Rules for Classification and Construction

VI  Additional Rules and Guidelines

4  Diesel Engines

2  Calculation of Crankshafts for Internal Combustion Engines
The following Rules come into force on 1 May 2012.

Alterations to the preceding Edition are marked by beams at the text margin.

Germanischer Lloyd SE

Head Office
Brooktorkai 18, 20457 Hamburg, Germany
Phone: +49 40 36149-0
Fax: +49 40 36149-200
headoffice@gl-group.com

www.gl-group.com

"General Terms and Conditions" of the respective latest edition will be applicable
(see Rules for Classification and Construction, I - Ship Technology, Part 0 - Classification and Surveys).

Reproduction by printing or photostatic means is only permissible with the consent of
Germanischer Lloyd SE.

Published by: Germanischer Lloyd SE, Hamburg
<table>
<thead>
<tr>
<th>Section 1</th>
<th>Calculation of Crankshafts for Internal Combustion Engines</th>
</tr>
</thead>
<tbody>
<tr>
<td>A.</td>
<td>General ..........................................................................</td>
</tr>
<tr>
<td>B.</td>
<td>Calculation of Stresses ............................................</td>
</tr>
<tr>
<td>C.</td>
<td>Calculation of Stress Concentration Factors ..................</td>
</tr>
<tr>
<td>D.</td>
<td>Additional Bending Stresses ........................................</td>
</tr>
<tr>
<td>E.</td>
<td>Calculation of Equivalent Alternating Stress ..................</td>
</tr>
<tr>
<td>F.</td>
<td>Calculation of Fatigue Strength ....................................</td>
</tr>
<tr>
<td>G.</td>
<td>Acceptability Criteria ...............................................</td>
</tr>
<tr>
<td>H.</td>
<td>Calculation of Shrink-fits of Semi-built Crankshafts ..........</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Annex A</th>
<th>Definition of Stress Concentration Factors in Crankshaft Fillets</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Annex B</th>
<th>Stress Concentration Factors and Stress Distribution at the Edge of Oil Drillings</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Annex C</th>
<th>Alternative Method for Calculation of Stress Concentration Factors in the Web Fillet Radii of Crankshafts by utilizing Finite Element Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>A.</td>
<td>General ..................................................................................................................</td>
</tr>
<tr>
<td>B.</td>
<td>Model Requirements .........................................................................................</td>
</tr>
<tr>
<td>C.</td>
<td>Load Cases .........................................................................................................</td>
</tr>
</tbody>
</table>
Section 1

Calculation of Crankshafts for Internal Combustion Engines

A. General

1. Scope

These Rules for the scantlings of crankshafts are to be applied to diesel engines for main propulsion and auxiliary purposes, where the engines are so designed as to be capable of continuous operation at their rated power when running at rated speed.

Crankshafts which cannot satisfy these Rules will be subject to special consideration as far as detailed calculations or measurements can be submitted.

In case of:

– surface treated fillets
– tested parameters influencing the fatigue behaviour
– measured working stresses

these data can be considered on special request.

2. Field of application

These Rules apply only to solid-forged and semi-built crankshafts of forged or cast steel, with one crank throw between main bearings.

3. Principles of calculation

The design of crankshafts are 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.

When journal diameter is equal or larger than the crankpin one, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores. Otherwise, the engine manufacturer if requested by GL shall submit separate documentation of fatigue safety.

Calculation of crankshaft strength consists initially in determining the nominal alternating bending and nominal alternating torsional stresses which, multiplied by the appropriate stress concentration factors using the theory of constant energy of distortion (v. Mises’ Criterion), result in an equivalent alternating stress (uni-axial stress). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material. This comparison will then show whether or not the crankshaft concerned is dimensioned adequately.

4. Drawings and particulars to be submitted

For the calculation of crankshafts, the documents and particulars listed in the following are to be submitted:

– crankshaft drawing which must contain all data in respect of the geometrical configuration 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, precombustion chamber, etc.)
– number of cylinders
– rated power [kW]
– rated engine speed [min⁻¹]
– sense of rotation (see Fig. 1.1)
– ignition sequence with the respective ignition intervals and, where necessary, V-angle α_v (see Fig. 1.1)
– cylinder diameter [mm]
– stroke [mm]
– maximum cylinder pressure p_{max} [bar]
– charge air pressure [bar] (before inlet valves or scavenge ports, whichever applies)
– nominal compression ratio [–]
– connecting rod length L_{H} [mm]
– oscillating weight of one crank gear [kg] (in case of V-type engines, where necessary, also for the cylinder unit with master and articulated-type connecting rod or forked and inner connecting rod)
– digitalized gas pressure curve presented at equidistant intervals (bar versus crank angle, but not more than 5° CA)
Fig. 1.1  Designation of the cylinders

- for engines with articulated-type connecting rod (see Fig. 1.2)
  - distance to link point $L_{A}$ [mm]
  - link angle $\alpha_{N}$ [$^\circ$]
  - connecting rod length $L_{N}$ [mm]

Fig. 1.2  Articulated-type connecting rod

- for the cylinder with articulated-type connecting rod
  - maximum cylinder pressure $p_{\text{max}}$ [bar]
  - charge air pressure [bar] (before inlet valves or scavenge ports, whichever applies)
  - nominal compression ratio [-]
  - digitalized gas pressure curve presented at equidistant intervals [bar/°CA]

__Details of crankshaft material__

- material designation (according to ISO, DIN, AISI, etc.)
- mechanical properties of material (minimum values obtained from longitudinal test specimens)

The minimum requirements of the GL Rules II – Materials and Welding must comply with:
- tensile strength [N/mm²]
- yield strength [N/mm²]
- reduction in area at fracture [%]
- elongation $A_5$ [%]
- impact energy – KV [J]
- type of forging (free form forged, continuous grain flow forged, drop-forged, etc., with description of the forging process)
- heat treatment
- surface treatment of fillets, journals and pins (induction hardened, flame hardened, nitrided, rolled, shot peened, etc. with full details concerning hardening)
- hardness at surface [HV]
- hardness as a function of depth of hardening
- extension of surface hardening
- particulars for alternating torsional stresses, see B.2.
B. Calculation of Stresses

1. Calculation of alternating stresses due to bending moments and radial forces

1.1 Assumptions

The calculation is based on a statically determinate system, so that only one single crank throw is considered of which the journals are supported in the centre of adjacent bearings and which is subject to gas and inertia forces. The bending length is taken as the length between the two main bearings (distance $L_3$) see Figs. 1.3 and 1.4.

The bending moments $M_{BR}$, $M_{BT}$ are calculated in the relevant section based on triangular bending moment diagrams due to the radial component $F_R$ and tangential component $F_T$ of the connecting-rod force, respectively (see Fig.1.3).

For crank throws with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see Fig.1.4).

---

**Fig. 1.3** Crankthrow for in-line engine  
**Fig. 1.4** Crank throw for Vee engine with 2 adjacent connecting rods
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 centre 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 are to 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 fig. 1.5).

Mean stresses are neglected.

Fig. 1.5 Reference area of crankweb cross section
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.

\[ M_{BRO} = \text{bending moment of the radial component of the connecting-rod force} \]
\[ M_{BTO} = \text{bending moment of the tangential component of the connecting-rod force} \]

**Fig. 1.6 Crankpin section through the oil bore**

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

Mean bending stresses are neglected.

### 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. A simplified calculation of the radial and tangential forces may be used at the discretion of GL.

Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments \( M_{BRF}, M_{BRO}, M_{BTO} \) and radial forces \( Q_{RF} \) (defined in 1.1) will then be calculated.

In case of V-type engines, the bending moments – progressively calculated from the gas and inertia forces – of the two cylinders acting on one crank throw are superposed according to phase, the 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 configuration (e.g. asymmetric cranks) in one crankshaft, the calculation is to cover all crank variants.

The decisive alternating values will then be calculated according to:

\[ X_N = \pm \frac{1}{2} (X_{\text{max}} - X_{\text{min}}) \]

- \( X_N \) = considered as alternating force, moment or stress
- \( X_{\text{max}} \) = maximum value within one working cycle
- \( X_{\text{min}} \) = minimum value within one working cycle

#### 1.2.1 Nominal alternating bending and compressive stresses in web cross section

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

\[ \sigma_{\text{BFN}} = \frac{M_{\text{BRF}}}{W_{\text{eqw}}} \cdot 10^3 \cdot \text{Ke} \]
\[ \sigma_{\text{QFN}} = \frac{Q_{\text{RF}}}{F} \cdot \text{Ke} \]

- \( \sigma_{\text{BFN}} \) = nominal alternating bending stress related to the web \([\text{N/mm}^2]\)
- \( M_{\text{BRF}} \) = alternating bending moment related to the centre of the web \([\text{Nm}]\) (see Fig. 1.3 and 1.4)
- \( W_{\text{eqw}} \) = section modulus related to cross-section of web \([\text{mm}^3]\)
- \( K_e \) = empirical factor considering to some extent the influence of adjacent crank and bearing restraint with:
  - \( K_e = 0.8 \) for 2-stroke engines
  - \( K_e = 1.0 \) for 4-stroke engines
- \( \sigma_{\text{QFN}} \) = nominal alternating compressive stress due to radial force related to the web \([\text{N/mm}^2]\)
- \( Q_{\text{RF}} \) = alternating radial force related to the web \([\text{N}]\) (see Fig. 1.3 and 1.4)

\[ Q_{\text{RF}} = \pm \frac{1}{2} (Q_{\text{RFmax}} - Q_{\text{RFmin}}) \]

- \( F \) = area related to cross-section of web \([\text{mm}^2]\)
- \( W_{\text{eqw}} = \frac{B \cdot W^2}{6} \)

#### 1.2.2 Nominal alternating bending stress in outlet of crankpin oil bore

The calculation of the nominal alternating bending stress is as follows:

\[ \sigma_{\text{BON}} = \frac{M_{\text{BON}}}{W_e} \cdot 10^3 \]
\[ \sigma_{\text{BON}} = \text{nominal alternating bending stress related to crank pin diameter} [\text{N/mm}^2] \]

\[ M_{\text{BON}} = \text{alternating bending moment calculated at the outlet of crankpin oil bore} [\text{N/mm}^2] \]

\[ M_{\text{BON}} = \pm \frac{1}{2} \left[ M_{\text{BOMax}} - M_{\text{BOMin}} \right] \]

\[ M_{\text{BO}} = M_{\text{BTO}} \cdot \cos \psi + M_{\text{BRQ}} \cdot \sin \psi \]

\[ \psi = \text{angular position} [^\circ] \text{ (see Fig. 1.6)} \]

\[ W_e = \text{section modulus related to cross-section of axially bored crankpin} [\text{mm}^3] \]

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

1.3 Calculation of alternating bending stresses in fillets

The calculation of stresses is to be carried out for the crankpin fillet as well as for the journal fillet.

For the crankpin fillet:

\[ \sigma_{\text{BH}} = \pm (\alpha_{\text{B}} \cdot \sigma_{\text{BFN}}) \]

\[ \sigma_{\text{BH}} = \text{alternating bending stress in crankpin fillet} [\text{N/mm}^2] \]

\[ \alpha_{\text{B}} = \text{stress concentration factor for bending in crankpin fillet} [-]\text{ (determination, see C.)} \]

For the journal fillet:

\[ \sigma_{\text{BG}} = \pm (\beta_{\text{B}} \cdot \sigma_{\text{BFN}} + \beta_{\text{Q}} \cdot \sigma_{\text{QFN}}) \]

\[ \sigma_{\text{BG}} = \text{alternating stresses in journal fillet} [\text{N/mm}^2] \]

\[ \beta_{\text{B}} = \text{stress concentration factor for bending in journal fillet} [-]\text{ (determination, see C.)} \]

\[ \beta_{\text{Q}} = \text{stress concentration factor for shearing} [-]\text{ (determination, see C.)} \]

1.4 Calculation of alternating bending stresses in outlet of crankpin oil bore

\[ \sigma_{\text{BO}} = \pm (\gamma_{\text{B}} \cdot \sigma_{\text{BON}}) \]

\[ \sigma_{\text{BO}} = \text{alternating bending stress in outlet of crankpin oil bore} [\text{N/mm}^2] \]

\[ \gamma_{\text{B}} = \text{stress concentration factor for bending in crankpin oil bore} \text{ (determination, see C.)} \]

2. Calculation of alternating torsional stresses

2.1 General

The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in 2.2.

The maximum value obtained from such calculations will be used by GL when determining the equivalent alternating stress, according to E. In the absence of such a maximum value it will be necessary for GL to incorporate a fixed value in the calculation for the crankshaft dimensions on the basis of an estimation.

In case GL is entrusted with carrying out a forced vibration calculation on behalf of the engine manufacturer to determine the torsional vibration stresses to be expected in the engine and possibly in its shafting, the following data are to be submitted to GL additionally to A.4.:

- Equivalent dynamic system of the engine comprising
  - mass moment of inertia of every mass point [kgm²]
  - inertialess torsional stiffnesses [Nm/rad]
- Vibration dampers
  - type designation
  - mass moments of inertia [kgm²]
  - inertialess torsional stiffnesses [Nm/rad]
  - damping coefficients [Nms]
- Flywheels
  - mass moment of inertia [kgm²]

If the whole installation is to be considered, the above information is to be extended by the following:

- Coupling
  - dynamic characteristics and damping data
- Gearing data
  - shaft diameter of gear shafts, thrust shafts, intermediate shafts and propeller shafts
- Shafting
  - diameter of thrust shafts, intermediate shafts and propeller shafts
- Propellers
  - propeller diameter
  - number of blades
  - pitch and area ratio
- Natural frequencies with their relevant modes of vibration and the vector sums for the harmonics of the engine excitation.
- Estimated torsional vibration stresses in all important elements of the system with particular reference to clearly defined resonance speeds of rotation and continuous operating ranges.
2.2 Calculation of nominal alternating torsional stresses

The maximum and minimum alternating torques are to be ascertained for every mass point of the 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 must be made for the dampings that exist in the system and for unfavourable conditions (misfiring in one of the cylinders). The speed ranges shall be selected in such a way that the transient response can be recorded with sufficient accuracy.

The values received from such calculation are to be submitted.

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

\[
\tau_N = \pm \frac{M_T}{W_p} \times 10^3
\]

\[
M_{TN} = \frac{1}{2} (M_{Tmax} - M_{Tmin})
\]

\[
W_p = \frac{\pi}{16} \left( \frac{D^4 - D_{hi}^4}{D} \right)
\]

\[
\sigma_{TO} = \pm \left( \gamma_T \cdot \tau_N \right)
\]

\[
\tau_N = \text{nominal alternating torsional stress referred to crankpin or journal [N/mm²]}
\]

\[
M_{TN} = \text{nominal alternating torque [Nm]}
\]

\[
W_p = \text{polar section modulus related to cross-sectional area of bored crankpin or bored journal [mm⁴]}
\]

\[
M_{Tmax}, M_{Tmin} = \text{extreme values of the torque with consideration of the mean torque [Nm]}
\]

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. Where barred speed ranges are necessary, the torsional stresses within these ranges are to be neglected in the calculation of the acceptability factor.

Barred speed ranges are to be so arranged that satisfactory operation is possible despite of their existence. There are to be no barred speed ranges above a speed ratio of \( \lambda \geq 0.8 \) of the rated speed.

The approval of crankshafts is to be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by engine manufacturer).

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

2.3 Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore

The calculation of stresses is to be carried out for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore.

For the crankpin fillet:

\[
\tau_H = \pm (\alpha_T \cdot \tau_N)
\]

\[
\tau_H = \text{alternating torsional stress in crankpin fillet [N/mm²]}
\]

\[
\alpha_T = \text{stress concentration factor for torsion in crankpin fillet [-] (determination, see C.)}
\]

\[
\tau_N = \text{nominal alternating torsional stress related to crankpin diameter [N/mm²]}
\]

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

\[
\tau_G = \pm (\beta_T \cdot \tau_N)
\]

\[
\tau_G = \text{alternating torsional stress in journal fillet [N/mm²]}
\]

\[
\beta_T = \text{stress concentration factor for torsion in journal fillet [-] (determination, see C.)}
\]

\[
\tau_N = \text{nominal alternating torsional stress related to crankpin diameter [N/mm²]}
\]

For the outlet of crankpin oil bore:

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

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

\[
\gamma_T = \text{stress concentration factor for torsion in outlet of crankpin oil bore [-] (determination, see C.)}
\]

\[
\tau_N = \text{nominal alternating torsional stress related to crankpin diameter [N/mm²]}
\]

C. Calculation of Stress Concentration Factors

1. General

The stress concentration factors are evaluated by means of the formulae according to 2., 3. and 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 must 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.

Where the geometry of the crankshaft is outside the boundaries of the analytical stress concentration fac-
tors (SCF) the calculation method detailed in Annex C may be undertaken.

All crank dimensions necessary for the calculation of stress concentration factors are shown in Fig. 1.7.

The stress concentration factors for bending ($\alpha_B$, $\beta_B$) are defined as the ratio of the maximum equivalent stress (von Mises) - occurring in the fillets under bending load - to the nominal stress related to the web cross-section, see Annex A.

The stress concentration factor for compression ($\beta_Q$) in the journal fillet is defined as the ratio of the maximum equivalent stress (von Mises) - occurring in the fillet due to the radial force - to the nominal compressive stress related to the web cross-section.

The stress concentration factor for torsion ($\alpha_T, \beta_T$) is defined as the ratio of the maximum equivalent shear stress - occurring in the fillets under torsional load - to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see Annex A).

The stress concentration factors for bending ($\gamma_B$) and torsion ($\gamma_T$) are defined as the ratio of the maximum principal stress - occurring at the outlet of the crankpin oil-hole under bending and torsional loads - to the corresponding nominal stress related to the axially bored crankpin cross section (see Annex B).

When reliable measurements and/or calculations are available, which can allow direct assessment of stress concentration factors, the relevant documents and their analysis method have to be submitted to Classification Societies in order to demonstrate their equivalence to present rules evaluation.

Actual dimensions:

- $D =$ crankpin diameter [mm]
- $D_{BH} =$ diameter of axial bore in crankpin [mm]
- $D_O =$ diameter of oil bore in crankpin [mm]
- $R_H =$ fillet radius of crankpin [mm]
- $T_H =$ recess of crankpin [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 [mm]
- $E =$ pin eccentricity [mm]
- $S =$ pin overlap [mm]

$$W^* = \frac{D + D_G}{2} - E$$

- $W^* =$ web thickness [mm]
- $B^* =$ web width [mm]

*) in case of semi-built crankshafts:
- when $T_H > R_H$
  - the web thickness must be considered as equal to
  $$W_{red} = W - (T_H - R_H)$$
  see Fig. 1.7
- web width $B$ must be taken in way of crankpin fillet radius centre acc. to Fig. 1.7

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

<table>
<thead>
<tr>
<th>Crankpin fillets</th>
<th>Journal fillets</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r = R_H/D$</td>
<td>$r = R_G/D$</td>
</tr>
<tr>
<td>$s = S/D$</td>
<td>crankshafts with overlap</td>
</tr>
<tr>
<td>$w = W/D$</td>
<td>$W_{red}/D$ crankshafts without overlap</td>
</tr>
<tr>
<td>$b = B/D$</td>
<td></td>
</tr>
<tr>
<td>$d_O = D_O/D$</td>
<td></td>
</tr>
<tr>
<td>$d_G = D_{BG}/D$</td>
<td></td>
</tr>
<tr>
<td>$d_H = D_{BH}/D$</td>
<td></td>
</tr>
<tr>
<td>$t_H = T_H/D$</td>
<td></td>
</tr>
<tr>
<td>$t_G = T_G/D$</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 1.7 Crank dimensions necessary for the calculation of stress concentration factors
Stress concentration factors are valid for the ranges of related dimensions for which the investigations have been carried out. Ranges are as follows:

\[
\begin{align*}
s & \leq 0.5 \\
0.2 & \leq w \leq 0.8 \\
1.1 & \leq b \leq 2.2 \\
0.03 & \leq r \leq 0.13 \\
0 & \leq d_G \leq 0.8 \\
0 & \leq d_H \leq 0.8 \\
0 & \leq d_O \leq 0.2 \\
\end{align*}
\]

Low range of s can be extended down to large negative values provided that:
- if calculated \( f(\text{recess}) < 1 \) then the factor \( f(\text{recess}) \) is not to be considered (\( f(\text{recess}) = 1 \))
- if \( s < -0.5 \) then \( f(s, w) \) and \( f(r, s) \) are to be evaluated replacing actual value of \( s \) by \(-0.5\)

2. Crankpin fillet

The stress concentration factor for bending \( \alpha_{B} \) is:

\[
\alpha_{B} = 2.6914 \cdot f(s, w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(\text{recess})
\]

\[
f(s, w) = -4.1883 + 29.2004 \cdot w - 77.5925 \cdot w^2 + 91.9454 \cdot w^3 - 40.416 \cdot w^4 + (1 - s) \cdot (9.5440 - 58.3480 \cdot w + 159.3415 \cdot w^2 - 192.5846 \cdot w^3 + 85.2916 \cdot w^4) + (1 - s)^2 \cdot (-3.8399 + 25.0444 \cdot w - 70.5571 \cdot w^2 + 87.0328 \cdot w^3 - 39.1832 \cdot w^4)
\]

\[
f(w) = 2.1790 \cdot w^{0.7171}
\]

\[
f(b) = 0.6840 - 0.0077 \cdot b + 0.1473 \cdot b^2
\]

\[
f(r) = 0.2081 \cdot r^{-0.5231}
\]

\[
f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3
\]

\[
f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241 \cdot d_H^2 + 2.4147 \cdot d_H^3
\]

\[
f(\text{recess}) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)
\]

The stress concentration factor for torsion \( \alpha_T \) is:

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

\[
f(r, s) = r^{-0.322} + 0.1015 \cdot (1 - s)
\]

\[
f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 - 0.857 \cdot b^3
\]

\[
f(w) = w^{-0.145}
\]

3. Journal fillet

(not applicable to semi-built crankshaft)

The stress concentration factor for bending \( \beta_B \) is:

\[
\beta_B = 2.7146 \cdot f_B(s, w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})
\]

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

\[
f_B(w) = 2.2422 \cdot w^{0.7348}
\]

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

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

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

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

\[
f(\text{recess}) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)
\]

The stress concentration factor for compression \( \beta_Q \) due to the radial force is:

\[
\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})
\]

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

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

\[
f_Q(b) = -0.5 + b
\]

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

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

\[
f(\text{recess}) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)
\]

The stress concentration factor for torsion \( \beta_T \) is:

\[
\beta_T = \alpha_T
\]

if the diameters and fillet radii of crankpin and journal are the same, and if crankpin and journal diameters and/or radii are of different sizes:

\[
\beta_T = 0.8 \cdot f(r, s) \cdot f(b) \cdot f(w)
\]

\( f(r, s) \), \( f(b) \) and \( f(w) \) are to be determined in accordance with 2. (see calculation of \( \alpha_T \)), however, the radius of the journal fillet is to be related to the journal diameter:

\[
r = \frac{R_G}{D_G}
\]
4. **Outlet of crankpin oil bore**

The stress concentration factor for bending \( \gamma_B \) is:

\[
\gamma_B = 3 - 5.88 \cdot d_O + 34.6 \cdot d_O^2
\]

The stress concentration factor for torsion \( \gamma_T \) is:

\[
\gamma_T = 4 - 6 \cdot d_O + 30 \cdot d_O^2
\]

**D. Additional Bending Stresses**

In addition to the alternating bending stresses in fillets (see B.1.3) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying \( \sigma_{\text{add}} \) as given by the following table:

<table>
<thead>
<tr>
<th>Type of engine</th>
<th>( \sigma_{\text{add}} ) [N/mm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crosshead engines</td>
<td>± 30 *)</td>
</tr>
<tr>
<td>Trunk piston engines</td>
<td>± 10</td>
</tr>
</tbody>
</table>

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

- an additional stress of ± 20 N/mm² resulting from axial vibration
- an additional stress of ± 10 N/mm² resulting from misalignment / bedplate deformation

It is recommended that a value of ± 20 N/mm² be used for the axial vibration component for assessment purpose where axial vibration calculation results of the complete dynamic system (engine / shafting / gearing / propeller) are not available.

Where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

**E. Calculation of Equivalent Alternating Stress**

1. **General**

In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (see Annex A).

As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the von Mises criterion.

At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see Annex B).

The above two different ways of equivalent stress evaluation both lead to stresses which may be compared to the same fatigue strength value of crankshaft assessed according to von Mises criterion.

2. **Equivalent alternating stress**

The equivalent alternating stress is calculated in accordance with the formulæ given.

For the crankpin fillet:

\[
\sigma_v = \pm \sqrt{(\sigma_{\text{add}})^2 + 3 \cdot \tau_H^2}
\]

For the journal fillet:

\[
\sigma_v = \pm \sqrt{(\sigma_{\text{add}})^2 + 3 \cdot \tau_G^2}
\]

For the outlet of crankpin oil bore:

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

\( \sigma_v = \text{equivalent alternating stress [N/mm²]} \)

For other parameters, see B.1.3, B.2.3 and D.

**F. Calculation of Fatigue Strength**

The fatigue strength is to be understood as that value of equivalent alternating stress (von Mises) which a crankshaft can permanently withstand at the most highly stressed points; the fatigue strength may be evaluated by means of the following formulæ:

Related to the crankpin diameter:

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

\( R_X = R_H \) in the fillet area

\( R_X = D_O/2 \) in the oil bore area

Related to the journal diameter:

\[
\sigma_{DW} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \\
\left[ 0.264 + 1.073 \cdot D_G^{-0.2} + \frac{785 - \sigma_B}{4900} \right] \\
\left[ 1 + \frac{196}{\sigma_B} \cdot \frac{1}{\sqrt{R_X}} \right]
\]
\[ \sigma_{DW} = \text{allowable fatigue strength of crankshaft} \quad [\text{N/mm}^2] \]

\[ K = \text{factor for different types of crankshafts without surface treatment} \quad [-] \]

Values greater than 1 are only applicable to fatigue strength in fillet area.

\[ = 1,05 \quad \text{for continuous grain flow forged or drop-forged crankshafts} \]

\[ = 1,0 \quad \text{for free form forged crankshafts} \]

\[ \text{factor for cast steel crankshafts with cold rolling treatment in fillet area} \quad [-] \]

\[ = 0,93 \quad \text{for cast steel crankshafts manufactured by companies using a GL approved cold rolling process} \]

\[ \sigma_B = \text{minimum tensile strength of crankshaft material} \quad [\text{N/mm}^2] \]

For other parameters see C.1.

When a surface treatment process is applied, it must be approved by GL.

These formulae are subject to the following conditions:

- surface of the fillet, the outlet of the oil bore and inside the oil bore (down to a minimum depth equal to 1,5 times the oil bore diameter) shall be smoothly finished.

- for calculation purposes \( R_H, R_G \) or \( R_X \) are to be taken as not less than 2 mm.

As an alternative the fatigue strength of the crankshaft can be determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow.

In any case the experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment have to be submitted for approval to GL (method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, confidence number …).

G. Acceptability Criteria

The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. This comparison has to be carried out both for the crankpin fillet, the journal fillet, the outlet of crankpin oil bore and is based on the formula:

\[ Q = \frac{\sigma_{DW}}{\sigma_v} \]

\( Q \) = acceptability factor \([-]\)

Adequate dimensioning of the crankshaft is ensured if the smaller of both acceptability factors satisfies the criterion:

\[ Q \geq 1,15 \]

H. Calculation of Shrink-fits of Semi-built Crankshafts

1. General

All crank dimensions necessary for the calculation of the shrink-fit are shown in Fig. 1.8.

\[ D_S = \text{shrink diameter} \quad [\text{mm}] \]

\[ L_S = \text{length of shrink-fit} \quad [\text{mm}] \]

---

![Fig. 1.8 Crank throw of semi-built crankshaft](image-url)
\( D_A \) = outside diameter of web [mm] or
twice the minimum distance \( x \) between centre-line of journals and outer contour of web, whichever is less.

\( y \) = distance between the adjacent generating lines of journal and pin [mm]
\( y \geq 0,05 \cdot D_S \)

Where \( y \) is less than \( 0,1 \cdot D_S \), special consideration is to be given to the effect of the stress due to the shrink on the fatigue strength at the crankpin fillet.

For other parameter, see C.1. (Fig. 1.7).

Regarding the radius of the transition from the journal to the shrink diameter, the following must be observed:

\[ R_G \geq 0,015 \cdot D_G \quad \text{and} \quad R_G \geq 0,5 \cdot (D_S - D_G) \]

where the greater value is to be considered.

The actual oversize \( Z \) of the shrink-fit must be within the limits \( Z_{\min} \) and \( Z_{\max} \) calculated in accordance with items 2. and 3.

In the case where H.2. condition cannot be fulfilled then H.3. and H.4. calculation methods of \( Z_{\min} \) and \( Z_{\max} \) are not applicable due to multizone-plasticity problems. In such case \( Z_{\min} \) and \( Z_{\max} \) have to be established based on FEM calculations.

2. Maximum permissible hole in the journal pin

The maximum permissible hole diameter in the journal pin is calculated in accordance with the following formula:

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

\( S_R \) = safety factor against slipping [-]
A value not less than 2 is to be taken unless documented by experiments.

\( M_{\max} \) = absolute maximum value of the torque \( M_{T\max} \) in accordance with B.2.2 [Nm]

\( \mu \) = coefficient for static friction [-]
A value not greater than 0.2 is to be taken unless documented by experiments.

\( \sigma_{SP} \) = minimum yield strength of material for journal pin [N/mm²]
This condition serves to avoid plasticity in the hole of the journal pin.

3. Necessary minimum oversize of shrink-fit

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

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

and

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

\( Z_{\min} \) = minimum oversize [mm]
\( E_m \) = Young’s modulus [N/mm²]
\( \sigma_{SW} \) = minimum yield strength of material for crank web [N/mm²]
\( Q_A \) = web ratio [-] \( Q_A = \frac{D_S}{D_A} \)
\( Q_S \) = shaft ratio [-] \( Q_S = \frac{D_{BG}}{D_S} \)

4. Maximum permissible oversize of shrink-fit

The maximum permissible oversize is calculated in accordance with the following formula:

\[ Z_{\max} \leq \frac{\sigma_{SW} \cdot D_S}{E_m} + \frac{0,8 \cdot D_S}{1000} \]

\( Z_{\max} \) = maximum oversize [mm]

The condition serves to restrict the shrinkage induced mean stress in the fillet.
Annex A

Definition of Stress Concentration Factors in Crankshaft Fillets

| Stress Location of maximal stresses | Max $||\sigma_2||$ | Max $\sigma_1$ | B |
|-----------------------------------|-----------------|---------------|---|
| Torsional loading                 | A               | C             | B |
| Typical principal stress system   |                 |               |   |
| Mohr’s circle diagram with $\sigma_2 = 0$ | $||\sigma_2|| > \sigma_1$ | $\sigma_1 > ||\sigma_3||$ | $\sigma_1 = ||\sigma_3||$ |
| Equivalent stress and S.C.F.      | $\tau_{equiv} = \frac{\sigma_1 - \sigma_2}{2}$ | S.C.F. = $\frac{\tau_{equiv}}{\tau_n}$ for $\alpha_T, \beta_T$ |
| Location of maximal stresses      | B               | B             | B |
| Bending loading                   |                 |               |   |
| Typical principal stress system   |                 |               |   |
| Mohr’s circle diagram with $\sigma_3 = 0$ | $\sigma_2 
eq 0$ | $\sigma_{equiv} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2}$ | S.C.F. = $\frac{\sigma_{equiv}}{\sigma_n}$ for $\alpha_B, \beta_B, \beta_Q$ |
Annex B

Stress Concentration Factors and Stress Distribution at the Edge of Oil Drillings

\[
\sigma_x = \sigma_n \gamma_l / 3 \left[ 1 + 2 \cos(2\alpha) \right]
\]

\[
\gamma_B = \sigma_{\text{max}} / \sigma_n \quad \text{for} \quad \alpha = k\pi
\]

\[
\gamma_T = \sigma_{\text{max}} / \tau_{\text{n}} \quad \text{for} \quad \alpha = \frac{\pi}{4} + k\frac{\pi}{2}
\]

\[
\sigma_{\text{max}} = \frac{1}{2} \sigma_n \left[ 1 + 2 \left( \cos(2\alpha) + \frac{3}{2} \frac{\tau_{\text{n}}}{\gamma_l} \frac{\tau_{\text{t}}}{\sigma_n} \sin(2\alpha) \right) \right]
\]

\[
\alpha = -\frac{1}{2} \tan\left( \frac{3\gamma_T \tau_{\text{t}}}{2\sigma_n \sigma_{\text{n}}^2} \right)
\]
Annex C

Alternative Method for Calculation of Stress Concentration Factors in the Web Fillet Radii of Crankshafts by utilizing Finite Element Method

A. General

1. The objective of the analysis is to develop Finite Element Method (FEM) calculated figures as an alternative to the analytically calculated Stress Concentration Factors (SCF) at the crankshaft fillets. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries and accordingly the application of these formulae is limited to those geometries.

2. The SCF’s calculated according to the rules of this Annex 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 or the alternative methods, von Mises stresses shall be calculated for bending and principal stresses for torsion.

3. The procedure as well as evaluation guidelines are valid for both solid cranks and semibuilt cranks (except journal fillets).

4. The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.

5. The calculation of SCF at the oil bores is not covered by this Annex.

6. It is advised to check the element accuracy of the FE solver in use, e.g. by modeling a simple geometry and comparing the stresses obtained by FEM with the analytical solution for pure bending and torsion.

7. Boundary Element Method (BEM) may be used instead of FEM.

B. Model Requirements

1. General

The basic recommendations and perceptions for building the FE-model are presented in 2. It is obligatory for the final FE-model to fulfill the requirement in 4.

2. Element mesh recommendations

In order to fulfill 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 the main bearing centreline to the opposite side main bearing centreline.

- 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 $a = r/4$ through the entire fillet as well as in the circumferential direction. When using 20 node hexahedral elements, the element size in the circumferential direction may be extended up to $5a$. In the case of multi-radii fillet $r$ 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 1
    - Second layer thickness equal to element to size of 2a
    - Third layer thickness equal to element to size of 3a
  - Minimum 6 elements across web thickness.
  - Generally the rest of the crank should be suitable for numeric stability of the solver.
  - Counterweights only have to be modeled only when influencing the global stiffness of the crank significantly.
  - Modeling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than 2r, see Fig. C.1.
  - Drillings and holes for weight reduction have to be modeled.
  - Sub-modeling may be used as far as the software requirements are fulfilled.
3. Material

These Rules do 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 have to be used, either as quoted in literature or as measured on representative material samples.

For steel the following is advised: $E = 2.05 \cdot 10^5$ MPa and $\nu = 0.3$.

4. Element mesh quality criteria

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

4.1 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 $\sigma_1$, $\sigma_2$ and $\sigma_3$ the following criterion is required:

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

4.2 Averaged/unaveraged stresses criterion

The criterion is based on observing the discontinuity of stress results over elements at the fillet for the calculation of 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.

C. Load Cases

1. General

To substitute the analytically determined SCF in these Rules the following load cases have to be calculated.

1.1 Torsion

In analogy to the testing apparatus used for the investigations made by FVV the structure is loaded 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 inline and V-type engines.

Multi-point constraint:
All nodes of cross section are rigidly connected to central node (= master)

Load:
Torque T applied to central node

Fig. C.1 Oil bore proximity to fillet

Fig. C.2 Boundary and load conditions for the torsion load case
For all nodes in both the journal and crank pin fillet principal stresses are extracted and the equivalent torsional stress is calculated:

\[
\tau_{\text{equiv}} = \max \left( \left| \sigma_1 - \sigma_2 \right| , \left| \sigma_2 - \sigma_3 \right| , \left| \sigma_1 - \sigma_3 \right| \right)
\]

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

\[
\alpha_T = \frac{\tau_{\text{equiv,}T}}{\tau_N}
\]

\[
\beta_T = \frac{\tau_{\text{equiv,}\beta}}{\tau_N}
\]

where \( \tau_N \) is nominal torsional stress referred to the crankpin and respectively journal as per Section 1, B.2.2 with the torsional torque \( T \):

\[
\tau_N = \frac{T}{W_p}
\]

1.2 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 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.

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

\[
\alpha_B = \frac{\sigma_{\text{equiv,}T}}{\sigma_N}
\]

\[
\beta_B = \frac{\sigma_{\text{equiv,}\beta}}{\sigma_N}
\]

Nominal stress \( \sigma_N \) is calculated as per Section 1, B.1.2.1 with the bending moment \( M \):

\[
\sigma_N = \frac{M}{W_{\text{eqw}}}
\]

1.3 Bending with shear force (3-point bending)

This load case is calculated to determine the SCF for pure transverse force (radial force, \( \beta_Q \)) 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 central node; boundary conditions are applied to the centre nodes. These nodes act as master nodes with 6 degrees of freedom.
Fig. C.4 Boundary and load conditions for the 3-point bending load case of an inline engine

Load:
Force $F_{3p}$ applied at central node at connecting rod centre line.

Boundary conditions:
Displacement in z direction for master node is restrained, $u_z = 0$; $u_x$, $u_y$ and $\varphi \neq 0$ (axial, vertical displacements and rotations are free)

Multi-point constraint:
All nodes of cross section are connected to a central node (= master)

Boundary conditions:
Displacements for master node are fully restrained $u_x, u_y, u_z = 0$; $\varphi \neq 0$ (rotations are free)

Boundary conditions:
Displacements in y and z directions for master node are restrained $u_y, u_z = 0$. $u_x, \varphi \neq 0$ (axial displacement and rotations are free)

Fig. C.5 Load applications for in-line and V-type engines
The force is applied to the central node located at the 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 modeled with one connecting rod force only. Using two connecting rod forces will make no significant change in the SCF.

The maximum equivalent von Mises stress $\sigma_{3P}$ in the journal fillet is evaluated. The SCF in the journal fillet can be determined in two ways as shown below.

1.3.1 Method 1

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

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

where:

$\sigma_{3P}$ as found by the FE calculation.

$\sigma_{N3P}$ Nominal bending stress in the web centre due to the force $F_{3P}$ [N] applied to the centre-line of the actual connecting rod, see Fig. C.5

$\beta_B$ as determined in 1.2.

$\sigma_{Q3P} = Q_{3P}/(B \cdot W)$ where $Q_{3P}$ is the radial (shear) force in the web due to the force $F_{3P}$ [N] applied to the centre-line of the actual connecting rod, see also Section 1, Fig. 1.3 and 1.4.

1.3.2 Method 2

This method is not analogous to the FVV investigation. In a statically determined system with one crank throw supported by 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 1.3.1.

When using this method the radial force and stress determination becomes superfluous. The alternating bending stress in the journal fillet as per Section 1, B.1.3 is then evaluated:

$$\sigma_{BG} = \pm \beta_{BQ} \cdot \sigma_{BNF}$$

Note that the use of this method does not apply to the crankpin fillet and that this SCF must not be used in connection with calculation methods other than those assuming a statically determined system.