18-410

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 2018 AMENDMENT NO.1
Guidance for the Survey and Construction of Steel Ships
Part D 2018 AMENDMENT NO.1

Rule No.100 / Notice No.52 29 June 2018
Resolved by Technical Committee on 31 January 2018

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NIPPON KAIJI KYOKAI

An asterisk (*) after the title of a requirement indicates that there is also relevant information in the corresponding Guidance.

RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

RULES

Part D Machinery Installations

2018 AMENDMENT NO.1

Rule No.100 29 June 2018

Resolved by Technical Committee on 31 January 2018

An asterisk (*) after the title of a requirement indicates that there is also relevant information in the corresponding Guidance.

Rule No.100 29 June 2018
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 1-1

Chapter 13 PIPING SYSTEMS

13.6 Air Pipes

Paragraph 13.6.1 has been amended as follows.

13.6.1 General

1 All tanks and cofferdams are to be provided with air pipes having sufficient cross sectional areas to permit easy venting from any part of the tank and cofferdam.
2 Tanks having top plates not less than 7 *meters* either in length or in width are to be provided with two or more pipes arranged a suitable distance apart. However, tanks having inclined top plates may be provided with one air pipe located at the highest part of the top plate.
3 For tanks requiring more than one air pipe, overflow pipes which comply with requirement **13.7.2** may be used in lieu of air pipes as long as proper air flow from the tank to the atmosphere is ensured; all tanks, however, are to be provided with at least one air pipe.
~~3~~4 In cases where tanks or cofferdams are of a complicated profile, special consideration is to be given to the number and positions of all air pipes.
~~4~~5 Air pipes are to be arranged to be self-draining.
~~5~~6 Vent pipes for fuel oil service, settling and lubrication oil tanks are to be located and arranged so that, in cases where such pipes break, there is no direct risk of any ingress of seawater or rainwater.

EFFECTIVE DATE AND APPLICATION (Amendment 1-1)

1. The effective date of the amendments is 29 June 2018.
2. Notwithstanding the amendments to the Rules, the current requirements 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.
3. Notwithstanding the provision of preceding **2.**, the amendments to the Rules 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 upon request by the owner.

Amendment 1-2

Chapter 14 PIPING SYSTEMS FOR TANKERS

14.6 Tests

14.6.2 Tests after Installation On Board

Sub-paragraph -2 has been amended as follows.

2 Heating pipes inside cargo oil tanks are to be subjected to leak tests at pressures of 1.5 times or greater their design pressures. <u>However, the test pressure is to be at least 0.4 *MPa* or more.</u>

EFFECTIVE DATE AND APPLICATION (Amendment 1-2)

1. The effective date of the amendments is 29 June 2018.
2. Notwithstanding the amendments to the Rules, the current requirements apply to the tests for which the application is submitted to the Society before the effective date.

Amendment 1-3

Chapter 18 AUTOMATIC AND REMOTE CONTROL

18.1 General

18.1.1 Scope*

Sub-paragraph -3 has been amended as follows.

3 ~~In cases where machinery and equipment which are deemed necessary by the Society use computer based systems for the proper achievement of their functions, the design, construction, commissioning and maintenance of such computer~~ Computer based systems, including the hardware and software which constitute such systems, are to be in accordance with requirements specified otherwise by the Society in addition to those specified in **-1** and **-2** above and throughout the rest of this chapter for design, construction, commissioning, maintenance, etc.

Sub-paragraph -4 has been added as follows.

4 The requirement in **-3** above is not applicable to equipment mentioned below:
(1) navigating equipment specified in the **Rules for Safety Equipment**,
(2) radio installations specified in the **Rules for Radio Installations**,
(3) stability instruments, and
(4) loading computers.

Paragraph 18.1.2 has been amended as follows.

18.1.2 Terminology*

Terms used in this Chapter are defined as follows:
((1) to (9) are omitted.)
(10) A system is defined as a combination of interacting programmable devices and/or sub-systems organized to achieve one or more specified purposes.
(11) A computer based system is defined as a system which provides control, alarm, monitoring, safety or internal communication functions and depends upon software for the proper achievement of these functions.
(12) A sub-system is defined as an identifiable part of a system, which may perform a specific function or set of functions.
(13) A programmable device is defined as a physical component where software is installed.
(~~10~~14) A safety system is defined as a system which operates automatically, in order to prevent damage to machinery and equipment in cases where serious impediments to functioning should occur during their operation so that one of the following actions will take place:
(a) Starting of stand-by machinery or equipment
(b) Reduction of output of machinery or equipment
(c) Shutting off fuel or power supplies, thereby stopping the machinery or equipment

18.2 System Design

18.2.6 Safety Systems*

Sub-paragraph -1 has been amended as follows.

1 ~~Constitution of~~ Independence of safety systems

~~Constitution~~ Independence of safety systems is to comply with the following requirements:

((1) and (2) are omitted.)

Paragraph 18.2.7 has been added as follows.

18.2.7 Use of Computers*

1 The reliability and maintainability of computer based systems are not to be inferior to those of systems not relying upon computers.

2 Control systems, alarm systems and safety systems which constitute computer based systems are to comply with the following **(1)** to **(3)**:

(1) Requirements for computers

- (a) The composition of computers is to be so planned that the extent of impact on the system as a whole of any failure in any part of a circuit or component is to be minimized as far as possible.
- (b) Each component is to be protected against any possibility of overvoltage (electronic noise) which may originate from input or output terminals.
- (c) Central processing units and important peripheral devices are to have self-monitoring functions.
- (d) Important programs and data are to be ensured against loss in cases where an external electrical power supply may be temporarily interrupted.
- (e) Computers are to be set up so they can be quickly re-started following planned procedures within a short period of time after electrical power has been restored after a power failure.
- (f) Spare parts for all important elements which require special techniques for repair work are to be kept in ample supply for easy replacement.
- (g) Change-over to back-up means is to be able to be performed easily and soundly.

(2) Back-up means

- (a) In cases where one computer simultaneously performs fuel control (governor control, electronic injection control, etc.) and remote control of main propulsion machinery in diesel or turbine ships, or output control (rotational speed control, load control, etc.) and remote control of main propulsion machinery in electric propulsion ships, one of the following systems is to be provided in the case of computer failure. However, where this requirement is impracticable, relevant systems are to comply with requirements deemed appropriate by the Society.
  - i) Stand-by computer
  - ii) Governor-controlled back-up systems operated at the main control station
- (b) Safety systems are to be provided with back-up means which can be used in a timely manner in the event of the failure of the computer in service.
  - i) Stand-by computer
  - ii) Safety systems that do not rely on computers
- (c) In cases where visual display units (VDU) are adopted as indicators for the alarm

systems stipulated in this chapter, at least two VDUs are to be installed, or other arrangements deemed appropriate by the Society are to be considered.

(3) Independence

Independence of computerized control systems and safety systems is to comply with the requirements in **18.2.4-1** and **18.2.6-1** respectively, except in cases where their constitution are comply with requirements specified below.

(a) In cases where secondary control systems or stand-by computers are installed for those control systems, the independence of such control systems may not be required for individual machinery or equipment. In such cases, local control equipment fitted to main propulsion machinery in accordance with the requirements given in **18.3.2-3(2)** are not regarded as secondary control systems.

(b) In cases where safety systems conform to the requirement given in **(2)(b)** above, the independence of individual machinery and equipment in systems, and their independence from other systems may not be required, notwithstanding the requirements in **18.2.6-1**.

(c) In cases where secondary systems or stand-by computers are installed in both control systems and safety systems, the independence of individual machinery and equipment in their systems including alarm systems, and their independence from the other systems may not be required.

18.3 Automatic and Remote Control of Main Propulsion Machinery or Controllable Pitch Propellers

Paragraph 18.3.3 has been amended as follows.

18.3.3 Bridge Control Devices *

Bridge control devices are to comply with the following ~~requirements~~ **(1)** through **(4)** as well as ~~those~~ requirements in **18.3.2**.

((1) and (2) are omitted.)

(3) Bridge control devices are to be provided with visual and audible alarms which give the officer in charge of the navigational watch enough time to assess navigational circumstances in an emergency before the safety systems of main propulsion machinery specified in **18.1.2(~~10~~14)(b)** or **(c)** go into effect.

(4) Bridge control devices are to be provided with an override arrangement specified in **18.2.6-3** for the following safety systems of main propulsion machinery:

(a) Safety systems ~~Systems~~ which perform as specified in **18.1.2(~~10~~14)(b)**~~.~~

(b) Safety systems ~~Systems~~ which perform as specified in **18.1.2(~~10~~14)(c)**~~.~~ (except in cases where the total failure of main propulsion machinery will occur within a short period of time.)

EFFECTIVE DATE AND APPLICATION (Amendment 1-3)

1. The effective date of the amendments is 29 June 2018.

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

Amendment 1-4

Chapter 16 has been amended as follows.

Chapter 16 WINDLASSES AND MOORING WINCHES

16.1 General

16.1.1 Scope

~~1~~ The requirements herein **Chapter 16** apply to ~~those~~ windlasses and mooring winches ~~that are driven by electric power, hydraulic power or steam~~.

~~**2** Any windlasses and mooring winches other than those specified in **-1** are to be subject to Society approval.~~

16.1.2 Terminology

Terms used in this chapter are defined as follows:

(1) "Prime mover" means electric motors, hydraulic motors, steam turbines and so on which drive cable lifters.

(2) "Torque-transmitting components" means components which transmit power from the prime movers to cable lifters when anchors and chain cables are paid out or hoisted; for example, shafts, gears, clutches, couplings and coupling bolts, etc. (includes components which constitute prime movers)

(3) "Load-bearing components" means components which are loaded; this, however, excludes torque-transmitting components such as shaft bearings, cable lifters, sheaves, drums, bed-frames, brakes, chain cable stoppers and foundations, etc.

16.2 Windlasses

16.2.1 General*

1 Windlasses fitted to the ship in order to handle anchors are to be suitable for the size of chain cable being used.

2 The design, construction and testing of windlasses are to conform to a standard or code of practice recognized by the Society in addition to requirements in this chapter. The standard or code of practice is to specify criteria for stress, performance and testing.

16.2.~~1~~2 Drawings and Data*

~~Drawings and data listed below~~ The following drawings and data showing design specifications, standards of compliance, engineering analyses and details of construction, are, in principle, to be submitted.

(1) Drawings and data for approval:

(a) ~~Particulars and those internationally recognized standards being applied~~ Windlass design specifications

(b) ~~General arrangement~~ Windlass arrangement plan

(c) ~~Material specification of essential parts~~ Dimensions, materials and welding details of torque-transmitting components and load-bearing components

(d) Drawings and data concerning hydraulic systems

(e) Control, monitoring and instrumentation arrangements

(f) Procedures for shop tests
(~~d~~g) Other drawings and data considered necessary by the Society

(2) ~~Data~~ Drawing and data for reference:
~~(a) Production test procedures~~
(~~b~~a) Calculated strength for ~~essential parts~~ torque-transmitting components and load-bearing components
(b) General arrangements and sectional assembly drawings of chain cable stoppers and documents which demonstrate the chain cable stoppers are in accordance with requirements specified in **16.2.4-2(6)** (in cases where chain cable stoppers are fitted)
(c) Load calculations of prime movers (in cases where the load test specified in **16.2.5-1(3)** is not carried out)
(d) Calculation sheets for cable lifter brake capacities (in cases where the cable lifter brake capacity test specified in **16.2.5-1(4)** is not carried out)
(e) Operation and maintenance procedures
(~~e~~f) Other drawings and data considered necessary by the Society

16.2.3 Materials and Fabrication*

1 Materials

Materials used in the construction of torque-transmitting and load-bearing components of windlasses are to comply with the following requirements:

(1) The materials are to be approved by the Society in accordance with the requirements in **Part K of the Rules**, except in cases where the material is comply with standards recognized by the Society.
(2) The Proposed materials are to be indicated in the drawings and data specified in **16.2.2(1)**.
(3) All such materials are to be certified by the material manufacturers and are to be traceable to the manufacturers' certificates.

2 Welded fabrication

Welded fabrication is to comply with the following requirements:

(1) Weld joint designs, the degree of non-destructive examination of welds and post-weld heat treatment, if any, are to be indicated in the drawings and data specified in **16.2.2(1)**.
(2) Welding procedures and welders are to be qualified in accordance with requirements in standards recognized by the Society.
(3) Welding consumables are to be type-approved by the Society in accordance with the requirements in **Part M of the Rules**.

16.2.~~2~~4 ~~Construction and Performance~~ Design*

1 Windlasses and their beds as well as any other accessories and facilities are to be installed effectively and securely onto the deck.

2 Mechanical designs of windlasses are to be according to the following requirements:

(1) Design loads are to comply with the following requirements:
(a) Holding loads
Calculations are to be made to show that, in the conditions specified in **i)** and **ii)** below, the maximum stress for each load bearing component do not exceed yield strength (or 0.2% proof stress) of the material.
i) The holding condition (single anchor, cable lifter brake fully applied and cable lifter declutched)
ii) Under a load equal to 80% of the specified breaking test load of the chain cable (For installations fitted with a chain cable stopper, 45% of the specified breaking test load of the chain cable may instead be used for the calculation.)
(b) Inertia loads

Designs for drive trains (including prime movers, reduction gears, bearings, clutches, shafts, cable lifters and bolting) are to consider the dynamic effects of the sudden stopping and starting of the prime movers or chain cables so as to limit inertial loads.

~~**1** Windlasses are to be capable of continuous operation over a period of 30 *minutes* while exerting the working load defined in **(1)** and at least two *minutes* overload pull defined in **(2)**.~~

~~(1) The working load is to be decided depending on grade of chain cables below.~~

(2) The continuous duty pull is to be decided in accordance with the following requirements:

(a) Prime movers are to be able to exert, for at least 30 minutes, a continuous duty pull corresponding to the grade and diameter of chain cable as follows:

i) Maximum anchorage depth is to be not deeper than 82.5 *m* for the following windlasses:

~~(a~~1) those using Grade 1 chain cables: $Z_{cont1}=37.5d^2$ (*N*) ($3.82\,d^2$ (*kgf*))

~~(b~~2) those using Grade 2 chain cables: $Z_{cont1}=42.5d^2$ (*N*) ($4.33\,d^2$ (*kgf*))

~~(c~~3) those using Grade 3 chain cables: $Z_{cont1}=47.5d^2$ (*N*) ($4.84\,d^2$ (*kgf*))

where

Z_{cont1}: the continuous duty pull

d ~~is~~: the nominal diameter of chain cable (*mm*)

ii) Maximum anchorage depth is to be deeper than 82.5 *m* for the following windlasses:

$$Z_{cont2}(N)=Z_{cont1}(N)+(D-82.5)\times 0.27d^2$$

$$\left(Z_{cont2}(kgf)=Z_{cont1}(kgf)+(D-82.5)\times 0.0275d^2\right)$$

where

Z_{cont2}: the continuous duty pull

d: the nominal diameter of chain cable (*mm*)

D: the maximum anchorage depth (*m*)

(b) In general, the stresses in each torque-transmitting component are not to exceed 40% of the yield strength (or 0.2% proof stress) of the material when the continuous duty pull is loaded.

~~(2) Overload is to be 1.5 times of working load.~~

(3) Prime movers are to be able to provide the necessary temporary overload capacity for breaking out the anchor. This temporary overload capacity or "short term pull" is to be at least 1.5 times the continuous duty pull applied for at least 2 *minutes*. The speed in this period may be lower than that specified in **(4)**.

(4) The mean speed of the chain cable during hoisting of the anchor and chain cable is to be at least 0.15 *m/s* when the windlass hoists over two shots of chain cable and initially with at least three shots of chain cable (82.5 *m*) with the anchor submerged and hanging free.

~~2~~(5) ~~The windlass is~~ Windlasses are to be fitted with ~~a brake~~ cable lifter brakes of capacities sufficient to stop ~~for the safe stopping of~~ the anchor and the chain cable when paying out the chain cable. Such brakes are to produce torques capable of withstanding the following loads without any permanent deformation of strength members and without brake slip. ~~Such a brake is to be capable of holding the following loads sufficiently.~~

(~~1~~a) with a chain cable stopper: 0.45 × the breaking test load of chain cable

(~~2~~b) without a chain cable stopper: 0.80 × the breaking test load of chain cable

~~**3** Windlasses and their beds as well as any other accessories and facilities are to be installed effectively and securely onto the deck.~~

(6) Chain cable stoppers, if fitted, along with their attachments are to be designed to withstand, without any permanent deformation, 80% of the specified minimum breaking strength of the chain cable.

(7) Hull supporting structures of windlasses and chain cable stoppers are to be according to the

following requirements:

(a) Hull supporting structures of windlasses and chain cable stoppers are to comply with the requirements specified in **Chapter 27, Part C** or **Chapter 23, Part CS**.

~~4~~(b) For those ships of 80 *m* or more in length L_1 that are specified in **15.2.1-1, Part C**, all windlass mounts on an exposed deck over the forward $0.25L_1$ line are to be of sufficient strength in cases where the height of the exposed deck in way of the item is mounted is less than $0.1L_1$ or 22 *m* above the designed maximum load line, whichever is lesser.

~~5~~(c) The strength of any above deck framing and hull structure supporting a windlass and its securing bolt is to be according to the requirements in **10.7.1, Part C** or **10.6.1, Part CS**.

3 Hydraulic systems where employed for driving windlasses are to comply with the requirements specified in other chapters of this Part in addition to those in this chapter.

4 Electrical systems (e.g., electric motors and electrical circuits) are to comply with the requirements specified in **Part H** in addition to those in this chapter.

5 The following protections are to be provided:

(1) Suitable protection systems which limit the speed and torque of the prime mover to protect mechanical parts, including component housings such as over pressure protection devices, slipping clutches between electric motors and gearing, torque limiting devices (for electrically driven windlasses only)

(2) Means to contain debris consequent to any severe damage to the prime mover due to over-speed in the event of uncontrolled rendering of the cable; for example, covers, particularly when an axial-piston-type hydraulic motor is used as the prime mover

(3) Devices or parts necessary for the safety of users; for example, covers for any exposed gearing, covers for any hot surfaces of steam cylinders, etc.

~~**6** Prime movers and gearing are to be provided with the following devices and parts for protection against excessive torque or shock and for safety on board.~~

~~(1) Over pressure protection devices~~

~~(2) Slipping clutches between electric motors and gearing~~

~~(3) Torque limiting devices (electrically driven windlasses only)~~

~~(4) Covers for any exposed gearing~~

~~(5) Covers for any hot surfaces of steam cylinders~~

~~**7** Windlasses are to be capable of lifting chain cables (3 links or more in length) and anchors being hoisted from the sea at a mean lifting speed of at least 0.15 *m/s*.~~

6 Windlasses are to be fitted with couplings which are capable of disengaging between cable lifters and drive shafts. Hydraulically or electrically operated couplings are to be capable of being disengaged manually.

7 Windlasses are to be permanently marked with the following information:

(1) Nominal size of chain cable; for example, 100/3/45 means the nominal diameter of 100 *mm* and grade 3, with a holding load of 45% of the breaking test load.

(2) Maximum anchorage depth (*m*)

16.2.~~3~~5 Tests*

1 Shop Tests

For conformance with approved plans, windlasses are to be inspected during fabrication at the manufacturer facilities and acceptance tests, as specified in the specified standard of compliance and including at least **(1)** to **(4)** below, are to be carried out in the presence of the Surveyor.

(1) Pressure tests

Before assembly, hydrostatic pressure tests are to be carried out for the following components in accordance with the requirements in **12.6.1** ~~**of Part D**~~. The test pressure is to be 1.5 times ~~the~~ designed pressure. However, the test pressure for steam cylinders may be 1.5 times ~~the~~ working pressure.

(a) Housings with covers for hydraulic motors and pumps
(b) Hydraulic pipes
(c) Valves and fittings
(d) Pressure vessels
(e) Steam cylinders

(2) No-load tests

Windlasses are to be run without loads at nominal speed for 15 *minutes* in each direction (for a total of 30 *minutes*). In cases where the windlass is provided with a gear change, additional runs in each direction for 5 *minutes* at each gear change are required.

(3) Load tests

Windlasses are to be tested to verify that the continuous duty pull, overload capacity and hoisting speed as specified in **16.2.4-2** can be attained. In case where the manufacturing works does not have adequate facilities, the following **(a)** or **(b)** may be complied instead of the load tests:

(a) Submission of the documents specified in **16.2.2(2)(c)**.
(b) To carry out the load tests, including adjustments of overload protection, on board ship. In such cases, functional testing at manufacturer works is to be performed under no-load conditions.

(4) Cable lifter brake capacity tests

It is to be verified that the holding power of the cable lifter brake complies with **16.2.4-2.(5)** either through testing or submission of the calculation sheet specified in **16.2.2(2)(d)**.

~~(2) Load tests, overload tests, operation tests and chain wheel brake tests are to be carried out on windlasses (including their prime movers). However, in cases where a windlass has been recognized by the Society, such tests may be omitted.~~

~~(3) In cases where a windlass has been given Society approval, part of or all of the tests required in above **(2)** may be omitted.~~

2 (Omitted)

16.3 Mooring Winches

16.3.1 Structure, etc.

(-1 and -2 are omitted.)

3 Mounts of mooring winches which are integrated with windlasses are to be in accordance with the requirements in ~~**16.2.2-4**~~**16.2.4-2(7)(b)** and ~~**-5**~~**(c)**.

EFFECTIVE DATE AND APPLICATION (Amendment 1-4)

1. The effective date of the amendments is 1 July 2018.

2. Notwithstanding the amendments to the Rules, the current requirements apply to windlasses for which the application for approval is submitted to the Society before the effective date and that are installed on ships for which the date of contract for construction* is before the effective date.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

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 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, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:

(1) such alterations do not affect matters related to classification, or

(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.

The optional vessels will be considered part of the same series of 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.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D **Machinery Installations**

GUIDANCE

2018 AMENDMENT NO.1

Notice No.52 29 June 2018
Resolved by Technical Committee on 31 January 2018

Notice No.52 29 June 2018
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 1-1

D5 POWER TRANSMISSION SYSTEMS

D5.3 Strength of Gears

Paragraph D5.3.1 has been amended as follows.

D5.3.1 Application

In the case of bevel gear, the wording "deemed appropriate by the Society" in **5.3.1, Part D of the Rules** means as follows:

(1) The bending stress at the root sections of gear teeth and limiting tooth surface stress are to be according to ISO~~AGMA~~ standards or as deemed appropriate by the Society.

(2) Strength of the interior of gear teeth
The Vickers hardness (*HV*) of the interior of gear teeth is not to be less than the value calculated by the following formula. However, this requirement does not apply to bevel gears for which the tip diameter (outer end) is smaller than 1,100 *mm*:

If $\frac{z}{w} < 0.79$ then $\frac{z}{w}$ is to be taken as 0.79.

$$HV = 1.11 S_H p \left[\frac{z}{w} - \frac{\left(\frac{z}{w}\right)^2}{\sqrt{1+\left(\frac{z}{w}\right)^2}} \right]$$

HV : Vickers hardness

S_H : Safety factor for contact stress, is to comply with the requirements in **Annex D5.3.5 "GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS" 1.6.3-9.**

p : Real hertzian stress (*MPa*). The upper limit of the value of p used in this calculation is to be 1500 *MPa.*

$p = AS_c$

S_c: Contact stress (*MPa*), to be calculated according to *ISO 10300*~~ANSI/AGMA 2003~~ standards.

A : If S_c is calculated according to *ISO 10300*~~ANSI/AGMA 2003~~ standards, then the coefficients are to be determined, in consideration of analysis results, by the Society on a case by case basis. In addition, if S_c is calculated according to *ISO*

~~ANSI/AGMA 2003-A86~~10300 standards, A is to taken as ~~1.7~~1.32

w: Half the hertzian contact width (mm), to be calculated by the following formula:

$$w = \frac{p\rho_C}{56300}$$

$$\rho_C = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$$\rho_1 = 0.5 d_{vn1} \sin\alpha_n$$

$$\rho_2 = 0.5 d_{vn2} \sin\alpha_n$$

$$d_{vn1} = d_{m1} \frac{\sqrt{1+u^2}}{u} \frac{1}{\cos^2 \beta_{vb}}$$

d_{m1} : Mean pitch diameter of pinion （mm）

u : Gear ratio

$$\beta_{vb} = \arcsin(\sin\beta_m \cos\alpha_n)$$

β_m : Mean spiral angle

α_n : Normal pressure angle

$$d_{vn2} = u^2 d_{vn1}$$

z : Depth from teeth surface to evaluation point (mm)

EFFECTIVE DATE AND APPLICATION (Amendment 1-1)

1. The effective date of the amendments is 29 June 2018.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to bevel gears for which the date of application for approval is before the effective date.

Amendment 1-2

D12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

D12.1 General

D12.1.6 Use of Special Materials

Sub-paragraph -5 has been deleted.

~~**5** "Where specially approved by the Society" stipulated in **12.1.6-2(3)(a) of the Rules** refers to the use of materials such as Teflon or nylon which are unable to be reinforced. However, the hoses are to have external wire braid protection as practicable.~~

D12.3 Construction of Valves and Pipe Fittings

D12.3.4 Flexible Hose Assemblies

Sub-paragraph -3 has been added as follows.

<u>**3** "Where specially approved by the Society" stipulated in **12.3.4-3(3)(a) of the Rules** refers to the use of materials such as Teflon or nylon which are unable to be reinforced. However, the hoses are to have external wire braid protection as practicable.</u>

D12.6 Tests

D12.6.1 Shop Tests

Sub-paragraph -1(1) has been amended as follows.

1 Testing of pipe joints of a butt welded type and pipe joints of a slip-on sleeve welded type (such as elbows, reducers, tees, bends and sockets, etc.)

(1) Materials and tests of pipe joints of a butt welded type and pipe joints of a slip-on sleeve welded type used for Group I or II pipes are to be in accordance with the following:

(a) Materials

i) Materials for pipe joints are to comply with the requirements in **Part K** (see **D1.1.4(7)**).

ii) Notwithstanding the requirement given in **i)**, materials complying with <u>international or</u> ~~a~~ national standard<u>s</u> such as <u>*ISO*,</u> *JIS*<u>, etc.</u> may be used for pipe joints for which hot forming or heat treatment is carried out during the manufacturing process, provided that they receive approval of use from the Society in accordance with **Chapter 12, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

((b) to (e) are omitted.)

EFFECTIVE DATE AND APPLICATION (Amendment 1-2)

1. The effective date of the amendments is 29 June 2018.

Amendment 1-3

D18 AUTOMATIC AND REMOTE CONTROL

D18.1 General

Paragraph D18.1.1 has been amended as follows.

D18.1.1 Scope

1 (Omitted)

~~**2** The "machinery and equipment which are deemed necessary by the Society" referred to in **18.1.1-3, Part D of the Rules** means machinery and equipment used for the purposes specified in **(1)** to **(4)** given below and includes programmable controllers such as sequencers.~~

~~(1) Control systems for the machinery and equipment specified in **18.1.1-1, Part D of the Rules**.~~

~~(2) Alarm systems specified in **18.2.5, Part D of the Rules**.~~

~~(3) Safety systems for the machinery and equipment specified in **18.1.1-1, Part D of the Rules**.~~

~~(4) Control, alarm, and safety systems related to **Table 2.1 of Annex D18.1.1 "COMPUTER BASED SYSTEMS"**.~~

~~**3** Notwithstanding the requirements in **-2** above, the "machinery and equipment which are deemed necessary by the Society" referred to in **18.1.1-3, Part D of the Rules** is not to include the machinery and equipment specified in the following **(1)** to **(4)**:~~

~~(1) navigating equipment specified in the **Rules for Safety Equipment**,~~

~~(2) radio installations specified in the **Rules for Radio Installations**,~~

~~(3) stability instruments, and~~

~~(4) loading computers.~~

~~4~~2 The "requirements specified otherwise by the Society" referred to in **18.1.1-3, Part D of the Rules** means **Annex D18.1.1** "**COMPUTER BASED SYSTEMS**"

Paragraph D18.1.2 has been added as follows.

D18.1.2 Terminology

The computer based system referred to in **18.1.2(11), Part D of the Rules** includes a system which contains programmable controllers such as sequencers.

D18.2 System Design

Paragraph D18.2.7 has been added as follows.

D18.2.7 Use of Computers

1 "The extent of impact on the system as a whole of any failure in any part of a circuit or component is to be minimized as far as possible" specified in **18.2.7-2(1)(a), Part D of the Rules** means, for example, that in a system always controlled by two or more computers, the system is able to cope with the failure of one computer without hindering overall performance.

2 "Deemed appropriate by the Society" specified in **18.2.7-2(2)(a), Part D of the Rules** means that the results of a failure analysis such as FMEA on the system are satisfactory and approved by the Society.

3 "Other arrangements deemed appropriate by the Society" specified in **18.2.7-2(2)(c), Part D of the Rules** means, for example, the combination of a VDU and an alarm printer.

Annex D18.1.1 COMPUTER BASED SYSTEMS

Chapter 1 INTRODUCTION

1.1 General

Paragraph 1.1.1 has been amended as follows.

1.1.1 Scope

The requirements in this annex ~~relate to the design, construction, commissioning and maintenance of computer based systems which depend upon software to automatically or remotely control the machinery and equipment fitted with such systems, and~~ apply to computer based systems, including the hardware and software which constitute such systems, ~~the software as well as any hardware supporting such software for said machinery and equipment~~ in accordance with **18.1.1-3, Part D of the Rules**.

Section 1.2 has been amended as follows.

1.2 Submission of Drawings and Data

The following drawings and data are, in principle, to be submitted. In cases where deemed necessary by the Society, other drawings and data may be required. However, no submission is required for category I systems unless it is specifically requested by the Society.

(1) Drawings and data for approval:
- (a) quality plan (**3.1.1-3**),
- (b) test programs and procedures for functional tests and failure tests in integration testing before installation on board (**3.1.3**),
- (c) test program for simulation tests for final integration (**3.1.5-1**),
- (d) test program for on board tests (includes tests related to wireless data links) (**3.1.5-2, -3** and **5.2.2(3)**), and
- (e) test reports of environmental tests according to ~~UR E10~~ **18.7.1(1), Part D of the Rules** (**3.1.4** and **Chapter 4**).

(2) Drawings and data for reference:
- (a) documents related to quality systems (includes security policies specified in **3.4.1-1**);
- (b) risk assessment report or justification for the omission of risk assessment (**3.1.2-1**);
- (c) documents related to software code creation and testing ~~such as the following~~ (**3.1.2-2**)~~:~~
  - ~~i) software module functional description and associated hardware description,~~
  - ~~ii) evidence of verification of software code, and~~
  - ~~iii) evidence of functional tests for elements at the level of software module, sub-system and system~~;
- (d) other drawings and data concerning systems such as the following (**3.1.3-4(3)**):
  - i) functional description of software;
  - ii) list and versions of software installed in system;
  - iii) user manual including instructions for use during software maintenance;
  - iv) list of interfaces between system and other vessel systems;
  - v) list of standards used for data links; and
  - vi) documentation which might include a FMEA or equivalent to demonstrate

~~demonstrating~~ the adequacy of failure test (in cases where deemed necessary by the Society).

Chapter 2 DEFINITIONS

2.1 Stakeholders

Paragraph 2.1.1 has been amended as follows.

2.1.1 Owner

The owner is responsible for contracting the system integrator and/or suppliers regarding the provision of a hardware system, including software, according to the owner's specification. The owner may be the "ship builder integrator" (builder or shipyard) during initial construction. After vessel delivery, the owner may delegate some responsibilities to the vessel operating company.

Paragraph 2.1.2 has been amended as follows.

2.1.2 System Integrator

At ship construction, the ~~The~~ role of the system integrator is to be taken by the shipyard unless an alternative ~~organisation~~ organization is specifically contracted or assigned this responsibility.

The system integrator is responsible for the integration of systems and products provided by suppliers into the system subject to the requirements specified herein and for providing the integrated system. The system integrator may also be responsible for integration of the systems in the vessel.

If there are multiple parties performing system integration at any one time, then a single party is to be responsible for overall system integration and coordinating the integration activities. If there are multiple stages of integration, then different system integrators may be responsible for the specific stages of integration; in such cases, however, a single party is to be responsible for defining and coordinating all of the stages of integration.

Paragraph 2.1.3 has been amended as follows.

2.1.3 Supplier

The supplier is any contracted or subcontracted provider of system components or software under the coordination of the system integrator or shipyard. The supplier is responsible for providing software, programmable devices, sub-systems or systems to the system integrator. The supplier is to provide a description of the software functionality which meets the owner's specification, applicable international and national standards, and the requirements specified herein.

Section 2.2 has been amended as follows.

2.2 Objects

2.2.1 Object Definitions

1 **Fig. 2.1** shows the hierarchy and relationships of a typical computer based system.

Fig. 2.1 (Omitted)

~~2.2.1 Object Definitions~~

~~1~~**2** "Vessel" is the ship or offshore unit where the system is to be installed.

~~2~~**3** "System", "sub-system" and "programmable device" are as specified in **18.1.2, Part D of the Rules.** ~~is a combination of interacting programmable devices and/or sub-systems organized to achieve one or more specified purposes.~~

~~**3** "Sub-system" is identifiable part of a system, which may perform a specific function or set of functions.~~

~~**4** "Programmable device" is the physical component where software is installed.~~

~~5~~**4** "Software module" is a standalone piece of code which provides specific and closely coupled functionality.

Sections 2.3 and 2.4 have been renumbered to Paragraph 2.2.2 and Section 2.3, and Section 2.3 has been amended as follows.

~~2.3~~2.2.2 System Categories

Systems are typically assigned category I, II or III as shown in **Table 2.1** based upon their effect upon system functionality. The exact category, however, is dependent upon the risk assessment for all operational scenarios.

Table 2.1 (Omitted)

2.~~4~~3 Other Terminology

2.~~4~~3.1 Simulation Tests

~~A simulation test is~~ "Simulation test" is a control system testing where the equipment under control is partly or fully replaced with simulation tools, or where parts of the communication network and lines are replaced with simulation tools.

Chapter 3 REQUIREMENTS FOR SOFTWARE AND SUPPORTING HARDWARE

Section 3.1 has been amended as follows.

3.1 Life Cycle Approach

A global top-to-bottom approach is to be undertaken regarding software and its integration into a system, ~~and is to span the software lifecycle~~ spanning the software lifecycle. This approach is to be accomplished according to software development standards as listed herein or other standards recognized by the Society.

3.1.1 Quality System

1 System integrators and suppliers are to operate a quality system regarding software development and testing and associated hardware such as *ISO* 9001 ~~which takes~~ taking into account *ISO* 90003.

2 Satisfaction of the requirement specified in **-1** above is to be demonstrated through either of the following **(1)** or **(2)**:

(1) The quality system ~~is~~ being certified as compliant to the recognized standard by an ~~organisation~~ organization with accreditation under a national accreditation scheme~~.~~ , or

(2) ~~Verification of~~ The quality system being confirmed compliance with a ~~standards~~ recognized standard by the Society through a specific assessment.

3 The quality system specified in **-1** above is to include a quality plan documenting the items listed in the following **(1)** to **(4)**:

((1) and (2) are omitted.)

(3) For category II and III systems, the information specified in the following **(a)** to **(c)**:

  (a) Specific procedures for verification of software code at the level of systems, sub-systems and programmable devices and modules,

  (b) Drawings and data submitted for the ~~Class~~ Society and tests witnessed by the Surveyor, and

  (c) Specific procedures for software modification and installation on board the vessel defining interactions with owners.

(4) (Omitted)

3.1.2 Design Phase

1 Risk assessments of systems are to be according to the following **(1)** to **(4)**:

((1) to (3) are omitted.)

(4) In cases where the risks associated with a computer based system are well understood, it is permissible for the risk assessment to be omitted; in such cases, however, the supplier or the system integrator is to provide a justification for the omission. The justification is to give consideration to the following **(a)** to **(c)**:

  (a) ~~Means to know the risks~~ How the risks are known.

  (b) The equivalence of the context of use of the current computer based system and the computer based system initially used to determine the risks.

  (c) The adequacy of existing control measures in the current context of use.

2 (Omitted)

3.1.3 Integration Testing before Installation On Board

1 Intra-system integration testing including functional and failure tests is to be done between system and sub-system software modules before being integrated on board. The objective is to

check the following **(1)** to **(3)**:
(1) the software functions are properly executed,
(2) the software and the hardware it controls interact and function properly together, and
(3) the software systems react properly in the case of failures.
(-2 and -3 are omitted.)

4 Category II and III systems are to comply with the following **(1)** to **(3)** in addition to the requirements in **-1** to **-3** above:

(1) Test programs and procedures for functional tests and failure tests are to be submitted to the Society. A FMEA may be requested by the Society in order to support containment of failure tests programs.
(2) Factory acceptance test including functional and failure tests is to be witnessed by the Society.
(3) The documentations specified in ~~**(a)** to **(g)** below~~ **1.2(2)(d)** are to be provided with the Society:
~~(a) functional description of software,~~
~~(b) list and versions of software installed in system,~~
~~(c) user manual including instructions for use during software maintenance,~~
~~(d) list of interfaces between system and other vessel systems,~~
~~(e) list of standards used for data links,~~
~~(f) additional documentation as requested by the Society which might include an FMEA or equivalent to demonstrate the adequacy of failure test case applied, and~~
~~(g) other documentation deemed necessary by the Society.~~

3.1.4 Approval of Programmable Devices for Category II and III Systems

~~**1** The application for the approval of a programmable device to be integrated inside a system is to be made by the system integrator or supplier.~~

~~**2**~~**1** Approval is to be granted on a case-by-case basis, except in cases where the programmable device has received approval of use in accordance with the requirements specified in **Chapter 1, Part 7 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

2 The application for the approval of a programmable device integrated inside a system is to be made by the system integrator or supplier.

3 With respect to **-~~2~~1** above, documentation for approval is recommended to address the information specified in the following **(1)** to **(3)**:
(1) the compatibility of the programmable device in the vessel's application
(2) the necessity to have on board tests during vessel integration, and
(3) the components of the systems using the approved programmable device.

3.1.5 Final Integration and On Board Testing

1 For computer based systems integrated with other computer based systems, simulation ~~Simulation~~ tests are to be undertaken before installation in cases where it is found necessary to check safe interaction with the other computer based ~~computerized~~ systems and functions which are unable to be previously tested. In such cases, the specifications of the simulation tests are to be submitted to the Society by the system integrator. The simulation tests are to be witnessed by the Surveyor.

2 On board tests are to check whether a computer based system in its final environment and which is integrated with all other systems with which it interacts is as follows:
(1) performing the functions for which it was designed,
(2) reacting safely in the case of failures originating internally or by devices external to the system, and
(3) interacting safely with other systems implemented into on board systems.

3 For category II and III systems, final integration and on board testing is to comply with the following requirements in addition to those specified in **-1** and **-2** above:

(1) Test specifications are to be submitted to the Society by the system integrator~~ for approval~~.

(2) The tests are to be witnessed by a surveyor assigned by the Society.

3.2 Limited Approval

Paragraph 3.2.1 has been amended as follows.

3.2.1 General

1 Sub-systems and programmable devices may be approved by the Society for limited applications with service restrictions in cases where the vessel systems in which they will be integrated into is not known. In such cases, sub-systems and programmable devices may be granted limited approval mentioning the required checks and tests performed.

2 In ~~such~~ cases specified in **-1** above, requirements about ~~related to the~~ quality systems specified in **3.1.1** may need to be satisfied as deemed necessary by the Society. Additional drawings, details, tests reports and surveys related to the standard declared by the supplier may be required by the Society upon request.

~~**2** With respect to **-1**, sub-systems and programmable devices may be granted limited approval based upon the carrying out of required checks and tests.~~

Section 3.3 has been amended as follows.

3.3 Modifications during Operation

3.3.1 Responsibilities

1 Organizations in charge of software modifications are to be clearly identified by owner to the Society.

2 A system integrator is to be designated by the owner as appropriate and is to satisfy the requirements specified in **3.1**.

3 ~~Software modifications are to be carried out for any modification already considered for each life cycle stage and in accordance with the scope of initial approval.~~ Limited life cycle steps may be considered for modifications already considered and accepted in the scope of initial approval.

4 The level of documentation necessary to be provided for modifications is to be determined by the Society on a case-by-case basis.

5 At the vessel level, it is the responsibility of the owner to manage traceability of ~~all~~ modifications. For category II and III systems, the software registry which contains the following **(1)** and **(2)** is to be updated. The achievement of this responsibility ~~is to~~ may be supported by system integrators updating the software registry~~. This software registry is to contain the following **(1)** and **(2)**~~:

(1) the lists and versions of software installed in systems ~~required by **3.1.3-4(3)(b)**~~, and

(2) the results of the security scans ~~required by~~ as described in **3.4.1-3**.

3.3.2 ~~Modification~~ Change Management

1 The owner is to ensure that necessary procedures for software and hardware ~~modification~~ change management ~~are maintained~~ exist on board, and that any software modifications or upgrades are performed according to ~~such~~ the procedures.

2 All ~~modifications~~ changes to computer based systems in the operational phase are to be recorded and be traceable.

3.4 System Security

Paragraph 3.4.1 has been amended as follows.

3.4.1 General

1 Owners, system integrators and suppliers are to adopt security policies and include these in their quality systems and procedures.

2 ~~For category I, II, and III systems, physical~~ Physical and logical security measures are to be in place to prevent unauthorized or unintentional modification of software, whether undertaken at the physical system or remotely.

3 Prior to installation, all artefacts (intermediate work products produced during the development of software), software code, executables and the physical medium used for installation on the vessel are to be scanned for viruses and malicious software. Results of the scan are to be documented and kept with the software registry.

~~**4** Results of the scan specified in -3 above are to be documented and added to the software registry.~~

Chapter 5 REQUIREMENTS FOR DATA LINKS

Section 5.1 has been amended as follows.

5.1 General Requirements

1 The requirements of this chapter apply to category II and III systems, unless otherwise specified.

~~**5.1.1 Loss of a Data Link**~~

2 Loss of a data link is to be specifically addressed in risk assessment analysis.

~~**5.1.2 Automatic Recovery after a Single Failure**~~

3 A single failure in data link hardware is to be automatically treated in order to restore ~~the system to~~ proper working of system ~~order~~. For category III systems, a single failure in data link hardware is not to influence the proper working of the system.

~~**5.1.3 Measures against Overload**~~

4 Characteristics of data links are to prevent overloading in any operational condition of system.

~~**5.1.4 Checking and Detecting Failures**~~

5 Data links are to be self-checking, detecting failures on the link itself and data communication failures on nodes connected to the link. Detected failures are to initiate an audible and visual alarm.

5.2 Specific Requirements for Wireless Data Links

Paragraph 5.2.2 has been amended as follows.

5.2.2 Requirements for Category II Systems

Category II systems may use wireless data links in accordance with the following **(1)** to **(3)** requirements:

(1) Recognised international wireless communication system protocols incorporating the following **(a)** to **(d)** are to be employed:

(a) Message integrity
Fault prevention, detection, diagnosis, and correction so that the received message is not corrupted or altered when compared to the transmitted message ~~in order to ensure message integrity (i.e., the received message is neither corrupted nor altered when compared to the transmitted message)~~.

(b) Configuration and device authentication
~~which only permit the~~ Only connection of devices included in the system design are to be permitted.

(c) Massage encryption
~~which is capable of protecting the contents of confidential and/or criticality data~~ Protection of the confidentiality and or criticality of the data content.

(d) Security management
~~which is capable of protecting the network and preventing unauthorized access~~ Protection of network assets, prevention of unauthorized access to network assets.

(2) The internal wireless system within the vessel is to comply with the radio frequency and power level ~~standards (e.g.,~~ requirements of the International Telecommunication Union and flag state requirements~~)~~.

(3) For wireless data communication equipment, tests during harbour and sea trials are to be conducted to demonstrate the following **(a)** and **(b)**:

(a) Radio-frequency transmission does not cause failure of any equipment during expected operations.

(b) Radio-frequency transmission does not cause itself to fail as a result of electromagnetic interference during expected operating conditions during expected operations.

EFFECTIVE DATE AND APPLICATION (Amendment 1-3)

1. The effective date of the amendments is 29 June 2018.

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

Amendment 1-4

D16 WINDLASSES AND MOORING WINCHES

Section D16.2 has been amended as follows.

D16.2 Windlasses

D16.2.1 General

As “standard or code of practice recognized by the Society” referred to in **16.2.1-2, Part D of the Rules**, the followings are examples of standard recognized but not limited to:

(1) *SNAME T & R Bulletin* 3-15 “Guide to the Design and Testing of Anchor Windlasses for Merchant Ships”
(2) *ISO* 7825 “Deck machinery general requirements”
(3) *ISO* 4568 “Shipbuilding - Sea-going vessels - Windlasses and anchor capstans”
(4) *JIS* F6714 “Windlasses”
(5) *BS MA*35 “Specifications for Ship Deck Machinery Windlass”

D16.2.2 Drawings and Data

1 “Windlass design specifications” specified in **16.2.2(1)(a), Part D of the Rules** are to be include the following in addition to windlass particulars:
(1) Anchor and chain cable particulars
(2) Maximum anchorage depth
(3) Performance criteria
(4) Standard or code of practice of compliance

2 “Windlass arrangement plan” specified in **16.2.2(1)(b), Part D of the Rules** are to be show all of the components of the anchoring/mooring system. The followings are examples of the components:
(1) Prime movers, shafting, cable lifters, anchors, chain cables, brakes and controls
(2) Mooring winches, wires and fairleads, if they form part of the windlass machinery
(3) Marking of nominal size of chain cable and maximum anchorage depth

3 “Dimensions, materials and welding details of torque-transmitting components and load-bearing components” specified in **16.2.2(1)(c), Part D of the Rules** are to comply with the followings:
(1) Information of mooring winches are to be included in case where the mooring winch is one with a windlass.
(2) Proposed materials are to be indicated.
(3) Weld joint designs, the degree of non-destructive examination of welds and post-weld heat treatment are to be indicated.

4 “Drawings and data concerning hydraulic systems” specified in **16.2.2(1)(d), Part D of the Rules** are to include the following:
(1) Piping diagram along with system design pressure
(2) Safety valves arrangement and settings
(3) Material specifications for pipes and equipment
(4) Typical pipe joints, as applicable
(5) Technical data and details for hydraulic motors

5 “Calculated strength for torque-transmitting components and load-bearing components” specified in **16.2.2(2)(a), Part D of the Rules** are to comply with the following:

(1) It is to be demonstrated that torque-transmitting components and load-bearing components comply with a standard or code of practice recognized by the Society.

(2) Analyses for gears are to be in accordance with a standard recognized by the Society.

6 "Load calculations" specified in **16.2.2(2)(c), Part D of the Rules** are to demonstrate that the prime mover is capable of attaining the hoisting speed, the required continuous duty pull and the overload capacity specified in **16.2.4, Part D of the Rules**.

7 "Operation and maintenance procedures" specified in **16.2.2(2)(e), Part D of the Rules** are to show the maximum anchorage depth.

D16.2.3 Materials and Fabrication

"Standards recognized by the Society" referred to in **16.2.3-1(1)** and **-2(2), Part D of the Rules** means national or international standard such as *JIS* or *ISO*.

D16.2.~~2~~4 ~~Construction and Performance~~ Design

1 The continuous duty pull specified in **16.2.4-2(2)(a)** is based on the following conditions:

(1) Ordinary stockless anchors are used.

(2) The anchor masses are assumed to be the masses as given in **Chapter 27, Part C of the Rules** and **Chapter 2, Part L of the Rules**.

(3) One anchor is hoisted at a time.

(4) The effects of buoyancy and hawse pipe efficiency (assumed to be 70%) have been accounted for.

2 The strength of windlass mounts in ~~**16.2.2-4**~~**16.2.4-2(7)(b), Part D of the Rules** is to comply with the following requirements:

((1) to (4) are omitted.)

~~D16.2.3 Tests~~

~~**1** Shop tests~~

~~Required tests in **16.2.3-1(2) of Part D of the Rules** may be dealt with as follows:~~

~~(1) As an operation test, windlasses are to be run without loads at speeds not less than nominal speeds for 30 *min*., 15 *min*. in each direction plus 5 *min*. in each direction on each additional gear change.~~

~~(2) Unless load tests and overload tests are practicable, load calculations using test results may be used in order to verify the ability of essential parts.~~

~~(3) Regarding chain wheel brake tests, calculation on braking force may be accepted.~~

EFFECTIVE DATE AND APPLICATION (Amendment 1-4)

1. The effective date of the amendments is 1 July 2018.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to windlasses for which the application for approval is submitted to the Society before the effective date and that are installed on ships for which the date of contract for construction* is before the effective date.

* "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

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 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, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:

(1) such alterations do not affect matters related to classification, or

(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.

The optional vessels will be considered part of the same series of 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.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.

Amendment 1-5

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

1.3 Calculation of Stresses

1.3.1 Alternating Bending Stress

Sub-paragraph -3 has been amended as follows.

3 Alternating Bending Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating bending stresses in crankpin fillets is as follows:

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

where

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

α_B : Stress concentration factor for bending in crankpin fillet (see **1.4.2** and <u>**3.1.2-2** of **Appendix D1**</u>~~**1.4.4-4(2)(b)**~~)

(2) Calculation of alternating bending stresses in journal fillets (not applicable to semi-built crankshafts) is as the following formulae in **(a)** or **(b)**:

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

where

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

β_B : Stress concentration factor for bending in journal fillet (see **1.4.3** and <u>**3.1.2-2** of **Appendix D1**</u>~~**1.4.4-4(2)(b)**~~)

β_Q : Stress concentration factor for compression due to radial force in journal fillet (see **1.4.3** and <u>**3.1.3-2(1)** of **Appendix D1**</u>~~**1.4.4-4(3)(b)i)**~~)

(b) $\sigma_{BG} = \pm(\beta_{BQ} \cdot \sigma_{BFN})$

β_{BQ} : Stress concentration factor for bending and compression due to radial force in journal fillet (see <u>**3.1.3-2(2)** of **Appendix D1**</u>~~**1.4.4-4(3)(b)ii)**~~)

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

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

where

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

γ_B : Stress concentration factor for bending in crankpin oil bore (see **1.4.**<u>**4**</u>~~**5**~~ <u>and **3.1.2-2** of **Appendix D4**</u>)

1.3.2 Alternating Torsional Stresses

Sub-paragraph -2 has been amended as follows.

2 Alternating Torsional Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating torsional stresses in crankpin fillets is as follows:

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

where

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

α_T : Stress concentration factor for torsion in crankpin fillet (see **1.4.2** and **3.1.1-3** of **Appendix D1**~~**1.4.4-4(1)(c)**~~)

τ_N : Nominal alternating torsional stress related to crankpin diameter (N/mm^2)

(2) Calculation of alternating torsional stresses in journal fillets (not applicable to semi-built crankshafts) is as follows:

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

where

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

β_T : Stress concentration factor for torsion in journal fillet (see **1.4.3** and **3.1.1-3** of **Appendix D1**~~**1.4.4-4(1)(c)**~~)

τ_N : Nominal alternating torsional stress related to journal diameter (N/mm^2)

(3) Calculation of alternating stresses in outlets of crankpin oil bores due to torsion (only applicable to radially drilled oil holes) is as follows:

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

where

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

γ_T : Stress concentration factor for torsion in outlet of crankpin oil bore (see **1.4.4**~~**5**~~ and **3.1.1-2** of **Appendix D4**)

τ_N : Nominal alternating torsional stress related to crankpin diameter (N/mm^2)

1.4 Stress Concentration Factors

1.4.1 Explanation of Terms and Symbols

Sub-paragraph -2 has been amended as follows.

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

D : Crankpin diameter (mm)

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

D_O : Diameter of oil bore in crankpin (mm)

R_H : Fillet radius of crankpin (mm)

T_H : Recess of crankpin fillet (mm)

D_G : Journal diameter (mm)

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

R_G : Fillet radius of journal (mm)

T_G : Recess of journal fillet (mm)

E : Pin eccentricity (mm)

S : Pin overlap (mm)

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

W : Web thickness (mm)

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

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

B : Web width (mm)

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

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

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

$w = W / D$ $(0.2 \le w \le 0.5)$

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

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

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

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

$t_H = T_H / D$

$t_G = T_G / D$

Where the geometry of crankshaft is outside the above ranges, stress concentration factors in crankpin fillets, ~~and~~ journal fillets and outlets of crankpin oil bores are to be calculated by utilizing the Finite Element Method (FEM) given in ~~**1.4.4**~~**Appendix D1** and **Appendix D4** or methods considered by the Society to be equivalent. In such cases, care should be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

Paragraph 1.4.4 has been deleted, and Paragraph 1.4.5 has been renumbered to Paragraph 1.4.4.

~~1.4.5~~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:

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

~~1.4.4 Stress Concentration Factors by Utilizing FEM~~

~~1 Finite Element Model~~

~~(1) The model is to consist of one complete crank, from the main bearing centreline to the opposite side main bearing centreline.~~

~~(2) Element type used in the vicinity of the fillets is to be as follows:~~

~~(a) 10-node tetrahedral elements~~

~~(b) 8-node hexahedral elements~~

~~(c) 20-node hexahedral elements~~

~~(3) Mesh properties in fillet radii applied to ±90 degrees in a circumferential direction from the crank plane are as follows:~~

~~(a) Maximum element size a through the entire fillet as well as in the circumferential direction is to be $a=R_H/4$ in crankpin fillets and $a=R_G/4$ in journal fillets. When using 20-node hexahedral elements, the element size in the circumferential direction may be extended up to 5a. In the case of multi-radii fillet, the local fillet radius is to be applied.~~

~~(b) Element size in fillet depth direction is to be of a to first layer thickness, 2a to second~~

~~layer thickness and $3a$ to third layer thickness. (see **Fig. 6**)~~

~~(4) A minimum of 6 elements are to be set across the web thickness.~~

~~(5) The rest of the crank is to be suitable for numeric stability of the solver.~~

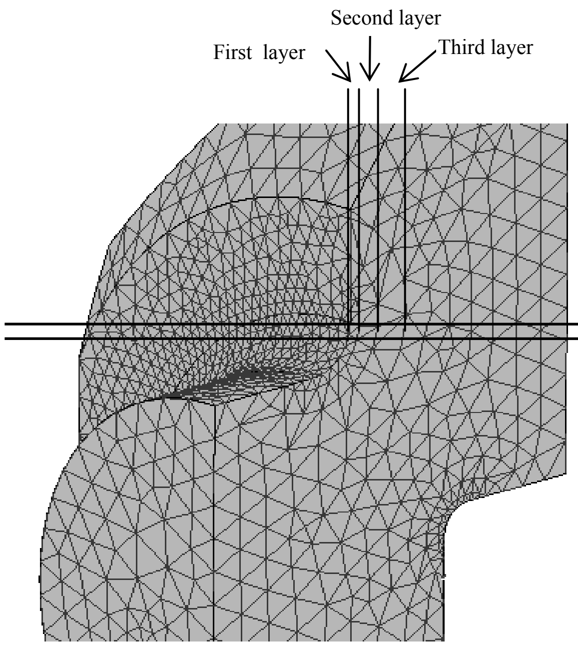
~~(6) Counterweights have to be modelled only when influencing the global stiffness of the crank significantly.~~

~~(7) Modelling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than $2R_H$ or $2R_G$ (see **Fig. 7**)~~

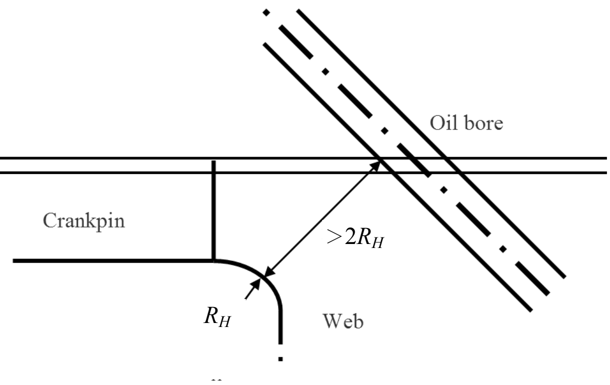
~~(8) Drillings and holes for weight reduction have to be modelled.~~

~~(9) Sub-modelling may be used as far as the software requirements are fulfilled.~~

~~Fig. 6 Element Size in Fillet Depth Direction~~



~~Fig. 7 Oil Bore Proximity to Fillet~~



~~**2** Material properties~~

~~Material properties applied to steels as follows.~~

~~Young's Modulus : $E = 2.05 \cdot 10^5$ MPa~~

~~Poisson's ratio : $\nu = 0.3$~~

~~**3** Element mesh quality criteria~~

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

~~(1) The quality of the mesh is to be assured by checking the stress component normal to the surface of the fillet radius. Ideally, this stress is to be zero. With principal stresses σ_1, σ_2 and σ_3, the following criterion is required:~~

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

~~(2) Unaveraged nodal stress results calculated from each element connected to a node is to differ less than by 5 % from the 100 % averaged nodal stress results at this node at the examined location.~~

~~**4** Load cases~~

~~The following load cases have to be calculated.~~

~~(1) Torsion~~

~~(a) Calculation is to be performed under the boundary and load conditions given in **Fig. 8** where the 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.~~

~~(b) For all nodes in both the journal and crankpin fillet, principal stresses are extracted and the equivalent torsional stress is calculated:~~

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

~~(c) The maximum value taken for the subsequent calculation of the stress concentration factors for torsion in crankpin and journal fillet.~~

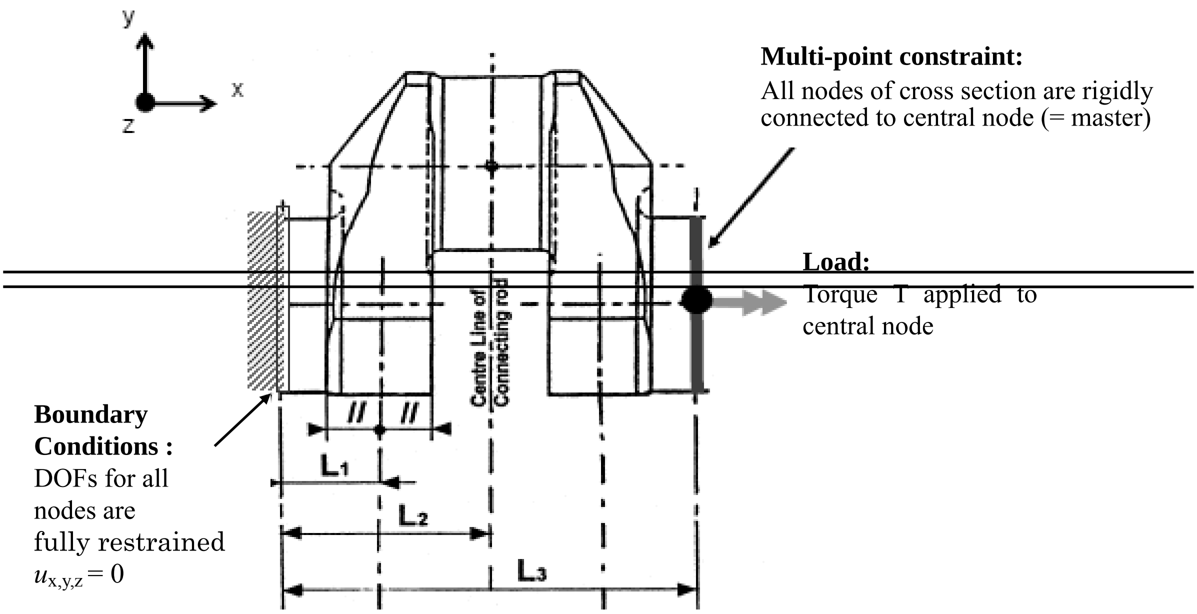
$$\cancel{\alpha_T = \frac{\tau_{equiv,a}}{\tau_N}}$$

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

~~where τ_N is nominal torsional stress for the crankpin and journal respectively and is calculated as follows (for W_P see **1.3.2**):~~

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

~~Fig. 8 Boundary and Load Conditions for the Torsion Load Case~~



~~(2) Pure bending (4 point bending)~~

~~(a) Calculation is to be performed under the boundary and load conditions given in **Fig. 9** where the bending moment is applied to the central node located at the crankshaft axis.~~

~~(b) For all nodes in both the journal and crankpin fillet, von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the stress concentration factors for bending in crankpin and journal fillet according to:~~

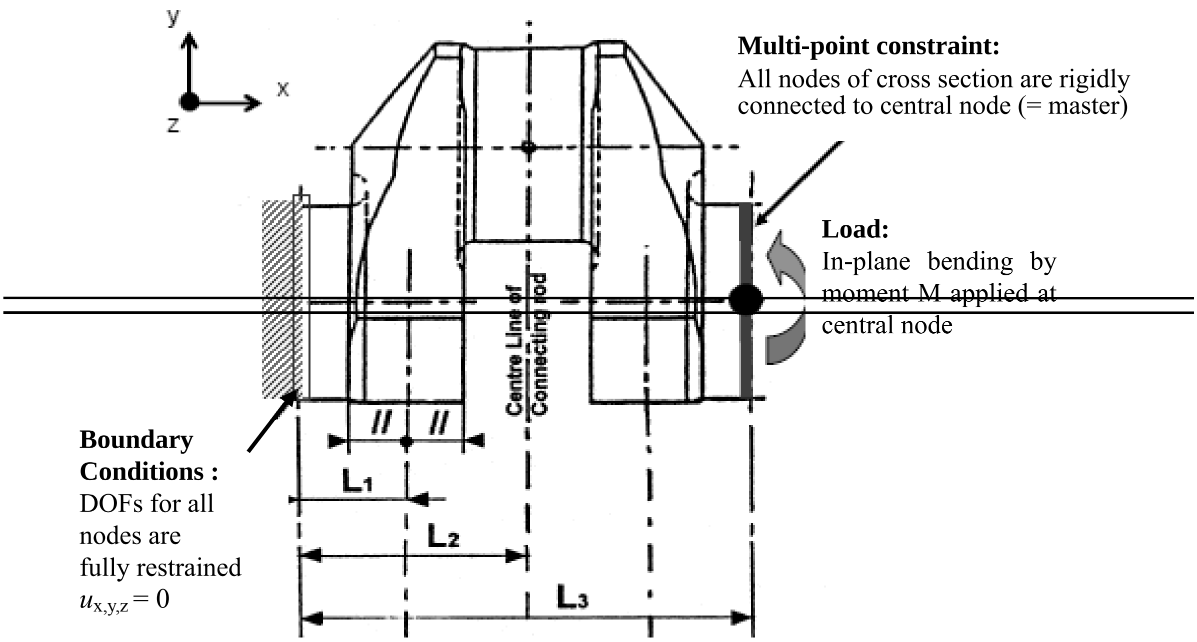
$$\cancel{\alpha_B = \frac{\sigma_{equiv,a}}{\sigma_N}}$$

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

~~where σ_N is nominal mending stress for the crankpin and journal respectively and is calculated as follows (for W_{eqw} see **1.3.1-2(2)**):~~

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

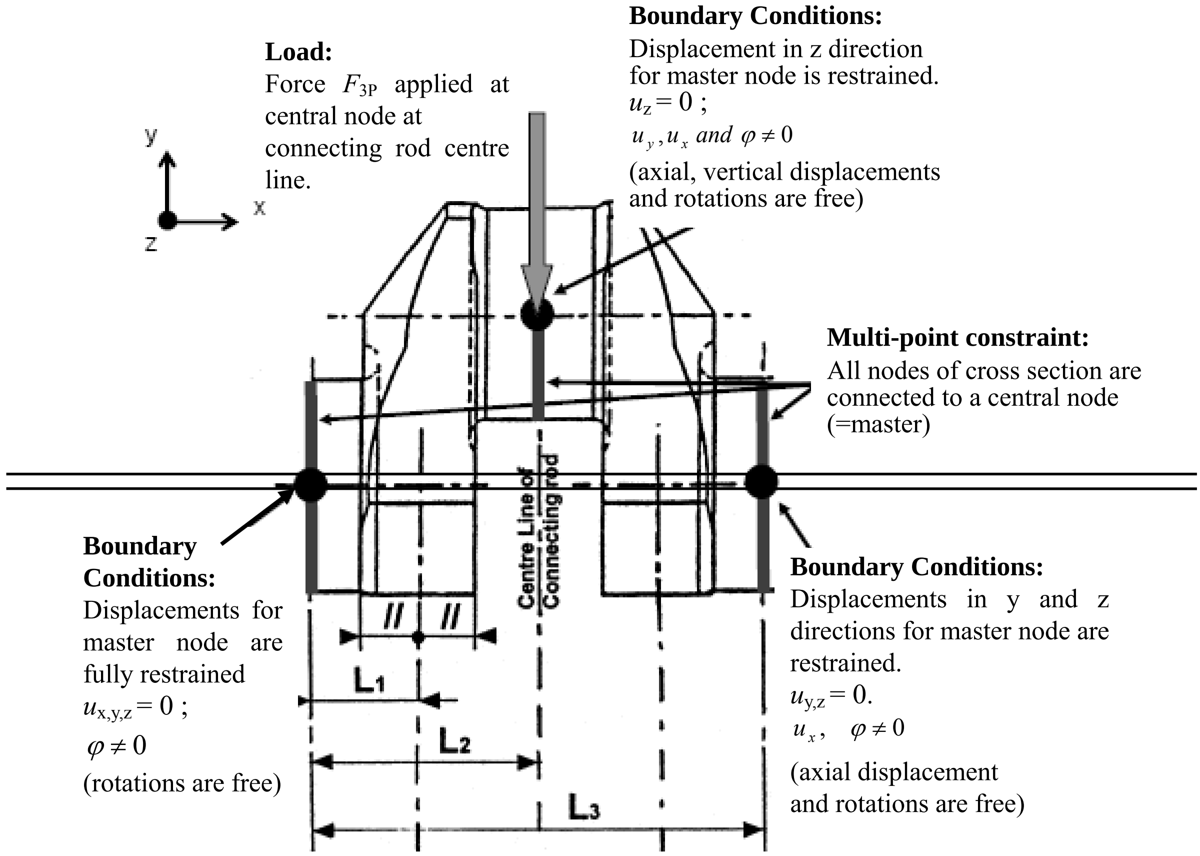
~~Fig. 9 Boundary and Load Conditions for the Pure Bending Load Case~~



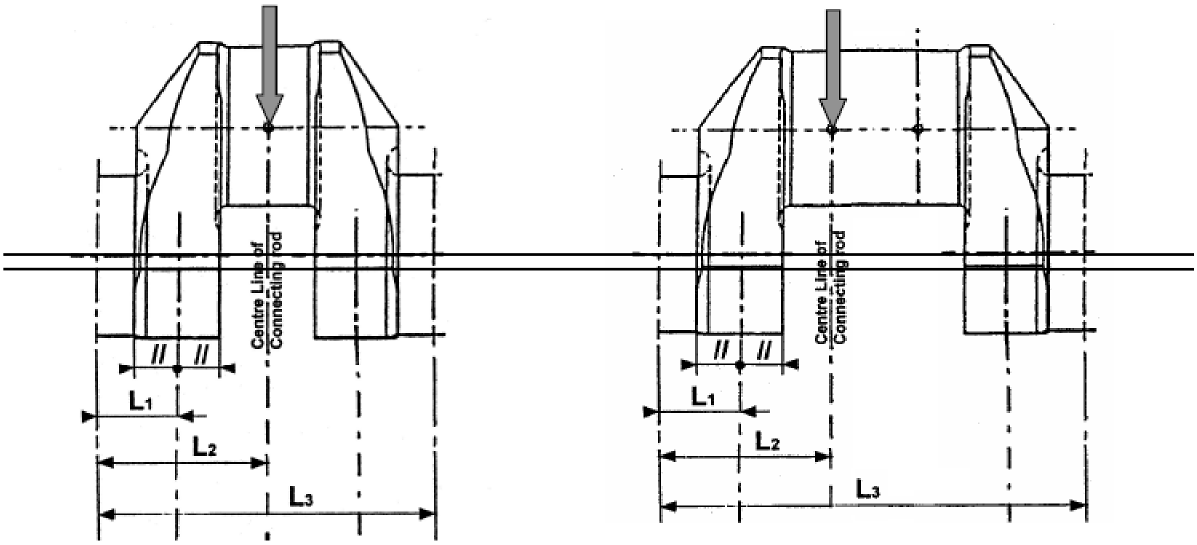
~~(3) Bending with shear force (3-point bending)~~

~~(a) Calculation is to be performed under the boundary and load conditions given in **Fig. 10** where the force is applied to the central node located at the pin centre-line of the connecting rod.~~

~~Fig. 10 Boundary and Load Conditions for the 3-point Bending Load Case of an Inline Engine.~~



~~Fig. 11 Load Applications for In-line and V-type Engines~~



~~(b) The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The stress concentration factors in the journal fillet can be determined as shown **i) or ii).**~~

~~i) Stress concentration factor for compression due to radial force in journal fillet β_Q is calculated in accordance with the following:~~

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

~~where~~

~~σ_{3P} : as found by the Finite Element Calculation~~

~~σ_{N3P} : Nominal bending stress in the web centre due to force F_{3P} applied to the centre-line of the actual connecting rod (see Figure 11)~~

~~β_B : as determined in **1.4.4-4(2)(b)**~~

~~$\sigma_{Q3P} = Q_{3P}/(B \cdot W)$~~

~~Q_{3P} : the radial (shear) force in the web due to the force F_{3P} applied to the centre-line of the actual connecting rod (see Figures 1 and 2)~~

~~ii) The stress concentration factor for bending and compression due to radial force in journal fillet β_{BQ} is calculated in accordance with the following:~~

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

~~for the relevant parameters see **i)**.~~

1.7 Fatigue Strength

Paragraph 1.7.1 has been amended as follows.

1.7.1 Fatigue Strength in Crankpin Fillets

1 The fatigue strength in crankpin fillets is evaluated in accordance with the following: (For calculation purposes, 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}) in cases where the surfaces of fillets, the outlets of oil bores and the insides of oil bores (down to a minimum depth equal to 1.5 times the oil bore diameter) are all smoothly finished

K : Factor for the different types of crankshafts without surface treatment

= 1.05 for continuous grain flow forged or drop-forged crankshafts

= 1.0 for free form forged crankshafts (without continuous grain flow)

Factor for cast steel crankshafts with cold rolling treatment in fillet areas

= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society

As an alternative, the value of K can be determined by experiments based either on full size crankthrows (or crankshafts) or on specimens taken from full size crankthrows.

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

for other parameters see **1.4**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow, evaluation of test results is to be carried out in accordance with **Appendix D2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the fillets, every surface-treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix D3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

Paragraph 1.7.3 has been amended as follows.

1.7.3 Fatigue Strength in Outlets of Crankpin Oil Bores

1 The fatigue strength in outlets of crankpin oil bores is evaluated in accordance with the following: (For calculation purposes, $D_O/2$ 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 areas
= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society
As an alternative, the value of K can be determined by experiments based either on full size crankthrows (or crankshafts) or on specimens taken from full size crankthrows.
for other parameters see **1.7.1**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full size crankthrow (or crankshaft), evaluation of test results is to be carried out in accordance with **Appendix D2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the outlets of oil bores, every surface treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix D3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

Appendix D1 has been added as follows.

Appendix D1 Guidance for Calculation of Stress Concentration Factors in the web fillet radii of crankshafts by utilizing Finite Element Method

1.1 General

The objective of the analysis is to develop Finite Element Method (FEM) calculated figures as an alternative to the analytically calculated Stress Concentration Factors (SCF) at the crankshaft fillets. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries and accordingly the application of these formulae is limited to those geometries.
The SCF's calculated according to the rules of this Appendix are defined as the ratio of stresses calculated by FEM to nominal stresses in both journal and pin fillets. When used in connection with the present method in **Annex D2.3.1-2(2)**, von Mises stresses is to be calculated for bending and principal stresses for torsion.
The procedure as well as evaluation guidelines are valid for both solid cranks and semi-built cranks (except journal fillets).
The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.
The calculation of SCF at the oil bores is covered by **Appendix D4**.
It is advised to check the element accuracy of the FE solver in use, e.g. by modeling a simple geometry and comparing the stresses obtained by FEM with the analytical solution for pure bending and torsion.
Boundary Element Method (BEM) may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and perceptions for building the FE-model are presented in **2.1.1**. It is obligatory for the final FE-model to fulfill the requirement in **2.2**.

2.1.1 Element Mesh Recommendations

1 In order to fulfil the mesh quality criteria, it is advised to construct the FE model for the evaluation of Stress Concentration Factors according to the following recommendations:

(1) The model consists of one complete crank, from the main bearing centreline to the opposite side main bearing centreline.

(2) Element type used in the vicinity of the fillets:
  - (a) 10-node tetrahedral elements
  - (b) 8-node hexahedral elements
  - (c) 20-node hexahedral elements

(3) Mesh properties in fillet radii applied to ±90 degrees in a circumferential direction from the crank plane are as follows:
  - (a) Maximum element size a through the entire fillet as well as in the circumferential direction is to be $a=R_H/4$ in crankpin fillets and $a=R_G/4$ in journal fillets. When using 20-node hexahedral elements, the element size in the circumferential direction may be extended up to $5a$. In the case of multi-radii fillet, the local fillet radius is to be applied.
  - (b) Element size in fillet depth direction (see **Fig. 1**):

i) First layer thickness equal to element size of a
ii) Second layer thickness equal to element to size of $2a$
iii) Third layer thickness equal to element to size of $3a$

(4) A minimum of 6 elements are to be set across the web thickness.
(5) The rest of the crank should be suitable for numeric stability of the solver.
(6) Counterweights have to be modelled only when influencing the global stiffness of the crank significantly.
(7) Modelling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than $2R_H$ or $2R_G$ (see **Fig. 2**)
(8) Drillings and holes for weight reduction have to be modelled.
(9) Sub-modelling may be used as far as the software requirements are fulfilled.

Fig. 1 Element Size in Fillet Depth Direction

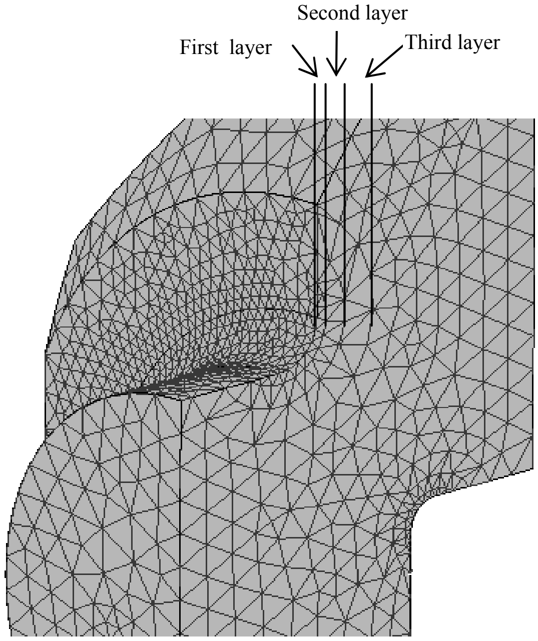
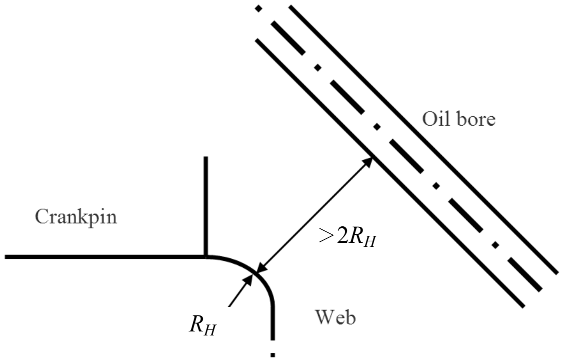


Fig. 2 Oil Bore Proximity to Fillet



2.1.2 Material

1 Material properties applied to steels are as follows:

Young's Modulus : $E = 2.05 \cdot 10^5\, MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters are to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

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

2.2.1 Principal Stresses Criterion

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

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

2.2.2 Averaged/Unaveraged Stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than by 5 % from the 100 % averaged nodal stress results at this node at the examined location.

3.1 Load Cases

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 3** where the torque is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, principal stresses are extracted and the equivalent torsional stress is calculated:

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

3 The maximum value taken for the subsequent calculation of the stress concentration factors for torsion in crankpin and journal fillet.

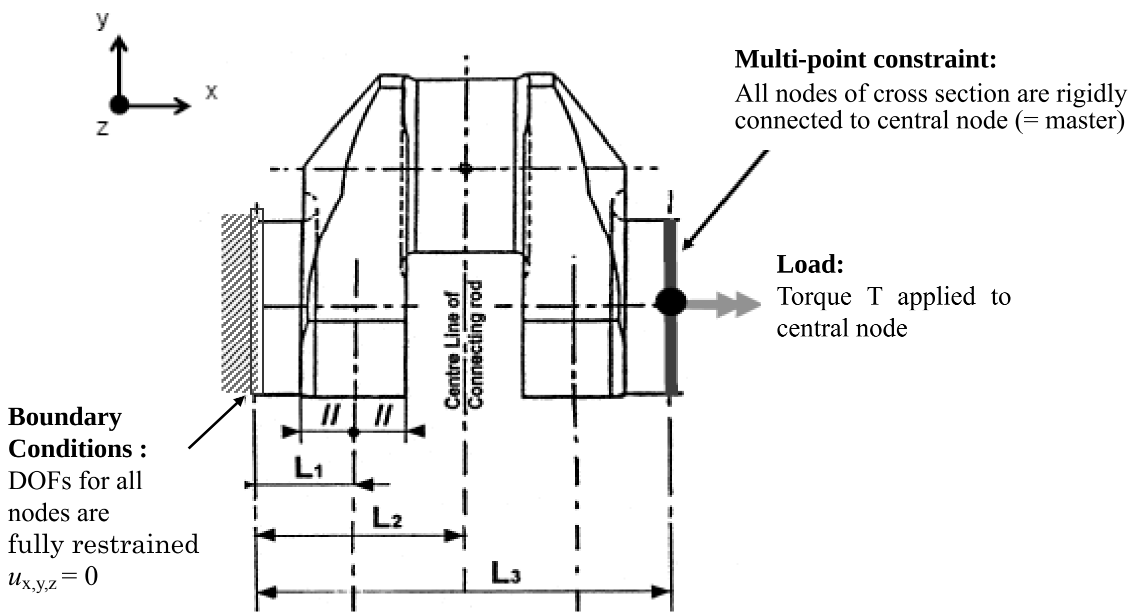
$$\alpha_T = \frac{\tau_{equiv,\alpha}}{\tau_N}$$

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

where τ_N is nominal torsional stress for the crankpin and journal respectively and is calculated as follows (for W_P see **1.3.2** of **Annex D2.3.1-2(2)**):

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

Fig. 3 Boundary and Load Conditions for the Torsion Load Case



3.1.2 Pure Bending (4-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 4** where the bending moment is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the stress concentration factors for bending in crankpin and journal fillet according to:

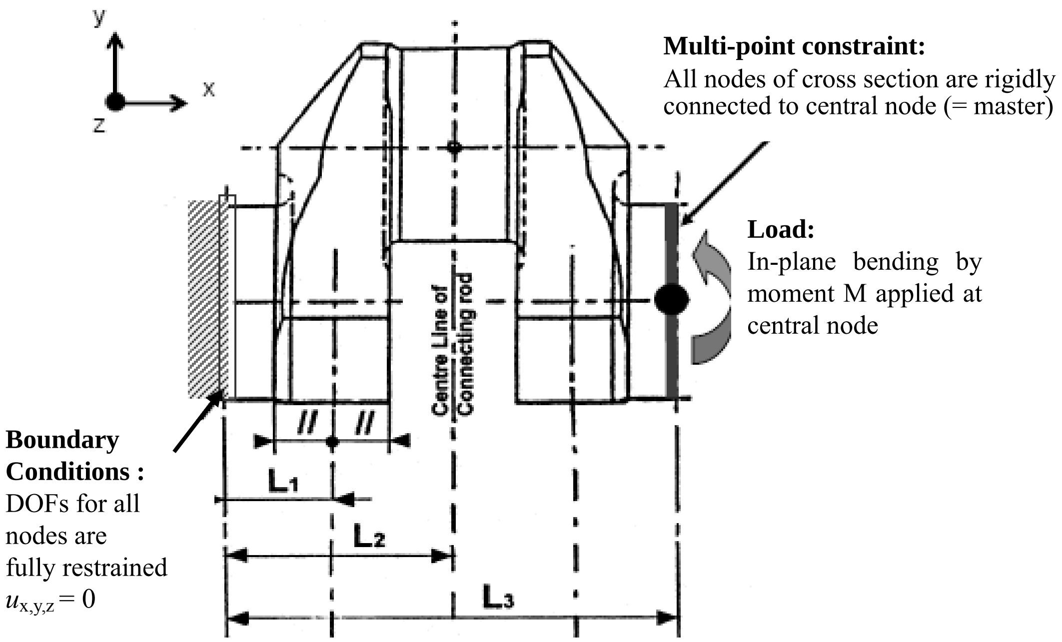
$$\alpha_B = \frac{\sigma_{equiv,a}}{\sigma_N}$$

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

where σ_N is nominal bending stress for the crankpin and journal respectively and is calculated as follows (for W_{eqw} see **1.3.1-2(2)** of **Annex D2.3.1-2(2)**):

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

Fig. 4 Boundary and Load Conditions for the Pure Bending Load Case



3.1.3 Bending with Shear Force (3-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 5** where the force is applied to the central node located at the pin centre-line of the connecting rod.

Fig. 5 Boundary and Load Conditions for the 3-point Bending Load Case of an Inline Engine.

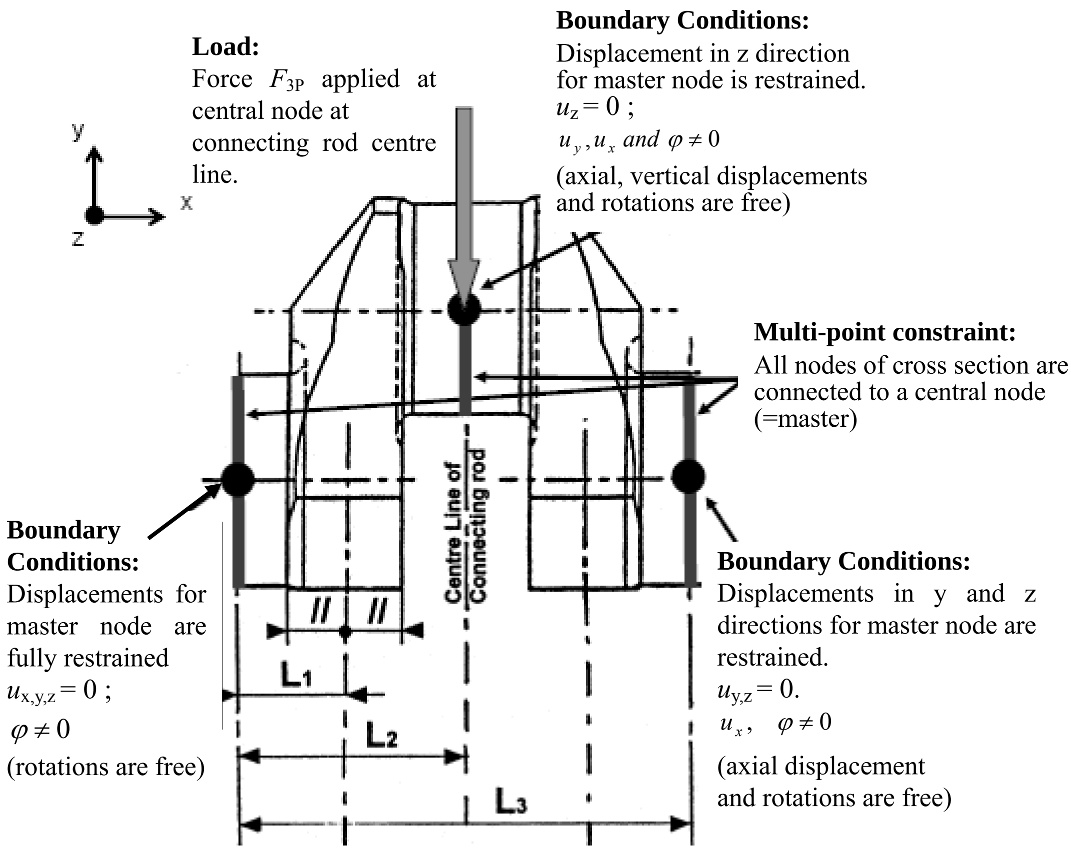
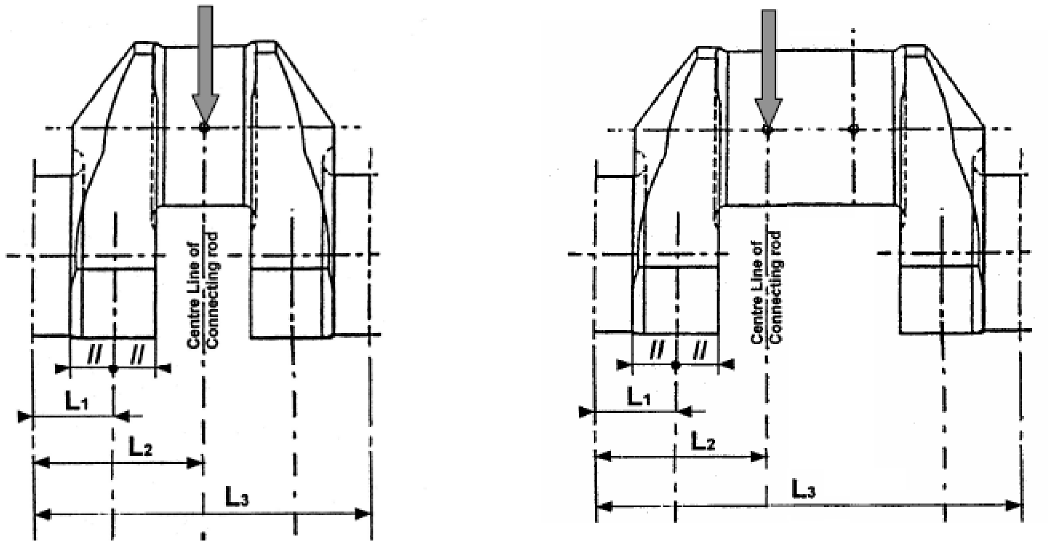


Fig. 6 Load Applications for In-line and V-type Engines



2 The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The stress concentration factors in the journal fillet can be determined as shown **i)** or **ii)**.

(1) Stress concentration factor for compression due to radial force in journal fillet β_Q is calculated in accordance with the following:

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

where

σ_{3P} : as found by the Finite Element Calculation

σ_{N3P}: Nominal bending stress in the web centre due to force F_{3P} applied to the centre-line of the actual connecting rod (see **Fig. 6**)

β_B : as determined in **3.1.2-2**

$\sigma_{Q3P} = Q_{3P} / (B.W)$

Q_{3P} : the radial (shear) force in the web due to the force F_{3P} applied to the centre-line of the actual connecting rod (see **Fig. 1** and **Fig. 2** of **Annex D2.3.1-2(2)**)

(2) The stress concentration factor for bending and compression due to radial force in journal fillet β_{BQ} is calculated in accordance with the following:

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

for the relevant parameters see **(1)**.

Appendix D2 has been added as follows.

Appendix D2 Guidance for Evaluation of Fatigue Tests

1.1 Introduction

Fatigue testing can be divided into two main groups; testing of small specimens and full-size crank throws. Testing can be made using the staircase method or a modified version thereof which is presented in this Appendix. Other statistical evaluation methods may also be applied.

1.2 Small Specimen Testing

1 For crankshafts without any fillet surface treatment, the fatigue strength can be determined by testing small specimens taken from a full-size crank throw.
2 When other areas in the vicinity of the fillets are surface treated introducing residual stresses in the fillets, this approach cannot be applied.
3 One advantage of this approach is the rather high number of specimens which can be then manufactured. Another advantage is that the tests can be made with different stress ratios (R-ratios) and/or different modes e.g. axial, bending and torsion, with or without a notch. This is required for evaluation of the material data to be used with critical plane criteria.

1.3 Full-size Crank Throw Testing

1 For crankshafts with surface treatment the fatigue strength can only be determined through testing of full size crank throws.
2 The load can be applied by hydraulic actuators in a 3- or 4- point bending arrangement, or by an exciter in a resonance test rig. The latter is frequently used, although it usually limits the stress ratio to R = -1.

2.1 Evaluation of Test Results

2.1.1 Principles

1 Prior to fatigue testing the crankshaft is to be tested as required by quality control procedures, e.g. for chemical composition, mechanical properties, surface hardness, hardness depth and extension, fillet surface finish, etc.
2 The test samples should be prepared so as to represent the "lower end" of the acceptance range e.g. for induction hardened crankshafts this means the lower range of acceptable hardness depth, the shortest extension through a fillet, etc. Otherwise the mean value test results should be corrected with a confidence interval: a 90 % confidence interval may be used both for the sample mean and the standard deviation.
3 The test results are to be evaluated to represent the mean fatigue strength, with or without taking into consideration the 90 % confidence interval as mentioned above. The standard deviation should be considered by taking the 90 % confidence into account. Subsequently the result to be used as the fatigue strength is then the mean fatigue strength minus one standard deviation.
4 If the evaluation aims to find a relationship between (static) mechanical properties and the fatigue strength, the relation is to be based on the real (measured) mechanical properties, not on the

specified minimum properties.

5 The calculation technique in **2.1.4** was developed for the original staircase method. However, since there is no similar method dedicated to the modified staircase method the same is applied for both.

2.1.2 Staircase Method

1 In the original staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress corresponding to the expected average fatigue strength.

(2) If the specimen specified in **(1)** survives 10^7 cycles, it is discarded and the next specimen is subjected to a stress that is one increment above the previous.

(3) A survivor is always followed by the next using a stress one increment above the previous, as speficied in **(2)**. The increment should be selected to correspond to the expected level of the standard deviation.

(4) When a specimen fails prior to reaching 10^7 cycles, the obtained number of cycles is noted and the next specimen is subjected to a stress that is one increment below the previous.

2 This original staircase method is only suitable when a high number of specimens are available.

3 The minimum number of test specimens is to be 25.

2.1.3 Modified Staircase Method

1 When a limited number of specimens are available, it is advisable to apply the modified staircase method.

2 In the modified staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress level that is most likely well below the average fatigue strength.

(2) When this specimen specified in **(1)** has survived 10^7 cycles, this same specimen is subjected to a stress level one increment above the previous. The increment should be selected to correspond to the expected level of the standard deviation. This is continued with the same specimen until failure.

(3) Then the number of cycles is recorded and the next specimen is subjected to a stress that is at least 2 increments below the level where the previous specimen failed.

3 The acquired result of a modified staircase method should be used with care, since some results available indicate that testing a runout on a higher test level, especially at high mean stresses, tends to increase the fatigue limit. However, this “training effect” is less pronounced for high strength steels (e.g. $UTS > 800$ MPa).

4 The minimum number of test specimens is to be 3.

2.1.4 Calculation of Sample Mean and Standard Deviation

A hypothetical example of tests for 5 crank throws is presented further in the subsequent text.

(1) When using the modified staircase method and the evaluation method of Dixon and Mood, the number of samples will be 10, meaning 5 run-outs and 5 failures, i.e.:
Number of samples, $n = 10$

(2) Furthermore, the method distinguishes between:
  (a) Less frequent event is failures: $C = 1$
  (b) Less frequent event is run-outs: $C = 2$
  The method uses only the less frequent occurrence in the test results, i.e. if there are more failures than run-outs, then the number of run-outs is used.

(3) In the modified staircase method, the number of run-outs and failures are usually equal. However, the testing can be unsuccessful, e.g. the number of run-outs can be less than the number of failures if a specimen with 2 increments below the previous failure level goes

directly to failure. On the other hand, if this unexpected premature failure occurs after a rather high number of cycles, it is possible to define the level below this as a run-out.

(4) Dixon and Mood's approach, derived from the maximum likelihood theory, which also may be applied here, especially on tests with few samples, presented some simple approximate equations for calculating the sample mean and the standard deviation from the outcome of the staircase test.

(a) The sample mean can be calculated as follows:

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2} \right) \quad \text{when} \quad C = 1$$

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} + \frac{1}{2} \right) \quad \text{when} \quad C = 2$$

(b) The standard deviation can be found by

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

where:

S_{a0} is the lowest stress level for the less frequent occurrence

d is the stress increment

$F = \sum f_i$

$A = \sum i \cdot f_i$

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

i is the stress level numbering

f_i is the number of samples at stress level i

The formula for the standard deviation is an approximation and can be used when

$$\frac{B \cdot F - A^2}{F^2} > 0.3$$

and

$$0.5 \cdot s < d < 1.5 \cdot s$$

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

(5) If increment d is greatly higher than the standard deviation s, the procedure leads to a lower standard deviation and a slightly higher sample mean, both compared to values calculated when the difference between the increment and the standard deviation is relatively small. Respectively, if increment d is much less than the standard deviation s, the procedure leads to a higher standard deviation and a slightly lower sample mean.

****

Example

Hypothetical test results are shown in **Fig. 1**. The processing of the results and the evaluation of the sample mean and the standard deviation are shown in **Fig. 2**.

Fig. 1 Log Sheet of a Modified Staircase Test.

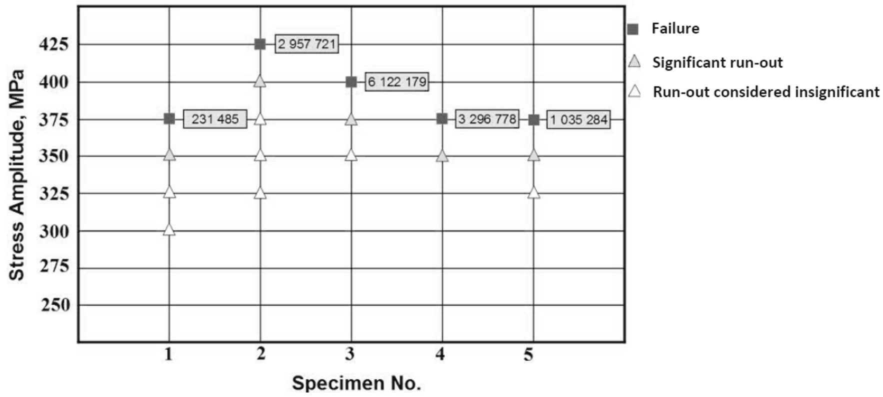
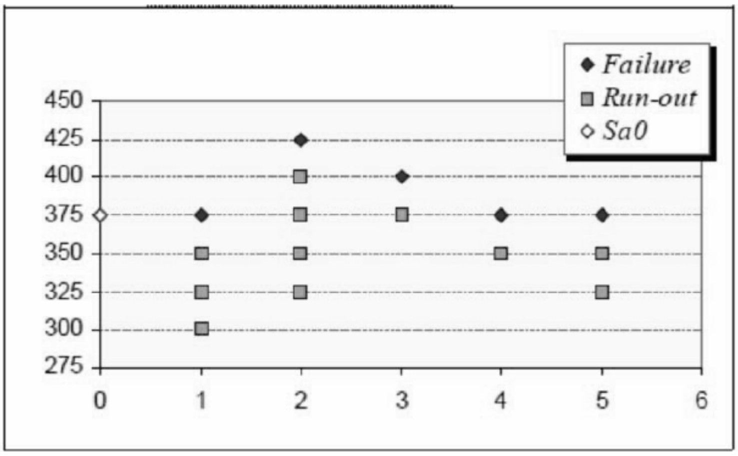


Fig. 2 Processing of a Modified Staircase Test Results.



i	fi	i*fi	i^2*fi
2	1	2	4
1	1	1	1
0	3	0	0
Σ	5	3	5
	F	*A*	*B*

$$F = \sum f_i$$

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

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

Notes:
i = 0, 1, 2… is the stress level numbering, the numbering usually starts from zero
f_i is number of test specimen at stress level, i

Sample mean and standard deviation are evaluated as follows based upon **Fig. 2**.

(1) Stress level 0, $S_{a0} := 375MPa$

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

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

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

(4) Calculation of sample mean is as follows:

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

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

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

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

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

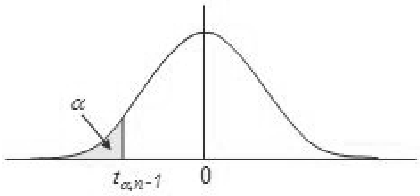
****

2.1.5 Confidence Interval for Mean Fatigue Limit

1 If the staircase fatigue test is repeated, the sample mean and the standard deviation will most likely be different from the previous test. Therefore, it is necessary to assure with a given confidence that the repeated test values will be above the chosen fatigue limit by using a confidence interval for the sample mean.

2 The confidence interval for the sample mean value with unknown variance is known to be distributed according to the *t*-distribution (also called student's *t*-distribution) which is a distribution symmetric around the average. (See **Fig. 3**)

Fig. 3 Student's *t*-distribution



Note:
The confidence level normally used for the sample mean is 90 %, meaning that 90 % of sample means from repeated tests will be above the value calculated with the chosen confidence level. The **Fig. 3** shows the t-value for (1-α)· 100 % confidence interval for the sample mean.

3 If S_a is the empirical mean and *s* is the empirical standard deviation over a series of *n* samples, in which the variable values are normally distributed with an unknown sample mean and unknown variance, the $(1-\alpha)\cdot 100\%$ confidence interval for the mean is:

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

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

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

which is the mean fatigue limit (population value) to be used to obtain the reduced fatigue limit where the limits for the probability of failure are taken into consideration.

****

Example

Applying a 90% confidence interval ($\alpha = 0.1$) and $n = 10$ (5 failures and 5 run-outs) leads to $t_{\alpha,n-1} = 1.383$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990. Note that $v = n-1$ in the tables.). Hence:

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

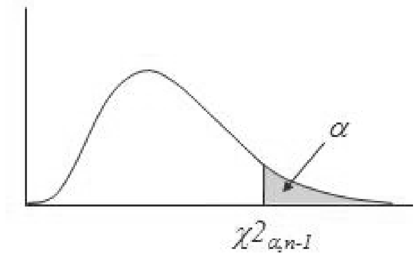
To be conservative, some authors would consider *n* to be 5, as the physical number of used specimen, then $t_{\alpha,n-1} = 1.533$.

****

2.1.6 Confidence Interval for Standard Deviation

1 The confidence interval for the variance of a normal random variable is known to possess a chi-square distribution with $n-1$ degrees of freedom. (See **Fig.4**)

Fig. 4 Chi-square distribution



Note:
The confidence level on the standard deviation is used to ensure that the standard deviations for repeated tests are below an upper limit obtained from the fatigue test standard deviation with a confidence level. The **Fig. 4** shows the chisquare for (1-α)· 100% confidence interval for the variance.

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

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

3 A $(1-\alpha)\cdot 100\%$ confidence interval for the standard deviation is obtained by the square root of the upper limit of the confidence interval for the variance and can be found by

$$S_{X\%} = \sqrt{\frac{n-1}{\chi^2_{\alpha,n-1}}}\cdot s$$

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

****

Example

Applying a 90% confidence interval ($\alpha = 0.1$) and n = 10 (5 failures and 5 run-outs) leads to $\chi^2_{\alpha,n-1} = 4.168$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990).
Hence:

$$S_{90\%} = \sqrt{\frac{n-1}{4.168}}\cdot s = 1.47\cdot s$$

To be conservative, some authors would consider *n* to be 5, as the physical number of the used specimen, then $\chi^2_{\alpha,n-1} = 1.064$.

****

3.1 Small Specimen Testing

3.1.1 General

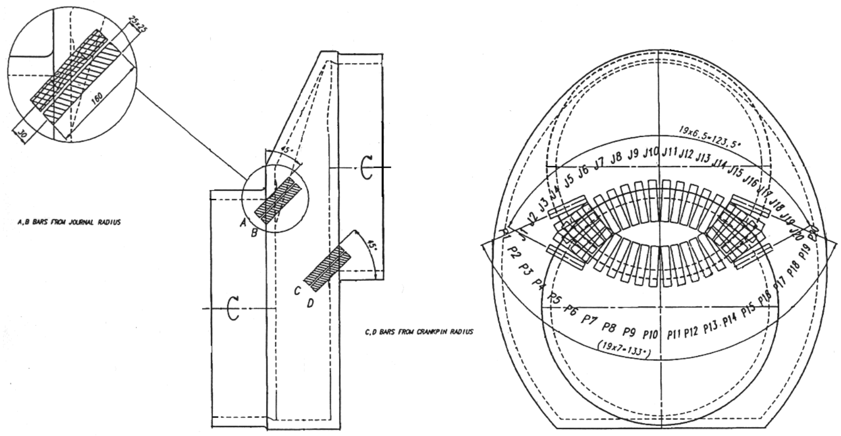
1 In this Appendix, a small specimen is considered to be one of the specimens taken from a crank throw.

2 Since the specimens are to be representative for the fillet fatigue strength, they should be taken out close to the fillets. (See **Fig. 5**)

3 It should be made certain that the principal stress direction in the specimen testing is equivalent to the full-size crank throw. The verification is recommended to be done by utilizing the finite element method.

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

Fig. 5 Specimen locations in a crank throw



3.1.2 Determination of Bending Fatigue Strength

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

2 In order to ensure principal stress direction in push-pull testing to represent the full-size crank throw principal stress direction and when no further information is available, the specimen is to be taken in 45 degrees angle as shown in **Fig. 5**. If the objective of the testing is to document the influence of high cleanliness, test samples taken from positions approximately 120 degrees in a circumferential direction may be used. (See **Fig. 5**) If the objective of the testing is to document the influence of continuous grain flow (cgf) forging, the specimens should be restricted to the vicinity of the crank plane.

3.1.3 Determination of Torsional Fatigue Strength

1 If the specimens are subjected to torsional testing, the selection of samples should follow the same guidelines as for bending above. The stress gradient influence has to be considered in the evaluation.

2 If the specimens are tested in push-pull and no further information is available, the samples should be taken out at an angle of 45 degrees to the crank plane in order to ensure collinearity of the principal stress direction between the specimen and the full size crank throw. When taking the specimen at a distance from the (crank) middle plane of the crankshaft along the fillet, this plane rotates around the pin centre point making it possible to resample the fracture direction due to torsion (the results are to be converted into the pertinent torsional values).

3.1.4 Other Test Positions

1 If the test purpose is to find fatigue properties and the crankshaft is forged in a manner likely to lead to cgf, the specimens may also be taken longitudinally from a prolonged shaft piece where specimens for mechanical testing are usually taken. The condition is that this prolonged shaft piece is heat treated as a part of the crankshaft and that the size is so as to result in a similar quenching rate as the crank throw.

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

3.1.5 Correlation of Test Results

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

2 When using the bending fatigue properties from tests mentioned in **3.1**, it should be kept in mind that successful continuous grain flow (cgf) forging leading to elevated values compared to other (non cgf) forging, will normally not lead to a torsional fatigue strength improvement of the same magnitude. In such cases it is advised to either carry out also torsional testing or to make a conservative assessment of the torsional fatigue strength. This approach is applicable when using the Gough Pollard criterion. However, this approach is not recognised when using the von Mises or a multi-axial criterion such as Findley.

3 If the found ratio between bending and torsion fatigue differs significantly from $\sqrt{3}$, one should consider replacing the use of the von Mises criterion with the Gough Pollard criterion. Also, if critical plane criteria are used, it has to be kept in mind that cgf makes the material inhomogeneous in terms of fatigue strength, meaning that the material parameters differ with the directions of the planes.

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

4.1 Full Size Testing

4.1.1 Hydraulic Pulsation

1 A hydraulic test rig can be arranged for testing a crankshaft in 3-point or 4-point bending as well as in torsion. This allows for testing with any *R*-ratio.

2 Although the applied load should be verified by strain gauge measurements on plain shaft sections for the initiation of the test, it is not necessarily used during the test for controlling load. It is also pertinent to check fillet stresses with strain gauge chains.

3 Furthermore, it is important that the test rig provides boundary conditions as defined in **3.1** of **Appendix D3**.

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

4.1.2 Resonance Tester

1 A rig for bending fatigue normally works with an *R*-ratio of -1. **Fig. 6** shows a layout of the testing arrangement.

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

3 Clamping around the journals is to be arranged in a way that prevents severe fretting which could lead to a failure under the edges of the clamps. If some distance between the clamps and the journal fillets is provided, the loading is consistent with 4-point bending and thus representative for the journal fillets also.

4 In an engine, the crankpin fillets normally operate with an *R*-ratio slightly above -1 and the journal fillets slightly below -1. If found necessary, it is possible to introduce a mean load (deviate from $R = -1$) by means of a spring preload.

5 A rig for torsion fatigue can also be arranged as shown in **Fig. 7**. When a crank throw is subjected to torsion, the twist of the crankpin makes the journals move sideways. If one single crank throw is tested in a torsion resonance test rig, the journals with their clamped-on weights will vibrate heavily sideways. This sideway movement of the clamped-on weights can be reduced by having two crank throws, especially if the cranks are almost in the same direction. However, the journal in the middle will move more.

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

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

Fig. 6 An example of testing arrangement of the resonance tester for bending loading

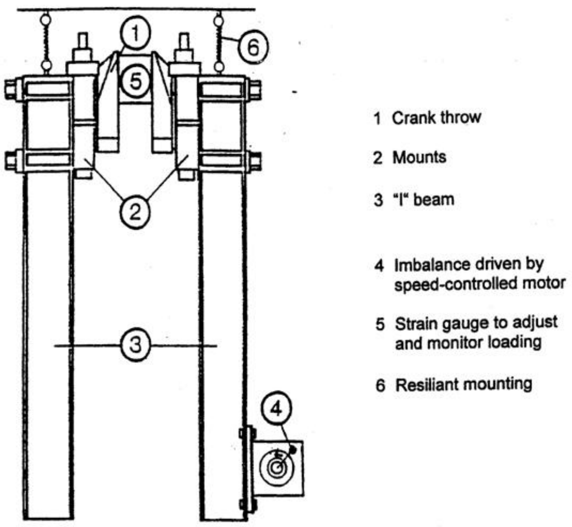
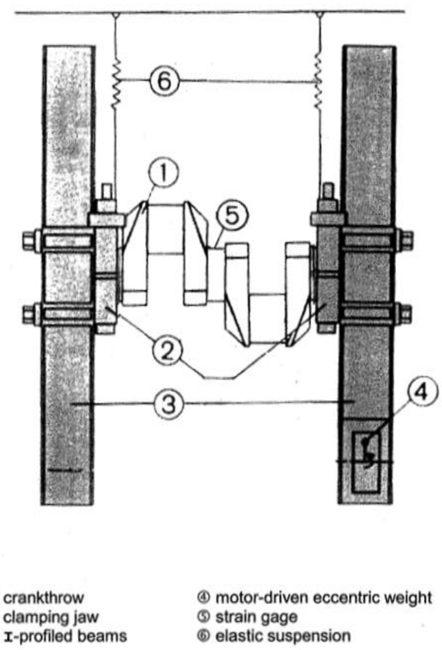


Fig. 7 An example of testing arrangement of the resonance tester for torsion loading with double crank throw section



4.1.3 Use of Results and Crankshaft Acceptability

1 In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability (See **1.8** of **Annex D2.3.1-2(2)**), the Gough-Pollard approach can be applied for the following cases:

(1) Related to the crankpin diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BH}}{\sigma_{DWCT}} \right)^2 + \left(\frac{\tau_{BH}}{\tau_{DWCT}} \right)^2} \right)^{-1}$$

where:

σ_{DWCT}: fatigue strength by bending testing

τ_{DWCT}: fatigue strength by torsion testing

(2) Related to crankpin oil bore:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BO}}{\sigma_{DWOT}} \right)^2 + \left(\frac{\tau_{BO}}{\tau_{DWOT}} \right)^2} \right)^{-1}$$

where:

σ_{DWOT}: fatigue strength by bending testing

τ_{DWOT}: fatigue strength by torsion testing

(3) Related to the journal diameter:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BG}}{\sigma_{DWJT}} \right)^2 + \left(\frac{\tau_{G}}{\tau_{DWJT}} \right)^2} \right)^{-1}$$

where:

σ_{DWJT}: fatigue strength by bending testing

τ_{DWJT}: fatigue strength by torsion testing.

2 In case increase in fatigue strength due to the surface treatment is considered to be similar between the above cases, it is sufficient to test only the most critical location according to the calculation where the surface treatment had not been taken into account.

5.1 Use of Existing Results for Similar Crankshafts

5.1.1 Use of Existing Results

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

(1) Material:
  (a) Similar material type
  (b) Cleanliness on the same or better level
  (c) The same mechanical properties can be granted (size versus hardenability)

(2) Geometry:
  (a) Difference in the size effect of stress gradient is insignificant or it is considered
  (b) Principal stress direction is equivalent. (See **3.1**)

(3) Manufacturing:
  (a) Similar manufacturing process

2 Induction hardened or gas nitrited crankshafts will suffer fatigue either at the surface or at the transition to the core. The surface fatigue strength as determined by fatigue tests of full size cranks, may be used on an equal or similar design as the tested crankshaft when the fatigue initiation occurred at the surface. With the similar design, it is meant that a similar material type and surface hardness are used and the fillet radius and hardening depth are within approximately $\pm 30\%$ of the tested crankshaft.

3 Fatigue initiation in the transition zone can be either subsurface, i.e. below the hard layer, or at the surface where the hardening ends. The fatigue strength at the transition to the core can be determined by fatigue tests as described above, provided that the fatigue initiation occurred at the transition to the core. Tests made with the core material only will not be representative since the tension residual stresses at the transition are lacking.

4 It has to be noted also what some recent research has shown:

The fatigue limit can decrease in the very high cycle domain with subsurface crack initiation due to trapped hydrogen that accumulates through diffusion around some internal defect functioning as an initiation point. In these cases, it would be appropriate to reduce the fatigue limit by some percent per decade of cycles beyond 10^7. Based on a publication by Yukitaka Murakami "Metal Fatigue: Effects of Small Defects and Non-metallic Inclusions" the reduction is suggested to be 5 % per decade especially when the hydrogen content is considered to be high.

Appendix D3 has been added as follows.

Appendix D3 Guidance for Calculation of Surface Treated Fillets and Oil Bore Outlets

1.1 Introduction

This appendix deals with surface treated fillets and oil bore outlets. The various treatments are explained and some empirical formulae are given for calculation purposes.
Please note that measurements or more specific knowledge should be used if available. However, in the case of a wide scatter (e.g. for residual stresses) the values should be chosen from the end of the range that would be on the safe side for calculation purposes.

2.1 Definition of Surface Treatment

"Surface treatment" is a term covering treatments such as thermal, chemical or mechanical operations, leading to inhomogeneous material properties - such as hardness, chemistry or residual stresses - from the surface to the core.

2.2 Surface Treatment Methods

The following list given in **Table 1** covers possible treatment methods and how they influence the properties that are decisive for the fatigue strength.

Table 1 Surface treatment methods and the characteristics they affect.

Treatment method	Affecting
Induction hardening	Hardness and residual stresses
Nitriding	Chemistry, hardness and residual stresses
Case hardening	Chemistry, hardness and residual stresses
Die quenching (no temper)	Hardness and residual stresses
Cold rolling	Residual stresses
Stroke peening	Residual stresses
Shot peening	Residual stresses
Laser pening	Residual stresses
Ball coining	Residual stresses

Note:
It is important to note that since only induction hardening, nitriding, cold rolling and stroke peening are considered relevant for marine engines, other methods as well as combination of two or more of the above are not dealt with in this appendix. In addition, die quenching can be considered in the same way as induction hardening.

3.1 Calculation Principles

3.1.1 General

1 The basic principle is that the alternating working stresses is to be below the local fatigue strength (including the effect of surface treatment) wherein non-propagating cracks may occur. (See also **6.1.2** for details) This is then divided by a certain safety factor.
2 This applies through the entire fillet or oil bore contour as well as below the surface to a depth

below the treatment - affected zone - i.e. to cover the depth all the way to the core.

3 Consideration of the local fatigue strength is to include the influence of the local hardness, residual stress and mean working stress.

4 The influence of the “giga-cycle effect”, especially for initiation of subsurface cracks, should be covered by the choice of safety margin.

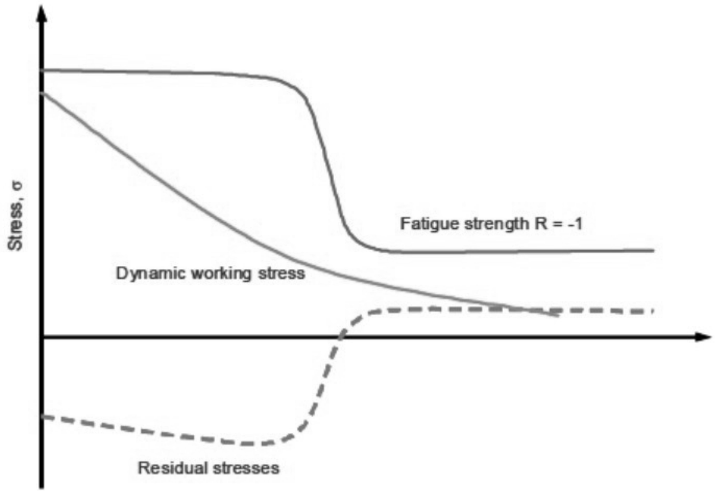
5 It is of vital importance that the extension of hardening/peening in an area with concentrated stresses be duly considered.

6 Any transition where the hardening/peening is ended is likely to have considerable tensile residual stresses. This forms a “weak spot” and is important if it coincides with an area of high stresses.

7 Alternating and mean working stresses are to be known for the entire area of the stress concentration as well as to a depth of about 1.2 times the depth of the treatment. (See **Fig. 1**)

8 The acceptability criterion should be applied stepwise from the surface to the core as well as from the point of maximum stress concentration along the fillet surface contour to the web.

Fig. 1 Stresses as functions of depth, general principles (In case of induction hardening)



Note:
The base axis is either the depth (perpendicular to the surface) or along the fillet contour.

3.2 Evaluation of Local Fillet Stresses

3.2.1 Evaluation Based upon FEM

It is necessary to have knowledge of the stresses along the fillet contour as well as in the subsurface to a depth somewhat beyond the hardened layer. Normally this will be found via FEA as described in **Appendix D3**. However, the element size in the subsurface range will have to be the same size as at the surface. For crankpin hardening only the small element size will have to be continued along the surface to the hard layer.

3.2.2 Evaluation Based upon a Simplified Approach

1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined stress concentration factors (SCFs), as in **1.4** of **Annex D2.3.1-2(2)** if within its validity range, and a relative stress gradient inversely proportional to the fillet radius. Bending and torsional stresses are to be addressed separately. The combination of these is addressed by the acceptability criterion.

2 The subsurface transition-zone stresses, with the minimum hardening depth, can be determined by means of local stress concentration factors along an axis perpendicular to the fillet surface.

(1) Calculation of the local SCFs $\alpha_{B\text{-}local}$ and $\beta_{B\text{-}local}$ for bending in crankpin and journal fillets is as follows: (See **Fig. 2**)

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

$$\beta_{B-local} = (\beta_B - 1) \cdot e^{\frac{-2\cdot t}{R_G}} + 1 - \left(\frac{2 \cdot t}{\sqrt{W^2 + S^2}} \right)^{\frac{0.6}{\sqrt{\beta_B}}}$$

For parameters see **1.3.1-3** and **1.4** of **Annex D2.3.1-2(2)**

(2) Calculation of the local SCFs $\alpha_{T\text{-}local}$ and $\beta_{T\text{-}local}$ for torsion in crankpin and journal fillets is as follows: (See **Fig. 3**)

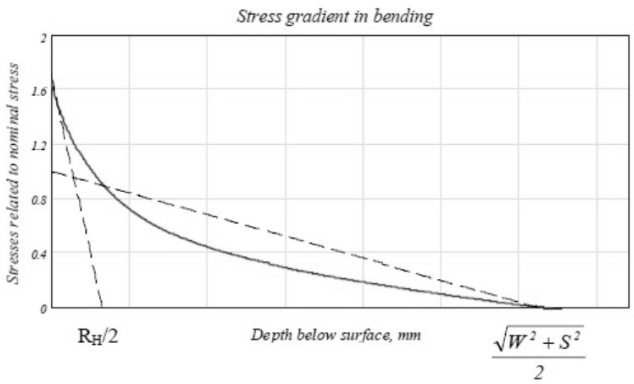
$$\alpha_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_H}} + 1 - \left(\frac{2 \cdot t}{D} \right)^{\frac{0.6}{\sqrt{\alpha_T}}}$$

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

For parameters see **1.3.1-3** and **1.4** of **Annex D2.3.1-2(2)**

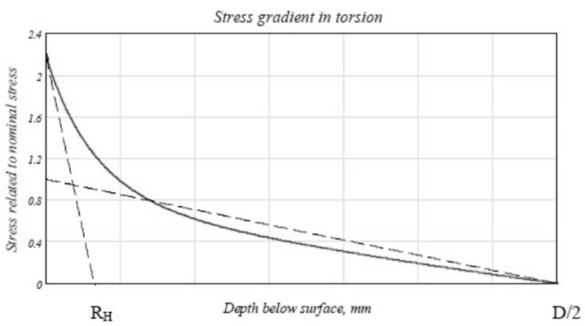
3 If the pin is hardened only and the end of the hardened zone is closer to the fillet than three times the maximum hardness depth, FEA should be used to determine the actual stresses in the transition zone.

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



Note:
The corresponding SCF for the journal fillet can be found by replacing R_H with R_G

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



Note:
The corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G

3.3 Evaluation of Oil Bore Stresses

3.3.1 Evaluation Based upon FEM

1 Stresses in the oil bores can be determined also by FEA.

2 The element size should be less than 1/8 of the oil bore diameter D_O and the element mesh quality criteria should be followed as prescribed in **Appendix D1**.

3 The fine element mesh should continue well beyond a radial depth corresponding to the hardening depth.

4 The loads to be applied in the FEA are the torque and the bending moment, with four-point bending. (See **3.1.1** and **3.1.2** of **Appendix D1**)

3.3.2 Evaluation Based upon a Simplified Approach

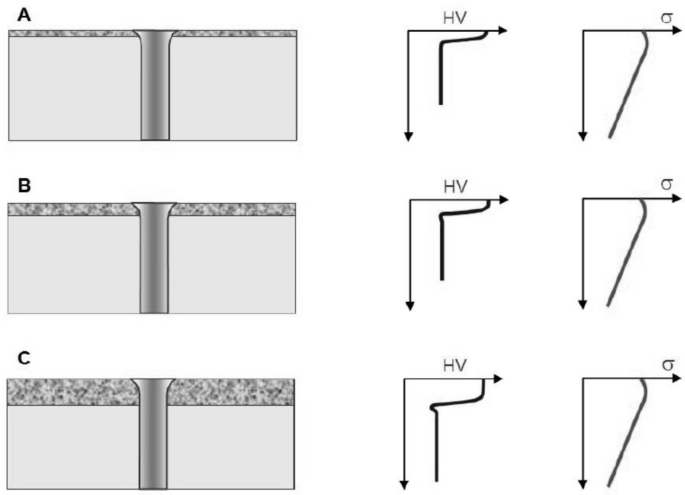
1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from **1.3** of **Annex D2.3.1-2(2)** if within its applicability range.

2 Bending and torsional stresses at the point of peak stresses are combined as in **1.6** of **Annex D2.3.1-2(2)**.

3 **Fig. 4** indicates a local drop of the hardness in the transition zone between a hard and soft material. Whether this drop occurs depends also on the tempering temperature after quenching in the QT process.

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

Fig. 4 Stresses and hardness in induction hardened oil holes



5 The subsurface transition-zone stresses (using the minimum hardening depth) can be determined by means of local stress concentration factors along an axis perpendicular to the oil bore surface.

(1) Calculation of the local SCF $\gamma_{B-local}$ for bending in crankpin oil bores is as follows:

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

For parameters see **1.3.1-3** and **1.4** of **Annex D2.3.1-2(2)**

(2) Calculation of the local SCF $\gamma_{T-local}$ for torsion in crankpin oil bores is as follows:

$$\gamma_{T-local} = (\gamma_T - 1) \cdot e^{\frac{-2t}{D_o}} + 1$$

For parameters see **1.3.1-3** and **1.4** of **Annex D2.3.1-2(2)**

3.4 Acceptability Criteria

The acceptability factors of crankpin fillets, journal fillets and the outlets of crankpin oil bores are to comply with the following criteria, which is specified in **1.8** of **Annex D2.3.1-2(2)**:

$Q \geq 1.15$

4.1 Induction Hardening

4.1.1 General

1 Generally, the hardness specification is to specify the surface hardness range i.e. minimum and maximum values, the minimum and maximum extension in or through the fillet and also the minimum and maximum depth along the fillet contour. The referenced Vickers hardness is considered to be *HV*0.5...*HV*5.

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

3 In the case of crankpin or journal hardening only, the minimum distance to the fillet is to be specified due to the tensile stress at the heat-affected zone as shown in **Fig. 5**.

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

(1) The hardness profile consists of two layers (See **Fig. 6**):
  (a) Constant hardness from the surface to the transition zone
  (b) Constant hardness from the transition zone to the core material

(2) Residual stresses in the hard zone of 200 *MPa* (compression)

(3) Transition-zone hardness as 90 % of the core hardness unless the local hardness drop is avoided

(4) Transition-zone maximum residual stresses (von Mises) of 300 *MPa* (tension)

5 If the crankpin or journal hardening ends close to the fillet, the influence of tensile residual stresses has to be considered. If the minimum distance between the end of the hardening and the beginning of the fillet is more than 3 times the maximum hardening depth, the influence may be disregarded.

Fig. 5 Residual stresses along the surface of a pin and fillet

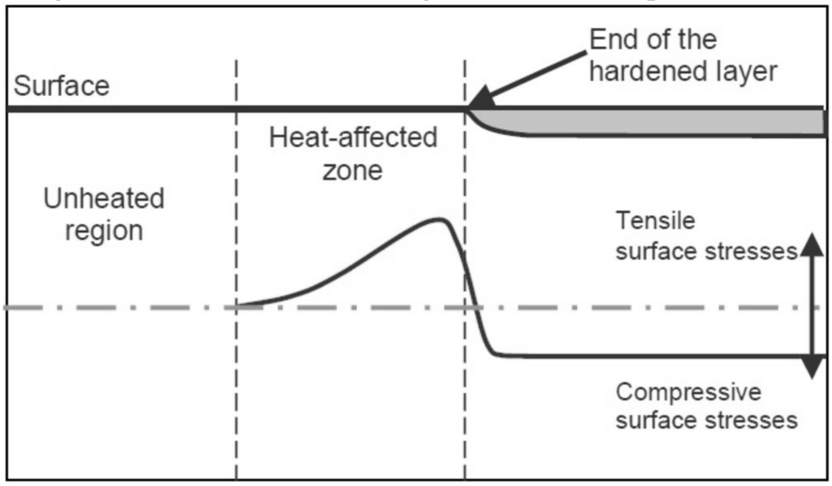
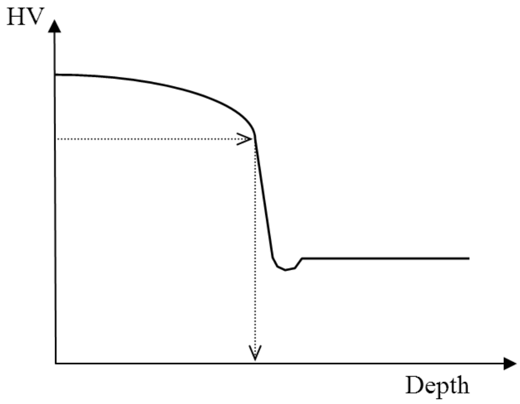


Fig. 6 Typical hardness as a function of depth.



Note:
The arrows indicate the defined hardening depth. Note the indicated potential hardness drop at the transition to the core. This can be a weak point as local strength may be reduced and tensile residual stresses may occur.

4.2 Local Fatigue Strength

4.2.1 General

Induction-hardened crankshafts will suffer fatigue either at the surface or at the transition to the core.

4.2.2 Evaluation Based upon Fatigue Testing

1 The fatigue strengths, for both the surface and the transition zone, can be determined by fatigue testing of full size cranks as described in **Appendix D2**.
2 In the case of a transition zone, the initiation of the fatigue can be either subsurface (i.e. below the hard layer) or at the surface where the hardening ends.
3 Tests made with the core material only will not be representative since the tensile residual stresses at the transition are lacking.

4.2.3 Evaluation Based upon Calculations

1 The surface fatigue strength can be determined empirically as follows:

$$\sigma_{Fsurface} = 400 + 0.5 \cdot (HV - 400) \quad [MPa]$$

where

HV : surface Vickers hardness

The equation provides a conservative value, with which the fatigue strength is assumed to include the influence of the residual stress. The resulting value is valid for a working stress ratio of $R = -1$. It has to be noted also that the mean stress influence of induction-hardened steels may be significantly higher than that for QT steels.

2 The fatigue strength in the transition zone, without taking into account any possible local hardness drop, is to be determined by the following:

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

where

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O / 2$ for oil bore outlet

For parameters see **1.4** of **Annex D2.3.1-2(2)**

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

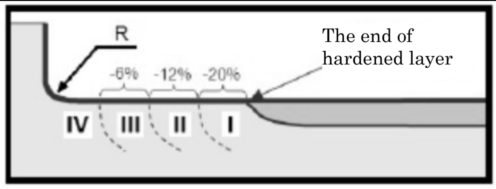
3 For the purpose of considering subsurface fatigue, below the hard layer, the disadvantage of tensile residual stresses has to be considered by subtracting 20% from the value determined above. This 20% is based on the mean stress influence of alloyed quenched and tempered steel having a residual tensile stress of 300 MPa.

4 When the residual stresses in **-3** are known to be lower, also smaller value of subtraction is to be used. For low-strength steels the percentage chosen should be higher.

5 For the purpose of considering surface fatigue near the end of the hardened zone - i.e. in the heat-affected zone shown in the **Fig. 5** - the influence of the tensile residual stresses can be considered by subtracting a certain percentage, in accordance with **Table 2**, from the value determined by the above formula.

Table 2 The influence of tensile residual stresses at a given distance from the end of the hardening towards the fillet

Area	Distance from the end of the hardening towards the fillet	Ratio
I	0 to 1.0 of the max. hardening depth	20%
II	1.0 to 2.0 of the max. hardening depth	12%
III	2.0 to 3.0 of the max. hardening depth	6%
IV	3.0 or more of the max. hardening depth	0%



5.1 Nitriding

5.1.1 General

1 The hardness specification is to include the surface hardness range (min and max) and the minimum and maximum depth.

2 Only gas nitriding is considered.

3 The referenced Vickers hardness is considered to be HV 0.5.

4 The nitriding depth tN is defined as the depth to a hardness of 50 HV above the core hardness.

5 The hardening profile should be specified all the way to the core.

6 If this is not known, it may be determined empirically via the following formula:

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

where:

t : The local depth

$HV(t)$: Hardness at depth t

HV_{core} : Core hardness (minimum)

$HV_{surface}$: Surface hardness (minimum)

tN : Nitriding depth as defined above (minimum)

5.2 Local Fatigue Strength

5.2.1 General

It is important to note that in nitrided crankshaft cases, fatigue is found either at the surface or at the transition to the core.

5.2.2 Evaluation Based on Fatigue Testing

The fatigue strength can be determined by tests as described in **Appendix D2**.

5.2.3 Evaluation Based on Calculations

1 Alternatively, the surface fatigue strength (principal stress) can be determined empirically and conservatively as follows:

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

This is valid for a surface hardness of 600 HV or greater.

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

2 The fatigue strength in the transition zone can be determined via the following formula:

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

where:

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O / 2$ for oil bore outlet

Note that this fatigue strength is not assumed to include the influence of the residual stresses.

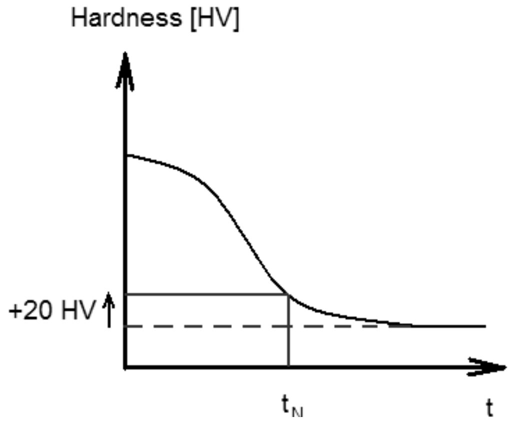
3 In contrast to induction-hardening the nitrited components have no such distinct transition to the core. Although the compressive residual stresses at the surface are high, the balancing tensile stresses in the core are moderate because of the shallow depth.

4 For the purpose of analysis of subsurface fatigue the disadvantage of tensile residual stresses in and below the transition zone may be even disregarded in view of this smooth contour of a nitriding hardness profile.

5 Although in principle the calculation should be carried out along the entire hardness profile, it can be limited to a simplified approach of examining the surface and an artificial transition point. (See **Fig. 7**)

6 This artificial transition point can be taken at the depth where the local hardness is approximately 20 HV above the core hardness. In such a case, the properties of the core material should be used. This means that the stresses at the transition to the core can be found by using the local SCF formulae mentioned in **3.2.2** or **3.3.2** when inserting $t = 1.2tN$.

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



6.1 Cold Forming

6.1.1 General

1 The advantage of stroke peening or cold rolling of fillets is the compressive residual stresses introduced in the high-loaded area.

2 The fatigue strength has to be determined by fatigue testing (See also **Appendix D2**). Such testing is normally carried out as four-point bending, with a working stress ratio of $R = -1$.

3 From these results, the bending fatigue strength - surface - or subsurface-initiated depending on the manner of failure - can be determined and expressed as the representative fatigue strength for applied bending in the fillet.

4 In comparison to bending, the torsion fatigue strength in the fillet may differ considerably from the ratio $\sqrt{3}$ (utilized by the von Mises criterion). The forming-affected depth that is sufficient to prevent subsurface fatigue in bending, may still allow subsurface fatigue in torsion. Another possible reason for the difference in bending and torsion could be the extension of the highly stressed area.

5 The results obtained in a full-size crank test can be applied for another crank size provided that the base material (alloyed Q+T) is of the similar type and that the forming is done so as to obtain the similar level of compressive residual stresses at the surface as well as through the depth. This means that both the extension and the depth of the cold forming are to be proportional to the fillet radius.

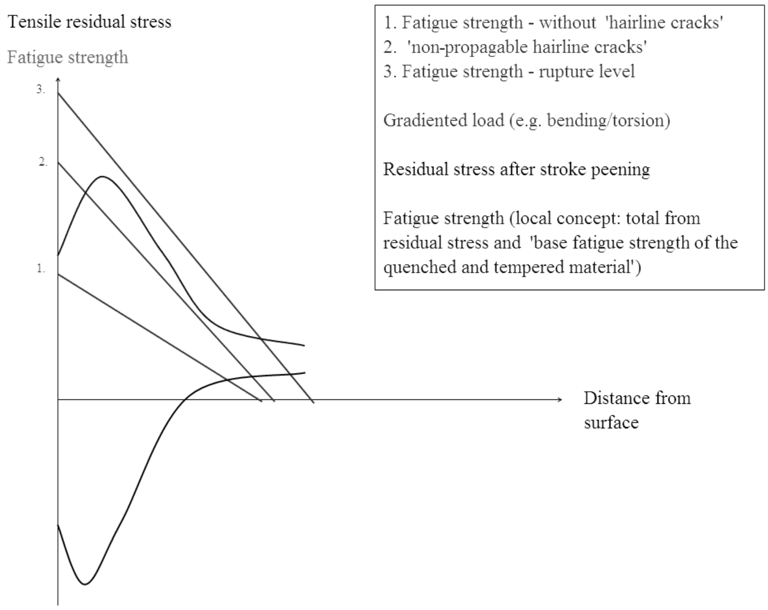
6.1.2 Stroke Peening by Means of a Ball

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

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

3 As a result of the stroke peening process the maximum of the compressive residual stress is found in the subsurface area. Therefore, depending on the fatigue testing load and the stress gradient, it is possible to have higher working stresses at the surface in comparison to the local fatigue strength of the surface. Because of this phenomenon small cracks may appear during the fatigue testing, which will not be able to propagate in further load cycles and/or with further slight increases of the testing load because of the profile of the compressive residual stress. Put simply, the high compressive residual stresses below the surface “arrest” small surface cracks. (See 2. in **Fig. 8**)

Fig. 8 Working and residual stresses below the stroke-peened surface.



Note:
Straight lines 1…3 represent different possible load stress gradients.

4 In fatigue testing with full-size crankshafts these small “hairline cracks” should not be considered to be the failure crack. The crack that is technically the fatigue crack leading to failure, and that therefore shuts off the test-bench, should be considered for determination of the failure load level. This also applies if induction-hardened fillets are stroke-peened.

5 In order to improve the fatigue strength of induction-hardened fillets it is possible to apply the stroke peening process in the crankshafts’ fillets after they have been induction-hardened and tempered to the required surface hardness. If this is done, it might be necessary to adapt the stroke peening force to the hardness of the surface layer and not to the tensile strength of the base material.

6 The effect on the fatigue strength of induction hardening and stroke peening the fillets is to be determined by a full-size crankshaft test.

6.1.3 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved by applying stroke peening, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:

(1) Ball size relative to fillet radius within ±10% in comparison to the tested crankshaft
(2) At least the same circumferential extension of the stroke peening
(3) Angular extension of the fillet contour relative to fillet radius within ±15% in comparison to the tested crankshaft and located to cover the stress concentration during engine operation
(4) Similar base material, e.g. alloyed quenched and tempered
(5) Forward feed of ball of the same proportion of the radius
(6) Force applied to ball proportional to base material hardness (if different)

(7) Force applied to ball proportional to square of ball radius

6.1.4 Cold Rolling

1 The fatigue strength can be obtained by means of full size crank tests or by empirical methods, if these are applied so as to be on the safe side.

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

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

6.1.5 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved applying cold rolling, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:

(1) At least the same circumferential extension of cold rolling

(2) Angular extension of the fillet contour relative to fillet radius within $\pm 15\%$ in comparison to the tested crankshaft and located to cover the stress concentration during engine operation

(3) Similar base material, e.g. alloyed quenched and tempered

(4) Roller force to be calculated so as to achieve at least the same relative (to fillet radius) depth of treatment

Appendix D4 has been added as follows.

Appendix D4 Guidance for Calculation of Stress Concentration Factors in the Oil Bore Outlets of crankshafts through utilisation of the Finite Element Method

1.1 General

The objective of the analysis described in this Appendix is to substitute the analytical calculation of the stress concentration factor (SCF) at the oil bore outlet with suitable finite element method (FEM) calculated figures. The former method is based on empirical formulae developed from strain gauge readings or photo-elasticity measurements of various round bars. In cases where these formulae are outside their applicable scope, the FEM-based method is to be used.
The SCF calculated according to the rules set forth in this Appendix is defined as the ratio of FEM-calculated stresses to nominal stresses calculated analytically. In use in connection with the present method in **Annex D2.3.1-2(2)**, principal stresses are to be calculated.
The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.
It is advisable to check the element accuracy of the FE solver in use, e.g. by modelling a simple geometry and comparing the FEM-obtained stresses with the analytical solution.
A boundary element method (BEM) approach may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and assumptions for building of the FE-model are presented in **2.1.1**. The final FE-model is to meet one of the criteria in **2.2**.

2.1.1 Element Mesh Recommendations

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

(1) The model consists of one complete crank, from the main bearing centre line to the opposite side's main bearing centre line.
(2) The following element types are used in the vicinity of the outlets:
  (a) 10-node tetrahedral elements
  (b) 8-node hexahedral elements
  (c) 20-node hexahedral elements
(3) The following mesh properties for the oil bore outlet are used:
  (a) Maximum element size $a = r/4$ through the entire outlet fillet as well as in the bore direction (if 8-node hexahedral elements are used, even smaller elements are required for meeting of the quality criterion)
  (b) Recommended manner for element size in the fillet depth direction:
    i) First layer's thickness equal to element size of a
    ii) Second layer's thickness equal to element size of $2a$
    iii) Third-layer thickness equal to element size of $3a$
(4) The rest of the crank should be suitable for numeric stability of the solver
(5) Drillings and holes for weight reduction have to be modelled
(6) Submodeling may be used as long as the software requirements are fulfilled.

2.1.2 Material

1 Material properties applied to steels as follows.

Young's Modulus : $E = 2.05 \cdot 10^{5}\,MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters have to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

If the actual element mesh does not fulfil any of the following criteria in the area examined for SCF evaluation, a second calculation, with a finer mesh is to be performed.

2.2.1 Principal-stresses Criterion

The quality of the mesh is to be assured through checking of the stress component normal to the surface of the oil bore outlet radius. With principal stresses σ_1, σ_2 and σ_3 the following criterion is to be met:

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

2.2.2 Averaged/Unaveraged-stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than 5 % from the 100 % averaged nodal stress results at this node at the location examined.

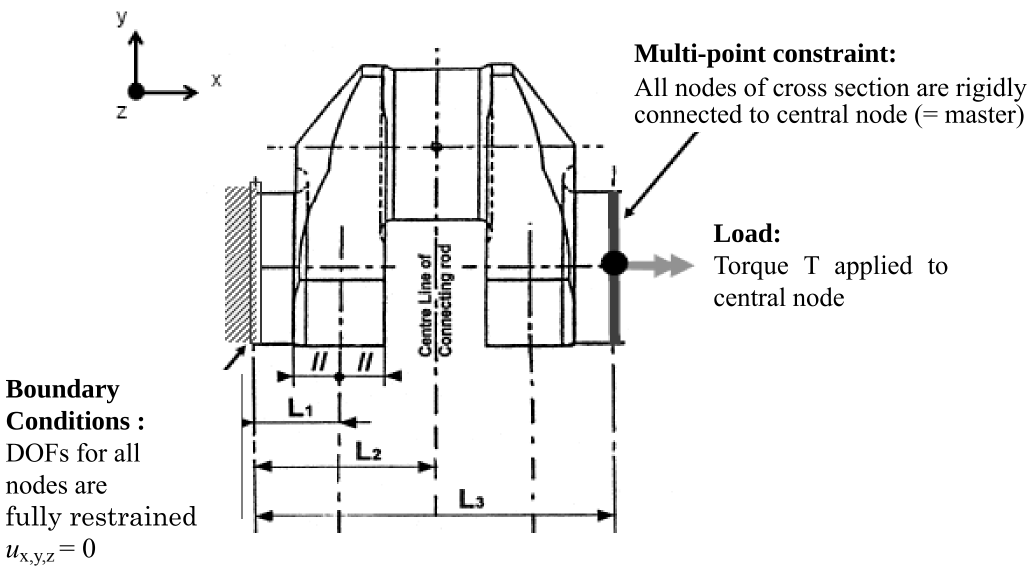
3.1 Load Cases and Assessment of Stress

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 1** where the torque is applied to the central node located at the crankshaft axis.

Fig. 1 Boundary and load conditions for the torsion load case



2 For all nodes in an oil bore outlet, the principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

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

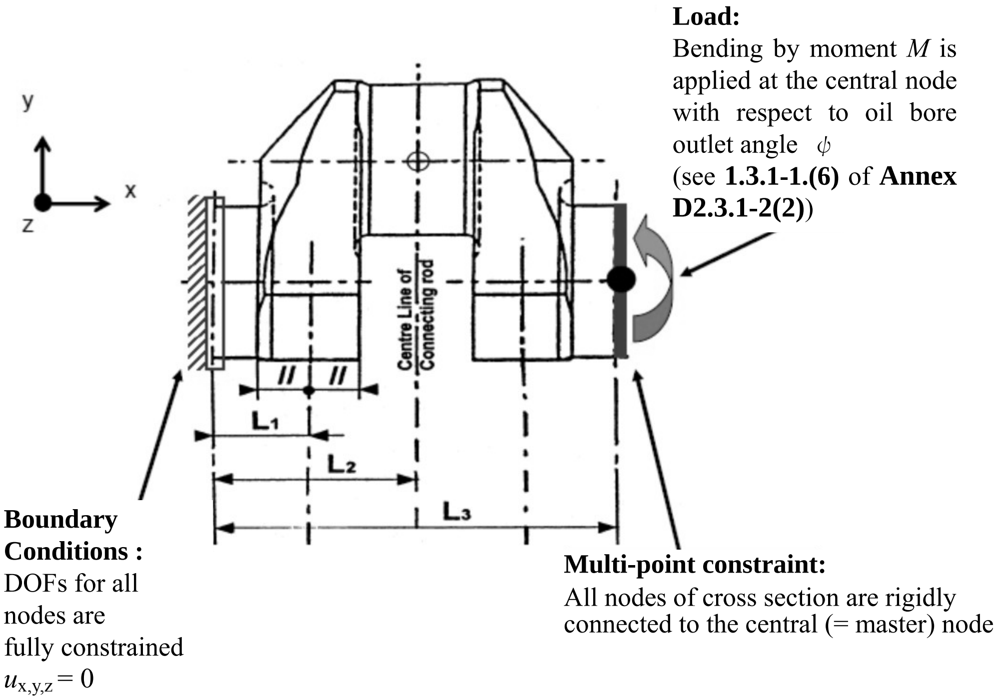
where the nominal torsion stress τ_N referred to the crankpin is calculated as follows (for W_P see **1.3.2** of **Annex D2.3.1-2(2)**) :

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

3.1.2 Bending

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 2** where the bending moment is applied to the central node located at the crankshaft axis.

Fig. 2 Boundary and load conditions for the pure bending load case



2 For all nodes in the oil bore outlet, principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_B = \frac{\max(|\sigma_1|,|\sigma_2|,|\sigma_3|)}{\sigma_N}$$

where the nominal bending stress σ_N referred to the crankpin is calculated as follows (for W_e see **1.3.2** of **Annex D2.3.1-2(2)**):

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

EFFECTIVE DATE AND APPLICATION (Amendment 1-5)

1. The effective date of the amendments is 1 July 2018.

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