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Metal industrial piping

Part 3: Design and calculation

National foreword

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Foreword

This document (EN 13480-3:2012) has been prepared by Technical Committee CEN/TC 267 “Industrial piping and pipelines”, the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by December 2012, and conflicting national standards shall be withdrawn at the latest by December 2012.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

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 EC Directive(s).

For relationship with EC Directive(s), see informative Annex ZA, which is an integral part of this document.

In this European Standard the Annexes A, C, F, G, I, K, L, M, P and Q are informative and the Annexes B, D, E, H, J, N and O are normative.

This European Standard EN 13480 for metallic industrial piping consists of eight interdependent and not dissociable Parts which are:

— *Part 1: General;*

— *Part 2: Materials;*

— *Part 3: Design and calculation;*

— *Part 4: Fabrication and installation;*

— *Part 5: Inspection and testing;*

— *Part 6: Additional requirements for buried piping;*

— CEN/TR 13480-7*: Guidance on the use of conformity assessment procedures;*

— *Part 8: Additional requirements for aluminium and aluminium alloy piping.*

Although these Parts may be obtained separately, it should be recognised that the Parts are interdependent. As such the manufacture of metallic industrial piping requires the application of all the relevant Parts in order for the requirements of the Standard to be satisfactorily fulfilled.

This European Standard will be maintained by a Maintenance MHD working group whose scope of working is limited to corrections and interpretations related to EN 13480.

The contact to submit queries can be found at http://www.unm.fr (en13480@unm.fr). A form for submitting questions can be downloaded from the link to the MHD website. After subject experts have agreed an answer, the answer will be communicated to the questioner. Corrected pages will be given specific issue number and issued by CEN according to CEN Rules. Interpretation sheets will be posted on the website of the MHD.

This document supersedes EN 13480-3:2002+A1:2005+A2:2006+A3:2009+A4:2010+A5:2012. This new edition incorporates the Amendments/the corrigenda which have been approved previously by CEN members, the corrected pages up to Issue 17 without any further technical change. Annex Y provides details of significant technical changes between this European Standard and the previous edition.

Amendments to this new edition may be issued from time to time and then used immediately as alternatives to rules contained herein. It is intended to deliver a new Issue of EN 13480:2012 each year, consolidating these Amendments and including other identified corrections. Issue 3 (2014-08) includes the corrected pages listed in Annex Y.

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, Former Yugoslav Republic of Macedonia, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and the United Kingdom.

1 Scope

This Part of this European Standard specifies the design and calculation of industrial metallic piping systems, including supports, covered by EN 13480.

2 Normative references

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

EN 287-1:2004+A2:2006, *Qualification test of welders — Fusion welding — Part 1: Steels*

EN 1515-2:2001, *Flanges and their joints — Bolting — Part 2: Combination of flange and bolting materials for steel flanges PN designated*

EN 1515-3:2005, *Flanges and their joints — Bolting — Part 3: Classification of bolt materials for steel flanges, Class designated*

EN 1515-4:2010, *Flanges and their joints — Bolting — Part 4: Selection of bolting for equipment subject to the Pressure Equipment Directive 97/23/EC*

EN 1591-1:2001+A1:2009+AC:2011, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*

EN 1591-2:2008, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 2: Gasket parameters*

EN 1993 (all parts), *Eurocode 3: Design of steel structures*

EN 10204:2004, *Metallic products — Types of inspection documents*

EN 12953-3:2002, *Shell boilers — Part 3: Design and calculation for pressure parts*

EN 13445-3:2009, *Unfired pressure vessels — Part 3: Design*

EN 13480-1:2012, *Metallic industrial piping — Part 1: General*

EN 13480-2:2012, *Metallic industrial piping — Part 2: Materials*

EN 13480-4:2012, *Metallic industrial piping — Part 4: Fabrication and installation*

EN 13480-5:2012, *Metallic industrial piping — Part 5: Inspection and testing*

EN ISO 15614-1:2004, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 1: Arc and gas welding of steels and arc welding of nickel and nickel alloys* (ISO 15614-1:2004)

EN ISO 5817:2007, *Welding — Fusion-welded joints in steel, nickel, titanium and their alloys (beam welding excluded) — Quality levels for imperfections* (ISO 5817:2003, corrected version:2005, including Technical Corrigendum 1:2006)

3 Terms, definitions, symbols and units

3.1 Terms and definitions

For the purposes of this Part of this European Standard, the terms and definitions given in EN 13480-1 apply.

3.2 Symbols and units

For the purposes of this Part of this European Standard, the symbols and units given in EN 13480-1 and in Table 3.2-1 apply.

Specific symbols are defined in the relevant sub-clauses.

Table 3.2-1 — General symbols and units

Symbol	Description	Unit
PS[a]	maximum allowable pressure	bar
R, r[b]	radii	mm
$R_{eH\,t}$	minimum specified value of upper yield strength at calculation temperature when this temperature is greater than the room temperature	MPa (N/mm²)
S_1	mean value of the stress which leads to a 1 % creep elongation in 100 000 h	MPa (N/mm²)
S_2	mean value of the stress which leads to a 1 % creep elongation in 200 000 h	MPa (N/mm²)
$S_{R\,T\,t}$	mean value of creep rupture strength as indicated by the standards, for the material in question at the considered temperature, t, and for the considered lifetime T (in hours) whereby the dispersion band of the results does not deviate by more than 20 % from the mean value.	MPa (N/mm²)
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/mm²)
f_{cr}	Design stress in the creep range	MPa (N/mm²)
f_f	Design stress for flexibility analysis	MPa (N/mm²)
p_c	calculation pressure (see 4.2.3.4)	MPa (N/mm²)
p_o	operating pressure (see 4.2.3.1)	MPa (N/mm²)
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/mm²) and *PS* is in bar.

[b] The following subscripts apply :

i inside

m mean

o outside

4 Basic design criteria

4.1 General

The calculation rules in this Part shall apply for operating and testing conditions as well as preset, cold pull conditions, flushing and cleaning conditions.

The scope of each of the calculation rules is limited on a case by case basis by geometrical characteristics which shall be taken into account for each component, loading, failure mode and material property.

NOTE Where this European Standard does not indicate a calculation rule, it is the designer's responsibility to use widely acknowledged calculation rules, or experimental methods to justify the dimensions and thicknesses selected.

Elastic calculation methods shall be used in this Part, although some components might exhibit plastic behaviour.

If shaping and assembly techniques specific to the pressure vessel industry are used for large diameter pipes, the rules applicable to such techniques shall be those relating to the design of shells for pressure vessels. For the general stability, the requirements of EN 13480 remain applicable if the structure, as a whole, behaves according to the beam theory.

For temporary piping, e.g. flushing, cleaning and blow-through systems, the nominal design stress for the design conditions shall be used.

4.2 Loadings

4.2.1 General

Any piping system is subjected to a number of loadings during its lifetime. These loadings can be one or a combination of the following loads:

— internal and/or external pressure;

— temperature;

— weight of piping and contents;

— climatic loads;

— dynamic effects of the fluid;

— movements of the ground and buildings;

— vibrations;

— earthquakes.

NOTE 1 This list is not exhaustive.

NOTE 2 Explanations of these loads are given in 4.2.3 and 4.2.4.

4.2.2 Combination of loads

The loads and their possible combinations, given in 4.2.5.1 to 4.2.5.4, shall be taken into account at the design stage of the piping system and its supports. Some unlikely combinations may be ruled out following a study considering both the likelihood of their occurrence, failure of the fluid containment boundary and the health and safety consequences.

Where a piping system is subjected to more than one pressure/temperature condition, the greatest calculated thickness determined for these conditions shall be used.

4.2.3 Loads for dimensioning

4.2.3.1 Operating pressure

The operating pressure, p_{o}, shall be below the maximum allowable pressure, *PS*, specified for the piping system.

4.2.3.2 Operating temperature

The operating temperature, t_{o}, shall be below the maximum allowable temperature, *TS*, specified for the piping system.

4.2.3.3 Sets

The set (p_o, t_o) to be considered for the dimensioning of the elements of a piping system shall correspond to the most severe conditions of pressure and temperature which prevail simultaneously over a long time in the piping section under consideration. Thus for the thickness calculation of a component, the simultaneous conditions of pressure and temperature to be considered are the conditions which lead to the greatest thickness.

For all piping system elements, an allowable maximum pressure, based on

a) specified material (mechanical properties),

b) a given temperature,

can be easily determined by taking into account the applicable safety factors.

Temporary deviations e.g. due to pressure surge or operation of control release valve (safety valve) shall not be taken into account if the calculated stresses from such variations do not exceed the allowable stress by more than 10 % for less than 10 % of any 24 h operating period.

4.2.3.4 Calculation pressure

For all pressure temperature conditions (p_o, t_o) specified in 4.2.3.3 calculation pressures p_c shall be determined.

The calculation pressure p_c shall be not less than the associated operating pressure p_o, taking into account the adjustments of the safety devices. The conditions (p_o, t_o) resulting in the greatest wall thickness shall be considered with both of the following minimum conditions:

1) $p_c = p_o = PS$ with the associated t_c as defined in 4.2.3.5;

2) t_c as defined in 4.2.3.5 for $t_o = TS$ with the associated $p_c = p_o$.

NOTE If there is a condition where $p_o = PS$ and $t_o = TS$ only this condition has to be calculated.

When the calculation temperature t_c is such that the creep rupture strength characteristics are relevant for the determination of the nominal design stress, the calculation pressure shall be considered equal to the operating pressure (p_o) which is associated with the corresponding temperature (t_o).

4.2.3.5 Calculation temperature

The calculation temperature, t_{c}, shall be the maximum temperature likely to be reached at the mid-thickness of the piping, under normal operating conditions, at the calculation pressure p_{c}. The calculation temperature shall be determined as indicated below. Any heat transfer calculation shall be performed on the assumption that there is no heat loss due to wind.

a) For externally uninsulated and internally unlined piping components, the calculation temperature shall be as follows :

   1) For fluid temperatures below 40 °C, the calculation temperature for the component shall be taken as the fluid temperature;

   2) For fluid temperatures of 40 °C and above, unless a lower average wall temperature is determined by test or heat transfer calculation, the calculation temperature for uninsulated components shall be not less than the following values, but not less than 40°C:

      i) 95 % of the fluid temperature for valves, pipes, ends, welding fittings, and other components having wall thickness comparable to that of the pipe;

      ii) 90 % of the fluid temperature for flanges (except lap joint flanges) including those on fittings and valves;

      iii) 85 % of the fluid temperature for lap joint flanges;

      iv) 80 % of the fluid temperature for bolting.

b) For externally insulated piping components, the component calculation temperature shall be the fluid temperature unless calculations, tests, or service experience based on measurements, support the use of another temperature. Where piping is heated or cooled by tracing or jacketing, this effect shall be considered in establishing component calculation temperatures;

c) For lined piping components, the component calculation temperature shall be specified or based on heat transfer calculations or tests taking into account the fluid temperature and the characteristics of the lining;

   NOTE The lining may be used for insulation.

d) When the calculated thickness is determined by the creep strength properties, the operating pressure (p_{o}) and the operating temperature (t_{o}) can exceed temporarily the values used for this calculation, see 4.2.5.2.1 and 12.3.3.

4.2.4 Other loads to be taken into account

4.2.4.1 Weight of piping and contents

Gravity loads acting on the piping system shall be taken into account in the design and include:

— the weight of the pipe, fittings, valves and insulation;

— the weight of the conveyed fluid;

— the weight of the test fluid.

4.2.4.2 Climatic loads

When the piping system is exposed to climatic loads, these shall be taken into account. The maximum loads shall be determined in accordance with the actual local climatic and piping system exposure conditions.

4.2.4.3 Dynamic effects of the fluid

The piping system shall be designed in order to avoid the accidental dynamic effects due to the fluid. Where such effects are impossible to avoid, they shall be taken into account. Where these effects result from a direct consequence of the process, or the use of the equipment supplied by the purchaser, they shall be defined quantitatively in the purchase specification.

The reaction forces of safety valves during their operation shall be taken into account. If these valves are not supplied by the piping installer, the purchaser shall define the reaction forces and their directions.

For the assessment of the dynamic effects of the fluid, see Annex A.

4.2.4.4 Movements of the ground and of the buildings

When such movements (movements of the ground, differential settlements of the buildings) are likely to occur during the lifetime of the piping system, the values to be used in the design shall be provided in the purchase specification.

4.2.4.5 Vibrations

The piping system shall be designed and supported so as to eliminate or minimize excessive and harmful effects of vibration which may arise from such sources as impact, pressure pulsations, resonance due to compressors, and wind loads.

Where vibrations can occur during operation, the piping layout shall be studied and supports, dampers, restraints, anchors etc. shall be provided in accordance with clause 13 to remove these effects. If such studies are insufficient, it shall be necessary to carry out a specific vibration analysis to ensure the piping system is not over-stressed.

NOTE It is not necessary to provide written proof of the vibration analysis.

4.2.4.6 Earthquake

If required in a specification, the piping system shall be designed to withstand seismic loads. The specification shall give details relating to the characteristics of the seismic conditions to be taken into account.

NOTE Guidance for assessment of the earthquake effects is given in Annex A.

4.2.5 Design conditions

4.2.5.1 Normal operating conditions

Normal operating conditions shall be the conditions met in steady state at constant power and in transient working conditions corresponding to normal operating processes. Full load, partial load and shut-downshut-down conditions shall be examined together with the associated start-up, switching and shut-down operations.

For the calculation under normal operating conditions, each of the following conditions shall be examined:

— internal pressure and/or external pressure, including static head, where relevant;

— dead weight of piping, including internal structures and attached equipment;

— weight of insulation;

— weight of fluid;

— thermal expansion;

— supporting conditions;

— reaction of spring and constant load supports;

— displacements and rotations of anchor points, supports and connected equipment;

— cold pull;

— settlement of buildings.

4.2.5.2 Occasional operating conditions

4.2.5.2.1 General loadings

Occasional operating conditions shall be normal operation incidents, for example the reaction effects due to operation of safety and relief devices, turbine load rejection, pump failure, opening and closing of shut-off valves.

For the calculation under occasional operating conditions, in addition to normal loadings given in 4.2.5.1, each of the following conditions shall be examined:

— safety valve operations;

— dynamic shock forces e.g. steam, water hammer;

— discharge reactions;

— temperatures deviating from normal operating conditions;

— restraining effects of dampers and snubbers;

— loading conditions of spring and constant effort supports;

— possible normal climatic effects e.g. normal snow load and wind load corresponding to the actual local conditions;

— seismic conditions (Design Basis Earthquake).

4.2.5.2.2 Cleaning loading

For the calculation under cleaning (acid-cleaning and rinsing) loading conditions, all static, dynamic and kinematic boundary conditions shall be examined.

Each of the following conditions shall be examined:

— internal pressure;

— dead weight of piping, including internal structures and attached equipment;

— weight of insulation (partially or completely applied);

— weight of cleaning fluid;

— thermal expansion at cleaning temperature;

— supporting conditions (including temporary supports);

— locked and unlocked spring and constant effort supports;

— displacements and rotations of anchor points, supports and connected equipment;

— cold pull.

4.2.5.2.3 Steam purge/blow loading

For the calculation under steam purge/blow loading conditions, all static and dynamic boundary conditions shall be taken into account.

Each of the following conditions shall be examined:

— changed system geometry;

— dead weight of piping system, including internal structure and attachments;

— weight of insulation (partially or completely applied);

— blow-through pressures;

— thermal expansion at blow-through temperature;

— supporting conditions (including temporary supports);

— locked and unlocked spring and constant effort supports;

— displacements and rotations of anchor points, supports and connected equipment;

— cold pull;

— forces at exit (discharge reaction force).

4.2.5.3 Exceptional operating conditions

Exceptional operating conditions shall be rarely occurring events.

For the calculation under exceptional operating conditions, in addition to normal loadings given in 4.2.5.1, each of the following conditions shall be examined:

- possible exceptional climatic effects, e.g. exceptional snow and wind corresponding to the local conditions;
- seismic conditions (Safe Shut-down Earthquake).

4.2.5.4 Test conditions

For the calculation under test conditions, all static, dynamic and kinematic boundary conditions shall be taken into account.

Each of the following conditions shall be examined:

- internal pressure (test gauge pressure and static head);
- dead weight of piping, including internal structures and attached equipment;
- weight of insulation (partially or completely applied);
- weight of test fluid;
- thermal expansion;
- supporting conditions (including temporary supports);
- blocked and unblocked spring and constant effort supports;
- displacements and rotations of anchor points, supports and connected equipment;
- cold pull.

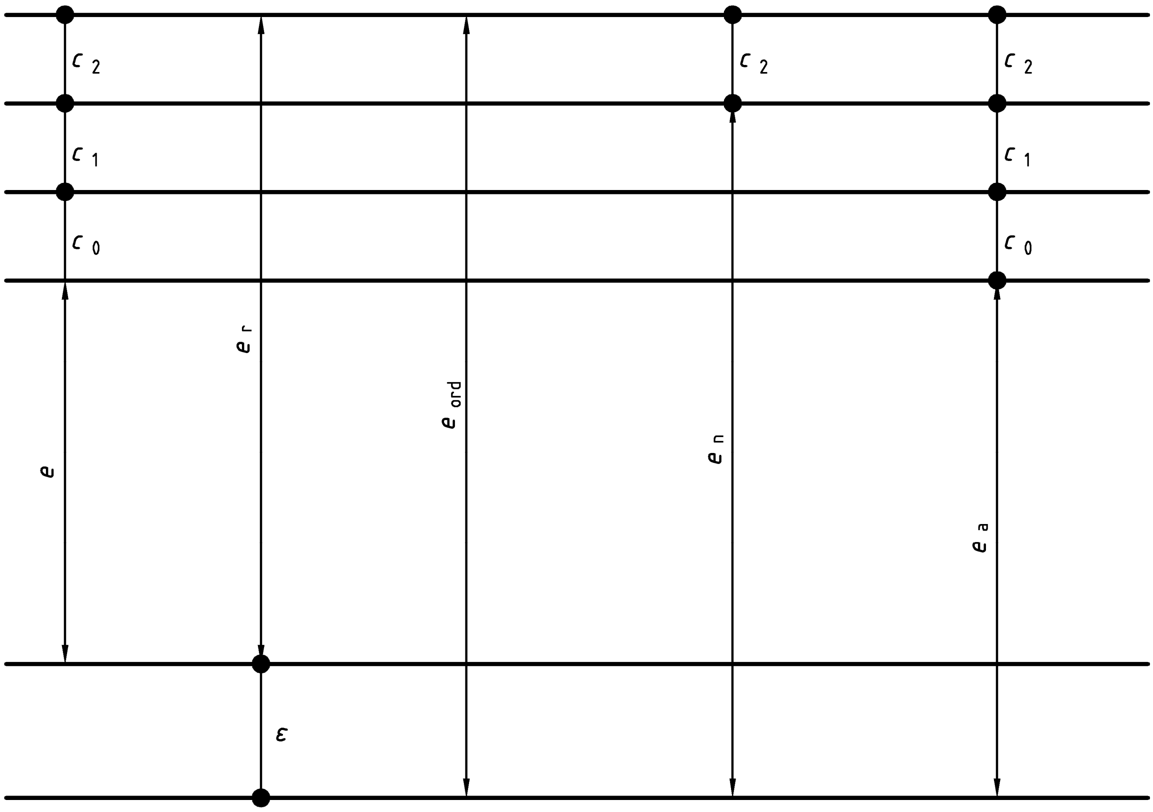
4.3 Thickness

The minimum thickness shall be determined with regard to the manufacturing process for pipes and fittings.

Corrosion can be internal or external or both at the same time (the term corrosion includes erosion).

The value of the corrosion allowance c_0 (which may be zero if no corrosion is to be expected) shall be determined by the manufacturer in accordance with the nature, temperature, velocity etc. of the products in contact with the wall.

All thicknesses, the corrosion allowance c_0, the tolerance c_1 and the thinning c_2 are shown in Figure 4.3-1.



Where

e is the minimum required thickness without allowances and tolerances to withstand pressure, calculated by the appropriate equations given in this standard;

c_0 is the corrosion or erosion allowance;

c_1 is the absolute value of the negative tolerance taken from the material standards or as provided by the pipe manufacturer;

c_2 is the thinning allowance for possible thinning during manufacturing process (e.g. due to bending, straving, threading, groaving, etc);

e_r is the minimum required thickness with allowances and tolerances;

ε is the additional thickness resulting from the selection of the ordered thickness e_{ord};

e_{ord} is the ordered thickness (where c_2 is often equal to 0; e.g. straight pipe);

e_n is the nominal thickness (on drawings);

e_a is the analysis thickness of a component, used for the check of the strength.

Figure 4.3-1 — Thickness (applicable to straight pipes as well as bends)

The analysis thickness e_a shall be the lowest thickness after corrosion and shall be given by:

$$e_a = e + \varepsilon \tag{4.3-1}$$

or

$$e_a = e_{ord} - c_0 - c_1 - c_2 \tag{4.3-2}$$

The ordered thickness e_{ord} of a component (pipe or fitting) shall be at least equal to:

— if the value of tolerance c_1 is expressed in unit of length

$$e_{ord} \geq e + c_0 + c_1 + c_2 \tag{4.3-3}$$

— if the value of tolerance c_1 is expressed as a percentage x of the ordered thickness e_{ord} :

$$e_{ord} \geq (e + c_0 + c_2)\,100/(100 - x) \tag{4.3-4}$$

4.4 Tolerances

The nominal dimensions shall be used in the calculations and the tolerances regarding thickness shall be fulfilled.

4.5 Joint coefficient

The joint coefficient z shall be used in the calculation of the thicknesses of components which include one or several butt welds, other than circumferential, and shall not exceed the following values:

— for equipment subject to destructive and non-destructive testing which confirms that the whole series of joints show no significant imperfections: 1;

— for equipment subject to random non-destructive testing: 0,85;

— for equipment not subject to non-destructive testing other than visual inspection: 0,7.

For the calculation of the strength of butt welded assemblies under exceptional operating conditions or under test conditions, it shall not be necessary to take a joint coefficient into account.

NOTE See EN 13480-5, Table 8.3-1.

4.6 Dimensioning of piping components subject to pressure

Clauses 6, 7, 8, 9, 10 and 11 describe the "design by rules" of piping components under static and cyclic loadings. This may be completed or replaced by a "design by analysis" as described in EN 13445-3, Annexes B and C. The requirements of clauses 6, 7, 8, 9 and 11 shall apply for loads of predominantly non-cyclic nature. It is assumed that a number of 1 000 full load pressure cycles do not lead to fatigue failure of the piping component under consideration. Where high strength materials with f greater than 250 N/mm^2 are used, a detailed load cycle analysis shall be carried out in accordance with clause 10.

Clauses 6, 7, 8, 9, 10 and 11 describe the "design by rules" of piping components under static and cyclic loadings. The « design by rule » can be completed or replaced by a « design by analysis » as described in EN 13445-3, Annex B and Annex C, where applicable.

In the case of cyclic loading (see clause 10), the geometry of the component under consideration shall be designed to avoid high stress concentrations. Where a higher number of pressure cycles shall be considered, the evaluation procedure outlined in 10.3 shall apply. For significant through-wall temperature gradients with cyclic nature in combination with cyclic pressure, the evaluation procedure outlined in 10.4 shall apply.

If the component under consideration is subjected to significant section moments resulting from connected piping, this shall be in accordance with 12.4.

The stress limits of components in accordance with European Standards with P/T ratings, e.g. flanges and components with wall thickness related to standard pipes, e.g. fittings, need not be recalculated.

5 Design stresses

5.1 General

The design stress shall be the lower of the time-independent stress value as calculated in 5.2 and the time-dependent stress value as calculated in 5.3 and shall be determined for each design and test condition.

The values of the design stress shall be determined from the material properties as calculated in accordance with the material standards and specifications given in EN 13480-2. These minimum values, specified for the delivery conditions, shall be used for design purpose, unless fabrication and/or heat treatment is known to lead to lower values. In such cases, the values to be used shall be determined by the manufacturer on the basis of the data given in the specification.

For steels used at low temperature (i.e. less than – 10 °C), the design stress shall be determined at room temperature and they shall have an impact energy at design reference temperature in accordance with EN 13480-2.

NOTE 1 For intermediate temperatures, linear interpolation may be used.

NOTE 2 For use of the specified value at room temperature for temperatures less than or equal to 50 ºC, see EN 13480-2, 4.2.2.1.

NOTE 3 For temporary piping, see 4.1.

For bolts, see additional requirements in 6.6.

5.2 Time-independent nominal design stress

5.2.1 Steels other than austenitic steels

5.2.1.1 Design conditions

The design stress shall be in accordance with the following:

$$f = \min\left\{\frac{R_{\mathrm{eH}\,t}}{1{,}5} \text{ or } \frac{R_{\mathrm{p0,2}\,t}}{1{,}5}; \frac{R_{\mathrm{m}}}{2{,}4}\right\} \tag{5.2.1-1}$$

5.2.1.2 Test conditions

The designer shall ensure that the nominal design stress f_{test} under the proof test conditions, given in EN 13480-5, does not exceed 95 % R_{eH} at specified test temperature.

5.2.2 Austenitic steels

5.2.2.1 Design conditions

The design stress shall be in accordance with the following:

— for $A \geq 35$ %

$$f = \frac{R_{\mathrm{p1,0}\,t}}{1{,}5} \tag{5.2.2-1}$$

or $f = \min\left(\frac{R_{m\,t}}{3};\frac{R_{p1.0\,t}}{1{,}2}\right)$ if $R_{m\,t}$ is available

— for 35 % > $A \geq$ 30 %

$$f = \min\left(\frac{R_{p1,0\,t}}{1{,}5};\frac{R_m}{2{,}4}\right) \tag{5.2.2-2}$$

— for A < 30 %, see 5.2.1.1.

5.2.2.2 Test conditions

For $A \geq$ 25 %, the designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed the greater of the two following values:

— 95 % $R_{p1,0}$ at specified test temperature;

— 45 % R_m at specified test temperature.

For A < 25 %, see 5.2.1.2.

5.2.3 Nickel and / or chromium alloy steels

Time-independent nominal design stress for Nickel and/or chromium alloy steels depend on the specified minimum elongation after rupture at room temperature.

5.2.4 Steels castings

5.2.4.1 Design conditions

The design stress shall be in accordance with the following:

$$f = \min\left(\frac{R_{eH\,t}}{1{,}9}\text{ or }\frac{R_{p0,2\,t}}{1{,}9};\frac{R_m}{3{,}0}\right) \tag{5.2.4-1}$$

5.2.4.2 5.2.4.2 Test conditions

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed R_{eH} or $R_{p0,2}$ at the specified test temperature divided by a safety factor of 1,4.

5.2.5 Additional requirements for steels with no specific control

5.2.5.1 General

Steels with no specific control are those not possessing at least a test report 2.2 in accordance with EN 10204, and shall only be used if permitted in the technical specification.

5.2.5.2 Design conditions

The design stress given in 5.2.1.1 shall be divided by an additional safety factor which shall not be less than 1,2.

Where the yield strength values at elevated temperatures for unalloyed steels are not specified in the material standards, the following equation may be used:

$$R_{p\ 0,2} = R_m \frac{720 - t}{1400} \tag{5.2.5-1}$$

where t is between 20 °C and 150 °C.

5.2.5.3 Test conditions

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed 95 % R_{eH} at the specified test temperature.

5.3 Time-dependent nominal design stress

5.3.1 General

For welds other than circumferential welds in welded pipes and fittings, the creep strength values of the base material shall be reduced by 20 %, except where ensured creep strength values have been determined for these pipes and fittings. This reduction is valid only for dimensioning.

NOTE If the creep characteristics determine the thickness, additional creep tests (based on extrapolations using the Larson-Miller formula) for instance on the welding consumables and the complete weld, should be performed.

5.3.2 Steels

5.3.2.1 Design conditions

The design stress in the creep range f_{cr} to be used for design under static loading shall be:

$$f_{cr} = \frac{S_{RTt}}{Sf_{cr}} \tag{5.3.2-1}$$

where

Sf_{cr} is a safety factor which depends on the time and shall be in accordance with Table 5.3.2-1.

Table 5.3.2-1 — Safety factor as a function of mean creep rupture strength related to time

Time T h	Safety factor Sf_{cr}
200 000	1,25
150 000	1,35
100 000	1,5

If the design lifetime is not specified, the mean creep rupture strength at 200 000 h shall be used.

In cases where the 200 000 h values are not specified in the material Standards, the mean creep rupture strength at 150 000 h or 100 000 h shall be used.

If a design lifetime between 100 000 h and 200 000 h is specified, and a lifetime monitoring system is provided, divergent from Table 5.3.2-1, a safety factor Sf_{cr} = 1,25 may be used.

In cases where design lifetimes shorter than 100 000 h are specified, one of the following methods shall be used:

a) If a lifetime monitoring System is not provided, the safety factor Sf_{cr} shall be equal to 1,5 and shall be applied to the mean creep rupture strength at the relevant lifetime of at least 10 000 h;

b) If a lifetime monitoring system is provided, a safety factor of Sf_{cr} = 1,25 may be specified with regard to the mean creep rupture strength at the relevant lifetime of at least 10 000 h. In no case shall the 1 % creep strain limit (mean value) at 100 000 h be exceeded.

The creep rupture strength associated to the specified lifetime shall be interpolated based on a logarithmic time axis as well as a logarithmic stress axis (double logarithmic interpolation scheme).

5.3.2.2 Test conditions

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed 95 % R_{eH} or 95 % $R_{p1,0}$ or 95 % $R_{p0,2}$ as applicable, at the specified test temperature .

5.3.3 Nickel and/or chromium alloy steels

The requirements of 5.3.2 shall apply unless otherwise stated in the specification.

6 Design of piping components under internal pressure

6.1 Straight pipes

The minimum required wall thickness for a straight pipe without allowances and tolerances, e, shall be calculated as follows:

— where $D_o/D_i \leq 1{,}7$:

$$e = \frac{p_c D_o}{2fz + p_c} \tag{6.1-1}$$

or

$$e = \frac{p_c D_i}{2fz - p_c} \tag{6.1-2}$$

— where $D_o/D_i > 1{,}7$:

$$e = \frac{D_o}{2}\left(1 - \sqrt{\frac{fz - p_c}{fz + p_c}}\right) \tag{6.1-3}$$

or

$$e = \frac{D_i}{2}\left(\sqrt{\frac{fz + p_c}{fz - p_c}} - 1\right) \tag{6.1-4}$$

NOTE This is Lame's equation.

6.2 Pipe bends and elbows

6.2.1 General

There are two methods for calculating the wall thickness of elbows (see 6.2.3.1 and Annex B) and three methods for the wall thickness of pipe bends (see 6.2.3.1, 6.2.3.2 and Annex B). The chosen method shall be used in its entirety.

The equations given in 6.2.3 shall be applicable only if the out-of-roundness of the pipes bends are within the tolerances given in EN 13480-4.

NOTE These calculation rules take into account [1] and [2] that upon applying internal pressure to a pipe bend, higher stresses occur on the intrados of the bend (and lower stresses on the extrados of the bend) than on a straight pipe with identical wall thickness.

6.2.2 Symbols

For the purposes of 6.2, the symbols given in Table 6.2.2-1 shall apply in addition to those given in Table 3.2-1.

Table 6.2.2-1 — Additional symbols for the purposes of 6.2

Symbol	Description	Unit
e_{int}	minimum required thickness without allowances and tolerances for a bend on the intrados	mm
e_{ext}	minimum required thickness without allowances and tolerances for a bend on the extrados	mm
R	radius of bend or elbow	mm
r	mean radius of the pipe	mm

6.2.3 Required wall thicknesses

6.2.3.1 Normal route

The minimum required wall thickness without allowances and tolerances shall be calculated by:

— on the intrados

$$e_{int} = e\frac{(R/D_o) - 0{,}25}{(R/D_o) - 0{,}5} \tag{6.2.3-1}$$

— on the extrados

$$e_{ext} = e\frac{(R/D_o) + 0{,}25}{(R/D_o) + 0{,}5} \tag{6.2.3-2}$$

where

e is calculated in accordance with 6.1 for straight pipe.

6.2.3.2 Alternative route

Pipes to be bent, by any manufacturing process, shall have a sufficient thickness such that it can be demonstrated that the following requirements are met after bending.

a) The minimum thickness at any point in a pipe bend (including the extrados) shall be not less than that required for the equivalent straight pipe.

b) Where the design stress is time-dependent, and the bend radius is less than six times the outside diameter of the pipe, the intrados thickness shall be not less than that calculated from the following equation:

$$e_{\text{int}} = e\frac{2R - r}{2R - 2r} \tag{6.2.3-3}$$

where

$$r = \frac{D_0 - e}{2} \tag{6.2.3-4}$$

c) Where the design stress is time-independent, and the bend radius is less than three times the outside diameter, the intrados thickness shall be not less than that calculated from the following equation:

$$e_{\text{int}} = \max\left(e; \frac{e}{1{,}25}\,\frac{2R - r}{2R - 2r}\right) \tag{6.2.3-5}$$

where

r is calculated in accordance with equation (6.2.3-4).

NOTE Manufacturing processes and experience should be taken into account for the determination of the thickness before bending. Table 6.2.3-1 gives guidance on the pipe wall thickness needed to meet the requirements of 6.2.3.2.

6.2.3.3 More accurate route

The more accurate method of calculating the wall thickness of pipe bends and elbows shall be in accordance with Annex B.

Table 6.2.3-1 — Minimum pipe wall thickness before bending by induction

Radius	Normal route 6.2.3.1	Alternative route 6.2.3.2
10 D_o	1,02 e	1,04 e
8 D_o	1,03 e	1,05 e
6 D_o	1,04 e	1,06 e
5 D_o	1,04 e	1,08 e
4 D_o	1,05 e	1,10 e
3 D_o	1,06 e	1,13 e
2,5 D_o	1,08 e	1,16 e
2 D_o	1,10 e	1,20 e
1,5 D_o	1,15 e	1,25 e

6.3 Mitre bends

6.3.1 General

The following rules for mitre bends (see Figure 6.3.2-1) shall only be used if the following conditions are met:

a) Time independent design stress:

— the calculation pressure p_c is less or equal to 20 bar (2,0 MPa);

b) Time dependent design stress:

— the piping system is fully compensated by expansion joints;

— the internal pressure is limited to 4 bar (0,4 MPa);

— the number of full pressure cycles is limited to 100;

— consideration is given to high temperatures cycling.

A mitre bend with an angle of change in direction at a single joint greater than 22,5 ° (see angle α in Figure 6.3.2-1) shall not be used under cyclic loadings (> 7 000 cycles).

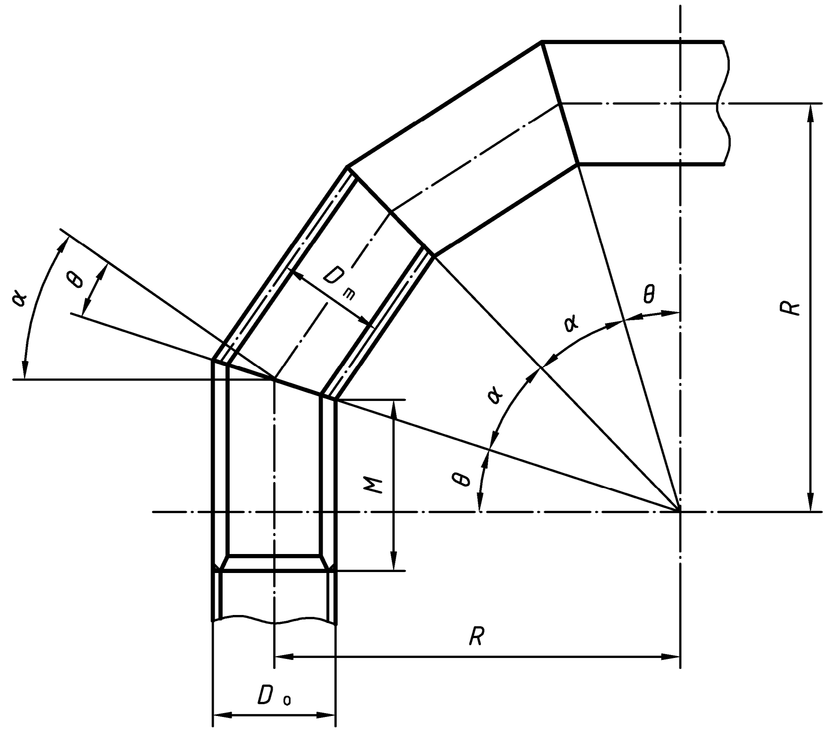
NOTE For an angle of change in direction of 3 ° or less at a single joint, the calculation method given in 6.1 may be used.

6.3.2 Symbols

For the purposes of 6.3, the symbols given in Table 6.3.2-1 shall apply in addition to those given in Table 3.2-1.

Table 6.3.2-1 — Additional symbols for the purposes of 6.3

Symbol	Description	Unit
R	Effective radius of mitre bend (see Figure 6.3.2-1)	mm
l_a	Empirical value as a function of wall thickness e_a (see Table 6.3.3-1)	mm
θ	Angle (see Figure 6.3.2-1)	°
α	Angle of change in direction at mitre joint	°



NOTE $\alpha = 2\,\theta$

Table 6.3.2-1 — Scheme for a mitre bend

6.3.3 Effective radius of mitre bend

The value of R shall not be less than the following

$$R = \frac{l_a}{\tan\theta} + \frac{D_0}{2} \tag{6.3.3-1}$$

where

l_a is given in Table 6.3.3-1.

Table 6.3.3-1 — Empirical values of l_a for given values of e_a

e_a mm	l_a mm
$e_a \leq 13$	25
$13 < e_a < 22$	$2\,e_a$
$e_a \geq 22$	$2/3\,e_a + 30$

6.3.4 Multiple mitre bends

The maximum allowable internal pressure, p_a, for multiple mitre bends (see Figure 6.3.2-1) shall be the lesser of the values calculated from equations (6.3.4-1) and (6.3.4-2) which shall only apply for an angle $\theta \leq 22,5$ °.

$$p_a = \frac{2fze_a}{D_m}\left(\frac{e_a}{e_a + 0{,}643\tan\theta\sqrt{0{,}5D_m e_a}}\right) \quad (6.3.4\text{-}1)$$

$$p_a = \frac{2fze_a}{D_m}\left(\frac{R - 0{,}5D_m}{R - 0{,}25D_m}\right) \quad (6.3.4\text{-}2)$$

6.3.5 Single mitre bends

A single mitre bend is a mitre bend with one angular offset only.

The maximum allowable internal pressure, p_a, for a single mitre bend with an angle θ not greater than 22,5 ° shall be calculated in accordance with 6.3.4.

The maximum allowable internal pressure, p_a, for a single mitre bend with an angle θ greater than 22,5 ° shall be calculated from equation (6.3.5-1).

$$p_a = \frac{2fze_a}{D_m}\left(\frac{e_a}{e_a + 1{,}25\tan\theta\sqrt{0{,}5D_m e_a}}\right) \quad (6.3.5\text{-}1)$$

6.3.6 Adjacent straight pipe sections of mitre bends

The wall thickness shall extend a distance not less than M (see Figure 6.3.2-1) from the inside crotch of the end mitre welds where

$$M = \max\left\{2{,}5\sqrt{0{,}5D_m e_a}\ ;\ \left(R - \frac{D_m}{2}\right)\tan\theta\right\} \quad (6.3.6\text{-}1)$$

6.4 Reducers

6.4.1 Conditions of applicability

Requirements are given in 6.4.4 to 6.4.8 for right circular cones and cone/cylinder intersections where the cone and the cylinder are on the same axis of rotation. Requirements for offset cones are given in 6.4.9.

The requirements do not apply to:

— cones for which the half angle at the apex of the cone is greater that 75°;

— cones for which;

$$\frac{e_a \cos\alpha}{D_c} \leq 0{,}001; \quad (6.4.1\text{-}1)$$

— short cones joining a jacket to a shell.

Limits on the minimum distance from other major discontinuities are given in individual clauses.

6.4.2 Specific definitions

6.4.2.1

junction between the cylinder and the cone
intersection of the mid-thickness lines of cylinder and cone, extended if necessary where there is a knuckle (see Figure 6.4.2-1 and Figure 6.4.2-2 for examples at the large end)

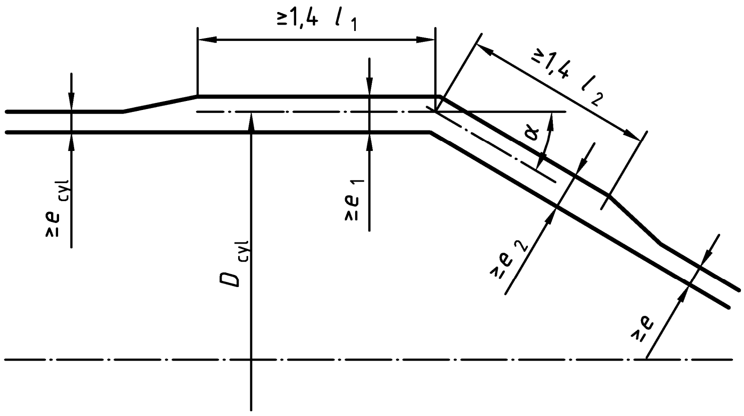


Figure 6.4.2-1 — Geometry of cone/cylinder intersection without knuckle – Large end

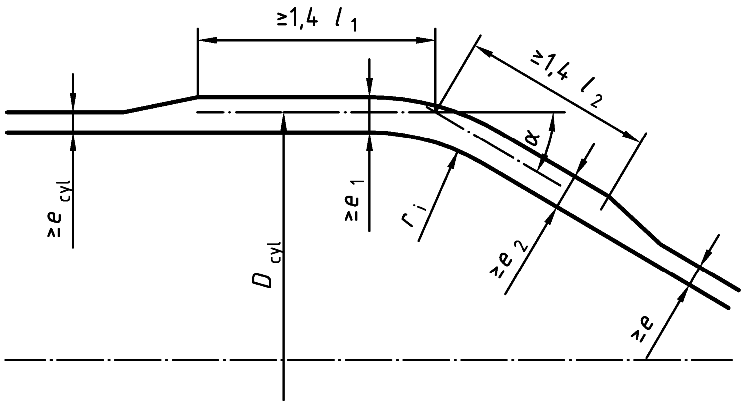


Figure 6.4.2-2 — Geometry of cone/cylinder intersection with knuckle – Large end

6.4.3 Specific symbols and abbreviations

For the purposes of 6.4, the symbols given in Table 6.4.3-1 shall apply in addition to those given in Table 3.2-1.

Table 6.4.3-1 — Additional symbols for the purposes of 6.4

D_c	the mean diameter of the cylinder at the junction with the cone;
D_e	the outside diameter of the cone;
D_i	the inside diameter of the cone;
D_K	a diameter given by equation (6.4.4-7);
D_m	the mean diameter of the cone;
e_{con}	required thickness of cone as determined in 6.4.4;
e_{cyl}	required thickness of cylinder as determined in 6.1;
e_j	a required or analysis thickness at a junction at the large end of a cone;
e_1	required thickness of cylinder at junction;
e_{1a}	analysis reinforcing thickness in cylinder;
e_2	required thickness of cone and knuckle at junction;
e_{2a}	analysis reinforcing thickness in cone;
f	the nominal design stress. In the design of junctions to 6.4.6 to 6.4.9 it is the lowest of the values for the individual component parts;
l_1	length along cylinder;
l_2	length along cone at large or small end;
r_i α	inside radius of knuckle the semi angle of cone at apex (degrees);
β	a factor defined in 6.4.6;
β_H	a factor defined in 6.4.8;
γ	a factor defined in 6.4.7;
ρ	a factor defined in 6.4.7;
τ	a factor defined in 6.4.8.

6.4.4 Conical shells

The required thickness at any point along the length of a cone shall be calculated from one of the following two equations:

$$e_{con} = \frac{p_c\, D_i}{2f\, z - P}\, \frac{1}{\cos\alpha} \qquad (6.4.4\text{-}1)$$

or

$$e_{con} = \frac{p_c\, D_e}{2f\, z + P}\, \frac{1}{\cos\alpha} \qquad (6.4.4\text{-}2)$$

where D_i and D_e are at the point under consideration.

For a given geometry:

$$PS = \frac{2f\, z\, e_{con} \cos\alpha}{D_m} \qquad (6.4.4\text{-}3)$$

where D_m is at the point under consideration.

At the large end of a cone attached to a cylinder it is permissible to make the following substitutions:

$$D_i = D_k \tag{6.4.4-4}$$

$$D_e = D_k + 2e_2 \cos \alpha \tag{6.4.4-5}$$

$$D_m = (D_i + D_e)/2 \tag{6.4.4-6}$$

where

$$D_K = D_c - e_1 - 2r\left(1 - \cos\alpha\right) - l_2 \sin\alpha \tag{6.4.4-7}$$

NOTE 1 The thickness given by this section is a minimum. Thickness may have to be increased at junctions with other components, or to provide reinforcement at nozzles or openings, or to carry non-pressure loads.

NOTE 2 Since the thickness calculated above is the minimum allowable at that point along the cone, it is permissible to build a cone from plates of different thickness provided that at every point the minimum is achieved.

6.4.5 Junctions - general

The requirements of 6.4.6, 6.4.7 and 6.4.8 apply when the junction is more than $2l_1$ along the cylinder and $2l_2$ along the cone from any other junction or major discontinuity, such as another cone/cylinder junction or a flange, where:

$$l_1 = \sqrt{D_c\ e_1} \tag{6.4.5-1}$$

$$l_2 = \sqrt{\frac{D_c\ e_2}{\cos\alpha}} \tag{6.4.5-2}$$

The length of the cone can be reduced to less than $2l_2$ if both of the following conditions are fulfilled:

— the wall thickness e_2, calculated in accordance with 6.4.6 or 6.4.7, is existent along the whole length of the cone;

— the junction at the small end of the cone is sufficiently dimensioned according to 6.4.8.

6.4.6 Junction between the large end of a cone and a cylinder without a knuckle

6.4.6.1 Conditions of applicability

The requirements of 6.4.6.2 and 6.4.6.3 apply provided that all the following conditions are satisfied:

1) the joint is a butt weld where the inside and outside surfaces merge smoothly with the adjacent cone and cylinder without local reduction in thickness; and

2) the weld at the junction shall be subject to 100 % non-destructive examination, either by radiography or ultrasonic techniques, unless the design is such that the thickness at the weld exceeds $1{,}4e_j$, in which case the normal rules for the relevant design shall be applied.

6.4.6.2 Design

The required thickness e_1 of the cylinder adjacent to the junction is the greater of e_{cyl} and e_j where e_j shall be detemined by the following procedure:

$$\beta = \frac{1}{3}\sqrt{\frac{D_c}{e_j}} \cdot \frac{\tan\alpha}{1 + 1/\sqrt{\cos\alpha}} - 0{,}15 \qquad (6.4.6\text{-}1)$$

$$e_j = \frac{p_c\ D_c\ \beta}{2f} \qquad (6.4.6\text{-}2)$$

The answer is acceptable if the value given by equation (6.4.6-2) is not less than assumed in equation (6.4.6-1).

β can also be read from the graph in Figure 6.4.6-1.

This thickness shall be maintained for a distance of at least $1{,}4l_1$ from the junction along the cylinder.

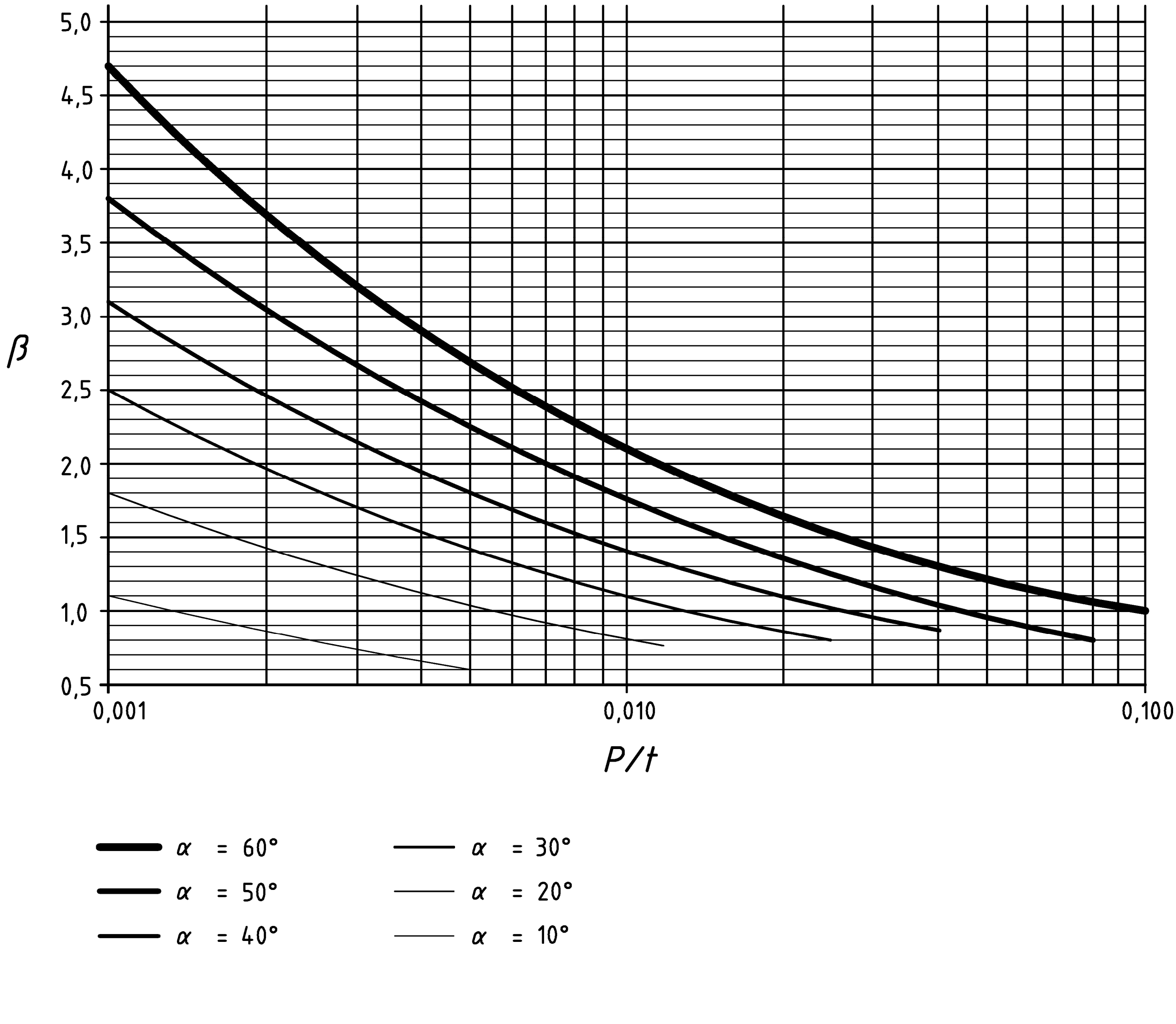


Figure 6.4.6-1 — Values of coefficient β for cone/cylinder intersection without knuckle

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 $1{,}4l_2$ from the junction along the cone, see Figure 6.4.6-1.

It is permissible to redistribute the reinforcement in the following way, provided that the minimum thicknesses given by 6.1 and 6.4.4 continue to be met.

The thickness for the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance 1,4l_1 from the junction is not less than 1,4$e_1 l_1$. In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance 1,4l_2 from the junction is not less than 1,4$e_2 l_2$.

6.4.6.3 Rating

The maximum permissible pressure for a given geometry shall be determined as follows:

a) apply equation (6.4.4-3) to cylinder;

$$PS = \frac{2f\,z\,e_a}{D_c} \qquad (6.4.6\text{-}3)$$

b) apply equation (6.4.4-3) to the cone;

c) determine the analysis reinforcing thickness e_{1a} of the cylinder at the junction;

d) determine the analysis reinforcing thickness e_{2a} of the cone at the junction;

e) apply equation (6.4.4-3) with thickness e_{2a} and diameter D_m;

f) find e_j, the lesser of e_{1a} and e_{2a};

g) calculate β from equation (6.4.6-1), then,

$$P_{max} = \frac{2f\,e_j}{\beta\,D_c} \qquad (6.4.6\text{-}4)$$

h) the maximum permissible pressure is the lowest of the pressures determined in a), b), e) and g).

NOTE The following procedure may be used to find the analysis reinforcing thickness at c) or d) above:

1) Assume e_{1a} (the initial choice should be the thickness at the junction).

2) Calculate

$$l_1 = 1{,}4\sqrt{D_C\,e_{1a}} \qquad (6.4.6\text{-}5)$$

3) If the thickness is constant within the distance l_1 then e_{1a} is confirmed.

4) If not, calculate the metal area A_1 within the distance l_1 from the junction.

5) Obtain a better estimate by.

$$e_{1a} = A_1 / l_1 \qquad (6.4.6\text{-}6)$$

The answer is acceptable if it is not greater than e_{1a} assumed in 1).

6) If the answer is unacceptable, return to 1).

7) Use a similar procedure to find e_{2a} making.

$$l_2 = 1{,}4\sqrt{\frac{D_c\, e_{2a}}{\cos\alpha}} \tag{6.4.6-7}$$

6.4.7 Junction between the large end of a cone and a cylinder with a knuckle

6.4.7.1 Conditions of applicability

This sub-clause applies provided that all the following conditions are satisfied:

a) the knuckle is of toroidal form and merges smoothly with the adjacent cone and cylinder; and

b) the inside radius of curvature of the knuckle, $r_i < 0{,}3\ D_c$.

NOTE This clause does not prescribe a lower limit to the radius of curvature of the knuckle.

6.4.7.2 Design

The value of e_j shall be determined by the following procedure:

Assume a value of e_j and calculate:

$$\beta = \frac{1}{3}\sqrt{\frac{D_c}{e_j}} \cdot \frac{\tan\alpha}{1 + 1/\sqrt{\cos\alpha}} - 0{,}15 \tag{6.4.7-1}$$

$$\rho = \frac{0{,}028 r_i}{\sqrt{D_c\, e_j}}\ \frac{\alpha}{1 + 1/\sqrt{\cos\alpha}} \tag{6.4.7-2}$$

$$\gamma = 1 + \frac{\rho}{1{,}2\left(1 + \frac{0{,}2}{\rho}\right)} \tag{6.4.7-3}$$

$$e_j = \frac{p_c\ D_c\ \beta}{2f\ \gamma} \tag{6.4.7-4}$$

The answer is acceptable if the value given by equation (6.4.7-4) is not less than that assumed.

The required thickness e_1 of the cylinder adjacent to the junction is the greater of e_{cyl} and e_j.

This thickness shall be maintained for a distance of at least $1{,}4 l_1$ from the junction and $0{,}5 l_1$ from the knuckle/cylinder tangent line along the cylinder.

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.

6.4.7.3 Rating

The maximum permissible pressure for a given geometry shall be found as follows.

a) Determine e_{1a}, the analysis thicknesses for the cylinder next to the knuckle, and e_{2a}, the analysis thickness for the knuckle and the adjacent part of the cone;

b) Check that the limitations of 6.4.7.1 are met;

c) Apply equation (6.4.6-3) to the cylinder with $e_a = e_{1a}$;

d) Apply equation (6.4.4-3) to the cone with $e_{con} = e_{2a}$;

e) Find e_j, the lesser of e_{1a} and e_{2a};

f) Find β and Υ from equations (6.4.7-1) and (6.4.7-3), then

$$PS = \frac{2f\ \gamma\ e_j}{\beta\ D_c} \tag{6.4.7-5}$$

g) The maximum permissible pressure is the lowest of the pressures determined in c), d) and f).

6.4.8 Junction between the small end of a cone and a cylinder

6.4.8.1 Conditions of applicability

The requirements of 6.4.8.2 and 6.4.8.3 apply provided that all the following conditions are satisfied:

a) the required thickness of the cylinder e_1 is maintained for a distance l_1 and that of the cone e_2 is maintained for a distance l_2 from the junction (see Figure 6.4.8-1); and

b) the thicknesses meet the requirements of 6.1 and 6.4.4;

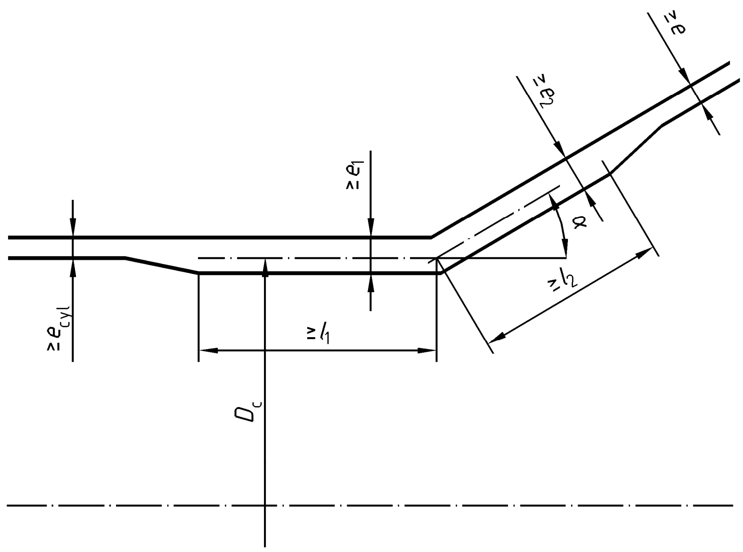


Figure 6.4.8.1-1 — Geometry of cone/cylinder intersection: small end

6.4.8.2 Design

Required thicknesses e_1 and e_2 shall be found by the following procedure:

Assume values of e_1 and e_2:

$$s = \frac{e_2}{e_1} \tag{6.4.8-1}$$

when $s < 1$

$$\tau = s\sqrt{\frac{s}{\cos\alpha}} + \sqrt{\frac{1+s^2}{2}} \tag{6.4.8-2}$$

when $s \geq 1$

$$\tau = 1 + \sqrt{s\left\{\frac{1+s^2}{2\cos\alpha}\right\}} \tag{6.4.8-3}$$

$$\beta_H = 0{,}4\sqrt{\frac{D_c}{e_1}}\frac{\tan\alpha}{\tau} + 0{,}5 \tag{6.4.8-4}$$

If

$$p_c \leq \frac{2f\ z\ e_1}{D_c\ \beta_H} \tag{6.4.8-5}$$

then e_1 and e_2 are acceptable. If not, repeat with increased values of e_1 and/or e_2.

NOTE The above procedure does not provide values for e_1 and e_2 independently. Any values may be selected to suit the needs of the design, for example to obtain a favourable value of l_1 or l_2.

Provided that the requirements of 6.1 and 6.4.4 continue to be met, it is permissible to modify a design according to the above rule in one of the following ways:

a) Where $e_1 = e_2$ a knuckle of the same thickness may be included. l_1 and l_2 continue to be measured from the junction (i.e. the point where the centre lines of cone and cylinder meet);

b) The thickness of the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance l_1 from the junction is not less than $l_1\ e_1$. In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance l_2 from the junction is not less than $l_2 \cdot e_2$.

6.4.8.3 Rating

The maximum permissible pressure for a given geometry shall be:

$$PS = \frac{2f\ z\ e_1}{D_c \beta_H} \tag{6.4.8-6}$$

β_H is found from equations (6.4.8-1) to (6.4.8-4) using e_{1a} and e_{2a} in place of e_1 and e_2 .

NOTE 1 The procedure for finding e_{1a} and e_{2a} is as provided in the note to 6.4.6.3.

NOTE 2 Analysis thicknesses may exceed the required thickness without leading to any increase in l_1 or l_2.

6.4.9 Offset reducers

This sub-clause shall apply to offset reducers (see Figure 6.4.9-1). The cylindrical parts shall have parallel centre lines offset from each other by a distance d_{off} not greater than the difference of their radius. The thicknesses shall be calculated in accordance with 6.4.6 or 6.4.7 for the junction at the large end. The minimum thicknesses shall be calculated in accordance with 6.4.8 for the junction at the small end. The greater of these shall apply to the whole reducer. The angle α shall be taken as the greatest angle between the conical and cylindrical parts.

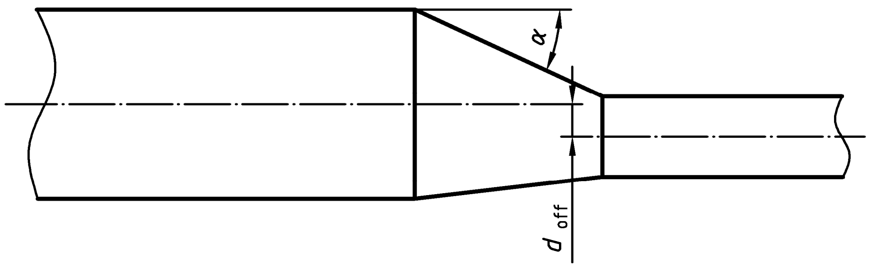


Figure 6.4.9-1 — Offset reducer

6.4.10 Special forged reducers

Special forged reducers, for example for very high temperature and/or very high internal pressure, which are not covered by the product standards may be designed as shown in Figure 6.4.10-1 where:

$$l_l \geq \sqrt{\frac{D_l\ e_l}{\cos\alpha}} \tag{6.4.10-1}$$

$$l_s \geq \sqrt{\frac{D_s e_s}{\cos\alpha}} \tag{6.4.10-2}$$

$$r \geq 10 \text{ mm} \tag{6.4.10-3}$$

$$r' \geq 100 \text{ mm} \tag{6.4.10-4}$$

$$e_{r=\max}\{e_{cyl}; e_j\} \tag{6.4.10-5}$$

With e_{cyl} according to 6.1 and e_j according to equation (6.4.7-4).

In such cases additional design calculations are not required.

Where reducers of other design are supplied, their suitability shall be demonstrated.

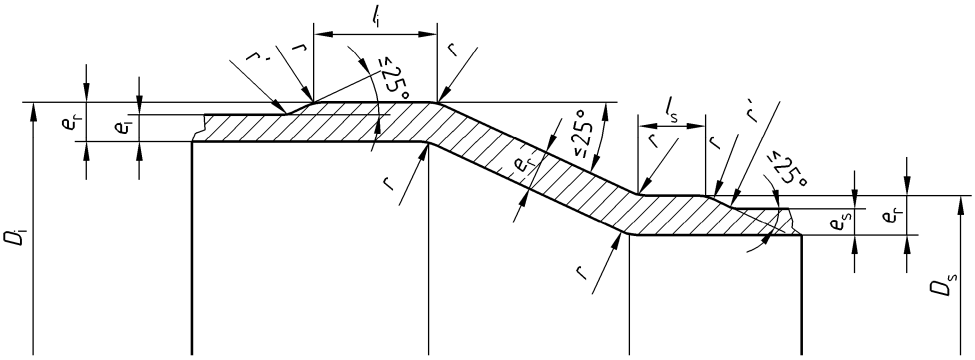


Figure 6.4.10-1 — Special forged reducer

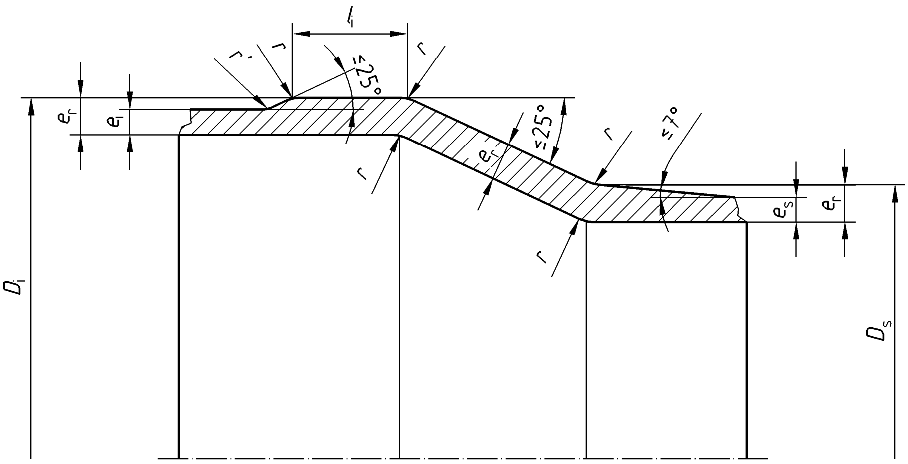


Figure 6.4.10-2 — Special forged reducer (alternative solution)

6.5 Flexible piping components

6.5.1 General

This sub-clause deals with bellows expansion joints and braided corrugated hose assemblies which are used to absorb relative movements and misalignments of piping or plant components, as well as to reduce forces and moments, i.e. stresses, in the pipes and their connections.

The manufacturer of such components has the responsibility for their correct design and manufacture.

6.5.2 Expansion joints

A piping system containing a bellows expansion joint is critically dependent upon a proper combination of each and every component. This requires that detailed consideration shall be given to the system, its supports and anchors and their interaction with the bellows expansion joints. The bellows expansion joints should not be considered as a commodity item.

For design and incorporation of expansion joints, see Annex C.

In order to design the expansion joints the following data shall be supplied:

- type of expansion joint (axial, angular, lateral, universal);
- end fitting (weld end, flange);
- nominal diameter DN or dimensions of end fittings (e.g. diameter, wall thickness);
- operating pressure or calculation pressure;
- operating temperature or calculation temperature;
- movements of the expansion joint:
  - total axial movement;
  - total angular rotation;
  - total lateral movement;
- presetting (magnitude, direction);
- number of load cycles (movement, pressure);
- fluid (type, density, additives);
- fluid velocity;
- material requirements;
- additional loads (see 4.2.4).

The individual movements may also be specified as a combined movement from which the relevant design shall be determined by the manufacturer.

NOTE The corrugations of metallic expansion joints generally have a wall thickness substantially less than that of the equipment in conjunction with which they are used. It is therefore essential to select material for their manufacture having adequate resistance to all the corrosive agents likely to be encountered in any particular application.

The manufacturer of the expansion joint shall provide, when requested, the following information for system analysis:

- reaction force rates for all directions of movement;
- reaction moment rates for all rotation axes;
- forces and moments resulting from the friction in the bearings of the hinges;
- pressure thrust, the pressure-induced axial force of unrestrained expansion joints, acting on anchors.

6.5.3 Corrugated metal hose assemblies

Braided pressurized corrugated hose assemblies are self restrained axially and do not exert pressure loads on adjacent pipework.

Straight hose assemblies shall not be used for axial movement. Movement should be accommodated laterally or by use of bends or loops (see Figures 6.5.3-1, 6.5.3-2, 6.5.3-3)

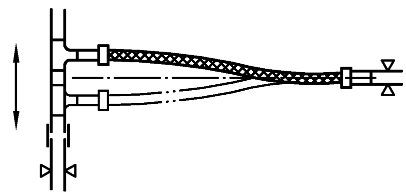


Figure 6.5.3-1

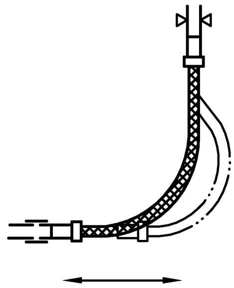


Figure 6.5.3-2

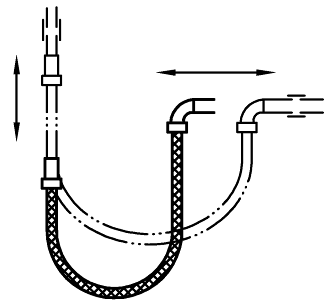


Figure 6.5.3-3

To design a corrugated hose assembly the following design date shall be supplied to and taken into consideration by the manufacturer:

- description of intended application;
- nominal size of the hose assembly;
- maximum operating pressure (internal and/or external);
- vacuum conditions where applicable;
- ambient and maximum operating temperature;
- materials specification;
- fluid to be conveyed;
- Information about possible corrosion, erosion, abrasion;
- fluid velocity;
- movements and/or vibrations (including torsion if any);
- specified life (e.g. number of cycles);
- type of fitting for hose assembly;
- external or internal protection if required;
- special requirements for heat treatment and/or cleaning;
- any other parameter or loadings which can influence design and life expectancy (e.g. water hammer).

Where corrugated metal hose assemblies can be subjected to different working conditions (e.g. for normal and cleaning operation) all these conditions shall be made known to and taken into consideration by the manufacturer.

6.6 Bolted flange connections

6.6.1 General

The rules of this sub-clause are to check the mechanical resistance of the flange connection subjected to static loads. It is also in the responsibility of the designer to ensure the adequacy of the flange connection (gasket type and characteristics, etc) with the operating conditions, in particular with regards to any specific required tightness.

If there is a specific requirement on tightness for the flange connection, this shall be calculated in accordance with EN 1591-1, using Annex P.

The designer shall consider section loadings caused by the connected piping system.

The classification of material for flanges, bolts and nuts is given by EN 1515-2 (PN flanges) and EN 1515-3 (Class flanges). The selection of bolting shall comply with Annex D or Annex P and EN 1515-4.

6.6.2 Symbols

For the purposes of 6.6, the symbols given in Table 6.6.2-1 shall apply in addition to those given in Table 3.2-1.

Table 6.6.2-1 — Additional symbols for the purposes of 6.6

Symbol	Description	Unit
P_{eq}	Equivalent design pressure	MPa (N/mm^2)
P	Internal calculation pressure	MPa (N/mm^2)
F	Pulling axial force (to be a positive value in equation)	N
M	External bending moment	N mm
G	Diameter of gasket load reaction	mm

6.6.3 Standard flange

A standard steel flange connection in accordance with defined material requirements, giving the maximum allowable pressure with regards to the flange materials and the design temperature, may be used within the construction of piping subjected to internal pressure, without the necessity of carrying out a calculation to verify its resistance when the following conditions are met:

a) For each normal working condition, the design pressure shall not exceed the maximum allowable pressure specified.

b) For conditions where the flange connection is simultaneously subjected to internal pressure, axial load and bending moment, the equivalent design pressure, P_{eq}, according to equation (6.6.2-1) shall not exceed the limits specified in a).

$$P_{eq} = P + \frac{4F}{\pi G^2} + \frac{16|M|}{\pi G^3} \tag{6.6.2-1}$$

Where:

G is the diameter of circle on which applies the compression load of the gasket (normally the mean diameter of the gasket).

c) The gasket types, for each PN, are specified in EN 1514-1 to EN 1514-8.

d) The strength of the bolting for the flange connection, for each PN, shall be as indicated in EN 1515-1 to 4.

e) The difference of temperature between the flanges and the bolting shall not exceed, 50 °C in any case.

f) If the design temperature is ≥ 120 °C, the thermal expansion coefficient of the flange material shall not exceed the thermal expansion coefficient of the bolt material by more than 10 %.

6.6.4 Non-standard flange

If a non-standard flange is used, the design shall be done by applying the calculation method in EN 1591-1, using for example Annex P, or by applying the algorithm shown in the Taylor-Forge method, using for example Annex D.

NOTE 1 The Taylor-Forge method does not ensure tightness.

NOTE 2 The algorithm given in EN 1591-1 includes a consideration of section loadings.

NOTE 3 The bolt torque should be specified by the designer. Attention should be paid in such cases to the method of tightening. Guidance of scatter band when applying the different methods of tightening are given in EN 1591-1.

7 Design of ends under internal pressure

7.1 Dished ends

7.1.1 Symbols

For the purposes of 7.1, the symbols given in Table 7.1.1-1 shall apply in addition to those given in Table 3.2-1:

Table 7.1.1-1 — Additional symbols for the purposes of 7.1

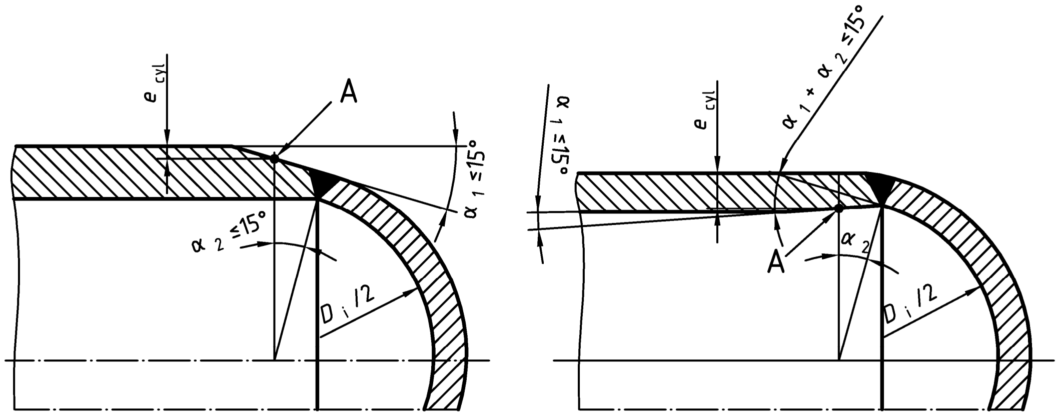
Symbol	Description	Unit
e_{kn}	thickness of knuckle	mm
e_s	minimum thickness of end to limit membrane stress in spherical part	mm
$e_{kn\ y}$	minimum thickness of knuckle to avoid axisymetric yielding	mm
$e_{kn\ b}$	minimum thickness of knuckle to avoid plastic buckling	mm
D_o	outside diameter of end	mm
D_i	inside diameter of end	mm
h_i	inside height of ellipsoidal end	mm
K	shape factor for ellipsoidal end	-
R_i	inside spherical radius for torispherical end	mm
r_i	inside knuckle radius	mm
p_s	maximum pressure of end to limit membrane stress in spherical part	MPa (N/mm²)
$p_{kn\ y}$	maximum pressure of knuckle to avoid axisymetric yielding	MPa (N/mm²)
$p_{kn\ b}$	maximum pressure of knuckle to avoid plastic buckling	MPa (N/mm²)

7.1.2 Hemispherical ends

The minimum required thickness of a hemispherical end shall be given by the following equation:

$$e = \frac{p_c D_i}{4fz - p_c} \tag{7.1.2-1}$$

The thickness of the cylindrical part, e_{cyl}, shall be not less than the minimum thickness of the connected pipe calculated in accordance with 6.1 up to the point A shown in Figure 7.1.2-1 a) and b).



a) External chamfer

b) Internal chamfer

Figure 7.1.2-1 — Hemispherical ends

7.1.3 Torispherical ends

This sub-clause shall apply provided that the following conditions are simultaneously fulfilled:

$r_i \leq 0{,}2D_i$

$r_i \geq 0{,}06D_i$

$r_i \geq 2e$

$0{,}001D_i \leq e \leq 0{,}08D_i$

$R_i \leq D_o$

If $e < 0{,}003D_i$ then the method shall only be applicable:

— for carbon steel and austenitic stainless steel ends; and

— with a calculation temperature $t_c \leq 100$ °C.

The minimum required thickness e shall be the greatest of e_s, $e_{kn\,y}$ and $e_{kn\,b}$, where

$$e_{\mathrm{s}} = \frac{p_{\mathrm{c}} R_{\mathrm{i}}}{2 f z - 0{,}5 p_{\mathrm{c}}} \tag{7.1.3-1}$$

$$e_{\mathrm{kn\,y}} = \frac{\beta p_{\mathrm{c}} \left(0{,}75 R_{\mathrm{i}} + 0{,}2 D_{\mathrm{i}}\right)}{f} \tag{7.1.3-2}$$

where

β is calculated in accordance with 7.1.5

and

$$e_{\mathrm{kn\,b}} = \left(0{,}75 R_{\mathrm{i}} + 0{,}2 D_{\mathrm{i}}\right) \left\{ \frac{p_{\mathrm{c}}}{111 f_{\mathrm{b}}} \left(\frac{D_{\mathrm{i}}}{r_{\mathrm{i}}}\right)^{0{,}825} \right\}^{\frac{1}{1{,}5}} \tag{7.1.3-3}$$

where

f_{b} is the design stress to prevent buckling:

— for all materials except cold formed austenitic stainless steel:

$$f_{\mathrm{b}} = f \tag{7.1.3-4}$$

— for cold formed austenitic stainless steel:

$$f_{\mathrm{b}} = 1{,}6 f \tag{7.1.3-5}$$

NOTE The factor of 1,6 takes account of the benefit of strain hardening.

For a given geometry, the maximum pressure *PS* shall be the minimum of p_{s}, $p_{\mathrm{kn\,y}}$ and $p_{\mathrm{kn\,b}}$,

where

$$p_{\mathrm{s}} = \frac{2 f e_{\mathrm{a}} z}{R_{\mathrm{i}} + 0{,}5 e_{\mathrm{a}}} \tag{7.1.3-6}$$

$$p_{\mathrm{kn\,y}} = \frac{f e_{\mathrm{a}}}{\beta \left(0{,}75 R_{\mathrm{i}} + 0{,}2 D_{\mathrm{i}}\right)} \tag{7.1.3-7}$$

where β is calculated in accordance with 7.1.5

$$p_{\mathrm{kn\,b}} = 111 f_{\mathrm{b}} \left(\frac{e_{\mathrm{a}}}{0{,}75 R_{\mathrm{i}} + 0{,}2 D_{\mathrm{i}}}\right)^{1{,}5} \left(\frac{r_{\mathrm{i}}}{D_{\mathrm{i}}}\right)^{0{,}825} \tag{7.1.3-8}$$

NOTE Where $e_{\mathrm{kn\,y}} > 0{,}005\, D_{\mathrm{i}}$, it is not necessary to calculate $e_{\mathrm{kn\,b}}$ or $p_{\mathrm{kn\,b}}$.

It shall be permissible to reduce the thickness of the spherical part of the end to the value of e_{s} over a circular area that is not closer to the knuckle than the distance $\sqrt{R_{\mathrm{i}} e_{\mathrm{kn}}}$ as shown in Figure 7.1.3-1.

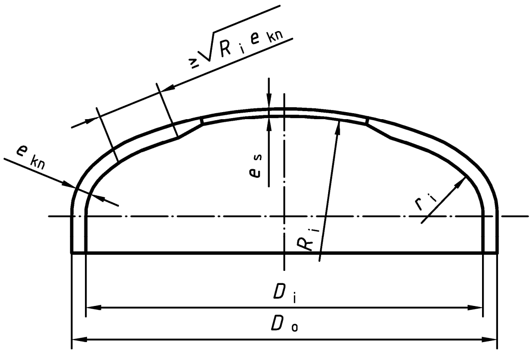


Figure 7.1.3-1 — Torispherical ends

Any cylinder skirt shall meet the requirements of 6.1 for a straight pipe, unless its length is not greater than $0{,}2\sqrt{D_i e_{kn}}$, in which case its thickness and that of the knuckle may be the same.

7.1.4 Ellipsoidal ends

This sub-clause shall apply to ends for which 1,7 < K < 2,2 and z = 1,

where

$K = \dfrac{D_i}{2\ h_i}$ (see Figure 7.1.4-1).

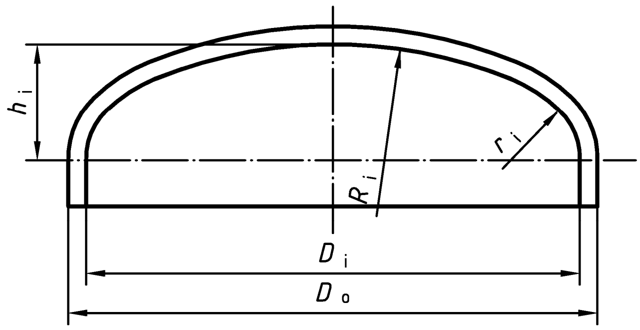


Figure 7.1.4-1 — Ellipsoidal ends

Ellipsoidal ends shall be designed as nominally equivalent torispherical ends with:

$$r_i = D_i\left(0{,}5/K - 0{,}08\right) \qquad (7.1.4\text{-}1)$$

and

$$R_i = D_i\left(0{,}44K + 0{,}02\right) \qquad (7.1.4\text{-}2)$$

7.1.5 Calculation of β

β shall be calculated from the following equations:

$$Y = \min\left(e / R_\mathrm{i} ; 0{,}04\right) \tag{7.1.5-1}$$

$$Z = \log\left(1 / Y\right) \tag{7.1.5-2}$$

$$X = r_\mathrm{i} / D_\mathrm{i} \tag{7.1.5-3}$$

$$N = \left(1{,}006 - \frac{1}{6{,}2 + \left(90Y\right)^4}\right) \tag{7.1.5-4}$$

For $X = 0{,}06$

$$\beta = \beta_{0{,}06} = N\left(-0{,}3635\ Z^3 + 2{,}2124\ Z^2 - 3{,}2937\ Z + 1{,}8873\right) \tag{7.1.5-5}$$

For $0{,}06 < X < 0{,}1$,

$$\beta = 25\left[\left(0{,}1 - X\right)\beta_{0{,}06} + \left(X - 0{,}06\right)\beta_{0{,}1}\right] \tag{7.1.5-6}$$

For $X = 0{,}1$

$$\beta = \beta_{0{,}1} = N\left(-0{,}1833\ Z^3 + 1{,}0383\ Z^2 - 1{,}2943\ Z + 0{,}8370\right) \tag{7.1.5-7}$$

For $0{,}1 < X < 0{,}2$,

$$\beta = 10\left[\left(0{,}2 - X\right)\beta_{0{,}1} + \left(X - 0{,}1\right)\beta_{0{,}2}\right] \tag{7.1.5-8}$$

For $X = 0{,}2$,

$$\beta = \beta_{0{,}2} = \max\left[\left(0{,}532 - 1{,}843\ Y - 78{,}375\ Y^2\right), \left(0{,}5\right)\right] \tag{7.1.5-9}$$

NOTE β may be taken from Figures 7.1.5-1 and 7.1.5-2.

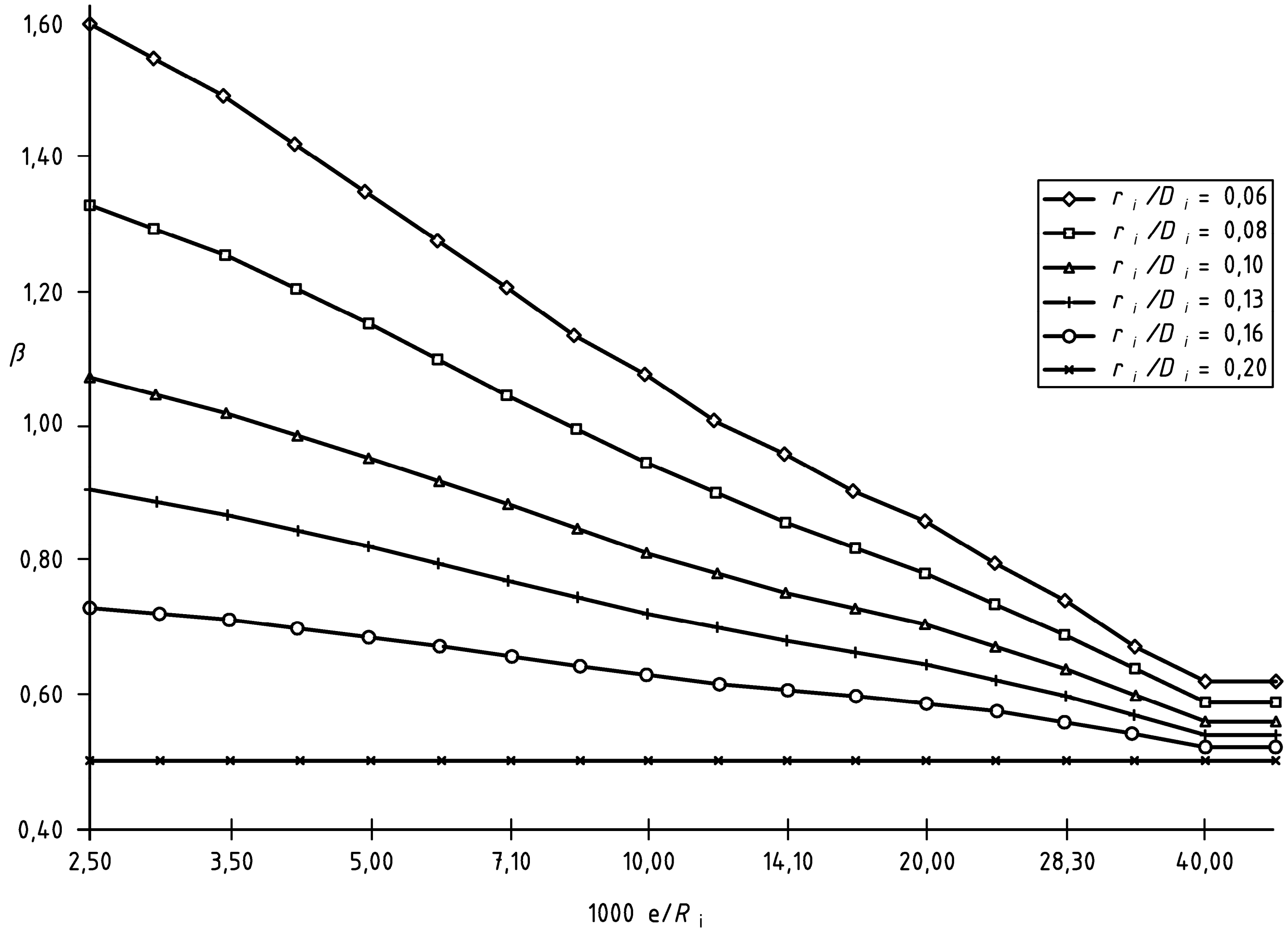


Figure 7.1.5-1 — Torispherical end design

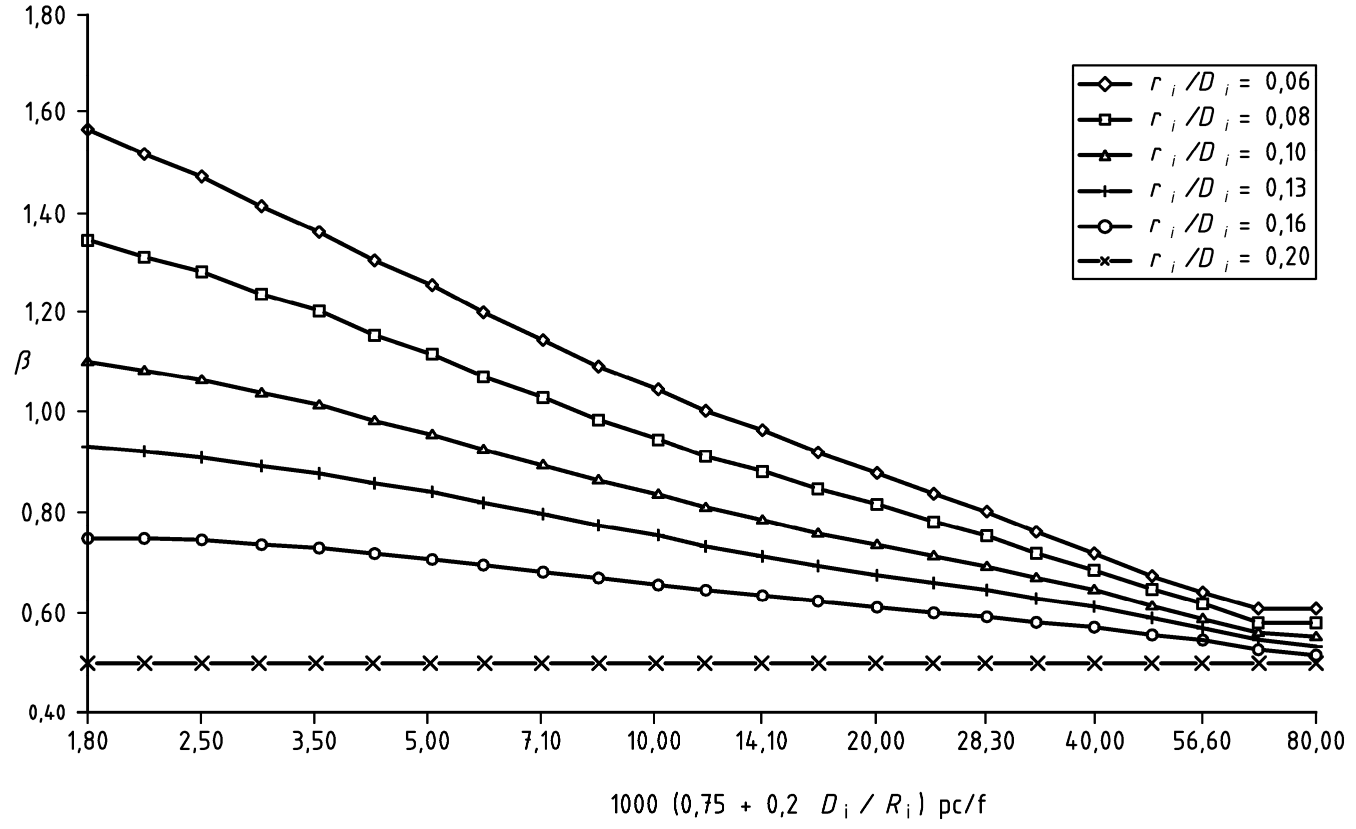


Figure 7.1.5-2 — Torispherical end rating

7.2 Circular flat ends

7.2.1 General

This sub-clause shall apply to the design of circular flat ends connected to piping, either by welding or by bolting, and take into account reinforcement of openings.

7.2.2 Symbols

For the purposes of 7.2 the symbols given in Table 7.2.2-1 shall apply in addition to those given in Table 3.2-1.

Table 7.2.2-1 — Additional symbols for the purposes of 7.2

Symbol	Description	Unit
b	Effective width of the gasket	mm
d	Diameter of the opening, or equivalent diameter of openings fitted with nozzles	mm
d_i	Nozzle inside diameter	mm
d_o	Nozzle outside diameter	mm
e	Minimum required wall thickness of the end without opening	mm
$e_{a\ b}$	Analysis wall thickness of the nozzle	mm
$e_{r\ b}$	Minimum required wall thickness of nozzle under internal pressure	mm
e_{op}	Minimum required wall thickness of end including reinforcement of opening	mm
$e_{a\ f}$	Analysis wall thickness of a flat end with a flanged edge	mm
e_{eq}	Equivalent thickness of the cylindrical shell close to the end	mm
e_{rg}	Minimum required wall thickness of the flat end at the stress-relief groove	mm
e_1	Minimum required wall thickness for the peripheral area of the end	mm
e_A	Minimum required wall thickness of the end for the gasket seating condition	mm
e_p	Minimum required wall thickness of the end for each pressure conditions	mm
f_1	Nominal design stress of end material	MPa (N/mm²)
f_2	Nominal design stress of material used for the cylindrical shell	MPa (N/mm²)
f_A	Nominal design stress of the material of the end for the gasket seating condition	MPa (N/mm²)
h	Distance between centre of opening and pipe inside diameter	mm
l	Length of nozzle contributing to reinforcement	mm
l_{cyl}	Length of cylindrical shell, measured as shown in Figures 7.2.3-1	mm
m	Gasket factor	-
r_i	Inside radius of the flanged edge	mm
y	Gasket seating pressure (see Table 7.2.4-1)	MPa (N/mm²)
A_r	Nozzle reinforcement area	mm
D_i	Inside diameter of the cylindrical shell/pipe When the thickness of the cylindrical shell is not constant close to the end, D_i is the inside diameter of the length of shell with equivalent thickness e_{eq}.	mm
D_{eq}	Equivalent diameter of a flat end with a flanged edge as shown in Figure 7.2.3-1	mm
D_p	Mean diameter of gasket	mm
D_t	Diameter of the bolt circle	mm
F_A	Tensile force of bolts for the gasket seating condition	N
K	Distance between centre lines of two adjacent openings	mm
Y_1	Calculation coefficient of opening reinforcement	-
Y_2	Another calculation coefficient of opening reinforcement	-

7.2.3 Unstayed flat circular ends welded to cylindrical shells/pipes

7.2.3.1 General

This sub-clause shall be used to determine the wall thickness of unstayed flat ends without an opening and welded to the end of a pipe.

The following ends are covered in 7.2.3:

a) flat ends with flanged edge joined to cylindrical shell by butt welding as shown in Figures 7.2.3-1;

b) flat ends joined to cylindrical shell by fillet welds, partial penetration welds and full penetration welds as shown in Figure 7.2.3-3;

c) flat ends with peripheral stress-relief grooves as shown in Figure 7.2.3-5.

NOTE The failure modes covered by 7.2.3 are gross plastic deformation at the centre of the ends, and incremental collapse of the pipe at its junction with the end.

7.2.3.2 Flanged flat ends

The minimum required wall thickness for a flat end with a flanged edge shall be given by the following equation

$$e = C_1 D_{eq} \sqrt{\frac{p_c}{f_1}} \tag{7.2.3-1}$$

where

C_1 is a factor determined from Figure 7.2.3-2 or the following equation:

$$C_1 = \max\left(0{,}40825 A_1 \frac{D_i + e_{eq}}{D_i} ; 0{,}299\left(1 + 1{,}7\frac{e_{eq}}{D_i}\right)\right) \tag{7.2.3-2}$$

where $A_1 = B_1\left(1 - B_1 \dfrac{e_{eq}}{2(D_i + e_{eq})}\right)$ (7.2.3-3)

$$B_1 = 1 - \frac{3f}{p_c}\left(\frac{e_{eq}}{(D_i + e_{eq})}\right)^2 + \frac{3}{16}\left(\frac{D_i}{(D_i + e_{eq})}\right)^4 \frac{p_c}{f_1} - \frac{3}{4}\frac{(2D_i + e_{eq})e_{eq}^{\ 2}}{(D_i + e_{eq})^3} \tag{7.2.3-4}$$

The flanged edge of the flat end shall be joined to the cylindrical part by a circumferential butt weld.

This equation shall apply where the radius $r_i > e_{a\,f}$. In all other cases, flanged ends shall be calculated as an unflanged flat end (see 7.2.3.3).

The mid-thickness lines may be offset, without however going beyond the alignment of the internal or external faces. The surface shall be blended with an angle not exceeding 30 °.

The analysis thickness of the cylindrical part shall be at least equal to e_{eq} over a length equal to:

$$l_{cyl} = 0{,}5\sqrt{\left(D_i + e_{eq}\right)e_{eq}} \tag{7.2.3-5}$$

Where the thickness of the cylindrical part is not constant close to the end, the equivalent thickness over the length shall be at least equal to e_{eq} as shown in Figure 7.2.3-1b and 7.2.3-1c, where A_i = A_o.

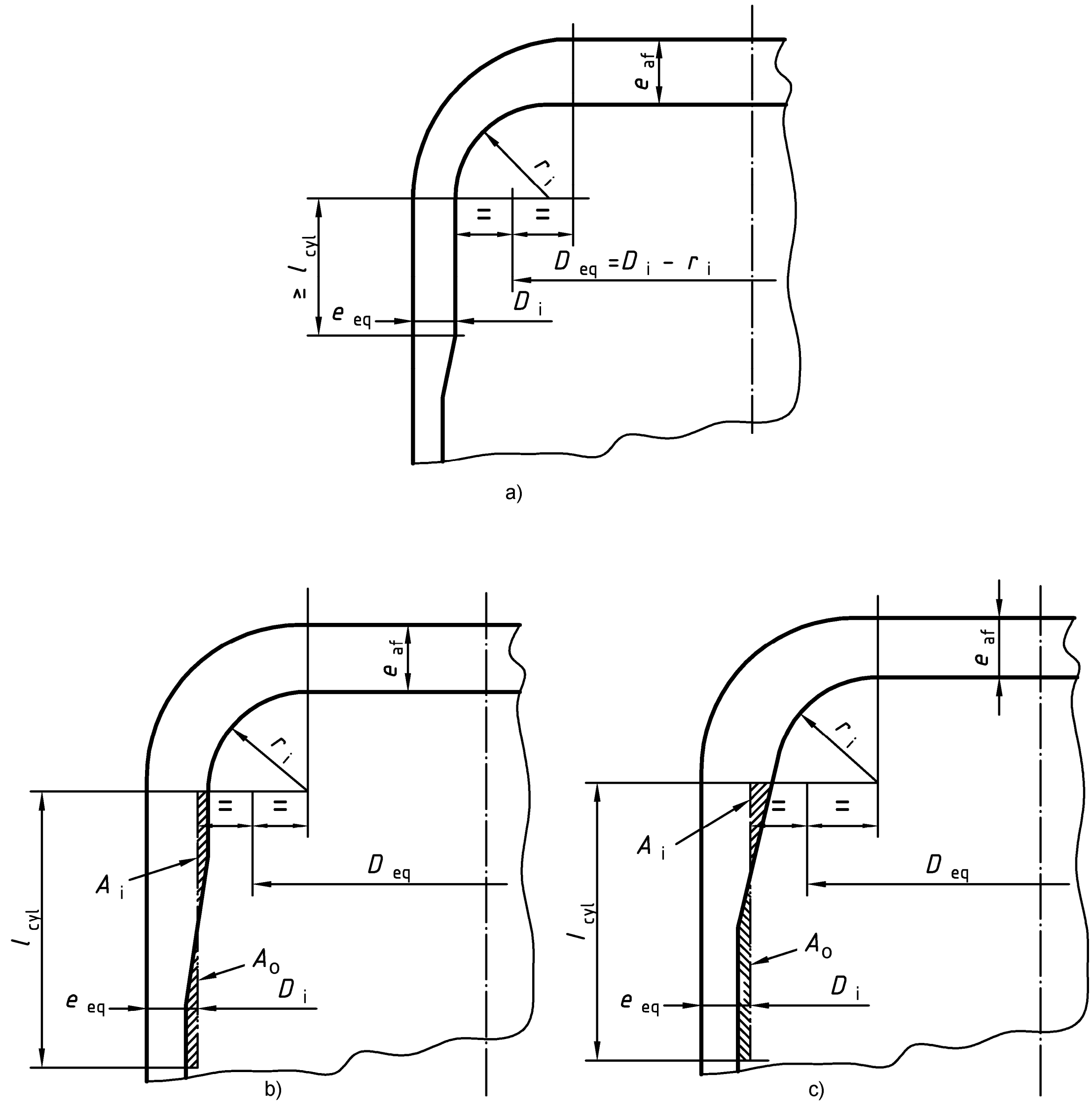
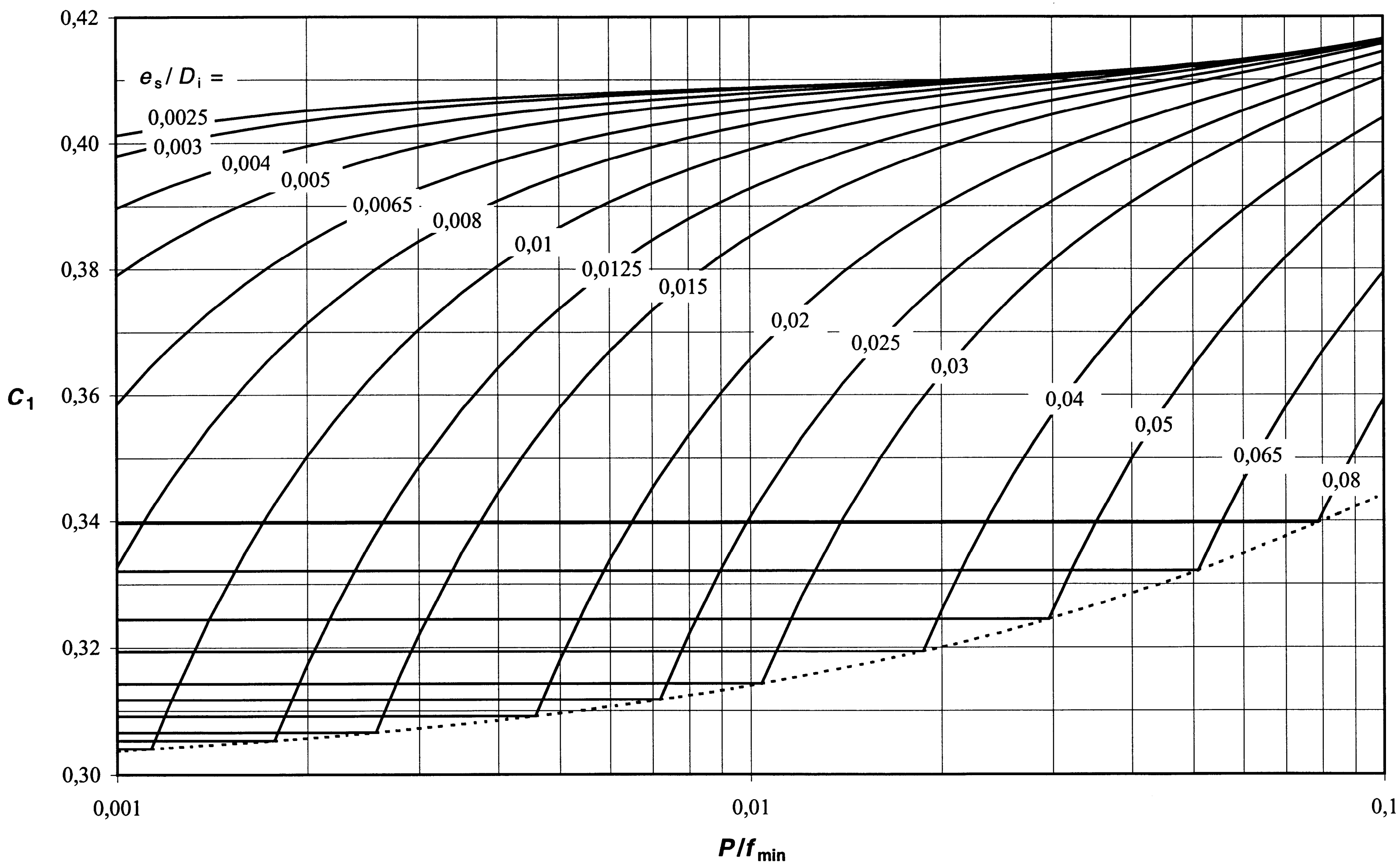


Figure 7.2.3-1 — Flanged flat ends

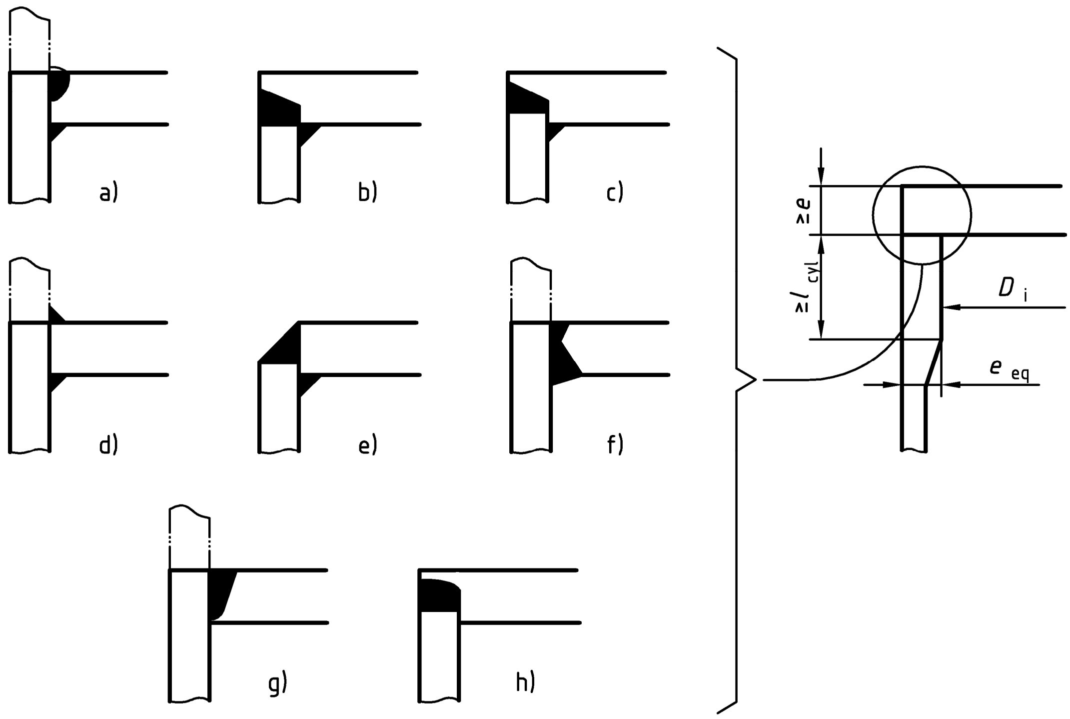


NOTE Where the line for a constant, given P/f_{min} does not intersect the relevant curve for constant e_s/D_i, C_1 is given by the bottom curve for the given P/f_{min}.

Figure 7.2.3-2 — Factor C_1

7.2.3.3 Unflanged flat ends

This sub-clause shall be used for unflanged flat ends (see Figure 7.2.3-3).



NOTE 1 The weld details are given in EN 13480-4.

NOTE 2 For design b), c) and h) attention should be given to sufficient strength values in normal-to-surface direction. Verification tests for absence of lamination are required.

Figure 7.2.3-3 — Corner joints at unflanged flat ends

The minimum required wall thickness of an unflanged flat end shall be given by the following equations:

— for normal operating conditions

$$e = \max\left(C_1 D_i \sqrt{\frac{p_c}{f_1}}\ ;\ C_2 D_i \sqrt{\frac{p_c}{f_{min}}} \right) \tag{7.2.3-6}$$

— for exceptional operating or test conditions

$$e = C_1 D_i \sqrt{\frac{p_c}{f_1}} \tag{7.2.3-7}$$

where

f_{min} is the minimum of f_1 and f_2;

C_1 is the coefficient determined from Figure 7.2.3-2, using the value f_{min} for f_1;

C_2 is the coefficient determined from Figure 7.2.3-4 or the following equations:

$$g = \frac{D_\mathrm{i}}{D_\mathrm{i} + e_\mathrm{eq}} \tag{7.2.3-8}$$

$$H = \sqrt[4]{12\left(1-\nu^2\right)}\sqrt{\frac{e_\mathrm{eq}}{D_\mathrm{i} + e_\mathrm{eq}}} \tag{7.2.3-9}$$

$$J = \frac{3 f_\mathrm{min}}{p_\mathrm{c}} - \frac{D_\mathrm{i}^2}{4\left(D_\mathrm{i} + e_\mathrm{eq}\right) e_\mathrm{eq}} - 1 \tag{7.2.3-10}$$

$$U = \frac{2\left(2 - \nu\, g\right)}{\sqrt{3\left(1-\nu^2\right)}} \tag{7.2.3-11}$$

Not used (7.2.3-12)

$$A = \left(\frac{3}{4}\frac{U\, D_\mathrm{i}}{e_\mathrm{eq}} - 2J\right)\left(1+\nu\right)\left[1 + \left(1-\nu\right)\frac{e_\mathrm{eq}}{D_\mathrm{i} + e_\mathrm{eq}}\right] \tag{7.2.3-13}$$

$$B = \left[\left(\frac{3}{8}\frac{U\, D_\mathrm{i}}{e_\mathrm{eq}} - J\right)H^2 - \frac{3}{2}\left(2 - \nu\, g\right)g\right]H \tag{7.2.3-14}$$

$$F = \left(\frac{3}{8}U\, g + \frac{3}{16}\left(2g^2 - g^4\right)\frac{D_\mathrm{i} + e_\mathrm{eq}}{e_\mathrm{eq}} - 2J\,\frac{e_\mathrm{eq}}{D_\mathrm{i} + e_\mathrm{eq}}\right)H^2 - 3\left(2 - \nu\quad g\right)g\,\frac{e_\mathrm{eq}}{D_\mathrm{i} + e_\mathrm{eq}} \tag{7.2.3-15}$$

$$G = \left[\frac{3}{8}\left(2g^2 - g^4\right) - 2J\left(\frac{e_\mathrm{eq}}{D_\mathrm{i} + e_\mathrm{eq}}\right)^2\right]H \tag{7.2.3-16}$$

$$a = \frac{B}{A} \tag{7.2.3-17}$$

$$b = \frac{F}{A} \tag{7.2.3-18}$$

$$c = \frac{G}{A} \tag{7.2.3-19}$$

$$N = \frac{b}{3} - \frac{a^2}{9} \tag{7.2.3-20}$$

$$Q = \frac{c}{2} - \frac{a\,b}{6} + \frac{a^3}{27} \tag{7.2.3-21}$$

$$K = \frac{N^3}{Q^2} \tag{7.2.3-22}$$

when $Q \geq 0$: $S = \sqrt[3]{Q\left[1 + (1+K)^{\frac{1}{2}}\right]}$ (7.2.3-23)

when $Q < 0$: $S = -\sqrt[3]{|Q|\left[1 + (1+K)^{\frac{1}{2}}\right]}$ (7.2.3-24)

The minimum required wall thickness given by the following equation in 7.2.3-6 for normal operating conditions:

$$e = C_2 D_i \sqrt{\frac{p_c}{f_{min}}} \tag{7.2.3-25}$$

is given by the following equation:

$$e = \left(D_i + e_{eq}\right)\left(\frac{N}{S} - S - \frac{a}{3}\right) \tag{7.2.3-26}$$

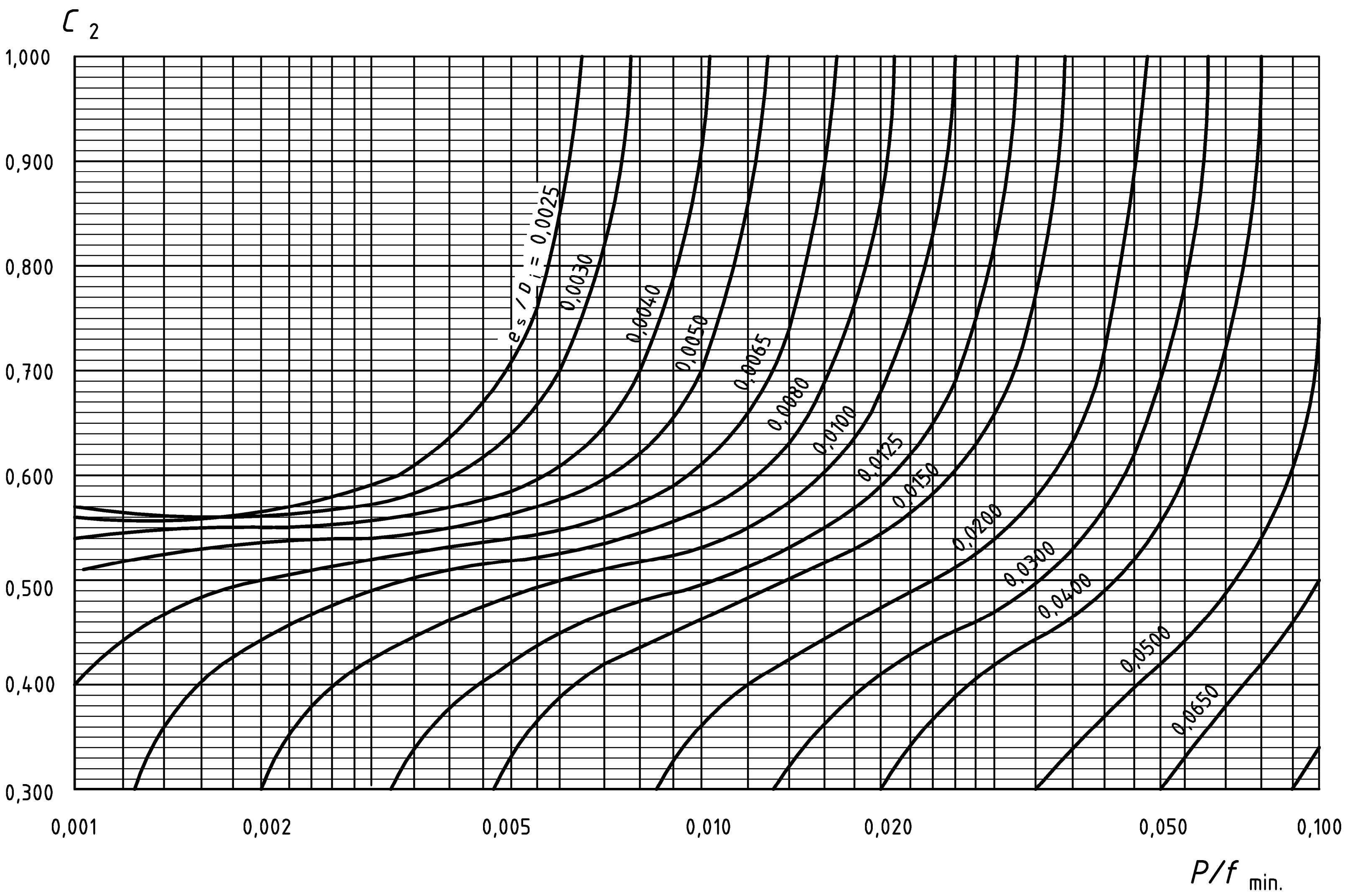


Figure 7.2.3-4 — Coefficient C_2

NOTE When the values of e_{eq}/D_i and p_c/f_1 result in a value of the coefficient C_2 less than 0,30, only the first term of equation (7.2.3-6) applies.

Where $r_i \leq e_{a\,f}$, the length l_{cyl} shall be as follows:

$$l_{cyl} = \sqrt{\left(D_i + e_{eq}\right) e_{eq}} \qquad (7.2.3\text{-}27)$$

7.2.3.4 Flat ends with a stress-relief groove

Flat ends with a stress-relief groove is prohibited in the creep range.

The minimum required wall thickness, e, of a flat end with a stress-relief groove shall be given by the equations (7.2.3-6) and (7.2.3-7).

The minimum required wall thickness, e_{rg}, at the bottom of the groove shall be given by:

$$e_{rg} = \max\left(e_{eq}\,;\, e_{eq}\frac{f_2}{f_1}\right) \qquad (7.2.3\text{-}28)$$

The minimum wall thickness of the cylindrical part, e_{eq}, shall be in accordance with 6.1 and for nominal design stress $f = \min(f_1\,;\, f_2)$. The minimum radius of the stress-relief groove, r_i, shall be max (0,25 e_{eq}, 5 mm) (see Figure 7.2.3-5).

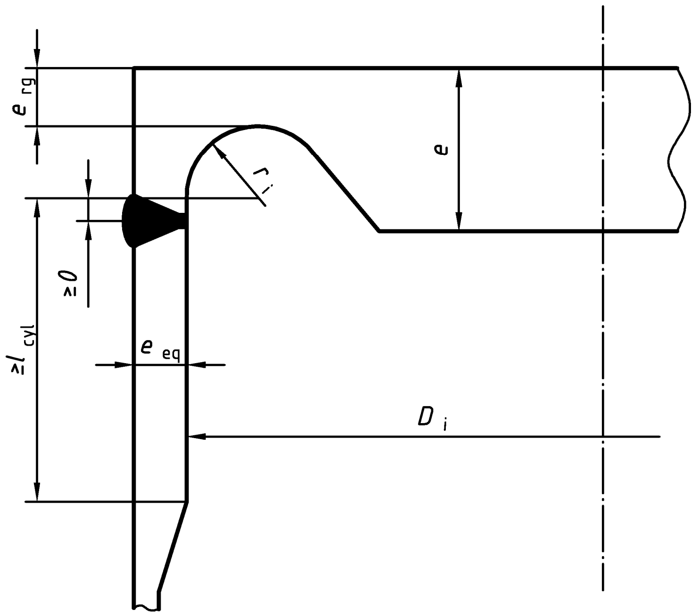


Figure 7.2.3-5 — Flat ends with peripheral stress relief groove

7.2.4 Unstayed flat circular bolted ends

7.2.4.1 General

This sub-clause shall be used to determine the wall thickness of bolted, flat circular ends without an opening.

The following ends are covered by 7.2.4:

— ends with gaskets entirely within the bolt circle (see Figure 7.2.4-1);

— ends with full faced gaskets (see Figure 7.2.4-2).

These ends may or may not be of uniform thickness. The minimum required wall thickness shall be extended to the entire surface located inside the gasket.

Ends with gaskets not force-balanced (flat face flanges), see EN 1591-1, are not covered by this sub-clause.

7.2.4.2 Circular ends with the gasket entirely within the bolt circle

The minimum required wall thickness, e, for the end shall be given by:

$$e = \max\left(e_{\mathrm{A}}\,;e_{\mathrm{p}}\right) \tag{7.2.4-1}$$

where

— for the gasket seating condition (bolting-up) :

$$e_{\mathrm{A}} = \sqrt{\frac{\frac{3}{\pi}\left(D_{\mathrm{t}} - D_{\mathrm{p}}\right)F_{\mathrm{A}}}{D_{\mathrm{p}}\,f_{\mathrm{A}}}} \tag{7.2.4-2}$$

where F_{A} is given by the following equation :

$$F_{\mathrm{A}} = \pi\ b\ D_{\mathrm{p}}\,y \tag{7.2.4-3}$$

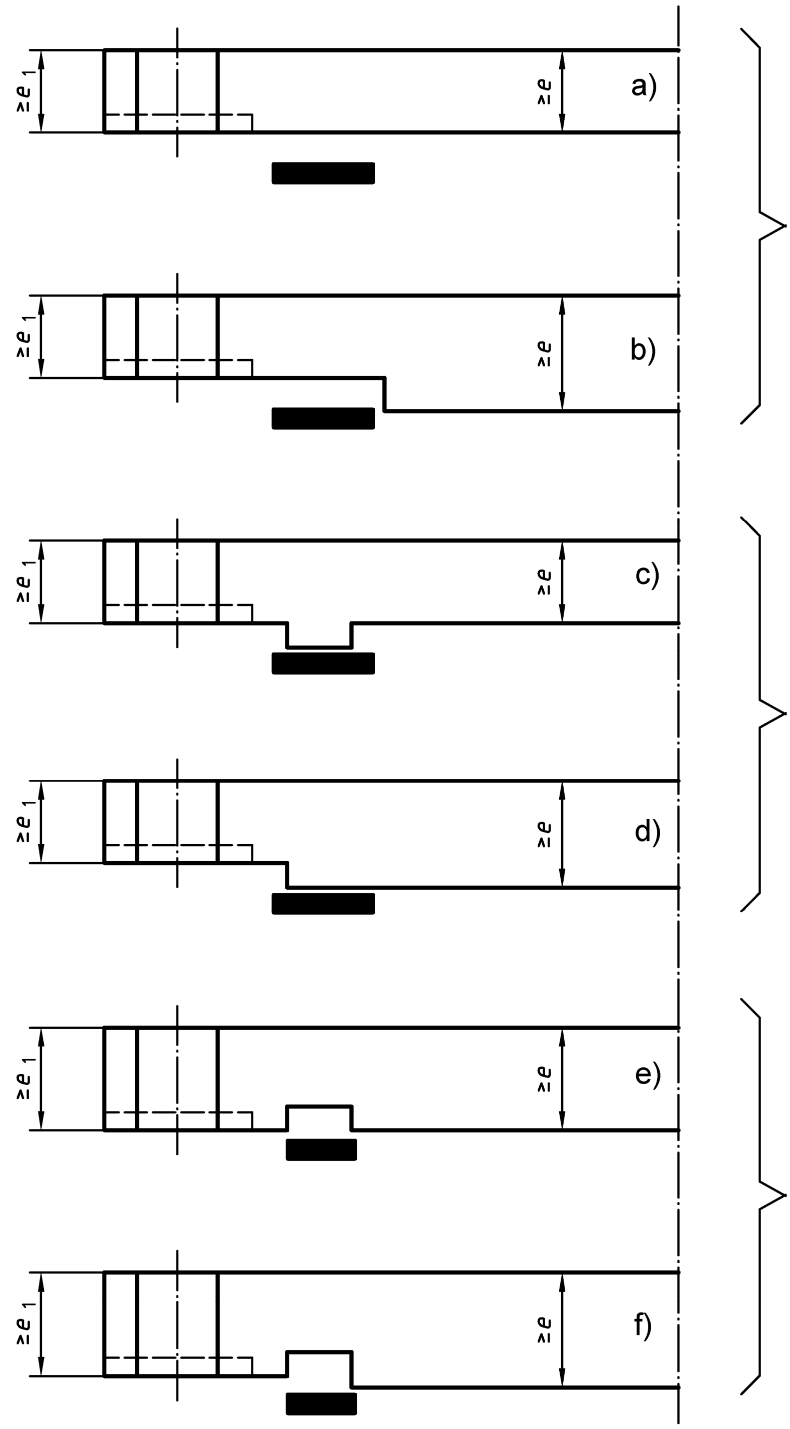
— for design conditions :

$$e_{\mathrm{p}} = \sqrt{\left\{0{,}31D_{\mathrm{p}}^{2} + 3\left(\frac{D_{\mathrm{p}}}{4} + 2b\ m\right)\left(D_{\mathrm{t}} - D_{\mathrm{p}}\right)\right\}\frac{p_{\mathrm{c}}}{f_{1}}} \tag{7.2.4-4}$$

The various values of m and y shall be as given in Table 7.2.4-1.

The minimum required wall thickness for the peripheral area of the end shall be given by equation (7.2.4-2) or equation (7.2.4-5) for each design condition, whichever is the greatest.

$$e_{1} = \sqrt{3\left(\frac{D_{\mathrm{p}}}{4} + 2b\ m\right)\left(D_{\mathrm{t}} - D_{\mathrm{p}}\right)\frac{p_{\mathrm{c}}}{f_{1}}} \tag{7.2.4-5}$$



a) and b) : ends with flat gasket face
c) and d) : ends with tongue joint
e) and f) : ends with groove joint

Figure 7.2.4-1 — Bolted circular flat ends with gasket entirely within the bolt circle

7.2.4.3 Flat ends with a full faced gasket

The minimum required wall thickness for a bolted flat end with a full faced gasket (see Figure 7.2.4-2) shall be given by the following equation:

$$e = 0{,}41 D_t \sqrt{\frac{p_c}{f_1}} \tag{7.2.4-6}$$

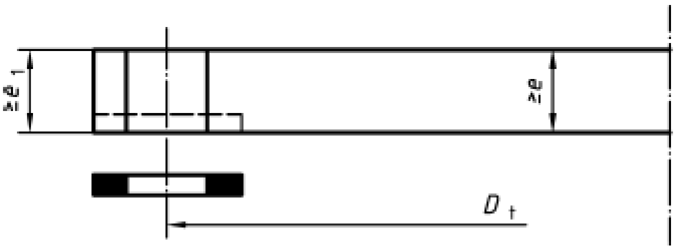


Figure 7.2.4-2 — Bolted flat end with a full faced gasket

The minimum required wall thickness for the peripheral area of the end shall be given by the following equation:

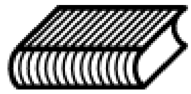
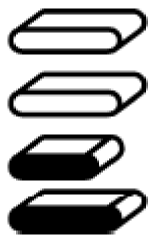
$$e_1 = 0{,}8\, e \tag{7.2.4-7}$$

Table 7.2.4-1 — Recommended gasket factors (*m*) and minimum design seating stress (*y*)

Gasket material		Gasket factor *m*	Minimum design seating stress *y* N/mm²	Sketches
Rubber without fabric or a high percentage of asbestos[a] fibre: below[b] 75°BS and IRH 75° BS and IRH or higher		 0,50 1,00	 0 1,4	
Asbestos[a] with a suitable binder for the operating conditions	3,2 mm thick 1,6 mm thick 0,8 mm thick	2,0 2,75 3,5	11,0 25,5 44,8	
Rubber with cotton fabric insertion		1,25	2,8	
Rubber with asbestos[a] fabric insertion, with or without wire reinforcement	3-ply 2-ply 1-ply	2,25 2,5 2,75	15,2 20,0 25,5	  

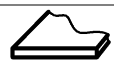
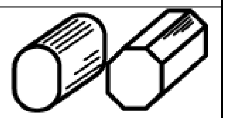
(continued)

Table 7.2.4-1 *(continued)*

Gasket material		Gasket factor *m*	Minimum design seating stress *y* N/mm^2	Sketches
Vegetable fibre		1,75	7,6	
Spiral-wound metal, asbestos[a] filled	Carbon Stainless or Monel	2,50 3,00	to suit application	
Corrugated metal, asbestos[a] inserted or corrugated metal jacketed asbestos[1] filled	Soft aluminium Soft copper or brass Iron or soft steel Monel or (4 to 6) % chromium alloy steel Stainless steels	2,50 2,75 3,00 3,25 3,50	20,0 25,5 31,0 37,9 44,8	
Flat metal jacketed asbestos[a] filled	Soft aluminium Soft copper or brass Iron or soft steel Monel (4 to 6) % chromium alloy steel Stainless steels	3,25 3,5 3,75 3,5 3,75 3,75	37,9 44,8 52,4 55,1 62,0 62	
Corrugated metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or (4 to 6) % chromium alloy steel Stainless steels	2,75 3,00 3,25 3,5 3,75	25,5 31,0 37,9 44,8 52,4	
Grooved metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4 to 6 % chromium alloy steel Stainless steels	3,25 3,5 3,75 3,75 4,25	37,9 44,8 52,4 62 69,5	

(continued)

Table 7.2.4-1 *(continued)*

Gasket material		Gasket factor *m*	Minimum design seating stress *y* N/mm^2	Sketches
Solid flat metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or (4 to 6) % chromium alloy steel Stainless steels	4,00 4,75 5,5 6,0 6,5	60,6 89,5 124 150 179	
Ring joint	Iron or soft steel Monel or (4 to 6) % chromium Stainless steels	5,50 6,00 6,50	124 150 179	
Rubber O-rings : below 75° BS 75° BS and 85° BS and higher		0 to 0,25	0,7 1,4	
Rubber square section rings : below 75° BS and IRH 75° BS and 85° BS and IRH		0 to 0,25	1,0 2,8[c]	
Rubber T-section rings : below 75° BS and IRH 75° and 85° BS and IRH		0 to 0,25	1,0 2,8	

NOTE 1 Asbestos gaskets are not allowed in many EU-countries.

NOTE 2 In selecting gasket materials for use with aluminium alloy flanges account should be taken of the relative hardness values of the gasket and flange materials.

[a] New non-asbestos bonded fibre sheet gaskets are not necessarily direct substitutes for asbestos based materials. In particular pressure, temperature and bolt load limitations may be applied. Use within the manufacturer's current recommendations.

[b] See BS 903 Part A26

[c] This value has been calculated.

7.2.5 Reinforcements of openings in unstayed flat ends

7.2.5.1 General

This sub-clause shall apply to reinforcement of single or multiple openings in unstayed flat ends (either bolted or welded to the adjacent pipe), provided that their diameter is not greater than 50 % of the pipe inside diameter D_i for welded ends, or not greater than 50 % of the gasket mean diameter D_p for bolted ends.

Blind screwed holes for stud-bolts for connection to standard pipe flanges shall be strong enough that reinforcements are not required, provided that they are located around an opening having a diameter not greater than the maximum bore diameter of the standard flange which should be bolted to that opening, and provided that the thickness at the bottom of the bore, e_{bb}, is at least 50 % of the stud-bolt diameter d_{bt} (see Figure 7.2.5-1).

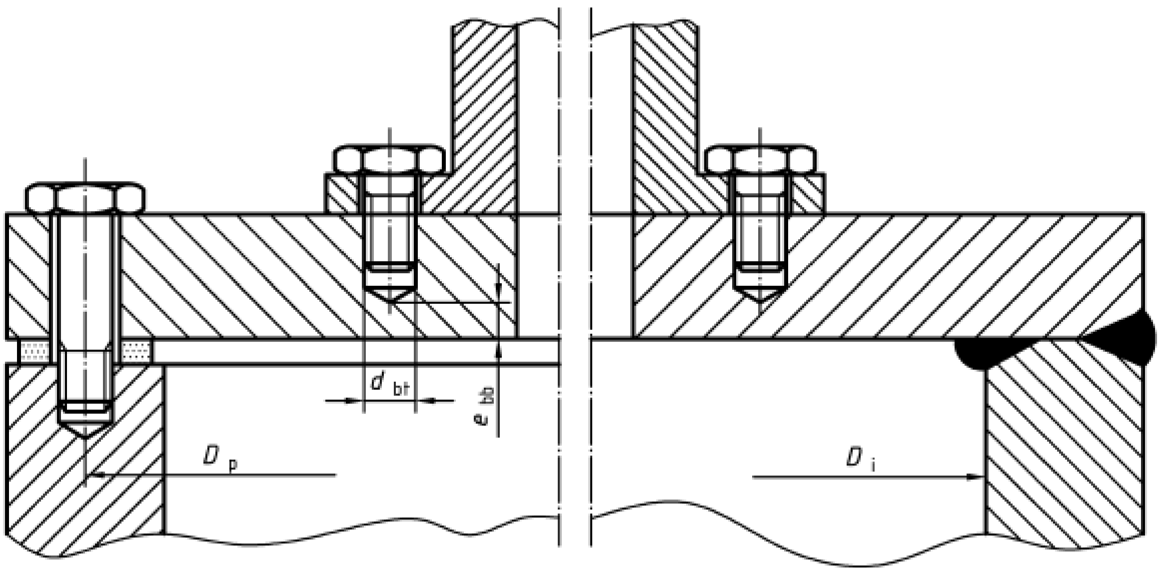


Figure 7.2.5-1 - Flat end with bolted opening

7.2.5.2 Flat ends with openings

For flat ends welded to the pipe as shown in Figures 7.2.3-1, 7.2.3-3 and 7.2.3-5 the wall thickness, e_{op}, shall be:

$$e_{op} = \max\left(Y_1\, e\ ;\ C_1\, Y_2\ D_i \sqrt{\frac{p_c}{f_1}} \right) \tag{7.2.5-1}$$

For bolted ends as shown in Figures 7.2.4-1 and 7.2.4-2 the wall thickness, e_{op}, shall be:

$$e_{op} = Y_2\, e \tag{7.2.5-2}$$

where Y_1 and Y_2 are defined as follows:

$$Y_1 = \min\left(2\ ;\ \sqrt[3]{\frac{K}{K-d}} \right) \tag{7.2.5-3}$$

$$Y_2 = \sqrt{\frac{K}{K-d}} \tag{7.2.5-4}$$

where

K is the distance between the centres of two adjacent openings (see Figure 7.2.5-2).

For a single opening:

— for equation (7.2.5-3), K is equal to twice the distance h from the centre of the opening to the circumference of diameter D_i ;

— for equation (7.2.5-4), K is equal to the diameter of the end D_i.

d is the equivalent diameter, which shall not be used if $d < 0$ and is given by:

— in case of set-on nozzles

$$d = d_i - \frac{2A_r}{e_{op}} \tag{7.2.5-5}$$

— in case of set-in nozzles

$$d = d_o - \frac{2A_r}{e_{op}} \tag{7.2.5-6}$$

where

A_r is the reinforcement area shown in Figure 7.2.5-3 or 7.2.5-4, respectively.

Where the nominal design stress of the nozzle is less than that of the flat end, the area A_r shall be multiplied by the ratio of the design stress of the flat end to that of the nozzle.

Where two openings of different (actual or equivalent) diameters are located in the flat end, the mean value of the two diameters in question shall be used for d when considering the corresponding pairs of bores for calculation of coefficients Y_1 and Y_2.

Where the openings are distributed irregularly, the pair of openings for which the coefficients Y_1 and Y_2 are the largest shall be chosen.

Every opening shall be checked as a single opening in all cases.

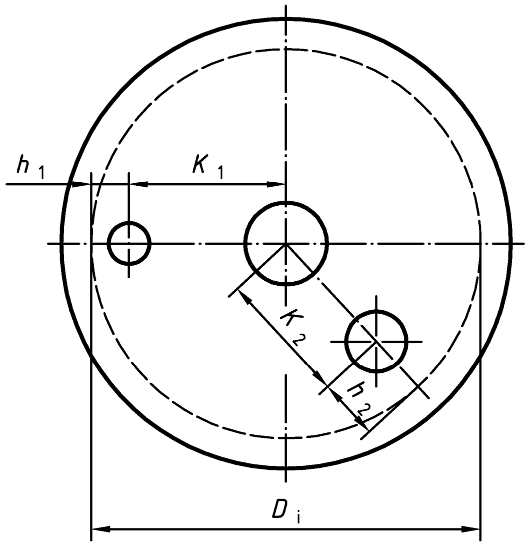


Figure 7.2.5-2 - Flat ends with multiple openings

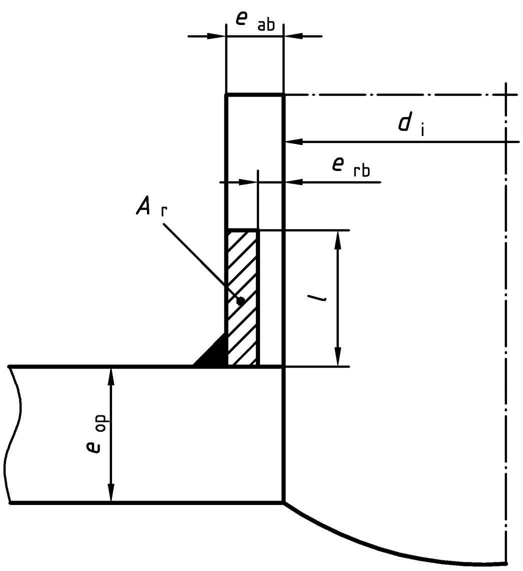


Figure 7.2.5-3 — Reinforcement area A_r for set-on nozzles

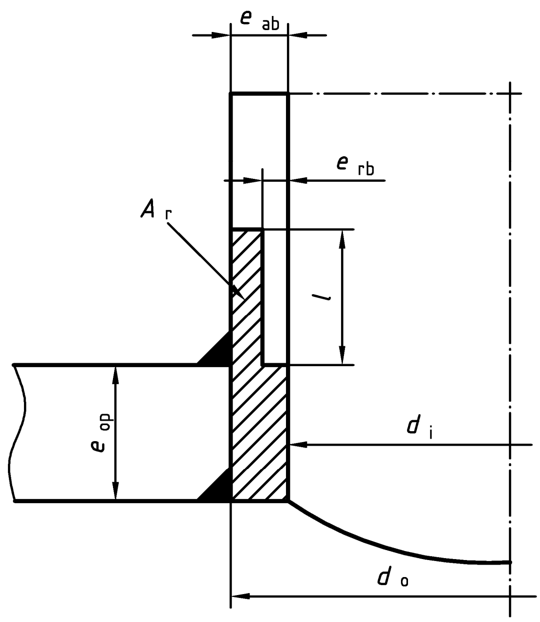


Figure 7.2.5-4 — Reinforcement area A_r for set-in nozzles

8 Openings and branch connections

8.1 General

This sub-clause shall apply to cylindrical shells, conical shells, spherical shells and dished ends having circular, elliptical or obround openings, provided that the assumptions and conditions specified in Clause 8 are satisfied.

For the purposes of Clause 8, the word “shell” shall apply to run pipes and headers in addition to shells.

NOTE Forces and/or moments due to loadings other than internal pressure are not considered in this design method.

An alternative method for the calculation of openings is given in Annex O (normative).

This new procedure is based on limit analysis and shakedown analysis and allows the connection to be designed as well as the reinforcement where necessary, and is particularly suitable for large openings.

As for Clauses 6, 7, 8, 9 and 11, the requirements of Annex O shall apply for loads of predominantly non-cyclic nature.

This method applies to connections that are self reinforced and also to those where reinforcing pads are used.

Oblique branch connections are also covered.

In addition, significant moments due to loadings other than internal pressure, as bending or torsion moments, can be considered by this new design method.

8.2 Symbols

For the purposes of clause 8, the symbols given in Table 8.2-1 shall apply in addition to those given in Table 3.2-1.

Table 8.2-1 — Additional symbols for the purposes of clause 8

Symbol	Description	Unit
A_f	effective cross sectional area of the considered wall without allowances and tolerances	mm²
A_p	pressured area	mm²
D	diameter of shell	mm
D_m	mean diameter as defined by the subscript [a]	mm
L_b	distance between the centers of adjacent branches	mm
R	radius of spherical or elliptical end	mm
d	diameter of branch and connected pipe	mm
d_m	mean diameter as defined by the subscript [a]	mm
e	minimum required wall thickness as defined by the subscript [a]	mm
e_a	analysis wall thickness as defined by the subscript [a]	mm
e_{ord}	ordered wall thickness as defined by the subscript [a]	mm
l	reinforcing length as defined by the subscripts [a]	mm
x	minimum distance to next discontinuity	mm
α	one-half apex angle of the reducer	°
φ	inclined angle of the branch axis to the normal of the shell or end	°
ψ	angle between the pipe axis and the line through the centres of two adjacent branches (see Figure 8.4.1-1)	°

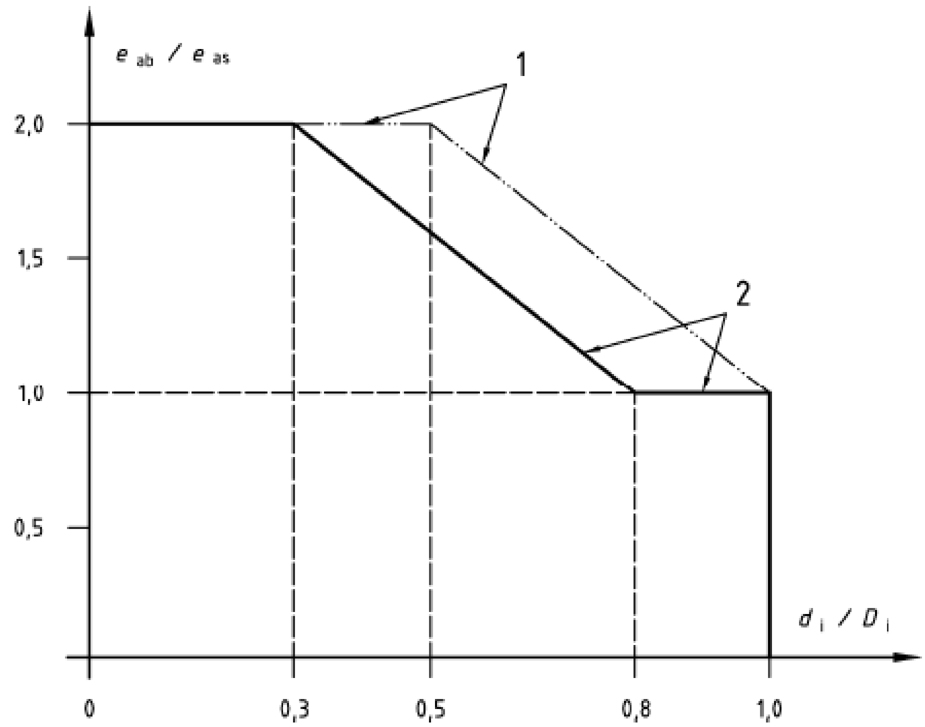
[a] The following subscripts shall apply:

- b branch
- s shell or end
- pl compensation plate
- φ angle

8.3 Limitations

8.3.1 Thickness ratio

The thickness ratio $e_{a\,b}$ / $e_{a\,s}$ used in the calculations shall not be greater than the value given in Figure 8.3.1-1 for cylindrical or conical shells and in Figure 8.3.1-2 for spherical shells or dished ends as a function of d_i/D_i.

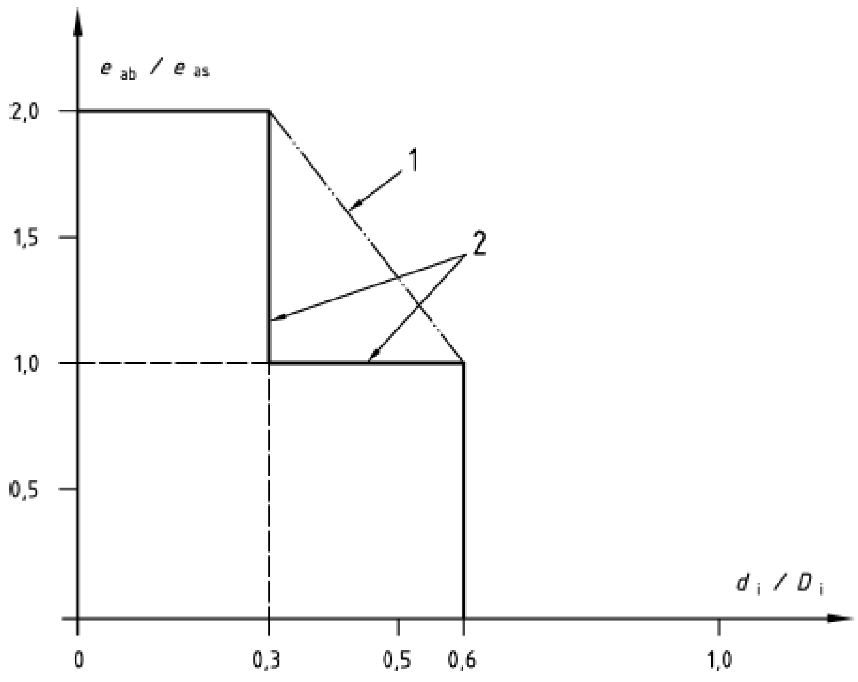


Key

1 for $f \leq$ 250 MPa (N/mm²) only

2 for f > 250 MPa (N/mm²) only

Figure 8.3.1-1 — Diagram of thickness ratio as a function of diameter ratio for cylindrical and conical shells



Key

1 for $f \leq$ 250 MPa (N/mm²) only

2 for $f >$ 250 MPa (N/mm²) only

Figure 8.3.1-2 — Diagram of thickness ratio as a function of diameter ratio for spherical and dished ends

For dished ends (hemispherical, torispherical or ellipsoïdal ends), openings with a ratio $d_i/D_i > 0{,}3$, but not greater than 0,6 shall be permitted provided that the following conditions are satisfied:

— the opening shall be reinforced in accordance with 8.4.3 or 8.4.4;

— the thickness ratio $e_{a\,b}/e_{a\,s}$ shall not be greater than the value given in Figure 8.3.1-2 as a function of d_i/D_i.

8.3.2 Openings in the vicinity of discontinuities

This sub-clause shall apply to cylindrical and conical shells, to ellipsoidal and torispherical ends where openings are distanced from shell discontinuity by a value x not less than those indicated for each type of shell.

a) Openings in cylindrical shells shall have a distance x as follows:

— for a cylindrical shell connected to a dished end, to a conical shell at the large base, to a flat end, to expansion joints and flanges:

$$x \geq \max\left(0{,}2 l_s\ ;\ 3{,}0\ e_{as}\right) \quad (8.3.2\text{-}1)$$

— for cylindrical shell connected to a conical shell at the small base, to an hemispherical shell and to other cylindrical non-coaxial shells :

$$x \geq l_{\text{s}} \tag{8.3.2-2}$$

where x is shown in Figure 8.3.2-1.

and l_{s} is given by equation (8.4.1-2).

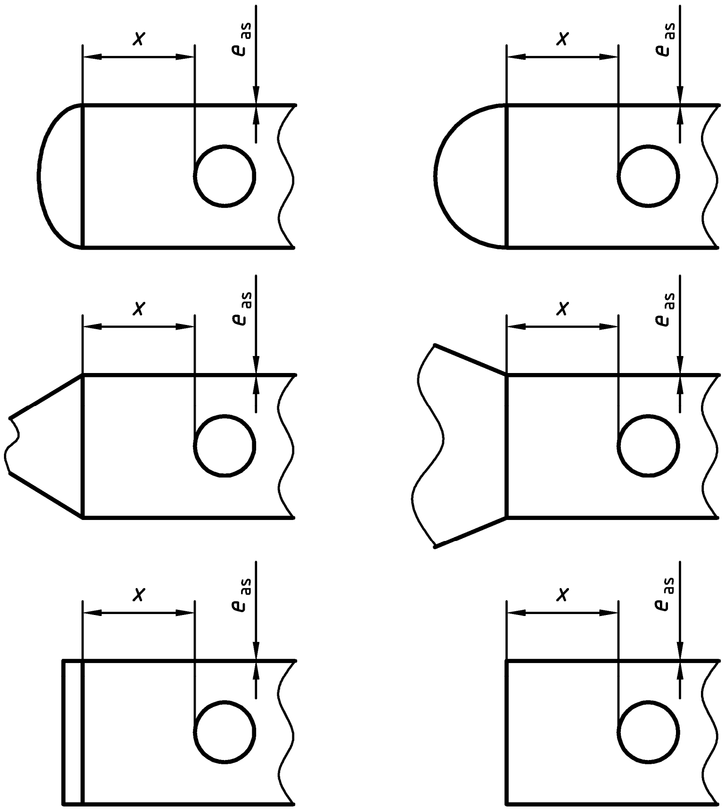


Figure 8.3.2-1 — Opening in a cylindrical shell

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

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

— for the small end

$$x_{\text{s}} \geq \max\left(0{,}2\sqrt{\frac{D_{\text{ms}}\ e_{\text{as}}}{\cos\alpha}}\ ; 3{,}0\ e_{\text{as}}\right) \tag{8.3.2-4}$$

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.

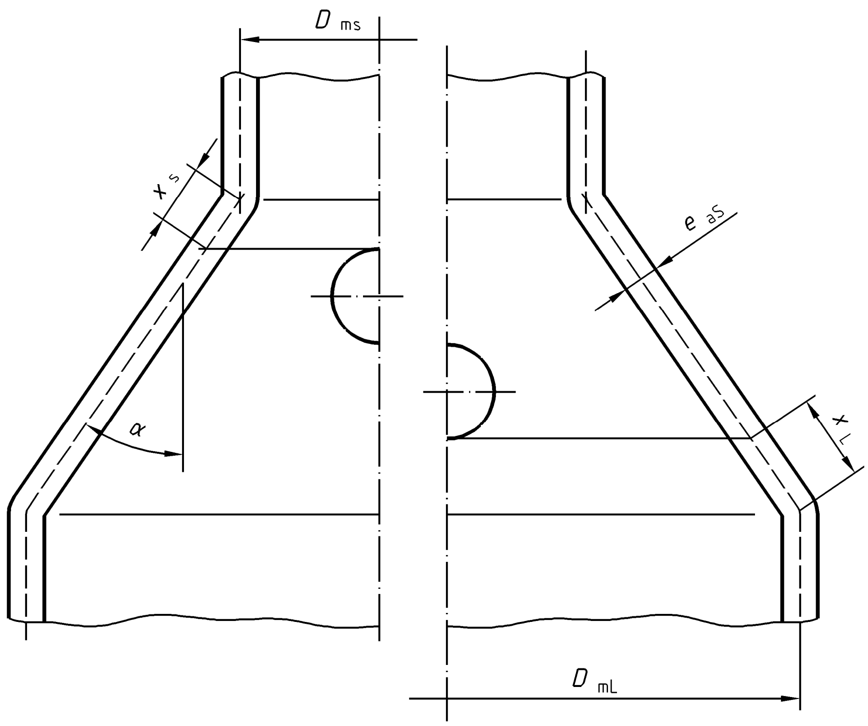


Figure 8.3.2-2 — Opening in a conical shell

8.3.3 Types of reinforcement

Cylindrical and conical shells, spherical shells and dished ends with openings shall be reinforced where required by clause 8. The reinforcement of the opening shall be obtained by one of the following method:

— an increased wall thickness of the shell compared with that of the shell without openings (see Figure 8.3.3-1);

— set-on welded reinforcing pads (see Figures 8.3.3-2);

— an increased wall thickness of the branch (see Figure 8.3.3-3);

— a combination of the above methods.

Where reinforcement is provided, it shall be the same in all planes through the axis of the opening or branch.

Set-on or set-in welded branches which are only seal-welded shall not be considered as reinforcement, and shall be calculated in accordance with 8.4.2.

8.3.4 Calculation method

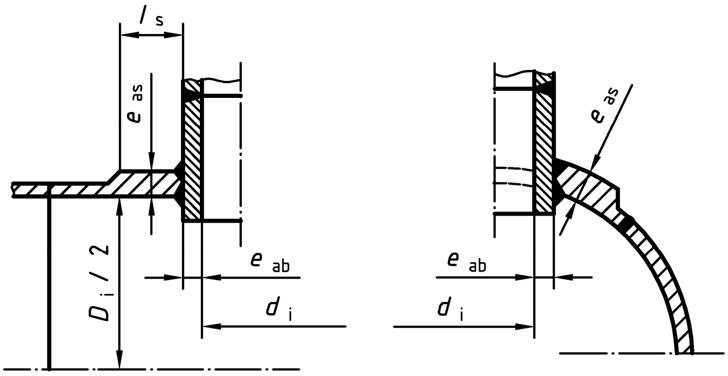
The reinforcement area of the shell with openings cannot be calculated directly, but shall be assumed in the first instance. That assumption shall be verified by means of the method given in the following sub-clauses. The applied method is derived from the requirements for cylindrical shells and for spherical shells, and spherical sections of dished ends respectively, and leads to relationships between the pressure loaded area A_p and the stress loaded cross sectional area A_f. Under certain circumstances, the calculation may need to be repeated using a corrected assumption of the reinforcement area.

8.3.5 Elliptical openings and oblique branch connections

In the case of elliptical or obround openings without branch connections, the ratio between the major and the minor diameters shall not exceed 2,0.

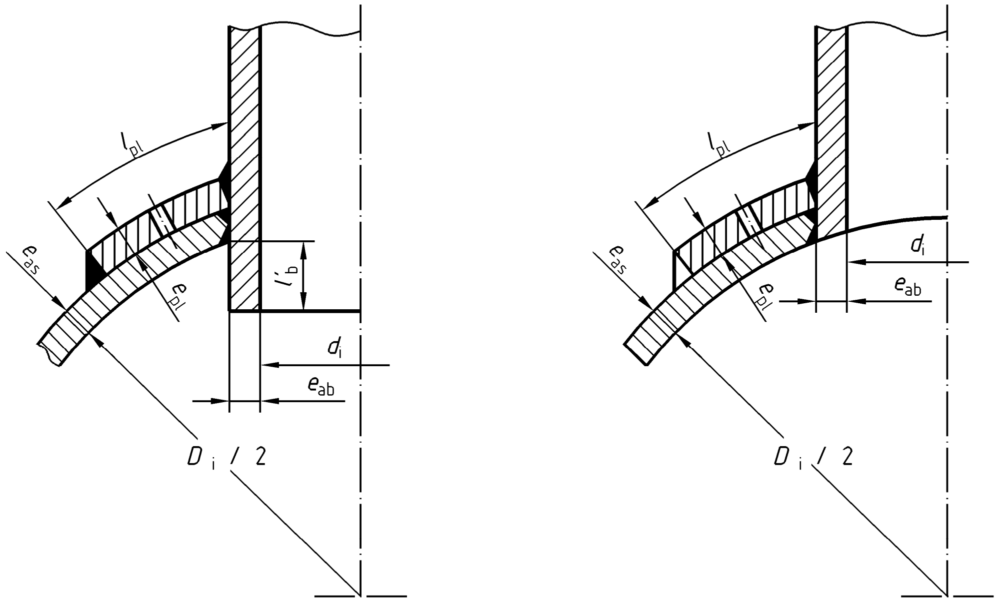
For design purposes, the diameter of elliptical or obround openings in cylindrical and conical shells shall be taken as the dimension extending in the direction of shell axis, while for spherical shells and dished ends, the major diameter shall be taken.

For oblique branch connections to cylindrical or spherical shells, the angle between the normal to the wall of the shell and the axis of the branch, φ, shall be between 0 ° and 45 ° (see Figures 8.4.3-3, 8.4.3-4 and 8.4.3-5).



a) cylindrical shell b) spherical shell

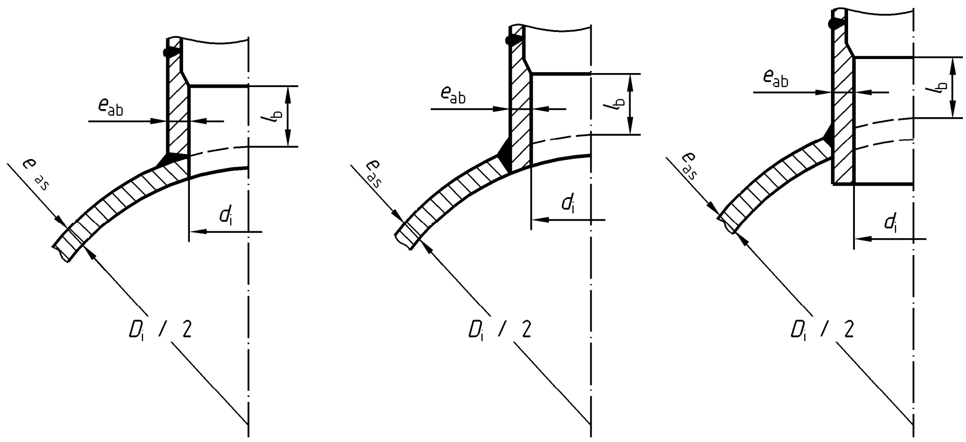
Figure 8.3.3-1 — Reinforcement by increase in wall thickness of the shell



a) set through **b) set in**

NOTE Consideration should be given to the effect of the flow for design of set through nozzles.

Figure 8.3.3-2 — Reinforcement by reinforcing pads



a) set on **b) set in** **c) set through**

NOTE 1 The increase wall thickness can be on the inside or the outside of the branch.

NOTE 2 Consideration should be given to the effect of the flow for design of set through nozzles.

Figure 8.3.3-3 — Reinforcement by increase in wall thickness of the branch

8.3.6 Reinforcing pads

Reinforcement of openings by reinforcing pads shall be limited to the conditions given in Figure 8.3.6-1 and to a diameter ratio of $d_i/D_i \leq 0{,}8$.

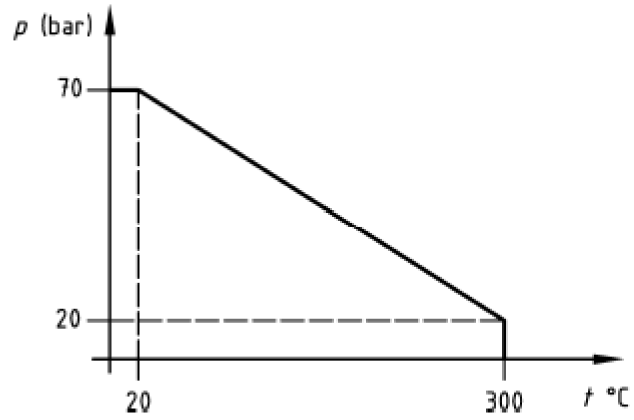


Figure 8.3.6-1 — Pressure and temperature limits for reinforcing pads

8.3.7 Dissimilar material of shell and reinforcements

Where the shell and the reinforcements consist of materials with different allowable design stresses and where the design stress of the shell is the lower, this lower value shall be used in calculating the reinforcement.

Material for reinforcements shall be selected in such a way that thermal stresses due to significant different thermal expansion coefficients are avoided.

8.3.8 Extruded outlets

The application of extruded outlets shall be restricted to a diameter ratio:

— for materials other than austenitic steels, $d_i/D_i \leq 0{,}8$;

— for austenitic steels, $d_i/D_i \leq 1{,}0$.

The areas A_{fs} and A_{fb} shall be multiplied by a factor of 0,9 if the actual wall thickness of the extrusion is unknown.

Applications within the creep range shall be limited to openings with $d_i/D_i \leq 0{,}7$ and the design stress shall be reduced to 90 % of that given in clause 5.

8.3.9 Branches in bends or elbows

The design of branch and support connections to piping accessories shall be in accordance with Annex E.

Branches in bends or elbows shall not be permitted for an application within the creep range.

8.3.10 Screwed-in branches

Screwed-in branches shall be limited to:

— a temperature of 400 °C;

— a pressure of 40 bar gauge;

— branch diameter ≤ DN 50.

In addition,

— the thread shall be not stripped;

— seal-welding shall be applied for a design temperature above 200 °C, or a design pressure above 16 bar;

— materials with similar thermal coefficients for pipe and screwed branches shall be used;

— due to the weldability of the screwed branch, a material with less than 0,25 % C shall be used in accordance with EN 13480-2;

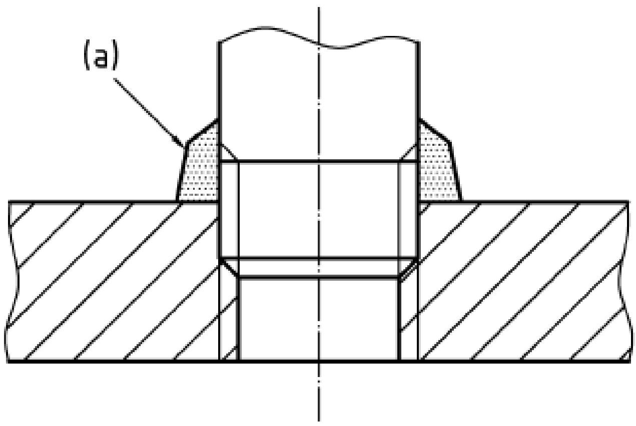
— the minimum thread engagement shall be not less than:

  — for ≤ DN 20, 6 times the pitch;

  — for > DN 20 ≤ DN 40, 7 times the pitch;

  — for > DN 40 ≤ DN 50, 8 times the pitch.

NOTE For piping containing group 1 fluids, and piping subjected to fatigue loading, extra care should be taken in the design and use of screwed branches. Even seal-welding a screwed branch is not considered to be a connection free of the risk of leakage.



(a) optional seal-welding

Figure 8.3.10-1 — Screwed-in branches

8.4 Isolated openings

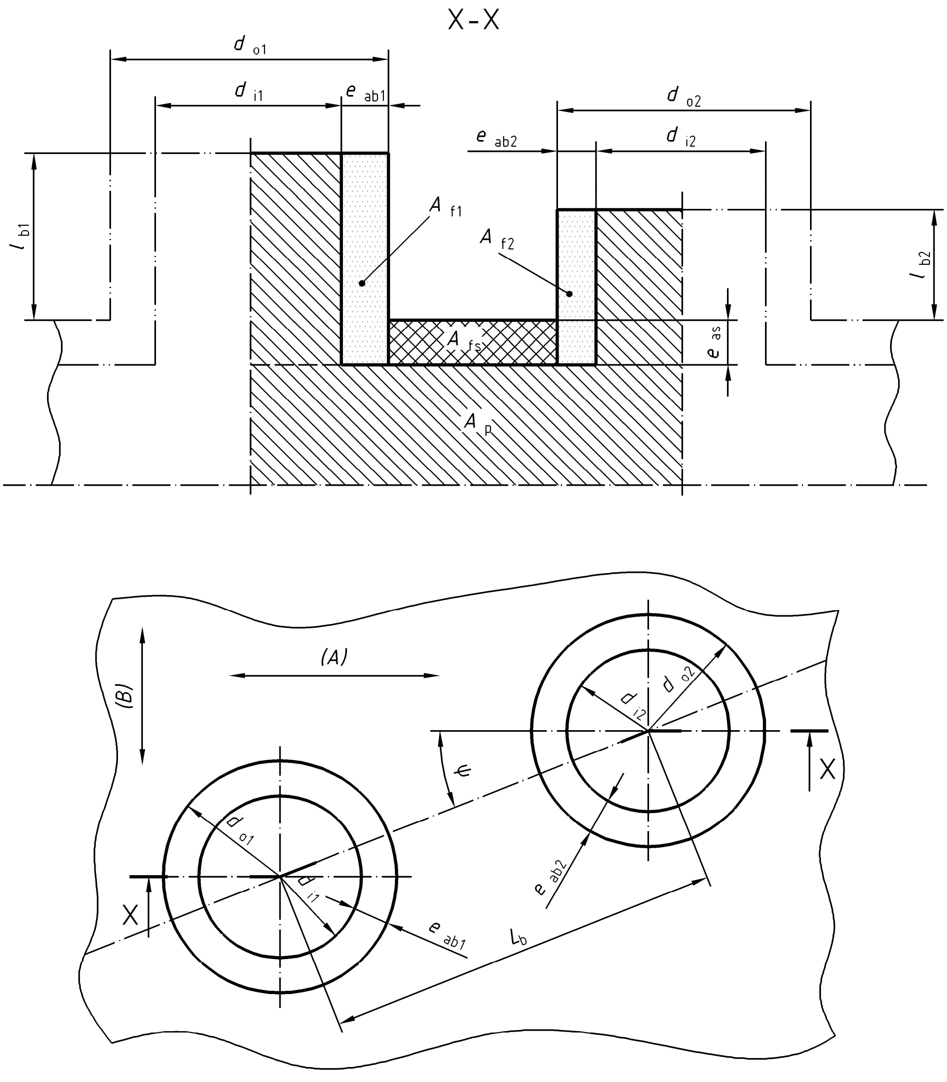
8.4.1 General

Adjacent openings or branches shall be considered as isolated openings if the centre to centre distance L_b between openings or branches, measured on the mean diameter of the shell, satisfies the following equation:

$$L_b \geq \frac{d_1}{2} + \frac{d_2}{2} + 2l_s \quad (8.4.1\text{-}1)$$

where d_1 and d_2 are given by:

- for cylindrical and conical shells, the diameters of the two openings or intersections of external diameter of branches and compensating rings with the main wall taken along the line of L_b (see Figure 8.4.1-1) ;
- for spherical shells and dished ends, the lengths of arcs on the wall at mid-thickness, subtended by chords d_{o1} and d_{o2} (see Figure 8.4.1-2) along L_b.



Key

(A) : longitudinal direction;

(B) : circumferential direction

Figure 8.4.1-1 — Cylindrical shell with adjacent branches with an angle ψ to the axis of the shell

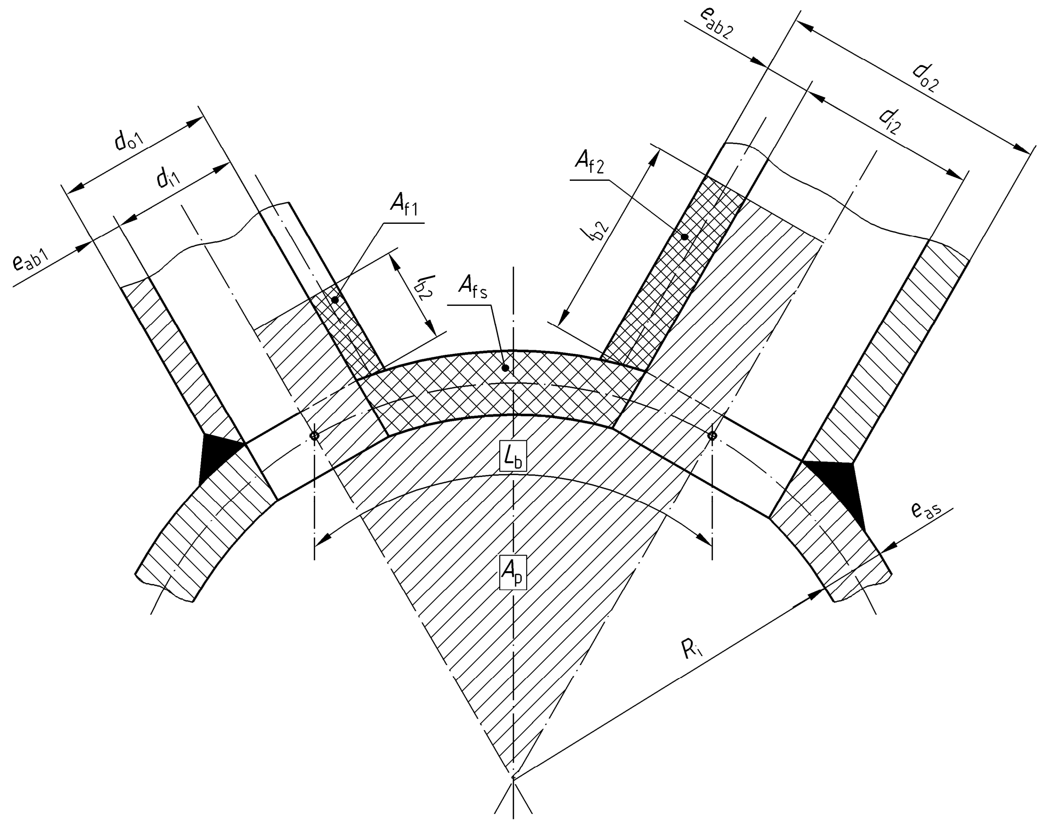


Figure 8.4.1-2 — Section view of spherical shell with adjacent branches

and where l_s for each opening is given by:

$$l_s = \sqrt{D_{eq} e_{as}} \tag{8.4.1-2}$$

where

D_{eq} is the equivalent diameter of the shell at the centre of each opening normal to the shell i.e.

— for cylindrical shells

$$D_{eq} = D_i + e_{as} \tag{8.4.1-3}$$

or

$$D_{eq} = D_o - e_{as} \tag{8.4.1-4}$$

— for hemispherical, torispherical or ellipsoidal ends

$$D_{eq} = 2R_i + e_{as} \tag{8.4.1-5}$$

NOTE See 7.1.4 for ellipsoidal end.

— for conical shells

$$D_{eq} = \frac{D_m}{\cos\alpha} - e_{as} \tag{8.4.1-6}$$

See Figure 8.3.2-2.

e_{as} is the analysis shell or end thickness without taking into account any reinforcing pad thickness.

8.4.2 Unreinforced openings

Reinforcement shall not be required if the following condition is met:

$$d_i \le 0{,}14\sqrt{(D_{eq}\, e_{as})} \tag{8.4.2-1}$$

8.4.3 Reinforced openings with d_i/D_i < 0,8

a) Reinforced openings with increased wall thickness

The reinforcement may be obtained by an increased wall thickness of the shell and/or branch. This reinforced wall thickness shall extend up to a minimum distance of l_s along the shell, and l_b along the branch, measured as shown in Figures 8.3.3-1, 8.3.3-3 and 8.4.3-1.

The length l_s shall be calculated from equation (8.4.1-2).

The lengths l_b and l'_b shall be calculated from:

$$l_b = \sqrt{d_{eqb}\, e_{ab}} \tag{8.4.3-1}$$

$$l'_b = 0{,}5\sqrt{d_{eq\,b}\, e_{ab}} \text{ but not greater than the actual length} \tag{8.4.3-2}$$

In addition, the following condition shall be satisfied:

$$\left(f_b - \frac{p_c}{2}\right)A_{f_b} + \left(f_s - \frac{p_c}{2}\right)A_{f_s} \ge p_c A_p \tag{8.4.3-3}$$

where

A_f is the cross sectional area effective as compensation (A_{fb} + A_{fs}) ;

A_p is the pressure loaded area.

l_b and l_s, given by equations 8.4.3-1 and 8.4.1-2 are maximum lengths for reinforcement calculation.

Where so calculated, the established dimension shall be verified on the manufactured part. If the design shows a shorter length as given by equations 8.4.1-2 and 8.4.3-1, this shall be considered by the reinforcement calculation.

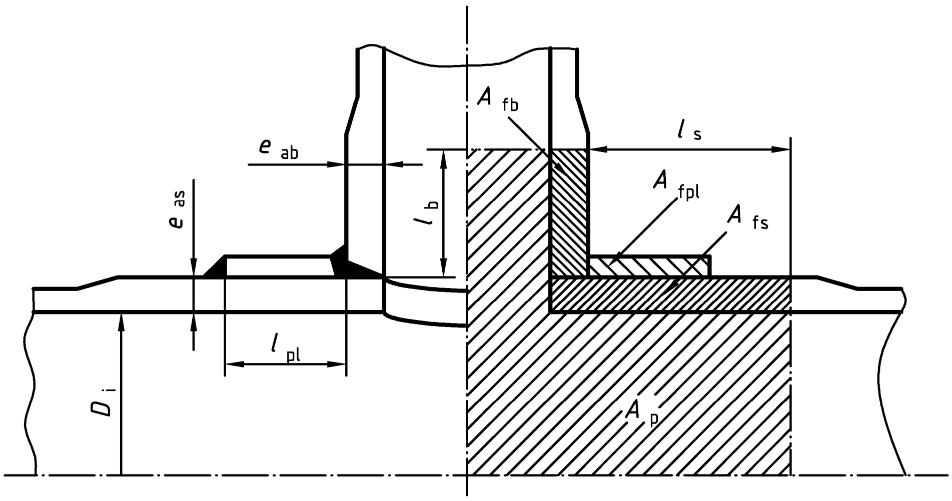


Figure 8.4.3-1 — Reinforcement by increasing the wall thicknesses of shell and/or branch

b) Reinforcement by reinforcing pads

Reinforcing pads shall be in close contact with the shell. The width of reinforcing pad l_{pl}, considered as contributing to the reinforcement shall not exceed l_s (see equation (8.4.1-2) and Figures 8.3.3-2 and 8.4.3-2).

$$l_{pl} \leq l_s \tag{8.4.3-4}$$

The value of $e_{a\,pl}$ used for the determination of $A_{f\,pl}$ in equations (8.4.3-6) and (8.4.3-7) shall not exceed the thickness e_{as} of the shell:

$$e_{a\,pl} \leq e_{a\,s} \tag{8.4.3-5}$$

In addition, the following condition shall be satisfied:

$$\left(f_s - \frac{p_c}{2}\right)\left(A_{f\,s} + A_{f\,b} + A_{f\,pl}\right) \geq p_c\,A_p\,. \tag{8.4.3-6}$$

where

$A_{f\,pl}$ is the cross-sectional areas of the reinforcing pad effective as compensation;

$A_{f\,s}$ is the cross-sectional area of the shell.

If the design stress of the branch, f_b, and/or of the reinforcing pad f_{pl}, is less than that of the shell, f_s, the following condition shall be satisfied instead of that of equation (8.4.3-6):

$$\left(f_s - \frac{p_c}{2}\right)A_{f\,s} + \left(f_b - \frac{p_c}{2}\right)A_{f\,b} + \left(f_{pl} - \frac{p_c}{2}\right)A_{f\,pl} \geq p_c\,A_p\,. \tag{8.4.3-7}$$

NOTE In no case a design stress of the branch f_b or a design stress of the reinforcing pad f_{pl} higher than f_s should be considered.

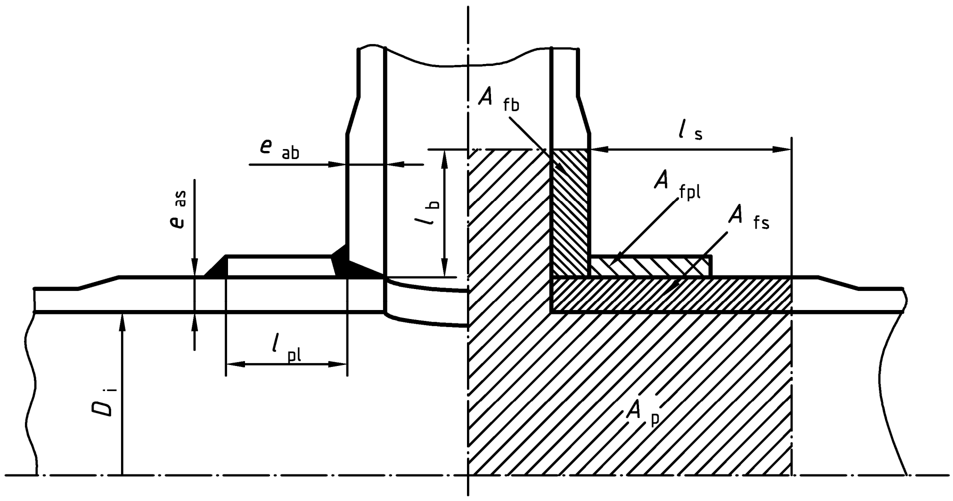


Figure 8.4.3-2 — Reinforcement by reinforcing pad for a cylindrical shell

c) Oblique branch connections in cylindrical and conical shells

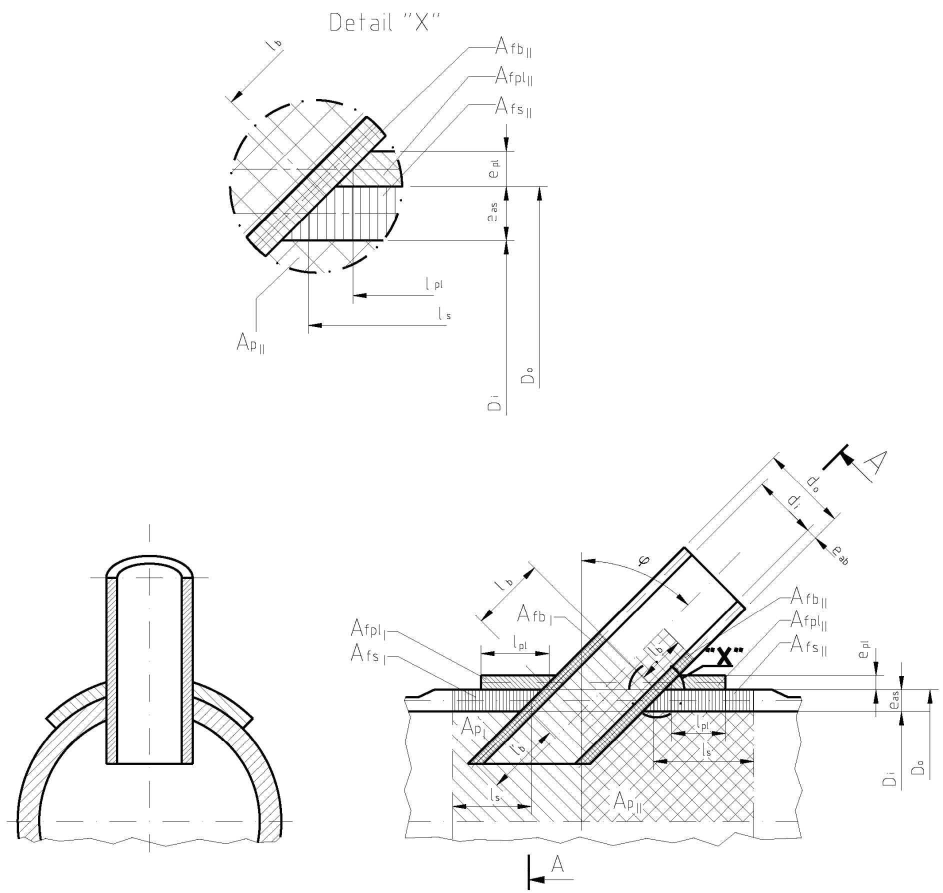
This sub-clause shall apply to branches not normal to the wall of cylindrical or conical shells, but with axis lying in a plane normal to the generating line of the shell at the centre of the opening, and having an angle φ formed with the normal or lying in a plane that contains the axis of the shell and having an angle φ formed with the normal. For non-radial branch connections, the reinforcement shall be calculated for the longitudinal and lateral section (see Figure 8.4.3-4).

Equations (8.4.3-3) or (8.4.3-6) and (8.4.3-7) shall apply, the area A_p is calculated with:

$$d = \frac{d_i}{\cos\varphi} \qquad (8.4.3\text{-}8)$$

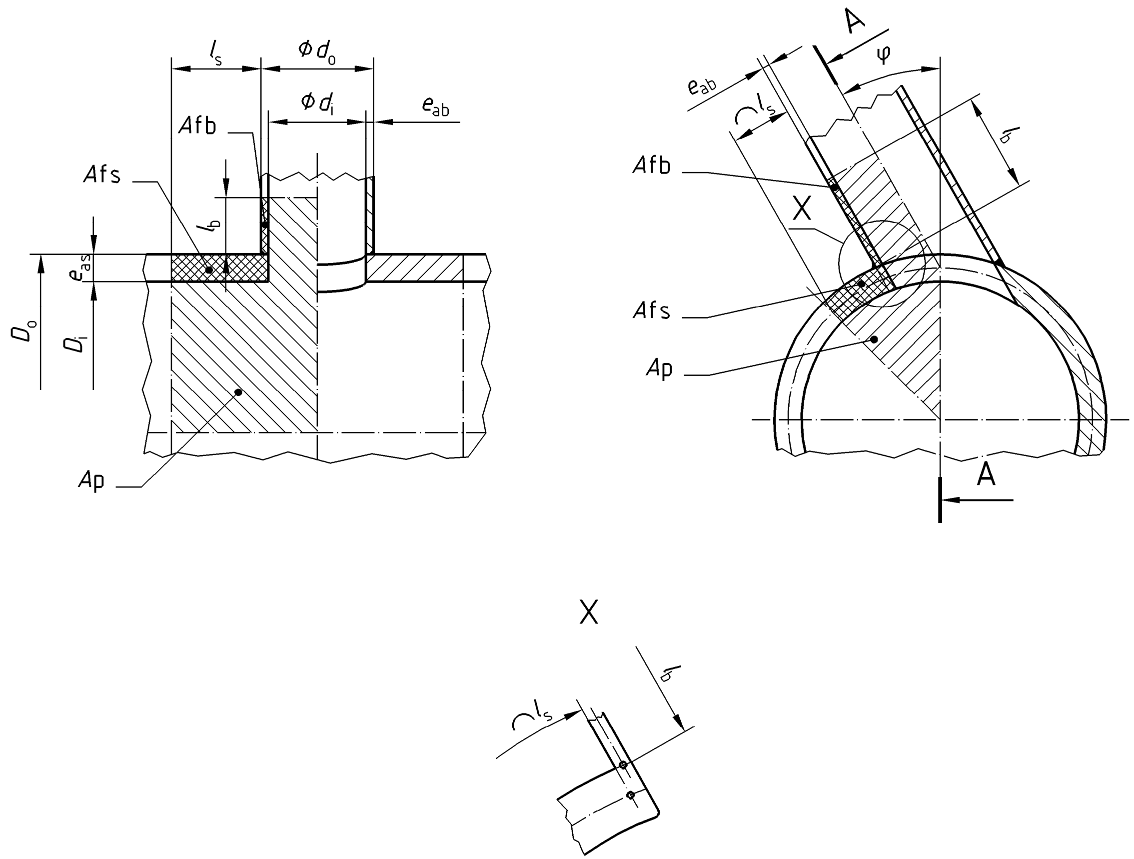
The maximum length considered as contributing to reinforcement shall be calculated in accordance with equation (8.4.1-2).

The angle φ shall be defined as shown in Figure 8.4.3-3 or 8.4.3-4: 0° < $\varphi \leq$ 45°.



NOTE Consideration should be given to the effect of the flow for design of set through nozzle.

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



a) Cross section view

b) Section X-X

Figure 8.4.3-4 — Reinforcement of non radial branch connection in cylindrical or conical shell

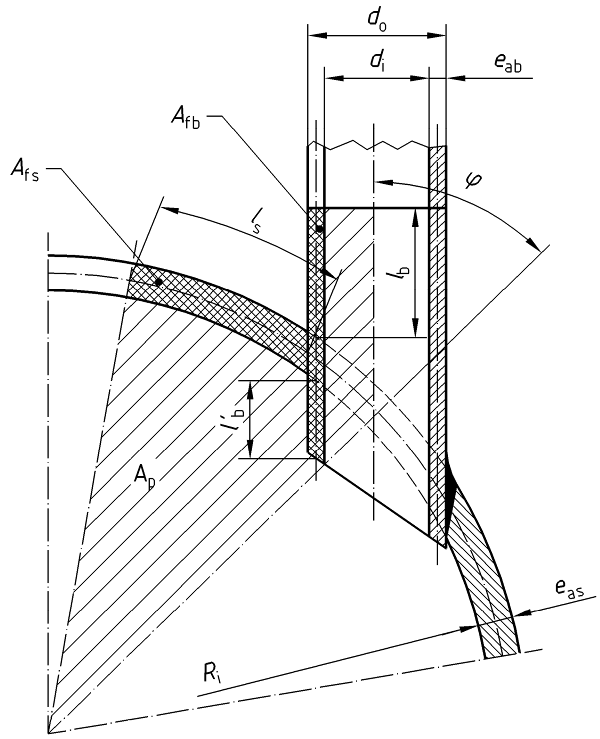
d) Oblique branch connections in spherical shells and dished ends

This sub-clause shall apply to branches in spherical shells and spherical parts of dished ends not normal to the wall of the shell and having an angle φ formed with the normal. The reinforcement shall be calculated in the section that contains the axis of branch and the normal to the shell at the centre of opening (see Figure 8.4.3-5).

The reinforcement shall be calculated in accordance with equations (8.4.3-3), (8.4.3-6) and (8.4.3-7), the area A_p is calculated in accordance with equation (8.4.3-8).

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

The angle φ shall be defined as shown in Figure 8.4.3-5.



NOTE Consideration should be given to the effect of the flow for design of set through nozzle.

Figure 8.4.3-5 — Reinforcement of oblique branch connection in spherical shells and dished ends

8.4.4 Reinforced single openings with 0,8 < d/D ≤ 1,0

The equations given in 8.4.3 shall apply with the restriction that large openings with d/D > 0,8 are not permissible for a design within the creep range of the chosen material.

8.5 Adjacent openings

8.5.1 Unreinforced openings

Unreinforced adjacent openings shall be permitted provided that the following restrictions are met:

— the centre lines of two openings shall be not closer to each other than the sum of their inner diameters, measured on the inside surface of the shell;

— if there are more than two openings within a circle area with a diameter D_{ca} given by

$$D_{\mathrm{ca}} = 2{,}0\sqrt{D_{\mathrm{m}} e_{\mathrm{as}}} \tag{8.5.1-1}$$

— the sum of their bore diameters shall satisfy the following relation:

$$\sum_{1}^{n} d_i \leq 0{,}175\sqrt{D_{\mathrm{m}} e_{\mathrm{as}}} \tag{8.5.1-2}$$

where *n* is the number of openings.

8.5.2 Reinforced openings with *d/D* ≤ 0,8

a) Reinforcement for adjacent openings in cylindrical shells

For the cross-section extending through the adjacent openings at the angle ψ with the longitudinal axis of the cylindrical shell in accordance with Figure 8.4.1-1 the following condition shall be satisfied if the design stress of branches and reinforcement pads are higher than or equal to that of the cylindrical or conical shell:

$$\left(f - \frac{p_{\mathrm{c}}}{2}\right)\left(A_{\mathrm{f\,s}} + A_{\mathrm{f\,b}} + A_{\mathrm{f\,pl}}\right) \geq p_{\mathrm{c}}\left[\frac{A_{\mathrm{p\,s}}}{2}\left(1 + \cos^2\psi\right) + A_{\mathrm{p\,b}}\right] \tag{8.5.2-1}$$

If the design stress of branches or reinforcing pads is lower than that of the cylindrical or conical shell, the following condition shall be satisfied :

$$\left(f_{\mathrm{s}} - \frac{p_{\mathrm{c}}}{2}\right)A_{\mathrm{f\,s}} + \left(f_{\mathrm{b}} - \frac{p_{\mathrm{c}}}{2}\right)A_{\mathrm{f\,b}} + \left(f_{\mathrm{pl}} - \frac{p_{\mathrm{c}}}{2}\right)A_{\mathrm{f\,pl}} \geq p_{\mathrm{c}}\left[\frac{A_{\mathrm{p\,s}}}{2}\left(1 + \cos^2\psi\right) + A_{\mathrm{p\,b}}\right] \tag{8.5.2-2}$$

For groups of openings, the strength consideration shall be performed for the ligaments extending in each direction and for each couple of adjacent openings.

The requirements of this sub-clause may be used for adjacent branches not normal to the shell wall and with the centres on the same generating line, using inclination angles resulting from projection on the plane containing the centre to centre distance and the normals to the shell at the centres of the openings. The value of the areas $A_{pb\varphi}$ shall be calculated in accordance with 8.4.3 c).

b) Reinforcement for adjacent openings in spherical shells and dished ends

For the cross-section extending through the adjacent openings in a spherical shell or in a spherical part of a dished end, in accordance with Figure 8.4.1-2, the following condition shall be satisfied if the design stress of branches and reinforcement pads are higher than or equal to that of the spherical shells or spherical part of dished ends:

$$\left(f - \frac{p_c}{2}\right)\left(A_{f\,s} + A_{f\,b} + A_{f\,pl}\right) \geq p_c\,A_p \tag{8.5.2-3}$$

If the design stress of the branches or reinforcing pad is lower than that of the spherical shell or spherical part of the dished end, then the following condition shall be satisfied:

$$\left(f_s - \frac{p_c}{2}\right)A_{f\,s} + \left(f_b - \frac{p_c}{2}\right)A_{f\,b} + \left(f_{pl} - \frac{p_c}{2}\right)A_{f\,pl} \geq p_c\,A_p \tag{8.5.2-4}$$

For groups of openings, the strength consideration shall be performed for the ligaments extending in each direction and for each couple of adjacent openings.

The requirements of this sub-clause may be used for two adjacent branches not normal to the shell wall using inclination angles resulting from projection on the plane containing the centre to centre distance of the branches and the centre of the spherical shell or of the spherical part of the dished end. The values of the areas $A_{pb\varphi}$ shall be calculated in accordance with 8.4.3 d).

8.6 Design of special piping components

8.6.1 Cylindrical Y-pieces

The equations of 8.4.3 shall apply. For l_s and l_{b1}, l_{b2} see Figure 8.6.1-1.

In cases with d/D > 0,8, the design stress shall be reduced to 90 % of that specified in clause 5. If applied at elevated temperatures, attention shall be paid to creep.

NOTE It is recommended, that such a design should not apply in the creep range. Attention should be paid to the welding process.

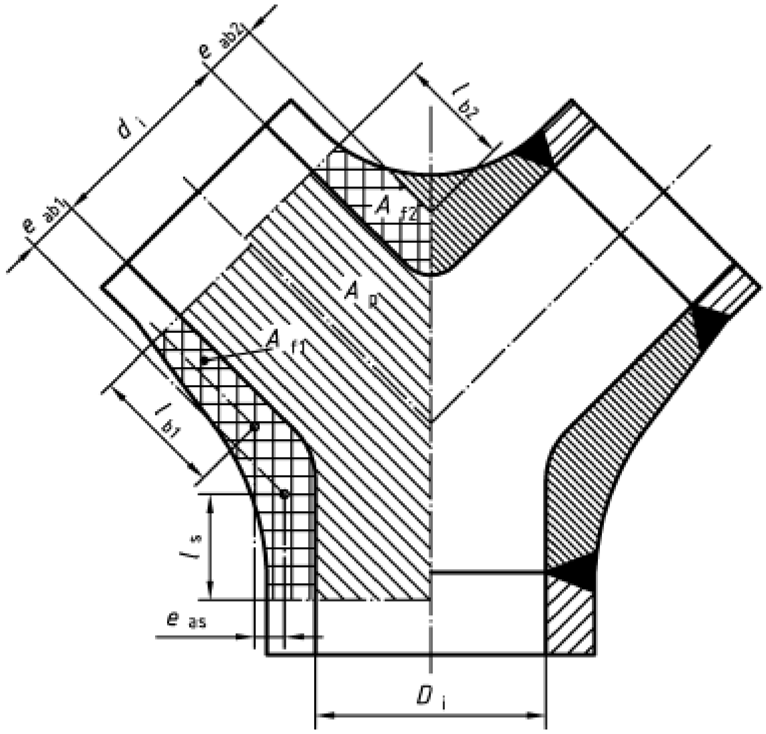


Figure 8.6.1-1 — Forged Y-piece

8.6.2 Spherical Y-pieces

The equations of 8.4.3 shall apply.

For l_s and l_b, see Figure 8.6.2-1.

NOTE It is recommended that the centre line of the branch should be normal to the spherical wall at the point of intersection.

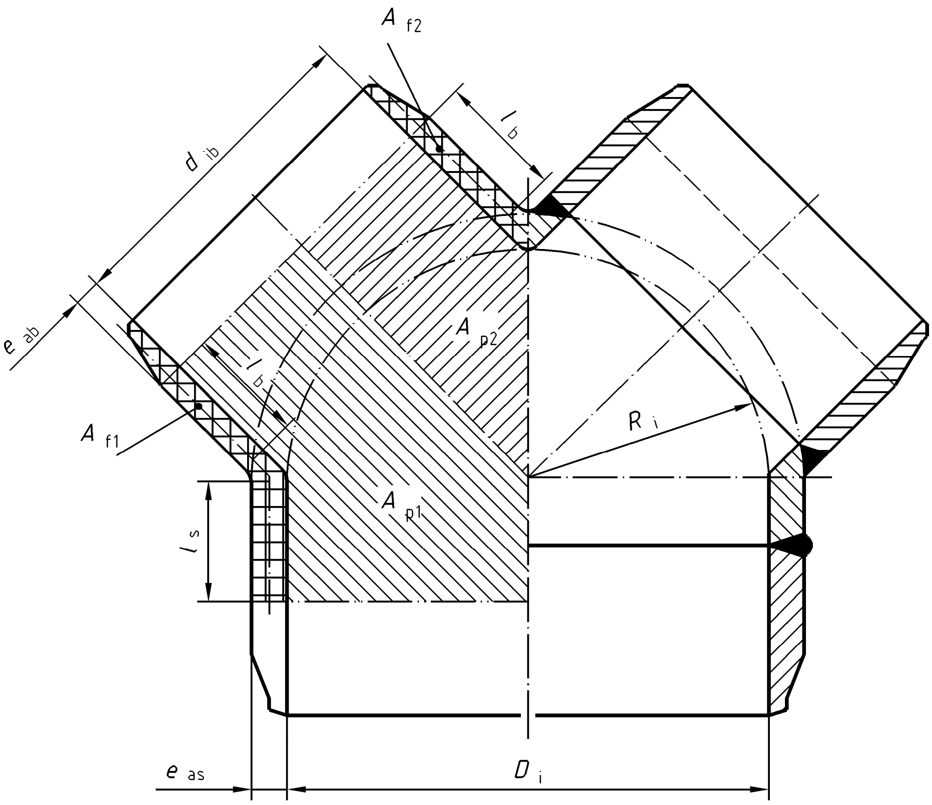


Figure 8.6.2-1 — Fabricated spherical Y-piece

8.6.3 Triform reinforced branches

The application of triform branches shall be limited to a maximum temperature of 200 °C.

This type of reinforcement shall be used only on piping where no significant thermal stresses occur during operation.

For notations, see Figure 8.6.3-1.

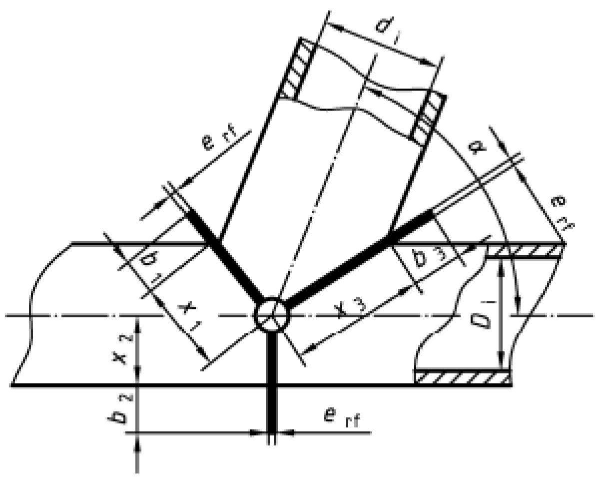


Figure 8.6.3-1 — Triform reinforced branches

The principle of calculation consists of a check of the strength of the external reinforcement provided for withstanding the load due to the pressure in the branch:

$$W \leq \sum_{1}^{n} W_j \tag{8.6.3-1}$$

$$W = 2R_{\mathrm{p0,2}}\, e_{\mathrm{as}} \frac{d_{\mathrm{i}}^2}{D_{\mathrm{i}}}(1 - 0{,}7\sin\alpha) \tag{8.6.3-2}$$

$$W_j = \frac{e_{\mathrm{rf}}\left(3{,}7R_{\mathrm{p0,2}}\right) b^2}{4b + 3x_j} \tag{8.6.3-3}$$

where

b_j is the height of reinforcement (j = 1 , 2 , 3);

D_{i} is the inside diameter of shell;

d_{i} is the diameter of branch;

e_{as} is the analysis thickness of the shell which may be taken as equal to $e_{\mathrm{ord,s}}$, defined in 3.2;

e_{rf} is the thickness of reinforcement;

n is the number of reinforcements;

W is the load carried by the reinforcements;

W_j is the strength of a reinforcement, calculated as a function of the values of b_j and x_j (j = 1 , 2 , 3);

x_j is the length of projection of reinforcement withstanding bending (j = 1 , 2 , 3);

α is the angle between the axes of shell and branch.

9 Design of piping components under external pressure

9.1 General

The rules in clause 9 shall take account of loading due to external pressure. These rules shall not apply within the creep range.

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.

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 $e/D_o \geq 0{,}01$; and

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

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.

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

NOTE 1 The rules of clause 9 apply to cylindrical shells that are circular to within 0,5 % on the radius, measured from the true centre.

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.

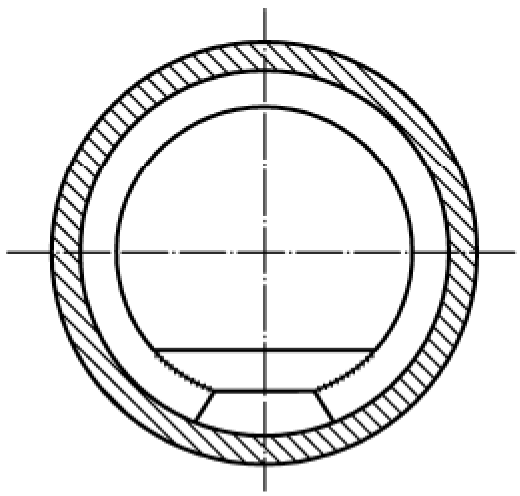


Figure 9.1-1 — Internal stiffening rings with a reinforced cut-out

9.2 Symbols and elastic stress limits

9.2.1 Symbols

For the purposes of clause 9, the symbols given in Table 9.2.1-1 shall apply in addition to those given in Table 3.2-1.

Table 9.2.1-1 — Additional symbols for the purposes of clause 9

Symbol	Description	Unit
A_e	cross-sectional area of stiffener plus effective length of shell	mm²
A_f	area of flange	mm²
A_s	cross-sectional area of stiffener	mm²
A_w	area of web	mm²
E_t	modulus of elasticity of material of part under consideration at design temperature, t	MPa (N/mm²)
L	unstiffened length of the shell	mm
L_c	see Figures 9.3.1-1	mm
R_f	radius to part of the stiffener furthest from the shell (see Figure 9.3.4-1)	mm
R_m	mean radius of the cylindrical, shells or sections	mm
R_s	radius of the centroid of the stiffener cross section (see Figure 9.3.4-1)	mm
$R_{p0,2\,t}$	minimum 0,2 % proof strength at temperature of pipe	MPa (N/mm²)
$R_{p0,2\,\mathrm{S}\,t}$	minimum 0,2 % proof strength at temperature of stiffener	MPa (N/mm²)
S, S_s	elastic stress limits for shell and stiffener, respectively	MPa (N/mm²)
e_a	Analysis wall thickness of the shell	mm
e_f	thickness of flange of stiffener section	mm
e_w	thickness of web of stiffener section	mm
h	external height of a dished end	mm
h_s	radial height of stiffener between flanges	mm
I_c	second moment of area of the composite cross section of the stiffener and effective length of shell acting with it about an axis parallel to the axis of the cylinder passing through the centroid of the combined section	mm^4
k	safety factor	-
k_s	factor depending on the fabrication of stiffener	-
n	number of circumferential waves for a stiffened cylinder	-
n_{cyl}	number of circumferential waves for a unstiffened part of cylinder	-
p	specified external design pressure	MPa (N/mm²)
p_n	theoretical elastic instability pressure of a stiffened cylinder	MPa (N/mm²)
p_m	theoretical elastic instability pressure for collapse of perfect cylindrical shell	MPa (N/mm²)

To be continued

Table 9.2.1-1 (continued)

Symbol	Description	Unit
p_r	calculated lower bound collapse pressure	MPa (N/mm²)
p_y	pressure at which mean circumferential stress in cylindrical shell midway between stiffeners reaches yield point of material	MPa (N/mm²)
p_{ys}	pressure causing circumferential yield of stiffener	MPa (N/mm²)
r_i	radius of the point on the stiffener web closest to the shell about which rotation is assumed in the stiffener tripping calculation (see Figure 9.3.4-1)	mm
R_i	Radius : see Figure 9.3.4-1	mm
b	width of the stiffener in contact with shell	mm
w_f	projecting width of flange of stiffener	mm
ε	mean elastic circumferential strain at collapse	-
λ	parameter for stiffeners	-
σ_s	maximum stress in heavy stiffener	MPa (N/mm²)
σ_i	instability stress at which sideways tripping occurs	MPa (N/mm²)
α	angle of inclination of conical shell to axis	°

9.2.2 Elastic stress limits

The elastic stress limits shall be:

— for non austenitic steels:

$$S = R_{p0,2\ t} \tag{9.2.2-1}$$

$$S_s = R_{p0,2\,s\ t} \tag{9.2.2-2}$$

— for austenitic steels:

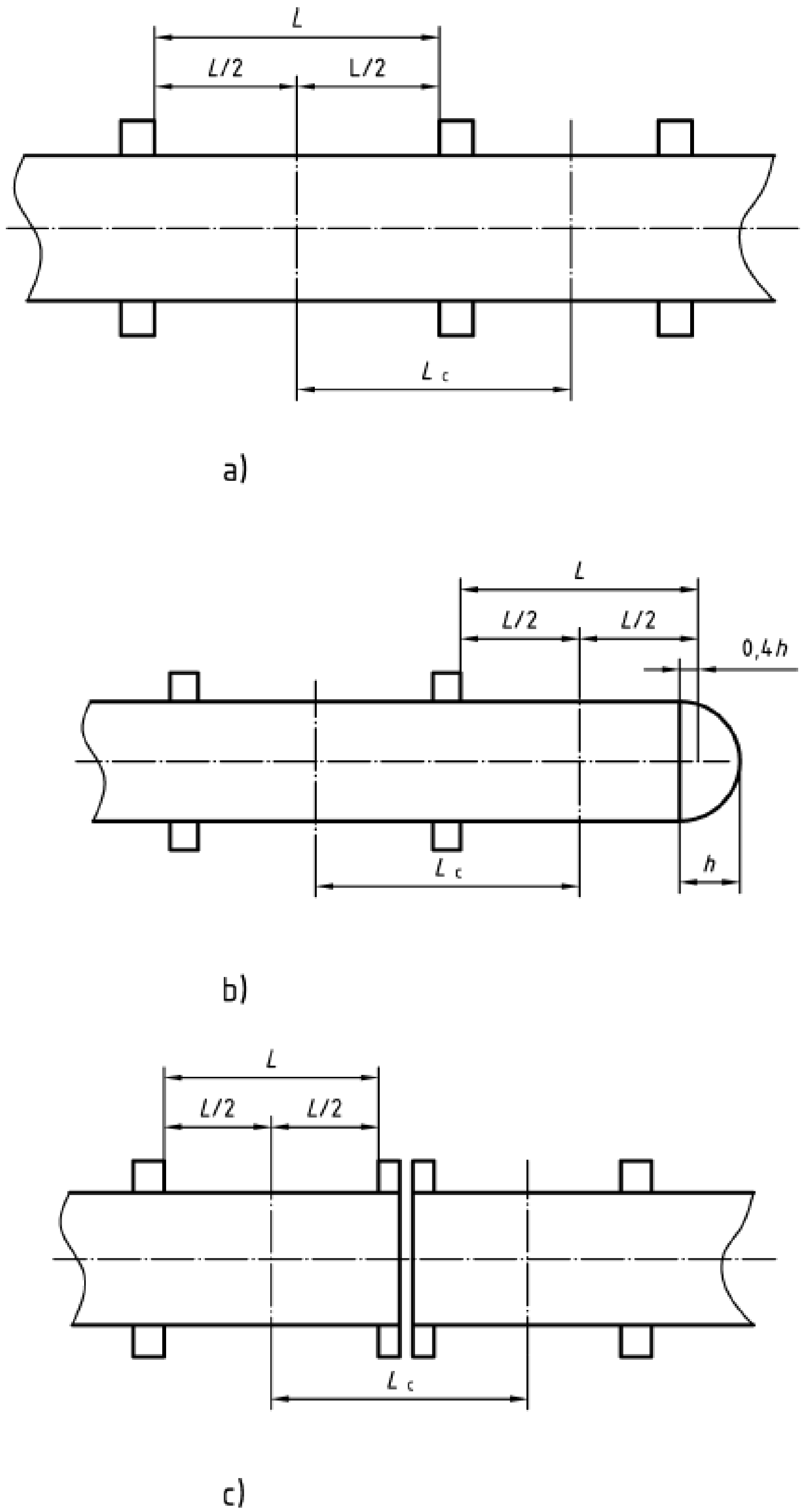
$$S = \frac{R_{p0,2\ t}}{1,25} \tag{9.2.2-3}$$

$$S_s = \frac{R_{p0,2\ s\ t}}{1,25} \tag{9.2.2-4}$$

9.3 Cylindrical pipes, elbows and mitre bends

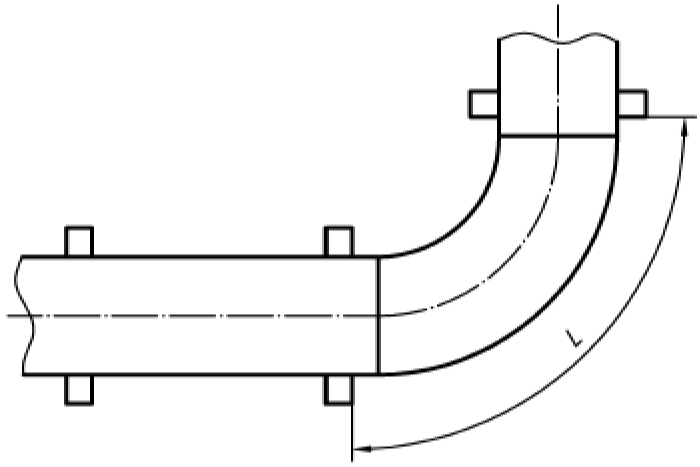
9.3.1 Determination of lengths

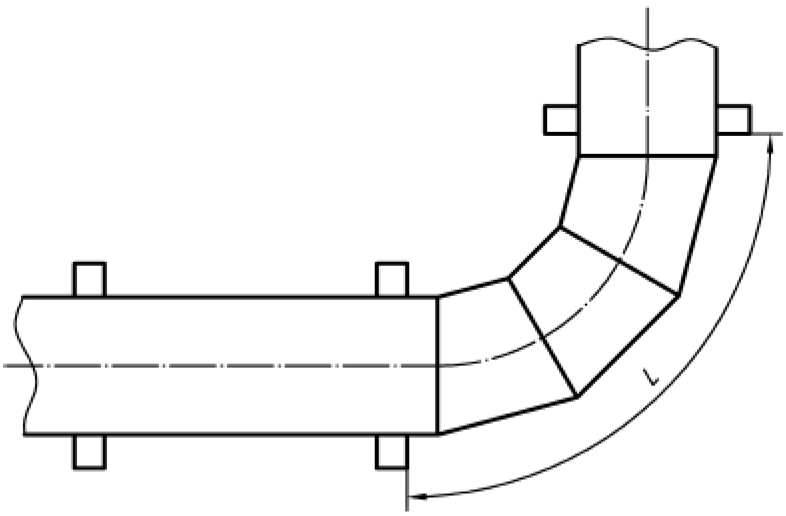
The lengths L and L_c shall be determined from Figure 9.3.1-1.



Key
a) Single pipe
b) Pipe with end
c) Pipes with flange connection

Figure 9.3.1-1 — Determination of lengths in cylindrical pipes *(continued)*





Key

d) Pipe with bend or elbow

e) Pipe with mitre bend

NOTE *L* measured on extrados

Figure 9.3.1-1 — Determination of lengths in cylindrical pipes

9.3.2 Interstiffener collapse

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

a) Estimate a value for e_a and calculate p_y as follows:

$$p_y = \frac{S\ e_a}{R_m} \tag{9.3.2-1}$$

b) Calculate p_m using the same value for e_a chosen for calculation of p_y:

$$p_m = \frac{E_t \, e_a \, \varepsilon}{R_m} \tag{9.3.2-2}$$

where

ε is calculated from:

$$\varepsilon = \frac{1}{n_{cyl}^2 - 1 + \frac{Z^2}{2}} \left\{ \frac{1}{\left(\frac{n_{cyl}^2}{Z^2} + 1 \right)^2} + \frac{e_a^2}{12 R_m^2 \left(1 - \nu^2\right)} \left(n_{cyl}^2 - 1 + Z^2 \right)^2 \right\} \tag{9.3.2-3}$$

where

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

$$Z = \frac{\pi \, R_m}{L} \tag{9.3.2-4}$$

and L is determined in accordance with 9.3.1

c) Calculate p_m / p_y and determine p_r / p_y from Table 9.3.2-1.

Table 9.3.2-1 — Cylindrical straight pipes and reducers (hoop stress governing)

p_m / p_y	*0*	0,25	0,5	0,75	1,0	1,25	1,5	1,75
p_r / p_y	*0*	0,1245	0,2505	0,375	0,4995	0,6045	0,6795	0,72
p_m / p_y		2,0	2,25	2,5	2,75	3,0	3,25	3,5
p_r / p_y		0,7545	0,78	0,8025	0,822	0,8355	0,849	0,861
p_m / p_y		3,75	4,0	4,25	4,5	4,75	5,0	5,25
p_r / p_y		0,87	0,879	0,8865	0,8955	0,9045	0,9135	0,9165
p_m / p_y		5,5	5,75	6,0	6,25	6,5	6,75	7,0 and above
p_r / p_y		0,9225	0,9285	0,9345	0,9405	0,9465	0,9525	0,9585

d) Calculate the pressure p_r from p_r / p_y and p_y which shall conform to the following:

$$p_r \geq k \, p \tag{9.3.2-5}$$

where

k = 1,5

except

— for specific applications where k may be increased by agreement between the parties involved;

— for steel castings where k = 1,5 × 1,25.

If p_r is less than $k\,p$, the assumed value of e_a shall be increased or the spacing of stiffeners shall be adjusted until the required value is obtained.

9.3.3 Overall collapse of stiffened pipes

The following calculations shall be performed.

a) Calculate p_n from:

$$p_n = \frac{3}{R_m{}^3 L_c} E_t I_c \tag{9.3.3-1}$$

The value of p_n shall conform to the following:

$$p_n \geq k\,k_s\,p \tag{9.3.3-2}$$

where

$k_s = 1{,}2$ for fabricated or hot formed stiffeners (low residual stresses);

$k_s = 1{,}33$ for cold formed stiffeners (high residual stresses).

If p_n is less than $k\,k_s\,p$, either additional or heavier stiffening shall be provided or the pipe thickness shall be increased.

b) Calculate p_{ys} from :

$$p_{ys} = \frac{S_s\, e_a\, R_f}{R_m^2 \left(1 - \frac{\nu}{2}\right)} \tag{9.3.3-3}$$

c) Calculate the maximum stress in the stiffener from:

$$\sigma_s = \frac{k k_s S_s p}{p_{ys}} + \frac{E_t\, \delta\, (n^2 - 1)\, 0{,}005 k k_s\, p}{R_m\, (p_n - k k_s\, p)} \tag{9.3.3-4}$$

where

$n = 2$

$$\delta = \max\{\lambda(R_m - R_f) - X_c + e_a/2; X_c\} \tag{9.3.3-5}$$

$$X_c = \frac{\left\{\left(\frac{e_a^2}{2}\right)L_c + A_s\left[\frac{e_a}{2} + \lambda(R_m - R_s)\right]\right\}}{A_e} \tag{9.3.3-6}$$

λ = 1 for internal stiffeners;

λ = -1 for external stiffeners.

The stress σ_s shall conform to:

$$0 \le \sigma_s \le S_s \tag{9.3.3-7}$$

If equation (9.3.3-7) is not fulfilled, either additional or heavier stiffening shall be provided or the pipe thickness shall be increased.

9.3.4 Stiffener stability

To ensure lateral stability, the following calculations shall be performed

a) For a stiffener other than flat bar

1) the stress σ_i shall conform to the following:

$$\sigma_i = E_t \ C \frac{p_{ys}}{p} > S_s \tag{9.3.4-1}$$

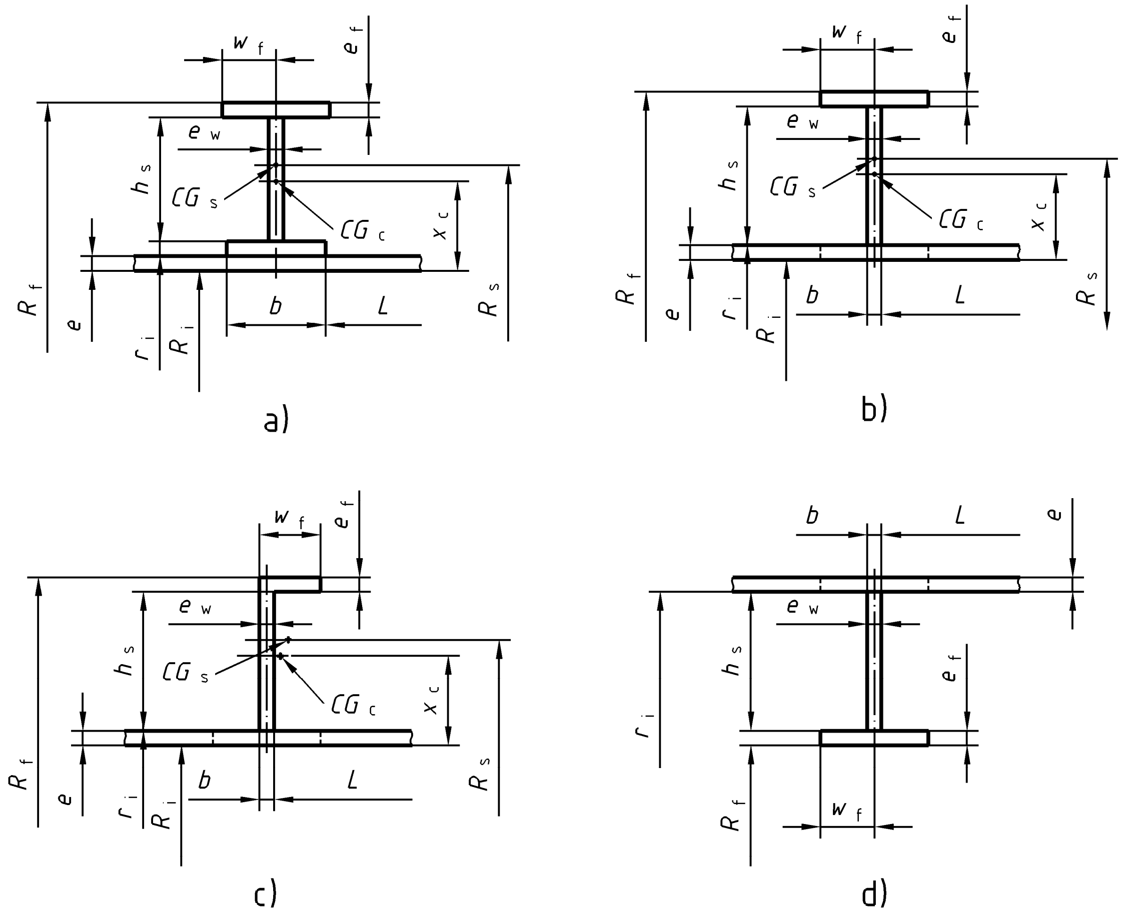
where *C* shall be:

— for Figures 9.3.4-1 a), b) and d):

$$C = \frac{h_s e_w^3 + 8 e_f w_f^3}{r_i \left[6 h_s^2 e_w + 12 e_f \ w_f \left(2h_s + e_f\right)\right]} \tag{9.3.4-2}$$

— for Figures 9.3.4-1 c):

$$C = \frac{e_f \ w_f^3}{r_i \left[6 \ h_s^2 \ e_w + 6 \ e_f \ w_f \left(2h_s + e_f\right)\right]} \cdot \left[\frac{4h_s \cdot e_w + 3e_f \cdot w_f}{h_s \cdot e_w + 3e_f \cdot w_f}\right] \tag{9.3.4-3}$$



Key

a) External I stiffener

b) External T stiffener

c) External L stiffener

d) Internal T stiffener

CG_s — Centroid of stiffener

CG_c — Centroid of stiffener plus effective length of shell

where the effective length, l_{ps}, is given by:

$$l_{ps} = 1{,}56\sqrt{R_i e} \tag{9.3.4-4}$$

Figure 9.3.4-1 — Types of stiffeners other than flat bar

2) If the stiffener is flanged at the edge farthest from the shell, proportions of stiffeners shall conform to the following :

$$\frac{h_s}{e_w} \leq \max\left(1{,}1\sqrt{\frac{E_t}{S_s}}\ ;\ 0{,}67\sqrt{\frac{E_t\ p_{ys}}{S_s\ p}}\right) \tag{9.3.4-5}$$

or

$$\frac{w_{\mathrm{f}}}{e_{\mathrm{f}}} \leq \max\left(0{,}51\sqrt{\frac{E_t}{S_{\mathrm{s}}}}\ ;\ 0{,}32\sqrt{\frac{E_t\ p_{\mathrm{ys}}}{S_{\mathrm{s}}\ p}}\right) \tag{9.3.4-6}$$

b) for a flat bar stiffener, the stress σ_{i} shall conform to the following:

$$\sigma_{\mathrm{i}} > \frac{4\,p\,S_{\mathrm{s}}}{p_{\mathrm{ys}}} \tag{9.3.4-7}$$

where σ_{i} shall be calculated from values obtained from Table 9.3.4-1 for internal stiffeners or from Table 9.3.4-2 for external stiffeners.

Table 9.3.4-1 — Values of $(\sigma_{\mathrm{i}}/E_t)(h_{\mathrm{s}}/e_{\mathrm{w}})^2$ for internal flat bar stiffeners

h_{s}/R_m n_{cyl}	0,01	0,02	0,04	0,06	0,08	0,10	0,12	0,14	0,16	0,18	0,20
2	0,0119	0,0236	0,0466	0,0691	0,0913	0,114	0,135	0,157	0,180	0,202	0,225
3	0,0239	0,0461	0,0865	0,123	0,156	0,187	0,217	0,247	0,276	0,305	0,334
4	0,0395	0,0734	0,130	0,176	0,216	0,252	0,286	0,319	0,353	0,386	0,421
5	0,0577	0,103	0,171	0,223	0,266	0,304	0,341	0,378	0,416	0,456	0,498
6	0,0778	0,132	0,208	0,262	0,306	0,347	0,387	0,428	0,472	0,517	0,570
7	0,0981	0,160	0,240	0,294	0,340	0,382	0,427	0,474	0,527	0,580	0,643
8	0,119	0,186	0,268	0,322	0,369	0,415	0,465	0,517	0,580	0,647	0,725
9	0,139	0,210	0,290	0,345	0,394	0,445	0,502	0,565	0,638	0,720	0,812
10	0,158	0,231	0,310	0,365	0,417	0,474	0,536	0,614	0,696	0,792	0,903
11	0,176	0,249	0,328	0,383	0,440	0,502	0,575	0,662	0,758	0,874	1,010
12	0,193	0,266	0,343	0,400	0,461	0,531	0,614	0,715	0,831	0,966	1,121
13	0,209	0,280	0,356	0,416	0,483	0,560	0,657	0,768	0,903	1,058	
14	0,224	0,293	0,368	0,431	0,502	0,594	0,700	0,831	0,981		
15	0,237	0,304	0,379	0,446	0,527	0,628	0,749	0,894	1,068		
16	0,249	0,314	0,389	0,461	0,551	0,662	0,797	0,961			
17	0,260	0,324	0,399	0,476	0,575	0,696	0,850	1,034			
18	0,270	0,332	0,409	0,493	0,599	0,734	0,903	1,106			
19	0,279	0,339	0,418	0,507	0,623	0,773	0,961				
20	0,287	0,346	0,427	0,522	0,652	0,816	1,019				

NOTE 1 For intermediate values of h_{s}/R_m use logarithmic interpolation.

NOTE 2 Since $(\sigma_{\mathrm{i}}/E_t)(h_{\mathrm{s}}/e_{\mathrm{w}})^2$ is limited to a maximum value of 1,14, values of the expression should not be extrapolated beyond that value.

Table 9.3.4-2 — Values of $(\sigma_i / E_t)(h_s / e_w)^2$ for external flat bar stiffeners

h_s / R_m / n_{cyl}	0,01	0,011	0,012	0,015	0,02	0,025	0,03	0,04	0,045
2	0,012	0,0132	0,0144	0,0180	0,0241	0,0303	0,0366	0,0492	0,0557
3	0,0257	0,0284	0,0311	0,0374	0,0537	0,0687	0,0846	0,119	0,138
4	0,0466	0,0517	0,0570	0,0734	0,103	0,137	0,175	0,268	0,326
5	0,0768	0,860	0,0955	0,126	0,187	0,263	0,361	0,679	0,965
6	0,120	0,136	0,153	0,211	0,340	0,537	0,881	1,44[a]	
7	0,183	0,211	0,242	0,356	0,677	1,48[a]			
8	0,279	0,331	0,390	0,648	1,92[a]				
9	0,438	0,541	0,676	1,49[a]					
10	0,736	0,998	1,420[a]						
11	1,490[a]								
h_s / R_m / n_{cyl}	0,05	0,06	0,08	0,10	0,12	0,14	0,16	0,18	0,20
2	0,0622	0,0755	0,103	0,133	0,164	0,198	0,236	0,277	0,324
3	0,157	0,201	0,310	0,462	0,695	1,10	1,99[a]		
4	0,395	0,581	1,44[a]						
5	1,46[a]								

[a] These values are provided to enable intermediate values to be interpolated

NOTE 1 For intermediate values of h_s / R_m use logarithmic interpolation.

NOTE 2 Since $(\sigma_i / E_t)(h_s / e_w)^2$ is limited to a maximum value of 1,14, values of the expression should not be extrapolated beyond that value.

NOTE 3 Buckling can not occur for n > 10, h_s / R_m > 0,01 under external pressure.

9.3.5 Heating/cooling channels

The rules in this sub-clause shall apply for the thickness of a cylinder to which circumferential heating or cooling channels are attached.

NOTE Such channels are also known as hemicoils or limpet coils. Typical forms of construction are shown in Figure 9.3.5-1.

The shell shall be designed, using the rules of 9.3.3 to resist overall collapse. The pressure inside the channels shall be ignored and the channels may be considered as stiffeners.

The shell shall also be designed to resist the pressure difference between the channel and the inside of the cylinder, i.e. the calculation pressure shall be checked in accordance with 9.3.2. In addition, the minimum thickness shall be not less than:

$$e \geq l \sqrt{\frac{p}{3Sk}} \tag{9.3.5-1}$$

where

$$l = \max\left(l_1 ; l_2\right) \tag{9.3.5-2}$$

p is the maximum internal pressure.

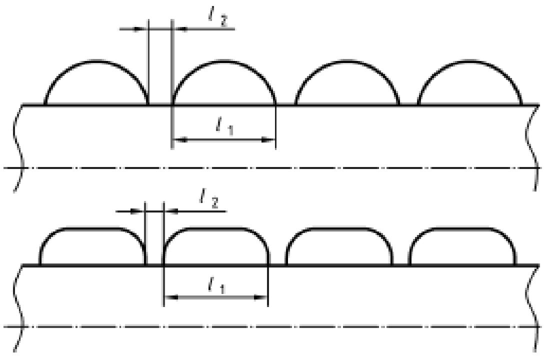


Figure 9.3.5-1 — Heating / cooling channels / stiffeners

Furthermore, the shell shall be designed for the pressure difference between the space outside the hemi-coil and the space inside the pipe.

9.4 Reducers (conical shells)

9.4.1 Conical shells shall be designed in accordance with 9.4.3 where the cone/cylinder junction may be considered as an effective stiffener, as defined in 9.4.2, and in accordance with 9.4.4 where the cone to cylinder junction does not meet the requirements of 9.4.2.

9.4.2 The cone/cylinder junction shall be considered as an effective stiffener where the moment of inertia, I_x, taken parallel to the axis of cylinder, of the part of the cone and cylinder within a distance $\sqrt{D_{eq} e}$ (see Figure 9.4.3-1) on either side of the junction is not less than :

$$I_x = 0{,}18 D_{eq} L \ D_s^2 \frac{p_c}{E_t} \tag{9.4.2-1}$$

where

D_s is the diameter of the centroid of the moment of inertia of the stiffening area cross section.

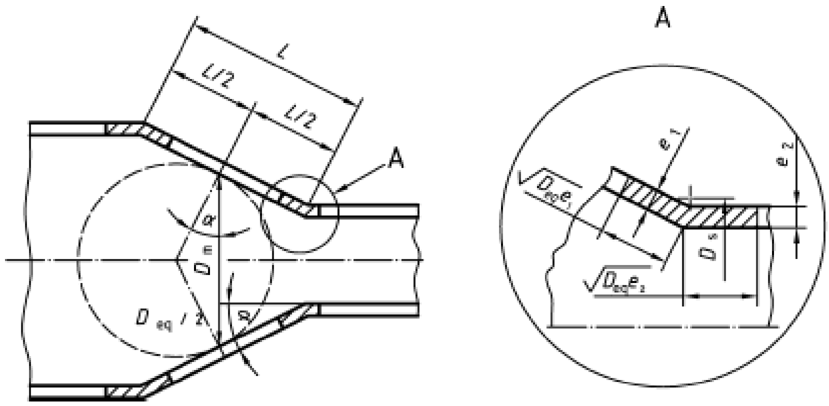
9.4.3 The design of conical section with effective cone to cylinder junctions shall be carried out as for cylindrical shells (see 9.3) using the following:

— *L* is the length of conical section (see Figure 9.4.3-1);

— D_{eq} is the equivalent diameter calculated in the following way:

$$D_{\mathrm{eq}} = \frac{D_{\mathrm{m}}}{\cos\alpha} \tag{9.4.3-1}$$

The design of this equivalent cylindrical shell shall be in accordance with 9.3.



Detail A

Figure 9.4.3-1 — Conical section with effective cone to cylinder junction

9.4.4 For conical shell which do not have effective cone to cylinder junctions the following values shall be used in the calculations for equivalent cylindrical shell in 9.3:

— L is the axial length between effective stiffeners (see Figure 9.4.4-1)

— D_{o} is the outside diameter of cylinder attached to the large diameter of the cone (see Figure 9.4.4-1)

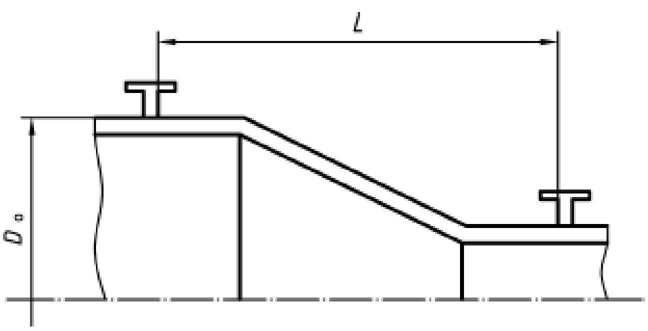


Figure 9.4.4-1 — Conical section without effective cone to cylinder junction

9.5 Dished ends

9.5.1 Hemispherical ends

9.5.1.1 Design procedure

The design thickness shall be not less than that determined by the following procedure.

a) Chose a value for *e* and calculate p_{y} as follows:

$$p_{\mathrm{y}} = \frac{2Se}{R_{\mathrm{m}}} \tag{9.5.1-1}$$

b) Calculate p_m using the same value for e chosen for the calculation of p_y :

$$p_m = \frac{1{,}21\, E_t\, e^2}{R_m^{\ 2}} \tag{9.5.1-2}$$

c) Calculate p_m/p_y and determine p_r / p_y from Table 9.5.1-1

Table 9.5.1-1 — Values of p_r / p_y versus p_m/p_y

p_m/p_y	0	0,5	1	1,5	2	2,5	3,0
p_r / p_y	0	0,09	0,18	0,255	0,324	0,3855	0,435
p_m/p_y	3,5	4	4,5	5,0	5,5	6	6,5 and above
p_r / p_y	0,4785	0,51	0,5325	0,5475	0,5595	0,567	0,57

d) Calculate the pressure p_r from p_r / p_y and p_y which shall conform to the following :

$$p_r \geq k\,p \tag{9.5.1-3}$$

If p_r is less than $k\,p$, the assumed value of e shall be increased.

9.5.1.2 Permissible shape deviations

The method in 9.5.1.1 shall apply only to hemispherical ends that are spherical to within 1 % on the radius and in which the radius of curvature based on an arc length of $2{,}4\sqrt{e\, R_{max}}$ does not exceed the nominal value by more than 30 %.

NOTE For some applications, this criterion for applicability can be too stringent to be met owing to difficulties of manufacture and measurement. In such cases, it is permissible to divide the pressure p_r obtained from the above procedure by the factor $(R_{max}/1{,}3\, R)^2$ where R_{max} is the maximum local radius of curvature either measured or estimated conservatively.

9.5.2 Torispherical ends

Torispherical ends shall be designed as hemispherical ends of mean radius *R* equal to the external dishing or crown radius.

9.5.3 Ellipsoidal ends

Ends to true semi-ellipsoidal form shall be designed as hemispherical ends of mean radius *R* equal to the maximum radius of the crown, i.e. $\dfrac{D^2}{4h}$.

10 Design for cyclic loading

10.1 General

Cyclic changes in pressure, temperature and external loading can cause damage by fatigue cracking at levels below those used for the static design conditions.

If the conditions in 10.2, 10.3 and 10.4 are not fulfilled, the static design shall be supplemented by a detailed fatigue analysis carried out in accordance with 12.4.

The analysis shall take into account the magnitude and frequency of all specified cyclic load conditions, and ensure that for each loading, the allowable fatigue cycles, N_i, of the piping exceeds the number of predicted loads cycles, n_i, and the accumulated fatigue damage for all these conditions does not exceed 1.

$$\sum_{i=1}^{m} n_i / N_i \leq 1 \tag{10.1-1}$$

where m is the number of elementary loadings.

Fatigue calculations shall be based upon cyclic variations in the normal range of operating conditions, including predicted excursions. The assessments shall be related to the actual component thicknesses and material properties at the relevant temperatures.

NOTE Surface smoothness and the presence of welds influence fatigue cracking behaviour, and should be considered when assessing the likelihood of fatigue. Where there is a significant risk of fatigue failure, the design should be reviewed to reduce the risk by considering changes to the configuration to lower peak stresses, the provision of smooth profiles, particularly at welds, and the use of less susceptible materials.

10.2 Exemption from detailed fatigue analysis

Detailed fatigue analysis (see 12.4) shall not be required if one of the following conditions is met:

a) the system design can be shown to replicate closely a previously analysed acceptable design;

b) the system design is similar to a currently operating system design;

c) the total number of alternating load cycles from all sources is less than 1 000;

d) the calculated maximum range of principal cyclic stress is less than 47 N/mm^2 for carbon and austenitic steels. When considering fillet welds, this value shall be reduced to 35 N/mm^2;

e) all the following conditions are met simultaneously :

   1) the equivalent number of full pressure cycles as defined in accordance with 10.3.1 does not exceed 1 000;

   2) the mechanical loading on branches is such that the maximum total stress range caused by mechanical loads, inclusive of stress concentration factors, does not exceed 1/3 of the design stress f (including stress range factors) at the calculation temperature;

   3) the thickness does not exceed 125 mm for ferritic steels and 60 mm for austenitic steels and the number of thermal cycles is less than 7 000.

10.3 Fatigue design for cyclic pressure

10.3.1 Equivalent full load cycles

Clause 10.2 e), 1) permits exemption from analysis where the number of full load cycles does not exceed 1 000. For pressure cycles with smaller ranges, an equivalent number of full load cycles may be determined according to the following equation:

$$N_{eq} = N_f + \sum_{i=1}^{n} \left\{ \left(\left[\hat{p}_i - \check{p}_i \right] / p_c \right)^{3,5} N_i \right\} \tag{10.3.1-1}$$

where

N_{eq} is the number of equivalent full load cycles;

n is the number of different pressure ranges $(\hat{p}_i - \check{p}_i)$;

N_f is the number of full pressure cycles with a range $\left(\hat{p} - \check{p}\right) = p_c$ (see 10.3.2.2);

N_i is the number of different pressure cycles with a range $\left(\hat{p}_i - \check{p}_i\right)$ less than p_c;

p_c is the calculation pressure (full load pressure range).

The design of the piping shall be considered satisfactory if the number of equivalent full pressure cycles is less than 1 000.

10.3.2 Simplified fatigue analysis

10.3.2.1 General

Where cyclic loading requiring calculation arises only from variations in pressure, the simplified fatigue analysis shall be permitted.

It uses the static design criteria and takes into account the relevant fatigue peak stresses by the use of a stress concentration factor η for a range of typical geometries. The method is approximate, and less conservative dimensioning may result from the use of more detailed analysis in accordance with 12.4.

The rules shall apply for pressure containing parts of piping of ferritic and austenitic rolled and forged steels manufactured and tested in accordance with EN 13480-2 and EN 13480-4.

The calculation only applies for components dimensioned on the basis of non-time-dependent strength characteristics and subjected to cyclic loadings only in the form of pressure fluctuations.

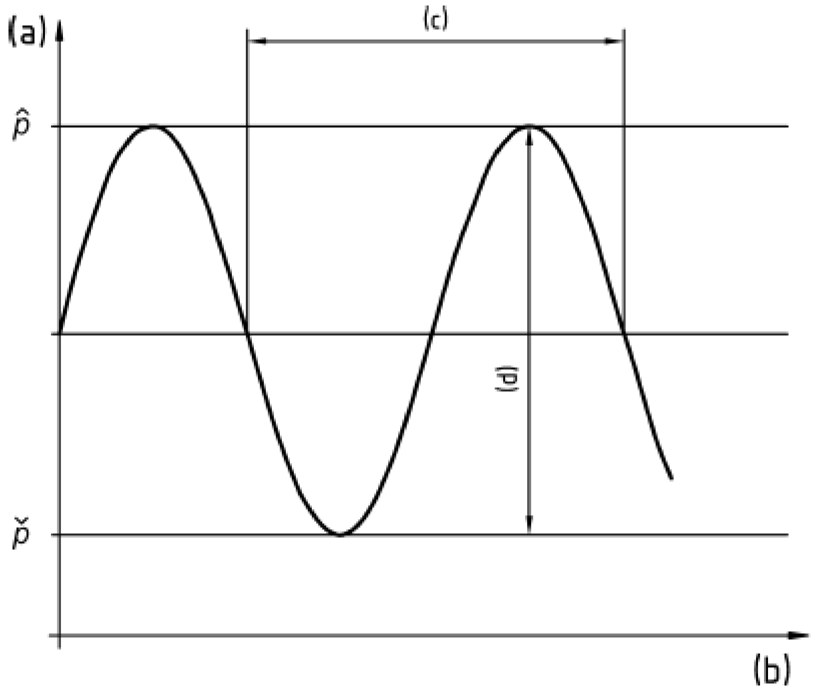
NOTE 1 The term « cyclic loading » is the change over time of a load regardless of the magnitude and arithmetic sign of the mean value.

Additional cyclic loads, for example loads due to rapid changes in temperature during operation or from external forces and moments, shall be assessed within the framework of a detailed fatigue analysis (see 12.4).

The rules shall apply if there are no influences from the fluid which will reduce the fatigue life (see 10.3.2.8).

NOTE 2 10.3.2 does not need to be applied if the pressure fluctuations superimposed on the service pressure do not exceed 10 % of the allowable operating pressure.

If the number of expected operational pressure fluctuations exceeds the allowable number of load cycles calculated in accordance with 10.3.2, the design shall be changed or a detailed fatigue analysis conducted. The criterion for failure due to cyclic loading shall be an incipient technical crack, which is a crack type material separation that can be detected by visual means or non-destructive testing. A measure for cyclic loading is the stress range (double load amplitude) arising from the action of the repeatedly changing pressure (see Figure 10.3.2-1).



Key

(a) pressure p

(b) time

(c) load cycle

(d) pressure fluctuation range $\left(\hat{p}-\check{p}\right)$

Figure 10.3.2-1 — Pressure curve and load cycle (schematic)

The allowable number of load cycles shall be linked to the results of the dimensioning and design of the piping components in accordance with 4.6 and clause 6.

NOTE 3 In the case of frequent load cycles with considerable cyclic loading, detailed fatigue analysis would be more practical to assess the design. As a rule, this yields a greater allowable number of load cycles than the calculation to 10.3.2.

NOTE 4 Of special importance are fluctuations between the unpressurized state and the allowable operating pressure (start-ups and shut-downs). These pressure fluctuations can be superimposed with low amplitude operating pressure fluctuation, or with varying amplitude with irregular sequence and differing frequency (operating load collective). An appropriate procedure should be adopted when vacuum load is imposed.

Pressure fluctuations of differing ranges and differing frequencies shall be combined by the linear accumulation law (see 10.3.2.3)

NOTE 5 The number and level of pressure fluctuations which a piping can withstand during its probable lifetime without damage to the pressure-containing parts depends on a large number of different influences, e.g.:

— design, e.g. configuration of component with regard to the avoidance of high stress peaks;

— manufacture, e.g. avoidance of damaging residual stresses and weld imperfections;

— material, e.g. softer steels are normally less notch-sensitive than harder ones. With the notch-sensitive steels it should be noted that the probability of failure is greater if a manufacturing defect is not noticed or operating conditions are unfavourable. The strength of the weld metal should be equal to or greater than that of the base metal;

— surface condition, e.g. smooth surface finish;

— wall thickness, e.g. with equal stress amplitude, increasing wall thickness reduces fatigue;

— temperature, e.g. higher temperatures reduce the cyclic strength of the materials and reduce the component's fatigue life.

NOTE 6 Corrosion arising during operation, especially in notch-sensitive materials, can reduce the number of load cycles which can be withstood. Operational measures and testing during the operating period (see 10.3.2.7.3) are of special importance here. Where a covering layer forms, this should be considered when dimensioning and designing in order to prevent the covering layer from tearing open.

In order to determine the allowable number of load cycles for the whole piping system, the calculations according to 10.3.2.3 shall be performed for the various components of the piping. The smallest value shall be used.

10.3.2.2 Symbols

For the purposes of 10.3.2, the symbols given in Table 10.3.2-1 shall apply in addition to those given in 3.2.

Table 10.3.2-1 — Additional symbols for the purposes of 10.3.2

Symbol	Description	Unit
F_{t^*}	temperature influence factor	-
k	number of intervals of differing pressure fluctuations which together form the load collective	-
p_r	notional pressure	bar
$\left(\hat{p} - \check{p}\right)$	pressure fluctuation range (double amplitude)	bar
F_d	correction factor to take account of the influence of the wall thickness	-
N	operating number of load cycles	-
N_{all}	allowable number of load cycles with a pressure fluctuation range $\left(\hat{p} - \check{p}\right)$	-
t^*	determining calculation temperature during the load cycle	°C
$2\,\sigma_a{}^*$	notional pseudo-elastic stress amplitude	MPa (N/mm²)
$2\,\sigma_{aD}$	notional endurance values	MPa (N/mm²)
η	stress factor	-
f_{20}	design stress f according to 5.2 at 20 °C	MPa (N/mm²)
Superscripts and subscripts: Superscript $\wedge$ maximum value, e.g. $\hat{p}$ Superscript $\vee$ minimum value, e.g. $\check{p}$ Subscript K number index e.g. N_K		

10.3.2.3 Determination of allowable number of load cycles

To determine the allowable number of load cycles, $2\,\sigma_a{}^*$ shall be calculated in accordance with the equation:

$$2\sigma_a^* = \frac{\eta}{F_d F_{t^*}} \frac{\left(\hat{p} - \check{p}\right)}{p_r} f_{20} \tag{10.3.2-1}$$

The notional pressure p_r shall be determined as allowable pressure with full utilisation of the nominal design stress f_{20} for a piping component from the dimensional equations in clauses 6 to 9 and 11, rearranged to give p.

The stress factor η shall represent the upper limit of the stress factors for dimensioning conditions of a certain component geometry arising in practical situations or selected from Table 10.3.2-4.

To take account of the influence of the component size on the cyclic load strength, a correction factor F_d shall be taken for wall thicknesses e_{ord} > 25 mm in accordance with:

$$F_d = \left(\frac{25}{e_{ord}}\right)^{0,25} \tag{10.3.2-2}$$

or from Figure 10.3.2-2. The factor F_d shall be limited to F_d = 0,64.

For forgings, the wall thickness shall be taken as the determining heat treatment diameter from the relevant material standards.

For the calculation, the following shall be defined as the determining calculation temperature during any load cycle under consideration:

$$t^* = 0,75\,\hat{t} + 0,25\,\breve{t} \tag{10.3.2-3}$$

All temperature related factors shall be related to this determining temperature t^* of the relevant load cycle.

With load cycle temperatures of non-time dependent strength characteristics and with, t^* > 100 °C a temperature influence factor F_{t^*} shall be determined:

— for ferritic steel

$$F_{t^*} = 1,03 - 1,5 \times 10^{-4}\ t^* - 1,5 \times 10^{-6}\ t^{*2} \tag{10.3.2-4}$$

— for austenitic steel

$$F_{t^*} = 1,043 - 4,3 \times 10^{-4}\ t^* \tag{10.3.2-5}$$

or shall be taken from Figure 10.3.2-3.

For $t^* \leq$ 100°C, F_{t^*} shall be 1.

The allowable number of load cycles, N_{all}, shall be calculated within the range $10^3 \leq N_{all} \leq 2 \times 10^6$ as a function of the stress amplitude 2 σ_a^* given in equation 10.3.2-1:

$$N_{all} = \left(\frac{B}{2\sigma_a^*}\right)^m \tag{10.3.2-6}$$

where

B is a constant, taken from Table 10.3.2-2

m = 3 for welded joints,

m = 3,5 for unwelded component areas with rolled or machined surfaces.

Alternatively, N_{all} shall be taken from Figure 10.3.2-4.

NOTE 1 The notch effects from weld seams or surface roughness, and the influence of residual welding stresses from operating pressure have been incorporated in m.

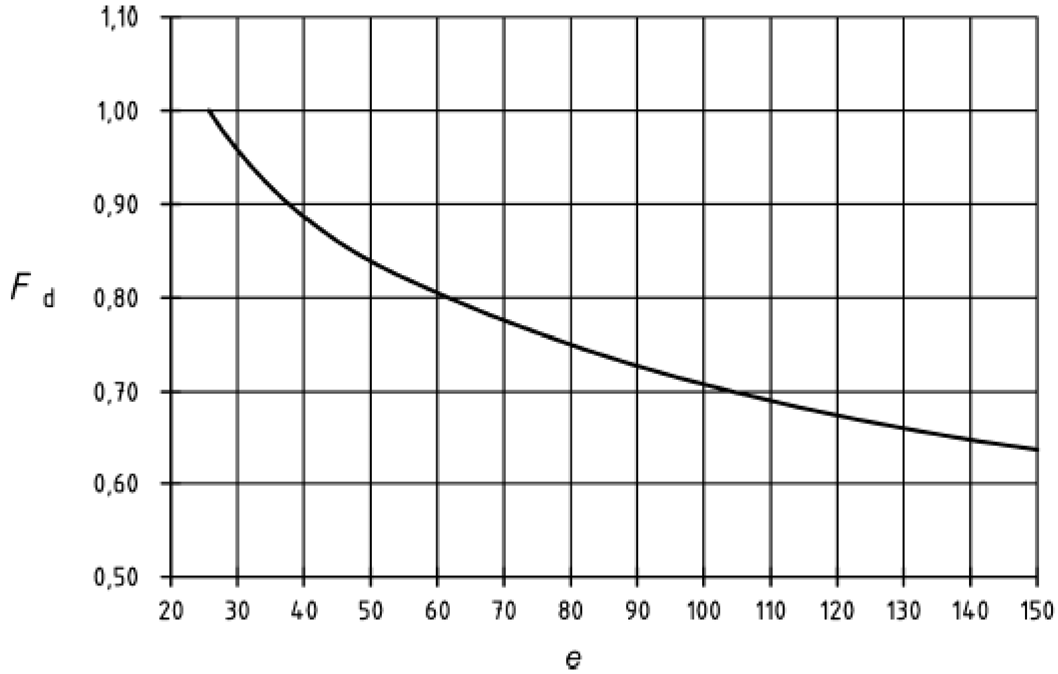
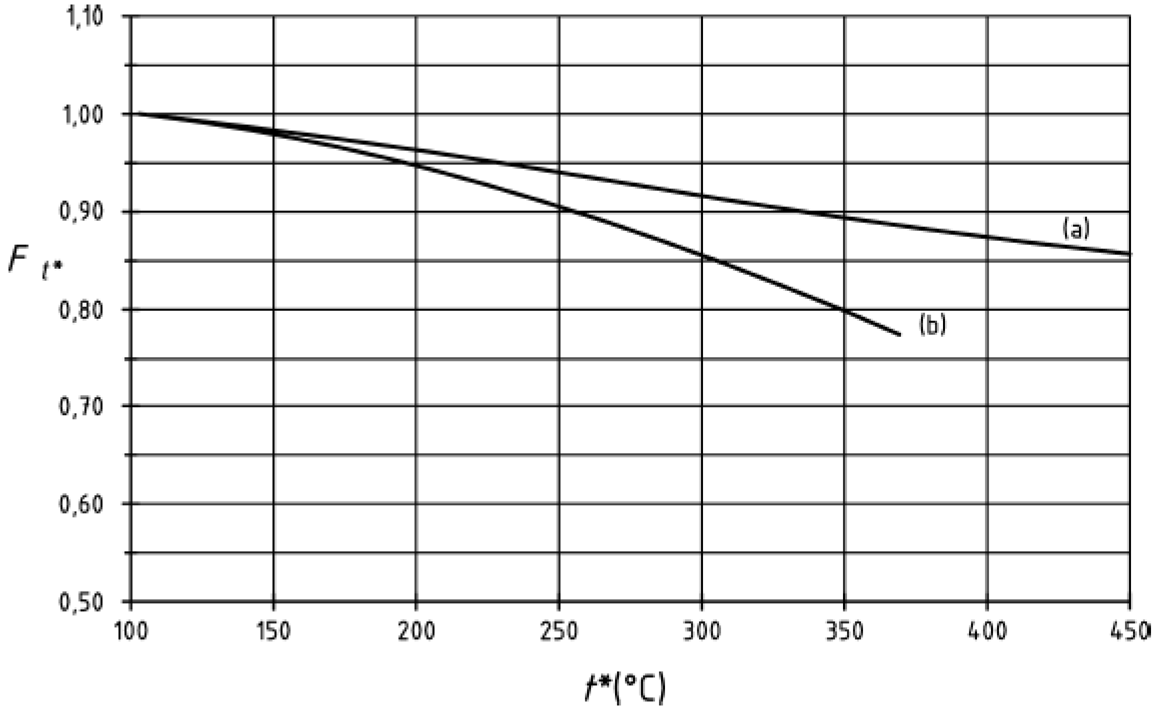


Figure 10.3.2-2 — Correction factor F_d to take into account the wall thickness



Key

(a) austenitic

(b) : ferritic

t* : determining calculation temperature

Figure 10.3.2-3 — Correction factor F_{t^*} to take into account temperature influence

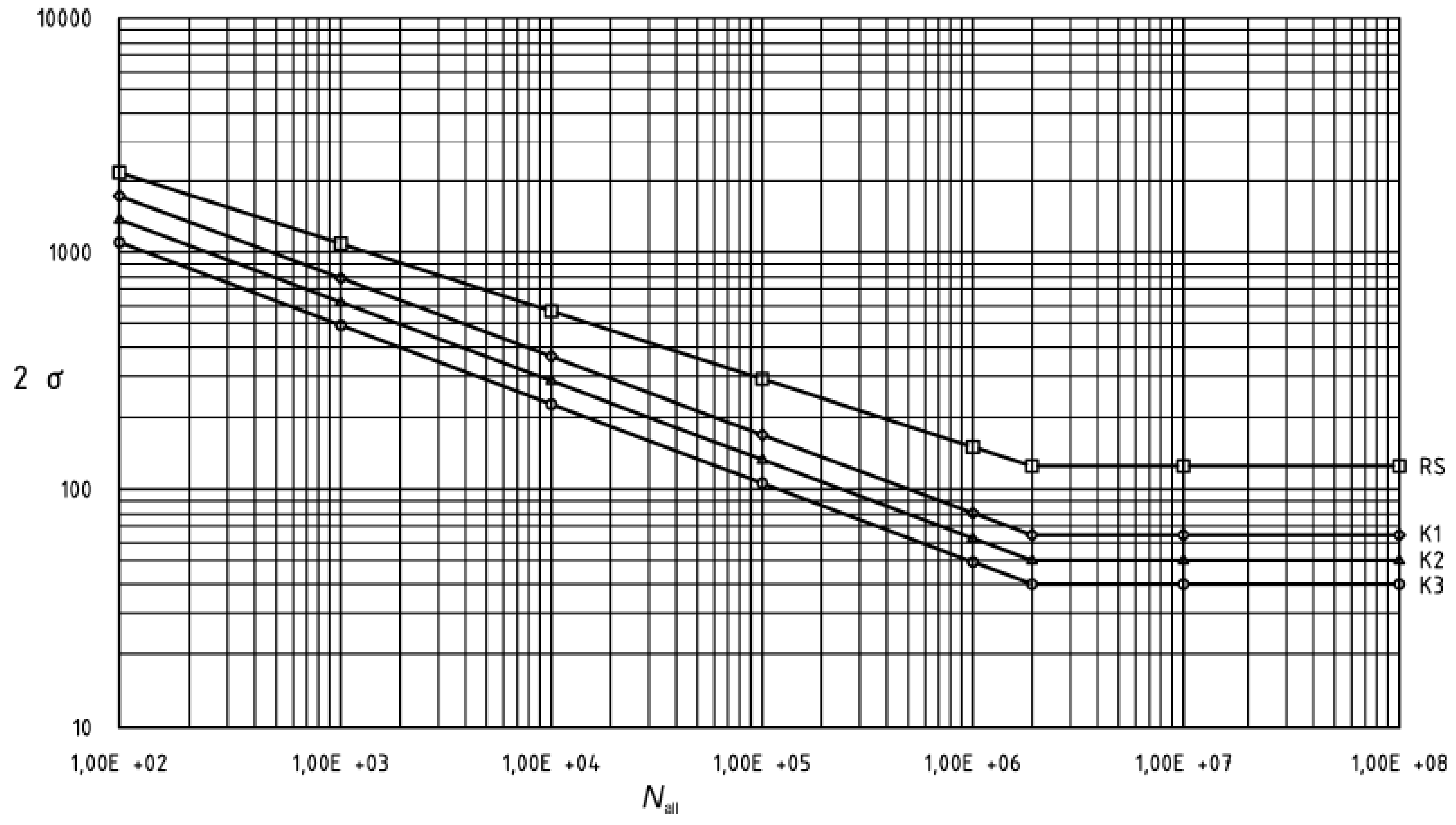


Figure 10.3.2-4 — Allowable number of load cycles for design temperature < 100 °C and wall thickness < 25 mm

The constants B shall be in accordance with Table 10.3.2-2 for class RS (rolled surface) for unwelded components, and weld classes K1, K2, K3.

Table 10.3.2-2 — Calculation constants *B*

Class	B N/mm^2
RS	7 890
K1	7 940
K2	6 300
K3	5 040

The three weld classes K1, K2 and K3, allocated to the usual welded joints for piping in accordance with their notch effect shall be as illustrated in Table 10.3.2-4.

NOTE 2 For additional applications, see EN 13445-3.

NOTE 3 The allowable number of load cycles for class RS is derived from hot rolled material with a roughness of 200 µm and cold rolled material.

The notional endurance limit, 2 σ_{aD}, is assumed to be $N = 2 \times 10^6$. With stress amplitudes 2 σ_a^* below the values 2 σ_{aD} in accordance with Table 10.3.2-3 endurance may be assumed.

Table 10.3.2-3 — Endurance limit values 2 σ_{aD}

Class	2 σ_{aD} N/mm^2
RS	125
K1	63
K2	50
K3	40

If pressure fluctuations of differing range and differing frequency occur (operating load collective), the allowable fatigue life shall be determined using the linear damage accumulation law

$$\sum_{i=1}^{k} \frac{N_i}{N_{i\ \text{all}}} = \left(\frac{N_1}{N_{1\ \text{all}}} + \frac{N_2}{N_{2\ \text{all}}} + \cdots + \frac{N_k}{N_{k\ \text{all}}} \right) \leq 1{,}0 \qquad (10.3.2\text{-}7)$$

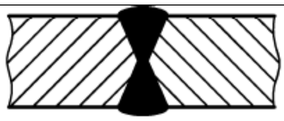
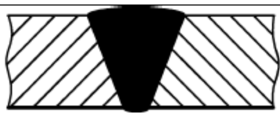
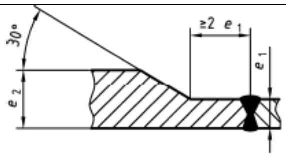
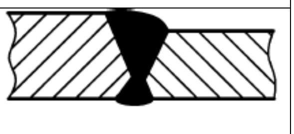
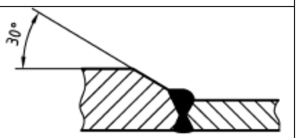
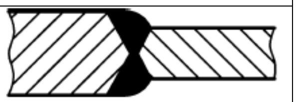
N_1, N_2, ...N_k are the numbers of load cycles to be expected in operation, the load cycles with the same pressure fluctuation range $\left(\hat{p} - \check{p}\right)$ being combined in each case. The related allowable number of load cycles $N_{1\ \text{all}}$, $N_{2\ \text{all}}$, ...$N_{k\ \text{all}}$ shall be taken from Figure 10.3.2-4 with the relevant stress amplitude 2 σ_a^* in accordance with equation (10.3.2-1) in the corresponding load cycle curves, or shall be calculated in accordance with equation (10.3.2-6).

If an operating load collective gives rise to stress amplitudes 2 σ_a* which are smaller than the endurance limits 2 σ_{aD} given in Table 10.3.2-3 for N > 2 x 10^6, the related allowable numbers of load cycles N_{all} = 2 x 10^6 shall be taken. The damage portions of collective stages whose stress amplitude 2 σ_a* is smaller than 50 % of the 2 σ_{aD} values may be ignored.

10.3.2.4 Stress factors η for structural forms

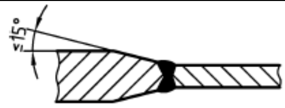
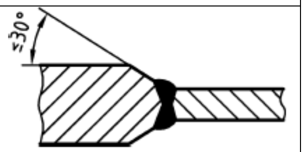
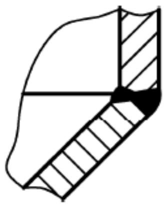
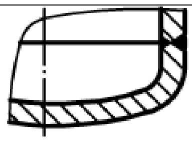
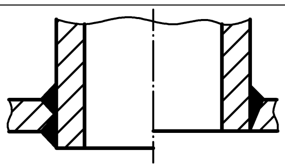
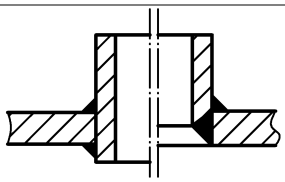
Examples of structural forms and welded joints with the corresponding classes (RS, K1, K2, K3) and the corresponding stress factor η are given in Table 10.3.2-4.

Table 10.3.2-4 — Stress factors η for structural forms

No	Diagram	Description	Conditions	Class	η
1. Cylindrical and conical shells					
1.1		Circumferential weld between walls of equal thickness	Welded from both sides	K1	1,3 [a]
1.2			Welded from one side with backing weld	K1	
1.3			Welded from one side without backing weld	K2	
1.4		Circumferential weld between walls of unequal thicknesses	Welded from both sides	K1	1,5 [a b]
1.5				K1	
1.6				K1	
1.7			Welded from both sides, edge offset, same inside and outside	K1	

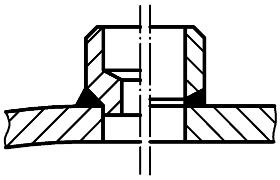
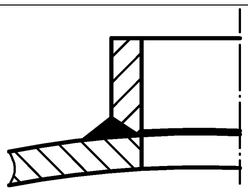
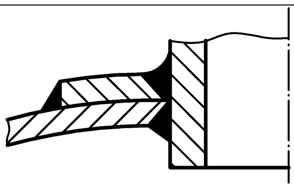
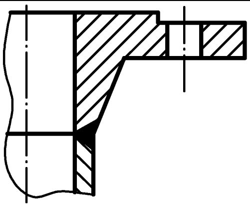
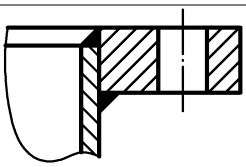
(continued)

Table 10.3.2-4 *(continued)*

No	Diagram	Description	Conditions	Class	η
1.8				K1	
1.9				K1	
1.10	see 1.1, 1.2 + 1.3	Longitudinal joint between walls of equal thicknesses	See No. 1.1, 1.2 and 1.3	see 1.1 1.2 1.3	1,6 [c] [b]
1.11		Cone with angle joint	Welded from both sides or from one side with backing weld	K1	2,7
1.12			Welded from one side without backing weld	K3	
1.13		Cone with knuckle and longitudinal weld	Form of weld and class allocation as No. 1.1 to –1.3	-	2,0
1.14		Weld between a cylinder and a dished end with cylindrical flange	Form of weld and class allo-cation as No. 1.1 to 1.9	-	1,5
1.15		Knuckle of dished end	Unwelded	RS	2,5
2. Nozzles					
2.1		Set-through or set-in nozzle	Full penetration welded from one or both sides or through-welded from one side with backing weld	K1	3,0
2.2			Full penetration welded from one side without backing weld	K2	
2.3		Set-through nozzle	Welded from both sides, but not fully penetrating	K2	
2.4		Set-in nozzle		K3	

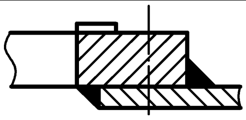
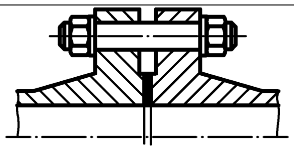
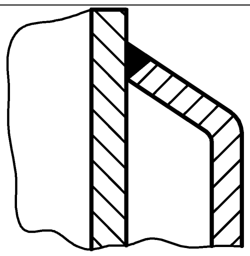
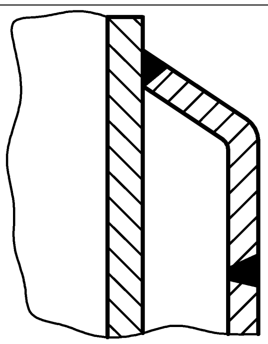
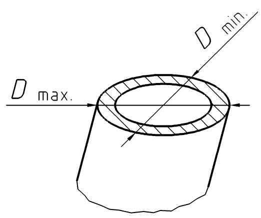
(continued)

Table 10.3.2-4 *(continued)*

No	Diagram	Description	Conditions	Class	η
2.5		Set-on nozzle	Full penetration welded from one side (without residual gap), nozzle bored or root ground over	K1	
2.6			Through-welded from one side without backing weld or without machining of the root	K2	
2.7		Nozzle with disk-shaped reinforcement weld: on the outside diameter of the disk		K3	3,0
2.8		Nozzle with disk-shaped reinforcement weld: nozzle-to-plate weld	Full-penetration welds for connection of nozzle to pad-weld or nozzle to shell weld	K1	
3. Flanges					
3.1		Welding neck flange	Welded from both sides or welded from one side with backing weld	K1	2,0
3.2			Welded from one side without backing weld	K2	
3.3		Welded slip-on flange	Fillet-weld throat at least 5mm	K2	3,0

(continued)

Table 10.3.2-4 *(continued)*

No	Diagram	Description	Conditions	Class	η
3.4		Set-on-pad; weld on inside diameter (in diagram left hand weld)		K3	4,0
		Set-on-pad; weld on outside diameter (in diagram right hand weld)		K2	
3.5		Bolts for flanged joints: verification normally only necessary if the bolts are frequently loosened. In such cases the values in brackets apply		(RS)	(5,0)
4. Double-shell connecting welds					
4.1		With formed knuckle: the evaluation applies both for the inner pipe as well as for the connecting weld itself	Through-welded from one side	K2	3,0
4.2		With separate knuckle : The assessment applies both for the inside vessel wall and for the connecting weld between the knuckle and the inner pipe wall. (The connecting weld between the knuckle and the outer shell is assessed according to serial No 1.3 with K2	Through-welded from both sides or through-welded from one side with backing weld	K1	
5. Pipes and bends with out-of-roundness					
5.1		Seamless or welded pipe with out-of-roundness	Weld form and classification as no. 1.1 to 1.3		see Table 10.3.2-5

(continued)

Table 10.3.2-4 *(continued)*

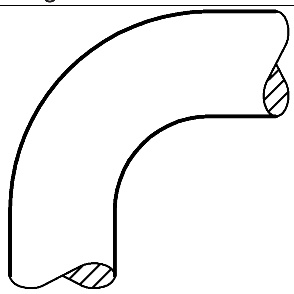
No	Diagram	Description	Conditions	Class	η
5.2		Seamless or welded bend with out-of-roundness	Weld form and classification as no. 1.1 to 1.3		see Table 10.3.2-5
[a] For maximum allowable weld offset $h \leq 0,5\ e$ with maximum of 3 mm according to EN ISO 5817, rating group B [b] For additional limits for weld offsets see EN 13480-4. [c] For maximum allowable weld offset $h \leq 0,1\ e$ according to EN ISO 5817, rating group B					

Table 10.3.2-5 — η-values for pipes and bends with out-of-roundness

	e/D_m						
u %	0,005	0,01	0,025	0,05	0,10	0,15	0,20
1 %	1,16	1,27	1,35	1,25	1,14	1,10	1,07
2 %	1,31	1,54	1,69	1,51	1,29	1,20	1,15
3 %	1,47	1,81	2,04	1,76	1,43	1,29	1,22
4 %	1,63	2,08	2,39	2,02	1,57	1,39	1,30
5 %	1,78	2,35	2,74	2,27	1,72	1,49	1,37
6 %	1,94	2,62	3,08	2,52	1,86	1,59	1,44
7 %	2,09	2,89	3,43	2,78	2,00	1,69	1,52
8 %	2,25	3,16	3,78	3,03	2,15	1,78	1,59
9 %	2,41	3,43	4,13	3,28	2,29	1,88	1,67
10 %	2,56	3,70	4,47	3,54	2,43	1,98	1,74

D = outside diameter of pipe/bend

$D_m = (D_{max} + D_{min})/2$

e = wall thickness

u = out –of roundness (%) with $u = 100\ (D_{max} - D_{min})/D_m$

NOTE Values not tabulated may be determined by linear interpolation.

10.3.2.5 Design

The fatigue life of cyclically loaded components depends largely on the dimensioning and design. Designs with high stress or strain concentration shall be avoided, e.g. by means of a stress-flow related design for cross-section transitions. Table 10.3.2-4 contains an assessment of frequently used weld details in piping systems.

NOTE 1 In the case of rigorous requirements regarding fatigue life, the weld designs of class K1 are recommended. The possibility of testing in accordance with 10.3.2.7 should be taken into account in the design. To assess the fatigue life of designs not given in Table 10.3.2-4, the anticipated value of η should be fixed by means of corresponding estimates via the structural stress multiplied by the notch factor.

NOTE 2 The fatigue life may be increased within the framework of the design evaluation in accordance with Table 10.3.2-4 by, for example, the following design measures :

a) avoidance of set-on reinforcement pads;

b) tapered transition between piping sections with unequal diameters and/or thicknesses;

c) full penetration welding, welding from two sides or one-side welding with backing strip.

NOTE 3 Over-dimensioning for predominantly static load also leads to a greater number of allowable load cycles.

10.3.2.6 Manufacture

For the manufacture of piping components, EN 13480-4 applies. In addition, for piping components calculated in accordance with clause 10 the following requirements shall be met.

NOTE In the case of cyclic loading, defects arising during production have a more unfavourable effect than with static loading. The fatigue life of components can be considerably shortened by notches and unfavourable residual stresses.

Special requirements shall be imposed on the form of welds. Residual welding stresses shall be minimized by heat control during welding and the welding sequence. All heat treatments shall be performed in accordance with EN 13480-4.

10.3.2.7 Testing

10.3.2.7.1 General

For the testing before, during and after manufacture, the following sub-clauses shall be observed in addition to the requirements of EN 13480-5.

For testing during operation, see Annex F.

10.3.2.7.2 Initial testing - Design review

Within the context of the design review, the points which shall be tested shall be established with regard to cyclic loading in the tests described under 10.3.2.7.3.

10.3.2.7.3 Testing during fabrication and final inspection

The testing to be performed during fabrication or within the framework of the final inspection shall ensure that there are no imperfections present in the piping parts which could grow rapidly in size with cyclic loading, and which could result in a failure of the pressure-containing parts before the allowable number of load cycles have been achieved.

For the non-destructive examination, the requirements of EN 13480-5 shall apply.

10.3.2.8 Consideration of special operating conditions

When the following are likely to occur, corrosion-induced crack formation, fatigue crack corrosion, strain-induced crack corrosion, hydrogen-induced crack formation in compressed hydrogen, or in the magnetite protective layer, additional provisions (e.g. calculations) shall be applied as appropriate.

10.4 Fatigue design for thermal gradients

10.4.1 General

The design of piping shall consider of the effects of through-wall thermal gradients and rapid changes in metal temperature.

Where practical, the design of piping systems shall be such that excessive thermal gradients and thermal shocks are avoided. Where this cannot be achieved, the design shall incorporate detailed features to minimise stress concentrations in areas of high thermal gradients. Design guidance is given in 10.4.2.

Where the additional stresses generated by thermal gradients or rapid changes in temperature are considered to be significant, detailed analysis shall be performed to determine the effect of these stresses (alone or in combination with others) on the fatigue life of the component.

10.4.2 Design guidance

For normal operations, the rate of temperature rise or fall in piping systems is generally governed by considerations other than the pipe or component dimensioning.

NOTE 1 Whilst this cannot be regarded as a guarantee of performance, rates of temperature change at start up or shut down up to 2°C/min have generally been acceptable for ferritic materials up to 125 mm in thickness.

NOTE 2 The effects of thermal gradients are generally most severe at branch connections. Insulation of pipe runs, nozzles and branch lines will reduce thermal gradients in these regions. Providing a radius to the branch hole on the inner surface of the run pipe and a smooth profile to any welding will reduce the effects of thermal gradients.

NOTE 3 Thermal shock can arise when fluids of different temperatures are mixed, and consideration should be given to providing a sufficient volume to ensure that mixing is rapid and away from the pipe wall. The use of non pressure-retaining thermal sleeves should also be considered to minimize shock to the piping.

NOTE 4 Condensate and condensate dripping in high temperature piping can be a serious source of thermal shock and reduced fatigue life. Adequate drainage arrangements should be made to avoid any accumulation of condensate. Where the layout makes it possibe for condensate to drip into piping (e.g. from a branch with low or no flow) the incorporation of insulated horizontal runs close to the connection should be considered.

10.5 Fatigue design for combined loads

Where a fatigue analysis for thermal loads in combination with pressure load is required, the method in EN 12953-3 may be used. Where piping is additionally subject to significant cyclic external mechanical loads, or where reduced conservatism over 10.3.2 is required, the analysis shall be in accordance with 12.4.

11 Integral attachments

11.1 General

Integral attachments are forged attachments or attachments welded on the pressure-loaded wall of a straight pipe which transfer piping loadings to the steel framework or concrete.

NOTE No major discontinuity, either with regard to the geometry and/or with regard to the material, should be closer to the attachment than $2{,}5\left(\frac{e_\mathrm{n}D_\mathrm{m}}{2}\right)^{0{,}5}$ where the dimensions of the pipe apply. The material of integral attachments should be chosen in such a way that no major difference exists with respect to the pipe material, the thermal expansion coefficient and the modulus of elasticity. Furthermore, the design stress of the attachment should be similar to that of the pipe material. If major deviations occur, special attention should be paid to choosing the appropriate design stress.

For piping operating in the creep range, it is highly recommended to use the same material for the integral attachment as for the pipe, welds shall be full penetration welds.

Attachments with small lever arms may be designed in shear only when the shear stress is similar to the bending stress. If they are welded to thin walled pipes with $D_m/e_n \geq 10$ bending stresses in the pipe wall shall be determined and assessed.

The calculation of hollow circular attachments is described in 11.4 and 11.6, for calculation of rectangular attachments, see 11.5 and 11.6.

Loads on the attachments cause stresses in the pipe wall. Equations to determine these stresses are given in 11.4 and 11.5. The attachment stresses are then added to the piping system stresses at the attachment. The piping system stresses are determined for straight pipe. The equations, including the attachment stress terms, are given in 11.6.

There are additional equations given in 11.4 and 11.5 for attachments that shall also be checked for attachment stresses. These are based on the absolute values of maximum loads occurring simultaneously for all specified service loading conditions.

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:

$P_m + P_b + P_L \leq 1{,}5\, f_h$ in case of sustained loads;

$P_m + P_b + P_L \leq 1{,}8\, f_h$ in case of sustained and occasional loads;

$P_m + P_b + P_L \leq 2{,}7\, f_h$ in case of exceptional loads;

$Q \leq f_a$ in case of restrained thermal expansion of the piping system;

$P_m + P_b + P_L + Q \leq f_h + f_a$ 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, f_h see equations (12.1.3.1) to (12.1.3.4), 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*.

11.3 Symbols

For the purposes of 11.4 to 11.6, the symbols given in Table 11.3-1 shall apply in addition to those given in 3.2.

Table 11.3-1 — Additional symbols for the purposes of 11.4 to 11.6

Symbol	Description	Unit
A_m	half cross section area of circular hollow attachment	mm²
A_t	cross section area of circular hollow attachment/rectangular attachment	mm²
A_w	total fillet weld throat area	mm²
d_i	attachment inside diameter for circular hollow attachment	mm
d_o	attachment outside diameter for circular hollow attachment	mm
D_o	outside diameter of run pipe	mm
e_n	nominal run pipe wall thickness	mm
$e_{n,t}$	nominal attachment wall thickness	mm
f	design stress (see 5.2)	MPa (N/mm²)
f_a	allowable stress range (see 12.1.3)	MPa (N/mm²)
f_{cr}	design stress in the creep range (see 5.3)	MPa (N/mm²)
f_h	allowable stress at maximum metal temperature (see 12.1.3)	MPa (N/mm²)
L_1	half length of attachment in circumferential direction of the run pipe for rectangular attachment	Mm
L_2	half length of attachment in longitudinal direction of the run pipe for rectangular attachment	Mm
M_L	longitudinal bending moment applied to the attachment (vector normal to the attachment and run pipe centre line)	N mm
M_N	circumferential bending moment applied to the attachment (vector parallel to the run pipe centre line)	N mm
M_T	torsional moment applied to the attachment (vector normal to the run pipe centre line)	N mm
p_c	calculation pressure	MPa (N/mm²)
Q_1	circumferential shear load applied to the attachment	N
Q_2	longitudinal shear load applied to the attachment	N
R_m	mean radius of run pipe	Mm
W	thrust load applied to the attachment (vector normal to the run pipe centre line)	N
Z	section modulus of run pipe	mm^3
Z_t	Section modulus of hollow attachment	mm^3
Z_{WL}	section modulus of fillet or partial penetration weld about the neutral axis of bending parallel to L_2	mm^3
Z_{WN}	section modulus of fillet or partial penetration weld about the neutral axis of bending parallel to L_1	mm^3
Z_{WT}	Torsional section modulus of fillet or partial penetration weld for torsional loading	mm^3
M_L, M_N, M_T, Q_1, Q_2 and W are determined at the surface of the pipe, associated to the load cases. M_L**, M_N**, M_T**, Q_1**, Q_2** and W** are absolute values of maximum loads occuring simultaneously at the surface of the pipe under all loading conditions.		

11.4 Hollow circular attachments

11.4.1 Limitations

The attachment shall be welded to the pipe by a full penetration weld or a fillet weld along the entire outside circumference (see Figure 11.4.1-1).

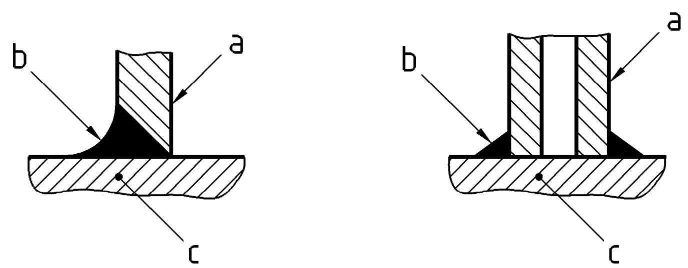
The axis of the attachment shall be normal to the run pipe.

The parameters, calculated in 11.4.2 shall conform to the following limitations:

$$4{,}0 \le \gamma \le 50{,}0 \quad (11.4\text{-}1)$$

$$0{,}2 \le \tau \le 1{,}0 \quad (11.4\text{-}2)$$

$$0{,}3 \le \beta \le 1{,}0 \quad (11.4\text{-}3)$$



Full penetration weld   Fillet weld

Key

a attachment

b weld

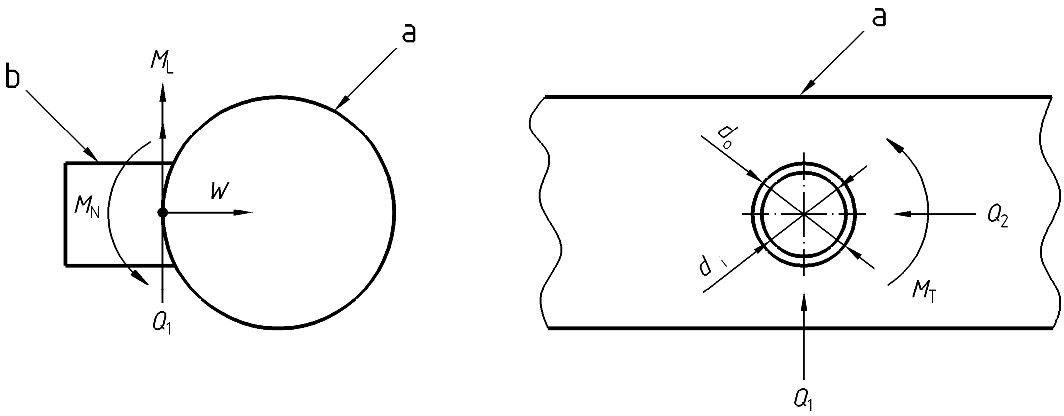
c pipe wall

Figure 11.4.1-1 — Hollow circular attachment welds

11.4.2 Preliminary calculations

M_L, M_N, M_T, Q_1, Q_2, and W are determined at the surface of the pipe associated to the relevant load cases. M_L**, M_N**, M_T**, Q_1**, Q_2** and W** are absolute values of maximum loads occurring simultaneously under all specified service loading conditions.

The dimensions d_i and d_o are defined in Figure 11.4.2-1.



Key

(a) pipe

(b) attachment

Figure 11.4.2-1 — Loading and dimensions of attachments

$$A_t = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) \tag{11.4.2-1}$$

$$Z_t = 2 \cdot I_t / d_o \tag{11.4.2-2}$$

$$I_t = (\pi / 4)\left[(d_o / 2)^4 - (d_i / 2)^4\right] \tag{11.4.2-3}$$

$$A_m = A_t / 2 \tag{11.4.2-4}$$

$$\gamma = D_o / (2e_n) \tag{11.4.2-5}$$

$$\tau = e_{n,t} / e_n \tag{11.4.2-6}$$

$$\beta = d_o / D_o \tag{11.4.2-7}$$

$$C = A_o (2\gamma)^{n_1} \beta^{n_2} \tau^{n_3} \text{, but not less than 1,0} \tag{11.4.2-8}$$

$$J = \min \left\{ Z_T ; \pi \left(\frac{d_o}{2} \right)^2 e_n \right\} \tag{11.4.2-9}$$

The equation (11.4.2-8) shall be used to calculate C_W, C_L and C_N using factors given in Table 11.4.2-1. The maximum values of C_W, C_L and C_N, calculated for the pipe and the attachment, shall be subsequently used.

Table 11.4.2-1 — Factors for hollow circular attachments

Index	part	β range	A_o	n_1	n_2	n_3
C_W	pipe	0,3 to 1,0	1,40	0,81	a	1,33
	attachment	0,3 to 1,0	4,00	0,55	b	1,00
C_L	pipe	0,3 to 1,0	0,46	0,60	-0,04	0,86
	attachment	0,3 to 1,0	1,10	0,23	-0,38	0,38
C_N	pipe	0,3 to 0,55	0,51	1,01	0,79	0,89
	attachment	0,3 to 0,55	0,84	0,85	0,80	0,54
C_N	pipe	>0,55 to 1,0	0,23	1,01	-0,62	0,89
	attachment	>0,55 to 1,0	0,44	0,85	-0,28	0,54

[a] replace β^{n_2} with $e^{-1,2\beta^3}$

[b] replace β^{n_2} with $e^{-1,35\beta^3}$

$C_T = 1,0$ for $\beta \leq 0,55$ (11.4.2-10)

$C_T = C_N$ for $\beta = 1,0$, but not less than 1,0 (11.4.2-11)

C_T should be linearly interpolated for $0,55 < \beta < 1,0$, but not less than 1,0.

$B_W = 0,5\ C_W$, but not less than 1,0 (11.4.2-12)

$B_L = 0,5\ C_L$, but not less than 1,0 (11.4.2-13)

$B_N = 0,5\ C_N$, but not less than 1,0 (11.4.2-14)

$B_T = 0,5\ C_T$, but not less than 1,0 (11.4.2-15)

$K_T = 2,0$ for fillet welds

$K_T = 1,8$ for full penetration, or partial penetration welds.

NOTE Fillet welds and partial penetration welds are not allowed for pipes in the creep range.

11.4.3 Analysis of attachments welded to pipe with a full penetration weld

The stresses σ_{MT}, σ_{NT}, and σ_{NT}** shall be calculated as follows.

$$\sigma_{MT} = \frac{B_W W}{A_t} + \frac{B_N M_N}{Z_t} + \frac{B_L M_L}{Z_t} + \frac{Q_1}{A_m} + \frac{Q_2}{A_m} + \frac{B_T M_T}{J} \quad (11.4.3\text{-}1)$$

$$\sigma_{NT} = \frac{C_W W}{A_t} + \frac{C_N M_N}{Z_t} + \frac{C_L M_L}{Z_t} + \frac{Q_1}{A_m} + \frac{Q_2}{A_m} + \frac{C_T M_T}{J} \quad (11.4.3\text{-}2)$$

$$\sigma_{PT} = K_T \sigma_{NT} \quad (11.4.3\text{-}3)$$

$$\sigma_{NT}^{**} = \frac{C_W W^{**}}{A_t} + \frac{C_N M_N^{**}}{Z_t} + \frac{C_L M_L^{**}}{Z_t} + \frac{Q_1^{**}}{A_m} + \frac{Q_2^{**}}{A_m} + \frac{C_T M_T^{**}}{J} \qquad (11.4.3\text{-}4)$$

11.4.4 Analysis of attachments welded to pipe with fillet or partial penetration weld

The following additional equations shall be satisfied.

$$\frac{W^{**}}{A_W} + \frac{M_L^{**}}{Z_W} + \frac{M_N^{**}}{Z_W} + \frac{\sqrt{Q_1^{**2} + Q_2^{**2}}}{A_W} + \frac{M_T^{**}}{Z_{WT}} \leq 2R_{eHt} \qquad (11.4.4\text{-}1)$$

$$\sqrt{\left(\frac{W^{**}}{A_W}\right)^2 + 4\left(\frac{Q_1^{**} + Q_2^{**}}{A_W} + \frac{M_T^{**}}{Z_{WT}}\right)^2} \leq R_{eHt} \qquad (11.4.4\text{-}2)$$

NOTE Fillet welds and partial penetration welds are not allowed for pipes in the creep range.

11.5 Rectangular attachments

11.5.1 Limitations

The attachment shall be welded to the pipe by:

— a full penetration weld along the two long sides of the attachment; or

— a fillet or partial penetration weld along four sides of the attachment; or

— a fillet or partial penetration weld along the two long sides of the attachment, where the length of the long side is at least three times the length of the short side in absence of fatigue.

The parameters, calculated in 11.5.2 shall conform to the following limitations:

$$\beta_1 \leq 0{,}5 \qquad (11.5.1\text{-}1)$$

$$\beta_2 \leq 0{,}5 \qquad (11.5.1\text{-}2)$$

$$\beta_1 \, \beta_2 \leq 0{,}075 \qquad (11.5.1\text{-}3)$$

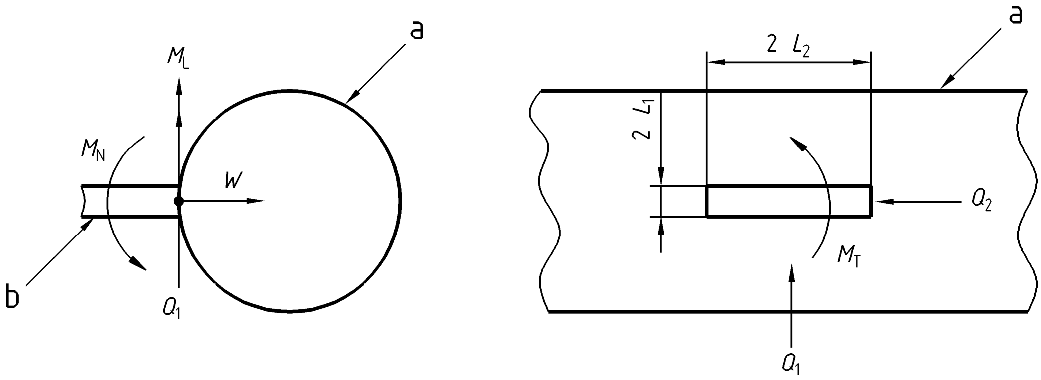
and

$$D_o \, / \, e_n \leq 100 \qquad (11.5.1\text{-}4)$$

11.5.2 Preliminary calculations

M_L, M_N, M_T, Q_1, Q_2, and W are determined at the surface of the pipe associated to the relevant load cases. M_L^{**}, M_N^{**}, M_T^{**}, Q_1^{**}, Q_2^{**} and W^{**} are absolute values of maximum loads occurring simultaneously under all specified service loading conditions.

The dimensions L_1 and L_2 are defined in Figure 11.5.2-1.



Key

(a) pipe

(b) attachment

Figure 11.5.2-1 — Loading and dimensions of attachments

The following preliminary calculations shall be made (L_1 and L_2 are shown in Figure 11.5.2-1):

$$\gamma = R_m / e_n \tag{11.5.2-1}$$

$$\beta_1 = L_1 / R_m \tag{11.5.2-2}$$

$$\beta_2 = L_2 / R_m \tag{11.5.2-3}$$

$$L_a = \min(L_2; e_n) \tag{11.5.2-4}$$

$$L_b = \min(L_1; e_n) \tag{11.5.2-5}$$

$$L_c = \min(L_1; L_2) \tag{11.5.2-6}$$

$$L_d = \max(L_1; L_2) \tag{11.5.2-7}$$

Calculate η, X_1 and Y_1 using the factors given in Table 11.5.2-1 for each case (C_T , C_L , C_N) as follows :

$$\eta = -(X_1 \cos\theta + Y_1 \sin\theta) - \frac{1}{A_o}(X_1 \sin\theta - Y_1 \cos\theta)^2 \tag{11.5.2-8}$$

$$X_1 = X_0 + \lg \beta_1 \tag{11.5.2-9}$$

$$Y_1 = Y_0 + \lg \beta_2 \tag{11.5.2-10}$$

NOTE lg X = $\log_{10}$ X (lg 10 = 1).

Table 11.5.2-1 — Factors for rectangular attachments

Index	A_0	θ	X_0	Y_0
C_T	2,2	40°	0	0,05
C_L	2,0	50°	- 0,45	- 0,55
C_N	1,8	40°	- 0,75	- 0,60

Then, with factor η, calculate C_T, C_L, C_N as follows:

$$C_T = 3{,}82(\gamma)^{1{,}64}\,\beta_1\,\beta_2\,\eta^{1{,}54}\text{, but not less than 1,0} \qquad (11.5.2\text{-}11)$$

$$C_L = 0{,}26(\gamma)^{1{,}74}\,\beta_1\,\beta_2^2\,\eta^{4{,}74}\text{, but not less than 1,0} \qquad (11.5.2\text{-}12)$$

$$C_N = 0{,}38(\gamma)^{1{,}90}\,\beta_1^2\,\beta_2\,\eta^{3{,}40}\text{, but not less than 1,0} \qquad (11.5.2\text{-}13)$$

$$B_T = (2/3)C_T\text{, but not less than 1,0} \qquad (11.5.2\text{-}14)$$

$$B_L = (2/3)C_L\text{, but not less than 1,0} \qquad (11.5.2\text{-}15)$$

$$B_N = (2/3)C_N\text{, but not less than 1,0} \qquad (11.5.2\text{-}16)$$

$$A_T = 4\,L_1\,L_2 \qquad (11.5.2\text{-}17)$$

$$Z_{tL} = (4/3)L_1(L_2)^2 \qquad (11.5.2\text{-}18)$$

$$Z_{tN} = (4/3)(L_1)^2\,L_2 \qquad (11.5.2\text{-}19)$$

— K_T = 2,0 for as-welded full penetration welds and fillet or partial penetration welds where the attachment is welded on four sides

— K_T = 3,6 for fillet or partial penetration welds where the attachment is welded on two or three sides

$$M_{TT} = \max\left(\frac{M_T}{L_c\,L_d\,e_n\left[1+(L_c/L_d)\right]};\frac{M_T}{\left[0{,}8+0{,}05(L_d/L_c)\right]L_c^2\,L_d}\right) \qquad (11.5.2\text{-}20)$$

NOTE *Fillet welds and partial penetration welds are not allowed for pipes in the creep range.*

11.5.3 Analysis of attachments welded to pipe with a full penetration weld

The stresses σ_{MT}, σ_{NT}, and σ_{NT}** shall be calculated as follows.

$$\sigma_{MT} = \frac{B_T W}{A_t} + \frac{B_L M_L}{Z_{tL}} + \frac{B_N M_N}{Z_{tN}} + \frac{Q_1}{2L_1 L_a} + \frac{Q_2}{2L_2 L_b} + M_{TT} \qquad (11.5.3\text{-}1)$$

$$\sigma_{NT} = \frac{C_T W}{A_t} + \frac{C_L M_L}{Z_{tL}} + \frac{C_N M_N}{Z_{tN}} + \frac{Q_1}{2L_1 L_a} + \frac{Q_2}{2L_2 L_b} + M_{TT} \qquad (11.5.3\text{-}2)$$

$$\sigma_{PT} = K_T\sigma_{NT} \qquad (11.5.3\text{-}3)$$

$$\sigma_{NT}^{**} = \frac{C_T W^{**}}{A_t} + \frac{C_L M_L^{**}}{Z_{tL}} + \frac{C_N M_N^{**}}{Z_{tN}} + \frac{Q_1^{**}}{2L_1 L_a} + \frac{Q_2^{**}}{2L_2 L_b} + M_{TT}^{**} \qquad (11.5.3\text{-}4)$$

11.5.4 Analysis of attachments welded to pipe with fillet or partial penetration weld

The following additional equations shall be satisfied.

$$\frac{W^{**}}{A_{\mathrm{W}}} + \frac{M_{\mathrm{L}}^{**}}{Z_{\mathrm{WL}}} + \frac{M_{\mathrm{N}}^{**}}{Z_{\mathrm{WN}}} + \frac{2(Q_1^{**} + Q_2^{**})}{A_{\mathrm{W}}} + \frac{M_{\mathrm{T}}^{**}}{Z_{\mathrm{WT}}} \le 2R_{\mathrm{eHt}} \tag{11.5.4-1}$$

$$\sqrt{\left(\frac{W^{**}}{A_{\mathrm{W}}}\right)^2 + 4\left(\frac{Q_1^{**} + Q_2^{**}}{A_{\mathrm{W}}} + \frac{M_{\mathrm{T}}^{**}}{Z_{\mathrm{WT}}}\right)^2} \le R_{\mathrm{eHt}} \tag{11.5.4-2}$$

11.6 Stress analysis of the run pipe

The following modified equations of clause 12 shall be satisfied.

a) For sustained loads

$$\sigma_1 = \frac{p_c\ D_{\mathrm{o}}}{4e_{\mathrm{n}}} + \frac{0{,}75i\ M_{\mathrm{A}}}{Z} + \sigma_{\mathrm{MT}} \le 1{,}5 f_h \text{, and } 0{,}75i \ge 1{,}0 \tag{11.6-1}$$

with:

M_A bending moment due to sustained loads

σ_{MT} additional stress resulting from sustained loads

b) For sustained and occasional loads

$$\sigma_{2\mathrm{a}} = \frac{p_c\ D_{\mathrm{o}}}{4e_{\mathrm{n}}} + \frac{0{,}75i\left(M_{\mathrm{A}} + M_{\mathrm{B}}\right)}{Z} + \sigma_{\mathrm{MT}} \le 1{,}8 f_h \text{, and } 0{,}75i \ge 1{,}0 \tag{11.6-2}$$

with:

M_B bending moment due to occasional loads

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

c) For exceptional loads

$$\sigma_{2\mathrm{b}} = \frac{p_c\ D_{\mathrm{o}}}{4e_{\mathrm{n}}} + \frac{0{,}75i\left(M_{\mathrm{A}} + M_{\mathrm{B}}\right)}{Z} + \sigma_{\mathrm{MT}} \le 2{,}7 f_h \text{, and } 0{,}75i \ge 1{,}0 \tag{11.6-3}$$

with:

M_B bending moment due to exceptional loads

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

The stress range σ_3, due to the resultant moment, M_c, from thermal expansion and alternate loads, e.g. seismic loads, shall either satisfy the following equation:

d) For loads caused by restrained thermal expansion

$$\sigma_3 = \frac{i\ M_{\mathrm{C}}}{Z} + \frac{\sigma_{\mathrm{PT}}}{2} \le f_a \tag{11.6-4}$$

with:

σ_{PT} additional stress resulting from restrained thermal expansion

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

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

$$\sigma_4 = \frac{p_c\ D_\text{o}}{4e_\text{n}} + 0{,}75i\frac{M_\text{A}}{Z} + i\frac{M_\text{C}}{Z} + \sigma_\text{MT} + \frac{\sigma_\text{PT}}{2} \leq f_h + f_a \text{ , and } 0{,}75i \geq 1{,}0 \qquad (11.6\text{-}5)$$

with:

σ_{MT} additional stress resulting from sustained loads

σ_{PT} additional stress resulting from restrained thermal expansion

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

$$\sigma_5 = \frac{p_c D_o}{4e_n} + 0{,}75\left[\frac{i \cdot M_A}{Z} + \frac{i \cdot M_c}{3Z}\right] + \sigma_{MT} + \frac{\sigma_{PT}}{2} \leq 1{,}25 f_{cr} \text{ , and } 0{,}75i \geq 1{,}0$$
(11.6-5a)

with:

σ_{MT} additional stress resulting from sustained loads

σ_{PT} additional stress resulting from restrained thermal expansion

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

$$\sigma^{**}_\text{NT} \leq 2R_\text{eHt} \qquad (11.6\text{-}6)$$

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

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

11.7 Shear stress analysis in attachment

11.7.1 Hollow circular attachments

For time independent design:

$$\frac{\sqrt{\left(Q_1^{**}\right)^2 + \left(Q_2^{**}\right)^2}}{A_m} + \frac{M_T^{**}}{J} \leq R_{eHt} \qquad (11.7.1\text{-}1)$$

For attachments in the creep range: limitation of the shear stress of the hollow circular attachment (less than or equal to mean value of (shear-) creep rupture strength according to von Mises hypothesis):

$$\frac{\sqrt{\left(Q_1^{**}\right)^2 + \left(Q_2^{**}\right)^2}}{A_m} + \frac{M_T^{**}}{J} \leq \frac{1{,}25}{\sqrt{3}} \cdot f_{cr} \tag{11.7.1-1a}$$

11.7.2 Rectangular attachments

For time independent design:

$$\sqrt{\left(\frac{Q_1^{**}}{2L_1L_a}\right)^2 + \left(\frac{Q_2^{**}}{2L_2L_b}\right)^2} + M_{TT}^{**} \leq R_{eHt} \tag{11.7.2-1}$$

For attachments in the creep range: limitation of the shear stress of a rectangular attachment (less than or equal to mean value of (shear-) creep rupture strength according to von Mises hypothesis):

$$\sqrt{\left(\frac{Q_1^{**}}{2L_1L_a}\right)^2 + \left(\frac{Q_2^{**}}{2L_2L_b}\right)^2} + M_{TT}^{**} \leq \frac{1{,}25}{\sqrt{3}} \cdot f_{cr} \tag{11.7.2-1a}$$

11.8 Alternative calculation methods

If the method described in the clause 11 gives no satisfactorily results, or in case of non-compliance with the given geometric limits of 11.4.1 and 11.5.1, alternative calculation methods shall be applied.

NOTE Some of these methods are described in a selection of well known literature [3], [4], [5] and PD 5500. It is the responsibility of the designer, to classify the stresses into the categories of primary, secondary and peak stresses, and to limit it to the corresponding allowable stress as given in 12.2.

12 Flexibility analysis and acceptance criteria

12.1 Basic conditions

12.1.1 General

In addition to the design requirements for pressure given in clauses 6 to 11, piping systems shall be designed to withstand the effects of weight and other loadings and shall be analysed for the effects of thermal expansion or contraction or to similar movements imposed by other sources. The influence of axial forces caused by internal pressure and bellows rigidities shall be considered when using unrestrained expansion joints to avoid buckling of the line. This clause deals with the stress analysis to be completed and the corresponding acceptance criteria to be met in order to achieve these requirements.

12.1.2 Loading conditions

The loading conditions to be considered are given in clause 4.

12.1.3 Allowable stresses

12.1.3.1 The basic allowable stresses are given in clause 5.

12.1.3.2 The allowable stress range f_a shall be given by:

$$f_a = U\left(1{,}25 f_c + 0{,}25 f_h\right)\frac{E_h}{E_c} \tag{12.1.3-1}$$

where

E_c is the value of the modulus of elasticity at the minimum metal temperature consistent with the loading under consideration;

E_h is the value of the modulus of elasticity at the maximum metal temperature consistent with the loading under consideration;

f_c is the basic allowable stress at minimum metal temperature consistent with the loading under consideration

$$f_c = \min\left(\frac{R_m}{3}\,;\,f\right) \tag{12.1.3-2}$$

where f is calculated according to 5.2 at room temperature.

f_h is the allowable stress at maximum metal temperature consistent with the loading under consideration

$$f_h = \min\left(f_c; f; f_{CR}\right) \tag{12.1.3-3}$$

where

f is calculated according to 5.2;

f_{CR} is calculated according to 5.3 at calculation temperature t_C.

U is the stress range reduction factor (see NOTE 1) taken from Table 12.1.3-1 or calculated from equation 12.1.3-3 (see NOTE 2):

$$U = 6{,}0N^{-0{,}2} \leq 1{,}0 \tag{12.1.3-4}$$

where

N is the number of equivalent full amplitude cycles during the expected service lifetime of the piping system (see NOTE 3).

NOTE 1 U applies essentially to non-corroded piping. Corrosion can sharply decrease cyclic lifetime. Therefore, corrosion resistant materials should be considered where a large number of major stress cycle are anticipated.

NOTE 2 Equation (12.1.3-3) does not apply beyond approximately 2×10^6 cycles. Selection of U factors beyond 2×10^6 cycles should be the designer's responsibility.

NOTE 3 The designer is cautioned that the fatigue lifetime of material operated at elevated temperatures may be reduced.

NOTE 4 12.1.3.2 is not applicable for expansion joints

If the range of temperature change varies, equivalent full temperature cycles shall be as follows:

$$N = N_E + \sum_{i=1}^{n}\left(r_i^5\ N_i\right) \tag{12.1.3-5}$$

where

N_E is the number of cycles at full temperature change Δt_E for which stress from thermal expansion σ_3 (see 12.3.4) has been calculated

N_i is the number of cycles at lesser temperature changes Δt_i

r_i is the ratio of lesser temperature changes to that for any which the stress σ_3 has been calculated $\Delta t_i/\Delta t_E$.

Table 12.1.3-1 — Stress range reduction factors

Number of equivalent full temperature cycles *N*	Factor *U*
N ≤ 7 000	1,0
7 000 < N ≤ 14 000	0,9
14 000 < N ≤ 22 000	0,8
22 000 < N ≤ 45 000	0,7
45 000 < N ≤ 100 000	0,6
100 000 < N	0,5

Table 12.1.3-1 shall only be used specifically for the calculation of allowable stress range f_a and shall not be substituted for any fatigue analysis deemed essential by this European Standard.

12.2 Piping flexibility

12.2.1 General

All piping systems shall have sufficient inherent flexibility to prevent the following during their design lifetime:

a) failure of piping or supports from overstress or fatigue;

b) leakage at any point in the piping;

c) detrimental stresses or distortion in the piping or in-line equipment (e.g. valves) or in connected equipment or plant (e.g. vessels, pumps or turbines) resulting from excessive thrusts and moments in the piping.

12.2.2 Basic conditions

The calculated stress range at any point due to displacements in the system shall not exceed the allowable stress range in accordance with 12.1.3.2.

The calculated movement of the piping shall be within any specified limits and taken into account in the flexibility analysis.

The flexibility of the piping shall be provided to ensure the following:

- the stresses shall not exceed the design limits;
- the computed stress range at any point due to displacements in the system shall not exceed the allowable stress range established in accordance with 12.1.3.2;
- that reaction forces shall not be detrimental to supports or connected equipment;
- the computed movement of the piping shall be within any prescribed limits and properly accounted for in the flexibility calculations.

Adequate flexibility shall be provided by changes in direction along the pipe runs (e.g bends, loops or offsets), by flexible joints (e.g. expansion joints, metal hoses) or other suitable devices.

12.2.3 Displacement strains

12.2.3.1 General

Specific consideration shall be given to displacement strains due to thermal or externally imposed displacements or by displacements of supports:

- Thermal displacements

  A piping system will undergo dimensional changes with any change in temperature. If it is constrained from free expansion or contraction by connected equipment and restraints such as guides and anchors, displacement stresses will be introduced;

- Externally imposed displacements

  Externally caused movement of restraints will impose displacements on the piping in addition to those related to thermal effects. Movements may result from tidal changes (dock piping), wind sway (e.g. piping supported from a tall slender tower), temperature changes in connected equipment, earthquake or other imposed dynamic loadings such as rapid valve closure;

  Movement due to earth settlement shall be taken into account where it is established that such an event has a sustained effect on stresses induced into the piping, or where such settlement can cause detrimental permanent deformation to the piping, either at localised positions or in the form of excessive end reactions.

- Displacement of supports

  If restraints are not considered rigid, it is permissible to incorporate their flexibility in determining the displacement range and reactions.

12.2.3.2 Total displacement strains

Thermal displacements, reaction displacements, and externally imposed displacements all have equivalent effects on the piping system, and shall be considered together in determining the total displacement strains (proportional deformations) in various parts of the piping system.

The effects of longitudinal expansion of the piping due to internal pressure shall be considered. This effect will be partially offset by the Poisson effect of hoop expansion.

12.2.4 Displacement stresses

12.2.4.1 Elastic behaviour

Stresses may be considered proportional to the total displacement strains in a piping system in which the strains are well distributed and not excessive at any point (a balanced system). Layout of systems shall aim for such a condition, which is assumed in flexibility analysis methods provided in this standard.

12.2.4.2 Overstrained behaviour

Stresses cannot be considered proportional to displacement strains throughout a piping system in which an excessive amount of strain may occur in localised portions of the system (an unbalanced system). Operation of an unbalanced system in the creep range may aggravate the deleterious effects due to creep strain accumulation in the most susceptible regions of the system. Imbalance can result from one or more of the following:

— highly stressed small size pipe runs in series with large or relatively stiff pipe runs;

— a local reduction in size or wall thickness or local use of material having reduced yield strength (for example girth welds of substantially lower strength than the base metal);

— a line configuration in a system of uniform size in which the expansion or contraction shall be absorbed largely in a short offset from the major portion of the run;

— variation of piping material or temperature in a line.

Unbalance, leading to localised plastic collapse, shall be avoided or minimised by design and layout of piping systems, particularly those using materials of low ductility. Many of the effects of imbalance can be mitigated by selective use of cold pull. If unbalance cannot be avoided, the designer shall use appropriate analytical methods to assure adequate flexibility.

12.2.5 Stress range

Stresses caused by thermal expansion, when of sufficient initial magnitude, relax in the hot condition as a result of local yielding or creep. A stress reduction takes place and usually appears as a stress of reversed sign when the component returns to the cold condition.

This phenomenon is designated as self-springing of the line and is similar in effect to cold pulling. The extent of self-springing depends on the material, the magnitude of the initial expansion, fabrication stress, the hot service temperature and the elapsed time.

While the expansion stress in the hot condition tends to diminish with time, the sum of the expansion strains for the hot and cold conditions during any one cycle remains substantially constant. This sum is referred to as the strain range. However, to permit convenient association with allowable stress, stress range is selected as the criterion for the thermal design of piping.

The allowable stress range shall be determined in accordance with 12.1.3.2.

When piping has to remain in the elastic range for the whole of its operating lifetime, the allowable stress range shall be determined in accordance with 12.1.3.2 and the sum of all stresses shall not exceed 0,95 of the minimum specified material yield strength.

12.2.6 Cold pull

Cold pull is the intentional elastic deformation of piping during assembly to produce a desired initial displacement and stress. Cold pull is beneficial in that it serves to balance the magnitude of stress under initial and extreme displacement conditions. When cold pull is properly applied, there is less likelihood of overstrain during initial operation. Hence, it is recommended for piping materials of limited ductility. There is also less deviation from installed dimensions during initial operation, so that hangers will not be displaced so far from their original settings.

NOTE When using expansion joints, cold pull is a useful method to optimise the movement capabilities of the joint and to reduce reaction forces and moments.

In as much as the service lifetime of a piping system is affected more by the range of stress variation than by the magnitude of stress at a given time, no account of cold pull shall be permitted in stress range calculations. However, in calculating the thrusts and moments where actual reactions as well as their range of variations are significant cold pull may be taken into account.

The benefits of cold pull shall not exceed 60 % of the original value when calculating reactions.

12.2.7 Properties for flexibility analysis

12.2.7.1 Thermal expansion data

The thermal expansion range can be determined in accordance with Annex G as the difference between the unit expansion shown for the highest metal temperature and that for the lowest metal temperature resulting from operating shut-down conditions.

NOTE For materials not included in Annex G, reference should be made to authoritative source data.

12.2.7.2 Modulus of elasticity

The values of the modulus of elasticity E_t used for flexibility analysis shall be the value taken at the temperature of the piping load under consideration.

The moduli of elasticity can be taken as the values shown in Annex E.

NOTE For materials not included in Annex G, reference should be made to authoritative source data.

12.2.7.3 Poisson's ratio

Poisson's ratio, when required for flexibility analysis, may be taken as 0,3 for all steel types at all temperatures.

12.2.7.4 Flexibility and stress factors

In the absence of more directly applicable data, the flexibility factors and stress intensification factors shown in Annex H, shall be used in flexibility calculations.

NOTE The stress intensification factors in Annex H have been developed from fatigue tests of representative piping components and assemblies manufactured from ductile ferrous materials. The allowable displacement stress range is based on tests of carbon and austenitic stainless steels.

For piping components or attachments (such as valves, strainers, anchor, rings or bands) not covered in Annex H, suitable stress intensification factors may be assumed by comparison of their significant geometry with that of the component shown.

12.2.8 Supporting conditions

Anchors, restraints, supports, pipe hangers and other externally connected control devices shall be provided, as required, in order to ensure the proper function of the expansion absorbing devices according to 12.2.9.

The term "support" comprises piping anchors, rigid supports, (e.g. slides and guides), pipe hangers and connections to components as well as constant/variable spring supports/hangers (see clause 13).

NOTE Supporting conditions describe the possibility of withstanding forces and moments at these supports and should be given in a mathematical model which sufficiently represents the design.

For each support, 12 mathematical functions exist. Where these are described in a suitable three-dimensional system of coordinates u, v, w, there is a direct dependence between the forces and displacements in equal direction on the one hand and between moments and rotation about the same axis on the other hand. A force that is not absorbed results in a displacement in its direction of application, and similarly a moment not absorbed results in a rotation about its axis of action.

The 12 functions are:

— absorption of forces in three directions F_u, F_v, F_w;

— absorption of moments M_u, M_v, M_w of axis u, v, w;

— displacement in three directions δ_u, δ_v, δ_w;

— rotations Φ_u, Φ_v, Φ_w about axis u, v, w.

To clearly determine the supporting conditions of a support, six functions independent of each other should be established.

The possibility of unrestrained displacement in the direction of one of the three axes as well as unrestrained rotation about one of the three axes is called "degree of freedom". Since six degrees of freedom define the possibility of fully unrestrained movement, one support can only have zero to five degrees of freedom.

Some typical supports are:

— anchors: Supports where forces and moments can be withstood in all directions;

— partial restraints : Supports which have 1 degree to 5 degrees of freedom;

— elastic connection of components: Points of attachment, e.g. to a vessel, equipment, pump etc., which allow the absorption of forces and moments depending on component rigidity;

— variable support: Supports which can absorb forces in linear relationship of spring rate and deflection;

— constant supports: Support which provide a constant supporting force throughout the total range of deflections;

— shock arrestors/absorbers and vibration dampers: Shock arrestors/absorbers are devices which in the case of dynamic loading (e.g. hydraulic shock) absorb a force in the direction of movement. Vibration dampers are used to reduce vibration. Shock arrestors/absorbers and vibration dampers will not absorb forces from static loading (e.g. from dead weight, thermal expansion, etc). Sway braces are two-directional acting spring supports with a preset force and a back placing characteristic;

— rigid struts : rigid struts form a subgroup of partial restraints and prevent displacement in the direction of connection between the joints (ball bushings).

The support structures shall be designed to withstand all the loads transmitted by the piping. The design of supports shall be in accordance with clause 13.

12.2.9 Expansion joints

Two categories of expansion joints need to be considered differently because of their different movement characteristics. These two categories are:

— Unrestrained (axial and universal);

— Restrained (angular and lateral).

Unrestrained expansion joints require appropriate anchors and additional guides to avoid buckling of the piping system. Recommended maximum spacings between these guides are given in Annex C.

The system shall be checked either according to the "rigid piping" method, or by analysis in accordance with 12.3 using the joint flexibility given by the manufacturer of the expansion joint.

12.2.10 Flexibility analysis

12.2.10.1 Formal analysis not required

No formal analysis of adequate flexibility shall be required for a piping system which meets one of the following criteria:

— duplicates or replaces without significant change a system operating with a satisfactory service record;

— can readily be judged adequate by comparison with previously analysed systems;

— is of uniform size, has no more than two anchors and no intermediate restraints or other movement controlling devices, and is designed for a service of not more than 7 000 full cycles (or in the case of fuel gas piping, 1 000 full cycles) and satisfies the following empirical equation

$$\frac{D_o Y}{(L-l)^2} \leq 208{,}3 \qquad (12.2.10\text{-}1)$$

where

D_o is the nominal outside diameter of pipe (mm);

L is the developped length of piping between anchors (m);

l is the anchor distance (length of straight line joining anchors) (m);

Y is the resultant of movements to be absorbed by piping (mm).

The formula given is an example of acceptable simplified methods of analysis.

NOTE No general proof can be offered that this equation will yield accurate or consistently conservative results. It was developed for ferrous materials and is not applicable to systems used under severe cyclic conditions. It should be used with caution in configurations such as unequal leg U-bends (L/l > 2,5), or near straight "saw-tooth" runs, or for large diameter thin-wall pipe, or where extraneous displacements (not in the direction connecting anchor points) constitute a large part of the total displacement. There is no assurance that terminal reactions will be acceptably low, even if a piping system falls within the above limitations.

12.2.10.2 Formal analysis required

Any piping system which does not meet the criteria in 12.2.10.1 shall be analysed by a method of analysis which is either simplified or approximate or comprehensive.

A simplified or approximate method may be applied only if used within its limits.

Acceptable comprehensive methods of analysis include analytical and chart methods which provide an evaluation of the forces, moments and stresses caused by displacement strains.

Comprehensive analysis shall take into account stress intensification factors for any component other than straight pipe. Credit may be taken for the flexibility of such a component.

12.2.10.3 Basic assumptions and requirements

12.2.10.3.1 Wherever possible, formal analysis shall be performed on complete systems between anchor points or points where boundary conditions are known. This may include axes of symmetry. Directions of free movements and fixation at supports, shall be simulated in the analysis.

NOTE Care should be taken to see that the designs are faithfully reproduced in construction.

Friction forces shall be considered. Where the effect of friction is considered to be significant, the additional forces shall be taken into account in the piping design.

12.2.10.3.2 Where it is necessary to simplify assumptions for the purposes of reducing the complexity of the flexibility analysis, particulars of such simplification shall be recorded in the design calculations. Where simplified assumptions are used in calculations or model tests, the likelihood of attendant underestimates of forces, moments and stresses, including the effects of stress intensification shall be evaluated.

12.2.10.3.3 The significance of all parts of the piping system to be analysed and of all restraints, such as supports or guides, including intermediate restraints introduced for the purposes of reducing moments and forces on equipment or small branch lines, shall be considered.

12.2.10.3.4 Linear and rotational behaviour of connecting equipment shall be taken into account.

12.2.10.3.5 Flexibility characteristics and stress intensification factors as shown in Annex H shall be used for the appropriate bend, branch, T-piece etc.

12.2.10.3.6 For the purposes of analysis and for denoting piping effects (forces, moments, deflections and rotations), on connecting equipment the system shall have a sign and axes convention.

12.2.10.3.7 Small components which have only little influence on the total rigidity shall be idealised as a beam with cross-sections that approximate to the effective rigidity.

Large components e.g. vessels, may have substantial influence on the total structure. Therefore, it is necessary to idealise the component with its specific rigidities in the model e.g. line of beams to point of support or representation of supporting structure by stiffness matrix.

12.2.10.3.8 The influence of supporting elements shall be considered to the extent required. Regarding the values of reactions and moments, the rigidities of support shall be considered by the model, if required.

12.2.10.3.9 The rigidities of expansion joints shall be considered as internal rigidities (stiffness matrices included directly).

NOTE Acoustic fatigue can occur in a piping system particularly when the natural frequency of the system matches the source frequency. This problem is not addressed in this clause, and specialist advice should be sought where it is considered that it can occur.

12.3 Flexibility analysis

12.3.1 General

The following determination and limitation of stresses shall be used to ensure the safe operation of the piping.

The equations (12.3.2-1) and (12.3.3-1) deal with the longitudinal stresses due to design and operating loadings, and the equations (12.3.4-1) and (12.3.4-2) with the stress range due to such loadings that gives rise to deformation of the total system.

In equation (12.3.5-1), one-third of the stress resulting from thermal expansion and alternating loadings are taken into consideration with respect to the material behaviour in the creep rupture stress range, assuming that two-thirds will be relieved by relaxation.

Equation (12.3.6-1) ensures that in the event of a single non-repeated load, no strain occurs which can adversely affect the material.

Stresses shall be determined for nominal thickness.

NOTE Wall thickness reductions, allowed by the technical conditions of delivery for seamless and welded pipes are covered by the stress limits.

The stress intensification factors, *i*, are given in Tables H-1 and H-2.

As an alternative route to equations given in 12.3.2 to 12.3.6, a more detailed determination of the stresses by separating in-plane and out-of-plane moments can be performed, using the corresponding stress intensity factors in Table H-3.

In this case the factor 0,75 *i* for moment M_A, M_B and M_C in equations (12.3.2-1), (12.3.3-1), (12.3.4-2) and (12.3.5-1) shall be replaced by i_O and i_i respectively, in accordance with Table H-3. In the same way, the factor *i* for moments M_C and M_D in equations (12.3.4-1), (12.3.4-2), (12.3.5-1) and (12.3.6-1) shall be replaced by i_O and i_i.

NOTE The pressure term $\frac{p_c d_o}{4 e_n}$ in the equations (12.3.2-1), (12.3.3-1), (12.3.4-1), (12.3.4-2) and (12.3.5-1) may be replaced by the alternative term $\frac{p_c d_i^2}{d_o^2 - d_i^2} + \frac{p_c}{2}$.

For the general and the alternative route, the stress intensity factors, *i*, including the reduction factor 0,75, if defined, shall be greater than or equal to 1,0 (0,75 $i \geq 1{,}0$). If a value less than 1 is obtained then the value 1,0 shall be used.

12.3.2 Stress due to sustained loads

The sum of primary stresses σ_1, due to calculation pressure, p_c, and the resultant moment, M_A, from weight and other sustained mechanical loads shall satisfy the following equation:

$$\sigma_1 = \frac{p_c d_o}{4 e_n} + \frac{0{,}75\, i\, M_A}{Z} \leq f_f \qquad (12.3.2\text{-}1)$$

where

M_A is the resultant moment from the sustained mechanical loads which shall be determined by using the most unfavourable combination of the following loads:

- piping dead weight including insulation, internals and attachments;
- weight of fluid;
- internal pressure forces due to unrelieved axial expansion joints etc.

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

12.3.3 Stress due to sustained and occasional or exceptional loads

The sum of primary stresses, σ_2, due to internal pressure, p_c, resultant moment, M_A, from weight and other sustained mechanical loads and resultant moment, M_B, from occasional or exceptional loads shall satisfy the following equation:

$$\sigma_2 = \frac{p_c d_o}{4 e_n} + \frac{0{,}75\, i\, M_A}{Z} + \frac{0{,}75\, i\, M_B}{Z} \leq k\ f_f \qquad (12.3.3\text{-}1)$$

where

M_B is the resultant moment from the occasional or exceptional loads which shall be determined by using the most unfavourable combination of the following loads:

- wind loads ($T \leq T_B/10$);
- snow loads;
- dynamic loads from switching operations ($T \leq T_B/100$);
- seismic loads ($T \leq T_B/100$);

f_f is the design stress for flexibility analysis in N/mm^2 (MPa) with $f_f = \min(f; f_{cr})$.

$k = 1$ if the occasional load is acting for more than 10 % in any 24 h operating period, e.g. normal snow, normal wind;

$k = 1{,}15$ if the occasional load is acting for less than 10 % in any 24 h operating period;

$k = 1{,}2$ if the occasional load is acting for less than 1 % in any 24 h operating period, e.g. dynamic loadings due to valve closing/opening, design basis earthquake;

$k = 1{,}3$ for exceptional loads with very low probability e.g. very heavy snow/wind (i.e. = 1,75 × normal);

$k = 1{,}8$ for safe shut-down earthquake;

p_c is the maximum calculation pressure occurring at the considered loading condition, the calculation pressure shall be taken as a minimum.

The effects of anchor displacements due to earthquake may be excluded if they are included in equation (12.3.4-1).

Unless specified otherwise, the following agreements apply:

a) the action time T corresponds to the bracketed values referring to the total operating time T_B;

b) snow and wind loads are not applied simultaneously;

c) loadings with $T \leq T_B/100$ are not applied simultaneously;

12.3.4 Stress range due to thermal expansion and alternating loads

The stress range, σ_3, due to the resultant moment, M_C, from thermal expansion and alternating loads, e.g. seismic loads, shall either satisfy the following equation:

$$\sigma_3 = \frac{i\,M_C}{Z} \leq f_a \qquad (12.3.4\text{-}1)$$

or where the conditions of equation (12.3.4-1) are not met, the sum of stresses, σ_4, due to calculation pressure p_c, resultant moment, M_A, from sustained mechanical loads and the resultant moment, M_C, from thermal expansion and alternating loads shall satisfy the following equation:

$$\sigma_4 = \frac{p_c\,d_o}{4e_n} + \frac{0{,}75\,i\,M_A}{Z} + \frac{i\,M_C}{Z} \leq f_f + f_a \qquad (12.3.4\text{-}2)$$

where

M_C is the range of resultant moment due to thermal expansion and alternating loads which shall be determined from the greatest difference between moments using the moduli of elasticity at the relevant temperatures.

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

Particular attention shall be given to:

— longitudinal expansion, including terminal point movements, due to thermal expansion and internal pressure;

— terminal point movements due to earthquake if anchor displacement effects were omitted from equation (12.3.3-1);

— terminal point movements due to wind;

— frictional forces;

The condition of the piping during shut-down shall be also considered. Cold pull, if any, applied during installation shall not be taken into account, i.e. the operating case pertinent to M_C shall be so designed as if no cold pulling was applied.

12.3.5 Additional conditions for the creep range

For piping operating within the creep range, the stresses, σ_5, due to calculation pressure, p_c, resultant moment, M_A, from weight and other sustained mechanical loadings, and to resultant moment, M_C, from thermal expansion and alternating loadings, shall satisfy the following equation:

$$\sigma_5 = \frac{p_c d_o}{4e_n} + \frac{0{,}75\, i\, M_A}{Z} + \frac{0{,}75\, i\, M_C}{3Z} \le f_{CR} \qquad (12.3.5\text{-}1)$$

In equation (12.3.5-1), one-third of the moment M_C is considered with respect to the material behaviour in the creep rupture stress range, unless confirmed otherwise by detailed inelastic analysis.

12.3.6 Stresses due to a single non-repeated support movement

Where the design stress as given in clause 5 is time-independent, the stress, σ_6, due to resultant moment, M_D, from a single non-repeated support movement, shall satisfy the following equation :

$$\sigma_6 = \frac{i\, M_D}{Z} \le \min(3f\,; 2R_{p0,2\ t}) \qquad (12.3.6\text{-}1)$$

where

M_D is the resultant moment from a single non-repeated anchor/restraint movement e.g. terminal point movements due to settlement of buildings, or due to damage caused by mining.

Where the design stress as given in clause 5 is time-dependent, the calculated stress shall not exceed the following:

— 0,3 times 0,2 % proof strength for ferritic steels at the calculation temperature;

— 0,3 times 1,0 % proof strength for austenitic steels at the calculation temperature.

NOTE It is possible that a small part of the piping system will undergo considerable inelastic strain when the rest of the system is almost entirely elastic. This happens when the part concerned is appreciably weaker than the rest, either due to reduced section size, weaker material or higher temperature. Conditions likely to cause significant inelastic strain should preferably be avoided. If this is not possible, a more complete inelastic analysis should be undertaken.

12.3.7 Determination of resultant moments

When determining the moment values M_A, M_B, M_C and M_D, used in equations given in 12.3.2 to 12.3.6, the following basic rules shall be considered.

For *n* simultaneously applied moments M_i (i = 1, 2, ..., n) with the co-ordinates M_{xi}, M_{yi}, M_{zi} referred to a right-angle co-ordinate system x, y, z the total resultant moment M is the sum of moments:

$$\vec{M} = \begin{pmatrix} M_{\mathrm{x}} \\ M_{\mathrm{y}} \\ M_{\mathrm{z}} \end{pmatrix} = \begin{pmatrix} \sum_1^n M_{\mathrm{x}i} \\ \sum_1^n M_{\mathrm{y}i} \\ \sum_1^n M_{\mathrm{z}i} \end{pmatrix} \qquad (12.3.7\text{-}1)$$

and

$$M = \sqrt{M_{\mathrm{x}}^2 + M_{\mathrm{y}}^2 + M_{\mathrm{z}}^2} \qquad (12.3.7\text{-}2)$$

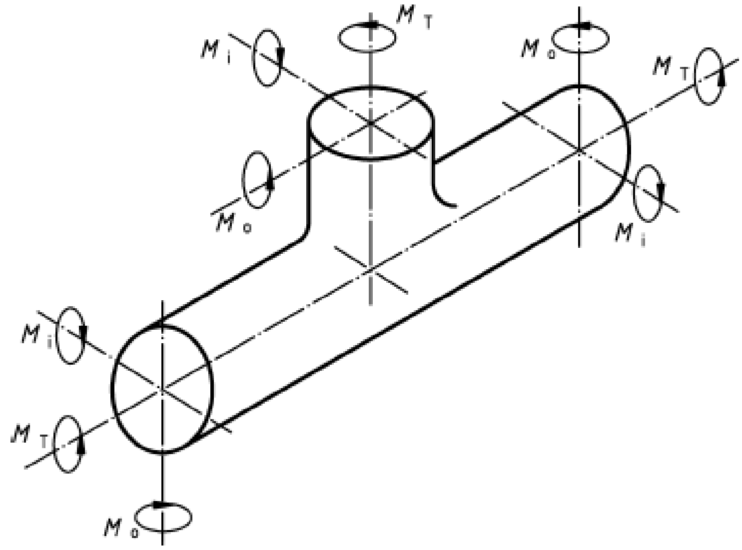
Where at a time T_{a}, the total moment $M_{T\mathrm{a}}$ and at another time T_{e} the total moment $M_{T\mathrm{e}}$ is applied, then the resultant alternating moments will be the difference between moments:

$$\vec{M} = \vec{M}_{T\mathrm{e}} - \vec{M}_{T\mathrm{a}} = \begin{pmatrix} M'_{\mathrm{x}} \\ M'_{\mathrm{y}} \\ M'_{\mathrm{z}} \end{pmatrix} = \begin{pmatrix} M_{\mathrm{x}\,T\mathrm{e}} - M_{\mathrm{x}\,T\mathrm{a}} \\ M_{\mathrm{y}\,T\mathrm{e}} - M_{\mathrm{y}\,T\mathrm{a}} \\ M_{\mathrm{z}\,T\mathrm{e}} - M_{\mathrm{z}\,T\mathrm{a}} \end{pmatrix} \qquad (12.3.7\text{-}3)$$

with the value

$$M' = \sqrt{M_{\mathrm{x}}'^2 + M_{\mathrm{y}}'^2 + M_{\mathrm{z}}'^2} \qquad (12.3.7\text{-}4)$$

Examples for moments on a T-piece and on a bend are given in Figures 12.3.7-1 and –2.



Key

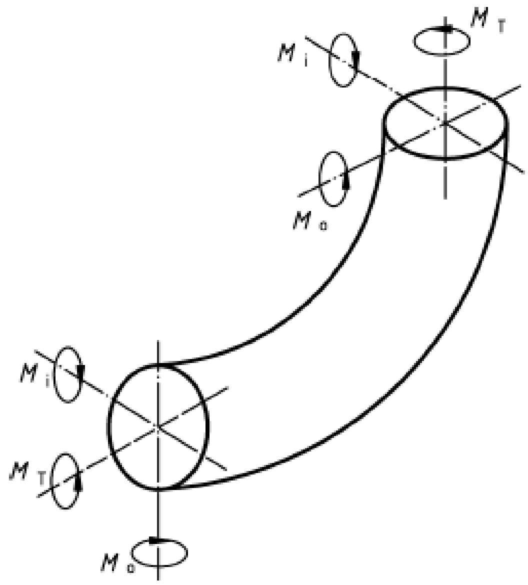
M_o is the out-of-plane bending moment;

M_i is the in-plane bending moment;

M_T is the torsional moment;

M_R is the resultant moment $\sqrt{M_o^2 + M_i^2 + M_T^2}$

Figure 12.3.7-1— Moments on a T-piece



Key

M_o is the out-of-plane bending moment;

M_i is the in-plane bending moment;

M_T is the torsional moment;

M_R is the resultant moment $\sqrt{M_o^2 + M_i^2 + M_T^2}$

Figure 12.3.7-2 — Moments on a bend

12.3.8 Reactions

The calculated reactions on connecting equipment shall not exceed the specified limits.

12.4 Fatigue analysis

Where the considerations of clause 10 require a fatigue analysis, the cyclic stress range and the corresponding allowable number of cycles for each of the specified loading conditions shall be determined at significant points in the piping system.

NOTE 1 The method of calculation in EN 12952-3 including appropriate annexes should be used to provide an acceptable analysis where external forces and moments are not significant. Alternatively, the method of calculation in EN 13445-3 may be used.

NOTE 2 Fatigue analysis of expansion joints is not subject to this clause.

12.5 Vibration

When vibration can occur, for example, due to the motion of the fluid in the pipe or due to externally imposed cyclic loads, the piping designer shall study the extent of the problem and examine the following as a means of eliminating or reducing the effects of vibration:

- provide piping route with alternative natural frequency;
- incorporate additional supports close to rotating / pulsating equipment;
- provide additional support in the vicinity of concentrated loads;
- additional anchors;
- fit shock arrestor or sway brace where thermal expansion is present;
- incorporate guides particularly at changes of direction;
- incorporate sliding supports in preference to hangers;
- change the number of spring supports.

13 Supports

13.1 General requirements

13.1.1 General

Clause 13 specifies the requirements for supporting and controlling the movement of piping systems subject to the requirements of EN 13480.

NOTE See also Annex I, Annex J, Annex K, Annex L, Annex M and Annex N.

It does not cover the main structures to which supports are attached, nor service conditions such as corrosion and erosion effects.

Supports are divided into:

- rigid supports;

- movable supports;
- intermediate (secondary) steel.

Supporting elements are those devices which connect the piping to the surrounding structure. They shall:

- carry the weight of the piping as well as that of any equipment connected to the piping;
- control the movement of the piping;
- direct and transfer static (or dynamic if they occur) loadings from the pipe to the surrounding structure, and generally remove or restrain one or more of the six degrees of freedom at particular points in the pipework.

13.1.2 Classification of supports

Supports shall be classified according to three levels, given in Table 13.1.2-1, depending upon the category of the piping according to PED being supported.

Table 13.1.2-1 — Classification of support

Category according to PED	Class of support
III	S 3
II	S 2
I / no [a]	S 1

[a] Including Sound Engineering Practice of a Member State according to PED, article 3.3.

Where equipment of different categories according to PED have a common support, the level of the support shall conform to the requirements of the most stringent support class.

NOTE For harmonization of fabrication, some supports may be supplied to a higher class than that required by the piping class.

13.1.3 Additional definitions

For the purposes of clause 13, the following definitions apply in addition to those given in 3.1:

13.1.3.1
anchor (fixed point)
rigid device used to prevent all relative pipe rotation and displacement at the point of application, under the design conditions of temperature and loading

13.1.3.2
line stop
device to restrain axial displacement of the piping

13.1.3.3
guide
device which permits pipe movement in a pre-determined direction whilst preventing movement in one or more other directions

13.1.3.4
sliding support or shoe
device to carry the vertical load component whilst restraining vertical downward movement but not significantly limiting planar displacements or rotations

13.1.3.5
roller support
base support with one or more roles having extremely small axial movement resistance

13.1.3.6
rigid support, rigid hanger
device to carry loads in one direction (vertical) whilst restraining movement in this direction

13.1.3.7
spring hanger, spring support, constant hanger, constant support
pipe support with variable or constant characteristic (spring hanger, constant hanger) to carry vertical loads whilst permitting vertical displacements, base mounted or suspended

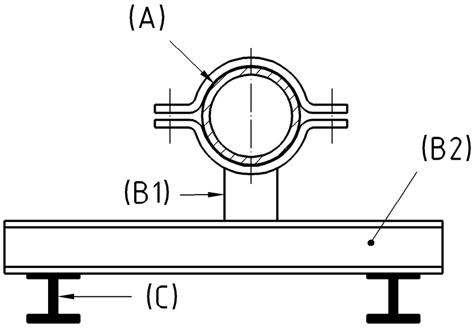
13.1.3.8
sway brace
preloaded device that exerts a restoring force to swaying pipework

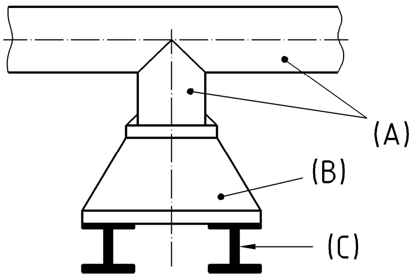
13.1.3.9
rigid strut
device to restrain the piping in a single direction, in many cases used for dynamic loading

13.1.3.10
shock arrestor (shock absorber, snubber)
device which is self-locking, or self-braking, to limit displacement in its direction of action, the rapid displacement of pipework subject to dynamic loadings, whilst permitting slow movements (such as those due to thermal expansion) in these directions

13.1.4 Boundaries

The boundaries between the support and the surrounding structure shall be as shown in Figures 13.1.4-1 to 13.1.4-3.





Pipe support made of standard components **Customised pipe support**

Key

(A) pipe

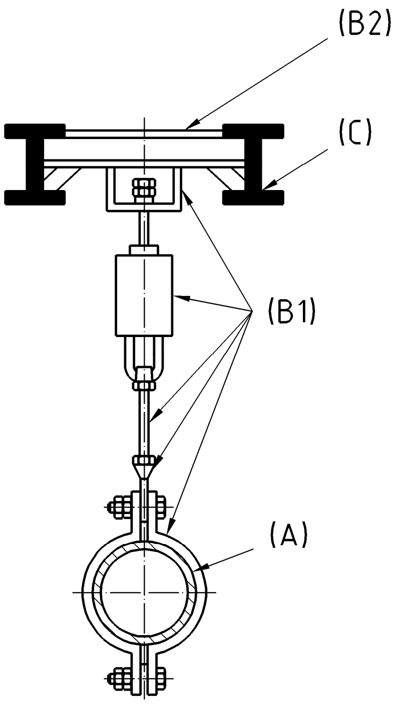
(B) pipe support

(B1) pipe support (e.g.clamp base)

(B2) pipe support (e.g. intermediate steel)

(C) structure

Figure 13.1.4-1 — Connections to steel structures



Key

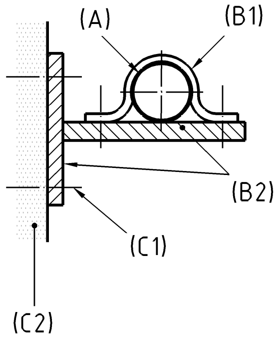
(A) pipe

(B1) pipe support (e.g. pipe clamp, spring support/hanger, threaded rods, connection parts)

(B2) pipe support (e.g. intermediate (secondary) steel work)

(C) structure

Figure 13.1.4-2 — Example of connection of a spring support to the structure



Key

(A) pipe

(B1) pipe support (e.g. pipe clamp)

(B2) pipe support (e.g. intermediate steel work)

(C1) bolts as part of structure

(C2) concrete structure

Figure 13.1.4-3 — Connections of a rigid support to concrete structures

13.1.5 Welded support attachments

13.1.5.1 Where support attachments are welded directly to the pipework (see Figure 13.1.5-1), the welding shall conform to EN 13480-4. The support attachment B shall conform to the requirements of clause 11. Stresses arising from any differential expansion between the pipe and a welded attachment shall be taken into account in the design of the support and of the piping.

13.1.5.2 Where an attachment B is forged or cast integrally with the pipe (see Figure 13.1.5-2), any welding to the attachment located within

$$l = \sqrt{2 d_{\mathrm{m}} e} \quad (13.1.5.2\text{-}1)$$

of the pipe surface shall conform to EN 13480-4, unless it can be demonstrated that the weld and heat affected zone have no effect on the mechanical properties of the pressure shell.

NOTE If the weld is farther from the pipe than this distance, the welding may alternatively conform to this Part of this European Standard.

13.1.5.3 Where a support component C is connected to a pipe A via an intermediate element or pad C1 (see Figure 13.1.5-3), the material of that pad shall be compatible with the pipe and welding to the pipe shall conform to the pipe welding requirements. The welding of the support to the pad shall conform to clause 11.

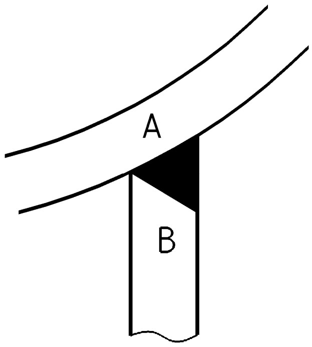


Figure 13.1.5-1 — Support welded to the piping

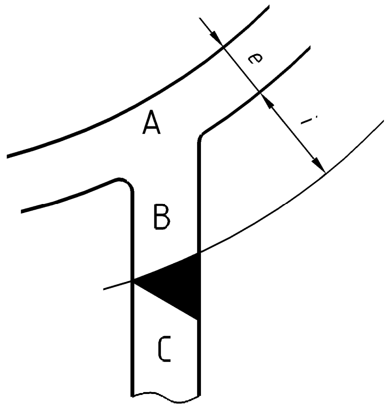


Figure 13.1.5-2 — Support integral with the piping

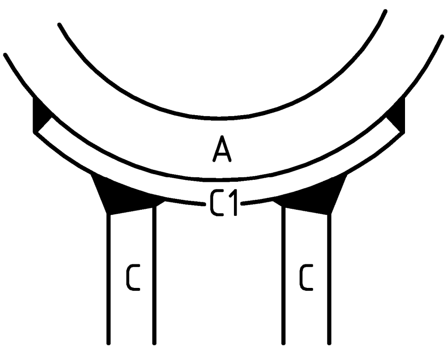


Figure 13.1.5-3 — Support with intermediate pad

13.2 Material requirements

The materials used in the manufacture of supports shall be acceptable for the operating and environmental conditions of the piping. Support materials in contact with the piping shall be compatible with, and shall not affect the required metallurgical characteristics of the pipe materials.

Materials shall conform to the requirements of EN 13480-2.

13.3 Design

13.3.1 General

Pipe supports can be defined as:

— standardised supports;

— customised (special designed) supports.

The form, dimension and load capacity of standardized supports were determined, demonstrated and catalogued.

Supports shall be designed to meet the requirements of the piping design. The support design shall be confirmed by calculation in accordance with this sub-clause or by type testing (rating). Type testing procedures shall be in accordance with Annex J. When applying type-tested parts, operating temperature shall be considered.

It shall be permissible to use standard types of support whose design has been previously verified by calculation or testing for the applicable conditions.

The piping engineer shall provide accurate details of loadings and movements at designated support points for the purpose of support design.

NOTE 1 This data is normally generated by means of a flexibility analysis of the piping (see 12.2.10).

The flexibility analysis of the piping shall include the piping weight together with the normal medium and insulation weights, and shall have considered all the additional factors outlined in the flexibility analysis. The effects on the loadings of pipe movement and angulation of hanger rods and friction shall be incorporated by the piping designer.

If the requirements of 12.2.10.1 are met, then no detailed analysis shall be required. In such a case, the designer of the piping wall shall ensure that all the following conditions are met:

— the angulation of the support rod from the vertical shall not exceed 4;

— the supports shall not significantly modify the pipe displacement;

— the variation in supporting effort shall not exceed 25 %.

— for spring and constant supports possible horizontal forces caused by lateral movement shall be considered, to maintain functionality: see 13.5;

— when selecting spring hangers, constant hangers, spring supports, constant supports or shock arrestors, a sufficient residual movement shall be ensured: see 13.5.

NOTE 2 Special consideration may be needed for supports close to sensitive equipment (e.g. rotating machinery) or where the piping would exert unacceptable loads on equipment connected to it.

When selecting supports, it shall take into account all anticipated external climatic loadings, such as wind, snow or ice.

Where piping is subject to other external effects (vibration, structural displacements, ground movement, earthquakes etc.), the purchaser shall specify those effects in the purchase order and shall define the relevant characteristics for incorporation into the design of the supports.

Any hydrostatic test loading shall be taken into account in the design of supports and the connected structures.

The design of the supports shall not produce stresses and deformations greater than the allowable stresses and deformations at any point in the pipework.

The support reactions shall not generate loadings at anchors or terminal points, when combined with thermal expansion and friction loads, which are greater than those calculated in clause 12.

The design of the supports shall not alter the predicted movement and loadings without the agreement of the pipework designer.

Any maintenance requirements shall be specified by the manufacturer and any lifetime restrictions shall be identified.

Where supports are subject to cyclic loads or movements (e.g. shock arrestors and struts), all the support components shall meet the specified design life and loadings / cycles.

The location, type, and identification mark of each support shall be provided on piping isometric drawings, schedules or by other means.

13.3.2 Design temperatures for support components

13.3.2.1 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 for temperatures in the range 0 °C to 80 °C. For system operating temperatures outside this range, the piping operating temperature shall be specified to the support designer and a specific justification shall be provided.

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.

13.3.2.2 Design temperature of components within the insulation

The temperature to be used for design purposes shall be in accordance with Table 13.3.2-1 and Figure 13.3.2-1.

Table 13.3.2-1 — Design temperature of components within the insulation

Type of component	Design temperature of support t
Components directly welded to the pipe, straps and clamps (i.e large face contact)	t_f
Components without direct contact with the pipe	t_f - 20 °C
Bolts, screws, nuts and pins	t_f - 30 °C
Where t_f is the temperature of the medium in the pipe.	

13.3.2.3 Design temperature of components outside the insulation

The temperature to be used for design purposes shall be in accordance with Table 13.3.2-2 and Figure 13.3.2-1.

Table 13.3.2-2 — Design temperature of components outside the insulation

Type of component	Temperature of the medium in the pipe t_f	Design temperature of support t
components directly connected to the pipe	> 80 °C	0,5 t_f but not less than 80 °C
	≤ 80 °C	80 °C
Bolts, screws, nuts and pins	> 80 °C	0,33 t_f but not less than 80 °C
	≤ 80 °C	80 °C
Where t_f is the temperature of the medium in the pipe.		

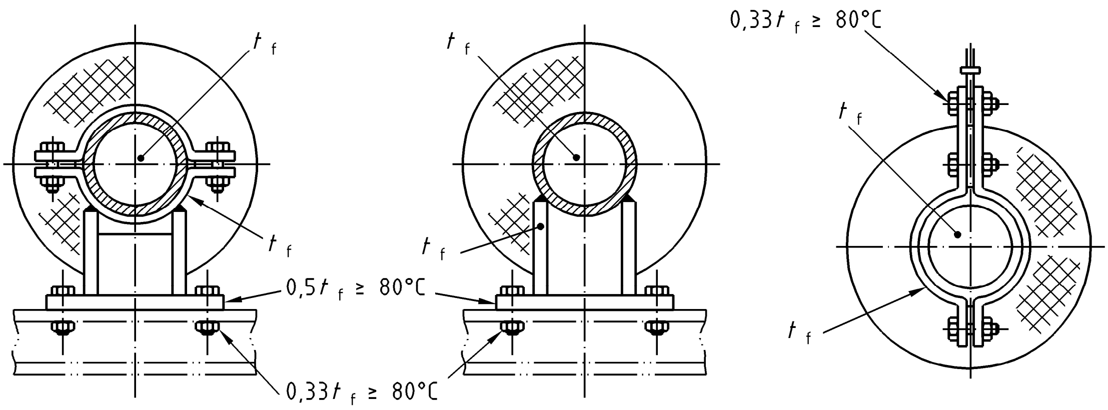


Figure 13.3.2-1 — Support design temperatures inside and outside insulation

13.3.2.4 Other components

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

13.3.2.5 High and low temperatures

For fluid temperatures above 600 °C and below - 20 °C, the support shall either be manufactured from material suitable for the design temperature and compatible with the pipe material, or shall incorporate a suitable thermal barrier.

13.3.3 Detail design

13.3.3.1 Components in contact with pipes shall be designed to avoid concentrated loadings on the pipe wall which could lead to localized plastic deformation of the pipe.

13.3.3.2 The detail design shall ensure that the pipe is firmly located within its support, and that where necessary, adjustment shall be provided to compensate for tolerances in pipe and support dimensions.

13.3.3.3 All suspensions (hangers) shall be provided with means of adjustment.

Hangers used for pipe sizes greater than DN 100 shall be designed to permit adjustment whilst carrying the specified load.

Where threaded adjustment is used, the minimum length of thread engagement shall be not less than 0,8 times the thread diameter.

All nuts, pins, or fixing devices shall be provided with a means of locking.

13.3.3.4 Components shall be designed to avoid the accumulation of water or debris.

NOTE Dust covers or similar protection may be specified by the purchaser.

13.3.3.5 Components shall be designed to minimise bending loads on threaded parts.

13.3.3.6 Where lugs are welded to a vertical pipe to carry the weight and other loads, an appropriate number of lugs shall be evenly spaced around the pipe and accurately located to ensure uniform contact.

NOTE Excessive bending moments on the pipe wall should be avoided, stress situation of pipe wall should be checked e.g according to Clause 11.

13.3.3.7 Where vertical pipes are supported by twin hanger rods as rigid supports, the support shall be designed to carry the total load safely in either rod, unless the design prevents the load distribution becoming unequal (e.g. trunnion type clamps).

13.3.3.8 Details of the attachment of supports to the main structure shall be agreed with the structural designer, who shall ensure that all loadings imposed on the structure by the piping are within acceptable limits (see Annex K for examples).

13.3.3.9 The dimensioning of intermediate or secondary steelwork supplied for supporting the pipe shall be based on good industrial practice as e.g. defined in EN 1993. Secondary steel work shall fulfil the requirements of 13.3.6.3.

13.3.3.10 The threads of bolts shall not be subjected to shear loads unless this is specifically included in the bolting design calculations.

13.3.4 Buckling

When considered necessary, the design of the support and any secondary or intermediate steelwork shall demonstrate resistance to buckling.

NOTE See Annex L for guidance.

13.3.5 Support location

Supports shall be located following flexibility analysis or by assessment of the loads to be supported. In either case the availability of suitable connections to the surrounding structure shall be considered, and intermediate or secondary steelwork or frames incorporated in the support design where required.

13.3.6 Determination of component sizes

13.3.6.1 General

Dimensioning of support components designed by calculation shall be based upon good engineering practice such as EN 1993. The requirements of 13.3.6.2 shall be considered. For further guidance, see Annexes I, J, K, L and M.

13.3.6.2 Stress levels

The individual or equivalent stress levels shall not exceed the allowable stresses given in Table 13.3.6-1.

The equivalent stress, σ_e is defined as:

$$\sigma_e = \sqrt{(\sigma_a + \sigma_b)^2 + 3\,\tau^2} \qquad (13.3.6\text{-}1)$$

where

σ_a is the calculated axial (membrane) stress;

σ_b is the calculated bending stress;

τ is the calculated shear stress.

The maximum permissible stress is:

$$f = \min\left(\frac{R_{eHt}}{1{,}5}\ or\ \frac{R_{p0,2t}}{1{,}5}\,;\ \frac{R_m}{2{,}4}\,;\ f_{cr}\right) \qquad (13.3.6\text{-}2)$$

NOTE 1 For occasional operating conditions, see 4.2.5.2.

NOTE 2 For creep data for time periods other than 200 000 h, see 5.3.2.

13.3.6.3 Allowable stress

Allowable stress of different loadings and stresses shall be in accordance with Table 13.3.6-1.

Table 13.3.6-1 — Allowable stress for pipe supports

Stresses	Normal operating conditions	Occasional operating conditions
σ_a	$\leq 1{,}0\ f$	$\leq 1{,}2\ f$
σ_b	$\leq 1{,}0\ f$	$\leq 1{,}2\ f$
τ	$\leq 0{,}6\ f$	$\leq 0{,}7\ f$
σ_e	$\leq 1{,}0\ f$	$\leq 1{,}2\ f$

NOTE 1 The allowable stress for pipe supports not operating in the creep range are:

- for pipe supports analysed with plate or shell theory:
  - normal operating conditions: $\sigma_b \leq 1{,}5\ f$; $\sigma_e \leq 1{,}5\ f$
  - occasional operating conditions: $\sigma_b \leq 1{,}8\ f$; $\sigma_e \leq 1{,}8\ f$
- for double symmetric solid sections:
  - normal operating conditions: $\sigma_b \leq 1{,}1\ f$; $\sigma_e \leq 1{,}1\ f$
  - occasional operating conditions: $\sigma_b \leq 1{,}3\ f$; $\sigma_e \leq 1{,}3\ f$

NOTE 2 Allowable stress for welded connection see 13.4.1, for occasional operating conditions the allowable stresses are 1,2 times those for the normal operating condition.

NOTE 3 For bolted connections, see 13.4.2.

13.4 Connections

13.4.1 Welded connections

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

The allowable stress of weld joints shall be multiplied compared to the base material by the weld efficiency factor z.

- For all welded joints, only inspected by visual examination the weld efficiency factor z = 0,7.
- The weld efficiency factor may be increased up to z = 0,85, if following conditions are fulfilled:

a) a type-testing has been performed;

b) partly mechanized or fully-mechanized welding processes are used;

c) weld joints are spot-checked by NDT (MT/PT).

- For butt welds and full penetration welds with 100 % NDT a weld efficiency factor z = 1,0 is permissible.

Additionally the permissible shear stress of the base material shall not be exceeded.

13.4.2 Bolted connections

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

13.5 Design requirements for special components

13.5.1 Constant load hangers and supports

13.5.1.1 General

The following requirement shall be met when the use of constant load hangers and supports are specified.

NOTE Constant load (constant effort) supports generally are specified by the piping designer where vertical pipe movements are relatively large and spring reaction forces would be high. They are also specified for particular locations where load change during movement of more than 5 % of the preset load would not be acceptable. Further information is given in Annex I.

13.5.1.2 Base mounted constant supports/constant hangers

Constant hangers shall be designed for a load inclined at an angle of 4°. Constant supports shall withstand a lateral loading of 10 % of the design load. Suitable sliding surfaces shall be provided if lateral movement occurs.

13.5.1.3 Load deviation from preset load

The deviation in supporting effort, including friction effects, shall not exceed ± 5 % of the preset load at any point in the total travel of the unit at vertical loading. Where a lower deviation is required, this shall be specified by the piping manufacturer.

13.5.1.4 Site adjustment of the preset load

Constant load units shall have provision for on site adjustment of the preset load by a minimum of ± 15 % . This adjustment shall not reduce the specified travel of the support.

13.5.1.5 Overtravel

Provision shall be made in the design of the support for travel in excess of the calculated piping movement under design conditions. Overtravel of 10 % of the calculated movement shall be specified, with a minimum of 25 mm. The distribution and direction of overtravel shall be determined by the piping designer.

13.5.1.6 Blocking

All units shall be provided with a means of locking the movement, and shall have stops to positively limit the upwards and downwards travel.

The units shall be supplied locked in the mounting position specified by the purchaser.

When the support is locked, it shall be capable of carrying twice the specified preset load.

13.5.1.7 Spring life

Springs shall retain the set load over the design life of the piping system, with variation due to ageing not exceeding ± 2,5 % of the set load.

13.5.1.8 Scale plate

Units shall be provided with a fixed non-corrodible scale plate to meet the anticipated environmental conditions, indicating as a minimum

- support reference number;
- unit type;
- unit size;
- total travel;
- mounting load;
- installed position;
- normal operating position;
- manufacturer's name.

13.5.2 Variable load hangers and supports

13.5.2.1 General

Variable load spring hangers and supports are normally used for relatively small vertical pipe movements. They can be used when a load variation of up to 25 % of the design load is acceptable during pipe movements. Greater variation may be acceptable if permitted by the piping analysis.

When spring supports whose supporting effort varies directly in relation to the vertical movement of the pipe are specified, the load variation shall conform to either equation:

$$\text{Load Variation} = \frac{\text{design load - preset load}}{\text{design load}} 100\%$$

or

$$\text{Load Variation} = \frac{\text{thermal displacement} \times \text{spring rate}}{\text{design load}} 100\%$$

NOTE Further information is given in Annex I.

13.5.2.2 Base mounted variable spring supports/variable spring hangers

Spring hangers shall be designed for a load inclined at an angle of 4°. Base mounted variable load spring supports shall withstand a lateral loading of 10 % of the design load. Suitable sliding surfaces shall be provided if lateral movement occurs.

13.5.2.3 Tolerance on spring rate

For vertical tension or for vertical compression the deviation of the load shall be less than 5 % compared with the theoretical load travel behaviour, Figure I.2-1 (10 % for support class S1), including variation due to ageing (see 13.5.1.7).

13.5.2.4 Overtravel

Provision shall be made in the design of the support for travel in excess of the calculated piping movement under design conditions. A minimum of 10 % overtravel of the calculated movement shall be provided with a minimum of 5 mm.

13.5.2.5 Blocking

All spring supports shall be provided with a means of locking the movement, and shall have stops to positively limit the upward and downward travel.

The spring supports shall be supplied locked in the monitoring position specified in the technical specification.

When the support is locked it shall be capable of carrying twice the maximum rated load.

13.5.2.6 Spring life

Springs shall retain the specified load over the design life of the piping system, with variation due to ageing not exceeding ± 2,5 % of the set load.

13.5.2.7 Scale plate

Units shall be provided with a fixed non-corrodible scale plate to meet the anticipated environmental conditions, indicating as a minimum:

- support reference number;
- unit type;
- unit size;
- total travel;
- preset load;
- installed position;
- normal operating position;
- manufacturer's name.

13.5.3 Rigid struts

The following requirements shall be met when the use of rigid struts is specified.

- Their characteristics shall be specified by the piping manufacturer;
- Rigid struts shall be operable in both tension and compression;
- Rigid struts shall be fitted with spherical bearings at the ends. The bearings shall have the minimum of free play, but permit the angulation of the strut of at least 6 ° from the operating plane;
- Rigid struts shall permit at least ± 25 mm adjustment in length;
- Structural brackets and pipe clamps used for rigid struts shall change clearances and stiffness of the supporting system only by a minimum amount.

When subjected to the design load, the buckling deflection of the strut from the axis between the centres of the bearings shall not exceed 1 mm for strut lengths up to 1 000 mm in length and not more than 1 mm per 1 000 mm for longer struts.

NOTE 1 Rigid struts are normally used for restraining piping during dynamic loading in a single direction.

NOTE 2 Rigid struts should have a high stiffness with minimum of free play in the assembly.

13.5.4 Shock arrestors (shock absorber, snubber)

13.5.4.1 The following requirements shall be met when the use of shock arrestors are specified.

- The use of a particular design of shock arrestors, e.g. mechanical or hydraulic, shall be specified by the piping manufacturer;
- The shock arrestors shall not restrict or limit thermal movement during normal operation;
- The shock arrestors shall operate equally in either tension or compression;
- The shock arrestors shall operate in any attitude or position in accordance with the purchase specification;
- The shock arrestors shall be fitted at each end with spherical bearings which have minimum of free play, but which permit angulation of a maximum of 6 ° of the unit from the normal operating plane;
- Structural brackets and pipe clamps used for shock arrestors shall change clearances and stiffness of the supporting system only by a minimum amount;
- The shock arrestors shall have the operating parameters given in Table 13.5.4-1.

NOTE Shock arrestors may be used in the piping design to control pipe movements during dynamic events such as earthquake or rapid closure of valves. They are not designed to carry the weight of the piping system and further information is given in Annex I.

Table 13.5.4-1 — Operating parameters for shock arrestors

Parameter	Values
Activation velocity range	3 mm/s to 5 mm/s
Bleed velocity range (if applicable)	0,2 mm/s to 2 mm/s
Operating frequency range	0,5 Hz to 50 Hz
Control valve release (where applicable)	The greater of 200 N or 2 % of the rated load
Drag load maximum (at 0,5 mm/s)	The greater of 200 N or 2 % of the rated load
NOTE 1 The lost motion during load reversal (from play in the bearings and other components) should not exceed 1,5 mm NOTE 2 Fatigue limits may be specified by the purchaser.	

13.5.4.2 Hydraulic shock arrestors shall be fitted with a fluid level indicator, and shall be provided with fluid and seals suitable for the anticipated environment.

13.5.4.3 A travel position indicator shall be fitted if specified by the purchaser.

13.5.4.4 Provision shall be made in the design of the shock arrestor for travel in excess of the calculated piping movement under design conditions. Overtravel of 10 % of the calculated movement shall be specified, with a minimum of 25 mm. The distribution and direction of overtravel shall be determined by the piping designer.

13.5.5 Sliding supports

13.5.5.1 Sliding supports shall be designed and installed such that they cannot become disengaged during normal operation. Regard shall also be paid to the possibility of up-lift causing separation of the sliding surfaces.

13.5.5.2 Unless devices are introduced to direct and limit the motion of a sliding support, the design and installation of the unit shall allow for the reversal of the predicted travel.

13.5.5.3 Sliding supports shall be designed to prevent contamination of the sliding surfaces during normal operation.

13.5.5.4 The sliding area shall be of sufficient size and proportion to accommodate all specified movements with a margin of safety of at least 25 mm in each direction.

13.5.5.5 The loads resulting from the friction of sliding surfaces shall be incorporated into the design of the supporting structure. Where steel on steel sliding surfaces are used, the coefficient of friction shall be 0,3, unless it can be shown that the choice of sliding surfaces will provide a smaller coefficient consistently over the specified operating life of the piping. Where lower frictional forces are required, low friction materials, e.g. PTFE, sized to suit the physical and ambient conditions in which the bearing is intended to operate shall be used.

13.5.6 Anchors

Anchors shall be designed to provide a fixed point for the piping connection to the structure.

Anchors shall carry all the predicted forces and moments, including those from sliding supports and pressure end loads, where appropriate.

13.6 Documentation of supports

The support manufacturer shall provide the purchaser of supports with a certificate confirming that the supports comply with the requirements of clause 13 and Annex N.

13.7 Marking of supports

All supports shall be marked in accordance with the purchaser's requirements. Where supports are not supplied fully assembled, each part or sub-assembly shall be identified.

Annex A
(informative)

Dynamic analysis

A.1 General

In addition to the static conditions and cyclic pressure and temperature loadings covered by 4.2, piping may be subjected to a variety of dynamic loadings. Dynamic events should be considered in the design of the piping. However, unless otherwise specified, such consideration may not require detailed analysis. The effects of significant dynamic loads should be added to the sustained stresses in the design of the piping. Continuous dynamic loads should be considered in a fatigue analysis.

Where the dynamic event produces reverse forces, it may be acceptable to derive maximum loadings by combining those forces whose direction makes them additive to the static loads. However, care should be taken regarding the displacements as both plus and minus movements may be needed for layout and supporting detail design.

There are a number of methods for the calculation of the effect of dynamic events, such as:

a) simplified static equivalent;

b) quasi-static equivalent;

c) shock response spectra modal analysis;

d) force time history.

Experience has shown that for properly supported piping, the use of simplified methods generally leads to acceptable engineering solutions for the prevention of damage during dynamic events. Where complex analysis is to be undertaken, care should be exercised in the selection of suitable programmes and consistent data for the derivation of forces and allowable loads.

Piping and piping components may also be analysed by subjecting full or part scale models to a vibratory regime comparable to the expected dynamic loading.

A.2 Analysis by calculation

A.2.1 Seismic events

A.2.1.1 General

Seismic events produce vibratory ground movements which are transmitted through the building structure to piping and other equipment. The structure and equipment respond by undergoing accelerations and displacements whose magnitude varies with their stiffness and natural resonance frequencies.

The analysis of the interaction of the building structure with the seismic driving forces is not within the scope of piping design and the associated response will normally be supplied by the purchaser or site owner, following earthquake assessment and structural analysis of the proposed building.

An analysis of the piping should be carried out to show the maximum forces and moments generated within the piping as a result of the structures response to the predicted earthquake.

The type of calculation determines the form and extent of seismic data to be made available to the piping designer.

A.2.1.2 Simplified static equivalent analysis

This method ignores the variation in the structure's response at different frequencies and damping rates, and calculates the displacements and forces in the piping using a single equivalent static accelerating force for each principal direction of seismic movements. This acceleration is based upon the maximum value arising from the earthquake. It may be presented to the designer as a ground base response spectrum, or calculated for each level within the building structure, or given as a single set of responses which are considered to envelope the different responses applicable to the piping.

Where no building related accelerations are available, the designer should use the peak ground acceleration as the maximum acceleration a_i.

The static equivalent acceleration, a_{cqi}, for direction i is calculated as follows:

$$a_{cqi} = k_i \ a_i \tag{A.2.1-1}$$

where

a_i is the maximum acceleration defined for the level in direction i;

k_i is a factor;

- $k_i = 1$ where the natural frequencies of the piping can be shown not to coincide within 10 % of the peak vibration frequencies in the response spectrum of the structure;
- $k_i = 1{,}5$ where no check on the coincidence of piping and building vibration characteristics has been undertaken.

A.2.1.3 Quasi-static equivalent analysis

This calculation applies a single static acceleration for each of the directions of the ground vibration equivalent to the highest acceleration in the building response spectrum which can excite the piping. For this method, the significant natural frequencies of the piping should be calculated.

The quasi-static equivalent acceleration $a_{qe\ i}$ for direction i is calculated as follows:

$$a_{qe\ i} = \bar{k}_i \ a_{fi} \tag{A.2.1-2}$$

where

a_{fi} is the maximum acceleration in the ground or level vibration spectrum at frequencies greater than or equal to the first own frequency of the piping;

$\bar{k}_i$ is a factor related to the contributions of multiple own frequencies for the shape of the piping system.

The factor $\bar{k}_i$ should be determined from Table A.2.1-1. Lower values of the factor may be used where their admissibility is demonstrated.

Table A.2.1-1 — Values of $\bar{k}_i$

Model	$\bar{k}_i$
Multi supported linear beam with equal span lengths	1,0
Cantilever beam	1,0
Single beam supported at both ends (maximum forces are to be applied at every cross section)	1,0
Single plane systems, e.g. frames, girder systems, single plane piping	1,2
3 dimensional systems with complex shapes	1,5

For rigid piping (i.e. where the lowest own frequency of the system is higher than or equal to the cut-off frequency of the ground vibration spectrum) the value of $\bar{k}_i$ may be taken as 1,0.

For the determination of support reactions, the value of $\bar{k}_i$ may be taken as 1,0 irrespective of which model is used from Table A.2.1-1.

A.2.1.4 Modal response spectra analysis

For modal response spectra analysis, the piping designer requires a building response spectrum for each level/location within the structure, or a spectrum which can be considered to envelope the responses within the structure. This modal response spectrum is derived from the maximum accelerations generated by the earthquake at differing frequencies over an appropriate period of time, and their interaction with the building structure. Vibration analysis of the piping should be carried out to determine the displacements, moments, and forces for the imposed accelerations at each significant frequency in the modal spectrum.

The total response of the piping (displacements, moments, forces) for each direction should be obtained by combining each peak modal response by the square root of the sum of the squares (SRSS) method, i.e.

$$R_i = \pm\sqrt{\sum_{m=1}^{n} R_{mi}^2} \qquad \text{(A.2.1-3)}$$

where

R_i is the total response in the principal direction i;

R_{mi} is the peak response due to the mode m;

n is the number of significant modes.

The combination of piping responses from the three principal directions should be based on the following assumptions:

— the piping responses to different building modal peaks do not occur at the same time;

— peak responses do not occur simultaneously in the three principal directions;

— peak stresses due to the different modes do not generally occur in the same place in the piping.

Consequently the maximum response of the system need not be calculated by applying the SRSS method to the three orthogonal directional maxima.

A.2.1.5 Force time history analysis

Where seismic displacements of the supporting structure are known with respect to time, the dynamic response of the piping system can be determined. This is done by imposing the pattern of accelerations or displacements at the support and terminal point locations onto a suitable model of the piping which incorporates the stiffness and masses of the piping and appropriate dynamic damping factors. The resultant stresses can be determined for the piping displacements as a series of calculations at discrete time intervals.

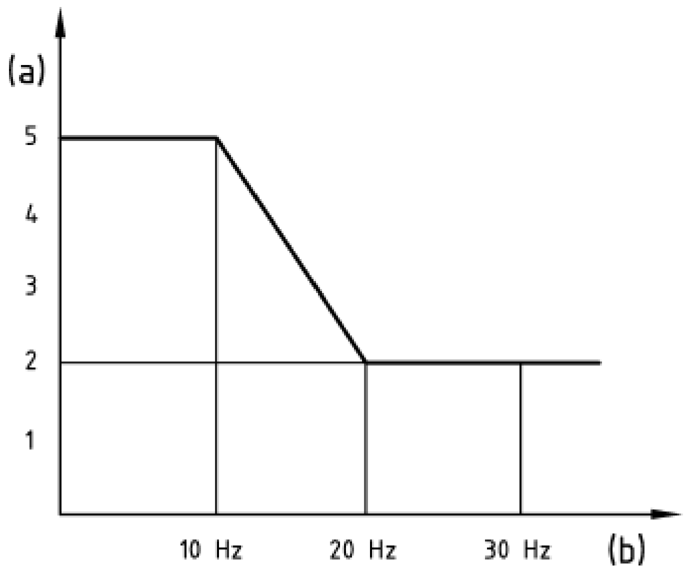
Various mathematical programmes are available for the solution of the dynamic problems, and the designer should ensure that the model and analytical methods are appropriate.

The time intervals should be chosen to ensure that no significant short term excitation is missed, and the number of steps should ensure that all significant displacements are included in the analysis.

The seismic stresses in each of the three principal directions should be combined using the SRSS method for each time step, and the highest resultant values generated during the seismic event should be added to the sustained loads to determine the total stress for design purposes.

A.2.1.6 Damping values

Vibration in piping and structures is subject to energy dissipation or damping. Sources of damping arise from the internal friction of the materials, imperfect connections between components, sliding friction, and other features. The assessment of the extent of damping for particular sources is complex and specific, so that for the purpose of this Annex the graph given in Figure A.2.1-1 should be used for those methods of dynamic analysis incorporating damping, unless other appropriate and reliable data is available.



Key

For all sizes of pipe:

(a) critical damping in %

(b) frequency in Hz

Figure A.2.1-1 — Damping for seismic events

A.2.1.7 Seismic support displacement

The effect of relative movement of supports and anchors during seismic events should be considered in the calculation of the total stresses. For piping supports in the same level of a single building, these effects may be small, but where there is no coupling between parts of the supporting structure, the relative displacements can be significant.

In such cases, the designer should use the absolute sum of the displacements at anchors in each of the three principal directions (discounting signs). As an alternative, detailed force time history analysis of the supporting structure may be used to determine the maximum relative displacements and consequent stresses. It should be noted that these relative movement stresses are self limiting and secondary.

A.2.2 Rapid valve closure

A.2.2.1 General

If the flow of a fluid in a piping system is interrupted by the rapid closure of a valve downstream of the source of the flow, a pressure wave can be generated in the fluid, travelling back from the valve to the source. Such a wave will interact with the piping and will be reflected from the source to create complex pressure patterns within the system. In multi-branched systems, these patterns are further influenced by waves travelling in pipes meeting others out of phase. Vibration is caused by the differential wave pressures generated in the system creating out of balance forces in the piping which can take several seconds to decay. This is called water hammer.

For this phenomenon to occur, the action of the valve will be sufficiently fast for it to close in less time than it takes for a wave travelling at sonic speed in the fluid to travel from the valve to the source and be reflected back to the valve. This is called rapid valve closure.

The way in which a valve closes can vary from one type to another. It is generally assumed that the rate of reduction of area is constant over a substantial part of the stroke with final closure at a reduced rate to minimise impact on valve seating. Such a tail to the closing curve will increase the total closure time with generally beneficial results for the effect of water hammer.

It should be noted, however, that fluid flow will not have the same characteristic curve, being proportionately higher than the area reduction at the same point in time. Consequently, the valve may close by a large proportion of its area without significantly reducing the fluid flow. Those calculations which model the valve closure characteristics need particular care in this respect.

The rise in pressure should be calculated to ensure that the piping can withstand the combined sustained and shock pressure stresses. Additionally, the magnitude of the out of balance forces should be determined and applied to the design of the piping to calculate the stresses within the pipes and nozzles, and at connections to the supporting structure.

In addition to the calculation of the forces in the system, the designer should determine the movement of the piping under this forced vibration to ensure adequate clearances.

It should be noted that in addition to the pressure wave upstream, there may be a rarefaction wave created downstream of the closing valve, and the resulting vacuum effects should be assessed.

NOTE Attention is also drawn to the effects of the sudden opening of valves. On the upstream of the valve similar effects to those due to valve closure may be seen due to a front of lower pressure passing backwards up the pipe. In the piping downstream of the valve, unbalanced momentum and pressure forces will act by turn on each section of straight pipe as the fluid or its pressure front progresses.

A.2.2.2 Simplified static analysis of rapid valve closure

This method considers only the initial pressure rise in the system following valve closure and assumes the stresses caused by this to be the maximum that the system will experience. It ignores the interactions and damping of the waves and the dynamic response of the system to the vibration. The analysis is conservative and can lead to an over protection of the piping which may conflict with the thermal or other design criteria.

a) Pressure rise assessment

Closure is rapid if the following equation is satisfied:

$$T < \frac{2L}{v_s} \tag{A.2.2-1}$$

where

— L is the length of the system;

— T is the effective valve closure time;

— v_s is the sonic velocity in the fluid.

The initial rise in pressure dP is given by:

$$\mathrm{d}P = v_s \, v \, \rho \tag{A.2.2-2}$$

where

— v is the velocity of the fluid;

— ρ is the density of the fluid under the calculation conditions.

NOTE This is Joukowsky' s formula.

The sonic velocity may be calculated as:

$$v_s = \sqrt{\frac{k}{\rho}} \tag{A.2.2-3}$$

where

k is the fluid bulk modulus.

For piping with significant elasticity, this may be modified to:

$$v_s = \sqrt{\frac{1}{\rho\left(\frac{1}{k} + \frac{D_o}{eE}\right)}} \tag{A.2.2-4}$$

The designer should ensure that the minimum design pipe wall thickness can withstand the operating pressure plus the maximum dynamic pressure rise dP.

b) Static assessment of dynamic loads

The effects of imbalance or surges on the piping system may be assessed by applying a calculated pressure differential to the ends of straight runs of pipe or at changes in direction. The differential pressure is the proportion of the peak pressure developed over the piping length under consideration and it is assumed to act over the internal area of the pipe. In calculating the resulting forces, factors should be applied which makes allowance for the variation in closure rate throughout the valve stroke and the dynamic nature of the actual loadings.

The maximum out of balance load, F, in a length of pipe section, L, may be calculated as follows:

— for stiff piping

$$F = 2\frac{M}{A}\frac{L}{\lambda}\,\mathrm{d}P\,\pi\,\frac{D_\mathrm{i}^2}{4} \tag{A.2.2-5}$$

— for flexible piping

$$F = 4\frac{M}{A}\frac{L}{\lambda}\,\mathrm{d}P\,\pi\,\frac{D_\mathrm{i}^2}{4} \tag{A.2.2-6}$$

$$\lambda = v_\mathrm{s} T \tag{A.2.2-7}$$

$$L/\lambda \; M/A \leq 1 \tag{A.2.2-8}$$

where

λ is the wavelength of the pressure wave;

M is the maximum rate of valve area closure;

A is the average rate of closure determined by the total closure time.

A.2.2.3 Advanced methods of calculation

The characteristics and effects of the pressure wave created by rapid valve closure may also be assessed by time history or modal analysis.

The development of the pressure pulse throughout the piping system can be idealised using mathematical modelling of the events, and these pressures used at a large number of time intervals to determine the forces at terminals, or changes of direction. The forces thus derived can be used as the driving factor in an analysis of the vibrational response of the piping to these forces.

If modal analysis is used, the designer should check that the cut off frequency does not exclude any significant higher modes resulting from the interaction of waves in the piping, as the system can be relatively stiff for these frequencies.

These advanced methods may incorporate coupling between the fluid and the piping and can thus incorporate the damping of the pressure wave by the transfer of energy to relatively stiff piping. For steam, or similar fluids where the mass of the fluid is negligible relative to that of piping, the advantage of the use of the advanced method is small.

Whilst these methods offer a potentially more accurate and less conservative solution to the problem of rapid valve closure, the advanced techniques for rapid valve closure analysis can be very sensitive to the modelling of the fluid source, the valve characteristics, the supports, and the fluid behaviour. The designer should be satisfied that the mathematical representations of all aspects are suitable and accurate.

A.2.2.4 Damping values

Vibration in piping and structures is subject to energy dissipation or damping. Sources of damping arise from the internal friction of the materials, imperfect connections between components, sliding friction, and other features. The assessment of the extent of damping for particular sources is complex and specific, so that for the purpose of this Annex the graph given in Figure A.2.1-1 should be used for those methods of dynamic analysis incorporating damping, unless other appropriate and reliable data is available.

A.2.3 Flow induced vibration

A.2.3.1 General

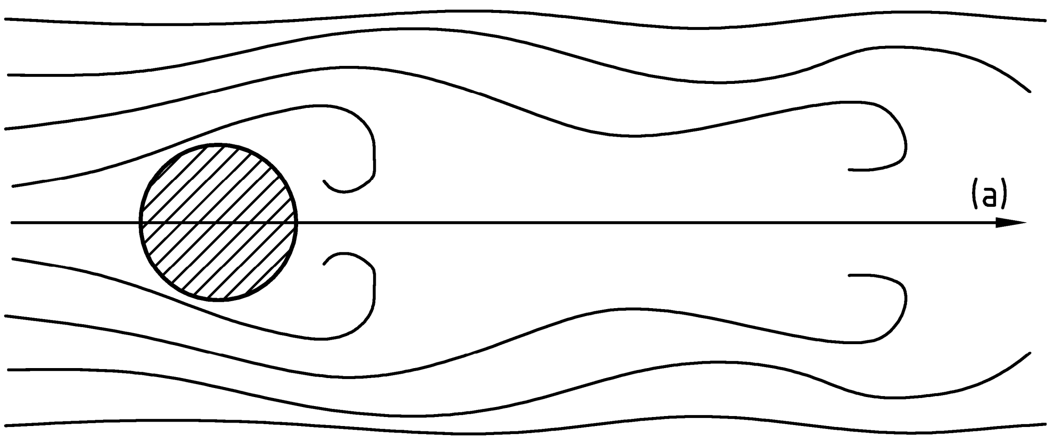
Disturbances to the smooth flow of fluids in piping systems can cause vibrations to be set up in the fluids. The fluid vibration can be transferred to the piping itself and in some circumstances large amplitude oscillations can be generated.

A piping system can be subject to a number of sources of excitation simultaneously and complex analysis may be required to assess the effect of these and the subsequent influence on the piping. Much of the data required to predict pipe movement is derived from experimental work and relates to particular conditions and geometry.

Unless reliable and appropriate data and mathematical models are available, the designer should consider the general mechanisms and problems posed by the more significant sources of flow related vibration in the design of the piping, and be prepared to make modifications if problems are experienced in operation.

A.2.3.2 Vortex shedding

The presence of a body in the path of fluid flow will create vortices downstream which are formed on alternate sides of the object in a regular pattern. This phenomenon will be found both internally, caused by the piping, and externally, caused by the passage of fluid (including wind) over the piping. A typical vortex pattern for a cylinder in a flow path is shown in Figure A.2.3-1. Such a pattern can be formed by a tube set into the flow, for example a thermometer or other measuring device. Similar patterns can be formed by arrays of tubes across the flow or non-circular shapes such as flat plates (in butterfly valves).



Key
(a) Flow

Figure A.2.3-1 — Typical vortex pattern

These vortices create an alternating force on the object normal to the flow and a smaller oscillating force in the direction of the flow.

The frequency, f_F , of the main force, *F*, can be expressed for a cylindrical object as:

$$f_F = S \frac{v}{D} \tag{A.2.3-1}$$

where

v is the velocity of the fluid;

D is the diameter of the cylinder;

S is the Strouhal Number which comes from appropriate literature.

S = 0,2 may be used for fluids with a Reynolds number between 10^3 and 2×10^5.

The magnitude of the force *F* may be expressed as:

$$F = C\, J\, \frac{1}{2}\, v^2\, D\, L\, \sin\left(2\pi\, f_F T\right) \tag{A.2.3-2}$$

where

L is the length of the system;

C, J and f_F are functions of the Reynolds number and need to be established for the fluid properties from appropriate literature or by experimental procedures.

Where the frequency of the vortex force lies within approximately ± 25 % of the natural frequency of the object in the flow, the two frequencies can tend towards synchronisation and large amplitude resonance can develop. The transmission of these vibrations to the piping depends upon the coupling of the object to the fluid and the pipe wall.

The strength of the vortex lift effect will be reduced in practice by turbulence around the object, by surface roughness disturbing the smooth flow of the fluid, by tapering the object, and by inclining the object to the flow. The proximity of other objects in the flow may also break up the development of strong vibrations.

A.2.3.3 Pump induced fluid pulsing

The operation of pumps does not generally produce a completely uniform delivery or suction pressure. The nature of the pressure variation in the fluid is dependent on the characteristics of the pump and the operating conditions.

Where possible, the designer should consider the layout of piping close to pumps, to dissipate the energy of the pulses and to avoid sharp changes of direction and the development of sympathetic vibration in the pipes.

If the frequency spectrum of the fluid pulses at the pump outlet is known, the response of the piping to this excitation can be modelled and analysed by either of the main methods of dynamic analysis. If such calculations are to be undertaken, the designer should ensure that the data and models accurately represent the operating conditions.

A.2.4 Safety valve discharge

A.2.4.1 General

The discharge of a safety valve will produce a reaction load on the piping to which it is connected. The initial rapid opening of the valve produces a dynamic component to the force which can be significant.

The effect should be treated as a localised event producing point loading at the nozzle connecting the valve to the piping, and should be incorporated into the design of the piping and the supporting arrangements. Where more than one valve is incorporated into a header, the designer should consider the reaction effects of combinations of valves opening.

Whilst it is possible to incorporate the valve opening characteristics into a mathematical model of the valve and discharge piping to determine the reaction force, it is generally satisfactory for atmospheric discharge to carry out a simple static analysis for the steady state and to apply a dynamic load factor.

A.2.4.2 Simple static analysis

The sustained reaction force F_r at the discharge into a vent pipe or atmosphere may be calculated as follows:

$$F_r = R\, v_e + (p_e - p_a)A \tag{A.2.4-1}$$

where

p_e is the calculated pressure at the exit point;

p_a is the atmospheric pressure;

A is the outlet flow area;

R is the relief mass flow rate;

v_e is the calculated exit velocity.

The initial dynamic force F_{dr} should be calculated as:

$$F_{dr} = F_r\ DLF \tag{A.2.4-2}$$

where

DLF is the dynamic factor load.

The dynamic load factor DLF should be related to the opening time of the valve, and may be determined by first calculating a safety valve period T as follows:

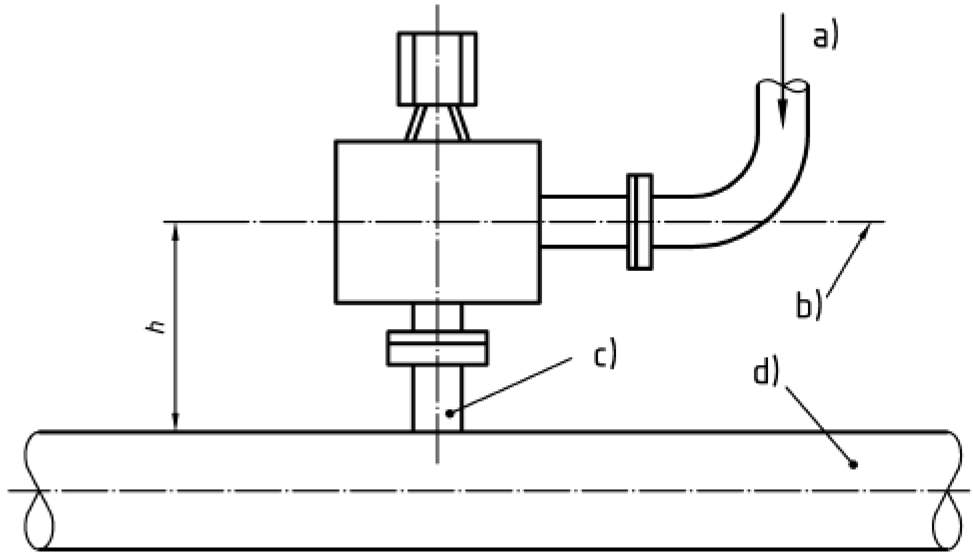
$$T = \frac{2\pi}{\sqrt{3}}\sqrt{\frac{W\ h^3}{E\ I}} \tag{A.2.4-3}$$

where

- h is the distance in mm from the surface of the main run pipe to the centre line of the outlet pipe, see Figure A.2.4-1;
- I is the second moment of area of the inlet pipe;
- W is the mass of the safety valve assembly, flanges etc.

The ratio of the safety valve opening time (from fully closed to fully open) to the calculated safety valve period should be use in conjunction with Figure A.2.4-2 to determine the dynamic load factor.

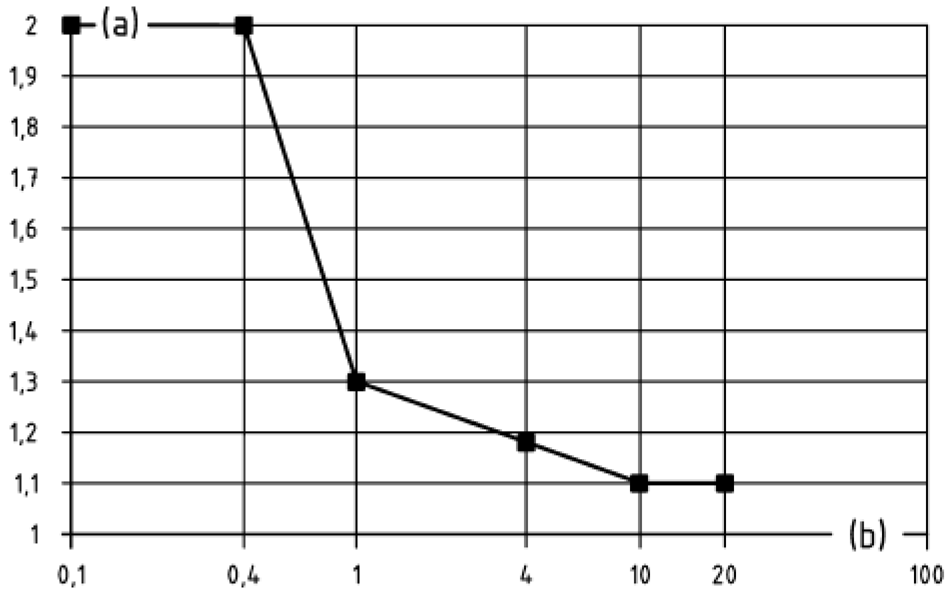
Alternative dynamic load factors may be used when supported by satisfactory experimental data.



Key

a) Reaction load
b) Centre line of valve outlet
c) Valve inlet pipe nozzle
d) Main run pipe

Figure A.2.4-1 — Typical arrangement of safety valve discharge



Key

(a) Dynamic load factor

(b) Ratio of valve opening time/valve period

Figure A.2.4-2 — Dynamic load factor

A.2.5 Allowable stresses

A.2.5.1 Pressure peaks occurring during dynamic events should be covered by the pressure design requirements in clauses 6 to 10.

A.2.5.2 For simple and quasi static methods the stress limitations of 12.3 should be used for primary and for secondary stresses.

A.2.5.3 For detailed dynamic analysis methods the primary stresses should be limited to the minimum material yield strength at the operating temperature, and primary plus secondary stresses should be limited to 2 times the minimum yield strength at the operating temperature.

A.3 Alternative means of design verification

A.3.1 Comparative studies

The design of a piping system may be verified by comparison to an existing arrangement which has been shown by calculation, testing, or operational events to be capable of withstanding the proposed design conditions. When using this approach, care should be taken to ensure that the comparator is similar in all significant details. In particular, the designer should consider the shape of the piping, the input excitations, the mechanical connections and the pressure, temperature and flow.

A.3.2 Full scale testing

Full scale testing of piping can be carried out to verify the design. Whilst this approach may not be practical for large piping systems, testing after installation or under laboratory conditions may be considered where serial production of the design is envisaged.

The designer should ensure that the conditions of test match those predicted for the design. Where vibrations of the system are artificially imposed these should include all significant frequencies and amplitudes.

Where the testing is carried out on components or sections of a piping system, the designer should ensure that the validity of the test is not adversely affected by differences between the end conditions for the test and the installation.

A.3.3 Reduced scale testing

Reduced scale testing of piping may be carried out to identify modes of behaviour for the system or to check the validity of the analytical models used in design calculations.

The model scale should be not less than 1/10 and the rules used for conforming similarity should be well defined. The characteristics of the imposed vibrations should be suitably scaled to ensure the validity of the piping response data.

Annex B
(normative)

More accurate calculation of bends and elbows

B.1 General

This Annex specifies a less conservative method for calculation of pipe bends and elbows compared with 6.2.3.1.

NOTE These calculation rules take into account [1] and [2] that upon applying internal pressure to a pipe bend, higher stresses occur on the inside of the bend (and lower stresses on the outside of the bend) than on a straight pipe with identical wall thickness.

B.2 Symbols and units

For the purposes of this Annex, the symbols given in Table B.2-1 shall apply in addition to those given in 3.2.

Table B.2-1 — Additional symbols for the purposes of this Annex

Symbol	Description	Unit
B	design coefficient for the determination of the wall thickness of elbows with uniform wall thickness	-
B_{ext}	design coefficient for the determination of the wall thickness on the extrados of the pipe bend and elbows	-
B_{int}	design coefficient for the determination of the wall thickness on the intrados of the pipe bend and elbows	-
e	minimum required wall thickness of straight pipes calculated in accordance with 6.2.1.	mm
$e_{a\ ext}$	analysis wall thickness on the extrados of the pipe bend and elbow	mm
$e_{a\ int}$	analysis wall thickness on the intrados of the pipe bend and elbow	mm
e_{ext}	minimum required wall thickness without allowances and tolerances on the extrados of the pipe bend and elbow	mm
e_{int}	minimum required wall thickness without allowances and tolerances on the intrados of the pipe bend and elbow	mm
$e_{ord,ext}$	ordered wall thickness on the extrados of the pipe bend and elbows	mm
$e_{ord,int}$	ordered wall thickness on the intrados of the pipe bend and elbow	mm
$e_{r\ ext}$	minimum required wall thickness with allowances and tolerances on the extrados of the pipe bend and elbow	mm
$e_{r\ int}$	minimum required wall thickness with allowances and tolerances on the intrados of the pipe bend and elbow	mm
$f_{m\ ext}$	mean stress on the extrados of the bend	MPa (N/mm²)
$f_{m\ int}$	mean stress on the intrados of the bend	MPa (N/mm²)
r	curvature radii of the pipe bend as defined in Figure B.2-1	mm

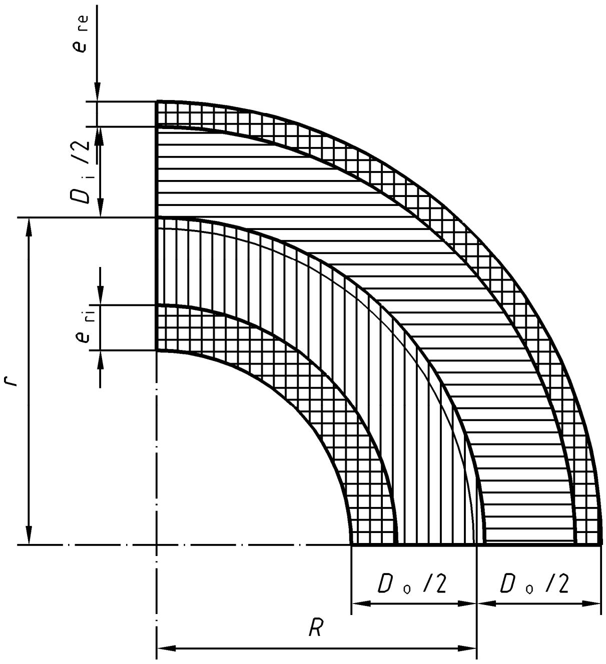


Figure B.2-1 — Notations used for pipe bends

B.3 Required wall thickness

The minimum required wall thickness with allowances and tolerances shall be:

— on the intrados of the bend

$$e_{\mathrm{r\ int}} = e_{\mathrm{int}} + c_0 + c_1 + c_2 \tag{B.3-1}$$

— on the extrados of the bend

$$e_{\mathrm{r\ ext}} = e_{\mathrm{ext}} + c_0 + c_1 + c_2 \tag{B.3-2}$$

For stress calculation of finished tube bends with the ordered wall thickness $e_{\mathrm{ord,int}}$ or $e_{\mathrm{ord,ext}}$ respectively, the following equations shall be used for calculation of the analysis wall thickness:

— on the intrados of the bend

$$e_{\mathrm{a\ int}} = e_{\mathrm{ord\ int}} - c_0 - c_1 - c_2 \tag{B.3-3}$$

— on the extrados of the bend

$$e_{\mathrm{a\ ext}} = e_{\mathrm{ord\ ext}} - c_0 - c_1 - c_2 \tag{B.3-4}$$

NOTE Bevels at the end on bends and elbows need not be considered in the calculation.

B.4 Calculation

B.4.1 Calculation of wall thickness

B.4.1.1 Wall thickness of the intrados of the bend

The minimum required wall thickness of the intrados of the bend without allowances and tolerances shall be calculated as:

$$e_{\text{int}} = e\,B_{\text{int}} \tag{B.4.1-1}$$

where

B_{int} is given by one of the following

— for bends with specified inside diameter

$$B_{\text{int}} = \frac{r}{e} - \frac{D_{\text{i}}}{2e} - \sqrt{\left(\frac{r}{e} - \frac{D_{\text{i}}}{2e}\right)^2 - 2\frac{r}{e} + \frac{D_{\text{i}}}{2e}} \tag{B.4.1-2}$$

NOTE The coefficient B_{int} as a function of r/D_{i} may be taken from Figure B.4.1-1.

— for bends with specified outside diameter

$$B_{\text{int}} = \frac{D_{\text{o}}}{2e} + \frac{r}{e} - \left(\frac{D_{\text{o}}}{2e} + \frac{r}{e} - 1\right)\sqrt{\frac{\left(\frac{r}{e}\right)^2 - \left(\frac{D_{\text{o}}}{2e}\right)^2}{\left(\frac{r}{e}\right)^2 - \frac{D_{\text{o}}}{2e}\left(\frac{D_{\text{o}}}{2e} - 1\right)}} \tag{B.4.1-3}$$

NOTE The coefficient B_{int} as a function of r/D_{o} may be taken from Figure B.4.1-2.

r/e shall be calculated from

$$\frac{r}{e} = \sqrt{\frac{1}{2}\left\{\left(\frac{D_{\text{o}}}{2e}\right)^2 + \left(\frac{R}{e}\right)^2\right\} + \sqrt{\frac{1}{4}\left(\left(\frac{D_{\text{o}}}{2e}\right)^2 + \left(\frac{R}{e}\right)^2\right)^2 - \frac{D_{\text{o}}}{2e}\left(\frac{D_{\text{o}}}{2e} - 1\right)\left(\frac{R}{e}\right)^2}} \tag{B.4.1-4}$$

The equations (B.4.1-2) and (B.4.1-3) will only produce identical results if

$$D_{\text{o}} = D_{\text{i}} + e_{\text{int}} + e_{\text{ext}} \tag{B.4.1-5}$$

and

$$R = r - \frac{e_{\text{int}} - e_{\text{ext}}}{2} \tag{B.4.1-6}$$

B.4.1.2 Wall thickness of the extrados of the bend

The minimum required wall thickness of the extrados of the bend without allowances and tolerances shall be calculated as:

$$e_{\text{ext}} = e\, B_{\text{ext}} \tag{B.4.1-7}$$

where

B_{ext} is given by one of the following:

— for bends with specified inside diameter

$$B_{\text{ext}} = \sqrt{\left(\frac{r}{e} + \frac{D_{\text{i}}}{2e}\right)^2 + 2\frac{r}{e} + \frac{D_{\text{i}}}{2e}} - \frac{D_{\text{i}}}{2e} - \frac{r}{e} \tag{B.4.1-8}$$

NOTE The coefficient B_{ext} as a function of r/D_{i} may be taken from Figure B.4.1-1.

— for bends with specified outside diameter

$$B_{\text{ext}} = \frac{D_{\text{o}}}{2e} - \frac{r}{e} - \left(\frac{D_{\text{o}}}{2e} - \frac{r}{e} - 1\right)\sqrt{\frac{\left(\frac{r}{e}\right)^2 - \left(\frac{D_{\text{o}}}{2e}\right)^2}{\left(\frac{r}{e}\right)^2 - \frac{D_{\text{o}}}{2e}\left(\frac{D_{\text{o}}}{2e} - 1\right)}} \tag{B.4.1-9}$$

NOTE The coefficient B_{ext} as a function of r/D_{o} may be taken from Figure B.4.1-2.

r/*e* shall be calculated from equation (B.4.1-4).

The equations (B.4.1-8) and (B.4.1-9) will only produce identical results if equations (B.4.1-5) and (B.4.1-6) are fulfilled.

B.4.1.3 Elbows with uniform wall thickness

For elbows with uniform wall thickness of the extrados and the intrados, the minimum required wall thickness shall be calculated as :

$$e_{\text{int}} = e_{\text{ext}} = e\, B \tag{B.4.1-10}$$

where B is given by one of the following:

— for elbows with specified internal diameter, the coefficient $B = B_{\text{int}}$ shall be in accordance with equation (B.4.1-2)

NOTE The coefficient B as function of R/D_{i} may be taken from Figure B.4.1-1.

— for elbows with specified outside diameter

$$B = \frac{D_{\text{o}}}{2e} - \frac{R}{e} + \sqrt{\left(\frac{D_{\text{o}}}{2e} - \frac{R}{e}\right)^2 + 2\frac{R}{e} - \frac{D_{\text{o}}}{2e}} \tag{B.4.1-11}$$

NOTE The coefficient B as function of R/D_{o} may be taken from Figure B.4.1-3.

The equation (B.4.1-2) in conjunction with equation (B.4.1-10) will only produce results identical to equation (B.4.1-11) if

$$D_\text{o} = D_\text{i} + 2e_\text{int} \tag{B.4.1-12}$$

and

$$R = r \tag{B.4.1-13}$$

B.4.2 Stress calculation

B.4.2.1 The mean stress for the intrados of the bend shall be:

— for bends with specified inside diameter:

$$f_\text{m int} = \left(\frac{p_\text{c}\ D_\text{i}}{2\,z\,e_\text{a int}} \quad \frac{2\,r - 0{,}5\,D_\text{i}}{2\,r - D_\text{i} - e_\text{a int}} \right) + \frac{p_\text{c}}{2} \leq f \tag{B.4.2-1}$$

— for bends with specified outside diameter:

$$f_\text{m int} = \left\{ \frac{p_\text{c}\left(D_\text{o} - e_\text{a int} - e_\text{a ext}\right)}{2\ z\ e_\text{a int}} \quad \frac{2\,R - 0{,}5\,D_\text{o} + 1{,}5\,e_\text{a int} - 0{,}5\,e_\text{a ext}}{2\,R - D_\text{o} + e_\text{a int}} \right\} + \frac{p_\text{c}}{2} \leq f \tag{B.4.2-2}$$

B.4.2.2 The mean stress for the extrados of the bend shall be:

— for bends with specified inside diameter:

$$f_\text{m ext} = \left(\frac{p_\text{c}\,D_\text{i}}{2\,z\,e_\text{a ext}} \quad \frac{2\,r + 0{,}5\,D_\text{i}}{2\,r + D_\text{i} + e_\text{a ext}} \right) + \frac{p_\text{c}}{2} \leq f \tag{B.4.2-3}$$

— for bends with specified outside diameter:

$$f_\text{m ext} = \left\{ \frac{p_\text{c}\left(D_\text{o} - e_\text{a int} - e_\text{a ext}\right)}{2\ z\ e_\text{a ext}} \ \frac{2\,R + 0{,}5\,D_\text{o} + 0{,}5\,e_\text{a int} - 1{,}5\,e_\text{a ext}}{2\,R + D_\text{o} - e_\text{a ext}} \right\} + \frac{p_\text{c}}{2} \leq f \tag{B.4.2-4}$$

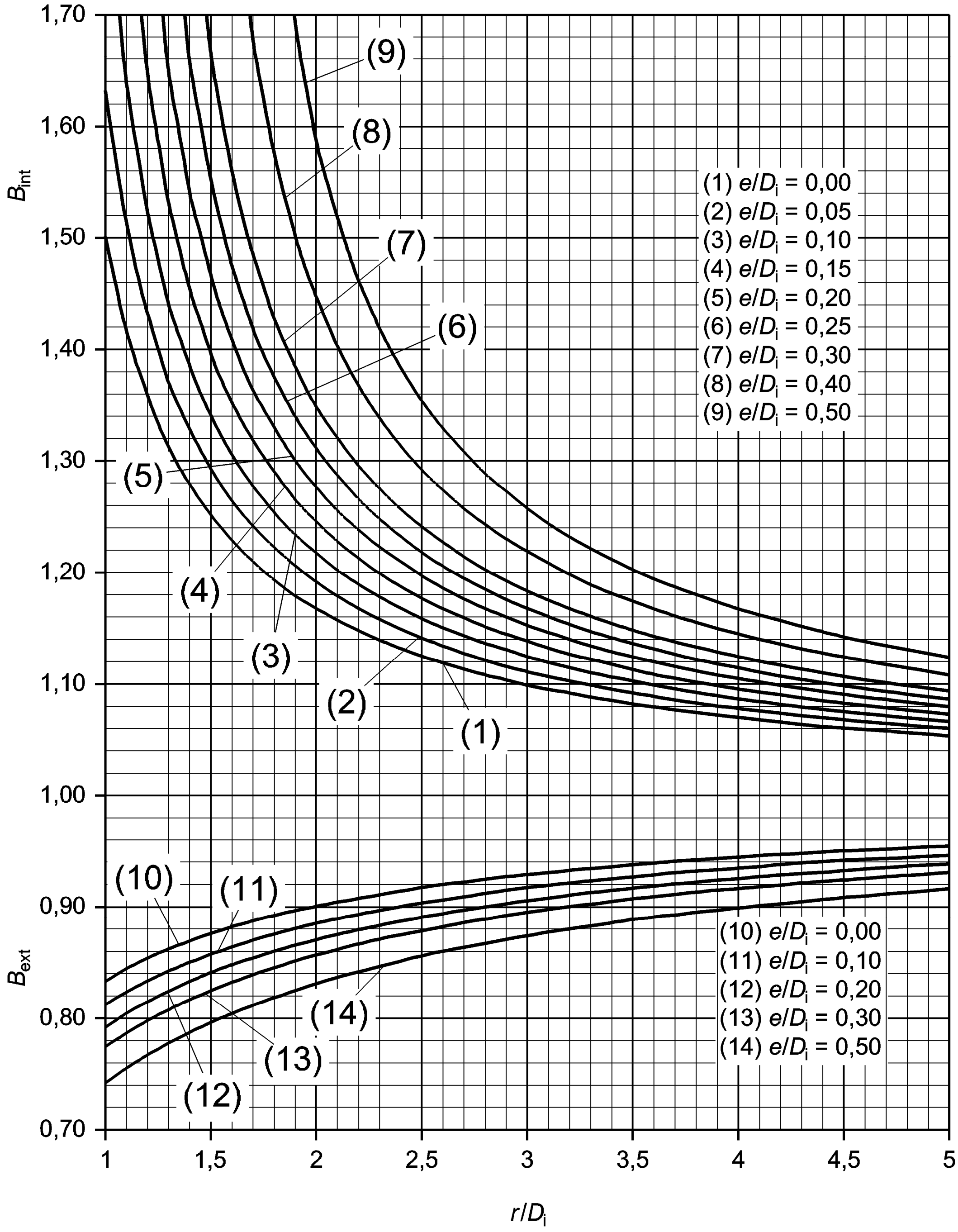


Figure B.4.1-1 — Design coefficients B_{int} and B_{ext} for bends with specified inside diameter in accordance with equation (B.4.1-2) for B_{int} and (B.4.1-8) for B_{ext}

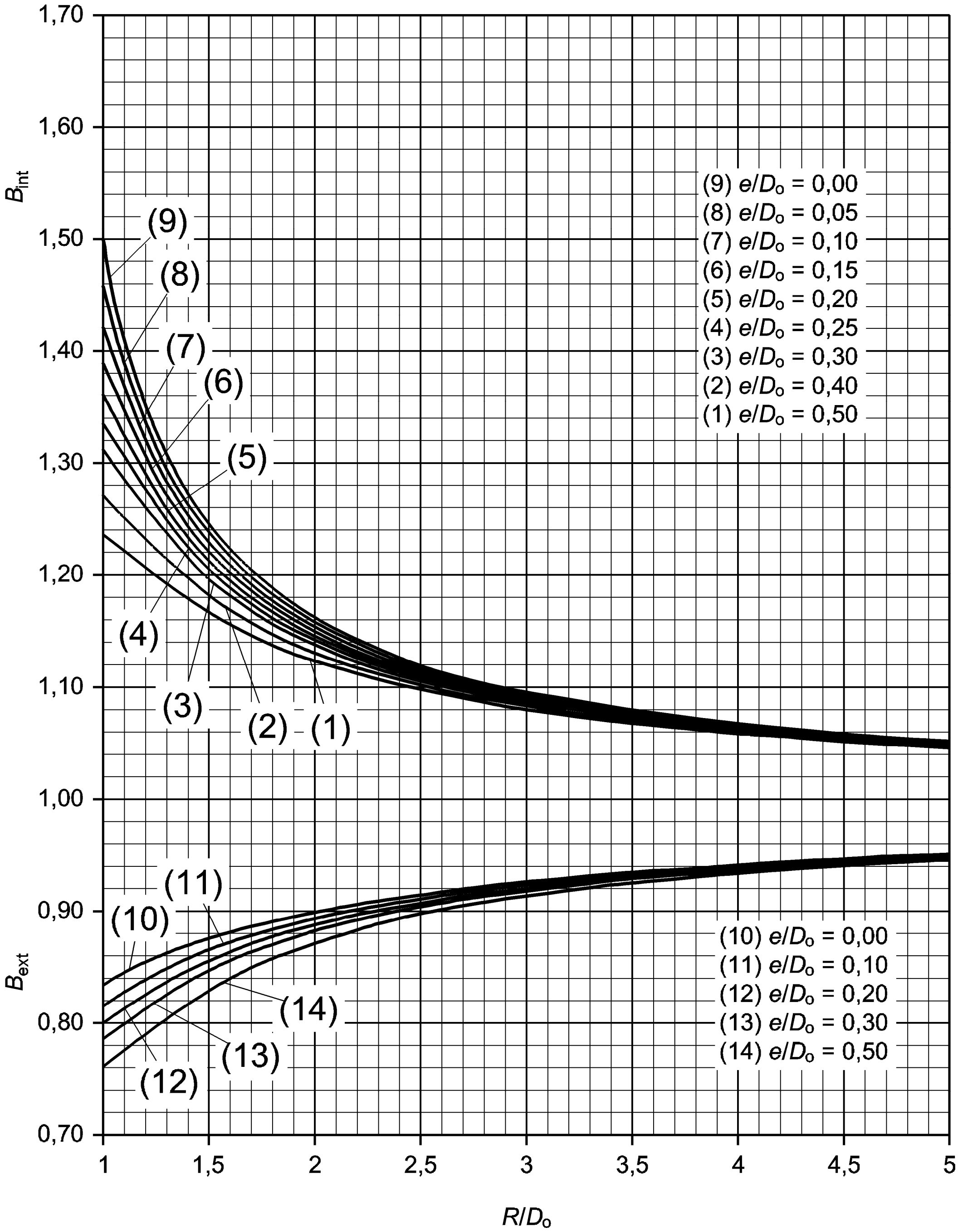


Figure B.4.1-2 — Design coefficients B_{int} and B_{ext} for bends with specified outside diameter in accordance with equation (B.4.1-3) for B_{int} and (B.4.1-9) for B_{ext}

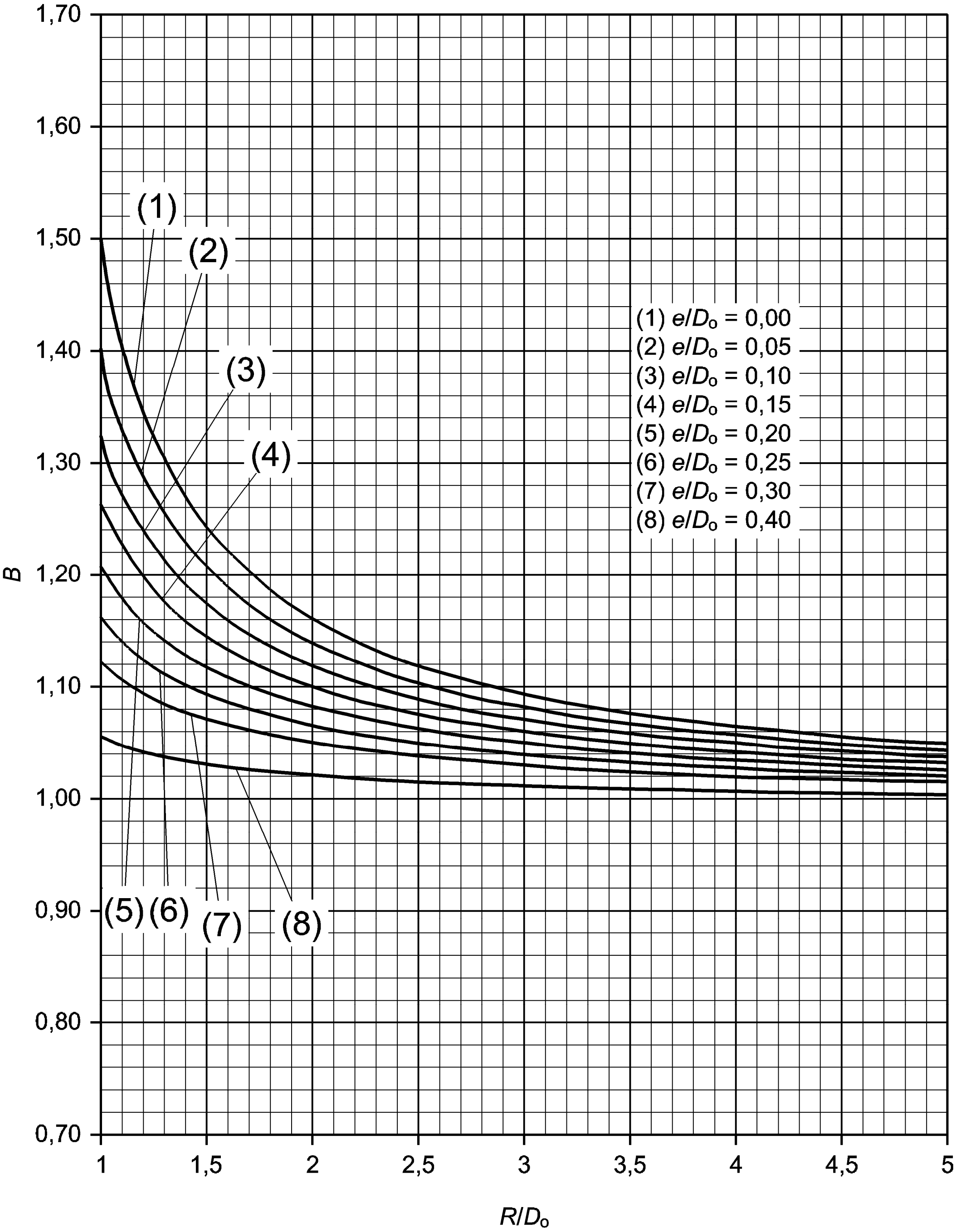


Figure B.4.1-3 — Design coefficients *B* for elbows of equal wall thickness ($e_{int} = e_{ext}$) with specified outside diameter

Annex C
(informative)

Expansion joints

C.1 Incorporation of expansion joints into piping systems

C.1.1 General

Movement within piping systems may be accommodated using the natural flexibility of the arrangement using the rules set out in clause 12. In addition, special devices such as expansion joints, hose assemblies etc. can be incorporated into the layout to withstand internal or external pressure and compensate for movement by providing the necessary flexibility to enable the system as a whole to meet the design requirements.

Expansion joints are commonly in the form of bellows with one or more convolutions which can extend or compress axially, the ends can be rotated angularly and the ends can be displaced laterally relative to each other.

The flexible convolutions are formed from a material which may be metal, rubber or similar elastomeric compounds. Metallic convolutions can be single or multiple ply, and a variety of elastomers and associated fabric constructions provide a range of "rubber" products. For the design of expansion joints, see EN 14917.

Movements can arise from thermal expansion, settlement or other terminal point movement, vibration and other externally imposed loading. Bellows expansion joints can be designed to accommodate axial, angular or lateral movement of the connected components, and arrangements are available to control the relative movements within safe limits, and to carry the axial pressure load, i.e. pressure thrust.

Several expansion joints at a single location may be needed to take up movements in piping systems.

Metallic bellows expansion joints can be designed to withstand high temperatures and pressures. Fatigue should to be considered.

Elastomeric expansion joints do not have such high temperature and pressure resistance.

Fatigue failure is unlikely but ageing limits life time.

Expansion joints can be fitted with sleeves or convolution fillers for abrasive fluids or where high fluid velocity could cause vibration problems, or with liners against corrosion. The incorporation of such devices can affect the performance of the expansion joint and the manufacturers specific advice should be sought in all cases.

A wide variety of special features can be added to expansion joints, for more detailed information see ISO 15348.

C.1.2 Types of expansion joints

C.1.2.1 General

Expansion joints can be made in many configurations to absorb different types of movements, axial, angular, lateral, or to restrain the pressure thrust. The most common expansion joints are described below.

C.1.2.2 Axial expansion joints

This type of expansion joint is fitted into a straight line to accommodate mainly axial movement by compression or extension of the bellows. Axial expansion joints are unable to resist the axial loads generated by the pressure of the fluid in the pipes, and some form of anchorage should be provided elsewhere in the system to counteract this pressure thrust. Several axial expansion joints may be needed in a long line or for large movements, and each must be separated from its neighbour by an intermediate anchor.

C.1.2.3 Angular expansion joints

This type of expansion joint is designed to absorb angular movement. When fitted with hinges, it allows movement in a single plane. When fitted with gimbal rings, it allows movement in any plane. It restrains pressure thrust.

Usually a system of at least two angular expansion joints is necessary to absorb the movements of pipe systems.

C.1.2.4 Lateral expansion joints

This type of expansion joint is designed to absorb lateral movement.

The ends are tied together by restraining parts consisting of at least two rigid rods (often fitted with spherical washers) or a pair of tie bars with hinge pins. The pressure thrust is taken by these restraining parts.

C.1.2.5 Universal expansion joints

This type of expansion joint is designed to permit any combination of movement mainly with two bellows joined by a short length of pipe. It does not restrain pressure thrust.

C.1.2.6 Pressure balanced expansion joints

This type of expansion joint is designed to accommodate axial and/or lateral movement and in addition

counteract the pressure thrust.

C.1.3 Design of expansion joints

Although the operating principles are universal, expansion joints are generally manufactured to proprietary designs. Consequently the characteristics of these components are specific to each manufacturer who should be consulted for detailed engineering information where required.

The key element is the design of the bellows, which should utilize the established code of calculation according to this Annex, or follow another approved method of calculation or should be verified by experimental demonstration. Structural components should be designed to carry all foreseeable loads using European Standard or other approved standard.

The flexibility of the bellows component increases with more convolutions, as each deflects less for a given overall movement. Where the design pressure is high the thickness of the bellows wall needs to be large and more convolutions are needed or multi-ply design is used to provide modest flexibility.

A secondary effect of using multi-ply construction is that the convolutions are smaller for a given flexibility and the reduced effective cross sectional area results in a lower pressure thrust.

C.1.4 Designing with expansion joints

C.1.4.1 General

The use of expansion joints is not a substitute for design analysis. Expansion joints should be treated as elements within the piping, and the designer should consider all the loadings likely to arise under the design conditions and ensure that the piping system behaves in an acceptable, predictable and controlled manner at all times.

C.1.4.2 Location of expansion joints

The piping designer should consider the use of expansion joints to provide a practical or an economical means of achieving compliance with the requirements of this standard where the calculated displacement and forces at points in the system cannot be readily accommodated by the natural flexibility of the piping alone, or where unacceptable loadings would otherwise be generated on connected equipment.

In general, expansion joints should be located where the pipe movements are simple to minimize complexity and cost. The system should be divided into sections, those requiring expansion joints and those where natural flexibility is adequate.

Bellows expansion joints tend to become instable when subjected to torsional loading beyond the allowable limits, and the designer should locate the units accordingly or take other measures to limit torsion.

The type of expansion joints used will depend on the size and direction of the pipe run in a system, the movements to be accommodated, and the working conditions. The amount of pressure thrust limits the use of axial expansion joints.

Expansion joints are generally arranged to encourage free axial movement of long pipe runs, with the movement of shorter connecting offsets and branches being controlled by natural flexibility of the pipe or by expansion joints selected according to the principal form of imposed movement.

In systems with acceptable pressure thrust, main run movement may be absorbed by one or more axial expansion joints with anchors at changes of direction. Expansion or movement of connecting runs may need to be accommodated by additional units reacting also against these anchors.

Alternatively, when pressure thrust is to high to be carried by the pipe anchors, main run movement may be accommodated by permitting lateral movement in offset legs through the use of restrained expansion joints.

If pipe movements are large, the use of restrained expansion joints in offset legs may also be required. One lateral or two angular expansion joints placed in the offset leg may permit significant lateral displacement of the leg. This displacement causes rotation of the intermediate pipe of the leg which results in lateral deflection of the main run pipes. The amount of deflection and loading on guides depends upon the geometry and design of the particular unit and the specific manufacturers data should be used to design proper supporting and guiding of the pipes.

A system of three angular expansion joints should be used, when main run movement is large and the offset leg is short. The third expansion joint than avoid deflection of the main run pipe.

C.1.4.3 Anchors

Because axial or universal expansion joints operate by reacting against pipe anchors, the system should be sectionalised to provide anchors to isolate each section containing an expansion joint. The designer should ensure that suitable structural points are available to carry the anchor load, which may reach unacceptable values at normal pressures and usual pipe diameters.

Unless the pressure load is carried by restraining parts within the expansion joint, the anchors should be designed to carry the pressure thrust on the effective area of the bellows, plus the bellows spring reaction load. In addition the friction forces generated within guides or at partial anchors should be considered. As the pressure thrust depends on the pipe diameter, intermediate anchors are needed at size changes in compensated runs. These anchors are subject to the differential pressure thrust. Similar requirements apply to points of pressure reduction (or increase).

In carrying out the design of anchors, consideration should be given to the directions in which movement is to be prevented. Partial anchors can be necessary to hold axial forces in a main run but accommodate lateral motion in an offset or branch.

C.1.4.4 Guides

Expansion joints are generally designed to operate in specific directions or planes. Consequently the designer should incorporate pipe guides into the supporting arrangement to control movement at the connection between pipe and expansion joint. It is usual to fit guide arrangements close to the expansion joint at about 3 to 4 diameters distance, with further guides along the main run to prevent lever moments.

Where expansion joints act by angular or lateral movement there will be a change to the effective length of the leg containing such units causing deflection of the main run. The pipe guidance needs to take this into account and partial or planar guides will be required with clearances to accommodate the anticipated motion.

Systems incorporating expansion joints rely on the proper action of these devices and the design intent can be destroyed if guide friction is excessive. Great care should therefore be given to the design or selection of guides, their installation and maintenance.

The designer should consider the effects of differential movements in the structures or plant to which supports and guides are fixed.

The loads acting on guides close to expansion joints can be high, and the designer should consider the use of two or more pairs of roller supports as a guide rather than plates or U bolts.

C.1.5 Analyses and calculation

Extension/compression, angular and lateral movement of expansion joints requires the action of a force and stores energy to restore equilibrium when the imposed loading is removed. In this respect they behave like springs, the spring rate being determined by type of expansion joint and the individual manufacturer. Each mode of action has its own spring effect, and the manufacturer should specify the data in the form of a force rate per mm of axial and lateral movement and per degree of rotation as appropriate.

In addition, angular movement of the bellows due to the operation of the expansion joint can generate a bending moment. The size of this loading depends on the design of the unit and the manufacturer’s data should be used when considering the significance if this factor.

The flexibility model used to represent the forces generated at the expansion joint should take into account the requirements and capabilities of the analytical programme and the level of accuracy warranted by the circumstances.

The designer may treat the expansion joint as a single element replaced by one or more equivalent springs. Greater confidence will be obtained if each bellows in a multi bellows unit is treated individually and separately from any intermediate pipe length. A flexibility matrix will be required for each bellows element, and whilst these are generally identical this may not be the case.

It should be noted that where an expansion joint contains two identical bellows in a symmetrical arrangement, each should be modelled as having twice the spring rate of the unit as a whole.

Where the movements are infrequent, it may be found that elastomeric materials acquire a „set“ at the sustained operating length. Further movement may need to be treated as commencing from a relaxed condition.

It is generally sufficient to model tie rods as preventing over the length of the expansion joint, and ignore the effects of friction in the movement of the unit. Where are more detailed calculation is considered necessary, the designer should review the results for each element of the analyses to ensure compatibility with the behaviour of the expansion joint as a whole.

Lateral displacements expansion joints will generate deflections in the adjacent piping which may be in plane and out of plane. These deflections should also be included in the analyses of the piping system, as should loads (including friction) at anchors and guides.

It is usual for piping designers to incorporate „typical“ equivalent spring data into trial analyses to access the effectiveness of proposed expansion joint configurations, and to check that all parts of the system including hinges, rods and supports can carry the loads likely to be imposed.

However, the final analyses should be carried out using data specific to the supplier and type actually being installed.

C.1.6 Cold pull

Because the action of a bellows expansion joint generates reaction loads proportional to the deflection from the neutral position, cold pull can be useful to minimize loading on sensitive terminal connections. Pre-setting expansion joints by half the predicted displacement will reduce the reaction load at the operating condition.

If the analyses depend on this procedure, the manufacturer of the piping should be notified.

C.2 Maximum spacing for unrestrained axially compensated straight runs

C.2.1 General

Straight pipe runs when compensated by unrestrained axial or universal expansion joints tend to buckle under the influence of internal pressure and/or axial compressing forces. This behaviour is similar to that of a straight bar objected to an axial compressing force. Buckling can be avoided, when the pipe is guided properly. Calculation rules for maximum allowable spacing of the guides are given below.

Proper weight support is unaffected by the rule and should be considered.

C.2.2 Calculation rules

Maximum allowable spacing L_G for pipe lines compensated with unrestrained expansion joints at any conditions:

$$L_{\mathrm{G}} = \frac{\pi}{\beta} \sqrt{\frac{\mathrm{E\ J}}{\mathrm{F_i\ S}}} \qquad \text{(C2.2-1)}$$

where

β is the guiding factor for the pipe line segment under consideration:

β = 1,0 for both sides simply supported;

β = 0,7 for one side simply supported, other side axially guided;

β = 0,5 for both sides axially guided.

For the guiding conditions see also Figures C.2.2-1 to C.2.2-3.

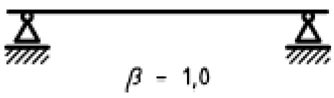


Figure C.2.2-1

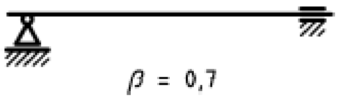


Figure C.2.2-2

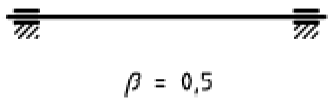


Figure C.2.2-3

E is the modulus of elasticity

J is the moment of inertia of the pipes cross section:

$$J = \frac{\pi}{8}\, e\, D_{mp}^3 \tag{C2.2-2}$$

where D_{mp} is the mean diameter of the pipe and e its wall thickness;

S is the safety factor (recommended : S = 3);

F_i is the buckling force consisting of the following components which may act on the pipe simultaneously:

$$F_i = F_p + F_B + F_F \tag{C2.2-3}$$

where

— the pressure thrust:

$$F_p = p \cdot a \tag{C2.2-4}$$

(bellows effective area a is normally supplied by the manufacturer)

— the axial displacement force of the bellows:

$$F_B = \pm\, x \cdot K_B \tag{C2.2-5}$$

(axial displacement x of the expansion joint is starting from the neutral position and positive for compression; bellows rigidity K_B is supplied by the manufacturer)

— the friction force of the pipe guides:

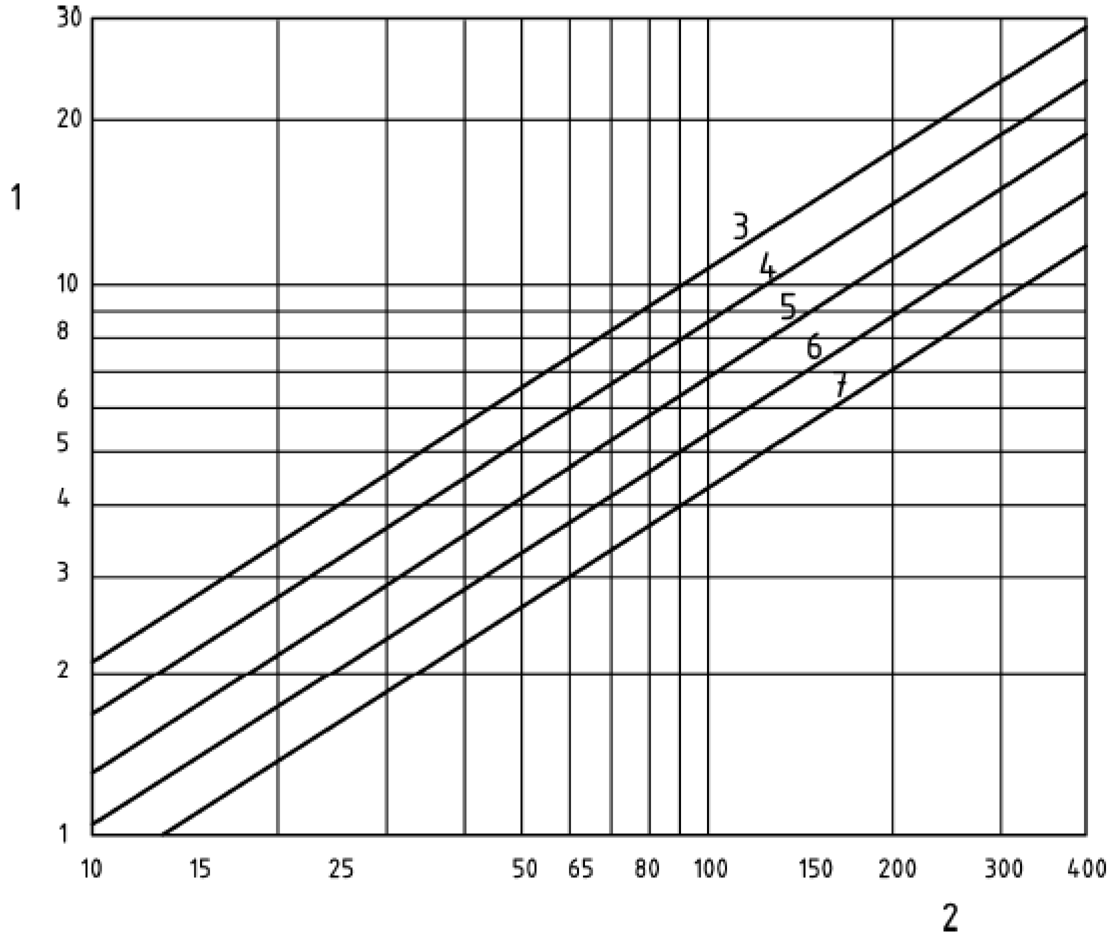
$$F_F = \pm \sum \mu \cdot F_N \tag{C2.2-6}$$

All the singular forces of every guide between expansion joint and anchor of a pipe line segment should be considered in order to calculate the friction force. They will arise when temperature changes and will be positive when temperature is increased (the friction coefficient μ in the guides is supplied by the support manufacturer and the vertical supporting force F_N at the supports by the system analyst).

C.2.3 Maximum spacing for defined conditions

The maximum spacing of unrestrained axially compensated straight runs operating under defined usual conditions may be taken from Figure C.2.3-1, which is based on the following defined conditions:

- Pipe simply supported in the guides;
- Pipe of steel (E = 210 000 MPa (N/mm²));
- Outside diameter D_a and standard wall thickness e_n (welded pipes);
- PN = p_c, p_{test} = 1,43 p_c is considered;
- axial expansion joint in neutral position during pressure test;
- Safety factor S = 3.



Key

1 Maximum spacing L_G in m

2 Nominal diameter DN

3 PN 6 : $L_G = 0{,}407{,}\times DN^{0{,}71}$

4 PN 10 : $L_G = 0{,}324{,}\times DN^{0{,}71}$

5 PN 16 : $L_G = 0{,}257{,}\times DN^{0{,}71}$

6 PN 25 : $L_G = 0{,}204{,}\times DN^{0{,}71}$

7 PN 40 : $L_G = 0{,}162{,}\times DN^{0{,}71}$

Figure C.2.3-1

C.3 Indication for the design of expansion joints

C.3.1 General

The detailed design of expansion joints should be carried out by the manufacturer of expansion joint on the basis of the design data according to 6.5.

C.3.2 Design data, Symbols

For the design of an expansion joint the symbols given in Table C.3.2-1 should apply. Further information can be taken from Figures C.3.2-1 to C.3.2-3.

Table C.3.2-1—Additional Symbols for this Annex

Symbol	Description	Unit
D	accumulative fatigue damage	—
D_m	mean bellows diameter	mm
DN	nominal diameter	—
F	reaction force	N
K	axial rigidity of convolution	N/mm
L_u	overall length of convoluted part, incl. intermediate pipe	mm
l	convolution length	mm
l^*	bellows centre distance	mm
Δl	equivalent axial movement	mm
M	reaction moment	Nm
N	number of load cycles	—
n	number of convolutions	—
x	total axial movement	mm
y	total lateral movement	mm
θ	total angular rotation	degree

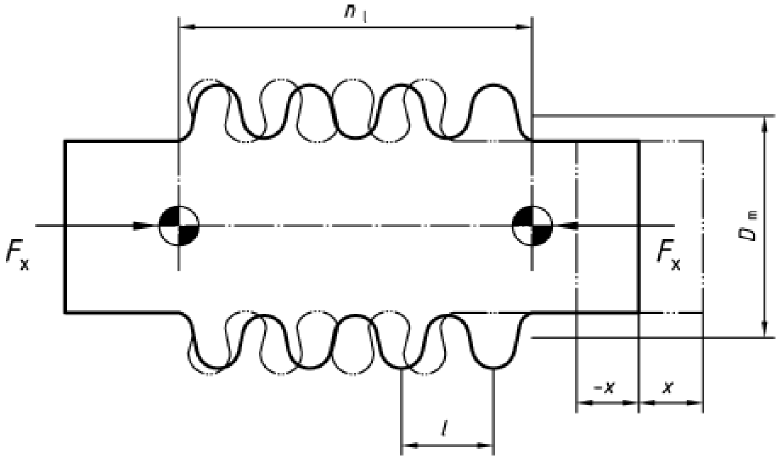


Figure C.3.2-1 — Bellows subjected to axial displacement *x*

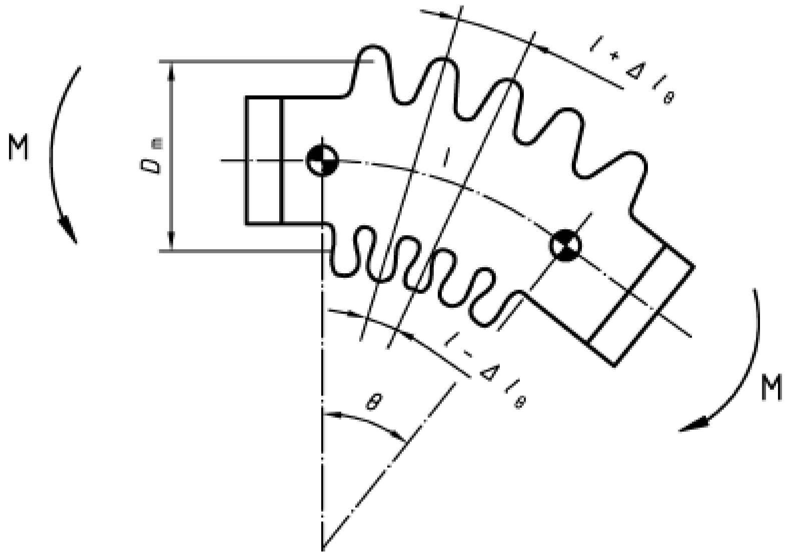


Figure C.3.2-2 — Bellows subjected to angular rotations θ

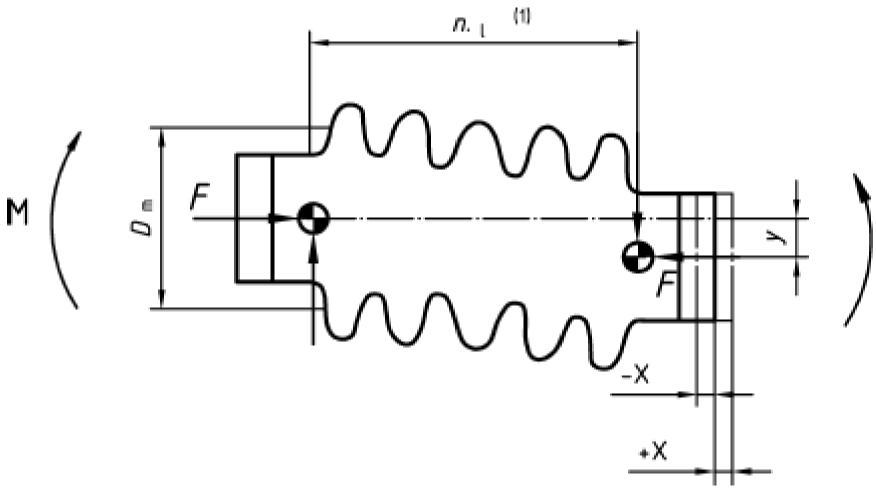


Figure C.3.2-3 — Bellows subjected to lateral deflection *y*

C.3.3 Design and calculation

C.3.3.1 General

The main characteristics for the design of an expansion joint are the design conditions, defined by the values design temperature t_c and pressure p_c and by the required movement of the expansion joint.

For a number of specified operating load cases with *n* couples of data the design conditions p_c, t_c are determined by the set $p_{o,j}$, $t_{o,j}$ which causes the highest stress to the expansion joint (see 4.2.2).

In addition hereto it should be made sure that the bellows material is suitable for the maximum operating temperature $t_{o\ max}$ and that the expansion joint is capable to absorb the thermal expansion resulting from this high temperature.

The number of allowable cycles should ensure that the cumulative fatigue damage *D*, which is the sum of the individual fatigue damages of all the loading conditions (index *j)* of the *n* load cases does not exceed unity:

$$D = \sum_{j=1}^{n} \left(N_{0,j} / N_{all,j} \right) \leq 1 \qquad \text{(C.3.3-1)}$$

where

$N_{o,j}$ is the number of load cycles to be expected at the considered operating conditions (*j*)

$N_{all,j}$ is the number of allowable cycles calculated at the same conditions (*j*).

C.3.3.2 Bellows

EN 13445-3, clause 14, provides a standardized calculation code and in addition a procedure for setting up a specific design fatigue curve to be applied. This calculation code deals only with single-bellows expansion joints, however it does not specify a calculation method for lateral and universal expansion joints consisting of two bellows and an intermediate pipe according to Figure C.3.3-1.

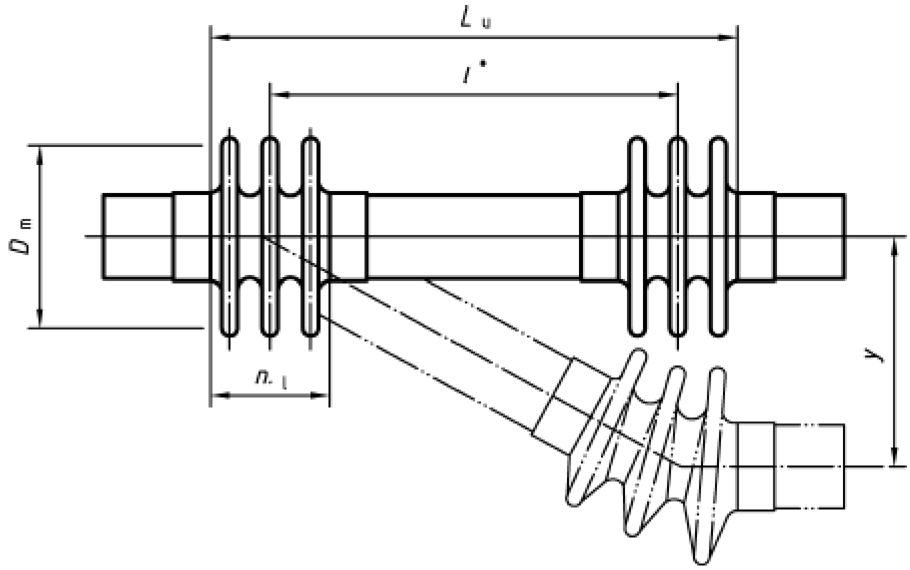


Figure C.3.3-1 — Lateral and universal expansion joint consisting of two bellows and an intermediate pipe

Additional rules for the two different designs, (a) unsupported and (b) guided intermediate pipe are given hereafter. Calculation results may also be used for further calculation according to EN 13445-3, clause 14.

a) Universal or lateral expansion joint with unsupported intermediate pipe and lateral movement *y:*

— equivalent axial movement of end convolutions:

$$\Delta l_{\text{y u}} = \frac{D_{\text{m}}}{2n} \frac{3(l\ n + l^*)}{(l\ n)^2 + 3l^{*2}} y \tag{C.3.3-2}$$

with $l^* = L_u - l\ n$

— corresponding moment from bellows rigidity at the ends of the bellows:

$$M_{\text{y u}} = \frac{D}{4} K\ \Delta l_{\text{y u}} \tag{C.3.3-3}$$

— corresponding lateral force from bellows rigidity at to the ends of the bellows:

$$F_{\text{y u}} = \frac{D_{\text{m}}}{2(l\ n + l^*)} K \cdot \Delta l_{\text{y u}} \tag{C.3.3-4}$$

NOTE The calculation of the allowable pressure to avoid columns squirm should take into account the total number of convolution of both bellows.

b) Lateral expansion joint with guided intermediate pipe and lateral movement *y:*

— equivalent axial movement of end convolutions:

$$\Delta l_{\text{y g}} = \frac{D}{2n} \frac{y}{l^*} \tag{C.3.3-5}$$

— corresponding moment from bellows rigidity at the ends of the bellows:

$$M_{\text{y g}} = \frac{D_{\text{m}}}{4} K \cdot \Delta l_{\text{y g}} \tag{C.3.3-6}$$

— corresponding lateral force from bellows rigidity at the ends of the bellows:

$$F_{\text{y g}} = \frac{D}{2l^*} K \cdot \Delta l_{\text{y g}} \tag{C.3.3-7}$$

The calculated values may be applied for further calculations accordingly.

C.3.3.3 Weld ends

The requirements of this standard for straight pipes (6.1) apply also to the dimensioning of weld ends. If there are additional forces and moments introduced by the restraining parts, they should be taken into consideration accordingly.

C.3.3.4 Restraining devices

The dimensioning of the restraining parts e.g. hinges, tie bars, gimbal rings, should comply with the requirements of this standard (see clause 13), as far as pressure parts other than main pressure bearing parts are concerned. The requirements of 13.3.2 may be applied to determine the calculation temperature of these parts.

C.3.4 Information for the system analyst

The designer of the expansion joint should provide the required information for the system analyst according to 6.5. The influence of pressure, temperature, movement or rotation on the individual values should be taken into consideration.

Annex D
(normative)

Flanges

D.1 Purpose

This Annex gives requirements for the design of circular bolted flange connections. Flanges with full face and narrow face gaskets, subject to internal and external pressure are included, as are reverse flanges and seal welded flanges. The requirements provided in this clause are based on the well established Taylor Forge rules.

D.2 Specific terms and definitions

For the purposes of this Annex the following terms and definitions given in EN 13480-1 apply.

D.2.1
assembly condition
condition applying when the gasket or joint contact surface is seated during assembly of the joint at ambient temperature and the only loading comes from the bolts

D.2.2
operating condition
condition when the hydrostatic end force due to the design pressure (internal or external) acts on the flange

D.2.3
narrow face flange
flange in which the gasket is entirely inside the circle enclosed by the bolts and there is no contact outside the bolt circle

D.2.4
full face flange
flange in which the face contact area, either direct or through a gasket or spacer, extends outside the circle enclosing the bolts

D.2.5
reverse flange
flange attached at its outside diameter to the shell

D.2.6
shell
pipe, vessel wall or other cylinder which is attached to and supports the flange

D.2.7
lap joint
flange assembly in which the bolt load is transmitted through a loose backing flange onto a stub flange

NOTE The stub flange incorporates the gasket contact face.

D.3 Specific symbols and abbreviations

The following symbols and abbreviations apply in addition to those in 3.1.

A	is the outside diameter of the flange or, where slotted holes extend to outside of flange, the diameter to bottom of slots;
A_B	is the total cross-sectional area of bolts at the section of least bolt diameter;
$A_{B,min}$	is the total required cross-sectional area of bolts;
A_2	is the outside diameter of the contact face between loose and stub flanges in a lap joint, see Figure D.5-9 (typical);
B	is inside diameter of flange;
B_2	is the inside diameter of the contact face between loose and stub flanges in a lap joint, see Figure D.5-9 (typical);
b	is the effective gasket or joint seating width;
b_0	is the basic gasket or joint seating width;
C	is the bolt pitch circle diameter;
C_F	is the bolt pitch correction factor;
D	is the inside diameter of shell;
d_b	is bolt outside diameter;
e	is the minimum flange thickness, measured at the thinnest section;
f_B	is the bolt nominal design stress at operating temperature (see D.4.3);
$f_{B,A}$	is the bolt nominal design stress at assembly temperature (see D.4.3);
f_H	is the nominal design stress of the hub – see D.5.4.2;
G	is the diameter of gasket load reaction, as given by requirements in D.5.2;
G_1	is the assumed diameter of load reaction between loose and stub flanges in a lap joint;
g_0	is the thickness of hub at small end;
g_1	is the thickness of hub at back of flange;
H	is the total hydrostatic end force;
H_D	is the hydrostatic end force applied via shell to flange;
H_G	is the compression load on gasket to ensure tight joint;
H_T	is the hydrostatic end force due to pressure on flange face;
h	is the hub length;
h_D	is the radial distance from bolt circle to circle on which H_D acts;
h_G	is the radial distance from gasket load reaction to bolt circle;
h_L	is the radial distance from bolt circle to circle on which load reaction acts for the loose flange in a lap joint;
h_T	is the radial distance from bolt circle to circle on which H_T acts;
K	is the ratio of the flange diameters – see equations D.5-21 and D.9-13;
k	is stress factor defined in D.5.4.2;
l_0	is a length parameter given by equation (D.5-22);
M	is the torsional moment exerted on the flange per unit of length, defined in D.5.4.1;
M_A	is the total moment acting upon flange for assembly condition;
M_{op}	is the total moment acting upon flange for operating condition;
m	is a gasket factor;
P	is the internal calculation pressure
P_e	is the external calculation pressure, expressed as a positive number;
W	is the design bolt load for asembly condition;
W_A	is the minimum required bolt load for assembly condition;
W_{op}	is the minimum required bolt load for operating condition;

w	is the contact width of gasket, as limited by gasket width and flange facing;
y	is the minimum gasket or joint seating pressure;
β_F	is a factor for integral method flange design as given in Figure D.5-4;
β_{FL}	is a factor for loose hubbed flanges as given in Figure D.5-7;
β_T	is a factor, given by equation (D.5-23);
β_U	is a factor, given by equation (D.5-24);
β_V	is a factor for the integral method, from Figure D.5-5;
β_{VL}	is a factor for loose hubbed flanges, from Figure D.5-8;
β_Y	is a factor, given by equation D.5-25;
δ	is the nominal gap between the shell and loose flange in a lap joint;
δ_b	is distance between centre lines of adjacent bolts;
λ	is a factor defined in D.5.4.1;
σ_b	is calculated bearing stress in a lap joint;
σ_H	is the calculated longitudinal stress in hub;
σ_r	is the calculated radial stress in flange;
σ_θ	is the calculated tangential stress in flange;
φ	is the hub stress correction factor for integral method flange design as given in Figure D.5-6.

D.4 General

D.4.1 Introduction

Circular bolted flanged connections, either sealed with a gasket or seal welded, used in the construction of vessels to this European Standard shall conform to either:

— an appropriate European Standard for pipework flanges, and the requirements of D.4.2, or

— the requirements for bolted flanged connections specified in this clause.

Alternative rules for bolted flanges connections are given in Annex P.

Both flanges of a mating pair shall be designed to the same standard or set of requirements. This applies when one of the pair is a bolted flat end or cover. The requirements for bolted flat ends in Clause 10 and bolted domed ends in Clause 12 are considered part of the same set of requirements as this clause.

Flanges made from plate are allowed if there is a safeguard regarding lamellar tearing (reduction of area within the thickness), that means ductility requirements in thickness direction shall be known.

D.4.2 Use of standard flanges without calculation

Flanges that conform to European Standards for pipework flanges may be used as without any calculation, provided all the following conditions are fulfilled:

c) Under normal operating conditions, the calculation pressure does not exceed the rating pressure given in the tables of the relevant European Standard, for the flange and material under consideration for the calculation temperature;

d) Under testing conditions or exceptional conditions, the calculation pressure does not exceed 1,5 times the rating pressure given in the same tables, at appropriate temperature;

e) The gasket is one of those permitted by Table D.4-1 for the relevant PN or Class series;

f) The bolts are of a strength class (see Table D.4-2) at least equal to the minimum required by Table D.4-1 as a function of the gasket type used in the connection;

g) The difference between mean temperatures of bolts and flange does not exceed 50 °C in any condition;

h) The bolt and flange materials have coefficients of thermal expansion at 20 °C that differ by more than 10 % (e.g. austenitic steel flanges with ferritic steel bolts) but the calculation temperature is < 120 °C, or the bolt and flange materials have coefficients of thermal expansion at 20 °C which do not differ by more than 10 %.

D.4.3 Bolting

There shall be at least four bolts.

In the case of small diameter bolts it can be necessary to use torque spanners or other means for preventing the application of excessive load on the bolt.

Special means may be required to ensure that an adequate preload is obtained when tightening bolts of nominal diameter greater than 38 mm.

Bolt nominal design stresses for determining the minimum bolt area in D.5.2 shall be:

— for carbon and other non-austenitic steels, the lesser of $R_{p0,2}/3$ measured at design temperature and $R_m/4$ measured at room temperature;

— for austenitic stainless steel, $R_m/4$ measured at design temperature.

Table D.4-1 — Gaskets for standard flanges

PN designated series [a]	Class designated series [a]	Gasket type	Minimum bolt strength class required (see Table D.4-2)
2,5 to 16	-	— Non-metallic flat gasket with or without jacket	Low strength
25	150	— Non-metallic flat gasket with or without jacket	Low strength
		— Spiral-wound metal with filler — Corrugated metal jacketed with filler — Corrugated metal with or without filler	Medium strength
40	-	— Non-metallic flat gasket with or without jacket	Low strength
		— Spiral-wound metal with filler — Corrugated metal jacketed with filler — Corrugated metal with or without filler	Medium strength
		— Flat metal jacketed with filler — Grooved or solid flat metal	High strength
63	300	— Non-metallic flat gasket with or without jacket	Low strength
		— Spiral-wound metal with filler — Corrugated metal jacketed with filler — Corrugated metal with or without filler	Medium strength
		— Flat metal jacketed with filler — Grooved or solid flat metal — Metal ring joint	High strength
100	600	— Non-metallic flat gasket with or without jacket — Spiral-wound metal with filler — Corrugated metal jacketed with filler — Corrugated metal with or without filler	Medium strength
		— Flat metal jacketed with filler — Grooved or solid flat metal — Metal ring joint	High strength

[a] The PN (or Class) values presented in this table are restricted to those existing in European Standards on Steel Flanges, up to PN 100 (or Class 600).

Table D.4-2 — Bolt strength classes

	Low strength	Medium strength	High strength
$\frac{R_{p,bolt}}{R_{p,flange}}$	≥ 1	≥ 1,4	≥ 2,5

NOTE R_p is $R_{p0,2}$ for non-austenitic steels, $R_{p1,0}$ for austenitic steels.

The assembly condition and operating condition are both normal design conditions for the purpose of determining nominal design stresses.

These allowable stresses may be multiplied by 1,5 for testing or exceptional conditions.

NOTE These stresses are nominal in so far as they may have to be exceeded in practice to provide against all conditions that tend to produce a leaking joint. However there is sufficient margin to provide a satisfactory closure without having to overload or repeatedly tighten the bolts.

D.4.4 Flange construction

A distinction is made between flanges in which the bore of the flange coincides with the bore of the shell (for example welded joints F1, F2, F4 and F5 as shown in Table A.7) and those with a fillet weld at the end of the shell (for example welded joint F3) in which the two bores are different. They are known as smooth bore (see Figure D.5-1) and stepped bore (see Figure D.5-2) respectively.

A further distinction is made between the slip-on hubbed flange (see Figure D.5-3), in which a forged flange complete with taper hub is slipped over the shell and welded to it at both ends, and other types of welded construction.

Any fillet radius between flange and hub or shell shall be not less than 0,25g_0 and not less than 5 mm.

Hub flanges shall not be made by machining the hub directly from plate material without special consideration.

Fillet welds shall not be used for design temperatures above 370 °C.

D.4.5 Machining

The bearing surface for the nuts shall be parallel within 1° to the flange face. Any back facing or spot facing to accomplish this shall not reduce the flange thickness nor the hub thickness below design values. The diameter of any spot facing shall be not less than the dimension across corners of the nut plus 3 mm. The radius between the back of the flange and the hub or shell shall be maintained.

The surface finish of the gasket contact face should be in accordance with the gasket manufacturers' recommendations or be based on experience.

D.4.6 Gaskets

The values of the gasket factors *m* and *y* should normally be provided by the gasket manufacturer but suggested values are given in Table 7.2.4-1.

Suggested minimum values of *w*, the assembly width, are also given in Annex K.

NOTE Asbestos containing gaskets are forbidden in most European countries.

D.5 Narrow face gasketed flanges

D.5.1 General

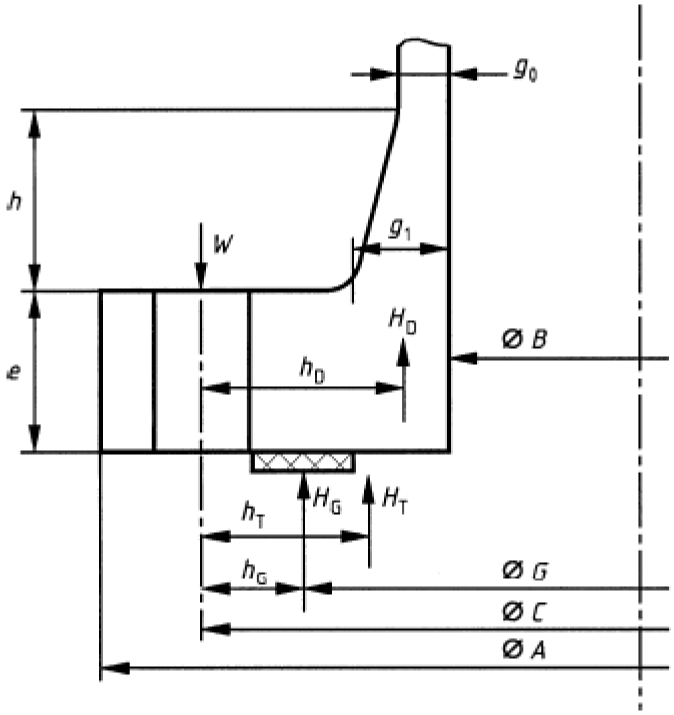


Figure D.5-1 — Narrow face flange - smooth bore

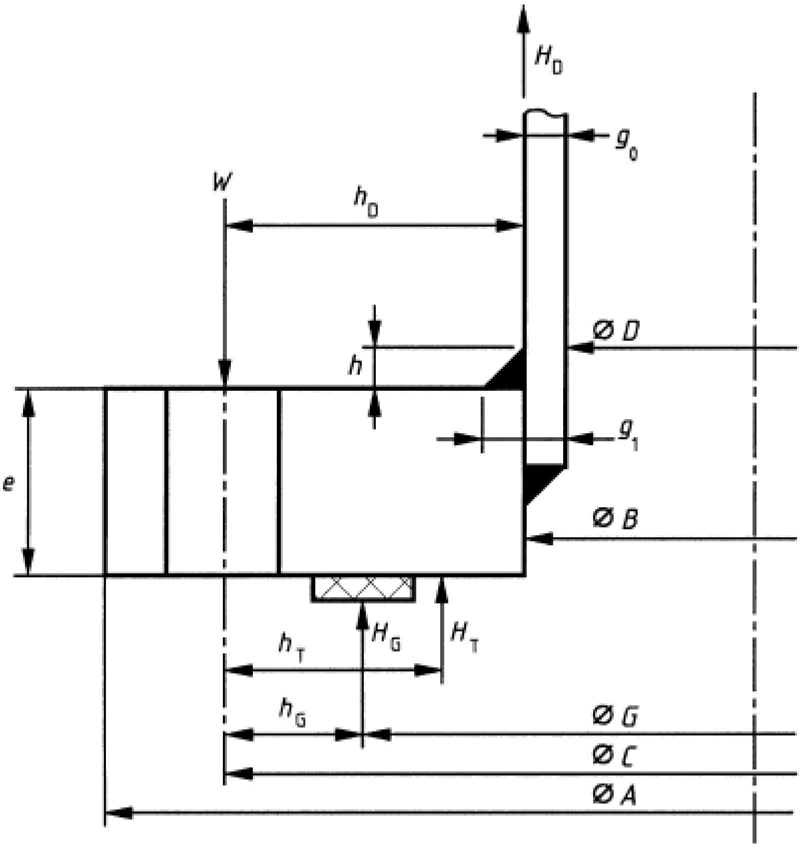


Figure D.5-2 — Narrow face flange — stepped bore

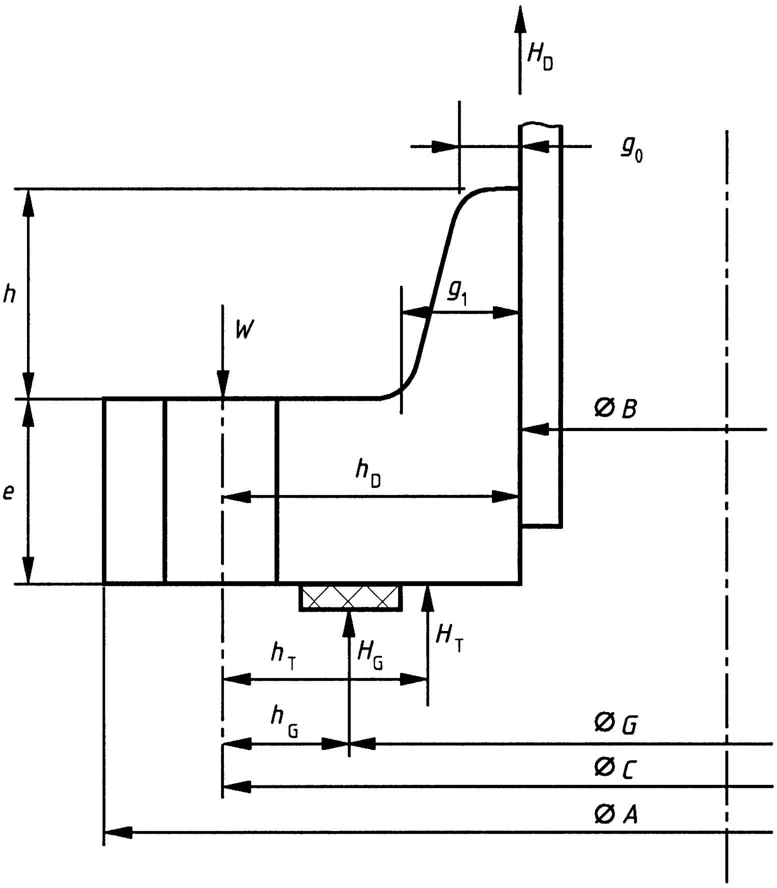


Figure D.5-3 — Narrow face flange - slip on hub type

One of the three following methods of stress calculation shall be applied in D.5.4. to narrow face flanges with gaskets or joints under internal pressure, taking account of the exceptions given.

a) Integral method. The integral method shall not be applied to the slip-on hubbed flange or to the loose flange in a lap joint. The integral design method allows for a taper hub, which may be a weld; the hub assumed for purposes of calculation shall not have a slope of more than 1:1, i.e. $g_1 \leq h + g_0$;

b) Loose method. The loose method shall only be applied, except for loose flanges in lap joints, if all of the following requirements are met:

   1) $g_o \leq 16$ mm;

   2) $P \leq 2$ N/mm^2;

   3) $B / g_o \leq 300$;

   4) operating temperature ≤ 370 °C.

c) Loose hubbed flange method. This shall be applied to the slip-on hubbed flange and the loose hubbed flange in a lap joint.

NOTE 1 In the integral method account is taken of support from the shell and stresses in the shell are calculated, but in the loose method the flange is assumed to get no support from the shell and shell stresses are ignored.

NOTE 2 In more unusual shapes of hub it may be necessary to choose values of g_1 and h defining a simple taper hub which fits within the profile of the actual assembly.

NOTE 3 There is no minimum value of h for a slip-on hubbed flange.

NOTE 4 The procedure for calculating the value of M is independent of the design method chosen.

D.5.2 Bolt loads and areas

$$b_0 = w/2 \tag{D.5-1}$$

except for the ring-joint (see Annex K), for which

$$b_0 = w/8; \tag{D.5-2}$$

When $b_0 \leq 6{,}3$ mm,

$$b = b_0 \tag{D.5-3}$$

When $b_0 > 6{,}3$ mm,

$$b = 2{,}52\sqrt{b_0} \tag{D.5-4}$$

(This expression is valid only with dimensions expressed in millimetres).

When $b_0 \leq 6{,}3$ mm, G is the mean diameter of gasket contact face,

when $b_0 > 6{,}3$ mm, G is the outside diameter of gasket contact face less $2b$:

$$H = \pi/4\ G^2\ P \tag{D.5-5}$$

$$H_G = 2\pi\ b\ G\ m\ P \tag{D.5-6}$$

Bolt loads and areas shall be calculated for both the assembly and operating conditions as follows.

a) *Assembly condition*. The minimum bolt load is given by:

$$W_A = \pi\ b\ G\ y \tag{D.5-7}$$

NOTE The minimum bolt loading to achieve a satisfactory joint is a function of the gasket and the effective gasket area to be seated.

b) *Operating condition*. The minimum bolt load is given by:

$$W_{op} = H + H_G \tag{D.5-8}$$

The required bolt area $A_{B,min}$ is given by:

$$A_{B,min} = \max\left(\frac{W_A}{f_{B,A}};\frac{W_{op}}{f_B}\right) \quad (D.5-9)$$

Bolting shall be chosen so that $A_B \geq A_{B,min}$.

NOTE Internal pressure tends to part the joint and the bolt load has to maintain sufficient pressure on the gasket to ensure a tight joint. The minimum bolt load under this condition is a function of design pressure, gasket material and the effective gasket contact area to be kept tight under pressure. More than one operating condition may require consideration.

D.5.3 Flange moments

$$H_D = \frac{\pi}{4}B^2P \quad (D.5-10)$$

$$H_T = H - H_D \quad (D.5-11)$$

$$h_D = (C - B - g_1)/2 \quad (D.5-12)$$

except for slip-on hubbed and stepped bore flanges for which:

$$h_D = (C - B) / 2 \quad (D.5-13)$$

$$h_G = (C - G) / 2 \quad (D.5-14)$$

$$h_T = (2C - B - G) / 4 \quad (D.5-15)$$

$$W = 0{,}5\ (A_{B,min} + A_B)\ f_{B,A} \quad (D.5-16)$$

a) Flange *Assembly condition*. The total flange moment shall be:

$$M_A = W \cdot h_G \quad (D.5-17)$$

b) *Operating condition*. The total flange moment shall be:

$$M_{op} = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad (D.5-18)$$

For flange pairs having different design conditions, as for example when they trap a tubesheet, bolt loads shall be calculated at assembly and operating conditions for each flange/gasket combination separately. W_{op} and W_A shall be taken as the greater of the two calculated values. For the flange for which W_{op} has the lower calculated value, the value of H_G shall be increased as follows:

$$H_{G,new} = H_G + W_{op,max} - W_{op,min} \quad (D.5-19)$$

D.5.4 Flange stresses and stress limits

D.5.4.1 Flange stresses

$$C_F = \max\left(\sqrt{\frac{\delta_b}{2\,d_b + \frac{6e}{m+0,5}}};1\right) \qquad \text{(D.5-20)}$$

$$K = A/B \qquad \text{(D.5-21)}$$

$$I_0 = \sqrt{Bg_0} \qquad \text{(D.5-22)}$$

$$\beta_T = \frac{K^2(1+8,55246\log 10(K))-1}{(1,0472+1,9448K^2)(K-1)} \qquad \text{(D.5-23)}$$

$$\beta_U = \frac{K^2(1+8,55246\log 10(K))-1}{1,36136(K^2-1)(K-1)} \qquad \text{(D.5-24)}$$

$$\beta_Y = \frac{1}{K-1}\left(0,66845+5,7169\frac{K^2\log 10(K)}{K^2-1}\right) \qquad \text{(D.5-25)}$$

Flange stresses shall be determined from the moment, M, as follows:

For the assembly condition,

$$M = M_A \frac{C_F}{B} \qquad \text{(D.5-26)}$$

For the operating condition,

$$M = M_{op} \frac{C_F}{B} \qquad \text{(D.5-27)}$$

a) Integral method

β_F β_V and φ are found from Figures N.5-4, N.5-5 and N.5-6.

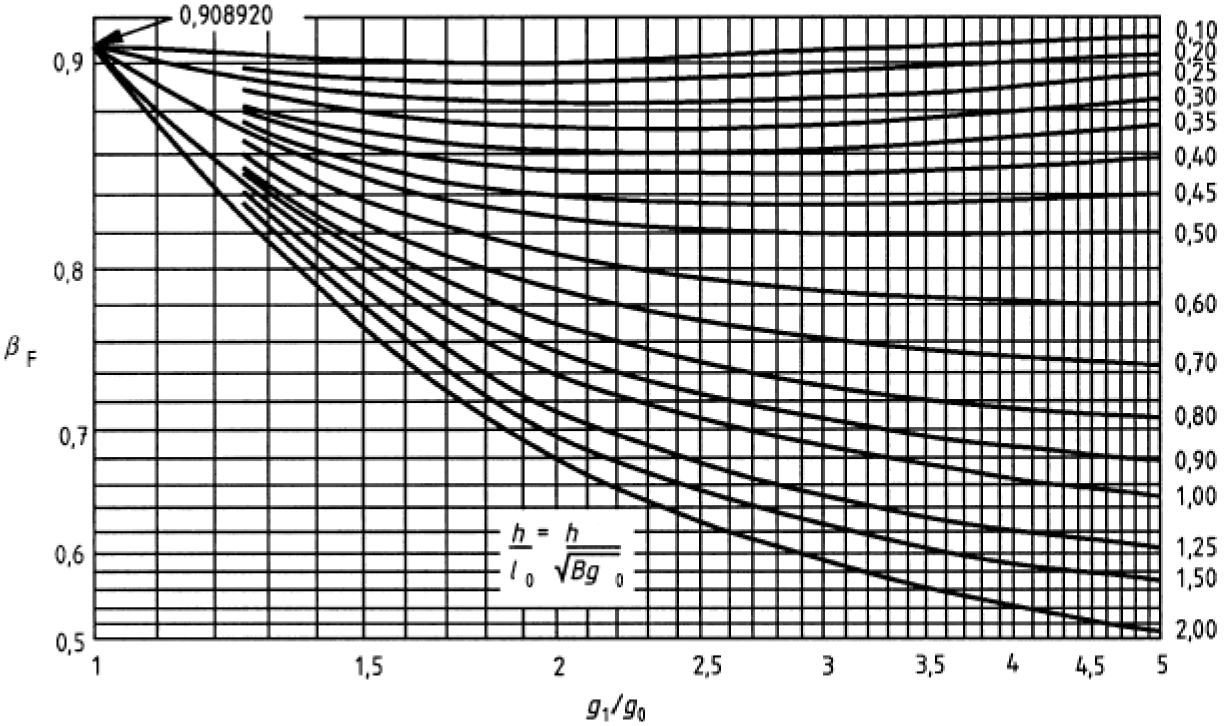


Figure D.5-4 — Value of β_F (integral method factor)

$$\lambda = \left(\frac{e \cdot \beta_F + l_0}{\beta_T \cdot l_0} + \frac{e^3 \cdot \beta_V}{\beta_U \cdot l_0 \cdot g_0^2} \right) \tag{D.5-28}$$

The longitudinal hub stress:

$$\sigma_H = \frac{\varphi M}{\lambda g_1^2} \tag{D.5-29}$$

The radial flange stress:

$$\sigma_r = \frac{(1{,}333 e \ \beta_F + l_0) M}{\lambda e^2 l_0} \tag{D.5-30}$$

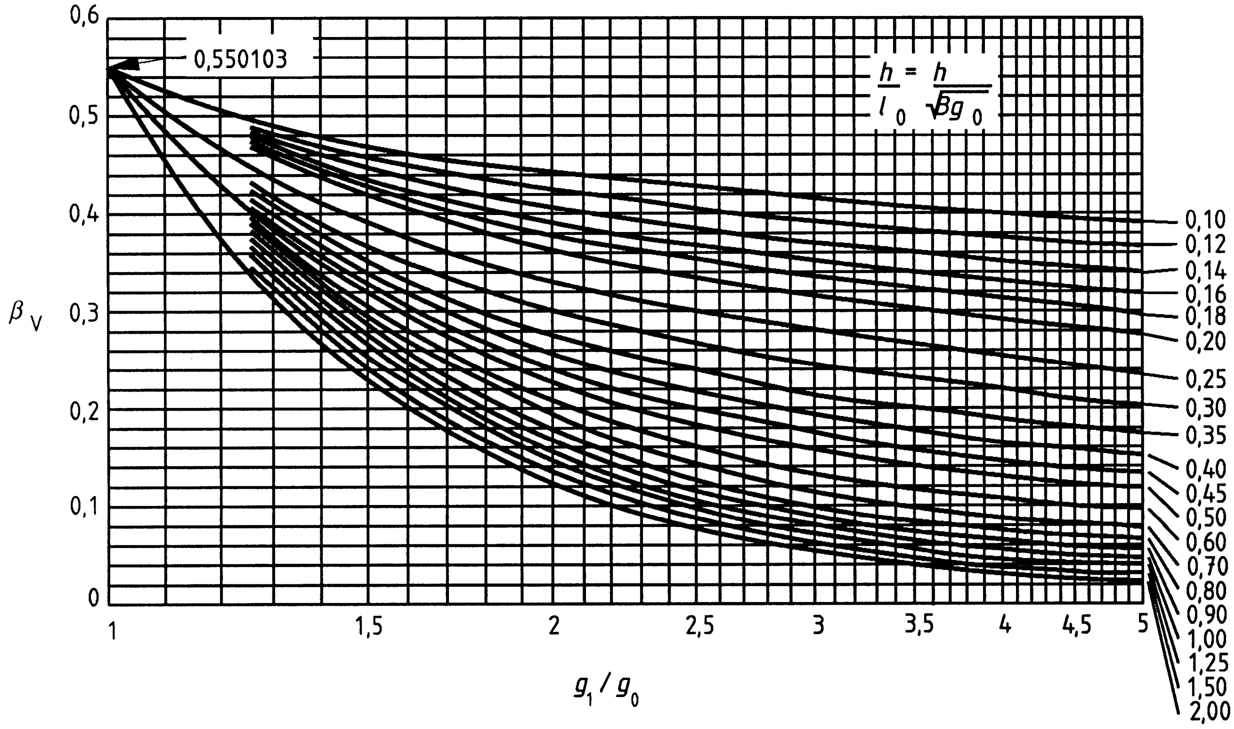
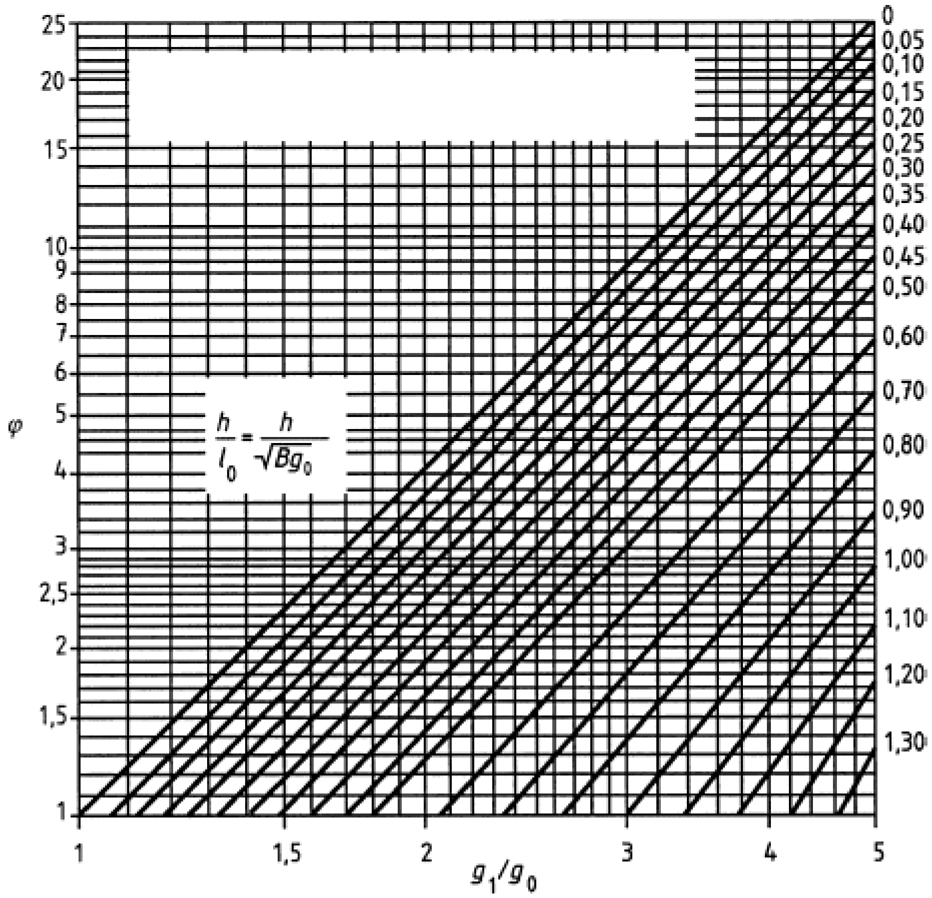


Figure D.5-5 — Value of β_V (integral method factor)



φ = 1 (minimum) for hubs of uniform thickness (g_1 / g_0 = 1)

Figure D.5-6 — Value of φ (hub stress correction factor)

The tangential flange stress:

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} \qquad \text{(D.5-31)}$$

b) Loose method

The tangential flange stress:

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} \qquad \text{(D.5-32)}$$

The radial stress in flange and longitudinal stress in hub are

$$\sigma_r = \sigma_H = 0 \qquad \text{(D.5-33)}$$

c) Loose hubbed flange method

β_{FL} and β_{VL} are found from Figures D.5-7 and D.5-8 respectively.

$$\lambda = \left[\frac{e\beta_{FL} + I_0}{\beta_T I_0} + \frac{e^3 \beta_{VL}}{\beta_U I_0 g_0^2} \right] \tag{D.5-34}$$

The longitudinal hub stress:

$$\sigma_H = \frac{M}{\lambda g_1^2} \tag{D.5-35}$$

The radial flange stress:

$$\sigma_r = \frac{(1{,}333 e \cdot \beta_{FL} + I_0) M}{\lambda \cdot e^2 \cdot I_0} \tag{D.5-36}$$

The tangential flange stress:

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} \tag{D.5-37}$$

D.5.4.2 Stress limits

The assembly condition and operating condition are both normal design conditions for the purpose of determining nominal design stresses.

Nominal design stresses f shall be obtained in accordance with clause 6 except that the rule based on R_m/3 (see clause 5) for austenitic stainless steel is not applicable.

f_H shall be the nominal design stress of the shell except for welding neck or slip-on hubbed construction where it is the nominal design stress of the flange.

If $B \leq$ 1 000 mm, then k = 1,0.

If $B \geq$ 2 000 mm, then k = 1,333.

For values of B between 1 000 and 2 000 mm:

$$k = \frac{2}{3}\left(1 + \frac{B}{2000}\right) \tag{D.5-38}$$

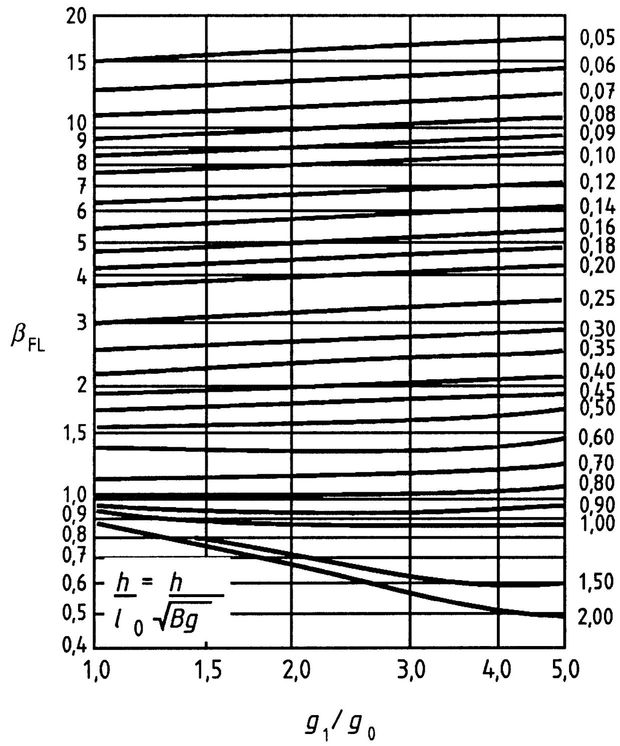


Figure D.5-7 — Value of β_{FL} (loose hub flange factor)

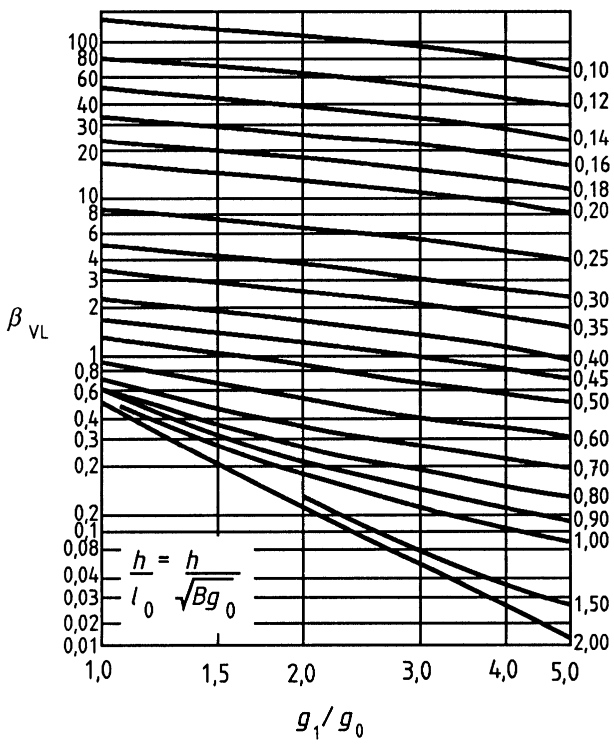


Figure D.5-8 — Value of β_{VL} (loose hub flange factor)

The flange stresses as calculated in D.5.4.1 shall meet the following requirements:

$$k\ \sigma_H \le 1{,}5 \min(f; f_H) \quad \text{(D.5-39)}$$

$$k\ \sigma_r \le f \quad \text{(D.5-40)}$$

$$k\ \sigma_\theta \le f \quad \text{(D.5-41)}$$

$$0{,}5k\ (\sigma_H + \sigma_r) \le f \quad \text{(D.5-42)}$$

$$0{,}5k\ (\sigma_H + \sigma_\theta) \le f \quad \text{(D.5-43)}$$

D.5.5 Narrow face flanges subject to external pressure

If the flange is subject to both internal and external pressure it shall be designed for both conditions, except that external pressure need not be considered where the external calculation pressure P_e is less than the internal calculation pressure.

The design of flanges for external pressure shall be in accordance with N.5.4 except that:

a) P_e replaces P;

$$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G) \quad \text{(D.5-44)}$$

and

b) $W_{op} = 0$ (D.5-45)

NOTE In the case of external pressure the bolts can be completely loose, leading to $W_{op} = 0$. This is a conservative assumption as any bolt load reduces the net moment on the flange.

Where a flange is being designed for external pressure and is one of a flange pair having different design conditions, W_{op} shall be that calculated for the other of the pair and M_{op} shall be the greater of M_{op} as calculated above and $W_{op}h_G$.

D.5.6 Lap joints

D.5.6.1 General

In a lap joint the loose flange may have a hub. The stub flange may be attached to the shell in any way permitted for a bolted flange.

Bolt loads and areas shall meet the requirements of D.5.2 or D.6.2 as appropriate, depending on which method is applied to the stub flange in D.5.6.2.

The diameter G_1 of the load reaction between stub and loose flanges shall be given a value lying between $(A_2–\delta)$ and $(B_2+\delta)$.

NOTE The value given by equation (11.5-46) should be used unless there is good reason to do otherwise.

$$G_1 = (A_2 + B_2)/2 \quad \text{(D.5-46)}$$

The area of the contact face between the two flanges shall be given by:

$$A_C = \frac{\pi}{2}\min\left[\left(A_2 - \delta\right)^2 - G_1^2; G_1^2 - \left(B_2 + \delta\right)^2\right] \quad \text{(D.5-47)}$$

If the diameters A_2 and B_2 are defined by the same component, as with the stepped flange shown in Figure N.5-9, δ shall be given the value zero in equation (D.5-47).

Bearing stress σ_b at the contact face shall be determined for both assembly and operating conditions using the following equation:

$$\sigma_b = \frac{W_{op} \text{ or } W}{A_c} \quad \text{(D.5-48)}$$

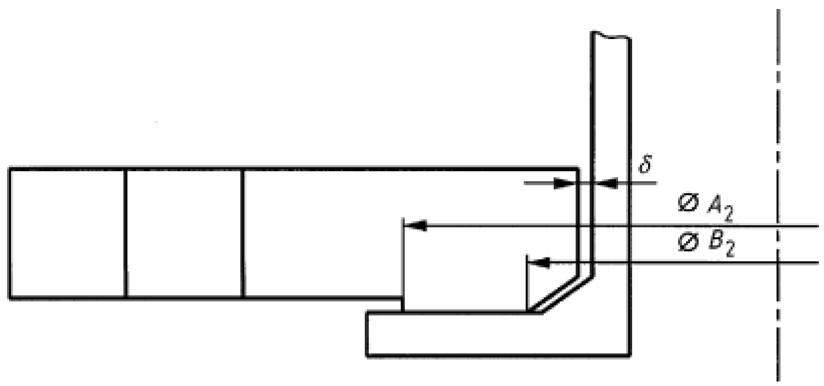


Figure D.5-9 — Stepped loose flange

The bearing stress shall not exceed 1,5 times the lower nominal design stress of the two flanges.

D.5.6.2 Stub flange

The stub flange shall take one of the forms listed in D.4.4 and either the narrow face (see clause D.5) or full face (see clause D.6) method shall be applied.

NOTE When G1 is greater than the OD of the gasket then the full face method is inapplicable. Even when G1 is less than the OD of the gasket the narrow face method is applicable though possibly less economic.

The stub flange shall meet the requirements for a flange loaded directly by the bolts as given in D.5.4 or clause D.6, except that the bolt load is assumed to be imposed at diameter G_1, which therefore replaces C in the calculation at the moment arms h_D, h_G and h_T. The diameter of the bolt holes, d_h, required in clause D.6, shall be taken as zero.

D.5.6.3 Loose flange

See Figures D.5-10 and D.5-11.

$$h_L = \left(C - G_1\right)/2 \quad \text{(D.5-49)}$$

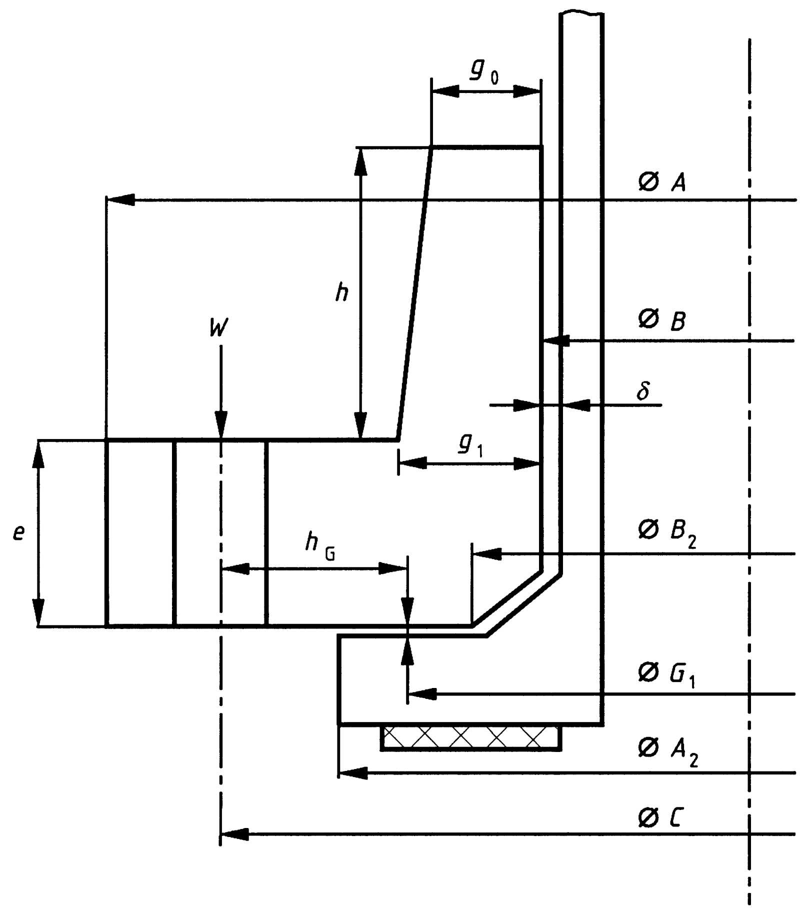
The moment arm on the loose flange for all components of load shall be h_L so that:

$$M_{op} = W_{op}\ h_L \tag{D.5-50}$$

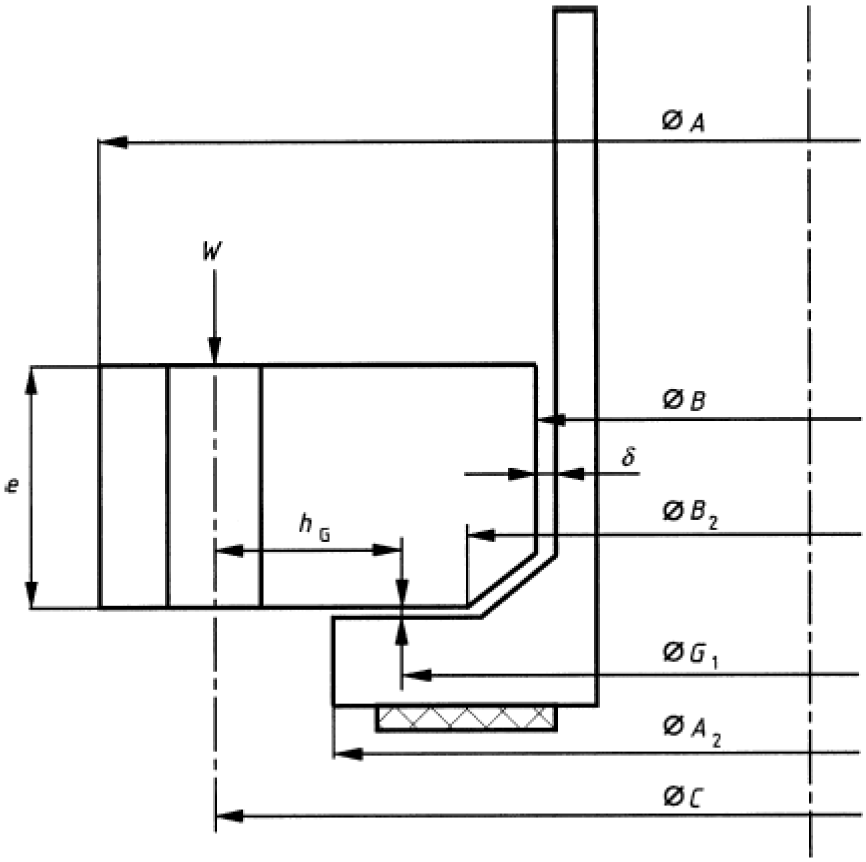
NOTE For external pressure, $W_{op} = 0$ – see D.6.4.

$$M_A = W\ h_L \tag{D.5-51}$$

The loose flange stresses and stress limits shall meet the requirements of D.5.4.



Figures D.5-10 — Lap type joint; loose flange with hub



Figures D.5-11 — Lap type joint; loose flange without hub

D.5.7 Split ring flanges

It is permissible to split the loose flange in a lap joint across the diameter to make it readily removable from the nozzle neck or vessel. The design shall be in accordance with D.5.6.3 modified as follows.

When the flange consists of a single split ring, it shall be designed as if it were a solid flange (without splits), using 200 % of the moment M_{op} and/or M_A required in D.5.6.3.

When the flange consists of two split rings, each ring shall be designed as if it were a solid flange (without splits), using 75 % of the moment required in D.5.6.3. The pair of rings shall be assembled so that the splits in one ring are 90° from the splits in the other ring. The splits shall be located midway between bolt holes.

D.6 Full face flanges with soft ring type gaskets

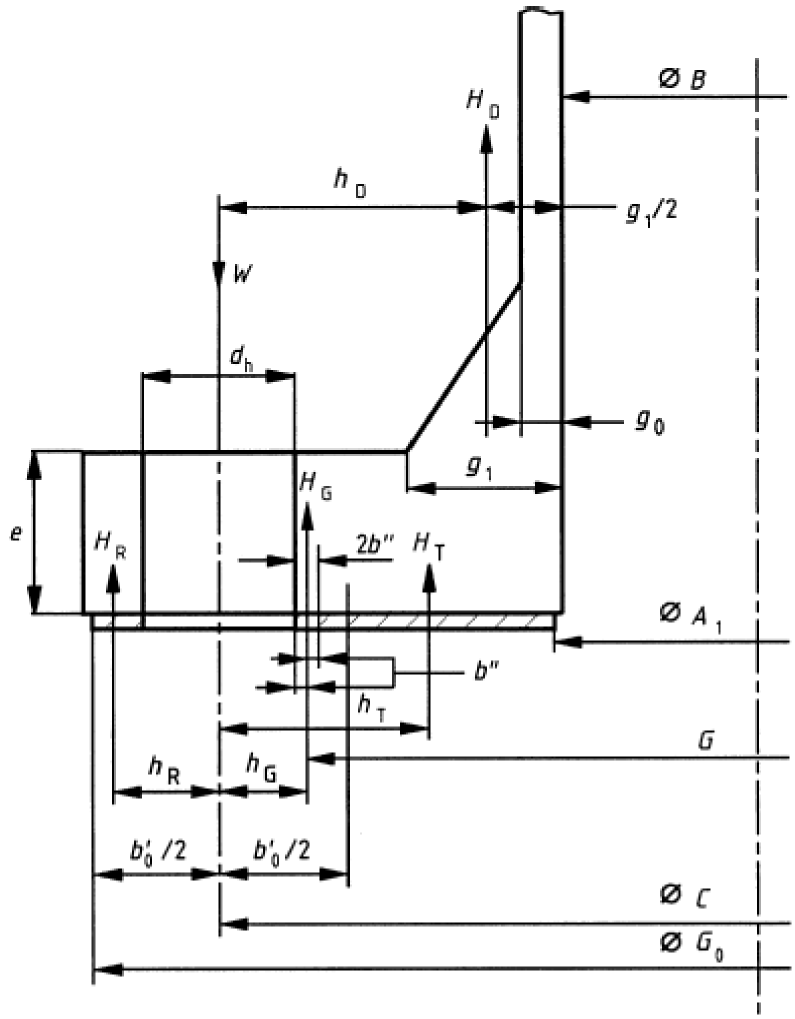


Figure D.6-1 — Full face flange (soft gasket)

D.6.1 Specific symbols and abbreviations

The following symbols and abbreviations apply in addition to those in clause D.3:

NOTE See Figure D.6-1 for an illustration of the various dimensions.

A_1 is inside diameter of gasket contact face;

$b^/$ is the effective assembly width;

$2b$" is the effective gasket pressure width, taken as 5 mm;

$b^/_0$ is the basic assembly width effective under initial tightening up;

d_h is diameter of bolt holes;

G is the diameter at location of gasket load reaction;

G_0 is outside diameter of gasket or outside diameter of flange, whichever is less;

H is the total hydrostatic end force;

H_G is compression load on gasket to ensure tight joint;

H_R is the balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle;

h_R is radial distance from bott circle to circle on which H_R acts;

h_S is radial distance from bolt circle to circle on which H_T acts;

h_T is radial distance from bolt circle to circle on which H_G acts;

M_R is balancing radial moment in flange along line of bolt holes;

n is number of bolts;

δ_b is bolt spacing.

D.6.2 Bolt loads and areas

$2b$" is given the value 5 mm

$$b^/_0 = \min\left(G_0 - C\ ;\ C - A_1\right) \quad \text{(D.6-1)}$$

$$b^/ = 4\sqrt{b'_0} \quad \text{(D.6-2)}$$

(This expression is valid only with dimensions expressed in millimetres);

$$G = C - (d_h + 2b") \quad \text{(D.6-3)}$$

$$H = \pi/4(C - d_h)^2 P \quad \text{(D.6-4)}$$

$$H_D = \frac{\pi}{4} B^2 \; P \qquad \text{(D.6-5)}$$

$$H_T = H - H_D \qquad \text{(D.6-6)}$$

$$H_G = 2b'' \; \pi \; G \; m \; P \qquad \text{(D.6-7)}$$

$$h_D = (C - B - g_1)/2$$

$$h_T = (C + d_h + 2b'' - B)/4 \qquad \text{(D.6-8)}$$

$$h_G = (d_h + 2b'')/2 \qquad \text{(D.6-9)}$$

$$h_R = (G_0 - C + d_h)/4 \qquad \text{(D.6-10)}$$

$$M_R = H_D \; h_D + H_T \; h_T + H_G \; h_G \qquad \text{(D.6-11)}$$

$$H_R = \frac{M_R}{h_R} \qquad \text{(D.6-12)}$$

Bolt areas shall be calculated in accordance with D.5.2, taking:

$$W_A = \pi \; C \; b' \; y \qquad \text{(D.6-13)}$$

$$W_{op} = H + H_G + H_R \qquad \text{(D.6-14)}$$

D.6.3 Flange design

The flange thickness shall be not less than the greatest value of *e* from the following three equations:

$$e = \sqrt{\frac{6M_R}{f\left(\pi C - n d_h\right)}} \qquad \text{(D.6-15)}$$

$$e = \frac{\left(m + 0{,}5\right)}{\left(E/200\,000\right)^{0{,}25}} \; \frac{\left(\delta_b - 2d_b\right)}{6} \qquad \text{(D.6-16)}$$

where E is expressed in N/mm^2

$$e = \frac{(A_1 + 2g_1)P}{2f} \qquad \text{(D.6-17)}$$

Where two flanges of different internal diameters, both designed to the rules of D.6.4, are to be bolted together to make a joint, the following additional requirements apply:

a) the value of M_R to be used for both flanges shall be that calculated with the smaller internal diameter;

b) the thickness of the flange with the smaller bore shall be not less than:

$$e = \sqrt{\frac{3(M_1 - M_2)(A + B)}{\pi f \times B(A - B)}} \qquad \text{(D.6-18)}$$

where

M_1 and M_2 are the values of M_R calculated for the two flanges.

D.6.4 Full face flanges subject to external pressure

If the flange is subject to both internal and external pressure it shall be designed for both conditions, except that external pressure need not be considered where the external calculation pressure is less than the internal.

The design of flanges for external pressure shall be in accordance with clause D.6 except that:

a) P_e replaces P;

b) Equation (D.6-16) does not apply;

c) W_{op} = 0.

D.7 Seal welded flanges

Seal welded flanges (as shown in Figure D.7-1) shall be designed in accordance with clause D.5, except that:

a) only the operating condition is to be considered;

b) $G = D_L$, the inside diameter of seal weld lip, as shown in Figure D.7-1;

c) $H_G = 0$;

d) flange thickness e shall be determined as the mean thickness of the flange.

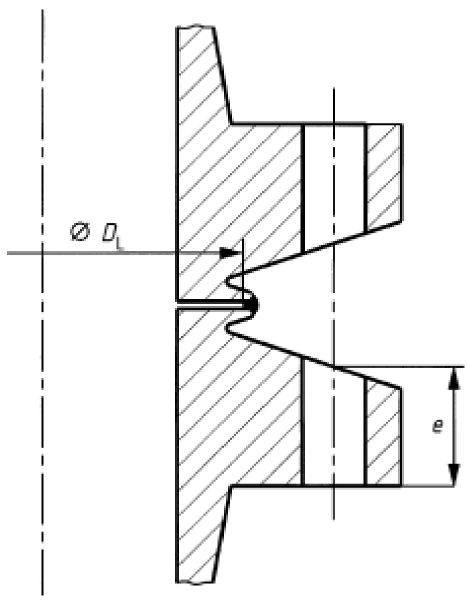


Figure D.7-1 — Seal welded flange

D.8 Reverse narrow face flanges

D.8.1 Internal pressure

Reverse flanges with narrow face gaskets (see Figures D.8-1 and D.8-2) under internal pressure shall be designed in accordance with clause D.5 with the following modifications.

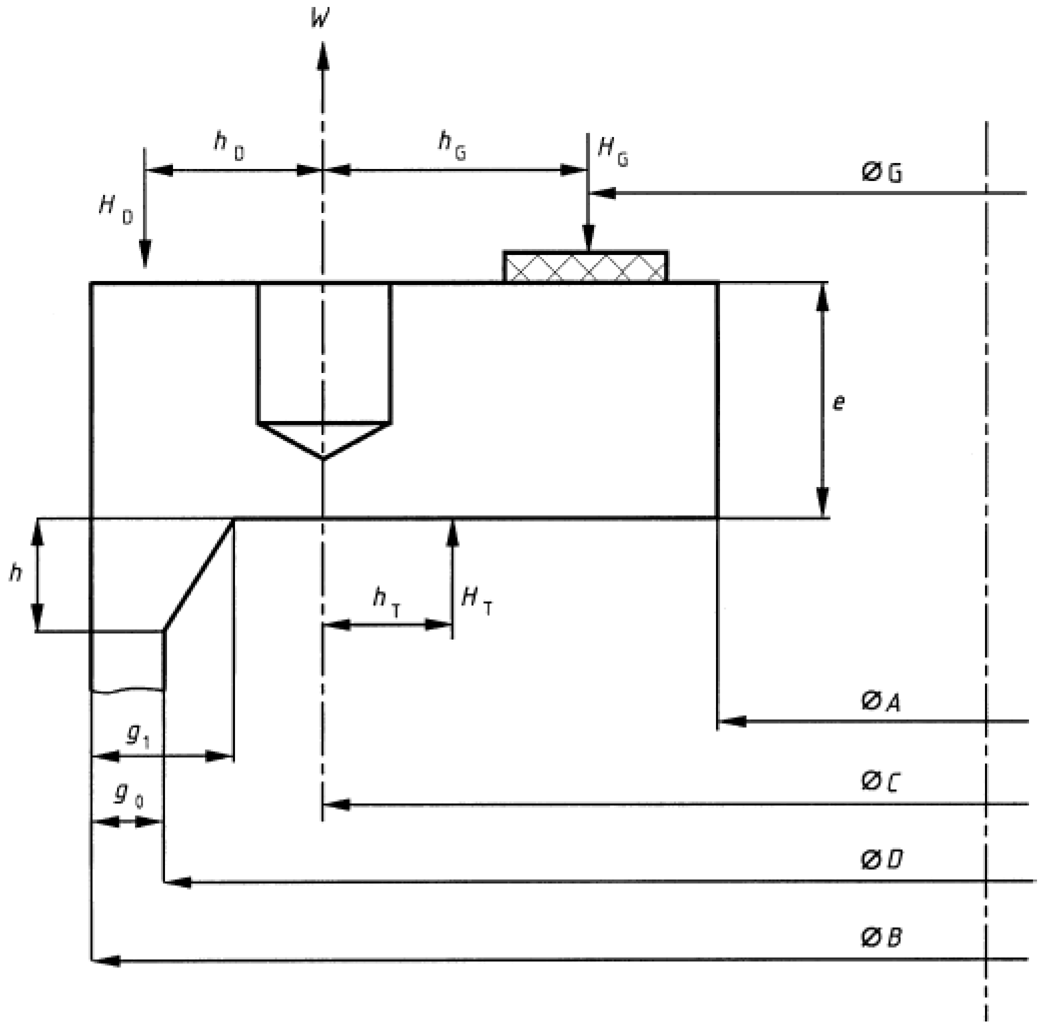
The limits on g_o and B/g_o to the application of the loose method of calculation do not apply.

The following symbols and abbreviations are in addition to or modify those in clause D.3:

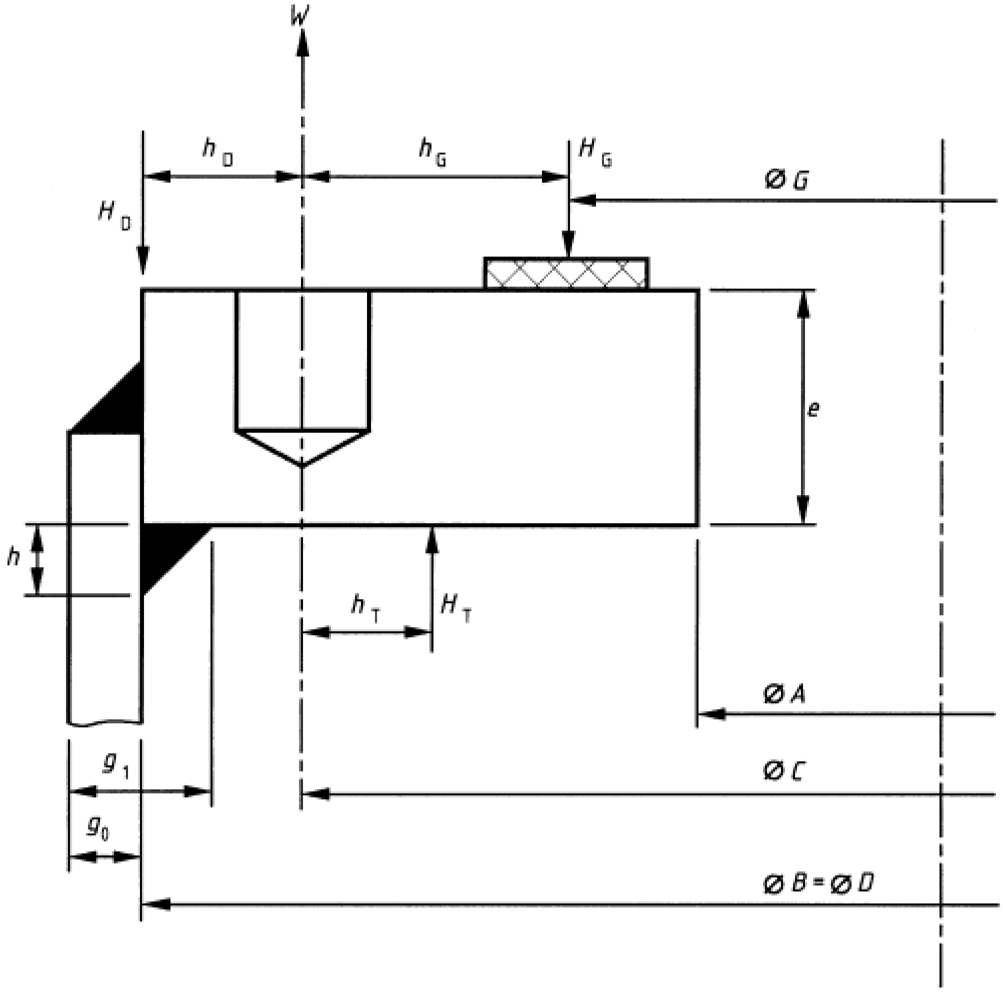
A is the inside diameter of the flange;

B is the outside diameter of the flange;

H_T is the net pressure load on the flange faces.



Figures D.8-1 — Reverse narrow face flange



Figures D.8-2 — Reverse narrow face flange; slip in type

The following equations replace the equations in clause D.5 for the given variables:

$$H_D = \pi/4PD^2 \quad \text{(D.8-1)}$$

$$H_T = H_D - H \quad \text{(D.8-2)}$$

$$h_D = (B - C - g_1) / 2 \quad \text{(D.8-3)}$$

except for slip-in type flange with fillet weld (so that $B = D$), when

$$h_D = (B - C) / 2 \quad \text{(D.8-4)}$$

$$h_T = (2C - G - D) / 4 \quad \text{(D.8-5)}$$

$$M_{op} = H_T\ \ h_T + H_D\ \ h_D \quad \text{(D.8-6)}$$

$$M = (M_A \text{ or } M_{op})\ C_F / A \quad \text{(D.8-7)}$$

$$K = B/A \quad \text{(D.8-8)}$$

The sign of h_T, which may be negative, has to be respected.

NOTE The moment due to gasket reaction is taken as zero for the operating condition. This is a conservative assumption since any gasket load reduces the moment in the flange.

D.8.2 External pressure

Reverse flanges with narrow face gaskets under external pressure shall be designed in accordance with D.8.1 modified by D.5.5, except that equation (D.5-5) is replaced by:

$$M_{op} = H_D(h_D + h_G) + H_T(h_G - h_T) \tag{D.8-9}$$

D.9 Reverse full face flanges

D.9.1 General

The design method shall be in accordance with either D.9.2 or D.9.3; both are equally valid. For both design methods gaskets and bolting loads at the assembly condition shall be in accordance with clause D.6.

NOTE Two alternative design methods are provided for reverse full face flanges. The first follows the approach of clause D.5 at the operating condition and assumes resistance to rotation comes from the flange itself; the second follows clause D.6 and requires a larger bolt area.

D.9.2 Design following method of D.5

NOTE See Figure 11.9-1 for an illustration of the loads and dimensions.

Design for the operating condition shall be in accordance with clause D.5 with the following modifications.

The following symbols and abbreviations are in addition to or modify those in clause D.3.

A is inside diameter of flange;

A_1 is inside diameter of gasket contact face;

B is outside diameter of flange;

H_S is the hydrostatic end force due to pressure on exposed flange face;

h_S is the radial distance from bolt circle to circle on which H_S acts.

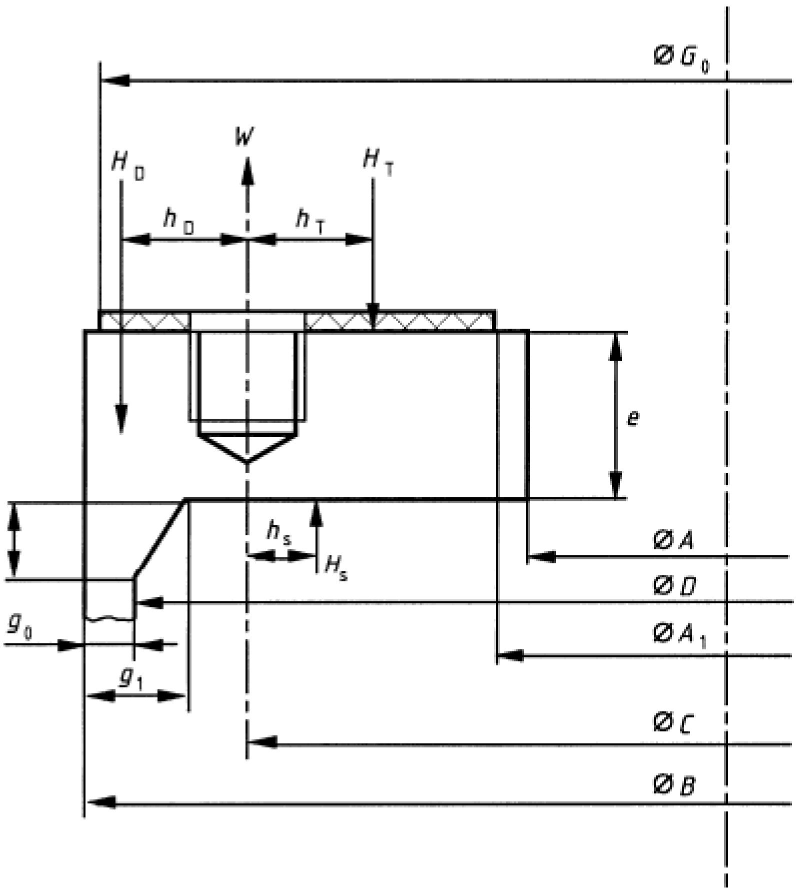


Figure D.9-1 — Reverse full face flange design to D.9.2

The following additional equations apply:

$$w = (C - A_1) / 2 \qquad \text{(D.9-1)}$$

$$H_S = H_D - \pi/4P \ \ A_1^2 \qquad \text{(D.9-2)}$$

$$h_S = (2C - D - A_1) / 4 \qquad \text{(D.9-3)}$$

The following equations replace the equations in clause D.5 for the given variable:

$$H = \pi/4P \ (C - d_h)^2 \ P \qquad \text{(D.9-4)}$$

$$H_D = \pi/4P \ \% \ D^2 \qquad \text{(D.9-5)}$$

$$H_G = 2\pi b \ \ C \ \ m \ \ P \qquad \text{(D.9-6)}$$

$$H_T = (H - H_D + H_S) / 2 \qquad \text{(D.9-7)}$$

$$h_D = (B - g_1 - C) / 2 \qquad \text{(D.9-8)}$$

except for the slip-in type flange (B≠D) for which,

$$h_D = (B - C) / 2 \tag{D.9-9}$$

$$h_T = (2C + d_h - 2A_1) / 6 \tag{D.9-10}$$

$$M_{op} = H_D\ h_D - H_T\ h_T + H_S\ h_S \tag{D.9-11}$$

$$M = M_{op}\ C_F / A \tag{D.9-12}$$

$$K = B/A \tag{D.9-13}$$

The sign of h_S, which may be negative, shall be respected.

NOTE The moment due to gasket reaction is taken as zero for the operating condition since this assumption gives higher stresses.

D.9.3 Design following method of D.6

NOTE See Figure D.9-1 for an illustration of loads and dimensions.

The rules in clause D.9.3 shall only be used for reverse flanges where the mating component is a tubesheet or flat plate.

Design for the operating condition shall be in accordance with clause D.6 with the following modifications.

The following symbols and abbreviations are in addition to or modify those in clause D.3:

A is inside diameter of flange;

A_1 is inside diameter of gasket contact face;

B is outside diameter of flange;

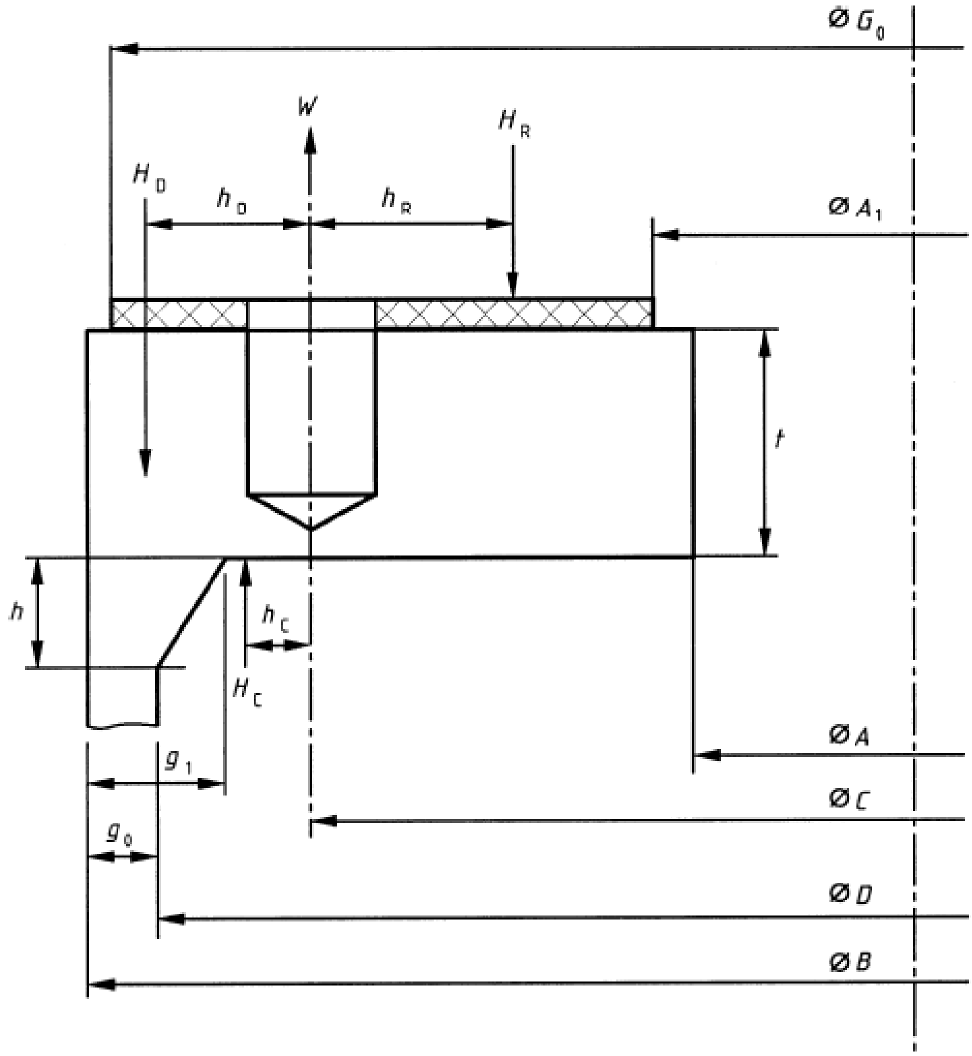


Figure D.9-2 — Reverse full face flange design to D.9.3

H_C is the pressure force on the flange face outside the bolt circle diameter;

h_C is radial distance from bolt circle to circle on which H_C acts;

The following additional equations apply:

$$H_C = H_D - \pi/4 \ P \ C^2 \quad \text{(D.9-14)}$$

$$h_C = (D - C) / 4 \quad \text{(D.9-15)}$$

The following equations replace the equations in clause D.6 for the given variable:

$$H_D = \pi/4P \ D^2 \quad \text{(D.9-16)}$$

$$h_D = (B - C - g_1) / 2 \quad \text{(D.9-17)}$$

$$M_R = H_D \ h_D - H_C \ h_C \quad \text{(D.9-18)}$$

$$W_{op} = H_D - H_C + H_R \quad \text{(D.9-19)}$$

D.10 Full face flanges with metal to metal contact

D.10.1 General

NOTE See Figure D.10-1 for an illustration of loads and dimensions.

The requirements of D.10.2 shall be applied when there is metal to metal contact both inside and outside the bolt circle before the bolts are tightened with more than a small amount of preload and the seal is provided by an O-ring or equivalent.

Manufacturing procedures and tolerances shall ensure that the flange is not dished in such a way as to give initial contact outside bolt circle.

NOTE 1 The rules are conservative where initial contact is at the bore.

NOTE 2 It is assumed that a self-sealing gasket is used approximately in line with the wall of the attached pipe or vessel and that the assembly load and any axial load from the seal may be neglected.

D.10.2 Specific symbols and abbreviations

The following symbols and abbreviations are in addition to those in clause D.3:

G is mean diameter of gasket;

H_R is the balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle;

h_R is radial distance from bolt circle to circle on which H_R acts;

M_R is balancing radial moment in flange along line of bolt holes;

n is number of bolts.

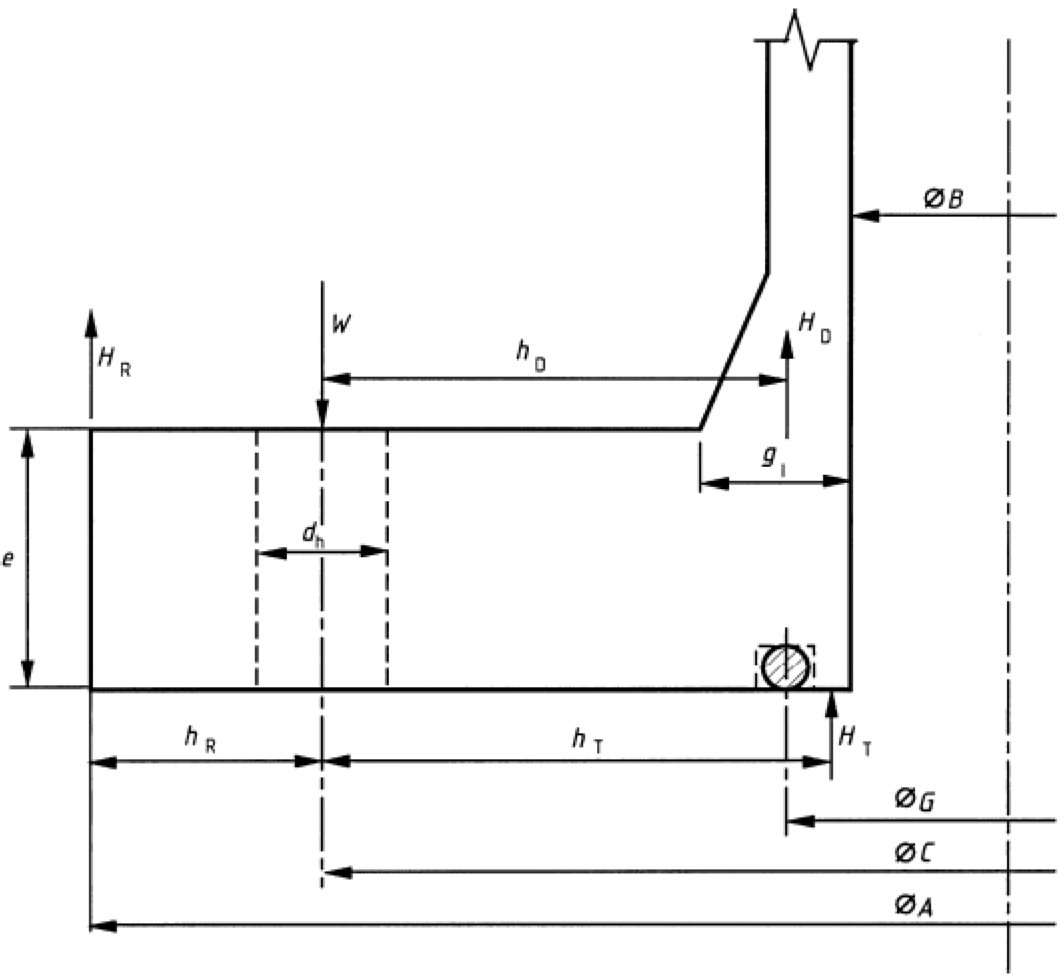


Figure D.10-1 — Flange with full face metal to metal contact and O-ring seal

D.10.3 Design

The following requirements apply where the flange is to be bolted to an identical flange or to a flat cover.

Bolt loads shall be calculated in accordance with D.5.2 taking:

$$h_R = (A - C) / 2 \quad \text{(D.10-1)}$$

$$M_R = H_D \cdot h_D + H_T \cdot h_T \quad \text{(D.10-2)}$$

$$H_R = M_R / h_R \quad \text{(D.10-3)}$$

$$W_A = 0 \quad \text{(D.10-4)}$$

$$W_{op} = H + H_R \quad \text{(D.10-5)}$$

The flange thickness shall be not less than:

$$e = \sqrt{\frac{6M_R}{f(\pi C - n\, d_h)}} \tag{D.10-6}$$

Where two flanges of different internal diameters, both designed to the rules of this clause, are to be bolted together to make a joint, the following additional requirements apply:

a) value of M_R to be used for both flanges shall be that calculated for the smaller internal diameter;

b) the thickness of the flange with the smaller bore shall be not less than:

$$t = \sqrt{\frac{3(M_1 - M_2)(A + B)}{\pi f\, B(A - B)}} \tag{D.10-7}$$

where M_1 and M_2 are the values of M_R calculated for the two flanges.

Annex E
(normative)

Design of branch connections in piping accessories

E.1 Scope

E.1.1 General

This Annex covers the design of branch connections in:

— seamless 3*d* and 5*d* elbows;

— pipe bends made from seamless or welded tubes (except those welded helicoidally).

The following are not covered by this Annex:

— moulded accessories;

— seamless 2*d* elbows.

NOTE Branch and support connections other than those covered by this Annex may be used subject to justification by calculation or by reference to satisfactory similar arrangements.

Attention is drawn to the fact that any branch increases the risks of turbulence. These phenomena shall not result in an increase in corrosion or erosion beyond the acceptable limits.

Unless specific justification the connections covered by this Annex shall fall within the limits given in Table E.1-1.

Table E.1.1-1 — Connections' limits

Pressure	≤ 40 bar
Temperature	Non insulated ≤ 200°C Insulated ≤ 350°C
Ratio of the nominal diameters	see Tables E.1.1-2 and E.1.1-3
Reinforcement	Without welded reinforcing ring (provide for an excess thickness for the accessory)
Maximum total number of equivalent cycles (see 10.3.1)	1 000

Table E.1.1-2 — Ratio of the nominal diameters – Radius of curvature *R* = 1,5 *d*

Maximum outside diameter of the branch										
Elbow : radius of curvature *R* = 1,5 *d* (Figures E.1.1-1 and E.1.1-2)										
elbow /pipe bend D_o mm		≤ 88,9	168,3	273	323,8	355,6	406,4	457,9	508	609,6
Branch d_e max	P ≤ 0,5 MPa	not allowed	26,7	33,4	60,3[a]	88,9	114,3	168,3	219,1	273
	0,5 < P ≤ 1 MPa	not allowed	26,7	33,4	33,4	60,3[a]	88,9	114,3	168,3	219,1
	P > 1 MPa	not allowed	not allowed	26,7	26,7	33,4	60,3[a]	88,9	114,3	168,3
Arrangement		Couplings or half couplings					Branches without reinforcing ring			
[a] 60,3 for tube, or half- coupling for tube d_o = 48,3										

Table E.1.1-3 — Ratio of the nominal diameters – Radius of curvature *R* >1,5

Maximum outside diameter of the branch		
Pipe bend : radius of curvature *R* > 1,5 *d* (Figure E.1.1-3)		
All D_o	$\frac{d_o}{2} + 50 \leq X \leq \frac{\sqrt{2}}{2}\left[R + \frac{Do}{e}\right] + \frac{do}{2}$	$Y \leq Do$

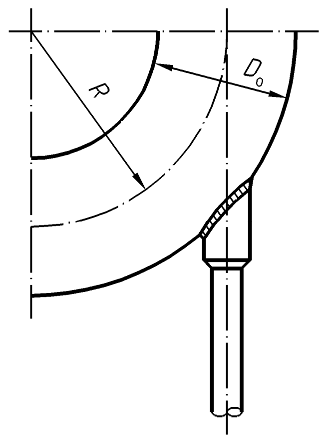


Figure E.1.1-1

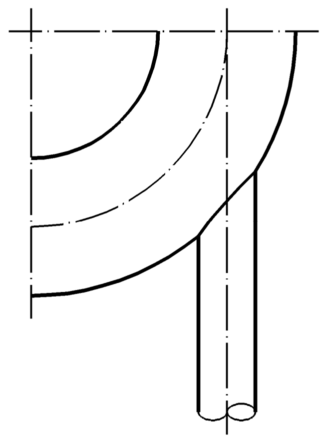


Figure E.1.1-2

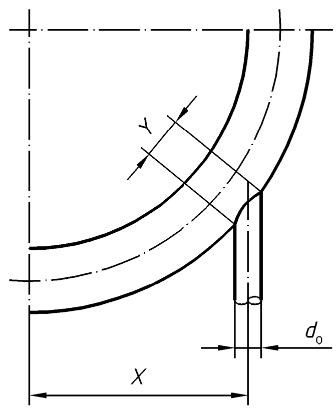


Figure E.1.1-3

E.2 Reinforcement

Connections shall be checked as follows where *L* and *l* shall be as defined in clause 8.4.

NOTE There is no need for reinforcement where the connection is made of a self-reinforcing piece (half-coupling or other) designed for the conditions of use.

E.2.1 Angles and areas

Angles and areas shall be calculated according to Figure E.2.1-1 and to the following:

E.2.1.1 Calculation of angles

$$\cos\alpha = \frac{X}{R + 0{,}5\,D\mathrm{o}} \qquad \text{(E.2.1-1)}$$

$$\cos\beta = \frac{X - 0{,}5\,d\mathrm{o}}{R + 0{,}5\,D\mathrm{o}} \qquad \text{(E.2.1-2)}$$

$$\cos\gamma = \frac{X + 0{,}5\,d\mathrm{o}}{R + 0{,}5\,D\mathrm{o}} \qquad \text{(E.2.1-3)}$$

$$A = 90° - \beta - \tau \qquad \text{(E.2.1-4)}$$

$$B = \gamma - \tau \qquad \text{(E.2.1-5)}$$

NOTE Angular area associated with an angle at the centre of 1°:

$$\Omega = \frac{\pi}{360}\left\{(R + 0{,}5\,D\mathrm{o} - 0{,}5\,e_a)^2 - R^2\right\} \qquad \text{(E.2.1-6)}$$

E.2.1.2 Area G_2(which is always greater than G_1) :

$$G_2 = \Omega\,(\alpha - \gamma + \tau) + 0{,}5(l + 0{,}5\,e_a)(d\mathrm{o} - e_{ap}) \qquad \text{(E.2.1-7)}$$

E.2.2 The following condition shall be satisfied:

$$p_c \le S_2\,\frac{f}{G_2} \qquad \text{(E.2.2-1)}$$

where *f* is the design stress defined in clause 5.

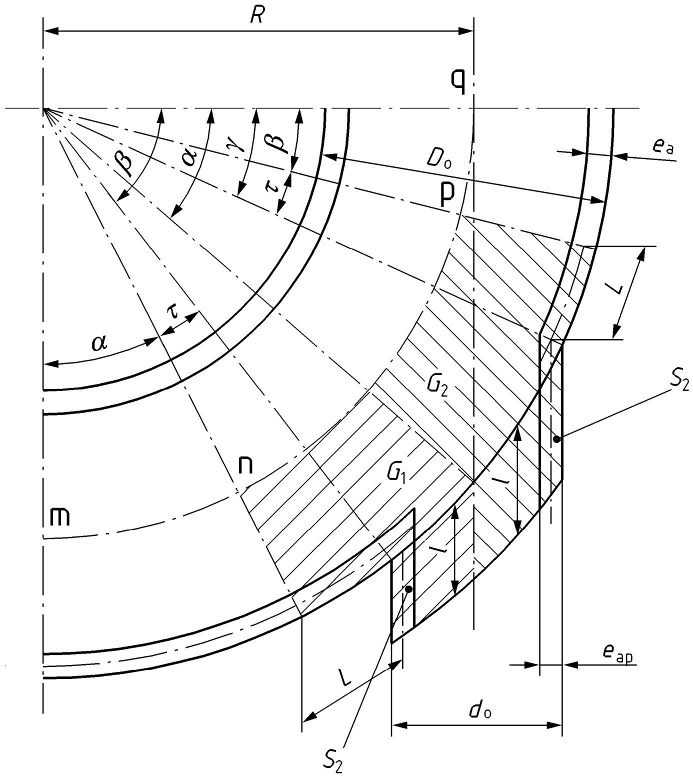


Figure E.2.1-1

E.3 Flexibility analysis

In addition to the stress intensification factor of elbow or pipe bend, a stress intensification factor due to the branch connection shall be applied to the stresses obtained from the flexibility analysis performed without the branch (see clause C.12).

The component shall be checked first according to C.12.3 and then the stresses increased by the stress intensification factor, i, shall meet the following relation:

$$i\,\sigma \leq 3\,f$$

The stress intensification factor, i, is given by the following:

$$i = 1.5\left(\frac{D_\mathrm{m}}{2e}\right)^{\frac{2}{3}}\left(\frac{d_\mathrm{m}}{D_\mathrm{m}}\right)^{\frac{1}{2}}\left(\frac{e_\mathrm{p}}{e}\right)\left(\frac{d_\mathrm{m}}{2r_\mathrm{p}}\right)$$

where

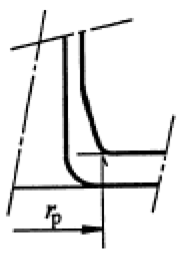
r_p is the external radius of the branch at the opening according to Figures E.3-1 to E.3-3;

e_p is the thickness of the branch;

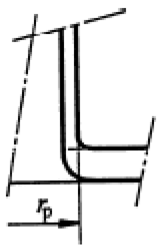
d_m is the mean diameter of the branch;

e is the thickness of the run;

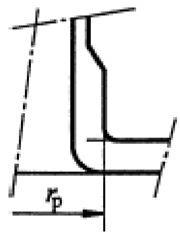
D_m is the mean diameter of the run.



Figures E.3-1



Figures E.3-2



Figures E.3-3

Annex F
(informative)

Testing during operation in the case of cyclic loading

F.1 Testing during operation

In the case of piping systems subject to cyclic loading, in-service inspections are of particular importance. They make it possible to detect incipient damage in good time. Therefore, the internal inspections should be supplemented by non-destructive examination (NDE) on highly loaded locations, especially by surface crack examination and ultrasonic examination. For monitoring of easily accessible areas, an ultrasonic examination from the outside surface of the piping can also be conducted.

Every piping system, for which the number of allowable load cycles (cycle number N) is fixed, should undergo an internal inspection at the latest after half the load cycles N have been reached. For piping designed for $N = 2 \times 10^6$ or higher, 2×10^6 should be taken. The operator is obliged to record the number of load cycles arising in a suitable fashion and, if necessary, to arrange for the internal inspections.

If the operating conditions assumed in the calculation in 10.3.2.3 deviate in terms of a greater cyclic loading or, if damage to the pressure containing wall is expected before the end of the inspection intervals due to other operation influences, the inspection intervals may be shortened. Longer inspection intervals will possibly result from calculations using detailed fatigue analysis methods.

If no incipient cracks are found during the regular inspection, the piping can be operated up to the next examination interval laid down or agreed between the parties involved, even if the allowable number of load cycles as calculated in accordance with 10.3.2.3 has already been reached or has been exceeded.

F.2 Measures to be taken when the calculated fatigue life has been reached

If the allowable number of load cycles for a component or the allowable value for overall damage according to 10.3.2.3 has been reached, NDE in accordance with 10.3.2.7 should be performed as completely as possible at a number of highly loaded locations.

If no cracks are found in the examination conducted under a continued operation is permissible. The prerequisite for this is that no fatigue damage is found in the NDE conducted in the inspection intervals which correspond to 50 % of the operating time according to 10.3.2.3. After this operating time has been reached, further procedure should be agreed between the parties involved.

Should cracks or crack-type defects or other more extensive damage be found in the examination conducted under a) or b), the component or the structural element concerned should be replaced, unless continued operation appears acceptable as a result of appropriate measures to be agreed between the parties involved.

The following design, manufacturing and process-related measures can be considered with regard to continued operation:

— removal of cracks by grinding. If the grinding leads to an insufficient wall thickness, repair welds are only to be applied by agreement between the parties involved;

— grinding the welds to remove all notches;

— change in mode of operation.

Annex G
(informative)

Physical properties of steels

G.1 General

The physical properties of steels are needed for stress analysis calculations.

G.2 Physical properties

G.2.1 Density

The density ρ depends on the temperature t. It may be calculated by

$$\rho_t = \frac{\rho_{20}}{\left[1 + \beta_{20,t} \cdot (t - 20)\right]^3} \tag{G.2-1}$$

In this equation the linear coefficient of thermal expansion from 20 °C to temperature t should be used. This is defined by

$$\beta_{20,t} = \frac{1}{l_{20}} \cdot \frac{l_t - l_{20}}{t - 20} \tag{G.2-2}$$

where

l_t is the length of a specimen at temperature t.

For the calculation of the mass of a component the density ρ_{20} at 20 °C should be used, see Table G.2.1-1.

Table G.2.1-1 — Density at 20°C

Steel group	Density ρ kg/m^3
1 to 4, 5.1 and 5.2	7 850
5.3, 5.4, 6 and 7	7 760
8.1 and 8.2	7 930

G.2.2 Differential coefficient of linear expansion

For the calculation of the thermal stress caused by a temperature difference $\Delta t = t_2 - t_1$, the differential coefficients of linear thermal expansion β_{diff,t^*} at temperature

$$t^* = 0{,}75\max(t_1, t_2) + 0{,}25\min(t_1, t_2) \tag{G.2-3}$$

should be used.

The relationship between $\beta_{20,t}$ and $\beta_{\text{diff},t}$ is

$$\beta_{\text{diff,t}} = \beta_{20,t} + \frac{\partial \beta_{20,t}}{\partial t}(t - t_0) \tag{G.2-4}$$

where:

t_o = 20 °C.

G.2.3 Specific thermal capacity

The relationship between the mean specific thermal capacity from 20 °C to temperature $C_{p,20,t}$ and the differential specific thermal capacity $C_{p,\text{diff},t}$ is (similar to the coefficient of linear thermal expansion):

$$C_{\text{p,diff,t}} = C_{\text{p,20,t}} + \frac{\partial C_{\text{p,20,t}}}{\partial t}(t - t_0) \tag{G.2-5}$$

G.2.4 Thermal diffusivity

The thermal diffusivity D_{th} is calculated by

$$D_{\text{th}} = \frac{\lambda_t}{\rho_t C_{\text{p,diff,t}}} \tag{G.2-6}$$

where λ_t is the temperature dependent thermal conductivity as given in G.5.3.

G.2.5 Poisson's ratio

The Poisson's ratio ν may be chosen for all steels independent of the temperature in the elastic state.

$$\nu = 0{,}3 \tag{G.2-7}$$

G.3 Physical properties of steels

NOTE For information on the grouping of steels see EN 13480-2.

The physical properties may be calculated by polynomials using equation (G.4-1) or may be read from Figures G.3-1 to G.3-4.

The calculated property *Z*, in units as given in the tables below for the temperature *t* in °C, is calculated by:

$$Z = c_0 + c_1 t + c_2 t^2 + c_3 t^3 + \ldots. \tag{G.3-1}$$

The polynomial coefficients are given in Tables G.3-1 to G.3-4.

t should not exceed the following limits:

ferritic steels, group 1.1 to 7: $20°C \leq t < 600°C$

austenitic stainless steels, group 8.1 and 8.2: $20°C \leq t < 800°C$.

Values of 20 °C may also be used for temperatures between 0 °C and 20 °C.

NOTE All values lie within less than 1 % of the tabulated data of the quoted literature.

When steels are selected in accordance with EN 13480-2 the physical properties can also be obtained from the relevant European Standard and linear interpolation used.

Table G.3-1 — Polynomial coefficients for modulus of elasticity E_t in kN/mm^2

	Coefficients for polynomials		
Steel group	c_0	c_1	c_2
1 to 4, 5.1 and 5.2	213,16	-6.91 E-2	-1,824 E-5
5.3, 5.4, 6 and 7	215,44	-4.28 E-2	-6,185 E-5
8.1 and 8.2	201,66	-8.48 E-2	0

Table G.3-2 — Polynomial coefficients for linear thermal expansion β_t in 10^{-6} K^{-1}

	Coefficients for polynomials		
Steel group	c_0	c_1	c_2
1 to 4, 5.1 and 5.2 $\beta_{20\ t}$	11,14	8,03 E-3	-4,29 E-6
$\beta_{diff\ t}$	10,98	1,623 E-2	-1,287 E-5
5.3, 5.4, 6 and 7 $\beta_{20\ t}$	10,22	5,26 E-3	-2,5 E-6
$\beta_{diff\ t}$	10,11	1,062 E-2	-7,5 E-6
8.1 and 8.2 $\beta_{20\ t}$	15,13	7,93 E-3	-3,33 E-6
$\beta_{diff\ t}$	14,97	1,599 E-2	-9,99 E-6

Table G.3-3 — Polynomial coefficients for thermal conductivity λ_t in W/mK

	Coefficients for polynomials		
steel group	c_0	c_1	c_2
1.1	55,72	-2,464 E-2	-1,298 E-5
1.2	49,83	-1,613 E-2	-1,372 E-5
2.1	39,85	1,111 E-2	-3,611 E-5
4	46,85	7,2 E-4	-3,305 E-5
5.1	45,0	-1,287 E-2	-1,075 E-5
5.2	36,97	6,40 E-3	-2,749 E-5
5.3 and 5.4	28,05	1,85 E-3	-5,58 E-6
6	22,97	8,73 E-3	-4,82 E-6
8.1 and 8.2	13,98	1,502 E-2	0

Table G.3-4 —Polynomial coefficients for specific thermal capacity $C_{p,xx,t}$ in J/kgK

	Coefficients for polynomials				
steel group	c_0	c_1	c_2	c_3	c_4
1 to 5 $C_{p,20\ t}$	454, 93	0,28139	-3,8815 E-4	4,7542 E-7	0
$C_{p\ diff\ t}$	449,30	0,57830	-1,1930 E-3	1,9017 E-6	0
6 $C_{p\ 20\ t}$	433,33	0,43342	-7,4702 E-4	8,0289 E-7	0
$C_{p\ diff\ t}$	424,66	0,89672	-2,2892 E-3	3,2116 E-6	0
8.1 and 8.2 $C_{p\ 20\ t}$	467,77	0,24905	-5,5393 E-4	8,3266 E-7	-4,3916 E-10
$C_{p\ diff\ t}$	462,69	0,52026	-1,7117 E-3	3,3658 E-6	-2,1958 E-9

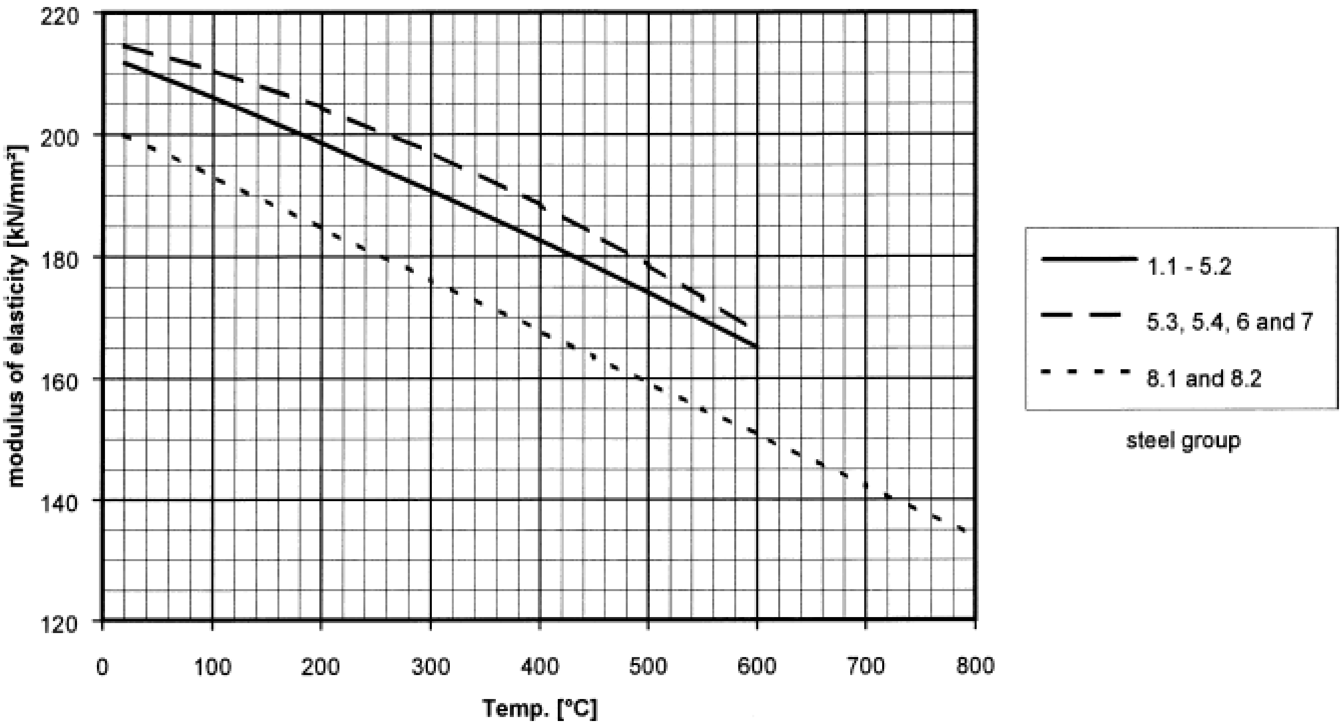


Figure G.3-1 — Modulus of elasticity E_t for steel

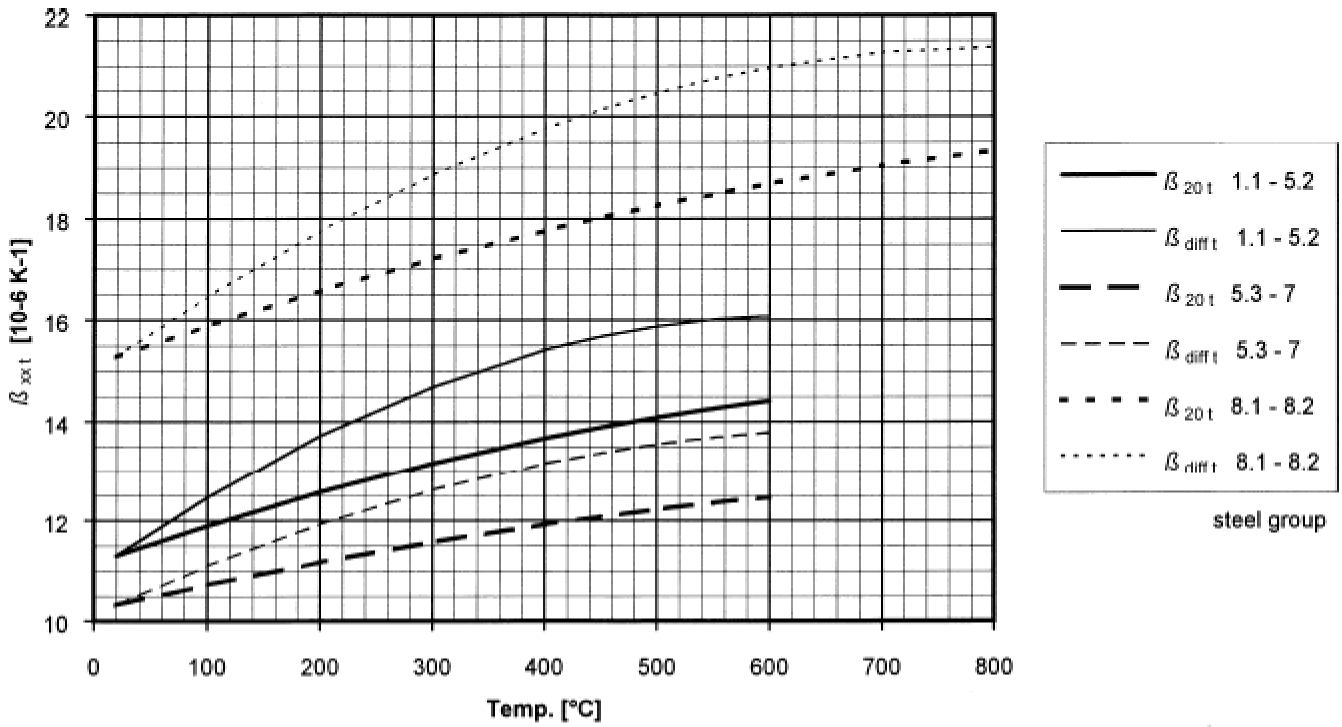


Figure G.3-2 — Coefficient of linear thermal expansion β_t

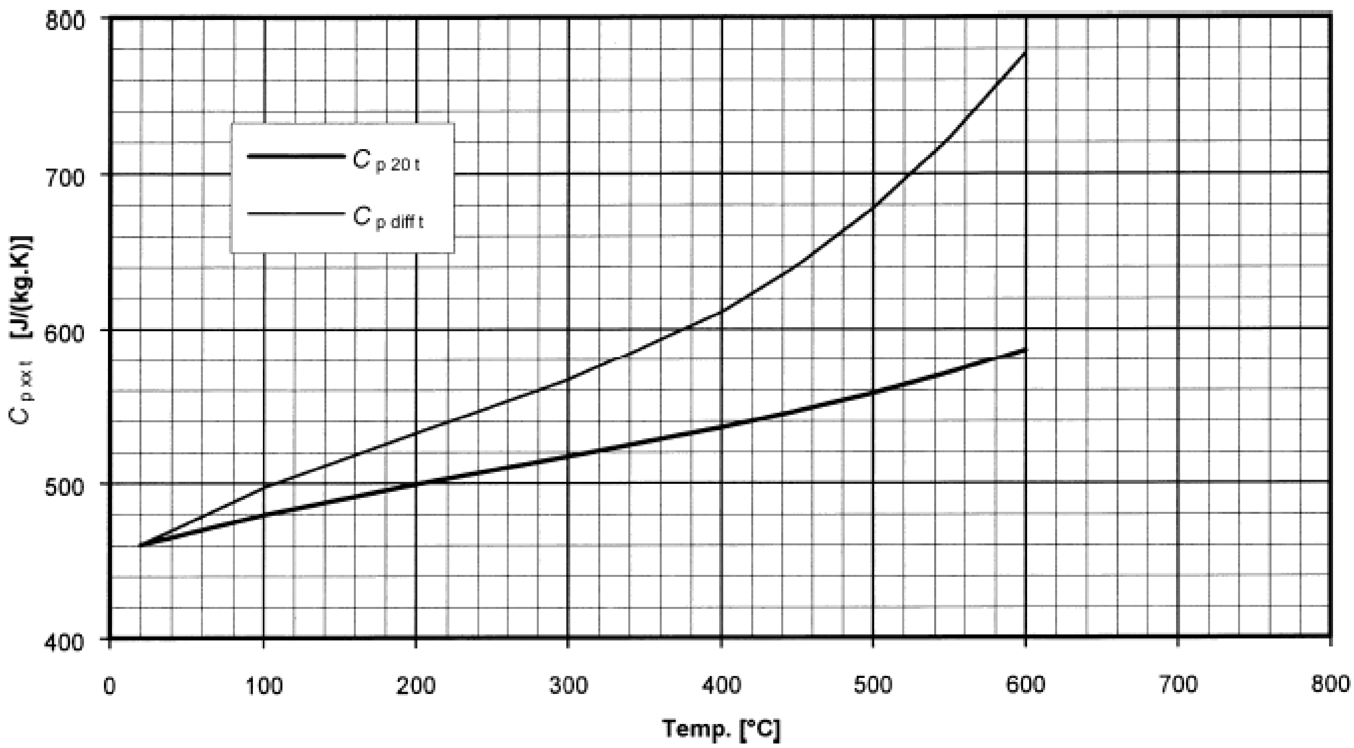


Figure G.3-3a — Specific thermal capacity for steel, groups 1 to 5.3

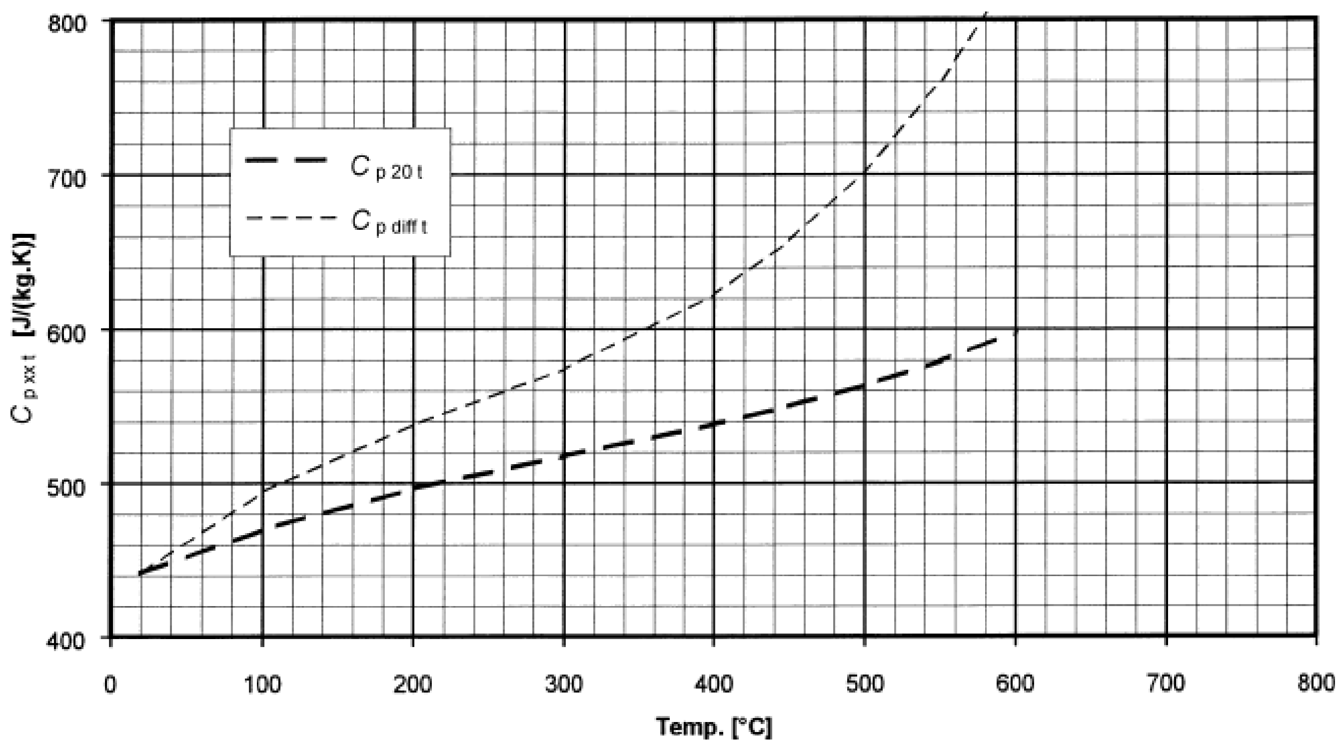


Figure G.3-3b — Specific thermal capacity for steel, groups 6.1 to 6.4

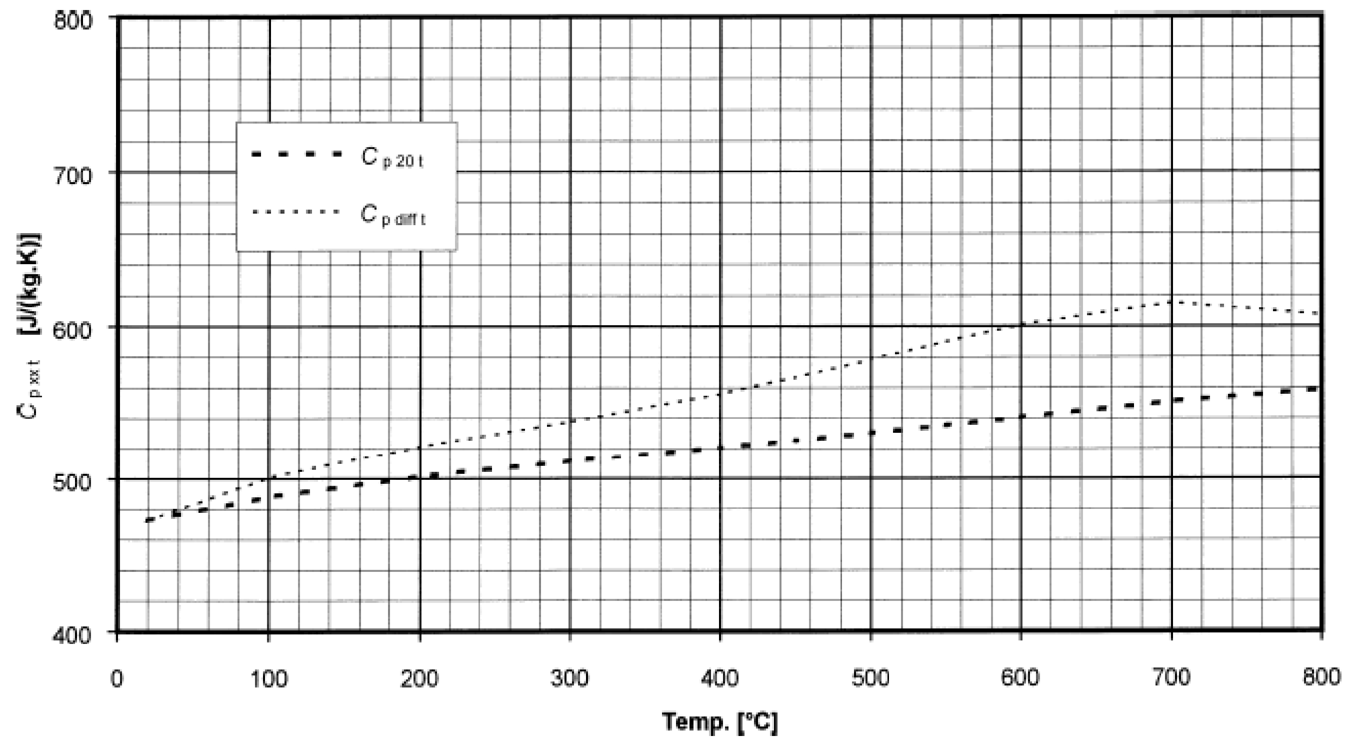


Figure G.3-3c —Specific thermal capacity for steel, groups 8.1 and 8.2

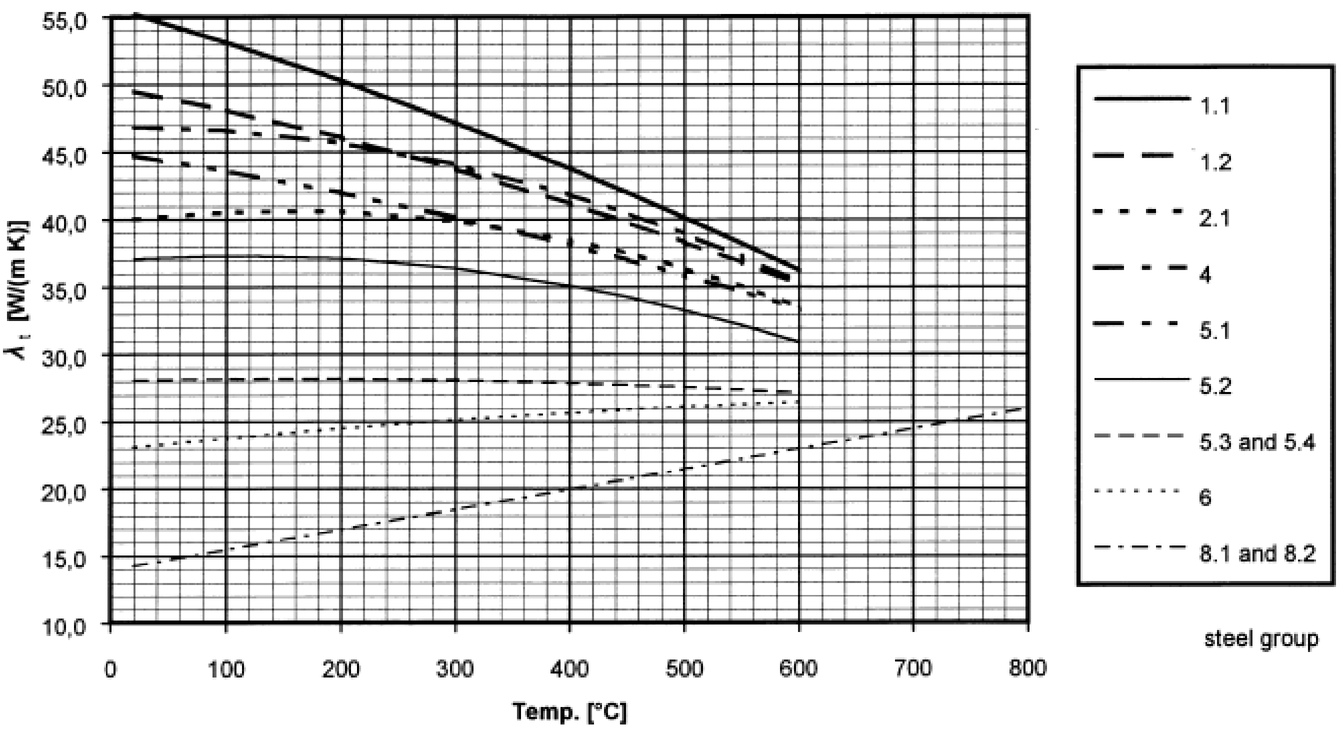


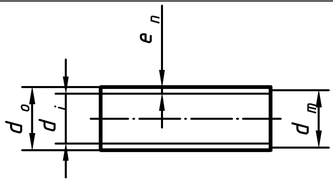
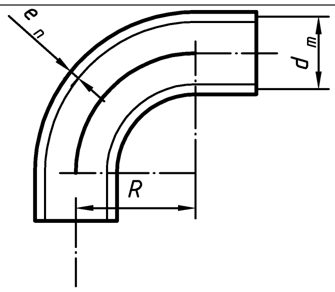
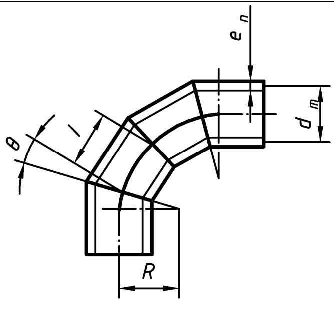
Figure G.3-4 — Coefficient of thermal conductivity λ_t

Annex H
(normative)

Flexibility characteristics, flexibility and stress intensification factors and section moduli of piping components and geometrical discontinuities

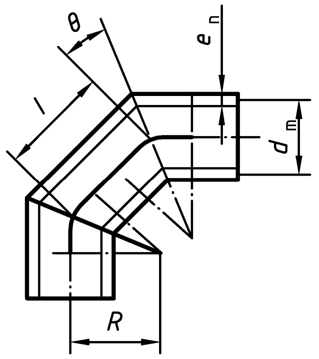
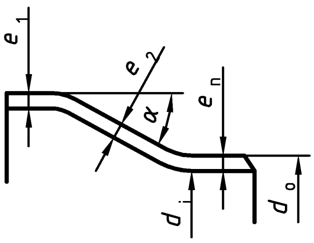
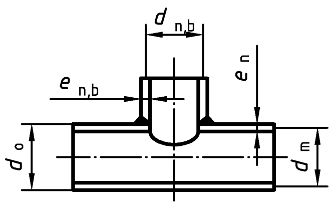
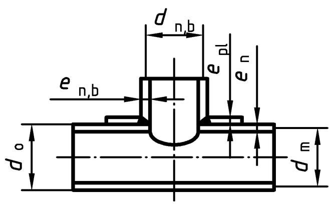
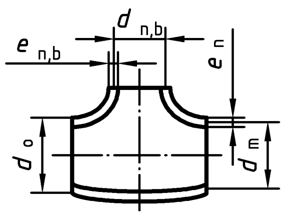
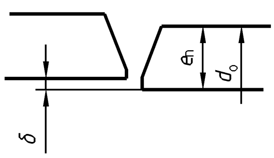
Piping component and geometrical disconuities characteristics for general cases, particular connections, and out of plane and in plane bending of the piping system shall be in accordance with Tables H.1 to H.3.

Table H.1 — Flexibility characteristics, flexibility and stress intensification factors and section moduli for general cases

N°	Designation	Sketch	Flexibility characteristic h	Flexibility factor k_B [a]	Stress intensification factor i	Section modulus Z
1	straight pipe		1	1	1	
2	plain bend		$\frac{4Re_n}{d_m^2}$	$\frac{1{,}65}{h}$	$\frac{0{,}9}{h^{2/3}}$ b c h i	$\frac{\pi}{32}\frac{d_o^4-d_i^4}{d_o}$
3	Closely spaced mitre bend $l < r\,(1 + \tan\theta)$ ($l = 2\,R\tan\theta$)		$\frac{4Re_n}{d_m^2}$ with $R = \frac{l\cot\theta}{2}$	$\frac{1{,}52}{h^{5/6}}$	$\frac{0{,}9}{h^{2/3}}$ b c h i	

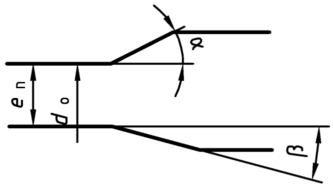
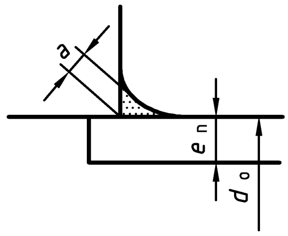
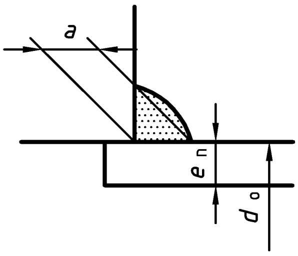
(to be continued)

Table H.1 *(continued)*

N°	Designation	Sketch	Flexibility characteristic *h*	Flexibility factor k_B [a]	Stress intensification factor *i*	Section modulus *Z*
4	Single mitre bend or widely spaced mitre bend $l \geq r\,(1 + \tan\theta)$		$\dfrac{4Re_n}{d_m^2}$ with $R = \dfrac{d_m(1+\cot\theta)}{4}$	$\dfrac{1{,}52}{h^{5/6}}$	$\dfrac{0{,}9}{h^{2/3}}$ [b h i]	
5	forged welded-in reducer		Shape conditions : $\alpha \leq 60°$ $e_n \geq d_o/100$ $e_2 \geq e_1$	1	$0{,}5 + \dfrac{\alpha}{100}\left(\dfrac{d_o}{e_n}\right)^{1/2}$ max. 2,0 (α in deg.) [d]	
6	tee with welded-on, welded-in or extruded nozzle		$\dfrac{2e_n}{d_m}$	1	$\dfrac{0{,}9}{h^{2/3}}$ [b e g]	Header $\dfrac{\pi}{32}\,\dfrac{d_o^4 - d_i^4}{d_o}$
7	as above, however, with additional reinforcing ring		$\dfrac{2(e_n + 0{,}5e_{pl})^{5/2}}{d_m e_n^{3/2}}$ with $e_{pl} \leq e_n$	1	$\dfrac{0{,}9}{h^{2/3}}$ [b e g]	Nozzle $\dfrac{\pi}{4} d_{m,b}^2 e_x$
8	forged welded-in tee with e_n and $e_{n,b}$ as connecting wall thickness		$\dfrac{8{,}8e_n}{d_m}$	1	$\dfrac{0{,}9}{h^{2/3}}$ [b g]	with e_x as smaller value of $e_{x1} = e_n$ and $e_{x2} = i\,e_{n,b}$ resp.
9	butt weld		$e_n \geq 5$ mm and $\delta \leq 0{,}1\,e_n$ [f] $e_n < 5$ mm or $\delta > 0{,}1\,e_n$ [f]	1 1	1,0 [f] 1,8 [f]	

(to be continued)

Table H.1 *(concluded)*

N°	Designation	Sketch	Flexibility characteristic h	Flexibility factor k_B [a]	Stress intensification factor i	Section modulus Z
10	wall thickness transitions		$\alpha \leq 30°$ $\beta \leq 15°$ (without circumferential weld at transitions $\delta = 0$)	1	1,3 + $0{,}0036 \dfrac{d_o}{e_n}$ + $3{,}6 \dfrac{\delta}{e_n}$ max 1,9 [f]	$\dfrac{\pi}{32} \dfrac{d_o^4 - d_i^4}{d_o}$
11	fillet welds at set-in connections		concave shape with continuous transition to pipe	1	1,3	smaller value of $\dfrac{\pi}{32} \dfrac{d_o^4 - d_i^4}{d_o}$ and $\dfrac{\pi}{4} d_o^2 a$
12				1	2,1	

[a] The flexibility factor k_B applies to bending in all planes. The factor related to torsion is equal to 1 in all cases.

[b] The factors k_B and i apply over the whole effective length of the elbows and bends and at the intersection of the axes in case of tees and nozzles.

[c] If these components are fitted with :

- flange at one extremity, k_B and i are multiplied by $h^{1/6}$;
- flange at each of the extremities, k_B and i are multiplied by $h^{1/3}$.

[d] The wall thickness of the reducer is not less than e_1 except in the vicinity of the small end where however the thickness is not less than e_n.

[e] Other values may be used subject to justification.

[f] The factor applies if the fabrication tolerances are met. Otherwise the determination of the factors is the responsibility of the designer.

[g] The factors only apply to nozzles with convergent axes, not applicable for instance for configurations according to Figure 8.4.3-5.

[h] If the pressure is likely to correct ovality (large diameter, small thickness), the factor i shall be divided by:

$$1 + 3{,}25 \left(\frac{p_o}{E_c}\right)\left(\frac{d_m}{2e_n}\right)^{5/2}\left(\frac{2R}{d_m}\right)^{2/3}$$

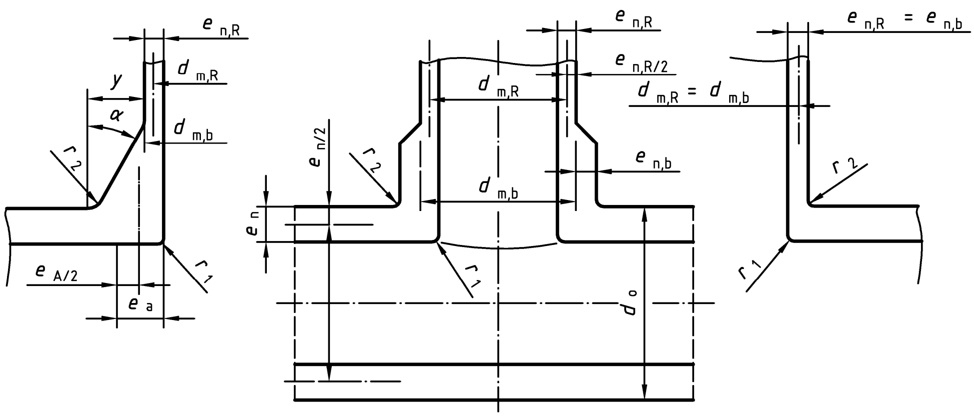
, where p_o is the operating pressure and E_c the modulus of elasticity at room temperature (20 °C).

[i] If the pressure is likely to correct ovality (large diameter, small thickness), the factor k shall be divided by:

$$1 + 6 \left(\frac{p_o}{E_c}\right)\left(\frac{d_m}{2e_n}\right)^{7/3}\left(\frac{2R}{d_m}\right)^{1/3}$$

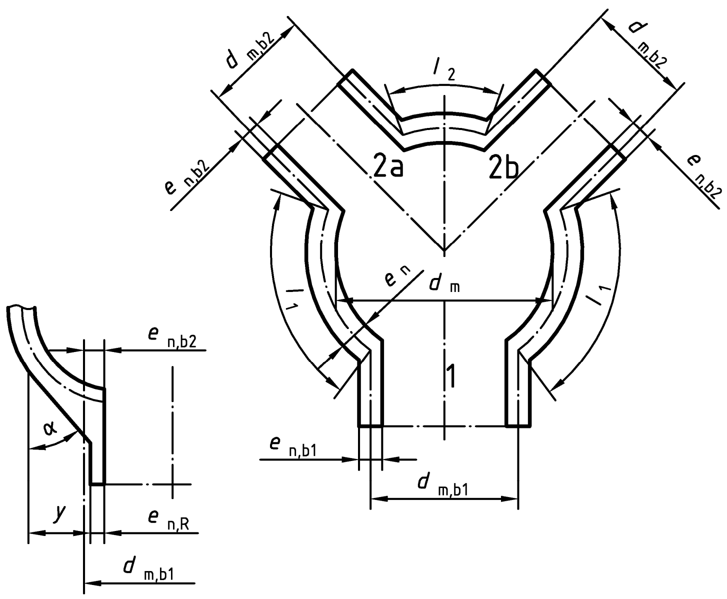
, where p_o is the operating pressure and E_c the modulus of elasticity at room temperature (20 °C).

Table H.2 — Stress intensification factors and section moduli for particular connections

Designation	Tee with special shape conditions
sketch	 $e_{n,b} = e_{n,R} + 2Y/3$ $d_{o,b} = d_{m,b} + e_{n,b}$
shape conditions	$\frac{d_{m,R}}{d_m} \leq 0{,}5$; $\frac{d}{e_n} \leq 100$; $0{,}1 e_n \leq r_1 \leq 0{,}5 e_n$ $r_2 \geq \max\left(\frac{e_{n,b}}{2}; \frac{e_n}{2}\right)$ $\alpha \leq 30°$ $r_3 \geq \max\left\{\alpha \frac{d_{m,R} + e_{n,R}}{500}; 2\sin^3 \alpha \left(d_{m,b} + e_{n,b} - d_{m,R} - e_{n,R}\right)\right\}$ For the conditions of r_3 α shall be in deg. For branches DN < 100 the conditions for r_1 can be omitted.

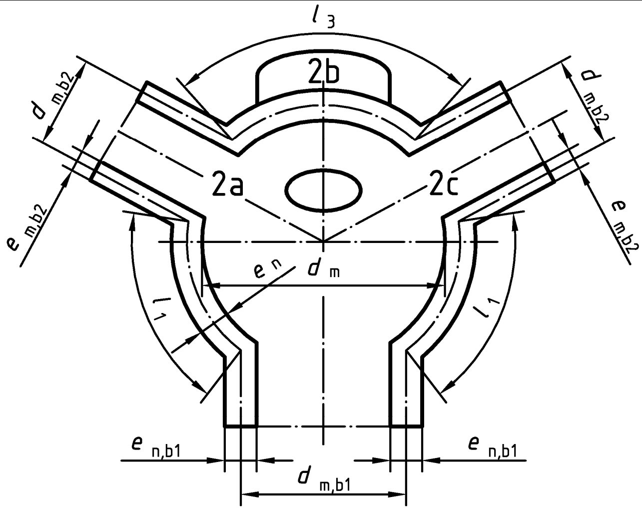
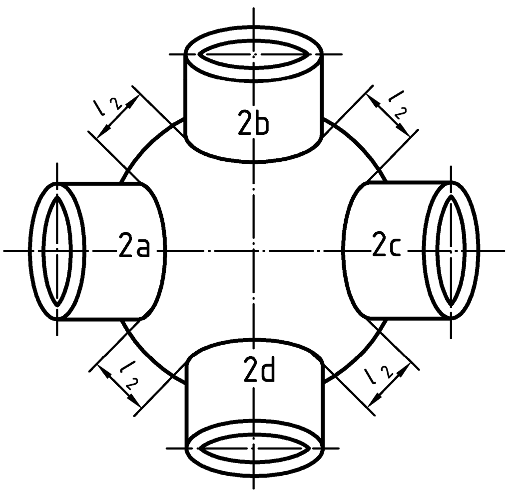
(to be continued)

Table H.2 *(continued)*

Designation	Y-spherical fitting	
	for header :	for branch :
stress intensification factors and section moduli	$i = 0{,}4\left(\frac{d_m}{2e_n}\right)^{\frac{2}{3}} \times \frac{d_{m,R}}{d_m}$ but at least *i* = 1,5 $Z = \frac{\pi}{4} d_m^2 e_n$	$i = 1{,}5\left(\frac{d_m}{2e_n}\right)^{\frac{2}{3}}\left(\frac{d_{m,R}}{d_m}\right)^{\frac{1}{2}} \times \frac{e_{n,R}}{e_n} \times \frac{d_{m,R}}{d_{m,b} + e_{n,b}}$ $Z = \frac{\pi}{4} d_{Rm}^2 e_{n,R}$
sketch		$e_{n,b1} = e_{n,R} + 2y/3$ $d_o = d_m + e_n$ $d_{o,b1} = d_{m,b1} + e_{n,b1}$ $d_{o,b2} = d_{m,b2} + e_{n,b2}$
factors of influence l_o, λ_1, λ_2	$l_o = 2\sqrt{d_m e_n}; \lambda = 1 - \sqrt{\frac{l_1}{l_o}}; \lambda = 1 - \sqrt{\frac{l_2}{l_o}}$ for $l_1 \geq l_o$, $\lambda_1 = 0$ and for $l_2 \geq l_o$, $\lambda_2 = 0$	
stress intensification factor *i*	$i = \frac{0{,}9}{h^{2/3}}$ with $h = \frac{2e_n}{d_m}$	
section moduli Z_1 Z_2	Nozzle 1 $Z_1 = \pi d_{m,b1}^2 e_{x1}/4$ with $e_{x1} = \min(e_n; ie_{n,b1})$	Nozzle 2a and 2b $Z_2 = \pi d_{m,b2}^2 e_{x2}/4$ with $e_{x2} = \min(e_n; ie_{n,b2})$

(to be continued)

Table H.2 *(continued)*

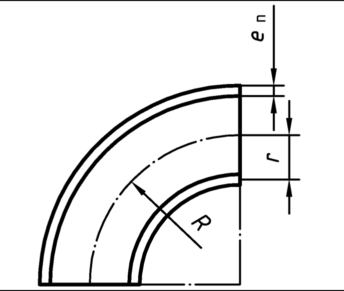
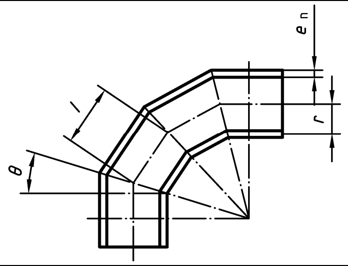
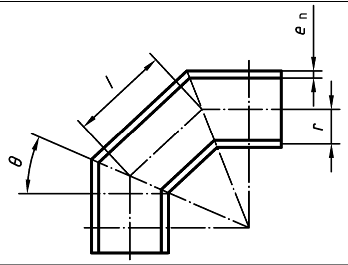
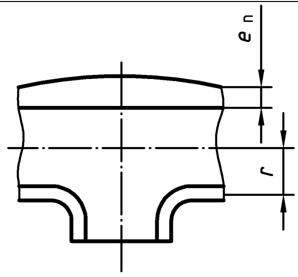
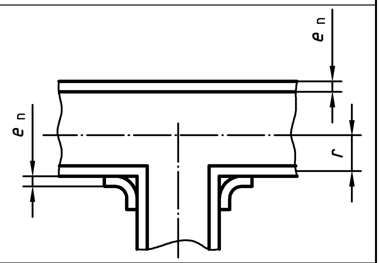
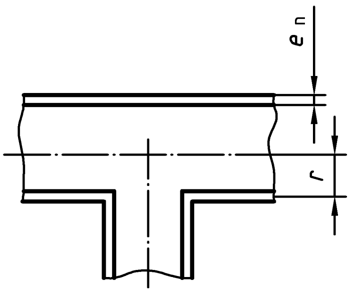
Designation	Double Y-spherical fitting		
substitute values	Nozzle 1	Nozzle 2a	Nozzle 2b
for the terms $\frac{M_A}{Z}\frac{M_B}{Z}$ $\frac{M_C}{Z}\frac{M_D}{Z}$	$\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2a}}{Z_2}$ $\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2b}}{Z_2}$	$\frac{M_{2a}}{Z_2}+\frac{\lambda_1 M_1}{Z_1}$ $\frac{M_{2a}+\lambda_2 M_{2b}}{Z_2}$	$\frac{M_{2b}}{Z_2}+\frac{\lambda_1 M_1}{Z_1}$ $\frac{M_{2b}+\lambda_2 M_{2a}}{Z_2}$
in equations given in 12.3.2 to 12.3.6	The greater value obtained from the two formulas shall be inserted in equations given in 12.3.2 to 12.3.6 for the respective branches instead of M/Z. Then the following applies : $M_1 = M_{A,B,C,D}$ at nozzle 1, $M_{2A} = M_{A,B,C,D}$ at nozzle 2a and $M_{2B} = M_{A,B,C,D}$ at nozzle 2b.		
sketch factors of influence l_o, λ_1, λ_2, λ_3 stress intensification factor i section moduli Z_1, Z_2	 		$l_o = 2\sqrt{d_m e_n}$ $\lambda_1 = 1-\sqrt{\frac{l_1}{l_o}}$ $\lambda_2 = 1-\sqrt{\frac{l_2}{l_o}}$ $\lambda_3 = 1-\sqrt{\frac{l_3}{l_o}}$ if $l_{1,2,3} \geq l_o$,then $\lambda_{1,2,3} = 0$ $i = \frac{0,9}{h^{2/3}}$ with $h = \frac{2e_n}{d_m}$ $e_{x1} = \min(e_n; ie_{n,b1})$ $Z_1 = \pi d_{m,b1}^2 e_{x1} / 4$ $e_{x2} = \min(e_n; ie_{n,b2})$ $Z_2 = \pi d_{m,b2}^2 e_{x2} / 4$

(to be continued)

Table H.2 *(concluded)*

Designation	Double Y-spherical fitting		
substitute values for the	Nozzle 1	Nozzle 2a	Nozzle 2b, c ,d
terms $\frac{M_A}{Z}\frac{M_B}{Z}$ $\frac{M_C}{Z}\frac{M_D}{Z}$ in equations given in 12.3.2 to 12.3.6	$\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2a}}{Z_2}$ $\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2b}}{Z_2}$ $\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2c}}{Z_2}$ $\frac{M_1}{Z_1}+\frac{\lambda_1 M_{2d}}{Z_2}$	$\frac{M_{2a}}{Z_2}+\frac{\lambda_1 M_1}{Z_1}$ $\frac{M_{2a}+\lambda_2 M_{2b}}{Z_2}$ $\frac{M_{2a}+\lambda_3 M_{2c}}{Z_2}$ $\frac{M_{2a}+\lambda_2 M_{2d}}{Z_2}$	$M_{2a} \cong M_{2b,c,d}$ $M_{2b} \cong M_{2c,d,a}$ $M_{2a} \cong M_{2d,a,b}$ $M_{2a} \cong M_{2a,b,c}$

Table H.3 — Flexibility characteristics and stress intensification factors for out-of-plane and in-plane bending

Component description	Out-of-plane i_o	In-plane i_i	Flexibility characteristic	Sketch
Welding elbow or pipe bend	$\frac{0{,}75}{h^{2/3}}$ [a,b,c,j]	$\frac{0{,}9}{h^{2/3}}$ [a,b,c,j]	$\frac{e_n R}{r^2}$	
Closely spaced mitre bend $l < r(1 + \tan\theta)$ ($l = 2R\tan\theta$)	$\frac{0{,}9}{h^{2/3}}$ [a,b,c,j]	$\frac{0{,}9}{h^{2/3}}$ [a,b,c,j]	$\frac{\cot\theta}{2}\,\frac{e_n l}{r^2}$	
Single mitre bend or widely spaced mitre bend $l \geq r(1 + \tan\theta)$	$\frac{0{,}9}{h^{2/3}}$ [a,b,c,j]	$\frac{0{,}9}{h^{2/3}}$ [a,b,c,j]	$\frac{e_n}{r}\left(\frac{1+\cot\theta}{2}\right)$	
Forged tee to be welded, designed with a burst pressure greater than or equal to the burst pressure of the connected pipes	$\frac{0{,}9}{h^{2/3}}$ [a,e,f,g,i]	$0{,}75 i_o + 0{,}25$ [a,e,f,g,i]	$\frac{4{,}4 e_n}{r}$	
Reinforced fabricated tee with pad or saddle	$\frac{0{,}9}{h^{2/3}}$ [a,d,e,i]	$0{,}75 i_o + 0{,}25$ [a,d,e,i]	$\frac{(e_n + 0{,}5 e_r)^{5/2}}{r\left(e_n^{3/2}\right)}$	
Unreinforced fabricated tee	$\frac{0{,}9}{h^{2/3}}$ [a,d,e,i]	$0{,}75 i_o + 0{,}25$ [a,d,e,i]	$\frac{e_n}{r}$	

(to be continued)

Table H.3 (*continued*)

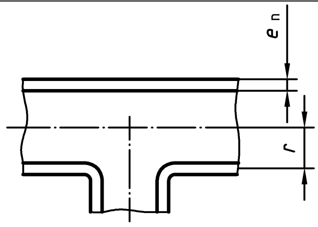
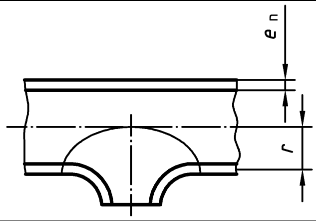
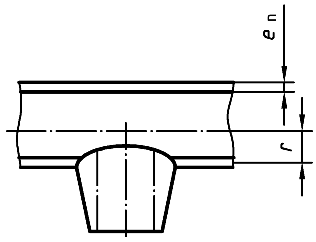
Component description	Out-of-plane i_o	In-plane i_i	Flexibility characteristic	Sketch
Extruded welding tee	$\frac{0{,}9}{h^{2/3}}$ [a e i]	$0{,}75 i_o + 0{,}25$ [a e i]	$\left(1 + \frac{r_1}{r}\right) \frac{e_n}{r}$	
Welded in contour insert	$\frac{0{,}9}{h^{2/3}}$ [a e f g i]	$0{,}75 i_o + 0{,}25$ [a e f g i]	$\frac{4{,}4 e_n}{r}$	
Branch welded on fitting (integrally reinforced)	$\frac{0{,}9}{h^{2/3}}$ [a d f h]	$0{,}75 i_o + 0{,}25$ [a d f h]	$\frac{3{,}3 e_n}{r}$	

Table H.3 (*concluded*)

[a] The factors i_O and i_i apply over the whole effective length of the elbows and bends and at the intersection of the axes in case of tees and nozzles. [b] If these components are fitted with : - flange at one extremity, i_O and i_i are multiplied by $h^{1/6}$; - flange at each of the extremities, i_O and i_i are multiplied by $h^{1/3}$. [c] If the pressure is likely to correct ovality (large diameter, small thickness), the factors i_O and i_i shall be divided by: $1 + 3{,}25 \left(\frac{p_o}{E_c}\right)\left(\frac{r}{e_n}\right)^{5/2}\left(\frac{R}{r}\right)^{2/3}$, where p_o is the operating pressure and E_c the modulus of elasticity at room temperature (20°C). [d] For a nozzle with a ratio of branch diameter to pipe diameter exceeding 0,5, the out-of-plane stress intensification factor may be non-conservative. In addition a smooth transition by a concave shaped weld is proved to reduce the value of this factor. Consequently the selection of an appropriate value for this factor remains the responsibility of the designer. [e] The stress intensification factors regarding the branch connections are based on tests carried out with at least two diameters of straight pipe on either side of the branch axis. The case of closer branches requires a particular attention. [f] The forgings shall be suitable with regard to the operating conditions. [g] When the limitations with respect to radius and thickness are not met and reliable data are not available, the flexibility characteristic is taken as $\frac{e_n}{r}$. [h] The designer shall check that the design against pressure is at least equivalent to that for a straight pipe. [i] The factors only apply to nozzles with convergent axes, and is not applicable for instance for configurations according to Figure 8.4.3-5. [j] If the pressure is likely to correct ovality (large diameter, small thickness), the factor k shall be devided by: $1+6 \left(\frac{p_o}{E_c}\right)\left(\frac{r}{e_n}\right)^{7/3}\left(\frac{R}{r}\right)^{1/3}$, where p_o is the operating pressure and E_c the modulus of elasticity at room temperature (20°C).

Annex I
(informative)

Production testing of spring supports and shock arrestors (shock absorbers)

I.1 Constant load supports

Features for the operation of constant load supports during testing are shown in the force travel characteristics (see Figure I.1-1).

The vertical compression and vertical tension force deviations should not exceed 5 % of the design load F_D, i.e.

$$\frac{\left|F_D - F_{act,d,max}\right|}{F_D} \le 0{,}05 \quad \text{and} \quad \frac{\left|F_D - F_{act,u,min}\right|}{F_D} \le 0{,}05$$

where

F_{act} is the actual force measured by the manufacturer's test;

$F_{act,d,max}$ is the maximum actual force in downwards travel (-);

$F_{act,d,min}$ is the minimum actual force in downwards travel (-);

$F_{act,u,max}$ is the maximum actual force in upwards travel (+);

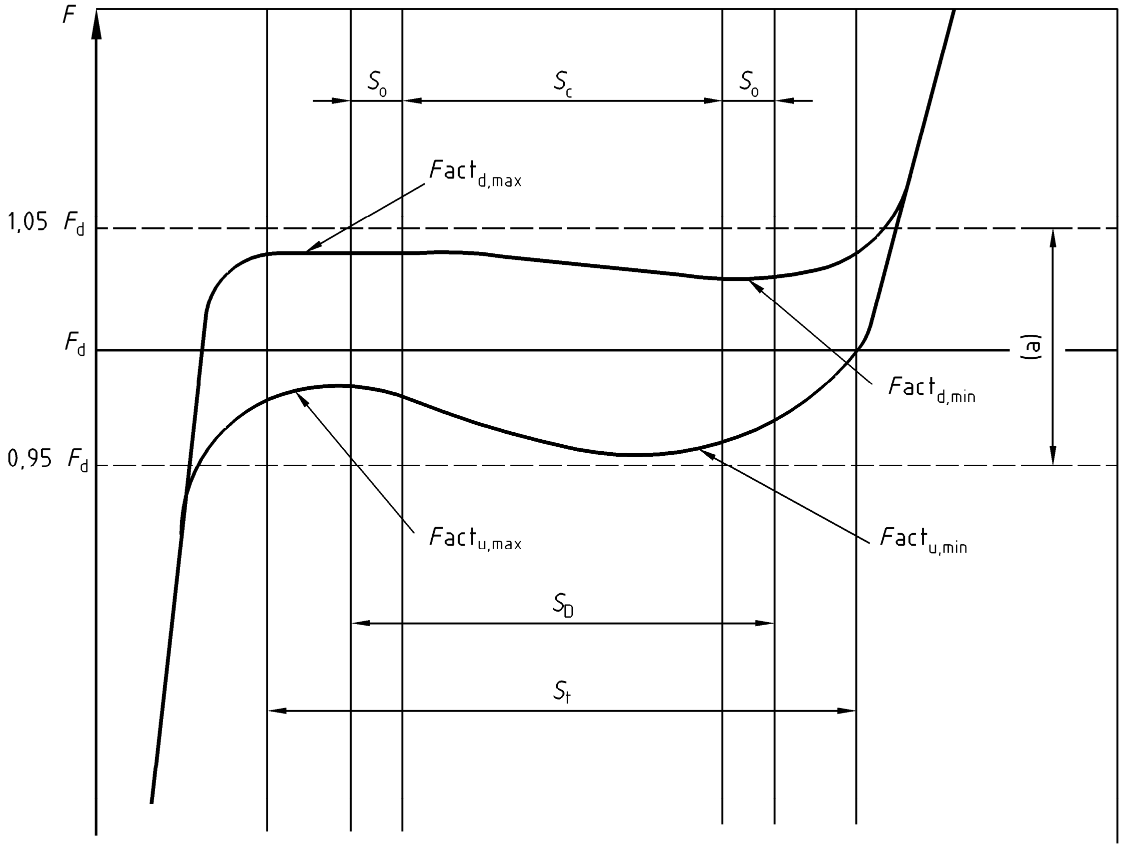
$F_{act,u,min}$ is the minimum actual force in upwards travel (+).

I.2 Variable spring supports

Features for the operation of variable spring supports during testing are shown in the force travel Figure I.2-1.

I.3 Shock arrestors

Features for the operation of shock arrestors and typical operating characteristics demonstrated during testing are shown in Figure I.3-1.



Key

(a) tolerance field

S_C = calculated travel

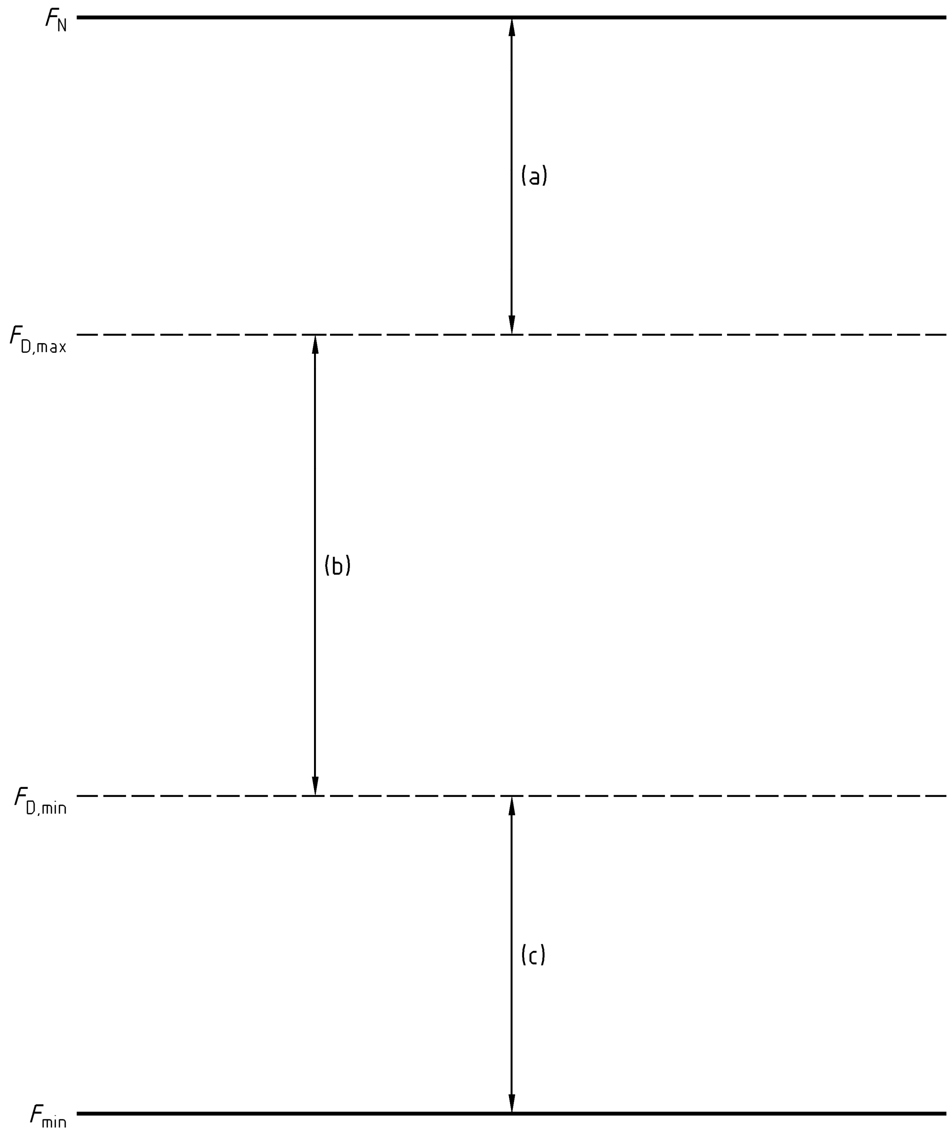
$\sum S_O$ = required overtravel

S_t = total designed travel

S_D = design travel

F_D = design load, including the weight of ancillary components where appropriate

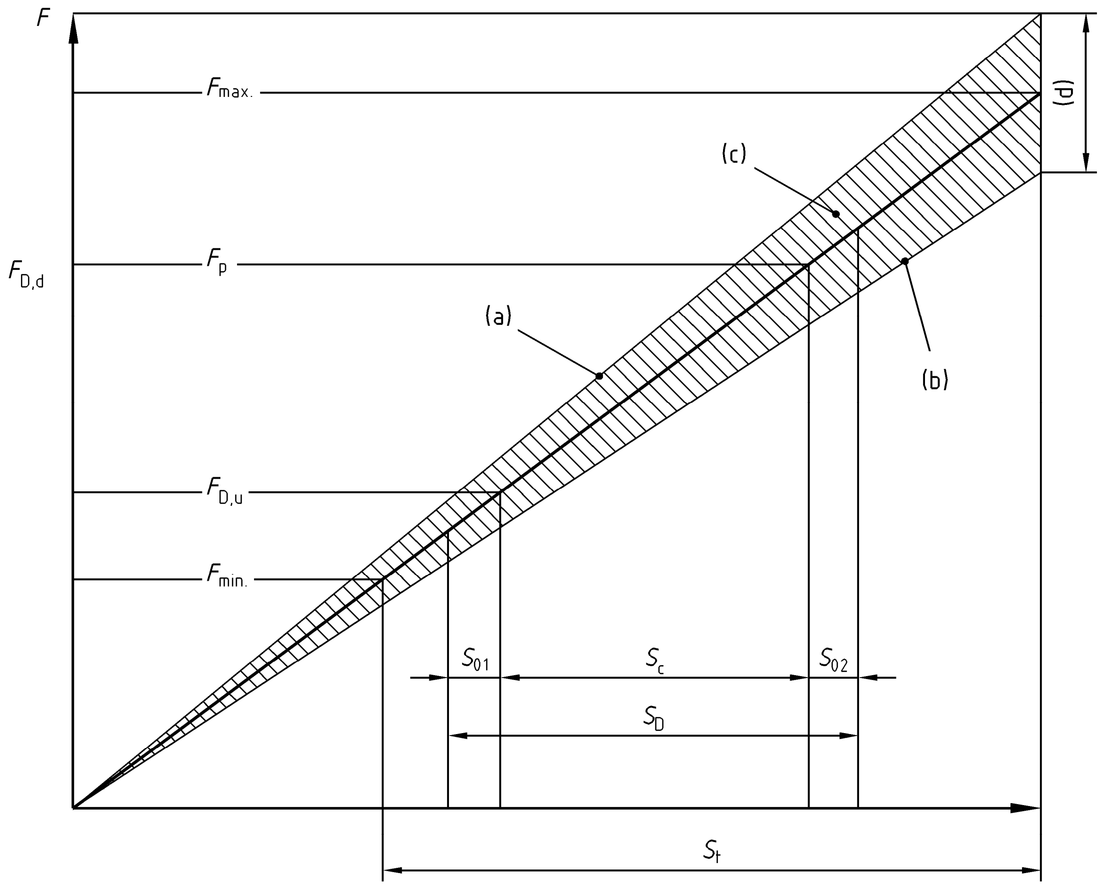
Figure I.1-1 — Force-travel characteristic for constant load supports (hangers)



Key

$F_{D.max}$ maximum adjustment load
$F_{D.min}$ minimum adjustment load
F_N maximum hanger load (nominal load) $F_N = 1,15\ F_{D.max}$
F_{min} minimum hanger load $F_{min} = 0,85\ F_{D.min}$
(a) possible re-adjustment +15 %
(b) scheduled adjustment range
(c) possible re-adjustment -15 %

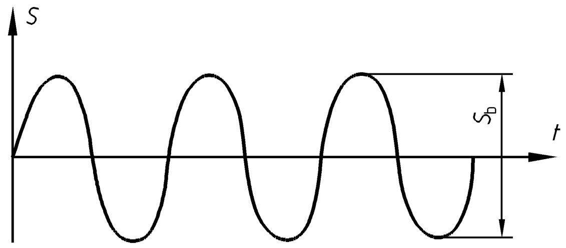
Figure I.1-2 — Adjustment range for constant load support (hangers)



Key

(a)	Downward movement
(b)	Upward movement
(c)	Tolerance field
(d)	Permitted tolerance of maximum spring load
F_P	Preset load
F_{max}	Maximum spring load
F_{min}	Minimum spring load
$F_{D,d}$	Design load (cold - including the weight of ancillaries) for downward pipe movement
$F_{D,u}$	Design load (hot - including the weight of ancillaries) for upward pipe movement
S_c	Calculated travel
$\sum S_o = S_{o1} + S_{o2}$	Total overtravel required
S_t	Total designed travel
S_D	Design travel

Figure I.2-1— Allowable force-displacement characteristic for variable spring supports (hangers)



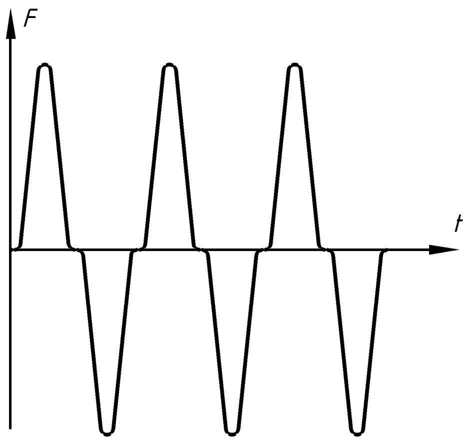
Key

S Piston rod movement

t Time

S_b Oscillating width

a) Piston rod movement — Time graph

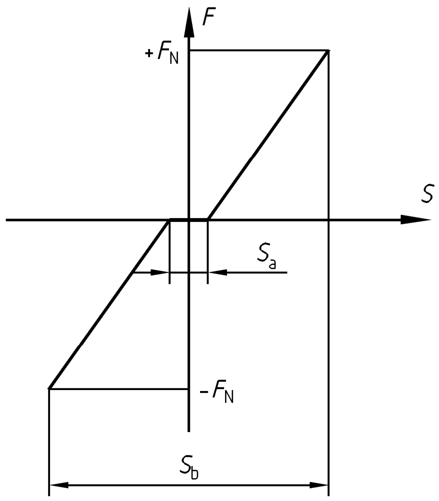


Key

F Force

t Time

b) Force — Time graph



Key

F Force

F_N Nominal force

S Piston rod movement

S_b Piston rod movement at nominal force

S_a Piston rod movement at force reversal (lost motion)

c) Schematic force — Displacement graph

Figure I.3-1 — Typical characteristics of shocks arrestors

Annex J
(normative)

Type testing of support components

J.1 When supports are tested by loading, the supports shall be such as to guarantee at least the same factor of safety as by calculation. The component to be tested shall be installed in a manner as similar as possible to the intended operational situation. Testing shall be carried out at room temperature, but to compensate for operation at elevated temperature, the load derived under test conditions shall be multiplied by the reducing factor $R_{p0,2,t} / R_{p0,2,RT}$, where RT is the room temperature during testing, and t the design temperature of the support under test. For components designed to operate in the creep range, the reducing factor shall be S_2 /1,25 $R_{p0,2,RT}$.

For components that have different temperatures during operation (e.g. pipe clamps), it shall be ensured that the real stress situation correlates to the stress situation during testing.

J.2 Load test qualification of a support component shall cease to be valid if any aspect of the design is changed (e.g. material specification, welding procedure, method of construction).

J.3 Testing should be carried out on a minimum of 2 samples of each component size. If only a single sample is used, the qualified rated load shall be reduced by 10 % from that derived from the test. If any sample fails the test, the support shall not be qualified.

J.4 The testing conditions shall match the physical arrangements of the intended use as far as is possible, in particular the fitting of attachments and the direction of loading. Where loadings have different directions and intensities, the most conservative combination shall be used.

The allowable rated load shall be based on the minimum test load to cause ultimate failure (F_u), to yield (F_y), to buckle (F_B) or limit of stability (F_S), if appropriate, using the lowest value for the type determined from Table J.4-1.

NOTE Failure by instability may occur in different ways such as buckling, lateral buckling, lateral torsional buckling, plate/shell buckling in the elastic as well as the plastic range.

The allowable occasional load shall be the derived rated load multiplied by 1,2.

Table J.4-1 — Rated load derived by testing

Type of support component	Rated load derived from		
	load to ultimate failure	load to yield	limit of stability, load to buckle
rigid support hanger clamps hangers	$F_u/4{,}0$ or $F_u/(2{,}4\ K_1)$	$F_y/(1{,}6\ K_2)$	$F_S/2{,}5$
rigid struts snubbers sway brace compression loaded pipe supports	$F_u/4{,}0$ or $F_u/(2{,}4\ K_1)$	$F_y/(1{,}6\ K_2)$	$F_B/2{,}5$
where $K_1 = \dfrac{R_{m,\text{tension test}}}{R_{m,\text{material specification}}}$		$K_2 = \dfrac{R_{p0,2,\text{tension test}}}{R_{p0,2,\text{material specification}}}$	

Annex K
(informative)

Attachment of supports to structures

K.1 Attachment of supports to concrete structures

Various methods are available for fixing supports to concrete structures. In general, cast in fixings are preferred, and for this reason, early assessment of support locations and loads is desirable. The following are examples of acceptable methods:

— Embedment plates:

Embedment plates are metallic (welded) assemblies intended to be encased in concrete during civil construction. They are generally composed of a steel surface plate and connecting devices (normally 4) which are anchored in the concrete.

Embedment plates are the preferred method of attachment when loads and locations can be defined at an early stage in the project.

When embedment plates are subject to high shear loads, they should be provided with stops made from flat bars or sections.

The supplying and fixing of embedment plates is generally the responsibility of the civil contractor.

— Anchor rods:

Anchor rods consist of metallic inserts (straight rods or tie bars) which are positioned in holes left by the civil contractor or core-drilled, into which a plastic material is subsequently poured.

This method should be agreed with the civil contractor.

— Expansion bolts and dowel:

— Expansion bolts and dowel inserted into concrete are generally used for fixing supports whose location was unknown at the stage of the development of the civil design.

Expansion bolts and dowels should be intended in accordance with the manufacturer's specification

Where the type of insert cannot be subject to significant shear loads, pre-tensioning should be considered to ensure that the sliding force balances the shear force.

— Channels:

Channels are fixed onto the surface of the concrete by inserted dowels. Stayed channels are fixed to the shuttering before the concrete is poured. The supports are connected to the unstayed or stayed channels by specific bolts and nuts. The different channel types have bolts and nuts associated to them according to their size which are only suitable for the specific channel type.

K.2 Attachment to metallic structures

K.2.1 Standard bolts

The threads of bolts should not be subjected to shear loads, unless this is taken into account in the design.

Nuts should be provided with locking devices.

K.2.2 Friction grip bolts

Friction grip bolts should be tightened sufficiently to produce the clamping force between the bearing faces necessary to prevent slippage.

K.2.3 Welding

All welded attachments should be approved by the structural designer and the purchaser. No welding should be allowed across the tension flanges of structural steelwork without the approval of the structural designer.

Annex L
(informative)

Buckling of linear type supports

L.1 General

Support components subject to compressive loads should be designed to resist collapse by buckling. For components with boundary conditions that are not covered with the formula given in Annex, the use of EN 1993 is recommended.

L.2 Symbols

For the purposes of this Annex, the symbols given in Table L.2-1 should apply in addition to those given in Table 3.2-1.

Table L.2-1 — Additional symbols for the purposes of this Annex

Symbol	Description	Unit
A	cross sectional area	mm²
$C_{m,y}; C_{m,z}$	coefficients of bending moment	-
I	moment of inertia of the cross section along the buckling direction	mm^4
K	coefficient dependent on end conditions	-
L	length of the bar	mm
L_b	buckling length of the bar	mm
ξ	longitudinal axis of the bar	mm
ψ, ζ	strong/weak bending axis of the bar	mm
λ	slenderness of the bar in the buckling direction	-
λ_c	slenderness of the bar corresponding to the commencement of buckling	-
ρ	radius of gyration of the cross section	mm
σ_a	compressive stress	MPa (N/mm²)
$\sigma_{a,per}$	permissible compressive stress (see 13.3.6)	MPa (N/mm²)
$\sigma_{b,y}; \sigma_{b,z}$	bending stresses	MPa (N/mm²)
$\sigma_{b,per}$	permissible bending stress (see 13.3.6)	MPa (N/mm²)
$\sigma_{cr,y}; \sigma_{cr,z}$	buckling stresses	MPa (N/mm²)

L.3 Basic formulae

The radius of giration is calculated from:

$$\rho = \sqrt{\frac{I}{A}} \tag{L.3-1}$$

The buckling length of the bar is given by:

$$L_\mathrm{b} = KL \tag{L.3-2}$$

The slenderness of the bar in the buckling direction is given by:

$$\lambda = \frac{L_\mathrm{b}}{\rho} \tag{L.3-3}$$

The slenderness should not exceed 200.

L.4 Allowable compressive stress

NOTE In the following equations (L.4-1 to L.5-4) for ferritic (non-alloyed and low alloyed) steels R_{eHt} is used instead of $R_{p0,2t}$.

Elastic buckling occurs for slenderness greater than or equal to the value given by:

$$\lambda_\mathrm{c} = \left(\frac{2\pi^2 E}{R_\mathrm{p0,2}}\right)^{0,5} \tag{L.4-1}$$

Plastic or elastoplastic buckling occurs at slenderness less than this value.

The compressive stress should remain less than the allowable values given in (L.4-2) and (L.4-3) and in no case should it be greater than the value of the permissible stress f according to 13.3.6.2.

— For $\lambda \geq \lambda_\mathrm{c}$

$$\sigma_\mathrm{a,per} = \frac{12}{23}\left(\frac{\pi^2 E}{\lambda^2}\right) \tag{L.4.-2}$$

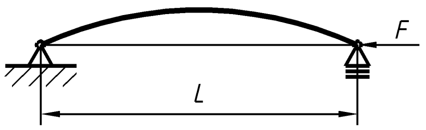
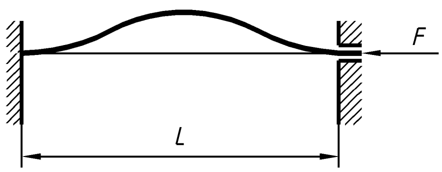
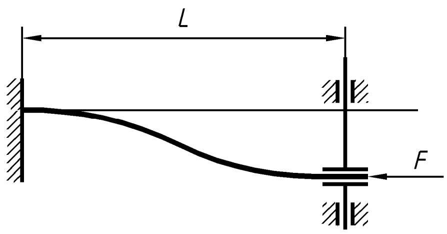
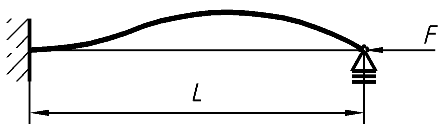
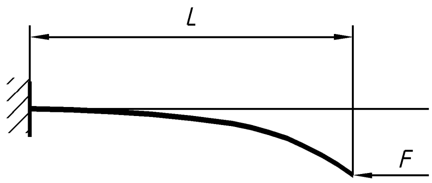
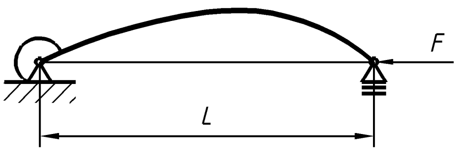
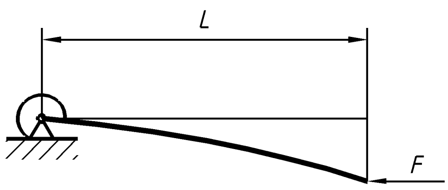
— For $\lambda < \lambda_\mathrm{c}$

$$\sigma_{a,per} = \min\left(f; \frac{R_{p0,2t}\left[1-\frac{1}{2}\left(\frac{\lambda}{\lambda_c}\right)^2\right]}{\left[\frac{5}{3}+\frac{3}{8}\left(\frac{\lambda}{\lambda_c}\right)-\frac{1}{8}\left(\frac{\lambda}{\lambda_c}\right)^3\right]}\right) \tag{L.4-3}$$

L.5 Buckling length

The buckling length takes into account the boundary conditions. Table L.5-1 gives the values of *K* for some cases.

Table L.5-1 — Values of *K*

Both ends supported/fixed	One end movable
 $K = 1$	
 $K = 0,5$	 $K = 1$
Both ends supported/fixed	One end movable
 $K = 0,7$	 $K = 2$
 $0,7 < K < 1$	 $K > 2$

For components subjected to both axial compression and bending moments, the stresses should be proportioned to satisfy:

$$\frac{\sigma_{\mathrm{a}}}{\sigma_{\mathrm{a,per}}} + \frac{C_{\mathrm{m,y}}\,\sigma_{\mathrm{b,y}}}{\left(1-\dfrac{\sigma_{\mathrm{a}}}{\sigma_{\mathrm{cr,y}}}\right)\sigma_{\mathrm{b,per}}} + \frac{C_{\mathrm{m,z}}\,\sigma_{\mathrm{b,z}}}{\left(1-\dfrac{\sigma_{\mathrm{a}}}{\sigma_{\mathrm{cr,z}}}\right)\sigma_{\mathrm{b,per}}} \le 1{,}0 \qquad \text{(L.5-1)}$$

and

$$\frac{\sigma_a}{f} + \frac{\sigma_{b,y} + \sigma_{b,z}}{\sigma_{b,per}} \le 1{,}0 \qquad \text{(L.5-2)}$$

When $\sigma_a / \sigma_{a,per} \le 0{,}15$ the following relation may be used:

$$\frac{\sigma_a}{\sigma_{a,per}} + \frac{\sigma_{b,y} + \sigma_{b,z}}{\sigma_{b,per}} \le 1{,}0 \qquad \text{(L.5-3)}$$

with

$$\sigma_{cr,y} = \frac{12}{23}\left(\frac{\pi^2 E}{\lambda_y^2}\right); \ \sigma_{cr,z} = \frac{12}{23}\left(\frac{\pi^2 E}{\lambda_z^2}\right) \qquad \text{(L.5-4)}$$

The values of the coefficients $C_{m,y}$ and $C_{m,z}$ are determined, for each direction, in accordance with the provisions of a) to c) below:

a) For compression members in frames subjected to joint translation (sideway): C_m = 0,85

b) For rotationally restrained compression members in frames braced against joint translation and not subject to transverse loading between their supports in the plane of bending:

— C_m = max (0,4; 0,6 – 0,4 (M1/M2)),

where M1/M2 is the ratio of the smaller to the larger bending moment (taken as an absolute value) at the ends of that portion of the member unbraced in the plan of bending under consideration. M1/M2 is positive when the two moments have different signs and negative otherwise.

c) For compression members in frames braced against joint translation in the plane of loading and subjected to transverse loading between their supports, the value of C_m may be determined by analysis, or failing this, by applicationof the following provisions:

— C_m = 0,85 for members whose ends are restrained,

— C_m = 1,00 for members whose ends are unrestrained.

Annex M
(informative)

Design guidance for structural components

M.1 Linear type components subjected to bending

M.1.1 General

The stability of a support can be checked by stability analysis of the compressed flange isolated from the beam and subject to uniform bending compressive stresses.

This verification is not required if the support is laterally supported, the length of each span being less than $\frac{200\,a}{\sqrt{R_{p0,2}}}$ (see Figure M.1.1-1)

With $R_{p0,2}$ in MPa (or N/mm^2) and a in mm.

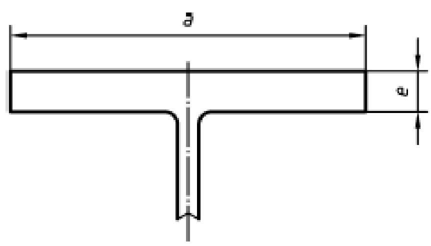


Figure M.1.1-1 — Flange dimensions

M.1.2 Supplementary verifications for linear type supports

M.1.2.1 General

These rules apply to the sections which are symmetrically arranged with regard to the axis of the greatest inertia and subjected to bending in this direction.

M.1.2.2 Welded assemblies

Welds should be continuous.

M.1.2.3 Free edges

The unstiffened part of the compressed flange should satisfy the following equation:

$$\frac{b}{e} \leq \frac{170}{\sqrt{R_{p0,2}}} \tag{M.1.2-1}$$

With $R_{p0,2}$ in MPa (or N/mm^2).

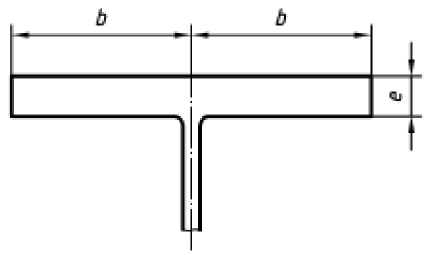


Figure M.1.2-1 – Stiffened flange dimensions

M.1.2.4 Stiffened edges

The stiffened part of the compressed flange should satisfy the following equation:

$$\frac{b}{e} \leq \frac{500}{\sqrt{R_{\mathrm{p0,2}}}} \tag{M.1.2-2}$$

With $R_{p0,2}$ in MPa (or N/mm^2).

Where *b* and *e* are given in Figure M.1.2-1.

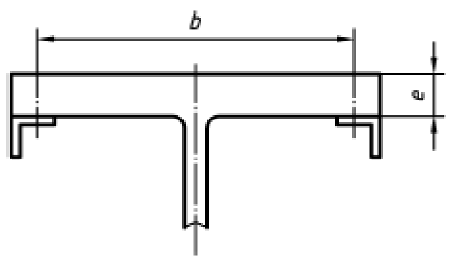


Figure M.1.2-2 — Stiffened flange dimensions

The components used as stiffeners should conform to the stability requirements in the direction perpendicular to the flanges where bending occurs. For this verification, these components should be isolated from the beam. For the calculation of slenderness, the inertia of the stiffener should be calculated taking into account the effective width of the flanges acting with it. This effective width should not exceed $\frac{b}{2}$ or $\frac{170\,e}{\sqrt{R_{\mathrm{p0,2}}}}$.

With $R_{p0,2}$ in MPa (or N/mm^2) and e in mm.

M.1.2.5 Shear load stability

Where shear loads occur, it is recommended that the panel web should be subdivided with a stiffener (see Figure M.1.2-2).

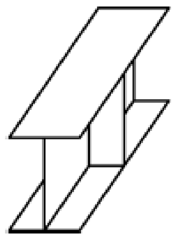


Figure M.1.2-3 — Panel web subdivision

M.2 Stability of plate type supports

Some plate type supports can be represented for the purpose of calculation as linear type supports assuming that the loading is fully supported by the stiffeners.

In this case, the stability of the stiffeners should only be verified in the perpendicular direction of the plate element, assuming for inertia calculations an effective width of the plate acting with the stiffeners. This effective width should not exceed 15 times the thickness.

M.3 Anchorage plates or equivalent anchorage components

M.3.1 General

Anchorage plates loaded by normal forces should be designed for at least 10 % of the normal loading at right angles to the normal plane.

M.3.2 Design of simple anchorage plates

When designing anchorage plates, allowance should be made for the strains occurring along lines tangential to the cross section of the beam which transfers the piping loads to the plate, see Figure M.3.2-1 a-a and b-b. The parts of the plate which lie outside these lines should be designed as cantilever beams and the cross section of the plate along these lines should withstand the moments of the reactions.

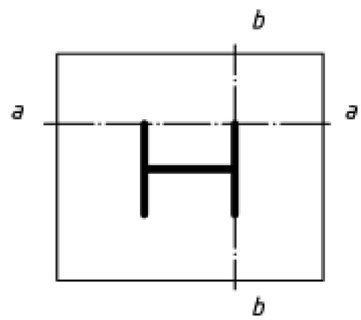


Figure M.3.2-1 — Simple anchorage plates

M.3.3 Fixing plates with stiffening gussets

Where it is necessary, due to the forces involved, stiffeners should be added. The calculations done in M.3.2 should be carried out along the lines a-a, b-b and c-c shown in Figure M.3.3-1.

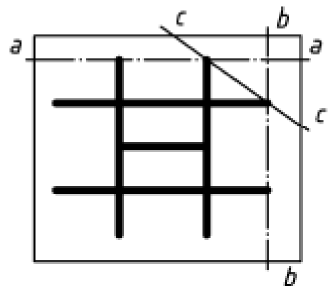


Figure M.3.3-1 — Fixing plates with stiffening gussets

M.3.4 Load calculations for anchorages fixed in concrete

Where anchorages are fixed in concrete, the support designer should provide the civil contractor with details of the loads and moments transferred to the structure.

Annex N
(normative)

Documentation of supports

The support manufacturer shall make available to the purchaser the documentation necessary to confirm that the supports conform to the requirements of clause 13. The extent of this documentation shall depend upon the class of the support as given in Table N.1 or as amended by agreement between the parties involved.

Table N.1 — Documentation of supports

	Documents	Support class		
		S1	**S2**	**S3**
Material	Springs – Inspection certificate 3.1 (EN 10204:2004)	-	Y	Y
	Springs – Test report 2.2 (EN 10204:2004)	Y	-	-
	Welding materials – Test report 2.2 (EN 10204:2004)	Y	Y	Y
	Flat products, long products, pipes, steel forgings			
	– Inspection certificate 3.1 (EN 10204:2004), if t > 300 °C	Y	Y	Y
	– Test report 2.2 (EN 10204:2004), if t ≤ 300 °C	-	Y	Y
	– Declaration of compliance with the order 2.1 (EN 10204:2004), if t ≤ 300 °C	Y	-	-
	Joining components (screws, nuts, studs, etc)			
	– Inspection certificate 3.1 (EN 10204:2004), if t > 300 °C	a	a	Y
	– Identification marking, if t ≤ 300 °C	a	a	a
	Small parts (washers, cotter pin, split pins, pins etc.) [b]	-	-	-
Design	Drawing, schedule or other means (e.g. catalogues) of providing the type, and identification of each support	X	X	X
	Drawing of each support [c]	-	X	X
	Proof of mechanical strength by analysis, testing or reference to a type-tested standard design	-	Y	Y

(*to be continued*)

Table N.1 (*concluded*)

	Documents	Support class		
		S1	S2	S3
Fabrication	Welder approval records in accordance with EN 287-1	-	Y	Y
	Welding procedure approval reports in accordance with EN ISO 15614-1	-	Y	Y
Inspection	Report on production testing in accordance with EN 13480-3, Annex I	-	X	X
Certification	Manufacturers certificate of compliance with EN 13480-3, Clause 13.	X	X	X

[a] Identification marking on the joining components.

[b] Documentation not necessary.

[c] For type-tested standard design information from catalogues or manufacturers standards are sufficient.

X = Documents to be supplied.

Y = Documents to be available for review.

Annex O
(normative)

Alternative method for checking branch connections

O.1 Scope

This Annex specifies a method for checking branch connections subjected to internal pressure and to moments (Figure O.1). Where external loads cannot be neglected, this method may be used in place of the method of EN 13480-3, 8.1. The rules of this Annex shall apply for temperatures below the creep range and for the following branch connections:

— connection of cylinders with intersecting axes;

— ratio of branch pipe to run pipe diameter within the range 0,1 to 1, 0,1 and 1 included;

— ratio of branch pipe to run pipe thickness within the range 0,2 to 1.5, 0,2 and 1,5 included;

— ratio of run pipe mean diameter to run pipe thickness within the range 10 to 125, 10 and 125 included;

— branch pipe self-reinforced or with complete encirclement pad (width = d_m / 2);

— angle φ_b between branch pipe and run pipe axes within the range 45° to 90°, 45° and 90° included;

— maximum thickness of reinforcing saddle = 1,5 times nominal thickness.

NOTE 1 The current developments included in this Annex do not deal with forged tees, considering the eventual reduction of thickness that could occur at the branch location (e.g. hot drawn tees).

NOTE 2 For the austenitic stainless steel with A ≥ 30%, $R_{p0,2t}$ in equations (O.3.1-1),(O.3.1-2), (O.3.3-1) to (O.3.3-4) may be replaced by $R_{p1,0t}$..

O.2 Symbols

For the purposes of Annex O, the symbols given below shall apply in addition to those given in Table 8.2-1 and in Table 3.2-1.

D_m	Mean diameter of the run pipe
d_m	Mean diameter of the branch pipe
e_s	Analysis thickness of the run pipe
e_b	Analysis thickness of the branch pipe
φ_b	Angle between the branch pipe axis and the run pipe axis (φ_b = 90° – φ)
p_c	Internal pressure

pln_s Limit pressure for the run pipe, in the absence of branch pipe

pln_b Limit pressure for the branch pipe considered separately

p_{max} Maximum permitted internal pressure when applied alone

Mfp_s Total bending moment acting on the run pipe and causing a rotation in the plane containing the run pipe and the branch pipe

Mfp_b Total bending moment acting on the branch pipe and causing a rotation in the plane containing the run pipe and the branch pipe

Mfh_s total bending moment acting on the run pipe and causing a rotation out of the plane containing the run pipe and the branch pipe

Mfh_b total bending moment acting on the branch pipe and causing a rotation out of the plane containing the run pipe and the branch pipe

Mt_s Torsional moment acting on the run pipe

Mt_b Torsional moment acting on the branch pipe

$Mfln_s$ Limit bending moment for the run pipe in the absence of branch pipe. This load is the nominal limit bending load corresponding to Mfp_s and Mfh_s

$Mtln_s$ Limit torsion moment for the run pipe in the absence of branch pipe

$Mfln_b$ Limit bending moment for the branch pipe considered separately. This load is the limit nominal bending load corresponding to Mfp_s and Mfh_s

$Mtln_b$ Limit torsional moment for the branch pipe considered separately

$Mflp_s$ Limit moment for the run pipe fitted with a branch pipe, corresponding to the loading Mfp_s

$Mflh_s$ Limit moment for the run pipe fitted with a branch pipe, corresponding to the loading Mfh_s

$Mflp_b$ Limit moment for the branch line in the branch connection, corresponding to the loading Mfp_b

$Mflh_b$ Limit moment for the branch line in the branch connection, corresponding to the loading Mfh_b

Mtl_s Limit moment for the run pipe fitted with a branch pipe, corresponding to the loading Mt_s

Mtl_b Limit moment for the branch line in the branch connection, corresponding to the loading Mt_b

$$\left.\begin{array}{l} Mfh_{\mathrm{b,max}} \\ Mfp_{\mathrm{b,max}} \\ Mt_{\mathrm{b,max}} \\ Mfh_{\mathrm{s,max}} \\ Mfp_{\mathrm{s,max}} \\ Mt_{\mathrm{s,max}} \end{array}\right\} \text{Maximum allowable value for each of the external loads when each load is applied alone}$$

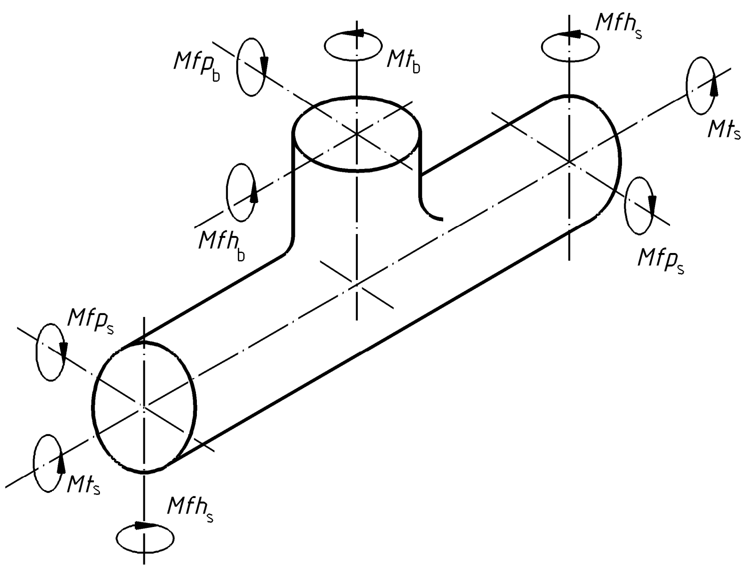


Figure O.1-1 – Location of moments

O.3 Design and checking of the branch connection

O.3.1 Limit value for the load due to pressure only for straight pipes without opening

$$pln_{\mathrm{s}} = \frac{2}{\sqrt{3}} R_{\mathrm{p\,0,2\,t}} \ln\left(\frac{D_m + e_{\mathrm{s}}}{D_m - e_{\mathrm{s}}}\right) \tag{O.3.1-1}$$

$$pln_{\mathrm{b}} = \frac{2}{\sqrt{3}} R_{\mathrm{p\,0,2\,t}} \ln\left(\frac{d_{\mathrm{m}} + e_{\mathrm{b}}}{d_{\mathrm{m}} - e_{\mathrm{b}}}\right) \tag{O.3.1-2}$$

O.3.2 Determination of the minimum thicknesses under loading due to pressure only

a) Weakening coefficient for the loading due to pressure only.

The Graphs O.3.2-1 to O.3.2-6 and Table O.3.2-1 make it possible to determine the weakening coefficient c as a function of e_b / e_s, d_m / D_m and D_m / e_s.

b) Minimum thicknesses of the run pipe and branch line.

The minimum thicknesses of the run pipe and of the branch line shall be determined from the following equations:

$$e_s = \frac{1}{c} \frac{p_c\ D_i}{2\ f\ z - p_c} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-1}$$

$$e_s = \frac{1}{c} \frac{p_c\ D_m}{2\ f\ z} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-2}$$

$$e_s = \frac{1}{c} \frac{p_c\ D_e}{2\ f\ z + p_c} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-3}$$

$$e_b = \frac{1}{c} \frac{p_c\ d_i}{2\ f\ z - p_c} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-4}$$

$$e_b = \frac{1}{c} \frac{p_c\ d_m}{2\ f\ z} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-5}$$

$$e_b = \frac{1}{c} \frac{p_c\ d_e}{2\ f\ z + p_c} (\sin \varphi_b)^{-\left(\frac{3}{2}\right)} \tag{O.3.2-6}$$

O.3.3 Checking of the thicknesses selected for the combination of pressure loading and loadings due to external loads

a) Limit values for the various external loadings applied separately.

For the various external loads applied separately, the limit values are given by the following formulae:

$$Mfln_s = R_{p\,0{,}2\,t} \frac{(D_m + e_s)^3}{6} \left(1 - \left(1 - \frac{2\ e_s}{D_m + e_s}\right)^3\right) \tag{O.3.3-1}$$

$$Mfln_b = R_{p\,0{,}2\,t} \frac{(d_m + e_b)^3}{6} \left(1 - \left(1 - \frac{2\ e_b}{d_m + e_b}\right)^3\right) \tag{O.3.3-2}$$

$$Mtln_{\mathrm{s}} = \frac{2}{\sqrt{3}} R_{\mathrm{p\,0,2\,t}} \left(\frac{\pi D_{\mathrm{m}}^{2}}{4} \right) e_{\mathrm{s}} \qquad \text{(O.3.3-3)}$$

$$Mtln_{\mathrm{b}} = \frac{2}{\sqrt{3}} R_{\mathrm{p\,0,2\,t}} \left(\frac{\pi d_{\mathrm{m}}^{2}}{4} \right) e_{\mathrm{b}} \qquad \text{(O.3.3-4)}$$

b) Weakening coefficients for the various external loads applied separately.

The Graphs O.3.2-7 to O.3.2-42 and Table O.3.2-2 make it possible to determine the weakening coefficients as a function of $e_{\mathrm{b}} / e_{\mathrm{s}}$, $d_{\mathrm{m}} / D_{\mathrm{m}}$ and $D_{\mathrm{m}} / e_{\mathrm{s}}$.

$$cfh_{\mathrm{b}} = \frac{Mflh_{\mathrm{b}}}{Mfln_{\mathrm{b}}} \qquad \text{(O.3.3-5)}$$

$$cfp_{\mathrm{b}} = \frac{Mflp_{\mathrm{b}}}{Mfln_{\mathrm{b}}} \qquad \text{(O.3.3-6)}$$

$$ct_{\mathrm{b}} = \frac{Mtl_{\mathrm{b}}}{Mtln_{\mathrm{b}}} \qquad \text{(O.3.3-7)}$$

$$cfh_{\mathrm{s}} = \frac{Mflh_{\mathrm{s}}}{Mfln_{\mathrm{s}}} \qquad \text{(O.3.3-8)}$$

$$cfp_{\mathrm{s}} = \frac{Mflp_{\mathrm{s}}}{Mfln_{\mathrm{s}}} \qquad \text{(O.3.3-9)}$$

$$ct_{\mathrm{s}} = \frac{Mtl_{\mathrm{s}}}{Mtln_{\mathrm{s}}} \qquad \text{(O.3.3-10)}$$

c) Maximum allowable loads if they are applied separately.

$$Mfh_{\mathrm{b,max}} = 0{,}5\ Mflh_{\mathrm{b}} \qquad \text{(O.3.3-11)}$$

$$Mfp_{\mathrm{b,max}} = 0{,}5\ Mflp_{\mathrm{b}} \qquad \text{(O.3.3-12)}$$

$$Mt_{\mathrm{b,max}} = 0{,}5\ Mtl_{\mathrm{b}} \qquad \text{(O.3.3-13)}$$

$$Mfh_{\mathrm{s,max}} = 0{,}5\ Mflh_{\mathrm{s}} \qquad \text{(O.3.3-14)}$$

$$Mfp_{\mathrm{s,max}} = 0{,}5\ Mflp_{\mathrm{s}} \qquad \text{(O.3.3-15)}$$

$$Mt_{\mathrm{s,max}} = 0{,}5\ Mtl_{\mathrm{s}} \qquad \text{(O.3.3-16)}$$

$$p_{\max} = \frac{\sqrt{3}}{3} MIN[z\ MIN\left(pln_{\mathrm{s}}\ ;\ pln_{\mathrm{b}}\right);\ c\ \mathrm{MIN}\left(pln_{\mathrm{s}}\ ;\ pln_{\mathrm{b}}\right)(\sin \varphi_{\mathrm{b}})^{\frac{3}{2}}] \qquad \text{(O.3.3-17)}$$

d) Checking of the admissibility of the applied loads.

$$\frac{Mfh_{\mathrm{b}}}{Mfh_{\mathrm{b,max}}} \leq 1 \qquad \text{(O.3.3-18)}$$

$$\frac{Mfp_{\mathrm{b}}}{Mfp_{\mathrm{b,max}}} \leq 1 \qquad \text{(O.3.3-19)}$$

$$\frac{Mt_{\mathrm{b}}}{Mt_{\mathrm{b,max}}} \leq 1 \qquad \text{(O.3.3-20)}$$

$$\frac{Mfh_{\mathrm{s}}}{Mfh_{\mathrm{s,max}}} \leq 1 \qquad \text{(O.3.3-21)}$$

$$\frac{Mfp_{\mathrm{s}}}{Mfp_{\mathrm{s,max}}} \leq 1 \qquad \text{(O.3.3-22)}$$

$$\frac{Mt_{\mathrm{s}}}{Mt_{\mathrm{s,max}}} \leq 1 \qquad \text{(O.3.3-23)}$$

$$\frac{p_{c}}{p_{\mathrm{max}}} \leq 1 \qquad \text{(O.3.3-24)}$$

$$\sqrt{\left(\frac{Mfh_{\mathrm{b}}}{Mfh_{\mathrm{b,max}}}\right)^2 + \left(\frac{Mfp_{\mathrm{b}}}{Mfp_{\mathrm{b,max}}}\right)^2 + \left(\frac{Mt_{\mathrm{b}}}{Mt_{\mathrm{b,max}}}\right)^2 + \left(\frac{Mfh_{\mathrm{s}}}{Mfh_{\mathrm{s,max}}}\right)^2 + \left(\frac{Mfp_{\mathrm{s}}}{Mfp_{\mathrm{s,max}}}\right)^2 + \left(\frac{Mt_{\mathrm{s}}}{Mt_{\mathrm{s,max}}}\right)^2 + \left(\frac{p_{\mathrm{c}}}{p_{\mathrm{max}}}\right)^2} \leq 1 \qquad \text{(O.3.3-25)}$$

If those criteria are not met, dimensions shall be modified and calculations repeated.

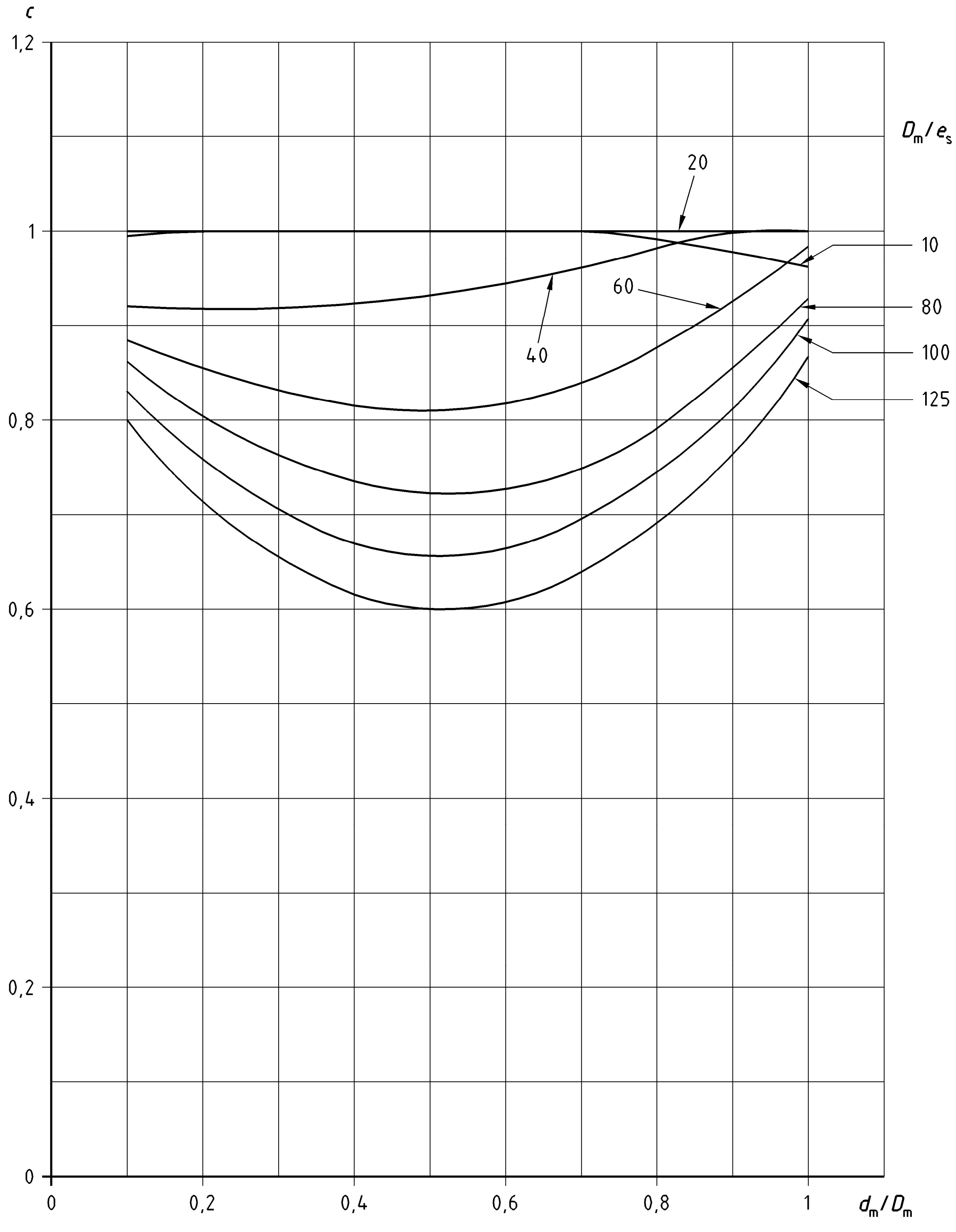


Figure O.3.2-1 — Coefficient c for e_b / e_s = 0,2

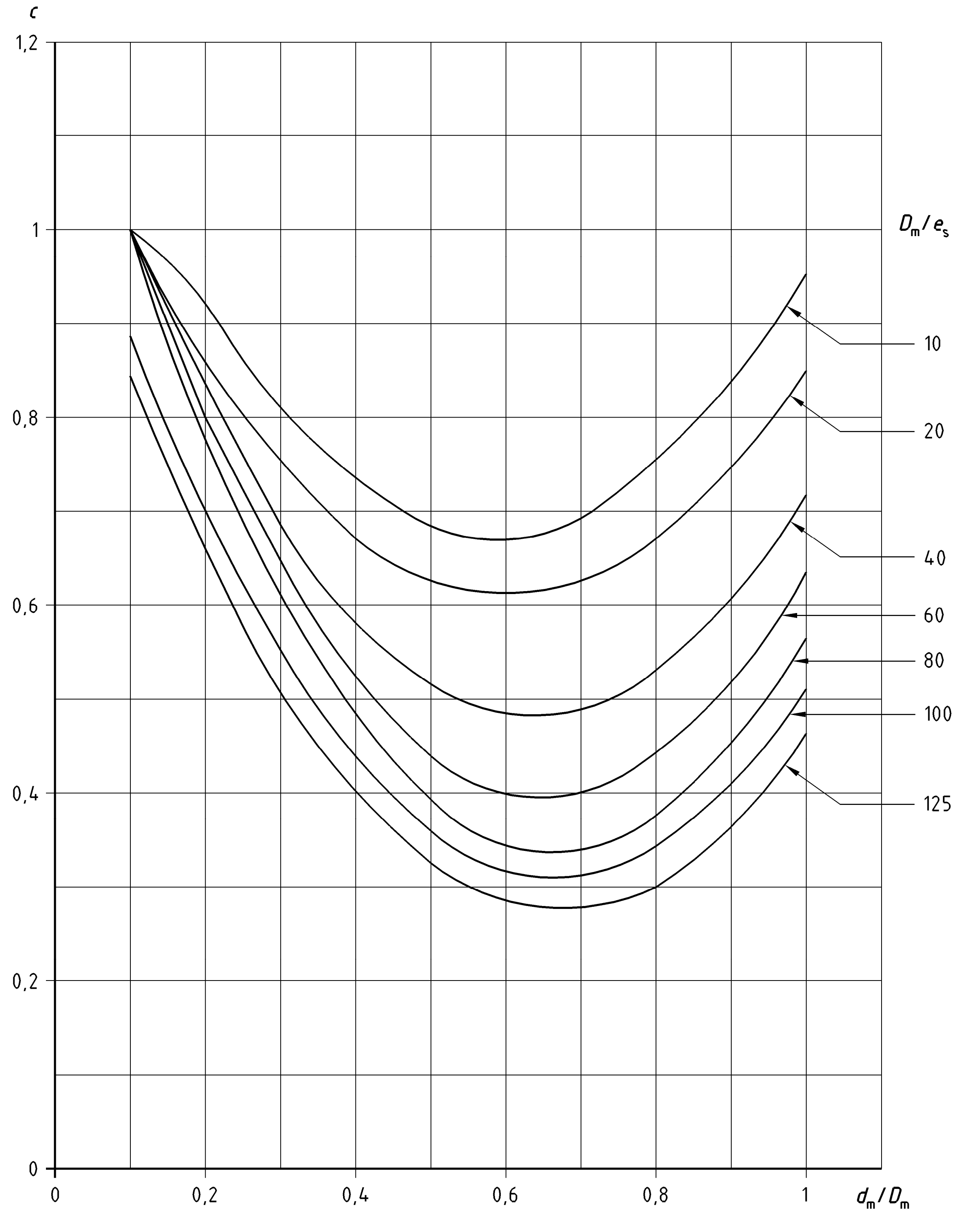


Figure O.3.2-2 — Coefficient c for e_b / e_s = 0,5

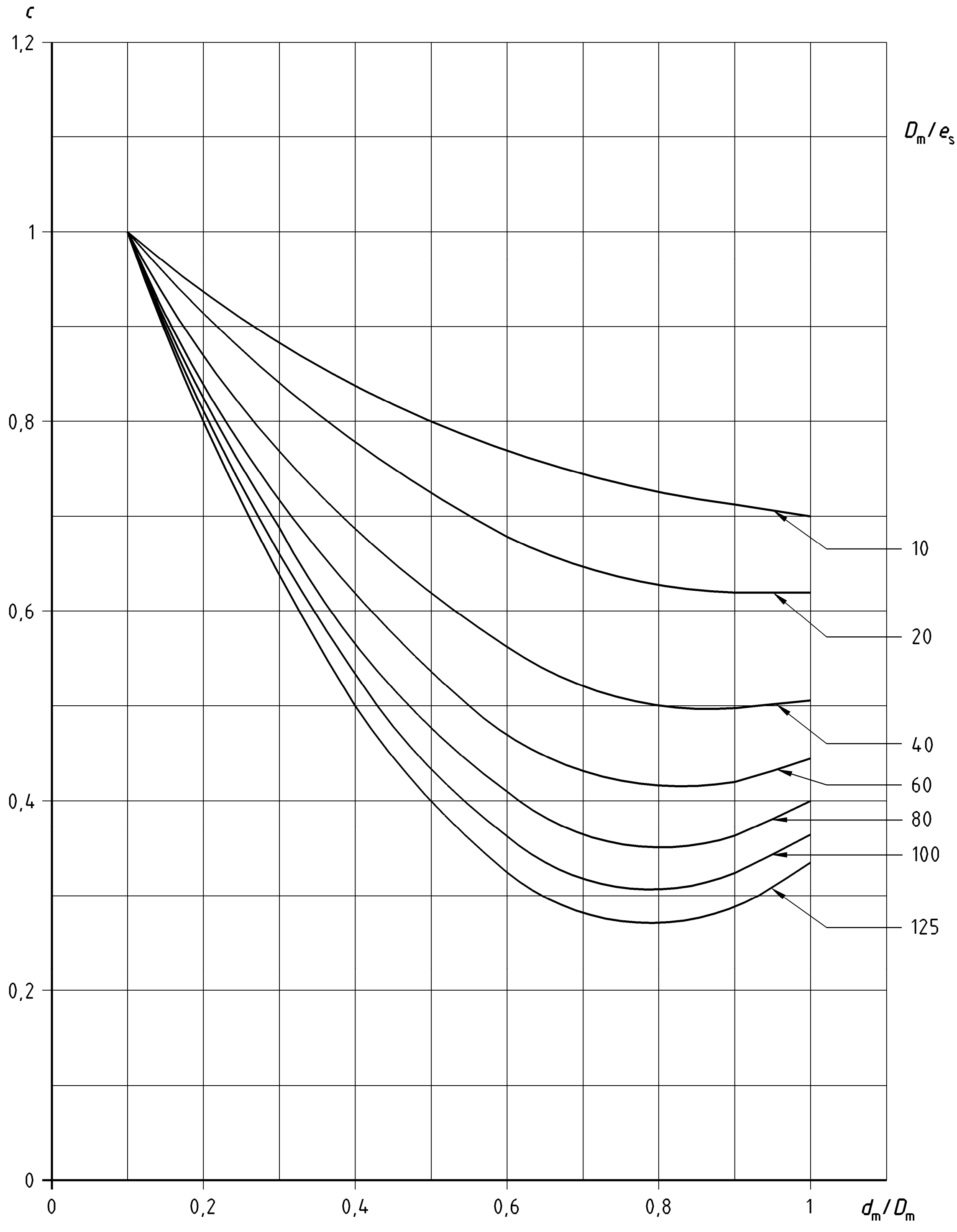


Figure O.3.2-3 — Coefficient c for e_b / e_s = 0,8

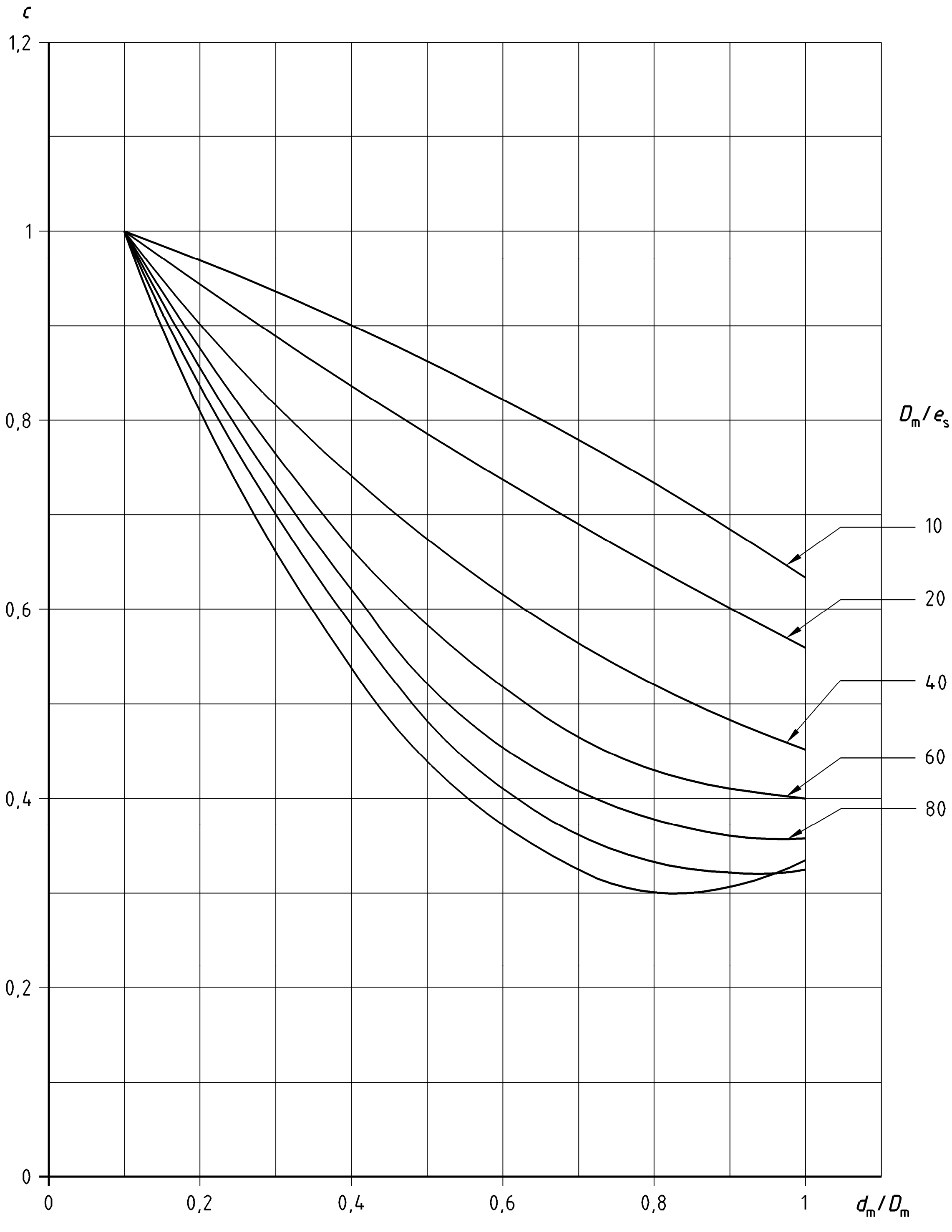


Figure O.3.2-4 — Coefficient c for e_b / e_s = 1,0

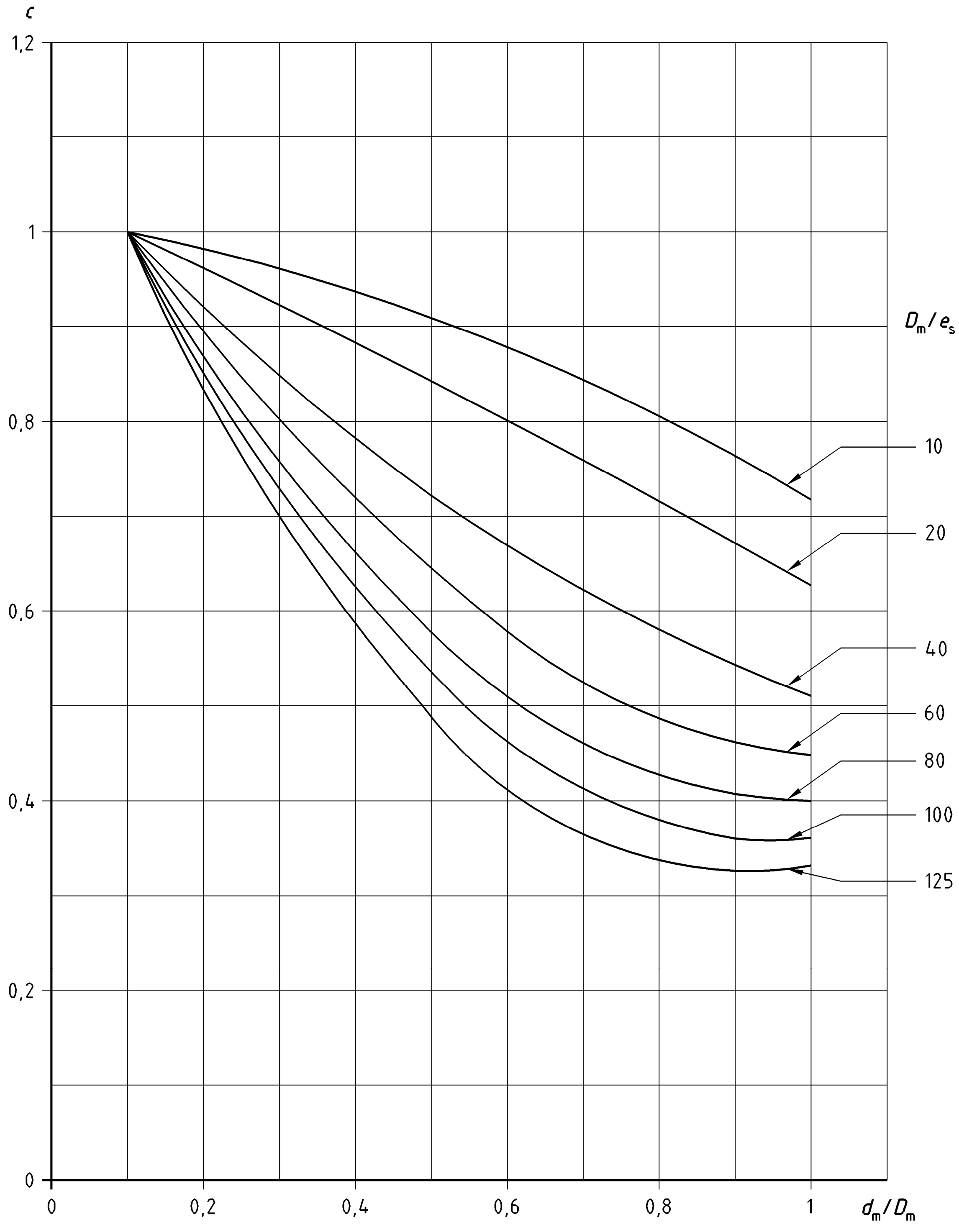


Figure O.3.2-5 — Coefficient c for e_b / e_s = 1,2

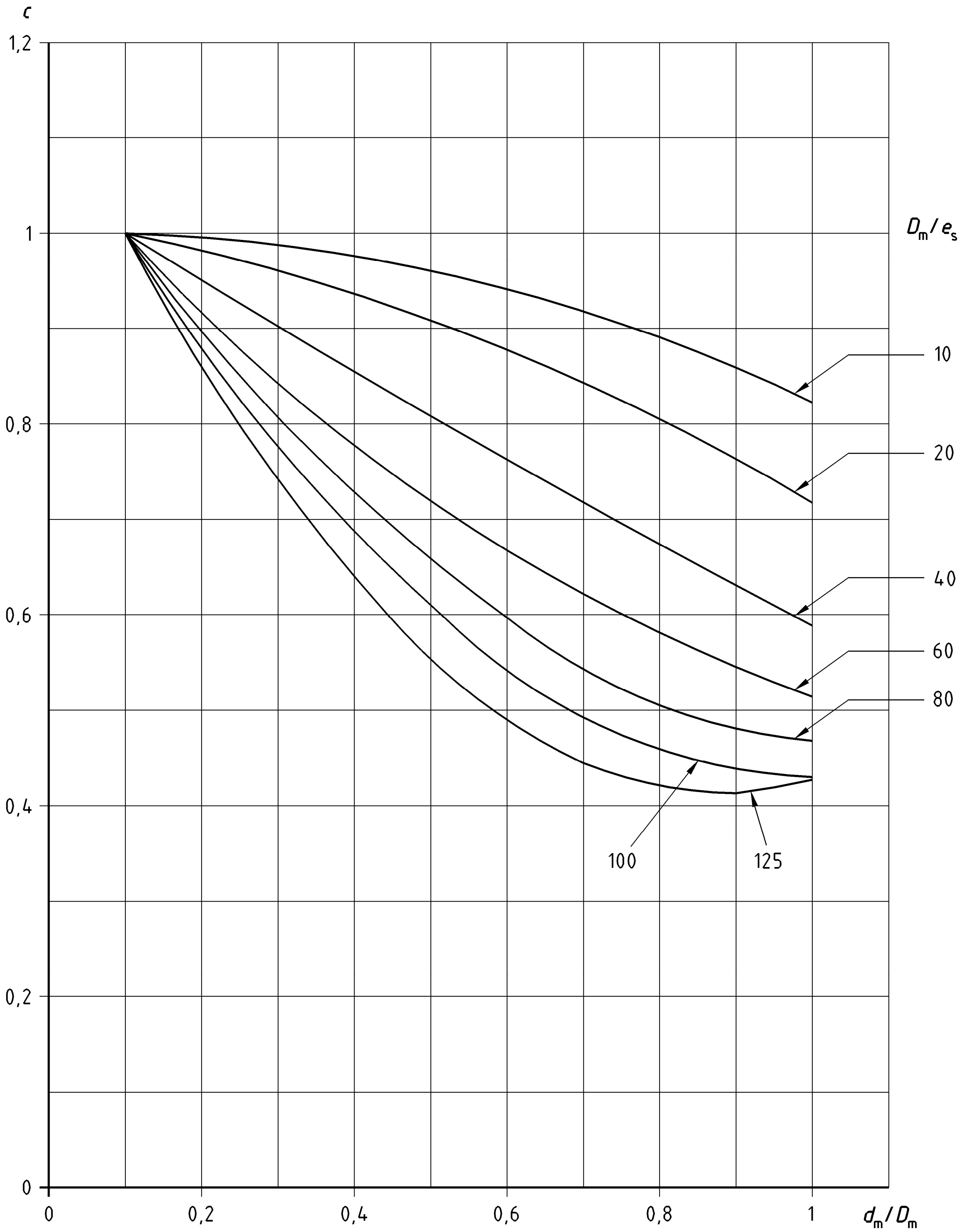


Figure O.3.2-6 – Coefficient c for e_b / e_s = 1,5

Table O.3.2-1 — Coefficient of the polynomial equations describing the curves of Figures O.3.2-1 to O.3.2-6

	D_m / e_s	*A*	*B*	*C*
Figure O.3.2-1 Coefficient *c* For e_b / e_s = 0,2	10	- 0,070 9	- 0,042 9	1,071 3
	20	- 0,100 5	0,133 8	0,984 5
	40	0,133 9	- 0,031 8	0,922 1
	60	0,596 7	- 0,546 7	0,937 1
	80	0,809 4	- 0,829 2	0,938 0
	100	1,021 4	- 1,044 1	0,928 7
	125	1,132 4	- 1,171 5	0,905 0
Figure O.3.2-2 Coefficient *c* For e_b / e_s = 0,5	10	1,666 4	- 1,954 9	1,247 3
	20	1,568 6	- 1,878 5	1,170 8
	40	1,780 2	- 2,268 4	1,206 7
	60	1,956 2	- 2,562 5	1,233 8
	80	2,056 5	- 2,741 3	1,249 5
	100	1,801 4	- 2,401 0	1,108 6
	125	1,769 4	- 2,383 9	1,075 9
Figure O.3.2-3 Coefficient *c* For e_b / e_s = 0,8	10	0,337 6	- 0,703 6	1,068 0
	20	0,509 6	- 0,983 8	1,093 1
	40	0,834 7	- 1,464 0	1,136 8
	60	1,110 2	- 1,838 0	1,170 9
	80	1,290 5	- 2,084 2	1,193 4
	100	1,418 2	- 2,259 9	1,209 3
	125	1,538 6	- 2,423 9	1,224 2
Figure O.3.2-4 Coefficient *c* For e_b / e_s = 1,0	10	- 0,106 0	- 0,285 0	1,030 5
	20	0,072 1	- 0,571 1	1,056 6
	40	0,430 3	- 1,075 2	1,102 6
	60	0,736 0	- 1,474 7	1,138 9
	80	0,944 3	- 1,746 1	1,163 6
	100	1,098 2	- 1,946 6	1,181 8
	125	1,309 9	- 2,173 8	1,202 1
Figure O.3.2-5 Coefficient *c* For e_b / e_s = 1,2	10	- 0,165 4	- 0,135 2	1,015 8
	20	- 0,039 1	- 0,375 5	1,038 4
	40	0,260 8	- 0,832 3	1,080 4
	60	0,553 2	- 1,226 2	1,116 4
	80	0,766 5	- 1,509 3	1,142 2
	100	0,925 8	- 1,719 8	1,161 4
	125	1,079 7	- 1,920 8	1,179 7
Figure O.3.2-6 Coefficient *c* For e_b / e_s = 1,5	10	- 0,242 4	0,063 9	0,995 9
	20	- 0,182 4	- 0,113 8	1,013 8
	40	0,045 2	- 0,508 7	1,050 7
	60	0,310 1	- 0,882 4	1,085 1
	80	0,516 5	- 1,166 8	1,111 1
	100	0,679 4	- 1,388 1	1,131 4
	125	0,913 6	- 1,647 9	1,154 8

NOTE The curves of Graphs O.3.2-1 to O.3.2-6 can be described by the following polynomial equation:

$$c = MIN[(Ax^2 + Bx + C) ; 1] \quad \text{(T.O.3.2-1.1)}$$

where

$$x = d_m / D_m$$

Coefficients A, B and C are given for each curve of each graph in function of the ratio D_m / e_s

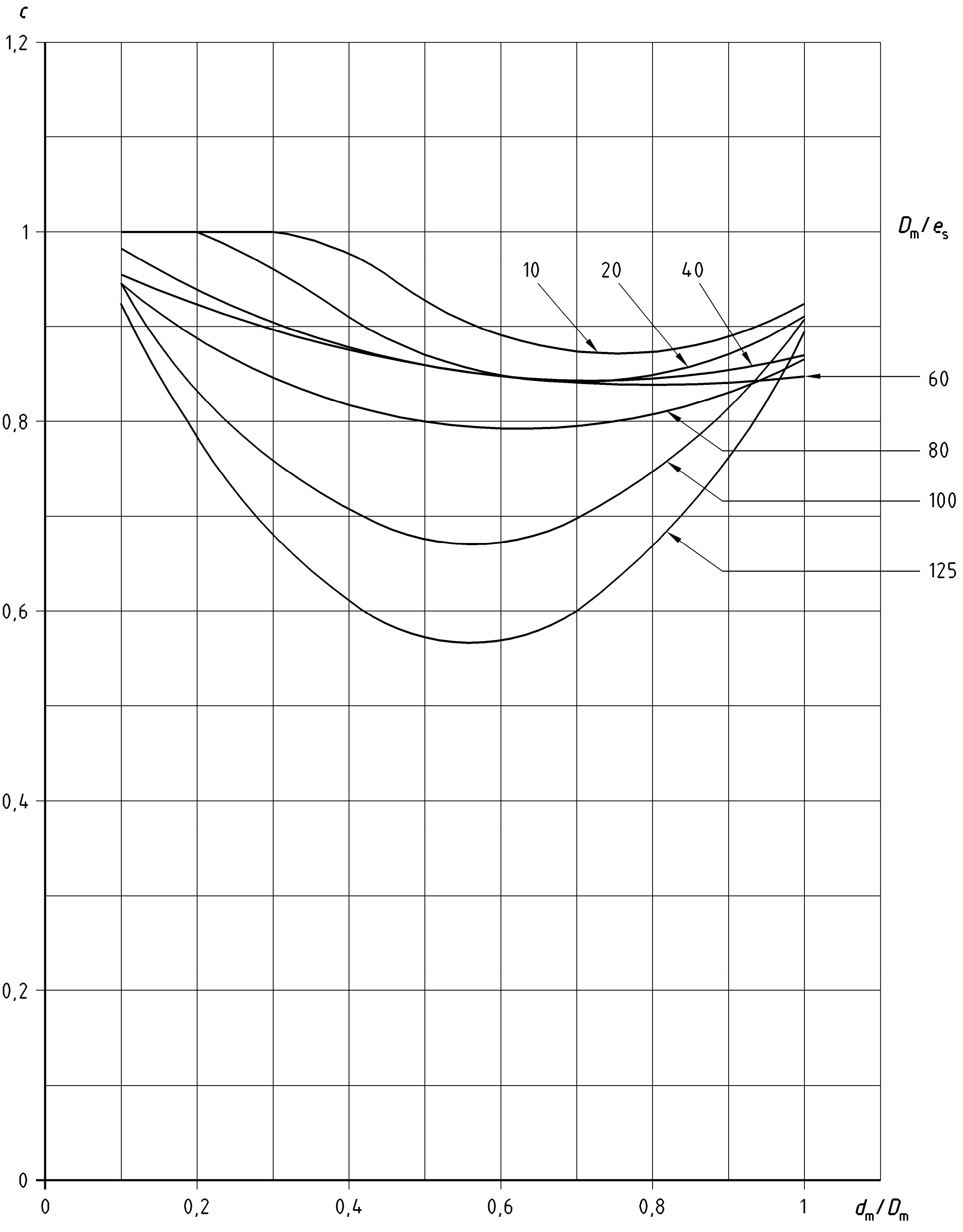


Figure O.3.2-7 — Coefficient cfh_b for e_b / e_s = 0,2

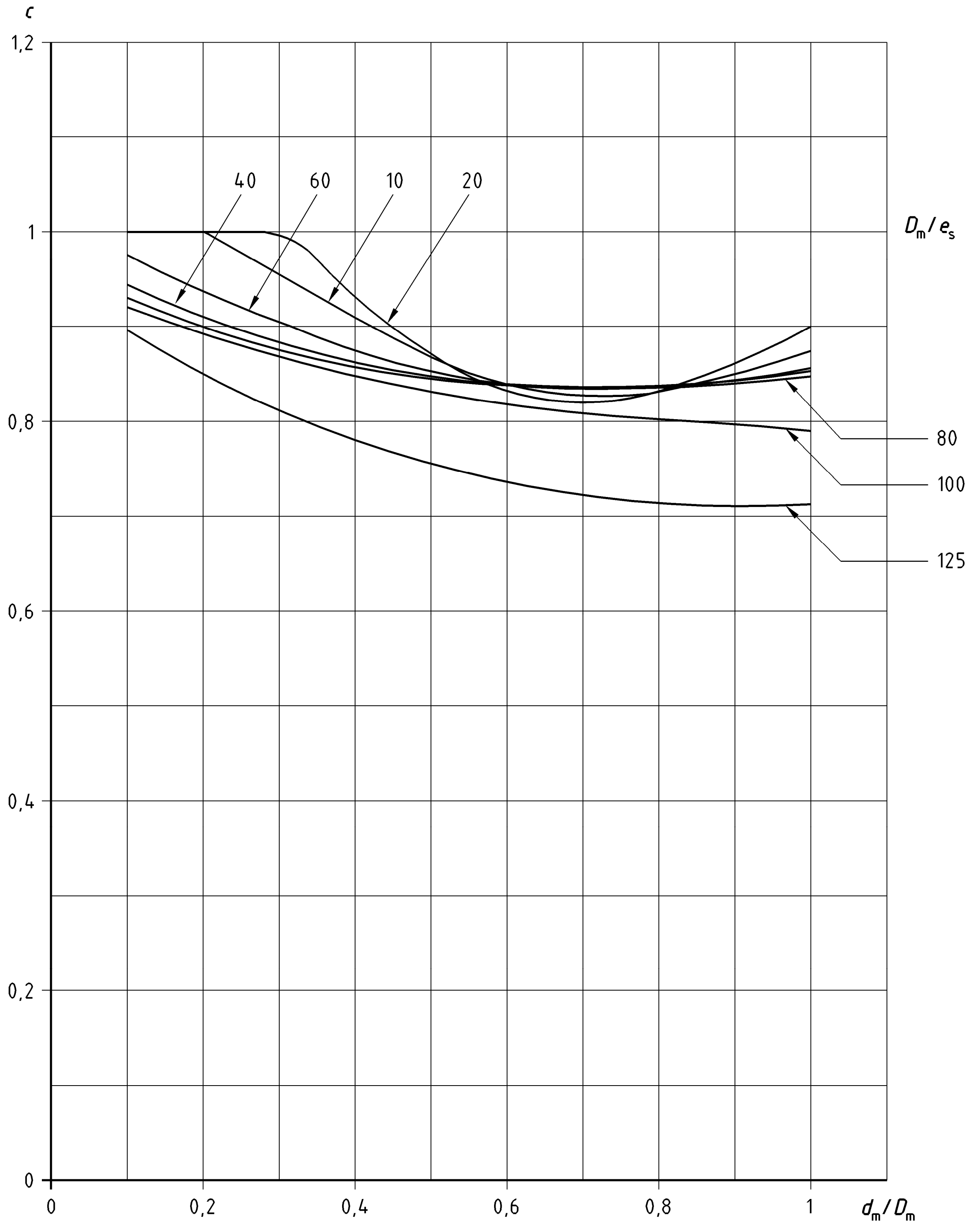


Figure O.3.2-8 — Coefficient cfp_b for e_b / e_s = 0,2

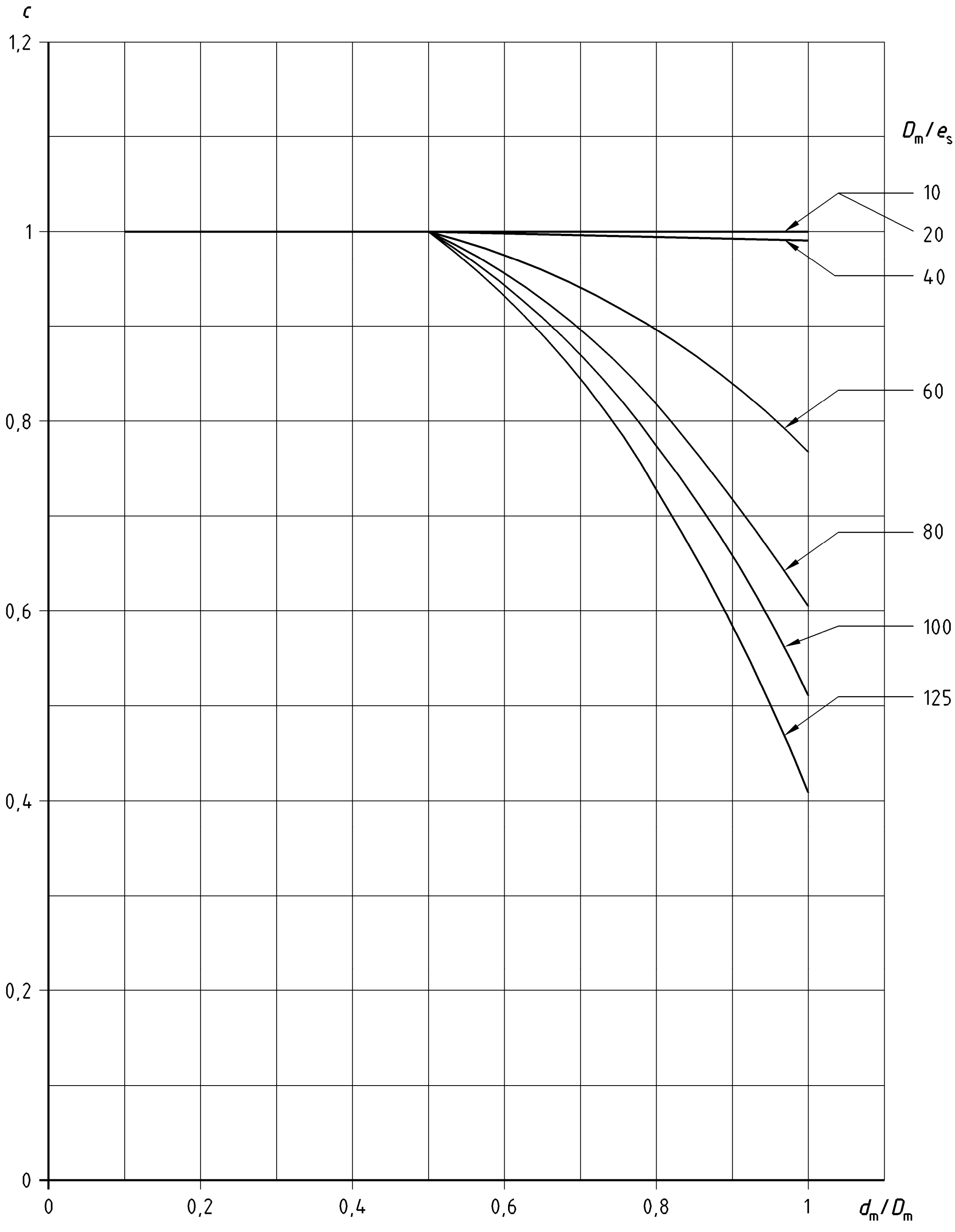


Figure O.3.2-9 — Coefficient ct_b for e_b / e_s = 0,2

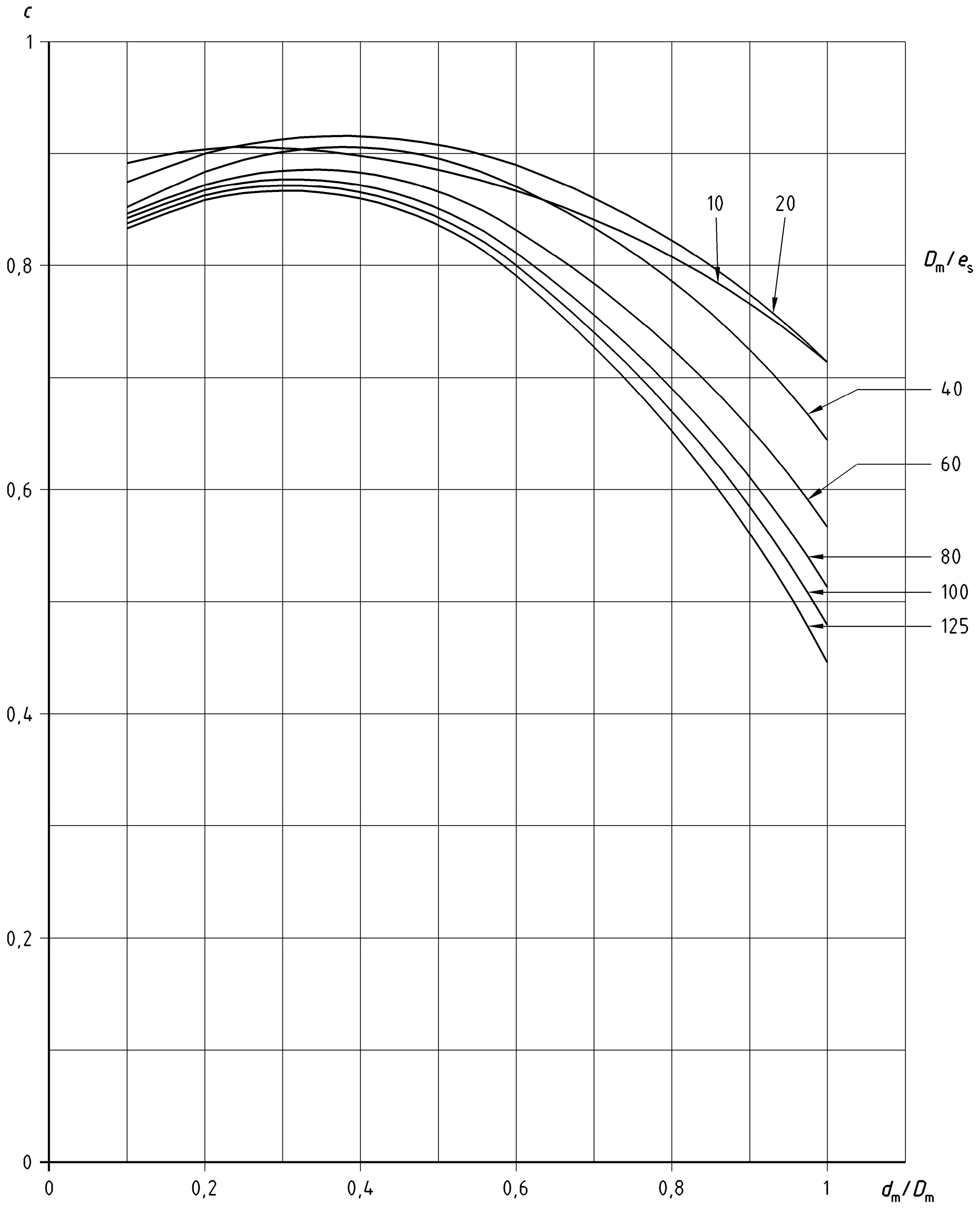


Figure O.3.2-10 — Coefficient cfh_s for e_b / e_s = 0,2

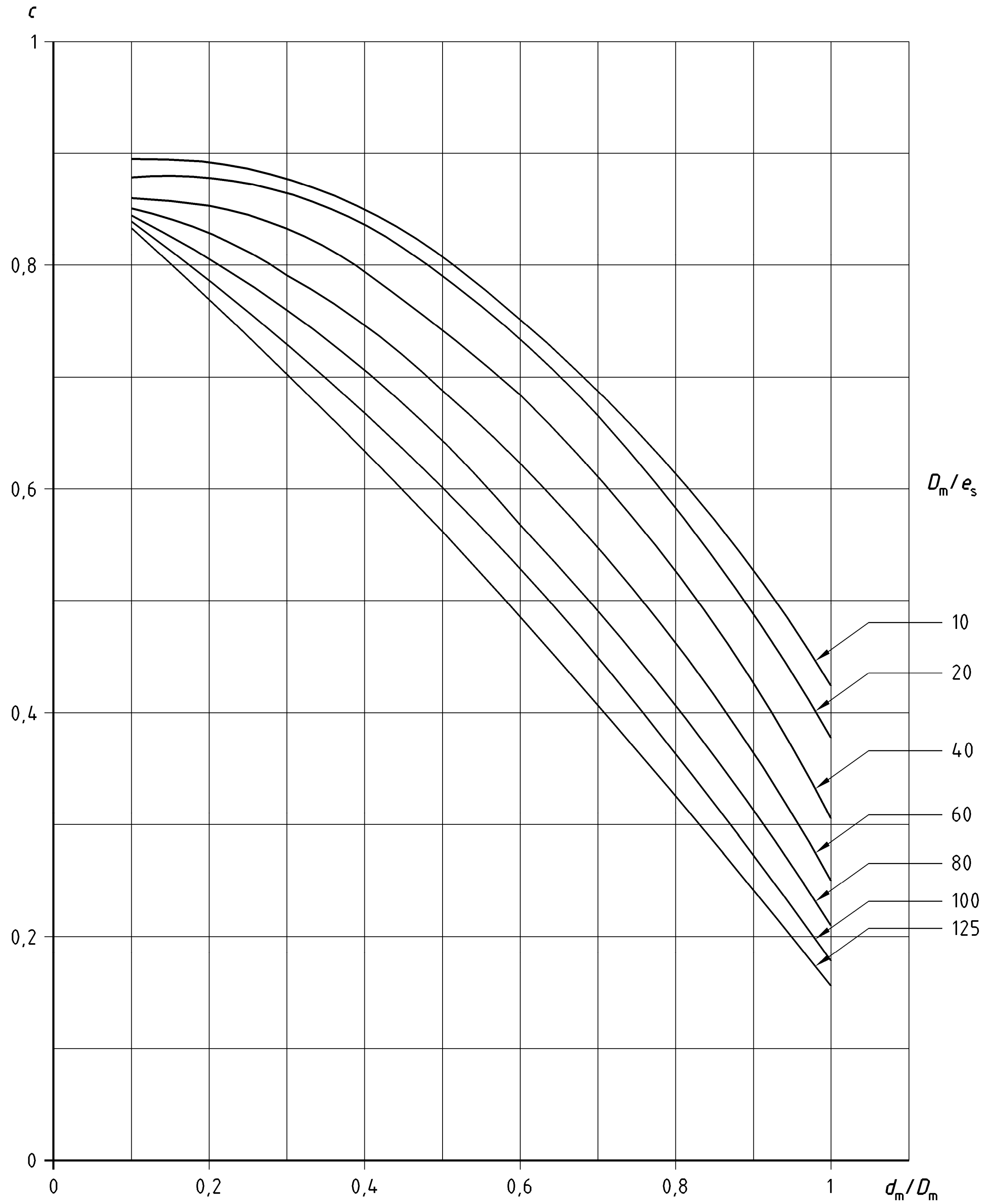


Figure O.3.2-11 — Coefficient cfp_s for e_b / e_s = 0,2

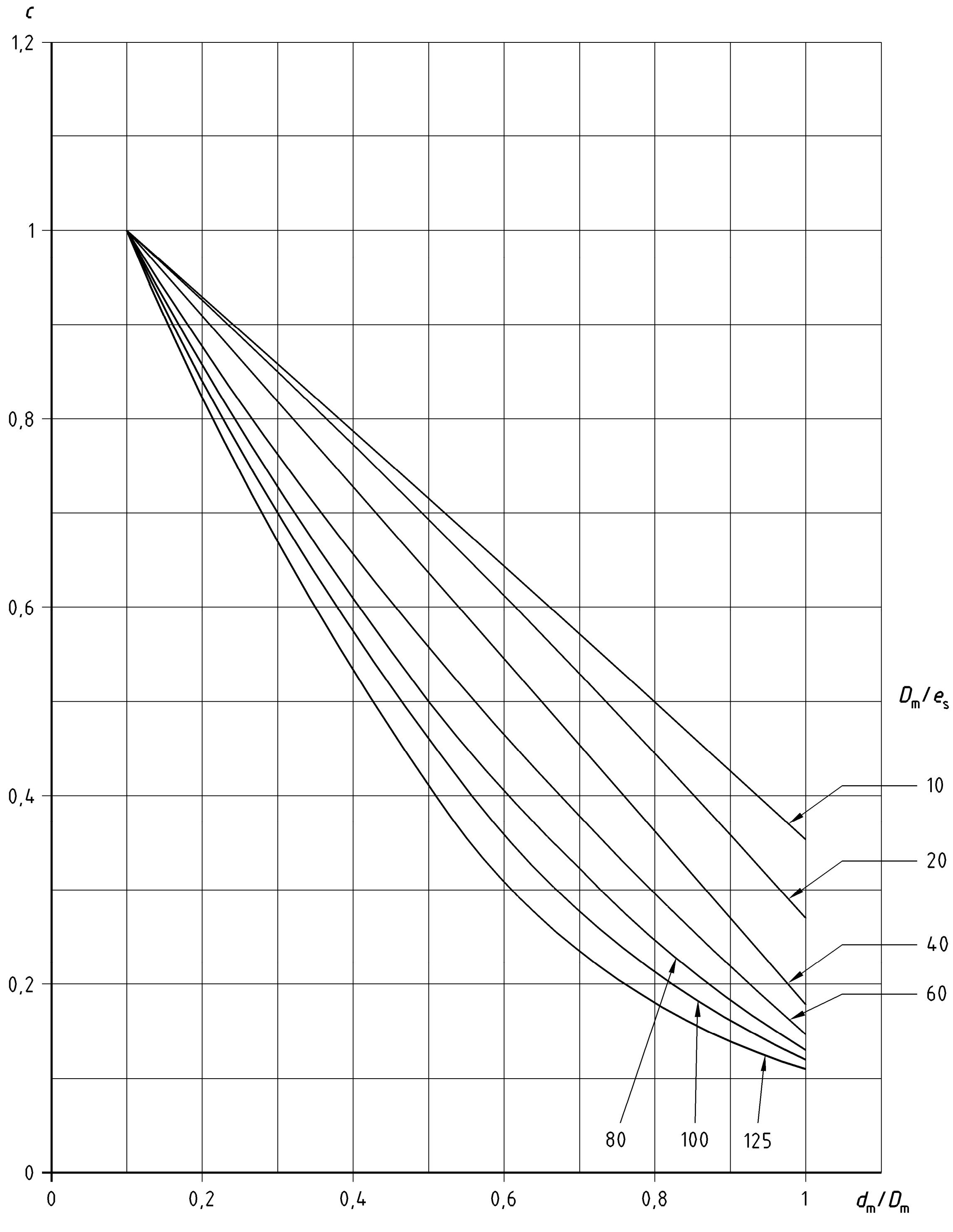


Figure O.3.2-12 — Coefficient ct_s for e_b / e_s = 0,2

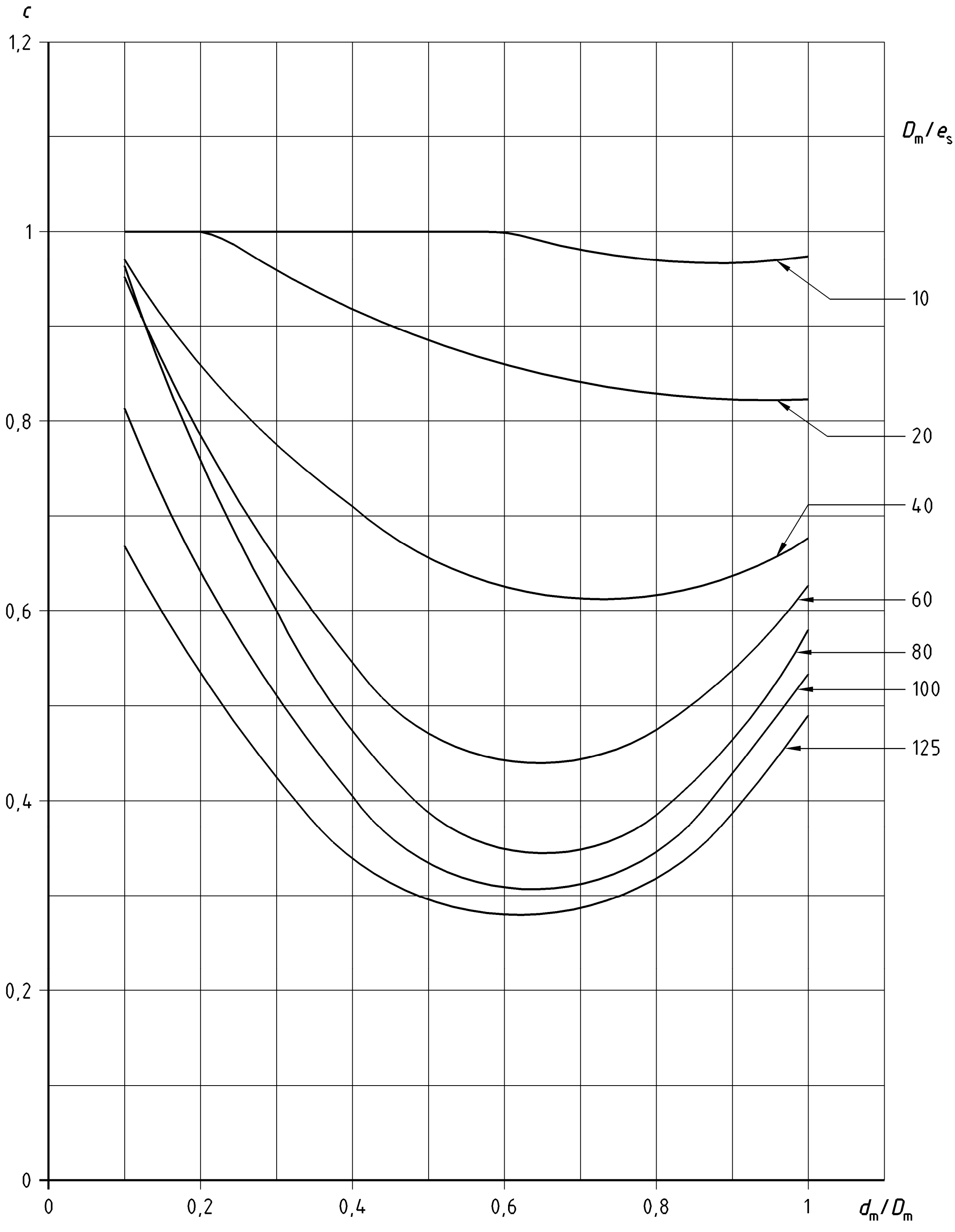


Figure O.3.2-13 — Coefficient cfh_b for e_b / e_s = 0,5

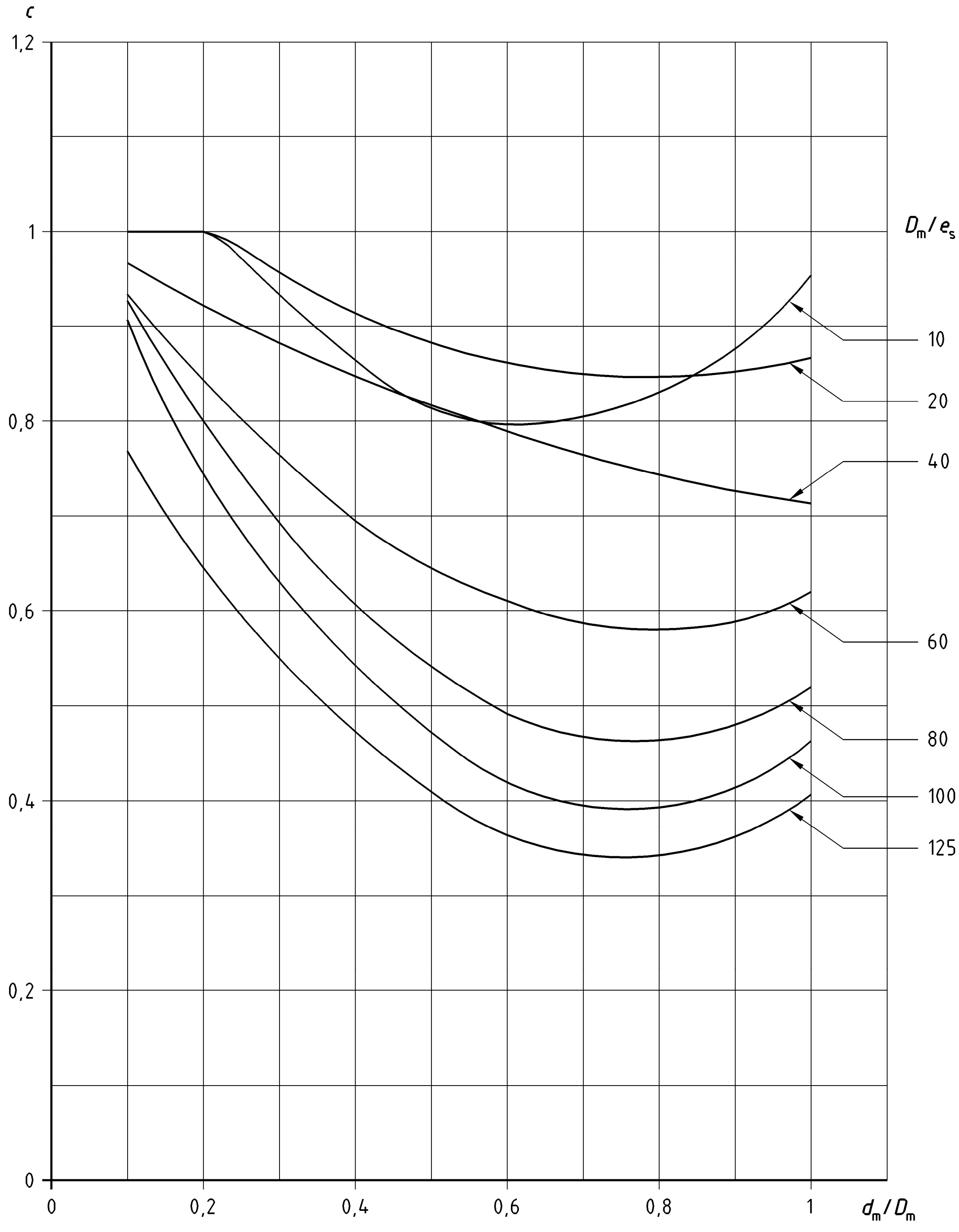


Figure O.3.2-14 — Coefficient cfp_b for e_b / e_s = 0,5

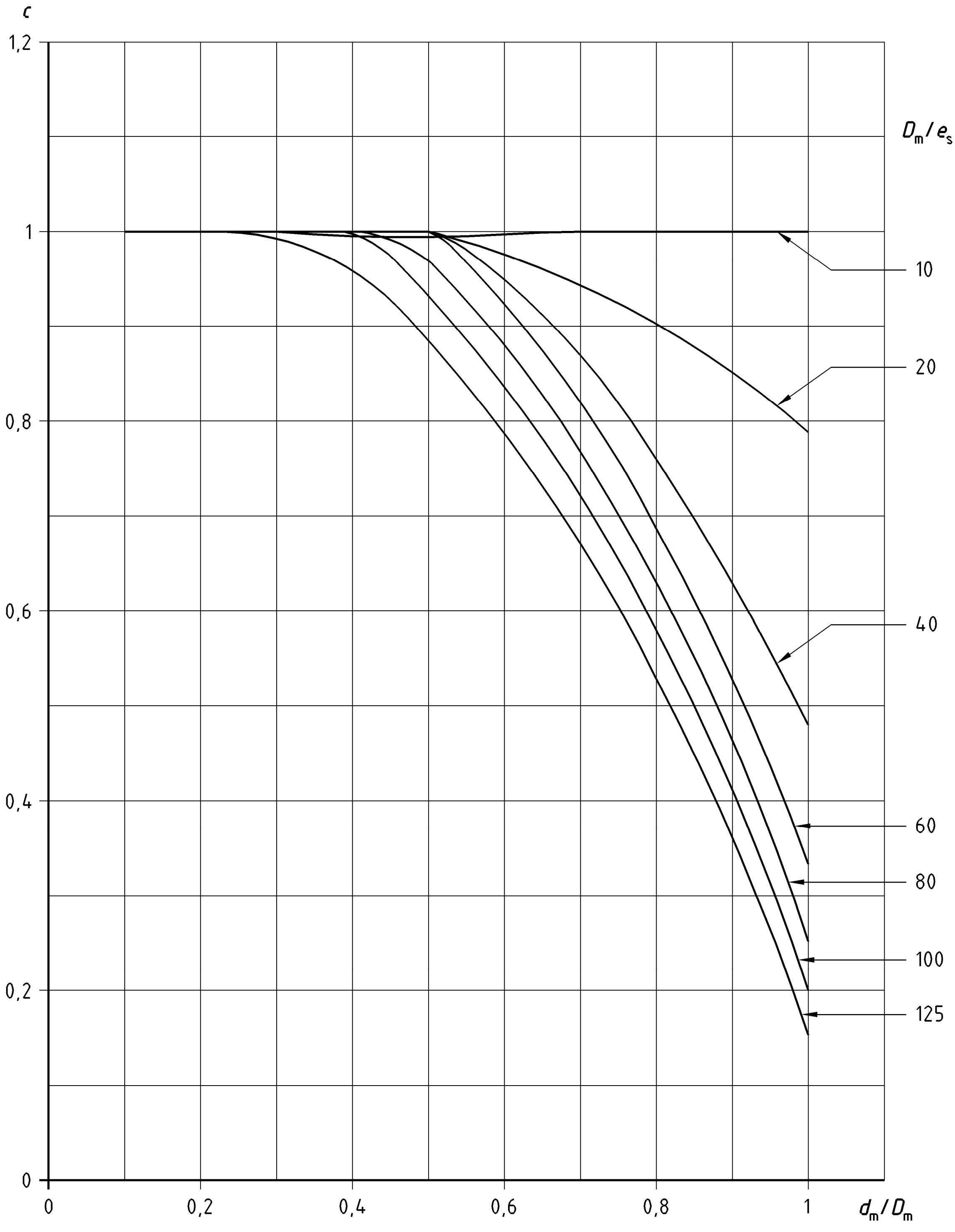


Figure O.3.2-15 — Coefficient ct_b for e_b / e_s = 0,5

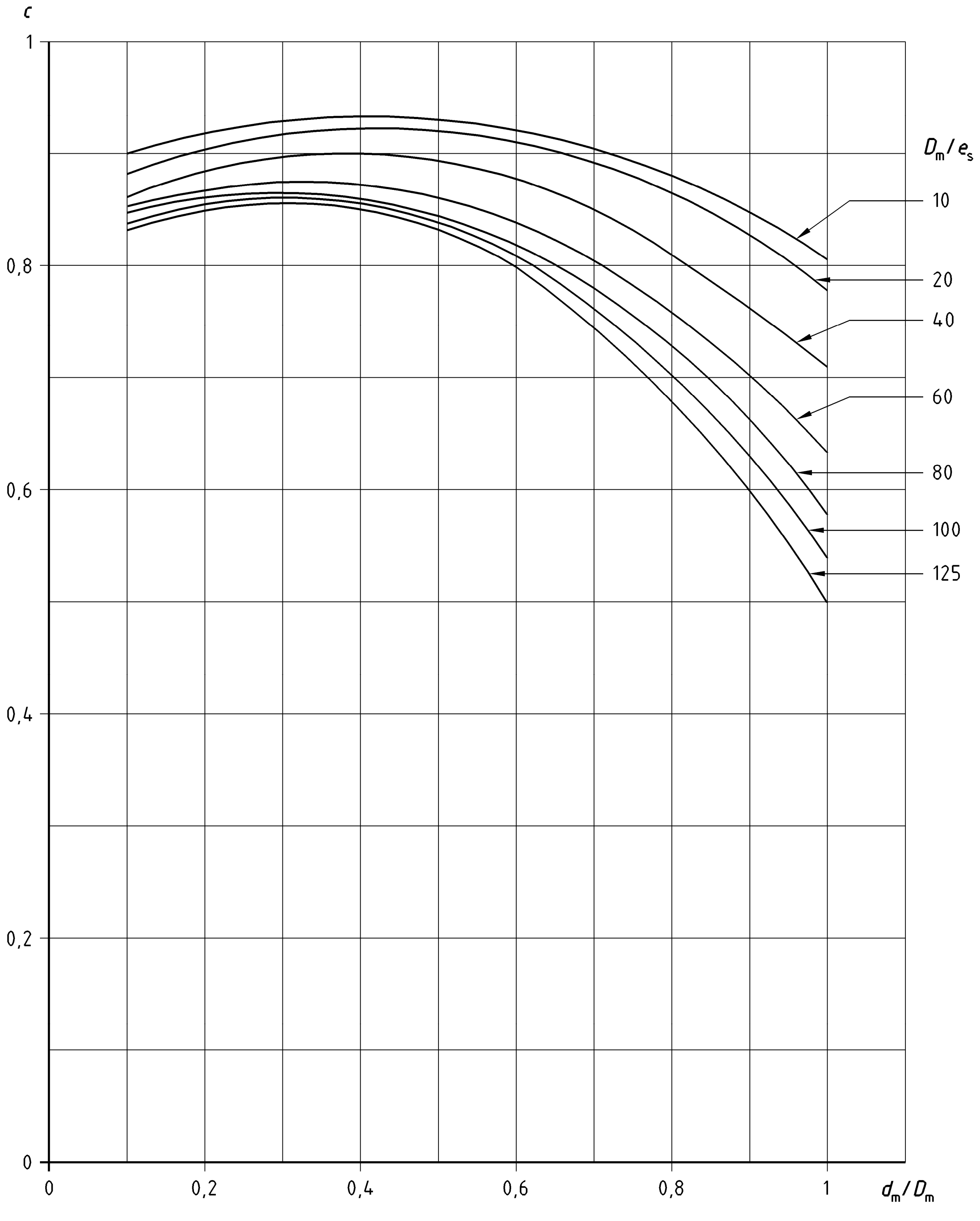


Figure O.3.2-16 — Coefficient cfh_s for e_b / e_s = 0,5

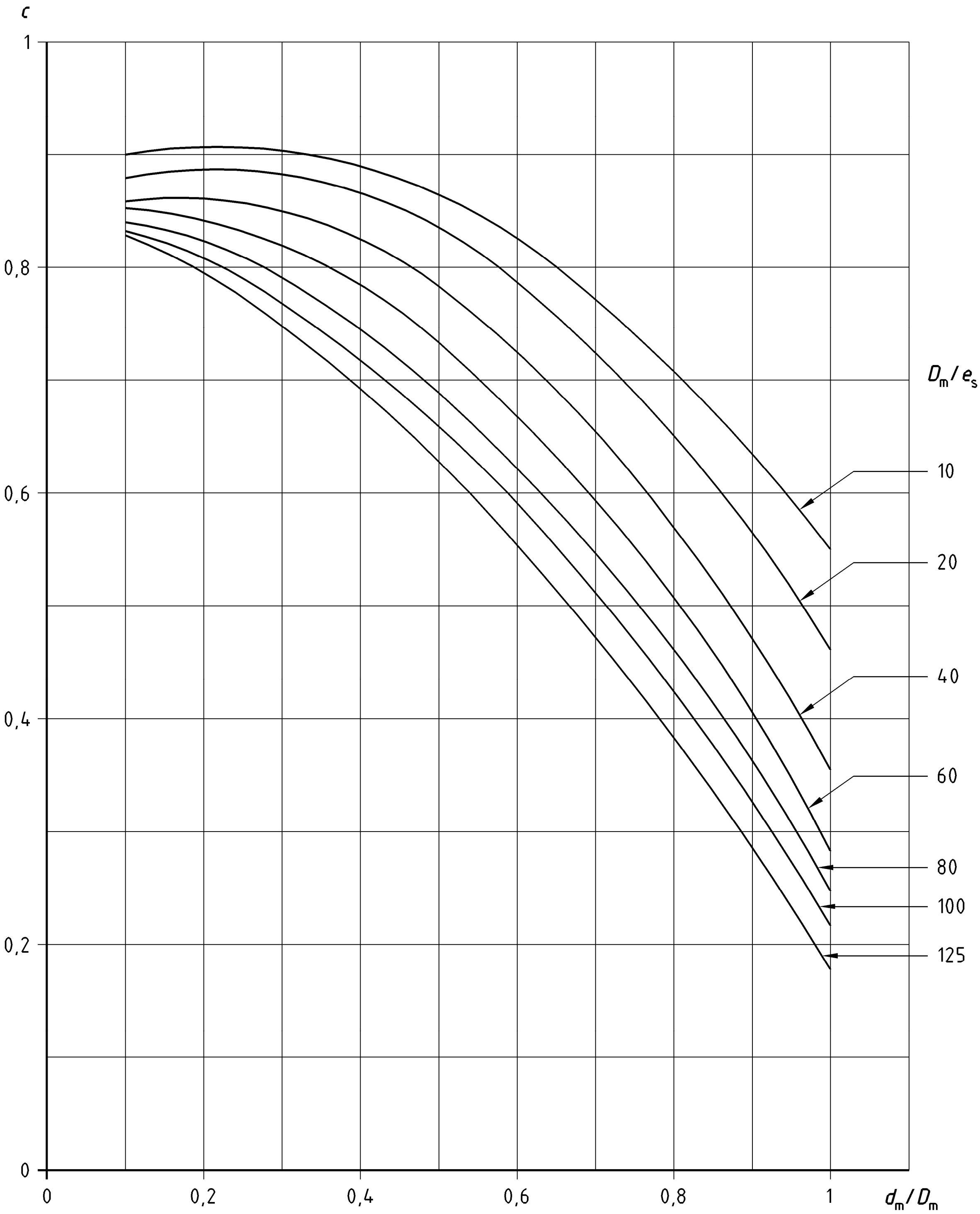


Figure O.3.2-17 — Coefficient cfp_s for e_b / e_s = 0,5

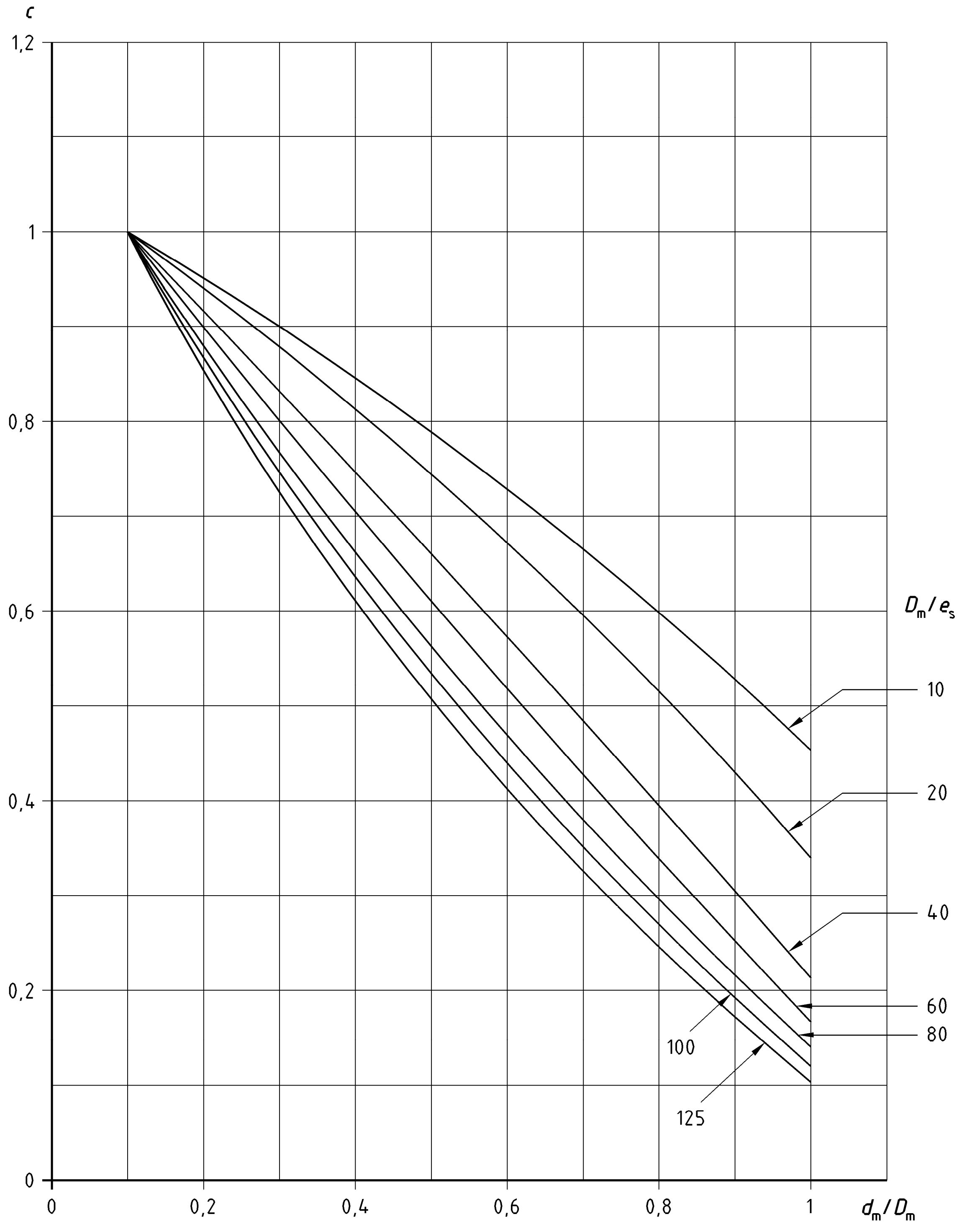


Figure O.3.2-18 — Coefficient ct_s for e_b / e_s = 0,5

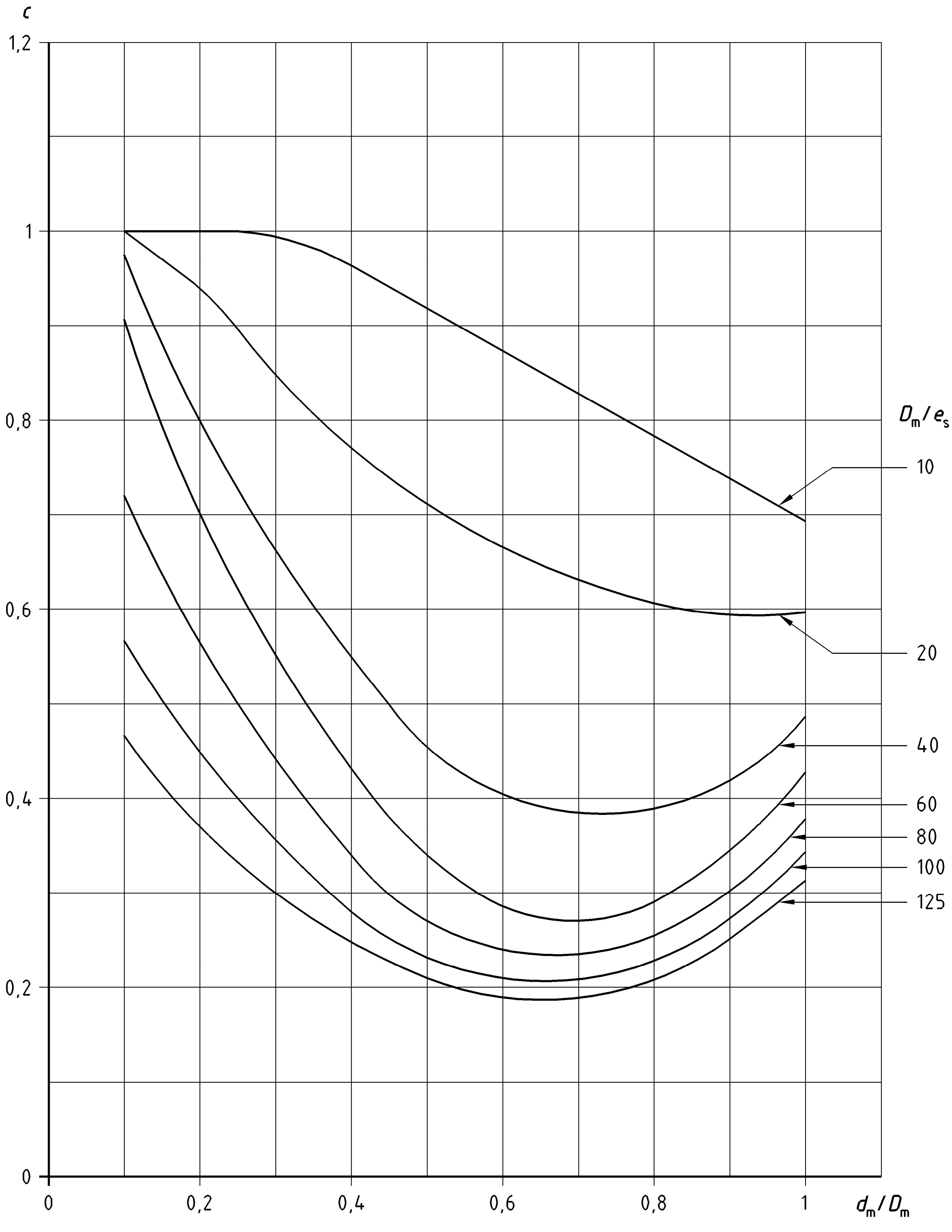


Figure O.3.2-19 — Coefficient cfh_b for e_b / e_s = 0,8

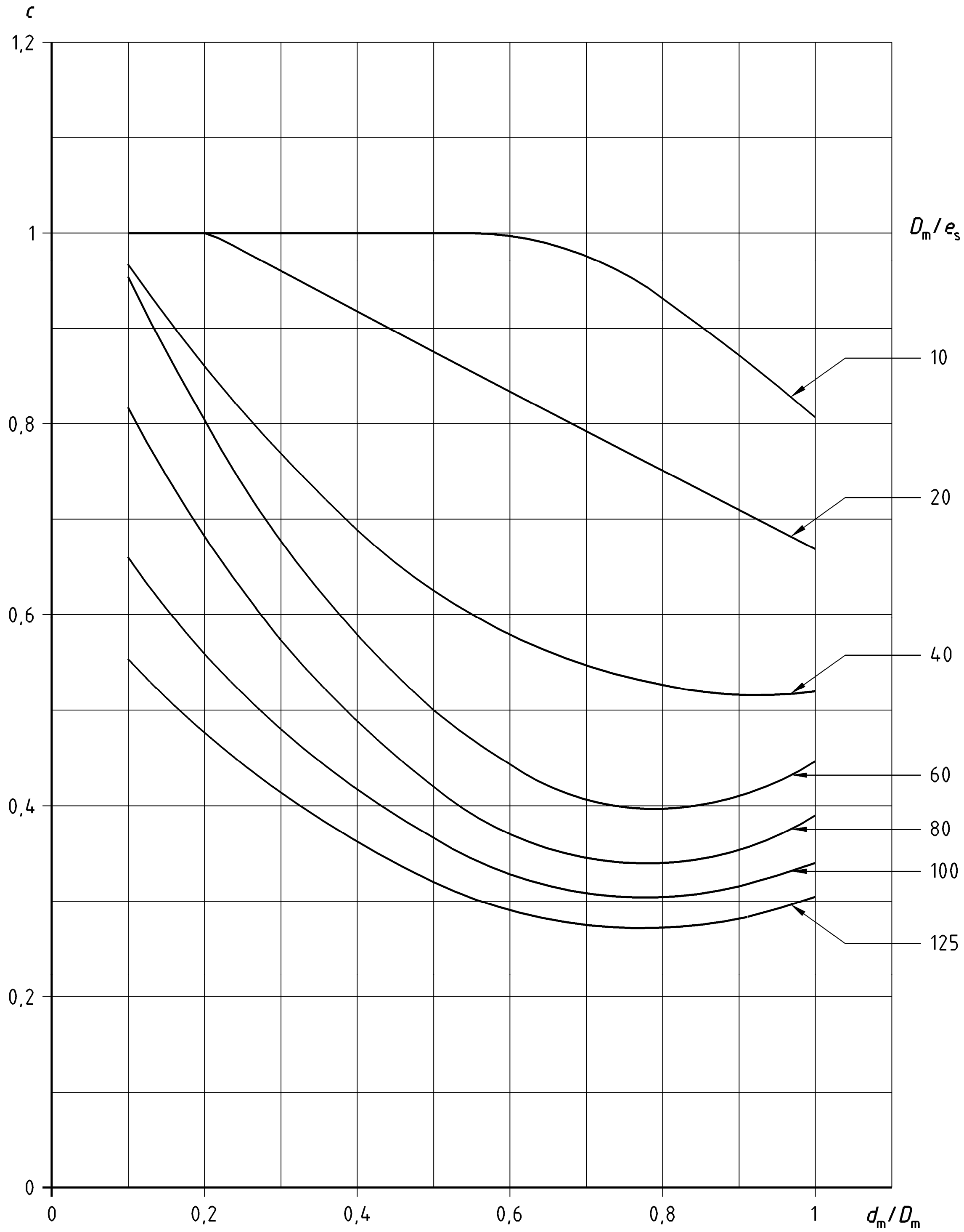


Figure O.3.2-20 — Coefficient cfp_b for e_b / e_s = 0,8

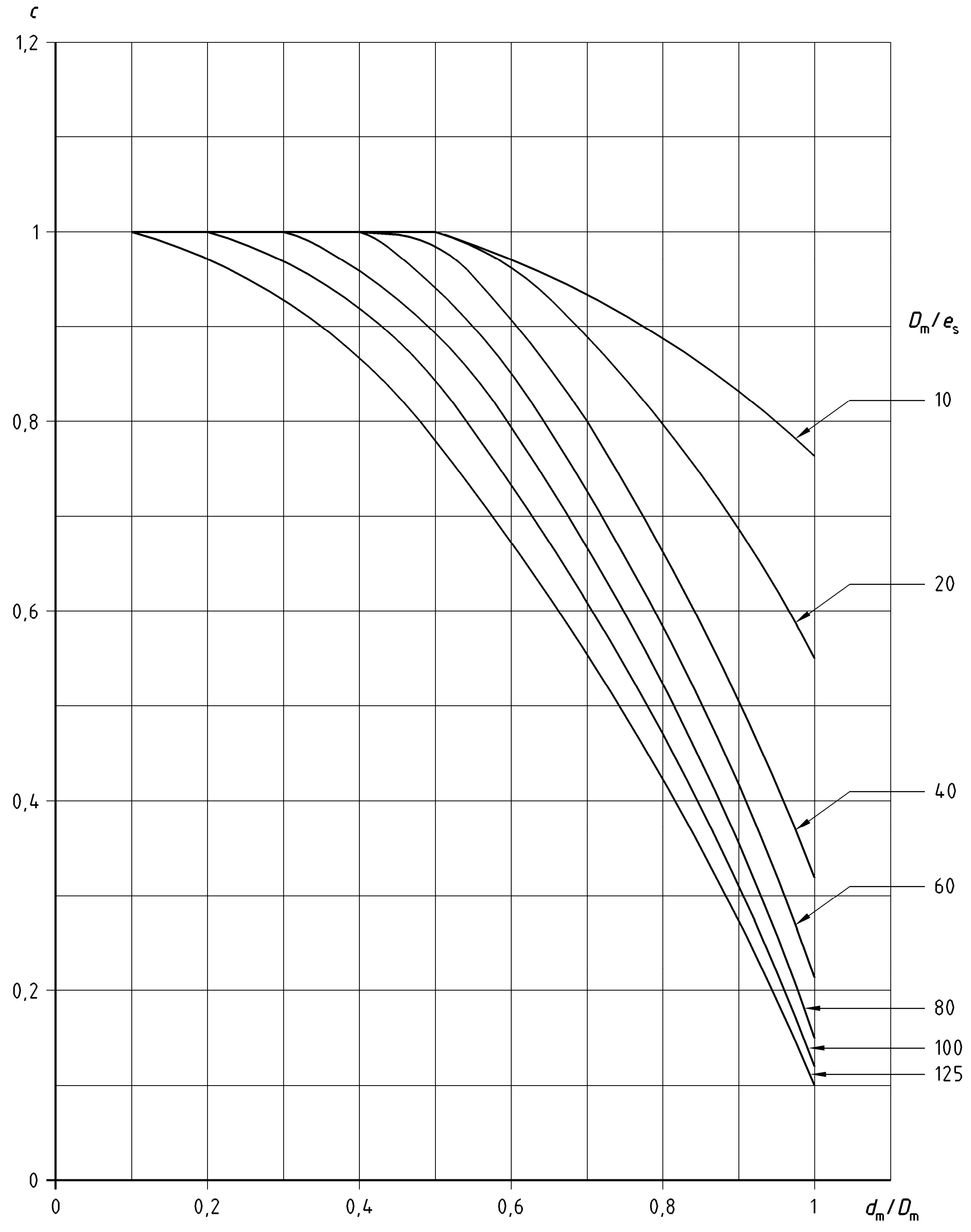


Figure O.3.2-21 — Coefficient ct_b for e_b / e_s = 0,8

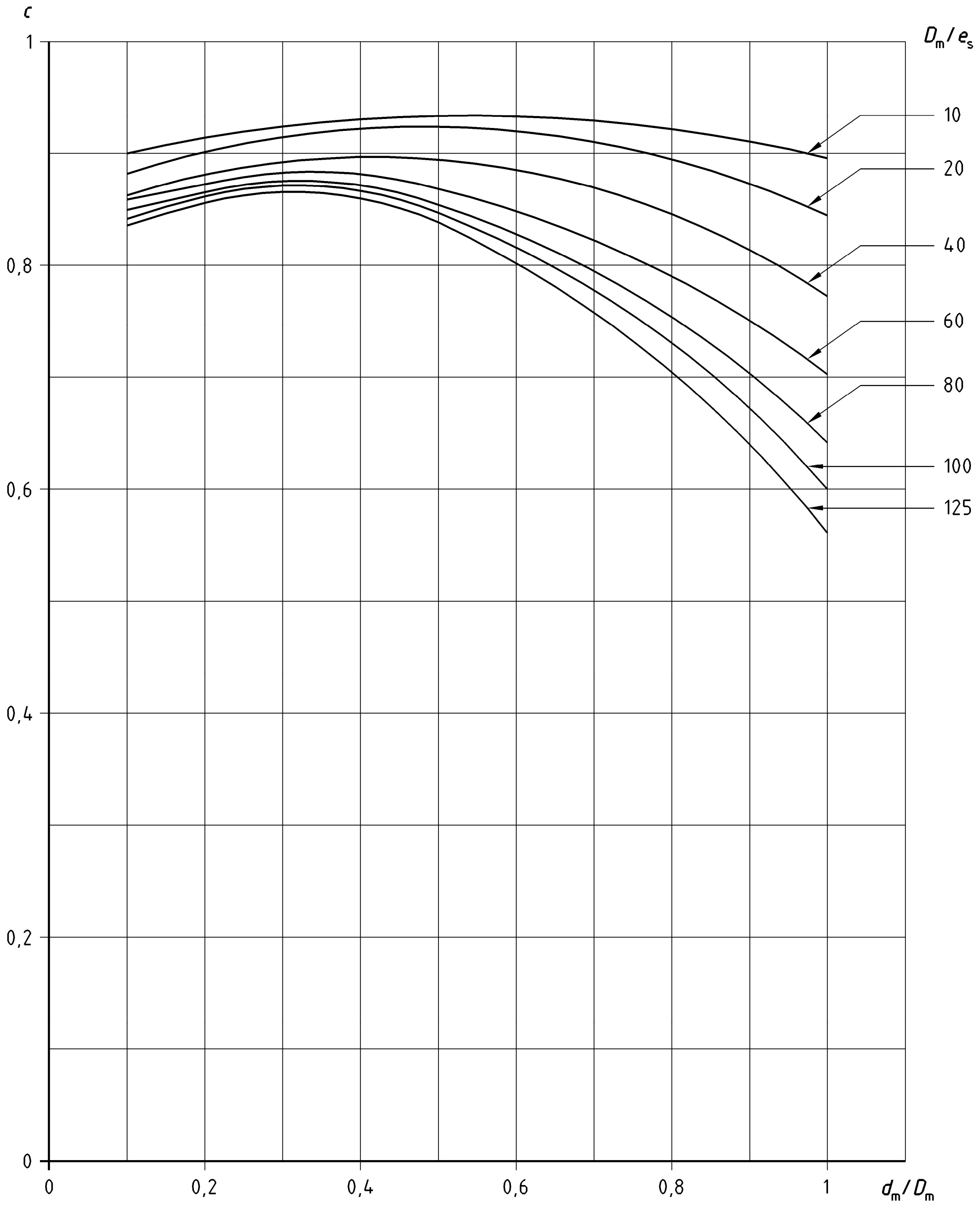


Figure O.3.2-22 — Coefficient cfh_s for e_b / e_s = 0,8

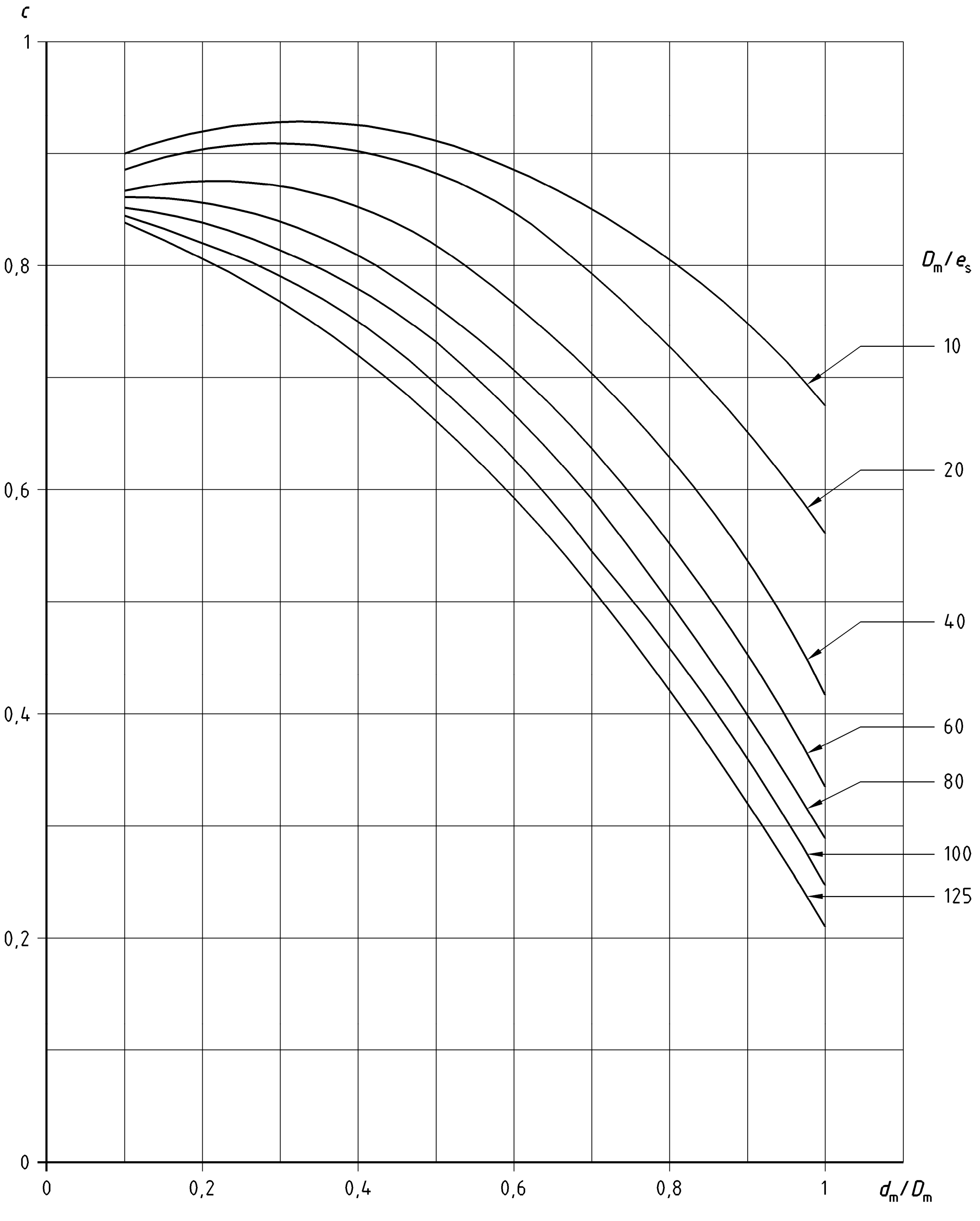


Figure O.3.2-23 — Coefficient cfp_s for e_b / e_s = 0,8

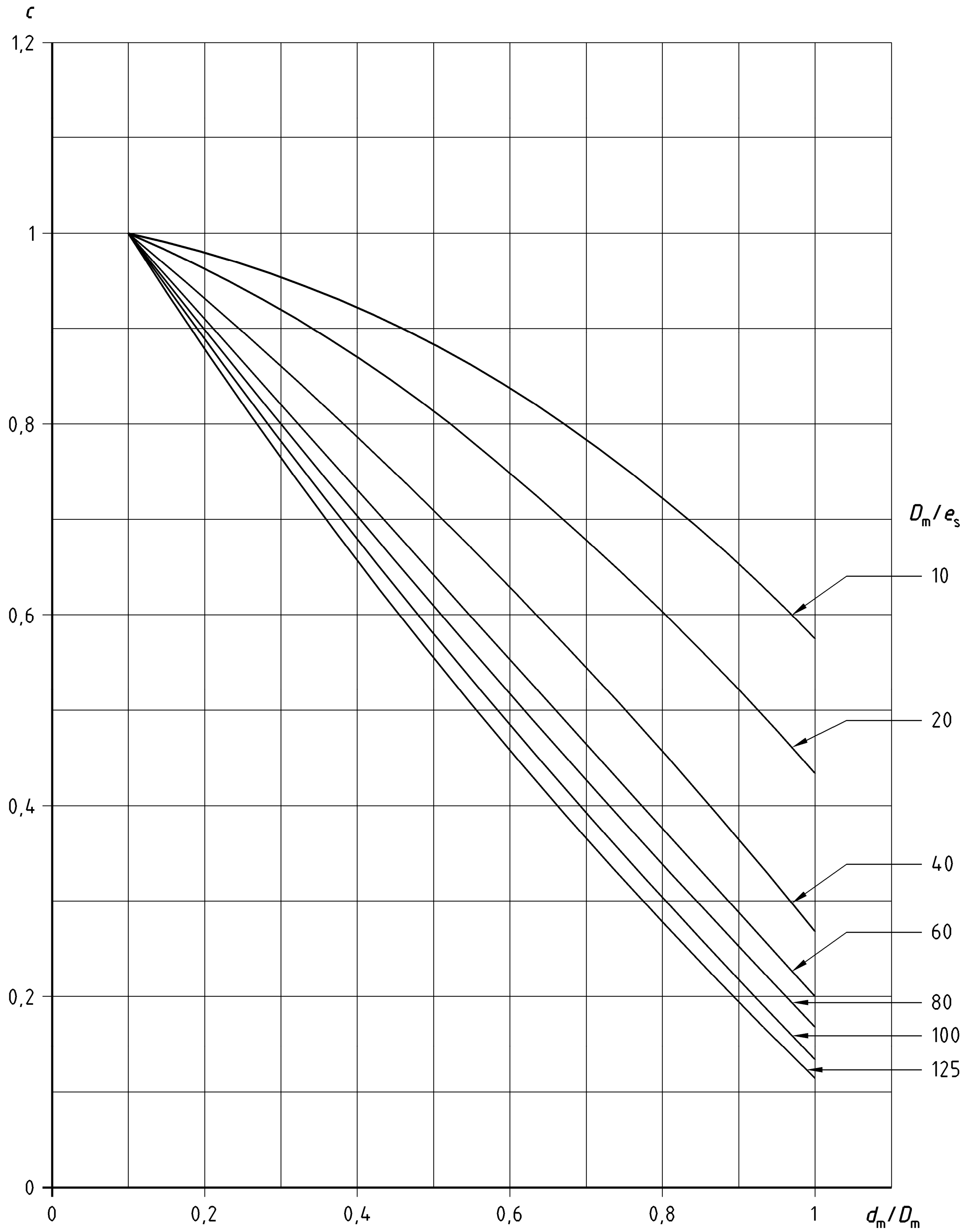


Figure O.3.2-24 — Coefficient ct_s for e_b / e_s = 0,8

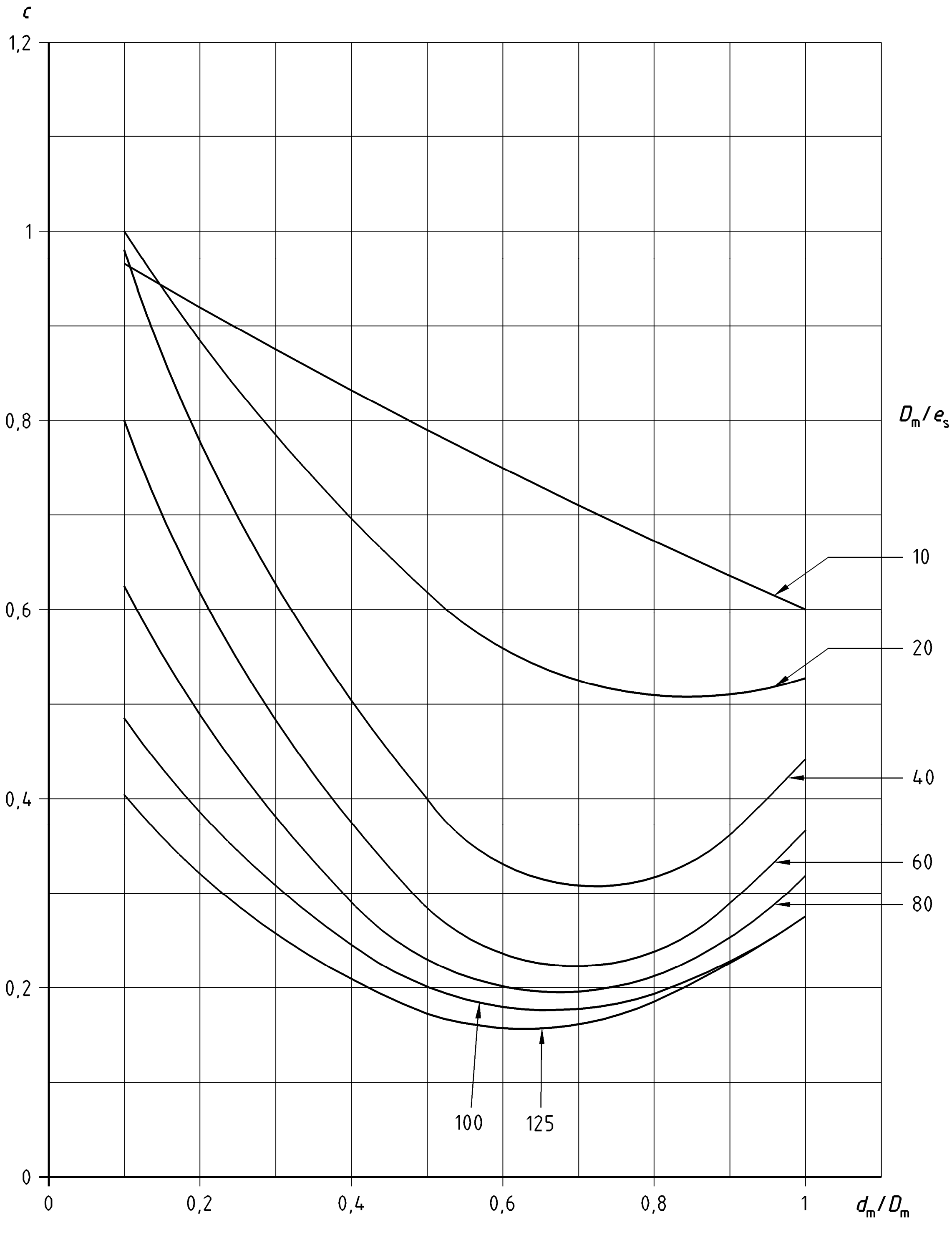


Figure O.3.2-25 — Coefficient cfh_b for e_b / e_s = 1

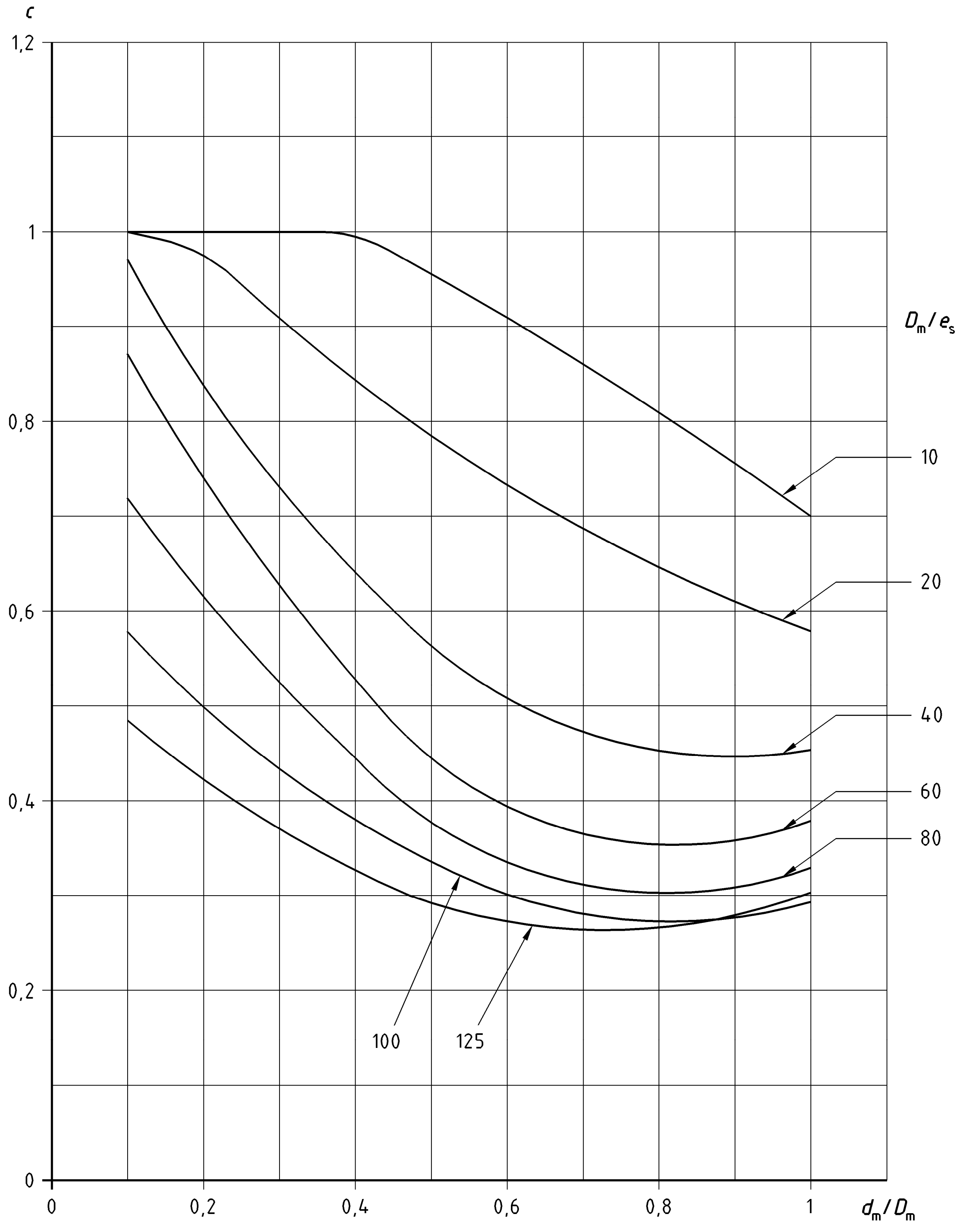


Figure O.3.2-26 — Coefficient cfp_b for e_b / e_s = 1

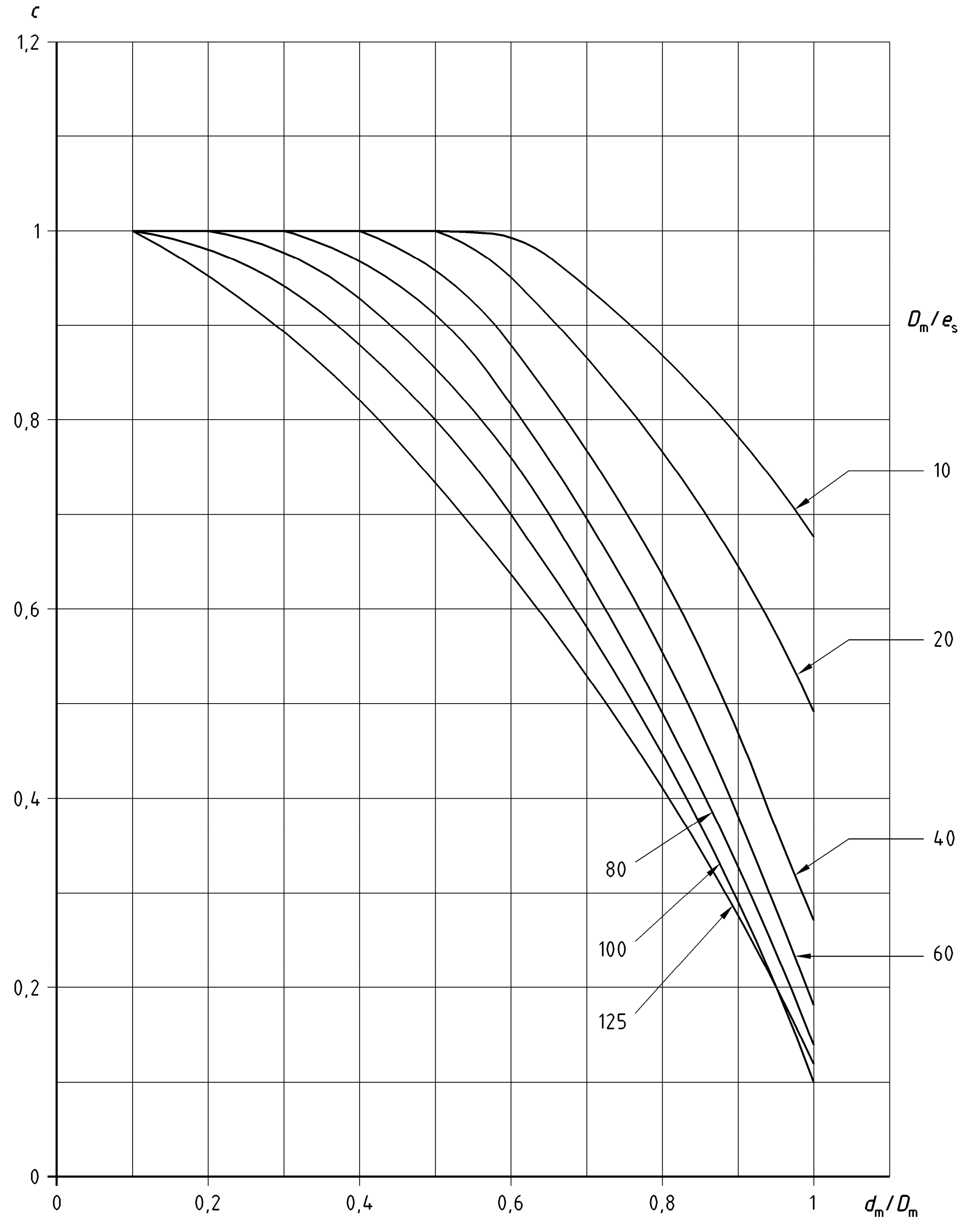


Figure O.3.2-27 — Coefficient ct_b for e_b / e_s = 1

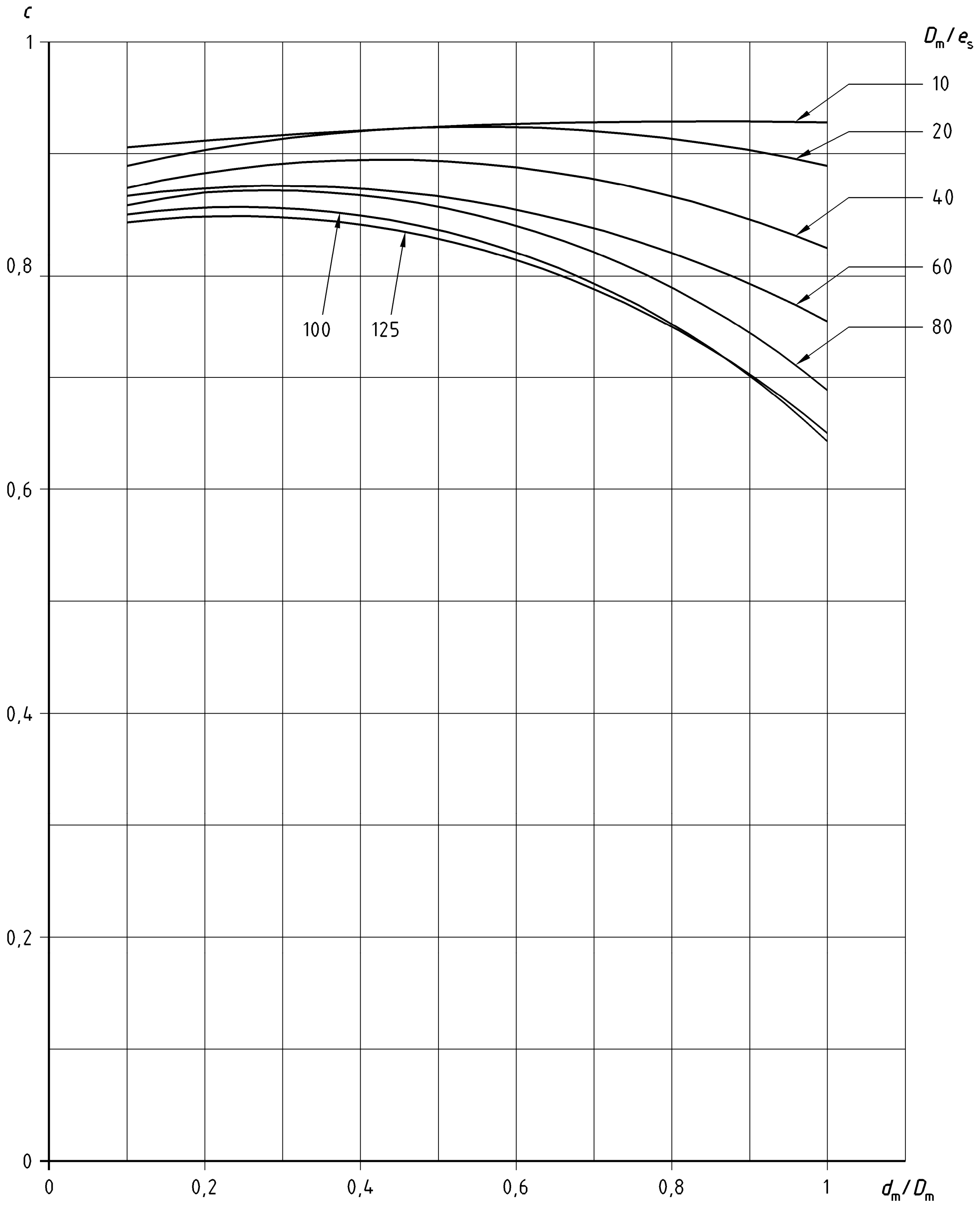


Figure O.3.2-28 — Coefficient cfh_s for e_b / e_s = 1

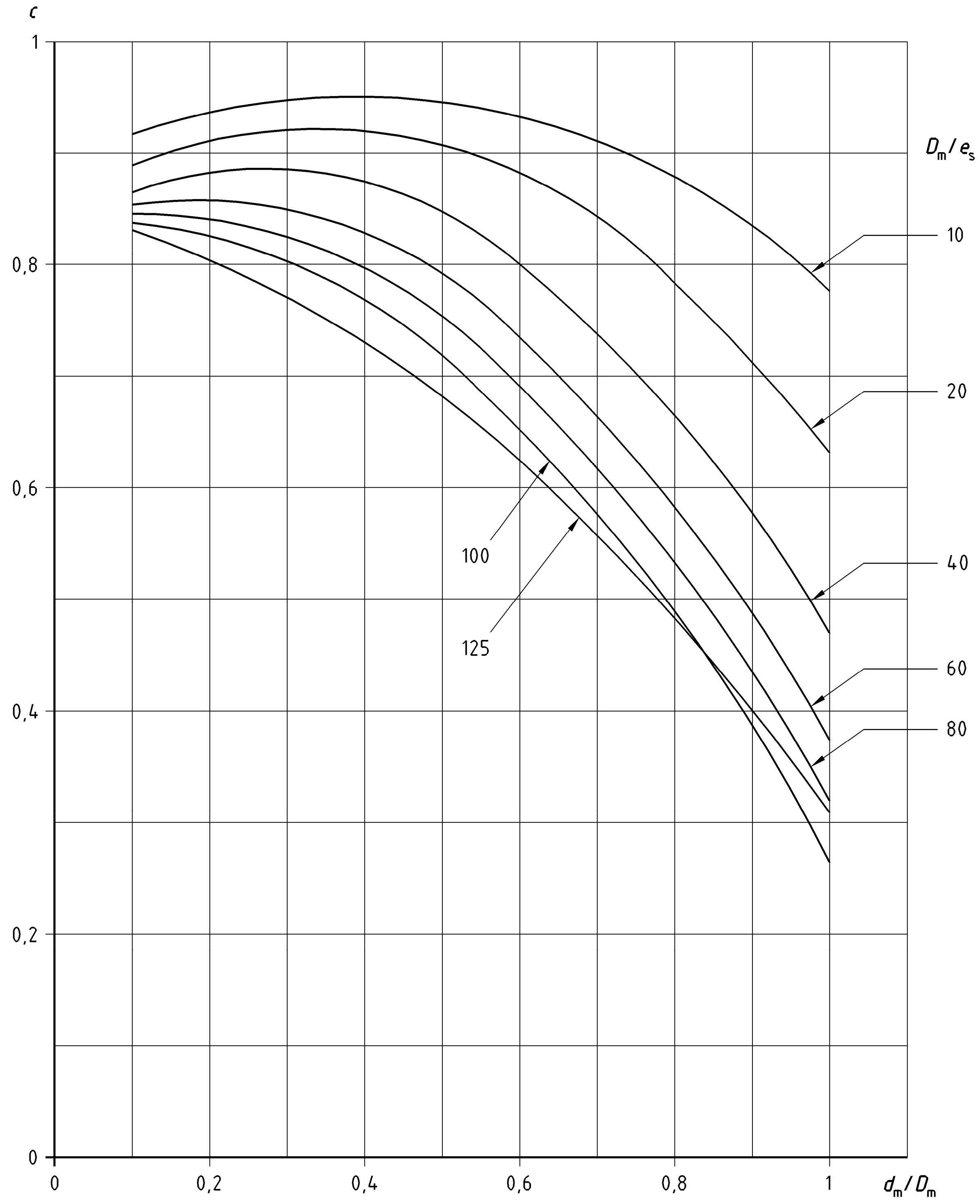


Figure O.3.2-29 — Coefficient cfp_s for e_b / e_s = 1

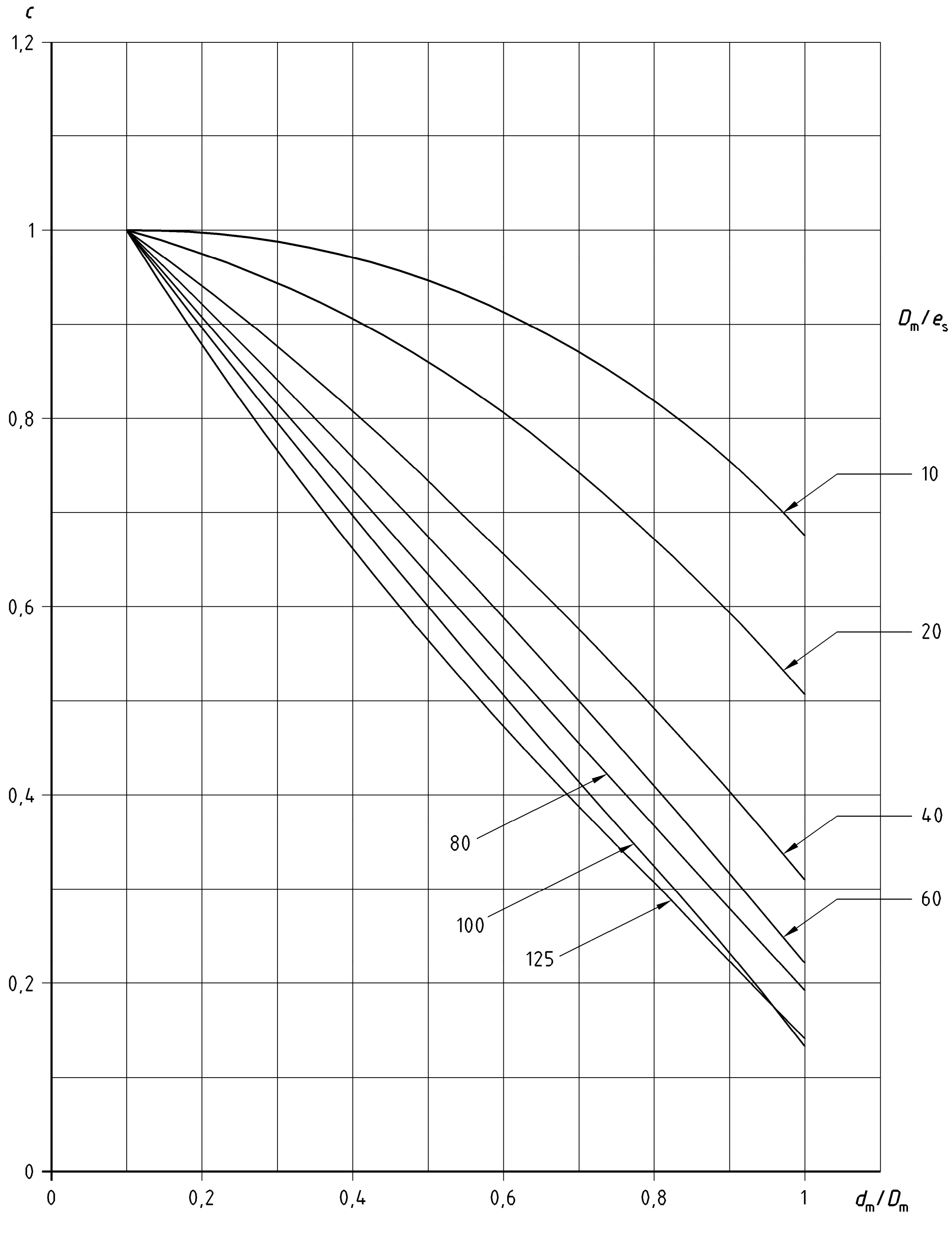


Figure O.3.2-30 — Coefficient ct_s for e_b / e_s = 1

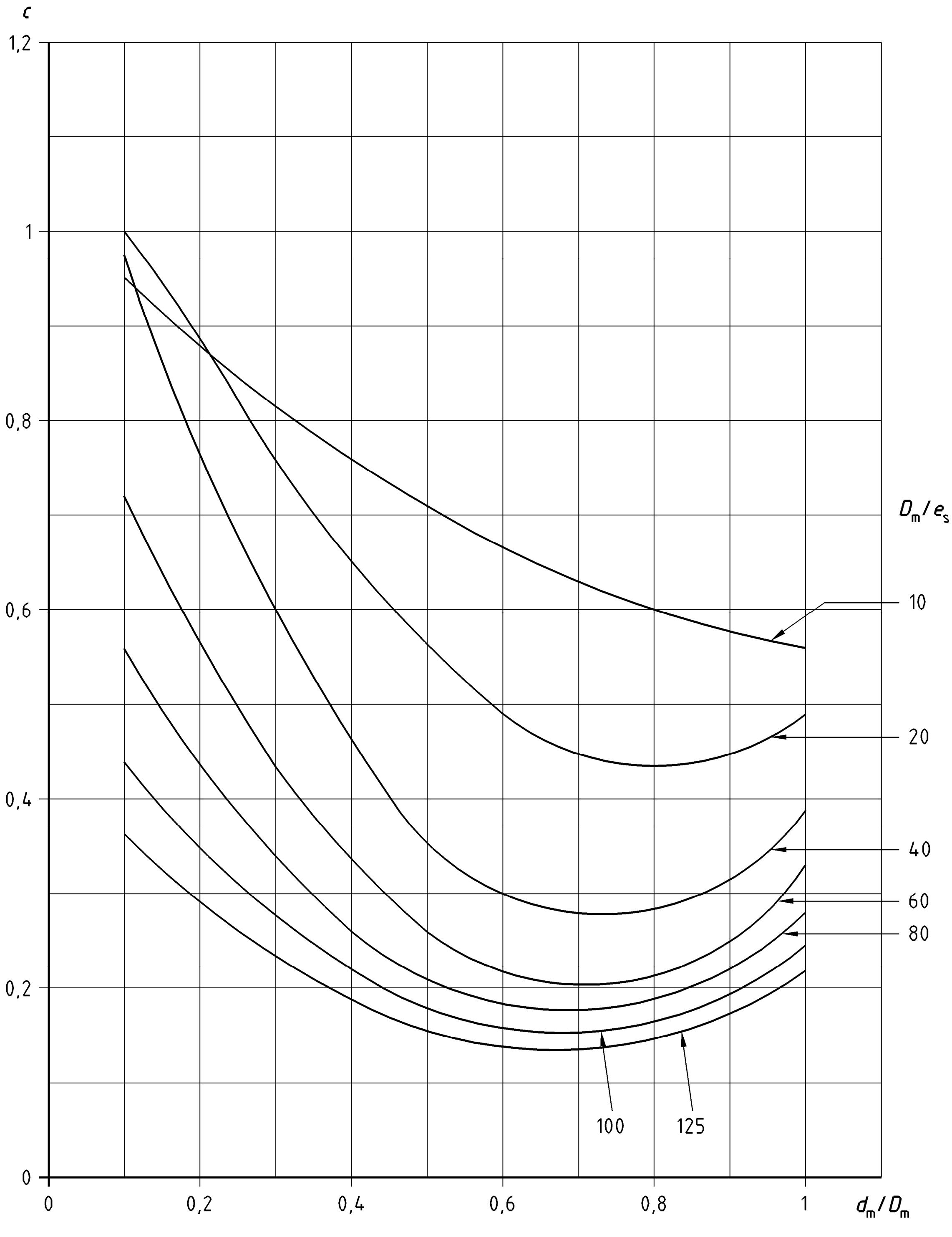


Figure O.3.2-31 — Coefficient cfh_b for e_b / e_s = 1,2

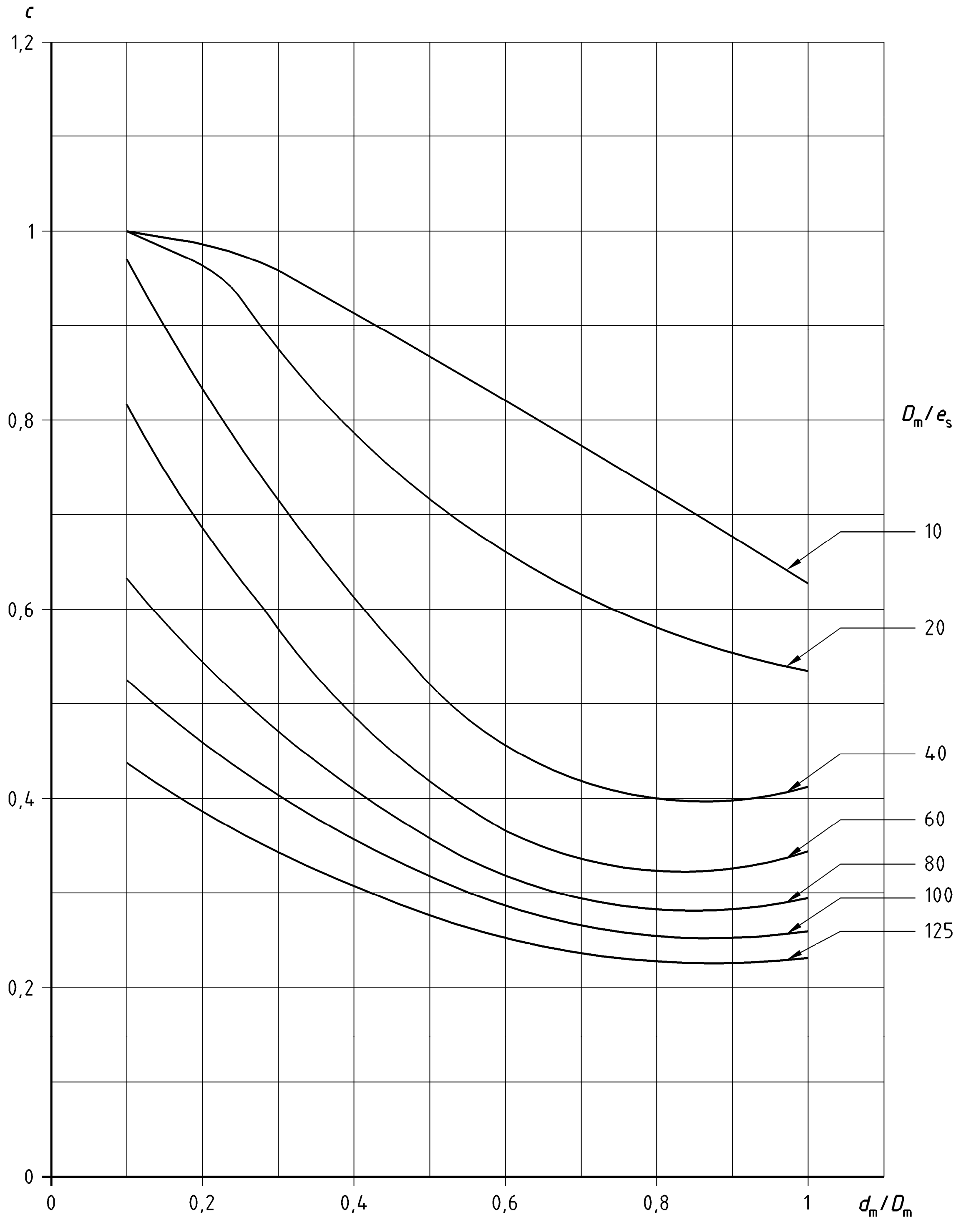


Figure O.3.2-32 — Coefficient cfp_b for e_b / e_s = 1,2

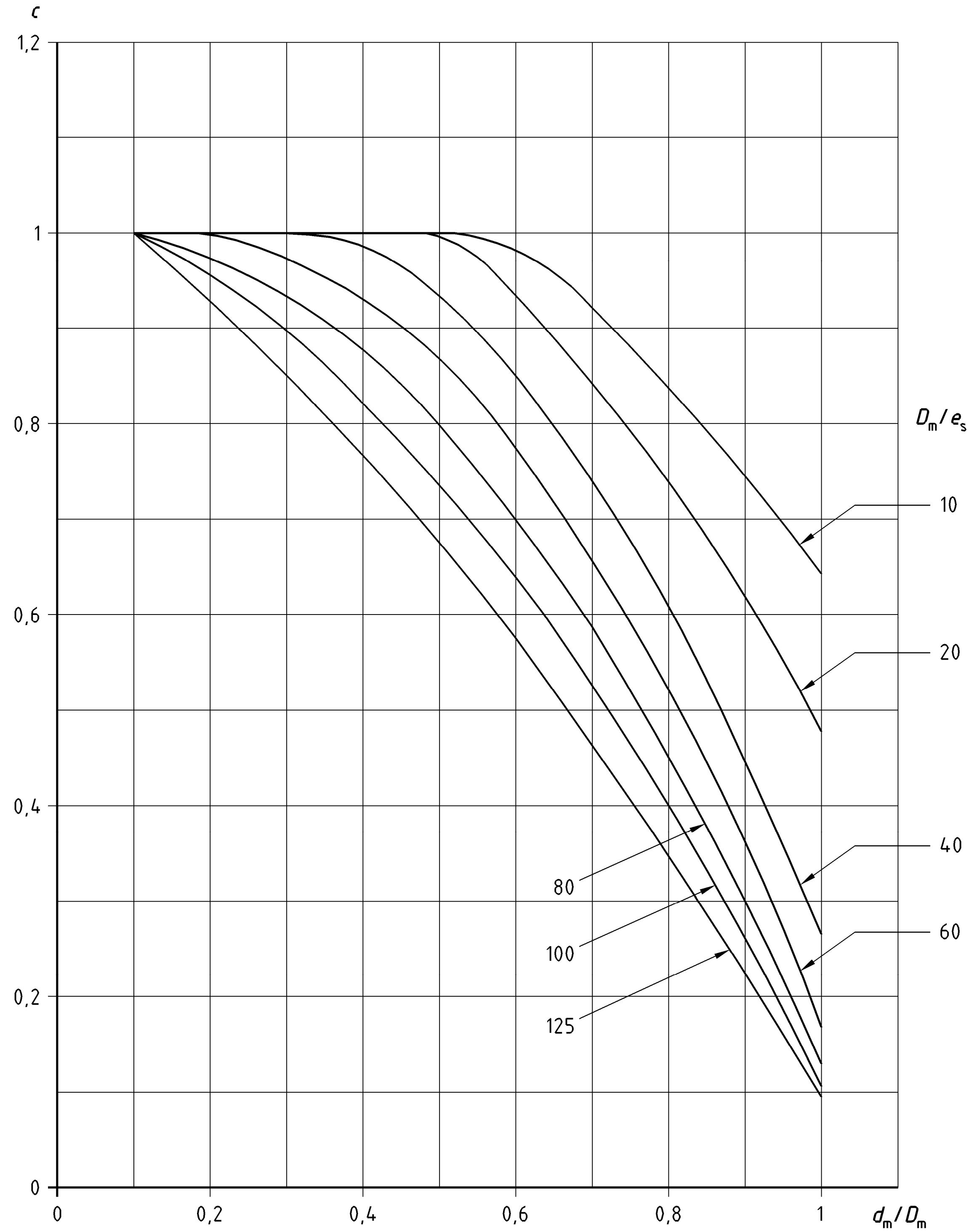


Figure O.3.2-33 — Coefficient ct_b for e_b / e_s = 1,2

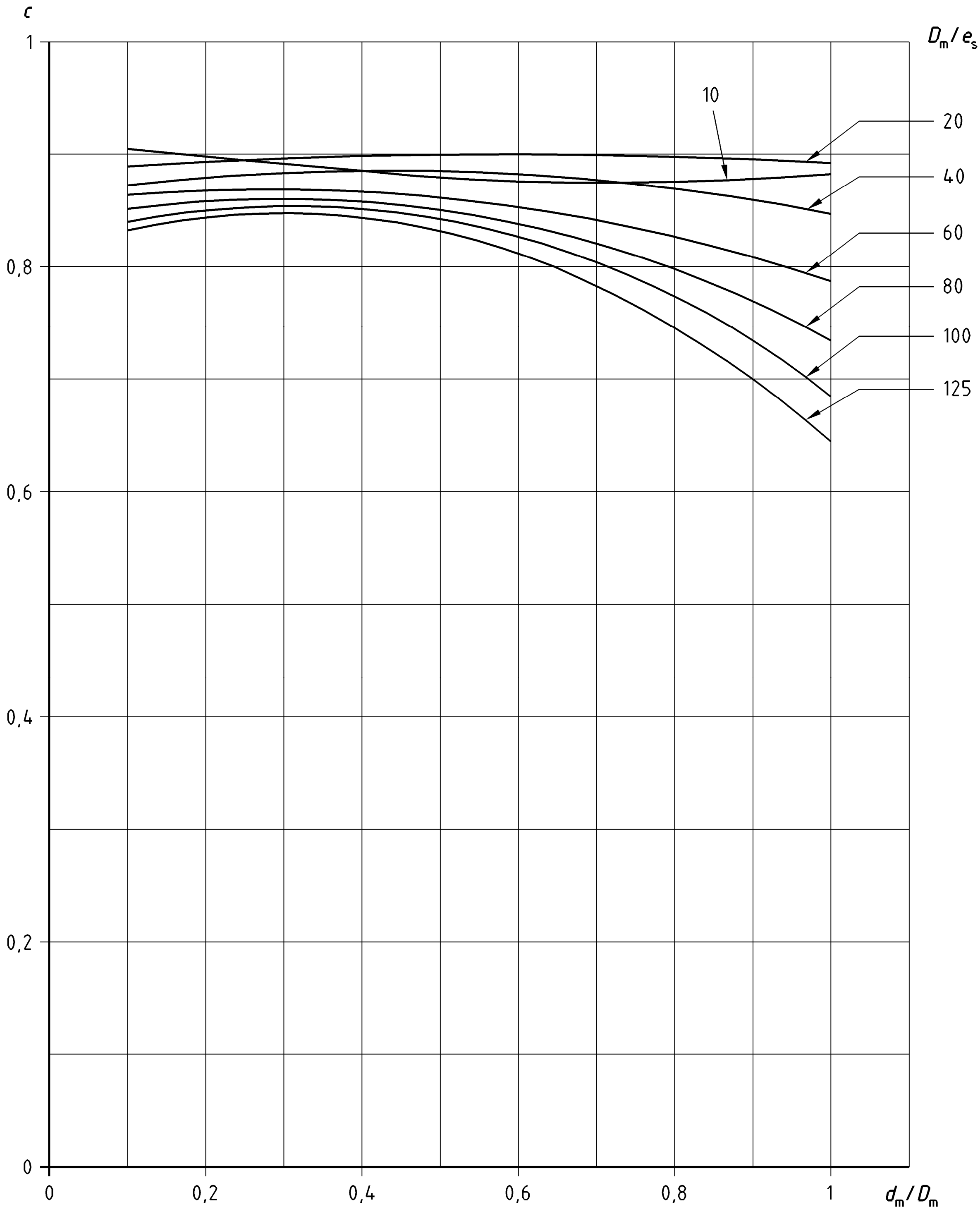


Figure O.3.2-34 — Coefficient cfh_s for e_b / e_s = 1,2

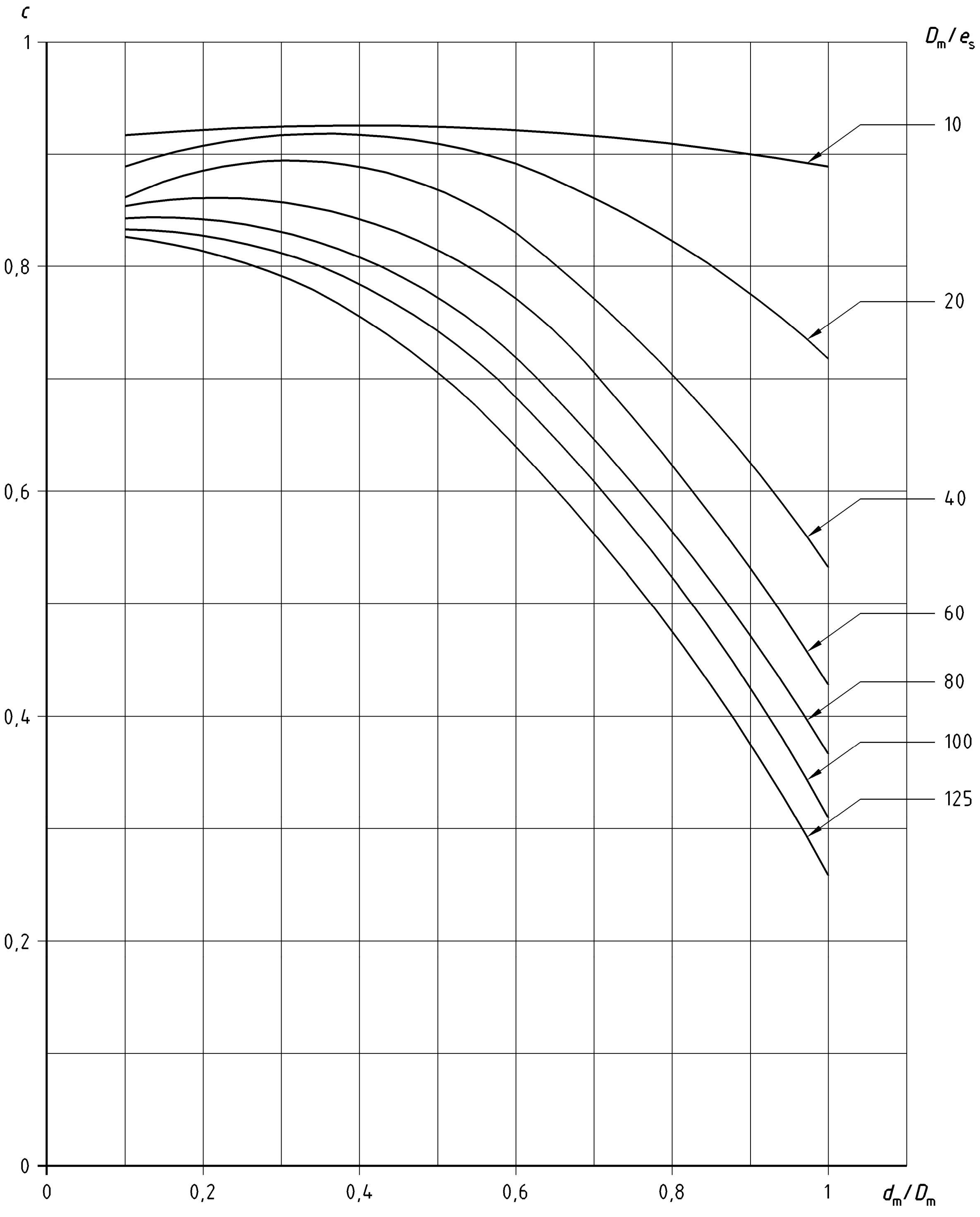


Figure O.3.2-35 — Coefficient cfp_s for e_b / e_s = 1,2

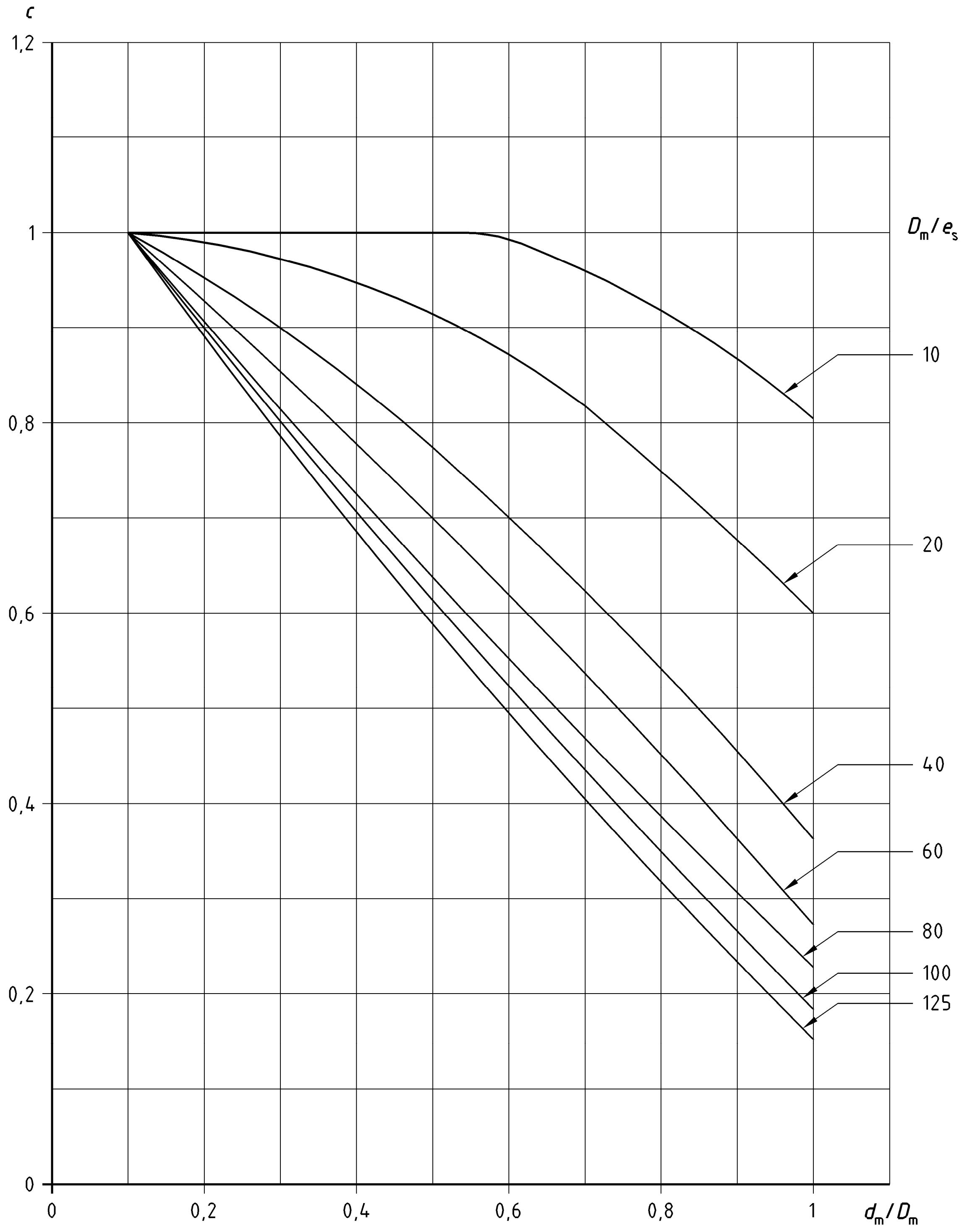


Figure O.3.2-36 — Coefficient ct_s for e_b / e_s = 1,2

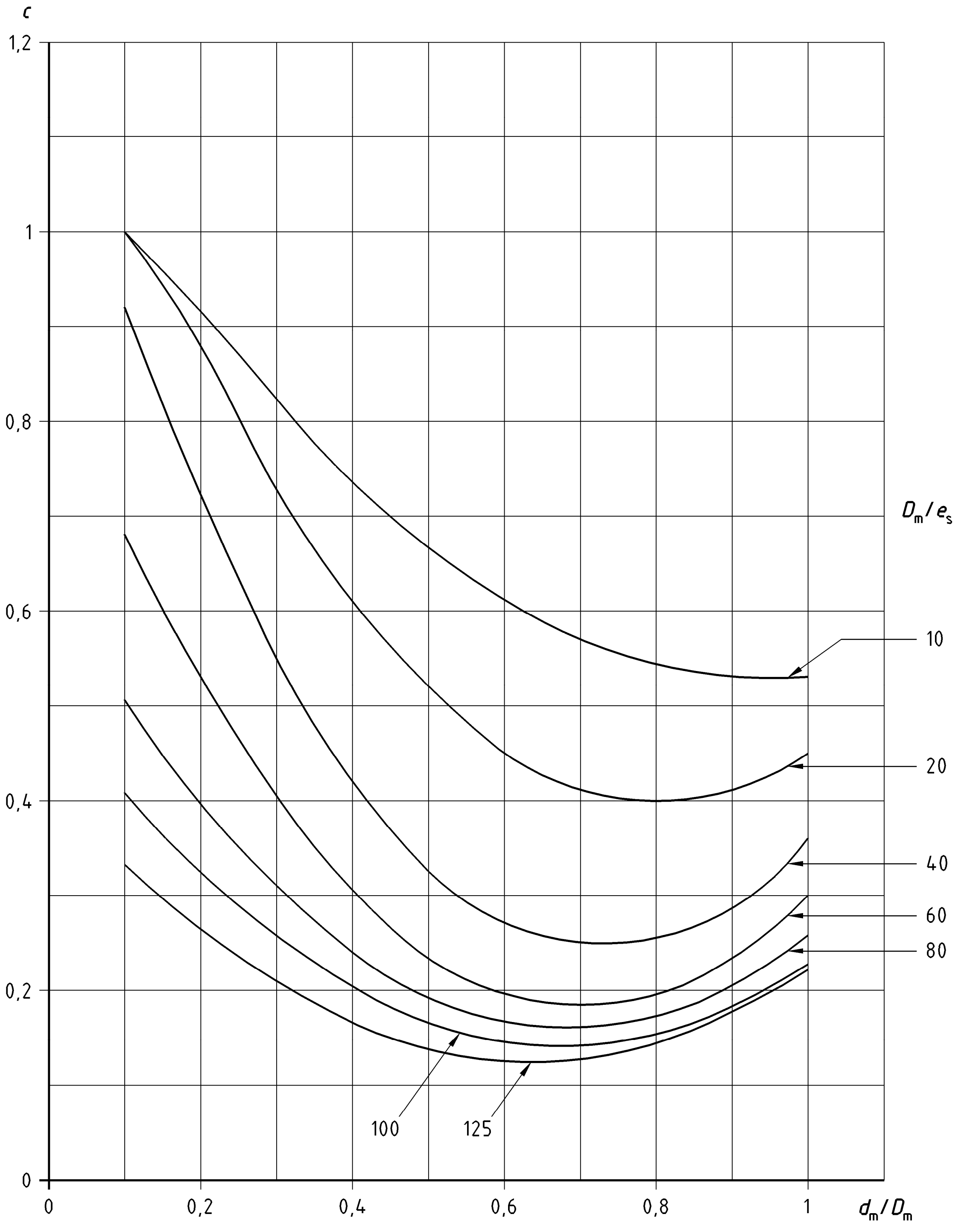


Figure O.3.2-37 — Coefficient cfh_b for e_b / e_s = 1,5

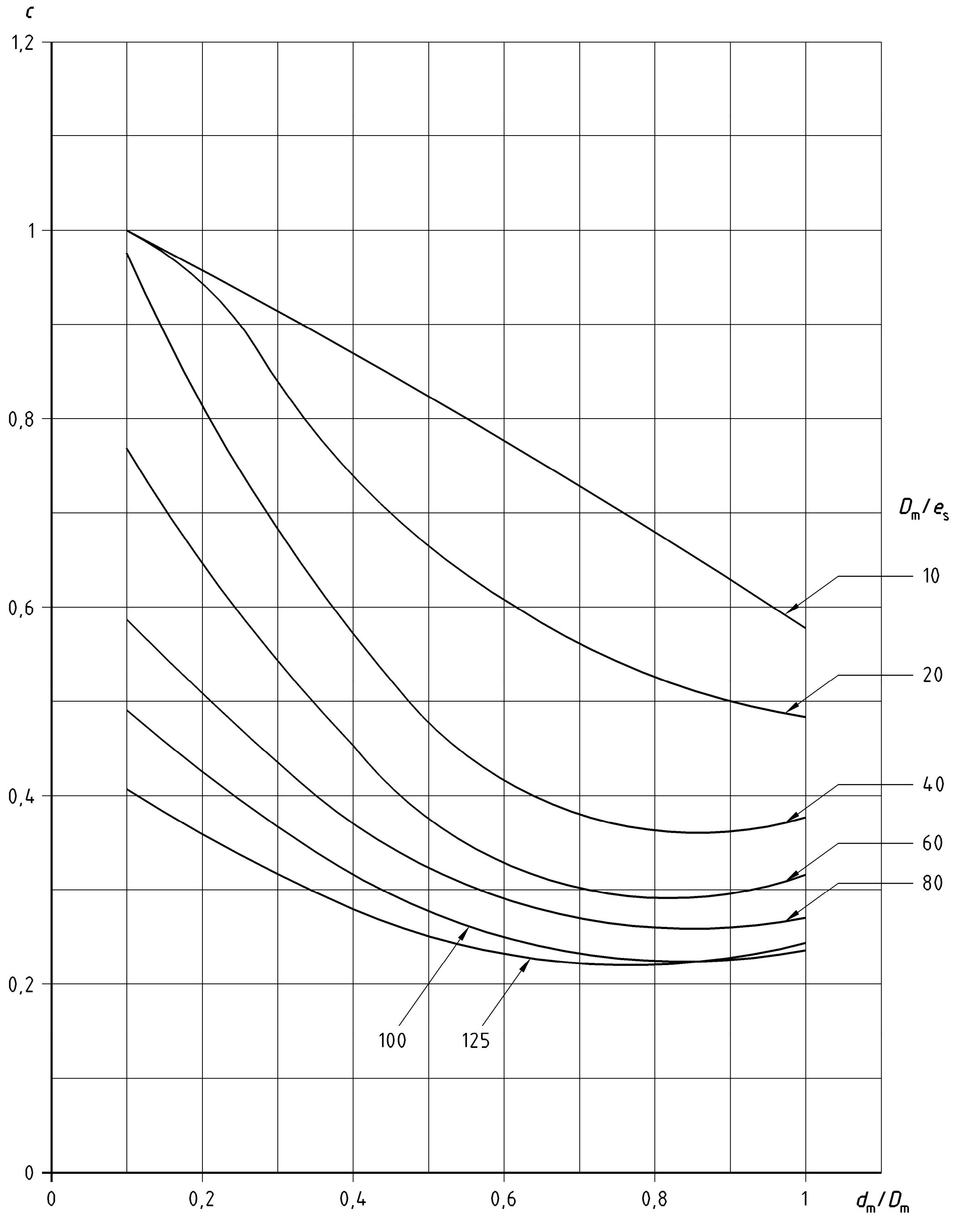


Figure O.3.2-38 — Coefficient cfp_b for e_b / e_s = 1,5

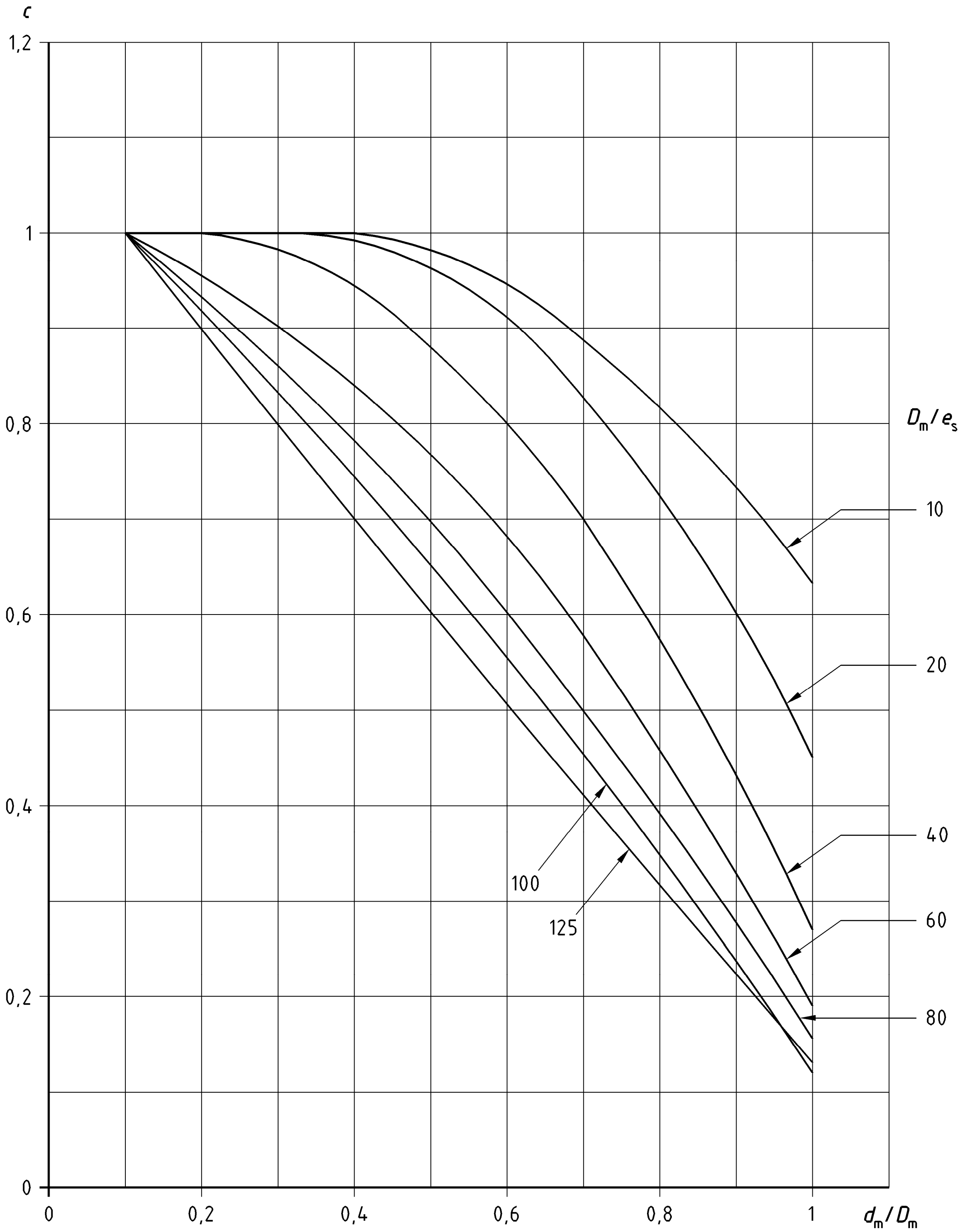


Figure O.3.2-39 — Coefficient ct_b for e_b / e_s = 1,5

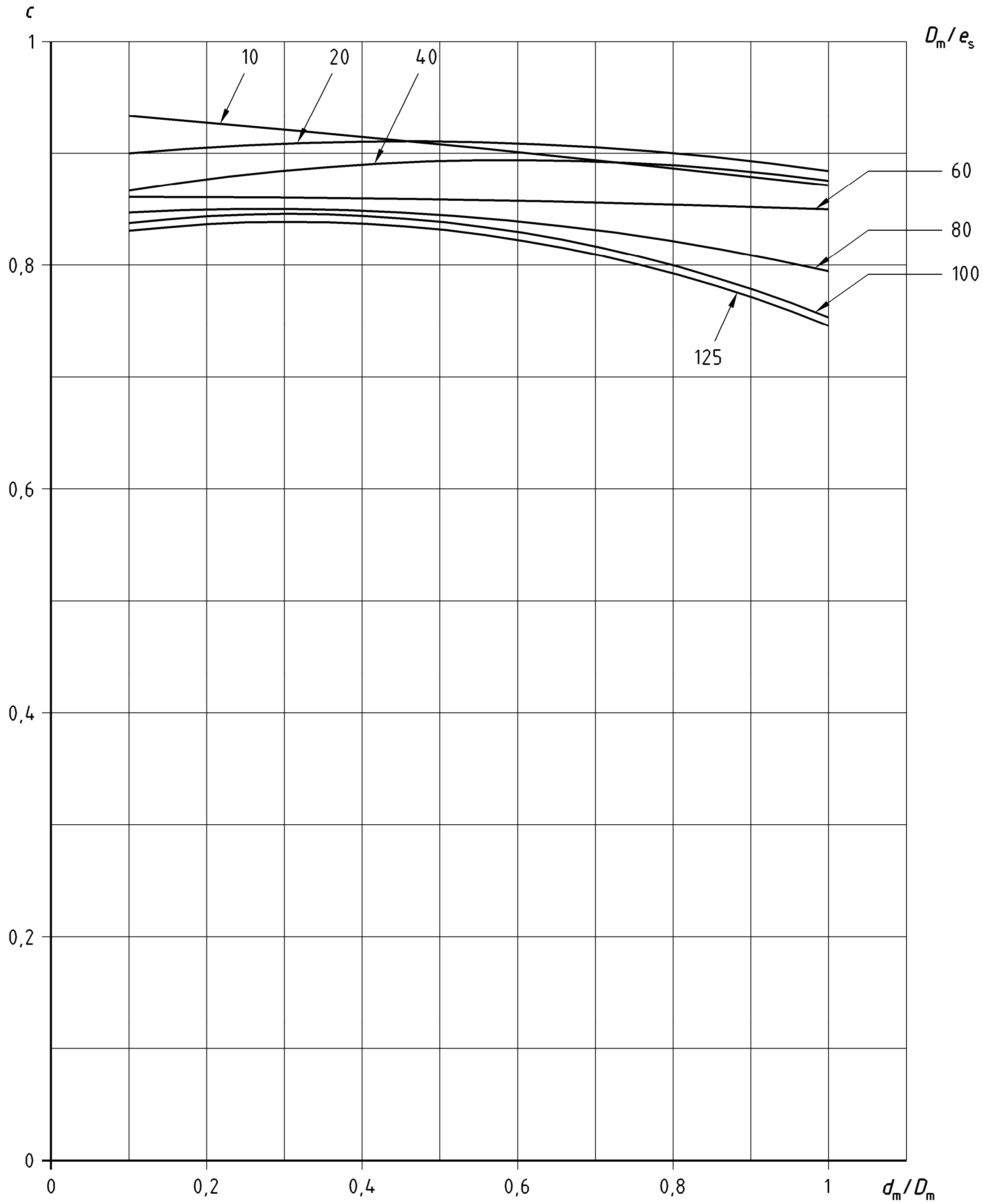


Figure O.3.2-40 — Coefficient cfh_s for e_b / e_s = 1,5

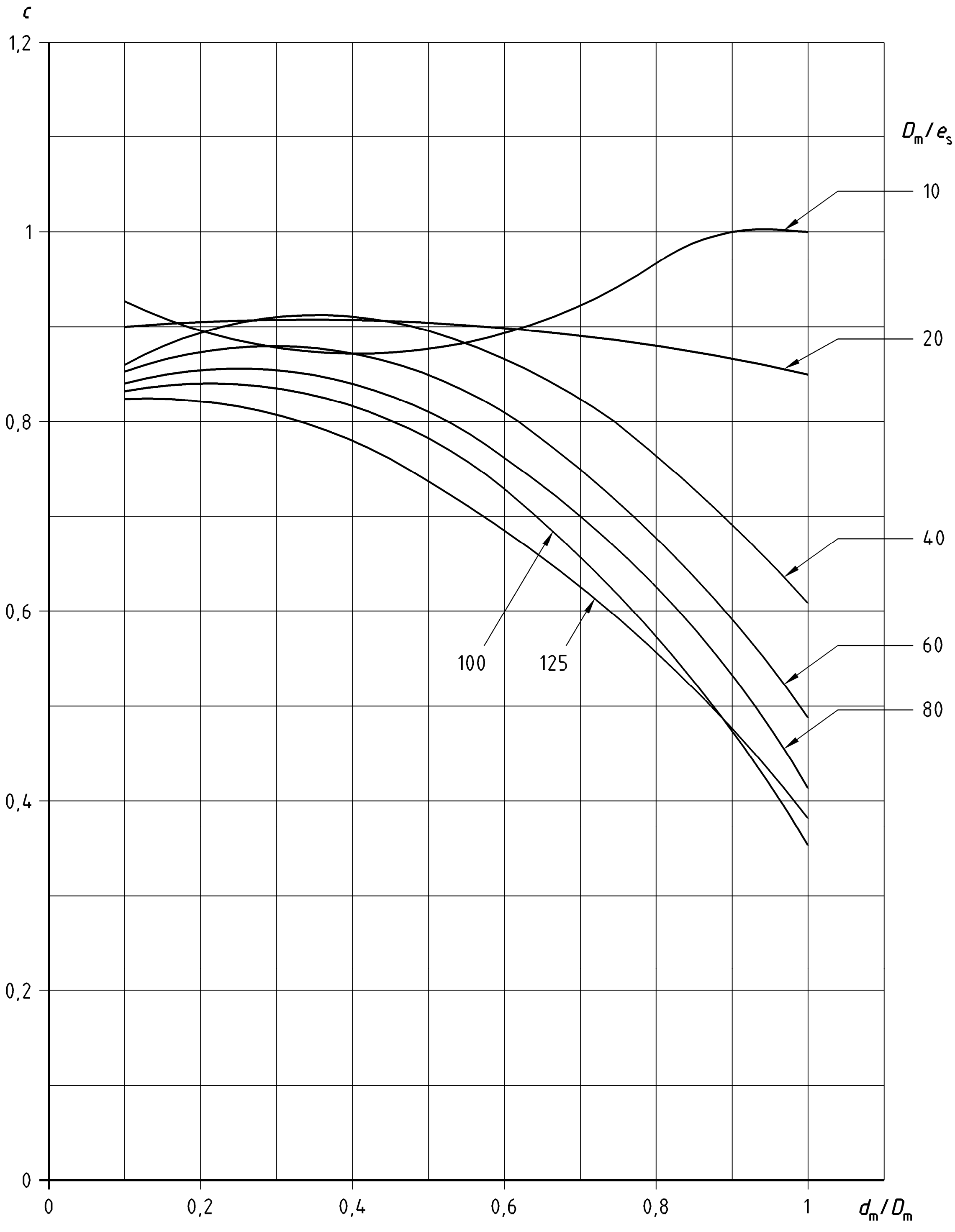


Figure O.3.2-41 — Coefficient cfp_s for e_b / e_s = 1,5

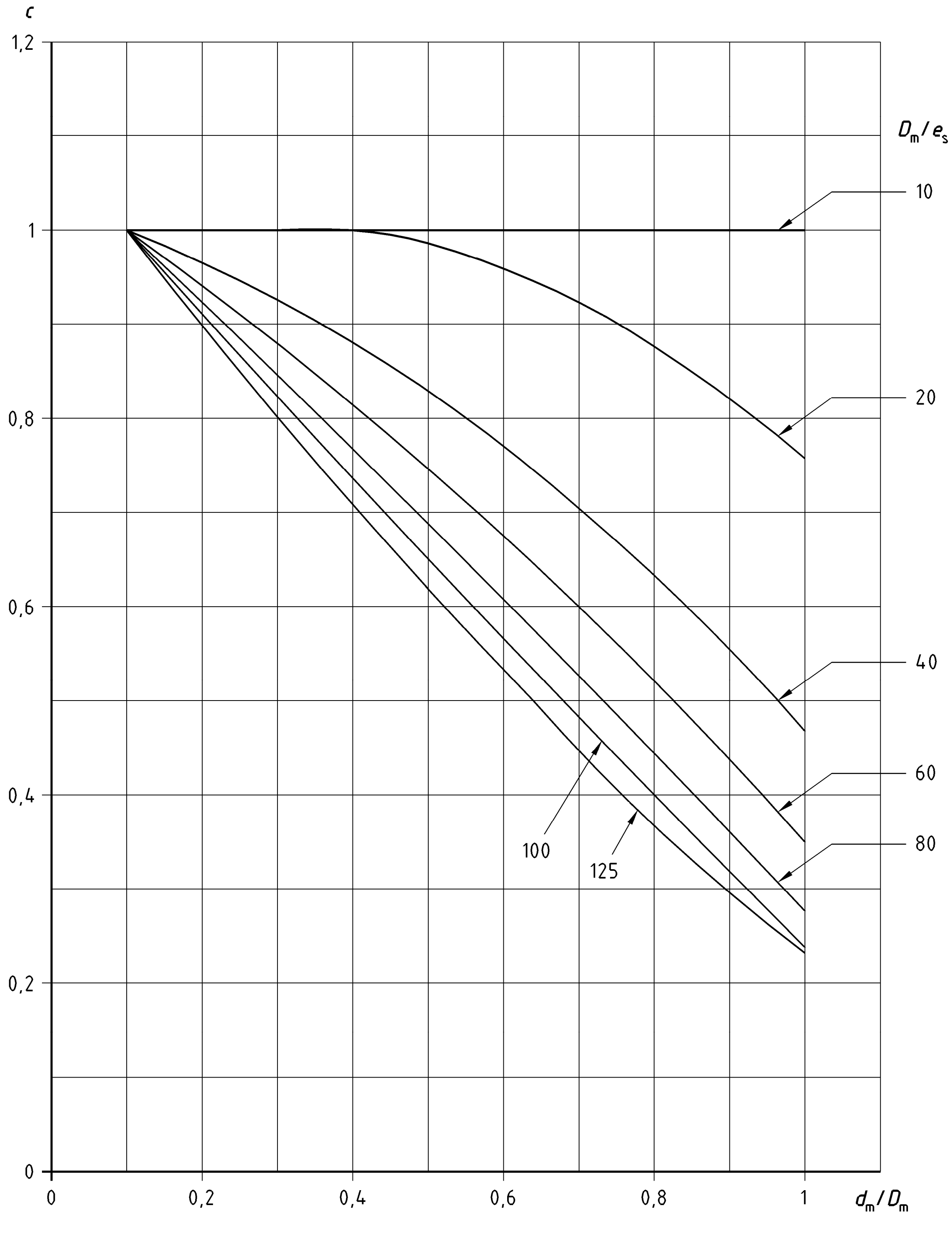


Figure O.3.2-42 — Coefficient ct_s for e_b / e_s = 1,5

Table O.3.2-2 — coefficient of the polynomial equations describing the curves of Graphs O.3.2-7 to O.3.2-42

	D_m / e_s	*A*	*B*	*C*
Figure O.3.2-7 Coefficient cfh_b for e_b / e_s = 0,2	10	0,763 7	- 1,155 6	1,310 4
	20	0,676 8	- 0,952 0	1,182 4
	40	0,322 6	- 0,475 4	1,023 7
	60	0,225 9	- 0,348 9	0,980 9
	80	0,542 1	- 0,675	1,001 3
	100	1,210 8	- 1,371 2	1,065 0
	125	1,675 7	- 1,875 5	1,093 6
Figure O.3.2-8 Coefficient cfp_b for e_b / e_s = 0,2	10	0,960 7	- 1,385 4	1,325 5
	20	0,604 2	- 0,917 2	1,178 4
	40	0,303 5	- 0,470 4	1,018 3
	60	0,206 9	- 0,332 8	0,969 4
	80	0,196 5	- 0,300 4	0,952 8
	100	0,130 5	- 0,282 5	0,946 2
	125	0,292 5	- 0,525 9	0,945 4
Figure O.3.2-9 Coefficient ct_b for e_b / e_s = 0,2	10	- 0,028 6	0,012 7	1,011 0
	20	0,057 2	- 0,072 3	1,028 4
	40	-0,003 4	- 0,024 5	1,017 2
	60	- 0,482 0	0,272 1	0,988 2
	80	- 0,877 9	0,516 0	0,966 0
	100	- 1,095 5	0,653 4	0,952 9
	125	- 1,348 9	0,818 2	0,938 4
Figure O.3.2-10 Coefficient cfh_s for e_b / e_s = 0,2	10	- 0,394 7	0,234 8	0,872 6
	20	- 0,519 4	0,388 9	0,843 4
	40	- 0,657 1	0,488 0	0,816 4
	60	- 0,702 1	0,453 3	0,814 1
	80	- 0,763 2	0,463 8	0,808 4
	100	- 0,809 2	0,488 2	0,799 2
	125	- 0,859 9	0,515 5	0,790 9
Figure O.3.2-11 Coefficient cfp_s for e_b / e_s = 0,2	10	- 0,604 7	0,138 3	0,887 1
	20	- 0,670 1	0,184 8	0,865 4
	40	- 0,661 8	0,113 6	0,853 5
	60	- 0,546 8	- 0,066 1	0,863 6
	80	- 0,390 0	- 0,279 0	0,877 8
	100	- 0,249 1	- 0,456 2	0,888 0
	125	- 0,128 8	- 0,610 9	0,896 6
Figure O.3.2-12 Coefficient ct_s for e_b / e_s = 0,2	10	- 0,022 1	- 0,690 7	1,067 2
	20	- 0,092 0	- 0,707 9	1,069 8
	40	0,021 4	- 0,933 2	1,090 6
	60	0,286 6	- 1,261 8	1,120 5
	80	0,539 9	- 1,560 7	1,147 6
	100	0,736 0	- 1,787 4	1,168 1
	125	0,924 8	- 2,006 3	1,187 6
Figure O.3.2-13 Coefficient cfh_b for e_b / e_s = 0,5	10	0,449 7	- 0,789 8	1,310 6
	20	0,304 5	- 0,594 6	1,112 1
	40	0,899 6	- 1,315 5	1,091 9
	60	1,617 9	- 2,143 4	1,151 3
	80	2,003 6	- 2,627 4	1,206 7
	100	1,741 7	- 2,223 8	1,015 4
	125	1,489 0	- 1,836 0	0,838 5
Figure O.3.2-14 Coefficient cfp_b for e_b / e_s = 0,5	10	1,191 3	- 1,520 5	1,282 7
	20	0,447 8	- 0,705 3	1,125 9
	40	0,153 7	- 0,446 5	1,006 5
	60	0,686 4	- 1,135 4	1,041 5
	80	1,034 0	- 1,592 6	1,077 4
	100	1,205 0	- 1,816 0	1,075 5
	125	1,002 5	- 1,510 6	0,912 5

(continued)

Table O.3.2-2 (*continued*)

	D_m / e_s	*A*	*B*	*C*
Figure O.3.2-15 Coefficient ct_b for e_b / e_s = 0,5	10	0,176 5	- 0,181 4	1,041 0
	20	- 0,439 2	0,230 8	0,999 2
	40	- 1,171 2	0,700 5	0,951 7
	60	- 1,460 5	0,859 1	0,935 6
	80	- 1,498 3	0,811 1	0,939 4
	100	- 1,438 7	0,686 9	0,950 4
	125	- 1,291 4	0,476 9	0,969 3
Figure O.3.2-16 Coefficient cfh_s for e_b / e_s = 0,5	10	- 0,371 1	0,300 2	0,875 2
	20	- 0,439 8	0,370 1	0,848 9
	40	- 0,505 4	0,386 9	0,826 7
	60	- 0,551 0	0,363 7	0,821 5
	80	- 0,609 7	0,369 9	0,818 8
	100	- 0,695 8	0,429 9	0,803 9
	125	- 0,757 6	0,461 3	0,795 3
Figure O.3.2-17 Coefficient cfp_s for e_b / e_s = 0,5	10	- 0,576 6	0,246 4	0,881 2
	20	- 0,701 2	0,306 7	0,857 3
	40	- 0,726 5	0,236 5	0,843 4
	60	- 0,639 0	0,080 8	0,849 9
	80	- 0,539 4	- 0,070 6	0,860 3
	100	- 0,494 4	- 0,149 1	0,859 1
	125	- 0,420 3	- 0,262 8	0,864 4
Figure O.3.2-18 Coefficient ct_s for e_b / e_s = 0,5	10	- 0,173 2	- 0,416 3	1,044 0
	20	- 0,185 4	- 0,530 0	1,054 5
	40	- 0,072 6	- 0,793 3	1,078 7
	60	0,097 8	- 1,033 2	1,100 4
	80	0,258 4	- 1,236 8	1,118 9
	100	0,354 7	- 1,364 2	1,130 4
	125	0,471 4	- 1,511 2	1,143 8
Figure O.3.2-19 Coefficient cfh_b for e_b / e_s = 0,8	10	- 0,053 8	- 0,363 7	1,110 3
	20	0,660 4	- 1,220 7	1,158 1
	40	1,505 3	- 2,199 9	1,183 0
	60	1,809 7	- 2,513 9	1,140 4
	80	1,485 5	- 2,005 7	0,905 5
	100	1,153 1	- 1,515 6	0,706 4
	125	0,960 4	- 1,227 3	0,579 6
Figure O.3.2-20 Coefficient cfp_b for e_b / e_s = 0,8	10	- 0,274 7	- 0,108 3	1,189 4
	20	0,027 1	- 0,452 9	1,094 8
	40	0,683 9	- 1,251 6	1,084 5
	60	1,163 5	- 1,843 1	1,127 5
	80	1,045 7	- 1,624 8	0,968 8
	100	0,768 6	- 1,201 7	0,773 8
	125	0,619 4	- 0,963 0	0,646 3
Figure O.3.2-21 Coefficient ct_b for e_b / e_s = 0,8	10	- 0,484 6	0,254 4	0,996 2
	20	- 0,986 9	0,576 5	0,966 7
	40	- 1,423 8	0,792 6	0,942 9
	60	- 1,414 9	0,671 5	0,952 7
	80	- 1,327 0	0,514 8	0,966 0
	100	- 1,173 2	0,308 4	0,984 8
	125	- 0,907 8	-0,005 9	1,013 3
Figure O.3.2-22 Coefficient cfh_s for e_b / e_s = 0,8	10	- 0,157 1	0,168 1	0,884 8
	20	- 0,256 7	0,246 9	0,861 6
	40	- 0,342 2	0,273 0	0,841 0
	60	- 0,372 6	0,233 4	0,840 7
	80	- 0,457 1	0,275 2	0,826 7
	100	- 0,554 8	0,342 4	0,811 2
	125	- 0,619 3	0,375 8	0,802 4

(*continued*)

Table O.3.2-2 (*continued*)

	D_m / e_s	A	B	C
Figure O.3.2-23 Coefficient cfp_s for e_b / e_s = 0,8	10	- 0,551 9	0,363 1	0,867 5
	20	- 0,706 6	0,419 0	0,847 6
	40	- 0,771 8	0,351 4	0,835 8
	60	- 0,695 6	0,191 0	0,845 1
	80	- 0,626 3	0,069 3	0,847 5
	100	- 0,584 9	- 0,011 0	0,845 7
	125	- 0,493 1	- 0,141 0	0,852 4
Figure O.3.2-24 Coefficient ct_s for e_b / e_s = 0,8	10	- 0,359 4	- 0,075 0	1,011 2
	20	- 0,317 4	- 0,283 7	1,031 9
	40	- 0,163 7	- 0,635 8	1,064 8
	60	- 0,008 5	- 0,878 0	1,086 9
	80	0,131 2	- 1,069 7	1,104 3
	100	0,209 7	- 1,184 4	1,114 7
	125	0,311 9	- 1,317 8	1,126 8
Figure O.3.2-25 Coefficient cfh_b for e_b / e_s = 1,0	10	0,084 8	- 0,508 8	1,024 7
	20	0,971 8	- 1,640 6	1,197 8
	40	1,684 0	- 2,472 7	1,214 2
	60	1,577 5	- 2,216 6	1,004 2
	80	1,258 6	- 1,724 3	0,783 7
	100	0,970 1	- 1,298 2	0,610 5
	125	0,858 3	- 1,080 2	0,502 8
Figure O.3.2-26 Coefficient cfp_b for e_b / e_s = 1,0	10	- 0,176 3	- 0,242 7	1,119 6
	20	0,284 1	- 0,835 0	1,133 0
	40	0,857 0	- 1,520 4	1,113 6
	60	1,052 5	- 1,708 0	1,041 3
	80	0,838 8	- 1,349 7	0,845 3
	100	0,597 4	- 0,977 4	0,672 4
	125	0,564 4	- 0,826 2	0,564 4
Figure O.3.2-27 Coefficient ct_b for e_b / e_s = 1,0	10	- 0,815 5	0,519 2	0,974 6
	20	- 1,117 3	0,650 8	0,958 9
	40	- 1,376 7	0,699 9	0,950 3
	60	- 1,339 9	0,556 8	0,962 7
	80	- 1,187 6	0,337 1	0,982 0
	100	- 0,968 1	0,065 8	1,006 9
	125	- 0,613 3	- 0,308 8	1,040 8
Figure O.3.2-28 Coefficient cfh_s for e_b / e_s = 1,0	10	- 0,029 2	0,058 5	0,900 5
	20	- 0,147 6	0,163 5	0,875 1
	40	- 0,234 1	0,198 9	0,850 7
	60	- 0,264 9	0,165 8	0,846 3
	80	- 0,353 6	0,212 1	0,831 9
	100	- 0,454 9	0,281 5	0,816 3
	125	- 0,421 4	0,256 4	0,812 4
Figure O.3.2-29 Coefficient cfp_s for e_b / e_s = 1,0	10	- 0,453 8	0,351 2	0,884 2
	20	- 0,662 3	0,445 8	0,850 4
	40	- 0,777 5	0,409 1	0,832 7
	60	- 0,735 5	0,275 8	0,837 5
	80	- 0,677 3	0,160 6	0,838 7
	100	- 0,641 9	0,083 0	0,836 7
	125	- 0,388 0	- 0,147 6	0,851 7
Figure O.3.2-30 Coefficient ct_s for e_b / e_s = 1,0	10	- 0,472 0	0,161 2	0,989 3
	20	- 0,399 3	- 0,110 8	1,015 7
	40	- 0,218 5	- 0,525 6	1,054 8
	60	- 0,063 8	- 0,782 5	1,078 4
	80	0,068 2	- 0,974 6	1,095 9
	100	0,145 9	- 1,093 4	1,106 7
	125	0,299 9	- 1,258 7	1,121 4

(*continued*)

Table O.3.2-2 (*continued*)

	D_m / e_s	A	B	C
Figure O.3.2-31 Coefficient cfh_b for e_b / e_s = 1,2	10	0,344 1	- 0,804 4	1,022 6
	20	1,220 8	- 1,961 3	1,232 8
	40	1,749 8	- 2,566 4	1,208 6
	60	1,442 9	- 2,032 3	0,918 5
	80	1,097 0	- 1,505 2	0,691 7
	100	0,874 6	- 1,176 8	0,553 4
	125	0,721 4	- 0,946 7	0,452 2
Figure O.3.2-32 Coefficient cfp_b for e_b / e_s = 1,2	10	- 0,093 8	- 0,340 1	1,061 1
	20	0,520 8	- 1,156 3	1,169 4
	40	0,981 1	- 1,704 8	1,134 4
	60	0,930 9	-1,547 1	0,962 3
	80	0,641 6	-1,084 1	0,739 5
	100	0,481 9	- 0,830 6	0,609 3
	125	0,367 5	- 0,641 5	0,505 3
Figure O.3.2-33 Coefficient ct_b for e_b / e_s = 1,2	10	-0,835 3	0,501 2	0,974 7
	20	- 1,126 2	0,635 1	0,959 3
	40	- 1,290 8	0,591 3	0,960 0
	60	- 1,166 3	0,358 2	0,980 5
	80	- 0,888 5	0,007 8	1,012 2
	100	- 0,662 2	- 0,269 6	1,037 9
	125	- 0,400 7	- 0,575 9	1,066 2
Figure O.3.2-34 Coefficient cfh_s for e_b / e_s = 1,2	10	0,055 5	- 0,082 9	0,913 4
	20	- 0,046 9	0,055 8	0,884 7
	40	- 0,139 1	0,139 7	0,855 7
	60	- 0,168 6	0,109 3	0,850 2
	80	- 0,256 1	0,154 7	0,836 1
	100	- 0,357 4	0,222 7	0,821 2
	125	- 0,433 9	0,263 9	0,811 7
Figure O.3.2-35 Coefficient cfp_s for e_b / e_s = 1,2	10	- 0,133 5	0,119 2	0,900 9
	20	- 0,486 9	0,348 1	0,858 4
	40	- 0,772 7	0,464 2	0,827 0
	60	- 0,764 4	0,354 5	0,829 8
	80	- 0,724 4	0,251 6	0,829 5
	100	- 0,692 7	0,172 2	0,828 8
	125	- 0,613 1	0,045 7	0,834 4
Figure O.3.2-36 Coefficient ct_s for e_b / e_s = 1,2	10	- 0,529 7	0,362 5	0,970 4
	20	- 0,445 6	0,043 9	1,000 9
	40	- 0,265 4	- 0,411 6	1,044 2
	60	- 0,111 5	- 0,686 4	1,069 7
	80	0,017 5	- 0,885 9	1,088 0
	100	0,094 8	- 1,010 7	1,099 4
	125	0,186 7	- 1,142 5	1,111 4
Figure O.3.2-37 Coefficient cfh_b for e_b / e_s = 1,5	10	0,675 6	- 1,300 2	1,147 0
	20	1,354 7	- 2,162 1	1,231 5
	40	1,687 4	- 2,475 2	1,146 7
	60	1,373 6	- 1,934 2	0,862 6
	80	1,014 6	- 1,390 9	0,634 5
	100	0,828 4	- 1,116 7	0,516 3
	125	0,702 3	- 0,891 4	0,412 4
Figure O.3.2-38 Coefficient cfp_b for e_b / e_s = 1,5	10	- 0,033 2	- 0,439 6	1,052 4
	20	0,696 3	- 1,412 8	1,205 8
	40	1,136 9	- 1,913 3	1,153 1
	60	0,899 0	- 1,490 3	0,905 4
	80	0,587 1	- 1,005 6	0,685 6
	100	0,422 0	- 0,753 1	0,563 5
	125	0,366 6	- 0,588 1	0,461 3

(continued)

Table O.3.2-2 (*concluded*)

	D_m / e_s	*A*	*B*	*C*
Figure O.3.2-39 Coefficient ct_b for e_b / e_s = 1,5	10	- 0,685 2	0,336 7	0,986 0
	20	- 1,039 4	0,530 7	0,967 2
	40	- 1,052 0	0,333 0	0,982 9
	60	- 0,657 7	- 0,182 4	1,029 9
	80	- 0,369 9	- 0,545 3	1,063 6
	100	- 0,182 8	- 0,782 2	1,086 2
	125	0,078 2	- 1,065 2	1,112 7
Figure O.3.2-40 Coefficient cfh_s for e_b / e_s = 1,5	10	0,007 0	- 0,068 2	0,940 3
	20	- 0,070 0	0,060 5	0,896 5
	40	- 0,076 3	0,099 6	0,858 9
	60	- 0,026 9	0,013 8	0,862 2
	80	- 0,116 6	0,070 4	0,843 1
	100	- 0,211 9	0,132 8	0,829 5
	125	- 0,210 5	0,129 7	0,822 9
Figure O.3.2-41 Coefficient cfp_s for e_b / e_s = 1,5	10	0,529 5	- 0,433 8	0,966 5
	20	- 0,150 4	0,108 8	0,889 6
	40	- 0,725 5	0,509 4	0,821 9
	60	- 0,770 3	0,435 4	0,824 1
	80	- 0,771 6	0,368 2	0,818 1
	100	- 0,754 0	0,295 6	0,817 6
	125	- 0,523 5	0,075 7	0,831 1
Figure O.3.2-42 Coefficient ct_s for e_b / e_s = 1,5	10	- 0,492 6	0,563 4	0,950 2
	20	- 0,476 6	0,253 8	0,980 5
	40	- 0,335 0	- 0,224 1	1,026 5
	60	- 0,174 6	- 0,535 6	1,055 7
	80	- 0,049 6	- 0,750 1	1,075 6
	100	0,025 6	- 0,884 5	1,088 1
	125	0,184 7	- 1,066 1	1,104 3

NOTE The curves of Graphs O.3.2-7 to O.3.2-42 can be described by the following polynomial equation:

$$c = MIN[(Ax^2 + Bx + C) ; 1] \qquad \text{(T.O.3.3-2.2)}$$

where

$$x = d_m / D_m$$

Coefficients A, B and C are given for each curve of each graph in function of the ratio D_m/e_s.

Annex P
(informative)

Bolted flange connections — Application of EN 1591

P.1 Introduction

According to EN 13480-3, two methods may be used to check bolted connections:

- the Taylor Forge method and
- the procedure detailed in EN 1591-1 and EN 1591-2.

However, the proper application of this European Standard to bolted connections in the field of piping requires additional explanations.

The following two parts of European Standard EN 1591, based on German developments, define an analytical procedure for the design of bolted flange connections with gasket:

- EN 1591-1, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*;
- EN 1591-2, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 2: Gasket parameters*.

This procedure allows the verification of the connection taking account of strength criteria and tightness criteria.

The parameters taken into account are as follows:

- fluid pressure;
- mechanical strength of flange, bolting and gasket;
- gasket coefficients;
- bolt nominal loads

and, other than the Taylor-Forge method (see 6.6), the following additional factors:

- operating conditions and specifically creep/relaxation behaviour;
- dispersions due to initial tightening where relevant;
- variations of gasket loading due to the deformation of the different components of the connection;
- effects of the connected shell or piping;
- effects of external axial forces and moments;
- effects of temperature difference between bolts and flanges.

P.2 Scope

P.2.1 General

This procedure shall apply to the following arrangements:

- two circular flanges (identical or different);
- four identical bolts, as a minimum, regularly spaced;
- a circular gasket entirely within the circle enclosed by the bolt holes.

The procedure does not apply to metal-metal connections.

P.2.2 Materials

Bolt and flange materials shall conform to the requirements of EN 13480-2 regarding ductility properties. Where these requirements are not fulfilled, lower nominal design stress shall be used.

P.2.3 Loadings

The following loadings are taken into account in this procedure:

- internal and external fluid pressure;
- external loads: axial forces and bending moments (equivalent axial load);
- thermal expansion of flanges, bolts and gasket.

P.2.4 Assumptions

P.2.4.1 The deformations of the cross-section of the plate are not taken into account. Only the rotation of the cross-section is considered.

P.2.4.2 The plate of an integral flange is connected to a cylindrical shell or to an equivalent cylindrical shell (conical or spherical shell).

P.2.4.3 The effective width b_{Ge} of contact between the gasket and the flanges may be less than the actual width of the gasket. This effective width shall be calculated for seating condition and considered as constant for all other conditions.

P.2.4.4 The modulus of elasticity E_G of the gasket is a function of the applied compressive stress.

P.2.4.5 Creep behaviour of the gasket is taken into account using the factor P_{QR}.

P.2.4.6 The thermal and mechanical deformations of flanges, bolts and gasket are considered.

P.2.4.7 External moments are taken into account as equivalent axial bolt loads.

P.2.4.8 Transitions between a condition to another lead to variations of bolt and gasket loads.

P.2.4.9 Acceptance of component loadings is based on limit analysis which covers failure by gross plastic deformation.

P.2.4.10 The following is not taken into account or covered by the procedure:

- bending stiffness of bolts;
- creep of flanges and bolts except through nominal design stress and thermal expansion factors;
- external torsion moments and external shear loads.

P.3 Application of EN 1591

P.3.1 Calculations

The minimum tightening load for the required tightening of bolts shall be calculated by successive iterations.

Internal loads due to initial tightening shall be calculated for each condition (initial tightening, proof test condition and operating conditions) and combined with external loads.

Safety factors shall be those defined by Clause 5. However for seating condition, factor for strength test condition shall apply.

P.3.2 Gasket coefficients

The recommended gaskets for industrial piping are given in Table P.1.

NOTE 1 For more information the gasket manufacturer should be contacted.

NOTE 2 Legend of tables:

- NA: not applicable;
- ND: not defined.

Gasket reference: Example: 1-09-101-1:

- 1-09 = see Table P.2;
- 101-1 = joint origin (manufacturer or other).

P.3.2.1 Gasket maximum allowable stress Q_{smax}.

The coefficients determined according to EN 13555 are given in Table P.2 to Table P.29 (room temperature and operating temperature).

In these tables:

- P is the test pressure in the sample;
- S_{ai} is the gasket pressure.

P.3.2.2 Minimum stress Q_{minL} to be applied at room temperature (seating condition) in order to fulfil the requirements regarding leak tightness class for the fluid under consideration.

The values determined according to EN 13555 (Helium tightness test at room temperature) are given in Table P.2 to Table P.29.

NOTE In Tables P.3 to P.29, Q_{minL} is given in MPa.

P.3.2.3 Minimum stress Q_{sminL} to be applied at room temperature (operating conditions) in order to fulfil the requirements regarding leak tightness class for the fluid under consideration.

The values determined according to EN 13555 (Helium tightness test at room temperature) are given in Table P.2 to Table P.29.

NOTE In Tables P.3 to P.29, Q_{sminL} is given in MPa.

P.3.2.4 Modulus of elasticity

E_G is the modulus of elasticity when compression is released and for a maximal stress equal to Q_0.

P.3.2.5 Creep/relaxation parameter

The parameter P_{QR} is given in Table P.2 to Table P.29. This parameter is used instead of the previous creep factor called g_c where permitted by the next revision of EN 1591-1.

P.3.3 Tightening

The initial tightening shall be greater than the minimum tightening required at room temperature to comply to the requirements of the tightness class for the fluid and pressure considered.

However this tightening shall not lead to a gasket stress greater than the allowable value at room temperature.

In addition, calculations shall take into account the tolerances on tightening due to tightening procedure and the used equipment.

Table P.1 — Recommended gaskets for industrial piping

Gasket type	EN 1514 c (PN flanges)	EN 12560 (CLASS Flanges)	Chemical compatibility	Maximum temperature	Maximum internal fluid pressure (bar)	PN max (EN 1514)	CLASS max (EN 12560)	Surface finish (Ra)
Fibre	1514-1	12560-1	All fluids (to be used carefully for steam: risk of hydrolysis)	250 °C	50	63	900	3,2 µm to 12,5 µm
Graphite	1514-1	12560-1	Risk of oxidation	350 °C (in oxidant environment) 550 °C with inhibitor or non oxidant environment	50	63	900	3,2 µm to 12,5 µm
PTFE	1514-1	12560-1	All fluids	Pure PTFE: 120 °C Modified PTFE 225 °C	50	63	900	3,2 µm to 12,5 µm
Spiral wound gasket	1514-2	12560-2	Depending on insert, spiral and ring materials (risk of oxidation with graphite insert)	> 600 °C (vermiculite insert) 600 °C (graphite insert) 250 °C (PTFE insert)	400	100 (with internal ring	900 (without internal ring) 2500 (with internal ring)	12,5 µm for P < 12 bar 6,4 µm for P > 20 bar 3,2 µm for hard conditions 1,6 µm for vacuum
Kammprofil	1514-6	12560-6	Depending on internal and covering materials (risk of oxidation with graphite covering)	600 °C (may be limited to 260 °C, by a PTFE covering for example)	400	100	2500	3,2 µm to 6,4 µm 1,6 µm for vacuum
Metal jacketed	1514-4 (Metal jacketed) 1514-7 (Covered metal jacketed)	12560-4 (Metal jacketed) 12560-7 (Covered metal jacketed)	Most of industrial fluids	Depending on covering	> 400	100	2500	0,8 µm to 3,2 µm (not covered) 1,6 µm to 12,5 µm (covered)
Solid metal	1514-4	12560-5 (RTJ) 12560-4 (Flat metallic gaskets)	Depending on material	High	500	100	2500	1,6 µm
Ring joint			Depending on material	Depending on material	> 400			1,6 µm
Expanded graphite			All fluids	600 °C	500			1,6 to 6,4 µm

Table P.2 — Gasket type code

Classification from EN 1514	Gasket type	Gasket family (EN 1591-2)	EN 1591-2 Table	Table of this European Standard
EN 1514-1	Modified PTFE	Non metallic flat gasket	1	P.5 and P.6
EN 1514-1	Non-asbestos fibre (aramid/glass)	Non metallic flat gasket	1	P.3
EN 1514-1	Expanded graphite with perforated metal insertion	Non metallic flat gasket	1	P.4
EN 1514-2	Standard spiral wound gasket with external ring	Spiral wound gasket	3	P.9
EN 1514-2	Standard spiral wound gasket with internal and external rings	Spiral wound gasket	3	P.8
EN 1514-2	Low stress spiral wound gasket with internal and external rings	Spiral wound gasket	3	P.7
EN 1514-4	Metal jacketed with graphite filler and stainless steel shell	Metal jacketed gasket	6	P.12
EN 1514-4	Corrugated inlaid gasket (graphite/stainless steel)	Non metallic flat gasket	7	P.11
EN 1514-6	Kammprofile gasket for use with steel flanges (graphite/stainless steel)	Grooved steel gasket with soft layers on both sides	2	P.10
EN 1514-7	Covered metal jacketed (graphite/graphite/stainless steel)	Covered metal jacketed gasket	5	P.13
EN 1514-1	Expanded graphite with metallic sheets laminated in thin layers withstanding high stresses	Flat gasket with metal insertion	1	P.14, P.27
EN 1514-1	Modified PTFE sheet material	Non metallic flat gasket	1	P.15, P.17, P.23
EN 1514-1	Non-asbestos fibre with binder $e_G \geq 1$ mm	Non metallic flat gasket	1	P.16, P.18, P.19, P.22
EN 1514-1	Expanded graphite with adhesive perforated metal insertion	Flat gasket with metal insertion	1	P.20
EN 1514-2	PTFE Standard spiral wound gasket with inner and outer support ring	Spiral wound gasket	3	P.24
EN 1514-3	PTFE envelope gasket	PTFE envelope gasket	1	P.28
EN 1514-4	Metal jacketed gasket with graphite	Metal jacketed gasket	6	P.26
EN 1514-4	Corrugated gasket with graphite	Corrugated gasket	7	P.29
EN 1514-4	Expanded graphite with metallic sheets laminated in thin layers withstanding high stresses	Flat gasket with metal insertion	1	P.14, P.27
EN 1514-6	Kammprofile gasket with bonded graphite layers	Grooved steel gasket with soft layers on both sides	2	P.21
EN 1514-7	Covered metal jacketed gasket with graphite (outer ring)	Covered metal jacketed gasket	5	P.25

Table P.3 — Gasket 1-09-101-1 – Non-asbestos fibre (aramid/glass) $e_G \geq 1$ mm

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	150 MPa
	200 °C	60 MPa
	250 °C	50 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 150 MPa Average for g_c: 0,72
	200 °C	Initial load: 60 MPa Average for g_c: 0,29
	250 °C	Initial load: 50 MPa Average for g_c: 0,28
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 150 MPa Average for P_{QR}: 0,985
	200 °C	Initial load: 60 MPa Average for P_{QR}: 0,805
	250 °C	Initial load: 50 MPa Average for P_{QR}: 0,775

Q_0	E_G		
	Room	**200 °C**	**250 °C**
20 MPa	979 MPa	4 898 MPa	3 731 MPa
30 MPa	1 414 MPa		
40 MPa	2 153 MPa	4 990 MPa	4 159 MPa
50 MPa	2 972 MPa		
60 MPa	4 182 MPa	6 023 MPa	4 024 MPa
80 MPa	8 412 MPa		
100 MPa	15 159 MPa		
120 MPa	26 392 MPa		
140 MPa	40 379 MPa		

Table P.3 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tighness Class	Q_{minL}	S_{a1}=320 Mpa
		$Q_{sminL,Sa1}$
High tightness	15,5	10
Very high tightness	60	10

P = 40 bar – Average of "full tests" values							
Tighness Class	Q_{minL}	S_{a1} = 20 MPa	S_{a2} = 40 MPa	S_{a3} = 60 MPa	S_{a4} = 80 MPa	S_{a5} = 105 MPa	S_{a6} = 160 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	11,6	10	10	10	10	10	10
High tightness	34,5	NA	10	10	10	10	10
Very high tightness	81	NA	NA	NA	40	33	17

P = 80 bar – Average of "full test" and "simplified test" values							
Tighness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2} = 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	10	10	10	10	10	10	10
High tightness	43	NA	10	10	10	10	10
Very high tightness	95	NA	NA	NA	NA	38	19,4

Table P.4 — Gasket 1-05-101-1 – Expanded graphite with perforated metal insertion

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	200 MPa
	200 °C	150 MPa
	300 °C	140 MPa
	450 °C	120 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 200 MPa Average for g_c: 0,98
	300 °C	Initial load: 140 MPa Average for g_c: 0,12
	450 °C	Initial load: 120 MPa Average for g_c: 0,08
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 200 MPa Average for P_{QR}: 1
	300 °C	Initial load: 140 MPa Average for P_{QR}: 0,775
	450 °C	Initial load: 120 MPa Average for P_{QR}: 0,62

Q_0	E_G			
	Room	**200 °C**	**300 °C**	**450 °C**
20 MPa	198 MPa	591 MPa	416 MPa	943 MPa
30 MPa	397 MPa			
40 MPa	675 MPa	1 579 MPa	1 396 MPa	2 482 MPa
50 MPa	1 043 MPa			
60 MPa	1 536 MPa	2 493 MPa	2 423 MPa	3 833 MPa
80 MPa	2 804 MPa	3 437 MPa	3 828 MPa	4 706 MPa
100 MPa	4 738 MPa	4 258 MPa	5 542 MPa	4 799 MPa
120 MPa	7 083 MPa	4 871 MPa	5 968 MPa	
140 MPa	10 447 MPa			
160 MPa	13 992 MPa			

Table P.4 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – Average of two "simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=140 MPa
		$Q_{sminL,Sa1}$
High tightness	23	10
Very high tightness	93	16

P = 40 bar – Average of "full tests" values							
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	10	10	10	10	10	10	10
High tightness	41	NA	10	10	10	10	10
Very high tightness	139	NA	NA	NA	NA	NA	95

P = 80 bar –"Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=140 MPa
		$Q_{sminL,Sa1}$
High tightness	60	10

Table P.5 — Gasket 1-10-100-1 – Modified PTFE

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	50 MPa
	175 °C	40 MPa
	225 °C	25 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 50 MPa Average for g_c: 0,31
	175 °C	Initial load: 40 MPa Average for g_c : 0,06
	225 °C	Initial load: 25 MPa Average for g_c : 0,08
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 50 MPa Average for P_{QR}: 0,84
	175 °C	Initial load: 40 MPa Average for P_{QR}: 0,41
	225 °C	Initial load: 25 MPa Average for P_{QR}: 0,365

Q_0	E_G		
	Room	**175 °C**	**225 °C**
20 MPa	2 170 MPa	826 MPa	614 MPa
30 MPa	2 986 MPa		
40 MPa	8 625 MPa	1 254 MPa	809 MPa
60 MPa		1 335 MPa	864 MPa

Table P.5 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}=20 MPa	S_{a2}=40 MPa	S_{a3}=60 MPa	S_{a4}=80 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
High tightness	10	10	10	10	10
Very high tightness	17	10	10	10	10

P = 40 bar – Average of the four tests values or choice among the four values					
Tightness Class	Q_{minL}	S_{a1}=20 MPa	S_{a2}=40 MPa	S_{a3}=60 MPa	S_{a4}=80 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
Normal	10	10	10	10	10
High tightness	12	10	10	10	10
Very high tightness	42	NA	38	10	10

P = 80 bar – Test values		
Tightness Class	Q_{minL}	S_{a1}=80 MPa
		$Q_{sminL,Sa1}$
High tightness	22	10
Very high tightness	37	10

Table P.6 — Gasket 1-10-102-1 – Modified PTFE

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	60 MPa
	175 °C	60 MPa
	225 °C	60 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 60 MPa Average for g_c : 0,44
	175 °C	Initial load: 60 MPa Average for g_c : 0,09
	225 °C	Initial load: 60 MPa Average for g_c : 0,06
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 60 MPa Average for P_{QR}: 0,895
	175 °C	Initial load: 60 MPa Average for P_{QR}: 0,5
	225 °C	Initial load: 60 MPa Average for P_{QR}: 0,42

Q_0	E_G		
	Room	**175 °C**	**225 °C**
20 MPa	1 924 MPa	1 164 MPa	1 263 MPa
30 MPa	2 587 MPa		1 569 MPa
40 MPa	3 894 MPa	1 682 MPa	2 178 MPa
50 MPa	6 378 MPa		2 553 MPa
60 MPa	9 750 MPa	2 217 MPa	3 170 MPa

Table P.6 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}=20 MPa	S_{a2}=40 MPa	S_{a3}=60 MPa	S_{a4}=80 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
Normal	10	10	10	10	10
High tightness	17,3	11	10	10	10
Very high tightness	38,3	NA	17,2	10	10

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}=20 MPa	S_{a2}=40 MPa	S_{a3}=60 MPa	S_{a4}=80 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
High tightness	16,4	10	10	10	10
Very high tightness	31	NA	10	10	10

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=80 Mpa
		$Q_{sminL,Sa1}$
High tightness	26,2	10
Very high tightness	39,3	10

Table P.7 — Gasket 3-05-102-1 – Low stress spiral wound modified gasket with internal and external rings

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	126 MPa
	300 °C	126 MPa
	450 °C	126 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 60 MPa Average for g_c: 0,98
	300 °C	Initial load: 60 MPa Procedure not suitable for this type of gasket at this temperature
	450 °C	Initial load: 60 MPa Procedure not suitable for this type of gasket at this temperature
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 60 MPa Average for P_{QR}: 0,995
	300 °C	-
	450 °C	-

Q_0	E_G			
	Room	**200 °C**	**300 °C**	**450 °C**
20 MPa	725 MPa	843 MPa	942 MPa	850 MPa
30 MPa	996 MPa			
40 MPa	1 207 MPa	1 809 MPa	1 988 MPa	2 259 MPa
50 MPa	1 703 MPa			
60 MPa	2 268 MPa	4 211 MPa	3 776 MPa	3 840 MPa
80 MPa		8 537 MPa	6 992 MPa	4 945 MPa

Table P.7 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=80 MPa
		$Q_{sminL,Sa1}$
Normal	10	10
High tightness	10	10

P = 40 bar – Average of "full tests" values								
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa	S_{a7}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$	$Q_{sminL,Sa7}$
High tightness	19	10	10	10	10	10	10	10
Very high tightness	140	NA	NA	NA	NA	NA	82,5	87,5

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=320 MPa
		$Q_{sminL,Sa1}$
Normal	10	10
High tightness	32,5	10
Very high tightness	230,6	229,6

Table P.8 — Gasket 3-04-104-1 – Standard spiral wound gasket with internal and external rings

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	138 MPa
	300 °C	250 MPa
	450 °C	220 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 300 MPa Average for g_c: 0,92
	300°C	Initial load: 250 MPa Average for g_c: 0,52
	450°C	Initial load: 220 MPa Average for g_c: 0,54
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 300 MPa Average for P_{QR}: 1
	300 °C	Initial load: 250 MPa Average for P_{QR}: 0,94
	450 °C	Initial load: 220 MPa Average for P_{QR}: 0,92

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	1 233 MPa	1 423 MPa	1 489 MPa
30 MPa	1 620 MPa		
40 MPa	1 916 MPa	2 790 MPa	3 013 MPa
50 MPa	2 316 MPa	3 997 MPa	
60 MPa	2 719 MPa	4 203 MPa	4 739 MPa
80 MPa	3 372 MPa	4 291 MPa	6 156 MPa
99 MPa	3 987 MPa	5 205 MPa	7 428 MPa
120 MPa	4 793 MPa	6 111 MPa	8 525 MPa
140 MPa	5 808 MPa	6 972 MPa	9 297 MPa
160 MPa	7 024 MPa	7 938 MPa	10 206 MPa
180 MPa	8 520 MPa	9 661 MPa	10 968 MPa
200 MPa		9 865 MPa	11 608 MPa
220 MPa	12 783 MPa	10 761 MPa	12 141 MPa
240 MPa	15 577 MPa	11 638 MPa	
260 MPa	19 291 MPa		
280 MPa	24 069 MPa		
300 MPa	30 036 MPa		

Table P.8 (*concluded*)

Class of tightness from tightness tests

P = 40 bar – Average of "full tests" values							
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 105 MPa	S_{a5}= 160 MPa	S_{a6}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	10	NA	10	10	10	10	10
High tightness	25	10	10	10	10	10	10
Very high tightness	81	NA	NA	NA	62,4	35	26

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=320 MPa
		$Q_{sminL,Sa1}$
Normal	10	10
High tightness	48	10
Very high tightness	143,2	51

P = 160 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
Normal	10	10
High tightness	44,6	10
Very high tightness	158,9	157,3

Table P.9 — Gasket 3-03-100-1 – Standard spiral wound modified gasket with external ring

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	125 MPa
	300 °C	125 MPa
	450 °C	125 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 125 MPa Average for g_c: 0,95
	300 °C	Initial load: 125 MPa Procedure not suitable for this type of gasket at this temperature
	450 °C	Initial load: 125 MPa Procedure not suitable for this type of gasket at this temperature
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 125 MPa Average for P_{QR}: 0,99
	300 °C	Initial load: 125 MPa
	450 °C	Initial load: 125 MPa

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	1 854 MPa	2 904 MPa	2 299 MPa
30 MPa	1 975 MPa		
40 MPa	2 158 MPa	3 359 MPa	4 094 MPa
49 MPa	2 563 MPa		
59 MPa	2 892 MPa	4 694 MPa	6 081 MPa
79 MPa	3 643 MPa	6 874 MPa	7 835 MPa
99 MPa	4 714 MPa	10 291 MPa	9 943 MPa
120 MPa	6 147 MPa	15 117 MPa	11 529 MPa

Table P.9 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
Normal	10	10
High tightness	63,4	10
Very high tightness	98	24

P = 40 bar – Average of "full tests" values							
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	30	NA	10	10	10	10	10
High tightness	62	NA	NA	NA	19	10	10
Very high tightness	126	NA	NA	NA	NA	NA	71

P = 80 bar – Average of "full test" and "simplified test" values						
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 105 MPa	S_{a5}= 160 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$
Normal	34,5	10	10	10	10	11
High tightness	71	NA	NA	25,2	17	15,5
Very high tightness	104,4	NA	NA	NA	100	40

Table P.10 — Gasket 2-05-104-1 – Kammprofile gasket for use with steel flanges (graphite/stainless steel)

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	600 MPa
	300 °C	450 MPa
	450 °C	400 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 600 MPa Average for g_c: 0,62
	300 °C	Initial load: 450 MPa Average for g_c: 0,20
	450 °C	Initial load: 400 MPa Average for g_c: 0,03
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 600 MPa Average for P_{QR}: 1
	300 °C	Initial load: 450 MPa Average for P_{QR}: 0,94
	450 °C	Initial load: 400 MPa Average for P_{QR}: 0,8

Table P.10 (*continued*)

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	3 273 MPa	13 379 MPa	12 923 MPa
30 MPa	3 598 MPa		
40 MPa	4 369 MPa	19 157 MPa	20 649 MPa
50 MPa	5 722 MPa		
60 MPa	7 391 MPa	30 932 MPa	58 406 MPa
80 MPa	12 085 MPa	52 885 MPa	73 918 MPa
100 MPa	16 774 MPa		68 786 MPa
119 MPa	22 854 MPa		141 110 MPa
140 MPa	32 441 MPa		
160 MPa	35 528 MPa		
180 MPa	38 537 MPa		
220 MPa	43 875 MPa		
240 MPa	45 988 MPa		
260 MPa	45 757 MPa		
280 MPa	46 222 MPa		
300 MPa	46 530 MPa		
320 MPa	46 662 MPa		
340 MPa	45 542 MPa		
360 MPa	46 350 MPa		
380 MPa	45 590 MPa		
400 MPa	44 702 MPa		

Table P.10 (*concluded*)

Class of tightness from tightness tests

P = 40 bar – Average of "full tests" values								
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa	S_{a7}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$	$Q_{sminL,Sa7}$
High tightness	11,7	12	16,5	18	14	13	16,9	10
Very high tightness	47	NA	NA	46	30,5	31	34	32

P = 80 bar – "Full test" values								
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa	S_{a7}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$	$Q_{sminL,Sa7}$
Normal	10	10	10	10	10	10	10	10
High tightness	44,7	NA	NA	31,6	24,3	22,8	10	20
Very high tightness	59,7	NA	NA	NA	50	46,1	42,2	54,9

P = 160 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
Normal	37,4	10
High tightness	62,1	42,8
Very high tightness	76	81,7

Table P.11 — Gasket 7-01-104-1 – Corrugated inlaid gasket (graphite/stainless steel)

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	400 MPa
	300 °C	200 MPa
	450 °C	180 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 400 MPa Average for g_c: 0,89
	300 °C	Initial load: 200 MPa Average for g_c: 0,05
	450 °C	Initial load: 180 MPa Average for g_c: 0,04
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 400 MPa Average for P_{QR}: 1
	300 °C	Initial load: 200 MPa Average for P_{QR}: 0,72
	450 °C	Initial load: 180 MPa Average for P_{QR}: 0,525

Table P.11 (*continued*)

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	1 498 MPa	3 559 MPa	2 933 MPa
30 MPa	1 822 MPa		
40 MPa	2 134 MPa	4 518 MPa	4 903 MPa
50 MPa	2 221 MPa		
60 MPa	1 968 MPa	4 823 MPa	5 113 MPa
80 MPa	2 824 MPa	6 942 MPa	5 530 MPa
100 MPa	3 968 MPa	7 662 MPa	5 528 MPa
119 MPa	5 185 MPa	7 821 MPa	5 394 MPa
140 MPa	6 804 MPa	7 812 MPa	5 302 MPa
160 MPa	8 046 MPa	7 388 MPa	5 061 MPa
180 MPa	9 489 MPa	7 292 MPa	4 968 MPa
220 MPa	11 783 MPa		
240 MPa	12 563 MPa		
260 MPa	13 503 MPa		
280 MPa	14 674 MPa		
300 MPa	15 757 MPa		
320 MPa	16 802 MPa		
340 MPa	17 370 MPa		
360 MPa	18 350 MPa		
380 MPa	18 806 MPa		
400 MPa	19 316 MPa		

Table P.11 (*concluded*)

Class of tightness from tightness tests

P = 40 bar – Average of "full tests" values								
Tightness Class	Q_{minL}	S_{a1}=20 MPa	S_{a2}=40 MPa	S_{a3}=60 MPa	S_{a4}=80 MPa	S_{a5}=105 MPa	S_{a6}=160 MPa	S_{a7}=320 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$	$Q_{sminL,Sa7}$
High tightness	11,2	10	10	10	10	10	10	17
Very high tightness	20,9	NA	33	13	17	27	23,8	31,6

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=320 Mpa
		$Q_{sminL,Sa1}$
Very high tightness	74,9	10

P = 160 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
Very high tightness	30	20

Table P.12 — Gasket 6-04-103-1 – Metal jacketed with graphite filler and stainless steel shell

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	400 MPa
	300 °C	400 MPa
	450 °C	400 MPa
g_{c} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 400 MPa Average for g_{c}: 0,77
	300 °C	Initial load: 400 MPa Average for g_{c}: 0,07
	450 °C	Initial load: 400 MPa Average for g_{c}: 0,03
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 400 MPa Average for P_{QR}: 1
	300 °C	Initial load: 400 MPa Average for P_{QR}: 0,93
	450 °C	Initial load: 400 MPa Average for P_{QR}: 0,865

Table P.12 (*continued*)

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	696 MPa	1 004 MPa	1 033 MPa
30 MPa	1 126 MPa		
40 MPa	1 718 MPa	2 120 MPa	2 434 MPa
50 MPa	2 435 MPa		
60 MPa	3 334 MPa	3 402 MPa	3 845 MPa
80 MPa	5 787 MPa	4 521 MPa	5 021 MPa
100 MPa	9 029 MPa	5 405 MPa	5 977 MPa
119 MPa	13 855 MPa	6 296 MPa	6 513 MPa
140 MPa	19 811 MPa	7 048 MPa	7 108 MPa
160 MPa	28 779 MPa	7 886 MPa	7 252 MPa
180 MPa	40 961 MPa	8 547 MPa	7 682 MPa
200 MPa		9 349 MPa	7 888 MPa
220 MPa		10 041 MPa	8 305 MPa
240 MPa		10 584 MPa	8 677 MPa
260 MPa		11 219 MPa	8 833 MPa
280 MPa		12 059 MPa	9 322 MPa
300 MPa		12 474 MPa	9 610 MPa
320 MPa		13 783 MPa	9 856 MPa
340 MPa		14 286 MPa	10 162 MPa
360 MPa		15 491 MPa	10 808 MPa
380 MPa		17 411 MPa	10 726 MPa
400 MPa		17 061 MPa	11 381 MPa

Table P.12 (*concluded*)

Class of tightness from tightness tests

P = 40 bar – Average of "full tests" values							
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 100 MPa	S_{a5}= 160 MPa	S_{a6}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$
Normal	54,7	36,4	26	23,4	21,3	10	10
High tightness	171	NA	NA	NA	NA	NA	34,8

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=320 Mpa
		$Q_{sminL,Sa1}$
Normal	62,6	20
High tightness	287,4	40,3

P = 160 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
Normal	120	NA

Table P.13 — Gasket 5-05-103-1 – Covered metal jacketed (graphite/graphite/stainless steel)

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	171 MPa
	300 °C	171 MPa
	450 °C	171 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 47 MPa Average for g_C: 0,92
		Initial load: 80 MPa Average for g_C: 0,95
	300 °C	Initial load: 47 MPa Not adapted procedure for this type of gasket at this temperature
		Initial load: 80 MPa Not adapted procedure for this type of gasket at this temperature
	450 °C	Initial load: 80 MPa Not adapted procedure for this type of gasket at this temperature
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Average for P_{QR}: 0,985

Q_0	E_G		
	Room	**300 °C**	**450 °C**
20 MPa	534 MPa	680 MPa	253 MPa
30 MPa	963 MPa		
40 MPa	1 243 MPa	1 637 MPa	758 MPa
50 MPa	1 629 MPa		
60 MPa	2 038 MPa	3 002 MPa	1 625 MPa
80 MPa	3 395 MPa	5 831 MPa	2 763 MPa
100 MPa	5 381 MPa	11 047 MPa	4 563 MPa

Table P.13 (*concluded*)

Class of tightness from tightness tests

P = 40 bar – Tests values								
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa	S_{a5}= 105 MPa	S_{a6}= 160 MPa	S_{a7}= 320 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$	$Q_{sminL,Sa6}$	$Q_{sminL,Sa7}$
High tightness	13,7	10	10	10	10	10	10	10
Very high tightness	217,5	NA	NA	NA	NA	NA	NA	130,8

P = 80 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=320 MPa
		$Q_{sminL,Sa1}$
High tightness	31,6	10

P = 160 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
High tightness	72,9	32

Table P.14 — Gasket 1-07-001-1 – Expanded graphite with metallic sheets laminated in thin layers with standing high stresses

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	300 °C	240 MPa
	400 °C	240 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 1
	300 °C	Initial load: 120 MPa Average for g_C: 1
	400 °C	Initial load: 120 MPa Average for g_C: 0,73
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	300 °C	Initial load: 120 MPa Average for P_{QR}: 1
	400 °C	Initial load: 120 MPa Average for P_{QR}: 0,98

Q_0	E_G		
	Room	**300 °C**	**400 °C**
20 MPa	352 MPa	371 MPa	438 Mpa
30 MPa	679 MPa	526 MPa	793 Mpa
40 MPa	1 041 MPa	734 MPa	958 Mpa
50 MPa	1 117 MPa	1 177 MPa	1 121 Mpa
60 MPa	1 424 MPa	1 287 MPa	1 912 Mpa
80 MPa	1 496 MPa	1 588 MPa	2 803 Mpa
100 MPa	1 803 MPa	2 107 MPa	2 057 Mpa
120 MPa	1 904 MPa	3 371 MPa	2 498 Mpa
140 MPa	2 340 MPa	2 853 MPa	2 948 Mpa
160 MPa	2 371 MPa	2 722 MPa	3 334 Mpa
180 MPa	2 272 MPa	3 567 MPa	3 145 Mpa

Table P.14 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
High tightness	10	NC
Very high tightness	83	10 [a]
[a] The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of "full tests" values						
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 100 MPa	S_{a5}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$
High tightness	30	10 [a]	10 [a]	10 [a]	10 [a]	NC
Very high tightness	120	NC	NC	NC	NC	38
[a] The minimum gasket stress was always 10 MPa.						

Table P.15 — Gasket 1-10-001-1 – Modified PTFE sheet material

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	150 °C	240 MPa
	225 °C	240 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,31
	150 °C	Initial load: 120 MPa Average for g_C: 0,07
	225 °C	Initial load: 120 MPa Average for g_C: 0,05
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 0,9
	150 °C	Initial load: 120 MPa Average for P_{QR}: 0,6
	225 °C	Initial load: 120 MPa Average for P_{QR}: 0,5

Q_0	E_G		
	Room	**150 °C**	**225 °C**
20 MPa	2 175 MPa	2 023 MPa	1 291 MPa
40 MPa	2 552 MPa	2 161 MPa	1 458 MPa
60 MPa	3 577 MPa	2 257 MPa	2 243 MPa
80 MPa	5 753 MPa	2 764 MPa	1 764 MPa
100 MPa	4 057 MPa	2 739 MPa	1 861 MPa
120 MPa	3 942 MPa	2 404 MPa	2 850 MPa
140 MPa	4 420 MPa	2 596 MPa	1 777 MPa
160 MPa	4 022 MPa	2 376 MPa	1 533 MPa
180 MPa	3 063 MPa	2 847 MPa	1 595 MPa

Table P.15 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
Very high tightness	12	NC

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 60 MPa	S_{a4}= 80 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
Very high tightness	20	14	10 [a]	10 [a]	10 [a]
[a] The minimum gasket stress was always 10 MPa.					

Table P.16 — Gasket 1-09-002-1 – Non-asbestos fibre with binder $e_G \geq 1$ mm

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	175 °C	240 MPa
	250 °C	80 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,72
	175 °C	Initial load: 80 MPa Average for g_C: 0,18
	250 °C	Initial load: 80 MPa Average for g_C: 0,08
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	175 °C	Initial load: 80 MPa Average for P_{QR}: 0,8
	250 °C	Initial load: 80 MPa Average for P_{QR}: 0,6

Q_0	E_G		
	Room	**175 °C**	**250 °C**
20 MPa	1 512 MPa	1 847 MPa	2 575 MPa
40 MPa	2 006 MPa	1 911 MPa	2 063 MPa
60 MPa	2 668 MPa	3 218 MPa	3 392 MPa
80 MPa	3 290 MPa	3 342 MPa	2 967 MPa
100 MPa	3 997 MPa	2 909 MPa	3 417 MPa
120 MPa	4 296 MPa	3 503 MPa	2 903 MPa
140 MPa	4 578 MPa	3 405 MPa	2 848 MPa
160 MPa	5 187 MPa	2 960 MPa	3 006 MPa
180 MPa	4 529 MPa	2 946 MPa	3 001 MPa

Table P.16 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
High tightness	15	NC
Very high tightness	47	NC

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 100 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
High tightness	25	10 [a]	NC	NC	NC
Very high tightness	54	NC	19	10 [a]	10 [a]
[a] The minimum gasket stress was always 10 MPa.					

Table P.17 — Gasket 1-10-004-1 – Modified PTFE sheet material

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	150 °C	240 MPa
	225 °C	240 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 80 MPa Average for g_{C}: 0,23
	150 °C	Initial load: 40 MPa Average for g_{C}: 0,12
	225 °C	Initial load: 40 MPa Average for g_{C}: 0,07
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 80 MPa Average for P_{QR}: 0,9
	150 °C	Initial load: 40 MPa Average for P_{QR}: 0,6
	225 °C	Initial load: 40 MPa Average for P_{QR}: 0,4

Q_0	E_{G}		
	Room	**150 °C**	**225 °C**
20 MPa	402 MPa	510 MPa	579 Mpa
40 MPa	883 MPa	1 092 MPa	553 Mpa
60 MPa	1 345 MPa	1 313 MPa	1 127 Mpa
80 MPa	1 889 MPa	2 538 MPa	990 Mpa
100 MPa	2 055 MPa	1 224 MPa	923 Mpa
120 MPa	1 663 MPa	1 212 MPa	779 Mpa
140 MPa	1 333 MPa	808 MPa	699 Mpa
160 MPa	1 145 MPa	617 MPa	762 Mpa
180 MPa	1 357 MPa	499 MPa	603 Mpa

Table P.17 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
High tightness	10	NC
Very high tightness	31	NC

P = 40 bar – Average of "full tests" values			
Tightness Class	Q_{minL}	S_{a1}= 60 MPa	S_{a2}= 80 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$
High tightness	25	NC	NC
Very high tightness	36	10 [a]	10 [a]

[a] The minimum gasket stress was always 10 MPa.

Table P.18 — Gasket 1-09-004-1 – Non-asbestos fibre with binder $e_G \geq 1$ mm

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	175 °C	240 MPa
	250 °C	
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_c: 0,63
	175 °C	Initial load: 80 MPa Average for g_c: 0,13
	250 °C	Initial load: 80 MPa Average for g_c: 0,07
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	175 °C	Initial load: 80 MPa Average for P_{QR}: 0,7
	250 °C	Initial load: 80 MPa Average for P_{QR}: 0,5

Q_0	E_G		
	Room	**175 °C**	**250 °C**
20 MPa	727 MPa	900 MPa	1 140 MPa
30 MPa	1 280 MPa	1 184 MPa	1 402 MPa
40 MPa	1 712 MPa	1 401 MPa	1 471 MPa
50 MPa	1 759 MPa	1 805 MPa	2 041 MPa
60 MPa	1 940 MPa	2 235 MPa	2 046 MPa
80 MPa	2 619 MPa	1 936 MPa	2 100 MPa
100 MPa	3 252 MPa	2 326 MPa	3 189 MPa
120 MPa	2 799 MPa	2 049 MPa	2 605 MPa
140 MPa	3 193 MPa	2 099 MPa	2 145 MPa
160 MPa	3 459 MPa	2 299 MPa	2 174 MPa
180 MPa	3 776 MPa	2 145 MPa	2 348 MPa

Table P.18 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
High tightness	40	NC
Very high tightness	100	17

P = 40 bar – Average of "full tests" values				
Tightness Class	Q_{minL}	S_{a1}= 80 MPa	S_{a2}= 100 MPa	S_{a3}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$
High tightness	64	14	10 [a]	10 [a]
Very high tightness	111	NC	NC	32

[a] The minimum gasket stress was always 10 MPa.

Table P.19 — Gasket 1-09-005-1 – Non-asbestos fibre with binder $e_G \geq 1$ mm

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	175 °C	120 MPa
	250 °C	100 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,71
	175 °C	Initial load: 120 MPa Average for g_C: 0,24
	250 °C	Initial load: 100 MPa Average for g_C: 0,22
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	175 °C	Initial load: 120 MPa Average for P_{QR}: 0,8
	250 °C	Initial load: 100 MPa Average for P_{QR}: 0,8

Q_0	E_G		
	Room	**175 °C**	**250 °C**
20 MPa	1 357 MPa	1 095 MPa	2 096 MPa
40 MPa	1 802 MPa	1 874 MPa	2 082 MPa
60 MPa	2 175 MPa	2 320 MPa	3 165 MPa
80 MPa	2 904 MPa	2 603 MPa	3 592 MPa
100 MPa	3 537 MPa	3 901 MPa	3 657 MPa
120 MPa	4 124 MPa	4 230 MPa	2 588 MPa
140 MPa	4 526 MPa	1 871 MPa	1 834 MPa
160 MPa	4 362 MPa	1 933 MPa	2 120 MPa
180 MPa	4 258 MPa	2 037 MPa	2 276 MPa

Table P.19 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
High tightness	12	NC
Very high tightness	84	10 [a]

[a] The minimum gasket stress was always 10 MPa.

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}= 60 MPa	S_{a2}= 80 MPa	S_{a3}= 100 MPa	S_{a4}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
High tightness	48	15	10 [a]	10 [a]	NC
Very high tightness	108	NC	NC	NC	15

[a] The minimum gasket stress was always 10 MPa.

Table P.20 — Gasket 1-05-005-1 – Expanded graphite with adhesive perforated metal insertion

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	200 MPa
	300 °C	180 MPa
	400 °C	180 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_{C}: 0,97
	300 °C	Initial load: 120 MPa Average for g_{C}: 0,70
	400 °C	Initial load: 120 MPa Average for g_{C}: 0,54
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	300 °C	Initial load: 120 MPa Average for P_{QR}: 1
	400 °C	Initial load: 120 MPa Average for P_{QR}: 1

Q_0	E_G		
	Room	**300 °C**	**400 °C**
20 MPa	411 MPa	499 MPa	484 Mpa
30 MPa	700 MPa	866 MPa	849 Mpa
40 MPa	1 019 MPa	1 023 MPa	1 097 Mpa
50 MPa	1 248 MPa	1 594 MPa	1 250 Mpa
60 MPa	1 438 MPa	1 363 MPa	1 708 Mpa
80 MPa	2 240 MPa	2 424 MPa	2 829 Mpa
100 MPa	2 411 MPa	2 404 MPa	1 943 Mpa
120 MPa	2 372 MPa	3 198 MPa	3 337 Mpa
140 MPa	2 783 MPa	3 397 MPa	3 389 Mpa
160 MPa	3 235 MPa	3 380 MPa	3 057 Mpa
180 MPa	3 081 MPa	4 246 MPa	3 148 Mpa

Table P.20 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
High tightness	28	NC
Very high tightness	94	10 [a]
[a] The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}= 60 MPa	S_{a2}= 80 MPa	S_{a3}= 100 MPa	S_{a4}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
High tightness	47	11	10 [a]	10 [a]	NC
Very high tightness	122	NC	NC	NC	29
[a] The minimum gasket stress was always 10 MPa.					

Table P.21 — Gasket 2-05-006-1 – Kammprofile gasket with bounded graphite layers

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	650 MPa
	300 °C	650 MPa
	400 °C	650 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,96
	300 °C	Initial load: 120 MPa Average for g_C: 0,77
	400 °C	Initial load: 120 MPa Average for g_C: 0,55
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 328 MPa Average for P_{QR}: 1
	300 °C	Initial load: 328 MPa Average for P_{QR}: 0,98
	400 °C	Initial load: 328 MPa Average for P_{QR}: 0,96

Q_0	E_G		
	Room	**300 °C**	**400 °C**
54 MPa	8 921 MPa	13 616 MPa	8 922 MPa
82 MPa	10 323 MPa	10 805 MPa	17 190 MPa
109 MPa	10 719 MPa	13 262 MPa	19 056 MPa
137 MPa	11 961 MPa	12 189 MPa	14 765 MPa
164 MPa	15 795 MPa	22 923 MPa	16 863 MPa
219 MPa	16 347 MPa	14 635 MPa	17 365 MPa
274 MPa	16 257 MPa	17 406 MPa	13 680 MPa
329 MPa	15 561 MPa	18 733 MPa	13 144 MPa
384 MPa	18 900 MPa	18 326 MPa	15 805 MPa
438 MPa	20 856 MPa	19 654 MPa	16 729 MPa
493 MPa	17 100 MPa	19 558 MPa	19 335 MPa
548 MPa	16 577 MPa	17 079 MPa	15 245 MPa
602 MPa	18 265 MPa	17 000 MPa	16 314 MPa
651 MPa	17 258 MPa	18 220 MPa	15 615 MPa

Table P.21 (*concluded*)

Class of tightness from tightness tests

P = 40 bar		
Tightness Class	Q_{minL}	S_{a1}=20 Mpa
		$Q_{\mathrm{sminL,Sa1}}$
Very high tightness	10	10

Table P.22 — Gasket 1-09-007-1 – Non-asbestos fibre with binder $e_G \geq 1$ mm

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	175 °C	240 MPa
	250 °C	240 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,80
	175 °C	Initial load: 120 MPa Average for g_C: 0,29
	250 °C	Initial load: 120 MPa Average for g_C: 0,24
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	175 °C	Initial load: 120 MPa Average for P_{QR}: 0,9
	250 °C	Initial load: 120 MPa Average for P_{QR}: 0,9

Q_0	E_G		
	Room	**175 °C**	**250 °C**
20 MPa	1 773 MPa	1 931 MPa	2 259 MPa
30 MPa	2 079 MPa	2 442 MPa	3 522 MPa
40 MPa	1 963 MPa	2 152 MPa	3 339 MPa
50 MPa	2 917 MPa	2 496 MPa	2 814 MPa
60 MPa	3 318 MPa	3 164 MPa	2 950 MPa
80 MPa	4 026 MPa	4 253 MPa	4 929 MPa
100 MPa	4 843 MPa	4 114 MPa	4 514 MPa
120 MPa	5 402 MPa	3 350 MPa	4 029 MPa
140 MPa	5 044 MPa	4 611 MPa	4 331 MPa
160 MPa	4 507 MPa	4 106 MPa	5 231 MPa
180 MPa	4 394 MPa	3 876 MPa	5 467 MPa

Table P.22 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
High tightness	22	NC
Very high tightness	66	10 [a]
[a] The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of "full tests" values						
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}=100 MPa	S_{a5}=160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$
High tightness	24	10 [a]	10 (*)	NC	NC	NC
Very high tightness	68	NC	NC	19	10 [a]	10 [a]
[a] The minimum gasket stress was always 10 MPa.						

Table P.23 — Gasket 1-10-007-1 – Modified PTFE sheet material

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	150 °C	160 MPa
	225 °C	100 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,90
	150 °C	Initial load: 120 MPa Average for g_C: 0,60
	225 °C	Initial load: 120 MPa Average for g_C: 0,18
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	150 °C	Initial load: 120 MPa Average for P_{QR}: 1
	225 °C	Initial load: 120 MPa Average for P_{QR}: 0,8

Q_0	E_G		
	Room	**150 °C**	**225 °C**
20 MPa	2 704 MPa	1 981 MPa	1 874 Mpa
30 MPa	3 283 MPa	2 833 MPa	2 166 Mpa
40 MPa	3 125 MPa	4 491 MPa	3 215 Mpa
50 MPa	4 286 MPa	4 276 MPa	3 551 Mpa
60 MPa	3 880 MPa	4 982 MPa	3 613 Mpa
80 MPa	4 413 MPa	3 663 MPa	4 035 Mpa
100 MPa	4 779 MPa	4 074 MPa	3 953 Mpa
120 MPa	4 684 MPa	4 422 MPa	4 174 Mpa
140 MPa	5 081 MPa	4 536 MPa	4 533 Mpa
160 MPa	5 205 MPa	5 629 MPa	3 797 Mpa
180 MPa	5 410 MPa	5 450 MPa	3 656 Mpa

Table P.23 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
High tightness	11	NC
Very high tightness	58	NC

P = 40 bar – Average of "full tests" values						
Tightness Class	Q_{minL}	S_{a1}= 40 MPa	S_{a2}= 60 MPa	S_{a3}= 80 MPa	S_{a4}= 100 MPa	S_{a5}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$	$Q_{sminL,Sa5}$
High tightness	25	10 [a]	10 [a]	10 [a]	NC	NC
Very high tightness	83	NC	NC	NC	27	10 [a]

[a] The minimum gasket stress was always 10 MPa.

Table P.24 — Gasket 3-02-007-1 – PTFE Standard spiral wound gasket with inner and outer support ring

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	690 MPa
	150 °C	690 MPa
	225 °C	690 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 80 MPa Average for g_C: 0,80
	150 °C	Initial load: 120 MPa Average for g_C: 0,80
	225 °C	Initial load: 120 MPa Average for g_C: 0,91
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 240 MPa Average for P_{QR}: 0,96
	150 °C	Initial load: 360 MPa Average for P_{QR}: 0,98
	225 °C	Initial load: 360 MPa Average for P_{QR}: 0,99

Q_0	E_G		
	Room	**150 °C**	**225 °C**
57 MPa	2 989 MPa	3 232 MPa	2 415 MPa
86 MPa	3 742 MPa	3 507 MPa	2 694 MPa
115 MPa	4 723 MPa	3 933 MPa	3 241 MPa
144 MPa	5 324 MPa	4 980 MPa	4 363 MPa
173 MPa	5 241 MPa	5 479 MPa	5 221 MPa
231 MPa	6 519 MPa	6 751 MPa	6 597 MPa
290 MPa	7 566 MPa	10 077 MPa	8 521 MPa
348 MPa	10 518 MPa	13 690 MPa	11 485 MPa
406 MPa	14 394 MPa	19 892 MPa	15 054 MPa
464 MPa	17 000 MPa	28 614 MPa	18 352 MPa
522 MPa	25 742 MPa	34 196 MPa	25 922 MPa

Table P.24 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=480 Mpa
		$Q_{sminL,Sa1}$
High tightness	39	30
Very high tightness	69	30

P = 40 bar – Average of "full tests" values		
Tightness Class	Q_{minL}	S_{a1}=480 Mpa
		$Q_{sminL,Sa1}$
High tightness	74	30
Very high tightness	105	30

Table P.25 — Gasket 5-05-103-2 – Covered metal jacketed gasket with graphite

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	300 °C	240 MPa
	400 °C	240 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_{C}: 1
	300 °C	Initial load: 120 MPa Average for g_{C}: 1
	400 °C	Initial load: 120 MPa Average for g_{C}: 1
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	300 °C	Initial load: 120 MPa Average for P_{QR}: 1
	400 °C	Initial load: 120 MPa Average for P_{QR}: 1

Q_0	E_G		
	Room	**300 °C**	**400 °C**
20 MPa	659 MPa	823 MPa	825 MPa
30 MPa	1 029 MPa	1 708 MPa	1 345 MPa
40 MPa	1 778 MPa	2 175 MPa	3 107 MPa
50 MPa	2 595 MPa	3 525 MPa	5 080 MPa
60 MPa	3 124 MPa	4 442 MPa	5 082 MPa
80 MPa	5 409 MPa	9 476 MPa	4 918 MPa
100 MPa	9 487 MPa	9 837 MPa	10 608 MPa
120 MPa	11 419 MPa	79 398 MPa	20 040 MPa
140 MPa	16 002 MPa	82 833 MPa	30 099 MPa
160 MPa	17 889 MPa	267 301 MPa	25 893 MPa
180 MPa	24 030 MPa	-	28 125 MPa

Table P.25 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
Very high tightness	10 [a]	NC

P = 40 bar – Average of "full tests" values		
Tightness Class	Q_{minL}	S_{a1}=20 MPa
		$Q_{sminL,Sa1}$
Very high tightness	15	10 [a]
[a] The minimum gasket stress was always 10 MPa.		

Table P.26 — Gasket 6-04-103-2 – Metal jacketed gasket with graphite

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	300 °C	240 MPa
	400 °C	240 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_{C}: 1,0
	300 °C	Initial load: 120 MPa Average for g_{C}: 0,84
	400 °C	Initial load: 120 MPa Average for g_{C}: 0,68
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	300 °C	Initial load: 120 MPa Average for P_{QR}: 1
	400 °C	Initial load: 120 MPa Average for P_{QR}: 1

Q_0	E_{G}		
	Room	**300 °C**	**400 °C**
20 MPa	709 MPa	798 MPa	796 Mpa
30 MPa	1 120 MPa	1 050 MPa	1 070 Mpa
40 MPa	1 344 MPa	1 531 MPa	1 372 Mpa
50 MPa	1 902 MPa	1 952 MPa	2 647 Mpa
60 MPa	2 424 MPa	2 509 MPa	2 519 Mpa
80 MPa	3 171 MPa	3 613 MPa	3 379 Mpa
100 MPa	3 495 MPa	3 621 MPa	4 099 Mpa
120 MPa	5 158 MPa	4 866 MPa	5 487 Mpa
140 MPa	5 876 MPa	4 975 MPa	5 282 Mpa
160 MPa	5 525 MPa	6 288 MPa	5 607 MPa
180 MPa	5 965 MPa	6 618 MPa	6 302 MPa

Class of tightness from tightness tests

Not available.

Table P.27 — Gasket 1-07-009-1 – Expanded graphite with metallic sheets laminated in thin layers with standing high stresses

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	300 °C	120 MPa
	400 °C	120 MPa
g_{C} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_{C}: 0,92
	300 °C	Initial load: 120 MPa Average for g_{C}: 0,48
	400 °C	Initial load: 120 MPa Average for g_{C}: NA
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 1
	300 °C	Initial load: 120 MPa Average for P_{QR}: 0,9
	400 °C	Initial load: 120 MPa -

Q_0	E_{G}		
	Room	**300 °C**	**400 °C**
20 MPa	532 MPa	544 MPa	471 MPa
30 MPa	752 MPa	670 MPa	1 091 MPa
40 MPa	1 101 MPa	960 MPa	848 MPa
50 MPa	1 148 MPa	1 124 MPa	1 398 MPa
60 MPa	1 681 MPa	1 788 MPa	1 586 MPa
80 MPa	1 828 MPa	1 750 MPa	1 692 MPa
100 MPa	2 451 MPa	2 723 MPa	2 202 MPa
120 MPa	3 403 MPa	2 420 MPa	3 487 MPa
140 MPa	3 717 MPa	1 651 MPa	1 203 MPa
160 MPa	3 169 MPa	1 749 MPa	1 530 MPa
180 MPa	3 345 MPa	2 264 MPa	1 718 MPa

Table P.27 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}
		$Q_{sminL,Sa1}$
Very high tightness	10 [a]	NC
A The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of "full tests" values					
Tightness Class	Q_{minL}	S_{a1}= 60 MPa	S_{a2}= 80 MPa	S_{a3}= 100 MPa	S_{a4}= 160 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
Very high tightness	10 [a]	10 [a]	10 [a]	10 [a]	10 [a]
A The minimum gasket stress was always 10 MPa.					

Table P.28 — Gasket 1-10-009-1 – PTFE envelope gasket

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	150 °C	240 MPa
	225 °C	240 MPa
g_C (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_C: 0,59
	150°C	Initial load: 120 MPa Average for g_C: 0,11
	225 °C	Initial load: 120 MPa Average for g_C: 0,08
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 40 MPa Average for P_{QR}: 0,9
	150 °C	Initial load: 40 MPa Average for P_{QR}: 0,5
	225 °C	Initial load: 40 MPa Average for P_{QR}: 0,4

Q_0	E_G		
	Room	**150 °C**	**225 °C**
2,5 MPa	23 MPa	27 MPa	36 MPa
5 MPa	74 MPa	72 MPa	76 MPa
10 MPa	219 MPa	230 MPa	207 MPa
20 MPa	434 MPa	511 MPa	520 MPa
30 MPa	658 MPa	893 MPa	564 MPa
40 MPa	750 MPa	810 MPa	1 296 MPa
50 MPa	883 MPa	999 MPa	677 MPa
60 MPa	1 124 MPa	1 357 MPa	930 MPa
80 MPa	1 378 MPa	912 MPa	1 938 MPa
100 MPa	1 671 MPa	1 497 MPa	
120 MPa	2 051 MPa	968 MPa	
140 MPa	2 034 MPa		
160 MPa	1 394 MPa		
180 MPa	1 629 MPa		

Table P.28 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – “Simplified test” values		
Tightness Class	Q_{minL}	S_{a1}=160 MPa
		$Q_{sminL,Sa1}$
Very high tightness	13	10 [a]
A The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of “full tests” values					
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 40 MPa	S_{a3}= 80 MPa	S_{a4}= 100 MPa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$	$Q_{sminL,Sa3}$	$Q_{sminL,Sa4}$
Very high tightness	10 [a]	10 [a]	10 [a]	10 [a]	10 [a]
A The minimum gasket stress was always 10 MPa.					

Table P.29 — Gasket 7-01-009-1 – Corrugated gasket with graphite

Gasket coefficients from mechanical tests

Coefficient	Temperature	Values
Q_{smax}	Room	240 MPa
	300 °C	240 MPa
	400 °C	240 MPa
g_c (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for g_c: 0,97
	300 °C	Initial load: 120 MPa Average for g_c: 0,28
	400 °C	Initial load: 120 MPa Average for g_c: 0,34
P_{QR} (for a simulated stiffness of 500 kN/mm)	Room	Initial load: 120 MPa Average for P_{QR}: 0,9
	300 °C	Initial load: 120 MPa Average for P_{QR}: 0,5
	400 °C	Initial load: 120 MPa Average for P_{QR}: 0,4

Q_0	E_G		
	Room	**300 °C**	**400 °C**
2,5 MPa		113 MPa	62 MPa
5 MPa		178 MPa	186 MPa
10 MPa		622 MPa	533 MPa
20 MPa	50 MPa	1 027 MPa	3 446 MPa
30 MPa	193 MPa	3 548 MPa	2 494 MPa
40 MPa	618 MPa	2 323 MPa	2 474 MPa
50 MPa	1 326 MPa	2 327 MPa	3 179 MPa
60 MPa	1 632 MPa	2 632 MPa	4 698 MPa
80 MPa	2 403 MPa	4 646 MPa	2 393 MPa
100 MPa	2 741 MPa	2 666 MPa	2 338 MPa
120 MPa	2 807 MPa	3 088 MPa	3 331 MPa
140 MPa	2 606 MPa		
160 MPa	3 127 MPa		
180 MPa	4 002 MPa		

Table P.29 (*concluded*)

Class of tightness from tightness tests

P = 10 bar – "Simplified test" values		
Tightness Class	Q_{minL}	S_{a1}=160 Mpa
		$Q_{sminL,Sa1}$
Very high tightness	14	11
A The minimum gasket stress was always 10 MPa.		

P = 40 bar – Average of "full tests" values			
Tightness Class	Q_{minL}	S_{a1}= 20 MPa	S_{a2}= 160 Mpa
		$Q_{sminL,Sa1}$	$Q_{sminL,Sa2}$
High tightness	10,8	10 [a]	NC
Very high tightness	28,2	NC	10 [a]
A The minimum gasket stress was always 10 MPa.			

Annex Q
(informative)

Simplified pipe stress analysis

Q.1 General

The pressure design of all piping components should be carried out in accordance with the rules of EN 13480-3. Stresses due to sustained loads, occasional and exceptional loads, thermal expansion and alternating loads have to be taken into account to meet the code stress requirements for their particular load case.

Q.2 Simplified procedure

Q.2.1 General

As a deviation from Q.1, the stresses from gravitational forces and temperature fluctuations can be simply determined (see 12.2.10) in accordance with Q.2.2 and Q.2.3 independent of the other stresses in each case.

This procedure should be applied by experienced personal only. It does not yield information about the loads at anchor points and should not be applied to piping, operating in the creep range.

This method only applies to piping which is not buried. For buried piping, additional considerations, for example with regard to vertical soil loads, restrained expansion due to soil resistance, or mining subsidence are necessary. These are not covered by the simplified method.

Q.2.2 Specification of allowable spacing of supports

The specification of the allowable spacing of supports, limits the effect of the dead weight of the piping on the deflections and the stresses. Thus it is possible to consider the internal pressure and the dead weight of the piping separate from each other. Proof of the permissibility of the support spacing is provided if, for steel pipes, the support distances given in Table Q.1 are not exceeded and the explanatory notes regarding the specification of support distances are considered. For other parameters, e.g. other materials, the Table Q.1 can be converted in accordance with the information in the explanatory notes of Table Q.1.

Q.2.3 Check of elasticity

To meet the stress limitations in the load case thermal expansion the piping need to have a sufficient elasticity. This is normally achieved by a routing that allows for bending and torsional deflection due to compensating measures. A design calculation for elasticity is not required if the leg lengths comply with the conditions of Figure Q.2. It is assumed that torsional stresses are less significant due to the routing.

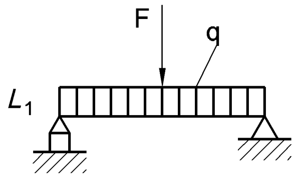
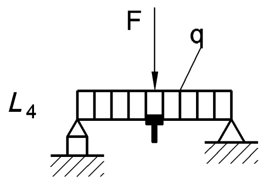
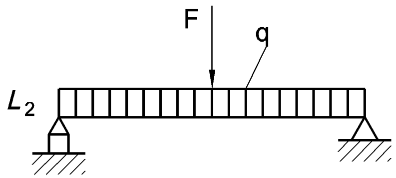
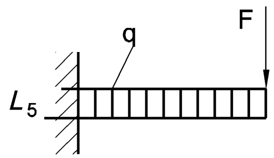
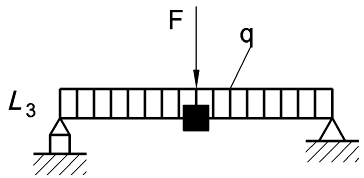
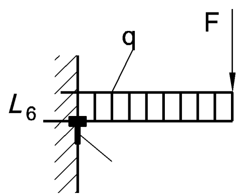
Examples of application of Figure Q.2 and explanatory notes are given in Q.9.

Table Q.1 — Allowable spacing of supports for steel pipes (for boundary conditions refer to Explanatory Notes for Table Q.1)

			Empty pipe, without insulation							Water-filled pipe, without insulation							Water-filled pipe, insulation thickness 40 mm							Water-filled pipe, insulation thickness 80 mm						
DN	d_a	s	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6
mm			kg/m	m						kg/m	m						kg/m	m						kg/m	m					
25	33,7	2,0	1,6	2,9	5,6	4,8	2,9	2,8	1,5	2,3	2,7	4,6	4,0	2,5	2,3	1,2	7,0	2,0	2,7	2,3	1,4	1,3	0,7	11,8	1,8	2,0	1,8	1,1	1,0	0,5
25	33,7	4,0	2,9	2,9	5,2	5,2	3,6	2,6	1,8	3,5	2,8	4,8	4,8	3,3	2,4	1,7	8,1	2,2	3,2	3,2	2,2	1,6	1,1	13,0	2,0	2,5	2,5	1,7	1,3	0,9
40	48,3	2,0	2,3	3,6	6,8	5,2	3,2	3,4	1,6	3,9	3,1	5,2	4,0	2,4	2,6	1,2	9,2	2,5	3,4	2,6	1,6	1,7	0,8	14,3	2,3	2,7	2,1	1,3	1,4	0,6
40	48,3	4,0	4,4	3,5	6,5	6,4	3,9	3,3	1,9	5,7	3,3	5,7	5,6	3,4	2,9	1,7	11,0	2,8	4,1	4,0	2,5	2,1	1,2	16,1	2,5	3,4	3,3	2,0	1,7	1,0
50	60,3	2,0	2,9	4,0	7,6	5,4	3,3	3,8	1,7	5,4	3,4	5,6	4,0	2,4	2,8	1,2	11,3	2,9	3,9	2,7	1,7	1,9	0,8	16,6	2,6	3,2	2,3	1,4	1,6	0,7
50	60,3	4,5	6,2	3,9	7,3	6,9	4,2	3,7	2,1	8,3	3,6	6,4	6,0	3,6	3,2	1,8	14,2	3,2	4,9	4,6	2,8	2,4	1,4	19,4	2,9	4,2	3,9	2,4	2,1	1,2
80	88,9	2,3	5,0	5,5	9,3	6,1	3,7	4,7	1,9	10,6	4,6	6,4	4,2	2,5	3,2	1,3	17,8	4,0	4,9	3,2	2,0	2,5	1,0	23,5	3,7	4,3	2,8	1,7	2,1	0,9
80	88,9	5,6	11,5	5,4	9,0	8,0	4,9	4,5	2,4	16,3	5,0	7,6	6,7	4,1	3,8	2,1	23,5	4,5	6,3	5,6	3,4	3,2	1,7	29,2	4,3	5,7	5,0	3,1	2,8	1,5
100	114,3	2,6	7,3	6,3	10,6	6,6	4,0	5,3	2,0	16,6	5,1	7,0	4,4	2,7	3,5	1,3	25,0	4,6	5,7	3,5	2,2	2,9	1,1	31,1	4,4	5,1	3,2	1,9	2,6	1,0
100	114,3	6,3	16,8	6,2	10,3	8,7	5,3	5,1	2,7	24,9	5,6	8,5	7,1	4,4	4,2	2,2	33,3	5,2	7,3	6,2	3,8	3,7	1,9	39,4	5,0	6,7	5,7	3,5	3,4	1,7
150	168,3	2,6	10,8	7,6	13,0	7,1	4,3	6,5	2,2	31,7	5,8	7,6	4,1	2,5	3,8	1,3	42,6	5,4	6,5	3,6	2,2	3,3	1,1	49,5	5,2	6,0	3,3	2,0	3,0	1,0
150	168,3	7,1	28,2	7,5	12,6	9,7	5,9	6,3	3,0	46,9	6,6	9,8	7,6	4,6	4,9	2,3	57,8	6,3	8,8	6,8	4,2	4,4	2,1	64,7	6,1	8,4	6,4	3,9	4,2	2,0
200	219,1	2,9	15,7	8,7	14,8	7,7	4,7	7,4	2,3	51,4	6,5	8,2	4,2	2,6	4,1	1,3	64,7	6,1	7,3	3,8	2,3	3,6	1,2	72,3	5,9	6,9	3,6	2,2	3,4	1,1
200	219,1	7,1	37,1	8,6	14,6	10,2	6,2	7,3	3,1	70,1	7,4	10,6	7,4	4,5	5,3	2,3	83,4	7,1	9,7	6,8	4,2	4,9	2,1	91,0	6,9	9,3	6,5	4,0	4,7	2,0
250	273,0	2,9	19,6	9,8	16,6	8,0	4,9	8,3	2,4	75,6	6,9	8,4	4,1	2,5	4,2	1,2	91,5	6,6	7,7	3,7	2,3	3,8	1,1	99,9	6,5	7,3	3,5	2,2	3,7	1,1
250	273,0	7,1	46,6	9,7	16,4	10,6	6,5	8,2	3,3	99,2	8,0	11,2	7,3	4,5	5,6	2,2	115,0	7,7	10,4	6,8	4,1	5,2	2,1	123,4	7,6	10,1	6,6	4,0	5,0	2,0
300	323,9	2,9	23,3	10,6	18,1	8,2	5,0	9,1	2,5	102,7	7,3	8,6	3,9	2,4	4,3	1,2	120,9	7,0	7,9	3,6	2,2	4,0	1,1	130,1	6,9	7,6	3,5	2,1	3,8	1,1
300	323,9	8,0	62,3	10,6	17,8	11,4	7,0	8,9	3,5	136,8	8,7	12,1	7,7	4,7	6,0	2,4	155,0	8,4	11,3	7,2	4,4	5,7	2,2	164,2	8,3	11,0	7,0	4,3	5,5	2,2
350	355,6	3,2	28,2	11,1	19,0	8,6	5,3	9,5	2,6	123,9	7,7	9,0	4,1	2,5	4,5	1,3	143,6	7,4	8,4	3,8	2,3	4,2	1,2	153,3	7,3	8,1	3,7	2,3	4,1	1,1
350	355,6	8,8	75,3	11,1	18,7	12,0	7,3	9,4	3,7	165,0	9,1	12,7	8,1	4,9	6,3	2,5	184,7	8,8	12,0	7,7	4,7	6,0	2,3	194,3	8,7	11,7	7,5	4,6	5,8	2,3
400	406,4	3,2	32,2	11,9	20,3	8,8	5,4	10,2	2,7	157,9	8,0	9,2	4,0	2,4	4,6	1,2	179,9	7,7	8,6	3,7	2,3	4,3	1,1	190,4	7,6	8,3	3,6	2,2	4,2	1,1
400	406,4	10,0	97,8	11,8	20,0	12,8	7,8	10,0	3,9	215,0	9,7	13,5	8,6	5,3	6,8	2,6	237,0	9,5	12,9	8,2	5,0	6,4	2,5	247,5	9,4	12,6	8,0	4,9	6,3	2,5
500	508,0	4,0	50,4	13,3	22,7	9,9	6,0	11,4	3,0	246,7	8,9	10,2	4,4	2,7	5,1	1,4	273,4	8,7	9,7	4,2	2,6	4,9	1,3	285,4	8,6	9,5	4,1	2,5	4,8	1,3
500	508,0	11,0	134,8	13,2	22,4	13,7	8,4	11,2	4,2	320,3	10,7	14,6	8,9	5,4	7,3	2,7	347,1	10,5	14,0	8,6	5,2	7,0	2,6	359,1	10,4	13,8	8,4	5,1	6,9	2,6

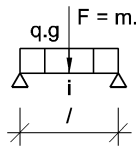
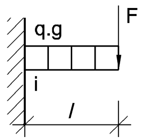
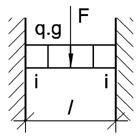
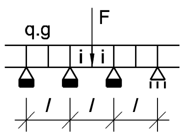
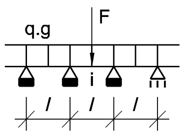
Q.3 Explanatory notes for Table Q.1

Table Q.2

L_1		f_{all} = 3 mm DN ≤ 50 f_{all} = 5 mm DN > 50		L_4	 Welded Tee	σ_{max} = MIN(40 N/mm^2 ; 0,4 f_h) i according to Annex F
L_2		σ_{max} = MIN(40 N/mm^2 ; 0,4 f_h)		L_5		σ_{max} = MIN(40 N/mm^2 ; 0,4 f_h)
L_3	 Forged Tee	σ_{max} = MIN(40 N/mm^2 ; 0,4 f_h) i according to Annex F		L_6	 Welded Tee	σ_{max} = MIN(40 N/mm^2 ; 0,4 f_h) i according to Annex F

Conditions:
Forged, or welded Tee with horizontal branch
Stress from internal pressure not considered
Tolerances and allowances not considered

Table Q.3

Case	System	Load	Criterion: Deflection	Criterion: Stress	Remarks	Index of curve in figure 1
A		q [kg/m] m [kg]	$f_A = \frac{l_{AF}^3 \cdot 9{,}81 \cdot 5 \cdot 10^6}{384\,EI}(q \cdot l_{AF} + 1{,}6\,m)$	$l_{AS} = -\frac{m}{q} + \sqrt{\left(\left(\frac{m}{q}\right)^2 + \frac{8 \cdot W \cdot \sigma_{max}}{9{,}81 \cdot 10^3 \cdot q \cdot i}\right)}$		1
B		q [kg/m] m [kg]	$f_B = \frac{l_{BF}^3 \cdot 9{,}81 \cdot 10^6}{24\,EI} \cdot (3q \cdot l_{BF} + 8\,m)$	$l_{BS} = -\frac{m}{q} + \sqrt{\left(\left(\frac{m}{q}\right)^2 + \frac{2 \cdot W \cdot \sigma_{max}}{9{,}81 \cdot 10^3 \cdot q \cdot i}\right)}$		1
C		q + Single load in all fields	$f_C = \frac{l_{CF}^3 \cdot 9{,}81 \cdot 10^6}{384\,EI} \cdot (q \cdot l_{CF} + 2\,m)$	$l_{CS} = -\frac{3m}{4q} + \sqrt{\left(\left(\frac{3m}{4q}\right)^2 + \frac{12 \cdot W \cdot \sigma_{max}}{9{,}81 \cdot 10^3 \cdot q \cdot i}\right)}$	Cont. support with equal field length (individual mass in each field)	4
D		q + Single load in particular field only	$f_D = \frac{l_{DF}^3 \cdot 9{,}81 \cdot 10^6}{384\,EI} \cdot (q \cdot l_{DF} + 6{,}1\,m)$	$l_{DS} = -\frac{126\,m}{265\,q} + \sqrt{\left(\left(\frac{126m}{265q}\right)^2 + \frac{12 \cdot W \cdot \sigma_{max}}{9{,}81 \cdot 10^3 \cdot q \cdot i}\right)}$	$\frac{m}{q} <= 0{,}38\ l^*$ $l^* = \sqrt{\frac{12 \cdot W \cdot \sigma}{9{,}81 \cdot 10^3 \cdot q \cdot i}}$	3
E		q + Single load in particular field only	$f_E = \frac{l_{EF}^3 \cdot 9{,}81 \cdot 10^6}{384\,EI} \cdot (q \cdot l_{EF} + 6{,}1\,m)$	$l_{ES} = -\frac{543m}{265q} + \sqrt{\left(\left(\frac{543m}{265q}\right)^2 + \frac{24 \cdot W \cdot \sigma_{max}}{9{,}81 \cdot 10^3 \cdot q \cdot i}\right)}$	$\frac{m}{q} > 0{,}38\ l^*$ $l^* = \sqrt{\frac{12 \cdot W \cdot \sigma}{9{,}81 \cdot 10^3 \cdot q \cdot i}}$	2

$$I = \frac{\pi}{64}\left(d_a^{\,4} - d_i^{\,4}\right)\left[mm^4\right];\ W = I\frac{2}{d_a}\left[mm^3\right];\ E\left[kN/mm^2\right]$$

Q.4 Symbols

d_{Am}	[mm]	mean diameter of branch
d_m	[mm]	mean diameter of pipe
d_a	[mm]	external diameter of pipe
d_i	[mm]	internal diameter of pipe
f	[mm]	deflection
l^*	[m] = m/q*	equivalent length
i	[-]	stress concentration factor
l	[m]	support spacing, cantilever length (general)
m	[kg]	additional (single) mass
q	[kg/m]	mass relative to length
s	[mm]	nominal wall thickness
v	[-]	weld efficiency
x	[-] = l/L	ratio of length with/without additional mass
y	[-] = l^*/L	ratio of equivalent length/length without additional mass
DN		nominal diameter
E	[kN/mm^2]	modulus of elasticity at calculation temperature
F	[N] = m · g	single load
I	[mm^4]	moment of inertia
f_h	[N/mm^2]	allowable stress at maximum metal temperature according to 12.1.3
L	[m]	length without additional mass
W	[mm^3]	section modulus
ρ	[kg/m^3]	density
σ_{max}	[N/mm^2]	maximum allowed bending stress due to weight
g	$\left[\frac{m}{s^2}\right]$	acceleration due to gravity

Q.5 Indices f_L

A, B, C, D, E	reference to cases in Table Q.3
F, S	reference to deflection/stress criterion
*	Parameter deviating from Table Q.1
-	relative to continuous support

Q.6 Explanatory notes to Q.2.2

Q.6.1 Specification of allowable spacing of supports

Q.6.1.1 General

Q.6.1.1.1 Values

The support distances in the Table Q.1 "Allowable spacing of supports of steel pipes" have been determined on the basis of the equations in the Q.3 "Explanatory notes to Table Q.1". The following data have been used for the mass q relative to length:

Medium	ρ_M	= 1 000 kg/m^3
Pipe material	ρ_R	= 7 900 kg/m^3
Thermal insulation	ρ_D	= 120 kg/m^3
Sheet covering	$\rho_S \cdot s_B$	= 10 kg/m^2

Overlaps and fasteners are included. The stiffening effect of the sheet covering has not been taken into account even though under certain circumstances it can be considerable. Additional loads $F = m \cdot g$ are not taken into account in the support distances in Table Q.1.

Q.6.1.1.2 Limitation of deflection – L_1

The support spacing L_1, has been specified in accordance with the "limitation of deflection" criterion. The limit deflection f has therefore been included as follows, from the point of view of avoiding possible "puddle formation":

— where DN ≤ 50 f = 3 mm;

— where DN > 50 f = 5 mm.

The calculation model for L_1 is the single field support, moments-free supported at both ends (case A in the Table Q.3 "Explanatory notes to Table Q.1"). An average value $E \approx 200$ kN/mm^2 has been assumed for the modulus for elasticity.

$$L_1 = l_{AF}\,(f, q, m = 0, E \cdot I) = L_{AF}\,(f, q, E \cdot I)$$

Q.6.1.1.3 Limitation of stress, L_2 to L_6

The support distances L_2, to L_6, have been determined in accordance with the "limitation of stress" criterion. By complying the support spacings L_2 to L_6, the stresses σ due to q at L_2 and L_5 in the undisturbed piping (without a tee) and at L_3 and L_6 in piping with a tee (welded or forged) at the point of maximum moment are limited to σ = MIN (40 N/mm^2; 0,4 f_h)

Q.6.1.2 Single field supports (moments free) L_2 to L_4

The support distances in Table Q.1 have been determined using the equation for l_{AS} in Table Q.3. For this, undisturbed piping with a stress concentration factor i = 1 has been assumed for L_2. For L_3, a forged tee with a stress concentration factor $i = 0{,}9/(8{,}8 \cdot s/d_m)^{2/3})$ has been assumed in the centre of the field.

For L_4, a welded tee in accordance with a stress concentration factor $i = 0{,}9/(2\ s/d_m)^{2/3}$ has been assumed in the centre of the field.

$L_2 = l_{AS}\ (\sigma, q, m = 0, W, i = 1) = L_{AS}\ (\sigma, q, W, i = 1)$

$L_3 = l_{AS}\ (\sigma, q, m = 0, W, i = 0{,}9/(8{,}8 \cdot s/d_m)^{2/3})$

$= l_{AS}\ (\sigma, q, W, i = 0{,}9/(8{,}8 \cdot s/d_m)^{2/3})$

$L_4 = l_{AS}\ (\sigma, q, m = 0, W, i = 0{,}9/(2 \cdot s/d_m)^{2/3})$

$= l_{AS}\ (\sigma, q, W, i = 0{,}9/(2 \cdot s/d_m)^{2/3})$

Q.6.1.3 Cantilever beam, L_5 and L_6

The cantilever beam lengths have been determined using the equation for l_{BS} in Table Q.3. For this, undisturbed piping where i = 1 has been assumed for L_5. For L_6, a welded tee in accordance with stress concentration factor $i = 0{,}9/(2 \cdot s/d_m)^{2/3}$) has been assumed.

$L_5 = l_{BS}\ (\sigma, q, m = 0, W, i = 1) = L_{BS}\ (\sigma, q, W, i = 1)$

$L_6 = l_{BS}\ (\sigma, q, m = 0, W, i = 0{,}9/(2s/d_m)^{2/3})$

$L_6 = l_{BS}\ (\sigma, q, W, i = 0{,}9/(2s/d_m)^{2/3})$

Q.7 Conversion of the allowable lengths

Q.7.1 Other support conditions

The support distances L_1 to L_4 are based on a moments-free fixed single-field support from case to case. The assumption of a centre field of a continuous support is frequently more realistic. For this support condition, the allowable support distance L^*_1 to L^*_4 can be derived from L_1 to L_4 as follows:

$$L^*_1 = \sqrt[4]{5} \cdot L_1 \sim 1{,}5 \cdot L_1$$

$$L^*_i = \sqrt{1{,}5} \cdot L_i \sim 1{,}225 \cdot L_i \qquad \text{(i = 2, 3 and 4)}$$

Q.7.2 Other parameters

If the moment of inertia I^* and the section modulus W^*, the uniformly distributed load q^*, the modulus of elasticity E^*, the default values f^* and σ^* or the stress concentration factor i^* deviate substantially, the allowable support distances or cantilever beam lengths can be derived from the lengths in Table Q.1.

Where the deflection is limited, the following applies:

$$L_1^* = \sqrt[4]{\frac{I^*}{I} \cdot \frac{E^*}{E} \cdot \frac{q}{q^*} \cdot \frac{f^*}{f}} \cdot L_1$$

Where the stress is limited, the following applies:

$$L_i^* = \sqrt{\frac{W^*}{W} \cdot \frac{q}{q^*} \cdot \frac{\sigma^*}{\sigma} \cdot \frac{i}{i^*}} \cdot L_i \qquad \text{(i = 2, 3, 4, 5 and 6)}$$

For other support conditions, the allowable lengths L* can be correspondingly converted from the length L as specified in Q.7.1.

Q.8 Additional single loads

Q.8.1 General

Single loads which are to be considered in addition to the uniformly distributed loads can be taken into account in cases L_1 to L_6 using the equations given in Table Q.3. The support spacings or cantilever beam lengths for the "stress limitations" criterion can also be determined in this case with the aid of Tables Q.1 and Q.6.1.3.

For this, the single load with $l^* = \frac{m}{q^*}$ is converted into an q equivalent length l^*. The relevant support distance or cantilever beam length without a single load is then determined in accordance with Table Q.1 - or by using the appropriate equations from Table Q.3. The value $x = l/L$ is taken from Figure Q.1 as a function of $y = l^*/L$. The allowable support distance taking account of the additional single load where $F = m \cdot g$ is then obtained as follows:

$$l = x \cdot L$$

If the parameters deviate from those used as a basis in Table Q.1, this deviation shall first be taken into account in accordance with Q.7.1 and then the influence of the single load is considered in accordance with Q.8.1.

EXAMPLE

DN 150 piping with s = 7,1 mm is designed as a continuously supported piping over several supports. The meter mass (mass per unit length) of the piping with filling q* = 60 kg/m.

In a centre field, a pipe branches off so that an additional mass m = 250 kg acts on this field. Assuming that the branch fitting is forged, then $i/i^* \approx 2{,}7$. As a result of the high operating temperatures, the stress σ^* should be limited to 30 N/mm^2.

A support distance L_4 = 4,2 m is obtained from the support distance table where q = 57,8 [kg/m].

$$L = L_4^* = \sqrt{\frac{W^*}{W} \cdot \frac{q}{q^*} \cdot \frac{\sigma^*}{\sigma} \cdot \frac{i}{i^*}} \cdot L_4 \cdot 1{,}225$$

$$= \sqrt{1 \cdot \frac{57{,}8}{60} \cdot \frac{30}{40} \cdot 2{,}7} \cdot 4{,}2 \cdot 1{,}225 = 7{,}2 \text{ m}$$

$$l^* = \frac{m}{q^*} = \frac{250}{60} = 4{,}17 m$$

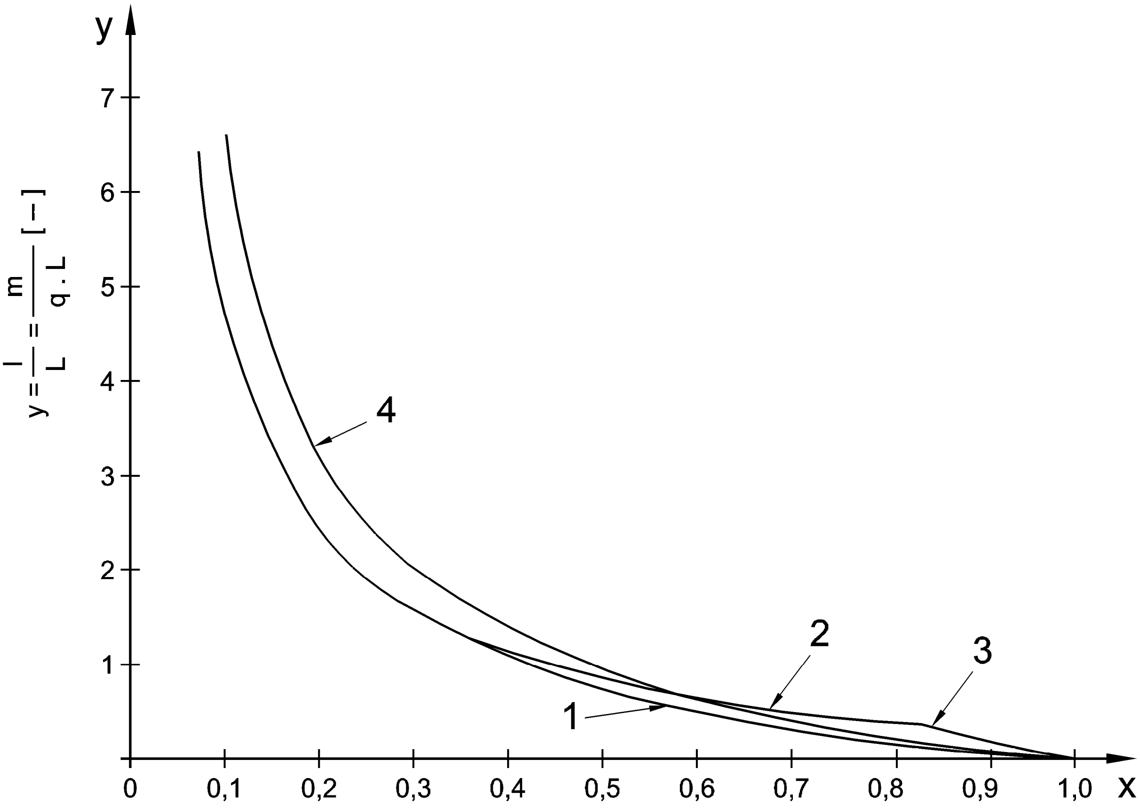
$$y = \frac{l^*}{L} = \frac{4{,}17}{7{,}2} = 0{,}58 > 0{,}38 \qquad \text{Curve 2 from Figure Q.1}$$

A value x = 0,65 is obtained with curve 2 for y = 0,58 from Figure Q.1 "Diagram taking account of single loads on the basis of the allowable stress".

The allowable support distance is as follows:

$$l = x \cdot L = 0{,}65 \cdot 7{,}2 = 4{,}7 \text{ [m]}$$

The deflection can be determined in accordance with case E in the Table Q.3 with $l_{requ.}$ = l_{EF} = 4,7 m and q = q*.



Key

1 beam on 2 supports and cantilever arm: $y = \frac{1-x^2}{2x}$

2 continuous beam with single mass: $y = \frac{265\left(2-x^2\right)}{1086x}$ for y > 0,380

3 continuous beam with single mass: $y = \frac{265\left(1-x^2\right)}{252x}$ for $y \leq$ 0,380

4 continuous beam with single mass in each field: $y = \frac{2\left(1-x^2\right)}{3x}$

max. bending moment:
in centre of field for 1 and 2
in support area for 3 and 4

$x = \frac{l}{L}$ (reduction factor)

Figure Q.1 — Diagram taking account of single load, starting from the allowable stress

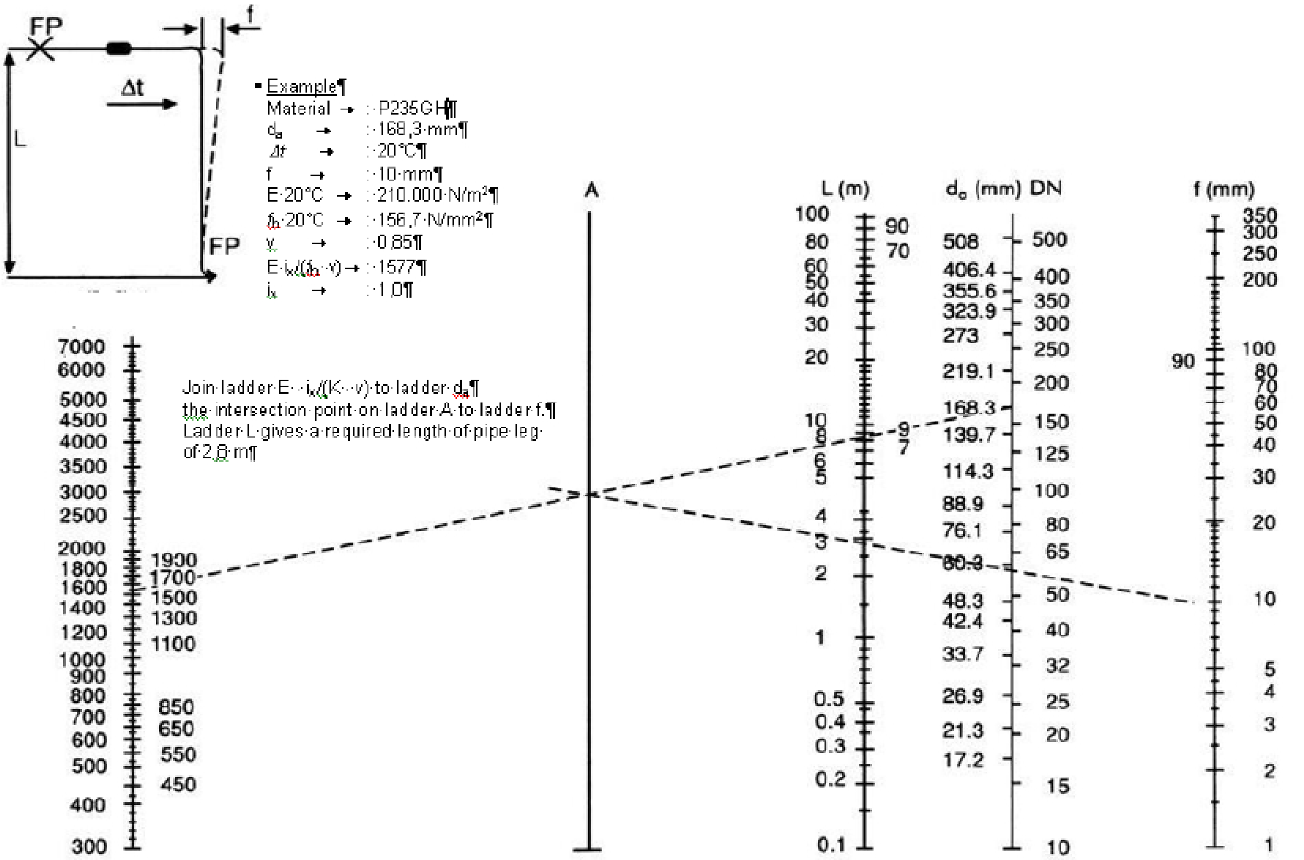

FP
f
Δt
L
FP
Example¶
Material → : P235GH¶
d_a → : 168,3 mm¶
Δt → : 20°C¶
f → : 10 mm¶
E 20°C → : 210.000 N/m²¶
f_h 20°C → : 156,7 N/mm²¶
v → : 0,85¶
E i_x/(f_h · v) → : 1577¶
i_x → : 1,0¶
Join ladder E · i_x/(K · v) to ladder d_a¶
the intersection point on ladder A to ladder f.¶
Ladder L gives a required length of pipe leg of 2,8 m¶
A
L (m)
d_a (mm)
DN
f (mm)
7000
6000
5000
4500
4000
3500
3000
2500
2000
1800
1600
1400
1200
1000
900
800
700
600
500
400
300
1900
1700
1500
1300
1100
850
650
550
450
100
80
60
50
40
30
20
10
8
6
5
4
3
2
1
0.5
0.4
0.3
0.2
0.1
90
70
9
7
508
406.4
355.6
323.9
273
219.1
168.3
139.7
114.3
88.9
76.1
60.3
48.3
42.4
33.7
26.9
21.3
17.2
500
400
350
300
250
200
150
125
100
80
65
50
40
32
25
20
15
10
350
300
250
200
100
90
80
70
60
50
40
30
20
10
5
4
3
2
1

Figure Q.2

Q.9 Explanatory note on Figure Q.2

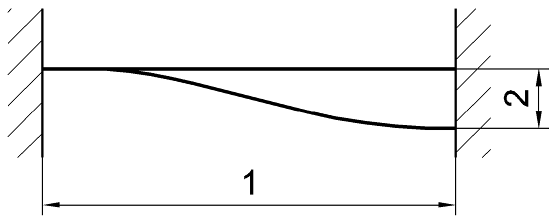
Q.9.1 General

Determination of the required compensating length for thermal expansion as a result of temperature for the nominal size range DN 10 to DN 500 with monogram.

Variables E, f_h, v and d_a

Monogram structure: pipe clamped on both sides

without bend



Key

1 pipe leg length L

2 displacement

Figure Q.3

The nomogram is based on the pipe clamped on both sides as a static system.

The effect of pipe bends on the stress has been taken into account by the stress concentration factor as specified in EN 13480-3, Annex H. It has been incorporated in the nomogram.

Branches can be covered by the nomogram by including the ratio of the pipe bend/branch stress concentration factors – called the reduction factor i_x - in the calculation.

E = modulus of elasticity [N/mm^2]

f_h = allowable stress at maximum metal temperature according to 12.1.3 [N/mm^2]

i_x = reduction factor

i_x = 1,0 for pipe bends with $R \geq 1{,}5 \cdot D$

i_x = 2,1 for welded pipe branches with same wall thickness/diameter ratio

v = weld efficiency

f = thermal expansion to be compensated [mm]

$$f = 10^3 \cdot L \cdot \alpha \cdot \Delta t$$

L = pipe leg length [m]

α = linear expansion coefficient [K^{-1}]

Δt = temperature difference [K]

d_a = external diameter of pipe [mm]

d_i = internal diameter of pipe [mm]

DN = nominal diameter

$$M = \frac{6 \cdot E \cdot I \cdot f}{L^2} \qquad [1]$$

$$M = f_h \cdot v \cdot W$$

$$W = \frac{\pi}{32} \cdot \frac{\left(d_a^4 - d_i^4\right)}{d_a}$$

$$I = \frac{\pi}{64} \cdot \left(d_a^4 - d_i^4\right)$$

$$L = \sqrt{\left(\left(\frac{3 \cdot E \cdot d_a \cdot f \cdot i_x}{10^6 \cdot f_h \cdot v}\right)\right)}$$

If the thermal expansion f is compensated by more than one pipe leg, the respective pipe leg lengths L_1, L_2, ..., L_i shall be added together to form an equivalent pipe leg length L* as follows for applying the monogram:

$$L^* = \sqrt{\left(L_1^2 + L_2^2 + \ldots + L_i^2\right)}$$

This procedure is explained in more detail in the following examples 1, 2 and 3.

EXAMPLE 1: Piping expansion in two directions

Determination of pipe leg lengths

Material:	P235GH
d_a	168,3 mm
Δt : :	200 °C
L	12,3 m
f_1	30 mm from L
E 200 °C	191 000 N/mm^2
f_h 200 °C	123,3 N/mm^2
v	0,85
$E \cdot i_x/(f_h \cdot v)$	1 822
α	$12,2 \cdot 10^{-6}$ K^{-1}
i_x	1,0

FP = anchor

FL = guide

LL = vertical stop

$f = 10^3 \cdot L \cdot \alpha \cdot \Delta t$

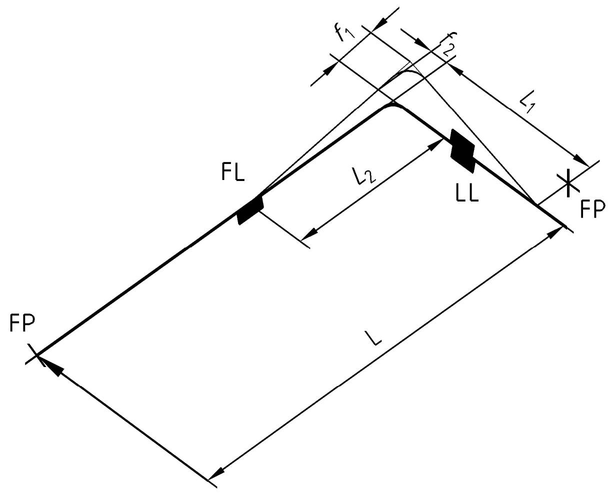


Figure Q.4

Q.9.2 Required pipe leg length L_1, for f_1 from the nomogram

Join ladder $E \cdot i_x/(f_h \cdot v)$ to ladder d_a, then the intersection point of ladder A to ladder f. Ladder L gives a required pipe leg length of L_1 = 5,3 m.

Q.9.3 Required pipe leg length L_2, for f_2 from the nomogram

Q.9.3.1 General

Expansion f_2 = 13 mm from L_1

Join the intersection point of ladder A to ladder f. Ladder L gives a required pipe leg length of $L_{requ.}$= 3,5 m.

EXAMPLE 2: Piping expansion in three directions

Check of existing pipe leg lengths

Material:	P235GH
d_a	168,3 mm
Δt	200 °C
L_1	9,4 m
f_1	23 mm from L_1
L_2	3 m
f_2	7,3 mm from L_2
L_3	7,5 m
f_3	18 mm from L_3
f_4	12 mm from expansion vessel
L_4	2,5 m
L_5	3,5 m
L_6	3,4 m

E 200 °C	191 000 N/mm^2
f_h 200 °C	123,3 N/mm^2
v	0,85
$E \cdot i_x/(f_h \cdot v)$	1 822
i_x	1,0
α	$12{,}2 \cdot 10^{-6}$ K^{-1}

FP = anchor

FL = guide

LL = vertical stop

$f = 10^3 \cdot L \cdot \alpha \cdot \Delta t$

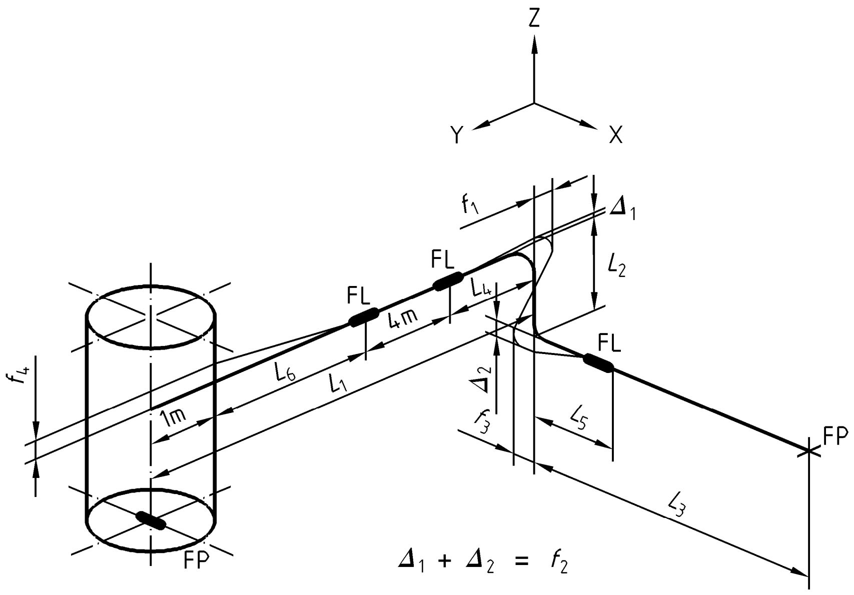


Figure Q.5

Q.9.3.2 Required pipe leg length for f_1 from the nomogram

Join ladder ($E \cdot i_x/(f_h \cdot$v) to ladder d_a, then the intersection point of ladder A to ladder f (f_1), ladder L gives a required pipe leg length of $L_{requ.}$= 4,6 m.

$$L^*_{avail.} = \sqrt{L_2^2 + L_5^2} = 4{,}6 = L_{requ.}$$

Q.9.3.3 Required pipe leg length for f_2 from the nomogram

Join the intersection point of ladder A to ladder f (f_2), ladder

$$L^*_{avail.} = \sqrt{L_4^2 + L_5^2} = 4{,}3 = L_{requ.}$$

Q.9.3.4 Required pipe leg length for f_2 from the nomogram

Join the intersection point of ladder A to ladder f (f_2), ladder, gives a required pipe leg length of $L_{requ.}$= 2,6 m.

$$L^*_{avail.} = \sqrt{L_4^2 + L_5^2} = 4{,}3 > L_{requ}$$

Q.9.3.5 Required pipe leg length for f_3 from the nomogram

Join the intersection point of ladder A to ladder f (f_3), ladder L gives a required pipe leg length of $L_{requ.}$= 4 m

$$L^*_{avail.} = \sqrt{L_2^2 + L_4^2} = 3{,}9 \cong L_{requ.}$$

Q.9.3.6 Required pipe leg length for f_4 from the nomogram

Join the intersection point of ladder A to ladder f (f_4), ladder L gives a required pipe leg length of $L_{requ.}$ = 3,4 m

$$L_{avail.} = 3{,}4 \text{ m} = L_{requ.}$$

EXAMPLE 3: Piping expansion in three directions

Check of existing pipe leg lengths

Material	P235GH
d_a	168,3 mm
Δt	200 °C
L_1	7 m
f_1	17 mm from L_1
L_2	3,5 m
f_2	8,5 mm from L_2
L_3	7 m
f_3	17 mm from L_3
L_4	5 m
f_4	12 mm from L_4
L_5	5 m
f_5	12 mm from L_5
L_6	4,5 m
L_7	5,0 m
E 200 °C	191 000 N/mm^2
f_h 200 °C	123,3 N/mm^2
α	$12{,}2 \cdot 10^{-6}$ K^{-1}
i_x	2,1
v	0,85
$E \cdot i_x/(f_h \cdot v)$	3 826

FP = anchor

FL = guide

LL = vertical stop

$f = 10^3 \cdot L \cdot \alpha \cdot \Delta t$

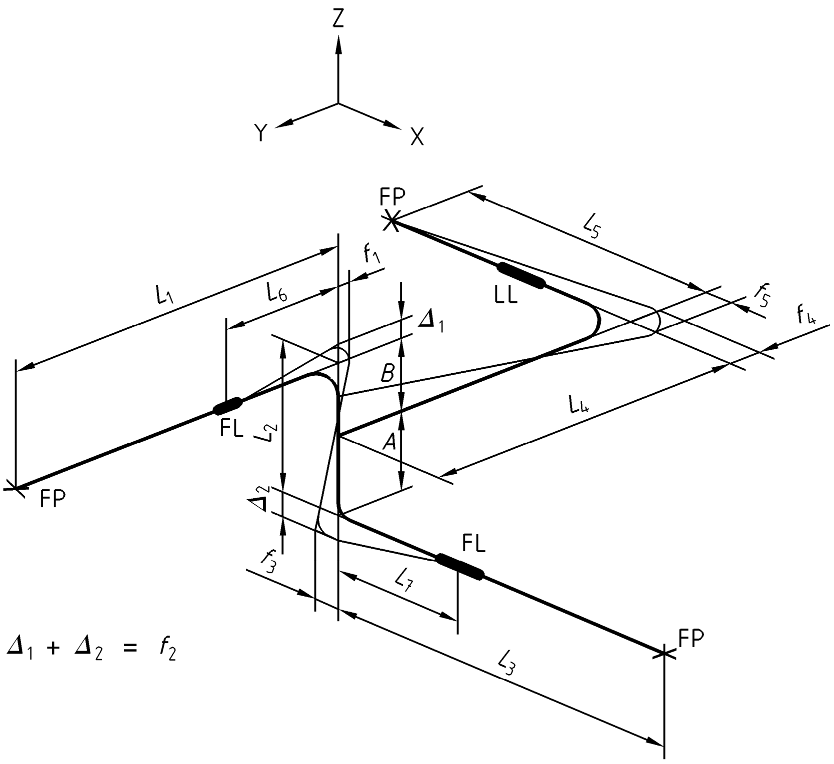


Figure Q.6

Q.9.3.7 Required pipe leg length for f_1 from the nomogram

Join ladder $E \cdot i_x/(f_h \cdot v)$ to ladder d_a, then the intersection point of ladder A to ladder f (f_1), ladder L gives a required pipe leg length of $L_{requ.}$ = 5,7 m.

$$L^*_{avail.} = \sqrt{L_2^2 + L_7^2} = 6{,}1 \text{ m} > L_{requ.}$$

For the example of f_1, the required elasticity for the displacement f_1 results if the whole length L_2 is effective in addition to L_7. This can be achieved by having as low flexural rigidity as possible for L_5 compared to L_2. The flexural rigidity depends approximately to the cube power on the pipe length. In this case, the leg L_5 has only 1/3 of the rigidity of leg L_2. Therefore, the requirement can be regarded as having been met.

Q.9.3.8 Required pipe leg length for f_2 from the nomogram

Join the intersection point of ladder A to ladder f (f_2), ladder L gives a required pipe leg length of $L_{requ.}$ = 4,1 m

$$L^*_{avail.}. = \sqrt{L_6^2 + L_7^2} = 6{,}7 \text{ m} > L_{requ.}$$

Because of the great length, the branching pipe L_4, L_5 does not represent any notable hindrance to expansion for f_2.

Q.9.3.9 Required pipe leg length for f_3 from the nomogram

Join the intersection point of ladder A to ladder f (f_3), ladder L gives a required pipe leg length of $L_{requ.}$ = 5,7 m

$$L^*_{avail.} = \sqrt{L_2^2 + L_6^2} = 5{,}7\text{ m} = L_{requ.}$$

The explanatory notes on f_1 are applicable here for uncoupling the branching pipe.

Q.9.3.10 Required pipe leg length for f_4 from the nomogram

Join the intersection point of ladder A to ladder f (f_4), ladder L gives a required pipe leg length of $L_{requ.}$ = 4,8 m

$$L_{avail.} = L_5 = 5\text{ m} > L_{requ.}$$

Q.9.3.11 Required pipe leg length for f_5 from the nomogram

Join the intersection point of ladder A to ladder f (f_5), ladder L gives a required pipe leg length of $L_{requ.}$ = 4,8 m

$$L_{avail.} = L_4 = 5\text{ m} > L_{requ.}$$

Annex Y
(informative)

History of EN 13480-3

Y.1 Differences between EN 13480-3:2002 and EN 13480-3:2012

The 2012 edition of EN 13480 contains the 2002 edition of the standard and all Amendment(s) and correction(s) issued in the meantime.

Significant technical changes include:

— Addition in clause 2 of normative references related to the types of inspection documents for metallic products and qualification of welding procedures for metallic materials.

— Revision of 5.3.2.1 concerning the design conditions related to the time-dependent nominal design stress, of 6.6 related to the bolted flange connections, of clause 8 related to openings and branch connections, of clause 11 related to integral attachments and of clause 13 related to supports.

— Revision of the Annex B related to more accurate calculation of bends and elbows.

— Revision of the Annex H related to the flexibility characteristics, flexibility and stress intensification factors and section moduli of piping components and geometrical discontinuities.

— Revision of the Annex L related to the buckling of linear type supports.

— Revision of the Annex N related to the documentation of supports.

— Addition of the new Annex O related to the alternative method for checking branch connections.

— Addition of the new Annex P related to the bolted flange connections (application of the European Standard concerning the calculation method for the design rules for gasketed circular flange connections).

— Addition of the new Annex Q related to the simplified pipe stress analysis.

— Revision of the Annexe ZA in relation with the Pressure Equipment Directive 97/23/EC.

NOTE The changes referred include the significant technical changes but is not an exhaustive list of all modifications.

Y.2 List of corrected pages of Issue 2 (2013-08)

Pages 9, 10, 246 and 406.

Y.3 List of corrected pages of Issue 3 (2014-08)

Pages 10, 140, 263, 406 and 407.

Annex ZA
(informative)

Relationship between this European Standard and the essential requirements of EU Directive 97/23/EC

This European Standard has been prepared under a mandate given to CEN by the European Commission to provide a means of conforming to Essential Requirements of the New Approach Directive 97/23/EC of the European Parliament and of the Council of 29 May 1997 on the approximation of the laws of the Member States concerning pressure equipment.

Once this standard is cited in the Official Journal of the European Union under that Directive and has been implemented as a national standard in at least one Member State, compliance with the clauses of this standard given in Table ZA.1 confers, within the limits of the scope of this standard, a presumption of conformity with the corresponding Essential Requirements of that Directive and associated EFTA regulations.

Table ZA.1 — Correspondence between this European Standard and Directive 97/23/EC

Clause(s)/sub-clause(s) of this EN	Essential Requirements (ERs) of Directive 97/23/EC, Annex I	Qualifying remarks/Notes
4.2	2.1-1st paragraph	Proper design using relevant factors
Clause 5	2.1-2nd paragraph	Safety coefficients and margins
Clause 4	2.2.1 to 2.2.3	Design for adequate strength
4.2	2.2.1-1st paragraph	Factors to be taken into account
4.2.2, 4.2.5	2.2.1-2nd paragraph	Simultaneous occurrence of loading
Clauses 6, 7, 8, 9, 11 and Annexes B, D, E, H and O	2.2.3	Calculation method
4.2.3.4	2.2.3 b) – 1st indent	Calculation pressure
4.2.3.5	2.2.3 b) – 2nd indent	Calculation temperature and margins
4.2.3.3	2.2.3 b) – 3rd indent	Combinations of pressure and temperature
Clause 5, 12.3	2.2.3 b) – 4th indent	Maximum stresses and stress concentrations
Clause 9, 12	2.2.3 c)	Stability aspects
4.3	2.6	Corrosion or other chemical attack
4.3	2.7	Wear
12.2	6 a)	Risk of overstressing
Clause 5, 12.3, 13.3	7.1	Allowable stresses
5.2.1	7.1.2-1st indent	Ferritic steels
5.2.2	7.1.2-2nd indent	Austenitic steels
5.2.4	7.1.2-3rd indent	Non-alloy or low alloy cast steel
4.5	7.2	Joint coefficients

WARNING — Other requirements and other EU Directives may be applicable to the product(s) falling within the scope of this European Standard.

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