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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

Reference number
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Contents

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/mm^2 and yield stress of not more than 600 N/mm^2.

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 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
kQ	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 × 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
β_n, β_{nS}, β_{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
γ_{Hf}, γ_{Sf}	Fatigue strength specific resistance factors
η_1	Edge distance of a hook body section
ν	Factor for load component
vh, vs	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
$\sigma_{\mathrm{Tmax}}, \sigma_{\mathrm{T1}}, \sigma_{\mathrm{T2}}$	Transformed stress amplitudes
$\Delta\sigma_{\mathrm{c}}$	Characteristic fatigue strength
$\Delta\sigma_{\mathrm{Rd}}$	Limit fatigue design stress
$\Delta\sigma_{\mathrm{Sd},i}$	Stress range in a lifting cycle i
$\Delta\sigma_{\mathrm{Sd,max}}$	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}
$\mathrm{T} \geq -10$ °C	0 °C	15 %	35 J
$\mathrm{T} \geq -20$ °C	− 10 °C		
− 30 °C > T ≥ − 40 °C	− 30 °C		
− 40 °C > T ≥ − 50 °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 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/mm²	Ultimate strength f_u N/mm²
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 ± 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

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

   Ramshorn hook

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

   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, α = 45° (see 5.5.3)

$M_{hf.}$ is derived from the formula

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

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.

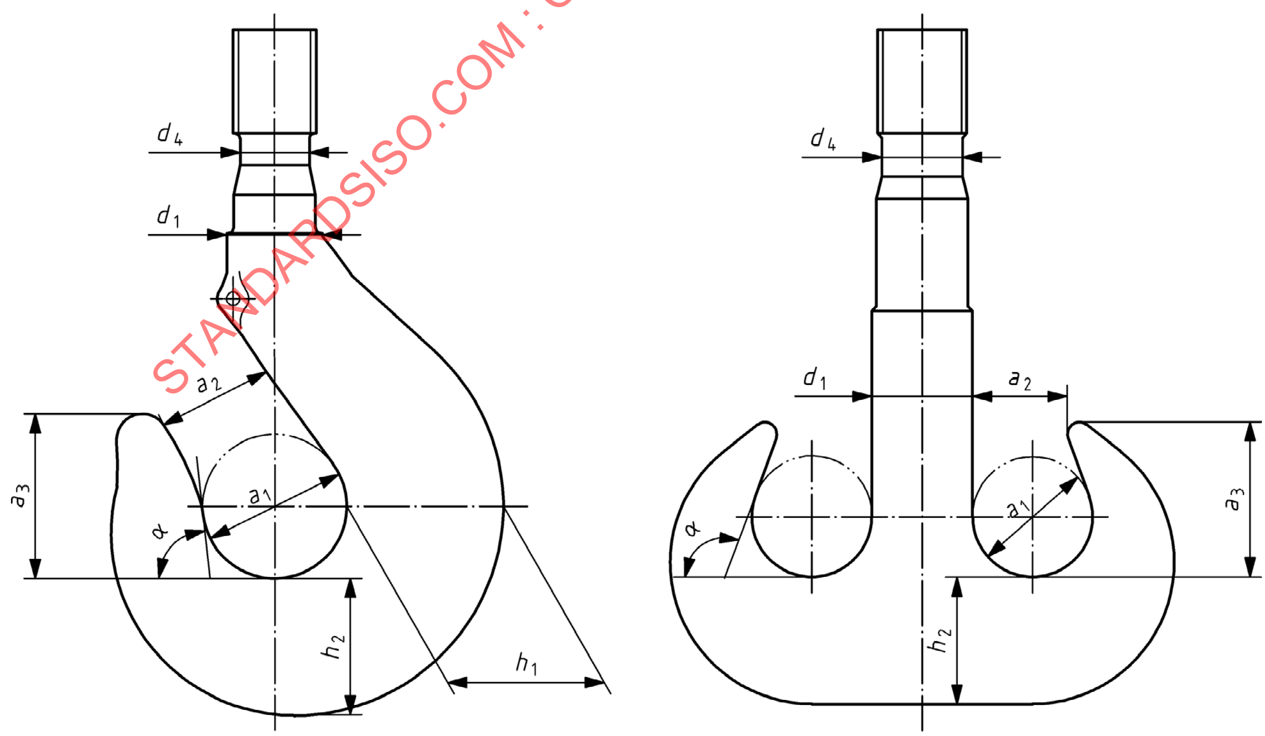


Figure 1 — Hook dimensions

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 $\alpha \geq 60°$, for a ramshorn hook $\alpha \geq 90°$

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

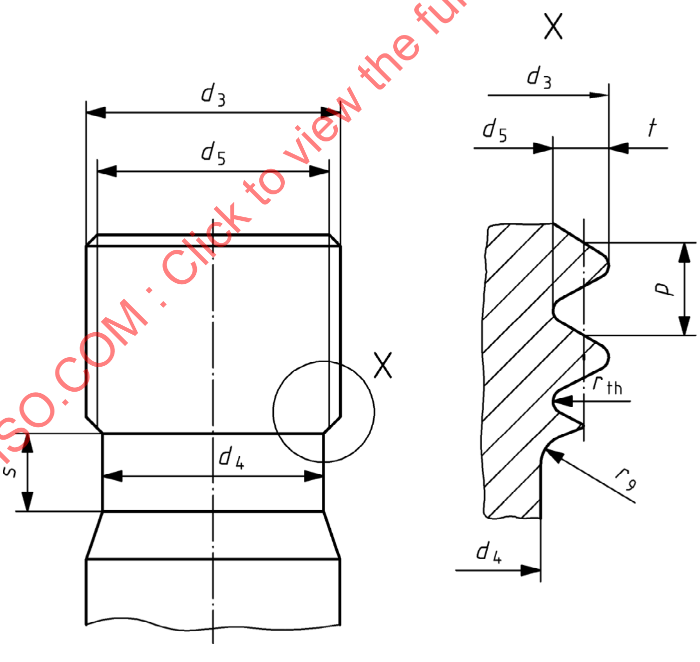


Figure 2 — Machined dimensions of shank

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:

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

with $\varphi = \max\left\{\varphi_2; \left(1 + \varphi_5 \times \frac{a}{g}\right)\right\}$

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 m/s^2;

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

γ_p = 1,34 for regular loads (load combinations A);

γ_p = 1,22 for occasional loads (load combinations B);

γ_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:

$$F_{\mathrm{Sd,s}} = F_{\mathrm{H}} \times \gamma_{\mathrm{p}} \times \gamma_{\mathrm{n}} \tag{2}$$

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 $H_{\mathrm{Sd,s}}$ due to horizontal accelerations shall be calculated as follows:

$$H_{\mathrm{Sd,s}} = \min \begin{Bmatrix} m_{\mathrm{RC}} \times a \times \varphi_5 \times \gamma_{\mathrm{p}} \times \gamma_{\mathrm{n}} \\ C_{\mathrm{t}} \times F_{\mathrm{Sd,s}} / h \end{Bmatrix} \tag{3}$$

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;

$F_{\mathrm{Sd,s}}$ is the vertical design force in accordance with 5.2, related to the loading condition where $H_{\mathrm{Sd,s}}$ 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 $H_{Sd,s}$.

$$M_1 = H_{Sd,s} \times h_s \tag{4}$$

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.

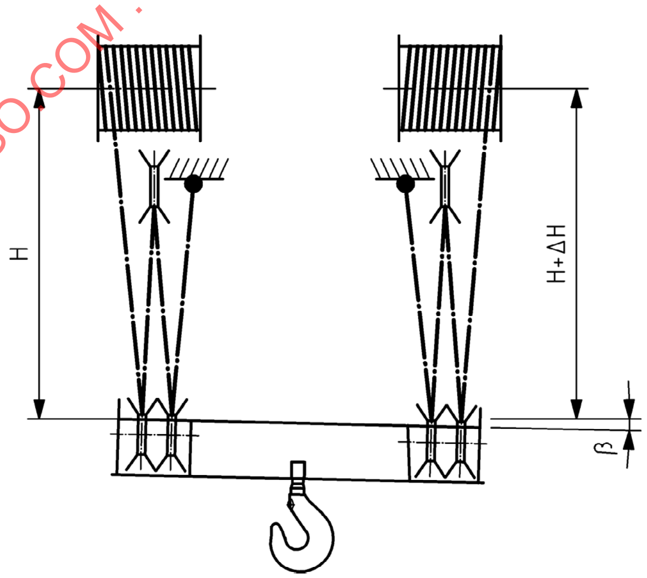


Figure 3 — Tilting of a hook in case of different hoist travel distances

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:

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

where

$F_{\mathrm{Sd,s}}$ 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:

$$\beta = \arctan\left(C_t / h\right) \qquad (6)$$

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.

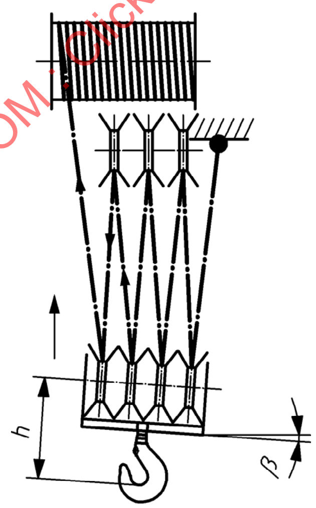


Figure 4 — Tilting of a hook suspension in a single rope reeving system

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:

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

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 $M_{\mathrm{Sd,s}}$ 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:

$$M_{\mathrm{Sd,s}} = \min \begin{Bmatrix} (M_1 + M_2 + M_3) \\ C_{\mathrm{t}} \times F_{\mathrm{Sd,s}} \end{Bmatrix} \tag{9}$$

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;

$F_{\mathrm{Sd,s}}$ 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:

$$M_{\mathrm{Sd,s}} = M_4 \tag{10}$$

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 $F_{\mathrm{Sd,s}}$ 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 $\alpha = 45°$.

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.

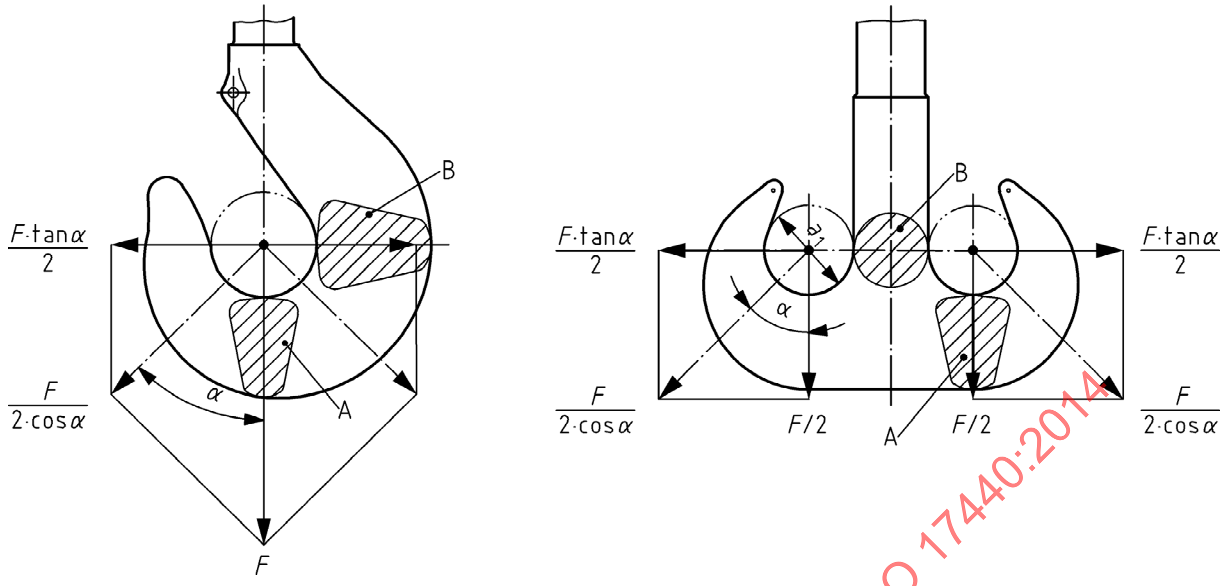


Figure 5 — Load actions on hook body and critical sections for calculation

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:

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

with

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

and

$v = 1$ for section B of single hooks;

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

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:

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

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:

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

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

$$\sigma_{\mathrm{Sd,s}} \leq f_{\mathrm{Rd}} = f_1 \times \frac{f_{\mathrm{y}}}{\gamma_{\mathrm{m}} \times \gamma_{\mathrm{sm}}} \tag{14}$$

where

$\sigma_{\mathrm{Sd,s}}$ 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 (γ_{m} = 1,1);

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

γ_{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.

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

γ_{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 °C ≤ T ≤ 100 °C:

$$f_1 = 1$$

For 100 °C < T ≤ 250 °C:

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

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 $F_{\mathrm{Rd,s}}$ 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:

$$F_{\mathrm{Rd,s}} = \frac{f_{\mathrm{y}}}{\gamma_{\mathrm{m}} \times \gamma_{\mathrm{sm}}} \times \frac{I \times (1 - \eta_1 / R)}{v \times R \times \eta_1} \tag{16}$$

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

$$F_{Sd,s} \le f_1 \times F_{Rd,s} \tag{17}$$

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:

$$F_{Sd,f,i} = \varphi_2 \times m_i \times g \tag{18}$$

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 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:

$$H_{Sd,f,i} = \min\begin{Bmatrix} m_i \times a \times \varphi_5 \\ C_t \times m_i \times g / h \end{Bmatrix} \tag{19}$$

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 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:

$$M_{1,f,i} = H_{Sd,f,i} \times h_s \tag{20}$$

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:

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

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:

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

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:

$$s_h = k_h \times v_h \tag{23}$$

with

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

and

$$v_h = \frac{N}{N_D} \tag{25}$$

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

$$s_Q = kQ \times N / N_D \tag{26}$$

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:

$$s_h = {s_Q}\Big/{\left(k_6^{\,*}\right)^m} \tag{27}$$

with

$$k_6^{\,*} = \sqrt[6]{\frac{kQ}{k_h}} \tag{28}$$

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(σ)/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:

$$\Delta\sigma_{Rd} = f_1 \times f_2 \times \Delta\sigma_c \tag{29}$$

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:

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

where f_u is the ultimate strength of the material in newtons per square millimetre (N/mm^2).

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

for 100 °C ≤ T ≤ 250 °C:

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

for – 50 °C ≤ T ≤ 100 °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 mm ≤ b_{max} ≤ 150 mm:

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

for b_{max} < 25 mm

$$f_2 = 1$$

for b_{max} > 150 mm:

$$f_2 = 0{,}74$$

where

b_{ref} is the reference width (b_{ref} = 25 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

$$\Delta\sigma_{Sd,max} \le \frac{\Delta\sigma_{Rd}}{\gamma_{Hf} \times \sqrt[m]{s_h}} = \frac{k_6^* \times \Delta\sigma_{Rd}}{\gamma_{Hf} \times \sqrt[m]{s_Q}} \quad (33)$$

where

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

$\Delta\sigma_{Rd}$ 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:

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

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:

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

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

$$F_{Sd,f} \le \frac{f_1 \times F_{Rd,f}}{\sqrt[m]{s_h}} = \frac{f_1 \times k_6{}^* \times F_{Rd,f}}{\sqrt[m]{s_Q}} \tag{36}$$

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:

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

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

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 $\sigma_{a1} = \sigma_a\left(F_{Sd,f,i}\right)$ (Formula (37)), where $F_{Sd,f,i}$ 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.

$\sigma_{Sd,a,psmin} = \sigma_{Sd,sh,pr}$ 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 $F_{Sd,f,i}$ (calculated as shown above) to the value of the initial residual stress $\sigma_{Sd,sh,pr}$:

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

b) Bending stress is $\sigma_{b1} = \sigma_b(M)$ [Formula (38)], where $M = \max\left[M_{2,f,i}, M_{3,f,i}\right]$ 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 $\sigma_{\mathrm{Sd,sh,pr}}$ 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 $\sigma_{\mathrm{Sd,ps1},min}$ equal to residual stress $\sigma_{\mathrm{Sd,sh,pr}}$ at zero hook load:

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

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

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

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 $\sigma_{\mathrm{Sd,sh,pr}}$ 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 $\sigma_{\mathrm{Sd,a2,ps},min}$ equal to residual stress $\sigma_{\mathrm{Sd,sh,pr}}$ at zero hook load

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

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

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

f) Bending stress is $\sigma_{\mathrm{b2},i} = \sigma_{\mathrm{b}}\left(M_{1,\mathrm{f},i}\right)$ [Formula (38)], where $M_{1,\mathrm{f},i}$ is in accordance with 6.4.1

Residual compressive stress $\sigma_{\mathrm{Sd,sh,pr}}$ 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 $\sigma_{\mathrm{m2},i} = \sigma_{\mathrm{a1},i}$ and stress amplitude $\sigma_{\mathrm{A2},i} = \sigma_{\mathrm{b2},i}$

With residual compressive stress $\sigma_{\mathrm{Sd,sh,pr}}$ 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 $\sigma_{\mathrm{Sd,ps2,min}}$ equal to $\left(\sigma_{\mathrm{Sd,sh,pr}} - \sigma_{\mathrm{b2}}\right)$ at zero hook load

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

to a maximum value of the design stress in the shank of “proofed” hook $\sigma_{\mathrm{Sd,ps1,max}}$ calculated by adding the stress σ_{a2} for the hook load $F_{\mathrm{Sd,f},i}$ (calculated as shown above) and bending stress σ_{b2} to the value of initial residual stress $\sigma_{\mathrm{Sd,sh,pr}}$

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

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:

$$\sigma_M = 0{,}45 \times f_u \tag{39}$$

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	$u_S = {(d_e - d_4)}/{2}$	$u_T = {(d_e - d_5)}/{2}$
Factor φ	$\phi = \dfrac{1}{2 + 4 \times \sqrt{\dfrac{u_S}{r_9}}}$	$\phi = \dfrac{1}{2 + 4 \times \sqrt{\dfrac{u_T}{r_{th}}}}$
Factor χ	$\chi = \dfrac{2 \times (1+\phi)}{r_9}$	$\chi = \dfrac{2 \times (1+\phi)}{r_{th}}$
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	$\beta_n = {\alpha_S}/{n}$	$\beta_n = {\alpha_T}/{n}$

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

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

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

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

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/mm^2).

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:

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

with

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

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 $0{,}4\ \mu m \leq R_a \leq 3{,}2\ \mu m$;

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

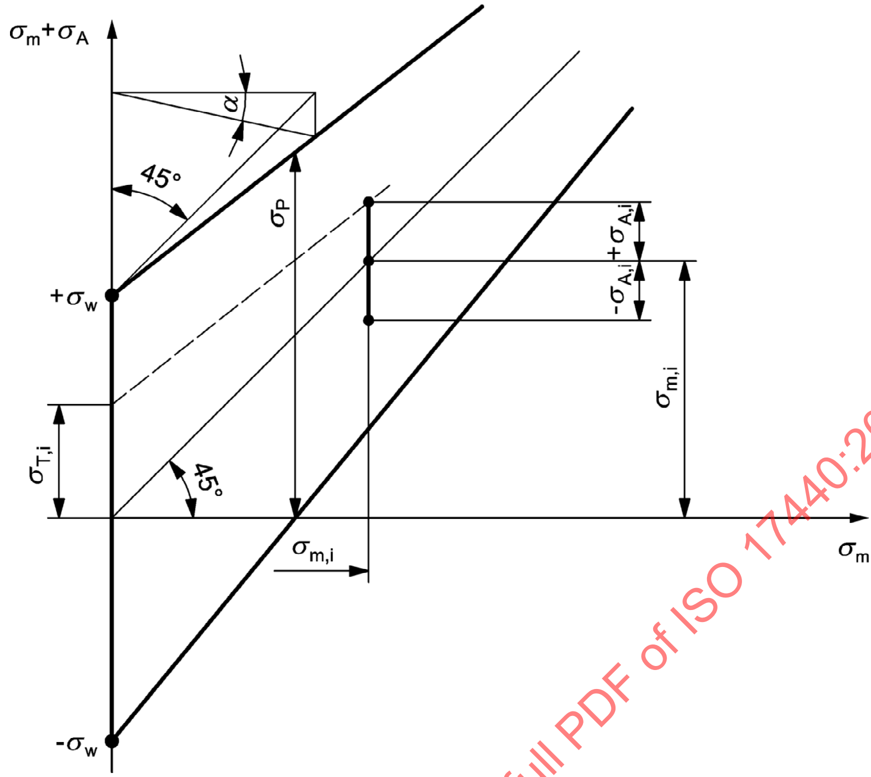


Figure 6 — Smith diagram and transformation of stress amplitude

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:

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

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

$$\sigma_{T1,i} = \sigma_{A1,i} + \mu \times \sigma_{m1,i} \tag{45}$$

Cycle Type 2

$$\sigma_{T2,i} = \sigma_{A2,i} + \mu \times \sigma_{m2,i} \tag{46}$$

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:

$$s_{\mathrm{S}} = k_{\mathrm{S}} \times v_{\mathrm{S}} \tag{47}$$

with

$$k_{\mathrm{S}} = \frac{1}{N + p_{\mathrm{a}} \times N} \times \left[\sum_{i=1}^{N} \left(\frac{\sigma_{\mathrm{T1},i}}{\sigma_{\mathrm{Tmax}}} \right)^{m} + p_{a} \times \sum_{i=1}^{N} \left(\frac{\sigma_{\mathrm{T2},i}}{\sigma_{\mathrm{Tmax}}} \right)^{m} \right] \tag{48}$$

and

$$v_{s} = \frac{N + p_{a} \times N}{N_{D}} \tag{49}$$

where

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

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:

$$s_{\mathrm{S}} = k_{\mathrm{S}} \times v_{\mathrm{S}} \tag{50}$$

with

$$k_{\mathrm{S}} = \frac{1}{1 + p_{\mathrm{a}}} \times \frac{kQ}{\left(k_{5}^{*}\right)^{m}} \left[\left(\frac{\sigma_{\mathrm{T1,max}}}{\sigma_{\mathrm{Tmax}}} \right)^{m} + p_{a} \times \left(\frac{\sigma_{\mathrm{T2,max}}}{\sigma_{\mathrm{Tmax}}} \right)^{m} \right] \tag{51}$$

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

and

$$k_5{}^* = \sqrt[5]{\frac{kQ}{k_5}} \tag{53}$$

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

$$\sigma_{Tmax} \le \frac{\sigma_W}{\gamma_{Sf} \times \sqrt[m]{s_s}} \tag{54}$$

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 (γ_{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 $a = 0{,}2$ 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.

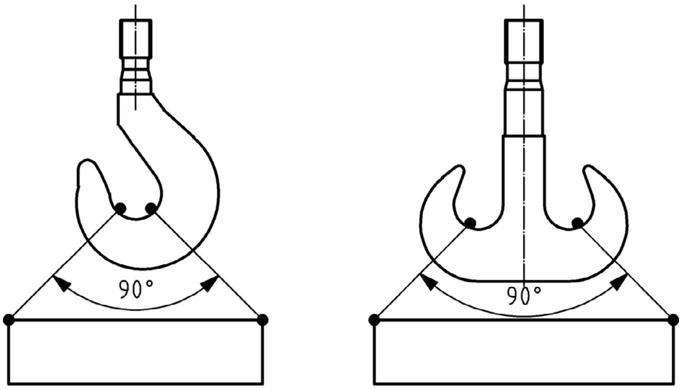


Figure 7 — Markings on a hook

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;

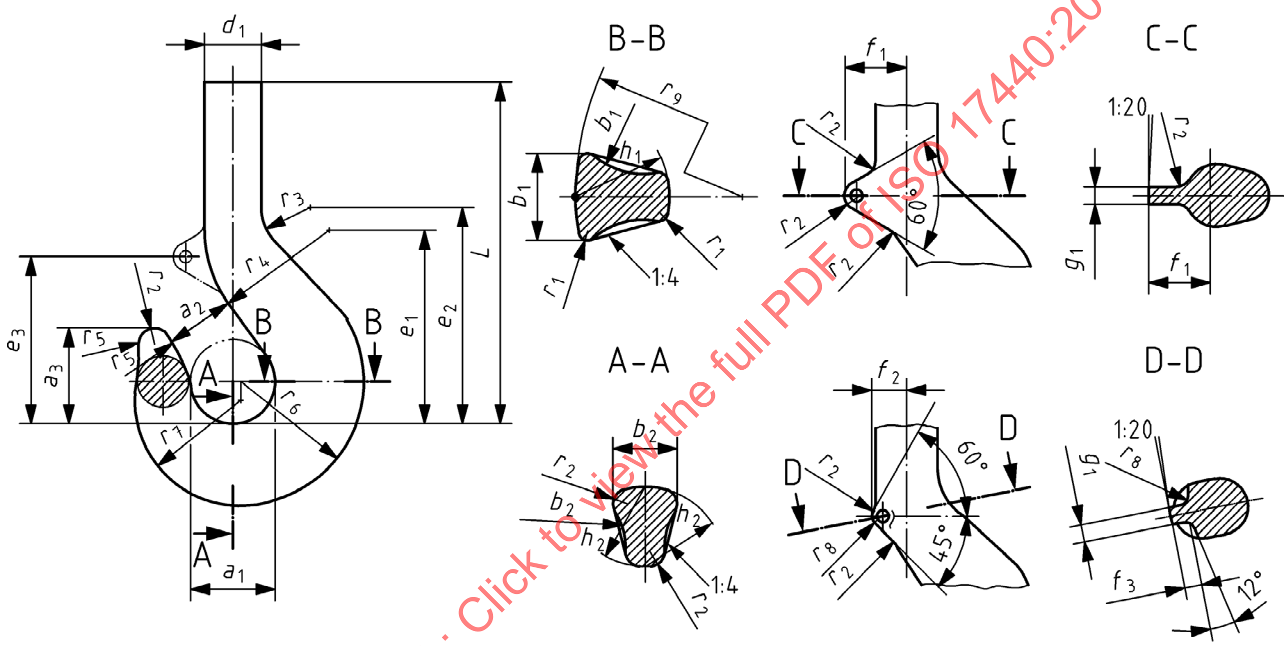
— temperature limits of the hook.

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						
																			e_3	f_1	f_2	f_3	g_1	r_8	L
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.

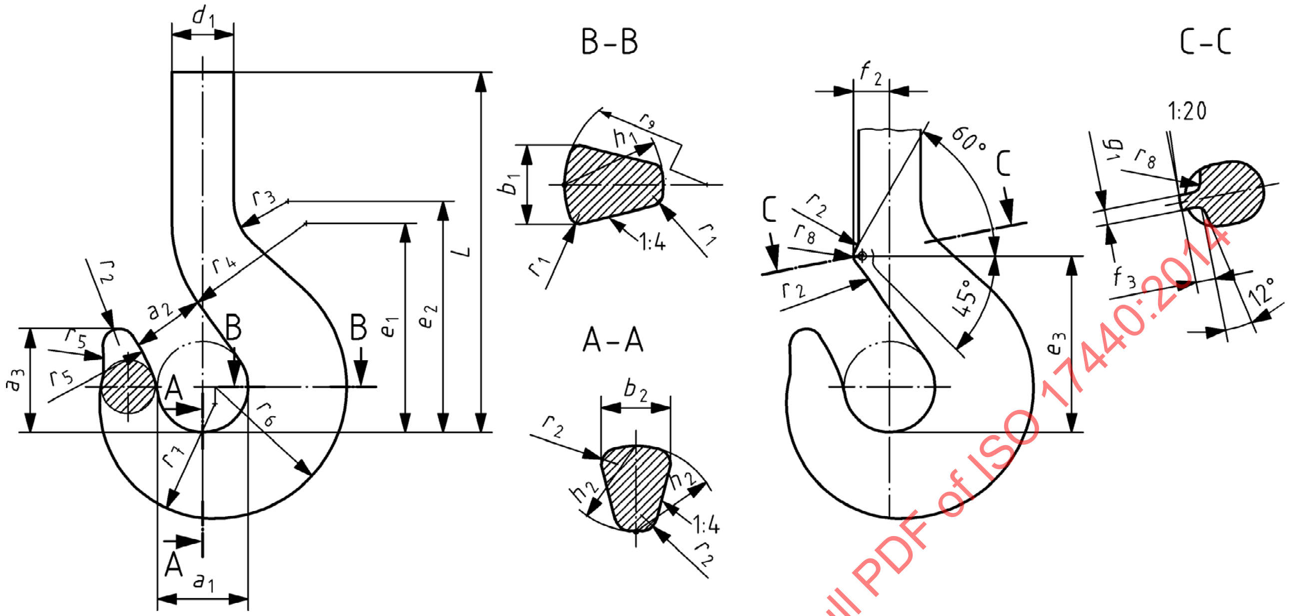
Table A.1 — (continued)

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 G.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.

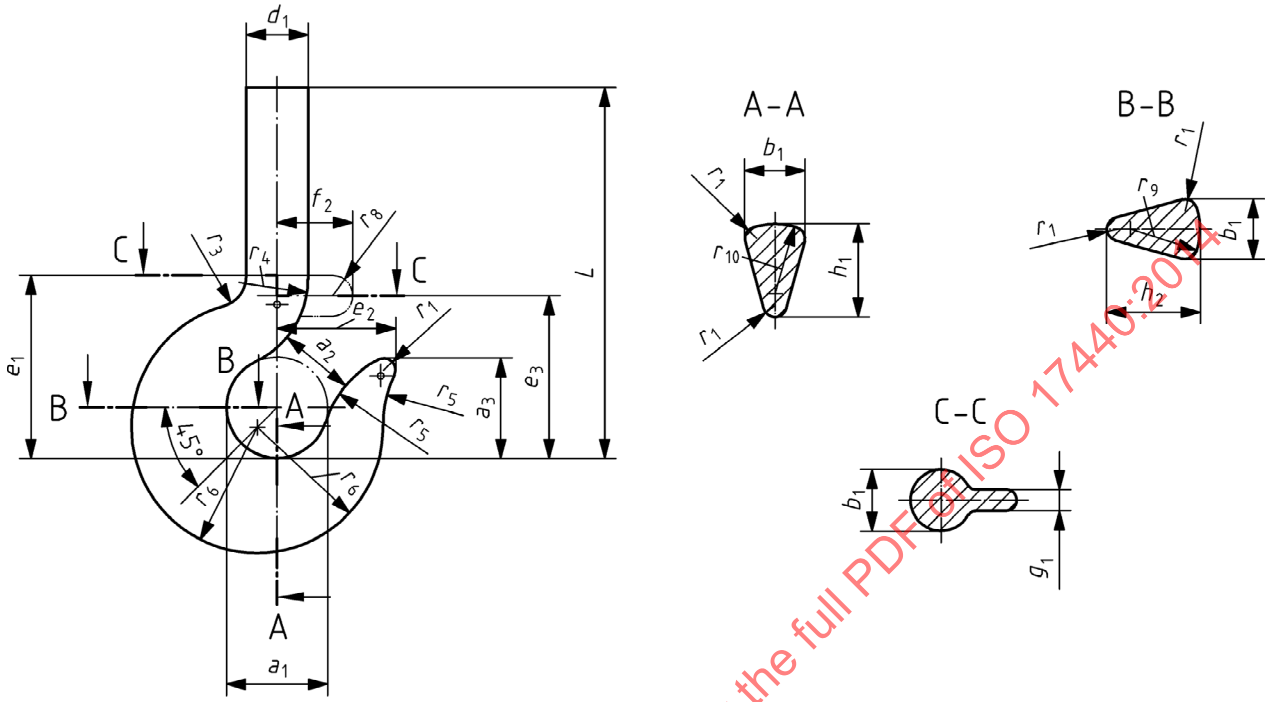


Figure A.3 — Symbols of dimensions for single point hooks

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.

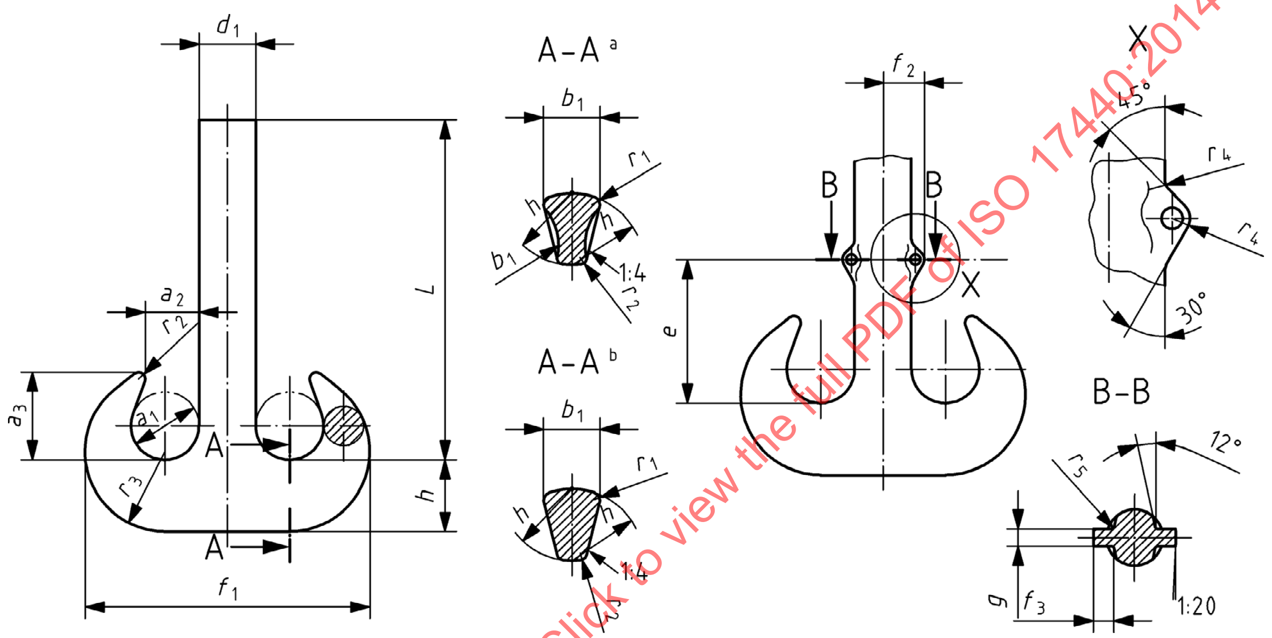
NOTE 4 Length of hook shank "L" to suit application.

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	125	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 Section B Hook factor $M_{hf.}$ mm^2	Classified material grades $F_{Rd,s}$ M	P	S	T	V	Ramshorn hooks, types RS/RSN and RF/RFN Section A Hook factor $M_{hf.}$ mm^2	Classified material grades $F_{Rd,s}$ M	P	S	T	V
006		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	0.75	399,61	104	153	184	242	291	217,87	95	139	167	220	264
5	single	506,63	132	193	233	307	368	268,78	117	171	206	272	326
6	hooks	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	1 242,5	324	474	572	753	904	680,27	295	433	522	687	825
16	ram-shorn	1 590,0	414	607	732	963	1 156	845,31	367	538	649	854	1 024
20	hooks	1 873,3	520	763	920	1 210	1 452	1056,5	459	672	811	1 067	1 281
25		2 532,0	660	967	1 166	1 535	1 841	1343,0	583	855	1 031	1 356	1 628
32		3 192,7	832	1 219	1 471	1 935	2 322	1700,5	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.	Single hooks, type B								
	All materials		Classified material grades $F_{Rd,s}$					Additional materials	
	Hook family factor γ_{sm}	Section B Hook factor M_{hf}. mm²	M	P	S	T	V	f_y N/mm²	$F_{Rd,s}$
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 ≤ 100 °C)

Hook no.	Single hooks, type B							
	All materials	Classified material grades $F_{Rd,f}$					Additional materials	
	Section B hook factor $M_{hf.}$ mm²	M	P	S	T	V	f_u N/mm²	$F_{Rd,f}$
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

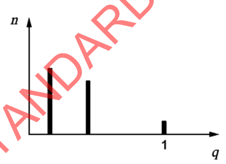
When classified duty is utilized, k_c shall be calculated in accordance with Table E.1.

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

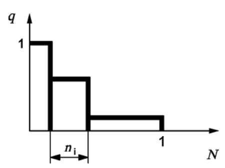
	Class Q	Q_0	Q_1	Q_2	Q_3	Q_4	Q_5
	$kQ(3)$	0,031 3	0,062 5	0,125	0,25	0,5	1,0
	Factor k_6*	1,348	1,343	1,259	1,172	1,084	1
Class U	**C [cycles]**						
U_0	16 000	5,37	4,77	3,98	3,30	2,72	2,24
U_1	31 500	4,8	4,26	3,56	2,95	2,43	2,00
U_2	63 000	4,27	3,79	3,17	2,63	2,17	1,78
U_3	125 000	3,81	3,38	2,83	2,34	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.

a) **Density function, where** $\int_{q_0}^{1} n(q) = 1$ b) **Accumulated density, where** $N(q) = \int_{q}^{1} n(q) \times dq$

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

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

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

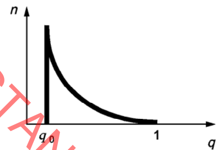
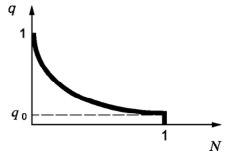
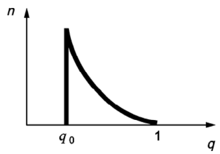
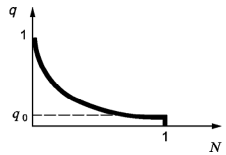
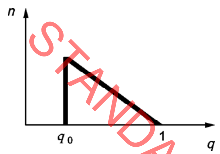
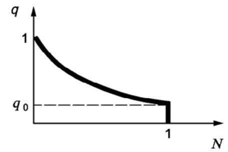
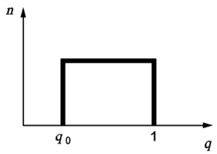
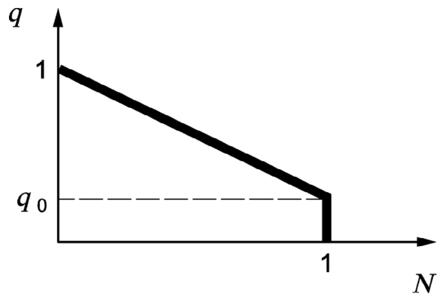
Class	Density function	Accumulated density function
Q_0, Q_1	 $n=4\times\frac{(1-q)^3}{(1-q_0)^4}$	 $N_{(q)}=\left(\frac{1-q}{1-q_0}\right)^4$ or $q_{(N)}=1-(1-q_0)\times\sqrt[4]{N}$
	$k_{(m)}=\frac{4}{(1-q_0)^4}\times\left[\frac{1-q_0^{m+1}}{m+1}-3\times\frac{1-q_0^{m+2}}{m+2}+3\times\frac{1-q_0^{m+3}}{m+3}-\frac{1-q_0^{m+4}}{m+4}\right]$ for Class Q_0: q_0 = 0,0226; for Class Q_1: q_0 = 0,1828	
Q_2	 $n=3\times\frac{(1-q)^2}{(1-q_0)^3}$	 $N_{(q)}=\left(\frac{1-q}{1-q_0}\right)^3$ or $q_{(N)}=1-(1-q_0)\times\sqrt[3]{N}$
	$k_{(m)}=\frac{3}{(1-q_0)^3}\times\left[\frac{1-q_0^{m+1}}{m+1}-2\times\frac{1-q_0^{m+2}}{m+2}+\frac{1-q_0^{m+3}}{m+3}\right]$ for Class Q_2: q_0 = 0,2765	

Table E.3 — (continued)

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

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:

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

NOTE Te denotes metric tonnes, 1 000 kgf.

Hook dimensions and section properties:

$a_1 = 92$ mm seat circle diameter

Section B-B:

$\eta_1 = 36{,}18$ mm coordinate y from centroid to the intrados, see Annex H

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

$I = 1{,}696 \times 10^6$ mm^4 reference moment of inertia of cross section, see Annex H

$v = 1$ factor for load component, section B-B see 5.5.3

Hook material: 150 M 19Q

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

$f_u = 620$ N mm^{-2} 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

$F_{Sd,f} = \phi_2 F_H = 127{,}49$ kN 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

$F_{PL} = 1{,}5 \times f_y \times M$ [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 $F_{PL} = 1{,}5 \times f_y \times M = 205{,}94$ 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}: $M_{PL} = F_{PL} \times R = 16{,}92$ 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:

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

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

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

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

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

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

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

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:

$$\sigma_{\text{a,p,t}} = \frac{\sigma_{\text{a,p}} - \mu \times \sigma_{\text{m,p}}}{1 - \mu \times \dfrac{(1+R_{\text{s}})}{(1-R_{\text{s}})}} \qquad \sigma_{\text{m,p,t}} = \sigma_{\text{a,p,t}}$$

where

$\sigma_{\text{a,p,t}}$, $\sigma_{\text{m,p,t}}$ are the amplitude and mean stress components of transformed stress cycle;

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:

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

the mean stress component

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

the amplitude stress component

Components of the transformed cycle:

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

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

Maximum stress range at intrados:

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

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

Stress cycling between 0 and maximum of 2 × 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:

$$\Delta\sigma_c = 0{,}315 \times f_u \times \lg\left[\frac{13\,000}{f_u}\right] = 258{,}11\ \text{N mm}^{-2}$$

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

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

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

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

$$N_D = 2 \times 10^6 \qquad s_Q = kQ \times \frac{N}{N_D} = 1$$

$$k_6^* = 1$$

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)]:

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

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

the fatigue strength resistance specific factor, see Table 7.

Hence:

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

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:

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

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)]:

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

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)

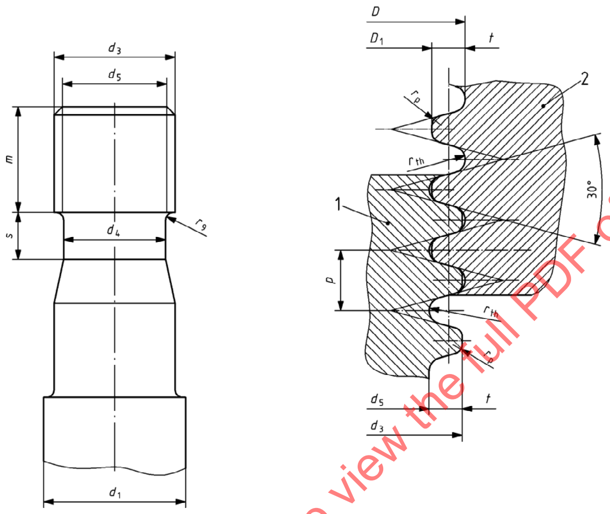
Alternatively, reduced life/ number of cycles to satisfy the inequality above can be calculated, using Formulae (26) and (27).

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.													

