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Proof of competence of hydraulic cylinders in crane applications

Vérification d'aptitude des vérins hydrauliques pour appareils de levage

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Foreword

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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 (see www.iso.org/directives).

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For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 96, *Cranes*, Subcommittee SC 10, *Design principles and requirements*.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Proof of competence of hydraulic cylinders in crane applications

1 Scope

This document applies to hydraulic cylinders that are part of the load carrying structure of cranes. It is intended to be used together with the ISO 8686 series and ISO 20332, and as such they specify general conditions, requirements and methods to prevent mechanical hazards of hydraulic cylinders, by design and theoretical verification.

This document does not apply to hydraulic piping, hoses, connectors and valves used with the cylinders, or cylinders made from other material than (carbon) steel.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. 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 683-1, *Heat-treatable steels, alloy steels and free-cutting steels — Part 1: Non-alloy steels for quenching and tempering·*

ISO 683-2, *Heat-treatable steels, alloy steels and free-cutting steels — Part 2: Alloy steels for quenching and tempering*

ISO 724, *ISO general-purpose metric screw threads — Basic dimensions*

ISO 5817:2014, *Welding — Fusion-welded joints in steel, nickel, titanium and their alloys (beam welding excluded) — Quality levels for imperfections*

ISO 8492, *Metallic materials — Tube — Flattening test*

ISO 8686 (all parts), *Cranes — Design principles for loads and load combinations*

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

ISO 20332:2016, *Cranes — Proof of competence of steel structures*

EN 10277:2018, *Bright steel products — Technical delivery conditions — Part 2: Steels for general engineering purposes*

EN 10297-1, *Seamless circular steel tubes for mechanical and general engineering purposes — Technical delivery conditions — Part 1: Non-alloy and alloy steel tubes*

EN 10305-1, *Steel tubes for precision applications — Technical delivery conditions — Part 1: Seamless cold drawn tubes*

EN 10305-2, *Steel tubes for precision applications — Technical delivery conditions — Part 2: Welded cold drawn tubes*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 12100 apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

4 Symbols

For the purposes of this document, the symbols given in Table 1 apply.

Table 1 — Symbols

Symbols	Description
$A\%$	Percentage elongation at fracture
a	Weld throat thickness
A_i, B_i, C_i, D_i	Constants
A_s	Stress area
D	Piston diameter
d	Rod diameter
$D_{a,i}$	Axles diameter
D_p	Pressure affected diameter
D_w	Weld diameter
E	Modulus of elasticity
F	Compressive force
F_A	Compressive force
FE	Finite elements
f_{Rd}	Limit design stress
$f_{Rd\sigma}$	Limit design stress, normal
$f_{Rd\tau}$	Limit design stress, shear
F_S	Lateral force
F_{Sd}	External compressive design force
$f_{w,Rd}$	Limit design weld stress
f_y	Yield strength
h	thickness of the cylinder bottom
I	Moment of inertia, generic
I_1	Moment of inertia of the tube
I_2	Moment of inertia of the rod
L	Overall length of the cylinder
L_1	Length of the cylinder tube
L_2	Length of the piston rod
m	Slope of the log $\Delta\sigma$ – log N curve
M_0	Shell section bending moment, acting at the intersection between tube and bottom
M_b	Bending moment
N	Compressive force
N_k	Critical buckling load
N_{Rd}	Limit compressive design force
p_{i1}	Maximum pressure in piston side chamber
p_{i2}	Maximum pressure in rod side chamber

Table 1 *(continued)*

Symbols	Description
p_o	Outer pressure
p_{Sd}	Design pressure
R	Middle radius of the tube ($R = r_i + t/2$)
r_i	Inner radius of the tube
r_o	Outer radius of the tube
r_r	Outer radius of the piston rod
s_3	Stress history parameter (see ISO 20332)
t	Wall thickness of the tube
T_0	Shell section transverse force, acting at the intersection between tube and bottom
x, y	Longitudinal and lateral coordinates
α	Angular misalignment, radians
γ_m	General resistance factor (γ_m = 1,1, see ISO 8686-1)
γ_{mf}	Fatigue strength specific resistance factor (see ISO 20332)
γ_R	Total resistance factor ($\gamma_R = \gamma_m \times \gamma_s$)
γ_s	Specific resistance factor
$\Delta\sigma$	Stress range
$\Delta\sigma_b$	Bending stress range in the tube
$\Delta\sigma_c$	Characteristic fatigue strength
$\Delta\sigma_m$	Membrane stress range in the tube (axial)
$\Delta\sigma_{Rd}$	Limit design stress range
$\Delta\sigma_{Sd}$	Design stress range
Δp_{Sd}	Design pressure range on piston side
δ_{max}	Maximum displacement
κ	Reduction factor for buckling
λ	Slenderness
λ_i	Friction parameters
μ_i	Friction factors
ν	Poisson's ratio (ν = 0,3 for steel)
σ_a	Axial stress in the tube
σ_b	Lower extreme value of a stress range
σ_r	Radial stress in the tube
σ_{Sd}	Design stress, normal
$\sigma_{w,Sd}$	Design weld stress, normal
σ_t	Tangential stress in the tube (hoop stress)
σ_u	Upper extreme value of a stress range
τ_{Sd}	Design stress, shear
$\tau_{w,Sd}$	Design weld stress, shear

5 General

5.1 Documentation

The documentation of the proof of competence shall include:

— design assumptions including calculation models;

— applicable loads and load combinations;

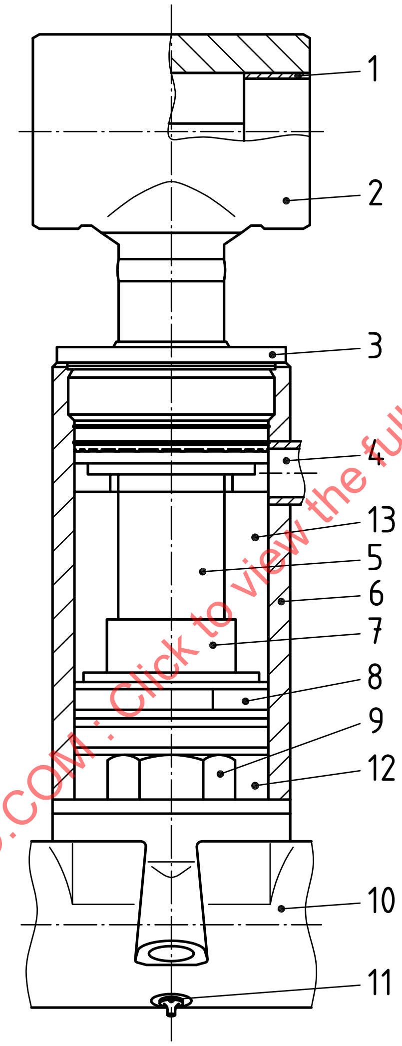
— material grades and qualities;

— weld quality levels, in accordance with ISO 5817 and ISO 20332;

— relevant limit states;

— results of the proof of competence calculation, and tests when applicable.

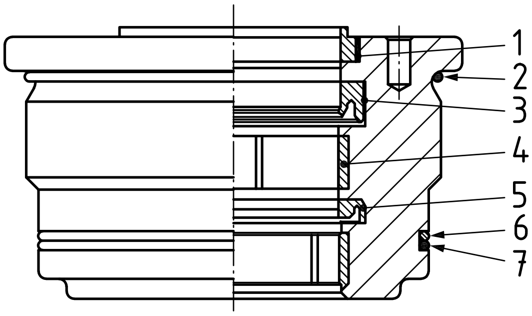
The main parts of hydraulic cylinder are indicated in Figure 1 to Figure 3.



Key

1	bushing	8	piston
2	rod head	9	nut
3	cylinder head	10	cylinder bottom
4	oil connector	11	grease nipple
5	piston rod	12	piston side chamber
6	cylinder tube	13	rod side chamber
7	spacer		

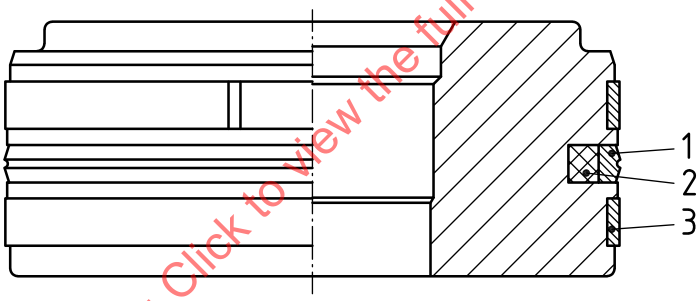
Figure 1 — Complete cylinder



Key

1 wiper
2 O-ring
3 secondary seal
4 guide ring (2 ×)
5 primary seal
6 backup ring
7 O-ring

Figure 2 — Cylinder head



Key

1 seal
2 pressure element
3 guide ring (2 ×)

Figure 3 — Piston

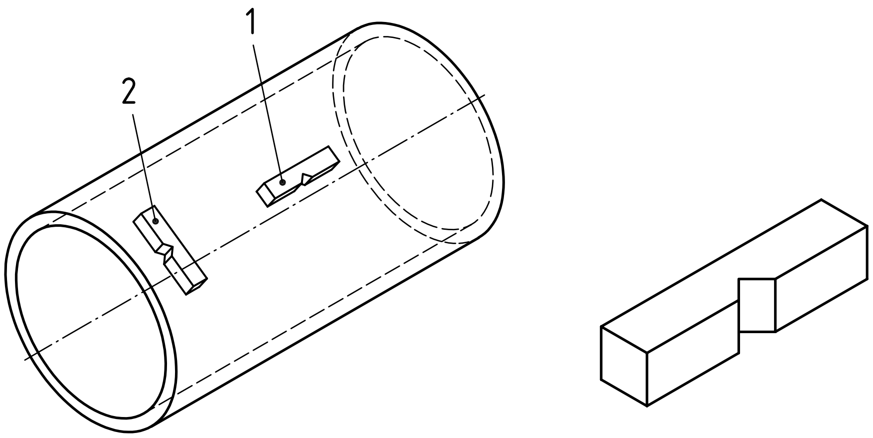
Figures 1 to 3 show some typical design features. Other designs may be used.

5.2 Materials for hydraulic cylinders

5.2.1 General requirements

The materials for load carrying cylinder tubes and piston rods shall fulfil the following requirements:

— The impact toughness in the transversal direction shall be tested in accordance with ISO 148-1 and shall meet the requirements stated in ISO 20332. Samples shall be cut out in the longitudinal direction. For cylinder tubes and pressurized piston rods, samples shall also be cut out in the transversal direction. The samples shall be prepared such that the axis of the notch is perpendicular to the surface of the tube.



Key

1 sample cut out in longitudinal direction

2 sample cut out in transversal direction

Figure 4 — Sample for impact toughness testing

— If the material thickness does not allow samples to be cut out in the transversal direction, the tube material shall pass a flattening test in accordance with ISO 8492. For welded tubes, two tests are required; one with the weld aligned with the press direction and one where the weld is placed 90° from the press direction, see Figure 4. The tube section shall be flattened down to a height H given by:

$$H = \frac{1{,}07 \cdot t}{C + \frac{t}{D_o}}$$

where

C	is a factor that depends on the yield strength of the material,
	C is 0,07 for $f_y \leq 400$ MPa and C is 0,05 for $f_y > 400$ MPa;
D_o	is the outer diameter of the tube;
t	is the wall thickness of the tube.

Material used in other parts shall meet the requirements specified in ISO 20332.

5.2.2 Grades and qualities

Steels in accordance with the following standards shall preferably be used as material for cylinder tubes and piston rods:

— ISO 683-1;

— ISO 683-2;

— EN 10277:2018;

— EN 10297-1:

— EN 10305-1;

— EN 10305-2.

Alternatively, other steel grades and qualities than those listed in this subclause may be used as material for cylinder tubes and piston rods, provided that the following conditions apply:

— the design value of f_y is limited to $f_u/1{,}1$ for materials with $f_u/f_y < 1{,}1$;

— the percentage elongation at fracture A % ≥ 14 % on a gauge length $L_0 = 5{,}65 \times \sqrt{S_0}$ (where S_0 is the original cross-sectional area).

Grades and qualities of materials used in other parts of cylinders or mounting interfaces of cylinders shall be selected in accordance with ISO 20332.

6 Proof of static strength

6.1 General

A proof of static strength by calculation is intended to prevent excessive deformations due to yielding of the material, elastic instability and fracture of structural members or connections. Dynamic factors given in the relevant part of ISO 8686 are used to produce equivalent static loads to simulate dynamic effects. Also, load increasing effects due to deformation shall be considered. The theory of plasticity for calculation of ultimate load bearing capacity is not considered acceptable for the purposes of this document. The proof shall be carried out for structural members and connections while taking into account the most unfavourable load effects from the load combinations A, B or C in accordance with the relevant part of ISO 8686 or relevant product standards.

This document considers only nominal stresses, i.e. those calculated using traditional elastic strength of materials theory; localized stress concentration effects are excluded. When alternative methods of stress calculation are used such as finite element analysis, using those stresses directly for the proof given in this document can yield inordinately conservative results as the given limit states are intended to be used in conjunction with nominal stresses.

Cylinder actions are either active or passive. The action is active when the force from the cylinder exerts a positive work on the crane structure, elsewise the action is passive.

As the forces applied to the cylinder by the crane structure are computed in accordance with ISO 8686, they are already increased by the partial safety factors γ_p and relevant dynamic factors. Formula (1) and Formula (2) give design pressures p_{Sd} caused by forces acting on the cylinder from the crane structure. In addition, additional pressures p_{Sde} caused by internal phenomena in the hydraulic circuit shall be considered and added to the design pressures p_{Sd}. Such internally generated pressures can be caused, for example, by regenerative connections, pressure drop in return lines or cushioning.

In case a cylinder is intended to be tested as a component at higher pressure than the design pressure p_{Sd}, this load case shall also be taken into account in the proof of static strength, and in which case the test pressure shall be multiplied by a partial safety factor γ_p equal to 1,05.

The design pressure p_{Sd} in the piston side chamber or in the rod side chamber shall be computed from the design force F_{Sd} taking into account the force direction and the cylinder efficiency η due to friction. An efficiency factor Ψ is used to handle the effect of cylinder friction. For active cylinders Ψ has the value of $1/\eta$ and for passive cylinders Ψ has the value of η.

For the piston side chamber, the design pressure is given by Formula (1):

$$p_{Sd} = \frac{4 \cdot F_{Sd}}{\pi \cdot D^2} \cdot \Psi \qquad (1)$$

where

F_{Sd} is the external design force;

D is the piston diameter;

Ψ is set to η for passive cylinders and to $1/\eta$ for active cylinders.

For the rod side chamber, the design pressure is given by Formula (2):

$$p_{\mathrm{Sd}} = \frac{4 \cdot F_{\mathrm{Sd}}}{\pi \cdot \left(D^2 - d^2\right)} \cdot \Psi + p_{\mathrm{Sde}} \tag{2}$$

where

F_{Sd} is the external design force;

D is the piston diameter;

d is the rod diameter;

Ψ is set to η for passive cylinders and to $1/\eta$ for active cylinders;

p_{Sde} is additional pressure caused by internal phenomena (e.g. regeneration).

Unless justified value of the efficiency η is available and used, Ψ shall be assigned the value of 1,1 for active cylinders and the value of 1,0 for passive cylinders.

6.2 Limit design stresses

6.2.1 General

The limit design stresses f_{Rd} shall be calculated from Formula (3):

$$f_{\mathrm{Rd}} = f_{\mathrm{n}}\left(f_{\mathrm{k}}, \gamma_{\mathrm{R}}\right) \tag{3}$$

where

f_{n} is a general function as described in 6.2.2;

f_{k} is the characteristic values (or nominal value);

γ_{R} is the total resistance factor.

6.2.2 Limit design stress in structural members

The limit design stress f_{Rd}, used for the design of structural members, shall be calculated from Formulae (4) and (5):

$$f_{\mathrm{Rd}\sigma} = \frac{f_{\mathrm{y}}}{\gamma_{\mathrm{Rm}}} \text{ for normal stresses} \tag{4}$$

$$f_{\mathrm{Rd}\tau} = \frac{f_{y}}{\gamma_{\mathrm{Rm}} \cdot \sqrt{3}} \text{ for shear stresses} \tag{5}$$

with $\gamma_{\mathrm{Rm}} = \gamma_{\mathrm{m}} \cdot \gamma_{\mathrm{sm}}$

where

f_{y} is the minimum value of the yield stress of the material;

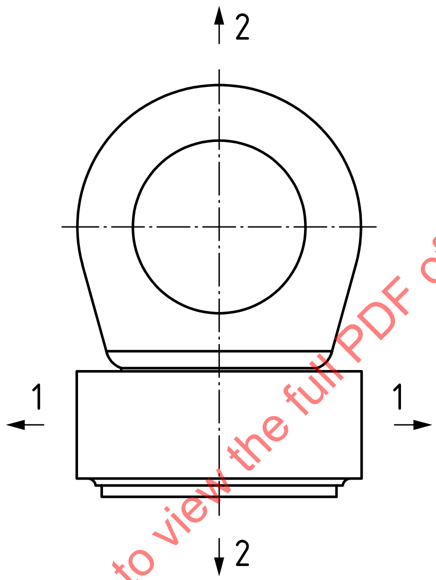
γ_m is the general resistance factor $\gamma_m = 1{,}1$ (see ISO 8686-1);

γ_{sm} is the specific resistance factor for material in accordance with ISO 20332;

γ_{sm} = 0,95 is the basic value for material not loaded perpendicular to the rolling plane.

For tensile stresses perpendicular to the plane of rolling (see Figure 5), the material shall be suitable for carrying perpendicular loads and be free of lamellar defects. ISO 20332 specifies the values of γ_{sm} for material loaded perpendicular to the rolling plane.

Figure 5 provides an example of a cylinder tube bottom where plate steel is used (eye is welded) and shows a tensile load perpendicular to plane of rolling.



Key

1 plane of rolling

2 direction of stress/load

Figure 5 — Tensile load perpendicular to plane of rolling

6.2.3 Limit design stresses in welded connections

The limit design weld stress $f_{w,Rd}$ used for the design of a welded connection shall be in accordance with ISO 20332.

6.3 Linear stress analysis

6.3.1 General

Subclause 6.3 comprises typical details for consideration that may be relevant for the proof of static strength. Details that are only relevant for fatigue analysis (e.g shell bending of tube) are not dealt with in 6.3. For cases or conditions not covered here, other recognized sources or static pressure/force testing may be used.

6.3.2 Typical cylinder arrangements

Before executing calculations, boundary conditions and loading shall be investigated. Typical conditions to be determined are:

— external forces and directions;

— type of cylinder;

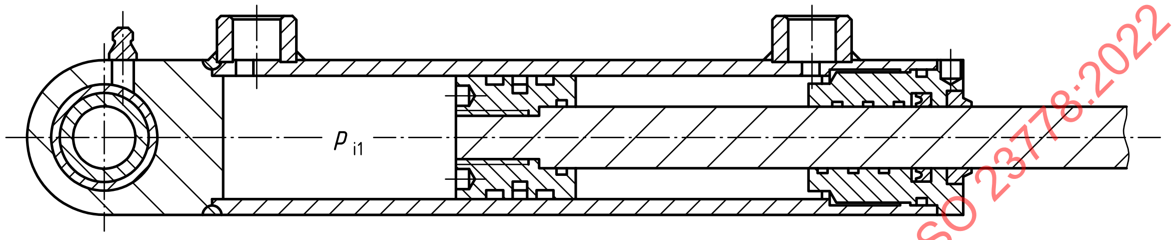
— cylinder tube and rod mounting to the machine;

— forces/stresses due to thread pre-tightening;

— direction of gravity.

Different pressurized cylinder arrangements shall be considered when calculating static strength for cylinders.

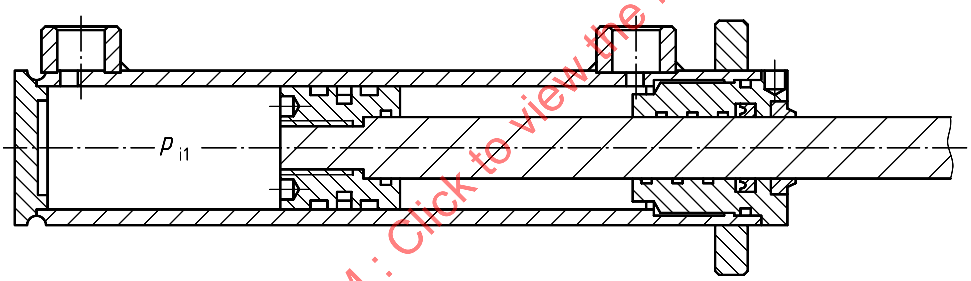
Typical pressurized cylinder arrangements are shown in Figure 6 to Figure 10.



Key

p_{i1} pressure in piston side chamber

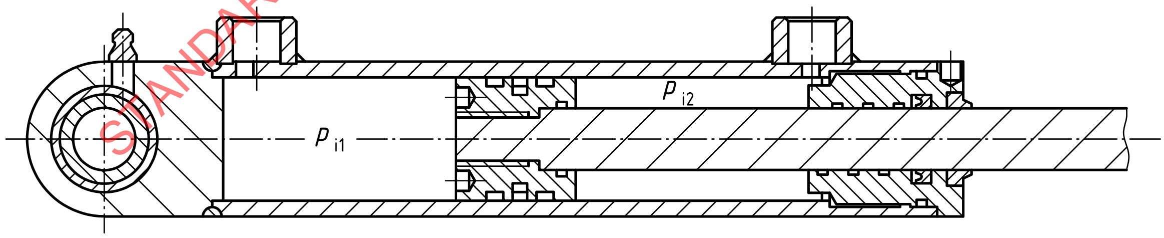
Figure 6 — Pushing cylinder with supported bottom



Key

p_{i1} pressure in piston side chamber

Figure 7 — Pushing cylinder, flange mounted with unsupported bottom

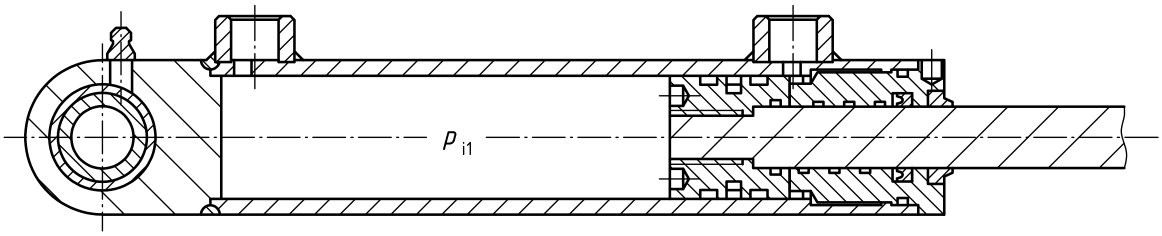


Key

p_{i1} pressure in piston side chamber

p_{i2} pressure in rod side chamber

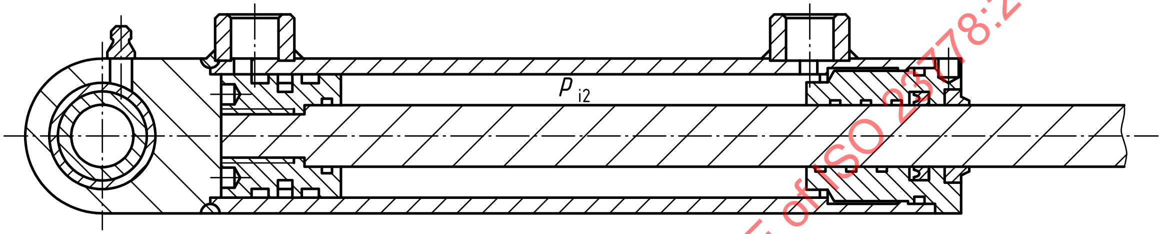
Figure 8 — Pulling cylinder or pushing cylinder with pressurized rod chamber



Key

p_{i1} pressure in piston side chamber

Figure 9 — Pushing cylinder at end of stroke



Key

p_{i2} pressure in rod side chamber

Figure 10 — Pulling cylinder at end of stroke

The worst load condition or combination shall be used when calculating stresses σ_{Sd} or $\sigma_{w,Sd}$ for a feature.

6.3.3 Cylinder tube

Cylinder tube stresses, see Figure 11, shall be computed from three components. For calculation of each component, forces and pressures shall be determined in accordance with 6.3.2.

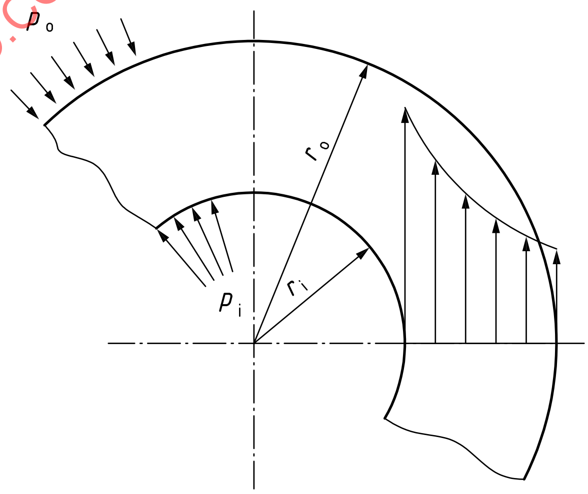


Figure 11 — Stresses in cylinder tube

The tangential stress (hoop stress) is given by Formula (6):

$$\sigma_t(r) = p_i \cdot \frac{\left(\frac{r_o}{r}\right)^2 + 1}{\left(\frac{r_o}{r_i}\right)^2 - 1} + p_o \cdot \frac{\left(\frac{r_i}{r}\right)^2 + 1}{\left(\frac{r_i}{r_o}\right)^2 - 1} \tag{6}$$

For cylindrical shells such as tubes or hollow rods that are also loaded by an outer pressure, the combination of inner and outer pressure that gives the largest absolute value of the tangential (hoop) stress shall be used.

Maximum radial stress magnitude in the tube occurs at the inner radius r_i or the at the outer radius r_o and is given by Formula (7):

$$\sigma_r = -p_i \text{ or } \sigma_r = -p_o \tag{7}$$

For the cylinder arrangement shown in Figure 6, maximum axial stress in the tube is given by Formula (8):

$$\sigma_a = M_b \cdot \frac{4 \cdot r_o}{\pi \cdot \left(r_o^4 - r_i^4\right)} \tag{8}$$

For the cylinder arrangements shown in Figure 8 and Figure 10, maximum axial stress in the tube is given by Formula (9):

$$\sigma_a = \frac{p_{i2} \cdot \left(r_i^2 - r_r^2\right)}{r_o^2 - r_i^2} + M_b \cdot \frac{4 \cdot r_o}{\pi \cdot \left(r_o^4 - r_i^4\right)} \tag{9}$$

For the cylinder arrangement shown in Figure 7 and Figure 9, maximum axial stress in the tube is given by:

$$\sigma_a = \frac{p_{i1} \cdot r_i^2}{r_o^2 - r_i^2} + M_b \cdot \frac{4 \cdot r_o}{\pi \cdot \left(r_o^4 - r_i^4\right)} \tag{10}$$

where

r is an arbitrary radius of the tube;

r_i is the inner radius of the tube;

r_o is the outer radius of the tube;

r_r is the outer radius of the piston rod;

p_i is the inner pressure;

p_{i1} is the inner maximum pressure in piston side chamber;

p_{i2} is the inner maximum pressure in rod side chamber;

p_o is the outer pressure;

M_b is any bending moment acting on the cylinder tube (e.g. dead weight).

The von Mises equivalent stress shall be computed for the location having the most severe stress as:

$$\sigma_{Sd} = \sqrt{\sigma_t^2 + \sigma_r^2 + \sigma_a^2 - \sigma_t\sigma_a - \sigma_t\sigma_r - \sigma_r\sigma_a} \tag{11}$$

6.3.4 Cylinder bottom

6.3.4.1 Bottom plate

The stress in an unsupported bottom plate, see Figure 12, in a cylinder with the ratio outer diameter to inner diameter in the range 1,07 to 1,24, shall be calculated as:

$$\sigma_{Sd} = p_i \cdot \left[\left(\frac{341}{350} - \frac{3}{7} \cdot \frac{D + 2 \cdot t}{D} \right) \cdot \frac{D}{h} \right]^2 \quad (12)$$

where

- p_i is the inner pressure;
- D is the inner diameter;
- t is the tube thickness;
- h is the bottom thickness.

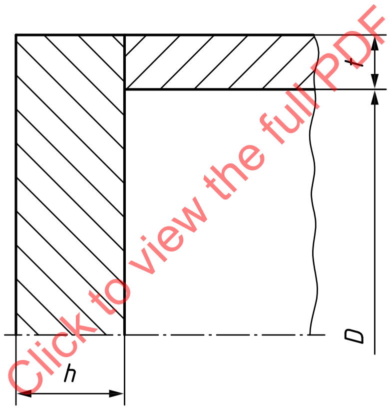


Figure 12 — Stresses in unsupported cylinder bottom

6.3.4.2 Bottom weld

Bottom welds, see Figure 13, shall be calculated for different load cases in accordance with 6.3.2.

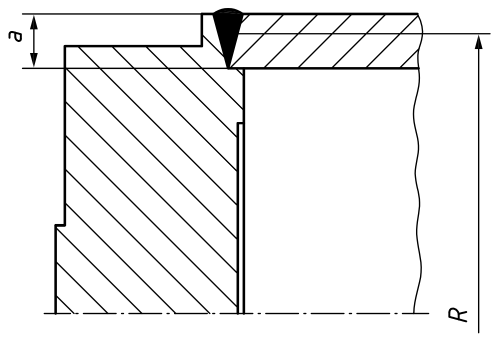


Figure 13 — Bottom weld

The bottom weld is loaded by the axial force in the tube, caused by internal pressure (see Figure 7 and Figure 8) or by pushing cylinder coming to end of stroke (see Figure 9).

$$\sigma_{w,Sd} = \frac{F_{Sdt}}{2 \cdot \pi \cdot R \cdot a} \quad (13)$$

where

F_{Sdt} is the design axial force acting in the tube;

a is the effective throat thickness of the weld, see ISO 20332:2016, Annex C;

R is the middle radius of the weld.

6.3.5 Piston rod welds

Piston rod welds shall be calculated for different load cases according to 6.3.2, in the same way as the calculation of bottom welds.

$$\sigma_{w,Sd} = \frac{F_{Sdw}}{2 \cdot \pi \cdot R \cdot a} \quad (14)$$

where

F_{Sdw} is the maximum design force acting in the rod;

a is the effective throat thickness of the weld, see ISO 20332:2016, Annex C;

R is the middle radius of the weld.

6.3.6 Cylinder tube and piston rod threads

Stresses in cylinder tube threads and piston rod threads shall be calculated for the different load cases in accordance with 6.3.2. The design stress shall be computed as:

$$\sigma_{Sd} = \frac{F_{Sdr}}{A_s} \quad (15)$$

$$\tau_{Sd} = \frac{2 \cdot F_{Sdr}}{\pi \cdot L \cdot d_2} \quad (16)$$

where

F_{Sdr} is the maximum design force acting on the cylinder head or the piston rod head;

A_s is the stress area of the threaded cylinder tube or piston rod (equivalent to stress area of bolt or nut);

L is the effective threaded length, maximum $0{,}9 \cdot d_2$;

d_2 is the pitch diameter of the thread in accordance with ISO 724.

It should be considered that the tube diameter can increase due to the internal pressure and thus decrease the shear area in Formula (16).

6.3.7 Thread undercuts and locking wire grooves

Stresses in thread undercuts or locking wire grooves (see Figure 14) shall be calculated for the different load cases in accordance with 6.3.2.

The design stress shall be computed as:

$$\sigma_{Sd} = \frac{F_{Sdu}}{A_c} \tag{17}$$

where

F_{Sdu} is the maximum design force acting at the undercut;

A_c is the critical stress area at the undercut or locking wire groove.

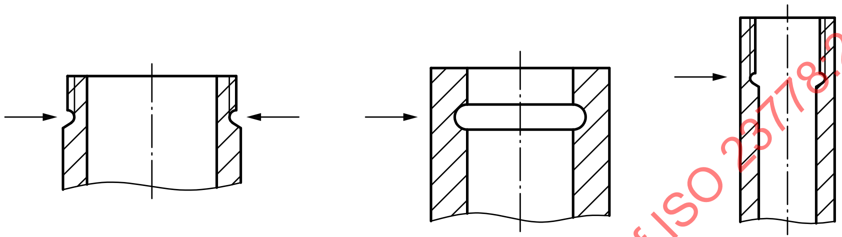


Figure 14 — Undercuts at thread run out

6.3.8 Oil connector welds

This subclause considers oil connectors welded to the tube (see Figure 15). The design stress $\sigma_{w,Sd}$ shall be computed as:

$$\sigma_{w,Sd} = \frac{F_{Sdo}}{A} \tag{18}$$

with

$$A = \pi \cdot D_w \cdot a \tag{19}$$

and

$$F_{Sdo} = \frac{p_{Sd} \cdot \pi \cdot D_p^2}{4} \tag{20}$$

where

p_{Sd} is the design pressure for chamber side;

D_p is the pressure affected diameter;

a is the effective throat thickness of the weld, see ISO 20332:2016, Annex C;

D_w is the effective weld diameter.

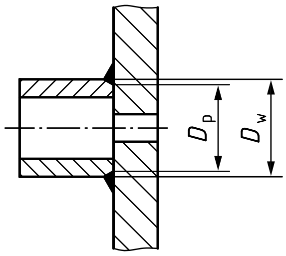


Figure 15 — Welded oil connector

6.3.9 Connecting interfaces to crane structure

The design stresses in parts connecting the cylinder to the crane structure shall be calculated in accordance with ISO 20332.

6.4 Nonlinear stress analysis

6.4.1 General

Nonlinear stress analysis takes into account the force balance in the deformed shape of the cylinder and can be governing when the compressive force acts together with bending moment or lateral force, or due to the angular misalignment α between rod and tube caused by the play at the guide rings. Nonlinear stress analysis may be omitted if lateral forces and bending moments are negligible, and if the maximum displacement δ_{max} due to the angular misalignment α is smaller than $L/600$, where L is the overall length of the cylinder. If the misalignment is unknown, δ_{max} shall be set to $L/300$. The omission of a second order analysis shall be justified.

In particular, the cases described in 6.4.2 and 6.4.3 may require nonlinear stress analysis. The nonlinear stress analyses may either be made with FE-analysis or by the analytical formulae given in Annex B.

6.4.2 Standard cylinder with end moments

Figure 16 shows a standard cylinder with the same configuration as in buckling case D (see 8.2), loaded by a compressive force F and by moments M_1 and M_2 caused by axle frictions acting at the bushings at the cylinder's ends, and with an angular misalignment α between the cylinder tube and the piston rod caused by play at guide rings.

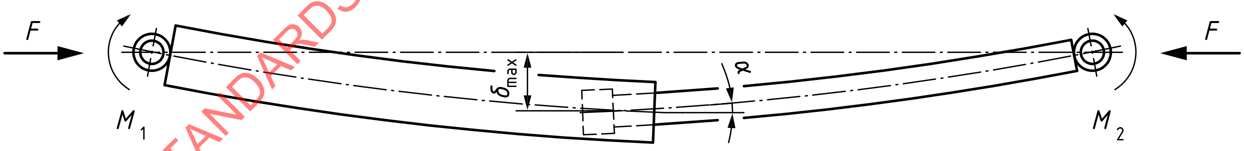


Figure 16 — Cylinder with end moments from axle frictions and angular misalignment

6.4.3 Support leg

Figure 17 shows a support leg cylinder loaded by a compressive force F_A and by a lateral force F_S, and with an angular misalignment α between the cylinder tube and the piston rod caused by play at guide rings.

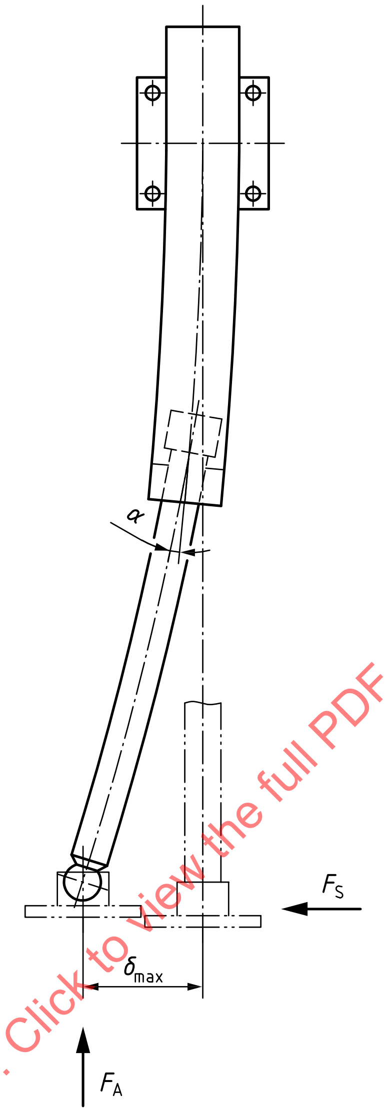


Figure 17 — Support leg cylinder with lateral force and angular misalignment

6.5 Execution of the proof

6.5.1 Proof for load bearing components

For the load bearing components (e.g. tube, rod, lugs) it shall be proven that:

$$\sigma_{Sd} \le f_{Rd\sigma} \quad \text{and} \quad \tau_{Sd} \le f_{Rd\tau} \tag{21}$$

where

σ_{Sd}	is the design normal stress or the von Mises equivalent stress;
τ_{Sd}	is the design shear stress;
$f_{Rd\sigma}$, $f_{Rd\tau}$	are the corresponding limit design stresses in accordance with 6.2.2.

6.5.2 Proof for bolted connections

Bolted connections shall be proofed in accordance with ISO 20332.

6.5.3 Proof for welded connections

For the weld it shall be proven that:

$$\sigma_{w,Sd} \leq f_{w,Rd} \tag{22}$$

where

$\sigma_{w,Sd}$ is the design weld stress;

$f_{w,Rd}$ is the limit design weld stress in accordance with ISO 20332.

7 Proof of fatigue strength

7.1 General

The proof of fatigue strength is intended to prevent risk of failure due to formation and propagation of critical cracks in load carrying part of a hydraulic cylinder under cyclic loading.

For the execution of the proof of fatigue strength, the cumulative damages caused by variable stress cycles shall be calculated. In this standard, Palmgren-Miner's rule of cumulative damage is reflected by use of the stress history parameters (see ISO 20332).

The fatigue strength specific resistance factor γ_{mf} is as defined in ISO 20332.

The limit design stress of a constructional detail is characterized by the value of the characteristic fatigue strength $\Delta\sigma_c$, which represents the fatigue strength at $2 \cdot 10^6$ cycles under constant stress range loading and with a probability of survival equal to P_S 97,7 % (see ISO 20332).

$\Delta\sigma_c$-values depend on the quality level of the weld. Quality levels shall be in accordance with ISO 5817:2014, Annex C. Weld quality lower than weld quality class C shall not be used.

Fatigue testing may be used to establish $\Delta\sigma_c$-values for details deviating from those given here below, or to prove higher $\Delta\sigma_c$-values than those given here. Such fatigue testing shall be done in accordance with ISO 20332.

7.2 Stress histories

The stress history is a numerical presentation of all stress variations that are significant for fatigue. Stress histories shall be determined either through stress calculations or measurements, in both cases simulating the loading imposed on the cylinder. The classification of load cycles in the ISO 4301 series can be used when estimating the number of relevant stress cycles.

For the proof of fatigue strength, stress histories are expressed as single-parameter representations of frequencies of occurrence of stress ranges by using methods such as the hysteresis counting method (Rain flow or Reservoir method) with the influence of mean stress neglected.

Each of the stress ranges is sufficiently described by its upper and lower extreme value.

$$\Delta\sigma = \sigma_u - \sigma_b \tag{23}$$

where

σ_u is the upper extreme value of a stress range;

σ_b is the lower extreme value of a stress range;

$\Delta\sigma$ is the stress range.

Stress history parameter s_3 is calculated as follows, based on a one-parameter presentation of stress histories during the design life of the cylinder:

$$s_3 = v \cdot k_3 \tag{24}$$

where

$$k_3 = \sum_i \left[\frac{\Delta\sigma_i}{\Delta\hat{\sigma}}\right]^3 \cdot \frac{n_i}{N_t} \tag{25}$$

$$v = \frac{N_t}{N_{ref}} \tag{26}$$

where

v	is the relative total number of occurrences of stress ranges;
k_3	is the stress spectrum factor;
$\Delta\sigma_i$	is the stress range i;
$\Delta\hat{\sigma}$	is the maximum design stress range;
n_i	is the number of occurrences of stress range i;
$N_t = \sum_i n_i$	is the total number of occurrences of stress ranges during the design life of the cylinder;
N_{ref} $2 \cdot 10^6$	is the reference number of cycles.

Depending on which part of a cylinder is considered, the stress range is proportional to either the external force range or the pressure range in either chamber. Therefore, the stress ranges in Formula (26) can be substituted with the corresponding force ranges ΔF or pressure ranges Δp.

In general, the stress history parameter s_3 has different values for different parts of a cylinder. These values are related to the duty and decisively depend on either:

— the number of working cycles and external force spectrum;

— the number of pressure cycles and related pressure spectrum in piston side chamber; or

— the number of pressure cycles and related pressure spectrum in rod side chamber.

For thermally stress relieved or non-welded components, the compressive portion of the stress range may be reduced to 60 %.

Different parts of cylinders may be arranged into classes S of the stress history parameter s_m. The classification is based upon m = 3 and is specified in Table 2. When a class S is referred to in the proof of fatigue strength for a cylinder part, the value of the stress history parameter s_3 shall be taken in accordance with Table 2. Proof of competence for fatigue may be omitted when the value of the stress history parameter s_3 is lower than 0,001.

Table 2 — Classes S of stress history parameter s_3

Class	*S*02	*S*01	*S*0	*S*1	*S*2	*S*3	*S*4	*S*5	*S*6	*S*7	*S*8	*S*9
s_3	0,002	0,004	0,008	0,016	0,032	0,063	0,125	0,25	0,5	1,0	2,0	4,0

When a single stress history class S is used to characterize a cylinder, the most severe class occurring within the cylinder shall be used.

7.3 Execution of the proof

For the detail under consideration, it shall be proven that:

$$\Delta\sigma_{Sd} \leq \Delta\sigma_{Rd} \tag{27}$$

$$\Delta\sigma_{Sd} = \max\sigma - \min\sigma \tag{28}$$

where

$\Delta\sigma_{Sd}$	is the design stresses range (the same as $\Delta\hat{\sigma}$ in 7.2);
max σ, min σ	are the extreme values of design stresses (compression stresses with negative sign);
$\Delta\sigma_{Rd}$	is the limit design stress range.

7.4 Limit design stress range

The limit design stress range is given by:

$$\Delta\sigma_{Rd} = \frac{\Delta\sigma_c}{\gamma_{mf} \cdot \sqrt[m]{s_3}} \tag{29}$$

where

$\Delta\sigma_{Rd}$	is the limit design stress range;
$\Delta\sigma_c$	is the characteristic fatigue strength;
γ_{mf}	is the fatigue strength specific resistance factor (see ISO 20332);
s_3	is the stress history parameter;
m	is the slope of the log $\Delta\sigma$ – log N-curve.

For the case of $m > 3$, Formula (29) is a conservative simplification. With knowledge of the actual stress spectrum, a more detailed calculation may be done in accordance with ISO 20332.

7.5 Details for consideration

7.5.1 General

This subclause deals with details where fatigue can occur and that can be relevant for the cylinder under consideration. The characteristic fatigue strengths are given for commonly used designs. For other details or for deviating conditions, other recognized sources or fatigue testing should be used.

7.5.2 Bottom weld

The cylinder bottom can either be supported or unsupported, see Figure 18. The bottom weld also transfers the axial load in the unsupported case.

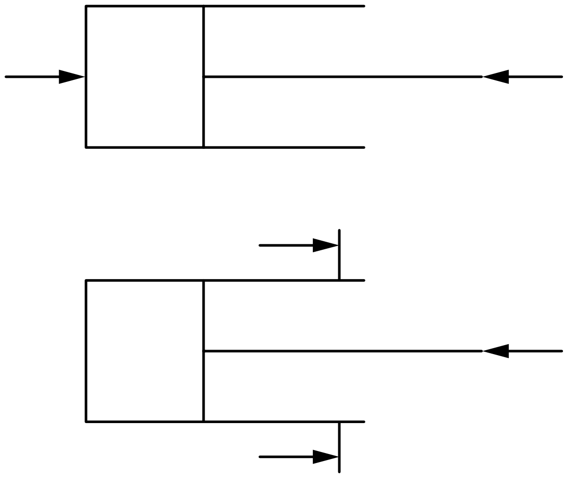


Figure 18 — Cylinder bottom, supported (upper) and unsupported (lower)

For the purpose of stress relieving the bottom weld, there may be a distance x between the cylinder bottom and the weld, see Figure 19. In the case without stress relieving, the distance x is set to zero.

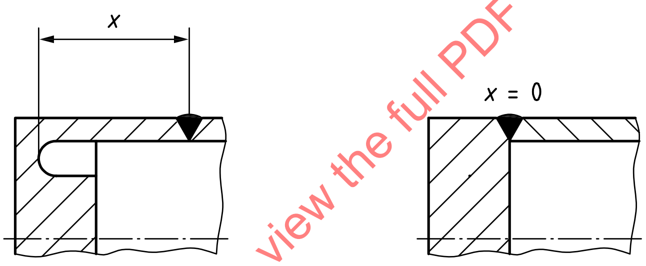


Figure 19 — Bottom weld

The shell section transverse force T_0 (force per length) and the shell section bending moment M_0 (moment per length) act at the intersection between the cylinder tube and the bottom (see Annex C). For the basic case of a bottom with constant thickness, there are two sets of formulae for T_0 and M_0 depending on whether the bottom is assumed to be supported by a constant pressure or unsupported. Formula (30) and Formula (31) give T_0 and M_0 for cylinders with supported bottom, whereas Formula (34) and Formula (35) give T_0 and M_0 for cylinders with unsupported bottom.

For supported cylinder bottom, the shell section transverse force T_0 and the shell section bending moment M_0 are given by:

$$T_0 = \frac{\rho \cdot r_i \cdot \left(\frac{1+\nu}{\kappa \cdot R} \cdot \rho^3 + 1 \right)}{\Omega} \cdot \Delta p_{Sd} \quad (30)$$

$$M_0 = \frac{\frac{\rho^2 \cdot r_i}{2} \cdot \left(\frac{1+\nu}{\kappa^2 \cdot R} \cdot \rho^2 - t \right)}{\Omega} \cdot \Delta p_{Sd} \quad (31)$$

For cylinders where the stiffness of the bottom is much higher than the stiffness of the tube, i.e. $h \gg t$, Formula (30) and Formula (31) may be well approximated by Formula (32) and Formula (33). As the approximated formulae yield conservative results at $x = 0$ when $h \geq t$, they may also be used in that case.

$$T_0 = \frac{r_i}{\kappa \cdot R} \cdot \Delta p_{Sd} \quad (32)$$

$$M_0 = \frac{T_0}{2 \cdot \kappa} \tag{33}$$

For unsupported cylinder bottom, the shell section transverse force T_0 and the shell section bending moment M_0 are given by:

$$T_0 = \frac{\rho \cdot \left[2 \cdot \left(\frac{1+\nu}{\kappa \cdot R} \cdot \rho^3 + 1 \right) \cdot \left(2 - \frac{\nu \cdot r_\mathrm{i}}{R} \right) \cdot r_\mathrm{i} + (h \cdot \kappa + 1) \cdot \kappa^2 \cdot R^3 \right]}{4 \cdot \Omega} \cdot \Delta p_\mathrm{Sd} \tag{34}$$

$$M_0 = \frac{\left(\frac{1+\nu}{\kappa^2 \cdot R} \cdot \rho^2 - t \right) \cdot \left(2 - \frac{\nu \cdot r_\mathrm{i}}{R} \right) \cdot \rho^2 \cdot r_\mathrm{i} + \left(\frac{h \cdot \kappa}{2} + 1 \right) \cdot \rho \cdot \kappa \cdot R^3 + \frac{1-\nu}{2} \cdot R^2}{4 \cdot \Omega} \cdot \Delta p_\mathrm{Sd} \tag{35}$$

where

$\Omega = (1+\nu) \cdot \rho^4 + \left(\frac{4}{3} \cdot h^2 \cdot \kappa^2 + 2 \cdot h \cdot \kappa + 2 \right) \cdot \rho \cdot \kappa \cdot R + 1 - \nu;$

$\kappa = \sqrt[4]{\frac{3 \cdot (1-\nu^2)}{t^2 \cdot R^2}};$

$\rho = \frac{h}{t};$

r_i is the inner radius of the tube;

R is the middle radius of the tube (i.e. $R = r_\mathrm{i} + t/2$);

t is the wall thickness of the tube;

h is the thickness of the cylinder bottom;

ν is Poisson's ratio (ν = 0,3 for steel);

Δp_Sd is the design pressure range on piston side.

For some more complex cases, T_0 and M_0 can be obtained by solving the formula systems given in Annex D.

The bending stress range $\Delta\sigma_\mathrm{b}(x)$ at the distance x from the cylinder bottom is given by:

$$\Delta\sigma_\mathrm{b}(x) = \pm 6 \cdot \mathrm{e}^{-\kappa \cdot x} \cdot \frac{M_0 \cdot (\sin(\kappa \cdot x) + \cos(\kappa \cdot x)) - \frac{T_0}{\kappa} \cdot \sin(\kappa \cdot x)}{t^2} \tag{36}$$

where the plus sign denotes the inside of the tube and the minus sign denotes the outside of the tube.

The membrane stress range $\Delta\sigma_\mathrm{m}$ depends on the bottom support and is given by:

$$\text{Supported bottom}: \Delta\sigma_\mathrm{m} = 0 \tag{37}$$

Unsupported bottom: $\Delta\sigma_{\text{m}} = \frac{r_{\text{i}}^2}{2 \cdot R \cdot t} \cdot \Delta p_{\text{Sd}}$

The total stress range $\Delta\sigma_{\text{Sd}}$ at the weld location x shall, for the outside of the cylinder tube, be computed as:

$$\Delta\sigma_{\text{Sd}}(x) = \Delta\sigma_{\text{m}} - \Delta\sigma_{\text{b}}(x) \tag{38}$$

The total stress range $\Delta\sigma_{\text{Sd}}$ at the weld location x shall, for the inside of the cylinder tube, be computed as:

$$\Delta\sigma_{\text{Sd}}(x) = \Delta\sigma_{\text{m}} + \Delta\sigma_{\text{b}}(x) \tag{39}$$

The following characteristic fatigue strengths with $m = 3$ shall be used:

Outside tube, weld toe in quality C:	$\Delta\sigma_{\text{c}} = 100$ MPa;
Outside tube, weld toe in quality B:	$\Delta\sigma_{\text{c}} = 112$ MPa;
Inside tube, weld root:	$\Delta\sigma_{\text{c}} = 71$ MPa.

The design stress range $\Delta\sigma_{\text{Sd}}$ may additionally be computed using a FE-analysis model for increased accuracy by applying one of the methods described in References [1] and [3].

7.5.3 Notch stress at oil connectors

Figure 20 illustrates an oil connector welded with all around fillet weld in quality C. The design stress range $\Delta\sigma_{\text{Sd}}$ is based on the pressure range at the oil connector's cylinder end, and, assuming that the diameter is much greater than the wall thickness, shall be computed as:

$$\Delta\sigma_{\text{Sd}} = \frac{\Delta p_{\text{Sd}} \cdot D}{2 \cdot t} \tag{40}$$

where

Δp_{Sd} is the design pressure range for that cylinder end;

D is the tube inner diameter;

t is the thickness of the cylinder tube.

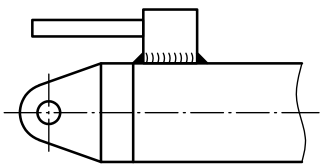


Figure 20 — Oil connector on piston side

The characteristic fatigue strength $\Delta\sigma_{\text{c}} = 80$ MPa with $m = 3$ shall be used.

The same calculation method shall be used for oil connectors both on piston side and rod side.

7.5.4 Cylinder head

7.5.4.1 General

The design stress range $\Delta\sigma_{Sd}$ is based on the force range ΔF resulting from the rod side's pressure range. The force range ΔF shall be computed as:

$$\Delta F_{Sd} = \frac{\Delta p_{Sd} \cdot \pi \cdot \left(D^2 - d^2\right)}{4} \quad (41)$$

where

Δp_{Sd} is the design pressure range on rod side;

D is the piston diameter;

d is the rod diameter.

7.5.4.2 Tube thread

This subclause deals with end of cylinder tube with machined threads, see Figure 21. The design stress range $\Delta\sigma_{Sd}$ shall be computed as:

$$\Delta\sigma_{Sd} = \frac{\Delta F_{Sd}}{A_s} \quad (42)$$

where

ΔF_{Sd} is the design force range acting on the cylinder head;

A_s is the stress area of the threaded tube.

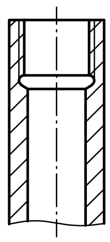


Figure 21 — Tubes with inner and outer threads

The characteristic fatigue strength (in MPa) for m = 3 is given by Formula (43):

$$\Delta\sigma_c = 44{,}5 + \frac{5{,}5}{1 - \frac{d_s}{D_s}}, \quad \max(\Delta\sigma_c) = 100 \quad (43)$$

where

d_s is the thread stress diameter for inner thread and inner tube diameter for outer thread;

D_s is the outer tube diameter for inner thread and thread stress diameter for outer thread.

7.5.4.3 Tube thread undercut

This subclause deals with half circular undercut at end of cylinder tube threads, see Figure 22. The design stress range $\Delta\sigma_{Sd}$ shall be computed as:

$$\Delta\sigma_{Sd} = \frac{\Delta F_{Sd}}{A} \tag{44}$$

where

ΔF_{Sd} is the design force range acting on the top nut;

A is the smallest stress area of tube at the undercut.

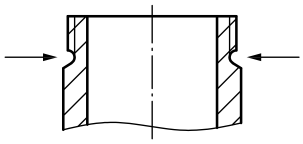


Figure 22 — Undercut for tube thread

The characteristic fatigue strength (in MPa) for m = 5 shall be computed as:

$$\Delta\sigma_c = 80 + 0{,}12 \cdot f_y \tag{45}$$

where f_y is the yield strength in MPa.

Formula (45) requires that the bottom radius of the undercut be at least 35 % of the undercut depth.

7.5.4.4 Locking wire groove

This subclause deals with stress concentration at locking wire groove, see Figure 23. The design stress range $\Delta\sigma_{Sd}$ is based on nominal stress at remaining area, and shall be computed as:

$$\Delta\sigma_{Sd} = \frac{\Delta F_{Sd}}{A} \tag{46}$$

where

ΔF_{Sd} is the design force range acting on the top nut;

A is the smallest stress area of the tube at the groove.

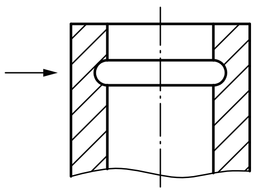


Figure 23 — Locking wire groove

The characteristic fatigue strength (in MPa) for m = 5 shall be computed as:

$$\Delta\sigma_c = 80 + 0{,}12 \cdot f_y \quad (47)$$

where f_y is the yield strength in MPa.

Formula (47) requires that the bottom radius of the undercut be at least 35 % of the undercut depth.

7.5.5 Piston rod

7.5.5.1 General

The design stress range $\Delta\sigma_{Sd}$ where the piston rod head and the piston are connected to the piston rod is based on the force range ΔF acting on the piston rod.

7.5.5.2 Piston rod threads

This subclause deals with end of piston rod with machined threads, see Figure 24. The design stress ranges $\Delta\sigma_{Sd}$ at the threads on the piston rod shall be computed as:

$$\Delta\sigma_{Sd} = \frac{\Delta F_{Sd}}{A_s} \quad (48)$$

where

ΔF_{Sd} is the design force range acting on the piston rod thread;

A_s is the stress area of the threaded piston rod.

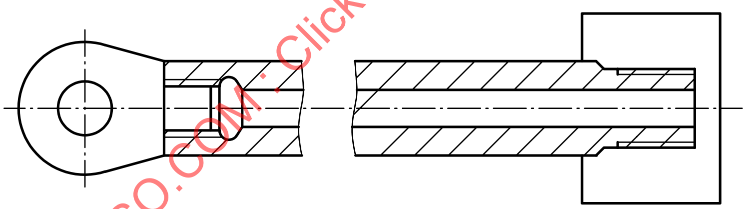


Figure 24 — Threads on piston rod

The characteristic fatigue strength (in MPa) for m = 3 is given by Formula (49):

$$\Delta\sigma_c = 44{,}5 + \frac{5{,}5}{1 - \frac{d_s}{D_s}}, \quad \max(\Delta\sigma_c) = 100 \quad (49)$$

where

d_s is thread stress diameter for inner thread and inner rod diameter for outer thread;

D_s is outer rod diameter for inner thread and stress thread diameter for outer thread.

7.5.5.3 Piston rod thread undercuts

This subclause deals with undercut at end of piston rod threads. The design stress range $\Delta\sigma_{Sd}$ shall be computed as:

$$\sigma_{Sd} = \frac{\Delta F_{Sd}}{A} \tag{50}$$

where

ΔF_{Sd} is the design force range acting on the piston rod;

A is the smallest stress area of the piston rod at the undercut.

The characteristic fatigue strength for m = 5 shall be computed as (in MPa):

$$\Delta\sigma_c = 80 + 0{,}12 \cdot f_y \tag{51}$$

where f_y is the yield strength in MPa.

Formula (51) requires that the bottom radius of the undercut be at least 35 % of the undercut depth.

7.5.5.4 Piston rod welds

This subclause deals with piston welded to rod with fillet weld, groove weld or friction weld, see Figure 25.

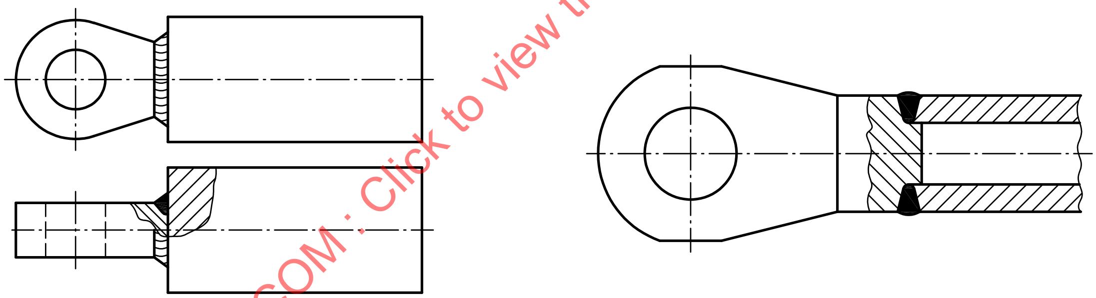


Figure 25 — Piston rod welds, fillet weld (left) and groove weld (right)

The design stress ranges $\Delta\sigma_{Sd}$ at the welds on the piston rod shall be computed as:

$$\sigma_{Sd} = \frac{\Delta F_{Sd}}{A} \tag{52}$$

where

ΔF_{Sd} is the design force range acting on the piston rod;

A is the cross-section area of the weld (rod area in case of friction weld).

The following characteristic fatigue strengths $\Delta\sigma_c$ with m = 3 shall be used:

Fillet weld in quality C	$\Delta\sigma_c$ = 45 MPa
Fillet weld in quality B	$\Delta\sigma_c$ = 50 MPa
Groove weld in quality C	$\Delta\sigma_c$ = 63 MPa

Groove weld in quality B	$\Delta\sigma_c$ = 71 MPa
Friction weld in quality C	$\Delta\sigma_c$ = 80 MPa
Friction weld in quality B	$\Delta\sigma_c$ = 90 MPa

7.5.6 Cylinder head bolts

The fatigue strength of the cylinder head bolts shall be assessed in accordance with ISO 20332.

7.5.7 Cylinder head flange weld

7.5.7.1 General

This subclause deals with flange at end of cylinder tube welded to tube with groove or fillet weld, see Figure 26.

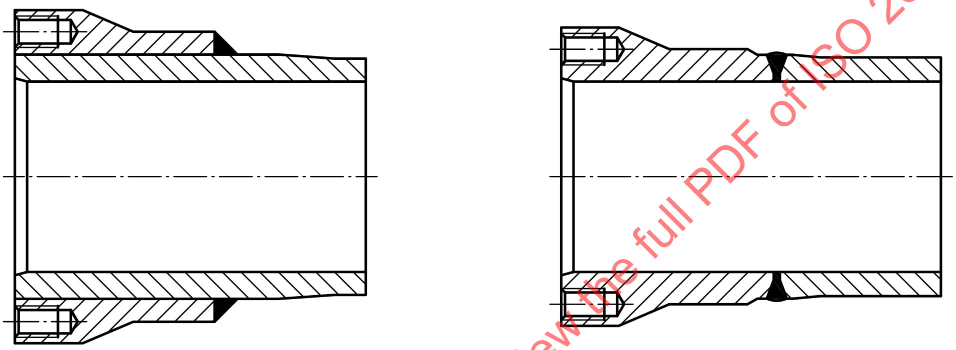


Figure 26 — Cylinder head flange welds, fillet weld (left) and butt weld (right)

A flange weld of a pulling cylinder is subjected to axial shell bending stress from the internal pressure, whereas the flange weld of a pushing cylinder is subjected to axial membrane stress from the axial load at end of stroke. The flange weld of a pulling cylinder shall in addition to axial stress also be assessed for tangential (hoop) stress. At least quality C shall be used for the flange weld and a butt weld shall be fully penetrated.

7.5.7.2 Pulling cylinder, axial stress

With the conservative assumption that the flange is rigid, the shell section transverse force T_0 and the shell section bending moment M_0 acting at the end of the tube are given by:

$$T_0 = \frac{r_i}{\kappa \cdot R} \cdot \Delta p_{Sd} \quad (53)$$

$$M_0 = \frac{T_0}{2 \cdot \kappa} \quad (54)$$

where

$\kappa = \sqrt[4]{\frac{3 \cdot (1-\nu^2)}{t^2 \cdot R^2}}$;

r_i is the inner radius of the tube;

R is the middle radius of the tube (i.e. $R = r_i + t/2$);

t is the wall thickness of the tube;

ν is Poisson's ratio ($\nu = 0{,}3$ for steel);

Δp_{Sd} is the design pressure range on rod side.

The bending stress range $\Delta\sigma_b(x)$ at the distance x into the cylindrical part is given by:

$$\Delta\sigma_b(x) = \left| 6 \cdot e^{-\kappa \cdot x} \cdot \frac{\frac{T_0}{\kappa} \cdot \sin(\kappa \cdot x) - M_0 \cdot (\sin(\kappa \cdot x) + \cos(\kappa \cdot x))}{t^2} \right| \tag{55}$$

The membrane stress range $\Delta\sigma_m$ is given by:

$$\Delta\sigma_m = \frac{r_i^2 - r_o^2}{2 \cdot R \cdot t} \cdot \Delta p_{Sd} \tag{56}$$

where

r_i is the inner radius of the tube;

r_o is the outer radius of the rod;

R is the middle radius of the tube (i.e. $R = r_i + t/2$);

t is the wall thickness of the tube;

Δp_{Sd} is the design pressure range on piston side.

The total design stress range $\Delta\sigma_{Sd}(x)$ at the weld location x shall be computed as:

$$\Delta\sigma_{Sd}(x) = \Delta\sigma_m + \Delta\sigma_b(x) \tag{57}$$

The characteristic fatigue strength $\Delta\sigma_c = 100$ MPa with $m = 3$ shall be used.

The design stress range $\Delta\sigma_{Sd}$ may additionally be computed using a FEM model for increased accuracy by applying one of the methods described in References [1] and [3].

7.5.7.3 Pushing cylinder, axial stress at end of stroke

The axial design stress range $\Delta\sigma_{Sd}$ at the flange weld shall be calculated as:

$$\Delta\sigma_{Sd} = \frac{r_i^2}{2 \cdot R \cdot t} \cdot \Delta p_{Sd} \tag{58}$$

where

r_i is the inner radius of the tube;

R is the middle radius of the tube (i.e. $R = r_i + t/2$);

t is the wall thickness of the tube;

Δp_{Sd} is the design pressure range on piston side.

The characteristic fatigue strength $\Delta\sigma_c = 100$ MPa with $m = 3$ shall be used.

7.5.7.4 Pushing and pulling cylinder, axial stress

For cylinders that are both pulling and pushing, and when both modes require to be considered in the fatigue assessment, the axial stress ranges $\Delta\sigma_{Sd}$ from Formula (57) and Formula (58) may, as a conservative assumption, be added and the sum be taken as an effective design stress range $\Delta\sigma_{Sd}$. A more precise analysis requires knowledge of the actual stress range spectrum at the weld.

7.5.7.5 Pulling cylinder, tangential stress

The tangential design stress range $\Delta\sigma_{Sd}$ at the flange weld shall be calculated as:

$$\Delta\sigma_{Sd} = \frac{R}{t} \cdot \Delta p_{Sd} \tag{59}$$

where

R is the middle radius of the tube (i.e. $R = r_i + t/2$);

t is the wall thickness of the tube;

Δp_{Sd} is the design pressure range on rod side.

The characteristic fatigue strength $\Delta\sigma_c$ = 180 MPa for weld quality B and $\Delta\sigma_c$ = 140 MPa for weld quality C and with m = 3 shall be used.

7.5.8 Mechanical interfaces

The fatigue strength of the mechanical interfaces between the hydraulic cylinder and the rest of the crane structure shall be assessed in accordance with ISO 20332.

8 Proof of elastic stability

8.1 General

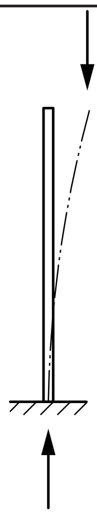
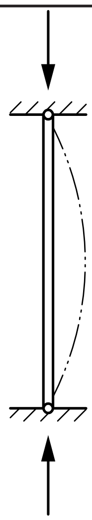
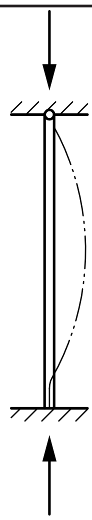
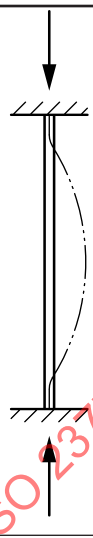
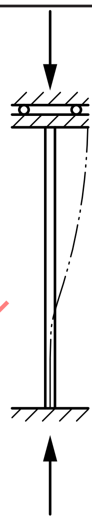
The proof of elastic stability is made to prove that ideally straight cylinders will not lose their stability due to lateral deformation caused solely by compressive forces or compressive stresses. Deformations due to compressive forces or compressive stresses in combination with bending moments caused by external forces or by initial geometric imperfections shall be assessed by the theory of second order (see 6.4) as part of proof of static strength. This clause covers buckling of complete cylinders and internal buckling of piston rods.

As the external compressive design force applied to the cylinder by the crane structure is computed in accordance with ISO 8686, it is already increased by the partial safety factors γ_p and relevant dynamic factors (see also 6.1).

8.2 Critical buckling load

The critical buckling load N_k is the smallest bifurcation load according to elastic theory. For cylinders having only the piston rod loaded in compression, N_k is given in Table 3 for a selection of boundary conditions, also known as Euler's buckling cases.

Table 3 — Critical buckling load N_k for Euler's buckling cases

Euler case no	1	2	3	4	5
Boundary conditions					
N_k	$\frac{\pi^2 \cdot E \cdot I}{4 \cdot L^2}$	$\frac{\pi^2 \cdot E \cdot I}{L^2}$	$\frac{2{,}05 \cdot \pi^2 \cdot E \cdot I}{L^2}$	$\frac{4 \cdot \pi^2 \cdot E \cdot I}{L^2}$	$\frac{\pi^2 \cdot E \cdot I}{L^2}$
E is the modulus of elasticity					
I is the moment of inertia of the member in the plane of the table					
L is the length of the member					

For other boundary conditions or for cylinders consisting of several parts i that are loaded in compression and with different cross-sections, N_k may be computed from the differential formula, or system of differential formulae, of the elastic deflection curve in its deformed state, which has the general solution:

$$y = A_i \cdot \cos(\omega_i \cdot x) + B_i \cdot \sin(\omega_i \cdot x) + C_i \cdot x + D_i , \omega_i = \sqrt{\frac{N}{E \cdot I_i}} \quad (60)$$

where

x is the longitudinal coordinate;

y is the lateral coordinate in the weakest direction of the member;

E is the modulus of elasticity;

i is an index running over the number of cylinder parts that are loaded in compression ($i \geq 1$);

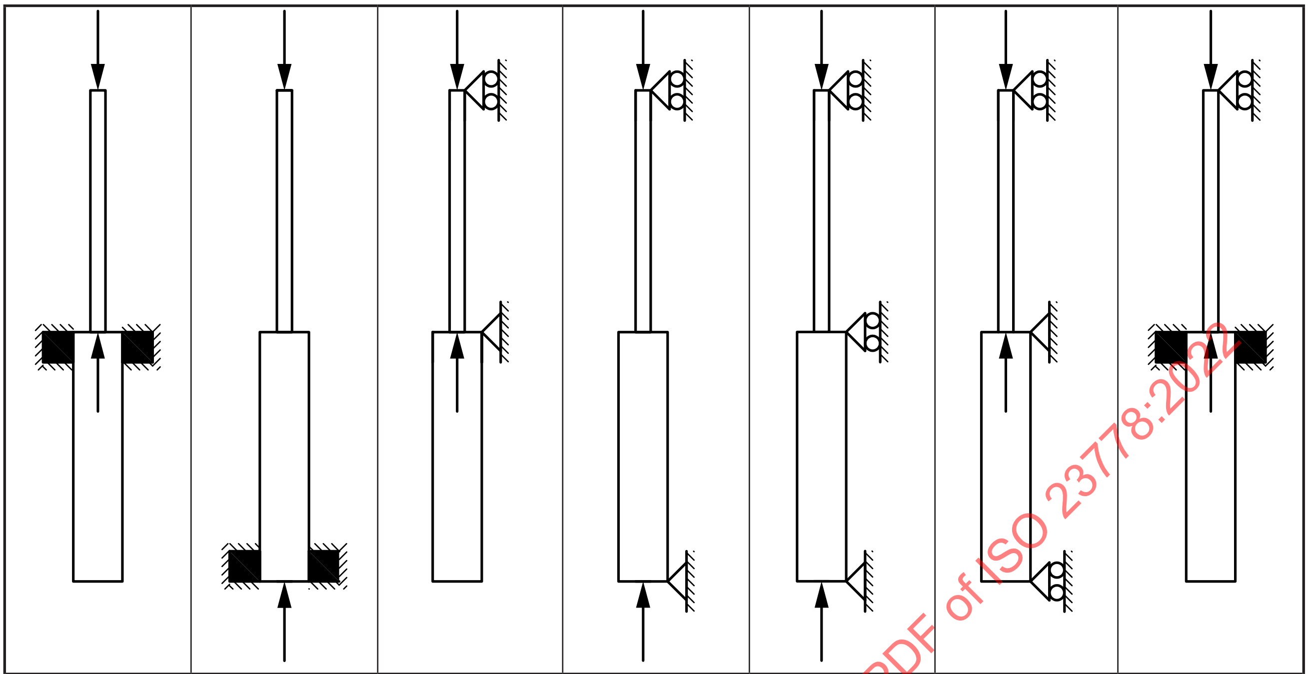
I_i is the moment of inertia of part i in the weakest direction of the member;

N is the compressive force;

A_i, B_i, C_i, D_i are constants to be found by applying appropriate boundary conditions.

The critical buckling load N_k is found as the smallest positive value N that satisfies Formula (61), or system of Formula (61), when solved with the appropriate boundary conditions applied. Formulae for the most common cylinder buckling cases A to G shown in Table 4 are given in Annex A. Alternatively, the critical buckling load N_k may be calculated using FE buckling analysis[2].

Table 4 — Common buckling cases for hydraulic cylinders



A	B	C	D	E	F	G
Regular Euler 1 case	As Euler 1 case, but with two different cross-sections	Regular Euler 2 case	As Euler 2 case, but with two different cross-sections	Two coupled Euler 2 cases	An Euler 2 case that is coupled with a rotational spring	Regular Euler 3 case

For the case when the rod buckles internally inside the cylinder tube, the critical buckling load N_k shall be calculated by using the appropriate Euler case from Table 3. However, a critical buckling load N_k resulting in an external design compressive force F_{Sd} that exceeds the limit design compressive force N_{Rd} may be acceptable if a second order calculation in accordance with 6.4 shows that the design stress does not exceed the limit design stress.

8.3 Limit compressive design force

The limit compressing design force $N_{Rd,i}$ for a cylinder part i is computed from the critical buckling load by:

$$N_{Rd,i} = \frac{\kappa_i \cdot f_{yk,i} \cdot A_i}{\gamma_m} \tag{61}$$

where

κ_i is a reduction factor for the evaluated cylinder part i;

$f_{yk,i}$ is characteristic yield stress of the evaluated cylinder part i;

A_i is the cross-section area of the evaluated cylinder part i;

γ_m is the general resistance factor, γ_m = 1,1 (see ISO 8686-1).

The reduction factor κ_i is computed from the slenderness λ_i, which is given by:

$$\lambda_i = \sqrt{\frac{f_{y,i} \cdot A_i}{N_k}} \quad (62)$$

where N_k is the critical buckling load in accordance with 8.2.

Depending on the value of λ_i, the reduction factor κ_i is given by:

$$\kappa_i = \frac{1}{\xi_i + \sqrt{\xi_i^2 - \lambda_i^2}} \quad \xi_i = 0{,}5 \cdot \left(0{,}96 + 0{,}2 \cdot \lambda_i + \lambda_i^2\right) \quad (63)$$

The overall limit compressing design force N_{Rd} is taken as the minimum value $N_{Rd,i}$ of all parts i:

$$N_{Rd} = \min\left(N_{Rd,i}\right) \quad (64)$$

If there is more than one cylinder part loaded in compression, the following condition shall also apply:

$$N_{Rd} \le \frac{N_k}{1{,}2 \cdot \gamma_m} \quad (65)$$

8.4 Execution of the proof

For the member under consideration, it shall be proven that:

$$F_{Sd} \le N_{Rd} \quad (66)$$

where

F_{Sd} is the compressive external design force;

N_{Rd} is the limit design compressive force according to 8.3.

Annex A
(informative)

Critical buckling load for common buckling cases

A.1 General

Depending on different mounting conditions, the common buckling cases shown in Figure A.1 can be found.

A. Regular Euler 1 case (see 8.2, Table 4).

B. As Euler 1 case, but with two different cross-sections.

C. Regular Euler 2 case (see 8.2, Table 4).

D. As Euler 2 case, but with two different cross-sections.

E. Two coupled Euler 2 cases.

F. An Euler 2 case that is coupled with a rotational spring.

G. Regular Euler 3 case (see 8.2, Table 4).

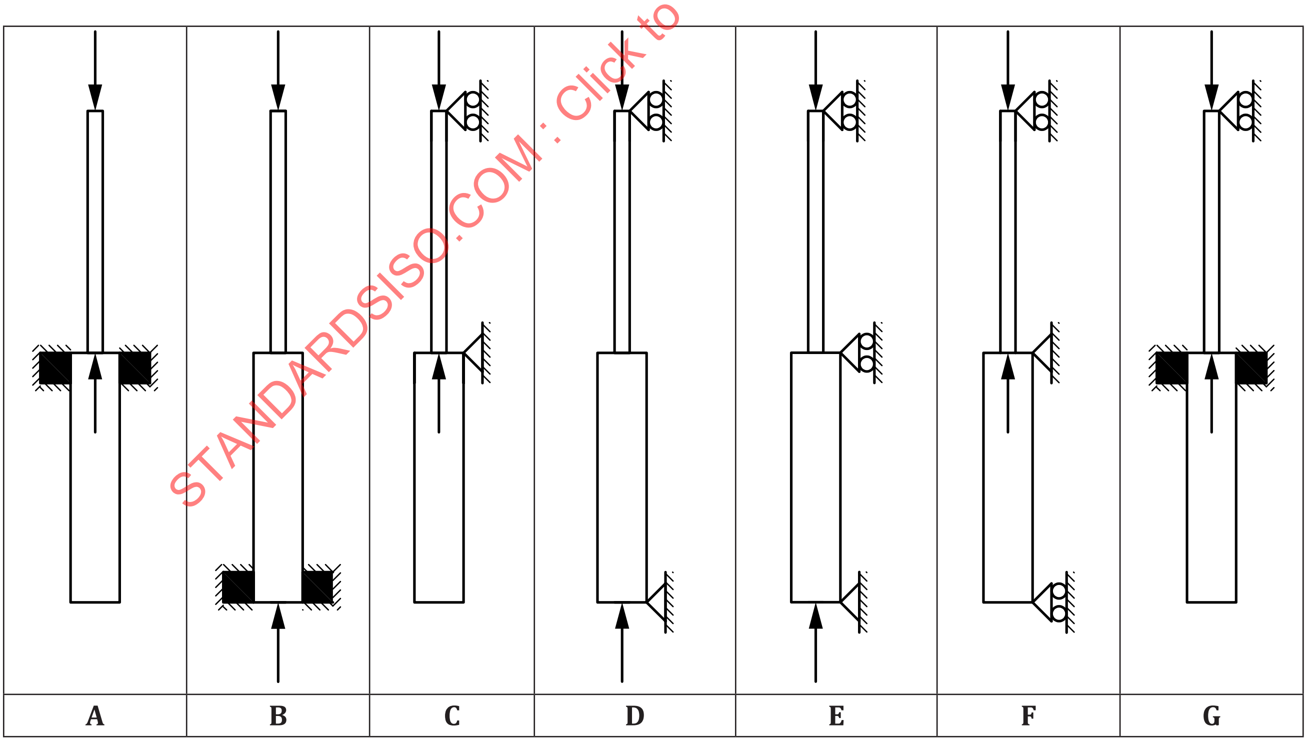


Figure A.1 — Common buckling cases for hydraulic cylinders

The critical buckling loads N_k for buckling cases A to G are obtained by applying appropriate boundary conditions to the differential formula or a system of differential formulae given in 8.2. The following part of this annex provides the relevant formulae that yield the critical buckling loads N_k for buckling cases A to G (Figure A.2 to Figure A.8), where:

N is the compressive force;

L_1 is the effective buckling length of the cylinder tube;

L_2 is the effective buckling length of the piston rod;

I_1 is the moment of inertia of the cylinder tube;

I_2 is the moment of inertia of the piston rod;

E is the elastic modulus of steel;

and where:

$$\omega_i = \sqrt{\frac{N}{E \cdot I_i}}, \qquad i = 1,\ 2$$

A.2 Buckling case A

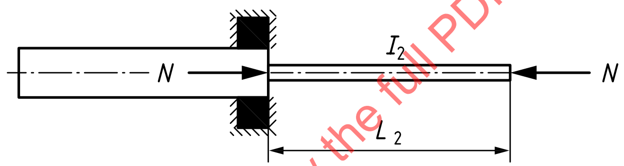


Figure A.2 — Buckling case A (regular Euler 1 case)

Regular Euler 1 case:

$$N_k = \frac{\pi^2 \cdot E \cdot I_2}{4 \cdot L_2^2} \tag{A.1}$$

A.3 Buckling case B

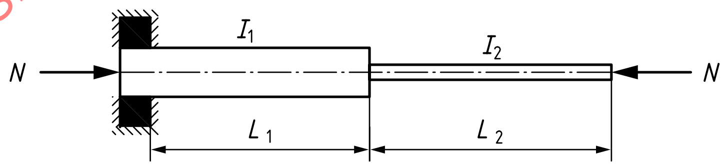


Figure A.3 — Buckling case B (as Euler 1 case, but with two different cross-sections)

N_k is given by solving for the smallest non-zero positive root N of:

$$\cos(\omega_i \cdot L_1) \cdot \cos(\omega_i \cdot L_2) - \sqrt{\frac{I_2}{I_1}} \cdot \sin(\omega_2 \cdot L_2) \cdot \sin(\omega_1 \cdot L_1) = 0 \tag{A.2}$$

A.4 Buckling case C

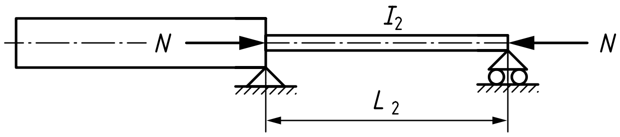


Figure A.4 — Buckling case C (regular Euler 2 case)

Regular Euler 2 case:

$$N_k = \frac{\pi^2 \cdot E \cdot I_2}{L_2^2} \quad (A.3)$$

A.5 Buckling case D

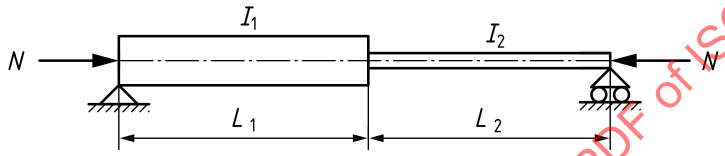


Figure A.5 — Buckling case D (as Euler 2 case, but with two different cross-sections)

N_k is given by solving for the smallest non-zero positive root N of:

$$\cos(\omega_1 \cdot L_1) \cdot \sin(\omega_2 \cdot L_2) + \sqrt{\frac{I_1}{I2}} \cdot \sin(\omega_1 \cdot L_1) \cdot \cos(\omega_2 \cdot L_2) = 0 \quad (A.4)$$

A.6 Buckling case E

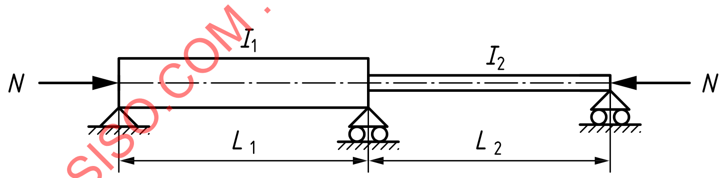


Figure A.6 — Buckling case E (two coupled Euler 2 cases)

N_k is given by solving for the smallest non-zero positive root N of:

$$\frac{L_2}{L_1} \cdot \sin(\omega_2 \cdot L_2) \cdot (\omega_1 \cdot L_1 \cdot \cos(\omega_1 \cdot L_1) - \sin(\omega_1 \cdot L_1)) \ldots$$
$$+ \sin(\omega_1 \cdot L_1) \cdot (\omega_2 \cdot L_2 \cdot \cos(\omega_2 \cdot L_2) - \sin(\omega_2 \cdot L_2)) = 0 \quad (A.5)$$

A.7 Buckling case F

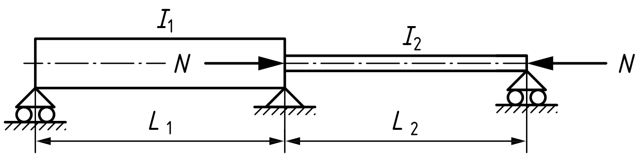


Figure A.7 — Buckling case F (Euler 2 case coupled with a rotational spring)

N_k is given by solving for the smallest non-zero positive root N of:

$$\cos(\omega_2 \cdot L_2)\cdot \omega_2 \cdot L_2 - \sin(\omega_2 \cdot L_2) - \frac{N \cdot L_2}{k_r} \cdot \sin(\omega_2 \cdot L_2) = 0 \qquad (A.6)$$

where the end support stiffness k_r of the rod provided by the cylinder tube is given by:

$$k_r = \frac{3 \cdot E \cdot I_1}{L_1}$$

A.8 Buckling case G

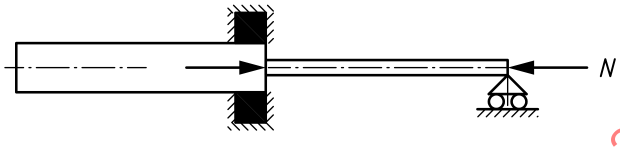


Figure A.8 — Buckling case G (regular Euler 3 case)

Regular Euler 3 case:

$$N_k = \frac{2{,}05 \cdot \pi^2 \cdot E \cdot I_2}{L_2^2} \qquad (A.7)$$

