



Rules for the Calculation of Crankshafts for Diesel Engines

1. General

1.1 Scope

These Rules for obtaining general approval are to be applied for checking the sufficient dimensioning of crankshafts for diesel engines for main propulsion and auxiliary purposes.

Diesel engines are to be so designed as to be capable of continuous operation at their rated power when running at rated speed.

Engines for special applications which cannot satisfy these Rules will be subject to special consideration if detailed calculations or measurements are submitted.

1.2 Field of application

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

The Rules do not apply to other types of crankshafts such as fully-built crankshafts or crankshafts of nodular cast iron, nor to engines having two or more crank throws between two main bearings, nor those with offset pins.

1.3 Calculation principles

The dimensioning of crankshafts is based on the assumption that the fillet transitions between the crankpin and web as well as between the journal and web are the areas with the highest stresses and for which a calculation of the stresses is to be carried out.

The outlets of oil bores into crankpins and journals are to be formed in such a way that the stresses at the oil bores are less than those in the fillets.

Calculation of crankshaft strength consists initially of determining the nominal alternating bending and nominal alternating torsional stresses which, multiplied by the appropriate stress concentration factors using the deformation hypothesis (von Mises' Criterion), give a comparative alternating stress. This comparative alternating stress is then compared with the fatigue strength of the selected crankshaft material. This comparison gives the solution to the question of sufficient dimensioning of a crankshaft.

It must be ensured that the maximum stresses which may arise from unfavourable operating conditions within the service range are determined.

1.4 Calculation data

In order to facilitate the calculation of crankshafts for the rated power at the rated speed specified by the engine manufacturer, the documents and data listed in the following are to be submitted to the Society and listed in the appropriate form in addition to the documents required for approval in accordance with the Construction Rules:

- crankshaft drawing

which must contain all data in respect of the geometrical configuration of the crankshaft

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- 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 [1/min]
- sense of rotation, see fig. 1
- firing order with the respective ignition intervals and, where necessary, V-angle α_V , see fig. 1.

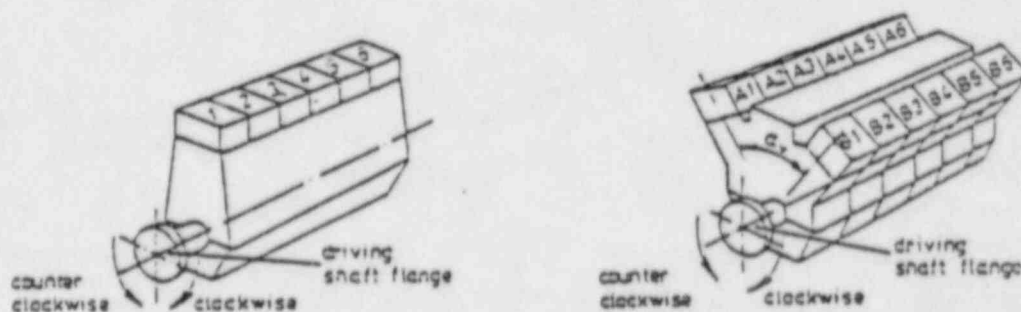


Fig. 1 Designation of the cylinders

- cylinder diameter [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)
- for engines with articulated-type connecting rod, see fig. 2.
 - distance to link point L_A [mm]
 - link angle α_N [°]
 - connecting rod length L_N [mm]

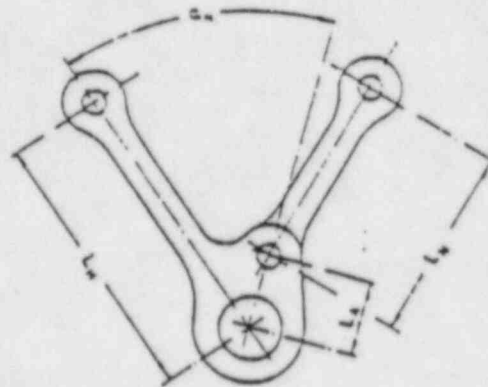


Fig. 2 Articulated-type connecting rod

- gas-pressure curve with values listed and presented at equidistant intervals

Wherever possible:

- gas pressure curve with values listed at equidistant intervals [$^{\circ}\text{CA}$ - degrees of the rotational angle of the crankshaft]
(intervals equidistant and integrally divisible by the V-angle, but not less than $2,5^{\circ}\text{CA}$ for 2-stroke nor 3°CA for 4-stroke cycle)

Or, if the gas-pressure curve has not been measured:

- indication of the constant-pressure combustion interval α_{gp} expressed in $^{\circ}\text{CA}$ measured from the T.D.C., as well as of the polytropic exponent for expansion (where necessary, up to 3 exponents for different ranges α_1 , α_2 and α_3) and compression ranges, see fig. 3.

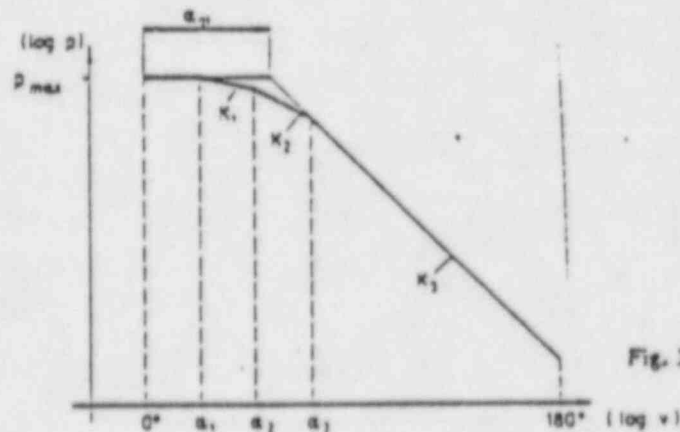


Fig. 3 Details of combustion sequence

- for engines with articulated-type connecting rod
(also the following details for the cylinder with articulated-type connecting rod)
 - maximum cylinder pressure p_{\max} [bar]
 - charge air pressure [bar]
(before inlet valves or scavenge ports, whichever applies)
 - nominal compression ratio [-]
 - gas pressure curve

- details of crankshaft material
 - material designation
(according to DIN, AISI, etc.)
 - mechanical properties of material
(minimum values obtained from longitudinal test specimens)
The minimum requirements of the Society's Rules for Materials must have been complied with:
 - tensile strength $[N/mm^2]$
 - yield strength $[N/mm^2]$
 - reduction in area at break $[\%]$
 - elongation A_5 $[\%]$
 - impact energy - KV $[J]$
 - method of material melting process
(open - hearth furnace, electric furnace, etc.)
 - type of forging
(free form forged, grain flow forged, drop-forged)
 - heat treatment
 - surface treatment
(induction hardened, nitrided, chromized, rolled, shot peened, etc.)
- particulars of the alternating torsional stresses, see item 2.2.

2. Calculation of nominal stresses

2.1 Calculation of alternating stresses due to bending moments and shearing forces

2.1.1 Assumptions

The calculation is to be based on a statically determined 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 figures 4 to 6.

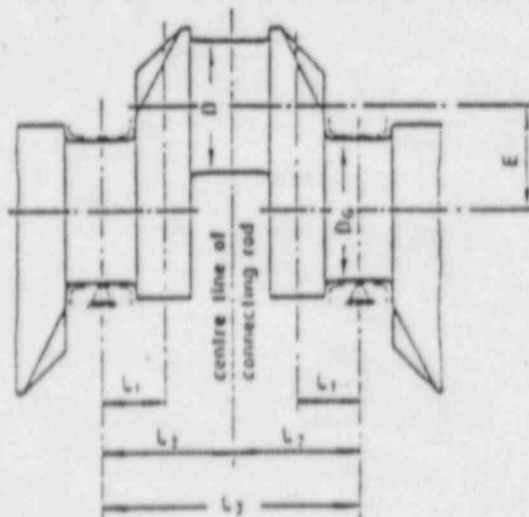


Fig. 4 Crank throw for in-line engine

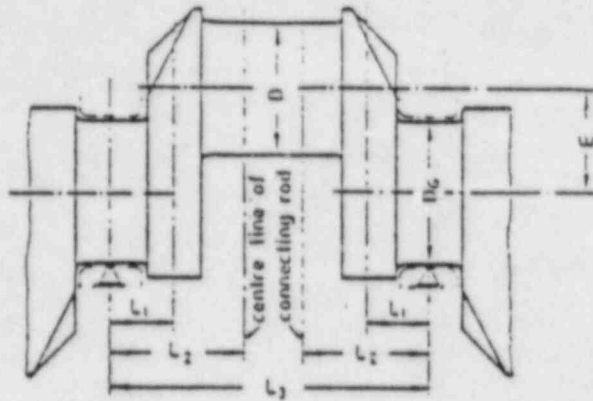


Fig. 5 Crank throw for engine with 2 adjacent connecting rods

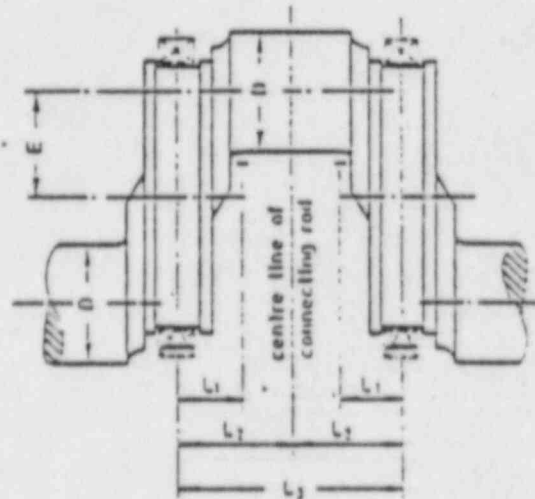


Fig. 6 Crank throw of disc-type crankshafts

The nominal bending moment is taken as the bending moment in the crank web cross-section at the middle of the solid web (distance L_1) with a triangular bending moment load due to bending moments and shearing forces resulting from the radial components of the connecting rod force.

The nominal alternating stresses due to bending moments and shearing forces are to be referred to the cross-sectional area of the crank web. This reference area of cross-section results from the web thickness W and the web width B in the centre of the overlap of the pins or, if appropriate, at the centre of the adjacent generatrices of the two pins if they do not overlap, see figure 7.

Nominal mean bending stresses are neglected.

2.1.2 Calculation of nominal alternating bending stresses

The calculation is carried out in such a way, that the radial forces P_R acting upon the crank pin owing to

gas and inertia forces will be calculated for all crank positions within one working cycle (smallest interval 2.5 °CA for 2-stroke and 5 °CA for 4-stroke cycle). From these individual values P_B , the highest positive value $P_{B,max}$ (radial force presses down upon the crank) and the highest negative value $P_{B,min}$ (radial force pulls at the crank) is taken and the nominal alternating bending force P_{BN} occurring during one working cycle determined by means of equation

$$P_{BN} = \pm \frac{1}{2} \cdot (P_{B,max} - P_{B,min})$$

By means of the nominal alternating force P_{BN} and the assumptions made in 2.1.1 the decisive nominal alternating bending moment will then be calculated

$$M_{BN} = \pm \frac{1}{2} \cdot (M_{B,max} - M_{B,min})$$

and, from the latter, the nominal alternating bending stress.

$$\sigma_{BN} = \pm \frac{M_{BN}}{W_{eq}} \cdot 10^3$$

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

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 differing designs (forked connecting rod, articulated-type connecting rod or adjacent connecting rods) being 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 calculation of the nominal alternating shearing force and stress is as follows:

$$Q_N = \pm \frac{1}{2} \cdot (Q_{max} - Q_{min})$$

$$\sigma_{QN} = \pm \frac{Q_N}{F}$$

$$F = B \cdot W$$

where:

P_{BN} [N] nominal alternating bending force

M_{BN} [Nm] nominal alternating bending moment

σ_{BN} [N/mm²] nominal alternating bending stress
 W_{eq} [mm³] equatorial moment of resistance referred to cross-sectional area of web
 Q_N [N] nominal alternating shearing force
 σ_{QN} [N/mm²] nominal alternating stress due to shearing force
 F [mm²] area referred to cross-section of web

2.1.3 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:

$$\sigma_{BH} = \pm (\sigma_B \cdot \sigma_{BN})$$

where:

σ_{BH} [N/mm²] alternating bending stress in crankpin fillet
 σ_B [-] stress concentration factor for bending in crankpin fillet (determination - see item 3)

For the journal fillet:

$$\sigma_{BG} = \pm (\sigma_B \cdot \sigma_{BN} + \sigma_Q \cdot \sigma_{QN})$$

where:

σ_{BG} [N/mm²] alternating stresses in journal fillet
 σ_B [-] stress concentration factor for bending in journal fillet (determination - see item 3)
 σ_Q [-] stress concentration factor for shearing (determination - see item 3)

2.2 Calculation of alternating torsional stresses

2.2.1 General

The calculation of the nominal alternating torsional stresses is to be performed by the engine manufacturer.

The approval of an engine is to be based on a value determined by the engine manufacturer. Only in the absence of such a value will it be necessary for the Society to include in the calculation a fixed value or a value found by forced vibration calculation.

2.2.2 Calculation of nominal alternating torsional stresses

The 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 zero order up to at least to the 12th order. Whilst doing so, allowance must be made for the dampings that exist in the system. The speed stages shall be selected in such a way that the transient response can be recorded with sufficient accuracy.

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

$$\tau_N = \pm \frac{M_T}{W_p} \cdot 10^3$$

$$M_T = \pm \frac{1}{2} \cdot (M_{Tmax} - M_{Tmin})$$

$$W_p = \frac{\pi}{16} \cdot \left(\frac{D^4 - D_{BH}^4}{D} \right) \quad \text{or} \quad W_p = \frac{\pi}{16} \cdot \left(\frac{D^4 - D_{AG}^4}{D} \right)$$

where:

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

M_T [Nm] nominal alternating torque

W_p [mm³] polar moment of resistance referred to cross-sectional area of bored crankpin or bored journal

M_{Tmax}, M_{Tmin} extreme values of the torque with consideration of the mean torque

To assess the crankshaft, that torsional stress is to be used that, in conjunction with the associated bending stress, results in the highest comparative alternating stress as per item 5. Where barred speed ranges are necessary, the torsional stresses within these ranges are to be neglected in the calculation of the comparative alternating stress.

Barred speed ranges are to be so arranged that satisfactory operation is possible despite their existence. There are to be no barred speed ranges above a speed ratio of $\lambda \geq 0,3$ of the rated speed.

The approval of crankshafts is to be based on the installation which gives the highest comparative strength according to item 5 and for the lowest safety factor according item 5.

Thus for each installation with a rigidly coupled engine, 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.

In the case of installation with elastically coupled engines, it will suffice to calculate the nominal alternating torsional stress of the system up to the primary part of the coupling only, provided the coupling sufficiently separates the engine from the machinery it is driving.

2.2.3 Calculation of alternating torsional 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:

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

where:

τ_H [N/mm²] alternating torsional stress in crankpin fillet

α_T [-] stress concentration factor for torsion in crankpin fillet (determination - see item 3)

For the journal fillet:

where:

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

S_T [-] stress concentration factor for torsion in journal fillet (determination - see item 3)

3. Calculation of stress concentration factors

3.1 Stress concentration factors for crank throws without recessed fillets

3.1.1 General

The calculation of the stress concentration factors is based on investigations by Forschungsvereinigung Verbrennungskraftmaschinen (FVV) and which apply to solid-forged crankshafts and to the crankpin fillets of semi-built crankshafts.

All crank dimensions necessary for the calculation of stress concentration factors are shown in figure 7.

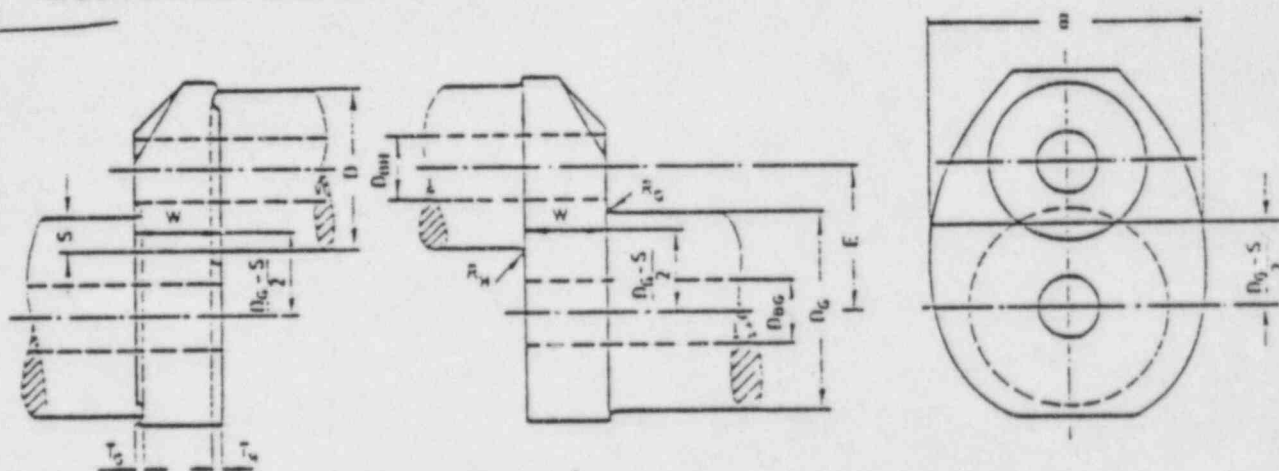


Fig. 7 Crank dimensions necessary for the calculation of stress concentration factors

Actual dimensions:

D [mm] crankpin diameter

D_{BH} [mm] diameter of bore in crankpin

R_H [mm] fillet radius of crankpin

T_H [mm] recess of crankpin

D_C [mm] journal diameter

D_{BC} [mm] diameter of bore in journal

R_C [mm] fillet radius of journal

T_C [mm] recess of journal

E [mm] pin eccentricity

S [mm] pin overlap

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

W [mm] web thickness

B [mm] web width

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

crankpin fillets	journal fillets
$r = R_H/D$	$r = R_C/D$
$s = S/D$	
$w = W/D$	
$b = B/D$	
$d_C = D_{BC}/D$	
$d_H = D_{BH}/D$	
$t_H = T_H/D$	
$t_C = T_C/D$	

They are valid for the ranges of related dimensions for which the investigations have been carried out. They are as follows:

$$0,5 \leq s \leq 0,7$$

$$0,2 \leq w \leq 0,3$$

$$1,2 \leq b \leq 2,2$$

$$0,03 \leq r \leq 0,13$$

$$0 \leq d_C \leq 0,3$$

$$0 \leq d_H \leq 0,3$$

Unless alternative values are furnished by reliable measurements, the stress concentration factors have to be calculated by means of the following formulae applicable to web-type crankshafts only. Stress concentration factors for disc-type crankshafts have in any case to be established and determined by tests.

The stress concentration factor for bending is defined as the ratio of the reference stress σ'_V - occurring in the fillets under bending load acting in the central cross-section of a crank - to the nominal stress referred to the web cross-section.

The reference stress σ'_V has to be determined from the principal stresses σ_1 and σ_2 by means of the equation:

$$\sigma'_V = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2}$$

The nominal stress has to be determined, for web-type crankshafts, under the bending moment in the middle of the solid web. For disc-type crankshafts, the nominal stress has to be established under the bending moment occurring at the distance L_1 from the disc centre line, see fig. 6.

The stress concentration factor for torsion is defined as the ratio of the maximum torsional stress occurring under torsional load in the fillets to the nominal stress referred to the bored crankpin or journal cross-section. The maximum torsional stress has to be determined from

$$\tau = \frac{\sigma'_V}{\sqrt{3}}$$

3.1.2 Stress concentration factor for bending in crankpin fillet

The stress concentration factor is calculated in accordance with the following formula:

$$a_B = 2,6914 \cdot f(s, w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H)$$

where:

$$f(s, w) = -4,1333 + 29,2004 \cdot w - 77,5925 \cdot w^2 + 91,9454 \cdot w^3 - 40,0416 \cdot w^4 + (1-s) \cdot (9,5400 - 58,3480 \cdot w + 159,3415 \cdot w^2 - 192,5846 \cdot w^3 + 85,2916 \cdot w^4) + (1-s)^2 \cdot (-3,3399 + 25,0444 \cdot w - 70,5571 \cdot w^2 + 87,0323 \cdot w^3 - 39,1832 \cdot w^4)$$

$$f(w) = 2,1790 \cdot w^{0,7171}$$

$$f(b) = 0,6840 - 0,2077 \cdot b + 0,1473 \cdot b^2$$

$$f(r) = 0,2031 \cdot r^{(-0,5231)}$$

$$f(d_G) = 0,9993 + 0,27 \cdot d_G - 1,0211 \cdot d_G^2 + 0,5306 \cdot d_G^3$$

$$f(d_H) = 0,9978 + 0,3145 \cdot d_H - 1,5241 \cdot d_H^2 + 2,4147 \cdot d_H^3$$

3.1.3 Stress concentration factor for bending in journal fillet

The stress concentration factors are calculated in accordance with the following formulae:

$$S_B = 2,7146 \cdot f_B(s, w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H)$$

where:

$$f_B(s, w) = -1,7625 + 2,9821 \cdot w - 1,5276 \cdot w^2 + (1-s) \cdot (5,1169 - 5,3029 \cdot w + 3,1391 \cdot w^2) + (1-s)^2 \cdot (-2,1567 + 2,3297 \cdot w - 1,2952 \cdot w^2)$$

$$f_B(w) = 2,2422 \cdot w^{0,7548}$$

$$f_B(b) = 0,5616 + 0,1197 \cdot b + 0,1176 \cdot b^2$$

$$f_B(r) = 0,1903 \cdot r^{(-0,5563)}$$

$$f_B(d_G) = 1,0012 - 0,6441 \cdot d_G + 1,2265 \cdot d_G^2$$

$$f_B(d_H) = 1,0012 - 0,1903 \cdot d_H + 0,0073 \cdot d_H^2$$

$$S_Q = 3,0123 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H)$$

where:

$$f_Q(s) = 0,4363 + 2,1630 \cdot (1-s) - 1,5212 \cdot (1-s)^2$$

$$f_Q(w) = \frac{w}{0,5557 + 0,5569 \cdot w}$$

$$f_Q(b) = -0,5 - b$$

$$f_Q(r) = 0,5331 \cdot r (-0,2033)$$

$$f_Q(d_H) = 0,9937 - 1,1949 \cdot d_H + 1,7373 \cdot d_H^2$$

3.1.4 Stress concentration factor for torsion in crankpin fillet

The stress concentration factor is calculated in accordance with the following formula:

$$\alpha_T = 0,923 \cdot f(r, s) \cdot f(b)$$

where:

$$f(r, s) = r(-0,322 + 0,1015 \cdot (1-s))$$

$$f(b) = 7,3955 - 10,654 \cdot b + 5,3482 \cdot b^2 - 0,357 \cdot b^3$$

3.1.5 Stress concentration factor for torsion in journal fillet

If the diameters and fillet radii of crankpin and journal are the same, then the stress concentration factor for torsion in journal fillet is the same as in crankpin fillet:

$$\beta_T = \alpha_T$$

If crankpin and journal diameters and/or radii are of different sizes, then the stress concentration factor is calculated with the following formula:

$$\beta_T = 0,923 \cdot f(r, s) \cdot f(b)$$

where $f(r, s)$ and $f(b)$ are to be determined in accordance with item 3.1.4, however, the radius of the journal fillet is to be referred to the journal diameter:

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

3.2 Stress concentration factors for crank throws with recessed fillets

The stress concentration factors applicable to crank throws with recessed fillets are obtained by multiplying the stress concentration factors for bending according to item 3.1.2 and 3.1.3 by a recess factor calculated by the following formula:

$$f = 1 + (t_H + t_G) \cdot (1,3 - 3,2 \cdot s)$$

where:

f [-] factor for the influence of recess

for other parameters see item 3.1.1.

The formula is valid if

$$r_H \leq R_H/D$$

$$r_G \leq R_G/D$$

and can be applied within the range

$$-0,3 \leq s \leq 0,3$$

All stress concentration factors according to item 3.1.2 and 3.1.3 are to be multiplied by the recess factor even if only one fillet is recessed.

4. Additional bending stresses

In addition to the alternating bending stresses in fillets (see item 2.1.3) further bending stresses due to misalignment and bedplate deformation are to be considered and added as additional bending stresses as given in the following Table:

Type of engine	σ_{add} [N/mm ²]	
	Misalignment	Bedplate deformation
2-stroke	≈ 20	≈ 10
4-stroke	≈ 20	0

where:

σ_{add} [N/mm²] additional stresses due to misalignment and/or bedplate deformation

Further additional stresses are likely to occur in the fillet as a result of bending vibrations and axial vibrations.

Bending vibrations require attention especially in the case of engine plants where the ratio of speed to natural bending frequency of the "flywheel/end crank system" is relatively high.

Where axial vibrations are concerned, those due to two types of excitation require attention:

The dominant excitation is due to the radial components of the connecting rod forces, which spread the crank throws and, whilst doing so, cause axial variations in length and bending stresses. Harmonic components of these forces, which excite an axial natural frequency of the system, result in additional bending stresses, which are not included in the alternating bending stress calculated in accordance with item 2.1.

Axial vibration amplitudes are also likely to occur as a result of torsional vibrations where twisting of the individual crank throws results in length variations, which add up to amplitudes at the free end which pulsate at twice the frequency of the torsional vibrations.

Axial vibrations of the first type can cause considerable additional forces in the crankshafts of large two-stroke engines. Contrary to torsional vibrations, there is at present no generally accepted method of calculation in use. However, this does not exonerate the engine manufacturer from the responsibility to make appropriate calculations.

5. Calculation of comparative alternating stress

5.1 General

The comparative alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet. For this calculation the deformation hypothesis (von Mises' Criterion) is to be used.

In this it is assumed that the maximum alternating bending stresses and maximum alternating torsional stresses within a crankshaft occur simultaneously and at the same point.

5.2 Comparative alternating stress

The comparative alternating stress is calculated in accordance with the formulae given.

For the crankpin fillet:

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

For the journal fillet:

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

where:

σ_v [N/mm²] comparative alternating stress

for other parameters see items 2.1.3, 2.2.3 and 4.

For the remainder of the calculation (see item 9), the larger of the two values is to be used.

6. Calculation of fatigue strength

The fatigue strength is to be understood as that value of alternating bending stress which a crankshaft can permanently withstand at the most highly stressed points of the fillets.

The allowable fatigue strength for a crankshaft can be calculated by the following formula:

$$\sigma_{DW} = \pm K \cdot [0,42 \cdot \sigma_B + 39,3] \cdot \left[0,264 + 1,073 \cdot D^{-0,2} + \frac{735 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{K_H}} \right]$$

where:

σ_{DW} [N/mm²] allowable fatigue strength of crankshaft

K [-] factor for different types of forged and cast crankshafts for which no surface treatment is done

= 1,05 for grain flow forged or drop-forged crankshafts

= 1,0 for free form forged crankshafts

= 0,93 for cast steel crankshafts

σ_B [N/mm²] minimum tensile strength of crankshaft material

for other parameters see item 3.1.1.

Where no results of the fatigue tests conducted on full size crank throws or crankshafts which have been subjected to surface treatment are available, the K-factors for crankshafts without surface treatment are to be used.

In each case the experimental values of fatigue strength carried out with full size crank throws or crankshafts are subject to special consideration of each Society.

7. Calculation of shrink-fits of semi-built crankshafts

7.1 General

All crank dimensions necessary for the calculation of the shrink-fit are shown in figure 3.

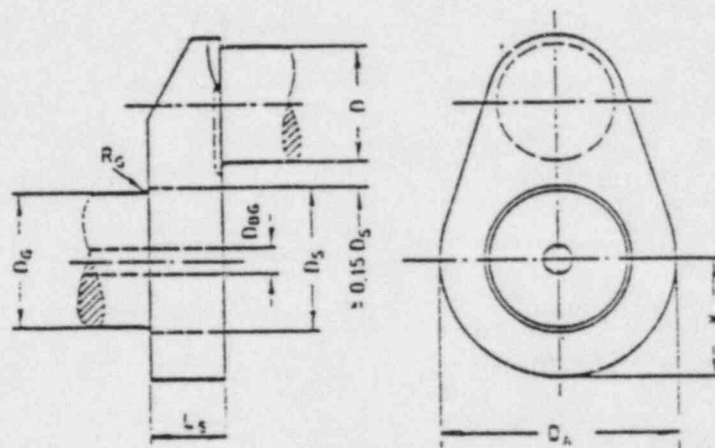


Fig. 3 Crank throw of semi-built crankshaft

where:

D_S [mm] shrink diameter

L_S [mm] length of shrink-fit

D_A [mm] outside diameter of web

or

twice the minimum distance x between centre-line of journals and outer contour of web, whichever is less

for other parameter see item 3.1.1.

For the radius of the transition from the journal to the shrink diameter, the following conditions must be maintained:

$$R_G \geq 0,015 D_G \text{ and } R_G \geq 0,5 (D_S - D_G)$$

The actual oversize Δ of the shrink-fit must be within the limits Δ_{\min} and Δ_{\max} , calculated in accordance with items 7.2 and 7.3.

7.2 Necessary minimum oversize of shrink-fit

The necessary minimum oversize is determined by the greater value calculated in accordance with items 7.2.1 and 7.2.2.

7.2.1 The calculation of the minimum oversize is to be carried out for the crank throw with the absolute maximum torque M_{\max} . The torque M_{\max} corresponds to the maximum value of the torque $M_{T\max}$ calculated as per item 2.2.2 for the various mass points of the crankshaft.

$$z_{\min} \geq \frac{4 \cdot 10^3}{\pi \cdot \mu} \cdot \frac{S_R \cdot M_{\max}}{E_m \cdot D_S \cdot L_S} \cdot \frac{1 - Q_A^2 \cdot Q_S^2}{(1 - Q_A^2) \cdot (1 - Q_S^2)}$$

with

$$Q_A = \frac{D_S}{D_A}, \quad Q_S = \frac{D_{3C}}{D_S}, \quad \mu = 0,20 \quad \text{for} \quad \frac{L_S}{D_S} \geq 0,40$$

where:

z_{\min} [mm] minimum oversize

S_R [-] safety factor against slipping, however a value not less than 2 is to be taken

Q_A, Q_S [-] ratio of different diameters

μ [-] coefficient for static friction

E_m [N/mm²] Young's modulus

7.2.2 In addition to item 7.2.1 the minimum oversize is also to be calculated according to the following formula:

$$z_{\min} \geq \frac{\sigma_S \cdot D_S}{E_m}$$

where:

σ_S [N/mm²] minimum yield strength of material for crank web or journal whatever is less.

7.3 Maximum permissible oversize of shrink-fit

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

$$z_{\max} \leq \frac{\sigma_S \cdot D_S}{E_m} + \frac{0,3 \cdot D_S}{1000}$$

where:

z_{\max} [mm] maximum oversize

This condition serves to restrict the shrinkage induced mean stress in the fillet.

Date: MARCH 5, 1984
To: G. E. RUSSELL
From: R. YANG
Subject: R-46 CRANKSHAFT

THE OBJECT OF THIS MEMO IS TO ILLUSTRATE WHY THE EXISTING CRANKSHAFT WITH 11" DIA. CRANKPIN IS PREFERABLE TO A 12" DIA. CRANKPIN AS USED FOR THE R-48. THE FOLLOWING COMPARISON WILL BE LIMITED TO ENGINES FOR DRIVING GENERATORS AT THE CURRENT RATED SPEED OF 450 RPM.

TORSIONAL CHARACTERISTICS

THE TYPICAL TORSIONAL SYSTEM OF A R-48 GENERATOR DRIVE HAVE EITHER NO COUNTERWEIGHTS OR ONE VERY SMALL COUNTERWEIGHT FOR EACH CRANK-THROW. THE REASON IS THAT THERE IS A FIRST MODE $\frac{1}{2}$ TH ORDER AND $\frac{1}{2}$ ORDER CRITICALS ABOVE THE RATED SPEED, MAKING IT MANDATORY TO HAVE A STIFF CRANKSHAFT WITH MINIMUM ROTATING INERTIA.

THE R-46 GENERATOR DRIVE ON THE OTHER HAND, REQUIRES TWO 12" COUNTERWEIGHTS PER CRANKTHROW TO POSITION THE FIRST MODE 6TH ORDER CRITICAL AT ABOUT 390 RPM. WITH A 12" DIA. CRANKPIN, THE SHAFT WILL BE STIFFER AND THEREFORE POSITION THE 6TH ORDER CRITICAL CLOSER TO 450 RPM (THERE IS NO ROOM AVAILABLE FOR LARGER COUNTERWEIGHTS).

TORQUE FLUCTUATION CALCULATED FROM NEAL SUPERPOSITION

FOR THE R-46 WE HAVE A RANGE OF 327759 FT.LB. (224240 TO -103519).
" " R-48 " " " " " " 503689 " " (312502 TO -191187).
CLEARLY THE R-46 IS STRESSED TO A MUCH LOWER TORQUE RANGE.

Memo *Handwritten signature*