RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D Machinery Installations

Rules for the Survey and Construction of Steel Ships
Part D 2006 AMENDMENT NO.3
Guidance for the Survey and Construction of Steel Ships
Part D 2006 AMENDMENT NO.2

Rule No.55 / Notice No.67 3rd October 2006
Resolved by Technical Committee on 6th July 2006
Approved by Board of Directors on 25th July 2006

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RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D Machinery Installations

RULES

2006 AMENDMENT NO.3

Rule No.55 3rd October 2006
Resolved by Technical Committee on 6th July 2006
Approved by Board of Directors on 25th July 2006

Rule No.55 3rd October 2006
AMENDMENT TO THE RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

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

Part D MACHINERY INSTALLATIONS

Amendment 3-1

Chapter 2 DIESEL ENGINES

2.3 Crankshafts

2.3.2 Built-up Crankshafts

Sub-paragraph -1 has been amended as follows.

1 The dimensions of crankpins and journals of built-up crankshafts are to comply with the followings.

(1) The diameters of crankpins and journals of built-up crankshafts are to comply with the requirements in **2.3.1-1**.

(2) The diameters of axial bores in journals of built-up crankshafts are to comply with the following formulae:

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

D_{BG} : Diameter of axial bore in journal (*mm*)
D_S : Journal diameter at the shrinkage fit (*mm*)
S_R : Safety factor against slipping (a value not less than 2 is to be taken)
M_{max} : Absolute maximum torque at the shrinkage fit (*N・mm*)
μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)
L_S : Length of shrinkage fit (*mm*)
σ_{SP} : Minimum yield strength of material for journal (N/mm^2)

EFFECTIVE DATE AND APPLICATION (Amendment 3-1)

1. The effective date of the amendments is 1 January 2007.
2. Notwithstanding the amendments to the Rules, the current requirements may apply to the crankshafts for which the application for design approval is submitted to the Society before the effective date.

Amendment 3-2

Chapter 9 BOILERS ETC. AND INCINERATORS

9.1 General

9.1.3 Drawings and Data to be submitted

Sub-paragraph (1)(k) has been renumbered to (l), and sub-paragraph (k) has been newly added as follows.

(k) Details of bursting disk (where fitted in accordance with the requirement in **9.9.3-12(4)**)

Sub-paragraph (2)(c) has been renumbered to (d), and sub-paragraph (c) has been newly added as follows.

(c) Operating instructions (shell type exhaust gas economizer only)

9.3 Design Requirements

9.3.5 Considerations for Installing

Sub-paragraph -3 has been newly added as follows.

3 Shell type exhaust gas economizer are to be installed as the tube plate to shell connection can be inspected easily.

Paragraph 9.3.6 has been amended as follows.

9.3.6 Protection against Flame

In case where part of the boiler drum and tube header is of the construction exposed to flames or high temperature gas, proper thermal insulation or other suitable means is to be provided thereto. For the shell type exhaust gas economizer, the insulation at the circumference of the tube end plate are to enable ultrasonic examination of the tube plate to shell connection.

9.9 Fittings, etc.

9.9.3 Safety Valves and Relief Valves

Sub-paragraph -11 has been amended as follows.

11 Where the economizer and exhaust gas economizer (including the heating element of the exhaust gas boiler) are equipped with an intervening valve between the boiler and the economizer or exhaust gas economizer, they are to be provided with at least one relief valve capable of discharging the quantity not less than that calculated from the maximum absorbable energy. However, the shell type exhaust gas economizer which has a total heating surface of 50 m^2 or more is to be provided with at least two relief valves.

Sub-paragraph -12 has been amended as follows.

12 The construction of safety valve and relief valve is to comply with the following requirements :

(1) The safety valve and the relief valve are to be so constructed that spring and valve are housed in a cage and they can not be overloaded intentionally from outside and that in case of spring failure they can not come out of their cage.

(2) The safety valve and the relief valve are to be fitted to the boiler shell, header, or outlet connection of the superheater by a flanged joint or welded joint. The valve chests are not to be made common to other valve chests. However, the safety valve of the superheater may be fitted with flanges to the distance pieces welded to the outlet connection.

(3) The safety valve and the relief valve are to be provided with an easing gears, and their handles are to be so arranged that they can be operated from an accessible place free from danger.

(4) The relief valves for shell type exhaust gas economizers are to incorporate features that ensure pressure relief even with solid matter deposits on the valve and guide, or features that prevent the accumulation of solid matter in way of the valve and the in the clearance between the valve spindle and guide. However, where no relief valves incorporating the features are fitted, a bursting disc discharging to suitable waste steam pipe is to be provided in addition to the valves. The alternative arrangements for ensuring pressure relief in the event of solid matter on the valve and guide are to function at a pressure not exceeding 1.25 times the economizer design pressure and are to have sufficient capacity to prevent damage to the economizer when operating at its design heat input level.

(5) The housings of the safety valve, relief valve and bursting disk are to be fitted with drainage arrangements from the lowest part, directed with continuous fall to a position clear of the boiler or exhaust gas economizer where it will not pose threats to either personnel or machinery. No valves or cocks are to be fitted in the drainage arrangements.

Sub-paragraph -13 has been amended as follows.

13 The waste steam pipes for safety valve and relief valve are to comply with the following requirements :

(1) The waste steam pipes for the safety valve and relief valve are to be of such construction that back pressure does not interfere with operation of the valves. The inside diameter of the waste steam pipes are not to be less than the diameter of the valve outlet, and are to

be designed at the pressure 1/4 or more of the set pressure of the valves.

(2) Where a common waste steam pipe is provided for two or more safety valves or relief valves, its cross sectional area is not to be less than the aggregate area of steam passage of each safety valve or relief valve. The waste steam pipes of boiler safety valves are to be separated from pipe lines likely to contain a large amount of drains such as steam blow-off pipes to the atmosphere or waste steam pipes of relief valves for exhaust gas economizer.

EFFECTIVE DATE AND APPLICATION (Amendment 3-2)

1. The effective date of the amendments is 1 January 2007.

2. Notwithstanding the amendments to the Rules, the current requirements may apply to ships for which the date of contract for construction* is before the effective date.

*“contract for construction” is defined in IACS Procedural Requirement(PR) No.29 (Rev.2).

IACS PR No.29 (Rev.2)

Unless specified otherwise:

1. The date of “contract for construction” of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of “contract for construction” of a series of sister vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.

For the purpose of this Procedural Requirement, a “series of sister vessels” is a series of vessels built to the same approved plans for classification purposes, under a single contract for construction. The optional vessels will be considered part of the same series of sister vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of “contract for construction” for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a “new contract” to which **1.** and **2.** above apply.

Notes:

1. This Procedural Requirement applies to all IACS Members and Associates.
2. This Procedural Requirement is effective for ships “contracted for construction” on or after 1 January 2005.
3. Sister vessels may have minor design alterations provided such alterations do not affect matters related to classification.
4. Revision 2 of this Procedural Requirement is effective for ships “contracted for construction” on or after 1 April 2006.

Amendment 3-3

Chapter 13 PIPING SYSTEMS

13.8 Sounding Pipes

Paragraph 13.8.6 has been newly added as follows:

13.8.6 Water Level Detection and Alarm Systems for Single Hold Cargo Ships

1 Cargo ships other than bulk carriers defined in **31A.1.2(1), Part C**, having a length (L_f) of less than 80 *m* and a single cargo hold below the freeboard deck or cargo holds below the freeboard deck which are not separated by at least one bulkhead made watertight up to that deck, are to be fitted in such space or spaces with water level detection and alarm systems in accordance with the following **(1)** to **(3)**.

(1) The water level detection and alarm systems are to give an audible and visual alarm at the navigation bridge when the water level above the inner bottom in the cargo hold reaches a height of not less than 0.3 *m*, and another when such level reaches not more than 15% of the mean depth of the cargo hold.

(2) The systems are to be fitted at the aft end of the hold, or above its lowest part where the inner bottom is not parallel to the designed waterline. Where webs or partial watertight bulkheads are fitted above the inner bottom, the fitting of additional detectors may be required.

(3) The systems are to have constructions and functions deemed appropriate by the Society.

2 Alarms given by the water level detection and alarm systems specified in **-1** are to be capable of identifying the space where the water level reaches the alarm level and the water level specified in **-1(1)** at the navigation bridge. The above alarms are also to be capable of being easily distinguishable from alarms given by other installations in the navigation bridge.

3 Manuals documented operating and maintenance procedures for the water level detection and alarm systems specified in **-1** are to be kept on board.

4 Notwithstanding the provisions of **-1**, the water level detection and alarm systems need not to be fitted in ships complying with the requirements of **13.8.5**, or in ships having watertight side compartments each side of the cargo hold length extending vertically at least from inner bottom to freeboard deck having a breadth deemed as appropriate by the Society.

Chapter 22 SPECIAL REQUIREMENTS FOR MACHINERY INSTALLED IN SHIPS WITH RESTRICTED AREA OF SERVICE AND SMALL SHIPS

22.2 Modified Requirements

22.2.3 Ships with a Gross Tonnage less than 500 *tons*, etc.

Sub-paragraph -2 has been amended as follows.

2 For ships which are not engaged on international voyages or whose gross tonnage is less than 500 *tons*, the requirements specified in **13.4.1-4** and **13.8.6** need not to apply.

EFFECTIVE DATE AND APPLICATION (Amendment 3-3)

1. The effective date of the amendments is 1 January 2007.

2. Notwithstanding the amendments to the Rules, the current requirements may apply to ships the keels of which were laid or which were at *a similar stage of construction* before the effective date.
(Note) The term "*a similar stage of construction*" means the stage at which the construction identifiable with a specific ship begins and the assembly of that ship has commenced comprising at least 50 *tonnes* or 1% of the estimated mass of all structural material, whichever is the less.

Amendment 3-4

Chapter 12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

12.1 General

12.1.5 Service Limitations for Materials

Sub-paragraph -2(3)(d) has been amended as follows.

(d) Valves fitted on the external wall of fuel oil tanks or lubrication oil tanks, and subjected to the static head of internal fluid.

Chapter 13 PIPING SYSTEMS

13.3 Sea Suction Valves and Overboard Discharge Valves

13.3.1 Sea Suction Pipe and Overboard Discharge Pipes Connections

The existing text has been amended as follows.

Sea inlet and overboard discharge pipes are to be connected to the valves or cocks which are fitted in accordance with the requirements in **13.3.2-2** and **-3**. However, for discharge pipes from positions above the freeboard deck, those having substantial wall thickness for the omission of a non-return valve in accordance with the provisions of **13.4.1-7**, up to appropriate level above the freeboard deck, need not to comply with the provisions of **13.3.2-3**.

EFFECTIVE DATE AND APPLICATION (Amendment 3-4)

1. The effective date of the amendments is 1 April 2007.
2. Notwithstanding the amendments to the Rules, the current requirements may apply to ships other than ships for which the application for Classification Survey during Construction is submitted to the Society on and after the effective date.

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D **Machinery Installations**

GUIDANCE

2006 AMENDMENT NO.2

Notice No.67 3rd October 2006
Resolved by Technical Committee on 6th July 2006

Notice No.67 3rd October 2006
AMENDMENT TO THE GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

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

Part D MACHINERY INSTALLATIONS

Amendment 2-1

D15 STEERING GEARS

D15.2 Performance and Arrangement of Steering Gears

D15.2.7 has been added as follows.

D15.2.7 Electrical Installations for Electric and Electrohydraulic Steering Gear

For steering gear circuits fed through an electronic inverter unit which makes steering gear turning speed control and their current are limited not more than the rated current of the electronic inverter, the requirement to provide a protection device against excess current specified in **15.2.7-6, Part D of the Rule** are exempted. In this case, they are to comply with the following requirements:

(1) The overload alarm specified in **15.2.7-5, Part D of the Rule** is to be set to a value not greater than the rated load of the electronic inverter.

(2) The over-current and over-voltage protection devices are to be provided in the electronic inverter unit. In case where the protection devices are operated, the audible and visual alarm is to be activated in a navigation bridge and at a position from which the main engine is normally controlled.

(3) The function to reduce the output power of the electronic inverter working before the protection devices specified in **(2)** above is to be provided in the electronic inverter unit, and where this function is operated, the audible and visual alarm is to be activated in a navigation bridge and at a position from which the main engine is normally controlled. However, in case where total failure of the semi-conductor rectifier cells is resulted within a short time, it is acceptable to cut off the output power of the electronic inverter unit.

EFFECTIVE DATE AND APPLICATION (Amendment 2-1)

1. The effective date of the amendments is 3 October 2006.

Amendment 2-2

D2 DIESEL ENGINES

D2.3 Crankshafts

D2.3.2 Built-up Crankshafts

Sub-paragraphs -1 and -2 have been renumbered to -2 and -3 respectively.

Sub-paragraph -1 has been added as follows.

1 The wording "maximum torque at the shrinkage fit" in **2.3.2-1(2), Part D of the Rules** means $M_{T\max}$ shown in **1.3.2-1 of the Annex D2.3.1-2(2)** "GUIDANCE FOR CALCULATON OF CRANKSHAFT STRESS II", in principle.

Sub-paragraph -2(2) has been amended as follows.

(2) In case where the maximum torque at the shrinkage fit is evaluated by carrying out a forced vibration calculation including the stern shaftings:

$$\alpha \geq \frac{4\times 10^3 S_R M_{T\max}\left(1-\frac{R^2}{r_s^2}\right)}{\pi\mu E d_h^2 t\left(1-\frac{1}{r_s^2}\right)\left(1-R^2\right)}$$

where :

$M_{T\max}$: Maximum torque at shrinkage fit, as shown in **1.3.2-1 of the Annex D2.3.1-2(2)** "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS** II"($N\cdot mm$)

E : Modulus of longitudinal elasticity (N/mm^2)

Other symbols are the same as those used in **2.3, Part D of the Rules**.

Annex D2.3.1-2(2) has been amended as follows.

Annex D2.3.1-2(2) GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS II

1.1 Scope

This Guidance applies to solid-forged and semi-built crankshafts of forged or cast steel, with one crankthrow between main bearings.

1.2 Principles of Calculation

The principles of calculation in this Guidance are as follows:

(1) The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas.

(2) The calculation is also based on the assumption that the areas exposed to highest stresses are those listed below. In addition, attention is to be paid to prevent excessive stress concentration in the outlets of journal oil bores.
   (a) Fillet transitions between the crankpin and web
   (b) Fillet transitions between the journal and web
   (c) Outlets of crankpin oil bores

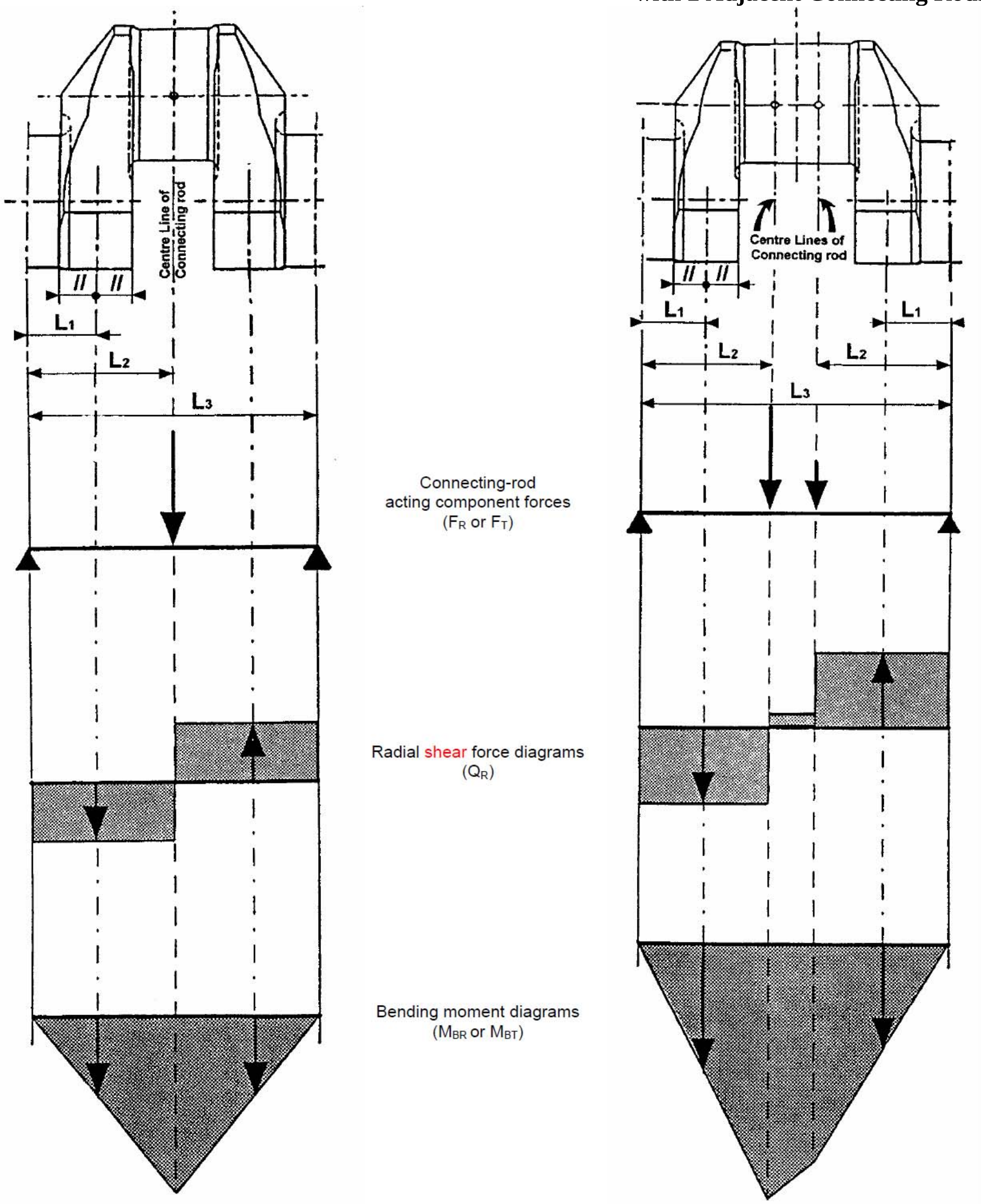
(3) The calculation of crankshaft strength consists initially in determining the nominal alternating bending (see **1.3.1**) and nominal alternating torsional stresses (see **1.3.2**) which, multiplied by the appropriate stress concentration factors (see **1.4**), result in an equivalent alternating stress (uni-axial stress) (see **1.6**).

(4) The equivalent alternating stress is evaluated in accordance with the followings.
   (a) 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.
   (b) 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.

(5) The equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (see **1.7**). The comparison will show whether or not the crankshaft concerned is dimensioned adequately (see **1.8**).

Fig.1 Crank Throw for In Line Engine

Fig.2 Crank Throw for Vee Type Engine with 2 Adjacent Connecting Rods

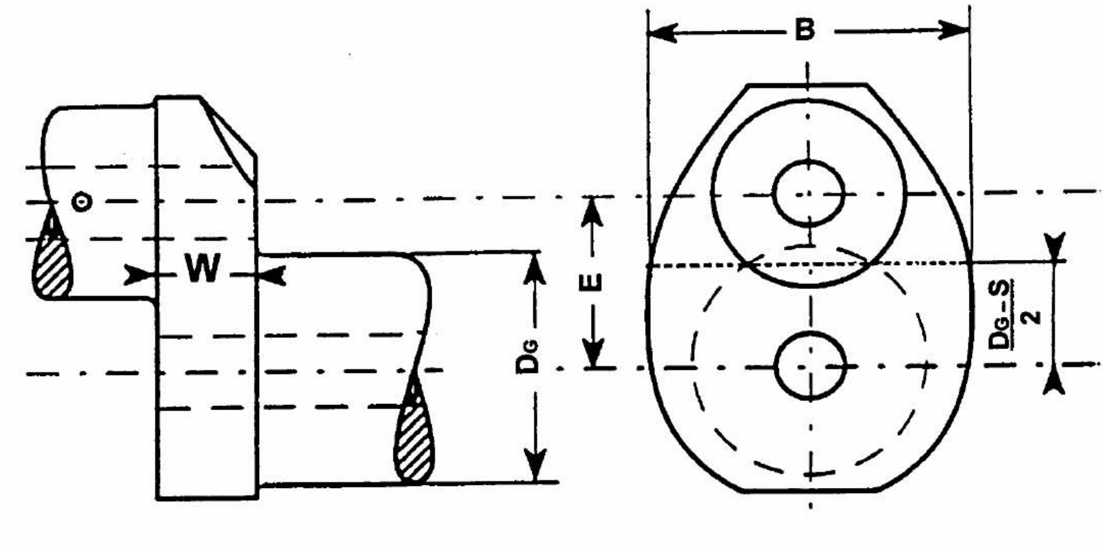


L_1 = Distance between main journal center line and crank web center (see also **Fig.3** for crankshaft without overlap)

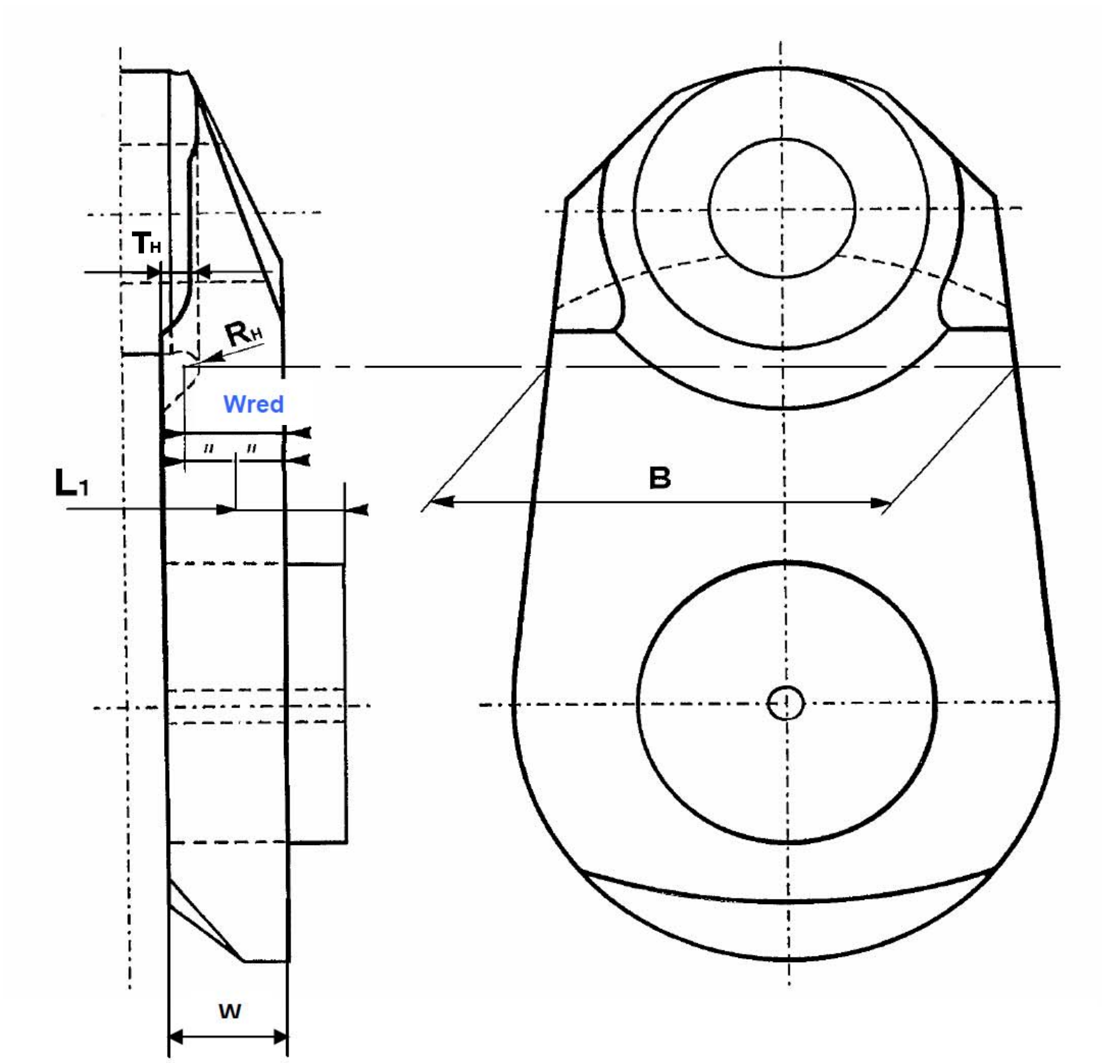
L_2 = Distance between main journal center line and connecting rod center

L_3 = Distance between two adjacent main journal center line

Fig.3　　Reference Area of Crank Web Cross Section

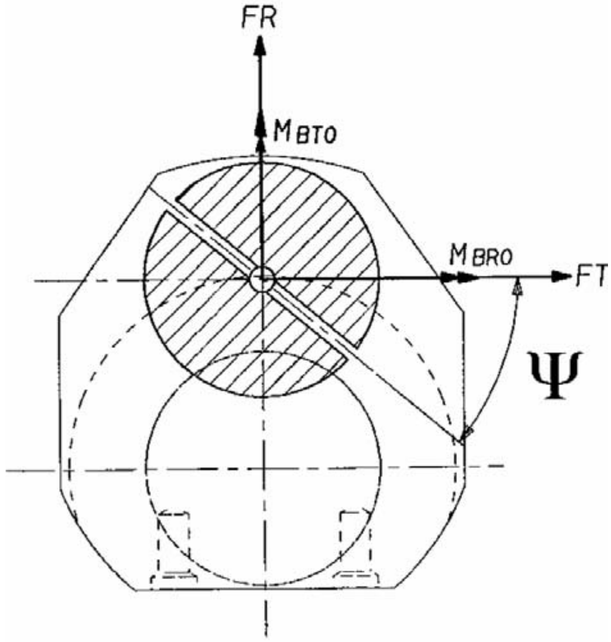


Overlapped crankshaft



Crankshaft without overlap

Fig.4 Crankpin Section through Oil Bore



M_{BRO} is the bending moment of the radial component of the connecting rod force.

M_{BTO} is the bending moment of the tangential component of the connecting rod force.

1.3 Calculation of Stresses

1.3.1 Alternating Bending Stress

1 Assumptions

The calculation of the alternating bending stress is based on the assumption as follows:

(1) The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces.

(2) The bending length is taken as the length between the two main bearing midpoints (distance L_3, see **Fig.1** and **Fig.2**).

(3) The bending moments M_{BR} and M_{BT} are calculated based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting rod force, respectively (see **Fig.1**).

(4) For crankthrows 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.2**).

(5) Bending moments and radial forces acting in web

(a) 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.

(b) 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.3**).

(c) Mean stresses are neglected.

(6) Bending moment acting in outlet of crankpin oil bore

(a) The two relevant bending moments are taken in the crankpin cross-section through the oil bore and are derived from the radial and tangential components of the connecting rod force (see **Fig.4**).

(b) The alternating stresses due to these bending moments are to be related to the cross-section of the axially bored crankpin.

(c) Mean bending stresses are neglected.

2 Nominal Alternating Bending and Compressive Stresses

(1) The calculation procedures are as follows:

(a) 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.

(b) Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments M_{BRF}, M_{BRO}, M_{BTO} and radial forces Q_{RF}, as defined in **-1(5)** and **(6)**, will then be calculated.

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

(d) Where there are cranks of different geometrical configuration in one crankshaft, the calculation is to cover all crank variants.

(2) Nominal alternating bending and compressive stresses in web cross-section

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

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

$$M_{BRFN} = \pm \frac{1}{2}\left(M_{BRF\max} - M_{BRF\min}\right)$$

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

where:

σ_{BFN} : Nominal alternating bending stress related to the web (N/mm^2)

W_{eqw} : Section modulus related to cross-section of web (mm^3)

Ke : Empirical factor considering to some extent the influence of adjacent crank and bearing restraint with:

Ke = 0.8 for 2-stroke engines

Ke = 1.0 for 4-stroke engines

M_{BRFN} : Alternating bending moment related to the center of the web ($N \cdot m$) (see **Fig.1** and **Fig.2**)

$M_{BRF\max}$: Maximum bending moment related to the center of the web within one working cycle ($N \cdot m$)

$M_{BRF\min}$: Minimum bending moment related to the center of the web within one working cycle ($N \cdot m$)

(b) The calculation of nominal alternating compressive stress is as follows:

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \cdot Ke$$

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

$$F = BW$$

where :

σ_{QFN} : Nominal alternating compressive stress due to radial force related to the web (N/mm^2)

Q_{RFN} : Alternating radial force related to the web (N) (see **Fig.1** and **Fig.2**)

$Q_{RF\max}$: Maximum radial force related to the web within one working cycle (N)

$Q_{RF\min}$: Minimum radial force related to the web within one working cycle (N)

F : Area related to cross-section of web (mm^2)

(3) Nominal alternating bending stress in outlet of crankpin oil bore

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

$$\sigma_{BON} = \pm \frac{M_{BON}}{We} \cdot 10^3$$

$$M_{BON} = \pm \frac{1}{2}\left(M_{BO\max} - M_{BO\min}\right)$$

$$We = \frac{\pi}{32}\left(\frac{D^4 - D_{BH}^4}{D}\right)$$

where :

σ_{BON} : Nominal alternating bending stress related to the crankpin diameter (N/mm^2)

M_{BON} : Alternating bending moment calculated at the outlet of crankpin oil bore ($N \cdot m$)

$M_{BO\max}$: Maximum value of bending moment M_{BO} within one working cycle ($N \cdot m$)

$M_{BO\min}$: Minimum value of bending moment M_{BO} within one working cycle ($N \cdot m$)

M_{BO} : Bending moment acting in outlet of crankpin oil bore ($N \cdot m$)

$$M_{BO} = \left(M_{BTO} \cdot \cos\psi + M_{BRO} \sin\psi\right)$$

ψ : Angular position (see **Fig.4**)

We : Section modulus related to cross-section of axially bored crankpin (mm^3)

D , D_{BH} : see **1.4.1**

3 Alternating Bending Stresses in Fillets and Outlet of Crankpin Oil Bore

(1) The calculation of the alternating bending stress in the crankpin fillet is as follows:

$$\sigma_{BH} = \pm\left(\alpha_B \cdot \sigma_{BFN}\right)$$

where:

σ_{BH} : Alternating bending stress in crankpin fillet (N/mm^2)

α_B : Stress concentration factor for bending in crankpin fillet (determination-see **1.4.2**)

(2) The calculation of the alternating bending stress in the journal fillet (not applicable to semi-built crankshafts) is as follows:

$$\sigma_{BG} = \pm\left(\beta_B \cdot \sigma_{BFN} + \beta_Q \cdot \sigma_{QFN}\right)$$

where:

σ_{BG} : Alternating bending stress in journal fillet (N/mm^2)

β_B : Stress concentration factor for bending in journal fillet (determination-see **1.4.3**)

β_Q : Stress concentration factor for compression due to radial force in journal fillet (determination-see **1.4.3**)

(3) The calculation of the alternating bending stress in the outlet of crankpin oil bore (only applicable to radially drilled oil hole)is as follows:

$$\sigma_{BO} = \pm\left(\gamma_B \cdot \sigma_{BON}\right)$$

where:

σ_{BO} : Alternating bending stress in outlet of crankpin oil bore (N/mm^2)

γ_B : Stress concentration factor for bending in crankpin oil bore (determination - see **1.4.4**)

1.3.2 Alternating Torsional Stresses

1 Nominal Alternating Torsional Stresses

The calculation for nominal alternating torsional stresses is to be undertaken according to the followings to specify the maximum nominal alternating torsional stress. The values received from such calculation are to be submitted to the Society.

(1) The maximum and minimum torques are to 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.

(2) Whilst doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring, which is defined as cylinder condition when no combustion occurs but only compression cycle, in one of the cylinders).

(3) The speed step calculation is to be selected in such a way that any resonance found in the operational speed range of the engine is to be detected.

The nominal alternating torsional stress in every mass point results from the following equation:

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

$$M_{TN} = \pm \frac{1}{2}\left(M_{\mathrm{Tmax}} - M_{\mathrm{Tmin}}\right)$$

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

where:

τ_N : Nominal alternating torsional stress referred to crankpin or journal (*N/mm*^{2})

W_P : Polar section modulus related to cross-section of axially bored crankpin or bored journal (*mm*^{3})

M_{TN} : Maximum alternating torque (*N*・*m*)

$M_{T\max}$: Maximum torque (*N*・*m*)

$M_{T\min}$: Minimum torque (*N*・*m*)

D, D_{BH}, D_{BG}, D_G : See **1.4.1**

Where barred speed ranges are necessary as the torsional stresses exceed the allowable limit τ_1 in **8.2.1** or **8.2.3-1, Part D of the Rules**, τ_1 is to be taken as the values of the torsional stresses within the range for the calculation.

In this connection, the barred speed ranges are to be so arranged that satisfactory operation is possible despite of their existence in accordance with **8.2.5** and **8.3.1, Part D of the Rules**.

2 Alternating Torsional Stresses in Fillets and Outlet of Crankpin Oil Bore

(1) The calculation of the alternating torsional stresses in the crankpin fillet is as follows:

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

where:

τ_H : Alternating torsional stress in crankpin fillet (*N/mm*^{2})

α_T : Stress concentration factor for torsion in crankpin fillet (determination-see **1.4.2**)

τ_N : Nominal alternating torsional stress referred to crankpin diameter (*N/mm*^{2})

(2) The calculation of the alternating torsional stresses in the journal fillet (not applicable to semi-built crankshafts) is as follows:

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

where:

τ_G : Alternating torsional stress in journal fillet (*N/mm*^{2})

β_T : Stress concentration factor for torsion in journal fillet (determination-see **1.4.3**)

τ_N : Nominal alternating torsional stress referred to journal diameter (*N/mm*^{2})

(3) The calculation of the alternating stresses in the outlet of crankpin oil bore due to torsion (only applicable to radially drilled oil hole) is as follows:

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

where:

σ_{TO} : Alternating stress in outlet of crankpin oil bore due to torsion (*N/mm*^{2})

γ_T : Stress concentration factor for torsion in outlet of crankpin oil bore (determination-see **1.4.4**)

τ_N : Nominal alternating torsional stress referred to crankpin diameter (*N/mm*^{2})

1.4 Stress Concentration Factors

1.4.1 Explanation of Terms and Symbols

1 The terms used in this **1.4** are defined as follows:

(1) The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (Von Mises), occuring in the fillets under bending load, to the nominal bending stress related to the web cross-section.

(2) The stress concentration factor for compression (β_Q) in the journal fillet is defined as the ratio of the maximum equivalent stress (Von Mises), occuring in the fillet due to the radial force, to the nominal compressive stress related to the web cross-section.

(3) The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum equivalent shear stress, occuring in the fillets under torsional load, to the nominal torsional stress related to the axially bored crankpin or journal cross-section.

(4) The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress, occuring at the outlet of the crankpin oil bore under bending and torsional loads, to the corresponding nominal stress related to the axially bored crankpin cross-section.

2 The symbols used in this **1.4** mean as follows (see **Fig.5**):

D : Crankpin diameter (*mm*)

D_{BH} : Diameter of axial bore in crankpin (*mm*)

D_O : Diameter of oil bore in crankpin (*mm*)

R_H : Fillet radius of crankpin (*mm*)

T_H : Recess of crankpin fillet (*mm*)

D_G : Journal diameter (*mm*)

D_{BG} : Diameter of axial bore in journal (*mm*)

R_G : Fillet radius of journal (*mm*)

T_G : Recess of journal fillet (*mm*)

E : Pin eccentricity (*mm*)

S : Pin overlap (*mm*)

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

W : Web thickness (mm)

In the case of 2 stroke semi-built crankshafts with $T_H > R_H$, the web thickness is to be considered as equal to:

$W_{red} = W - (T_H - R_H)$ (refer to **Fig.3**)

B : Web width (mm)

In the case of 2 stroke semi-built crankshafts, the web width is to be taken in way of crankpin fillet radius center according to **Fig.3**.

$r = R_H / D$ (in crankpin fillet), R_G / D (in journal fillet) $(0.03 \le r \le 0.13)$

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

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

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

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

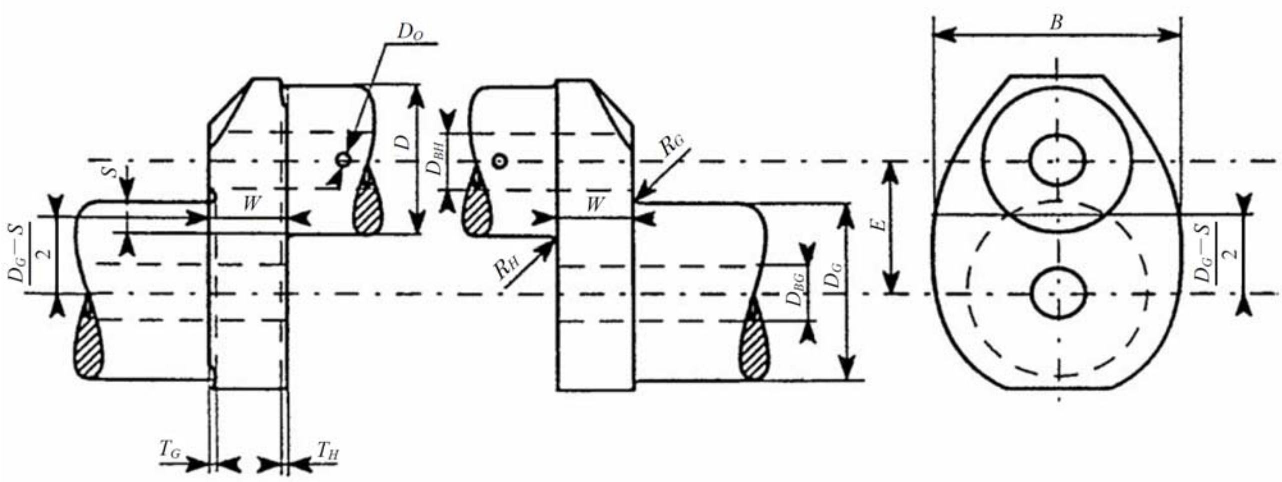
$d_G = D_{BG} / D \quad (0 \le d_G \le 0.8)$

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

$t_H = T_H / D$

$t_G = T_G / D$

Fig.5 Crank Dimensions



1.4.2 Stress Concentration Factors in Crankpin Fillet

1 The stress concentration factor for bending (α_B) is:

$$\alpha_B = 2.6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(recess)$$

where:

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

If $s < -0.5$ then $f(s,w)$ is to be evaluated replacing actual value of s by -0.5.

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

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

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

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

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

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

If calculated $f(recess) < 1$ then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

2 The stress concentration factor for torsion (α_T) is:

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

where:

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

If $s < -0.5$ then $f(r,s)$ is to be evaluated replacing actual value of s by -0.5.

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

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

1.4.3 Stress Concentration Factors in Journal Fillet

1 The stress concentration factor for bending (β_B) is:

$\beta_B = 2.7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(recess)$

where:

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

$f_B(w) = 2.2422 \cdot w^{0.7548}$

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

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

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

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

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

If calculated $f(recess) < 1$ then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

2 The stress concentration factor for compression (β_Q) due to the radial force is:

$\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(recess)$

where:

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

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

$f_Q(b) = -0.5 + b$

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

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

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

If calculated $f(recess) < 1$ then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

3 The stress concentration factor for torsion (β_T) is:

$\beta_T = \alpha_T$ if the diameters and fillet radii of crankpin and journal are the same.

$\beta_T = 0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$ if crankpin and journal diameters and/or radii are of different sizes.

where:

$f(r,s)$, $f(b)$ and $f(w)$ are to be determined in accordance with **1.4.2** (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter:

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

1.4.4 Stress Concentration Factors in Outlet of Crankpin Oil Bore

1 The stress concentration factor for bending (γ_B) is:

$$\gamma_B = 3 - 5.88 \cdot d_O + 34.6 \cdot d_O{}^2$$

2 The stress concentration factor for torsion (γ_T) is:

$$\gamma_T = 4 - 6 \cdot d_O + 30 \cdot d_O{}^2$$

1.5 Additional Bending Stresses

In addition to the alternating bending stresses in fillets (σ_{BH} and σ_{BG}) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying σ_{add} as follows:

σ_{add} = ± 30 *N/mm*^{2} for crosshead engines

= ± 10 *N/mm*^{2} for trunk piston engines

1.6 Equivalent Alternating Stress

1.6.1 Equivalent Alternating Stress in Crankpin Fillet

The equivalent alternating stress in the crankpin fillet is calculated in accordance with the followings.

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

where:

σ_v : Equivalent alternating stress (*N/mm*^{2})

for other parameters see **1.3.1-3**, **1.3.2-2** and **1.5**.

1.6.2 Equivalent Alternating Stress in Journal Fillet

The equivalent alternating stress in the journal fillet is calculated in accordance with the followings.

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

for parameters see **1.6.1**.

1.6.3 Equivalent Alternating Stress in Outlet of Crankpin Oil Bore

The equivalent alternating stress in the outlet of crankpin oil bore is calculated in accordance with the followings.

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

for parameters see **1.6.1**.

1.7 Fatigue Stength

1.7.1 Fatigue Strength in Crankpin Fillet

The fatigue strength in the crankpin fillet is evaluated in accordance with the followings. For calculation purpose, R_H is to be taken as not less then 2 *mm*.

$$\sigma_{DW} = \pm K\left[0.42\sigma_B + 39.3\right]\times\left[0.264 + 1.073D^{-0.2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_H}}\right]$$

where:

σ_{DW} : Allowable fatigue strength of crankshaft (*N/mm*^{2}) where the surfaces 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) are smoothly finished

K : Factor for different types of crankshafts without surface treatment
= 1.05 for continuous grain flow forged or drop-forged crankshafts
= 1.0 for free form forged crankshafts (without continuous grain flow)
Factor for cast steel crankshafts with cold rolling treatment in fillet area
= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society
As an alternative, the value of K can be determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow.

σ_B : Minimum tensile strength of crankshaft material (*N/mm*^{2})

for other parameters see **1.4**

1.7.2 Fatigue Strength in Journal Filet

The fatigue strength in the journal fillet is evaluated in accordance with the followings. For calculation purpose, R_G is to be taken as not less then 2 *mm*.

$$\sigma_{DW} = \pm K\left[0.42\sigma_B + 39.3\right]\times\left[0.264 + 1.073D^{-0.2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_G}}\right]$$

for parameters see **1.7.1**

1.7.3 Fatigue Strength in Outlet of Crankpin Oil Bore

The fatigue strength in the outlet of crankpin oil bore is evaluated in accordance with the followings. For calculation purpose, $D_O/2$ is to be taken as not less then 2 *mm*.

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

K : Factor for forged crankshafts without surface treatment
= 1.0
Factor for cast steel crankshafts with cold rolling treatment in fillet area
= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society
As an alternative, the value of K can be determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow.

for other parameters see **1.7.1**

1.8 Acceptability Criteria

The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. The acceptability factors of the crankpin fillet, the journal fillet and the outlet of crankpin oil bore are to comply with the following criteria:

$$Q \geq 1.15$$

where

Q : Acceptability factor

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

EFFECTIVE DATE AND APPLICATION (Amendment 2-2)

1. The effective date of the amendments is 1 January 2007.

2. Notwithstanding the amendments to the Guidance, the current requirements may apply to the crankshafts for which the application for design approval is submitted to the Society before the effective date.

Amendment 2-3

D9 BOILERS ETC. AND INCINERATORS

Section D9.1 has been newly added as follows.

D9.1 General

D9.1.3 Drawings and Data to be submitted

Operating Instructions specified in **9.1.3(2)(c) of the Rules** are to include the following information:

(1) Feed water treatment and sampling arrangements
(2) Operating temperatures (exhaust gas and feed water temperatures)
(3) Operating pressure
(4) Inspection and cleaning procedures
(5) Records of maintenance and inspection
(6) The need to maintain adequate water flow through the economizer under all condition
(7) Periodical operational checks of the safety devices to be carried out by the operating personnel and to be documented accordingly
(8) Procedures for using the exhaust gas economizer in the dry condition
(9) Procedures for maintenance and overhaul of the relief valves

D9.9 Fittings, etc.

D9.9.3 Safety Valves and Relief Valves

In sub-paragraph -2, the wording “safety valve” has been amended to “safety valve or relief valve”.

EFFECTIVE DATE AND APPLICATION (Amendment 2-3)

1. The effective date of the amendments is 1 January 2007.

2. Notwithstanding the amendments to the Guidance, the current requirements may apply to ships for which the date of contract for construction* is before the effective date.
*"contract for construction" is defined in IACS Procedural Requirement(PR) No.29 (Rev.2).

IACS PR No.29 (Rev.2)

Unless specified otherwise:

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of "contract for construction" of a series of sister vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.
For the purpose of this Procedural Requirement, a "series of sister vessels" is a series of vessels built to the same approved plans for classification purposes, under a single contract for construction. The optional vessels will be considered part of the same series of sister vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.

Notes:

1. This Procedural Requirement applies to all IACS Members and Associates.
2. This Procedural Requirement is effective for ships "contracted for construction" on or after 1 January 2005.
3. Sister vessels may have minor design alterations provided such alterations do not affect matters related to classification.
4. Revision 2 of this Procedural Requirement is effective for ships "contracted for construction" on or after 1 April 2006.

Amendment 2-4

D13 PIPING SYSTEMS

D13.8 Sounding Pipes

D13.8.5 Water Level Detection and Alarm Systems for Bulk Carriers and Ore Carriers

The reference resolution of “Resolution MSC.145(77)” in sub-paragraph -3 has been replaced with “Resolution MSC.188(79)”.

Sub-paragraph -3(3) has been amended as follows.

(3) Electrical installations for the systems installed in the following areas are to be of an intrinsically safe type of *Exib* and the maximum surface temperature of the installations is not to exceed 85℃, except those for a ship designed only for the carriage of cargoes which can not create a combustible or explosive atmosphere. Where a ship is designed only for the carriage of limited kinds of cargoes, the maximum surface temperature may be relaxed appropriately depending on the kinds of cargoes. Such limitations relating to cargoes are to be documented in the booklet for cargo operations. Those fitted in a boundary of the following areas are to be at the discretion to the Society, upon consideration of the design regarding gas-tightness, etc.
- (a) Cargo holds
- (b) Enclosed spaces adjacent to a cargo hold having an opening without a gas-tight or watertight door/hatch and the like into the hold
- (c) Areas within 3 *m* of any cargo hold mechanical exhaust ventilation outlet

Sub-paragraph -3(6) has been amended as follows.

(6) The systems are to be of a type continuously monitoring the system itself including detectors, and an audible and visual alarm is to be activated when detecting a fault on the systems. For the purpose of this regulation, such fault monitoring is to address faults associated with the system, e.g. open circuit, short circuit, loss of power supplies, CPU failure. The audible alarm is to be capable of being muted but the visual alarm is to remain active until the malfunction is cleared and not to be capable of being turned off by manual operations. In addition, the systems are to have means of testing function of detectors in-situ when the hold is empty.

Sub-paragraphs -5 and -6 have been renumbered to -6 and -7 respectively, sub-paragraph -5 has been added as follows.

5 With respect to the provisions of **13.8.5-2, Part D of the Rules**, one sensor capable of detecting both preset levels specified in **13.8.5-1(1)(a)** and **(b)**, **Part D of the Rules** may be allowed.

Sub-paragraph -6 has been newly amended as follows.

6 The wording “override devices deemed as appropriate by the Society” in **13.8.5-3, Part D of the Rules** means those complying with the following requirements:

(1) the alarm for each tank/cargo hold is to be capable of stopping independently;

(2) a visual override indication is to be given to the navigation bridge throughout deactivation of the water level detectors for the tanks/cargo holds;

(3) the override devices are to be arranged so that the alarm system is reactivated upon completion of de-ballasting automatically; and

(4) where the water level detection and alarm system is of a type enabling the facility to override alarm and to be customized for each specific ship, an override function for spaces other than ballast tanks nor cargo holds designed for the carriage of water ballast is to be modified so as to be ineffective, when the installation to the ship. The above modification and any subsequent modifications are to be subject to the confirmation by the Surveyor. A caution plate which prohibits personnel from overriding such alarm is not an acceptable alternative to the above modification to be ineffective.

The reference resolution of “Resolution MSC.145(77)” in sub-paragraph -7(2) has been replaced with “Resolution MSC.188(79)”.

Paragraph D13.8.6 ha s been newly added as follows.

D13.8.6 Water Level Detection and Alarm Systems for Single Hold Cargo Ships

1 Water level detection and alarm systems required in **13.8.6, Part D of the Rules** are in accordance with the provisions of **D13.8.5**.

2 For the purpose of the provisions of **13.8.6-3, Part D of the Rules**, “having a breadth deemed as appropriate by the Society” means that the distance between the side shell and the inner shell in any part of the watertight compartments is not less than 760 *mm* measured perpendicular to the side shell.

EFFECTIVE DATE AND APPLICATION (Amendment 2-4)

1. The effective date of the amendments is 1 January 2007.

2. Notwithstanding the amendments to the Guidance, the current requirements may apply to ships the keels of which were laid or which were at *a similar stage of construction* before the effective date.

(Note) The term “*a similar stage of construction*” means the stage at which the construction identifiable with a specific ship begins and the assembly of that ship has commenced comprising at least 50 *tonnes* or 1% of the estimated mass of all structural material, whichever is the less.

Amendment 2-5

D13 PIPING SYSTEMS

D13.2 Piping

D13.2.5 Bulkhead Valves

Sub-paragraph -3 has been added as follows.

3 The requirements for pipes piercing the collision bulkhead specified in **13.2.5-1 and -2, Part D of the Rules** apply to the extent below the bulkhead deck. In the extension part of the collision bulkhead above the bulkhead deck to be made weathertight in accordance with the provisions of **13.1.5(2), Part C of the Rules**, for pipes piercing such part and being open to enclosed space(s) arranged after the such bulkhead, a non-return valve is to be fitted on the after side of the bulkhead..

D13.3 Sea Suction Valves and Overboard Discharge Valves

Paragraph D13.3.1 has bee added as follows.

D13.3.1 Sea Suction Pipe and Overboard Discharge Pipes Connections

The term of "the extent up to appropriate level above the freeboard deck" as specified in **13.3.1, Part D of the Rules** is to be in accordance with the provisions of **D13.4.1-3**.

D13.4 Scuppers and Sanitary Discharges

D13.4.1 General

Sub-paragraph -3 has been added as follows.

3 With respect to the provisions of **13.4.1-7, Part D of the Rules**, the extent of pipes to have substantial wall thickness in accordance with **Table D12.6(1)** and **Table D12.6(2)**, may be limited to:

(1) an extent up to the freeboard deck, where the vertical distance from the load line to the freeboard deck is not less than 600 *mm*; or

(2) an extent up to the deck just above the freeboard deck, where such distance is less than 600 *mm*.

EFFECTIVE DATE AND APPLICATION (Amendment 2-5)

1. The effective date of the amendments is 1 April 2007.

2. Notwithstanding the amendments to the Guidance, the current requirements may apply to ships other than ships for which the application for Classification Survey during Construction is submitted to the Society on and after the effective date.