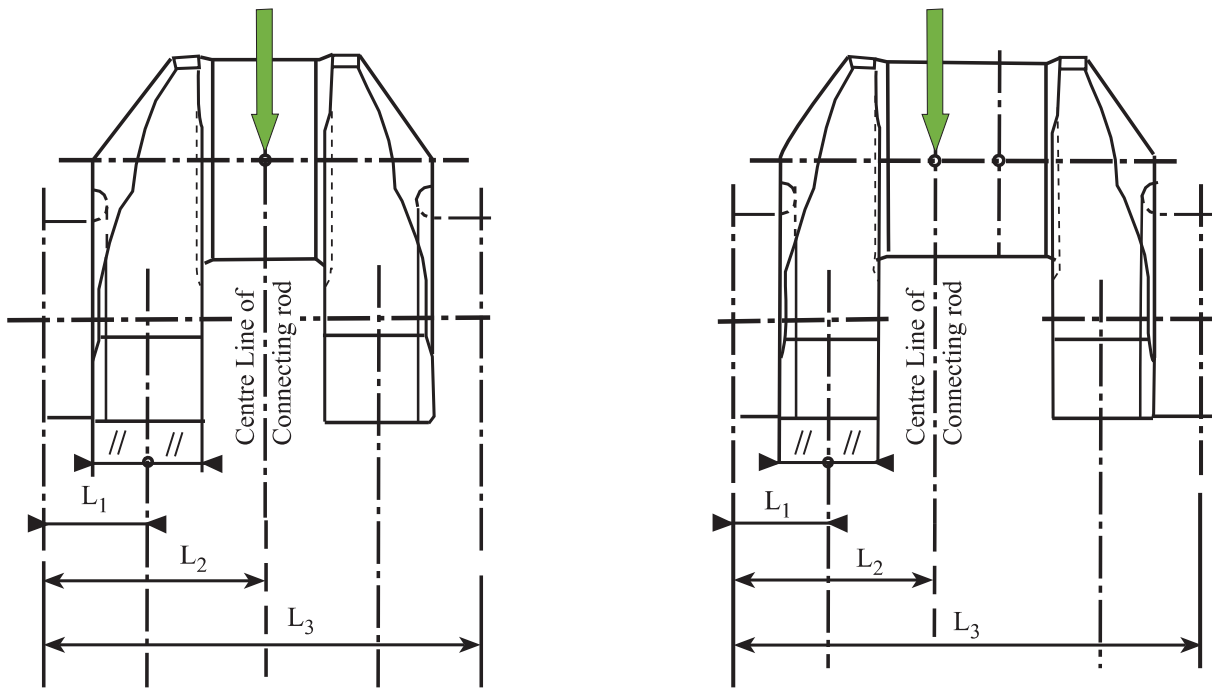


Figure 5 Load applications for in-line and V-type engines

The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The SCF in the journal fillet can be determined in two ways as shown in [3.3.1] and [3.3.2].

3.3.1 Method 1

This method is analogue to the FVV investigation. The results from 3-point and 4-point bending are combined as follows:

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

where:

σ_{3P} = as found by the FE calculation

σ_{N3P} = nominal bending stress in the web centre due to the force F_{3P} [N] applied at the centre-line of the actual connecting rod, see Figure 4

β_B = as determined in [3.2]

σ_{Q3P} = $\sigma_{Q3P} = \frac{Q_{3P}}{B \times W}$ where Q_{3P} is the radial (shear) force in the web due to the force F_{3P} applied at the centre-line of the actual connecting rod, see Sec.1 Figure 3 and Sec.1 Figure 4.

3.3.2 Method 2

This method is not analogous to the FVV investigation. In a statically determined system as one crank throw supported on two bearings, the bending moment and radial (shear) force are proportional. Therefore the journal fillet SCF can be found directly by the 3 point bending FE calculation.

The SCF is then calculated according to:

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

For symbols see [3.3.1].

When using this method the radial force and stress determination in [Sec.1](#). The alternating bending stress in the journal fillet as per [Sec.1 \[2.1.5\]](#) is then evaluated.

$$\sigma_{BG} = \pm \left| \beta_{BQ} \times \sigma_{BFN} \right|$$

Note:

The use of this method does not apply to the crankpin fillet and that this SCF shall not be used in connection with calculation methods other than those assuming a statically determined system as in [Sec.1](#).

---e-n-d---o-f---n-o-t-e---

APPENDIX D GUIDANCE FOR EVALUATION OF FATIGUE TESTS

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 - 1.2. Full-size crank throw testing
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 - 2.4. Calculation of sample mean and standard deviation
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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 document. Other statistical evaluation methods may also be applied.

1.1 Small specimen testing

For crankshafts without any fillet surface treatment, the fatigue strength can be determined by testing small specimens taken from a full-size crank throw. When other areas in the vicinity of the fillets are surface treated introducing residual stresses in the fillets, this approach cannot be applied.

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.

1.2 Full-size crank throw testing

For crankshafts with surface treatment the fatigue strength can only be determined through testing of full size crank throws. For cost reasons, this usually means a low number of crank throws. 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 Evaluation of test results

2.1 Principles

Prior to fatigue testing, the crankshaft shall 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.

The test samples should 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 should be corrected with a confidence interval: a 90% confidence interval may be used both for the sample mean and the standard deviation.

The test results, when applied in [Sec.1](#), shall be evaluated to represent the mean fatigue strength, with or without taking into consideration the 90% confidence interval as mentioned above. The standard deviation should 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.

If the evaluation aims to find a relationship between (static) mechanical properties and the fatigue strength, the relation shall be based on the real (measured) mechanical properties, not on the specified minimum properties.

The calculation technique presented in [\[2.4\]](#) 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.

Normally, small specimen tests shall be done with tensile load. The shear (torsion) fatigue strength can then be found by using the equivalent stress (von Mises) approach (observing the restriction for CGF forging mentioned in [\[3.4\]](#)).

If torsion fatigue tests are used as reference, the method used for determining the bending and tensile fatigue strength shall be specially considered. Unless the relation between the two is documented by special testing, the following relation may be used for steels:

$$\frac{\tau_{tw}}{\sigma_{zdw}} = \left[0.423 \times \left(\frac{\sigma_{0.2}}{\sigma_B} \right)^5 + \frac{1}{\sqrt{3}} \right]$$

If the fatigue strength both in torsion and in bending is tested independently, the two fatigue strengths can be combined by using the Gough and Pollard approach (see [\[4.3\]](#)), or other recognized method for calculation of multiaxial fatigue. Many such methods are described in the literature, and the choice of method (other than Gough and Pollard) for a specific application shall be justified by considering material, stress state, stress condition, geometry, results from full scale test or full scale stress measurements etc. (see also [\[3.4\]](#)).

2.2 Staircase method

In the original staircase method, the first specimen is subjected to a stress corresponding to the expected average fatigue strength. If the specimen survives 10^7 cycles, it is discarded and the next specimen is subjected to a stress that is one increment above the previous, i.e. a survivor is always followed by the next using a stress one increment above the previous. The increment should be selected to correspond to the expected level of the standard deviation.

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. With this approach the sum of failures and run-outs is equal to the number of specimens.

This original staircase method is only suitable when a high number of specimens are available. Through simulations it has been found that the use of about 25 specimens in a staircase test leads to a sufficient accuracy in the result.

2.3 Modified staircase method

When a limited number of specimens are available, it is advisable to apply the modified staircase method. Here the first specimen is subjected to a stress level that is most likely well below the average fatigue strength. When this specimen has survived 10^7 cycles, this same specimen is subjected to a stress level one increment above the previous. The increment should be selected to correspond to the expected level of the standard deviation. This is continued with the same specimen until failure. 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.

With this approach the number of failures usually equals the number of specimens. The number of run-outs, counted as the highest level where 10^7 cycles were reached, also equals the number of specimens.

The acquired result of a modified staircase method should 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). If the confidence calculation is desired or necessary, the minimum number of test specimens is 3.

2.4 Calculation of sample mean and standard deviation

A hypothetical example of tests for 5 crank throws is presented further in the subsequent text. 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: $n = 10$.

Furthermore the method distinguishes between:

- Less frequent event is failure: $C = 1$
- 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, and vice versa.

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.

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. The sample mean can be calculated as follows:

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

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

The standard deviation can be found by:

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

Where:

- S_{a0} = the lowest stress level for the less frequent occurrence
- d = the stress increment
- F = $\sum f_i$
- A = $\sum_i \cdot f_i$
- B = $\sum_i^2 \cdot f_i$
- i = 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:

$$\frac{B \times F - A^2}{F^2} > 0.3 \quad \text{and} \quad 0.5 \times s < d < 1.5 \times s$$

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

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.

Guidance note:

Example 1: hypothetical test results may look as shown in Figure 1. The processing of the results and the evaluation of the sample mean and the standard deviation are shown in Figure 2.

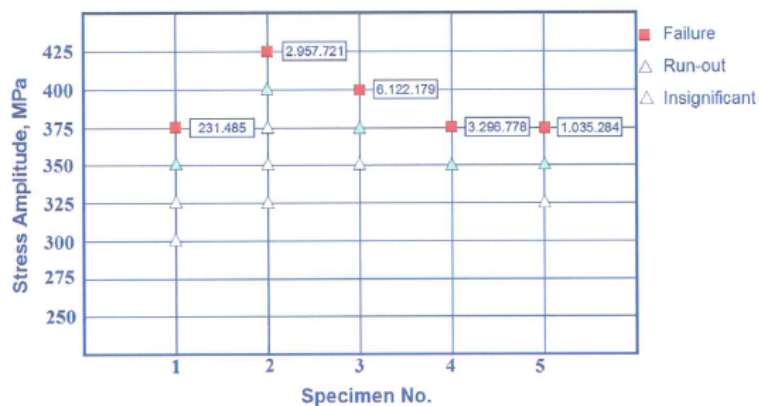


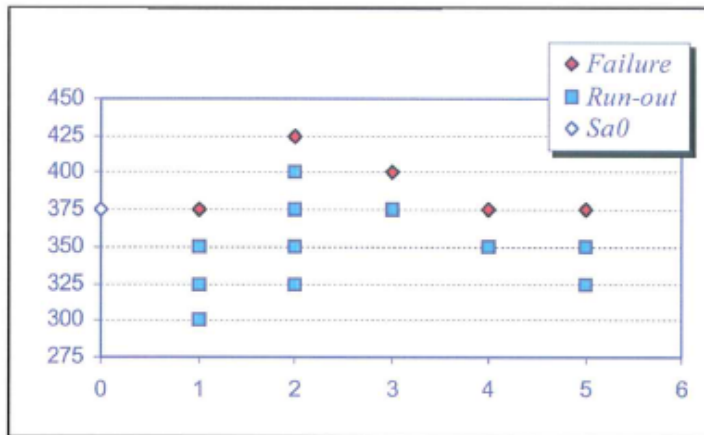
Figure 1 Log sheet of a modified staircase test

Stress level 0,

$$S_{a0} := 375 \cdot \text{MPa}$$

Stress increment,

$$d := 25 \cdot \text{MPa}$$



i	f _i	i·f _i	i ² ·f _i
2	1	2	4
1	1	1	1
0	3	0	0
Σ	5	3	5
	F	A	B

$$F = \sum f_i$$

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

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

$i = 0, 1, 2, \dots$ is the stress level numbering, the numbering usually starts from zero
 f_i is number of test specimen at stress level, i

$$F = 5$$

$$A = 3$$

Level 0 is the lowest value of the less frequent occurrence in the test results.

$$B = 5$$

Sample mean

Sample mean,

$$S_a = \begin{cases} S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2} \right) & \text{if } C = 1 \\ S_{a0} + d \cdot \left(\frac{A}{F} + \frac{1}{2} \right) & \text{if } C = 2 \end{cases}$$

$$S_a = 377.5 \text{ MPa}$$

Standard deviation

Sample standard deviation,

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

$$s = 27.09 \text{ MPa}$$

Standard deviation ratio,

$$s_r = \frac{s}{S_a} \quad s_r = 0.072$$

Figure 2 Processing of the staircase test results

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.5 Confidence interval for mean fatigue limit

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.

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

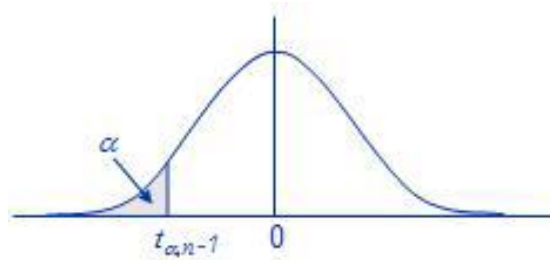


Figure 3 Student's t-distribution

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 figure shows the t -value for $(1 - \alpha) \times 100\%$ confidence interval for the sample mean.

---e-n-d---o-f---n-o-t-e---

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 $(1 - \alpha) \times 100\%$ confidence interval for the mean is:

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

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

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

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.

2.6 Confidence interval for standard deviation

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.

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. The figure shows the chi-square for $(1 - \alpha) \times 100\%$ confidence interval for the variance.

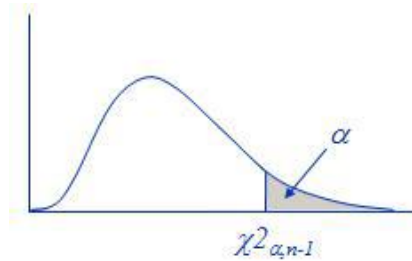


Figure 4 Chi-square distribution

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. The figure shows the chi-square for $(1 - \alpha) \times 100\%$ confidence interval for the variance.

---e-n-d---o-f---n-o-t-e---

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

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

A $(1 - \alpha) \times 100\%$ 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 found by

$$s_{X\%} = \sqrt{\frac{n-1}{\chi_{\alpha, n-1}^2}} \times s$$

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

3 Small specimen testing

In this connection a small specimen is considered to be one of the specimens taken from a crank throw. Since the specimens shall be representative for the fillet fatigue strength, they should be taken out close to the fillets, as shown in Figure 5.

It should 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 utilising the finite element method.

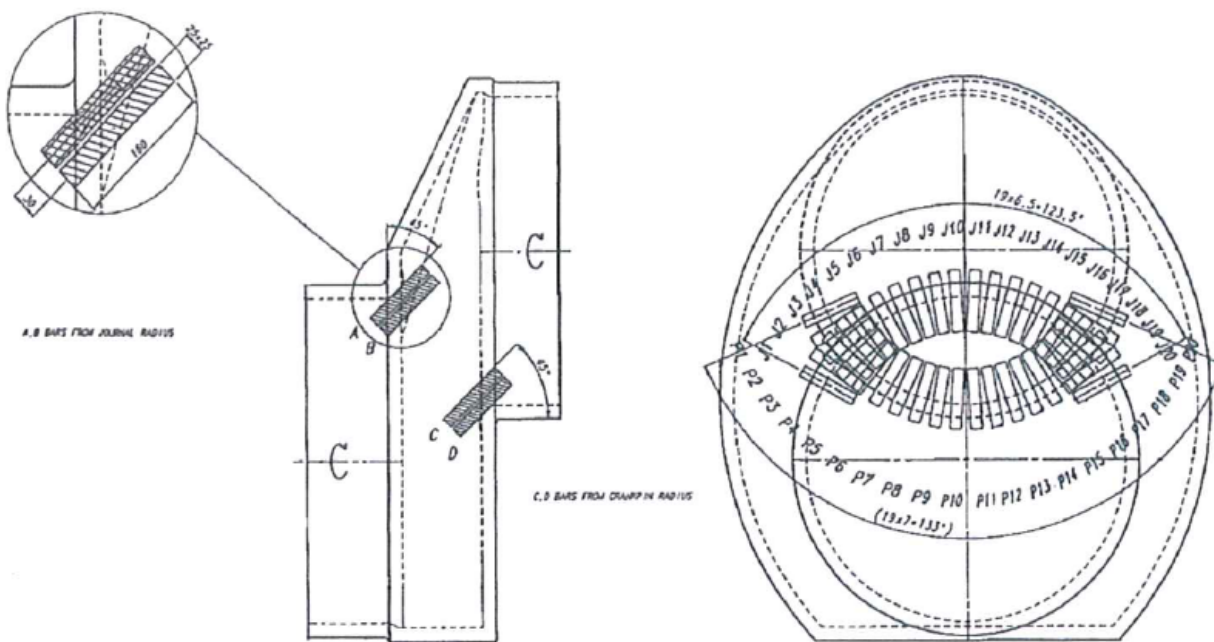


Figure 5 Specimen locations in a crank throw

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

3.1 Determination of bending fatigue strength

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

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 shall be taken in 45 degrees angle as shown in Figure 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 Figure 5.
- If the objective of the testing is to document the influence of continuous grain flow (cgf) forging, the specimens should be restricted to the vicinity of the crank plane.

3.2 Determination of torsional fatigue strength

- a) If the specimens are subjected to torsional testing, the selection of samples should follow the same guidelines as for bending above. The stress gradient influence shall be considered in the evaluation.
- b) If the specimens are tested in push-pull, the samples should be taken out at an angle of 45 degrees to the crank plane. When taking the specimen at a distance from the (crank) middle plane of the crankshaft along the fillet, this plane rotates around the pin center point making it possible to resample the fracture direction due to torsion (the results shall be converted into the pertinent torsional values).

3.3 Other test positions

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.

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

3.4 Correlation of test results

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

Here under, the test position shall be considered and it shall be noted that the test position of specimen in [3.1] is considered most representative.

When using the bending fatigue properties from tests mentioned in this section, it should be kept in mind that successful continuous grain flow (cgf) forging leading to elevated values compared to other (non cgf) forging (or empirical approaches), 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, e.g. by using no credit for cgf. 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.

If the found ratio between bending and torsion fatigue differs from $\sqrt{3}$, one should consider replacing the use of the von Mises criterion with the gough pollard criterion. Also, if critical plane criteria are used, it shall 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.

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

4 Full size testing

4.1 Hydraulic pulsation

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.

Although the applied load should 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. Furthermore, it is important that the test rig provides boundary conditions as defined in App.C [3.1] to App.C [3.3]. The (static) mechanical properties shall be determined as stipulated by the quality control procedures.

4.2 Resonance tester

A rig for bending fatigue normally works with an R-ratio of -1. Due to operation close to resonance, the energy consumption is moderate. Moreover, the frequency is usually relatively high, meaning that 10^7 cycles can be reached within some days. Figure 6 shows a layout of the testing arrangement.

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

Clamping around the journals shall 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.

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 pre-load.

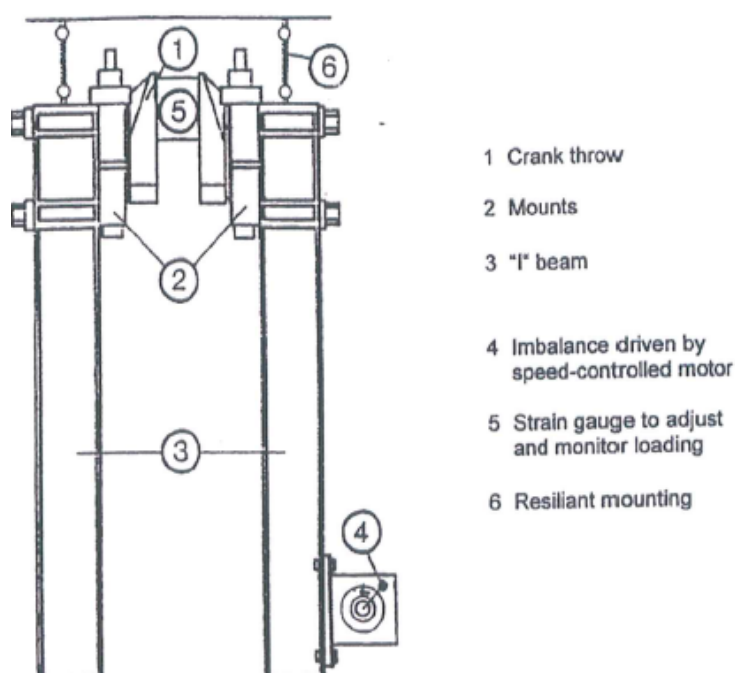


Figure 6 A testing arrangement of the resonance tester for bending loading

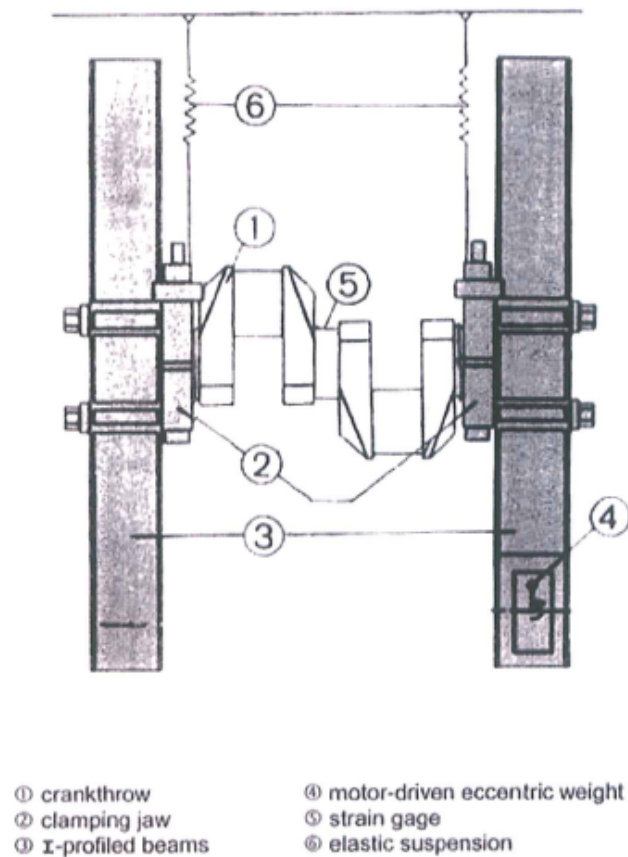


Figure 7 A testing arrangement of the resonance tester for torsion loading

A rig for torsion fatigue can also be arranged as shown in Figure 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 **sideways** 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.

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

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

4.3 Use of results and crankshaft acceptability

In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability, see Sec.1 [7.2], the Gough-Pollard approach can be applied for crankpin and journal fillets as follows:

Crankpin fillet :

$$\left(\frac{\sigma_H}{\sigma_{fHT}} \right)^2 + \left(\frac{\tau_H}{\tau_{fHT}} \right)^2 \leq \frac{1}{s^2}$$

where:

σ_{fHT} = fatigue strength by bending testing

τ_{fHT} = fatigue strength by torsion testing.

Journal fillet (not applicable to semi built crankshafts):

$$\left(\frac{\sigma_G}{\sigma_{fGT}}\right)^2 + \left(\frac{\tau_G}{\tau_{fGT}}\right)^2 \leq \frac{1}{S^2}$$

where:

σ_{fGT} = fatigue strength by bending testing, may be set equal to σ_{fHT}

τ_{fGT} = fatigue strength by torsion testing.

Oil bores can be calculated based on the largest alternating torsional stress as follows:

Crankpin oil bore:

$$\frac{1}{3} \times \sigma_{BO} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \times \left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2} \right] \leq \frac{\sigma_{fOT}}{S}$$

where:

σ_{fOT} = alternating largest principal stress from torsional testing.

Journal oil bore:

$$\frac{1}{3} \times \sigma_{BOG} \times \left[1 + 2 \times \sqrt{1 + \frac{9}{4} \times \left(\frac{\sigma_{TOG}}{\sigma_{BOG}}\right)^2} \right] \leq \frac{\sigma_{fOGT}}{S}$$

where:

σ_{fOGT} = alternating largest principal stress from torsional testing.

5 Use of existing results for similar crankshafts

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

Material:

- the same material type
- cleanliness on the same or better level
- the same mechanical properties can be granted (size versus hardenability).

Geometry:

- difference in the size effect of stress gradient is insignificant or it is considered
- principal stress direction is equivalent. See [Sec.2](#).

Manufacturing:

- similar manufacturing process.

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 the same material type and surface hardness are used and the fillet radius and hardening depth are within approximately $\pm 30\%$ of the tested crankshaft.

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.

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 special heat treatment to remove hydrogen content is not carried out.

APPENDIX E GUIDANCE FOR CALCULATION OF SURFACE TREATED FILLETS AND OIL BORE OUTLETS

Contents

1. Introduction
2. Definition of surface treatment
 - 2.1. Surface treatment methods
3. Calculation principles
 - 3.1. Evaluation of local fillet stresses
 - 3.2. Evaluation of oil bore stresses
 - 3.3. Acceptability criteria
4. Induction hardening
 - 4.1. Local fatigue strength
5. Nitriding
 - 5.1. Local fatigue strength
6. Cold forming
 - 6.1. Stroke peening by means of a ball
 - 6.1.1. Use of existing results for similar crankshafts
 - 6.2. Cold rolling
 - 6.2.1. Use of existing results for similar crankshafts

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. Conservative empiricism has been applied intentionally, in order to be on the safe side from a calculation standpoint.

Note:

Measurements or more specific knowledge should be used if available. However, in the case of a wide scatter (e.g. for residual stresses) the values should be chosen from the end of the range that would be on the safe side for calculation purposes.

---e-n-d---o-f---n-o-t-e---

2 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.

2.1 Surface treatment methods

The following list 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

<i>Treatment method</i>	<i>Affecting</i>
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 coning	Residual stresses

Note:

Since only induction hardening, nitriding, cold rolling and stroke peening are considered relevant for marine engines, other methods are not dealt with in this document. In addition die quenching can be considered in the same way as induction hardening.

---e-n-d---o-f---n-o-t-e---

3 Calculation principles

The basic principle is that the alternating working stresses shall be below the local fatigue strength (including the effect of surface treatment) wherein non-propagating cracks may occur, see also [6.1] for details. This is then divided by a certain safety factor. 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.

Consideration of the local fatigue strength shall include the influence of the local hardness, residual stress and mean working stress. The influence of the giga-cycle effect, especially for initiation of subsurface cracks, should be covered by the choice of safety margin.

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

Alternating and mean working stresses shall 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. The following figure indicates this principle in the case of induction hardening. The base axis is either the depth (perpendicular to the surface) or along the fillet contour.

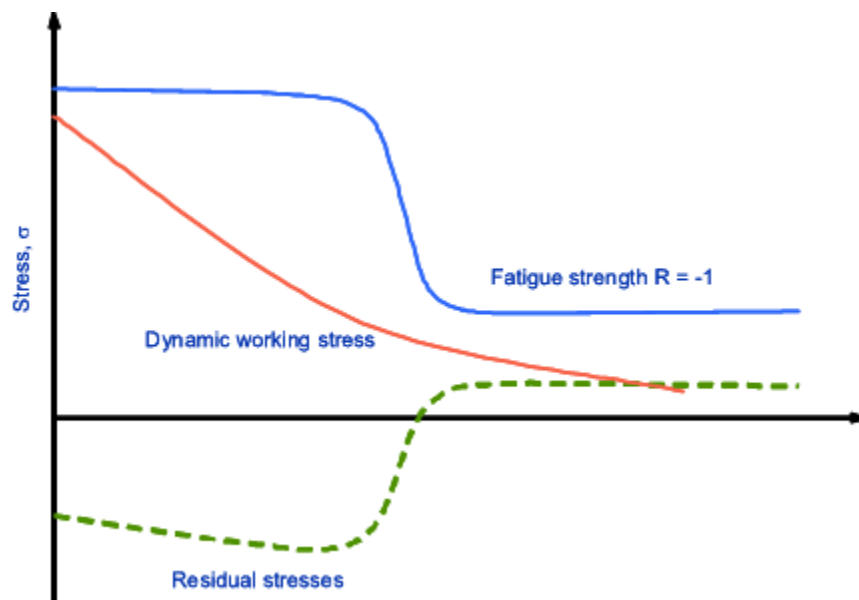


Figure 1 General principles of stresses as functions of depth

The acceptability criterion shall 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.

3.1 Evaluation of local fillet stresses

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 [App.C](#). 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.

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 IACS UR M53.3 if within its validity range, and a relative stress gradient inversely proportional to the fillet radius. Bending and torsional stresses shall be addressed separately. The combination of these is addressed by the acceptability criterion.

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. These functions $\alpha_{B-local}$ and $\alpha_{T-local}$ have different shapes due to the different stress gradients.

The SCFs α_β and α_T are valid at the surface, the local $\alpha_{\beta-local}$ and $\alpha_{T-local}$ drop with increasing depth. The relative stress gradients at the surface depend on the kind of stress raiser, but for crankpin fillets they can be simplified to $\frac{2}{R_H}$ in bending and $\frac{1}{R_H}$ in torsion. The journal fillets are handled analogously by using R_G . The nominal stresses are assumed to be linear from the surface to a midpoint in the web between the crankpin fillet and the journal fillet for bending and to the crankpin or journal center for torsion.

The local SCFs are then functions of depth t according to [Equation E-1](#) as shown in [Figure 2](#) for bending and respectively for torsion in [Equation E-2](#) and [Figure 3](#).

$$\alpha_{B-local} = \left(\alpha_B - 1 \right) e^{\frac{-2 \cdot t}{R_H}} + 1 - \left(\frac{2 \cdot t}{\sqrt{W^2 + S^2}} \right)^{\frac{0.6}{\alpha_\beta}}$$

Equation E-1

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

Equation E-2

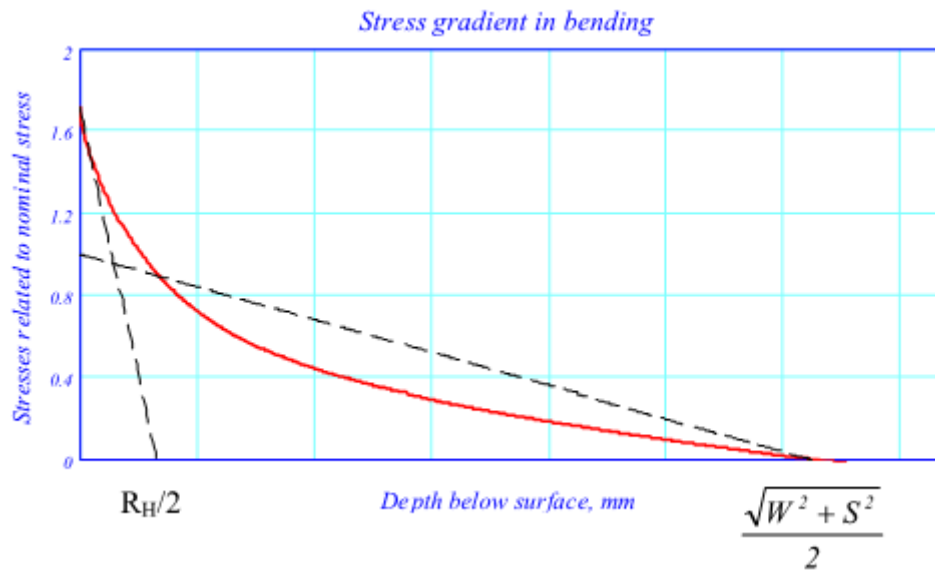


Figure 2 Bending SCF in the crankpin fillet as a function of depth

In [Figure 2](#) the corresponding SCF for the journal fillet can be found by replacing R_H with R_G .

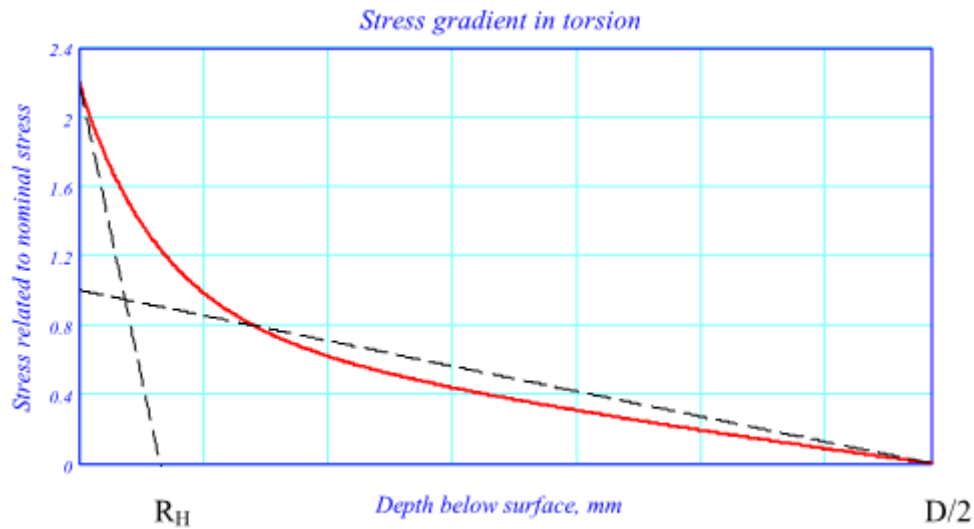


Figure 3 Torsional SCF in the crankpin fillet as a function of depth

In [Figure 3](#) the corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G . 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 should be used to determine the actual stresses in the transition zone.

3.2 Evaluation of oil bore stresses

Stresses in the oil bores can be determined also by FEA. The element size should be less than $1/8$ of the oil bore diameter D_O and the element mesh quality criteria should be followed as prescribed in [App.C](#). The fine element mesh should continue well beyond a radial depth corresponding to the hardening depth.

The loads to be applied in the FEA are the torque, [App.C \[3.1\]](#), and the bending moment, with four-point bending as in [App.C \[3.2\]](#).

If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from [Sec.1 \[3\] of this guideline](#) if within its applicability range. Bending and torsional stresses at the point of peak stresses are combined as in [Sec.1 \[5\]](#).

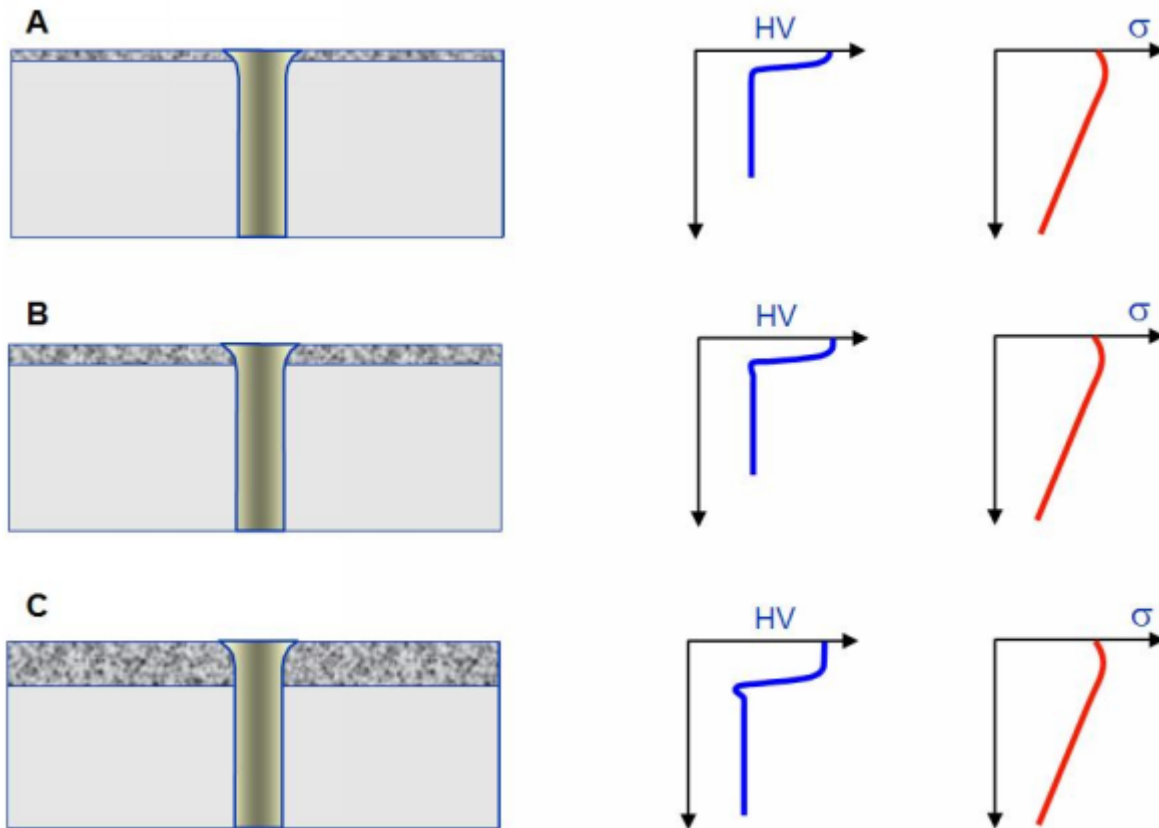


Figure 4 Stresses and hardness in induction hardened oil holes

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.

The peak stress in the bore occurs at the end of the edge rounding. Within this zone the stress drops almost linearly to the center of the pin. As can be seen from Figure 4, for shallow (A) and intermediate (B) hardening, the transition point practically coincides with the point of maximal stresses. For deep hardening the transition point comes outside of the point of peak stress and the local stress can be assessed as a portion $(1-2 \cdot t_H/D)$ of the peak stresses where t_H is the hardening depth.

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. These functions $\gamma_{B-local}$ and $\gamma_{T-local}$ have different shapes, because of the different stress gradients.

The stress concentration factors γ_B and γ_T are valid at the surface, the local SCFs $\gamma_{B-local}$ and $\gamma_{T-local}$ drop with increasing depth. The relative stress gradients at the surface depend on the kind of stress raiser, but for crankpin oil bores they can be simplified to $4/D_O$ in bending and $2/D_O$ in torsion. The local SCFs are then functions of the depth t :

$$\gamma_{B-local} = (\gamma_B - 1)e^{\frac{-4 \cdot t}{D_0}} + 1$$

Equation E-3

$$\gamma_{B-local} = (\gamma_B - 1)e^{\frac{-2 \cdot t}{D_0}} + 1$$

Equation E-4

3.3 Acceptability criteria

Acceptance of crankshafts is based on fatigue considerations, [Sec.1 of this guideline](#) compares the equivalent alternating stress and the fatigue strength ratio to an acceptability factor of $Q \geq 1.15$ for oil bore outlets, crankpin fillets and journal fillets. This shall be extended to cover also surface treated areas independent of whether surface or transition zone is examined.

4 Induction hardening

Generally the hardness specification shall 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 **HV0.5...HV5**.

The induction hardening depth is defined as the depth where the hardness is 80% of the minimum specified surface hardness.

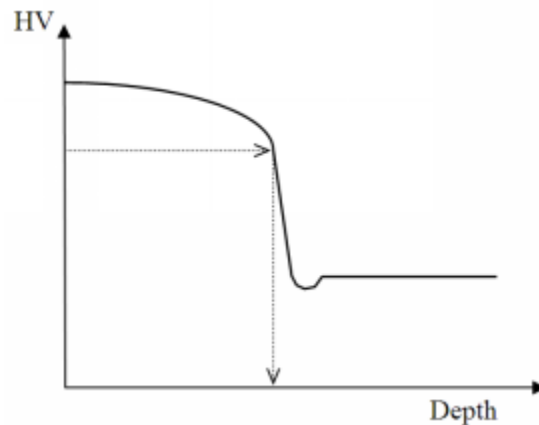


Figure 5 Typical hardness as a function of depth

In [Figure 5](#) 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.

In the case of crankpin or journal hardening only, the minimum distance to the fillet shall be specified due to the tensile stress at the heat-affected zone as shown in [Figure 6](#).

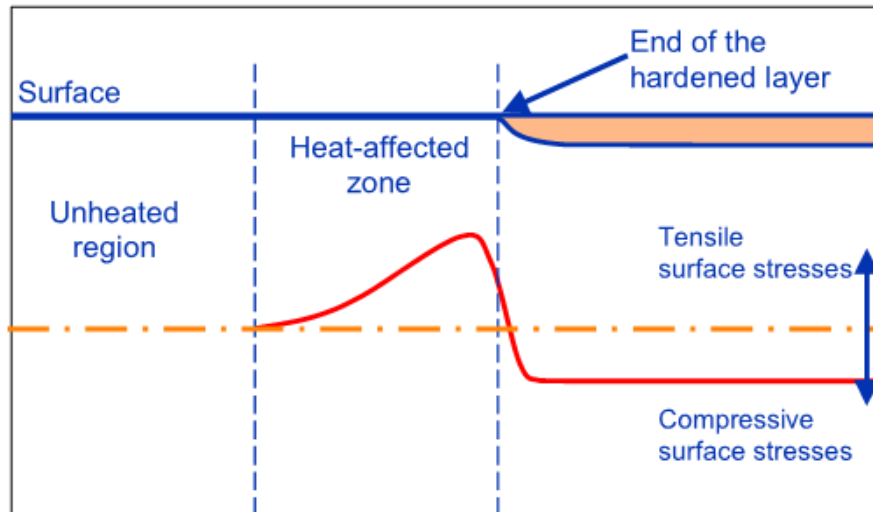


Figure 6 Residual stresses along the surface of a pin and fillet

If the hardness-versus-depth profile and residual stresses are not known or specified, one may assume the following:

- the hardness profile consists of two layers [Figure 5]:
 - constant hardness from the surface to the transition zone
 - constant hardness from the transition zone to the core material
- residual stresses in the hard zone of 200 MPa (compression)
- transition-zone hardness as 90% of the core hardness unless the local hardness drop is avoided
- transition-zone maximum residual stresses (von Mises) of 300 MPa tension.

If the crankpin or journal hardening ends close to the fillet, the influence of tensile residual stresses shall 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.

4.1 Local fatigue strength

Induction-hardened crankshafts will suffer fatigue either at the surface or at the transition to the core. The fatigue strengths, for both the surface and the transition zone, can be determined by fatigue testing of full size cranks as described in App.D. 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. Tests made with the core material only will not be representative since the tensile residual stresses at the transition are lacking. Alternatively, the surface fatigue strength (principal stress) can be determined empirically as follows where HV is the surface (Vickers) hardness. Equation E-5 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 $R = -1$:

$$\sigma_{F_{surface}} = 400 + 0.5 \times (HV - 400) \text{ [MPa]} \quad \text{Equation E-5}$$

It shall be noted also that the mean stress influence of induction-hardened steels may be significantly higher than that for QT steels.

The fatigue strength in the transition zone, without taking into account any possible local hardness drop, shall be determined by the equation introduced in [Sec.1 \[6\]](#) . For journal and respectively to crankpin fillet applies:

$$\sigma_{Ftransition, cpin} = \pm K \times \left(0.42 \times \sigma_B + 39.3 \right) \times \left[0.264 + 1.073 \times Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \times \sqrt{\frac{1}{X}} \right] \quad \text{Equation E-6}$$

where:

- Y = D_G and X = R_G for journal fillet
Y = D and X = R_H for crankpin fillet
Y = D and X = $\frac{D_O}{2}$ for oil bore outlet.

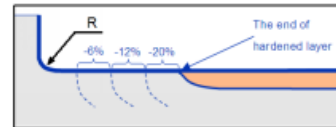
The influence of the residual stress is not included in equation E-6.

For the purpose of considering subsurface fatigue, below the hard layer, the disadvantage of tensile residual stresses shall 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. When the residual stresses are known to be lower, also smaller value of subtraction shall be used. For low-strength steels the percentage chosen should be higher.

For the purpose of considering surface fatigue near the end of the hardened zone, i.e. in the heat-affected zone shown in the [Figure 6](#), 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

- | | |
|----------------------------------------------|-----|
| 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 Nitriding

The hardness specification shall include the surface hardness range (min and max) and the minimum and maximum depth. Only gas nitriding is considered. The depth of the hardening is defined in different ways in the various standards and the literature. The most practical method to use in this context shall define the nitriding depth t_N as the depth to a hardness of 50 HV above the core hardness. The hardening profile should be specified all the way to the core. If this is not known, it may be determined empirically via the following formula:

$$HV \left(t \right) = HV_{core} + \left(HV_{surface} - HV_{core} \right) \times \left(\frac{50}{HV_{surface} - HV_{core}} \right)^{\left(\frac{t}{t_N} \right)^2} \quad \text{Equation E-7}$$

where:

- t = the local depth
 $HV(t)$ = hardness at depth t

HV_{core}	= core hardness
HV_{surface}	= surface hardness
T_N	= nitriding depth as defined above.

5.1 Local fatigue strength

It is important to note that in nitrided crankshaft cases, fatigue is found either at the surface or at the transition to the core. This means that the fatigue strength can be determined by tests as described in [App.D](#).

Alternatively, the surface fatigue strength (principal stress) can be determined empirically and conservatively as follows. This is valid for a surface hardness of 600HV or greater:

$$\sigma_{\text{Fsurface}} = 450 \text{ MPa}$$

Note:

This fatigue strength is assumed to include the influence of the surface residual stress and applies for a working stress ratio of $R = -1$.

---e-n-d---o-f---n-o-t-e---

The fatigue strength in the transition zone can be determined by the equation introduced in IACS UR M53.6. For crankpin and respectively to journal applies:

$$\sigma_{F\text{transition}, \text{cpin}} = \pm K \times \left(0.42 \times \sigma_B + 39.3 \right) \times \left[0.264 + 1.073 \times Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \times \sqrt{\frac{1}{X}} \right] \quad \text{Equation E-8}$$

where:

$Y = D_G$ and $X = R_G$	for journal fillet
$Y = D$ and $X = R_H$	for crankpin fillet
$Y = D$ and $X = \frac{D_o}{2}$	for oil bore outlet.

Note:

This fatigue strength in equation E-8 is **not** assumed to include the influence of the residual stresses.

---e-n-d---o-f---n-o-t-e---

In contrast to induction-hardening the nitrided 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. 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.

Nevertheless, in principle the calculation should be carried out along the entire hardness profile and it can be limited to a simplified approach of examining the surface and an artificial transition point. 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 should be used. This means that the stresses at the transition to the core can be found by using the local SCF formulae mentioned earlier when inserting $t = 1.2t_N$.

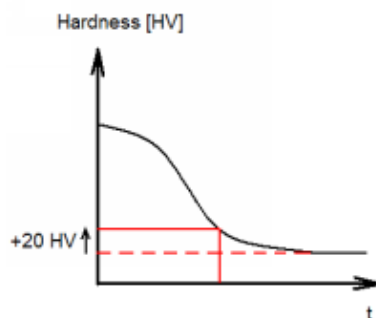


Figure 7 Sketch of the location for the artificial transition point in the depth direction

6 Cold forming

The advantage of stroke peening or cold rolling of fillets is the compressive residual stresses introduced in the high-loaded area. Even though surface residual stresses can be determined by X-ray diffraction technique and subsurface residual stresses can be determined through neutron diffraction, the local fatigue strength is virtually non-assessable on that basis since suitable and reliable correlation formulae are hardly known.

Therefore the fatigue strength shall be determined by fatigue testing, see also [App.D](#). Such testing is normally carried out as four-point bending, with a working stress ratio of $R = -1$. 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.

In comparison to bending, the torsion fatigue strength in the fillet may differ considerably from the ratio (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.

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 same type and that the forming is done so as to obtain the same 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 shall be proportional to the fillet radius.

6.1 Stroke peening by means of a ball

The fatigue strength obtained can be documented by means of full size crank tests or by empirical methods if applied on the safe side. If both bending and torsion fatigue strengths have been investigated and differ from the ratio $\sqrt{3}$, the von Mises criterion should be excluded.

If only bending fatigue strength has been investigated, the torsional fatigue strength should 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 should not be assumed to be more than $\frac{2}{3}$ of $x\%$ above that of the non-peened material.

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. This is illustrated in [Figure 8](#) as gradient load 2. Straight lines 1, 2 and 3 represent different possible load stress gradients.

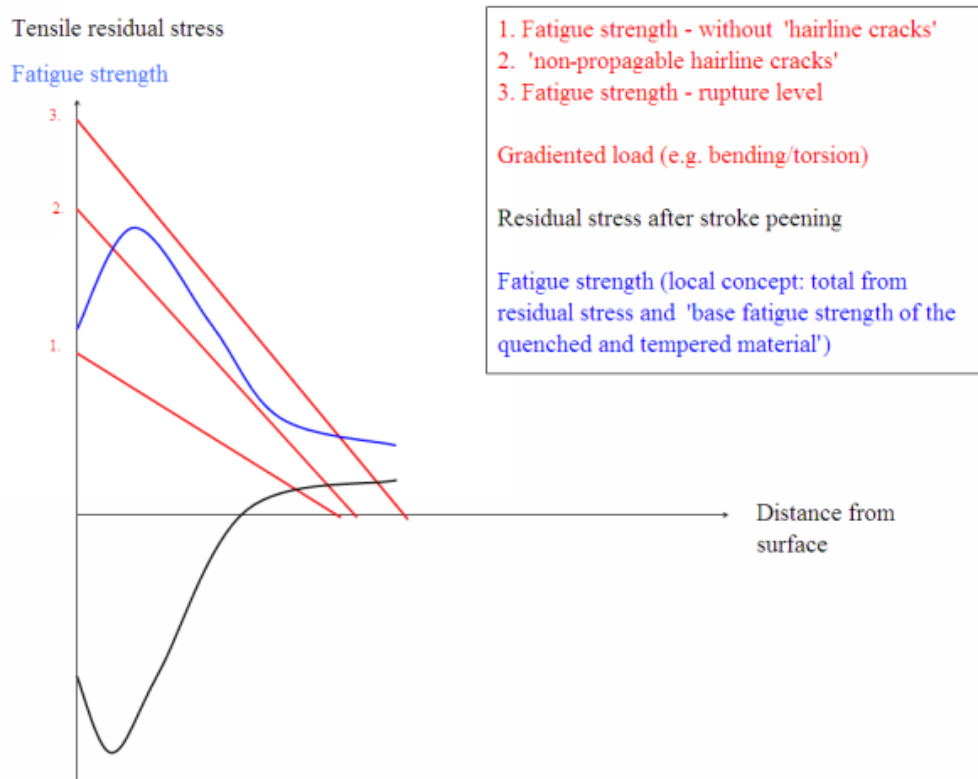


Figure 8 Working and residual stresses below the stroke-peened surface

In fatigue testing with full-size crankshafts these small hairline cracks should 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, should be considered for determination of the failure load level. This also applies if induction-hardened fillets are stroke-peened.

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. The effect on the fatigue strength of induction hardening and stroke peening the fillets shall be determined by a full size crankshaft test. See [App.D](#).

6.1.1 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:

- ball size relative to fillet radius within $\pm 10\%$ in comparison to the tested crankshaft
- at least the same circumferential extension of the stroke peening
- 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
- similar base material, e.g. alloyed quenched and tempered
- forward feed of ball of the same proportion of the radius
- force applied to ball proportional to base material hardness (if different)
- force applied to ball proportional to square of ball radius.

6.2 Cold rolling

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. If both bending and torsion fatigue strengths have been investigated, and differ from the ratio $\sqrt{3}$, the von Mises criterion should be excluded.

If only bending fatigue strength has been investigated, the torsional fatigue strength should 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 should not be assumed to be more than 2/3 of x% above that of the non-rolled material.

6.2.1 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:

- at least the same circumferential extension of cold rolling
- 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
- similar base material, e.g. alloyed quenched and tempered
- roller force to be calculated so as to achieve at least the same relative (to fillet radius) depth of treatment.

APPENDIX F STRESS CONCENTRATION FACTORS IN THE OILBORE OUTLETS OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD

Contents

1. General
2. Model requirements
 - 2.1. Element mesh recommendations
 - 2.2. Material
 - 2.3. Element mesh quality criteria
 - 2.3.1. Principal stresses criterion
 - 2.3.2. Averaged/ unaveraged stresses criterion
3. Load cases and assessment of stress
 - 3.1. Torsion
 - 3.2. Bending

1 General

The objective of the analysis is to substitute the analytically calculated stress concentration factor (SCF) at the oilbore outlet by suitable finite element method (FEM) calculated figures. The analytical method is based on empirical formulae developed from strain gauge or photo-elasticity measurements of various round bars. Use of these formulae beyond any of the various validity ranges can lead to erroneous results in either direction, i.e. results that more inaccurate than indicated by the mentioned standard deviations. Therefore, the FEM-based method is highly recommended.

The SCF calculated according to the rules set forth in this document is defined as the ratio of FEM-calculated stresses to nominal stresses calculated analytically. In use in connection with the present method in [Sec.1](#), principal stresses shall be calculated. The analysis shall be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude shall be applied for all load cases.

It is advised to check element accuracy of the FE solver in use, e.g. by modelling a simple geometry and comparing the stresses obtained by FEM with the analytical solution. Boundary element method (BEM) may be used instead of FEM.

2 Model requirements

The basic recommendations and perceptions for building the FE-model are presented in [\[2.1\]](#). It is obligatory for the final FE-model to fulfil a requirement in [\[2.3\]](#).

2.1 Element mesh recommendations

For the mesh quality criteria to be met, construction of the FE model for the evaluation of stress concentration factors according to the following recommendations is advised:

- The model consists of one complete crank, from the main bearing centerline to the opposite side main bearing centerline.
- Element types used in the vicinity of the outlets:
 - 10 node tetrahedral elements
 - 8 node hexahedral elements

- 20 node hexahedral elements.
- The following mesh properties for the oil bore outlet are used:
 - 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)
 - Recommended manner for element size in the fillet depth direction
 - First layer's thickness equal to element size of **a**
 - Second layer's thickness equal to element size of **$2a$**
 - Third -layer thickness equal to element size of **$3a$**
 - In general, the rest of the crank should be suitable for numeric stability of the solver
 - Drillings and holes for weight reduction have to be modelled Submodeling may be used as long as the software requirements are fulfilled.

Sub-modeling may be used as far as the software requirements are fulfilled.

2.2 Material

This guideline does not consider material properties such as Young's modulus (E) and Poisson's ratio (ν). In FE analysis those material parameters are required, as primarily strain is calculated and stress is derived from strain using the Young's modulus and Poisson's ratio. Reliable values for material parameters shall be used, either as quoted in literature or measured on representative material samples. For steel the following is advised:

$$E = 2.05 \cdot 10^5 \text{ MPa and } \nu = 0.3.$$

2.3 Element mesh quality criteria

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

2.3.1 Principal stresses criterion

The quality of the mesh should be assured by checking the stress component normal to the surface of the oil bore outlet radius. Ideally, this stress should be zero. With principal stresses σ_1 , σ_2 and σ_3 the following criterion must be met:

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

2.3.2 Averaged/ un-averaged stresses criterion

The averaged/ unaveraged -stresses criterion is based on observation of the discontinuity of stress results over elements at the fillet for the calculation of the SCF:

- unaveraged nodal stress results calculated from each element connected to a node_i should differ less than by 5% from the 100% averaged nodal stress results at this node i at the examined location.

3 Load cases

To substitute the analytically determined SCF in [Sec.1](#), calculation shall be performed for the following load cases.

3.1 Torsion

The structure is loaded in pure torsion. In the model surface warp at the end faces is suppressed. Torque is applied to the central node located at the crankshaft axis. This node acts as the master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

Boundary and load conditions are valid for both in-line and V- type engines.

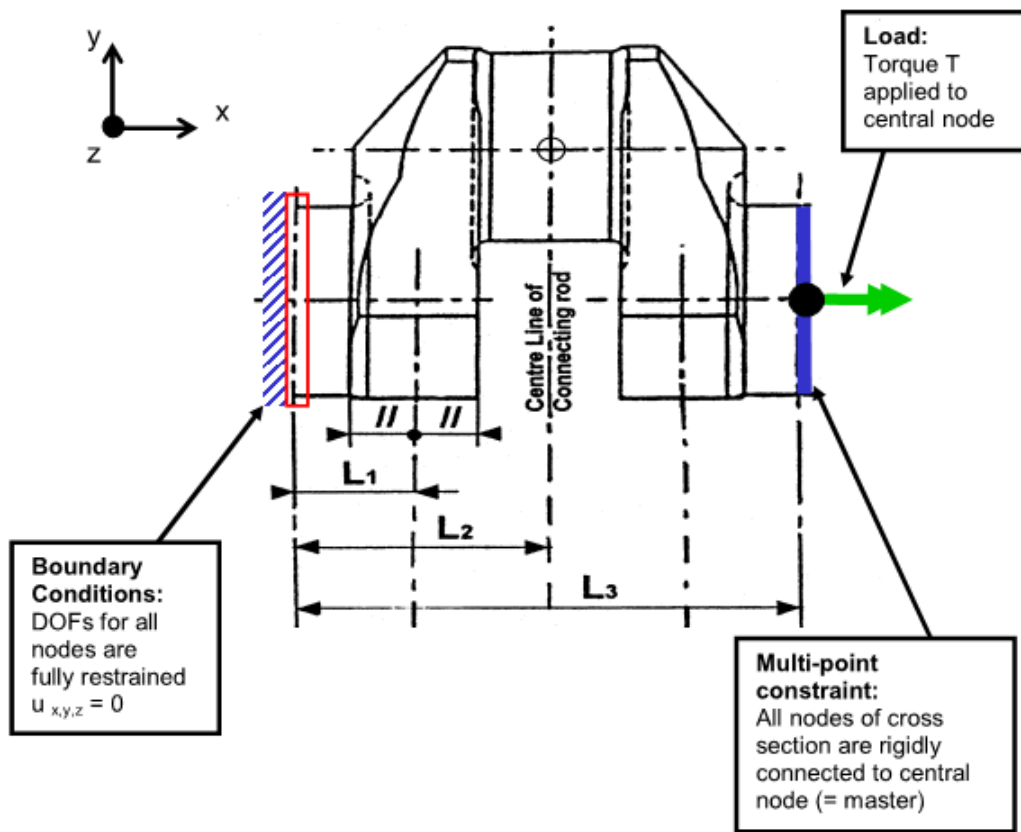


Figure 1 Boundary and load conditions for the torsion load case

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 evaluated per [Sec.1 \[2.2.2\]](#) with torque T :

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

3.2 Bending

The structure is loaded in pure bending. In the model surface warp at the end faces is suppressed. The bending moment is applied to the central node located at the crankshaft axis. This node acts as master node with 6 degrees of freedom and is connected rigidly to all nodes of the end face.

Boundary and load conditions are valid for both in-line and V-type engines.

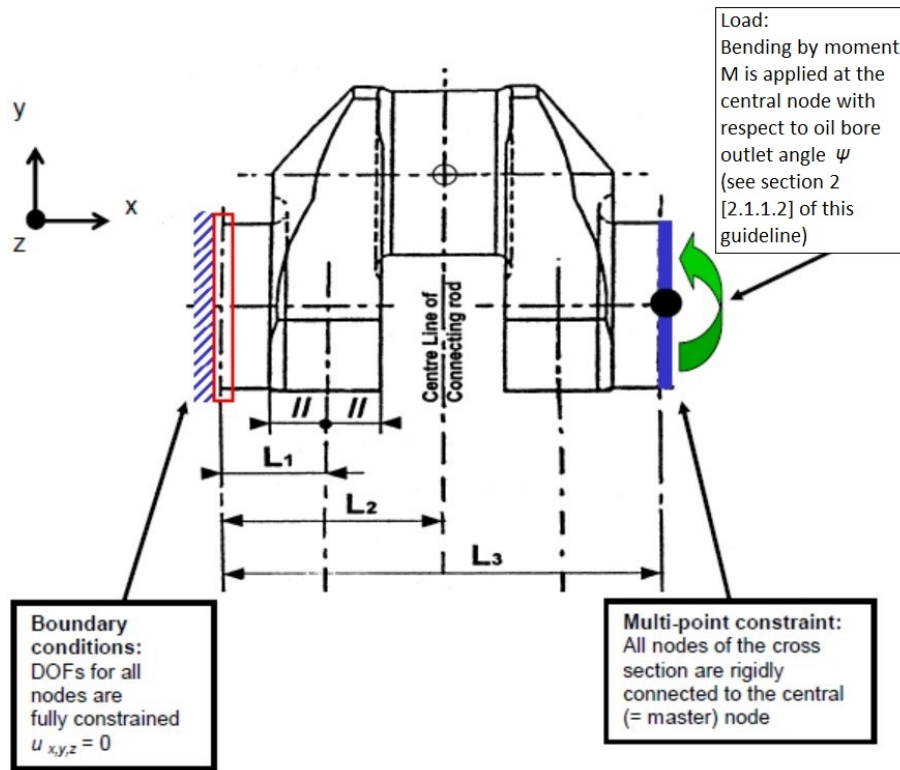


Figure 2 Boundary and load conditions for the pure bending load case

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 per [Sec.1 \[2.1.2.2\]](#) with bending moment M :

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

CHANGES – HISTORIC

November 2015 edition

Amendments July 2017

- App.E Surface tested fillets and oil bore outlets
 - App.E [3.1]: Misprint in Equation E-2, changed from 10,6 to 1.

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