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Metallic industrial piping - Part 3: Design and calculation

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Metallische industrielle Rohrleitungen - Teil 3: Konstruktion und Berechnung

This draft amendment is submitted to CEN members for enquiry. It has been drawn up by the Technical Committee CEN/TC 267.

This draft amendment A5, if approved, will modify the European Standard EN 13480-3:2017. If this draft becomes an amendment, CEN members are bound to comply with the CEN/CENELEC Internal Regulations which stipulate the conditions for inclusion of this amendment into the relevant national standard without any alteration.

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EUROPEAN COMMITTEE FOR STANDARDIZATION
COMITÉ EUROPÉEN DE NORMALISATION
EUROPÄISCHES KOMITEE FÜR NORMUNG

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European foreword

This document (EN 13480-3:2017/prA5:2021) has been prepared by Technical Committee CEN/TC 267 "Industrial piping and pipelines", the secretariat of which is held by AFNOR.

This document is currently submitted to the CEN Enquiry.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive(s).

For relationship with EU Directive(s), see informative Annex ZA, which is an part of EN 13480-3:2017.

This document includes the text of the amendment itself. The amended/corrected pages of EN 13480-3:2017 will be published as Issue 2 of the new Edition 2022 of the European Standard.

1 Modification to Clause 2, Normative references

Add the following normative reference:

"EN 1993-1-8:2005, *Eurocode 3: Design of steel structures — Part 1-8: Design of joints*".

2 Modification to 3.2, Symbols and units

Table 3.2-1 shall read as follows:

Table 3.2–1 — General symbols and units

Symbol	Description	Unit
A	elongation at rupture	%
E	modulus of elasticity	MPa (N/mm^2)
P_{max}	maximum pressure obtained from the design by formulae or relevant procedures for a given component	MPa (N/mm^2)
PS[a]	maximum allowable pressure	bar
R, r [b]	radii	mm
R_{eH}	upper yield strength at room temperature	MPa (N/mm^2)
$R_{eH\ t}$	upper yield strength at calculation temperature t [c]	MPa (N/mm^2)
R_m	tensile strength at room temperature	MPa (N/mm^2)
$R_{m\ t}$	tensile strength at calculation temperature t [c]	MPa (N/mm^2)
$R_{p0,2}$	0,2 % proof strength at room temperature	MPa (N/mm^2)
$R_{p0,2\ t}$	0,2 % proof strength at calculation temperature t [c]	MPa (N/mm^2)
$R_{p1,0}$	1,0 % proof strength at room temperature	MPa (N/mm^2)
$R_{p1,0\ t}$	1,0 % proof strength at calculation temperature t [c]	MPa (N/mm^2)
S_1	mean value of the stress which leads to a 1 % creep elongation in 100 000 h	MPa (N/mm^2)
S_2	mean value of the stress which leads to a 1 % creep elongation in 200 000 h	MPa (N/mm^2)
$S_{R\ T\ t}$	Mean value of creep rupture strength according to the material standards, for material temperature t, and lifetime T (in hours) under consideration whereby the scatter band does not deviate by more than ± 20 % from the mean value.	MPa (N/mm^2)
T	time	h

t	temperature	°C
TS	maximum allowable temperature	°C
Z	section modulus for a pipe	mm^3
c_0	corrosion or erosion allowance (see Figure 4.3–1)	mm
c_1	absolute value of the negative tolerance taken from the material standard (see Figure 4.3–1)	mm
c_2	thinning allowance for possible thinning during manufacturing process (see Figure 4.3–1)	mm
e_a	analysis thickness of a component used for the check of the strength (see Figure 4.3–1)	mm
e_n	nominal thickness on drawings (see Figure 4.3–1)	mm
e_{ord}	ordered thickness (see Figure 4.3–1)	mm
e_r	minimum required thickness with allowances and tolerances (see Figure 4.3–1)	mm
f	design stress (see Clause 5)	MPa (N/mm^2)
f_{cr}	Design stress in the creep range	MPa (N/mm^2)
f_f	Design stress for flexibility analysis	MPa (N/mm^2)
p_c	calculation pressure (see 4.2.3.4)	MPa (N/mm^2)
p_o	operating pressure (see 4.2.3.1)	MPa (N/mm^2)
t_c	calculation temperature (see 4.2.3.5)	°C
t_o	operating temperature (see 4.2.3.2)	°C
z	joint coefficient (see 4.5)	-
ε	additional thickness resulting from the selection of the ordered thickness (see Figure 4.3–1)	mm

[a] All pressures for calculation purposes are in MPa (N/mm^2) and *PS* is in bar.

[b] The following subscripts apply:

i inside

m mean

o outside

[c] When t is greater than the room temperature.

3 Modification to 4.5, Joint coefficient

At the end of 4.5, the paragraph shall read as follows:

See EN 13480-5:2017, Table 8.3-1. In case of the supply of a welded product, the joint coefficient for the wall thickness calculation should be taken equal to z = 1,0 if the material standard gives the appropriate requirements concerning destructive tests and non-destructive tests (e.g. EN 10217 series).

4 Modification to 6.1, Straight pipes

In 6.1, Formulae (6.1-1) and (6.1-2) shall read as follows:

— where $D_o/D_i \leq 1,7$:

$$e = \frac{p_c \, D_o}{2 f \, z + p_c} \qquad (6.1\text{-}1)$$

or

$$e = \frac{p_c \, D_i}{2 \, f \, z - p_c} \qquad (6.1\text{-}2)$$

5 Modification to 6.4.6.2, Design

The 5th paragraph of 6.4.6.2 shall read as follows:

The required thickness e_2 of the cone adjacent to the junction is the greater of e_{con} and e_j. This thickness shall be maintained for a distance of at least $1,4 l_2$ from the junction along the cone, see Figure 6.4.2-1.

6 Modification to 6.4.7.2, Design

The 6th paragraph of 6.4.7.2 shall read as follows:

The required thickness e_2 of the knuckle and the cone adjacent to the junction is the greater of e_{con} and e_j. This thickness shall be maintained for a distance of at least $1,4 l_2$ from the junction and $0,7 l_2$ from the cone/knuckle tangent line along the cone, see Figure 6.4.2-2.

7 Modification to 6.4.9, Offset reducers

The 5th sentence of 6.4.9 shall read as follows:

The greater of these shall apply to the cone section of the reducer.

8 Modification to 8.3.2, Openings in the vicinity of discontinuities

Indent b) of 8.3.2 shall read as follows:

b) Openings in conical shells connected to cylindrical shells shall have the distances x_L *and* x_S shown in Figure 8.3.2-2 as follows:

 — for the large end

$$x_L \geq \max\left(0{,}2\sqrt{\frac{D_{mL}\ e_{as}}{\cos\alpha}}\ ;\ 3{,}0\ e_{as}\right) \quad (8.3.2\text{-}3)$$

— for the small end

$$x_S \geq \max\left(\sqrt{\frac{D_{mS}\ e_{as}}{\cos\alpha}}\ ;\ 3{,}0\ e_{as}\right) \quad (8.3.2\text{-}4)$$

where

D_{mL} is the mean diameter of cylindrical shell at the large end;

D_{mS} is the mean diameter of cylindrical shell at the small end.

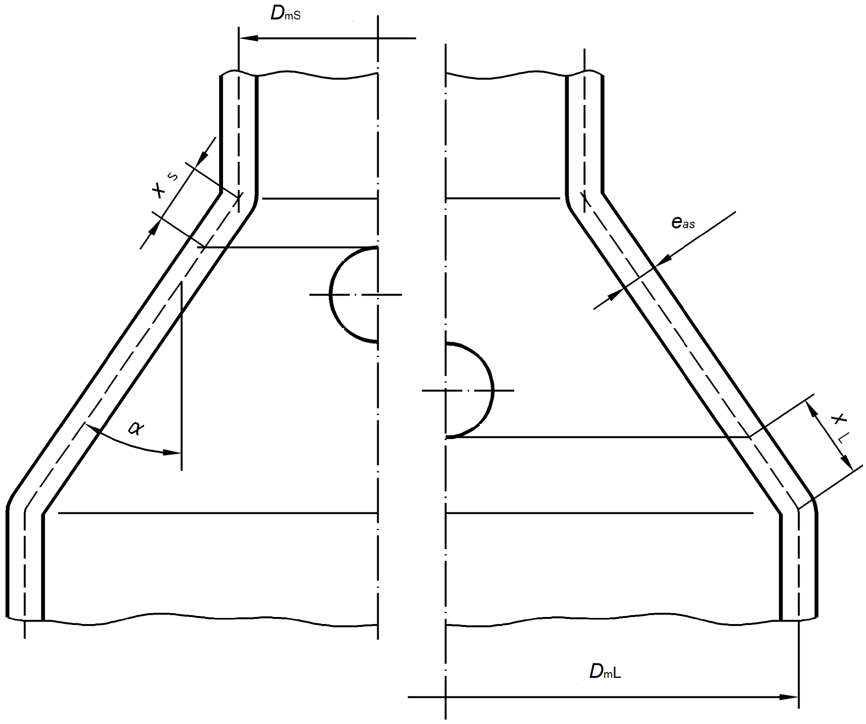


Figure 8.3.2-2 — Opening in a conical shell

9 Modification to 8.4.3, Reinforced openings with $d_i/D_i < 0{,}8$

After Formula (8.4.3-2), the following sentence shall be added:

d_{eqb} is the equivalent diameter of the branch at the intersection calculated according to Formulas (8.4.1-3) and (8.4.1-4) using the dimensions of the branch instead of the shell.

Indent c), 2nd paragraph of subclause 8.4.3, Formula (8.4.3-8) shall be deleted, new Figure 8.4.3-3 *shall be inserted, and the paragraph shall read as follows:*

Formulae (8.4.3-3) or (8.4.3-6) and (8.4.3-7) shall apply.

The maximum length of the shell considered as contributing to reinforcement shall be evaluated in accordance with the Formula (8.4.1-2) and for the branches in accordance with Formulae (8.4.3-1) and (8.4.3-2).

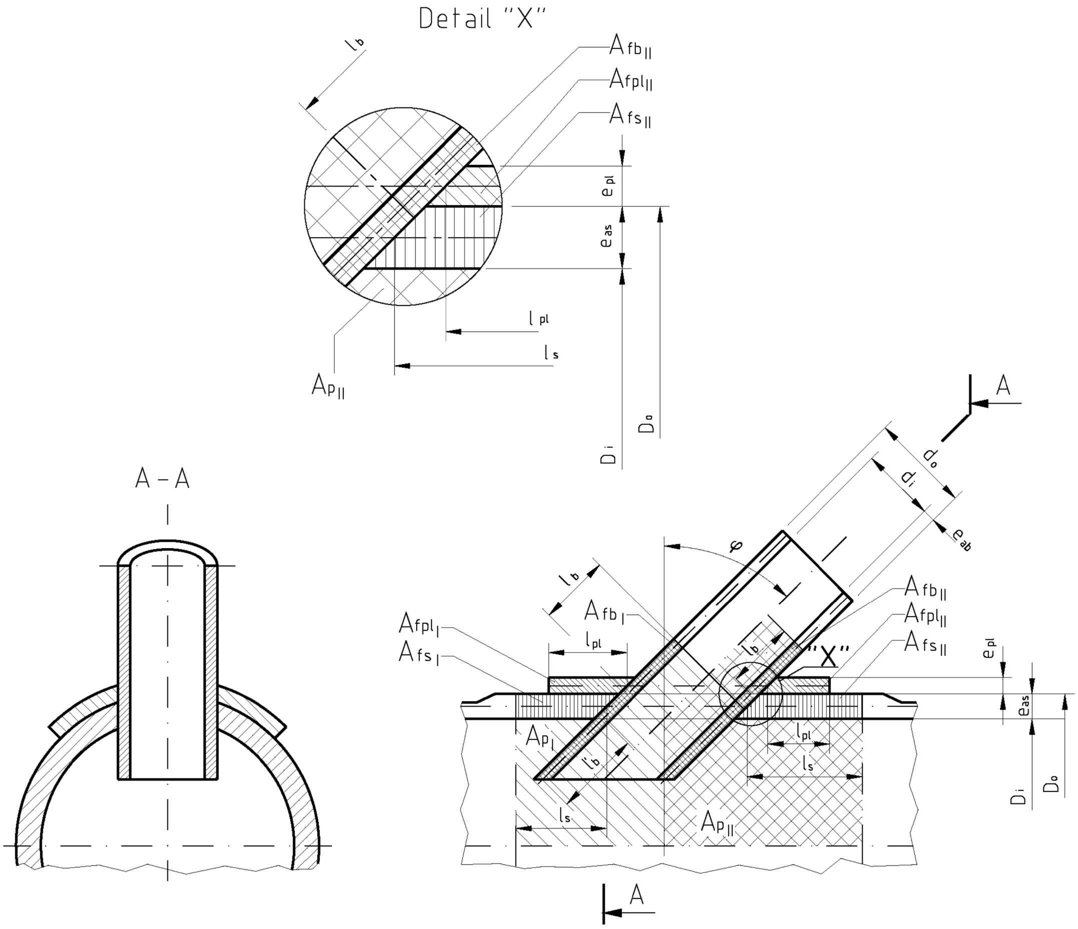


Figure 8.4.3-3 — Reinforcement of oblique branch connection in cylindrical or conical shell

Indent d), 2nd and 3rd paragraph of subclause 8.4.3 shall read as follows:

The reinforcement shall be calculated in accordance with Formulae (8.4.3-3), (8.4.3-6) and (8.4.3-7).

The maximum length of the shell considered as contributing to reinforcement shall be evaluated in accordance with the Formula (8.4.1-2) and for the branches in accordance with Formulae (8.4.3-1) and (8.4.3-2).

10 Modification to 9.1, General

9.1 shall read as follows:

"**9.1 General**

The rules in Clause 9 shall take account of loading due to external pressure.

9.1.1 External calculation pressure

The external pressure to be taken into account for calculation purpose shall be the maximum external pressure under operating conditions, or test conditions whichever is the greater.

Where internal pressure may decrease below atmospheric pressure due to fluid cooling, the external pressure to be used in calculation shall be equal to:

— 1 bar for single piping subject to external pressure; or

— the pressure between the two jackets, plus 1 bar for jacketed piping.

If pressure relief devices are fitted and where internal pressure may decrease below atmospheric pressure due to fluid cooling, the external pressure to be used in the calculation shall be at least the set pressure of the device.

9.1.2 Exception from verification against external pressure

For piping operating with external pressure not exceeding 1 bar, a check of design adequacy shall not be required where the following requirements are met:

— piping made of carbon steels or low alloy steels at a temperature less than or equal to 150 °C, or made of austenitic steel at a temperature less than or equal to 50 °C; and

— where $e/D_{\mathrm{O}} \geq 0{,}01$; and

— where out-of-roundness, *u* (see EN 13480-4:2017, 7.4.1), is less than or equal to 1 %, and local flat deviation is less than or equal to *e.*

9.1.3 General acceptance criteria

The thickness of a component under external pressure shall be not less than the thickness required by this standard for similar components under the same internal pressure using a joint coefficient of 1, (i.e. without any joint coefficient) or the thickness required by Clause 9 whichever is the greater.

There are two additional acceptability criteria which need to be checked for pipes / pressure vessels subjected to external pressure:

a) Sufficient safety margin against linear buckling: The existing external pressure p shall be smaller than the theoretical limit of stability of the perfect shape of the piping p_m divided by a safety factor of k_m = 3.0.

$$p \leq p_m / k_m \qquad (9.1.3\text{-}1)$$

The pressure p_m may be calculated using the formulas given below for the piping elements or by linear buckling analysis (bifucation load).

b) Sufficient saftety against over-stresses due to imperfections (e.g. ovalization). The existing external pressure p shall be smaller than pressure p_{yo} at which the mean circumferential stress in the shell reaches yield point of material divided by a safety factor of k_y = 1.5.

$$p \leq p_{yo} / k_y . \qquad (9.1.3\text{-}2)$$

The pressure p_{yo} shall be calculated taking into account the initial out of roundness of the piping as well as the increase of the imperfections due to the external pressure. The calculation can be done using the provisions of this chapter or using a detailed (e.g FE) analysis including the effects of geometric distortion under pressure (geometric nonlinearity / large deformation) and the material nonlinearities in case of piping subject to external pressure in the creep range.

The allowable deviation from the design shape shall be specified on the drawing or in associated documents.

The joint coefficient of welds shall not be taken into account.

Stiffening rings and other features used as stiffeners shall extend and be completely attached around the circumference. Any joint shall be so designed as to develop the full stiffness of the ring. Where internal stiffening rings arranged with local spaces between the shell and the ring are used (see Figure 9.1-1), in no case shall the length of the unsupported shell plate exceed the piping circumference divided by the coefficient (4 n_{cyl}).

Intermittent welds shall not be used where crevice corrosion can occur.

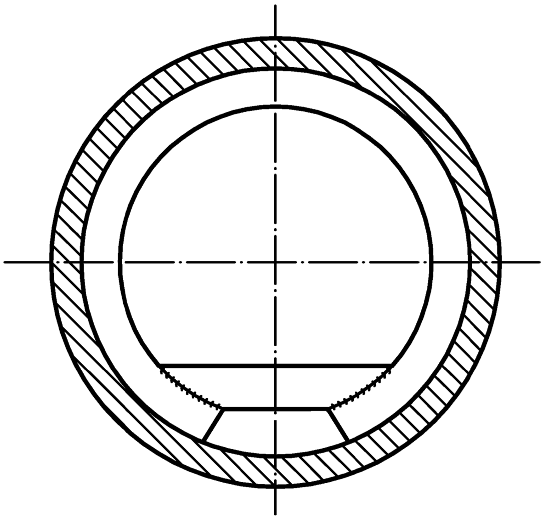


Figure 9.1-1 — Internal stiffening rings with a reinforced cut-out

"

11 Modification to 9.2.1, Symbols

The line concerning p_m *in the Table 9.2.1-1 shall be read as follows:*

p_m	theoretical elastic instability pressure for collapse	MPa (N/mm^2)
		To be continued

12 Modification to 9.2.2, Elastic stress limits

9.2.2 shall read as follows:

"**9.2.2 Stress limits**

For pipes not in the creep range the elastic stress limits shall be:

— for non austenitic steels:

$$S = R_{p0,2t} \quad (9.2.2\text{-}1)$$

$$S_s = R_{p0,2st} \quad (9.2.2\text{-}2)$$

— for austenitic steels:

$$S = \frac{R_{p1,0t}}{1,25} \quad (9.2.2\text{-}3)$$

$$S_S = \frac{R_{p1,0st}}{1,25} \quad (9.2.2\text{-}4)$$

For pipes in the creep range the stress limits shall be:

— for non austenitic steels:

$$S = min\left(R_{p0,2t}, \frac{R_{p1,0Tt}}{1,5} \right) \quad (9.2.2\text{-}5)$$

$$S_S = min\left(R_{p0,2st}, \frac{R_{p1,0sTt}}{1,5} \right) \quad (9.2.2\text{-}6)$$

— for austenitic steels:

$$S = min\left(\frac{R_{p1,0t}}{1,25}, \frac{R_{p1,0Tt}}{1,5} \right) \quad (9.2.2\text{-}7)$$

$$S_S = min\left(\frac{R_{p1,0st}}{1,25}, \frac{R_{p1,0sTt}}{1,5} \right) \quad (9.2.2\text{-}8)$$

where T is a time greater or equal than the design lifetime for operation under external pressure and t is the highest metal temperature while the pipe is under external pressure. Lifetime where the pipe is not operated in the creep range is not taken into account for Clause 9.

13 Modification to 9.3.2, Interstiffener collapse

9.3.2 shall read as follows:

The thickness of the shell within the unstiffened length L shall be not less than that determined by the following procedure.

a) calculate the theoretical elastic instability pressure for collapse:

$$p_{\mathrm{m}} = \frac{E_{\mathrm{t}}\, e_{\mathrm{a}}\, \varepsilon}{R_{\mathrm{m}}} \quad (9.3.2\text{-}2)$$

where

∑ is calculated from:

$$\varepsilon = \frac{1}{n_{\mathrm{cyl}}^2 - 1 + \frac{Z^2}{2}} \left\{ \frac{1}{\left(\frac{n_{\mathrm{cyl}}^2}{Z^2} + 1 \right)^2} + \frac{e_{\mathrm{a}}^2}{12 R_{\mathrm{m}}^2 \left(1 - \nu^2 \right)} \left(n_{\mathrm{cyl}}^2 - 1 + Z^2 \right)^2 \right\} \quad (9.3.2\text{-}3)$$

where

n_{cyl} is an integer ≥ 2 to minimize the value of p_{m};

$$Z = \frac{\pi \ R_{\text{m}}}{L} \tag{9.3.2-4}$$

and L is determined in accordance with 9.3.1

Check that $p \le p_m / k_m$ with k_m = 3 and if not, increase the wall thickness e_a .

b) Determine the maximum pressure at onset of plastic deformation for the ideal pipe:

$$p_y = \frac{S\, e_a}{R_m}$$

c) Determine the onset of plastic deformation using the given external pressure and the radius tolerance $\frac{\Delta r}{R_m}$ of the manufacturing specification:

$$p_{yo} = \left(\left(1 + 4 \frac{\Delta r}{R_m} \frac{R_m}{e_a} \right) p_m + p_y \right) / 2 - \sqrt{ \left(\left(1 + 4 \frac{\Delta r}{R_m} \frac{R_m}{e_a} \right) p_m + p_y \right)^2 / 4 - p_y\, p_m }$$

Check that $p \le p_{yo} / k_y$ with k_y = 1.5 and if not increase the wall thickness e_a or reduce the spacing of stiffeners until the required value is obtained.

14 Modification to 9.3.3, Overall collapse of stiffened pipes

The indent c) in the 9.3.3 shall read as follows:

c) Calculate the maximum stress in the stiffener from:

$$\sigma_s = \frac{k\, k_s\, S_s\, p}{p_{ys}} + \frac{E_t\ \delta \left(n^2 - 1 \right) \frac{\Delta r}{R_m} k\, k_s\, p}{R_m \left(p_n - k\, k_s\, p \right)}$$

15 Modification to 11.2, Allowable stresses

11.2 shall read as follows:

"11.2 Allowable stresses

The design stress shall be calculated in accordance with Clause 5.

Membrane stresses due to integral attachments shall be considered as local. Bending stresses caused by the same source and acting across the wall thickness of the pipe shall be classified as secondary stresses.

Stresses acting over the wall thickness of the pipe shall be combined with stresses resulting from:

— internal pressure;

— external loadings;

and shall comply with the following:

$P_m + P_b + P_L \leq 1{,}5\, f_f$	in case of sustained loads;
$P_m + P_b + P_L \leq 1{,}8\, f_f$	in case of sustained and occasional loads;
$P_m + P_b + P_L \leq 2{,}7\, f_f$	in case of exceptional loads;
$Q \leq f_a$	in case of restrained thermal expansion of the piping system;
$P_m + P_b + P_L + Q \leq f_f + f_a$	in case of sustained loads and restrained thermal expansion of the piping system.

where

P_m is the primary membrane stress;

P_L is the primary local membrane stress;

P_b is the primary bending stress;

Q is the secondary bending stress.

For determination of f_a, see Formulae (12.1.3.1) to (12.1.3.4). For the determination of f_f, see 12.3.2. The design stress f is defined in Clause 5.

For pure shear stresses (average value), the equivalent stress σ_{eq} shall be calculated according to the von Mises theory, and shall be limited to 1,5 f for time-independent design.".

16 Modification to 11.6, Stress analysis of the run pipe

11.6 shall read as follows:

"**11.6 Stress analysis of the run pipe**

The following modified formulas of Clause 12 shall be satisfied.

f_f is the design stress for flexibility analysis in N/mm2 (MPa) with $f_f = \min\left(f; f_{cr}\right)$.

a) For sustained loads

$$\sigma_1 + \sigma_{MT} \leq 1{,}5\, f_f \text{ and } 0{,}75i \geq 1{,}0 \qquad (11.6\text{-}1)$$

with:

σ_1 stress due to sustained loads according to (12.3.2-1) or (12.3.2-2)

σ_{MT} additional stress resulting from sustained loads

b) For sustained and occasional loads

$$\sigma_2 + \sigma_{MT} \leq 1{,}8\, f_f \text{ and } 0{,}75i \geq 1{,}0 \qquad (11.6\text{-}2)$$

with:

σ_2 stress due to sustained and occasional loads according to (12.3.3-1) or (12.3.3-4)

σ_{MT} additional stress resulting from sustained loads and occasional loads

c) For sustained and exceptional loads

$$\sigma_2 + \sigma_{MT} \leq 2,7\, f_f \text{ and } 0{,}75i \geq 1{,}0 \quad (11.6\text{-}3)$$

with:

σ_2 stress due to sustained and exceptional loads according to (12.3.3-1) or (12.3.3-4)

σ_{MT} additional stress resulting from sustained loads and exceptional loads

The stress range due to thermal expansion and alternate loads, e.g. seismic loads, shall either satisfy the following formula:

d) For loads caused by restrained thermal expansion

$$\sigma_3 + \frac{\sigma_{PT}}{2} \leq f_a \quad (11.6\text{-}4)$$

with:

σ_3 stress range due to thermal expansion and alternate loads according to (12.3.4-1) or (12.3.4-3)

σ_{PT} additional stress resulting from restrained thermal expansion

If the requirement of Formula (11.6-4) is not met, the sum of stresses due to sustained loads (Formula (11.6-1)) and restrained thermal expansion (Formula (11.6-4)) shall satisfy the following condition:

e) For the combination of sustained loads and restrained thermal expansion loads

$$\sigma_1 + \sigma_{MT} + \sigma_3 + \frac{\sigma_{PT}}{2} \leq f_f + f_a \text{ and } 0{,}75i \geq 1{,}0 \quad (11.6\text{-}5)$$

with:

σ_{MT} additional stress resulting from sustained loads

σ_{PT} additional stress resulting from restrained thermal expansion

The following formula limits the stress caused in the pipe wall to the mean value of the (associated) creep rupture strength in a similar way as Formula (12.3.5-1) or (12.3.5-2).

$$\sigma_5 + \sigma_{MT} + \frac{\sigma_{PT}}{2} \leq 1{,}25\, f_{cr} \text{ and } 0{,}75i \geq 1{,}0 \quad (11.6\text{-}5a)$$

with:

σ_5 stress in creep range according to (12.3.5-1) or (12.3.5-2)

σ_{MT} additional stress resulting from sustained loads

σ_{PT} additional stress resulting from restrained thermal expansion

In addition to the modified formulas above, the following formulas shall be also satisfied:

$$\sigma_{\mathrm{NT}}^{**} \leq 2R_{\mathrm{eHt}} \quad (11.6\text{-}6)$$

Limitation of the equivalent stress for pipes operating in the creep range (less than or equal to mean value of creep rupture strength):

$$\sigma_{NT}^{**} \leq 1{,}25 f_{cr} \qquad (11.6\text{-}6a)”$$

17 Modification to 13.11.2, Design temperatures for support components

13.11.2 shall read as follows:

“13.11.2 Design temperatures for support components

13.11.2.1 General

The temperature to be considered in the design of supports shall be related to that of the piping. All support components shall be designed as a minimum for temperatures in the range −20 °C to 80 °C. For ambient or system operating temperatures outside this range, the temperature shall be specified to the pipe support manufacturer.

Parts which may be adversely affected by excessively high or low pipe temperatures, such as springs or sliding material, shall be kept outside any insulation.

Design temperatures of supports shall be determined by detailed calculation or by testing or by reference to Table 13.11.2-1, Table 13.11.2-2 and Figure 13.11.2-1.

All support components including intermediate steelwork shall have a design temperature t of at least 80 °C.

Temperature shall be considered when designing or selecting individual hanger or support components.

13.11.2.2 Design temperature of components within the insulation

The temperature to be used for design purposes shall be in accordance with Table 13.11.2-1 and Figure 13.11.2-1.

Table 13.11.2–1 — Design temperature of components within the insulation

Type of component	Design temperature of support *t* but not less than 80 °C
Components directly welded to the pipe, straps and clamps (i.e. large face contact)	t_C
Components without direct contact with the pipe	t_C - 20 °C
Bolts, screws, nuts and pins	t_C - 30 °C
Where t_C is the design temperature of the pipe.	

13.11.2.3 Design temperature of components outside the insulation

The temperature to be used for design purposes shall be in accordance with Table 13.11.2-2 and Figure 13.11.2-1.

Table 13.11.2–2 — Design temperature of components outside the insulation

Type of component	Design temperature of support *t* but not less than 80 °C
Components directly connected to the pipe	0,5 t_C
Bolts, screws, nuts and pins	0,33 t_C
Where t_C is the design temperature of the pipe.	

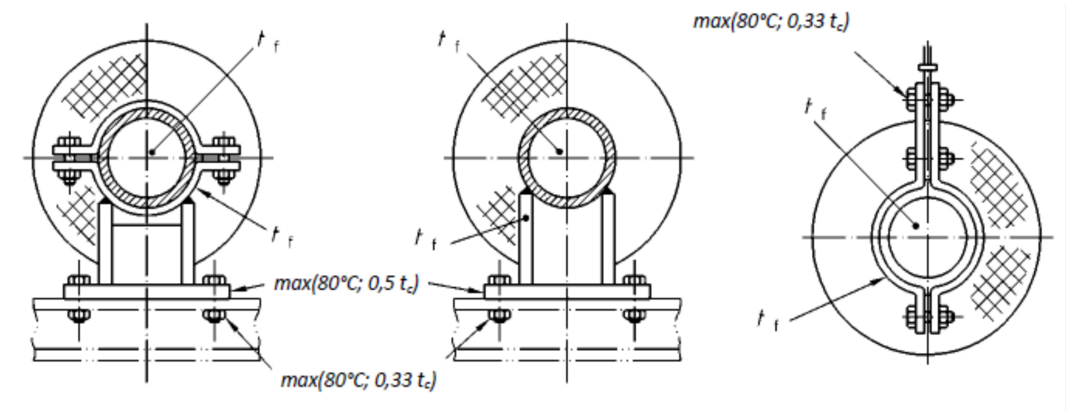


Figure 13.11.2-1 — Support design temperatures inside and outside insulation

18 Modification to 13.11.4, Determination of component sizes

13.11.4.1 and 13.11.4.2 shall read as follows:

"13.11.4.1 General

Dimensioning of support components designed by calculation shall be based upon sound engineering practice. The requirements of 13.11.4.2 shall be considered. For further guidance, see Annexes G.4, I, J, K, L and M.

13.11.4.2 Stress levels

The individual or equivalent stress levels shall not exceed the allowable stresses given in Table 13.11.4.3-1.

The equivalent stress, σ_e is defined as:

$$\sigma_e = \sqrt{(\sigma_a + \sigma_b)^2 + 3\tau^2} \quad (13.11.4.2\text{-}1)$$

where

- σ_a is the calculated axial (membrane) stress;
- σ_b is the calculated bending stress;
- τ is the calculated shear stress.

— The design stress is:

$$f = \min\left(\frac{R_{eHt}}{1,5} or \frac{R_{p0,2t}}{1,5}; \frac{R_m}{2,4}; f_{cr}\right) \quad (13.11.4.2\text{-}2)$$

alternative for austenitic steels Formula (13.11.4.2-2b), (13.11.4.2-2c) or (13.11.4.2-2d) can be applied:

for A ≥ 35 %

$$f = \min\left(\frac{R_{p\,1,0\,t}}{1,5}; f_{cr}\right) \quad (13.11.4.2\text{–}2b)$$

or $f = \min\left(\frac{R_{p\,1,0\,t}}{1,2}; \frac{R_{m\,t}}{3}; f_{cr}\right)$ if $R_{m\,t}$ is available (13.11.4.2–2c)

for 35 % > A ≥ 30 %

$$f = \min\left(\frac{R_{p\,1,0\,t}}{1,5}; \frac{R_m}{2,4}; f_{cr}\right) \quad (13.11.4.2\text{–}2d)$$

for A < 30 % see Formula (13.11.4.2–2).

NOTE For creep data for time periods other than 200 000 h, see 5.3.2.

19 Modification to 13.11.5, Welded connections

13.11.5 shall read as follows:

"13.11.5 Welded connections

13.11.5.1 General

Weld metal shall have a composition compatible with the parent material and shall not have a yield strength and minimum tensile strength below the lowest specified minimum value of the components being welded.

13.11.5.2 Full penetration welds

A full penetration weld has a complete penetration of weld and parent metal throughout the thickness of the joint.

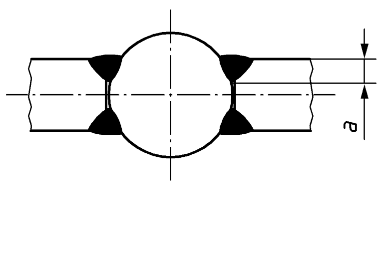
The allowable stress of a full penetration weld shall be taken as equal to the allowable stress of the weaker of the parts connected (see Table 13.11.4.3-1).

13.11.5.3 Partial penetration welds

A partial penetration weld is defined as a weld that has joint penetration which is less than the full thickness of the parent metal.

The allowable stress of a partial penetration weld shall be determined using the method described for a fillet weld.

The throat thickness of a partial penetration weld shall not be greater than the depth of penetration that can be consistently achieved (see Figure 13.11.5.3-1).



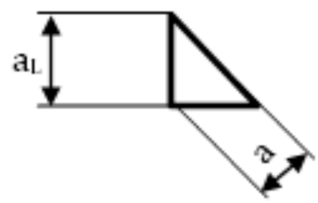
Key

a Throat thickness

Figure 13.11.5.3-1 — Throat thickness of a partial penetration weld

13.11.5.4 Fillet welds

A fillet weld has fusion faces that form an angle of between 60° and 120°. For angles of less than 60° the weld should be considered to be a partial penetration weld. A fillet weld shall be defined by its throat thickness *a* or by its leg length a_L (see Figure 13.11.5.4-1).



Key

a	Throat thickness
a_L	Leg length

Figure 13.11.5.4-1 — Dimensions of a fillet weld

Single sided fillet welds shall be avoided where practicable. If it is necessary to use a single sided fillet weld then the effects of bending shall be considered.

Fillet welds terminated at corners should be continued, full size, around the corner for a distance of at least twice the throat thickness *a* of the weld unless access or the configuration of the joint renders this impracticable.

The effective length of the weld l_{eff} shall be taken as the length over which the weld is full size. This may be taken as the overall length of the weld reduced by twice the effective throat thickness $a\ (= a_L/\sqrt{2})$. If the weld is full size throughout its entire length no reduction in length shall be necessary.

The stress in the weld, calculated with the throat thickness *a*, shall be determined in accordance with the Formula (13.11.5.4-1)

$$\sigma_{weld} = \sqrt{\left[\left(\sigma_{\mathrm{a}} + \sigma_{\mathrm{b}}\right)^2 + \tau^2\right]} \qquad (13.11.5.4\text{-}1)$$

The allowable stress of a fillet weld shall be determined in accordance with Table 13.11.5.4-1. Additionally the permissible shear stress of the base material shall not be exceeded.

Table 13.11.5.4–1 — Allowable stress of a fillet weld

	Normal operating loads	Occasional loads	Faulted condition loads
Fillet weld	0,7 *f*	0,84 *f*	1,05 *f*
Where *f* is the design stress of the weaker of the parts connected			

NOTE It is recommended to check the weld size of fillet welds to the main structure according to Table 13.11.5.4-2.

Table 13.11.5.4–2 — Allowable stress of a fillet weld to the main structure

<table>
<tr><th colspan="2">Normal operating loads</th><th colspan="2">Occasional loads</th><th colspan="2">Faulted condition loads</th></tr>
<tr><td colspan="2">$\sigma_{w,per}$</td><td colspan="2">1,2 $\sigma_{w,per}$</td><td colspan="2">1,5 $\sigma_{w,per}$</td></tr>
<tr><td colspan="6">Where $\sigma_{w,per}$ is the value of the of the weaker of the parts connected
$\sigma_{w,per} = \min (0{,}307\ R_{mt} / \beta_w\ ;\ 0{,}7\ f)$.</td></tr>
<tr><td rowspan="2">With</td><td></td><td>S235</td><td>S275</td><td>S355</td><td>all other</td></tr>
<tr><td>β_W =</td><td>0,8</td><td>0,85</td><td>0,9</td><td>1,0</td></tr>
<tr><td colspan="6">NOTE 1 For guidance, ($0{,}307\ R_{mt} / \beta_w$) is derived from the requirements of EN 1993-1-8:2005, 4.5.3.3.</td></tr>
<tr><td colspan="6">NOTE 2 For carbon steel / non-alloy structural steels (S235, S275, S355) see Annex G.4 - Material properties of carbon steel (structural steel) at elevated temperatures</td></tr>
</table>

20 Modification to 13.11.6, Threaded connections

13.11.6 shall read as follows:

"13.11.6 Bolted connections

13.11.6.1 General

Bolted connections shall be designed in accordance with appropriate European Standards. Attention shall be paid to the effects of temperature.

For:

— commercial grade bolt classes 4.6, 5.6, 8.8 and 10.9;

— bolted connections made of other ferritic or austenitic material;

The allowable stress in the thread of the bolt is defined in Tables:

— 13.11.6.1-1, for the time independent design range,

— 13.11.6.1-2, recommended for bolting to the main structure, and

— 13.11.6.1-3, for the time dependent design range,

where the stress is calculated using the tensile stress area for the thread and the gross area for the shank.

Depending on the position of the shear plane (perpendicular to the axis of the thread) the tensile stress area or the shank area shall be used to calculate the shear stress. Shear in the threaded portion of bolts should be avoided.

13.11.6.2 Time-independent nominal design stress

The design stress shall be in accordance with the following:

— for ferritic bolts: $f_B = R_{eH\,t}$ (13.11.6.1-1)

— for austenitic bolts: $f_B = R_{p0,2\,t}$ (13.11.6.1-2)

Table 13.11.6.1–1 — Allowable stress in bolted connections (time independent design)

Stress	Normal operating loads	Occasional loads	Faulted condition loads	Application range
Tension stress $\frac{\sigma_{B,per}}{f_B}$	0,600	0,720	0,900	for bolts with $R_{eH} \leq 450$ N/mm² $R_{p0,2} \leq 450$ N/mm²
Shear stress (per shear area) $\frac{\tau_{B,per}}{f_B}$	0,470	0,564	0,705	
Tension stress $\frac{\sigma_{B,per}}{f_B}$	0,400	0,480	0,600	for bolts with $R_{eH} > 450$ N/mm² $R_{p0,2} > 450$ N/mm²
Shear stress (per shear area) $\frac{\tau_{B,per}}{f_B}$	0,267	0,320	0,401	
Combined shear and tension	$\frac{\tau}{\tau_{B,per}} + \frac{\sigma}{1,4\sigma_{B,per}} \leq 1$			
Bearing stress $\frac{p_{per}}{\min(f;f_B)}$	0,70	0,84	1,05	$e/d_0 \geq 1,2$
	1,31	1,57	1,97	$e/d_0 \geq 1,5$
	1,75	2,10	2,63	$e/d_0 \geq 2$
with: d_0 hole diameter e edge distance (centre hole to edge)				

NOTE It is recommended to check bolting to the main structure in accordance with Table 13.11.6.1-2.

Table 13.11.6.1–2 — Allowable stress in bolted connections to the main structure

Stress		Normal operating loads	Occasional loads	Faulted condition loads	Application range
Tension stress	$\frac{\sigma_{B,per}}{R_{m\,B\,t}}$	0,242	0,290	0,363	
Shear stress (per shear area)	$\frac{\tau_{B,per}}{R_{m\,B\,t}}$	0,436	0,523	0,654	
Combined shear and tension		$\frac{\tau}{\tau_{B,per}} + \frac{\sigma}{1,4\sigma_{B,per}} \leq 1$			
Bearing stress	$\frac{p_{per}}{\max(R_{m\,B\,t};R_{m\,t})}$	0,320	0,384	0,480	$e/d_0 \geq 1{,}2$
		0,600	0,720	0,900	$e/d_0 \geq 1{,}5$
		0,810	0,972	1,215	$e/d_0 \geq 2$
Additionally, the stress in the bolt shank shall fulfil the requirements of 13.11.4.3.					

with:
- $R_{m\,B\,t}$: tensile strength of bolt at service temperature. If $R_{m\,B\,t}$ is not available the following value can be used $R_{m\,B\,t} = R_{m\,B}\,(R_{eH\,B\,t} / R_{eH\,B})$.
- $R_{m\,t}$: tensile strength of bolted material at service temperature. If $R_{m\,t}$ is not available the following value can be used $R_{m\,t} = R_m\,(R_{eH\,t} / R_{eH})$.
- d_0 hole diameter
- e edge distance (centre hole to edge)

NOTE Values above are derived to fulfil the requirements of EN 1993-1-8:2005, 3.6.1.

13.11.6.3 Time-dependent nominal design stress

Austenitic bolts shall not be used in the time-dependent design range. The design stress shall be in accordance with the following:

$$f_{B\,cr} = \min\,(S_{R\,200\,000\,h\,t}\,;\,0{,}7\,R_{eH\,t}) \qquad (13.11.6.2\text{-}1)$$

Table 13.11.6.2–3 — Allowable stress in bolted connections (time dependent design)

Stress	Normal operating loads	Occasional loads	Faulted condition loads
Tension stress $\dfrac{\sigma_{B,per}}{f_{B\,cr}}$	0,571	0,686	0,857
Shear stress (per shear area) $\dfrac{\tau_{B,per}}{f_{B\,cr}}$	0,381	0,457	0,571
Combined shear and tension	$\dfrac{\tau}{\tau_{B,per}} + \dfrac{\sigma}{1,4\sigma_{B,per}} \leq 1$		

In case of shear loaded bolted connections in the range of time-dependent nominal design stress the connection shall have sufficient edge distance to avoid tear-out of the threaded part.