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Cranes — General design — Limit states and proof of competence of forged steel hooks

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National foreword

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**Cranes — General design — Limit
states and proof of competence of
forged steel hooks**

*Appareils de levage à charge suspendue — Conception générale —
États limites et vérification d'aptitude des crochets forgés*



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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2. www.iso.org/directives

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: [Foreword - Supplementary information](#)

The committee responsible for this document is ISO/TC 96, *Cranes*, Subcommittee SC 8, *Jib cranes*.

Cranes — General design — Limit states and proof of competence of forged steel hooks

1 Scope

This International Standard is intended to be used together with the other relevant International Standards in its series. As such, they specify general conditions, requirements and methods to prevent hazards in hooks as part of all types of cranes.

This International Standard covers the following parts of hooks and types of hooks:

- bodies of any type of point hooks made of steel forgings;
- machined shanks of hooks with a thread/nut suspension.

NOTE 1 The principles of this International Standard can be applied to other types of shank hooks and also where stress concentration factors relevant to that shank construction are determined and used. Plate hooks, which are those assembled from one or several parallel parts of rolled steel plates are not covered in this International Standard.

This International Standard is applicable to hooks from materials with ultimate strength of not more than 800 N/mm² and yield stress of not more than 600 N/mm².

The following is a list of significant hazardous situations and hazardous events that could result in risks to persons during normal use and foreseeable misuse. [Clauses 4 to 8](#) of this document are necessary to reduce or eliminate the risks associated with the following hazards:

- a) exceeding the limits of strength (yield, ultimate, fatigue);
- b) exceeding temperature limits of material;
- c) unintentional disengagement of the load from the hook.

The requirements of this International Standard are stated in the main body of the document and are applicable to hook designs in general. The hook body and shank designs listed in [Annexes A, B and G](#) are only examples and should not be referred to as requirements of this International Standard.

This International Standard is applicable to cranes manufactured after the date of its publication, and serves as a reference base for product standards of particular crane types.

NOTE 2 This International Standard deals only with the limit state method in accordance with ISO 8686-1.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 148-1, *Metallic materials — Charpy pendulum impact test — Part 1: Test method*

ISO 148-2, *Metallic materials — Charpy pendulum impact test — Part 2: Verification of testing machines*

ISO 643, *Steels — Micrographic determination of the apparent grain size*

ISO 965-1, *ISO general purpose metric screw threads — Tolerances — Part 1: Principles and basic data*

ISO 4287, *Geometrical Product Specifications (GPS) — Surface texture: Profile method — Terms, definitions and surface texture parameters*

ISO 4306-1, *Cranes — Vocabulary — Part 1: General*

ISO 4301-1, *Cranes and lifting appliances — Classification — Part 1: General*

ISO 6892-1, *Metallic materials — Tensile testing — Part 1: Method of test at room temperature*

ISO 8686-1, *Cranes — Design principles for loads and load combinations — Part 1: General*

ISO 9327-1, *Steel forgings and rolled or forged bars for pressure purposes — Technical delivery conditions — Part 1: General requirements*

ISO 7500-1, *Metallic materials — Verification of static uniaxial testing machines — Part 1: Tension/compression testing machines — Verification and calibration of the force-measuring system*

ISO 12100, *Safety of machinery — General principles for design — Risk assessment and risk reduction*

ISO 15579, *Metallic materials — Tensile testing at low temperature*

EN 10228-3, *Non-destructive testing of steel forgings — Part 3: Ultrasonic testing of ferritic or martensitic steel forgings*

EN 10243-1, *Steel die forgings — Tolerances on dimensions — Part 1: Drop and vertical press forgings*

3 Terms, definitions and symbols

For the purposes of this document, the terms and definitions given in ISO 12100 and ISO 4306-1 and the following terms, definitions and symbols (see [Table 1](#)) apply.

3.1

hook shank

upper part of the hook, from which the hook is suspended to the hoist media of the crane

3.2

hook body

lower, curved part of the hook below the shank

3.3

hook seat

bottom part of the hook body, where the load lifting attachment is resting

3.4

hook suspension articulation

feature of the hook suspension, allowing the hook to tilt along the inclined load line

Table 1 — Symbols

| Symbol | Description |
|----------------------|---|
| A_{d1} | Cross section area of the forged, shank |
| A_{d4} | Cross section area of the critical section of hook shank |
| A_V | Minimum impact toughness of material |
| a | Acceleration |
| a_1 | Seat circle diameter |
| a_2 | Throat opening |
| a_3 | Height of the hook point |
| b_{max} | Maximum width in the critical hook body section |
| b_{ref} | Reference width |
| C | Total number of working cycles during the design life of crane |
| C_t | Relative tilting resistance of the hook suspension |
| c_e | Coefficient for load eccentricity |
| D | Cumulative damage in fatigue (Palmgren-Miner hypothesis) |
| d_1 | Diameter of the forged shank |
| d_3 | Principal diameter of thread |
| d_4 | Diameter of the undercut section of the shank |
| d_5 | Thread core diameter |
| e_R | Distance of the vertical load line from the centre line of the shank |
| F | Vertical force |
| F_H | Vertical force on hook due to occasional or exceptional loads |
| $F_{Rd,s}, F_{Rd,f}$ | Limit design forces, static / fatigue |
| $F_{Sd,s}$ | Vertical design force for the proof of static strength |
| $F_{Sd,f}$ | Vertical design force for the proof of fatigue strength |
| f_1, f_2, f_3 | Factors of further influences |
| f_{Rd} | Limit design stress |
| f_y | Yield stress |
| f_u | Ultimate strength |
| g | Acceleration due to gravity, $g = 9,81 \text{ m/s}^2$ |
| $H_{Sd,s}$ | Horizontal design force of hook |
| $H_{Sd,f}$ | Horizontal design force for the proof of fatigue strength |
| h_1, h_2 | Section heights of the hook body |
| h | Vertical distance from the seat bottom of the hook body to the centre of the articulation |
| h_s | Vertical distance from the seat bottom of the hook body to critical section of hook shank |
| i | Index for a lifting cycle or a stress cycle |
| I | Reference moment of inertia for curved beam |
| I_{d1} | Moment of inertia of the forged shank |
| I_{d4} | Moment of inertia of the critical section of hook shank |
| k_C | Conversion factor for stress spectrum and classified duty |
| k_h, k_s | Stress spectrum factors |
| k_Q | Load spectrum factor, in accordance with ISO 8686-1 |

Table 1 (continued)

| Symbol | Description |
|-----------------------------------|--|
| k_5^*, k_6^* | Specific spectrum ratio factors, $m = 5 / 6$ |
| lg | Log to the base of 10 |
| M_1, M_2, M_3, M_4 | Bending moments of hook shank |
| $M_{1,f,i}, M_{2,f,i}, M_{3,f,i}$ | Bending moments of hook shank for the proof of fatigue strength, lifting cycle i |
| $M_{Sd,s}$ | Static design bending moment |
| m | Slope parameter of the characteristic fatigue design curve |
| m_{RC} | Mass of rated hoist load |
| m_i | Mass of the hook load in a lifting cycle i |
| N | Total number of stress cycles / lifting cycles |
| N_D | Reference number of stress cycles, $ND = 2 \times 10^6$ |
| p | Pitch of thread |
| p_a | Average number of accelerations related to one lifting cycle |
| R | Radius of hook body curvature |
| R_a | Average depth of surface profile according to ISO 4287 |
| R_z | Maximum depth of surface profile according to ISO 4287 |
| r_9 | Relief radius of the undercut |
| r_{th} | Thread bottom radius |
| s | Length of undercut |
| s_h, s_s | Stress history parameters |
| s_Q | Load history parameter |
| t | Depth of thread |
| T | Operation temperature |
| u_S, u_T | Depths of notches |
| α | Angle |
| α_S, α_T | Stress concentration factors |
| β | Angle or direction of hook inclination |
| $\beta_n, \beta_{nS}, \beta_{nT}$ | Notch effect factors |
| ϕ_2 | Dynamic factor for hoisting an unrestrained grounded load |
| ϕ_5 | Dynamic factor for changes of acceleration of a movement |
| γ_n | Risk coefficient |
| γ_p | Partial safety factor |
| γ_m | General resistance coefficient |
| γ_{sm} | Specific resistance coefficient |
| $\gamma_{H\&Sf}$ | Fatigue strength specific resistance factors |
| η_1 | Edge distance of a hook body section |
| ν | Factor for load component |
| ν_h, ν_s | Relative numbers of stress cycles |
| μ | Factor for mean stress influence |
| σ_a | Shank stress due to axial force |
| σ_b | Shank stress due to bending moment |
| σ_m | Mean stress in a stress cycle |

Table 1 (continued)

| Symbol | Description |
|---|--|
| σ_A | Stress amplitude in a stress cycle |
| σ_{Sd} | Design stress |
| σ_M | Basic fatigue strength amplitude, un-notched piece |
| σ_p | Total stress range in a pulsating stress cycle |
| σ_W | Fatigue strength amplitude, notched piece |
| $\sigma_{Tmax}, \sigma_{T1}, \sigma_{T2}$ | Transformed stress amplitudes |
| $\Delta\sigma_c$ | Characteristic fatigue strength |
| $\Delta\sigma_{Rd}$ | Limit fatigue design stress |
| $\Delta\sigma_{Sd,i}$ | Stress range in a lifting cycle i |
| $\Delta\sigma_{Sd,max}$ | Maximum stress range |

4 General requirements

4.1 Materials

The hook material in the finished product shall have sufficient ductility to avoid brittle fracture at the temperature range specified for the use of the hook. Hook material, after forging and heat treatment, shall have minimum elongation and Charpy-V impact toughness in accordance with [Table 2](#).

Table 2 — Impact test and elongation requirements for hook material

| Operation temperature | Impact test temperature | Minimum elongation, A_5 | Minimum impact toughness, A_v |
|--|-------------------------|---------------------------|---------------------------------|
| $T \geq -10\text{ °C}$ | 0 °C | 15 % | 35 J |
| $T \geq -20\text{ °C}$ | -10 °C | | |
| $-30\text{ °C} > T \geq -40\text{ °C}$ | -30 °C | | |
| $-40\text{ °C} > T \geq -50\text{ °C}$ | -40 °C | | |

To satisfy the requirements of the operating temperature, the manufacturer shall select an alloyed or non-alloyed steel, as appropriate, which after suitable heat treatment, shall be consistent with achieving the chosen mechanical property grade for the selected hook form, taking into account its individual ruling thickness.

The steel shall be produced by an electric process or by one of the oxygen processes.

The steel shall be fully killed, stabilized against strain age embrittlement and have an austenitic grain size of 6 or finer when tested in accordance with ISO 643. This shall be accomplished, by ensuring that the steel contains sufficient aluminium (minimum 0,025 %) to permit the manufacture of hooks stabilized against strain-age-embrittlement during service.

The steel shall contain no more sulfur and phosphorus than the limits given in [Table 3](#).

Table 3 — Sulfur and phosphorus content

| Element | Maximum mass content as determined by | |
|----------------|---------------------------------------|---------------------|
| | Cast analysis % | Check analysis % |
| Sulfur (S) | 0,020 | 0,025 |
| Phosphorus (P) | 0,020 | 0,025 |
| Sum of S + P | 0,035 | 0,045 |

The mechanical properties (*yield stress, ultimate strength*) are dependent upon the thickness of the forged hook body. As a ruling thickness, either the largest width of the hook seat or the diameter of the shank shall be used, whichever is greater

For standardization purposes, a classification of material grades for forged hooks is specified in [Table 4](#). The values of mechanical properties given in [Table 4](#) shall be used as design values and shall be guaranteed as minimum values by the hook manufacturer.

Table 4 — Material properties for classified material grades

| Material class reference | Mechanical properties | |
|---|--|---|
| | Upper yield stress or 0,2 % proof stress f_y N/mm ² | Ultimate strength f_u N/mm ² |
| M | 215 | 340 |
| P | 315 | 490 |
| S | 380 | 540 |
| T | 500 | 700 |
| V | 600 | 800 |
| All materials selected shall fulfil the following requirement: $f_u/f_y \geq 1,2$ | | |

4.2 Workmanship

The manufacturing process, factory tests and delivery conditions shall meet the requirements of ISO 9327-1.

Each hook body shall be forged hot in one piece. The macroscopic flow lines of the forging shall follow the body outline of the hook. Excess metal from the forging operation shall be removed cleanly leaving the surface free from sharp edges.

Profile cutting from a rolled plate is not permissible for forged hooks.

The surface roughness of the hook seat in the finished product shall be equal to or better than R_z 500 μm . Grinding may be used to reach the required surface quality. Any grinding marks shall be in a circumferential direction in respect to the seat circle.

After heat treatment, furnace scale shall be removed and the hook body shall be free from harmful defects, including cracks. Hook forging shall be inspected for defects using appropriate NDT-methods according to EN 10228-3. Requirements of quality class 1 of EN 10228-3: shall be met.

No welding shall be carried out at any stage of manufacture.

4.3 Manufacturing tolerances

The dimensional tolerances according to EN 10243-1 for forging grade F shall be fulfilled, except as modified herein.

The seat circle diameter and the throat opening shall be within [0; +7 %] of the nominal dimension. The point height dimension a_3 shall be within [- 7 %; +7 %] of the nominal dimension.

The centre line of the machined shank shall not deviate from the seat centre more than $\pm 0,02 a_1$.

The shape of the hook in its own plane shall be such that the centres of the material sections specified by the two flanks of a section shall fall between two parallel planes with a spacing of $0,05 d_1$.

4.4 Heat treatment

Each forged hook shall either be hardened from a temperature above the AC_3 point and tempered, or normalized from a temperature above the AC_3 point. The tempering temperature shall be at least 475 °C.

The normalizing or tempering conditions shall be at least as effective as a temperature of 475 °C maintained for a period of 1 h.

4.5 Proof loading

As part of the manufacturing process, a hook may be proof loaded. This initial proof loading should be conducted at ambient room temperature and can further assist the Quality Assurance Management process as well as improve the fatigue performance of the hook in general. If proof loading is applied, the process of proof loading shall be as follows:

- a) Proof loading shall be applied after the complete manufacturing process. (forging, heat treatment and machining)
- b) The proof load force shall be applied between shank suspension nut and either:
 - i) the base of the hook seat, for a straight line pull, parallel with the vertical axis of the shank, in the case of a single point hook.
 - ii) two opposite contact points on the hook bowl surface consistent with a symmetrical 90 degree sling spread, and with load lines passing thro' the hook bowl centre(s), in the case of ramshorn hooks.
- c) A relative permanent set due to proof loading measured at the gap opening shall not exceed 0,25 %; For batch-produced hooks the proof loading shall be applied to each and every hook in the batch;
- d) The magnitude of the proof load (F_{PL}) should reflect a $1,5f_y$ theoretical maximum tensile stress in the body fibres in section B for single point and section A for ramshorn hooks for the chosen material. The value of this proof load shall be determined as follows relative to either section A(ramshorn) or B(single point) as the case may be:

Single point hook

$$F_{PL,sp} = \frac{1,5f_y M_{hf}}{1\ 000}$$

Ramshorn hook

$$F_{PL,rh} = \frac{1,5f_y M_{hf}}{1\ 000v}$$

where F_{PL} is expressed in kilonewtons (kN), f_y is the yield stress of the chosen material, and M_{hf} is a hook factor, i.e. for the hook intradoses of either section A or B, as the case may be, sample data are depicted within [Annex C](#) for individual hooks of their particular family.

$$v = 0,5x \tan \alpha$$

for section A of ramshorn hooks, $\alpha = 45^\circ$ (see [5.5.3](#))

M_{hf} is derived from the formula

$$I \frac{(1-\eta_1/R)}{(R\eta_1)}$$

All definitions are as per [Annex H](#).

- e) After proof loading, the hook shall be inspected for defects using appropriate NDT-methods and found free from harmful flaws, defects and cracks;
- f) Proof loaded hook shall be stamped with symbol “PL” adjacent to the hook type marking.
- g) The application of proof loading will affect (beneficially) subsequent fatigue performance of the hook. Calculation methodology of an example in [Annex F](#) can be used to quantify this effect.

Steels and in particular high strength steels for hooks due to be subjected to proof loading should be selected with due attention to the need of their adequate ductility.

NOTE 1 Additional benefits derived from the application of proof loading to the QA Management process is not addressed within this standard.

NOTE 2 The maximum stressed tensile fibres under F_{PL} will of course yield and a redistribution of stress will occur, resulting in a permanent compressive stress in this tensile area when the proof load is removed

4.6 Hook body geometry

Proportions of hook sections shall be such that stresses do not exceed stresses in the critical sections specified in [5.5.1](#).

The seat of a hook shall be of circular shape. In a single hook, the centre of curvature shall coincide with the centreline of the machined shank. In a ramshorn hook, the seat circle shall be tangential in respect to the outer edge of the forged shank.

A ramshorn hook shall be symmetrical with respect to the centre line of the shank.

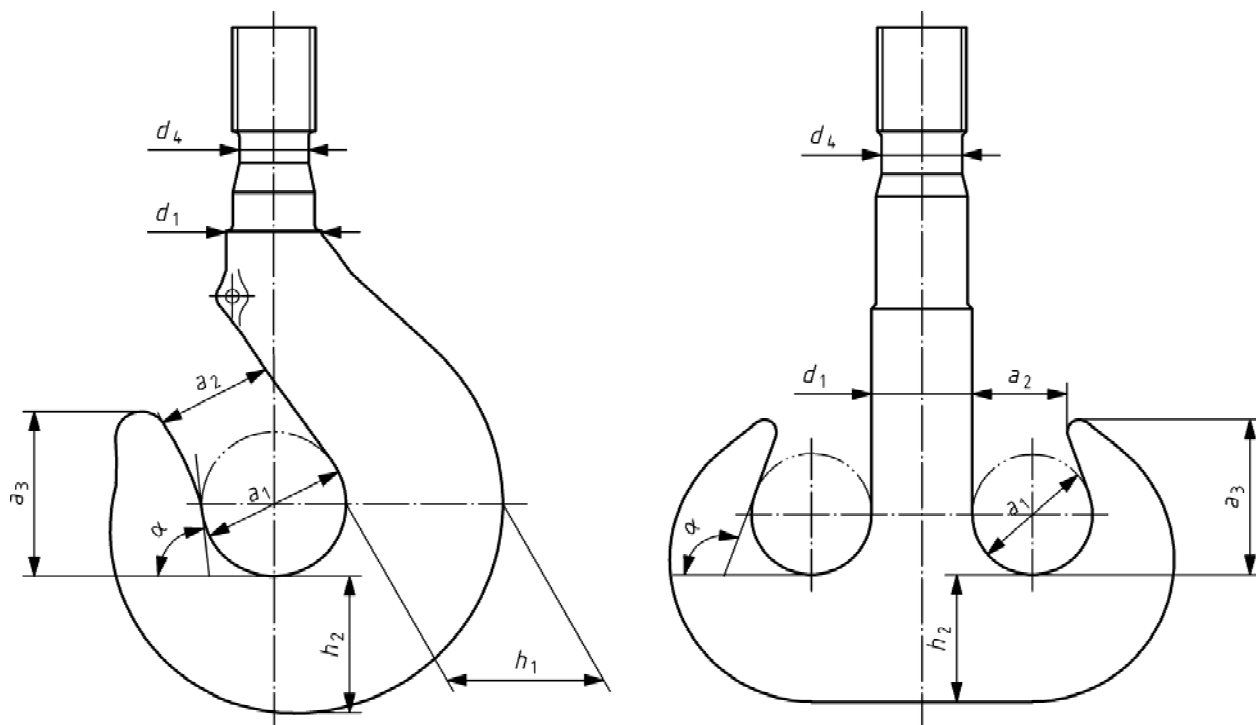


Figure 1 — Hook dimensions

The diameter of the forged shank (d_1) shall be proportioned to circle diameter (a_1) as follows:

$$d_1 \geq 0,55 a_1$$

The bifurcation point between the inner edge and the seat circle (a_1) shall be from the horizontal in minimum as follows: for a single hook $\alpha \geq 60^\circ$, for a ramshorn hook $\alpha \geq 90^\circ$

The full throat opening (a_2), without consideration to a latch shall be proportioned to the seat circle diameter as follows: $a_2 \leq 0,85 a_1$. The effective throat opening with a latch shall be in minimum $a_0 \geq 0,7 a_1$.

The point height of a hook (a_3) shall be in minimum as follows: $a_3 \geq a_1$.

[Annexes A](#) and [B](#) present example sets of hook body dimensions, which fulfil the requirements of this clause.

Other hook bodies differing from those shown within [Annexes A](#) and [B](#) can be technically assessed, either individually or as national groups to the requirements of this standard, provided dimensional characteristics shown within this clause and material requirements are fulfilled.

Furthermore, it is expected that other hook body sets in addition to those currently shown can and will be put forward for inclusion as national groups, within [Annexes A](#) and [B](#). in the future.

4.7 Hook shank machining

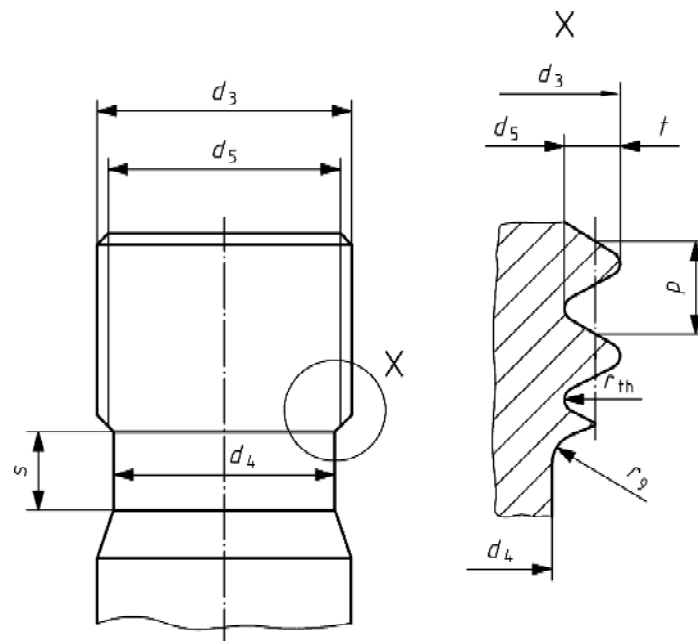


Figure 2 — Machined dimensions of shank

The length of the threaded portion of the shank shall be not less than $0,8d_3$.

The pitch of the thread (p) shall be proportioned to the principal diameter of the thread (d_3) as follows:

$$0,055d_3 \leq p \leq 0,15d_3$$

The depth of the thread (t) shall be proportioned to the pitch of the thread (p) as follows:

$$0,45p \leq t \leq 0,61p$$

The bottom radius of the thread profile (r_{th}) shall be no less than $0,14p$. A thread type, where the bottom radius is not specified, shall not be used.

The shank shall be undercut (with a diameter d_4) below the last threads for a length (s) proportioned to the undercut depth as follows: $s \geq 2(d_3 - d_4)$. The undercut shall reach deeper than the core diameter of the thread profile (d_5), in minimum as follows: $d_4 \leq (d_5 - 0,3 \text{ mm})$. The undercut shall be machined with a form ground tool to a surface finish of $R_a \leq 3,2 \mu\text{m}$ and shall be free from machining marks and defects.

There shall be a relief radius in a transition from the threaded part to the undercut part. The relief radius (r_9) shall be proportioned to the diameter of the undercut (d_4) as follows: $r_9 \geq 0,06 d_4$. The shape of the relief transition need not be a complete quadrant of a circle.

The thinnest section of the machined shank (consequently d_4) shall fulfil the condition $d_4 \geq 0,65d_1$, where d_1 is the diameter of the forged part of the shank, see [Figure 1](#).

The whole machined section of the shank shall have a radius at each change in diameter. The machined section shall not reach the curved part of the forged body.

Screwed threads shall conform to the tolerance requirements of ISO 965-1 (coarse series) and be of medium fit class 6g.

NOTE [Annex G](#) presents example sets of machined shank and thread dimensions, which fulfil the geometric requirements. Other hook shank and thread designs differing from those shown within [Annex G](#) can be utilized and technically assessed to the requirements of this standard, provided dimensional parameters fulfil the requirements of this clause. Furthermore, it is expected that other hook shank and thread designs in addition to those currently shown can and will be put forward for inclusion as national groups within [Annex G](#) in the future.

4.8 Nut

The material grade of the nut shall be equal to that of the hook

The height of the nut shall be such that the threaded length of the hook shank is fully engaged with the nut thread.

The nut shall be positively locked to the shank against rotation to prevent the nut from unscrewing. The locking shall not interfere with the lower two thirds of the nut/shank thread connection. The locking shall allow relative axial movement between the shank and the nut due to play in the threaded connection. Alternatively, if the nut is locked by a dowel or other similar fixing media, it is essential during the locking process that the nut/shank load bearing thread flanks are in direct contact to ensure resultant unimpaired load transmission.

The nut shall rest on an anti-friction bearing, enabling the hook body to rotate about the vertical axis. The contact surface of the nut resting on the bearing shall meet the requirements as stipulated by the related bearing. The height position of the contact surface shall fall within the lower half of the thread connection.

Screwed threads of the nut shall comply with the tolerance requirements of ISO 965-1 (coarse series) and be of medium fit class 6H. The bottom radius of the thread profile for the nut shall be not less than $0,07 p$, where p is the pitch of the thread. A thread type, where the bottom radius is not specified, shall not be used.

4.9 Hook suspension

In general, and always for serially produced hook blocks, the hook suspension together with hoist rope reeving system shall be such that the system allows free tilting of the hook in any inclined direction of the load line. In cases where this articulation of the hook suspension is not provided, this shall be specially taken into consideration in the design calculations of the hook. In cases, where by changing the crane/hook block configuration or position the hook suspension can be brought to a rigid position, this shall be taken into account in the design calculation of the hook.

The same load actions as specified for the hook shall be taken into account in the design of the hook suspension.

5 Static strength

5.1 General

The proof of static strength for hooks shall be carried out in accordance with principles of ISO 8686-1. The general design limit for static strength is yielding of the material.

The proof shall be delivered for the specified critical sections of the hook, taking into account the most unfavourable load effects from the load combinations A, B or C in accordance with ISO 8686-1. The relevant partial safety factors γ_p shall be applied. The risk coefficients γ_n shall be applied when required in the specific application or as specified in the relevant European crane type standard.

5.2 Vertical design load

The vertical design force for a hook $F_{Sd,s}$ when hoisting the rated hook load, shall be calculated as follows:

$$F_{Sd,s} = \varphi \times m_{RC} \times g \times \gamma_p \times \gamma_n \quad (1)$$

$$\text{with } \varphi = \max \left\{ \phi_2; \left(1 + \phi_5 \times \frac{a}{g} \right) \right\}$$

where

ϕ_2 is the dynamic factor, when hoisting an unrestrained grounded load, see ISO 8686-1

ϕ_5 is the dynamic factor for loads caused by hoist acceleration, see ISO 8686-1

a is the vertical acceleration or deceleration;

m_{RC} is the mass of the rated hook load;

g is the acceleration due to gravity, $g = 9,81 \text{ m/s}^2$;

γ_p is the partial safety factor, see ISO 8686-1:

$\gamma_p = 1,34$ for regular loads (load combinations A);

$\gamma_p = 1,22$ for occasional loads (load combinations B);

$\gamma_p = 1,10$ for exceptional loads (load combinations C);

γ_n is the risk coefficient.

Other load actions and combinations of ISO 8686-1 may produce vertical forces on the hook, whose load actions shall also be analysed. The vertical design force in such cases is expressed in a general format as follows:

$$F_{Sd,s} = F_H \times \gamma_p \times \gamma_n \quad (2)$$

where

F_H is a vertical force on hook due to other load action than hoisting a rated load; e.g. a test load or a peak load in an overload condition;

γ_p is the partial safety factor as above, see ISO 8686-1;

γ_n is the risk coefficient.

5.3 Horizontal design force

The Horizontal forces that are most significant for the strength of hooks are those caused by horizontal accelerations of the crane motions and these shall be taken into account. Other horizontal forces e.g. due to wind or sideways pull actions shall be taken into account, if significant. The horizontal force shall be assumed to act at the bottom of the hook seat.

The horizontal design force of hook $H_{Sd,s}$ due to horizontal accelerations shall be calculated as follows:

$$H_{Sd,s} = \min \left\{ \begin{array}{l} m_{RC} \times a \times \phi_5 \times \gamma_p \times \gamma_n \\ C_t \times F_{Sd,s} / h \end{array} \right\} \quad (3)$$

where

m_{RC} is the mass of the rated hook load;

a is the acceleration or deceleration of a horizontal motion;

ϕ_5 is the dynamic factor for loads caused by horizontal acceleration, see ISO 8686-1. For hook suspensions, which are not rigidly connected in horizontal direction to the moving part of the crane, it shall be set $\phi_5 = 1$;

γ_p is the partial safety factor as for Formula (1);

γ_n is the risk coefficient;

C_t is the relative tilting resistance of the hook suspension in accordance with [Annex I](#);

$F_{Sd,s}$ is the vertical design force in accordance with [5.2](#), related to the loading condition where $H_{Sd,s}$ is specified;

h is the vertical distance from the seat bottom of the hook body to the centre of the articulation.

5.4 Bending moment of the shank

5.4.1 General

The following load action shall be taken into consideration, when determining the total bending moment of the hook shank:

- a) horizontal forces, see [5.4.2](#);
- b) inclination of the hook suspension, see [5.4.3](#);

- c) eccentric action of vertical force in the hook seat, see 5.4.4;
- a) ramshorn hook, half of the rated load on one prong, see 5.4.5.

The bending moments caused through these load actions shall be addressed to the same load combinations, which the primary loads or operational conditions causing the bending belong to.

5.4.2 Bending moment due to horizontal force

This clause covers the shank bending moment due to external horizontal forces. The moment M_1 shall be calculated at the critical hook shank section (see 5.6) due to the horizontal design force $H_{Sd,s}$.

$$M_1 = H_{Sd,s} \times h_s \quad (4)$$

where

$H_{Sd,s}$ is the horizontal design force in accordance with 5.3;

h_s is the vertical distance from the seat bottom of the hook body to the upper end of the thinnest part of the hook shank. Bending moment due to inclination of hook suspension.

Where the arrangement of the hoist mechanism or hook/hook block is such that the hook suspension may be brought to an inclined position in a loaded condition, the bending moment at the shank caused by this inclination shall be considered in the design calculations. Such an inclination may be caused e.g. by:

- a) Differences in hoist travel distances between two separate hoist drives carrying a load beam with a hook, see Figure 3;
- b) Tilting of a single rope reeving during hoisting/lowering motion, see Figure 4;
- c) Tilting of a crane part, to which a hook is rigidly attached; or
- d) Two-blocking of a bottom block in the uppermost hoist position with a crane part, after which this crane part is tilted.

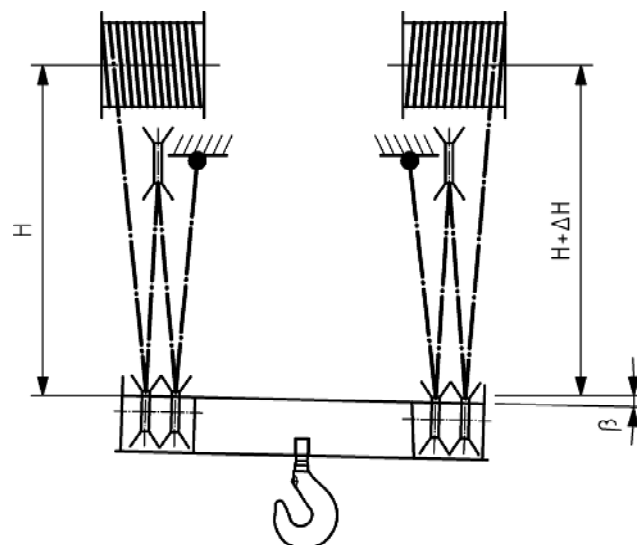


Figure 3 — Tilting of a hook in case of different hoist travel distances

Due to an inclination, the vertical force has a force component perpendicular to the axis of the hook shank. This force shall be taken into account in the same way as the horizontal forces. The bending

moment M_2 caused at the critical hook shank section is proportional to the vertical design force as follows:

$$M_2 = F_{Sd,s} \times h_s \times \sin(\beta) \quad (5)$$

where

$F_{Sd,s}$ is the vertical design force in accordance with 5.2, related to the condition with a hook inclination β ;

β is the maximum, total inclination in each relevant load combination;

h_s is the vertical distance from the seat bottom of the hook body to the upper end of the thinnest part of the hook shank.

In a rope balanced hook suspension with multiple rope falls and a single running rope coming from the drum, the hoisting/lowering movement causes the hook suspension to tilt, see Figure 4. The inclination is calculated as follows:

$$\beta = \arctan(C_t/h) \quad (6)$$

where

C_t is the relative tilting resistance of the hook suspension in accordance with Annex I;

h is the vertical distance from the seat bottom of the hook body to the centre of the articulation.

The maximum inclination, the related vertical force and the consequent moment M_2 shall be calculated separately for all relevant loading conditions of the crane.

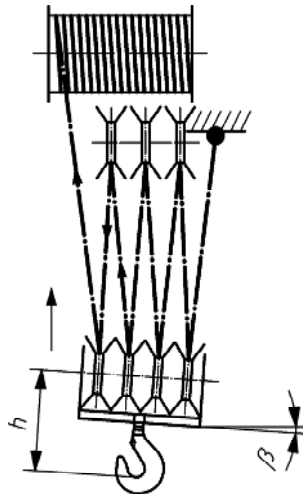


Figure 4 — Tilting of a hook suspension in a single rope reeving system

5.4.3 Bending moment due to eccentricity of vertical force

A hoist load attachment may not always settle in the middle of the hook seat. The deviation of the vertical load action line from the centre line of the shank causes a bending moment, which shall be calculated as follows:

$$M_3 = c_e \times F_{Sd,s} \times a_1 \quad (7)$$

where

$F_{Sd,s}$ is the vertical design force in accordance with [5.2](#);

a_1 is the seat circle diameter of the hook body;

c_e is a coefficient for the eccentricity ($c_e = 0,05$).

NOTE A smaller eccentricity may be used in the design calculations, if a positive, mechanical means is provided ensuring that the hoist load attachment settles closer to the hook seat centre.

5.4.4 Special case for a ramshorn hook

As a special loading case for ramshorn hooks it shall be assumed, that half of the vertical force acts on one prong while the other prong is unloaded. This loading case is addressed in the calculations to the load combination C.

For a ramshorn hook with one-sided loading, the bending moment M_4 caused at the critical hook shank section shall be calculated as follows:

$$M_4 = F_{Sd,s}/2 \times \left[e_R \times (1 - h_s/h) + h_s/h \times \min \left\{ \begin{matrix} e_R \\ C_t \end{matrix} \right\} \right] \quad (8)$$

with

$$e_R = (a_1 + d_1)/2$$

where

$F_{Sd,s}$ is the vertical design force in accordance with [5.2](#) and γ_p for load combination C;

d_1 is the diameter of the forged shank;

a_1 is the seat circle diameter of the hook;

e_R is the distance of the vertical load line from the centre line of the shank;

h is the vertical distance from the seat bottom of the hook body to the centre of the articulation;

h_s is the vertical distance from the seat bottom of the hook body to the upper end of the thinnest part of the hook shank;

C_t is the relative tilting resistance of the hook suspension, see [Annex I](#).

5.4.5 Design bending moment of the shank

In general, the design bending moment at the critical shank section $M_{Sd,s}$ for the loading conditions in accordance with [5.4.2](#) to [5.4.4](#) shall be calculated separately for each relevant load combination as follows:

$$M_{Sd,s} = \min \left\{ \begin{array}{l} (M_1 + M_2 + M_3) \\ C_t \times F_{Sd,s} \end{array} \right\} \quad (9)$$

where

M_1 to M_3 are the bending moments in accordance with [5.4.2](#) to [5.4.4](#);

C_t is the relative tilting resistance of the hook suspension, see [Annex I](#);

$F_{Sd,s}$ is the vertical design force in accordance with [5.2](#).

Additional to the above, the design bending moment in accordance with the special case of [5.4.5](#) shall be taken into account in a load combination C as follows:

$$M_{Sd,s} = M_4 \quad (10)$$

where M_4 is the bending moment in accordance with [5.4.5](#).

5.5 Hook body, design stresses

5.5.1 Loadings

The vertical design force $F_{Sd,s}$ shall be divided into two force components, acting in the centre of the seat circle symmetrically on the opposite sides of the vertical centre line and in an angle α in respect to vertical, see [Figure 5](#).

As a minimum value of α , it shall be assumed that $\alpha = 45^\circ$.

For ramshorn hooks, an equal load distribution shall be assumed between the two prongs in load combinations A and B.

As a special loading case, for ramshorn hooks it shall be assumed that half of the vertical force acts on one prong while the other prong is unloaded. This loading case is addressed in the calculations to the load combination C.

The horizontal forces shall be neglected in the hook body calculations.

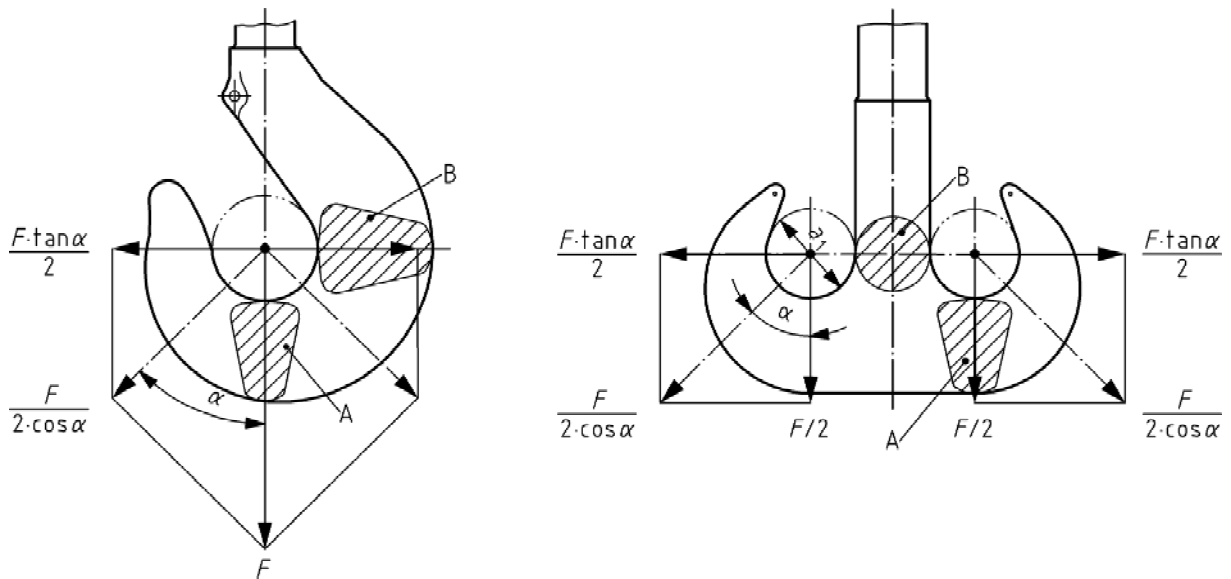


Figure 5 — Load actions on hook body and critical sections for calculation

5.5.2 Stress calculation methods

Stresses in the designated sections of a hook body shall be analysed either by the theory of curved beam bending in accordance with [Annex H](#), by finite element methods or by full scale experiments. Stresses in section B of a ramshorn hook may, however be analysed by the conventional beam bending theory.

The following subclause is based upon the theory of curved beam bending.

5.5.3 Design stresses

The design stresses σ_{Sd} in sections A and B of single hooks and in the section A of ramshorn hooks shall be calculated as follows:

$$\sigma_{Sd,s} = \frac{v \times F_{Sd,s} \times R \times \eta_1}{I} \times \frac{1}{1 - \eta_1/R} \quad (11)$$

with

$$R = a_1 / 2 + \eta_1$$

and

$$v = 1 \quad \text{for section B of single hooks;}$$

$$v = 0,5 \times \tan \alpha \quad \text{for section A of single and ramshorn hooks, } \alpha = 45^\circ;$$

where

- R is the hook curvature radius determined by the centroid of the section;
- $F_{Sd,s}$ is the vertical design force in accordance with 5.2;
- I is the reference moment of inertia for curved beam;
- a_1 is the seat circle diameter of the hook;
- η_1 is the absolute value of the coordinate y at inner edge of the particular section;
- α is the angle of the load action lines in respect to vertical, see Figure 5.

The quantities η_1 and I are section specific values and shall be calculated in accordance with Annex H.

The design stress in section B of ramshorn hook for the special case of 5.4.5 shall be calculated as follows:

$$\sigma_{Sd,s} = \frac{F}{A_{d1}} + \frac{F \times (a_1 + d_1) \times d_1 / 4}{I_{d1}} \quad (12)$$

with

$$F = F_{Sd,s} / 2$$

where

- $F_{Sd,s}$ is the vertical design force in accordance with 5.2 and γ_p for load combination C;
- A_{d1} is the cross section area of the forged shank;
- I_{d1} is the moment of inertia of the forged shank;
- d_1 is the diameter of the forged shank;
- a_1 is the seat circle diameter of the hook.

5.6 Hook shank, design stresses

The vertical design forces in accordance with 5.2 and the design bending moments in accordance with 5.4 shall be taken into account in the proof calculations of a hook shank. In general, the critical section of the shank is the undercut part immediately below the threaded section with a diameter d_4 , see Figure 1. Maximum design stress $\sigma_{Sd,s}$ is calculated as a nominal stress without stress concentration factors and using conventional beam bending theory as follows:

$$\sigma_{Sd,s} = \frac{F_{Sd,s}}{A_{d4}} + \frac{M_{Sd,s} \times d_4 / 2}{I_{d4}} \quad (13)$$

where

- $F_{Sd,s}$ is the vertical design hook force;
- $M_{Sd,s}$ is the design bending moment in the critical section, see 5.4.5;
- A_{d4} is the cross section area of the critical section of the hook shank;
- I_{d4} is the moment of inertia of the critical section of the hook shank.

5.7 Hook, proof of static strength

5.7.1 General for hook body and shank

Both for the hook body and the hook shank, it shall be proven for relevant load actions specified in 5.2 to 5.4 that

$$\sigma_{Sd,s} \leq f_{Rd} = f_1 \times \frac{f_y}{\gamma_m \times \gamma_{sm}} \quad (14)$$

where

$\sigma_{Sd,s}$ is the maximum design stress in accordance with 5.5 and 5.6 in the critical section;

f_{Rd} is the limit design stress;

f_y is the yield stress of the material in the finished product;

f_1 is the influence factor for the operation temperature;

γ_m is the general resistance coefficient in accordance with ISO 8686-1 ($\gamma_m = 1,1$);

γ_{sm} is the specific resistance coefficient for the section, as follows:

$\gamma_{sm} = 0,75$ for the hook body section B of single hooks, or

alternatively, if addressing a national standard particular hook form, γ_{sm} can be evaluated by superimposing Formula (14) characteristics within Formula (11) and applying that national standard's noted maximum rated capacity together with its nominated yield stress. Sample indications are contained within Annex C.

$\gamma_{sm} = 0,90$ for hook body section A of single point and ramshorn type hooks

$\gamma_{sm} = 1,0$ for all shank sections

In the absence of other data, the factor f_1 taking into consideration the reduction of the yield stress in high temperatures shall be calculated as follows:

For $-50 \text{ °C} \leq T \leq 100 \text{ °C}$:

$$f_1 = 1$$

For $100 \text{ °C} < T \leq 250 \text{ °C}$:

$$f_1 = 1 - 0,25 \times (T - 100) / 150 \quad (15)$$

where T is the operation temperature in degrees Celsius (°C).

5.7.2 The use of static limit design force for verification of the hook body

The static limit design force $F_{Rd,s}$ covers the proof of static strength for sections A and B of single hooks and for section A of ramshorn hooks. It shall be calculated as follows:

$$F_{Rd,s} = \frac{f_y}{\gamma_m \times \gamma_{sm}} \times \frac{I \times (1 - \eta_1 / R)}{v \times R \times \eta_1} \quad (16)$$

For a single hook the static limit design force is calculated separately for the two sections A and B and the smaller value of the two is used.

It shall also be proven for all relevant load actions and combinations specified in 5.2 that

$$F_{Sd,s} \leq f_1 \times F_{Rd,s} \quad (17)$$

where

$F_{Sd,s}$ is the vertical design force in accordance with 5.2;

$F_{Rd,s}$ is the static limit design force

f_1 is the influence factor for the operation temperature in accordance with Formula (15).

NOTE 1 Additionally, the proof of static strength for the shank is carried out in accordance with 5.7.1.

NOTE 2 In cases where the selected hook body is in accordance with Annexes A or B and the yield and ultimate strength values are those specifically noted within Table 4, the proof of static strength can be based on the static limit design force shown in Annex C.

NOTE 3 The dimensions of the ramshorn hooks in Annex B are proportioned such that section B of the hook body does not become governing in respect to static strength of the body, i.e. the proof of a special case in 5.4.5 is not required.

6 Fatigue strength

6.1 General

The proof of fatigue strength for hooks shall be carried out in accordance with principles of ISO 8686-1. A hook shall have the design life in minimum equal to that of the related crane or hoist.

The proof shall be delivered for the specified critical sections of the hook, taking into account the most unfavourable load effects from the load combinations A in accordance with ISO 8686-1, setting all partial safety factors $\gamma_p = 1$ and the risk coefficients $\gamma_n = 1$.

The number of stress cycles for the proof shall be based on the total number of working cycles during the design life of the crane, as specified in ISO 4301-1. In general, for the hook body, one lifting cycle induces one stress cycle. If a working cycle consists of several lifting cycles, this shall be taken into account, when counting the stress cycles. For the hook shank, additionally the number of positioning movements shall be taken into account, when counting the number of bending stress cycles.

6.2 Vertical fatigue design force

The vertical design force $F_{Sd,f,i}$ for a lifting cycle i shall be calculated as follows:

$$F_{Sd,f,i} = \phi_2 \times m_i \times g \quad (18)$$

where

ϕ_2 is the dynamic factor, when hoisting an unrestrained grounded load, see ISO 8686-1;

m_i is the mass of the hook load in a lifting cycle i ;

g is the acceleration due to gravity, $g = 9,81 \text{ m/s}^2$.

6.3 Horizontal fatigue design force

The horizontal design force $H_{Sd,f,i}$ for a lifting cycle i due to horizontal accelerations shall be calculated as follows:

$$H_{Sd,f,i} = \min \left\{ \begin{array}{l} m_i \times a \times \phi_5 \\ C_t \times m_i \times g / h \end{array} \right\} \quad (19)$$

where

m_i is the mass of the hook load in a lifting cycle i ;

a is the acceleration or deceleration of a horizontal motion;

ϕ_5 is the dynamic factor for loads caused by horizontal acceleration, see ISO 8686-1. For hook suspensions, which are not rigidly connected in horizontal direction to the moving part of the crane, it shall be set $\phi_5 = 1$;

C_t is the relative tilting resistance of the hook suspension, see [Annex I](#);

g is the acceleration due to gravity, $g = 9,81 \text{ m/s}^2$;

h is the vertical distance from the seat bottom of the hook body to the centre of the articulation.

6.4 Fatigue design bending moment of shank

6.4.1 Bending moment due to horizontal force

The moment $M_{1,f,i}$ shall be calculated at the critical hook shank section, due to the horizontal design force $H_{Sd,f,i}$ in accordance with [6.3](#):

$$M_{1,f,i} = H_{Sd,f,i} \times h_s \quad (20)$$

where

$H_{Sd,f,i}$ is the horizontal design force in a lifting cycle i in accordance with [6.3](#);

h_s is the vertical distance from the seat bottom of the hook body to the upper end of the thinnest part of the hook shank.

6.4.2 Bending moment due to inclination of hook suspension

The basis causing inclination and the method of calculation shall be taken into consideration in analogy to [5.4.3](#). For the proof of fatigue strength the consideration, which of the loading events of load combination A are regular loadings, shall be based upon the crane configuration and application.

As a minimum, the following shall be considered to occur regularly in each lifting cycle: In a rope balanced hook suspension with multiple numbers of falls and a single running rope coming from the

drum, the hoisting/lowering movement causes the hook suspension to tilt, see [Figure 4](#). The bending moment $M_{2,f,i}$ caused at the critical hook shank section is calculated as follows:

$$M_{2,f,i} = F_{Sd,f,i} \times h_s \times \sin(\beta) \quad (21)$$

where

$F_{Sd,f,i}$ is the vertical design force in a lifting cycle i , in accordance with [6.2](#);

h_s is the vertical distance from the seat bottom of the hook body to the upper end of the thinnest part of the hook shank;

β is the inclination of the hook suspension in accordance with Formula (6).

6.4.3 Bending moment due to eccentricity of vertical force

A hoist load attachment may not always settle in the middle of the hook seat. The deviation of the vertical load action line from the centre line of the shank causes a bending moment, which shall be calculated as follows:

$$M_{3,f,i} = c_e \times F_{Sd,f,i} \times a_1 \quad (22)$$

where

$F_{Sd,f,i}$ is the vertical design force in a lifting cycle i , in accordance with [6.2](#);

a_1 is the seat circle diameter of the hook body;

c_e is a coefficient for the eccentricity, $c_e = 0,05$.

NOTE A smaller eccentricity can be used in the design calculations if a positive, mechanical means is provided that ensures the hoist load attachment settles closer to the hook seat centre.

6.5 Proof of fatigue strength, hook body

6.5.1 Design stress calculation

The proof of fatigue strength shall be based upon cumulative effect of stress ranges in the critical sections. It shall be assumed that the load is grounded in each lifting cycle, i.e. the hook load range is from zero to full load, with dynamic factor inclusion.

Calculation of the stress ranges is comparable to that of the static design stress in [5.5.3](#), when applying the vertical fatigue design load from [6.2](#):

$$\Delta\sigma_{Sd,i} = \sigma_{Sd,s}$$

in accordance with Formula (11) in [5.5.3](#), when setting $F_{Sd,s} = F_{Sd,f,i}$

where

i is the index of a lifting cycle;

$\Delta\sigma_{Sd,i}$ is the stress range in a cycle i ;

$F_{Sd,f,i}$ is the vertical fatigue design force in accordance with [6.2](#).

In the proof of fatigue strength of “proofed” hooks, with residual compressive stress $\sigma_{Sd,pr}$ (due to the effect of proof load) in the region of maximum stress at the intrados of the hook bowl, the design stress will cycle/range from the minimum value of the design stress in the body of “proofed” hook $\sigma_{Sd,pb,min}$ which is equal to residual stress $\sigma_{Sd,pr}$ at zero hook load

$$\sigma_{Sd,pb,min} = \sigma_{Sd,pr}$$

to a maximum value of the design stress in the body of “proofed” hook $\sigma_{Sd,max}$ calculated by adding the stress range $\Delta\sigma_{Sd,i}$ for the hook load applied $F_{Sd,s} = F_{Sd,f,i}$ (calculated as shown above) to the initial residual stress $\sigma_{Sd,pr}$.

$$\sigma_{Sd,pbmax} = \sigma_{Sd,pr} + \Delta\sigma_{Sd,i}$$

In the proof of fatigue strength of “non-proofed” hooks, the design stress will cycle/range from the minimum value of zero at the intrados of the hook bowl at zero hook load. This will also be the case for the “proofed” hooks if the beneficial effects of this process are neglected, in which case the results will be conservative.

The calculation procedure outlined in 6.5 is based on a pulsating stress cycle (stress cycling from zero to a maximum value) and the stress cycle in “proofed” hook body has to be transformed to an equivalent pulsating cycle first, for instance using a procedure outlined in Annex F before the procedures of 6.5 can be applied.

6.5.2 Stress history in general

The cumulative fatigue effect of the stress history from all of the stress cycles is condensed to a single stress history parameter s_h . This is calculated as follows:

$$s_h = k_h \times v_h \quad (23)$$

with

$$k_h = \frac{1}{N} \sum_{i=1}^N \left(\frac{\Delta\sigma_{Sd,i}}{\Delta\sigma_{Sd,max}} \right)^m \quad (24)$$

and

$$v_h = \frac{N}{N_D} \quad (25)$$

where

- k_h is the stress spectrum factor;
- v_h is the relative number of stress cycles;
- i is the index of a lifting cycle;
- N is the total number of lifting cycles;
- N_D is the reference number of cycles ($N_D = 2 \times 10^6$);
- $\Delta\sigma_{Sd,i}$ is the stress range in a cycle i ;
- $\Delta\sigma_{Sd,max}$ is the maximum stress range;
- m is the slope parameter of the characteristic fatigue design curve ($m = 6$).

The total number of lifting cycles (N) shall conform to the total number of working cycles (C) during the design life of the crane as specified in ISO 4301-1.

6.5.3 Stress history based upon classified duty

The hook body represents a special case, where the stress variations depend upon the hoist load variations, only. Because of this, the stress history parameter can be derived directly from the classes Q and U of ISO 4301-1, instead of using a case specific stress history and detailed calculation of s_h in accordance with 6.5.2. In cases where the intended duty is specified through the classes Q and U only, the calculation of s_h shall be carried out as presented in Formula (27).

A load history parameter s_Q is defined by the equation

$$s_Q = kQ \times N / N_D \quad (26)$$

where

kQ is the load spectrum factor in accordance with [Table 5](#), see also ISO 4301-1; extended to include Q_0 – Q_5 category status;

N is the total number of lifting cycles. Typically, for a hook this shall be taken as the number work cycles (C) specified for the crane through the class U of ISO 4301-1. Each intermediate grounding of the load within a work cycle shall, however, be counted as an additional lifting cycle and added to the value of N ;

N_D is the reference number of cycles ($N_D = 2 \times 10^6$).

The load spectrum factor (kQ) is calculated by a Wöhler curve slope with an exponent of 3, whereas the hook body fatigue is related to a slope $m = 6$. For a load distribution with a given shape, a conversion factor can be calculated to create a connection between the load spectrum of ISO 4301-1 and the stress history parameter of hooks. For a classified duty shapes of load distributions given in [Annex E](#) shall be applied

In cases where the load spectrum is specified through the classification of ISO 4301-1, the stress history parameter s_h for a hook body shall be calculated as follows:

$$s_h = \frac{s_Q}{(k_6^*)^m} \quad (27)$$

with

$$k_6^* = \sqrt[6]{\frac{kQ}{k_h}} \quad (28)$$

where k_6^* is the specific spectrum ratio factor.

Standardized, conventional values in accordance with [Table 5](#) shall be used for design of hook body. See also [Annex E](#).

Table 5 — Specific spectrum ratio factors k_6^*

| Class Q of ISO 4301-1 extended to $Q_0 - Q_5$ | Load spectrum factor kQ | Factor k_6^* for $m = 6$ |
|---|---------------------------|----------------------------|
| Q ₀ | 0,031 3 | 1,348 |
| Q ₁ | 0,062 5 | 1,343 |
| Q ₂ | 0,125 | 1,259 |
| Q ₃ | 0,250 | 1,172 |
| Q ₄ | 0,500 | 1,084 |
| Q ₅ | 1,000 | 1,00 |

6.5.4 Limit fatigue design stress

The basic assumption is that the fatigue strength curves in the $\log(\sigma)/\log(N)$ -scale are straight lines, with the same slope (m) for all material grades. This is a reasonable approximation in the range of high number of stress cycles, where fatigue is the governing design criteria.

The limit fatigue design stress at the reference point N_D is calculated as follows:

$$\Delta\sigma_{Rd} = f_1 \times f_2 \times \Delta\sigma_c \quad (29)$$

where

$\Delta\sigma_{Rd}$ is the limit fatigue design stress;

$\Delta\sigma_c$ is the characteristic fatigue strength at $N_D = 2 \times 10^6$ cycles, dependent on the material;

f_1 is the influence factor for the operation temperature, in accordance with Formula (31);

f_2 is the influence factor for the material thickness, in accordance with Formula (32).

The characteristic fatigue strength $\Delta\sigma_c$ is dependent upon the ultimate strength of the material. For the classified material grades in accordance with [Table 4](#), the fatigue strength shall be taken from [Table 6](#).

Table 6 — Characteristic fatigue strength of forged hook materials

| Material class | $\Delta\sigma_c$ N/mm ² |
|----------------|---------------------------------------|
| M | 170 |
| P | 220 |
| S | 235 |
| T | 280 |
| V | 305 |

For other materials and in cases where the classified material grades are not applied, the characteristic fatigue strength $\Delta\sigma_c$ shall be calculated as follows:

$$\Delta\sigma_c = 0,315 \times f_u \times \lg \frac{13\,001}{f_u} \quad (30)$$

where f_u is the ultimate strength of the material in newtons per square millimetre (N/mm²).

The influence factor f_1 for the operation temperature is calculated as follows:

for $100\text{ °C} \leq T \leq 250\text{ °C}$:

$$f_1 = 1 - 0,1 \times (T - 100) / 150 \quad (31)$$

for $-50 \text{ °C} \leq T \leq 100 \text{ °C}$:

$$f_1 = 1$$

where T is the operation temperature in degrees Celsius (°C).

The influence factor f_2 for the material thickness is calculated as follows:

for $25 \text{ mm} \leq b_{\max} \leq 150 \text{ mm}$:

$$f_2 = \left(\frac{b_{\text{ref}}}{b_{\max}} \right)^{0,167} \quad (32)$$

for $b_{\max} < 25 \text{ mm}$

$$f_2 = 1$$

for $b_{\max} > 150 \text{ mm}$:

$$f_2 = 0,74$$

where

b_{ref} is the reference width ($b_{\text{ref}} = 25 \text{ mm}$);

b_{\max} is the maximum width in the critical hook body section, see [Figure 5](#).

6.5.5 Execution of the proof

The proof shall be carried out separately for all relevant sections of the hook body.

For the proof of fatigue strength, it shall be proven that

$$\Delta\sigma_{\text{Sd,max}} \leq \frac{\Delta\sigma_{\text{Rd}}}{\gamma_{\text{Hf}} \times \sqrt[m]{s_{\text{h}}}} = \frac{k_6^* \times \Delta\sigma_{\text{Rd}}}{\gamma_{\text{Hf}} \times \sqrt[m]{s_{\text{Q}}}} \quad (33)$$

where

$\Delta\sigma_{\text{Sd,max}}$ is the maximum stress range within the total stress history;

$\Delta\sigma_{\text{Rd}}$ is the limit fatigue design stress in accordance with Formula (29);

γ_{Hf} is the fatigue strength specific resistance factor in accordance with [Table 7](#);

m is the slope parameter of the characteristic fatigue design curve ($m = 6$);

s_{h} is the stress history parameter;

k_6^* is the specific spectrum ratio factor;

s_{Q} is the load history parameter.

Table 7 — Fatigue strength specific resistance factor

| Section of hook | γ_{Hf} |
|--|---------------|
| Section A of single and ramshorn hooks | 1,35 |
| Section B of single hooks | 1,25 |

For the calculation utilizing the classification in accordance with ISO 4301-1, the values for the following conversion factor are given in [Annex E](#):

$$k_C = k_6^* / \sqrt[m]{s_Q} \quad (34)$$

6.5.6 The use of fatigue limit design force for verification of the hook body

The fatigue limit design force $F_{Rd,f}$ shall be calculated as follows:

$$F_{Rd,f} = \frac{f_2 \times \Delta \sigma_c}{\gamma_{Hf}} \times \frac{I \times (1 - \eta_1 / R)}{v \times R \times \eta_1} \quad (35)$$

For a single hook, the fatigue limit design force shall be calculated separately for the two sections A and B and the smaller value of the two is used.

It shall also be proven for all relevant load actions and combinations specified in [6.2](#) that

$$F_{Sd,f} \leq \frac{f_1 \times F_{Rd,f}}{\sqrt[m]{s_h}} = \frac{f_1 \times k_6^* \times F_{Rd,f}}{\sqrt[m]{s_Q}} \quad (36)$$

where

$F_{Sd,f}$ is the maximum vertical fatigue design load in accordance with [6.2](#);

$F_{Rd,f}$ is the fatigue limit design force;

f_1 is the influence factor for the operation temperature in accordance with Formula (31).

In cases where the selected hook body is in accordance with [Annexes A](#) or B, the proof of fatigue strength may be based on the fatigue limit design force shown in [Annex D](#).

6.6 Proof of fatigue strength, hook shank

6.6.1 General

The number of stress cycles is derived from the total number of lifting cycles (N), which shall conform to the total number of working cycles (C) during the design life of the crane as specified in accordance with ISO 4301-1.

6.6.2 Design stress calculation

The design stresses shall be calculated in the undercut section of the shank immediately below the threads with a diameter d_4 , see [Figure 1](#). Basic stresses are calculated without stress concentration factors and using conventional beam bending theory. The following equations are general and apply in [6.6](#) for any vertical design force and design bending moment:

$$\sigma_a(F) = \frac{F}{A_{d4}} \quad (37)$$

$$\sigma_b(M) = \frac{M \times d_4 / 2}{I_{d4}} \quad (38)$$

where

- σ_a is the shank stress (axial) due to vertical design force;
- σ_b is the shank stress (bending) due to design bending moment;
- F is the vertical design force in a fatigue load cycle;
- M is the design bending moment in a fatigue load cycle;
- A_{d4} is the cross section area of the critical section of the hook shank;
- I_{d4} is the moment of inertia of the critical section of the hook shank.

6.6.3 Applied stress cycles

Within each lifting cycle, the two types of stress cycles shown below shall be considered, as relevant.

In the Proof of fatigue strength of “non-proofed” hooks residual compressive stress $\sigma_{Sd,sh,pr}$ in the equations below is set to zero. For “proofed” hooks residual compressive stress $\sigma_{Sd,sh,pr}$ set up in the undercut region where maximum stress during proof loading has exceeded the yield stress (e.g. the run out of shank thread) can be neglected (by setting in the equations below the value of the residual stress $\sigma_{Sd,sh,pr}$ to zero — a conservative approach) or the beneficial effects taken into account as shown in both types of stress cycles below.

Cycle Type 1: A stress cycle due to lifting a load and lowering it down on the ground, with due consideration to the bending stress due to inclination of hook suspension and eccentricity of the vertical load. The specifics of each stress cycle (i) are as follows:

- a) Axial stress is $\sigma_{a1} = \sigma_a(F_{Sd,fi})$ (Formula (37)), where $F_{Sd,fi}$ is in accordance with 6.2.

Residual compressive stress $\sigma_{Sd,sh,pr}$ set up in the region of shank undercut where maximum stress during proof loading has exceeded the yield stress has to be taken into account. The design axial stress will cycle/range from the minimum value of the axial design stress in the shank of “proofed” hook $\sigma_{Sd,ps,min}$ equal to residual stress $\sigma_{Sd,sh,pr}$ at zero hook load.

$\sigma_{Sd,apsmin} = \sigma_{Sd,sh,pr}$ to a maximum value of the axial design stress in the shank of “proofed” hook $\sigma_{Sd,ps,max}$ calculated by adding the stress σ_{a1} for the hook load $F_{Sd,fi}$ (calculated as shown above) to the value of the initial residual stress $\sigma_{Sd,sh,pr}$:

$$\sigma_{Sd,apsmax} = \sigma_{Sd,sh,pr} + \sigma_{a1}$$

- b) Bending stress is $\sigma_{b1} = \sigma_b(M)$ [Formula (38)], where $M = \max[M_{2,fi}, M_{3,fi}]$ is in accordance with 6.4.2 and 6.4.3.

Residual compressive stress $\sigma_{Sd,sh,pr}$ set up in the region of shank undercut where maximum stress during proof loading has exceeded the yield stress has to be taken into account and the design bending stress type 1, σ_{b1} has to be added to the axial stress including residual stress as calculated above.

- c) Pulsating stress cycle from 0 to $\sigma_{a1} + \sigma_{b1}$, mean stress $\sigma_{m1,i} = (\sigma_{a1} + \sigma_{b1})/2$, stress amplitude $\sigma_{A1,i} = \sigma_{m1,i}$.

With residual compressive stress $\sigma_{Sd,sh,pr}$ present in the region of “proofed” hook shank undercut the design stress Type 1 will cycle/range from the minimum value of the design stress $\sigma_{Sd,ps1,min}$ equal to residual stress $\sigma_{Sd,sh,pr}$ at zero hook load:

$$\sigma_{Sd,ps1,min} = \sigma_{Sd,sh,pr}$$

to a maximum value of the design stress in the shank of “proofed” hook $\sigma_{Sd,ps1,max}$ calculated by adding the stress σ_{a1} for the hook load $F_{Sd,f,i}$ (calculated as shown above) and bending stress σ_{b1} to the value of initial residual stress $\sigma_{Sd,sh,pr}$.

$$\sigma_{Sd,ps1,max} = \sigma_{Sd,sh,pr} + \sigma_{a1} + \sigma_{b1}$$

With the procedures of [6.6.8](#) being based on the cycle in [6.6.3](#) being a pulsating stress cycle (stress cycling from zero to a maximum value) the stress cycle in “proofed” hook shank has to be first transformed to an equivalent pulsating cycle, e.g. using the transformation equation shown in [Annex I](#) before the procedures of [6.6.8](#) can be applied.

- d) The total number of stress cycles is $N_1 = N$

Cycle Type 2: A stress cycle due to horizontal acceleration and resulting load sway shall be taken into consideration as follows.

- e) Axial stress as in Cycle Type 1

Residual compressive stress $\sigma_{Sd,sh,pr}$ set up in the region of shank undercut during proof loading has to be taken into account. The extreme values of the design axial stress Type 2, will be calculated in the manner analogous described in Paragraph a) for Type 1, viz. design stress cycling/ranging from the minimum value of the axial design stress in the shank of “proofed” hook $\sigma_{Sd,a2,ps,min}$ equal to residual stress $\sigma_{Sd,sh,pr}$ at zero hook load

$$\sigma_{Sd,a2,ps,min} = \sigma_{Sd,sh,pr}$$

to a maximum value of the axial design stress in the shank of “proofed” hook $\sigma_{Sd,a2,ps,max}$ calculated by adding the stress σ_{a2} to the value of the initial residual stress $\sigma_{Sd,sh,pr}$

$$\sigma_{Sd,a,psmax} = \sigma_{Sd,sh,pr} + \sigma_{a1}$$

- f) Bending stress is $\sigma_{b2,i} = \sigma_b(M_{1,f,i})$ [Formula (38)], where $M_{1,f,i}$ is in accordance with [6.4.1](#)

Residual compressive stress $\sigma_{Sd,sh,pr}$ set up in the region of shank undercut during proof loading has to be taken into account and the design bending stress type 2, σ_{b2} has to be added to the axial stress including residual stress as calculated above.

- g) Each stress cycle with a mean stress $\sigma_{m2,i} = \sigma_{a1,i}$ and stress amplitude $\sigma_{A2,i} = \sigma_{b2,i}$

With residual compressive stress $\sigma_{Sd,sh,pr}$ present in the region of “proofed” hook shank undercut the design stress Type 2 will cycle/range from the minimum value of the design stress $\sigma_{Sd,ps2,min}$ equal to $(\sigma_{Sd,sh,pr} - \sigma_{b2})$ at zero hook load

$$\sigma_{Sd,ps2,min} = \sigma_{Sd,sh,pr} - \sigma_{b2}$$

to a maximum value of the design stress in the shank of “proofed” hook $\sigma_{Sd,ps2,max}$ calculated by adding the stress σ_{a2} for the hook load $F_{Sd,f,i}$ (calculated as shown above) and bending stress σ_{b2} to the value of initial residual stress $\sigma_{Sd,sh,pr}$

$$\sigma_{Sd,ps2,max} = \sigma_{Sd,sh,pr} + \sigma_{a2} + \sigma_{b2}$$

With the procedures of 6.6.8 being based on the assumption of the Type 2 stress cycle in 6.6.3 being a pulsating stress cycle (stress cycling from zero to a maximum value) the stress cycle in “proofed” hook shank has to be first transformed to an equivalent pulsating cycle, e.g. using the transformation equation shown in Annex I before the procedures of 6.6.8 can be applied.

h) The total number of stress cycles is $N_2 = p_a \times N$

Within each lifting cycle, the hook load specific for that cycle shall be used.

NOTE The axial stresses within the Cycle Type 2 may be calculated without the effect of the factor ϕ_2 in 6.2.

The parameter p_a shall be selected in accordance with Table 8.

Table 8 — Average number of horizontal accelerations p_a

| Type of application | p_a |
|---|-------|
| 1. Process applications, where horizontal load movements are regularly a part of each work cycle | 8 |
| 2. Special applications, where horizontal movements are operated at all times under control of a signaller, with low speeds and short distances | 2 |
| 3. A special load sway control is used in the drive system of the horizontal movement | 2 |
| 4. All other applications and serially produced hooks, where the application is not known | 4 |

6.6.4 Basic fatigue strength of material

The basic, alternating fatigue strength of the material, for zero mean stress ($\sigma_m = 0$) and for the reference number of stress cycles $N_D = 2\,000\,000$ is calculated based upon the ultimate strength of the material as follows:

$$\sigma_M = 0,45 \times f_u \quad (39)$$

6.6.5 Stress concentration effects from geometry

The factors calculated within this clause are the stress concentration factor α and, as a final outcome, the notch effect factor β_n . Both of them shall be calculated separately for the shoulder and for the thread bottom in accordance with the equations in Table 9. The maximum value of the two β_n shall be used in the proof of fatigue strength of the shank.

NOTE The thread is assumed to be of a single lead type.

Table 9 — Parameters for calculation of stress concentration factors

| | Shoulder | Thread |
|---------------------------------------|--|---|
| Mean thread diameter d_e | $d_e = 0,6 \times d_3 + 0,4 \times d_5$ | |
| Depth of notch | $u_S = \frac{(d_e - d_4)}{2}$ | $u_T = \frac{(d_e - d_5)}{2}$ |
| Factor ϕ | $\phi = \frac{1}{2 + 4 \times \sqrt{\frac{u_S}{r_9}}}$ | $\phi = \frac{1}{2 + 4 \times \sqrt{\frac{u_T}{r_{th}}}}$ |
| Factor χ | $\chi = \frac{2 \times (1 + \phi)}{r_9}$ | $\chi = \frac{2 \times (1 + \phi)}{r_{th}}$ |
| Support factor n | $n = 1 + \sqrt{\chi} \times 10^{-(0,33 + f_y/712)}$ | |
| Geometric stress concentration factor | α_S [Formula (40)] | α_T [Formula (41)] |
| Notch effect factor | $\beta_n = \frac{\alpha_S}{n}$ | $\beta_n = \frac{\alpha_T}{n}$ |

The shoulder stress concentration factor α_S shall be calculated as follows:

$$\alpha_S = 1 + \frac{1,1}{\sqrt{0,22 \times \frac{r_9}{u_S} + 2,74 \times \frac{r_9}{d_4} \times \left(1 + 2 \frac{r_9}{d_4}\right)^2}} \quad (40)$$

The thread stress concentration factor α_T shall be calculated as follows:

$$\alpha_T = 1,8 \times \left(\frac{p}{d_5}\right)^{0,3} \times \left(\frac{u_T}{r_{th}}\right)^{0,2} \times \left(\frac{p}{u_S}\right)^{0,1} \times \left(\frac{d_4}{d_5}\right)^3 \times \left(1 + \frac{1}{\sqrt{0,22 \times \frac{r_{th}}{u_T} + 2,74 \times \frac{r_{th}}{d_5} \times \left(1 + 2 \frac{r_{th}}{d_5}\right)^2}}\right) \quad (41)$$

The geometric symbols in [Table 9](#) and in Formulae (40) and (41) are according to those in [Figure 2](#). The yield stress f_y in the equation for n shall be in newtons per square millimetre (N/mm²).

6.6.6 Fatigue strength of notched shank

The further following calculation shall be performed for the more critical of the two shank sections. The basic material fatigue strength shall be reduced to a comparable value in respect to nominal stresses in the shank.

The fatigue strength amplitude, notch piece factor σ_W shall be calculated as follows:

$$\sigma_W = f_1 \times \frac{\sigma_M}{\left(\beta_n + \frac{1}{f_3} - 1 \right)} \quad (42)$$

with

$$f_3 = 1 - 0,29 \times \lg \frac{R_a}{0,4} \times \lg \frac{f_u}{200} \quad (43)$$

where

σ_M is the basic fatigue strength of the material;

β_n is the maximum of β_{nS} and β_{nT} ;

f_1 is the influence factor for the operation temperature, in accordance with Formula (31);

f_3 is the factor for the surface roughness influence;

R_a is the surface finish grade in micrometres (μm) within the limits $0,4 \mu\text{m} \leq R_a \leq 3,2 \mu\text{m}$;

f_u is the ultimate strength of the material in newtons per square millimetre (N/mm^2),
 $f_u \geq 300 \text{ N}/\text{mm}^2$.

NOTE The factor for reducing the material strength by increasing diameter is not applicable in this document. The true material properties for the actual diameter are used in the design calculations.

6.6.7 Mean stress influence

The above σ_W values apply for a pure alternating stress with zero mean stress. Hook shank represents a type of component, where the reduction of fatigue strength by increasing mean stress shall be considered. The mean stress influence is illustrated through a principal Smith diagram in [Figure 6](#). Characteristics of the diagram are as follows:

- a) The fatigue strength σ_W is fixed at mean stress $\sigma_m = 0$;
- b) Upper limit line of the diagram is specified through a mean stress influence factor μ ;
- c) Stress amplitudes at a related mean stress shall be within the lower and upper limits of the diagram.

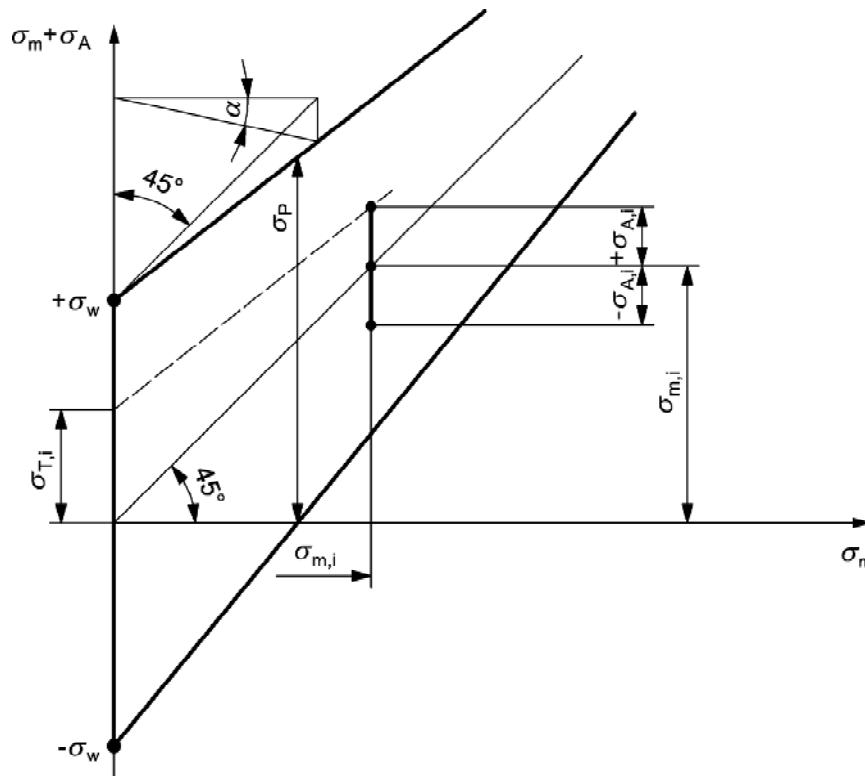


Figure 6 — Smith diagram and transformation of stress amplitude

The upper limit line of the diagram is specified through the assumption that with pulsating stress, the total stress variation is limited to $\sigma_p = 1,7 \sigma_w$. From this rule, the mean stress influence factor is calculated as follows:

$$\mu = \tan(\alpha) = \frac{\sigma_w}{1,7 \times \sigma_w / 2} - 1 = 0,1765 \quad (44)$$

NOTE The mean stress influence parameters μ and α correspond to the parameters μ_1 and α_1 , so that in this document α is counted always positive.

6.6.8 Transformation of stresses to a constant mean stress

The stress amplitudes with related mean stresses as specified in 6.6.3 are transformed to a stress amplitude with an equal fatigue influence. Transformation to a stress cycle with a mean stress zero is made as follows:

Cycle Type 1

$$\sigma_{T1,i} = \sigma_{A1,i} + \mu \times \sigma_{m1,i} \quad (45)$$

Cycle Type 2

$$\sigma_{T2,i} = \sigma_{A2,i} + \mu \times \sigma_{m2,i} \quad (46)$$

where $\sigma_{T1,i}$ and $\sigma_{T2,i}$ are the transformed stress amplitudes at zero mean stress, and μ is the mean stress influence factor.

The transformation of $\sigma_{A,i}$ to $\sigma_{T,i}$ is illustrated in Figure 6.

6.6.9 Stress history parameter in general

The cumulative fatigue effect of the stress history from all of the stress cycles is condensed into a single stress history parameter s_s . This is calculated as follows:

$$s_s = k_s \times v_s \quad (47)$$

with

$$k_s = \frac{1}{N + p_a \times N} \times \left[\sum_{i=1}^N \left(\frac{\sigma_{T1,i}}{\sigma_{Tmax}} \right)^m + p_a \times \sum_{i=1}^N \left(\frac{\sigma_{T2,i}}{\sigma_{Tmax}} \right)^m \right] \quad (48)$$

and

$$v_s = \frac{N + p_a \times N}{N_D} \quad (49)$$

where

σ_{Tmax} is the maximum of the transformed stress amplitudes $\sigma_{T1,i}$ and $\sigma_{T2,i}$,

k_s is the stress spectrum factor for the hook shank;

v_s is the relative number of stress cycles;

i is the index of a lifting cycle

N is the total number of lifting cycles

m is the slope parameter of the characteristic fatigue design curve ($m = 5$).

6.6.10 Stress history parameter based upon classified duty

The hook shank represents a special case, where the magnitudes of stress variations are directly proportional to the hoist load variations. Because of this, the stress history parameter can be derived directly from the classes Q and U of ISO 4301-1, instead of using a case specific stress history and detailed calculation in accordance with [6.6.9](#).

The load spectrum factor (kQ) is calculated by a Wöhler curve slope with an exponent of 3, whereas the hook shank fatigue is related to a slope $m = 5$. For a load distribution with a given shape, a conversion factor can be calculated to create a connection between the load spectrum of ISO 4301-1 and the stress history parameter of hook shank. For a classified duty shapes of load distributions given in [Annex E](#) shall be applied

In cases where the intended duty is specified through the classes Q and U of ISO 4301-1, the stress history parameter shall be calculated as follows:

$$s_s = k_s \times v_s \quad (50)$$

with

$$k_s = \frac{1}{1 + p_a} \times \frac{kQ}{(k_5^*)^m} \left[\left(\frac{\sigma_{T1,max}}{\sigma_{Tmax}} \right)^m + p_a \times \left(\frac{\sigma_{T2,max}}{\sigma_{Tmax}} \right)^m \right] \quad (51)$$

$$v_s = \frac{N}{N_D} \times (1 + p_a) \quad (52)$$

and

$$k_5^* = \sqrt[5]{\frac{kQ}{\sigma_{T1,max} \sigma_{T2,max}}} \quad (53)$$

where

kQ is the load spectrum factor in accordance with [Table 10](#);

$\sigma_{T1,max}$, $\sigma_{T2,max}$ are the maximums of the transformed stress amplitudes in Cycle Types 1 and 2;

N is the total number of lifting cycles, which, typically for a hook, shall be taken as equal to the number of work cycles (C) specified for the crane through the class U of ISO 4301-1; each intermediate grounding of the load within a work cycle shall, however, be counted as an additional lifting cycle and added to the value of N;

k_5^* is the specific spectrum ratio factor.

Standardized, conventional values in accordance with [Table 10](#) shall be used for design of hook shank. See also [Annex E](#).

Table 10 — Specific spectrum ratio factors k_5^*

| Class Q of ISO 4301-1 extended. to Q ₀ -Q ₅ | Load spectrum factor kQ | Factor k_5^* for $m = 5$ |
|---|---------------------------|----------------------------|
| Q ₀ | 0,031 3 | 1,292 |
| Q ₁ | 0,062 5 | 1,286 |
| Q ₂ | 0,125 | 1,22 |
| Q ₃ | 0,250 | 1,14 |
| Q ₄ | 0,500 | 1,07 |
| Q ₅ | 1,000 | 1,00 |

6.6.11 Execution of the proof

For the proof of fatigue strength it shall be proven that

$$\sigma_{Tmax} \leq \frac{\sigma_W}{\gamma_{Sf} \times \sqrt[m]{s_s}} \quad (54)$$

where

σ_{Tmax} is the maximum, transformed stress amplitude within the total stress history;

σ_W is the limit fatigue design stress in accordance with Formula (42);

γ_{Sf} is the fatigue strength specific resistance factor for hook shank ($\gamma_{Sf} = 1,35$);

m is the slope parameter of the characteristic fatigue design curve ($m = 5$);

s_s is the stress history parameter.

6.7 Fatigue design of hook shanks for serially produced hooks

The following design assumptions shall be used as a minimum for the design of hook shanks in serially produced hooks with a finished shank:

- a) Fatigue limit design force of the hook body shall be used as a fatigue design force for the shank;
- b) Number of shank bending cycles due to horizontal load sway is $p_a = 4$;
- c) For calculation of horizontal fatigue design force in [6.3](#) the horizontal acceleration is set $a = 0,2 \text{ m/s}^2$ and $\phi_5 = 1$;
- d) Hook suspension tilting resistance is assumed to correspond to a horizontal force in the hook seat equal to 2 % of the vertical force.

7 Verification of conformity with the requirements

7.1 General

Conformity with the requirements given in [Clauses 5](#) and [6](#) shall be verified by design calculations.

The design assumptions, e.g. intended duty and the intended hook capacity, shall conform to the corresponding design parameters of the related crane. This conformity shall be verified by engineering assessment.

All verifications in accordance with [Clause 7](#) shall be documented as a part of the technical file. See also [Annex K](#) for required documentation.

7.2 Verification of manufacture

Manufacturing conformity shall be verified through adherence to a written description of the hook manufacturing process, documented and certified by the manufacturer.

Conformity with the dimensional and material requirements shall be verified by measurements and tests. The results shall be recorded and retained by the hook manufacturer.

The test pieces for tensile, elongation and impact testing shall be taken longitudinally at the upper part of the hooks shank, preferably at a distance of 1/3 radius from the shanks surface. As an alternative, e.g. where the shank is too small, tests may be carried out on sample material selected from the same material melt and subjected to identical heat treatment. The required tensile/elongation qualities shall be specified by the manufacturer on the basis of ISO 7500-1 and either ISO 6892-1 or ISO 15579 as the case dictates.

The required Charpy impact qualities shall be specified by the manufacturer on the basis of ISO 148-1 and ISO 148-2.

7.3 Proof loading

A hook shall be able to withstand the proof load without excessive permanent deflection. The throat-opening dimension shall be measured before and after the test loading, using the specific measure points (1), see [Figure 7](#). A permanent set shall not exceed 0,25 %.

7.4 None destructive testing (NDT)

Hooks shall be examined, both for surface and internal defects using NDT methods that permit reliable detection.

Hook forging shall be inspected for defects using appropriate NDT-methods according to EN 10228-3, quality class 1 of this standard shall be met.

7.5 Test sampling

Material tests are to be carried out on either each individual hook or on the production batch principle.

Hooks having a ruling thickness 150 mm or greater, shall have all tests carried out on each and every hook.

Hooks having a ruling thickness less than 150 mm may be batch tested. The maximum batch size shall comprise the number of hooks, which can be manufactured from the same raw material cast or billet and undergoes identical heat treatment.

8 Information for use

8.1 Maintenance and inspection

The hook shall be handled as an issue of its own in the maintenance and inspection manuals of the related crane.

Maintenance of the following items shall be addressed as a minimum in the maintenance manual:

- a) thrust bearing under the nut;
- b) crosshead hinge.

The following items shall be addressed as a minimum in the inspection instruction, with their related frequency of inspection and rejection criteria:

- c) deformation (gap opening) of the hook body;
- d) wear of the hook body;
- e) inspection for surface defects, cracks and corrosion, with the hook suspension disassembled for the inspection of shank;
- f) safety locking of the nut;
- g) safety latch, if provided.

The acceptable wear depth of the hook body at the bottom of the seat is 5 % of the nominal height of the body section, dimension h_2 in [Annex A](#). The worn areas shall have smooth transition to adjacent areas. They shall be free of any sharp marks or edges, or defects opening onto the surface.

8.2 Marking

The hook body shall have a permanent marking positioned as item 2 in [Figure 7](#) specifying the following:

- a) size and shape of hook using a unique identification, e.g. hook number in accordance with [Annex A](#) or B;
- b) material designation, either the class symbol in accordance with [4.1](#) or other documented designation;
- c) number of the reference standard or specification;
- d) if relevant, marking of the proof loading, see [4.5](#).

EXAMPLE A hook fulfilling the requirements of this International Standard, size and shape being according to number 12 of [Annex A](#) and being made from material P has the marking:

RS12 - P - ISO 17440

Additionally the hook may be marked with a designation or symbol identifying the manufacturer. The hook body itself shall have no marking indicating either the load or the duty classification.

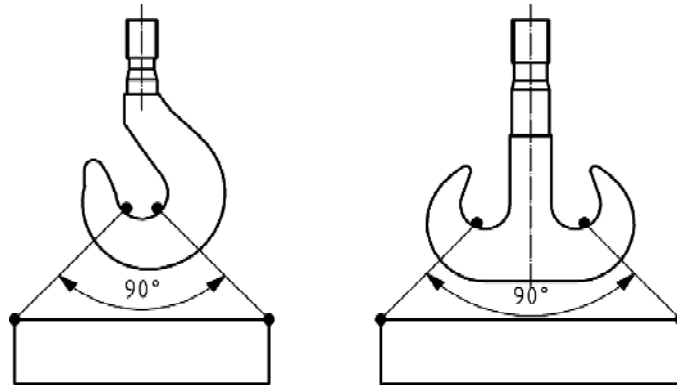


Figure 7 — Markings on a hook

The hook shall have permanent centre punch markings placed as item 1 in [Figure 7](#). As appropriate dimensions y or y_1 and y_2 shall be recorded and placed within hook documentation.

The fixed hoist media, from which the hook is suspended, should be marked as a part of the crane, indicating the mass of the rated capacity and the related “A” series duty classification (as per ISO 4301-1), as required by the relevant crane type standard.

8.3 Safe use

The following issues of safe use shall be, as a minimum, addressed in the user’s manual of either the related crane or the stand alone hook/hook block as a component:

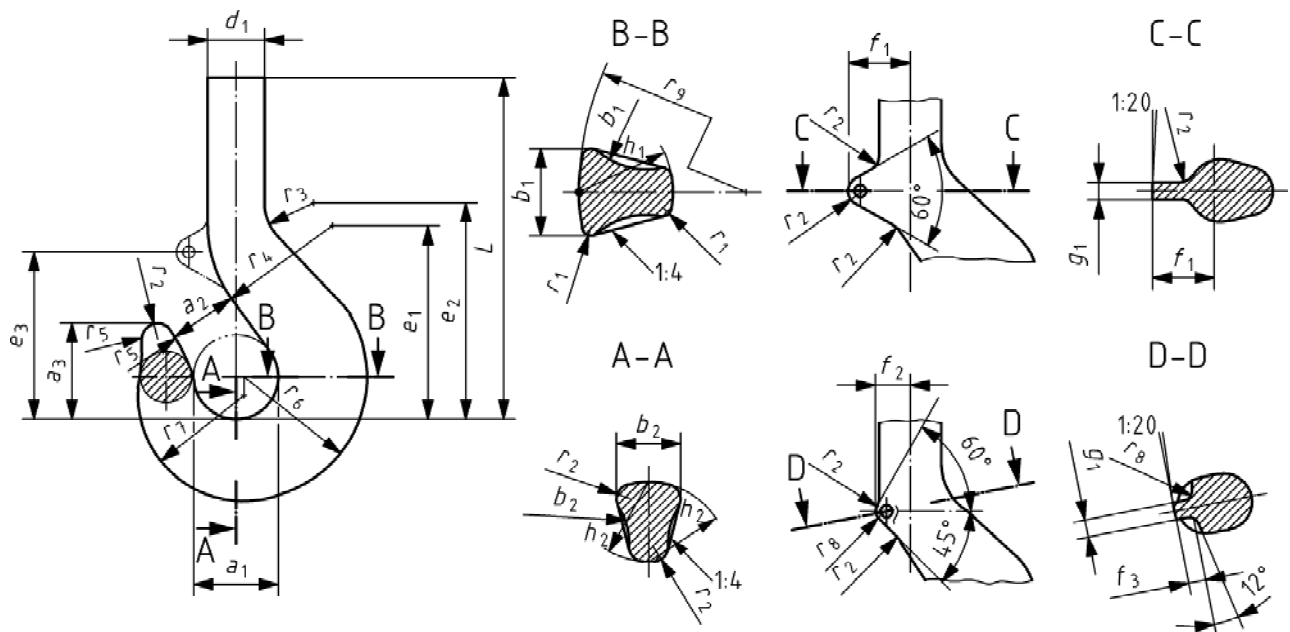
- free functioning of the hook suspension articulation (hinge), allowing the hook to align without obstacles in the direction of the load, either vertically or inclined during load sway;
- instructions for lashing a load on the hook, maximum 90° angle between the slings;
- shape requirement of the load attachment set on the hook, to avoid damage of the surface of the hook seat;
- the two prongs of a ramshorn hook shall be loaded symmetrically and equally;
- in cases, where a safety latch is provided, it shall be allowed to close freely after the load is attached;
- temperature limits of the hook.

Annex A (informative)

Sample sets of single point hooks

A.1 A series of single point hooks of type RSN, dimensions of forgings

See [Figure A.1](#) and [Table A.1](#).



Key

Designation:

RF without forged nose for latch

Figure A.1 — Symbols of dimensions for single point hooks with concave flanks

For hooks type RSN, see also [C.1](#), [D.1](#) and either [G.1](#) or [G.2](#) dependent upon size.

Table A.1 — Dimensions of forgings for single point hooks in millimetres (mm)

| Single hook no. | a_1 | a_2 | a_3 | b_1 | b_2 | d_1 | e_1 | e_2 | h_1 | h_2 | r_1 | r_2 | r_3 | r_4 | r_5 | r_6 | r_7 | r_9 | Dimensions for guidance | | | | | L |
|-----------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------------------------|-------|-------|-------|-------|-----|
| | | | | | | | | | | | | | | | | | | | e_3 | f_1 | f_2 | f_3 | g_1 | |
| 006 | 25 | 20 | 28 | 13 | 11 | 14 | 60 | 60 | 17 | 14 | 2 | 3 | 32 | 53 | 53 | 27 | 26 | 34 | 52 | 14,5 | - | 6,5 | - | 100 |
| 010 | 28 | 22 | 32 | 16 | 13 | 16 | 67 | 68 | 20 | 17 | 2 | 3,5 | 35 | 60 | 60 | 31 | 30 | 40 | 60 | 16,5 | - | 7 | - | 109 |
| 012 | 30 | 24 | 34 | 19 | 15 | 16 | 71 | 73 | 22 | 19 | 2,5 | 4 | 37 | 63 | 63 | 34 | 33 | 44 | 63 | 18 | - | 7,5 | - | 115 |
| 020 | 34 | 27 | 39 | 21 | 18 | 20 | 81 | 82 | 26 | 22 | 2,5 | 4,5 | 40 | 71 | 71 | 39 | 37 | 52 | 70 | 20 | - | 8,5 | - | 138 |
| 025 | 36 | 28 | 41 | 22 | 19 | 20 | 85 | 88 | 28 | 24 | 3 | 5 | 43 | 75 | 75 | 42 | 40 | 56 | 74 | 22 | - | 9 | - | 144 |
| 04 | 40 | 32 | 45 | 27 | 22 | 24 | 96 | 100 | 34 | 29 | 3,5 | 5,5 | 46 | 85 | 85 | 49 | 45 | 68 | 83 | 25 | - | 10 | - | 155 |
| 05 | 43 | 34 | 49 | 29 | 24 | 24 | 102 | 108 | 37 | 31 | 4 | 6 | 48 | 90 | 90 | 53 | 48 | 74 | 89 | 26 | - | 10,5 | - | 167 |
| 08 | 48 | 38 | 54 | 35 | 29 | 30 | 115 | 120 | 44 | 37 | 4,5 | 7 | 52 | 100 | 100 | 61 | 56 | 88 | 100 | 29 | - | 12 | - | 186 |
| 1 | 50 | 40 | 57 | 38 | 32 | 30 | 120 | 128 | 48 | 40 | 5 | 8 | 55 | 106 | 106 | 65 | 60 | 96 | 105 | 31 | - | 12,5 | - | 197 |
| 1.6 | 56 | 45 | 64 | 45 | 38 | 36 | 135 | 146 | 56 | 48 | 6 | 9 | 60 | 118 | 118 | 76 | 68 | 112 | 118 | 35 | - | 14 | - | 224 |
| 2.5 | 63 | 50 | 72 | 53 | 45 | 42 | 152 | 167 | 67 | 58 | 7 | 10 | 65 | 132 | 132 | 90 | 78 | 134 | 132 | 40 | - | 16 | - | 253 |
| 4 | 71 | 56 | 80 | 63 | 53 | 48 | 172 | 190 | 80 | 67 | 8 | 12 | 71 | 150 | 150 | 103 | 90 | 160 | 148 | 45 | - | 16 | - | 285 |

NOTE Dimensions based upon DIN 15400 series of hooks.

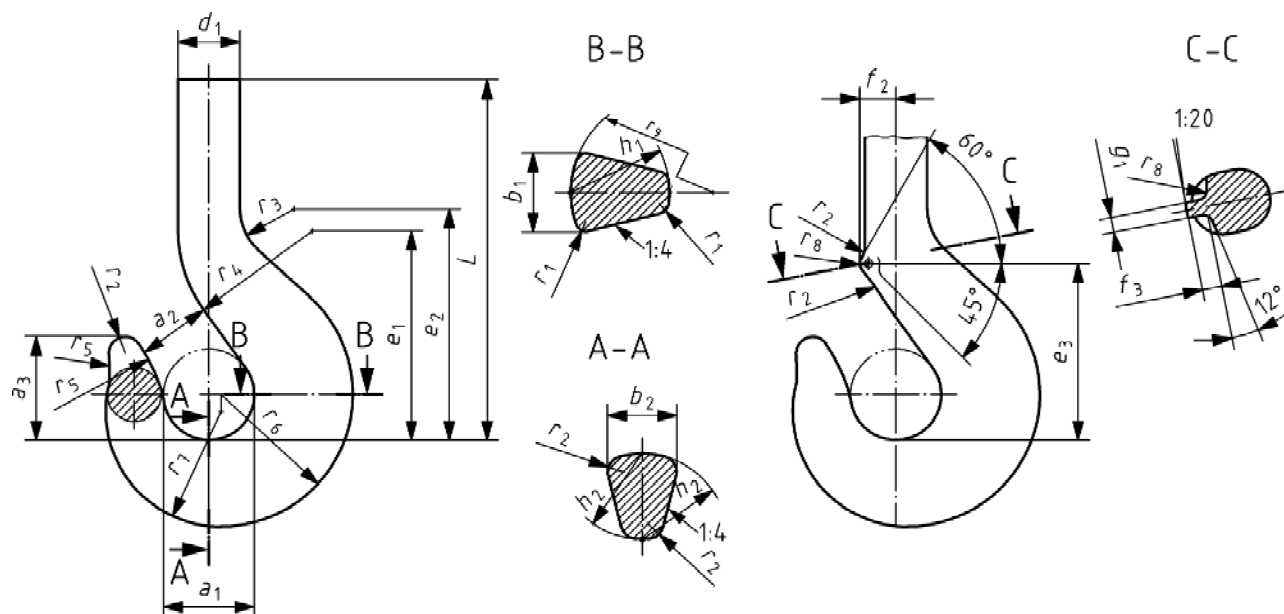
Table A.1 — (continued)

| | | | | | | | | | | | | | | | | | | | | | | | | | |
|----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|----|----|-----|-----|-----|-----|-----|-----|-----|----|----|----|----|----|-----|
| 5 | 80 | 63 | 90 | 71 | 60 | 53 | 194 | 215 | 90 | 75 | 9 | 14 | 80 | 170 | 170 | 114 | 100 | 180 | 165 | 51 | - | - | 18 | - | 318 |
| 6 | 90 | 71 | 101 | 80 | 67 | 60 | 218 | 240 | 100 | 85 | 10 | 16 | 90 | 190 | 190 | 131 | 112 | 200 | 185 | 57 | - | - | 18 | - | 380 |
| 8 | 100 | 80 | 113 | 90 | 75 | 67 | 242 | 268 | 112 | 95 | 11 | 18 | 100 | 212 | 212 | 146 | 125 | 224 | 210 | 64 | - | - | 23 | - | 418 |
| 10 | 112 | 90 | 127 | 100 | 85 | 75 | 256 | 286 | 125 | 106 | 12 | 20 | 65 | 165 | 236 | 163 | 140 | 250 | 221 | - | 46 | 26 | 23 | 12 | 452 |
| 12 | 125 | 100 | 143 | 112 | 95 | 85 | 292 | 316 | 140 | 118 | 14 | 22 | 70 | 185 | 265 | 182 | 160 | 280 | 252 | - | 53 | 34 | 28 | 16 | 510 |
| 16 | 140 | 112 | 160 | 125 | 106 | 95 | 325 | 357 | 160 | 132 | 16 | 25 | 80 | 210 | 300 | 204 | 180 | 320 | 280 | - | 58 | 35 | 33 | 16 | 582 |
| 20 | 160 | 125 | 180 | 140 | 118 | 106 | 370 | 405 | 180 | 150 | 18 | 28 | 90 | 240 | 335 | 232 | 200 | 360 | 330 | - | 68 | 45 | 33 | 20 | 653 |
| 25 | 180 | 140 | 202 | 160 | 132 | 118 | 415 | 455 | 200 | 170 | 20 | 32 | 100 | 270 | 375 | 262 | 224 | 400 | 360 | - | 74 | 45 | 38 | 20 | 724 |
| 32 | 200 | 160 | 225 | 180 | 150 | 132 | 465 | 510 | 224 | 190 | 22 | 36 | 115 | 300 | 425 | 292 | 250 | 448 | 400 | - | 80 | 45 | 38 | 20 | 796 |
| 40 | 224 | 180 | 252 | 200 | 170 | 150 | 517 | 567 | 250 | 212 | 25 | 40 | 130 | 335 | 475 | 326 | 280 | 500 | 447 | - | 93 | 55 | 42 | 25 | 893 |

NOTE Dimensions based upon DIN 15400 series of hooks.

A.2 A series of single point hooks of type RF/RFN, dimensions of forgings

See [Figure A.2](#) and [Table A.2](#).



Key

Designation:

RF without forged nose for latch

RFN with forged nose for latch

Figure A.2 — Symbols of dimensions for single hooks with straight flanks

For hooks type RF and RFN, see also [C.1](#), [D.1](#) and either [G.1](#) or [G.2](#) dependent upon size.

Table A.2 — Dimensions of forgings for single point hooks in millimetres (mm)

| Single hook no. | a_1 | a_2 | a_3 | b_1 | b_2 | d_1 | e_1 | e_2 | h_1 | h_2 | r_1 | r_2 | r_3 | r_4 | r_5 | r_6 | r_7 | r_9 | Dimensions for guidance | | | | | | | |
|-----------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------------------------|-------|-------|-------|-------|-------|--|--|
| | | | | | | | | | | | | | | | | | | | e_3 | f_2 | f_3 | g_1 | r_8 | L | | |
| 10 | 112 | 90 | 127 | 100 | 85 | 75 | 256 | 286 | 125 | 106 | 12 | 20 | 65 | 165 | 236 | 163 | 140 | 250 | 221 | 46 | 26 | 23 | 12 | 460 | | |
| 12 | 125 | 100 | 143 | 112 | 95 | 85 | 292 | 316 | 140 | 118 | 14 | 22 | 70 | 185 | 265 | 182 | 160 | 280 | 252 | 53 | 34 | 28 | 16 | 525 | | |
| 16 | 140 | 112 | 160 | 125 | 106 | 95 | 325 | 357 | 160 | 132 | 16 | 25 | 80 | 210 | 300 | 204 | 180 | 320 | 280 | 58 | 35 | 33 | 16 | 595 | | |
| 20 | 160 | 125 | 180 | 140 | 118 | 106 | 370 | 405 | 180 | 150 | 18 | 28 | 90 | 240 | 335 | 232 | 200 | 360 | 330 | 68 | 45 | 33 | 20 | 665 | | |
| 25 | 180 | 140 | 202 | 160 | 132 | 118 | 415 | 455 | 200 | 170 | 20 | 32 | 100 | 270 | 375 | 262 | 224 | 400 | 360 | 74 | 45 | 38 | 20 | 735 | | |
| 32 | 200 | 160 | 225 | 180 | 150 | 132 | 465 | 510 | 224 | 190 | 22 | 36 | 115 | 300 | 425 | 292 | 250 | 448 | 400 | 80 | 45 | 38 | 20 | 810 | | |
| 40 | 224 | 180 | 252 | 200 | 170 | 150 | 517 | 567 | 250 | 212 | 25 | 40 | 130 | 335 | 475 | 326 | 280 | 500 | 447 | 93 | 55 | 42 | 25 | 905 | | |
| 50 | 250 | 200 | 285 | 224 | 190 | 170 | 575 | 635 | 280 | 236 | 28 | 45 | 150 | 370 | 530 | 363 | 315 | 560 | 485 | 100 | 55 | 42 | 25 | 990 | | |
| 63 | 280 | 224 | 320 | 250 | 212 | 190 | 655 | 710 | 315 | 265 | 32 | 50 | 160 | 420 | 600 | 408 | 355 | 630 | 550 | 108 | 60 | 45 | 25 | 1 120 | | |
| 80 | 315 | 250 | 358 | 280 | 236 | 212 | 727 | 802 | 355 | 300 | 36 | 56 | 180 | 470 | 670 | 460 | 400 | 710 | 598 | 113 | 60 | 45 | 25 | 1 270 | | |
| 100 | 355 | 280 | 402 | 315 | 265 | 236 | 827 | 902 | 400 | 335 | 40 | 63 | 200 | 530 | 750 | 516 | 450 | 800 | 688 | 130 | 70 | 50 | 30 | 1 415 | | |
| 125 | 400 | 315 | 450 | 355 | 300 | 265 | 920 | 1 020 | 450 | 375 | 45 | 71 | 230 | 600 | 850 | 579 | 500 | 900 | 750 | 138 | 70 | 50 | 30 | 1 590 | | |
| 160 | 450 | 355 | 505 | 400 | 335 | 300 | 1 035 | 1 145 | 500 | 425 | 50 | 80 | 250 | 675 | 950 | 654 | 560 | 1 000 | 825 | 147 | 70 | 55 | 30 | 1 790 | | |
| 200 | 500 | 400 | 565 | 450 | 375 | 335 | 1 150 | 1 275 | 560 | 475 | 56 | 90 | 285 | 750 | 1 060 | 729 | 630 | 1 120 | 900 | 154 | 70 | 55 | 30 | 2 048 | | |
| 250 | 560 | 450 | 635 | 500 | 425 | 375 | 1 280 | 1 430 | 630 | 530 | 63 | 100 | 320 | 840 | 1 180 | 815 | 710 | 1 260 | 980 | 164 | 70 | 60 | 30 | 2 305 | | |

NOTE Dimensions based upon DIN 15400 series of hooks.

A.3 A series of single point hooks of type B, dimensions of forgings

See [Figure A.3](#) and [Table A.3](#).

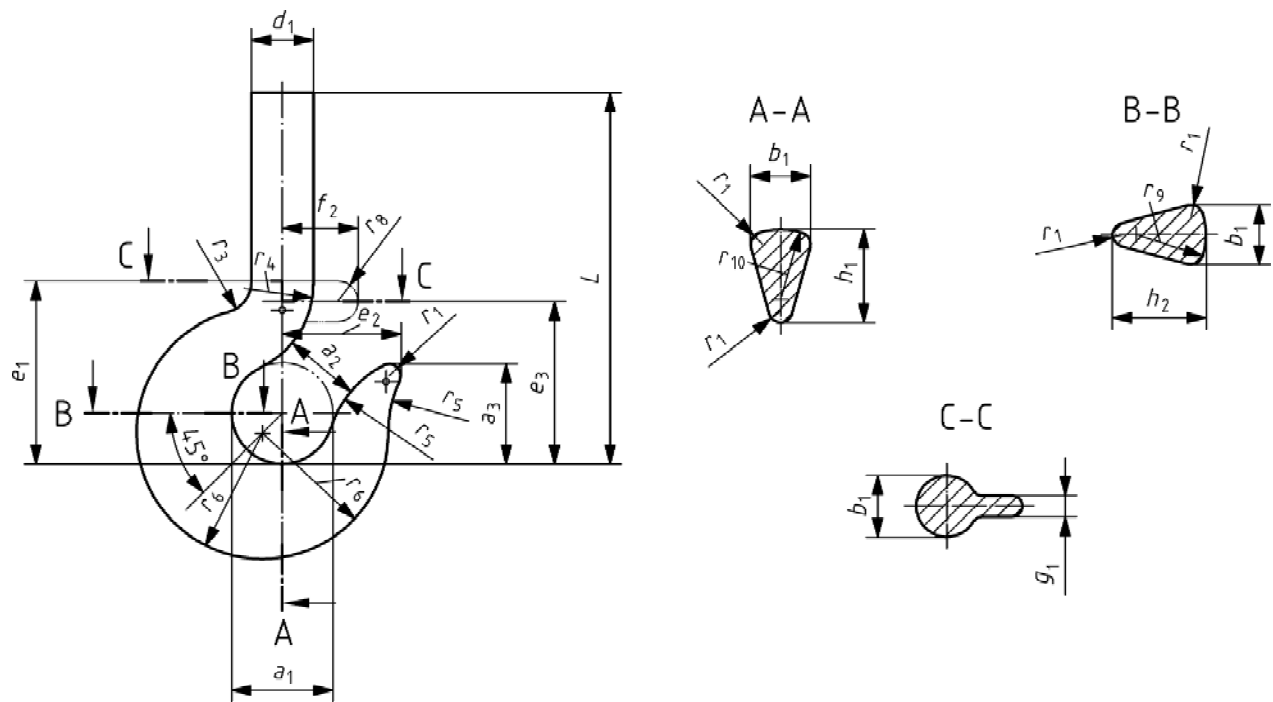


Figure A.3 — Symbols of dimensions for single point hooks

For hook type B, see also [C.2](#), [D.2](#) and [G.4](#).

Table A.3 — Dimensions of forgings for single point hooks in millimetres (mm)

| Single hook no. | a_1 | a_2 | a_3 | b_1 | d_1 | e_1 | e_2 | h_1 | h_2 | r_1 | r_9 | r_{10} | r_3 | r_4 | r_5 | r_6 | Dimensions for guidance | | | |
|-----------------|-------|------------|-------|------------|-------|-------|-------|------------|------------|------------|-------|----------|------------|------------|------------|------------|-------------------------|-------|-------|-------|
| | (1) | $0,75 a_1$ | | $0,60 a_1$ | | | | $0,93 a_1$ | $0,93 a_1$ | $0,12 a_1$ | | | $0,30 a_1$ | $0,92 a_1$ | $1,00 a_1$ | $1,25 a_1$ | e_3 | f_2 | r_8 | g_1 |
| B 0.8 | 26 | 20 | 26 | 16 | 17 | 47 | 31 | 24 | 24 | 3 | 18 | 13 | 8 | 24 | 26 | 32 | 41,8 | 19,7 | 5,2 | 5,2 |
| B 1.6 | 37 | 28 | 37 | 22 | 23 | 66,5 | 44 | 34 | 34 | 4 | 26 | 18 | 11 | 34 | 37 | 46 | 59,1 | 27,7 | 7,4 | 7,4 |
| B 2.5 | 46 | 34 | 46 | 28 | 28 | 83 | 55 | 43 | 43 | 6 | 32 | 23 | 14 | 42 | 46 | 58 | 73,8 | 34,7 | 9,2 | 9,2 |
| B 4 | 58 | 44 | 58 | 35 | 33 | 105 | 70 | 54 | 54 | 7 | 41 | 29 | 17 | 53 | 58 | 72 | 93,4 | 43,6 | 11,6 | 11,6 |
| B 5 | 65 | 49 | 65 | 39 | 38 | 118 | 78 | 60 | 60 | 8 | 46 | 32 | 20 | 60 | 65 | 81 | 105 | 48,8 | 13 | 13 |
| B 6.3 | 73 | 55 | 73 | 44 | 44 | 139 | 88 | 68 | 68 | 9 | 51 | 13 | 22 | 67 | 73 | 91 | 124 | 54,9 | 14,6 | 14,6 |
| B 8 | 83 | 62 | 84 | 50 | 50 | 151 | 100 | 77 | 77 | 10 | 58 | 18 | 25 | 76 | 83 | 104 | 134 | 62,4 | 16,6 | 16,6 |
| B 10 | 92 | 69 | 92 | 55 | 55 | 167 | 110 | 86 | 86 | 11 | 64 | 46 | 28 | 85 | 92 | 115 | 149 | 68,9 | 18,4 | 18,4 |
| B 12.5 | 103 | 77 | 104 | 62 | 60 | 187 | 124 | 96 | 96 | 12 | 72 | 52 | 31 | 95 | 103 | 129 | 166 | 77,4 | 20,6 | 20,6 |
| B 16 | 117 | 88 | 117 | 70 | 65 | 212 | 140 | 109 | 109 | 14 | 82 | 58 | 35 | 108 | 117 | 146 | 189 | 87,7 | 23,4 | 23,4 |
| B 20 | 131 | 98 | 132 | 79 | 75 | 237 | 157 | 122 | 122 | 16 | 92 | 66 | 39 | 120 | 131 | 164 | 211 | 98,5 | 26,2 | 26,2 |
| B 25 | 146 | 110 | 146 | 88 | 85 | 264 | 175 | 136 | 136 | 18 | 102 | 73 | 44 | 134 | 146 | 182 | 235 | 110 | 29,2 | 29,2 |
| B 32 | 159 | 119 | 160 | 95 | 90 | 288 | 191 | 148 | 148 | 19 | 111 | 80 | 48 | 146 | 159 | 199 | 256 | 119 | 31,8 | 31,8 |
| B 40 | 173 | 130 | 173 | 104 | 105 | 314 | 208 | 161 | 161 | 21 | 121 | 86 | 52 | 159 | 173 | 216 | 279 | 130 | 34,6 | 34,6 |
| B 50 | 191 | 143 | 191 | 115 | 115 | 346 | 229 | 178 | 178 | 23 | 134 | 95 | 57 | 176 | 191 | 239 | 308 | 143 | 38,2 | 38,2 |
| B 63 | 205 | 154 | 205 | 123 | 125 | 372 | 246 | 191 | 191 | 25 | 144 | 102 | 62 | 189 | 205 | 256 | 331 | 154 | 41 | 41 |

NOTE 1 Dimensions based upon BS 2903 series of hooks.

NOTE 2 The second row of dimensions gives a ratio to the measurement a_1 , internal diameter of hook bowl in mm.

For hooks complying with these proportions, the value of a_1 in mm is calculated from the equation, $a_1 = 6,62 \sqrt{P}$, where P is the proof load in kN. See C.2 for value of this proof load.

NOTE 3 Nose for attachment of latch is optional.

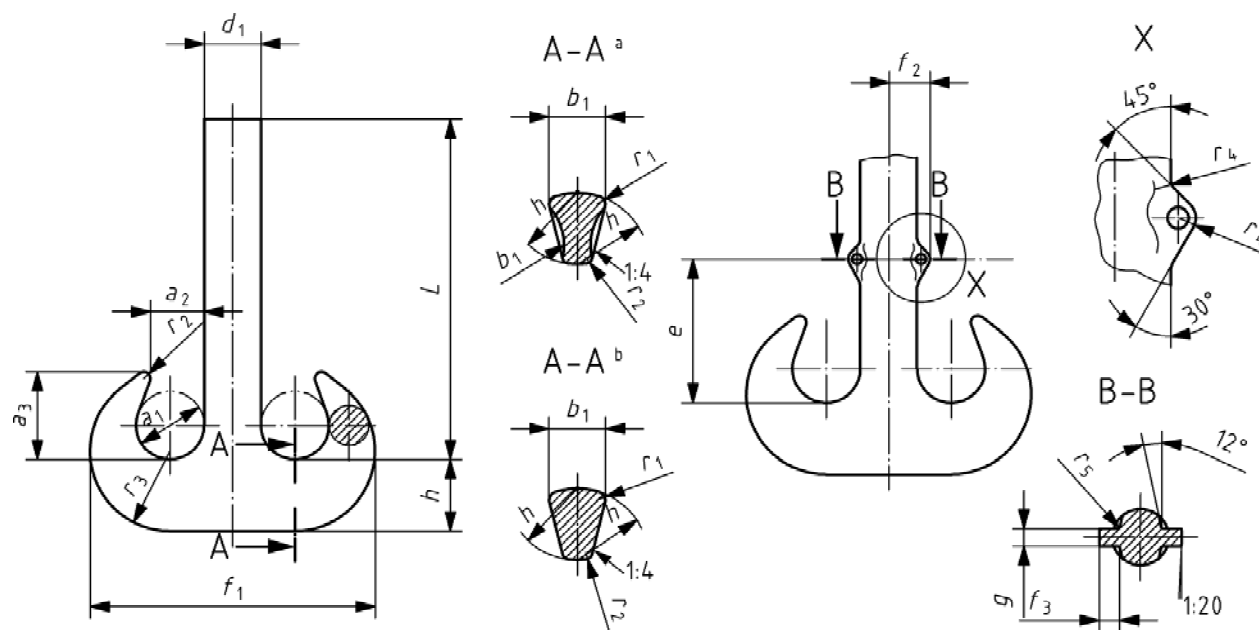
NOTE 4 Length of hook shank "L" to suit application.

Annex B (informative)

Sample set of ramshorn hooks

A series of ramshorn hooks of type RS/RSN and RF/RFN, dimensions of forgings

See [Figure B.1](#) and [Table B.1](#).



Key

Designation:

RS/RSN concave flanks (a), without or with nose

RF/RFN straight flanks (b), without or with nose

Figure B.1 — Symbols of dimensions for ramshorn hooks

Table B.1 — Dimensions of forgings for ramshorn hooks in millimetres (mm)

| Ramshorn hook no. | a_1 | a_2 | a_3 | b_1 | d_1 | f_1 | H | r_1 | r_2 | r_3 | Dimensions for guidance | | | | | | |
|-------------------|-------|-------|-------|-------|-------|-------|-----|-------|-------|-------|-------------------------|-------|-------|------|-------|-------|-----|
| | | | | | | | | | | | e | f_2 | f_3 | g | r_4 | r_5 | L |
| 05 | 34 | 27 | 44 | 22 | 24 | 130 | 27 | 3 | 3 | 36 | 80 | 20 | 12 | 10 | 6 | 1,6 | 165 |
| 08 | 38 | 30 | 49 | 26 | 30 | 150 | 33 | 4 | 3 | 41 | 83 | 22 | 12 | 10,5 | 6 | 1,6 | 183 |
| 1 | 40 | 32 | 52 | 28 | 30 | 158 | 36 | 4 | 3,5 | 44 | 96 | 22 | 14 | 12 | 7 | 1,6 | 195 |
| 1.6 | 45 | 36 | 59 | 34 | 36 | 183 | 43 | 5 | 4 | 51 | 100 | 28 | 14 | 12,5 | 7 | 1,6 | 222 |
| 2.5 | 50 | 40 | 65 | 40 | 42 | 208 | 50 | 6 | 4,5 | 58 | 112 | 30 | 14 | 14 | 7 | 1,6 | 250 |
| 4 | 56 | 45 | 73 | 48 | 48 | 238 | 60 | 7 | 5,5 | 67 | 124 | 33 | 23 | 16 | 10 | 2,5 | 280 |
| 5 | 63 | 50 | 82 | 53 | 53 | 266 | 67 | 8 | 6,5 | 75 | 143 | 40 | 23 | 16 | 10 | 2,5 | 312 |
| 6 | 71 | 56 | 92 | 60 | 60 | 301 | 75 | 9 | 7 | 85 | 160 | 44 | 23 | 18 | 10 | 2,5 | 375 |

Hook sizes 50–250 preferably with straight flanks.

NOTE Dimensions based upon DIN 15400 series of hooks.

Table B.1 (continued)

| Ramshorn hook no. | a_1 | a_2 | a_3 | b_1 | d_1 | f_1 | H | r_1 | r_2 | r_3 | Dimensions for guidance | | | | | | |
|---|-------|-------|-------|-------|-------|-------|-----|-------|-------|-------|-------------------------|-------|-------|-----|-------|-------|-------|
| | | | | | | | | | | | e | f_2 | f_3 | g | r_4 | r_5 | L |
| 8 | 80 | 63 | 103 | 67 | 67 | 337 | 85 | 10 | 8 | 95 | 182 | 48 | 23 | 18 | 10 | 2,5 | 415 |
| 10 | 90 | 71 | 116 | 75 | 75 | 377 | 95 | 11 | 9 | 106 | 192 | 54 | 27 | 23 | 12 | 3 | 450 |
| 12 | 100 | 80 | 130 | 85 | 85 | 421 | 106 | 12,5 | 10 | 118 | 210 | 60 | 27 | 23 | 12 | 3 | 510 |
| 16 | 112 | 90 | 146 | 95 | 95 | 471 | 118 | 14 | 11 | 132 | 237 | 69 | 36 | 28 | 16 | 4 | 580 |
| 20 | 125 | 100 | 163 | 106 | 106 | 531 | 132 | 16 | 12,5 | 150 | 265 | 75 | 36 | 33 | 16 | 4 | 650 |
| 25 | 140 | 112 | 182 | 118 | 118 | 598 | 150 | 18 | 14 | 170 | 315 | 86 | 45 | 33 | 20 | 5 | 715 |
| 32 | 160 | 125 | 205 | 132 | 132 | 672 | 170 | 20 | 16 | 190 | 335 | 94 | 45 | 38 | 20 | 5 | 790 |
| 40 | 180 | 140 | 230 | 150 | 150 | 754 | 190 | 22 | 18 | 212 | 375 | 104 | 45 | 38 | 20 | 5 | 885 |
| 50 | 200 | 160 | 260 | 170 | 170 | 842 | 212 | 25 | 20 | 236 | 420 | 120 | 56 | 42 | 25 | 6 | 965 |
| 63 | 224 | 180 | 292 | 190 | 190 | 944 | 236 | 28 | 22 | 265 | 460 | 131 | 56 | 42 | 25 | 6 | 1 090 |
| 80 | 250 | 200 | 325 | 212 | 212 | 1 062 | 265 | 32 | 25 | 300 | 515 | 144 | 56 | 45 | 25 | 6 | 1 235 |
| 100 | 280 | 224 | 364 | 236 | 236 | 1 186 | 300 | 36 | 28 | 335 | 575 | 157 | 56 | 45 | 25 | 6 | 1 375 |
| 125 | 315 | 250 | 408 | 265 | 265 | 1 330 | 335 | 40 | 32 | 375 | 645 | 178 | 68 | 50 | 30 | 8 | 1 550 |
| 160 | 355 | 280 | 458 | 300 | 300 | 1 505 | 375 | 45 | 36 | 425 | 725 | 198 | 68 | 50 | 30 | 8 | 1 745 |
| 200 | 400 | 315 | 515 | 335 | 335 | 1 685 | 425 | 50 | 40 | 475 | 800 | 218 | 68 | 55 | 30 | 8 | 1 998 |
| 250 | 450 | 355 | 580 | 375 | 375 | 1 885 | 475 | 56 | 45 | 530 | 875 | 240 | 68 | 55 | 30 | 8 | 2 250 |
| Hook sizes 50–250 preferably with straight flanks. | | | | | | | | | | | | | | | | | |
| NOTE Dimensions based upon DIN 15400 series of hooks. | | | | | | | | | | | | | | | | | |

For hooks type RS/RF and RSN/RFN, see also [C.1](#), [D.1](#) and either [G.1](#) or [G.2](#) dependent upon size.

Annex C (informative)

Annexes A and B static limit design forces for hook bodies

C.1 Static limit design forces of hook bodies for hook types RS/RSN and RF/RFN (see Table C.1.)

Table C.1 — Static limit design forces $F_{Rd,s}$ in kilonewtons (kN) — Valid for temperature influence factor $f_1 = 1$ ($T \leq 100$ °C)

| Hook no. | Factor γ_{sm} | Single hooks, types RS/RSN and RF/RFN | | | | | | Ramshorn hooks, types RS/RSN and RF/RFN | | | | | | |
|----------|-------------------------|--|---------------------------------------|--------|--------|--------|--------|--|---------------------------------------|--------|--------|--------|--------|-------|
| | | Section B Hook factor M_{hf} , mm ² | Classified material grades $F_{Rd,s}$ | | | | | Section A Hook factor M_{hf} , mm ² | Classified material grades $F_{Rd,s}$ | | | | | |
| | | | M | P | S | T | V | | M | P | S | T | V | |
| 006 | 0.75 single hooks | 15,37 | 4,0 | 5,9 | 7,1 | 9,3 | 11,2 | - | - | - | - | - | - | |
| 010 | | 22,58 | 5,9 | 8,6 | 10,4 | 14 | 16 | - | - | - | - | - | - | |
| 012 | | 30,05 | 7,8 | 11,5 | 14 | 18 | 22 | - | - | - | - | - | - | |
| 020 | | 39,37 | 10,3 | 15 | 18 | 24 | 29 | - | - | - | - | - | - | |
| 025 | | 44,78 | 11,7 | 17 | 21 | 27 | 33 | - | - | - | - | - | - | |
| 04 | | 68,37 | 18 | 26 | 32 | 41 | 50 | - | - | - | - | - | - | |
| 05 | | 80,33 | 21 | 31 | 37 | 49 | 58 | 41,71 | 18 | 27 | 32 | 42 | 51 | |
| 08 | | 117,00 | 30 | 45 | 54 | 71 | 85 | 62,13 | 27 | 40 | 48 | 63 | 75 | |
| 1 | | 140,17 | 37 | 54 | 65 | 85 | 102 | 73,43 | 32 | 47 | 56 | 74 | 89 | |
| 1.6 | | 195,75 | 51 | 75 | 90 | 119 | 142 | 108,20 | 47 | 69 | 83 | 109 | 131 | |
| 2.5 | | 279,05 | 73 | 107 | 129 | 169 | 203 | 149,45 | 65 | 95 | 115 | 151 | 181 | |
| 4 | | 399,61 | 104 | 153 | 184 | 242 | 291 | 217,87 | 95 | 139 | 167 | 220 | 264 | |
| 5 | | 506,63 | 132 | 193 | 233 | 307 | 368 | 268,78 | 117 | 171 | 206 | 272 | 326 | |
| 6 | | 633,00 | 165 | 242 | 292 | 384 | 460 | 339,90 | 148 | 216 | 261 | 343 | 412 | |
| 8 | | 798,18 | 208 | 305 | 368 | 484 | 580 | 430,94 | 187 | 274 | 331 | 435 | 522 | |
| 10 | | 987,90 | 257 | 377 | 455 | 599 | 719 | 538,00 | 234 | 342 | 413 | 543 | 652 | |
| 12 | | 0.90 ram- shorn hooks | 1 242,5 | 324 | 474 | 572 | 753 | 904 | 680,27 | 295 | 433 | 522 | 687 | 825 |
| 16 | | | 1 590,0 | 414 | 607 | 732 | 963 | 1 156 | 845,31 | 367 | 538 | 649 | 854 | 1 024 |
| 20 | | | 1 873,3 | 520 | 763 | 920 | 1 210 | 1 452 | 1 056,5 | 459 | 672 | 811 | 1 067 | 1 281 |
| 25 | | | 2 532,0 | 660 | 967 | 1 166 | 1 535 | 1 841 | 1 343,0 | 583 | 855 | 1 031 | 1 356 | 1 628 |
| 32 | | | 3 192,7 | 832 | 1 219 | 1 471 | 1 935 | 2 322 | 1 700,5 | 739 | 1 082 | 1 305 | 1 718 | 2 061 |
| 40 | | | 3 732,4 | 1 032 | 1 512 | 1 824 | 2 400 | 2 880 | 2 152,0 | 935 | 1 369 | 1 652 | 2 174 | 2 609 |
| 50 | | | 5 020,5 | 1 308 | 1 917 | 2 313 | 3 043 | 3 651 | 2 725,4 | 1 184 | 1 734 | 2 092 | 2 753 | 3 304 |
| 63 | | | 6 306,7 | 1 644 | 2 408 | 2 905 | 3 822 | 4 587 | 3 385,9 | 1 471 | 2 155 | 2 599 | 3 420 | 4 104 |
| 80 | | | 7 962,7 | 2 075 | 3 040 | 3 667 | 4 826 | 5 791 | 4 251,1 | 1 846 | 2 705 | 3 263 | 4 294 | 5 153 |
| 100 | 10 093 | | 2 630 | 3 854 | 4 649 | 6 117 | 7 341 | 5 376,8 | 2 335 | 3 422 | 4 128 | 5 431 | 6 517 | |
| 125 | 12 794 | | 3 334 | 4 885 | 5 893 | 7 754 | 9 305 | 6 729,4 | 2 923 | 4 282 | 5 166 | 6 797 | 8 156 | |
| 160 | 15 987 | | 4 167 | 6 105 | 7 364 | 9 690 | 11 628 | 8 507,3 | 3 695 | 5 414 | 6 531 | 8 593 | 10 311 | |
| 200 | 20 174 | | 5 257 | 7 702 | 9 292 | 12 226 | 14 671 | 10 792 | 4 688 | 6 868 | 8 285 | 10 901 | 13 081 | |
| 250 | 25 240 | 6 575 | 9 634 | 11 622 | 15 292 | 18 350 | 13 484 | 5 857 | 8 581 | 10 352 | 13 621 | 16 345 | | |

C.2 Static limit design forces of hook bodies for a series of hooks of type B, with additional materials (see [Table C.2](#))

Table C.2 — Static limit design forces $F_{Rd,s}$ in kilonewtons (kN) — Valid for temperature influence factor $f_1 = 1$ ($T \leq 100$ °C)

| Hook no. | All materials | | Single hooks, type B | | | | | Additional materials | |
|---------------|----------------------------------|--|---------------------------------------|------|------|------|------|---------------------------|------------|
| | Hook family factor γ_{sm} | Section B Hook factor M_{hf} , mm ² | Classified material grades $F_{Rd,s}$ | | | | | f_y , N/mm ² | $F_{Rd,s}$ |
| | | | M | P | S | T | V | | |
| B 0.8 | 0,86 | 25,392 | 5,75 | 8,43 | 10,2 | 13,4 | 16,1 | 430 | 11,5 |
| B 1.6 | | 49,722 | 11,3 | 16,5 | 19,9 | 26,2 | 31,5 | | 22,5 |
| B 2.5 | | 81,543 | 18,5 | 27,1 | 32,6 | 43 | 51,6 | | 36,9 |
| B 4 | | 127,31 | 28,8 | 42,3 | 51 | 67,1 | 80,5 | | 57,7 |
| B 5 | | 157,79 | 35,8 | 52,4 | 63,2 | 83,2 | 99,8 | | 71,5 |
| B 6.3 | | 201,91 | 45,8 | 67 | 80,9 | 106 | 128 | | 91,5 |
| B 10 | | 319,35 | 72,4 | 106 | 128 | 168 | 202 | | 144 |
| B 8 | | 258,77 | 58,6 | 86 | 104 | 136 | 164 | | 117 |
| B 12.5 | | 399,34 | 90,5 | 133 | 160 | 210 | 253 | | 181 |
| B 16 | 0,83 | 514,76 | 121 | 179 | 216 | 284 | 340 | 400 | 227 |
| B 20 | | 649,89 | 154 | 226 | 272 | 358 | 430 | | 286 |
| B 25 | | 807,66 | 191 | 280 | 338 | 445 | 534 | | 356 |
| B 32 | 0,75 | 948,18 | 245 | 358 | 432 | 569 | 683 | 455 | |
| B 40 | 0,71 | 1128,78 | 310 | 454 | 547 | 720 | 864 | 576 | |
| B 50 | 0,68 | 1379,07 | 399 | 585 | 706 | 929 | 1114 | 742 | |
| B 63 | 0,64 | 1587,16 | 487 | 713 | 880 | 1132 | 1358 | 905 | |

All hooks of this family shall be proof loaded as per [4.5](#).

When dimensionally sizing all hooks of greater capacity than those listed within [Table C.2](#) it shall be assumed, for this calculation only, that the proof load F_{PL} is equal 1,5 times their required maximum static rated capacity.

For hook sizes greater than B 63, the value of γ_{sm} factor shall be taken as 0,64.

Annex D (informative)

Annexes A and B fatigue limit design forces for hook bodies

D.1 Fatigue limit design forces of hook bodies for hooks of type RS and RF (see [Table D.1](#))

Table D.1 — Fatigue limit design forces $F_{Rd,f}$ in kilonewtons (kN) — Factors f_2 and γ_{Hf} incorporated, temperature influence factor $f_1 = 1$ ($T \leq 100$ °C)

| Hook no. | Single hooks, types RS and RF | | | | | Ramshorn hooks, types RS and RF | | | | |
|----------|-------------------------------|-------|-------|-------|-------|---------------------------------|-------|-------|-------|-------|
| | Classified material grades | | | | | Classified material grades | | | | |
| | M | P | S | T | V | M | P | S | T | V |
| 006 | 2,1 | 2,7 | 2,9 | 3,4 | 3,8 | - | - | - | - | - |
| 010 | 3,1 | 4,0 | 4,2 | 5,1 | 5,5 | - | - | - | - | - |
| 012 | 4,1 | 5,3 | 5,6 | 6,7 | 7,3 | - | - | - | - | - |
| 020 | 5,4 | 6,9 | 7,4 | 8,8 | 9,6 | - | - | - | - | - |
| 025 | 6,1 | 7,9 | 8,4 | 10 | 11 | - | - | - | - | - |
| 04 | 9,2 | 12 | 13 | 15 | 16 | - | - | - | - | - |
| 05 | 11 | 14 | 15 | 18 | 19 | 11 | 14 | 15 | 17 | 19 |
| 08 | 15 | 19 | 21 | 25 | 27 | 16 | 20 | 22 | 26 | 28 |
| 1 | 18 | 23 | 25 | 29 | 32 | 18 | 23 | 25 | 30 | 33 |
| 1.6 | 24 | 31 | 33 | 40 | 43 | 26 | 33 | 36 | 43 | 46 |
| 2.5 | 33 | 43 | 46 | 55 | 60 | 35 | 45 | 48 | 57 | 62 |
| 4 | 47 | 60 | 64 | 77 | 84 | 49 | 64 | 68 | 81 | 88 |
| 5 | 58 | 75 | 80 | 95 | 104 | 60 | 77 | 83 | 98 | 107 |
| 6 | 71 | 92 | 98 | 117 | 127 | 74 | 96 | 102 | 122 | 133 |
| 8 | 88 | 113 | 121 | 144 | 157 | 92 | 119 | 127 | 152 | 165 |
| 10 | 107 | 138 | 147 | 176 | 191 | 113 | 146 | 156 | 186 | 202 |
| 12 | 132 | 170 | 182 | 217 | 236 | 140 | 181 | 193 | 230 | 251 |
| 16 | 165 | 214 | 228 | 272 | 296 | 170 | 220 | 235 | 281 | 306 |
| 20 | 204 | 264 | 282 | 336 | 365 | 209 | 271 | 289 | 344 | 375 |
| 25 | 255 | 330 | 352 | 420 | 457 | 261 | 338 | 361 | 430 | 468 |
| 32 | 321 | 416 | 444 | 529 | 576 | 324 | 420 | 448 | 534 | 582 |
| 40 | 398 | 516 | 551 | 656 | 715 | 402 | 520 | 555 | 662 | 721 |
| 50 | 505 | 654 | 698 | 832 | 907 | 508 | 657 | 702 | 837 | 911 |
| 63 | 635 | 821 | 877 | 1 045 | 1 139 | 631 | 817 | 872 | 1 039 | 1 132 |
| 80 | 801 | 1 037 | 1 108 | 1 320 | 1 438 | 792 | 1 025 | 1 095 | 1 305 | 1 421 |
| 100 | 1 016 | 1 315 | 1 404 | 1 673 | 1 822 | 1 002 | 1 297 | 1 385 | 1 650 | 1 798 |
| 125 | 1 288 | 1 666 | 1 780 | 2 121 | 2 310 | 1 254 | 1 623 | 1 734 | 2 066 | 2 250 |
| 160 | 1 609 | 2 082 | 2 224 | 2 650 | 2 887 | 1 585 | 2 052 | 2 192 | 2 611 | 2 844 |
| 200 | 2 030 | 2 627 | 2 806 | 3 344 | 3 642 | 2 011 | 2 603 | 2 780 | 3 313 | 3 609 |
| 250 | 2 539 | 3 286 | 3 510 | 4 182 | 4 556 | 2 513 | 3 252 | 3 474 | 4 139 | 4 509 |

D.2 Fatigue limit design forces of hook bodies for a series of hooks of type B, with additional materials (see [Table D.2](#))

Table D.2 — Fatigue limit design forces $F_{Rd,f}$ in kilonewtons (kN) — Factors f_2 , γ_{Hf} and γ_{SH} included and temperature influence factor $f_1 = 1$ ($T \leq 100$ °C)

| Hook no. | All materials Section B hook factor M_{hf} mm ² | Single hooks, type B Classified material grades $F_{Rd,f}$ | | | | | Additional materials | |
|---------------|--|--|-------|-------|-------|-------|----------------------------|------------|
| | | M | P | S | T | V | f_u N/mm ² | $F_{Rd,f}$ |
| B 0.8 | 25,392 | 3,0 | 3,9 | 4,1 | 4,9 | 5,4 | 620 | 4,6 |
| B 1.6 | 49,722 | 5,9 | 7,61 | 8,1 | 9,7 | 10,5 | | 8,93 |
| B 2.5 | 81,543 | 9,5 | 12,2 | 13,1 | 15,6 | 17,0 | | 14,4 |
| B 4 | 127,31 | 14,2 | 18,4 | 19,7 | 23,4 | 25,5 | | 21,6 |
| B.5 | 157,79 | 17,3 | 22,4 | 23,9 | 28,5 | 31,1 | | 26,3 |
| B 8 | 258,77 | 27,3 | 35,3 | 37,7 | 44,9 | 48,9 | | 41,4 |
| B 12.5 | 399,34 | 40,6 | 52,5 | 56,10 | 66,8 | 72,8 | | 61,6 |
| B 16 | 514,76 | 53,6 | 69,3 | 74,1 | 88,3 | 96,1 | 625 | 81,8 |
| B 20 | 649,89 | 66,3 | 85,8 | 91,6 | 109,2 | 119,0 | | 101,2 |
| B 25 | 807,66 | 80,9 | 104,7 | 111,9 | 133,3 | 145,2 | | 123,5 |
| B 32 | 948,18 | 103,2 | 133,5 | 142,6 | 169,5 | 185,1 | | 157,5 |
| B 40 | 1128,78 | 127,4 | 164,8 | 176,1 | 209,8 | 228,5 | | 194,4 |
| B 50 | 1379,07 | 161,5 | 209,0 | 223,3 | 266,0 | 289,8 | | 246,5 |
| B 63 | 1587,16 | 194,6 | 258,9 | 269,0 | 320,5 | 349,2 | | 297,1 |

The section B type modulus is as defined below [Table C.2](#). Its values for hook sizes B 0.8 to B 63 are shown in [Table D.2](#). For hook sizes greater than B 63, its value can be calculated from the dimensions of the hook and of its section B.

Annex E (normative)

Hook body calculation and specific spectrum ratio factors

E.1 Conversion factor for hook body, k_c

When classified duty is utilized, k_c shall be calculated in accordance with [Table E.1](#).

Table E.1 — Conversion factor $k_c = k_6^* / \sqrt[6]{s_Q}$

| | Class Q | Q ₀ | Q ₁ | Q ₂ | Q ₃ | Q ₄ | Q ₅ |
|----------------|----------------|----------------|----------------|----------------|----------------|----------------|----------------|
| | $k_Q(3)$ | 0,031 3 | 0,062 5 | 0,125 | 0,25 | 0,5 | 1,0 |
| | Factor k_6^* | 1,348 | 1,343 | 1,259 | 1,172 | 1,084 | 1 |
| Class U | C [cycles] | | | | | | |
| U ₀ | 16 000 | 5,37 | 4,77 | 3,98 | 3,30 | 2,72 | 2,24 |
| U ₁ | 31 500 | 4,8 | 4,26 | 3,56 | 2,95 | 2,43 | 2,00 |
| U ₂ | 63 000 | 4,27 | 3,79 | 3,17 | 2,63 | 2,17 | 1,78 |
| U ₃ | 125 000 | 3,81 | 3,38 | 2,83 | 2,34 | 1,93 | 1,59 |
| U ₄ | 250 000 | 3,40 | 3,01 | 2,52 | 2,09 | 1,72 | 1,41 |
| U ₅ | 500 000 | 3,02 | 2,69 | 2,24 | 1,86 | 1,53 | 1,26 |
| U ₆ | 1 000 000 | 2,70 | 2,39 | 2,00 | 1,66 | 1,37 | 1,12 |
| U ₇ | 2 000 000 | 2,40 | 2,13 | 1,78 | 1,48 | 1,22 | 1,00 |
| U ₈ | 4 000 000 | 2,14 | 1,90 | 1,58 | 1,32 | 1,08 | 0,89 |
| U ₉ | 8 000 000 | 1,91 | 1,69 | 1,41 | 1,17 | 0,97 | 0,79 |

E.2 Specific spectrum ratio factors

The presentation of distributions with discrete values can be shown as spectrum or as accumulated spectrum. [Figure E.1](#) illustrates both presentations for discrete distributions.



a) Spectrum, where n is the relative number of cycles with amplitude q : $\sum_{n=1}$

b) Accumulated spectrum

Figure E.1 — Discrete distributions

The presentation of distributions given by continuous functions can be shown as density function or as accumulated density function. [Figure E.2](#) illustrates both presentations for distributions given by continuous functions.

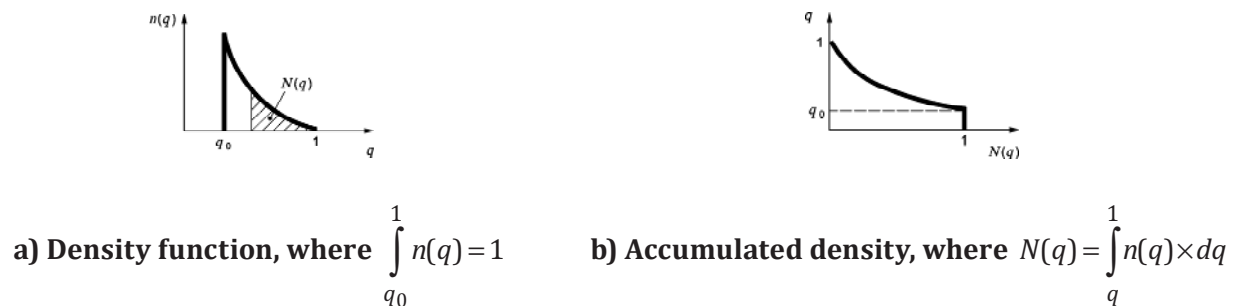


Figure E.2 — Continuous distributions

NOTE Whereas $n(q)$ gives the relative number of cycles with amplitude q , the accumulated value of $N(q)$ gives the number of cycles with amplitudes greater than q .

The stress spectrum factor k_m shall be calculated from the density function or from the accumulated density by

$$k_m = \int_{q_0}^1 q^m \times n \times dq = \int_0^1 q^m \times dN$$

The specific spectrum ratio factors (see [6.5.3](#)) then follows as

$$k_m^* = m \sqrt[m]{\frac{kQ}{k_m}} = m \sqrt[m]{\frac{k_3}{k_m}}$$

E.3 Underlying spectra for the specific spectrum ratio factors

In cases where the load spectrum of the crane is specified through the class Q only, the shape of load distribution in accordance with [Table E.3](#) shall be assumed. Consequently, the specific spectrum ratio factors as given in [Table E.2](#) can be derived and shall be used for calculation of the stress history parameter s_h for the hook.

Table E.2 — Specific spectrum ratio factors

| Class Q of ISO 4301-1 | Load spectrum factor kQ | Factor k^*_5 for $m = 5$ | Factor k^*_6 for $m = 6$ |
|-----------------------|------------------------------|----------------------------|----------------------------|
| Q ₀ | 0,031 3 | 1,292 | 1,348 |
| Q ₁ | 0,062 5 | 1,286 | 1,343 |
| Q ₂ | 0,125 | 1,217 | 1,259 |
| Q ₃ | 0,25 | 1,144 | 1,172 |
| Q ₄ | 0,5 | 1,070 | 1,084 |
| Q ₅ | 1 | 1 | 1 |

[Table E.3](#) gives the underlying density functions and accumulated density functions for [Table E.2](#).

Table E.3 — Underlying density and accumulated density functions


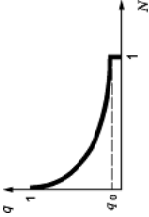
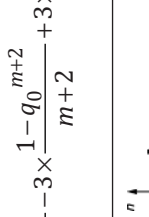
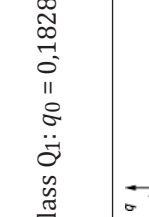
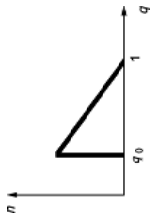

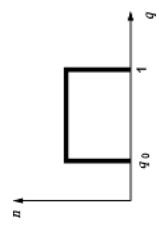
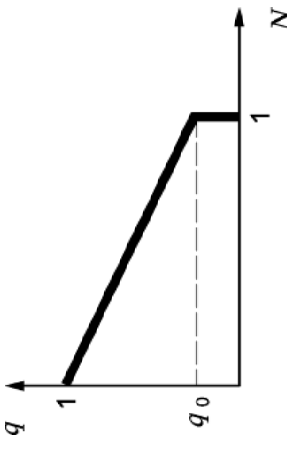
| Class | Density function | Accumulated density function |
|----------------------|--|--|
| <p>Q0, Q1</p> |  $n = 4 \times \frac{(1-q)^3}{(1-q_0)^4}$ |  $N(q) = \left(\frac{1-q}{1-q_0} \right)^4 \text{ or } q_{(N)} = 1 - (1-q_0) \times \sqrt[4]{N}$ |
| <p>Q2</p> |  $k_{(m)} = \frac{4}{(1-q_0)^4} \times \left[\frac{1-q_0^{m+1}}{m+1} \times \frac{1-q_0^{m+2}}{m+2} + 3 \times \frac{1-q_0^{m+3}}{m+3} - \frac{1-q_0^{m+4}}{m+4} \right]$ $n = 3 \times \frac{(1-q)^2}{(1-q_0)^3}$ | <p>for Class Q0: $q_0 = 0,0226$; for Class Q1: $q_0 = 0,1828$</p>  $N(q) = \left(\frac{1-q}{1-q_0} \right)^3 \text{ or } q_{(N)} = 1 - (1-q_0) \times \sqrt[3]{N}$ |
| | $k_{(m)} = \frac{3}{(1-q_0)^3} \times \left[\frac{1-q_0^{m+1}}{m+1} - 2 \times \frac{1-q_0^{m+2}}{m+2} + \frac{1-q_0^{m+3}}{m+3} \right]$ <p>for Class Q2: $q_0 = 0,2765$</p> | |

Table E.3 — (continued)

| | | |
|------------------|--|--|
| <p>Q3</p> |  $n = 2 \times \frac{1-q}{(1-q_0)^2}$ |  $N_{(q)} = \left(\frac{1-q}{1-q_0} \right)^2 \text{ or } q_{(N)} = 1 - (1-q_0) \times \sqrt{N}$ |
| <p>Q4</p> |  $n = \frac{1}{1-q_0}$ |  $N_{(q)} = \frac{1-q}{1-q_0} \text{ or } q_{(N)} = 1 - (1-q_0) \times N$ |
| | $k_{(m)} = \frac{2}{(1-q_0)^2} \times \left[\frac{1-q_0^{m+1}}{m+1} - \frac{1-q_0^{m+2}}{m+2} \right]$ <p>for Class Q3: $q_0 = 0,3943$</p> | |
| | $k_{(m)} = \frac{1}{1-q_0} \times \frac{1-q_0^{m+1}}{m+1}$ <p>for Class Q4: $q_0 = 0,5437$</p> | |

Annex F (informative)

Sample fatigue strength calculations of proofed hooks (with proof load applied)

F.1 Data

NOTE Calculations for single point hooks with dimensions in accordance with [A.3, Table A.3](#).

Hook size: Size B10, [Table A.3](#)

Nominal hook load:

$$F_H = 10 T_e = 98,07 \text{ kN}$$

NOTE T_e denotes metric tonnes, 1 000 kgf.

Hook dimensions and section properties:

$$a_1 = 92 \text{ mm} \quad \text{seat circle diameter}$$

Section B-B:

$$\eta_1 = 36,18 \text{ mm} \quad \text{coordinate } y \text{ from centroid to the intrados, see [Annex H](#)}$$

$$R = \frac{a_1}{2} + \eta_1 = 82,18 \text{ mm} \quad \text{radius to centroid of section}$$

$$I = 1,696 \times 10^6 \text{ mm}^4 \quad \text{reference moment of inertia of cross section, see [Annex H](#)}$$

$$v = 1 \quad \text{factor for load component, section B-B see [5.5.3](#)}$$

Hook material: 150 M 19Q

$$f_y = 430 \text{ N mm}^{-2} \quad \text{min. yield stress of hook material, for RS < 150}$$

$$f_u = 620 \text{ N mm}^{-2} \quad \text{min. ultimate strength of hook material}$$

Fatigue and service duty parameters:

Utilization class U7 (to ISO 4301-1:1986, Table 1) hence $N = 2 \times 10^6$, total number of load cycles

Load spectrum class Q4 (to ISO 4301-1:1986, Table 2) hence $kQ = 1$, load spectrum factor, max.

All load cycles are assumed to be the same and at full load.

F.2 Proof of competence in fatigue for hooks which have had proof load applied

F.2.1 Stresses in the bowl of “Non-proofed” hook (no residual stress present):

$\phi = 1,3$ dynamic factor

$$F_{Sd,f} = \phi_2 F_H = 127,49 \text{ kN} \quad \text{vertical design force for fatigue, Formula (18)}$$

Moment on hook section due to vert. design force $F_{Sd,f}$:

$$M_{Sd,f} = F_{Sd,f}R = 10,48 \text{ kN m}$$

Maximum design stress at the intrados of the hook due to vertical design force, Formula (11):

$$\sigma_{Sd,f} = \frac{v \times M_{Sd,f}}{I} \times \frac{R \times \eta_1}{R - \eta_1} = 399,28 \text{ N mm}^{-2}$$

Stress range at the intrados of the hook for fatigue (due to load $F_{Sd,f}$, cycling from 0 to maximum):

$$\Delta\sigma_{Sd,f} = \sigma_{Sd,f} = 399,28 \text{ N mm}^{-2}$$

With loads of one magnitude only assumed, viz. $F_{Sd,f}$, max. stress range at the intrados of the hook for fatigue:

$$\Delta\sigma_{Sd,max} = \Delta\sigma_{Sd,f} = 399,28 \text{ N mm}^{-2}$$

F.2.2 Stresses in the bowl of “proofed” hook (residual stress present)

F.2.2.1 Proof load for the hook

Proof load F_{PL} for the hook is calculated from

$$F_{PL} = 1,5 \times f_y \times M \quad [\text{see 4.5 e)]}$$

where

$$M = I \times \left[\frac{1 - \frac{\eta_1}{R}}{R \times \eta_1} \right] = 319,29 \text{ mm}^2$$

and proof load $F_{PL} = 1,5 \times f_y \times M = 205,94 \text{ kN}$

F.2.2.2 Stresses in the section of “proofed” hook (residual stress present)

Moment on hook section due to proof load force F_{PL} : $M_{PL} = F_{PL} \times R = 16,92 \text{ kN m}$

Maximum virtual design stress at the intrados of the hook with proof load applied, Formula (11):

$$\sigma_{p,max,v} = \frac{v \times M_{PL}}{I} \times \frac{R \times \eta_1}{R - \eta_1} = 645 \text{ N mm}^{-2}$$

NOTE Maximum stress at the intrados of the hook with proof load applied and on the assumption of no plastic deformation.

An “ideal elastic-plastic” stress model for the material behaviour is assumed, with the onset of plastic deformation occurring at the yield stress and the maximum stress remaining constant at the yield stress level. Hence max. stress at proof load:

$$\sigma_{p,\max} = f_y = 420 \text{ N mm}^{-2}$$

Stress change during unloading when the proof load is removed, is assumed to take place along the elastic line. Hence residual stress after full unloading will be

$$\sigma_{r,p,\max} = \sigma_{p,\max} - \sigma_{p,\max,v} = -225 \text{ N mm}^{-2}$$

Maximum stress range due to full service load $F_{Sd,f} = 127,49 \text{ kN}$ applied is

$$\Delta\sigma_{Sd,\max} = 399,28 \text{ N mm}^{-2}$$

giving a rise to a maximum stress at the intrados due to full service load

$$\sigma_{sd,p,\max} = \sigma_{r,p,\max} + \Delta\sigma_{Sd,\max} = 174,28 \text{ N mm}^{-2}$$

and to a stress cycling between the extreme values of

$$\sigma_{r,p,\max} = -225 \text{ N mm}^{-2} \quad \text{and} \quad \sigma_{sd,p,\max} = 174,28 \text{ N mm}^{-2}$$

The stress cycle must be transformed to an equivalent pulsating cycle (0 to max. stress), done below using the standard equation derived from the Haigh/Smith diagram:

$$\sigma_{a,p,t} = \frac{\sigma_{a,p} - \mu \times \sigma_{m,p}}{1 - \mu \times \frac{(1+R_s)}{(1-R_s)}} \quad \sigma_{m,p,t} = \sigma_{a,p,t}$$

where

$$\sigma_{a,p,t}$$

are the amplitude and mean stress components of transformed stress cycle;

$$\sigma_{m,p,t}$$

R_s stress ratio of the transformed cycle, $\sigma_{\min}/\sigma_{\max}$:

$R_s = 0$, for pulsating cycle 0 to max.;

$\mu = -0,1765$ from Formula (44), 6.6.7 (note the negative sign).

Component of the stress cycle:

$$\sigma_{m,p} = \frac{\sigma_{sd,p,\max} + \sigma_{r,p,\max}}{2} = -25,36 \text{ N mm}^{-2} \quad \text{the mean stress component}$$

$$\sigma_{a,p} = \frac{\sigma_{sd,p,\max} - \sigma_{r,p,\max}}{2} = 199,64 \text{ N mm}^{-2} \quad \text{the amplitude stress component}$$

Components of the transformed cycle:

From the equations above, amplitude and mean stress components of the transformed cycle

$$\sigma_{a,p,t} = \frac{\sigma_{a,p} - \mu \times \sigma_{m,p}}{1 - \mu \times \frac{(1+R_s)}{(1-R_s)}} = 165,89 \text{ N mm}^{-2} \quad \text{the amplitude stress component}$$

Maximum stress range at intrados:

$$\Delta\sigma_{Sd,p,f,max} = 2 \times \sigma_{a,p,t} = 331,77 \text{ N mm}^{-2} \quad \text{the maximum stress range at the intrados of the "proofed"}$$

$$\sigma_{m,p,t} = \sigma_{a,p,t} = 165,89 \text{ N mm}^{-2} \quad \text{the mean stress component for pulsating cycle}$$

Stress cycling between 0 and maximum of $2 \times$ amplitude stress component

Hook for fatigue, due to $F_{Sd,f}$ load cycling from 0 to maximum.

F.2.3 Execution of proof for fatigue strength

F.2.3.1 Limit design stress, load history parameter and stress history parameter

$$\Delta\sigma_{R,d} = f_1 \times f_2 \times \Delta\sigma_c$$

the limit fatigue design stress, Formula (29), [6.5.4](#).

Take $f_1 = 1$, the temperature factor, Formula (31), [6.5.4](#), and with $b_{max} = 55$ mm maximum width of hook cross section:

$$f_2 = \left[\frac{25 \text{ mm}}{b_{max}} \right]^{0,167} = 0,877$$

the material thickness influence factor, Formula (32), [6.5.4](#).

Characteristic fatigue strength, Formula (30), [6.5.4](#):

$$\Delta\sigma_c = 0,315 \times f_u \times \lg \left[\frac{13000}{f_u} \right] = 258,11 \text{ N mm}^{-2}$$

and the limit fatigue design stress range, Formula (29), [6.5.4](#):

$$\Delta\sigma_{R,d} = f_1 \times f_2 \times \Delta\sigma_c = 226,27 \text{ N mm}^{-2}$$

$$s_Q = k_Q \times \frac{N}{N_D}$$

the load history parameter, Formula (26), 6.5.3, with $N_D = 2 \times 10^6$ reference number of cycles

$$N_D = 2 \times 10^6 \quad s_Q = k_Q \times \frac{N}{N_D} = 1$$

$$k_6^* = 1$$

the specific spectrum ratio factor, Formula (28), 5.3, and Table 5 for standardized classification

$$s_h = \frac{s_Q}{\left(k_6^*\right)^m}$$

the stress history factor, with $m = 6$ for hook bowl/material, Formula (27), 6.5.2, 6.5.3

$$s_h = \frac{s_Q}{\left(k_6^*\right)^m} = 1$$

F.2.3.2 Proof of competence for the fatigue strength of “non-proofed” hook

Inequality to be satisfied [6.5.5, Formula (33)]:

$$\Delta\sigma_{Sd,max} \leq \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_h}} \quad \text{or} \quad \Delta\sigma_{Sd,max} \leq \frac{k_6^* \times \Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_Q}}$$

$$\gamma_{Hf} = 1,25$$

the fatigue strength resistance specific factor, see Table 7.

Hence:

$$\Delta\sigma_{Sd,max} = 399,28 \text{ N mm}^{-2} \quad \text{vs.} \quad \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_h}} = 181,01 \text{ N mm}^{-2}$$

i.e. the inequality and hence also proof of fatigue strength for the load stated are not satisfied; either the load applied or life, in number of cycles, has to be reduced (from F_H or N of 2×10^6 cycles, resp.).

Reduced hook load:

$$F_{H,red} = F_H \times \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_h}} \times \frac{1}{\Delta\sigma_{Sd,max}} = 44,46 \text{ kN}$$

Reduced life/number of cycles, can be calculated using Formulae (26) and (27).

F.2.3.3 Proof of competence for the fatigue strength of “proofed” hook

Inequality to be satisfied [6.5.5, Formula (33)]:

$$\Delta\sigma_{Sd,p,max} \leq \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_h}} \quad \text{or} \quad \Delta\sigma_{Sd,p,max} \leq \frac{k_6^* \times \Delta\sigma_{Rd}}{\gamma_{Hf} \times m \sqrt{s_Q}}$$

NOTE That there is no change on the R.H.S. of the inequality expression.

The limit fatigue design stress range:

$$\Delta\sigma_{Rd} = 226,27 \text{ N mm}^{-2}$$

load history parameter, Formula (26), 6.5.3, with $N_D = 2 \times 10^6$ reference number of cycles, $s_Q = 1$;

$k_6^* = 1$ the specific spectrum ratio factor, Formula (28), 6.5.3, and Table 5 for standardized classification;

$s_h = 1$ the stress history factor, with $m = 6$ for hook/bowl material, Formula (27), 6.5.2 and 6.5.3;

$N_D = 2\,000\,000$ $m = 6$;

$\gamma_{Hf} = 1,25$ the fatigue strength resistance specific factor, see [Table 7](#).

Hence:

$$\Delta\sigma_{Sd,p,f,max} = 331,77 \text{ N mm}^{-2} \text{ vs. } \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times \sqrt[m]{s_h}} = 181,01 \text{ N mm}^{-2}$$

i.e. the inequality and hence also the proof of competence for fatigue for the load stated has not been satisfied; either the load applied or the life required, in number of cycles has to be reduced (from F_H or N of 2×10^6 cycles, resp.).

Reduced hook load:

$$F_{H,p,red} = F_H \times \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \sqrt[m]{s_h}} \times \frac{1}{\Delta\sigma_{Sd,p,f,max}} = 53,5 \text{ kN}$$

(compared to 44,5 kN for un-proofed hook)

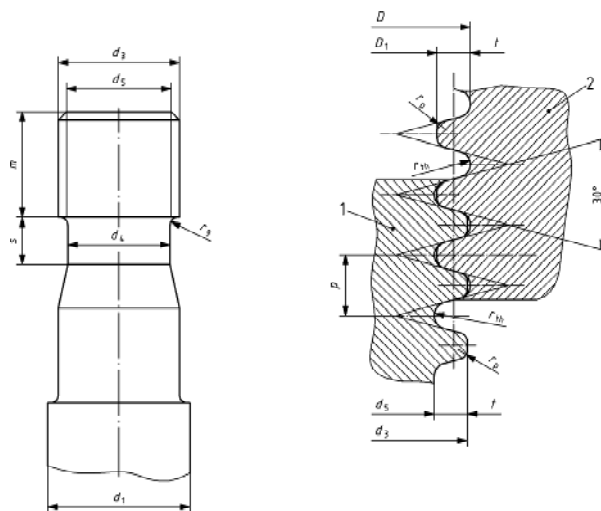
Alternatively, reduced life/ number of cycles to satisfy the inequality above can be calculated, using Formulae (26) and (27).

Annex G (informative)

Sample set of hook shank and thread designs

G.1 A series of hook shank and thread designs, a knuckle thread

See [Figure G.1](#) and [Table G.1](#).



Key

- 1 hook shank
- 2 nut

Figure G.1 — Symbols of dimensions for hook shank and thread

Table G.1 — Dimensions of hook shank and thread in millimetres (mm)

| (d_1) | Thread designation | d_3 | p | d_4 | d_5 | r_9 | s | r_{th} | r_p | t | m | (D) | (D_1) |
|-----------|--------------------|-------|-----|-------|-------|-------|-----|----------|-------|-----|-----|---------|-----------|
| 60 | Rd 50 × 6 | 50 | 6 | 42 | 43,4 | 4 | 20 | 1,33 | 0,92 | 3,3 | 45 | 50,6 | 44 |
| 67 | Rd 56 × 6 | 56 | 6 | 48 | 49,4 | 4 | 20 | 1,33 | 0,92 | 3,3 | 50 | 56,6 | 50 |
| 75 | Rd 64 × 8 | 64 | 8 | 54 | 55,2 | 4 | 25 | 1,77 | 1,23 | 4,4 | 56 | 64,8 | 56 |
| 85 | Rd 72 × 8 | 72 | 8 | 62 | 63,2 | 4 | 25 | 1,77 | 1,23 | 4,4 | 63 | 72,8 | 64 |
| 95 | Rd 80 × 10 | 80 | 10 | 68 | 69,0 | 6 | 30 | 2,21 | 1,54 | 5,5 | 71 | 81 | 70 |
| 106 | Rd 90 × 10 | 90 | 10 | 78 | 79,0 | 6 | 30 | 2,21 | 1,54 | 5,5 | 80 | 91 | 80 |
| 118 | Rd 100 × 12 | 100 | 12 | 85 | 86,8 | 6 | 40 | 2,65 | 1,84 | 6,6 | 90 | 101,2 | 88 |
| 132 | Rd 110 × 12 | 110 | 12 | 95 | 96,8 | 6 | 40 | 2,65 | 1,84 | 6,6 | 100 | 111,2 | 98 |
| 150 | Rd 125 × 14 | 125 | 14 | 108 | 109,6 | 8 | 45 | 3,10 | 2,15 | 7,7 | 112 | 126,4 | 111 |
| 170 | Rd 140 × 16 | 140 | 16 | 120 | 122,4 | 10 | 50 | 3,54 | 2,46 | 8,8 | 125 | 141,6 | 124 |
| 190 | Rd 160 × 18 | 160 | 18 | 138 | 140,2 | 10 | 55 | 3,98 | 2,77 | 9,9 | 140 | 161,8 | 142 |

The nut dimensions (D and D_1) and the forged shank diameter (d_1) are for guidance only. The shank machining and thread may be applied for any forged sizes and types within the requirements of this standard.

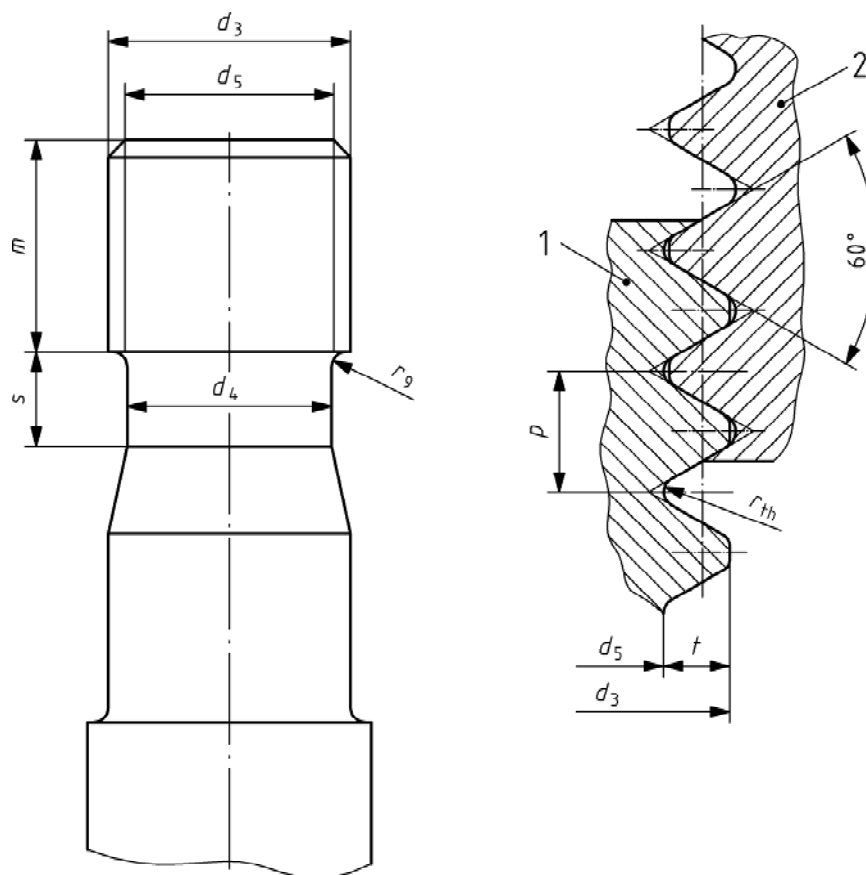
Table G.1 (continued)

| (d_1) | Thread designation | d_3 | p | d_4 | d_5 | r_9 | s | r_{th} | r_p | t | m | (D) | (D_1) |
|---------|--------------------|-------|-----|-------|-------|-------|-----|----------|-------|------|-----|-------|---------|
| 212 | Rd 180 × 20 | 180 | 20 | 156 | 158,0 | 12 | 60 | 4,42 | 3,07 | 11,0 | 160 | 182 | 160 |
| 236 | Rd 200 × 22 | 200 | 22 | 173 | 175,8 | 12 | 70 | 4,86 | 3,38 | 12,1 | 180 | 202,2 | 178 |
| 265 | Rd 225 × 24 | 225 | 24 | 196 | 198,6 | 12 | 80 | 5,31 | 3,69 | 13,2 | 200 | 227,4 | 201 |
| 300 | Rd 250 × 28 | 250 | 28 | 217 | 219,2 | 15 | 90 | 6,19 | 4,30 | 15,4 | 225 | 252,8 | 222 |
| 335 | Rd 280 × 32 | 280 | 32 | 242 | 244,8 | 18 | 100 | 7,07 | 4,92 | 17,6 | 250 | 283,2 | 248 |
| 375 | Rd 320 × 36 | 320 | 36 | 278 | 280,4 | 20 | 110 | 7,96 | 5,53 | 19,8 | 280 | 323,6 | 284 |

The nut dimensions (D and D_1) and the forged shank diameter (d_1) are for guidance only. The shank machining and thread may be applied for any forged sizes and types within the requirements of this standard.

G.2 A series of hook shank and thread designs, a metric thread

See [Figure G.2](#) and [Table G.2](#).



Key

- 1 hook shank
- 2 nut

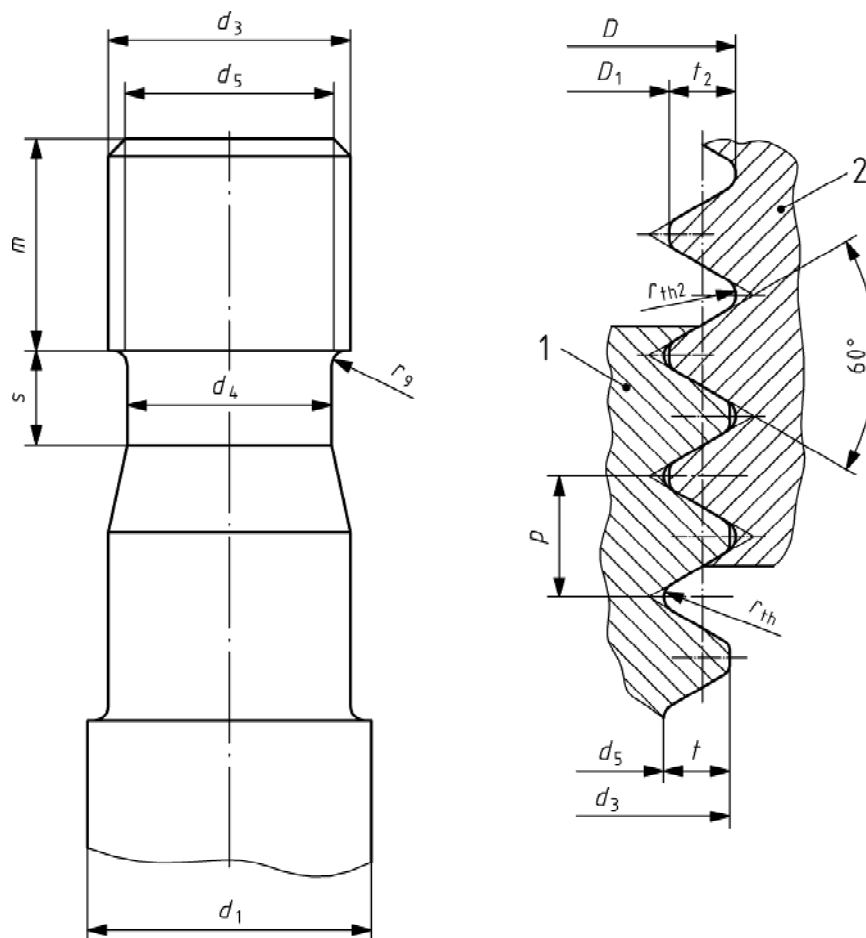
Figure G.2 — Symbols of dimensions for a hook shank and thread

Table G.2 — Dimensions of hook shank and thread in millimetres (mm)

| Thread designation | d_3 | p | d_4 | d_5 | r_9 | s | r_{th} | t | m |
|--------------------|-------|------|-------|--------|-------|-----|----------|------|-----|
| M 10 × 1,5 | 10 | 1,5 | 7,5 | 8,16 | 1,0 | 5 | 0,21 | 0,92 | 9 |
| M 12 × 1,75 | 12 | 1,75 | 9,0 | 9,85 | 1,2 | 6 | 0,25 | 1,07 | 11 |
| M 14 × 2 | 14 | 2 | 11,2 | 11,55 | 1,4 | 6 | 0,28 | 1,23 | 14 |
| M 16 × 2 | 16 | 2 | 12,5 | 13,55 | 1,2 | 7 | 0,28 | 1,23 | 15 |
| M 20 × 2,5 | 20 | 2,5 | 16,6 | 16,93 | 1,6 | 8 | 0,35 | 1,53 | 18 |
| M 24 × 3 | 24 | 3 | 20,0 | 20,32 | 2,0 | 9 | 0,42 | 1,84 | 22 |
| M 30 × 3,5 | 30 | 3,5 | 25,4 | 25,71 | 2,0 | 11 | 0,49 | 2,15 | 27 |
| M 33 × 3,5 | 33 | 3,5 | 28,4 | 28,71 | 3,3 | 11 | 0,49 | 2,15 | 33 |
| M 36 × 4 | 36 | 4 | 30,0 | 31,09 | 2,0 | 12 | 0,56 | 2,45 | 32 |
| M 39 × 4 | 39 | 4 | 33,8 | 34,09 | 3,9 | 12 | 0,56 | 2,45 | 39 |
| M 42 × 4,5 | 42 | 4,5 | 36,2 | 36,48 | 3,0 | 13 | 0,63 | 2,76 | 36 |
| M 45 × 4,5 | 45 | 4,5 | 38,5 | 39,48 | 3,0 | 13 | 0,63 | 2,76 | 40 |
| M 48 × 5 | 48 | 5 | 41,6 | 41,87 | 4,8 | 15 | 0,70 | 3,07 | 48 |
| M 52 × 5 | 52 | 5 | 45,6 | 45,87 | 5,2 | 15 | 0,70 | 3,07 | 52 |
| M 60 × 5,5 | 60 | 5,5 | 53,0 | 53,25 | 6,0 | 17 | 0,77 | 3,37 | 60 |
| M 68 × 6 | 68 | 6 | 60,3 | 60,64 | 6,8 | 18 | 0,84 | 3,68 | 68 |
| M 72 × 6 | 72 | 6 | 64,3 | 64,64 | 7,2 | 18 | 0,84 | 3,68 | 72 |
| M 80 × 6 | 80 | 6 | 72,3 | 72,64 | 8,0 | 18 | 0,84 | 3,68 | 80 |
| M 90 × 6 | 90 | 6 | 82,3 | 82,64 | 9,0 | 18 | 0,84 | 3,68 | 90 |
| M 100 × 8 | 100 | 8 | 89,9 | 90,19 | 10,0 | 24 | 1,12 | 4,91 | 100 |
| M 110 × 8 | 110 | 8 | 99,9 | 100,19 | 11,0 | 24 | 1,12 | 4,91 | 110 |

G.3 A series of hook shank and thread designs, a modified metric thread

See [Figure G.3](#) and [Table G.3](#).



Key

- 1 hook shank
- 2 nut

Figure G.3 — Symbols of dimensions for a hook shank and thread

Table G.3 — Dimensions of hook shank and thread in millimetres (mm)

| (d_1) | Thread designation | d_3 | p | d_4 | D_5 | r_9 | s | r_{th} | t | m | (D) | (D_1) | (r_{th2}) | (t_2) |
|-----------|--------------------|-------|-----|-------|--------|-------|-------|----------|------|-----|---------|-----------|---------------|-----------|
| 60 | ATS 52 × 5 | 52 | 5 | 46 | 46,587 | 10 | 15,41 | 1,083 | 2,71 | 43 | 52,361 | 47,670 | 0,361 | 2,35 |
| 67 | ATS 56 × 5,5 | 56 | 5,5 | 50 | 50,046 | 11 | 18,41 | 1,191 | 2,98 | 48 | 56,396 | 51,237 | 0,397 | 2,58 |
| 75 | ATS 64 × 6 | 64 | 6 | 57 | 57,505 | 12 | 21,48 | 1,299 | 3,25 | 54 | 64,433 | 58,804 | 0,433 | 2,81 |
| 85 | ATS 72 × 6 | 72 | 6 | 65 | 65,505 | 14 | 26,48 | 1,299 | 3,25 | 61 | 72,430 | 66,804 | 0,430 | 2,81 |
| 95 | ATS 80 × 7 | 80 | 7 | 72 | 72,422 | 16 | 29,55 | 1,516 | 3,79 | 67 | 80,505 | 73,938 | 0,505 | 3,28 |
| 106 | ATS 90 × 7 | 90 | 7 | 82 | 82,422 | 19 | 34,55 | 1,516 | 3,79 | 77 | 90,505 | 83,938 | 0,505 | 3,28 |
| 118 | ATS 100 × 8 | 100 | 8 | 91 | 91,340 | 20 | 37,61 | 1,732 | 4,33 | 85 | 100,577 | 93,072 | 0,577 | 3,75 |

The nut dimensions and the forged shank diameter (d_1) are for guidance only. The shank machining and thread may be applied for any forged sizes and types within the requirements of this standard.

Table G.3 (continued)

| (d_1) | Thread designation | d_3 | p | d_4 | D_5 | r_9 | s | r_{th} | t | m | (D) | (D_1) | (r_{th2}) | (t_2) |
|-----------|--------------------|-------|-----|-------|---------|-------|--------|----------|------|-----|---------|-----------|---------------|-----------|
| 132 | ATS 110 × 8 | 110 | 8 | 101 | 101,340 | 22 | 42,61 | 1,732 | 4,33 | 96 | 110,577 | 103,072 | 0,577 | 3,75 |
| 150 | ATS 125 × 10 | 125 | 10 | 113 | 114,175 | 25 | 46,82 | 2,165 | 5,41 | 106 | 125,722 | 116,340 | 0,722 | 4,69 |
| 170 | ATS 140 × 10 | 140 | 10 | 128 | 129,175 | 28 | 56,82 | 2,165 | 5,41 | 120 | 140,722 | 131,340 | 0,722 | 4,69 |
| 190 | ATS 160 × 12 | 160 | 12 | 146 | 147,010 | 32 | 62,95 | 2,598 | 6,50 | 134 | 160,866 | 149,608 | 0,866 | 5,63 |
| 212 | ATS 180 × 12 | 180 | 12 | 166 | 167,010 | 36 | 72,95 | 2,598 | 6,50 | 154 | 180,866 | 169,608 | 0,866 | 5,63 |
| 236 | ATS 200 × 14 | 200 | 14 | 183 | 184,845 | 40 | 77,16 | 3,031 | 7,58 | 170 | 201,000 | 187,876 | 1,01 | 6,56 |
| 265 | ATS 225 × 14 | 225 | 14 | 208 | 209,845 | 45 | 92,16 | 3,031 | 7,58 | 192 | 226,010 | 212,876 | 1,01 | 6,57 |
| 300 | ATS 250 × 16 | 250 | 16 | 230 | 232,680 | 50 | 101,36 | 3,464 | 8,66 | 211 | 251,155 | 236,144 | 1,155 | 7,51 |
| 335 | ATS 280 × 16 | 280 | 16 | 260 | 262,680 | 56 | 121,36 | 3,464 | 8,66 | 240 | 281,155 | 266,144 | 1,155 | 7,51 |
| 375 | ATS 320 × 18 | 320 | 18 | 298 | 300,515 | 63 | 137,50 | 3,897 | 9,74 | 268 | 321,199 | 304,412 | 1,299 | 8,39 |

The nut dimensions and the forged shank diameter (d_1) are for guidance only. The shank machining and thread may be applied for any forged sizes and types within the requirements of this standard.

G.4 Hook shank and thread designs for hooks of type B

See [Figure G.4](#) and [Table G.4](#).

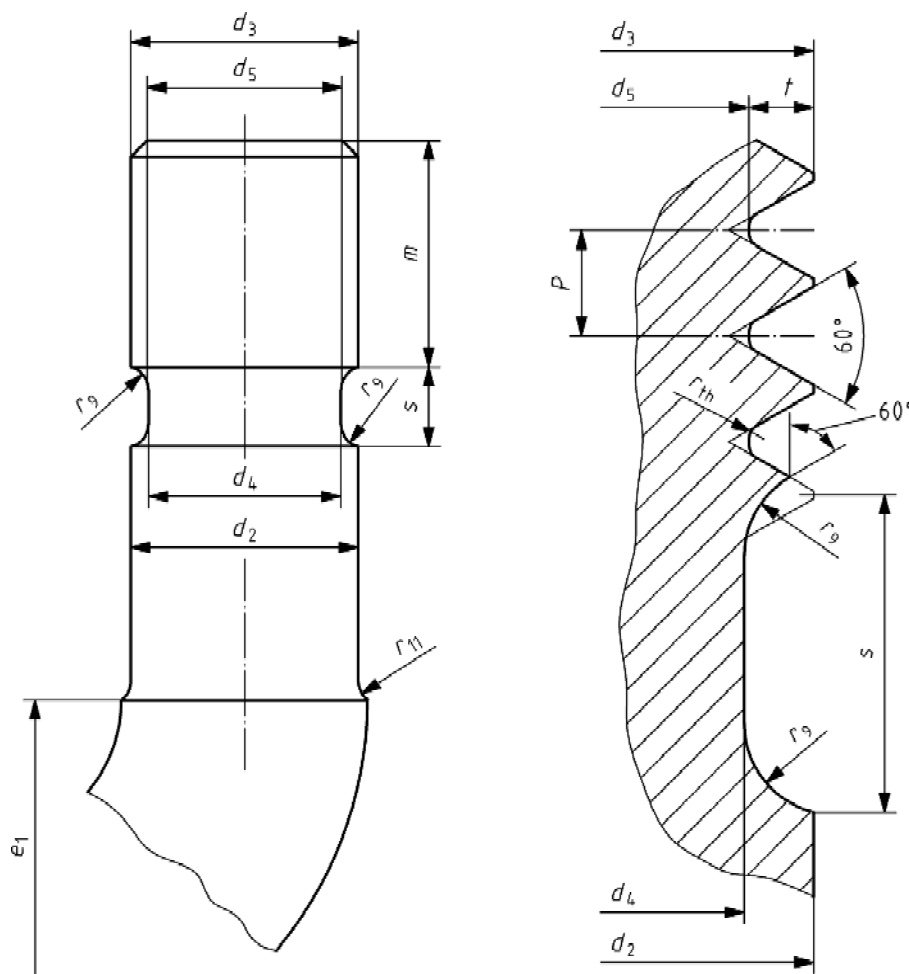


Figure G.4 — Symbols of dimensions for a hook shank and thread

NOTE Width of undercut, s , is measured from the shoulder of the shank to the crest of the last full thread.

Table G.4 — Dimensions of hook shank and thread in millimetres (mm)

| Hook no. | d_2 | Thread designation | d_3 | p | d_4 | d_5 | r_9 | s | r_{th} | t | r_{th11} | m |
|---------------|-------|--------------------|-------|-----|-------|--------|-------|------|----------|------|------------|-----|
| B 0.8 | 15 | M14 × 2 | 14 | 2 | 11,25 | 11,55 | 1,5 | 6 | 0,28 | 1,23 | 2 | 14 |
| B 1.6 | 20 | M20 × 2,5 | 20 | 2,5 | 16,63 | 16,93 | 2 | 7,5 | 0,35 | 1,53 | 2 | 20 |
| B 2.5 | 25 | M24 × 3 | 24 | 3 | 20,02 | 20,32 | 2,5 | 9 | 0,42 | 1,84 | 3 | 24 |
| B 4 | 30 | M30 × 3,5 | 30 | 3,5 | 25,41 | 25,71 | 3 | 10,5 | 0,49 | 2,15 | 3 | 30 |
| B 5 | 35 | M33 × 3,5 | 33 | 3,5 | 28,41 | 28,71 | 3,5 | 10,5 | 0,49 | 2,15 | 4 | 33 |
| B 6.3 | 40 | M39 × 4 | 39 | 4 | 33,79 | 34,09 | 4 | 12 | 0,56 | 2,45 | 4 | 39 |
| B 8 | 45 | M42 × 4,5 | 42 | 4,5 | 36,18 | 36,48 | 4,5 | 13,5 | 0,63 | 2,76 | 5 | 42 |
| B 10 | 50 | M48 × 5 | 48 | 5 | 41,57 | 41,87 | 5 | 15 | 0,7 | 3,07 | 5 | 48 |
| B 12.5 | 55 | M52 × 5 | 52 | 5 | 45,57 | 45,87 | 5,5 | 15 | 0,7 | 3,07 | 6 | 52 |
| B 16 | 60 | M60 × 5,5 | 60 | 5,5 | 52,95 | 53,25 | 6 | 16,5 | 0,77 | 3,37 | 6 | 60 |
| B 20 | 70 | M68 × 6 | 68 | 6 | 60,34 | 60,64 | 7 | 18 | 0,84 | 3,68 | 7 | 70 |
| B 25 | 80 | M72 × 6 | 72 | 6 | 64,34 | 64,64 | 8 | 18 | 0,84 | 3,68 | 8 | 72 |
| B 32 | 85 | M80 × 6 | 80 | 6 | 72,34 | 72,64 | 8,5 | 18 | 0,84 | 3,68 | 9 | 80 |
| B 40 | 100 | M 90 × 6 | 90 | 6 | 82,34 | 82,64 | 10 | 18 | 0,84 | 3,68 | 10 | 90 |
| B 50 | 110 | M100 × 8 | 100 | 8 | 89,89 | 90,19 | 11 | 24 | 1,12 | 4,91 | 11 | 100 |
| B 63 | 120 | M110 × 8 | 110 | 8 | 99,89 | 100,19 | 12 | 24 | 1,12 | 4,91 | 12 | 110 |

NOTE 1 For an explanation of dimension e_1 see [A.3](#).

NOTE 2 Width of undercut s is measured from the shoulder of the shank to the crest of the last full thread.

Annex H (normative)

Bending of curved beams

H.1 Basic equations for stresses

The stresses calculated below represent true primary tensile stresses in the cross section of a curved beam, derived from the theory of elasticity and with the assumption of no plastic behaviour of material.

The equations below are in a general format of equations for a curved beam and they apply to an analogous problem of stress distribution in a cross section of the curved part of the hook bowl body. The designations refer to [Figure H.1](#).

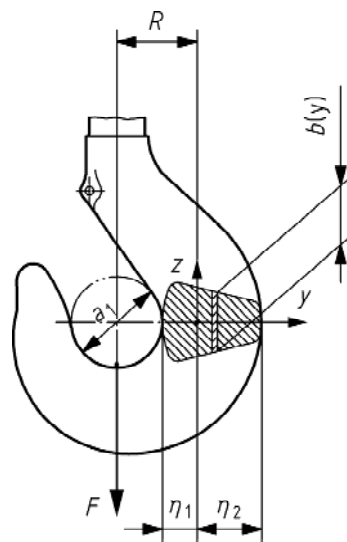


Figure H.1 — Symbols for curved beam bending calculation

For a curved beam with a solid section, a reference moment of inertia shall be calculated as follows:

$$I = \int_{-\eta_1}^{+\eta_2} \frac{y^2 \times b(y)}{1 + \frac{y}{R}} dy \quad (\text{H.1})$$

where

- y is the distance in radial direction, measured from the centroid of cross section; taken as positive between the centroid and the extrados of the section and negative between the intrados and centroid of the section;
- b is the width of the section at a location y ;
- R is the radius of curvature of centroid axis of the curved beam at the cross section under consideration;
- η_1 is the absolute value of the coordinate y at the inside radius;
- η_2 is the absolute value of the coordinate y at the outside radius.

The maximum tensile stress σ in the cross section shown is at the intrados and shall be calculated as follows:

$$\sigma = \frac{F \times R \times \eta_1}{I} \times \frac{1}{1 - \eta_1/R} \quad (\text{H.2})$$

where

- F is the force acting perpendicularly to the plane of the section and through the centre of the curvature;
- R is the radius of curvature of centroid axis of the hook bowl at the cross section under consideration;
- η_1 is the absolute value of the coordinate y at the intrados;
- I is the a reference moment of inertia of the section in accordance with Equation (H.1).

NOTE Equation (H.2) contains the combined effect of direct tension and bending moment. The equation is only valid in the case where the force causing the bending acts through the centre of the beam curvature. Under such a loading configuration the neutral axis coincides with the centroid of the section.

H.2 Approximation of the reference moment of inertia

The reference moment of inertia of a curved beam shall be expressed and calculated as follows:

$$I = k \times I_z = k \times \int_{-\eta_1}^{+\eta_2} y^2 \times b(y) \times dy \quad (\text{H.3})$$

where

- I_z is the conventional moment of inertia of the section about the z-axis;
- k is a conversion factor depending of the shape of the section and the relative curvature, see [Figure H.2](#).

with $0,45 < b_{\min}/b_{\max} < 0,55$, where b_{\max} is the larger, inner edge width of the section and b_{\min} is the smaller, outer edge width of the section.

For cross sections of hooks of type B, k shall be taken as $k = 1,05$ both on Section A and B in [Figure 5](#).

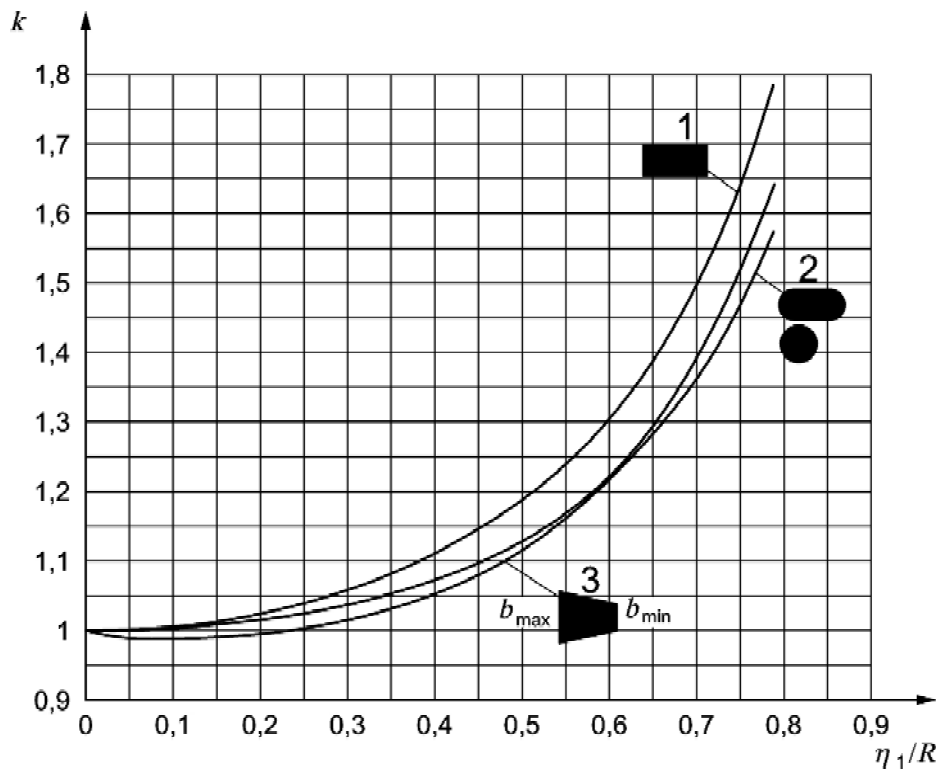


Figure H.2 — Factor k for a selection of section types

Key

- 1 rectangular section
- 2 circular and rounded sections
- 3 trapezoidal section

Annex I (normative)

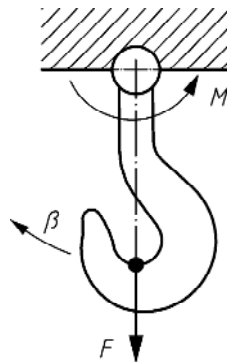
Calculation of hook suspension tilting resistance, articulation by a hinge or rope reeving system

I.1 General

The hook tilting resistance factor C_t represents the friction in the hook suspension resisting the tilting of the hook. It is a characteristic of the suspension only and its value is independent upon external loadings. The factor C_t is defined through Equation (I.1) as follows:

$$C_t = \frac{M}{F} \quad (\text{I.1})$$

where



C_t is the hook tilting resistance factor;

M is the moment resisting the tilting movement of the hook, see [Figure I.1](#);

F is the vertical force acting on the hook.

Key

β direction of the tilting movement

Figure I.1 — General presentation of hook tilting resistance

I.2 Articulation of hook by a hinge

See [Figure I.2](#).

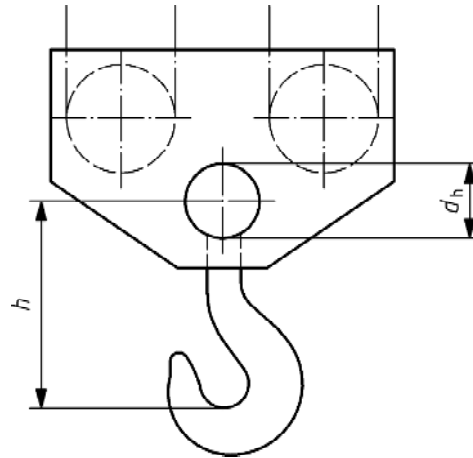


Figure I.2 — A hook suspension with a hook articulation by a hinge

Rope reeving system is such that the hook suspension stays horizontal, when the hook is tilted along the load line. Tilting resistance factor is calculated as follows:

$$C_t = \frac{M}{F} = \mu \times \frac{d_h}{2} \quad (I.2)$$

where

d_h is the diameter of the sliding surface of the hinge;

μ is the friction in the hinge, to be taken as follows:

$\mu = 0,1$ for non-grease able, special coated bearing bushings;

$\mu = 0,25$ for re-grease able, bronze/steel bearing bushings;

$\mu = 0,4$ for non-grease able steel/steel hinges;

$\mu = 0$ (zero) for hinges with anti-friction bearings.

As the tilting resistance is significant for the fatigue of the hook shank, the friction figures represent long term values to be expected in practice. This shall be taken into consideration, if the friction values should be based upon tests.

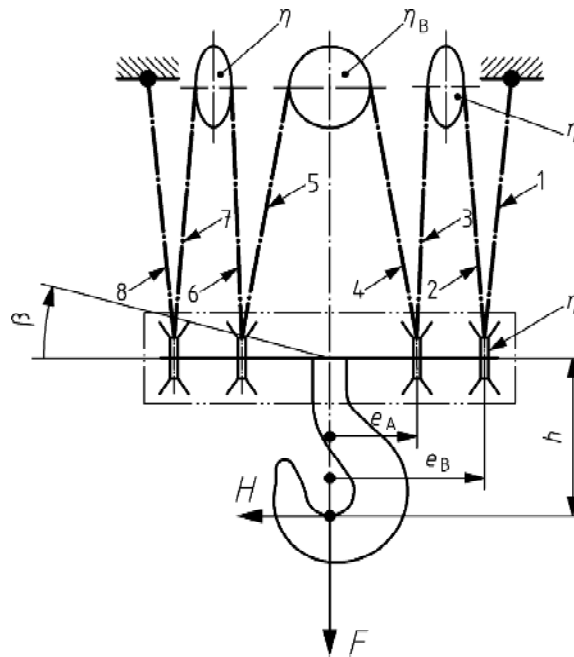
I.3 Articulation of a hook suspension by a balanced rope reeving

Due to a wide variety of possible configurations, an example only is given here, i.e. the calculation of the resistance factor for an eight-fall rope system.

In a balanced rope reeving system, for example that shown in [Figure I.3](#), differential tensions are generated within the individual rope falls due to sheave losses, when the hook suspension is tilted e.g. due to a horizontal force on the hook.

The sample reeving system assumes:

- a) symmetry about the hook centre line;
- b) sheaves are similar except that the middle sheave has a different efficiency than the others.



Key

- β direction of tilting
- h distance from the hook seat to the centre of the articulation
- H horizontal force causing the tilting

Figure I.3 — An example of a rope reeving system

When the hook suspension is tilted, maximum force is generated in rope fall no. 1, minimum force in fall no. 8, and the rope forces fulfil a relationship as depicted within [Table I.1](#).

Table I.1 — Rope forces in a tilted condition

| Rope no. (i) | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|-------------------|-------|-------------------|---------------------|---------------------|-----------------------------------|-----------------------------------|-----------------------------------|-----------------------------------|
| Rope force, R_i | R_1 | $\eta \times R_1$ | $\eta^2 \times R_1$ | $\eta^3 \times R_1$ | $\eta_B \times \eta^3 \times R_1$ | $\eta_B \times \eta^4 \times R_1$ | $\eta_B \times \eta^5 \times R_1$ | $\eta_B \times \eta^6 \times R_1$ |

where

- R_1 is the force in rope fall no. 1, the extreme rope on the descending side of the reeving;
- η is the efficiency of a basic sheave;
- η_B is the efficiency of the middle sheave.

Equilibrium condition in vertical direction presumes that the sum of the rope forces equals to the vertical force acting on the hook. From this we can derive Equation (I.3) for the rope force R_1 .

$$R_1 = \frac{F}{1 + \eta + \eta^2 + \eta^3 + \eta_B \times \eta^3 + \eta_B \times \eta^4 + \eta_B \times \eta^5 + \eta_B \times \eta^6} \quad (I.3)$$

where F is the vertical force acting on the hook.

The other rope forces R_i are derived based upon Equation (I.3) and the equations in [Table I.1](#). Forces R_i combine with distances e_i and generate a turning moment about the central vertical axis; this moment is derived as per Equation (I.4):

$$M = \sum_{i=1}^8 R_i \times e_i \quad (\text{I.4})$$

where e_i is the horizontal coordinate of the rope (i) from the hook suspension centre. Coordinates are taken positive on the descending side of the hook suspension, see [Figure I.3](#).

By placing the contents of [Table I.1](#) and Equation (I.3) to Equation (I.4), we get finally the tilting resistance C_t for this specific rope reeving system as follows:

$$C_t = \frac{M}{F} = \frac{e_B \times (1 + \eta - \eta_B \times \eta^5 - \eta_B \times \eta^6) + e_A \times (\eta^2 + \eta^3 - \eta_B \times \eta^3 - \eta_B \times \eta^4)}{1 + \eta + \eta^2 + \eta^3 + \eta_B \times \eta^3 + \eta_B \times \eta^4 + \eta_B \times \eta^5 + \eta_B \times \eta^6} \quad (\text{I.5})$$

where e_A and e_B are the horizontal distances of the sheaves from the hook suspension centre, see [Figure I.3](#).

Annex J (informative)

Guidance for selection of hook size using [Annexes C to E](#)

J.1 General

In cases where a hook body in accordance with [Annex A](#) or B is used, the proofs of static and fatigue strength of the hook body can be done using the limit design forces in [Annexes C](#) and [D](#).

Additionally, the proof of competence for the machined shank with a chosen design shall be carried out in accordance with appropriate sub-clauses in [Clauses 5](#) and [6](#).

All the assessments, choices of values and conclusions concerning the case specific loads and factors are given as an example only. Those shall not be considered as a general guidance for the same.

J.2 Case description

Search for a minimum size hook for the following case:

- mass of the rated load of the crane/hook: $m_{RC} = 50$ t;
- governing load factor ϕ : $\phi_2 = 1,21$;
- operating temperature for the hook: $T = 150$ °C;
- governing load case is regular loads: $\gamma_p = 1,34$;
- normal crane application, the risk factor: $\gamma_n = 1$;
- chosen to use a single hook from [Annex A](#);
- chosen to use the material class T;
- the use of the crane is specified through the classification as follows: number of cycles in accordance with class U₅ and load spectrum class Q₄.

J.3 Proof of static strength

Starting point is the condition given in Formula (16). The influence factor f_1 for the operation temperature is calculated in accordance with Formula (15):

$$f_1 = 1 - 0,25 \times (T - 100) / 150 = 0,9167 \quad (J.1)$$

The minimum, required static limit design force of the hook can be written, based upon Formulae (1) and (16) as follows:

$$F_{Rd,s} \geq \frac{F_{Sd,s}}{f_1} = \frac{\phi_2 \times m_{RC} \times g \times \gamma_p \times \gamma_n}{f_1} = \frac{1,21 \times 50 \times 9,81 \times 1,34}{0,9167} = 868 \text{ [kN]} \quad (J.2)$$

The hook is then searched from [Table C.1](#) from the material column "T". The hook number 16 has a static limit design force $F_{Rd,s} = 963$ kN, which fulfils the requirement in Equation (J.2).

J.4 Proof of fatigue strength

Starting point is the condition given in Formula (35). The influence factor f_1 for the operation temperature is calculated in accordance with Formula (31):

$$f_1 = 1 - 0,1 \times (T - 100) / 150 = 0,967 \quad (\text{J.3})$$

The value for the conversion factor k_C is found from [Annex E](#) at intersection of row U5 and column Q4.

$$k_C = k_6^* / \sqrt[6]{s_Q} = 1,53 \quad (\text{J.4})$$

The minimum, required fatigue limit design force of the hook can be written, based upon Formulae (18) and (35) as follows:

$$F_{\text{Rd},f} \geq \frac{F_{\text{Sd},f}}{f_1 \times \left(k_6^* / \sqrt[6]{s_Q} \right)} = \frac{\varphi_2 \times m_{\text{RC}} \times g}{f_1 \times k_C} = \frac{1,21 \times 50 \times 9,81}{0,967 \times 1,53} = 401 \quad [\text{kN}] \quad (\text{J.5})$$

The hook is then searched from [Table D.1](#) from the material column "T". The hook number 25 has a fatigue limit design force $F_{\text{Rd},f} = 420$ kN, which fulfils the requirement in Formula (J.5).

J.5 Final selection of hook

The selection of hook body shall meet both design criteria for static and fatigue strength. Consequently, the larger of the resulting hooks in [J.3](#) and [J.4](#) shall be chosen. For this case the minimum single hook of those in [Annex A](#) is the hook number 25, of material class T.

The final selection of the hook, which depends on the detailed shank design, shall have to take into account also the results of calculations in accordance with [Clauses 5](#) and [6](#).

Annex K (normative)

Information to be provided by the hook manufacturer

The hook manufacturer shall provide the following information for the technical file of the crane.

| Information to be provided by the hook manufacturer | |
|--|---|
| General | Manufacturer |
| | Unique manufacturers traceability hook marking |
| | Serial or lot number of the hook / date of manufacturing |
| | Single / ramshorn hook |
| | With or without a safety latch |
| | Mass of the hook, in kilograms (kg) |
| Dimensions | Size code and standard for forged dimensions |
| | Length of hook: L , in millimetres (mm) |
| | Size code and standard for shank dimensions (if hook is supplied with a finished shank) |
| | Certificate of dimensions |
| | Gap measure(s): y, y_1, y_2 , in millimetres (mm) |
| Material | Grade and standard |
| | Raw material certificate |
| | Heat treatment certificate |
| | Finished forging material certificate |
| Manufacturing | Process type / description |
| | NDT inspection certificate |
| | Load test certificate (if relevant) |
| | Proof loading certificate, if applied |
| Operational | Working temperature range(s) °C — selected as per Table 2 |
| Optional info | Static limit design force of the hook body, in kilonewtons (kN) |
| | Fatigue limit design force of the hook body, in kilonewtons (kN) |

Bibliography

- [1] DIN 15400, *Lifting hooks; materials, mechanical properties, lifting capacity and stresses*
- [2] BS 2903, *Specification for higher tensile steel hooks for chains, slings, blocks and general engineering purposes*¹⁾

1) Withdrawn. Replaced by BS EN 1677-5:2001+A1:2008.

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