Safety Standards

of the
Nuclear Safety Standards Commission (KTA)

KTA 3211.2 (2013-11)

Pressure and Activity Retaining Components of Systems Outside the Primary Circuit

Part 2: Design and Analysis

(Druck- und aktivitätsführende Komponenten von Systemen außerhalb des Primärkreises;
Teil 2: Auslegung, Konstruktion und Berechnung)

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KTA SAFETY STANDARD

2013-11	Pressure and Activity Retaining Components of Systems Outside the Primary Circuit; Part 2: Design and Analysis	KTA 3211.2

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PLEASE NOTE: Only the original German version of this safety standard represents the joint resolution of the 35-member Nuclear Safety Standards Commission (Kerntechnischer Ausschuss, KTA). The German version was made public in Bundesanzeiger BAnz of January 17th 2014. Copies may be ordered through the Wolters Kluwer Deutschland GmbH, Postfach 2352, 56513 Neuwied, Germany (Telefax +49 (0) 2631 801-2223, E-Mail: info@wolterskluwer.de).

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Comments by the editor:

Taking into account the meaning and usage of auxiliary verbs in the German language, in this translation the following agreements are effective:

shall	indicates a mandatory requirement,
shall basically	is used in the case of mandatory requirements to which specific exceptions (and only those!) are permitted. It is a requirement of the KTA that these exceptions - other than those in the case of **shall normally** - are specified in the text of the safety standard,
shall normally	indicates a requirement to which exceptions are allowed. However, the exceptions used, shall be substantiated during the licensing procedure,
should	indicates a recommendation or an example of good practice,
may	indicates an acceptable or permissible method within the scope of this safety standard.

Fundamentals

(1) The safety standards of the Nuclear Safety Standards Commission (KTA) have the objective to specify safety-related requirements, compliance of which provides the necessary precautions in accordance with the state of the art in science and technology against damage arising from the construction and operation of the facility (Sec. 7 para. 2 subpara. 3 Atomic Energy Act - AtG) in order to achieve the fundamental safety functions specified in the Atomic Energy Act and the Radiological Protection Ordinance (StrlSchV) and further detailed in the Safety Criteria for Nuclear Power Plants, in the Incident Guidelines and in the Safety Requirements for Nuclear Power Plants.

(2) Criterion 1.1, "Principles of Safety Precautions", of the Safety Criteria requires, among other things, a comprehensive quality assurance for fabrication, erection and operation, and Criterion 2.1, "Quality Assurance", requires, among other things, the application, preparation and observation of design rules, material specifications, construction rules, testing and inspection as well as operating instructions and the documentation of quality assurance.

Further requirements relating to design and condition to be met by the safety systems are specified in Criterion 4.2, "Residual Heat Removal in Specified Normal Operation"; Criterion 4.3, "Residual Heat Removal after Loss of Coolant"; Criterion 5.3, "Equipment for the Control and Shutdown of the Nuclear Reactor"; and Criterion 8.5, "Heat Removal from the Containment".

Safety Standards KTA 3211.1 to KTA 3211.4 are intended to specificy detailed measures which shall be taken to meet these requirements within the scope of their application. For this purpose, a large number of standards from conventional engineering, in particular DIN standards, are also used; these are specified in each particular case.

(3) The scope of application as defined in this Safety Standard comprises the pressure and activity-retaining systems and components outside the pressure-retaining boundary (Safety Standards of the KTA 3201 series) which are specifically significant in terms of reactor safety in accordance with the RSK Guidelines (Section 4.2).

(4) KTA 3211.2 specifies the detailed requirements to be met by

a) the classification into test groups, load case classes and level loadings,

b) the design and analysis of components,

c) the calculation procedures and design principles for obtaining and maintaining the required quality of the components,

d) the documents for the certificates and demonstrations to be submitted.

1 Scope

This Safety Standard applies to the manufacture of pressure-retaining walls of pressure and activity-retaining systems and components of light water reactors which are not part of the reactor coolant pressure boundary, are operated up to design temperatures of 673 K (400 °C) and which are specifically significant to reactor safety. This is the case, if one of the following criteria is fulfilled:

a) The plant component is needed to cope with incidents regarding shutdown, maintenance of long-term subcriticality and direct residual heat removal.

   Requirements to be met by components in systems which only indirectly serve residual heat removal, i.e. the non-activity-retaining closed cooling water systems and service cooling water systems, shall be specified in a plant-specific manner, taking into consideration multiple design (e.g. redundancy, diversity).

b) If the plant component fails, great amounts of energy are released and the consequences of failure are not limited by structural measures, physical separation or other safety measures to an extent which is reasonable in terms of nuclear safety.

c) The failure of the plant component may lead, either directly or in a chain of consequential events to be assumed, to an incident as defined in § 49 of the Radiological Protection Ordinance.

(2) The following components fall under the scope of this Safety Standard:

a) pressure vessels,

b) pipes and pipe components,

c) pumps, and

d) valves

including the integral areas of component support structures.

(3) This Safety Standard does not apply to:

a) pipes and valves equal to or smaller than DN 50, but may apply to the performance of stress and fatigue analyses for piping and valves with ≤ DN 50,

   Note:
   Simplified procedures are given in cl. 8.5.1 (5).
   Requirements for instrument lines are laid down in KTA 3507.

b) internals of the components (which are not part of the pressure-retaining wall) and accessories,

c) systems and plant components performing auxiliary functions for the systems dealt with by this safety standard,

d) subsystems whose system pressure is determined exclusively by the static head in the suction area,

e) parts used for the transmission of forces and power in pumps and valves as well as tests to demonstrate functional capability.

2 General requirements and Definitions

2.1 Definitions

(1) Functional capability

Functional capability means the capability of the component beyond the stability and integrity requirements to fulfil the specified task at the respective event.

Regarding functional capability distinction is made whether it is to be ensured during or after the event or during and after the event in which case distinction is also made between active and passive functional capability as well as between active and passive components.

a) Active functional capability ensures that the specified mechanical movements (relative movements between parts) can be made (consideration of the possibility of closing clearances, generating or altering frictional forces).

b) Passive functional capability means that distortions and displacement limits are not exceeded.

c) Active components are components for which mechanical movements are specified to satisfy safety requirements, e.g. pumps, valves. All other components are passive components, e.g. vessels, piping systems.

(2) Integrity

Integrity is the condition of a component or barrier, at which the safety requirements with regard to strength, resistance to fracture and leak tightness are met.

(3) Stability

Stability means the safety against inadmissible changes in position and location of installation (e.g. overturning, fall, inadmissible displacement).

2.2 General requirements

(1) For the components a classification regarding test groups and materials shall be made in dependence of the design data and dimensions, taking the specified materials and stress limits into account. When classifying components, different test groups may be selected for components within a system, among certain circumstances also for parts of a component.

(2) The allowable classification regarding test groups and materials is to be made in accordance with **Table 2-1**.

(3) The stress intensity value in test group A1 is S_m. In test groups A2 and A3 the stress intensity value is S.

(4) For the load cases of the total plant or system the loadings on the component shall be indicated in accordance with Section 3 and be classified into operational load cases A, B, P, C, D in dependence of the safety-criteria to be satisfied. For the purpose of dimensioning the effective cross-sections, a pertinent load case shall be determined from these loadings which shall be assigned to design loading (Level 0). These data shall be specified for each component to form the basis for design and calculation (see Section 4).

<table>
<tr><th rowspan="3">Test group</th><th colspan="2">Classification Criteria</th><th colspan="5">Allocation of Materials</th></tr>
<tr><th rowspan="2">Design stress intensity</th><th rowspan="2">Size limitation</th><th colspan="3">Ferritic Materials</th><th colspan="2">Austenitic Materials</th></tr>
<tr><th colspan="2">Materials in accordance with KTA 3211.1</th><th>Materials under the scope of AD 2000-Merkblatt W 0</th><th>Materials in accordance with KTA 3211.1</th><th>Materials 1.4550, 1.4580, 1.4541, 1.4571 under the scope of AD 2000-Merkblatt W 0</th></tr>
<tr><td>A 1</td><td>S_m</td><td>—</td><td>W I</td><td rowspan="4">(1) W II for:
- PG 1-small items
- integral supports
(2) Materials for special application upon specific agreement</td><td rowspan="4">Materials where $R_{p0.2RT} \leq 370$ N/mm² for PG 1-small items</td><td rowspan="4">Permitted for all dimensions</td><td rowspan="4">for:
a) PG 1-small items,
b) integral supports</td></tr>
<tr><td rowspan="2">A 2</td><td rowspan="2">S</td><td>—</td><td>W I
$R_{p0.2RT}$
≤ 370 N/mm²</td></tr>
<tr><td>for vessels:
s ≤ 16 mm
for pipes, pumps, valves:
≤ DN 150 1)</td><td rowspan="2">W II</td></tr>
<tr><td>A 3</td><td>S
in addition:
P_{mNB}
≤ 50 N/mm²</td><td>—</td></tr>
<tr><td colspan="8">(1) These test groups are based on the same basic safety (see RSK Guidelines) in accordance with the varying hazard potential (stress, dimensions), observing the materials used.
(2) Components within a system and subunits within a component may be allocated to different test groups.
1) In the case of pumps: nominal diameter of the largest pressure nozzle
In the case of valves: nominal diameter of the inlet nozzle.</td></tr>
</table>

Table 2-1: Test Groups: Classification criteria and allocation of materials

(5) With respect to the safety criteria to be satisfied by the component the stability, structural integrity and functional capability shall be verified as explained hereinafter.

a) Stability of the component

Stability is mainly proved by a verification of strength of the support, in which case the connection of the support to the component and the anchorage (support, component) shall be taken into account.

b) Structural integrity of the component

When verifying the structural integrity the generally accepted verification procedures shall be used and it shall be proved for the part or component that they are capable of withstanding the loadings occurring during their service life.

When verifying the structural integrity, the stability of the component and, where required (e.g. in case of flanged joints) the leak tightness shall also be taken into account.

c) Functional capability of the component

When verifying the functional capability it shall be proved for the part or component that the required distortion limits for the pressure-retaining walls are satisfied with regard to the loadings occurring during the service life.

Note:

This safety standard only considers the requirements for pressure retaining walls for safeguarding the functional capability of the component.

(6) Components shall be designed in accordance with the rules of Section 5 "Design" according to which component-specific design rules shall be considered in addition to the general rules. The use of other designs is subject to specific verifications.

(7) The loadings applied on the component shall be evaluated in dependence of the level loadings and be limited in accordance with Sections 6 to 8.

(8) The service limits given in Section 7 apply to loadings that were determined on the basis of linear elastic material laws, unless other stipulations are made in the following Sections.

(9) All level loadings specified in Sections 6 and 7 were fixed from the viewpoint of strength calculation and quality assurance such that a comparable safety is obtained for test groups A1, A2 and A3.

(10) The extent of the required verifications of strength is laid down in dependence of the test groups. The verifications of strength for test group A1 shall be performed by way of dimensioning according to Section 6 and as general analysis of the mechanical behaviour according to Section 7 or as component-specific analysis of the mechanical behaviour according to Section 8. Alternately, the use of equivalent design formulae is permitted (see clause 7.1.1 (6)). For test groups A2 and A3 dimensioning in accordance with Section 6, or if required with Section 8, and depending on the individual case, a simplified analysis according to Section 7 or 8 shall be performed.

(11) The components of test groups A2 and A3 shall be designed and calculated under the same considerations in which case the calculation shall include a verification that the laws of equilibrium of external fores are adhered to and a fatigue evaluation (in test group A3 only for piping systems).

(12) Within the verification of strength the stability of the structure shall also be proved, if required.

(13) The verifications of strength, stability and functional capability required according to this safety standard shall be made by way of calculation or experiments.

(14) Component-specific rules for dimensioning are contained in Annex A and, depending on the respective case, Section 8.

3 Load case classes as well as design, service and test loadings and limits of components

3.1 General

(1) Loadings on the systems resulting from the operation of systems and the events occurring in the total plant shall be listed according to their physical occurrence and chronological order to form load cases which then shall be classified into the load case classes described in 3.2 with respect to their importance for the whole plant and adherence of protective goals. Corresponding to the load case classes the loadings resulting from the load cases will be assigned to the level loadings described in section 3.3 for the individual components of a system.

(2) Connected to these level loadings are criteria for deciding whether to further operate the components or on the measures to be taken in the event of the respective load case occurring (see Section 3.3).

(3) Where loadings of considerable extent arise due to other load cases (e.g. transport, assembly and repair cases) they shall be verified by means of a strength calculation. The allowable service limits shall be determined for each individual case.

3.2 Load case classes

3.2.1 General

(1) Load case means a condition or change in condition of a system which lead to loadings on the various components. The design, dimensioning, and analysis of the mechanical behaviour of the components and parts shall be based on exact or conservative values.

(2) The load cases are assigned to one of the load case classes described hereinafter with respect to their importance for the whole plant and adherence to the protective goals.

3.2.2 Design load cases (AF)

Design load cases are considered to be load cases which cover the normal operational load cases (NB) according to clause 3.2.3 as far as they cause maximum primary stresses in the components or parts.

3.2.3 Specified operation

3.2.3.1 Normal operational load cases (NB)

Normal operational load cases are operating conditions or changes in operating conditions intended for the plant with the systems being in a functionally fit condition. They especially comprise start-up of the system, full-load operation, part-load operation, and shutdown of the system, examinations of functional capability including the transients occurring during these load variations.

3.2.3.2 Anomalous operational load cases (AB)

Anomalous operational load cases refer to deviations from the normal operating load cases which are caused by functional disturbance or control error of the component or adjacent components. There are no objections to continue the operation after such load cases.

3.2.3.3 Test load cases (PF)

These load cases comprise the first pressure test (component and system pressure test) as well as periodic pressure and leakage tests.

3.2.4 Incidents

3.2.4.1 General

Incidents are deviations from specified operation in the event of which the operation of the plant cannot be continued for safety reasons and for which the plant is designed.

3.2.4.2 Emergencies (NF)

Emergencies are incidents having very little probability of occurrence.

3.2.4.3 Accidents (SF)

Accidents are incidents having an extremely little probability of occurrence, or are postulated load cases.

3.3 Loadings levels

3.3.1 General

Distinction shall be made between the various loading levels of the components with regard to a continuation of operation and the measures to be taken where the fulfilment of the safety criteria (stability and structural integrity) will be ensured at each loading level if the loading limits specified in Sections 6 to 8 have been adhered to.

3.3.2 Design loading (level 0)

3.3.2.1 General

(1) Loadings covered by design load cases (AF) shall be assigned to Level 0. Level 0 loadings therefore are due to the effect of design pressure and additional design mechanical loads so that the maximum primary stresses resulting from the load cases under Level A according to clause 3.3.3.2, including the pertinent stability cases in the components and their parts are covered. The load case data comprise the design pressure (see clause 3.3.2.2), the design temperature (see clause 3.3.2.3) and additional design loads (see clause 3.3.2.4).

Note:

(1) The loading limits of Level 0 are fixed such that the loadings generate equilibrium with the external mechanical loads in such a manner that neither deformation nor fast fracture occurs if the required safety factors are considered.

(2) In component specifications and design data sheets the term "maximum allowable working pressure" is used in lieu of "design pressure" and "allowable operating temperature" in lieu of "design temperature".

3.3.2.2 Design pressure

(1) The design pressure to be specified for a component or part shall be not less than the maximum difference in pressure between the pressure-loaded surfaces according to Level A (see clause 3.3.3.2).

(2) For parts where the pressure on the inside is independent from the pressure on the outside, the largest value of the values indicated hereinafter shall be taken as the design pressure:

a) maximum difference between internal and atmospheric pressure,

b) maximum difference between external and atmospheric pressure to take the stability behaviour into account,

c) maximum difference between internal and external pressure to take the stability behaviour into account.

(3) For parts where the pressure on the inside depends on the pressure on the outside, the design pressure shall be the maximum pressure difference.

(4) Hydrostatic pressures shall be taken into account if they exceed 5 % of the design pressure.

(5) It is assumed that safety valves and other safety devices are designed and set such that the pressure, in the case of anomalous operation, does not exceed the design pressure until release of the full discharge area and during discharge operation by more than 10 %. In this case, however, the Level B service limits (see clause 3.3.3.3) shall be satisfied.

(6) Where a part is selected on account of a certain nominal pressure rating PN, the nominal pressure shall be at least equal to the design pressure. Further conditions if required, shall be taken from this safety standard or the respective standard of the part.

3.3.2.3 Design temperature

The design temperature is used to determine the design strength values and shall not be less than the highest temperature according to Level A (see clause 3.3.3.2) to be expected in the wall at the point under consideration.

3.3.2.4 Additional design mechanical loads

Additional design mechanical loads shall be selected to be at least so high that, when combined with the design pressure, they cover the simultaneously acting unfavourable primary stresses of Level A service limits.

Note:
In individual cases the loadings under Levels B, C and D may govern the design. These loadings shall be verified taking the respective allowable primary loading into account.

3.3.3 Service levels

3.3.3.1 General

The loadings for the various service limits shall be determined and limited within the analysis of the mechanical behaviour in which case the respective actual loadings and temperatures may be used.

3.3.3.2 Level A

(1) The loadings resulting from normal operational load cases (NB) are assigned to Level A.

(2) It shall be verified in accordance with clause 7.7.3 that the stress intensities and equivalent stress ranges are permitted.

Note:
The loading limits of Level A are fixed such that if the required safety factors are considered, neither deformation nor fast fracture and no progressive deformation nor fatigue occur.

3.3.3.3 Level B

(1) If the loadings are not classified under Level A, the loadings from anomalous operational load cases (AB) shall be assigned to Level B.

(2) For load cases assigned to Level B it shall be verified in accordance with clause 7.7.3 that the stress intensities and equivalent stress ranges are permitted.

(3) Primary stresses need only be verified if the Level 0 design loadings or Level A loadings are exceeded.

Note:
The loading limits of Level B are fixed such that if the required safety factors are considered, neither deformation nor fast fracture and no progressive deformation nor fatigue occur.

3.3.3.4 Level C

(1) If the loadings are not classified under Level B, the loadings from emergencies (NF) shall be assigned to Level C.

(2) Only primary stresses shall be considered within the stress analysis for the load cases assigned to Level C service limits. If the total number of stress cycles of all specified events of Level C for the respective components exceeds 25, the stress cycles exceeding the number of 25 shall be taken into account in the fatigue analysis for the respective component.

Note:
These sets of Level C service limits permit large deformations in areas of structural discontinuity and exclude fast fracture. Where such a case occurs, inspection of the respective component may become necessary.

(3) 120 % of the allowable external pressure according to Level 0 are permitted as external pressure without additional proof of stability. Where the respective verifications are made for Level A, this requirement shall apply accordingly.

3.3.3.5 Level D

(1) If the loadings are not classified under Level B or C, the loadings from accidents (SF) shall be assigned to Level D.

(2) Only primary stresses shall be considered within the stress analysis for the load cases assigned to Level D service limits.

Note:
The limits of this loading level exclude fast fracture. Here, it is accepted that gross general deformations may occur which may necessitate repair or replacement of the respective component.

3.3.3.6 Level P

(1) Test Level P applies to loadings from test load cases (pressure and leakage tests of components)

(2) Only primary stresses shall be considered within the stress analysis for the load cases assigned to Level P service limits. If the number of pressure tests does not exceed 10 they shall not be considered in the fatigue analysis. If the number of pressure tests exceeds 10, all pressure tests shall be considered in the fatigue analysis.

(3) The first pressure test of a component not installed in the system shall be conducted with a test pressure p' of 1.3 times the design pressure for rolled and forged steels, with 1.5 times

the design pressure for cast steel and with 2 times the design pressure for nodular-graphite cast iron.

4 Effects on the components due to mechanical and thermal loadings and fluid effects

4.1 General

(1) All relevant effects on the components due to mechanical and thermal loadings as well as fluid effects shall be taken into account in the design and calculation.

(2) Mechanical and thermal loadings are the effects on the component resulting from the load cases as defined in Section 3. These effects lead to loadings in the component for which the component has to be designed.

(3) Fluid effects on the component may

a) lead to local or large-area wall thinning (corrosion and erosion),

b) reduce the fatigue strength,

c) in connection with stresses, also lead to cracking.

4.2 Mechanical and thermal loadings

(1) Mechanical and thermal loadings comprise forces and moments, imposed deformations and temperature differentials as far as they cause loadings in the components.

(2) Mechanical and thermal loadings are the following:

a) loadings caused by the fluid, e.g. by its pressure, temperature, pressure transients, temperature transients, fluid forces, vibrations,

b) loadings caused by the component itself, e.g. dead weight, cold-spring, deviations from specified shape, internals,

c) loadings imposed by adjacent components, caused e.g. by pipe forces applied due to restraint to thermal expansion,

d) ambient loadings imposed e.g. by anchor displacement, vibrations due to earthquake.

4.3 Documentation of component loadings

All data relevant to design and construction shall be taken from the component-specific documents which, besides the data required for verification of strength, also contain data on the type and installation of the component.

4.4 Superposition of loadings and assignment to loading levels

The superposition of loadings on one component and the assignment of superpositioned loadings to loading levels shall be made for each specific plant and be laid down for each specific component.

Note:

Table 4-1 *gives an example of the combination of component loadings and their classification into loading levels.*

4.5 Fluid effects

(1) Special measures shall be taken to withstand fluid effects by selecting suitable materials, dimensioning, design or stress-reducing fabrication measures (e.g. stress relieving, cladding or deposition welding, avoidance of narrow gaps).

(2) Where uncertainty exists regarding the fluid effect on the structural integrity of the component, this shall be considered by limiting the allowable cumulative usage factor D (see cl. 7.8.3), by operation-simulating experiments or by suitable measures to be taken within operational monitoring and in-service inspections.

Note:

KTA safety standard 3211.4 lays down requirements for operational monitoring and in-service inspections.

Service loading levels	Loadings [1]												
	Static loadings							Transient loadings			Vibration and dynamic loadings		
	Design pressure	Design temperature [2]	Pressure	Temperature [2]	Dead weight and other loads	Mechanical loads, reaction forces	Restraint to thermal expansion	Transient loads (pressure, temperature, mechanical loads), dynamic loading	Anomalous loadings (static and dynamic)	Test loadings (static and dynamic)	Design basis earthquake	Effects from the inside	Other effects from the outside
Level 0	X	X			X								
Level A			X	X	X	X	X	X					
Level B			X	X	X	X	X		X				
Level P			X	X	X					X			
Level C			X	X	X	X							
			X	X	X	X						X	
Level D			X	X	X	X					X		
			X	X	X	X						X	
			X	X	X	X							X

[1] In each load case the type of loadings imposed shall be checked.

[2] To determine the design stress intensity value at the temperature governing the respective loadings.

Table 4-1: Example for the superposition of component loadings and assignment to service loading levels

5 Design

5.1 General requirements

5.1.1 Principles

(1) The design of the components shall

a) meet the functional requirements,

b) not lead to an increase of loadings/stresses,

c) meet the specific requirements of the materials,

d) meet fabrication and inspection and testing requirements,

e) be amenable to maintenance.

(2) The aforementioned general requirements are correlated and shall be harmonized with respect to the component-specific requirements. In this respect, the requirements and examples contained in Sections 5.2 and 5.3 will concretize the basic requirements of Section 5.1.

5.1.2 Design meeting functional requirements and not leading to an increase of loadings/stresses

Components shall be designed and constructed such as to meet the specific functional requirements. The following principles are based hereupon:

a) favourable conditions for component service loadings taking the loadings imposed by the system into account (e.g. actuating, closing, fluid forces, thermal stratification),

b) favourable distribution of stresses, especially in areas of structural discontinuity (nozzles, wall thickness transitions, points of support),

c) avoidance of abrupt changes at wall thickness transitions, especially in the case of components subject to transient temperature loadings (see clause 5.2.6),

d) avoidance of welds in areas of high local stresses,

e) pipe laying at a specified slope.

5.1.3 Design meeting the specific requirements for materials

(1) The following criteria shall be satisfied regarding the selection of materials and the product form:

a) strength,

b) ductility,

c) corrosion resistance,

d) amenability to repair,

e) construction (minimization of fabrication defects),

f) capability of being inspected and tested.

(2) The materials specified by KTA 3211.1 shall be used. For special loadings, such as erosion, corrosion or increased wear, "materials for special use" may be permitted.

(3) The materials shall be used in a product form suitable for the loadings occurring (e.g. plates, forgings, castings, seamless tubes)

(4) The use of dissimilar materials in one component shall be limited to the extent required.

5.1.4 Design meeting fabrication requirements

5.1.4.1 Design meeting manufacture and workmanship requirements

The following principles apply to design meeting manufacture and workmanship requirements:

a) Product forms and wall thicknesses shall be selected to ensure favourable conditions for processing and non-destructive testing.

b) The number of welds shall be minimized accordingly. Welds shall be located such as to consider accessibility during welding (taking heat treatment into account) and minimization of weld residual stresses.

c) Fixing clamps (temporary attachments) shall be welded on ferritic walls by double-layer welds. The final run shall not come into contact with the base metal of the component.

d) The structure shall be so designed that repairs, if any, can be done as simply as possible.

5.1.4.2 Design meeting testing and inspection requirements

(1) The shaping of the parts as well as the configuration and location of the welds shall permit the performance of non-destructive tests with sufficient defect interpretation on product forms, welds and installed components in accordance with KTA 3211.1, KTA 3211.3 and KTA 3211.4. The requirements for test and inspection depending on the type of the procedure used shall be taken from KTA 3211.3, Section 11.1 for each individual case.

(2) The following principles apply to design meeting test and inspection requirements

a) Attachment welds on pressure-retaining walls shall basically be full-penetration welds so that non-destructive testing of the welded joint is possible. Clause 5.2.2.2 (4) defines the permissibility of fillet welds.

b) The structure shall basically be designed such that all accessible welded joints on pressure parts can be machined flush, and attachment welds on pressure retaining walls having a notch-free contour. The surface finish of welded joints is specified in KTA 3211.3, Table 5-4.

c) Single-side welds are permitted if they can be subjected to the non-destructive testing procedures prescribed by KTA 3211.3. The design shall ensure that the conditions of KTA 3211.3 are met.

d) Forgings shall be so designed and constructed that the non-destructive testing (e.g. radiography, surface crack detection) is basically possible also on the inner surface. Only in areas of low nominal operating stress (equal to or less than 50 N/mm^2) restrictions are permitted. Design measures shall be taken to ensure that surface crack-detection is possible on the outside and where accessible on the inside of the part.

5.1.5 Design amenable to maintenance

(1) When designing pressure-retaining walls of components care shall be taken to ensure that they are easily accessible and in-service inspections can be adequately performed.

(2) The following principles shall be observed:

a) Adequate accessibility for maintenance (especially examination, visual inspection, repair or replacement) shall be ensured. The geometries in the areas to be non-destructively tested shall be simple.

b) Adequate accessibility for repairs, if any, shall be ensured taking the radiation protection requirements into account.

c) Activity-retaining components shall be so designed that deposits are avoided as far as possible and decontamination can be performed.

d) Welds in the controlled area shall be located and designed in accordance with the Radiation Protection Ordinance so that setting-up and inspection times for periodic inspections are as short as possible.

e) For components subject to periodic inspection which are not accessible for non-destructive testing, the nominal operating stress shall be limited to a value 50 N/mm^2 or less.

5.2 General requirements for components and their welds

5.2.1 General

Besides the requirements laid down hereinafter additional geometric conditions shall be taken into account when applying special calculation procedures, if any.

5.2.2 Welds

5.2.2.1 Butt welds

Butt weld shall be full-penetration welds. Cruciform joints, weld crossings and built-up weld deposits shall normally be avoided. If the thickness of two parts to be joined by butt welding differs, the thicker part shall be trimmed to a taper extending at least three times the offset between the abutting surfaces; the length of the taper, however, shall not exceed 150 mm. **Figure 5.2-1** shows single-sided weld configurations.

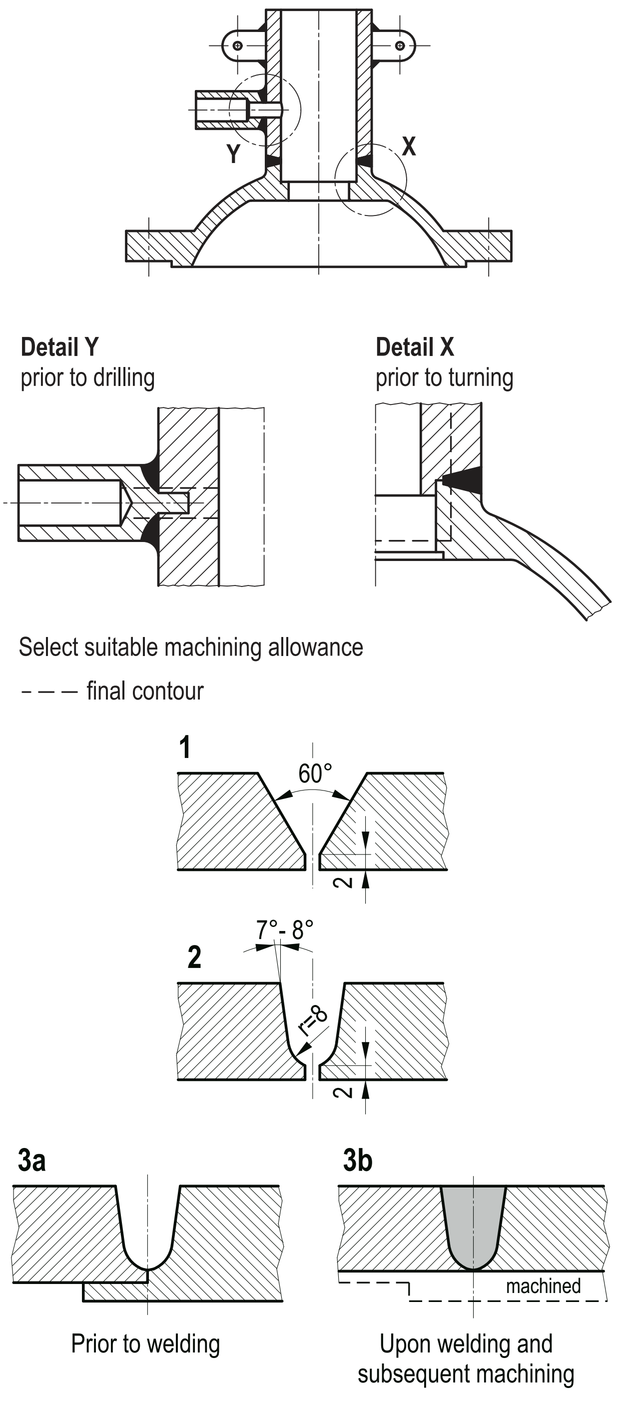


Figure 5.2-1: Examples of single-side butt welds

5.2.2.2 Attachment welds

(1) Attachment welds on pressure-retaining walls shall basically be welded with a length not less than 50 mm. Exceptions to this rule (e.g. pads for piping) are permitted if corresponding verification is made.

(2) Corner joints and welding-over of butt joints are not permitted. To avoid such welding-over, unwelded areas shall be left at the junction of brackets and support lugs, excluding parts with a wall thickness s less than 16 mm.

(3) Double-bevel butt welds and single-bevel butt welds with backing run according to **Figure 5.2-2** are permitted without restriction. Single-bevel butt joints without backing run are permitted in the case of restricted accessibility if the welds are of the full-penetration type and can be subjected to non-destructive testing.

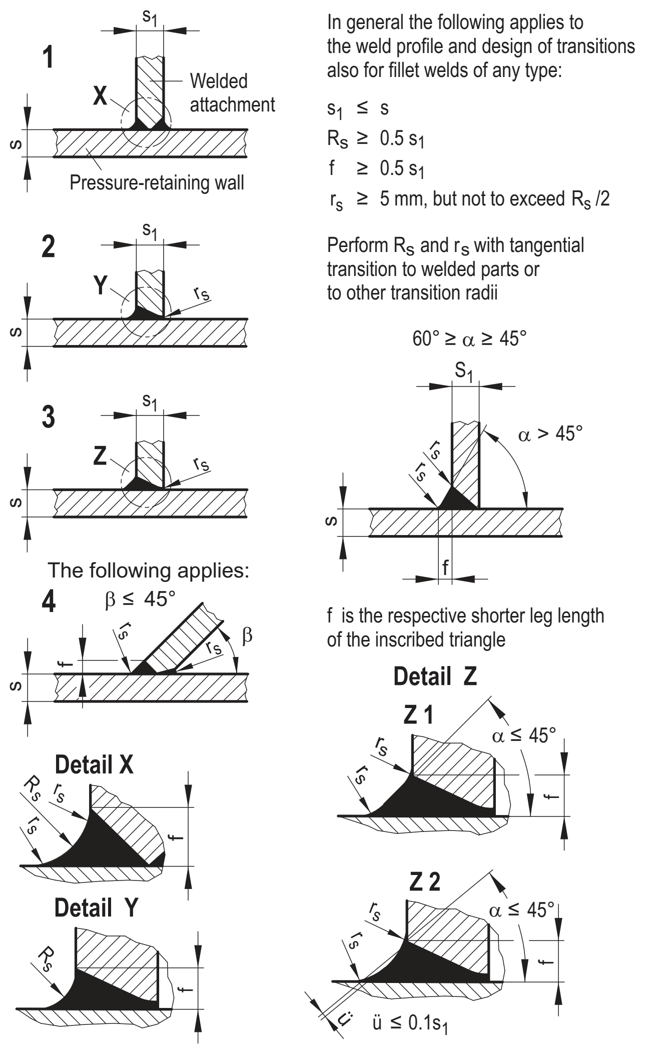


Figure 5.2-2: Examples of single-bevel and double-bevel butt welds for attachment welds

(4) Fillet welds shall be welded over the full circumference and are permitted in the following cases:

a) In test group A2 on the assumption that austenitic or ferritic materials with $R_{p0.2\ RT} \leq 300$ N/mm² are used and predominantly static loading is applied.

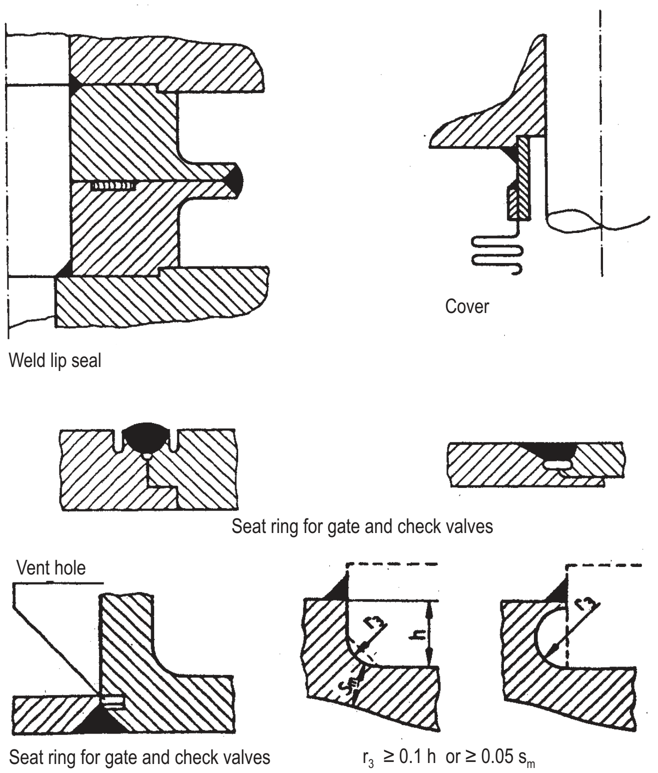
   Fluid-wetted fillet welds in test group A2 are only permitted if, in addition to the above requirements, they are subjected to flush-grinding and undergo an examination of surfaces and, in the case of ferritic materials, the hardness values in the heat-affected zone (HAZ) are clearly (approx. 10%) less than 350 HV 10. This shall be verified during welding procedure qualification.

b) In test group A3.

c) Where full-penetration welds lead to a cleary more favourable design than it would be the case if fillet welds were used.

d) As seal welds (see **Figure 5.2-3**).

e) As attachment welds on austenitic weld claddings.



s_m : minimum wall thickness

Figure 5.2-3: Examples of welds primarily having sealing function

5.2.2.3 Nozzle welds

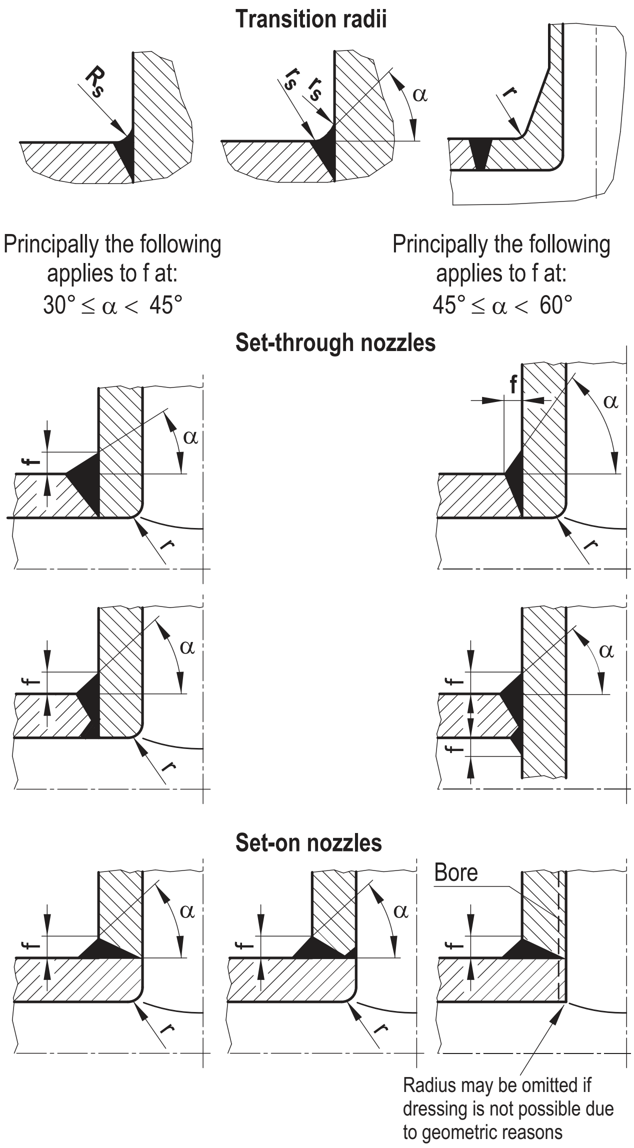
(1) The allowable configurations of welds and welded transitions are shown in **Figure 5.2-4**.

(2) Welded set-in nozzles shall be back welded where possible on account of dimensions. Single-side welds are permitted if the root has been dressed. Where in exceptional cases dressing of the root is not possible it shall be ensured that the weld can be tested.

5.2.3 Transitions in diameter

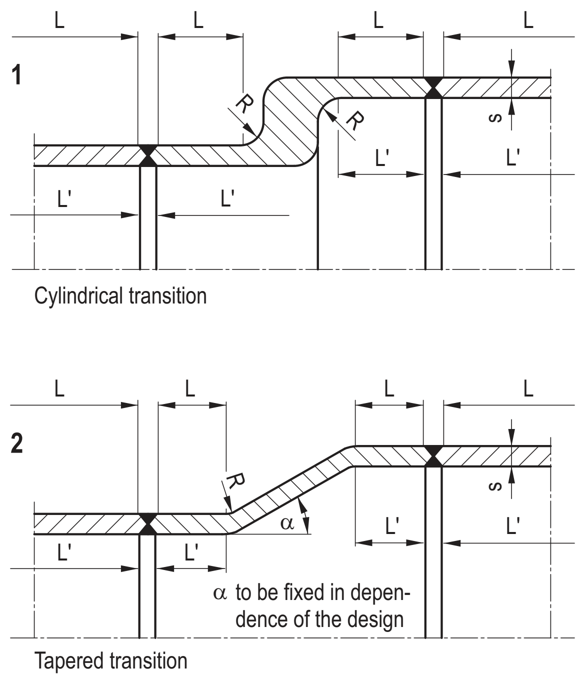
(1) **Figure 5.2-5** shows examples of allowable configurations of transitions in diameter.

(2) The scan lengths L or L* and L' according to **Figure 5.2-5** shall be provided on both sides of the weld depending on the test procedure selected, in accordance with KTA 3211.3, Section 11. Available radii at the edges or transitions shall be considered. The scan length shall begin at the run-out of the transition radius and end at the edge of the weld. In the case of configuration (2) the cylindrical section for components of test group A3 between the weld and conical shell (corner weld) may be omitted if R > 3 · s, $\alpha \leq 30°$, the weld is back-welded, and the weld surfaces are flush-ground on the inside and outside.



For notations see **Table 5.2-1**

Figure 5.2-4: Designs of welds on nozzles



L , L´ : scan lengths for ultrasonic testing

Figure 5.2-5: Examples of allowable configurations of transitions in diameter

5.2.4 Flanges and gaskets

(1) Flanges shall only be of the forged or cast type, in the case of loose-type flanges also rolled without seam.

(2) For flanged connections on nozzles and piping systems welding-neck flanges to DIN Standards or welding-neck flanges with standard sizes shall normally be used. Regarding the assignment to pressure rating clause A 2.10.3 shall be taken into account.

(3) For austenitic and ferritic flanged connections which cannot be constructed as standard flanges, the design of the flange faces and the bolt pitch circle shall be selected in correspondence with the standard flanged connections. The thickness of the flange ring and the transition to the attached piping shall preferably be adapted. For the radius r between flange ring and the conical or cylindrical hub the following condition applies:

The radius shall be at least $0.25 \cdot s_R$, but not be less than 6 mm (s_R = wall thickness of attached piping).

(4) In the case of non-standard flanges the number of bolts shall be so great that uniform pressure of the bolts and thus positive sealing is obtained. At least 4 bolts meeting the requirements of clause 5.2.5 shall be provided. The number of bolts shall normally be dividable by four. The ratio of bolt hole centre distance to bolt hole diameter shall be equal to or less than 5.

(5) When selecting the gasket, its capability to withstand the pressure, temperature, radiation, fluid and external loadings as well as the compatibility of materials shall be taken into account. The design of the faces shall consider the specific features of the gaskets.

5.2.5 Bolts and nuts

(1) Bolts and nuts complying with DIN standards shall be used as far as the design permits. Necked-down or reduced-shank bolts are to be preferred. The effective thread length shall be adapted to the combination of materials (e.g. bolts - body). Reduced-shank bolts to DIN 2510-1 to DIN 2510-4 or necked-down bolts shall be used at design temperatures above 300 °C or design pressures above 4 MPa.

(2) Ferritic bolts shall be provided with an adequate surface protection.

(3) Bolts and nuts for connection with austenitic parts shall be made, if possible, of the same or similar material as the parts to be joined. Where materials with different coefficients of thermal expansion are used, the effect of differential thermal expansion shall be taken into account.

(4) Bolts smaller than M 10 or respective diameter at root of thread are basically not permitted. In special cases (e.g. in the case of bolts for valves) smaller bolts may be used, however, their dimension shall not be less than M 6 or respective diameter at root of thread.

(5) The design of threaded connections shall ensure a mainly tensile loading of the bolts.

5.2.6 Nozzles

(1) The geometric conditions (wall thickness ratios, weld radii, nozzle lengths) are contained in **Table 5.2-1**. The definition of the units contained in Table 5.2-1 can be taken from **Figures 5.2-4** and **5.2-6**.

(2) For nozzles not less than DN 125 and a nozzle wall thickness s_A not less than 15 mm the main shell shall normally be reinforced, taking a favourable distribution of stresses into account. At a diameter ratio q_A above 0.8 a stress analysis shall be performed additionally unless this area has been covered by adequate dimensioning procedures. The diameter ratio q_A is defined as the ratio of the mean diameter of branch piping to the mean diameter of the reinforced area of the run pipe.

$$q_A = \frac{d_{Ai} + s_A}{d_{Hi} + s_H} \qquad (5.2\text{-}1)$$

Wall thickness ratios	s_A/s_H	≤ 1.3 : 1 additional requirements see clause 5.2.6 (4)
	s_A/s_R at PB ≥ 4 MPa	≥ 1.5 : 1 (nominal sizes) and ≥ 2 : 1 referred to the calculated wall thickness of the connected piping
Configuration of welds and transitions	Basic requirements R_S	$\geq 0.5 \cdot s_H$ for set-through nozzles and set-on nozzles
	Basic requirements f	$\geq 0.4 \cdot s_H$ for set-through nozzles $\geq 0.8 \cdot s_A$ for set-on nozzles
	Welds extending the weld edges shall be avoided	
	r_S	≥ 5 mm, however not exceeding $0.25 \cdot s_H$
	α	30° up to 60°
	r (beyond the welds)	Transitions shall be smooth and edges be rounded. The radii shall be fixed depending on the design and on a favourable distribution of stresses.
Minimum nozzle lengths l for vessels		100 mm for DN ≤ 80 150 mm for DN > 80

s_A wall thickness of branch (nozzle)
s_H wall thickness of body
s_R wall thickness of connected to the nozzle piping
r radius at intersection
R_S weld radius
r_S transition radius at weld
f width across corners
α weld angle
l nozzle length

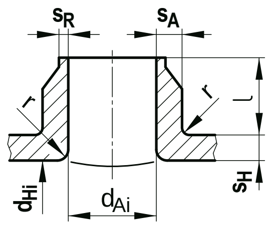


Table 5.2-1: Requirements for wall thickness ratios, welds and nozzle transitions

(3) The wall thickness ratio of nozzle to shell shall basically be selected to be equal to or less than 1.3 (see **Table 5.2-1**). This wall thickness ratio may be exceeded in the following cases:

a) the additional wall thickness of the nozzle is not used to reinforce the nozzle opening, but is selected for design reasons (e.g. manhole nozzle),

b) the nozzle is fabricated with reduced reinforcement area (e.g. nozzles which are conical to improve test conditions for the connecting pipe),

c) the ratio of nozzle-to-main shell diameter does not exceed 1:10 and the nozzle diameter does not exceed DN 125,

d) in individual cases up to a wall thickness ratio s_A/s_H not exceeding 1.5 if a general analysis of the mechanical behaviour as per Section 7 is made.

(4) Where the nozzle diameter is great in relation to the main shell, the wall thickness ratio shall be reduced. In the case of a branch with q_A exceeding 0.8 the wall thickness ratio s_A/s_H shall be ≤ 1.0.

(5) Nozzles shall be made from forged bars (limitation of diameter depending on analysis), seamless forged tubular products or seamless pipes. Longitudinally welded nozzles are permitted if the longitudinal weld is back-welded or the root is dressed.

(6) For nozzles ≥ DN 125 with a nozzle wall thickness s_A exceeding 15 mm the scan lengths laid down in KTA 3211.3 for ultrasonic testing of the nozzle weld shall be provided.

(7) For nozzles ≥ DN 125 made of austenitic steels and for nozzles ≥ DN 125 made of ferritic steels with a nozzle wall thickness $s_A \leq 15$ mm the scan lengths laid down in KTA 3211.3 for radiography of nozzle welds shall be provided.

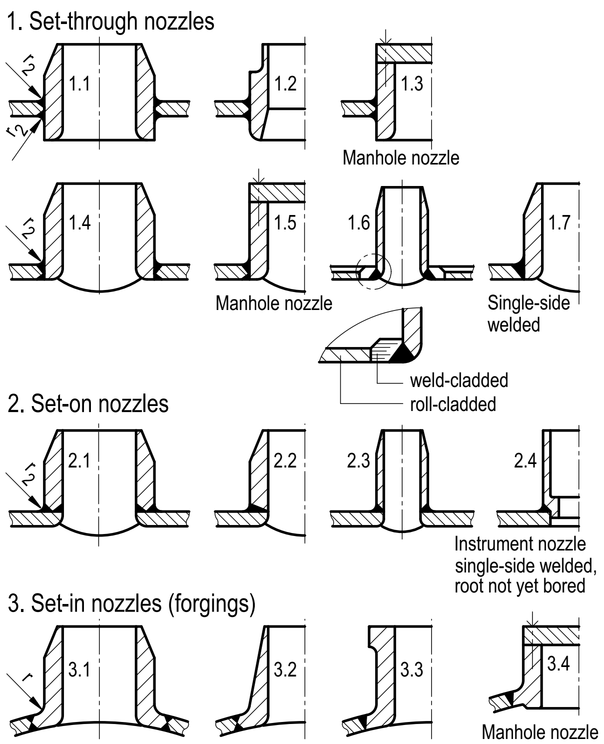


Figure 5.2-6: Examples of nozzle designs

5.2.7 Dished and flat heads

The following types of heads shall preferably be used:

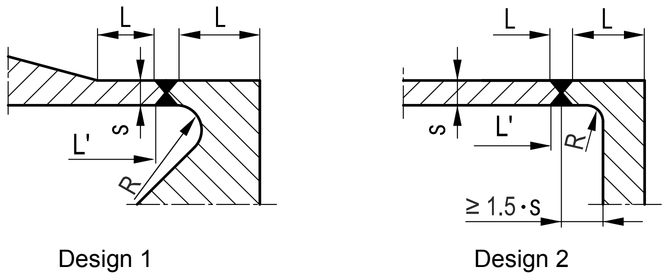
a) flanged flat heads

b) torispherical heads

c) semi-ellipsoidal heads

d) hemispherical heads.

Figure 5.2-7 shows permissible types of welded flat heads (e.g. end caps). Design types 1 and 2 are permitted for forgings or parts fabricated by a combination of forging and rolling. Type 2 may also be made of forged bars for diameters not exceeding DN 150. Plates are permitted for flanged flat covers only subject to pressure perpendicular to the surface. For pressure tests blanks made from plate are permitted.



Wall thickness s in mm	Design	Condition for R in mm	Condition for L, L'
s ≤ 40	1	R = max.{5; 0.5 · s}	acc. to KTA 3211.3
s ≤ 40	2	R = max.{8; 0.5 · s}	
s > 40	1 and 2	R ≥ 0.3 · s	

Figure 5.2-7: Allowable designs of welded flat heads

5.3 Component-specific requirements

5.3.1 General

The requirements of Sections 5.1 to 5.2 regarding the design apply primarily to all types of components. In the following, component-specific design requirements are additionally given to be met by various structural elements of apparatus and vessels, pumps, valves, and piping systems.

5.3.2 Pressure vessels

5.3.2.1 Nozzles

(1) For the design of nozzles on vessels the requirements of clause 5.2.6 apply.

(2) The portion of the nozzle calculated as reinforcement of opening shall be considered to be part of the pressure-retaining wall of the vessel. The portion belonging to the vessel may be extended to the first nozzle attachment weld or, in the case of flanged attachments, to the interface between the flanges.

5.3.2.2 Inspection openings

(1) Inspection openings shall be provided to meet the requirements of the AD 2000-Merkblatt A 5.

(2) Nozzles for inspection openings shall meet the design requirements of clause 5.2.6. Covers and sealings (e.g. manhole) shall be so designed that multiple opening for inspection and repair purposes is possible without affecting the tightness; weld lip seals shall be avoided.

(3) Vessels filled with radioactive fluids shall be provided with access openings with DN 600, if required by AD 2000-Merkblatt A5.

5.3.2.3 Tubesheets

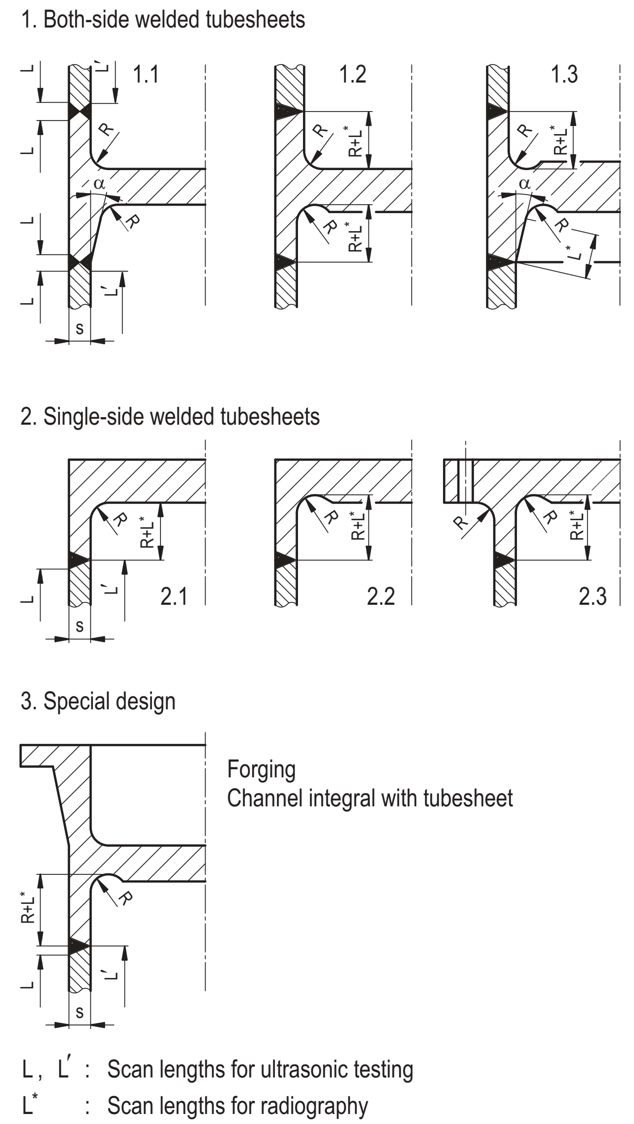
(1) **Figure 5.3-1** shows examples of typical designs of tubesheets with hubs for connection to cylindrical sections. These examples apply to ferritic and austenitic materials.

(2) The weld joining the cylindrical section and the tubesheet shall be back-welded, i.e. it shall, basically, never be welded as final weld. Exceptions to this rule are permitted in the case of small dimensions where access from the inside is not possible. Dressing on the inside shall basically be possible during fabrication.

(3) Where weld surfaces are ground flush both on the inside and outside, a scan length for ultrasonic testing at the outside at one side of the weld will suffice

(4) Other designs than those shown in **Figure 5.3-1** are permitted if is has been proved that the stresses are allowable

and the geometric conditions for performing non-destructive testing are given.



s in mm	R [1)]	α
≤ 40	0.5 · s	≤ 10°
> 40	0.3 · s	≤ 10°
[1)] Minimum values		

Figure 5.3-1: Design types of tubesheets (general examples), design requirements.

5.3.2.4 Expansion joints

(1) Single-or multi-ply bellows expansion joints with longitudinal or circumferential welds on the bellows may be used. The requirements set forth hereinafter apply to single-ply bellows; multi-ply bellows expansion joints shall be verified in each individual case.

(2) Expansion joints shall be butt-welded to weld ends or vessel sections; see design example in **Figure 5.3-2**; welds joining dissimilar materials shall be avoided.

(3) To make testing of the attachment weld possible a scan length $L \geq 3 \times s$ (see **Figure 5.3-2**) shall be provided. For reasons of stability the total length of the tangent (see **Figure 5.3-2**) with the wall thickness s shall not exceed the value

$$\ell = 0.5 \cdot \sqrt{s \cdot d / 2}$$

(see clause 8.2.5.2).

(4) Expansion joints are not suited to withstand considerable transverse and axial forces as well as bending moments. Therefore, the design shall ensure, e.g. by means of guides, internals and support structures that such loadings are not applied on expansion joints.

Note:
Clause 8.2.5.4.2.5 contains requirements for consideration of the weld position in case of high-cycle fatigue loading.

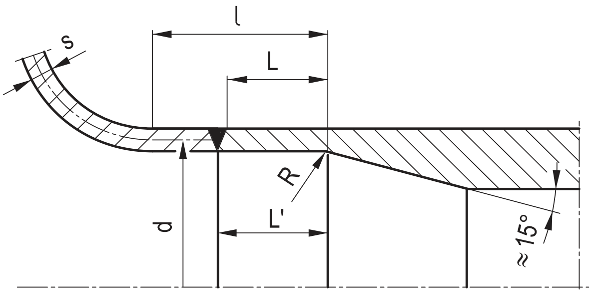


Figure 5.3-2: Example of connection between expansion joint and vessel

5.3.3 Pump casings

Pump casings may be of the forged, cast or welded design. Some typical design types are shown in Section A 3. The design requirements of Sections 5.1 and 5.2 apply. The following shall be considered additionally:

a) The pump casing shall be so designed that the required functional capability is maintained in the event of pipe forces and moments and as well as loadings from external events occurring in addition to the operational hydraulic and thermal loadings. In each case the pump support, the stability of the support structure and the anchorage in the building shall be taken into account.

b) The design of the pump casing and the pertinent systems shall permit adequate accessibility for maintenance, replacement of wear parts and repair purposes.

5.3.4 Valve bodies

Value bodies may be of the forged, cast or welded design. Some typical design types are shown in Section A 4. The design requirements of Sections 5.1 and 5.2 apply. The following shall be considered additionally:

a) The valve body shall be designed to be so stiff that the required stability is maintained in the event of pipe forces and moments as well as loadings from external events occurring in addition to the operational hydraulic loadings. In each case the supports shall be taken into account.

b) The design of the valve body and the pertinent systems shall permit adequate accessibility for maintenance, replacement of wear parts and repair purposes.

c) The design of the valve body shall especially provide smooth tapers at cross-sectional transitions.

5.3.5 Piping systems

5.3.5.1 Pipes, pipe bends and elbows

(1) Pipes, bends and elbows with PN exceeding 40 shall be seamless.

(2) The ratio R_m/d_a of elbows shall be not less than 1.5. A ratio R_m/d_a equal to or greater than 2 is desirable.

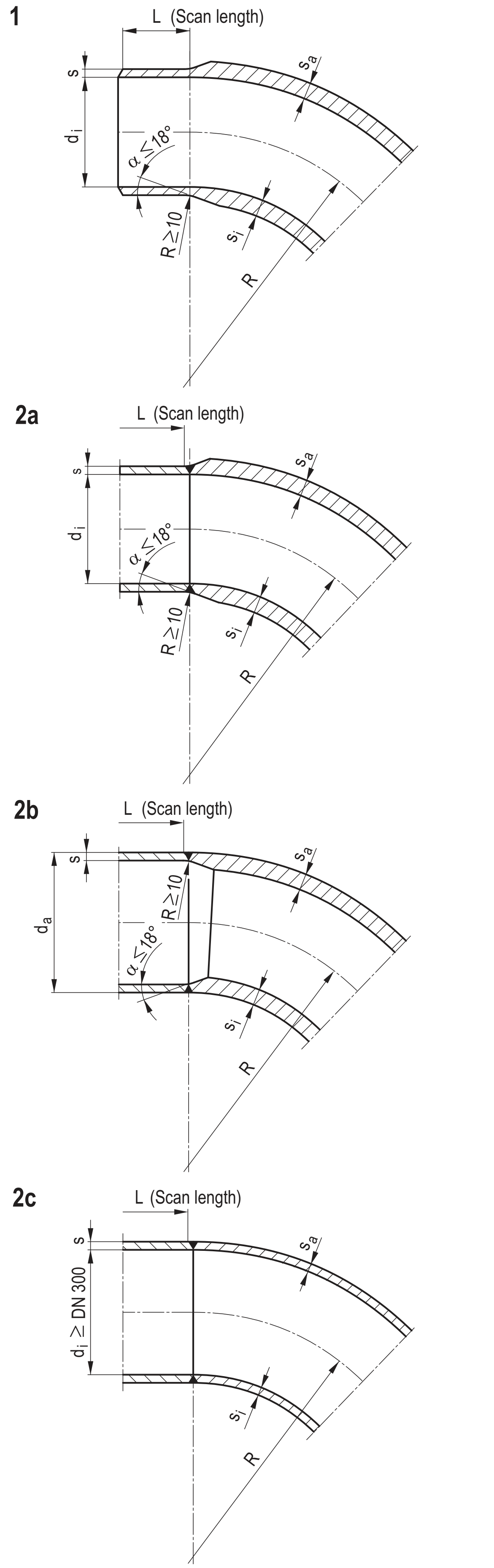
(3) Bends with straight ends shall be used if this is required to make the performance of non-destructive tests possible. Pipe bends without straight ends are permitted:

a) in the case of austenitic and ferritic joints that are subjected to radiography,

b) in the case of equal wall thickness of pipe and pipe bend ≥ DN 300 and R ≥ 1.5 x D,

c) in the case of a weld ground on the inside (ultrasonic testing from one weld side possible).

(4) Allowable designs of pipe bends are shown in **Figure 5.3-3**.



In the case of internal and external tapered transitions the sum of the angles shall be ≤ 18°.

Figure 5.3-3: Design of pipe bends

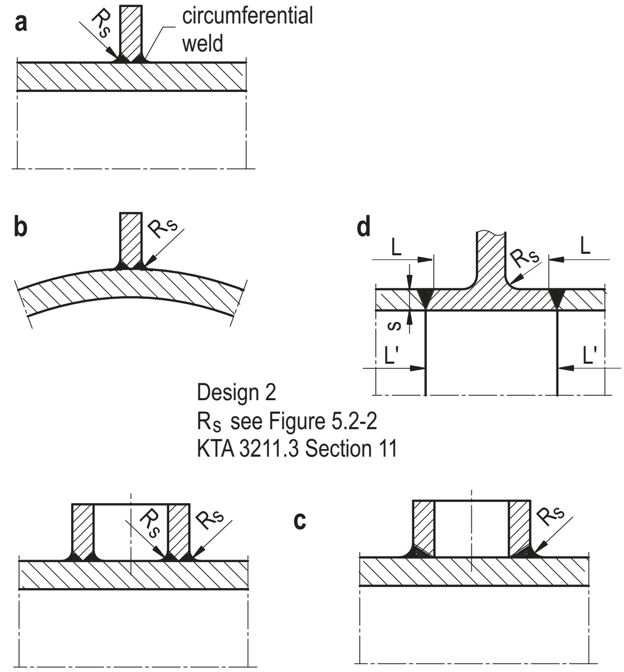
5.3.5.2 Welded attachments to pipe walls for anchors and intermediate anchors

The following requirements apply to anchors and intermediate anchors:

a) Design a) and b) of **Figure 5.3-4** are permitted for piping systems up to PN 40. Design c) of **Figure 5.3-4** may also be used for piping systems with nominal pressure ≥ 4 MPa.

b) For piping systems ≥ DN 250 and nominal pressure not less than 4 MPa as well as operating temperature not less than 100 °C forged fittings basically shall be used (design d) of **Figure 5.3-4**). Upon agreement with the authorised inspector design c) of **Figure 5.3-4** may be used for the abovementioned range of application

c) It shall be ensured that the attachment weld and the component can be tested.



Permitted if the requirements of cl. 5.2.2.2 are met

Design a, b and c: Wall thickness ratio of attachment/pipe ≤ 1.3:1

L , L´ : scan lengths for ultrasonic testing

Figure 5.3-4: Welded attachments of pipe walls for anchors and intermediate anchors

5.3.6 Component support structures

5.3.6.1 General

(1) Component support structures may be designed as supporting structures with integral or non-integral areas.

(2) The integral area of a supporting structure comprises the parts rigidly attached to the component (e.g. welded, cast, machined from the solid) with support function.

The non-integral area of a supporting structure comprises parts detachably connected or not connected (e.g. bolted, studded, simply supported) having supporting functions as well as those parts with supporting functions of a supporting structure rigidly attached to the component outside the area of influence (see Section 8.6).

Note:

Non-integral areas of a supporting structure shall be classified as structural steel components and fall under the scope of KTA 3205.1, and in the case of standard supports fabricated in series (with approval test) fall under the scope of KTA 3205.3.

(3) For welded integral support structures the same requirements as for the pressure-retaining wall apply. Attachment welds on the pressure-retaining wall shall be full-penetration welded unless they are fillet welds according to clause 5.2.2.2 (4).

5.3.6.2 Vessels

(1) Allowable design types are shown in **Figures 5.3-5** to **5.3-7**.

(2) In the case of elevated temperature components the differing thermal expansions of components and support structures shall be taken into account.

(3) In the case of horizontal loadings (e.g. external events) lateral supports may be required in the case of vertical vessels to ensure stability. Depending on the design these supports may also reduce vertical forces.

Examples:

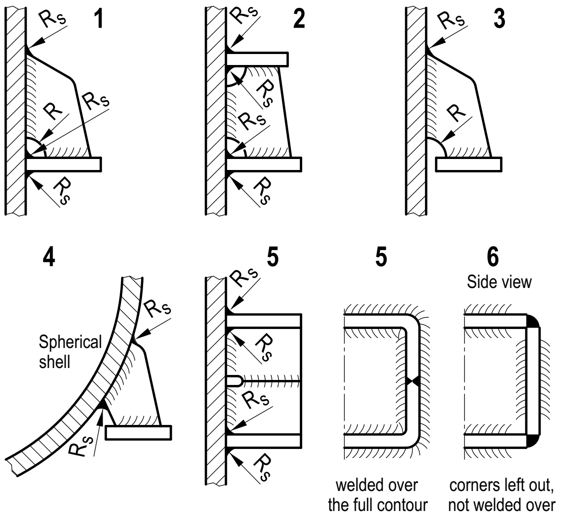
a) Skirt supports with or without support ring (see **Figure 5.3-6**),

b) Forged ring in the cylindrical shell (see **Figure 5.3-7**),

c) Guide pins (see **Figure 5.3-7**),

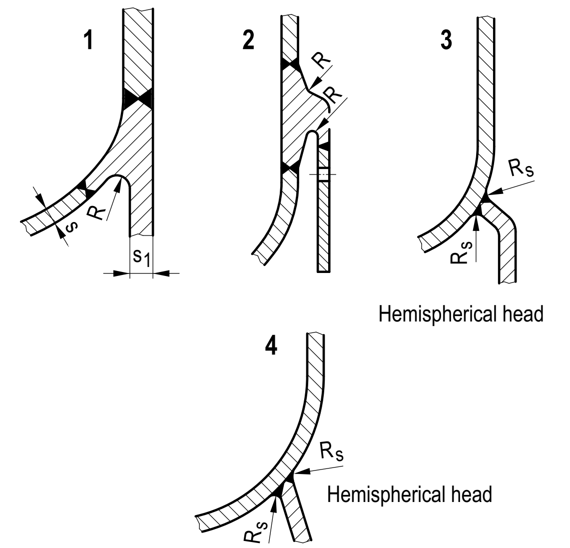
d) Brackets (see **Figure 5.3-5**).

(4) At suitable locations lifting lugs or hooks shall be provided on the components.



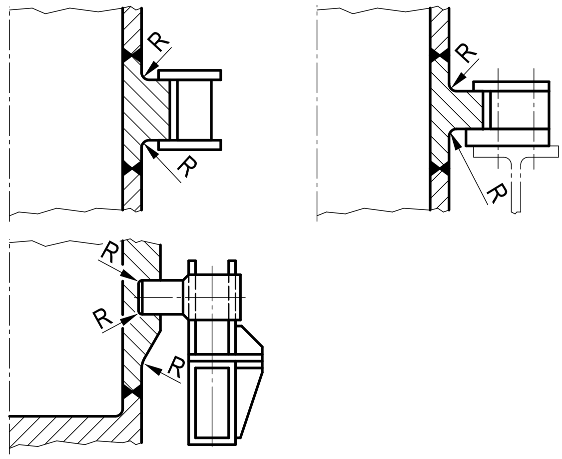
For the design types 1 to 4 two webs each per support skirt are provided. The radius R_S shall be fixed according to Figure 5.2-2. The radius R shall be selected with regard to a favourable distribution of stresses. The design types 1, 2, 3 and 4 shall only be selected for small components or at working pressure not exceeding 4 MPa.

Figure 5.3-5: Examples of component support structures with integral attachment of vertical pressure vessels to bracket supports



The radii R_S shall be fixed according to Figure 5.2-2.

Figure 5.3-6: Examples of component support structures with integral attachment of vertical pressure vessels with skirt supports

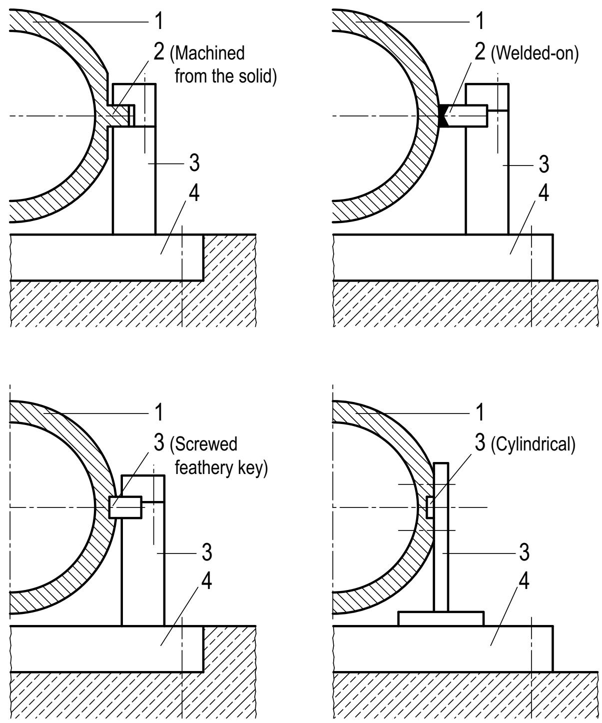


The transitional radii shall be smooth to avoid stresses.

Figure 5.3-7: Examples of component support structures of vertical vessels with forged rings

5.3.6.3 Pumps

For welded integral component support structures the same requirements as for pressure parts apply (full-penetration welds, test requirements). Examples of component support structures for pumps are given in **Figure 5.3-8**.

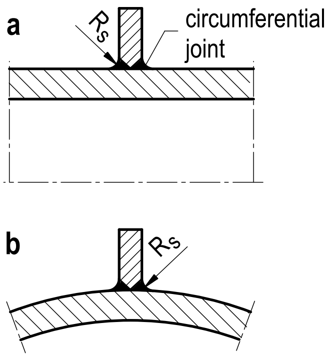


1 Casing
2 Integral component support structure
3 Non-integral component support structure
4 Foundation plate including anchor bolts

Figure 5.3-8: Examples of component support structures for pumps

5.3.6.4 Valves

For valve supports not less than DN 250, nominal pressures not less than 4 MPa and operating temperatures not less than 100 °C forged fittings shall be used. In the other cases, forged and welded attachments are permitted. Examples of welded joints on valves for component support structures are given in **Figure 5.3-9**.



Wall thickness ratio of attachment/pipe ≤ 1,3:1
R_s see Figure 5.2-2

Figure 5.3-9: Welded attachments on valves for component support structures

6 Dimensioning

6.1 General

(1) Dimensioning shall be effected on the basis of the design loading level (Level 0) in accordance with clause 3.3.2.

Note:

***Annex B** contains requirements for an analytical confirmation in case of a numerical reassessment of a component.*

(2) Dimensioning shall be effected using one of the following procedures:

a) in accordance with Annex A,

b) verification of primary stresses, where the primary stress shall be limited using the primary stress intensities laid down in clause 7.7.3.4,

c) as limit analysis where, for the purpose of calculating the lower bound collapse load, $\sigma_F = 1.5 \cdot S_m$ at design temperature shall be used as yield stress value, and the specified loading shall not exceed 67 % of the lower bound collapse load as per cl. 7.7.4.1.

In specific cases other suitable methods may be applied if it is proved by means of analytical and/or experimental analyses that in due consideration of interacting damage mechanisms, if any, the limit of stress intensities (safety factors) derivable from Section 7.7.3.4 are obtained. In this case, input data (e.g. wall thicknesses) measured or verified in detail may form the basis.

The components for which pertinent design rules are available in **Annex A** shall be dimensioned to these design rules.

(3) In addition, a proof of stability, if required, shall also be performed (see clause 7.10).

6.2 Welds

(1) Full-penetration welds

As the welds have to meet the requirements of KTA 3211.1 and 3211.3, they need not be considered separately in the dimensioning of the parts.

(2) Fillet welds

For attachment welds to cl. 5.2.2.2 (4) the reduced load-carrying capacity of fillet welds shall be considered in the dimensioning, e.g. in accordance with KTA 3205.1. In this case, the allowable stresses shall be taken from the respective part of Table 7-4 of KTA 3205.1 (serial no. 7 to 9). The design loading levels shall be assigned accordingly (H = Levels 0 and H; HZ = Levels B and P; HS1 = Level C and HS2/HS3 = Level D). The stresses shall be determined to Section E3 of KTA Safety Standard 3205.1 to consider the limitations to cl. 7.2.2 (3) of same standard.

6.3 Claddings

(1) When determining the required wall thicknesses and cross-sections, claddings, if any, shall be considered not to be contributing to the strength.

(2) The design against internal pressure shall take the internal diameter of the unclad part into account.

(3) Shape weldings on the base metal which meet the requirements of KTA 3211.3, are not considered claddings.

6.4 Wall thickness allowances

(1) When determining the nominal wall thickness the fabrication tolerances shall be considered by a respective allowance c_1 which is equal to the absolute value of the minus tolerance of the wall thickness in accordance with the acceptance specification.

(2) An allowance c_2 shall take wall thickness reductions due to chemical or mechanical wear into account. This applies both to the wall thickness reduction and the extension of the internal diameter. The allowance c_2 may be omitted if no wear is expected or a cladding is provided.

6.5 Wall thicknesses

(1) Irrespective of the dimensioning method used the following conditions shall be satisfied:

The nominal wall thickness s_n shall satisfy the following condition in consideration of the allowances c_1 And c_2:

$$s_n \geq s_0 + c_1 + c_2 \qquad (6.5\text{-}1)$$

where s_0 is the calculated wall thickness.

This shall be verified by a recalculation with the wall thickness $s_{0n} = s_n - c_1 - c_2$, see **Figure 7.1-1**.

When determining the wall thickness by means of the nominal external diameter d_{an}

$$d_a = d_{an} \qquad (6.5\text{-}2)$$

shall be taken and when determining the wall thickness by means of the nominal internal diameter d_{in} shall be taken as follows:

$$d_i = d_{in} + 2 \cdot c_2 \qquad (6.5\text{-}3)$$

(2) Pipes with nominal diameter not less than DN 150 and pressure-retaining components with comparable dimensions shall be designed for a wall thickness not less than 10 mm when ferritic materials are used and for a wall thickness not less than 5 mm when austenitic materials are used. Exemptions herefrom are components subject to low temperatures (design pressure not exceeding 2.5 MPa and design temperature not exceeding 100 °C).

This requirement shall not apply with regard to the minimum wall thickness of expansion joints and heat exchanger tubes.

6.6 Design stress intensities

(1) The design stress intensity values are separately fixed for test groups A1, A2 and A3 (see Section 2) in **Table 6.6-1**.

(2) The design stress intensity value for components of test group A1 is S_m. The design stress intensity value for components of test groups A2 and A3 is S.

(3) The S_m value and the S value are obtained on the basis of the temperature T of the respective component and the ambient temperature RT. For the service levels the temperature T occurring at the respective location and time shall be taken. For the design loading level (Level 0), however, the design temperature shall be taken.

Test group	A1	A2/A3
Material	Design stress intensity	
	S_m	S
Ferritic	$\min.\left\{\begin{matrix} R_{mRT}/3 \\ R_{mT}/2.7 \\ R_{p0.2T}/1.5 \end{matrix}\right.$	$\min.\left\{\begin{matrix} R_{mRT}/4 \\ R_{p0.2T}/1.6 \\ R_{p0.2RT}/1.6 \end{matrix}\right.$
Austenit	$\min.\left\{\begin{matrix} R_{mRT}/3 \\ R_{mT}/2.7 \\ R_{p0.2RT}/1.5 \\ R_{p0.2T}/1.1 \\ R_{p0.2T}/1.5^{1)} \end{matrix}\right.$	$\min.\left\{\begin{matrix} R_{mRT}/4 \\ R_{p0.2T}/1.1 \\ R_{p0.2RT}/1.6 \\ R_{p0.2T}/1.5^{1)} \end{matrix}\right.$
Austenitic cast steel [2)]	$\min.\left\{\begin{matrix} R_{mRT}/4 \\ R_{mT}/3.6 \\ \text{Ferritic:} \\ R_{p0.2T}/2 \\ \text{Austenitic:} \\ R_{p0.2T}/2^{1)} \end{matrix}\right.$	$\min.\left\{\begin{matrix} R_{mRT}/4 \\ R_{mT}/3.6 \\ \text{Ferritic:} \\ R_{p0.2T}/2.5 \\ \text{Austenitic:} \\ R_{p0.2T}/2.5^{1)} \end{matrix}\right.$

The material design strength values indicated shall be taken as minimum values.

Note:
The design strength values refer to the wall thickness at the final heat treatment (quenching and tempering, normalising or solution-annealing).

Footnotes:

1) This criterion only applies to the dimensioning.
For austenitic material with a ratio $R_{p0,2RT}/R_{mRT} \le 0.5$ $R_{p1,0T}$ may be taken in lieu of $R_{p0.2T}$ as far as guaranteed values are available for $R_{p1.0T}$.

2) For weld ends made of ferritic cast steel (e.g. on valve bodies) which are additionally subject to ultrasonic testing, the stress intensities may be 25 % higher.

Table 6.6-1: Formation of design stress intensity values for pressure parts

6.7 Allowable stresses for dimensioning

(1) The allowable stresses used for the dimensioning of pressure-retaining walls are determined in dependence of the test group, service level loading and stress category in **Table 6.7-1**.

(2) The allowable stresses used for the dimensioning of pressure-retaining bolts are laid down in **Table 6.7-2** for all test groups in the same manner, but in dependence of the loading level and the type of loading.

6.8 Nominal operating stress

The nominal operating stress is the general primary membrane stress (P_{mNB}) due to internal pressure loading during normal operation (service loading level A) and shall be determined in pressure-retaining shells of the component according to the following equations:

a) Cylindrical shell:

$$P_{mNB} = p \cdot \left(\frac{d_i}{2 \cdot s_{0n}} + 0.5 \right) \qquad (6.8\text{-}1)$$

b) Spherical shell:

$$P_{mNB} = p \cdot \left(\frac{d_i^2}{4 \cdot (d_i + s_{0n}) \cdot s_{0n}} + 0.5 \right) \qquad (6.8\text{-}2)$$

Notations in equations (6.8-1) and (6.8-2):

d_i = Shell internal diameter

s_{0n} = Nominal shell wall thickness minus the allowances according to Section 6.5

p = Operating pressure of Level A

To obtain a low stress level in transitional zones and at discontinuities the design requirements of Section 5 shall be taken into account.

Stress category		Loading levels	Test group A1	Test group A2, A3 [3)]
Primary stresses	General primary membrane stress P_m	0, A	S_m	S [1)]
		B	$1.1 \cdot S_m$	$1.1 \cdot S$
		Pressure test P	$0.9 \cdot R_{p0.2T}$	$0.9 \cdot R_{p0.2T}$
		C	Greater value of: [2)] $1.2 \cdot S_m$ and $R_{p0.2T}$	$1.5 \cdot S$
		D	Smaller value of: $2.4 \cdot S_m$ and $0.7 \cdot R_{mT}$	$2.0 \cdot S$
	General primary membrane plus primary bending stress P_m+P_b or local primary membrane plus primary bending stress P_l+P_b	0, A	$1.5 \cdot S_m$	$1.5 \cdot S$
		B	$1.65 \cdot S_m$	$1.65 \cdot S$
		Pressure test P	$1.35 \cdot R_{p0.2T}$	$1.35 \cdot R_{p0.2T}$
		C	Greater value of: [2)] $1.8 \cdot S_m$ and $1.5 \cdot R_{p0.2T}$	$1.8 \cdot S$
		D	Smaller value of: $3.6 \cdot S_m$ and R_{mT}	$2.4 \cdot S$

The material design strength values indicated are minimum values.

1) The conditions of Table 2-1 regarding the limits of nominal operating stresses shall be taken into account.

2) However, not more than 90 % of the allowable value of Level D.

3) For austenitic materials it shall be ensured that the allowable stresses for test group A1 are not exceeded.

Table 6.7-1: Allowable stresses for the dimensioning of pressure-retaining walls

Ser. no.	Bolt loading [1)]	Type of bolt [2)]	Allowable stress σ_{zul}				
			Bolting-up condition	Loading level			
				0	A, B	P	C, D
1	Average tensile stress due to internal pressure only $F_S = F_{RP} + F_F$	—	—	$\frac{1}{3} R_{p0.2T}$	—	—	—
2	Average tensile stress due to internal pressure, required gasket load reaction and external loads $F_S = F_{RP} + F_F + F_{DB} + F_{RZ} + F_{RM}$	Reduced-shank bolt	—	$\frac{1}{1.5} R_{p0.2T}$	$\frac{1}{1.5} R_{p0.2T}$	—	$\frac{1}{1.1} R_{p0.2T}$
		Full shank bolt	—	$\frac{1}{1.8} R_{p0.2T}$ [3)]	$\frac{1}{1.8} R_{p0.2T}$	—	$\frac{1}{1.3} R_{p0.2T}$
3	Average tensile stress at test condition F_{SP}	Reduced-shank bolt	—	—	—	$\frac{1}{1.1} R_{p0.2T}$	—
		Full shank bolt	—	—	—	$\frac{1}{1.3} R_{p0.2T}$	—
4	Average tensile stress in the bolting-up condition [4)] F_{S0}	Reduced-shank bolt	$\frac{1}{1.1} R_{p0.2RT}$ [5)]	—	—	—	—
		Full shank bolt	$\frac{1}{1.3} R_{p0.2RT}$	—	—	—	—
5	Average tensile stress due to internal pressure, external loads, residual gasket load, and differential thermal expansion [6)], if any, taking the bolts stress and residual gasket load at the respective pressure load condition into account	—	—	—	$\frac{1}{1.1} R_{p0.2T}$ [7)]	—	—
6	Total stress [8)] (including peak stresses)	—	—	—	$2 \cdot S_a$ [9)] $D \leq 1.0$	—	—

1) See clause A 2.9.1 for definition of notations used. For F_{DB} the respective unit shall be used ($F_{DBU/L}$ for floating type joints and "$g_{KNS} \cdot F_{DKU}$" for metal-to-metal contact type joints).

2) This stress limit shall only be observed for flanged joints where the working pressure exceeds 2.5 MPa which are loaded by external forces and moments in addition to the internal pressure.

3) The design allowance to clause A 2.9.4.3 shall be considered.

4) The differing application of forces on the bolts depending on torque moment and friction shall be conservatively considered in strength verifications (maximum bolt load).

5) In addition, the equivalent stress shall be limited to $R_{p0.2RT}$ where bolt assembly is performed by torque wrench. The calculated torsional strength may be determined by the polar resistance moment $W_p = (\pi/12) \cdot d_0^3$ (with d_0 = reduced-shank diameter).

6) Consideration of differential thermal expansion at a design temperature > 120 °C. This temperature limit does not apply to combinations of austenitic and ferritic materials for flange and bolts.

7) Where bending stresses occur, the sum of average tensile stress and bending stress (dependent on internal pressure, bolt pre-tensioning, temperature influence and additional loads) shall be limited to $R_{p0.2T}$.

8) To be determined by strain analysis [e.g. correlation of gasket seating load, gasket compression load for operating condition and internal pressure (rigging diagram)]; as regards the fatigue analysis also see sub-clause 7.11.2 (2).

9) The stress amplitude S_a and the cumulative usage factor D shall be limited to satisfy Section 7.8.

Table 6.7-2: Allowable stresses σ_{zul} for pressure-retaining bolted joints of test groups A1, A2 and A3

7 General analysis of the mechanical behaviour

7.1 General

7.1.1 Objectives

(1) It shall be demonstrated by means of the analysis of the mechanical behaviour that the components are capable of withstanding all loadings in accordance with the loading levels in Section 3.3.

(2) The extent of verification depends on the test group. Section 2, especially paragraphs (10) and (11) contains respective information.

(3) Within the analysis of the mechanical behaviour the loadings and, if required, the forces and moments as well as deformations due to loadings of the component to be analysed shall be determined by satisfying the boundary conditions and taking into account the mutual influence of adjacent components and individual parts in accordance with Section 7.6, if required by applying Annexes C and D of KTA 3102.2. The determination may be effected by way of calculation or experiments, or a combination of calculation and experiments.

(4) The loadings and deformations thus determined shall be examined for acceptability in accordance with Sections 7.7 to 7.12.

(5) Here, it shall be taken into account that the exactness of the determined forces and moments depends on the ideal geometric shape of the component or part, the exactness of assuming loadings, boundary conditions and material properties as well as the features and performance of the calculation method selected.

(6) The analysis of the mechanical behaviour may alternatively be made by means of design formulae if, in the case of sufficiently exact and complete consideration of the loading conditions and geometric shape the objectives of verification according to Section 7 are obtained. If applicable, the design formulae will suffice for dimensioning.

7.1.2 Welds

(1) As the welds have to meet the requirements of KTA 3211.1 and KTA 3211.3, their influence on the mechanical behaviour need not be considered separately when determining the allowable stresses.

(2) Within the fatigue analyses the strength-reducing influences of welds depending on weld dressing shall be taken into account as regards the reduction of fatigue strength.

Note:

*Stress indices for fatigue strength reduction (K values) are contained in **Table 8.5-1**.*

7.1.3 Claddings

(1) When determining the required wall thicknesses and sections, claddings, if any, shall not be considered to be contributing to the strength. Deposition welds made on the base metal with equivalent materials are not considered to be claddings.

(2) For the thermal analysis the cladding may be considered. If the cladding thickness exceeds more than 10 % of the wall thickness, the cladding shall be taken into account when analysing the mechanical behaviour. The stress classification and evaluation shall be made separately for the base material and the cladding.

7.1.4 Wall thickness used for analysing the mechanical behaviour

(1) For the analysis of the mechanical behaviour of a part the average wall thickness to be effected (or effective average wall thickness) shall be taken as s_c by subtracting the wear allowance c_2 according to Section 6.4:

$$s_c = s_n + \frac{c_3 - c_1}{2} - c_2 \qquad (7.1\text{-}1)$$

where s_n is defined in equation (6.5-1). c_3 is equal to the plus tolerance. c_1 is equal to the absolute value of the minus tolerance in accordance with Section 6.4; see also **Figure 7.1-1**.

The design wall thickness s_c according to equation (7.1-1) shall be fixed such that it lies in the centre of the tolerance field minus the wear allowance c_2.

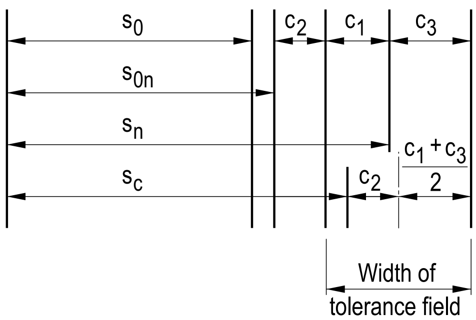


Figure 7.1-1: Wall thicknesses

(2) Where adequate reason is given, e.g. due to an asymmetrical tolerance field or in the case of forgings, another wall thickness may be taken as s_c if it is not less than the required wall thickness ($s_0 + c_2$).

(3) Where the wall thickness tolerances c_1 and c_3 each are not more than 2 % of the nominal wall thickness s_n they need not be considered in the determination of s_c.

7.1.5 Deviations from specified shape and dimensions

7.1.5.1 General

(1) The limitation of deviations from specified shape and dimensions for the purpose of fabrication and inspection/testing is laid down in KTA 3211.3, Section 9. Within the analysis of the mechanical behaviour deviations from shape and dimensions with the values indicated in Section 9 of KTA 3211.3 are negligible unless defined otherwise in the following Sections of this Safety Standard.

(2) All values refer to the unpenetrated membrane area of shell-type parts.

7.1.5.2 Cylindrical parts

7.1.5.2.1 Deviations from wall thickness

(1) Deviations of the effective wall thickness minus the allowance c_2 from the design wall thickness s_c need not be considered separately in the analysis of the mechanical behaviour if they are less than ± 5 % of s_c.

(2) For piping systems a deviation of the effective wall thickness minus the allowance c_2 from the design wall thickness s_c shall only be considered if this deviation lies outside the tolerance field in accordance with a component specification or comparable documents.

(3) For thin-walled ($s_c \leq 5$ mm) and multi-layer components the wall thickness of which shall meet further requirements in addition to the strength requirements (e.g. heat exchanger tubes, expansion joint bellows), the values on which the analysis of the mechanical behaviour are to be based shall be fixed for each individual case. This also applies to wall thickness tolerances in areas with structural discontinuity (e.g. penetrated area of a tee).

7.1.5.2.2 Ovalities

(1) Internal pressure

Ovalities and flattenings in longitudinal direction shall not show a deviation exceeding 1 % from the internal diameter up to and including an internal diameter d_i = 1000 mm. Where the inside diameter exceeds 1000 mm, the value $(d_i + 1000)/(2 \cdot d_i)$ [%] shall not be exceeded.

In this case, the ovality shall be determined as follows:

a) Ovality

$$U = 2 \cdot \frac{d_{i,max} - d_{i,min}}{d_{i,max} + d_{i,min}} \cdot 100\,[\%] \qquad (7.1\text{-}2)$$

b) Flattenings

$$U = 4 \cdot \frac{q}{d_i} \cdot 100\,[\%] \qquad (7.1\text{-}3)$$

where q is shown in **Figure 7.1-2**.

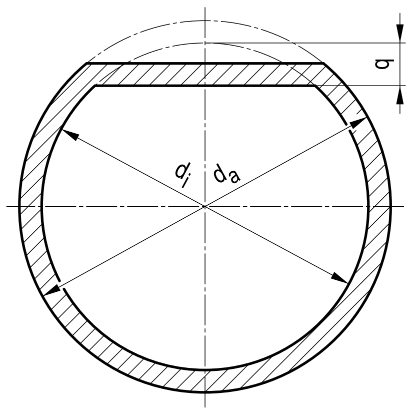


Figure 7.1-2 : Flattening q

(2) External pressure

The allowable ovalities may be taken from KTA 3211.3, clause 9.3.4.2. Where these values are exceeded, a proof of stability shall be furnished.

(3) For pipes the following ovalities are permitted:

for internal pressure: 2 %,

for external pressure: 1 %.

7.1.5.3 Spherical shells

7.1.5.3.1 Deviations from wall thickness

The requirements of clause 7.1.5.2.1 apply.

7.1.5.3.2 Ovalities

(1) Internal pressure

Ovalities and flattenings normally shall not show a deviation from the internal diameter which is greater than one of the following values

$(d_i + 1000)/2 \cdot d_i)$ [%] and $(d_i + 300)/(d_i)$ [%].

The allowable values can be taken from **Figure 7.1-3**.

Ovalities shall be determined in accordance with clause 7.1.5.2.2 (1).

(2) External pressure

The criteria of clause 7.1.5.2.2 (2) may be used.

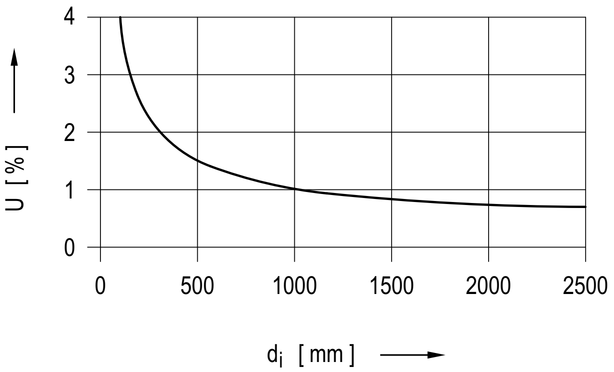


Figure 7.1-3: Ovalities

7.1.5.4 Conical shells

Conical shells shall be treated like cylindrical parts. Ovalities shall be referred to circular cross-sections vertical to the axis of symmetry.

7.1.5.5 Pipe bends and elbows

(1) For ovalities in the bent area of the pipe bend after bending the following applies:

$$U = \frac{d_{max} - d_{min}}{d_0} \cdot 100\ [\%] \qquad (7.1\text{-}5)$$

where

d_{max} : maximum outside diameter after bending or forming

d_{min} : minimum outside diameter after bending or forming

d_0 : pipe outside diameter prior to bending.

For internal pressure, U normally shall not exceed 5 %.

(2) For external pressure the requirements of clause 7.1.5.2.2 (2) apply.

7.1.6 Misalignment of welds

The limitation of weld misalignments for the purposes of fabrication and inspection/testing is laid down in KTA 3211.3 Section 5.7.1.2. Within the analysis of the mechanical behaviour weld misalignments with the values indicated in KTA 3211.3 are negligible provided that the required wall thickness in the weld area is adhered to.

The rules of Section 8.4 are not covered hereby.

7.2 Loadings

(1) Loadings are assumed to be all effects on the component or part which cause stresses in this component or part.

(2) The loadings result from load cases in accordance with Section 3 and are explained in Section 4.

(3) For all static and dynamic loadings of a load case the unit shear forces and unit moments shall be summed-up vectorially. In the cases where this is not possible the dynamic portions shall be formed for each component by the SRSS method (square root of sum of squares) and be superposed the static portions in positive and negative direction.

7.3 Stress/strain loadings

(1) These are stresses or strains or a combination of stresses and strains and are evaluated as equivalent stress or equivalent strain. In the case of a linear-elastic relationship stresses and strains are proportional to each other. In the stress and fatigue analysis according to Sections 7.7 and 7.8 respectively this proportional ratio even when in excess of the yield strength or proof stress of the material shall basically be the basis of analysis (fictitious stresses).

In the case of elastic-plastic analyses to clauses 7.7.4, 7.8.1, 7.8.4 or 7.12 the procedure described in the respective clause shall be followed.

(2) The loadings are (primarily) static loadings, cyclic loadings or dynamic loadings. Pulsating loads are considered to be a specific case of cyclic loading.

(3) The (primarily) static loadings shall be limited within the stress analysis according to Section 7.7. The limitation of cyclic loadings shall additionally be made within the fatigue analysis according to Section 7.8.

7.4 Resulting deformations

(1) Resulting deformations can be determined by means of the integrals calculated for strain and are changes in geometry of the component or the idealized structure due to loadings.

(2) Resulting deformations can be described by displacements and values derived therefrom (e.g. twisting). They shall be limited if required such that the functional capability of the component and its adjacent components is not impaired.

7.5 Determination, evaluation and limitation of mechanical forces and moments

(1) The mechanical forces and moments mentioned in clause 7.1.1 shall be determined by way of calculation according to the methods laid down in Annex C of KTA 3201.2 or by experiments or by a combination of calculation and experiments.

(2) In the case of comparable physical conditions, suitability of methods and adherence to the pertinent requirements the results obtained from various methods can be considered to be equivalent.

(3) Section 8 contains alternative requirements which completely or in part replace the requirements set forth in this Section 7.5 within the applicability of Section 8.

(4) The forces and moments thus determined shall be assessed and be limited such that ductile fracture and fatigue failure as well as inadmissible deformations and instability are avoided.

7.6 Mechanical system analysis

7.6.1 General

(1) The external loadings (e.g. forces, moments, displacements, temperature distributions) shall be used to determine the influence coefficients (e.g. unit shear forces, unit moments, and displacements) for the points under consideration in the system to be evaluated or at the adjoining edges between component and adjacent component.

(2) External system-independent loadings which do not change the behaviour of the system (e.g. radial temperature distribution and internal pressure, if applied) need only be considered when determining and evaluating the stresses.

7.6.2 Modelling

7.6.2.1 General

The system shall be transformed into an idealised model having the properties mentioned in clauses 7.6.2.2 to 7.6.2.5. In addition, the requirements for the respective calculation method and its pertinent modelling in accordance with Annex C of KTA 3201.2 shall be met.

7.6.2.2 System geometry

The system geometry shall comprise the components and parts which considerably influence the structure to be evaluated. The geometry of a piping system may be shown as a chain of bars by means of straight and curved bars which corresponds to the pipe axis routing.

7.6.2.3 Flexibilities

(1) Piping components

Piping components shall normally be considered in the analysis of the mechanical behaviour of the structure with the flexibilities according to their geometry (average dimensions including cladding).

Note:

In the case of symmetrical tolerances these are nominal dimensions.

(2) Small components

Small components are parts of the piping system (e.g. valves, header drums, manifolds, branches, and special parts). Where these components only have little influence on the flexibility of the total structure, suitable flexibility factors (limit values) (e.g. valves: rigid; insulation: without influence on the rigidity) shall be selected.

(3) Expansion joints

The working spring rates of expansion joints shall be taken into account.

(4) Large components

The influence of large components (e.g. vessels) shall be taken into account by suitable modelling in consideration of the anchor function of the vessel.

(5) Component supports and buildings

The influence of component supports and the building shall be considered.

7.6.2.4 Distribution of masses

(1) The masses in the system comprise the masses of each component or their parts, the fluid, the insulation, and other additional masses.

(2) A system with uniform distribution of masses may also treated like a system with discrete masses.

(3) The distribution of masses shall satisfy the requirements regarding the distribution of unit shear forces and unit moments and the type of vibrations.

(4) In the case of essential eccentricity the mass moments of inertia for the rotational degrees of freedom shall also be taken into account.

7.6.2.5 Edge conditions

Forces and moments and displacements shall be taken into account as edge conditions with respect to their effects for the considered load case.

7.6.2.6 Subdivision of structures (uncoupling) into sections to avoid interaction of loadings

(1) Static method

Structures may be subdivided into sections if at the interface between any two sections the respective conditions are satisfied:

a) the ratio of second moments of area does not exceed 0.01,

b) the ratio of these elements in a flexibility matrix which govern the considered deformations is sufficiently small.

(2) Dynamic method

In the case of dynamic loadings, structures may be subdivided into sections if the interaction between the sections is taken into account or the vibration behaviour is not inadmissibly changed.

7.6.3 Calculation methods

(1) The calculation methods to be used depend on the selected mathematical approach as well as on the loading to be evaluated (static or dynamic). When evaluating dynamic load cases the following methods may be used:

a) equivalent statical load method,

b) response spectrum method,

c) time history method.

(2) The requirements of KTA 2201.4 shall be considered specifically for earthquake load cases.

7.7 Stress analysis

7.7.1 General

(1) By means of a stress analysis along with a classification of stresses and limitation of stress intensities it shall be proved, in conjunction with the material properties, that no inadmissible distortions and especially only limited plastic deformations occur.

(2) The stress analysis for bolts shall be made in accordance with Section 7.11.2.

7.7.2 Classification of stresses

7.7.2.1 General

(1) Stresses shall be classified in dependence of the cause of stress and its effect on the mechanical behaviour of the structure into primary stresses, secondary stresses and peak stresses and be limited in different ways with regard to their classification.

(2) Where in special cases the classification into the aforementioned stress categories is unclear the effect of plastic deformation on the mechanical behaviour shall be determining where an excess of the intended loading is assumed.

Note:

The definitions and terms used hereinafter are taken from the theory of plane load-bearing structures (shells, plates, disks, etc.) and shall be applied accordingly to other load-bearing structures and components (bars, pipes considered to be bars, beams, bolts, fittings, circular ring subject to twisting, etc.). For the stresses mentioned hereinafter distinction is to be made between the various components of the stress tensor.

7.7.2.2 Primary stresses

(1) Primary stresses P are stresses which satisfy the laws of equilibrium of external forces and moments (loads).

(2) Regarding the mechanical behaviour of a structure the basic characteristic of this stress is that in case of (an inadmissibly high) increment of external loads the distortions upon full plastification of the section considerably increase without being self-limiting.

(3) Regarding primary stresses distinction shall be made between membrane stresses (P_m, P_l) and bending stresses (P_b) with respect to their distribution across the cross-section governing the load-bearing behaviour. Here, membrane stresses are defined as the average value of the respective stress component distributed over the section governing the load-bearing behaviour, in the case of plane load-bearing structures the average value of the stress component distributed across the thickness. Bending stresses are defined as stresses that can be altered linearly across the considered section and proportionally to the distance from the neutral axis, in the case of plane load-bearing structures as the portion of the stresses distributed across the thickness, that can be altered linearly.

(4) Regarding the distribution of membrane stresses across the wall distinction is to be made between general primary membrane stresses (P_m) and local primary membrane stresses (P_l). While general primary membrane stresses are distributed such that no redistribution of stresses due to plastification occurs into adjacent regions, plastification in the case of local primary membrane stresses at discontinuities will lead to a redistribution of stresses. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary membrane stress.

At stressed regions (discontinuities) a primary membrane stress may be considered a local primary membrane stress, if the distance over which the membrane stress intensity exceeds 1.1 times the allowable general membrane stress does not extend in the meridional direction more than $1 \cdot \sqrt{R \cdot s_c}$, where R is the minimum mid-surface radius of curvature and s_c is the minimum thickness in the region considered.

Two adjacent regions of local primary membrane stress intensity involving axysymmetric membrane stress redistributions that exceed 1.1 · S_m, shall not be closer in the meridional direction than $2.5 \cdot \sqrt{R \cdot s_c}$ where $R = (R_1 + R_2)/2$ and $s_c = (s_{c,1} + s_{c,2})/2$ where for the radii R_i and the wall thicknesses $s_{c,i}$ of the two regions 1 and 2 considered the locally available values are to be used in accordance with the definition of local primary membrane stress.

Discrete regions of local primary membrane stress intensity resulting from concentrated loads (e.g. acting on brackets) shall be spaced so that there is no overlapping of the areas in which the membrane stress intensity exceeds 1.1 of the allowable general membrane stress.

For components for which the above conditions cannot be satisfied or which do not satisfy the above conditions, the local character of membrane stresses may also be verified by means of a limit analysis as per clause 7.7.4.

7.7.2.3 Secondary stresses

(1) Secondary stresses (Q) are stresses developed by constraints due to geometric discontinuities or by the use of materials of different elastic moduli under external loads, or by constraints due differential thermal expansions. Only stresses that are distributed linearly across the cross-section are considered to be secondary stresses.

(2) With respect to the mechanical behaviour of the structure the basic characteristics of secondary stresses are that they lead to plastic deformation when equalizing different local distortions in the case of excess of the yield strength. Secondary stresses are self-limiting.

(3) Stresses in piping systems developed due to constraints in the system or generally due to fulfilment of kinematic boundary conditions are defined as P_e. Under unfavourable conditions regions with major distortions may develop in relatively long systems, and the constraints thus occurring will then act as external loads. In addition, it shall be demonstrated for these locations that yielding is limited locally.

7.7.2.4 Peak stresses

(1) Peak stress (F) is that increment of stress which is additive to the respective primary and secondary stresses. Peak stresses do not cause any noticeable distortion and are only important to fatigue in conjunction with primary and secondary stresses.

(2) Peak stresses also comprise deviations from nominal stresses at hole edges not reinforced by tubes within tubehole fields due to pressure and temperature in which case the nominal stresses shall be derived from equilibrium of forces considerations.

7.7.3 Superposition and evaluation of stresses

7.7.3.1 General

(1) As shown hereinafter, for each load case the stresses acting simultaneously in the same direction shall be added separately or for different stress categories (e.g. primary and secondary stresses) be added jointly.

(2) The classification of stresses with regard to loads on vessels and piping in dependence of the component and location is shown in **Tables 7.7-1** to **7.7-3**.

(3) From these summed-up stresses the stress intensity for the primary stresses and the equivalent stress range each for the sum of primary and secondary stresses or the sum of primary stresses, secondary stresses and peak stresses shall be derived.

(4) In clauses 7.7.3.2 and 7.7.3.3 the determination of stress intensities and equivalent stress ranges shall be based on the stress theory of von Mises or alternately on the theory of Tresca.

7.7.3.2 Stress intensities

(1) Having chosen a three-dimensional set of coordinates the algebraic sums of all normal and shear stresses acting simultaneously and in consideration of the respective axis direction shall be calculated for

a) the general primary membrane stresses or
b) the local primary membrane stresses or
c) the sum of primary bending stresses and either the general or local primary membrane stresses.

(2) From the superpositioned stress components the stress intensity according to von Mises shall be derived as follows

$$\sigma_{V,v.Mises} = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - (\sigma_x \cdot \sigma_y + \sigma_x \cdot \sigma_z + \sigma_y \cdot \sigma_z) + 3 \cdot (\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2)} \qquad (7.7\text{-}1)$$

(3) When deriving the stress intensity in accordance with the theory of Tresca, the principal stresses shall be determined for each of the three cases (1) a) to c) taking the respective primary shear stresses into account unless the primary shear stresses disappear or are negligibly small so that the effective normal stresses are the principal stresses. In each case the stress intensity then equals the difference between the maximum and minimum principal stress.

$$\sigma_{V,Tresca} = \sigma_{max} - \sigma_{min} \qquad (7.7\text{-}2)$$

(4) For the three cases (1) a) to c) thus the stress intensity is obtained from P_m, P_l and $P_m + P_b$ or $P_l + P_b$.

7.7.3.3 Equivalent stress ranges

(1) To avoid failure due to

a) progressive distortion (ratcheting)
b) fatigue

the stress ranges pertinent to the stress categories shall be determined and be limited in accordance with clause 7.7.3.4.

(2) In case (1) a) the required stress tensors shall be formed taking the simultaneously acting stresses from primary and secondary stress categories, and in case (1) b) taking the simultaneously acting stresses from all stress categories.

(3) From the number of service loadings to be considered two service loadings shall be selected by using one fixed co-ordinate system so that the stress intensity derived from the difference of the pertinent stress tensors becomes a maximum in accordance with the stress theory selected. This maximum value is the equivalent stress range.

(4) Where, upon application of Tresca`s maximum shear stress theory, the loading conditions to be considered show no change in the direction of principal stresses it will suffice to form the maximum value of the differences of any two principal stress differences of equal pairs of principal stress directions. This maximum value then is the equivalent stress range (according to the stress theory of Tresca).

7.7.3.4 Limitation of stress intensities and equivalent stress ranges

(1) The extent of verification for components of test group A1 shall be taken from **Table 7.7-4** insofar as for each stress category the limits of stress intensities and equivalent stress ranges are given in this Table.

(2) For components of test groups A2 and A3 **Table 6.7-1** applies accordingly. For bolts of test groups A1, A2 and A3 **Table 6.7-2** applies.

(3) The limits fixed in **Tables 6.7-1** and **7.7-4** only apply to full rectangular sections, as they are based e.g. on the considered distribution of stresses in shell structures. For other sections the shape factors shall be fixed in dependence of the respective load behaviour.

(4) Where a three-axial tensile stress state is produced, the sum of primary principal stresses shall be limited, except for Loading Level D, to

$$\sigma_1 + \sigma_2 + \sigma_3 \leq 4 \cdot S_m$$

Vessel Part	Location	Origin of Stress	Type of stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane	P_m
			Gradient through plate thickness	Q
		Axial thermal gradient	Membrane	Q
			Bending	Q
	Junction with head or flange	Internal pressure	Membrane [3]	P_l
			Bending	Q [1]
Any shell or head	Any section across entire vessel	External load or moment, or internal pressure [2]	General membrane averaged across full section. (Stress component perpendicular to cross section)	P_m
		External load or moment [2]	Bending across full section. (Stress component perpendicular to cross section)	P_m
	Near nozzle or other opening	External load or moment, or internal pressure [2]	Local membrane [3]	P_l
			Bending	Q
			Peak (fillet or corner)	F
	Any location	Temperature difference between shell and head	Membrane	Q
			Bending	Q
Dished head or conical head	Crown	Internal pressure	Membrane	P_m
			Bending	P_b
	Knuckle or junction to shell	Internal pressure	Membrane	P_l [4]
			Bending	Q
Flat head	Centre region	Internal pressure	Membrane	P_m
			Bending	P_b
	Junction to shell	Internal pressure	Membrane	P_l
			Bending	Q [1]
Perforated head	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section)	P_m
			Bending (averaged through width of ligament, but gradient through plate)	P_b
			Peak	F
	Isolated or atypical ligament	Pressure	Membrane (as above)	Q
			Bending (as above)	F
			Peak	F

Table 7.7-1: Classification of stress intensity in vessels for some typical cases

Vessel Part	Location	Origin of Stress	Type of stress	Classification
Nozzle	Cross section perpendicular to nozzle axis	Internal pressure or external load or moment 2)	General membrane, averaged across full cross section (Stress component perpendicular to section)	P_m 5)
		External load or moment 2)	Bending across nozzle section	P_m 5)
	nozzle wall	Internal pressure	General membrane Local membrane Bending Peak	P_m 5) P_l 5) Q 5) F
		Differential expansion	Membrane Bending Peak	Q Q F
Cladding	Any	Differential expansion	Membrane Bending	F F
Any	Any	Radial temperature distribution 5)	Equivalent linear stress 7) Non-linear stress distribution	Q F
Any	Any	Any	Stress concentration by notch effect	F

1) If the bending moment at the edge is required to maintain the bending stress in the middle of the head or plate within acceptable limits, the edge bending is classified as P_b.

2) To include all pipe end forces resulting from dead weight, vibrations and restraint to thermal expansion as well as inertial forces.

3) Outside the area containing the discontinuity the membrane stress in meridional and circumferential direction of the shell shall not exceed $1{,}1 \cdot S_m$ and the length of this area in meridional direction shall not exceed $1.0 \cdot \sqrt{R \cdot s_c}$.

4) Consideration shall be given to the possibility of wrinkling and excessive deformation in thin-walled vessels (large diameter-to-thickness ratio).

5) The P_m classification for stresses resulting from external forces and moments shall be used for that nozzle area within the limits for reinforcement of openings, as given in Annex A, irrespective of the fact whether the opening is reinforced or not. Outside the limits for reinforcement of openings the P_m classification applies to the membrane stress averaged through the cross section (and not through the wall thickness) resulting from internal pressure and sustained mechanical loads.

6) Consider possibility of failure due to thermal stress ratcheting.

7) The equivalent linear stress is defined as the linear stress distribution which has the same net bending moment as the actual stress distribution.

Table 7.7-1: Classification of stress intensity in vessels for some typical cases (continued)

Piping component	Location	Origin of stress	Type of stress	Classification
Straight pipe or tube, reducers, intersections and branch connections, except in crotch regions	Location remote from discontinuities	Internal pressure	Average membrane stress	P_m
		Sustained mechanical loads incl. dead weight and inertial forces	Bending across section (stress component perpendicular to cross section)	P_b
	Location with discontinuities (wall thickness transitions, connection of different piping components)	Internal pressure	Membrane (through wall thickness) Bending (through wall thickness)	P_l Q
		Sustained mechanical loads incl. dead weight and inertial forces	Membrane (through wall thickness) Bending (through wall thickness)	P_l Q
		Restraint to thermal expansion	Membrane Bending	P_e P_e
		Axial thermal gradient	Membrane Bending	Q Q
	Any	Any	Peak	F
Branch connections and tees	In crotch region	Internal pressure, sustained mechanical loads incl. dead weight and inertial forces as well as restraint to thermal expansion	Membrane Bending	P_l Q
		Axial thermal gradient	Membrane Bending	Q Q
		Any	Peak	F
Bolts and flanges	Remote from discontinuities	Internal pressure, gasket compression, bolt loads	Average membrane	P_m
	Wall thickness transitions	Internal pressure, gasket compression, bolt loads	Membrane Bending	P_l Q
		Axial or radial thermal gradient	Membrane Bending	Q Q
		Restraint to thermal expansion	Membrane Bending	P_e P_e
		Any	Peak	F
Any	Any	Radial thermal gradient 1)	Bending through wall Peak	F F

1) Consider possibility of failure due to thermal stress ratcheting.

Table 7.7-2: Classification of stress intensity in piping for some typical cases

Type of component support structures	Location	Origin of stress	Type of stress	Classification
Any shell	Any section through the entire component support structure	Force or moment to be withstood	General membrane, averaged across full section (stress component perpendicular to cross section)	P_m
		Force or moment to be withstood	Bending across full section (stress component perpendicular to cross section)	P_b
	Near discontinuity 1) or opening	Force or moment to be withstood	Membrane Bending	P_m Q 2)
	Any	Restraint 3)	Membrane Bending	P_e P_e
Any plate or disk	Any	Force or moment to be withstood	Membrane Bending	P_m P_b
	Near discontinuity 1) or opening	Force or moment to be withstood	Membrane Bending	P_m Q 2)
	Any	Restraint 3)	Membrane Bending	P_e P_e

1) Discontinuities mean essential changes in geometry such as wall thickness changes and transitions between different types of shells. Local stress concentrations, e.g. on edges and boreholes are no discontinuities.

2) Calculation not required.

3) These are stresses resulting from restraints of free end displacements or different displacements of component support structures or anchors, including stress intensifications occurring at structural discontinuities, but excluding restraint due to thermal expansion of piping systems. The forces and moments from re-strained thermal expansions of piping systems are considered to be "forces or moments to be withstood" by the component support structure.

Table 7.7-3: Classification of stress intensity of integral areas of component support structures for some typical cases

Stress category		Loading levels					
		Level 0 [1]	Level A	Level B	Level P [2]	Level C [3]	Level D
Primary stresses	P_m	S_m	—	$1.1 \cdot S_m$	$0.9 \cdot R_{p0.2T}$	Higher value of: $R_{p0.2T}$ [4] and $1.2 \cdot S_m$	Smaller value of: $0.7 \cdot R_{mT}$ and $2.4 \cdot S_m$
	P_l	$1.5 \cdot S_m$	—	$1.65 \cdot S_m$	$1.35 \cdot R_{p0.2T}$	Higher value of: $1.5 \cdot R_{p0.2T}$ [4] and $1.8 \cdot S_m$	Smaller value of: R_{mT} and $3.6 \cdot S_m$
	$P_m + P_b$ or $P_l + P_b$	$1.5 \cdot S_m$	—	$1.65 \cdot S_m$	$1.35 \cdot R_{p0.2T}$	Higher value of: $1.5 \cdot R_{p0.2T}$ [4] and $1.8 \cdot S_m$	Smaller value of: R_{mT} and $3.6 \cdot S_m$
Primary plus secondary stresses	P_e	—	$3 \cdot S_m$ [5]	$3 \cdot S_m$ [5] [6]	—	—	—
	$P_m + P_b + P_e + Q$ or $P_l + P_b + P_e + Q$	—	$3 \cdot S_m$ [5]	$3 \cdot S_m$ [5] [6]	—	—	—
Primary plus secondary stresses plus peak stresses	$P_m + P_b + P_e + Q + F$ or $P_l + P_b + P_e + Q + F$	—	$2 \cdot S_a$ [8] $D \leq 1.0$	$2 \cdot S_a$ [7] [8] $D \leq 1.0$	—	—	—

The material strength values shown shall be taken as minimum values.
When using the component specific analysis of the mechanical behaviour in accordance with Section 8 the values indicated in this Section shall apply.
The determination of design stress intensity values S_m is given in **Table 6.6-1**.

1) See **Annex B** as regards the analytical confirmation in case of a numerical reassessment of a component.
2) If the allowable number of cycles of 10 is exceeded, all cycles of this loading level shall be incorporated in the fatigue analysis according to Levels A and B.
3) If the allowable number of cycles of 25 is exceeded, the cycles of this loading level exceeding 25 shall be incorporated in the fatigue analysis according to Levels A and B.
4) However, not more than 90 % of the allowable value in Level D.
5) If the $3 \cdot S_m$ limit (for cast steel $4 \cdot S_m$) is exceeded an elastic plastic analysis shall be made taking the number of cycles into account (see clause 7.8.1). Where the respective conditions are given, this analysis may be a simplified elastic plastic analysis according to clause 7.8.4.
6) Verification is not required for those cases where the loadings from load cases "emergency" and "damage" have been assigned to this level for reasons of functional capability or other reasons.
7) A fatigue evaluation is not required for those cases where the loading from load cases "emergency" and "damage" have been assigned to this level for reasons of functional capability or other reasons and these load cases belong to the group with 25 load cycles of Level C for which no fatigue analysis is required.
8) The stress amplitude S_a and the cumulative usage factor D shall be limited to satisfy Section 7.8.

Table 7.7-4: Limitation of stress intensities and equivalent stress ranges: Allowable values depending on stress category and loading levels for ferritic and austenitic steels including cast steel of test group A1

7.7.4 Limit analysis

Note:
*See **Annex B** as regards the analytical confirmation in case of a numerical reassessment of a component.*

7.7.4.1 General

(1) The requirements given hereinafter apply to plate and shell type components of test groups A1, A2 and A3. They shall not apply to

a) threaded fasteners,

b) structures (e.g. fillet welds) where failure due to local damage may occur,

c) if the possibility of instability of the structure exists.

(2) The limit values for the general primary membrane stress, the local primary membrane stress as well as the primary membrane plus bending stress (elastic analysis) need not be satisfied at any point if it can be proved by means of limit analysis that the specified loadings multiplied with the safety factors given in 7.7.4.2 are below the respective lower bound collapse load.

(3) The lower bound collapse load is that load which is calculated with a fictitious yield stress σ_F as the lower bound (lower bound theorem of limit analysis) by assuming an ideally elastic-plastic behaviour of the material in which case any system of stresses in the structure must satisfy equilibrium. Multi-axial stress conditions shall be calculated by means of the von Mises theory.

7.7.4.2 Allowable loadings

(1) Loading Level 0

For this loading level the following yield stress value shall be used for calculating the lower bound collapse load:

test group A1 components : $\sigma_F = 1.5 \cdot S_m$
test group A2 and A3 components : $\sigma_F = 1.5 \cdot S$

The use of the S_m value may lead, in the case of non-linear elastic materials, to small permanent strains during the first load cycles. If these strains are not acceptable the value of the stress intensity factor shall be reduced by using the strain limiting factors as per **Table 7.7-5**.

The specified load shall not exceed 67 % of the lower bound collapse load.

(2) Loading Level B

For this loading level the following yield stress value shall be used for calculating the lower bound collapse load:

test group A1 components : $\sigma_F = 1.65 \cdot S_m$
test group A2 and A3 components : $\sigma_F = 1.65 \cdot S$

The use of 1.1 times the S_m value may lead, in the case of non-linear elastic materials, to small permanent strains during the first load cycles. If these strains are not acceptable the value of the design stress intensity shall be reduced by using the strain limiting factors as per **Table 7.7-5**.

The specified load shall not exceed 67 % of the lower bound collapse load.

(3) Loading Level C

For this loading level the following yield stress value shall be used for calculating the lower bound collapse load:

test group A1 components : $\sigma_F = 1.8 \cdot S_m$

test group A2 and A3 components : $\sigma_F = 1.8 \cdot S$

The specified load shall not exceed 67 % of the lower bound collapse load.

(4) Loading Level D

For this loading level the following yield stress value shall be used for calculating the lower bound collapse load:

test group A1 components: $\sigma_F = \min \{2.3 \cdot S_m; 0.7 \cdot R_{mT}\}$

test group A2 and A3 components : $\sigma_F = 2.0 \cdot S$

The specified load shall not exceed 90 % of the lower bound collapse load.

(5) Test Level P

For this loading level the following yield stress value shall be used for calculating the lower bound collapse load:

test group A1 components : $\sigma_F = 1.5 \cdot S_m$

Bauteile der Prüfgruppe A2 und A3 : $\sigma_F = 1.5 \cdot S$

The specified load shall not exceed 80 % of the lower bound collapse load.

(6) Where the conditions hereafter are satisfied, the lower bound collapse load obtained from one single calculation with perfect elastic-plastic material behaviour may be converted to the various loading levels proportional to the differing yield stresses:

a) the calculation is based on a geometrically linear calculation model (e.g. no non-linear bearing conditions),

b) the loading is proportional (e.g. if the structure is loaded by pressure and external loads both load portions increase at the same ratio),

c) where more than one material is used, the lowest yield stress applies to the entire component analysed.

Permanent strain %	Factors
0.20	1.00 *)
0.10	0.90
0.09	0.89
0.08	0.88
0.07	0.86
0.06	0.83
0.05	0.80
0.04	0.77
0.03	0.73
0.02	0.69
0.01	0.63

*) For non-linear elastic materials the S_m value may exceed 67 % of the proof stress $R_{p0.2T}$ and attain 90 % of this value at temperatures above 50 °C which leads to a permanent strain of approx. 0.1%. If this strain is not acceptable the S_m value may be reduced by using the factors of this table.

Table 7.7-5: Factors for limiting strains for non-linear elastic materials

7.8 Fatigue analysis

7.8.1 General

7.8.1.1 Objectives and methods to be used

(1) A fatigue analysis shall be made in dependence of the test group and of the type of component to avoid fatigue failure due to cyclic loading.

(2) The basis for fatigue evaluation are the design fatigue curves (**Figures 7.8-1 to 7.8-4**) based on tests carried out at ambient air.

Note:

Cf. Section 4, esp. clause 4.5.

(3) The fatigue curves shown in **Figure 7.8-2** for temperatures equal to or less than 80 °C as well as for temperatures exceeding 80 °C shall apply to the austenitic steels X6CrNiNb18-10 (1.4550) and X6CrNiTi18-10 (1.4541). The fatigue curve shown in **Figure 7.8-3** shall apply to all other austenitic steels.

(4) The equations of the fatigue curves for the steels 1.4550 and 1.4541 shown in **Figure 7.8-2** are:

a) as function $S_a = f(\hat{n}_i)$

$$S_a = 10^{-2} \cdot E \cdot \left[\left(\frac{e^a}{\hat{n}_i} \right)^{\frac{1}{b}} + c \right] \qquad (7.8.1\text{-}1)$$

b) as function $N = f(S_a)$

$$\hat{n}_i = \frac{e^a}{\left(\frac{S_a}{10^{-2} \cdot E} - c \right)^b} \qquad (7.8.1\text{-}2)$$

where

S_a : half stress intensity range in N/mm²

$\hat{n}_i$: allowable number of load cycles

E : modulus of elasticity

The modulus of elasticity $E = 1.79 \cdot 10^5$ N/mm² was used as reference value for the pictured fictitious elastic stress ranges.

The constants a, b and c have the following values:

a) a = 4.400 at T ≤ 80 °C and 4.500 at T > 80 °C

b) b = 2.450 at T ≤ 80 °C and 2.365 at T > 80 °C

c) c = 0.071 at T ≤ 80 °C and 0.0478 at T > 80 °C.

7.8.1.2 Test groups A1 and A2

7.8.1.2.1 Conditions required for performing a fatigue analysis

(1) Fatigue analysis shall be made if at least one of the following conditions applies:

a) The minimum tensile strength R_{mRT} of the material exceeds 550 N/mm².

b) The sum of the cycles due to the loadings mentioned in ba) to bd) hereinafter exceeds 1000 cycles.

ba) expected number of cycles over full range of pressure cycles including start-up and shutdown.

bb) expected number of pressure cycles where the range of pressure variation exceeds 0.2 · p (p = design pressure).

bc) effective number of weighted cycles of changes in metal temperature between any two adjacent points (see sub-clause (2)). Depending on the load case, those two adjacent points shall be selected from all adjacent points for which the highest temperature difference is obtained for the considered point of time. The number of cycles with this temperature difference depends on the specific component. The number of weighted temperature cycles is obtained by multiplying, for each load case, the component-specific number ob cycles with the factor given in **Table 7.8-1** relating to the pertinent highest temperature difference and then summing up the resulting number of cycles of all load cases.

Metal temperature differential [K]	Factor
$\Delta T < 30$	0
$30 \leq \Delta T < 60$	1
$60 \leq \Delta T < 80$	2
$80 \leq \Delta T < 140$	4
$140 \leq \Delta T < 190$	8
$190 \leq \Delta T < 250$	12
$250 \leq \Delta T$	20

Table 7.8-1: Factor for fatigue analysis to consider temperature differences in the material

bd) For vessels with welds joining materials of different thermal expansion coefficients (α_1, α_2) the number of temperature cycles shall be taken into account where the product of $\Delta T \cdot (\alpha_1 - \alpha_2)$ exceeds 0.00034. Here ΔT is the range of operating temperature cycles at the point under consideration.

(2) For adjacent points the following applies:

a) For surface temperature differences:

aa) For surface temperature differences on shells forming surfaces of revolution in the meridional direction, adjacent points are defined as points that are less than the distance $2 \cdot \sqrt{R \cdot s_c}$, where R is the radius measured normal to the surface from the axis of rotation to the midwall and s_c is the thickness of the part at the point under consideration. If the product $R \cdot s_c$ varies, normally the average value of the points shall be used.

ab) For surface temperature differences on surfaces of revolution in the circumferential direction and on flat parts (e.g. flanges and flat heads), adjacent points are defined as any two points on the same surface.

b) For surface temperature differences on surfaces of revolution in the circumferential direction, adjacent points are defined as any two points on the same surface.

7.8.1.2.2 Fatigue analysis methods to be used

(1) The following fatigue analysis methods are permitted:

a) Simplified fatigue evaluation in accordance with clause 7.8.2

This evaluation is based on a limitation of pressure cycle ranges, temperature differences and load stress cyclic ranges with regard to magnitude and number of cycles. If these limits are adhered to, safety against fatigue failure is obtained. This evaluation method is based on a linear-elastic stress strain relationship.

b) Elastic fatigue analysis in accordance with clause 7.8.3

This analysis method shall be used especially if the safety against fatigue failure according to clause 7.8.2 cannot be demonstrated. The elastic fatigue analysis is only permitted if the equivalent stress range resulting from primary and secondary stresses does not exceed a value of $3 \cdot S_m$ for steels and $4 \cdot S_m$ for cast steel.

c) Simplified elastic-plastic fatigue analysis in accordance with clause 7.8.4

This analysis method may be used for load cycles where the equivalent stress range resulting from all primary and secondary stresses exceeds the limit value of $3 \cdot S_m$ for steel and $4 \cdot S_m$ for cast steel, however, these limit values are adhered to by the equivalent stress range resulting from primary and secondary stresses due to mechanical loads. The influences of plastification are considered by using the factor K_e according to clause 7.8.4. In lieu of this K_e value other values may be used in individual cases, which have been proved by experiments or calculation or have been taken from literature. Their applicability shall be verified.

Note:

The literature referenced in [1] contains a proposal for the determination of K_e values.

In addition, it shall be demonstrated that no ratcheting (progressive distortion) occurs.

d) General elastic-plastic fatigue analysis

While the abovementioned methods are based on linear-elastic material behaviour, a fatigue analysis based on the elasto-plastic behaviour of the material may be made in lieu of the abovementioned methods in which case it shall be demonstrated that no progressive distortion (ratcheting) occurs.

(2) For piping the component-specific fatigue analysis of Section 8.5 may be used in lieu of the analysis methods of clauses 7.8.3 and 7.8.4.

(3) For valves the component-specific fatigue analysis of clause 8.4.6 may be used.

(4) For the fatigue analysis of bolts Section 7.11.2 applies.

7.8.1.3 Test group A3

For components of test group A3, except for piping, a fatigue analysis shall be waived. Piping of test group A3 shall be treated like piping of code class A2 with regard to fatigue analysis.

7.8.2 Simplified evaluation of safety against fatigue failure

The peak stresses need not be considered separately in the fatigue evaluation if for the service loadings of level A of the part the following conditions of sub-clauses a) to f) are satisfied.

Note:

Where load cases of level B are to be analysed regarding their fatigue behaviour, the same conditions as for level A apply.

a) Atmospheric to service pressure cycles

The specified number of times (including start-up and shutdown) that the pressure will be cycled from atmospheric pressure to service pressure and back to atmospheric pressure does not exceed the number of cycles on the applicable fatigue curves (see **Figures 7.8-1** and **7.8-3**) corresponding to an S_a value of three times (for steels) and four times (for cast steels) to the S_m value for the material at service temperature.

b) Normal service pressure fluctuations

The specified range of pressure fluctuations during level A Service does not exceed 1/3 times the design pressure, multiplied with the (S_a/S_m) ratio, where S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant pressure fluctuations and S_m is the design stress intensity for the material at service temperature. If the total specified number of significant pressure fluctuations exceeds the maximum number of load cycles obtained from the applicable fatigue curve, the S_a value may be used for maximum number of load cycles in the applicable fatigue curve. Significant pressure fluctuations are those for which the total excursion exceeds the quantity of 1/3 times the design pressure, multiplied by the S/S_m ratio. Here, S is defined as follows:

ba) If the specified number of load cycles is 10^6 or less, the value of S_a at 10^6 load cycles of the applicable fatigue curve applies to S,

bb) If the specified number of load cycles exceeds 10^6, the value of S_a at the maximum number of load cycles in the applicable fatigue curve applies to S.

c) Temperature difference - start-up and shutdown

The temperature difference, K (Kelvin) between any two adjacent points of the component during level A service does not exceed the value of $S_a/(2 \cdot E \cdot \alpha)$, where S_a is the value obtained from the applicable design fatigue curve for the specified number of start-up-shutdown cycles, α is the value of the instantaneous coefficient of thermal expansion at the mean value of the temperatures at the two points, and E is the modulus of elasticity at the mean value of the temperatures at the two points.

Note:

Adjacent points are explained in clause 7.8.1.2.1 (2).

d) Temperature difference for services other than start-up and shutdown

The temperature difference, K (Kelvin), between any two adjacent points is smaller than the value of $S_a/2 \cdot E \cdot \alpha$, where S_a is the value obtained from the applicable design fatigue curve for the total number of significant temperature fluctuations. A temperature difference fluctuation shall be considered to be significant if its total algebraic range exceeds the quantity $S/(2 \cdot E \cdot \alpha)$. Here, S is defined as follows:

da) If the specified number of load cycles is 10^6 or less, the value of S_a at 10^6 load cycles of the applicable fatigue curve applies to S,

db) If the specified number of load cycles exceeds 10^6, the value of S_a at the maximum number of load cycles in the applicable fatigue curve applies to S.

e) Temperature differences for dissimilar materials

For components fabricated from materials of differing moduli of elasticity or coefficients of thermal expansion, the total algebraic range of temperature fluctuation experienced by the component during normal service does not exceed the magnitude $S_a/[2 \cdot (E_1 \cdot \alpha_1 - E_2 \cdot \alpha_2)]$.

Here S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant temperature fluctuations, E_1 and E_2 are the moduli of elasticity, and α_1 and α_2 are the values of the instantaneous coefficients of thermal expansion at the mean temperature value for the two materials. A temperature fluctuation shall be considered to be significant if its total algebraic range exceeds the quantity $S/[2 \cdot (E_1 \cdot \alpha_1 - E_2 \cdot \alpha_2)]$. Here, S is defined as follows:

ea) If the specified number of load cycles is 10^6 or less, the value of S_a at 10^6 load cycles of the applicable fatigue curve applies to S,

eb) If the specified number of load cycles exceeds 10^6, the value of S_a at the maximum number of load cycles in the applicable fatigue curve applies to S.

If the two materials used have different design fatigue curves the smaller value of S_a shall be used when applying this sub-clause.

f) Mechanical loads

The specified full range of mechanical loads, excluding internal pressure, but including pipe reactions, does not result in load stresses whose range exceeds the value of S_a obtained from the applicable design fatigue curve for the total specified number of significant load fluctuations. If the total specified number of significant load fluctuations exceeds the maximum number of load cycles obtained from the applicable fatigue curve, the S_a value may be used for maximum number of load cycles in the applicable fatigue curve. A load fluctuation shall be considered to be significant if the total excursion of load stress exceeds the value S of the applicable fatigue curve. Here, S is defined as follows:

fa) If the specified number of load cycles is 10^6 or less, the value of S_a at 10^6 load cycles of the applicable fatigue curve applies to S,

fb) If the specified number of load cycles exceeds 10^6, the value of S_a at the maximum number of load cycles in the applicable fatigue curve applies to S.

7.8.3 Elastic fatigue analysis

(1) Prerequisite to the application of the elastic fatigue analysis is that the $3 \cdot S_m$ criteria for steels and the $4 \cdot S_m$ criteria for cast steel are satisfied in accordance with clause 7.7.3.4.

(2) As the stress cycles $\sigma_V = 2 \cdot \sigma_a = 2 \cdot E_T \cdot \varepsilon_a$ in level A and B service assume different magnitudes they shall be subdivided in an enveloping manner into several steps $2 \cdot \sigma_{ai}$ and their cumulative damage effect shall be evaluated as follows:

For each type of cycle $\sigma_{ai} = S_a$ the allowable number of cycles $\hat{n}_i$ shall be determined by means of **Figure 7.8-1**, **Figure 7.8-2** or **Figure 7.8-3** and be compared with the specified number of cycles n_i or number of cycles n_i verified by calculation.

The sum of these ratios $n_i/\hat{n}_i$ is the cumulative usage factor D for which the following applies within the design:

$$D = \frac{n_1}{\hat{n}_1} + \frac{n_2}{\hat{n}_2} + \ldots \frac{n_k}{\hat{n}_k} \le 1.0 \qquad (7.8\text{-}1)$$

Where a reduction of fatigue strength due to fluid effects cannot be excluded, then the following measures shall be taken at a threshold for cumulative damage of D = 0.4 to ensure consideration of fluid influence on the fatigue behaviour:

a) the components considered shall be included in a monitoring program to KTA 3211.4, or

b) experiments simulating operating conditions shall be performed, or

c) verifications by calculation shall be made in due consideration of fluid-effected reduction factors and realistic boundary conditions.

Note:

*See explanations regarding section 7.8 in **Annex D** with regard to attention thresholds for austenitic steels in the case that fatigue evaluations are not made on the basis of the fatigue curves in **Figures 7.8-2** and **7.8-3**.*

7.8.4 Simplified elastic plastic fatigue analysis

Within the simplified elastic-plastic analysis the $3 \cdot S_m$ limit for steels and $4 \cdot S_m$ limit for cast steel with a stress cycle range resulting from primary and secondary stresses may be exceeded if the requirements in a) to e) hereinafter are met.

a) The equivalent stress range resulting from primary and secondary membrane and bending stresses without thermal bending stresses across the wall shall be not greater than $3 \cdot S_m$ for steel and $4 \cdot S_m$ for cast steel. The limitation of thermal stress ratcheting shall be demonstrated (compare e.g. clause 7.12.3 and 8.5.2.4.4.1 b).

b) The value of half the equivalent stress range S_a to be compared with the design fatigue curve acc. to **Figure 7.8-1,**

Figure 7.8-2 or **Figure 7.8-3** shall be multiplied with the factor K_e where K_e is to be determined for steel as follows:

$K_e = 1.0$ for $S_n \le 3 \cdot S_m$ (7.8-2)

$$K_e = 1.0 + \frac{(1-n)}{n \cdot (m-1)} \cdot \left(\frac{S_n}{3 \cdot S_m} - 1 \right) \quad \text{for } 3 \cdot S_m < S_n < m \cdot 3 \cdot S_m \qquad (7.8\text{-}3)$$

$K_e = 1/n$ for $S_n \ge m \cdot 3 \cdot S_m$ (7.8-4)

S_n : Range of primary plus secondary stress intensity

In the foregoing equations the $3 \cdot S_m$ value shall be substituted by $4 \cdot S_m$ for cast steel.

The material parameters m and n shall be taken from **Table 7.8-2**.

The temperature for the material used shall not exceed the value of T_{max} in **Table 7.8-2**.

Type of material	m	n	T_{max} (°C)
Low alloy carbon steel	2.0	0.2	370
Martensitic stainless steel	2.0	0.2	370
Carbon steel	3.0	0.2	370
Austenitic stainless steel	1.7	0.3	425
Nickel based alloy	1.7	0.3	425

Table 7.8-2: Material parameter

For local thermal stresses the elastic equations may be used in the fatigue analysis. The Poisson`s ratio ν shall be determined as follows:

$$\nu = 0.5 - 0.2 \cdot \left(\frac{R_{p0.2T}}{S_a} \right), \text{ but not less than } 0.3 \qquad (7.8\text{-}5)$$

where

$$T = 0.25 \cdot \breve{T} + 0.75 \cdot \hat{T} \qquad (7.8\text{-}6)$$

with

$\hat{T}$ maximum temperature at the considered load cycle

$\breve{T}$ minimum temperature at the considered load cycle

c) Deviating from b) K_e values may be determined by calculation or experiments or be taken from literature. Their usability shall be demonstrated.

d) The limitation of the cumulative usage factor due to fatigue shall be in acc. with clause 7.8.3.

7.9 Strain analysis

A strain analysis shall only be made if specified strain limits are to be adhered to for functional reasons.

7.10 Structural analysis

Where under the effect of loading a sudden deformation without considerable increase in load may be expected, a structural analysis shall be performed.

7.11 Stress, strain and fatigue analyses for flanged joints

7.11.1 General

(1) The loading conditions of flanged joints shall be determined for the governing load cases. The verification by calculation of the strength and deformation conditions may be made by approximation in accordance with the simplified procedure of clause A 2.10.5. The exact verification shall be made according to this section in consideration of the elastic behaviour of the structure. Dimensioning and strain analysis may be made in accordance with Sections A 2.9 and A 2.10.

(2) The following shall be included, where required, in the structure:

a) identical flange pairs, non-identical flange pairs or the flange with flat or dished cover

b) bolts

c) the gasket and

d) the connected shell.

(3) The following load cases shall be examined:

a) the bolting-up condition(s)

b) the conditions of specified operation

c) upset conditions (incidents), if any.

(4) The loadings on the flanged joint in the load cases of specified operation and incidents, if any, shall be calculated in connection with the respective bolting-up condition e.g. taking consistent bolt elongation into account (definition see under clause A 2.10.6.1).

(5) For the flanges, the covers, if any, belonging to the flanged joint and the connected shell a stress analysis and limitation as per Section 7.7 and a fatigue analysis as per Section 7.8 shall be performed. The stresses shall be limited in accordance with **Table 6.7-2**. For bolts a stress and fatigue analysis as per clause 7.11.2 is required.

(6) The assessment of the gasket loading condition shall be made based on verified data of the gasket manufacturer, e.g. from gasket-data-sheets (see Section A 2.11). The residual gasket load shall be controlled according to the respective requirements in due consideration of the seating conditions.

7.11.2 Stress and fatigue analysis for bolts

(1) When evaluating stress limits for bolts the following stresses are referred to: average tensile stresses, bending stresses, torsional stresses, and peak stresses.

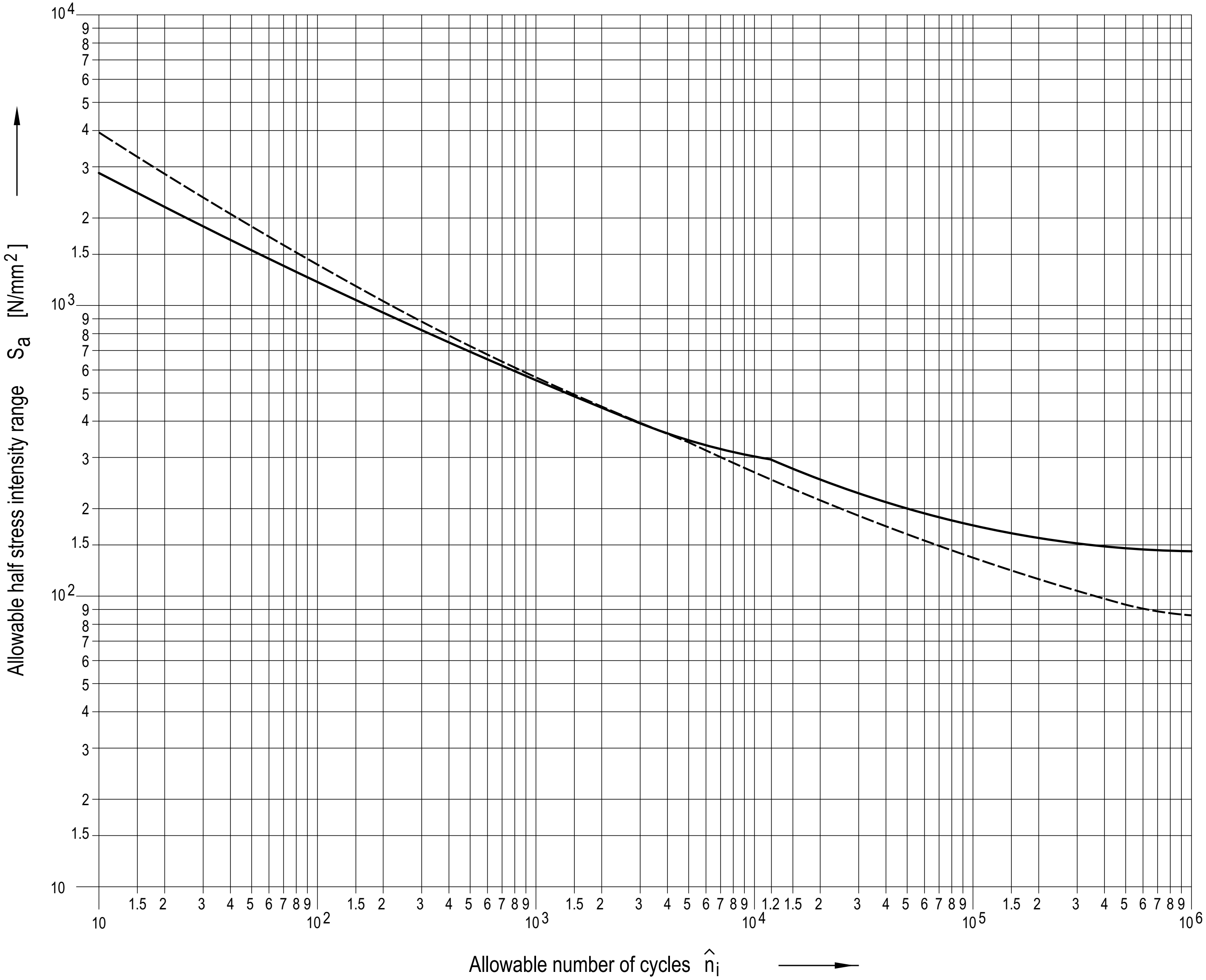
(2) A specific fatigue analysis shall be made if the bolts are not covered by the simplified evaluation of safety against fatigue failure of the component in acc. with clause 7.8.2. In this fatigue analysis the material properties and geometric boundary conditions of threaded members shall be considered e.g. when determining the load cycles resulting from pressure fluctuations and temperature differences.

(3) The allowable stress limits for bolts are contained in **Table 6.7-2**.

(4) The fatigue behaviour shall be evaluated on the basis of the range of maximum stress intensity in due consideration of the elasticity of threaded members, in which case the range of normal stress intensity shall be multiplied with a fatigue strength reduction factor of not exceeding 4. The usage factor shall be accumulated and be limited in acc. with equation (7.8.-1).

Fatigue strength reduction factors smaller than 4 shall be verified.

(5) For bolts with a specified tensile strength R_{mRT} not exceeding 690 N/mm^2 the design fatigue curves acc. to **Figures 7.8-1**, **7.8-2** or **7.8-3** apply, and for high-strength bolts with specified tensile strength R_{mRT} above 690 N/mm^2 the design fatigue curve for temperatures up to and including 370 °C of **Figure 7.8-4** applies. These bolts shall be designed as necked-down bolt in accordance with clause A 2.9.3 (2).



- - - - $R_m \leq 550$ N/mm^2
——— $R_m = 790$ to 900 N/mm^2
$E = 2.07 \cdot 10^5$ N/mm^2

Values for tensile strengths between 550 N/mm^2 and 790 N/mm^2 may be subject to straight interpolation.

Where the calculated stress intensity range is based on strains with an elastic modulus $E_T \neq E$ the calculated stress intensity range shall be multiplied with the ratio E/E_T.

Note:
The exact values to be used for the relationship between S_a and $\hat{n}_i$ are given in Table 7.8-3.

Figure 7.8-1: Design fatigue curves for ferritic steels

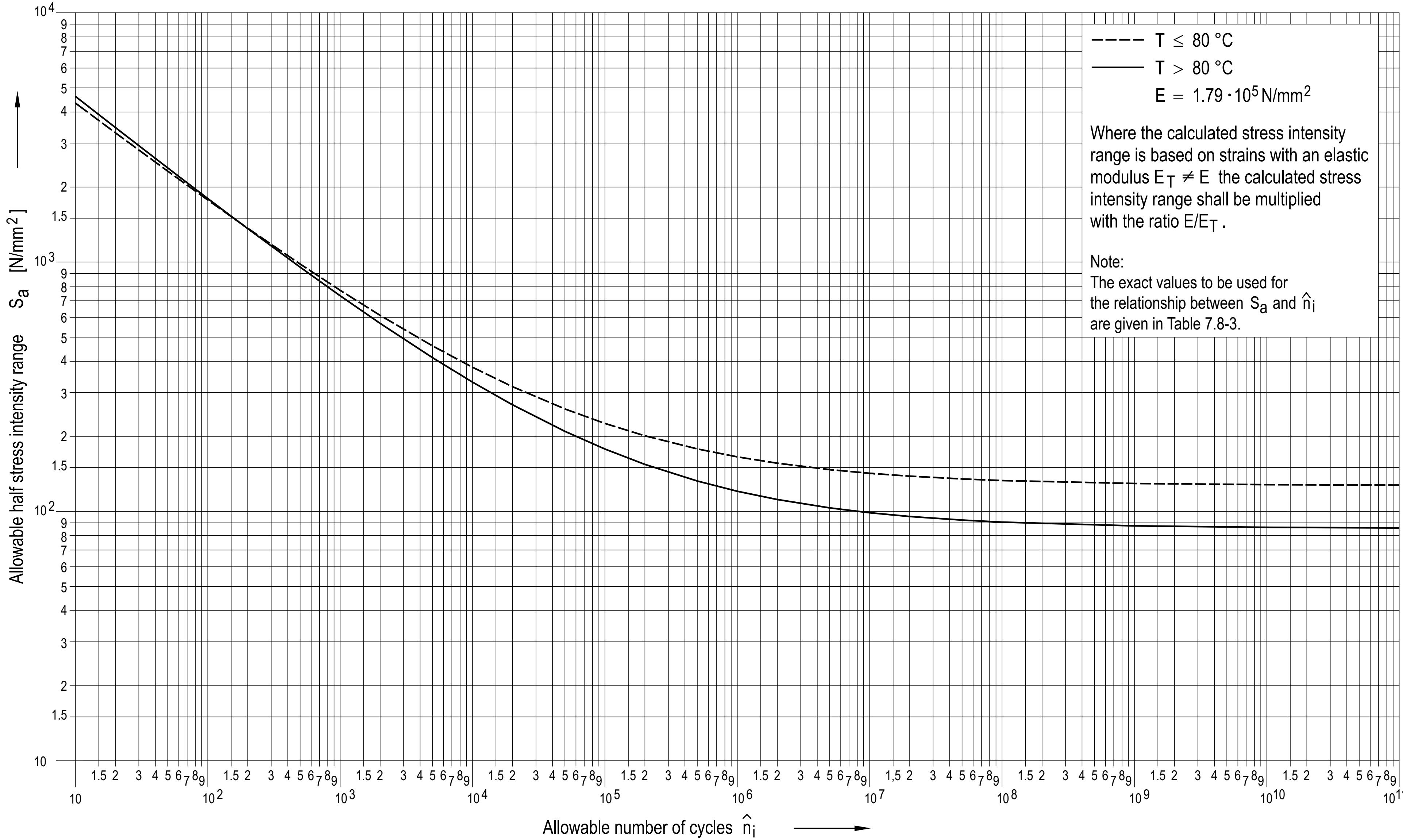


Figure 7.8-2: Design fatigue curves for the austenitic steels 1.4550 and 1.4541

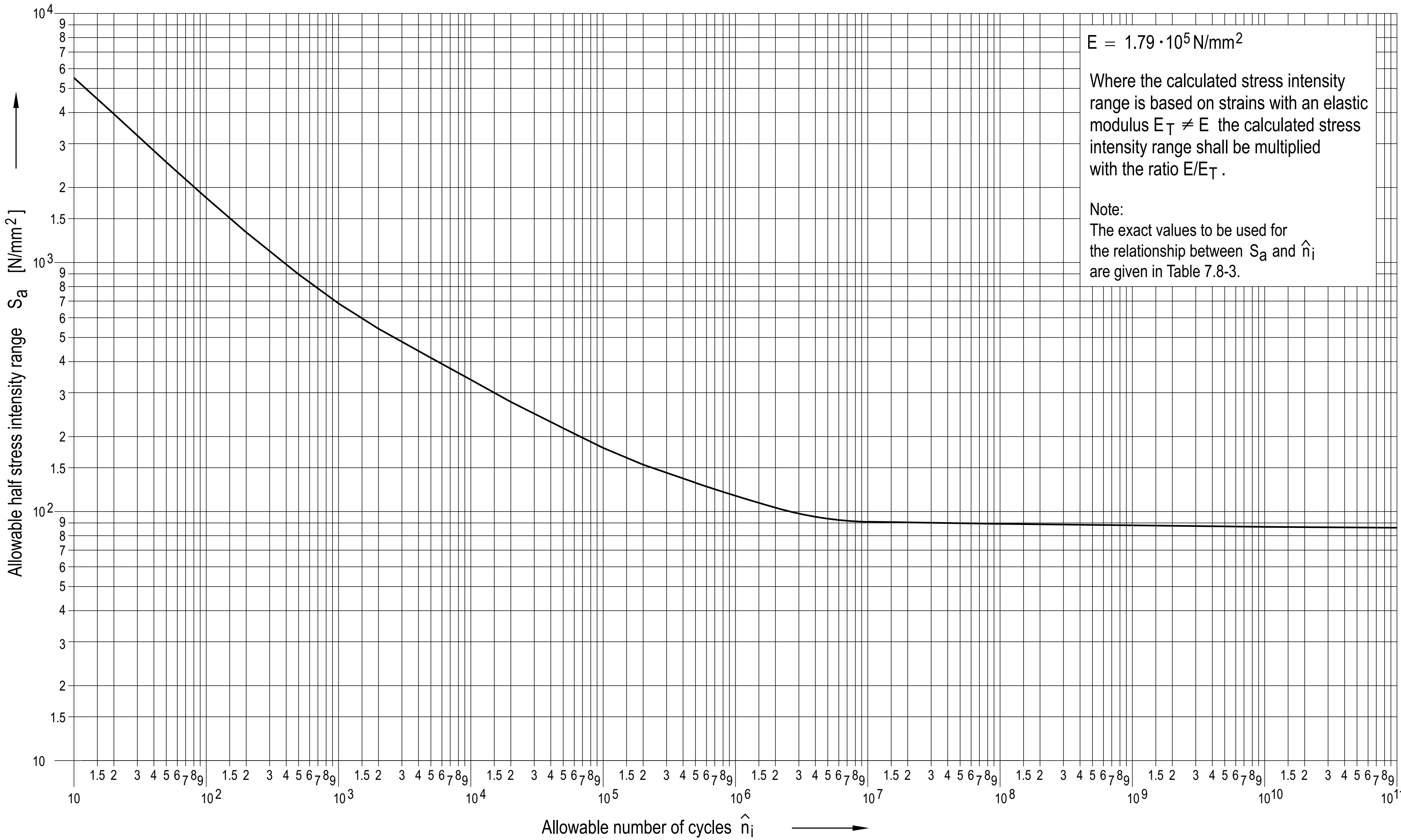


Figure 7.8-3: Design fatigue curve for austenitic steels except the steels 1.4550 and 1.4541

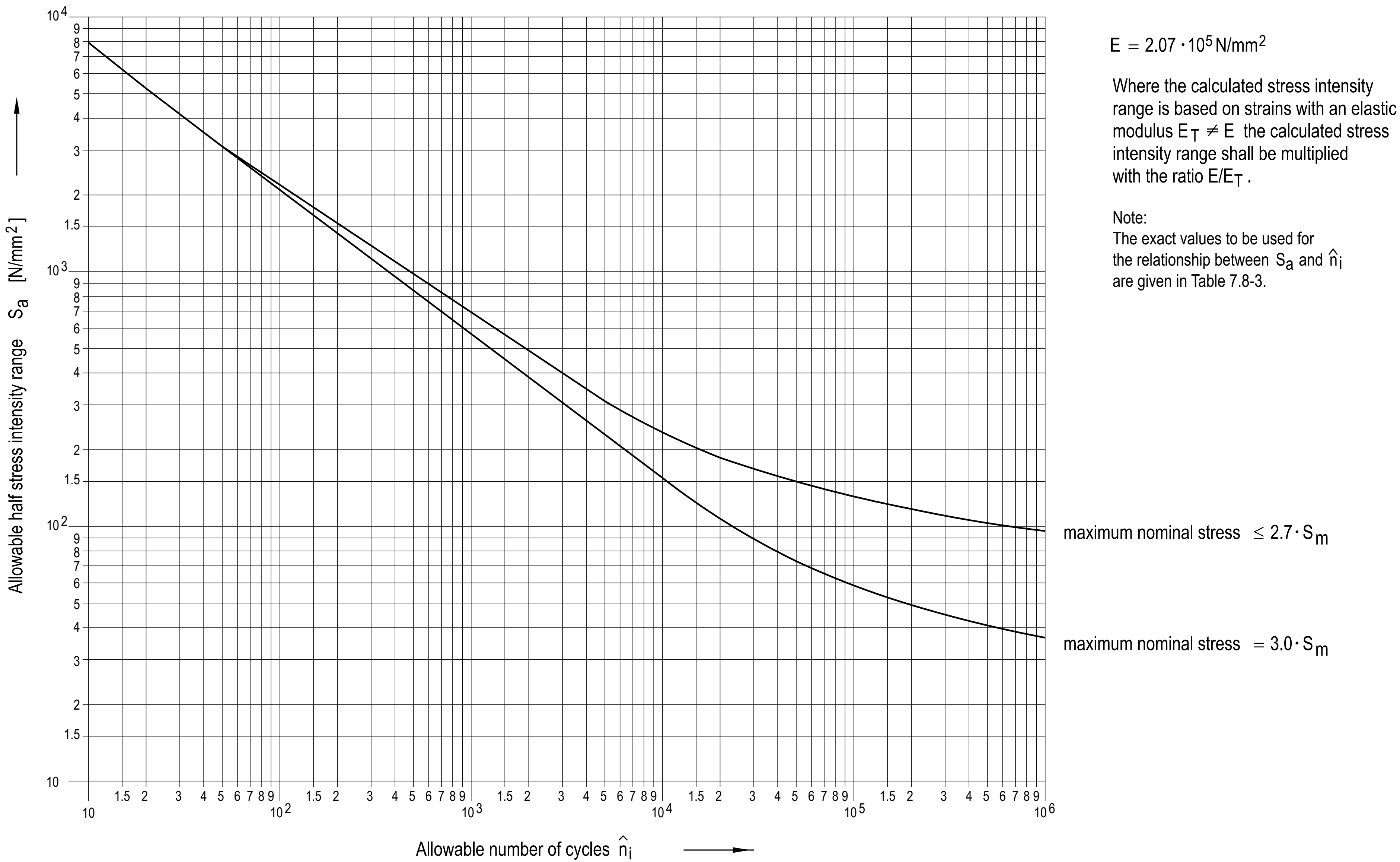


Figure 7.8-4: Design fatigue curve for high strength steel bolting for temperatures ≤ 370 °C

Figure	Allowable half stress intensity range S_a 1)2) at allowable number of load cycles $\hat{n}$																									
	$1 \cdot 10^1$	$2 \cdot 10^1$	$5 \cdot 10^1$	$1 \cdot 10^2$	$2 \cdot 10^2$	$5 \cdot 10^2$	$1 \cdot 10^3$	$2 \cdot 10^3$	$5 \cdot 10^3$	$1 \cdot 10^4$	$1.2 \cdot 10^{4*}$	$2 \cdot 10^4$	$5 \cdot 10^4$	$1 \cdot 10^5$	$2 \cdot 10^5$	$5 \cdot 10^5$	$1 \cdot 10^6$	$2 \cdot 10^6$	$5 \cdot 10^6$	$1 \cdot 10^7$	$2 \cdot 10^7$	$5 \cdot 10^7$	$1 \cdot 10^8$	$1 \cdot 10^9$	$1 \cdot 10^{10}$	$1 \cdot 10^{11}$
7.8-1: curve tensile strength 790 - 900 N/mm²	2900	2210	1590	1210	931	689	538	427	338	303	296	248	200	179	165	152	138									
7.8-1: curve tensile strength ≤ 550 N/mm²	4000	2830	1900	1410	1070	724	572	441	331	262	—	214	159	138	114	93.1	86.2									
7.8-2 T ≤ 80 °C	4341	3302	2312	1773	1368	981	770	612	461	378	—	316	257	225	201	178	165	156	147	142	138	135	133	129	128	127
7.8-2 T > 80 °C	4618	3467	2381	1798	1363	953	732	568	413	330	—	268	209	178	154	132	120	112	103	99	95	92	91	87	86	86
7.8-3	5508	3947	2522	1816	1322	894	684	542	413	338	—	275	216	180	154	130	116	104	94	91	—	—	89	88	87	86
7.8-4: curve maximum nominal stress 3) ≤ 2.7 · S_m	7930	5240	3100	2210	1550	986	689	490	310	234	—	186	152	131	117	103	93.1									
7.8-4: curve maximum nominal stress 3) = 3.0 · S_m	7930	5240	3100	2070	1415	842	560	380	230	155	—	105	73	58	49	42	36.5									

1) The values of S_a shown here are based on the respective elastic moduli of **Figures 7.8-1** to **7.8-4**.

2) Straight interpolation between tabular values is permitted based upon a double logarithmic representation: (straight lines between the data points on the log log plot). Where for a given value of S_a = S the pertinent number of load cycles $\hat{n}$ is to be determined, this shall be done by means of the adjacent data points $S_j < S < S_i$ and $n_j > n > n_i$ as follows:

$$\hat{n} / \hat{n}_i = \left(\hat{n}_j / \hat{n}_i\right)^{\log\frac{S_i}{S} / \log\frac{S_i}{S_j}}$$

Example: Given: Steel with tensile strength ≤ 550 N/mm², S_a = 370 N/mm²

from which follows: S_i = 441 N/mm², S_j = 331 N/mm², $\hat{n}_i = 2 \cdot 10^3$, $\hat{n}_j = 5 \cdot 10^3$

$$\hat{n} / 2000 = (5000 / 2000)^{\log\frac{441}{370} / \log\frac{441}{331}}$$

$$\hat{n} = 3500$$

3) Nominal stress = tensile stress + bending stress

* This data point is included to provide accurate representation of the curve.

Table 7.8-3: Table of values for the design fatigue curves of Figures 7.8-1 to 7.8-4

7.12 Avoidance of thermal stress ratcheting for components of test group A1

7.12.1 General

(1) Where the equivalent stress intensity range derived from primary stresses P and secondary stresses Q exceeds the value of $3 \cdot S_m$ for steels and $4 \cdot S_m$ for cast steels (see clause 7.8.1), it shall be proved my means of the following stipulations that the distortions developing as a result of stress ratchet remain within acceptable limits.

(2) When evaluating the limitation of progressive distortions under cyclic loading the same load cases and combination of these load cases as verified by means of fatigue analysis shall be considered.

(3) The evaluation of limitation of thermal stress ratcheting may be a simplified evaluation (clause 7.12.2) using approximation formulae; more exact evaluations require verification of strains by elasto-plastic analysis (clause 7.12.3) or by means of measurements (clause 7.12.4).

7.12.2 Simplified evaluation by approximation formulae

7.12.2.1 Range of application

(1) The simplified evaluation may be used for:

a) axisymmetric structures under axisymmetric loading conditions, which are located sufficiently away from local structural discontinuities, or

b) general structures where thermal peak stresses are negligible (i.e. linear thermal stress distribution through the wall).

(2) The evaluations are based on the results of elastic analysis and a stress classification in accordance with clause 7.7.3; here the following stress parameters referred to the elevated temperature proof stress $R_{p0.2T}$ are used:

$$X = (P_l + P_b/K)_{max}/R_{p0.2T} \qquad (7.12\text{-}1)$$

$$Y = (Q_R)_{max}/R_{p0.2T} \qquad (7.12\text{-}2)$$

$$\text{where } T = 0.25 \cdot \breve{T} + 0.75 \cdot \hat{T} \qquad (7.12\text{-}3)$$

(referred to the respective load cycle considered) with

$(P_l + P_b/K)_{max}$	maximum value of primary stress intensity where the portion of bending stress P_b has been adjusted with the factor K,
$(Q_R)_{max}$	maximum secondary stress intensity,
$\hat{T}$	maximum temperature,
$\breve{T}$	minimum temperature,
K	factor, e.g. K = 1.5 for rectangular cross-sections.

(3) Where the conditions of clause 7.12.2.1 (1) a) are satisfied, the stress relationships are simplified as follows:

X: maximum general membrane stress due to internal pressure, divided by $R_{p0.2T}$, and

Y: maximum allowable range of thermal stress, divided by $R_{p0.2T}$.

(4) The use of the yield strength instead of the proportional elastic limit allows a small amount of growth during each cycle until strain hardening raises the proportional elastic limit to the yield strength.

(5) This evaluation procedure can be applied as long as the load cycle number to be assessed does not exceed the value

$$n = \hat{n}\,(2 \cdot S_a = R_{p0.2T}) \qquad (7.12\text{-}4)$$

7.12.2.2 Evaluation by limitation of stresses

(1) If the evaluation requirements are met thermal stress ratcheting can definitely be excluded.

(2) When calculating the allowable secondary stress intensity the secondary stress parameter Y may be multiplied with the higher value of $R_{p0.2T}$ or $1.5 \cdot S_m$.

(3) At given primary stress parameter X the following secondary stress parameter Y is permitted for the stress intensity range:

Case 1: Linear variation of temperature or linear variation of secondary stress through the wall:

for $0.0 < X \leq 0.5$, $\quad Y = 1/X \qquad (7.12\text{-}5)$

for $0.5 < X < 1.0$, $\quad Y = 4\,(1-X) \qquad (7.12\text{-}6)$

Case 2: Parabolic constantly increasing or constantly decreasing variation of temperature through the wall:

for $0.615 \leq X \leq 1.0$, $Y = 5.2\,(1-X) \qquad (7.12\text{-}7)$

for $X < 0.615$, $\quad Y\,(X=0.5) = 2.70$

$Y\,(X=0.4) = 3.55$

$Y\,(X=0.3) = 4.65$

Case 3: Any component geometry and any loading:

for $X \leq 1.0$, $Y = 3.25\,(1-X) + 1.33\,(1-X)^3 + 1.38\,(1-X)^5 \qquad (7.12\text{-}8)$

Guide values: $\quad Y\,(X=1.0) = 0.00$

$Y\,(X=0.0) = 5.96$

7.12.2.3 Evaluation by limitation of strains

(1) This evaluation shall only be used for conditions as per clause 7.12.2.1 (1) a).

(2) When determining the strains, the following conditions identified by the index i are considered:

Index 1 the lower bound at extreme value formation of the range of thermal stresses or secondary stresses (low temperature) and with

Index 2 the upper bound at extreme value formation of the range of thermal stresses or secondary stresses (high temperature).

(3) Where the stress parameters

X_1, Y_1 are determined by using the elevated temperature proof stress $R_{p0.2T_1}$ at temperature T_1 averaged across the wall for condition 1

X_2, Y_2 are determined by using the elevated temperature proof stress $R_{p0.2T_2}$ at temperature T_2 averaged across the wall for condition 2,

distinction shall be made between the following cases when determining the auxiliary values of Z_i (i=1.2):

a) for $Y_i \cdot (1-X_i) > 1$, $\quad Z_i = X_i \cdot Y_i \qquad (7.12\text{-}9)$

b) for $Y_i \cdot (1-X_i) \leq 1$ and $X_i + Y_i > 1$,

$$Z_i = Y_i + 1 - 2 \cdot \sqrt{(1 - X_i) \cdot Y_i} \qquad (7.12\text{-}10)$$

c) for $X_i + Y_i \leq 1$, $\quad Z_i = X_i \qquad (7.12\text{-}11)$

(4) From this the plastic strain increment $\Delta\varepsilon$ for each cycle can be derived in dependence of the auxiliary value Z_i and in consideration of the ratio of the proof stress values $\rho = R_{p0.2T_2} / R_{p0.2T_1}$

$$Z_1 \leq \rho: \quad \Delta\varepsilon = 0 \qquad (7.12\text{-}12)$$

$\rho < Z_1 \leq 1$:

$$\Delta\varepsilon = \frac{R_{p0.2T_2} \cdot (Z_1 / \rho - 1)}{E_{T_2}} \qquad (7.12\text{-}13)$$

if $(Z_2 \cdot \rho - 1) \leq 0$

$$\Delta\varepsilon = \frac{R_{p0.2T_2} \cdot (Z_1 / \rho - 1) + R_{p0.2T_1} \cdot (Z_2 \cdot \rho - 1)}{E_{T_2}} \qquad (7.12\text{-}14)$$

if $(Z_2 \cdot \rho - 1) > 0$

$Z_1 > 1$:

$$\Delta\varepsilon = \frac{R_{p0.2T_1} \cdot (Z_1 - 1)}{E_{T_1}} \qquad (7.12\text{-}15)$$

if $(Z_2 - 1) \leq 0$

$$\Delta\varepsilon = \frac{R_{p0.2T_1} \cdot (Z_1 - 1)}{E_{T_1}} + \frac{R_{p0.2T_2} \cdot (Z_2 - 1)}{E_{T_2}} \qquad (7.12\text{-}16)$$

if $(Z_2 - 1) > 0$

(5) The sum of plastic strain increments $\Delta\varepsilon$ to the end of service life shall not exceed the value 2 %.

7.12.3 General evaluation by elastic-plastic analysis

(1) For the determination of plastic strains at cyclic loading an elasto-plastic analysis may be made. The material model used in this analysis shall be suited to realistically determine the cyclic strains.

(2) Where in the case of strain hardening materials the decrease of the strain increment from cycle to cycle is to be taken for the determination of the total strain, the load histogram shall comprise several cycles. From the strain history determined from the respective load histogram the maximum accumulated strain may be calculated by conservative extrapolation.

(3) At the end of service life, the locally accumulated principal plastic tensile strain shall not exceed, at any point of any cross section, the following maximum value: 5.0% in the base metal, 2.5% in welded joints.

7.12.4 Specific evaluation by measurement

(1) The cyclic accumulated strain may also be determined by means of measurements.

(2) Regarding an extrapolation for accumulated total plastic strain as well as the limits of allowable strain clause 7.12.3 applies.

8 Component-specific analysis of the mechanical behaviour

8.1 General

(1) The following component-specific analyses and verifications of strength are recognised and usually applied calculation methods. Where several methods are given, they are permitted within their application limits.

Note:

These procedures are usually based on different principles and contain varying conservative approaches so that non-identical results may be obtained.

(2) The component-specific analyses of the mechanical behaviour are intended to evaluate loadings and replace, fully or in part, the verification by the general analysis of the mechanical behaviour in acc. with Section 7 on the condition that the respective design and loading limit requirements as well as the pertinent specified stress limits are met.

(3) Where effective loading cannot fully be determined by one of the following component-specific analyses, the stresses resulting from partial loadings may be evaluated separately and be determined accordingly by superposition.

(4) As welds have to meet the requirements of KTA 3211.1 and KTA 3211.3 the effects of the welds on the allowable stresses in Section 8 need not be considered separately.

(5) Where fatigue analyses are performed, the fatigue strength-reducing influences of welds in dependence of weld dressing shall be taken into account.

Note:

*Stress indices are contained in **Table 8.5-1**.*

(6) Where a component-specific analysis is performed, the wall thickness s_c as per clause 7.1.4 shall be used. In such case, a cladding shall be considered in conformance with clause 7.1.3.

8.2 Vessels

8.2.1 Radial nozzles subject to internal pressure and external nozzle loadings due to connected piping

8.2.1.1 General

(1) Nozzles in pressure-retaining cylindrical or spherical shells including the attachment-to-shell juncture shall be able to withstand all loadings applied simultaneously, such as internal pressure and external nozzle loadings.

(2) Depending on the respective service limits, test group and stress category the allowable stress intensities shall be determined in accordance with **Tables 6.7-1** and **7.7-4**.

(3) The requirements regarding the design according to Section 5.2 shall be met.

(4) The methods indicated in clause 8.2.1.4 do not consider the effects of mutual influence by adjacent openings which, however, are to be taken into account if the distance between adjacent openings is less than $2 \cdot \sqrt{d_{Hm} \cdot s_H}$.

8.2.1.2 Nozzles not exceeding DN 50

If nozzles $\leq$ DN 50 are dimensioned to withstand internal pressure by means of the equations of Annex A 2.8, analyses of the mechanical behaviour are not required.

8.2.1.3 Nozzles exceeding DN 50 mainly subject to internal pressure

(1) If nozzles that are mainly subject to internal pressure, such as manhole, blanked-off and other nozzles not connected to piping, are dimensioned in accordance with Annex A 2.8, analyses of the mechanical behaviour are not required.

(2) This shall also apply to nozzles with connected piping if the additional primary stresses resulting from external loads do not exceed 5 % of the allowable primary stress values given in **Table 6.7-1**.

8.2.1.4 Nozzles exceeding DN 50 subject to internal pressure and external nozzle loadings

(1) The opening reinforcement shall first be dimensioned for internal pressure in acc. with Annex A 2.8 to include reserves for external nozzle loadings.

The nozzle wall thickness shall be at least 1.5 times the nominal wall thickness of a connected piping (see also **Table 5.2-1**).

To verify the acceptability of external nozzle loads a supplementary stress evaluation shall be made to cover stresses due to internal pressure and external nozzle loadings.

(2) To determine the stresses due to internal pressure the two methods described hereinafter are permitted:

a) Method 1: It is based on a parameter study assuming ideally elastic material behaviour. With this method the stress compo-nents of membrane as well as membrane plus bending stresses can be determined using stress indices. These stress indices refer to planes normal to the vessel wall which govern the combination of stresses resulting from mechanical loads and internal pressure. This method shall preferably be used for components of test group A1.

b) Method 2: This method exclusively leads to maximum total stress intensities due to internal pressure at certain given locations.

(3) External loads may be considered separately using the methods described in clause 8.2.2.4.

(4) The combination of stresses due to internal pressure and external loads and their limitation shall be in accordance with clause 8.2.3.

(5) The calculation methods 1 and 2 do not cover stresses in the nozzle wall outside the nozzle-to-shell transition. For nozzles with a wall thickness ratio $s_A/s_R \leq 1.5$ according to **Figure 8.2-1** or **8.2-2** the stress in the nozzle wall shall therefore be evaluated separately.

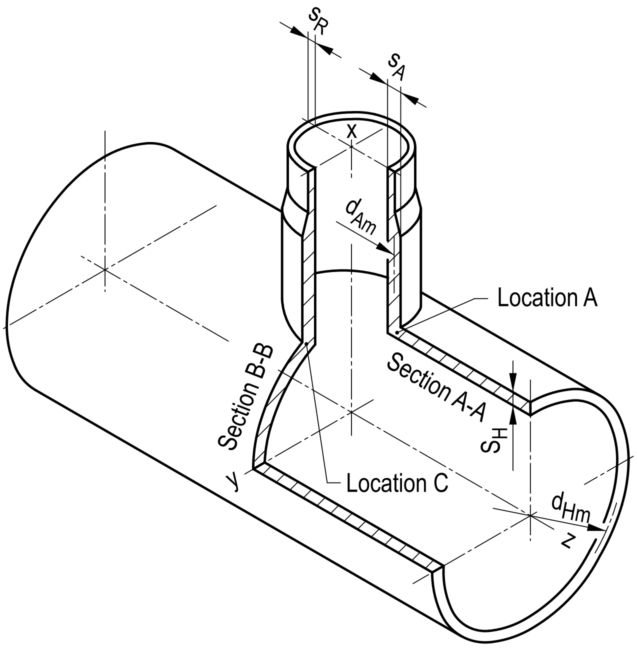


Figure 8.2-1: Nozzle in cylindrical shell

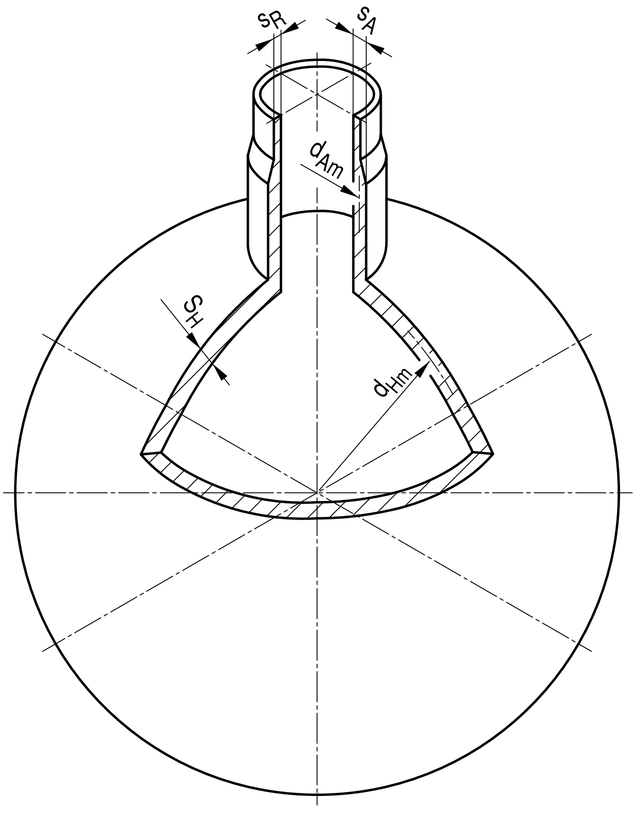


Figure 8.2-2: Nozzle in spherical shell

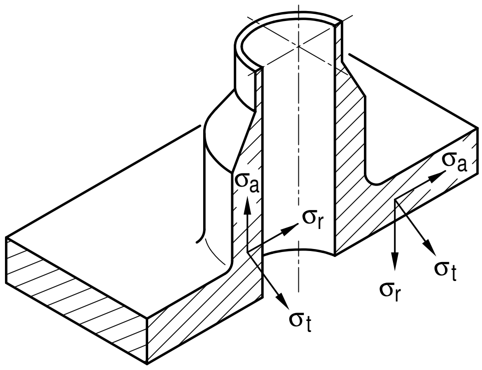


Figure 8.2-3: Direction of stress components

8.2.2 Method of analysis for radial nozzles

8.2.2.1 Design values and units relating to clause 8.2.2

Notation	Design value	Unit
d_{Am}	mean diameter of nozzle	mm
d_{HM}	mean diameter of unpenetrated shell	mm
d_i	internal diameter or spherical radius of head	mm
p	internal pressure	MPa
$p_{zul\ s}$	allowable pressure at shell-to-nozzle transition for a given geometry by utilisation of design stress	MPa
$p_{zul\ u}$	allowable pressure for unpenetrated shell by utilisation of allowable stress	MPa
s_A	nozzle wall thickness	mm
s_c	wall thickness in unreinforced region in acc. with clause 7.1.4	mm
s_H	thickness of shell	mm
s_R	nominal wall thickness of connected pipe	mm
Q	secondary membrane or bending stress	N/mm^2
P_L	local membrane stress	N/mm^2
S_m	design stress intensity for components of test group A1	N/mm^2
α	Stress index for P_L or P_L + Q depending on location and direction of stress in acc. with Figures 8.2-6 to 8.2-13	—
β_k	stress index for radial nozzles in spherical shell	—
β_z	stress index for radial nozzles in cylindrical shell	—
φ	angle between branch and run	degree
σ_a	stress component in axial direction (in the plane of the section under consideration and parallel to the boundary of the section)	N/mm^2
σ_r	stress component in radial direction (normal to the boundary of the section)	N/mm^2
σ_t	stress component in circumferential direction (normal to the plane of the section)	N/mm^2
σ_V	stress intensity (combined stress)	N/mm^2

8.2.2.2 Method 1: Stress index method for primary and secondary stress due to internal pressure

Note:

This method is suited to determine stresses for superposition with stresses resulting from external loadings. It does not result in peak stresses and therefore no total stess intensity is obtained.

To determine primary or primary plus secondary stresses in the shell e.g. for cylindrical and spherical shells, the following stress index method may be used:

a) Radial nozzles in cylindrical shells

The following dimensional ratios shall be adhered to:

Diameter-to-wall thickness ratio $30 \leq d_{Hm}/s_H \leq 200$

Wall thickness ratio $0.75 \leq s_A/s_H \leq 1.3$

Diameter ratio $d_{Am}/d_{Hm} \leq 0.6$

To cover stresses in the transitional area of shell-to-nozzle juncture the strebssses at the locations A and C shall be determined and limited in accordance with **Figure 8.2-1**.

The stresses due to internal pressure are determined as follows:

$$\sigma = \alpha \cdot \frac{d_{Hm}}{2 \cdot s_H} \cdot p \qquad (8.2\text{-}1)$$

The stress indices α shall be taken from the figures laid down in **Table 8.2-1** depending on the referred nozzle diameter $d_{Am} / \sqrt{d_{Hm} \cdot s_H}$ and the wall thickness ratio s_A/s_H.

Location	Stress category	Figure
A	P_L	8.2-4
C	P_L	8.2-5
A Inside	$P_L + Q$	8.2-6
C Inside	$P_L + Q$	8.2-7
A Outside	$P_L + Q$	8.2-8
C Outside	$P_L + Q$	8.2-9

Table 8.2-1: Assignment of stress indices α for cylindrical shells

b) Radial nozzles in spherical shells

The following dimensional ratios shall be adhered to:

Diameter-to-wall thickness ratio $50 \leq d_{Hm}/s_H \leq 400$

Wall thickness ratio $0.77 \leq s_A/s_H \leq 1.3$

The stresses due to internal pressure are determined as follows:

$$\sigma = \alpha \cdot \frac{d_{Hm}}{4 \cdot s_H} \cdot p \qquad (8.2\text{-}2)$$

The stress indices α shall be taken from the figures laid down in **Table 8.2-2** depending on the referred nozzle diameter $d_{Am} / \sqrt{d_{Hm} \cdot s_H}$ and the wall thickness ratio s_A/s_H.

Stress category	Figure
P_L	8.2-10
$P_L + Q$	8.2-11

Table 8.2-2: Assignment of stress indices α for spherical shells

8.2.2.3 Method 2: Stress index method for total peak stress due to internal pressure

The following dimensional ranges shall be adhered to:

a) $\frac{d_{Am}}{\sqrt{d_{Hm} \cdot s_H}} \leq 0.8$

b) $\frac{d_{Hm}}{s_H} \leq 100$

c) $\frac{d_{Am}}{d_{Hm}} \leq 0.5$

Stress indices are defined as the respective numerical ratio of the normal stress component under consideration or the stress intensity (maximum total stress intensity incl. peak stress F) to the mean circumferential stress (membrane hoop stress) in the unpenetrated shell. The stress intensity values and ranges determined by using stress indices shall be limited in accordance with Section 7.

Nozzles in spherical shells and formed heads				
Stress	Inside		Outside	
σ_t	2.0		2.0	
σ_a	- 0.2		2.0	
σ_r	$-4 \cdot s_c/d_i$		0	
S	2.2		2.0	
Nozzles in cylindrical shells				
Stress	Longitudinal plane		Lateral plane	
	Inside	Outside	Inside	Outside
σ_t	3.1	1.2	1.0	2.1
σ_a	- 0.2	1.0	- 0.2	2.6
σ_r	$-2 \cdot s_c/d_i$	0	$-2 \cdot s_c/d_i$	0
S	3.3	1.2	1.2	2.6

Table 8.2-3: Stress indices for nozzles

If the design conditions of clause 5.2.6 have been satisfied, the stress indices of **Table 8.2-3** may be used. These stress indices deal only with the maximum stresses at certain nozzle locations due to internal pressure. Often it is necessary to consider the effects of stresses due to external loadings or thermal stresses. In such cases, the total stress for each direction of stress shall be determined by superposition.

For nozzles whose axes make an angle with the shell within the limits of 5.2.6 the stress indices for tangential stress on the inside shall be multiplied with the following values:

$1 + 2 \cdot \sin^2\varphi$ for hillside branches in cylinders or spheres (non-radial connection),

$1 + \tan^{4/3}\varphi$ for lateral branches in cylinders (lateral connections).

8.2.2.4 Determination of stresses due to external loadings by connected pipe

Suitable methods for determining stresses may be taken from

a) WRC Bulletin 297 [2]

and, if required, from

b) WRC Bulletin 107 [3] and

c) PD 5500:2000 [4], Annex G

8.2.3 Superposition of stresses due to internal pressure and external nozzle loads

(1) Design values and units relating to clause 8.2.3

Notation	Design value	Unit
p	allowable working pressure	MPa
$p_{zul\,s}$	allowable pressure in transitional area of shell-to-nozzle juncture for a given geometry and by utilisation of design stress	MPa
F_{ax}	axial force in nozzle	N
$F_{ax\,zul}$	*)	N
F_c, F_l	transverse forces in nozzle	N
M_b	bending moment on nozzle	Nmm
$M_{b\,zul}$	*)	Nmm
M_c	circumferential moment on nozzle	Nmm
M_l	longitudinal moment on nozzle	Nmm
M_t	torsional moment on nozzle	Nmm
$M_{t\,zul}$	*)	Nmm

*) The index zul means allowable value if this loading alone is effective thus taking full use of the design stress.

(2) In the methods 1 and 2 the stress components resulting from internal pressure shall be superposed on the stress components resulting from external loads, and then the stress intensity value or equivalent stress intensity range shall be formed. The equivalent stress intensities or equivalent stress ranges thus formed which result from internal pressure and external loads due to connected piping shall be evidenced separately, with regard to the service limits, taking the allowable limit values of **Tables 6.7-1** or **7.7-4** into account.

(3) Where individual loadings act on the nozzle which are smaller than the computed allowable limit loadings, these individual loadings can be simplified and be superpositioned in a conservative manner. The limit loading of nozzles and shell under internal pressure is adhered to if the following condition is satisfied:

a) nozzles in cylindrical shells:

$$\frac{p}{p_{zul\,s}} + \frac{F_{ax}}{F_{ax\,zul}} + \frac{M_b}{M_{b\,zul}} \le 1 \qquad (8.2\text{-}3)$$

b) nozzles in spherical shells:

$$\frac{p}{p_{zul\,s}} + \frac{F_{ax}}{F_{ax\,zul}} + \frac{M_b}{M_{b\,zul}} \le 1 \qquad (8.2\text{-}4)$$

$$\text{where } M_b = \sqrt{M_l^2 + M_c^2} \qquad (8.2\text{-}5)$$

For a) and b) the additional condition applies:

$$M_t \le \min\begin{cases} M_{c\,zul} \\ M_{l\,zul} \end{cases}$$

and

$$F_l,\ F_c \le F_{ax\,zul}$$

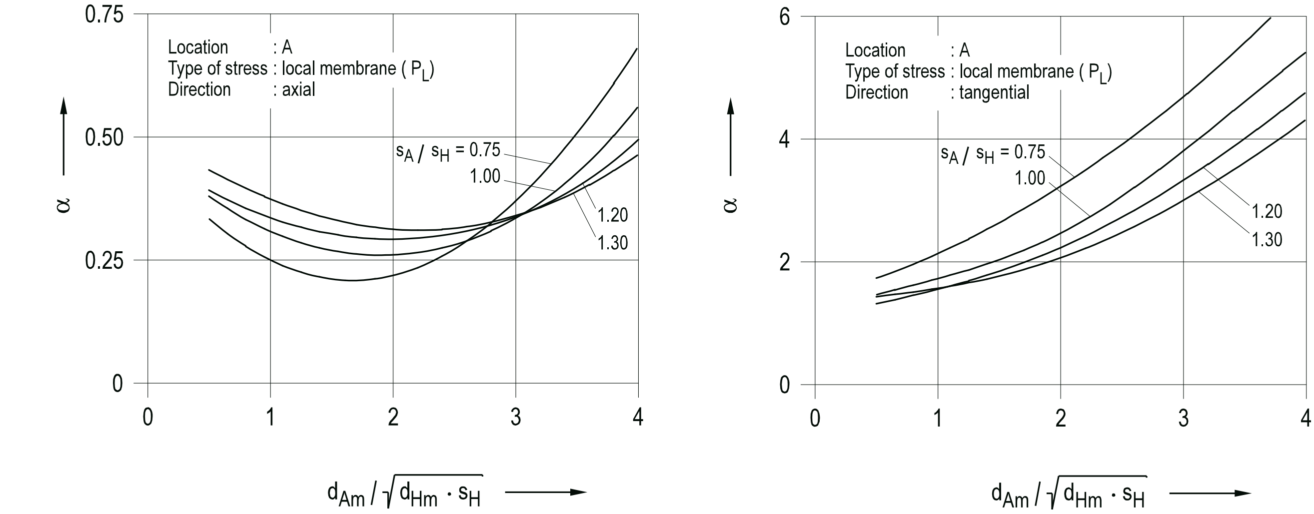


Figure 8.2-4: Stress index α for nozzle in cylindrical shell subject to internal pressure

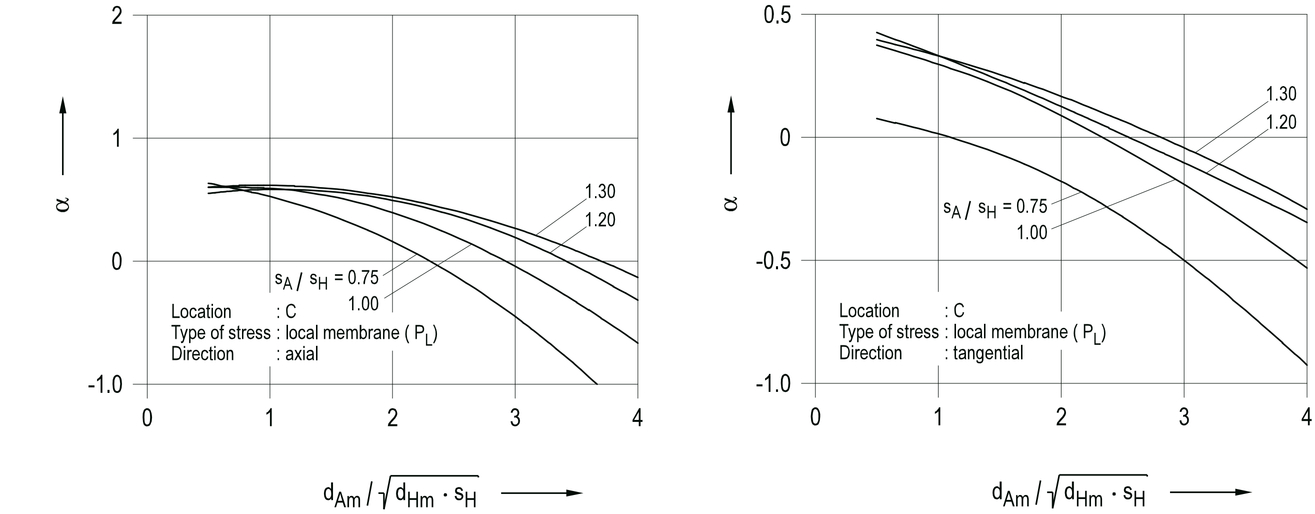


Figure 8.2-5: Stress index α for nozzle in cylindrical shell subject to internal pressure

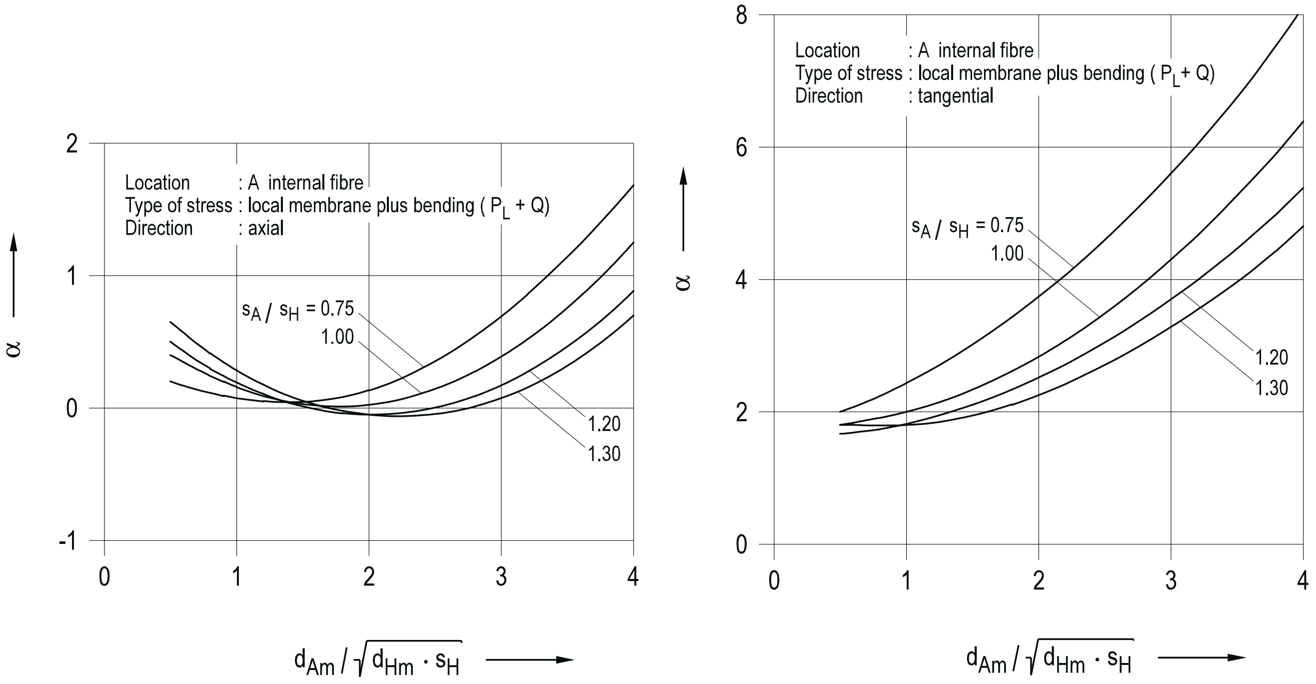


Figure 8.2-6: Stress index α for nozzle in cylindrical shell subject to internal pressure

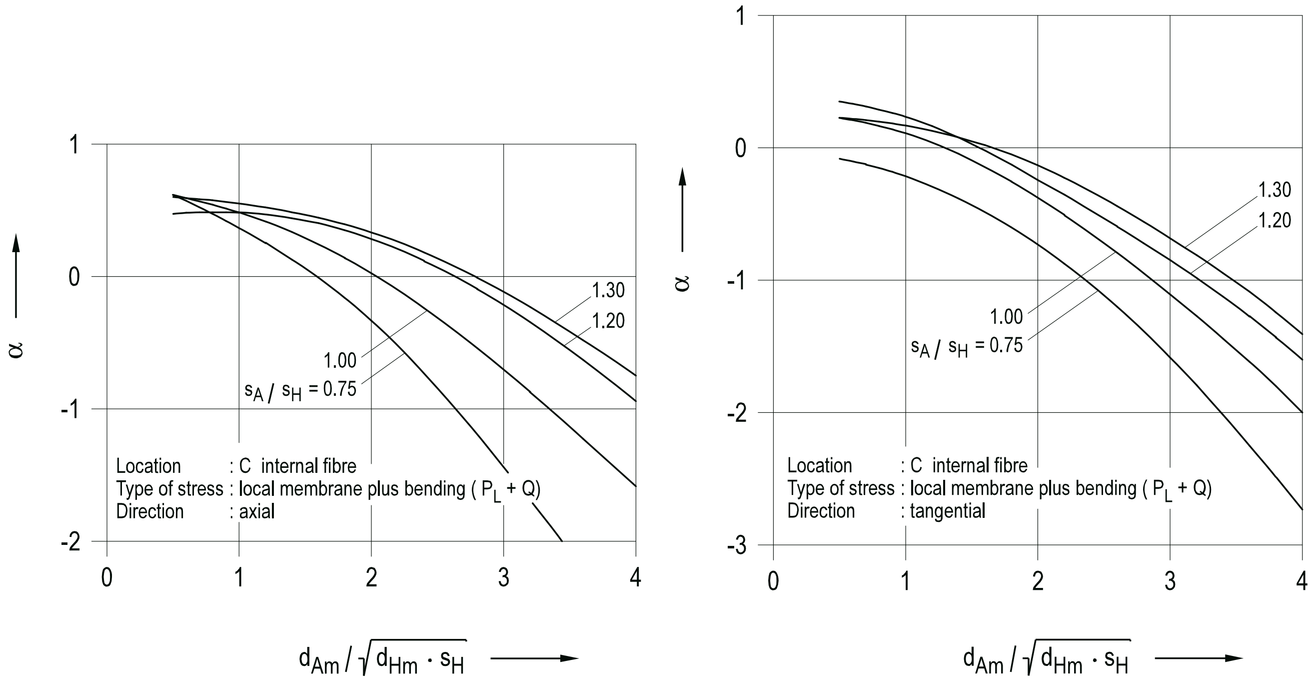
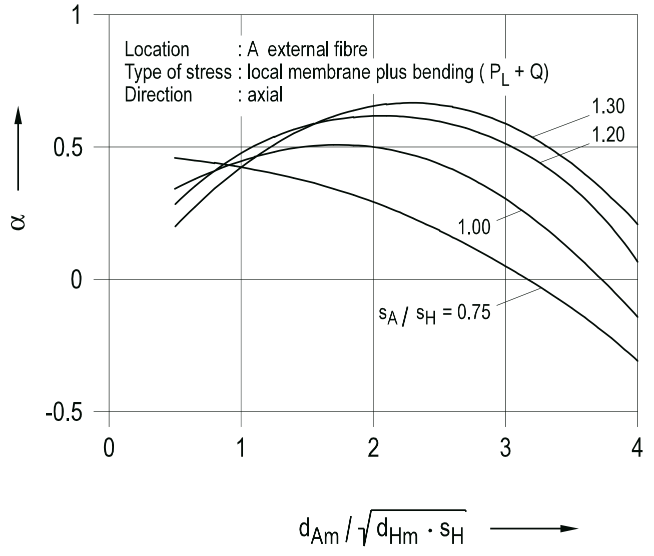


Figure 8.2-7: Stress index α for nozzle in cylindrical shell subject to internal pressure



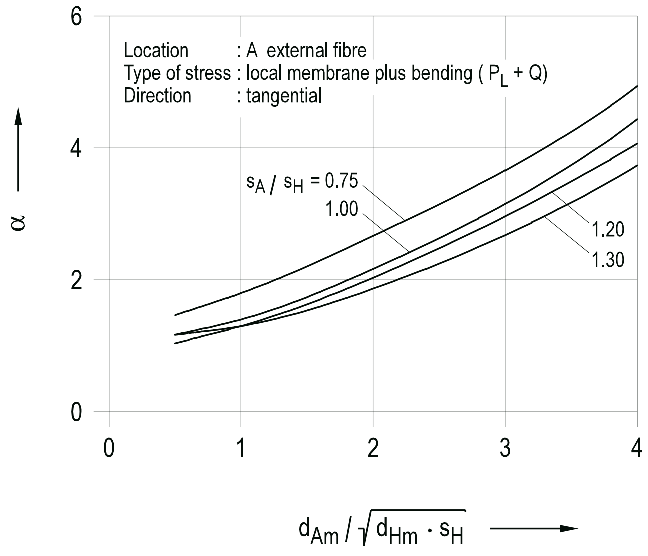
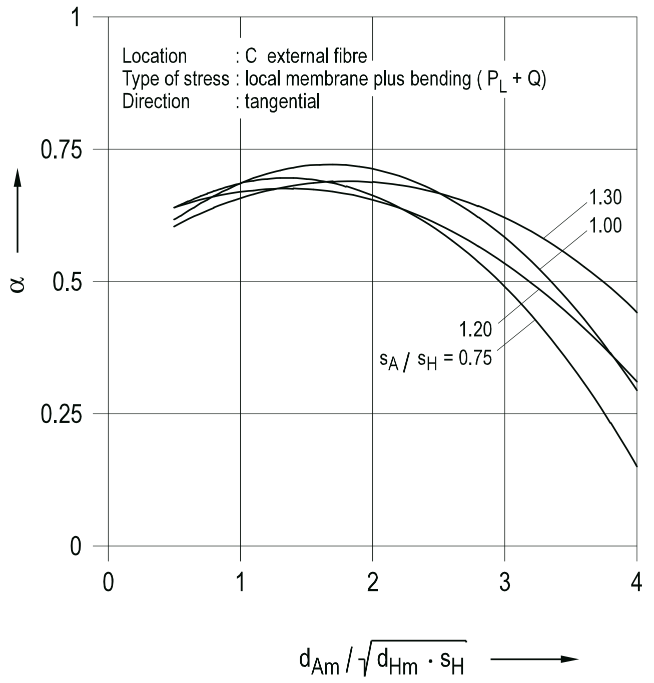


Figure 8.2-8: Stress index α for nozzle in cylindrical shell subject to internal pressure



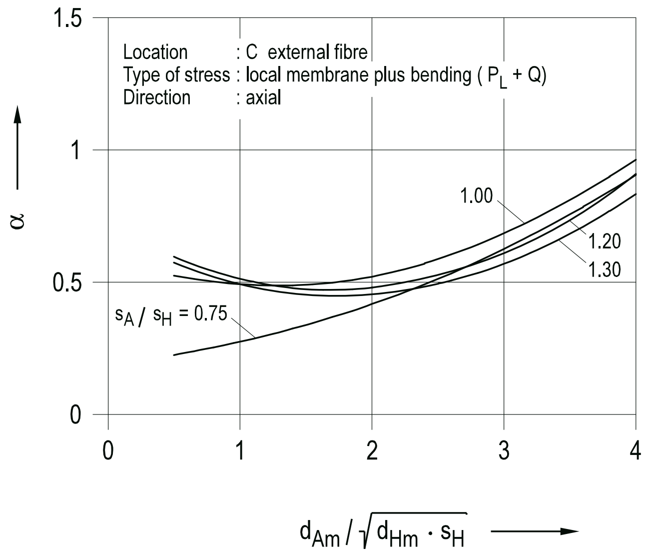


Figure 8.2-9: Stress index α for nozzle in cylindrical shell subject to internal pressure

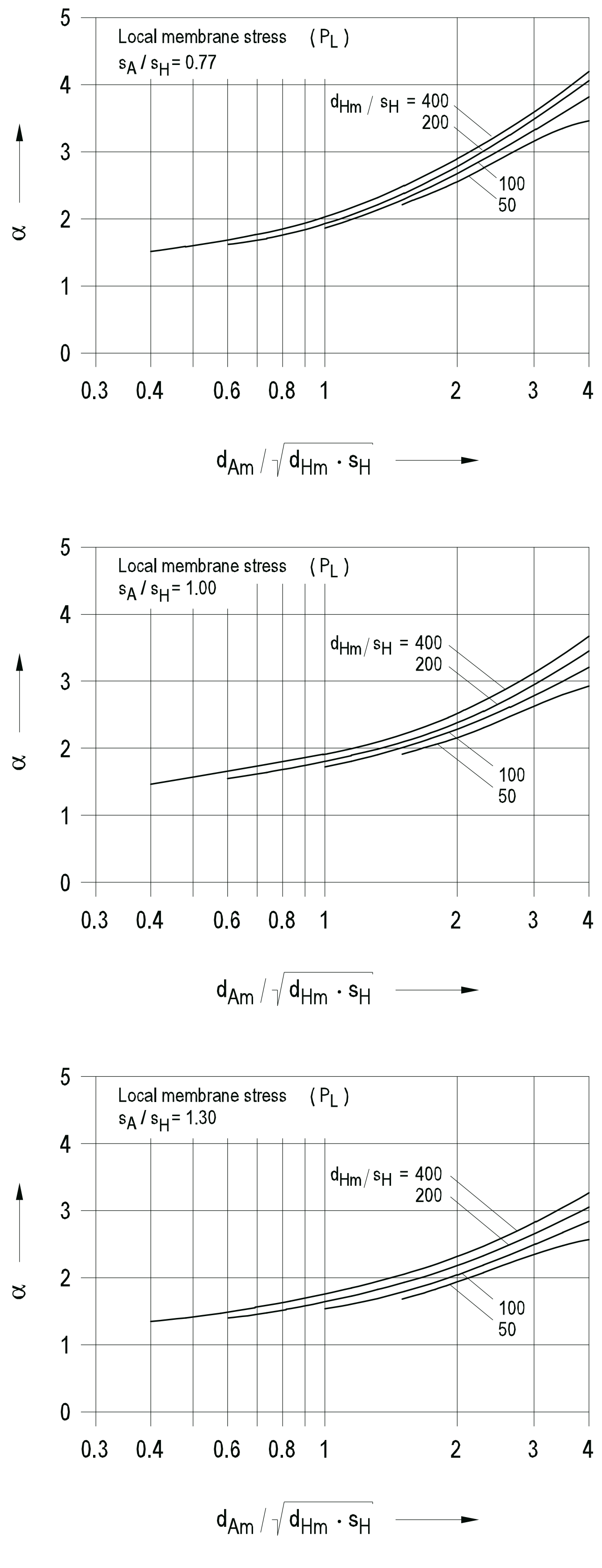


Figure 8.2-10: Stress index α for nozzle in spherical shell subject to internal pressure for P_L

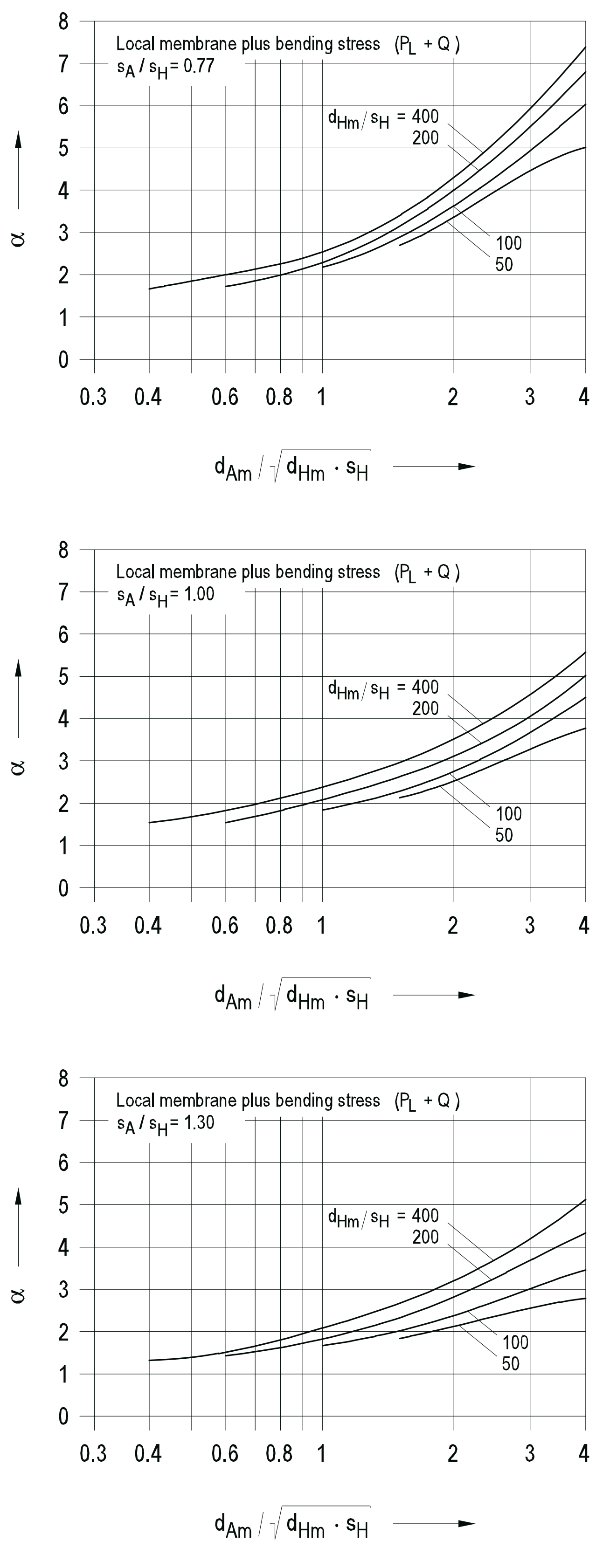


Figure 8.2-11: Stress index α for nozzle in spherical shell subject to internal pressure for $P_L + Q$

8.2.4 Fatigue analysis

The fatigue analysis shall be performed in accordance with Section 7.8.

8.2.5 Single-ply bellows expansion joints for pressure vessels

8.2.5.1 Design values and units relating to clause 8.2.5

Notation	Design value	Unit
d, h, l, r, s	dimensions in acc. with Figures 8.2-12 and 8.2-13	mm
f_1	cyclic strength factor for welded joint in acc. with clause 8.2.5.4.2.5	—
f_2	factor for partial plastic deformation in acc. with clause 8.2.	—
$f_{2(w)}$´ $f_{2(\alpha, \lambda)}$	factor f_2 with C values for axial displacement or deflection in acc. with equation (8.2-17)	—
n	shape factors	—
n_i	number of cycles in individual cyclic stress range „i"	—
p	pressure of the respective loading category	MPa
p_{krs}	instability pressure	MPa
Δp	range of pressure cycles	N/mm²
w	axial deformation of a bellows convolution	mm
z	number of bellows convolutions	—
C_w	axial working spring rate	N/mm
C'_w	axial working spring rate for multi-ply bellows	N/mm
C_α	angular working spring rate	Nm/grd
C'_α	angular working spring rate for multi-ply bellows	Nm/grd
C_λ	lateral working spring rate	N/mm
$C_{\alpha, \lambda}$	flexural rigidity due to lateral deformation	Nm/mm
D	usage factor	—
E	modulus of elasticity	N/mm²
E_T	modulus of elasticity at the highest temperature of the considered load case	N/mm²
L	overall length of bellows	mm
M_t	external torsional moment	Nmm
N	member of cycles to failure	—
N_{zul}	allowable number of cycles	—
$N_{i\ zul}$	allowable number of cycles in individual cyclic range „i"	—
$R_{(Cw)}$	design factor for axial working spring rate acc. to Tables 2 to 13 of AD 2000-Merkblatt B 13	—
$R_{(p)}$	design factor for pressure loading acc. to Tables 2 to 25 of AD 2000-Merkblatt B 13	—
$R_{p0.2}$	0.2 % proof stress	N/mm²
$R_{p1.0}$	1.0 % proof stress	N/mm²
$R_{p1.0T}$	1.0 % proof stress at temperature	N/mm²
$R_{p0.2T}$	0.2 % proof stress of the connected piping at design temperature	—
S_L	safety factor for cyclic loading	—
S_m	design stress intensity factor in acc. with Table 6.6-1	N/mm²
α	angular deformation of a bellows convolution	degree
λ'	lateral extension in one direction of an expansion joint with one bellows or with two bellows and intermediate pipe section, measured from neutral (straight) position, as per AD 2000-Merkblatt B 13, Figure 6	m
σ_{um}	mean circumferential stress	N/mm²
$\sigma_{v(p)}$	maximum stress intensity due to internal or external pressure	N/mm²
$\sigma_{V(p)}$	equivalent stress range due to pressure loading	N/mm²
$\sigma_{V(w)}$	stress intensity due to axial movement	N/mm²
$\sigma_{V(\alpha, \lambda)}$	stress intensity due to angular rotation or lateral deflection on the prerequisite that only one bellows is installed	N/mm²
$\Delta\sigma_{Vges}$	total equivalent stress range acc. to clause 8.2.5.4.2.3	N/mm²
τ	stress intensity due to torsion	N/mm²
$\Delta\tau$	equivalent stress range due to torsion	N/mm²

8.2.5.2 General

(1) On the assumption that the expansion joints are designed in accordance with clause 5.3.2.4 the procedures described hereinafter for the dimensioning and component-specific stress and fatigue analysis are permitted.

The procedures refer to the convoluted portion of the bellows. For the cylindrical portion of the bellows with equal wall thickness a specific verification may be waived if the cylindrical portion does not exceed a length $l = 0.5 \cdot \sqrt{s \cdot d_H / 2}$ (also compare **Figure 5.3-2**).

For the area beyond that portion the respective rules of this Safety Standard regarding the connecting component shall apply for stress classification and limitation.

(2) The following procedures apply to single-ply expansion joint bellows with parallel-sidewall or lyra-shaped convolutions (**Figures 8.2-12** and **8.2-13**) within the following limits:

$3 \leq d/h \leq 100$

$0.1 \leq r/h \leq 0.5$

$0.018 \leq s/h \leq 0.1$

(3) The procedures consider bellows loaded by pressure, forced movement (axial or lateral movements, angular rotation) and additional external loads.

(4) Regarding the essential influences of cyclic loading, test group A1 shall preferably be used for expansion joints. The following requirements apply to test group A1.

(5) Suitable materials shall be selected to withstand corrosion; wall thickness allowances are not appropriate.

8.2.5.3 Dimensioning

8.2.5.3.1 General

(1) Dimensioning shall be effected on the basis of the loadings and stress limits of the service condition (level 0) according to clause 3.3.2 and in consideration of the loadings and stress limits of the other service limits according to Section 3.3 insofar as they govern dimensioning.

(2) In this case the primary stresses of all service limits shall be taken into account and be limited with respect to the allowable values of **Table 6.7-1.**

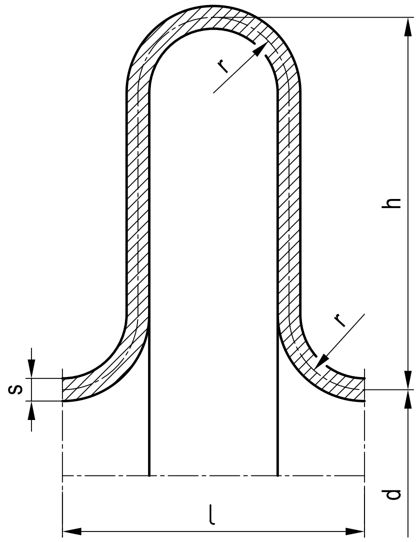


Figure 8.2-12: Parallel-sidewall convolution of bellows expansion joint

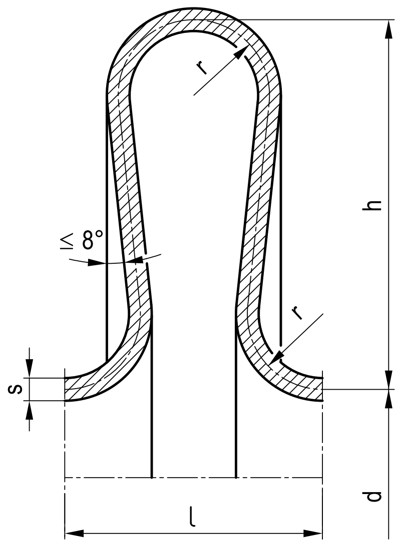


Figure 8.2-13: Slightly lyra-shaped convolution of bellows expansion joint (angle ≤ 8°)

8.2.5.3.2 Calculation of the bellows against internal and external pressure

A minimum wall thickness shall be fixed for dimensioning for the purpose of fabrication and design, and its usability shall be checked by means of the following conditions:

a) Mean circumferential stress due to pressure loading

$$\sigma_{um} = \frac{(d+h)\cdot l \cdot p}{4\cdot(1{,}14\cdot r+h)\cdot s} \le S_m \qquad (8.2\text{-}6)$$

b) Maximum stress intensity due to pressure loading

$$\sigma_{v(p)} = 10 \cdot R_{(p)} \cdot p \le n \cdot S_m \qquad (8.2\text{-}7)$$

For ferritic materials the shape factor is
$n = 1.55 - 2.8 \cdot 10^{-4} \cdot S_m$.

For austenitic materials the shape factor is

$n = 1.55 - 2{,}8 \cdot 10^{-4} \cdot S_m$,

if $R_{p0.2}$ governs the determination of the S_m value and

$n = 1.55$

if $R_{p1,.0}$ governs the determination of the $S_m$value in acc. with **Table 6.6-1**.

8.2.5.3.3 Consideration of additional external loads

Additional external loads (e.g. weight loads) shall not be considered if the requirements of clause 5.3.2.4 (4) have been met.

Otherwise, the resulting loadings and deformations shall be considered in the calculation.

It shall be ensured that bellows deformations do not impair the function of the expansion joint, component and support structures.

8.2.5.4 Analysis of mechanical behaviour

8.2.5.4.1 General

On the assumption that dimensioning was effected in acc. with clause 8.2.5.3 the analysis of the mechanical behaviour of expansion joint bellows is limited to a fatigue evaluation in acc. with clause 8.2.5.4.2 as well as to a proof of stability in acc. with clause 8.2.5.4.3.

8.2.5.4.2 Fatigue evaluation

8.2.5.4.2.1 General

The fatigue evaluation shall be performed taking into account loadings of the service levels in accordance with Section 3.3 where relevant to fatigue.

8.2.5.4.2.2 Number of cycles to failure

(1) The expected number of cycles to failure N (until onset of leakage) is obtained from:

$$N = \left(\frac{E_T}{10 \cdot \Delta\sigma_{Vges} \cdot f_1 \cdot f_2}\right)^{3.45} \qquad (8.2\text{-}8)$$

(2) The allowable number of cycles N_{zul} is obtained from:

$$N_{zul} = \frac{N}{S_L} \qquad (8.2\text{-}9)$$

(for $N_{zul} > 10^6$ may be taken $N_{zul} = \infty$ for the calculation of the usage factor)

with the safety factor for cyclic loading $S_L = 5.0$.

Where it is proved by own representative service life tests for at least 25 geometrically comparable expansion joints showing the same material behaviour and fabricated with the same manufacturing process, that the number of pressure cycles N according to equation (8.2-8) is obtained, a safety factor for cyclic loading $S_L = 2.0$ will suffice.

(3) In the case of stress cycles with variable amplitudes and cycle numbers, new individual stress cycles relevant to fatigue shall be formed (compare with clause 7.7.3.3 or also 8.4.6.3 (2)). The usage factors of these individual stress cycles shall be accumulated according to the linear damage rule to form

$$D = \sum \frac{n_i}{N_{i\,zul}} \le 1 \qquad (8.2\text{-}10)$$

where n_i is the cycle number in the respective individual stress cycles and $N_{i_{zul}}$ the allowable number of cycles.

8.2.5.4.2.3 Total equivalent stress range

(1) The total equivalent stress range $\Delta\sigma_{Vges}$ is formed in accordance with clause 7.7.3.3 and considers the stresses resulting from variable portions of the following loadings:

a) internal pressure,

b) axial, angular, lateral displacements,

c) torsional moments as far as design cannot prevent them from being applied on the expansion joint.

(2) The total equivalent stress range $\Delta\sigma_{Vges}$ shall be calculated from the individual stress ranges (approach):

$$\sigma_{Vges} = \sqrt{\left(\Delta\sigma_{V\,(p)} + \Delta\sigma_{V\,(w)} + \Delta\sigma_{V\,(\alpha,\lambda)}\right)^2 + 3\cdot\Delta\tau^2} \qquad (8.2\text{-}11)$$

The individual stress ranges shall be determined as follows:

$\Delta\sigma_{V\,(p)}$ stress range due to pressure

$$\Delta\sigma_{V\,(p)} = 10 \cdot R_{(p)} \cdot \Delta p \qquad (8.2\text{-}12)$$

$\Delta\sigma_{V\,(w)}$ stress range due to axial deformation

$$\Delta\sigma_{V\,(w)} = 2.4 \cdot 10^{-4} \cdot \frac{E_T}{h} \cdot R_{(w)} \cdot w \qquad (8.2\text{-}13)$$

$\Delta\sigma_{V\,(\alpha,\lambda)}$ stress range due to angular rotation and/or lateral deformation (only applicable if only one bellows is installed).

$$\Delta\sigma_{V(\alpha,\lambda)} = 2.4\cdot10^{-4}\cdot\frac{E_T}{h}\cdot R_{(w)}\cdot(d+2\cdot h)\cdot\left(\frac{\alpha}{114}+\frac{3}{z\cdot l}\cdot\frac{\lambda'}{z}\right) \qquad (8.2\text{-}14)$$

Stress range due to torsional moment

$$\Delta\tau = \frac{2\cdot\Delta M_t}{\pi\cdot d^2\cdot s} \qquad (8.2\text{-}15)$$

(3) It shall be considered that the maximum wall thickness s of the bellows (nominal wall thickness plus positive allowance) is to be used when forming $R_{(w)}$ from Table 2-13 and Table 14-25 of AD 2000-Merkblatt B13.

8.2.5.4.2.4 Consideration of partial plastic cyclic deformations

(1) Partial plastic cyclic deformations do not occur if the condition

$$\frac{\Delta\sigma_{Vges}}{R_{pT}} \le 2$$

has been satisfied. In this condition $\Delta\sigma_{Vges}$ from equation (8.2-11) shall be used. For R_{pT}, $R_{p0.T}$ or $R_{p1.0T}$ shall be taken depending on the value used for obtaining the S_m value. In this case $f_2 = 1$ for calculating the number of cycles to failure acc. to equation (8.2-8) shall be taken.

(2) Partial plastic cyclic deformations shall be considered if the condition

$$\frac{\Delta\sigma_{Vges}}{R_{pT}} > 2$$

has been satisfied. In this case, f_2 shall be calculated from

$$f_2 = 1 + C\left(\frac{\Delta\sigma_{Vges}}{R_{pT}} - 2\right) + 0.1\cdot B \qquad (8.2\text{-}16)$$

where

$$B = \max\left(\frac{\sigma_{V(p)}}{n\cdot\frac{R_{pT}}{1.2}};\frac{\sigma_{um}}{S_m}\right)$$

with $\sigma_{V(p)}$ acc. to equation (8.2-7) and σ_{um} acc. to equation (8.2-6). The C values for axial displacement and deflection shall be taken from **Table 8.2-4**.

Material group	C (bracketed values apply to deflection)		
	Bellows with circumferential welds in highly loaded zone	Bellows without circumferential welds in highly loaded zone	
		strain hardened	hot formed or normalised
Austenite	0.127 (0.101)	0.105 (0.86)	0.085 (0.067)
Ferrite	0.155 (0.127)	0.155 (0.127)	0.133 (0.109)

Table 8.2-4: C values

(3) In the case of superposition of axial deformation, deflection and torsion $\Delta\sigma_{Vges}\cdot f_2$ is broken down as follows:

$$\Delta\sigma_{Vges}\cdot f_2 = \left\{\left[\left(\Delta\sigma_{V(p)}\cdot\frac{\Delta\sigma_{V(w)}}{\Delta\sigma_{V(w)}+\Delta\sigma_{V(\alpha,\lambda)}}+\Delta\sigma_{V(w)}\right)\cdot f_{2(w)} + \left(\Delta\sigma_{V(p)}\cdot\frac{\Delta\sigma_{V(\alpha)}}{\Delta\sigma_{V(w)}+\Delta\sigma_{V(\alpha,\lambda)}}+\sigma_{V(\alpha,\lambda)}\right)\cdot f_{2(\alpha,\lambda)}\right] + 3\cdot\left(\Delta\tau\cdot f_{2(w)}\right)^2\right\}^{1/2} \qquad (8.2\text{-}17)$$

$f_{2(w)}$ and $f_{2(\alpha,\lambda)}$ are the respective factors f_2 with the corresponding C values for axial displacement or deflection acc. to equation (8.2-16).

8.2.5.4.2.5 Requirements for the calculation of the point of maximum loading of the bellows, taking the cyclic strength of the weld into account

To determine the point of maximum loading the design factors of Table 2 13 of AD 2000-Merkblatt B13 shall be used in connection with the weld factor $f_1 = 1.0$.

In addition it shall be checked, for the locations of welds mentioned hereinafter by means of the design factors of Table 14-25 of AD 2000-Merkblatt B13 in conjunction with a weld factor $f_1 = 2.0$ whether the weld is relevant to fatigue considerations. This additional requirement applies to welds

a) at the crest of the external knuckle
b) at the crest of the internal knuckle
c) at the bellows end as circumferential weld to the connecting component if the cylindrical end between internal bellows convolution and the weld is shorter than the greater value of $3\cdot s$, $0.25\cdot\sqrt{s\cdot d}$ or 10 mm.

8.2.5.4.3 Proof of stability

The examination for proof of sufficient safety against instability is made for column instability (squirm) by

$$p_{KrS} = \frac{2\cdot\pi}{z\cdot L}\cdot C_w \ge 3\cdot p \qquad (8.2\text{-}18)$$

Note:
The safety against in-plane squirm can be proved by the procedure given in Section III, NC 3649 of the ASME Code.

8.2.5.4.4 Spring rates

8.2.5.4.4.1 General

Bellows spring rates shall be known to determine the forces at the point of connection or to use them for static and dynamic analysis for the expansion joint.

8.2.5.4.4.2 Axial working spring rate

For axial movement the following spring rate may be used:

$$C_w = 0.15\cdot10^{-4}\cdot R_{(cw)}\,(d+h)\cdot E \qquad (8.2\text{-}19)$$

In the case of multi-convolute bellows the working spring rate for axial movement is

$$C_w' = \frac{C_w}{z} \qquad (8.2\text{-}20)$$

8.2.5.4.4.3 Angular working spring rate

For angular deflection the following spring rate may be used:

$$C_\alpha = 2.2\cdot10^{-6}\cdot(d+2\cdot h)^2\cdot C_w \qquad (8.2\text{-}21)$$

For multi-convolute bellows the angular working spring rate is

$$C_\alpha{}' = \frac{C_\alpha}{z} \quad (8.2\text{-}22)$$

8.2.5.4.4.4 Lateral working spring rate

(1) Lateral movement of the expansion joint is only possible if the bellows has at least two convolutions.

The lateral working spring rate is

$$C_\lambda = 1.5 \cdot \frac{(d + 2 \cdot h)^2}{(z \cdot l)^2} \cdot C_w{}' \quad (8.2\text{-}23)$$

(2) Deflection of the bellows due to lateral deformation can also be due to maximum axial deformation. Then the working spring rate is calculated as follows:

$$C_{\alpha,\lambda} = 0.75 \cdot 10^{-3} \cdot \frac{(d + 2 \cdot h)^2}{z \cdot l} \cdot C_w{}' \quad (8.2\text{-}24)$$

8.2.6 Heat exchanger tubesheets

8.2.6.1 Design values and units relating to clause 8.2.6

Notation	Design value	Unit
a, b, c, d, e	values for determining the buckling length L_K (Figure 8.2-18)	mm
d	outside diameter of heat exchanger tubes	mm
d_h	tubehole diameter in tubesheet	mm
e	tubesheet thickness (exclusive of allowances)	mm
e_{Bieg}	wall thickness of tubesheet for design against bending	mm
e_s	shell thickness (nominal)	mm
e_{Schub}	wall thickness of tubesheet for design against shear	mm
e_t	tube thickness (nominal)	mm
f_s	pressure factor acc. to equation (8.2-40)	—
f_t	pressure factor acc. to equation (8.2-41)	—
p	tubesheet design pressure acc. to clause 8.2.6.4	MPa
p_1	shell side design pressure of heat exchanger	MPa
p'_1	effective shell side design pressure for heat exchanger with fixed tubesheet acc. to clause 8.2.6.5.1	MPa
p_2	tube side design pressure of heat exchanger	MPa
p_4	design pressure acc. to equation (8.2-43)	MPa
p_5	design pressure acc. to equation (8.2-44)	MPa
p'_2	effective tube side design pressure for heat exchanger with fixed tubesheet acc. to clause 8.2.6.5.1	MPa
p_e	equivalent (fictitious) pressure differential due to restrained differential thermal expansion of heat exchanger shell and tubes acc. to clause 8.2.6.5.1	MPa
p'_S	shell-side effective design pressure	MPa
p'_t	tube-side effective design pressure	MPa
r	radius of gyration	mm
C	design factor	—
D	outside diameter of heat exchanger shell	mm
D_o	diameter of outer tube limit circle	mm
D_1	diameter to which shell fluid pressure is exerted	mm
D_2	diameter to which tube fluid pressure is exerted	mm
D_J	effective pressurised diameter of expansion joint bellows	Nmm
E, E_s, E_t	elastic modulus of tubesheet, shell and tube material, respectively	N/mm²
F	support factor for considering tubesheet clamping acc. to Figure 8.2-17	—
F_e	factor for considering final tube expansion	—
F_q	value for calculating equivalent pressures, where expansion joints are installed, acc. to equation (8.2-39)	—
F_r	factor for considering the type of tube-to-tubesheet joint, acc. to Table 8.2-5	—
F_y	factor for considering different mechanical properties	—
J	strain factor for expansion joint, if installed in the shell J = 1.0 for shell without expansion joint $J = \frac{1}{1 + (\pi \cdot D \cdot E_s \cdot e_s \cdot S)/L}$ for shell with expansion joint	—
K	mean strain ratio, tube bundle/shell acc. to equation (8.2-38)	—
L	tube length between inner faces of tubesheets	mm
L_K	buckling length acc. to Figure 8.2-18	mm
N	number of tubes for straight-tube heat exchangers	—
P	tube pitch (spacing between centres)	mm
P_m	general primary membrane stress from Table 6.7-1.	N/mm²
R_{pT}	proof stress at temperature	N/mm²
$R_{p0.2T}$	0.2 % proof stress at temperature	N/mm²
S	spring rate of expansion joint	mm/N
S_m	design stress intensity value acc. to Table 6.6-1	N/mm²
α_s, α_t	thermal expansion coefficient of shell and tube material, respectively	1/K
λ	ligament efficiency of tubesheet in shear	—
μ	ligament efficiency of tubesheet in bending	—
v, v*	Poisson's ratio for unperforated and perforated plate	—
σ_L	tube longitudinal stress	N/mm²
σ_{zul}	allowable stress	N/mm²
τ_{zul}	allowable shear stress	N/mm²
Θ_s, Θ_t	mean shell or tube wall temperature	°C
Ω	design stress factor	—

8.2.6.2 Scope

(1) This clause contains the dimensioning and component-specific stress analysis of heat exchanger tubesheets and tubes in accordance with clause 5.3.2.3. A schematic representation of the heat exchanger types considered herein with welded-in (fixed) tubesheets is shown in **Figure 8.2-14**, where the shells can be provided with an expansion joint or not. The arrangement of tubes in the tubesheet is shown in **Fig. 8.2-16**. The tube bundles either consist of straight or U-tubes.

(2) The equations indicated are based on the following preconditions:

a) The tubes and the shell are subjected to pressure and temperature distributions.

b) All tubes have the same cross-sectional dimensions (nominal wall thickness, nominal diameter).

c) For heat exchangers with two tubesheets, both tubesheets shall have the same thickness.

d) The tubesheet shall have the same thickness over the full array of tubes (tube field).

e) The tubesheet shall be uniformly tubed and the tubed area shall be circular or approximately circular.

f) The unperformated (outer) rim of the tubesheet shall be small enough to consider it like a ring (without considerable cross-sectional deformation). Here, the following condition additionally applies:

$D_1, D_2 \leq = D_0 + 6e$.

g) The tubesheet thickness (less corrosion allowance) shall not be less than

ga) 0.75 · tube outer diameter d for tubes with $d \leq 25$ mm

gb) 22 mm for tubes with 25 mm < $d \leq 30$ mm

gc) 25 mm for tubes with 30 mm < $d \leq 40$ mm

gd) 30 mm for tubes with 40 mm < $d \leq 50$ mm

(3) The strength of heat exchanger tubesheets which do not satisfy the conditions of (2), shall be demonstrated separately.

(4) Where, in case of heat exchanger tubes to **Figure 8.2-14**, configuration b, the tube-side pressure is higher than double the shell-side pressure ($p_2 > 2 \cdot p_1$), it shall be verified that the shell is capable of withstanding the resulting axial force.

8.2.6.3 Characteristics of perforated plates

8.2.6.3.1 Ligament efficiency of plate

For tubes which are expanded for at least 60 % of the tubesheet depth or are explosion welded, the following applies:

$$\mu = \lambda = \frac{P-(d_h - e_t)}{P} \quad (8.2\text{-}25)$$

For other tube-to-tubesheet joints the following applies:

$$\mu = \lambda = \frac{P-d_h}{P} \quad (8.2\text{-}26)$$

8.2.6.3.2 Effective elastic constant of perforated plate

The effective elastic constant ν* of the perforated plate is obtained from **Figure 8.2-15**, where distinction is made between thinner ($e < 2 \cdot P$) and thicker ($e \geq 2 \cdot P$) tubesheets. The symbols □ and Δ represent the tube pattern in the tubesheet (see **Figure 8.2-16**).

8.2.6.4 Tubesheets for U-tube heat exchangers

The heat exchanger tubesheets shall be designed for a pressure

$$p = p_2 - p_1 \quad (8.2\text{-}27)$$

Here, it must be taken into account that pressure alone will be applied.

The minimum tubesheet wall thickness is the greater of the two design values for bending or shear, respectively:

$$e_{Bieg} = C \cdot D_o \cdot \left(\frac{p}{\Omega \cdot \mu \cdot \sigma_{zul}}\right)^{1/2} \quad (8.2\text{-}28)$$

$$e_{Schub} = 0.155 \cdot D_o \cdot \frac{p}{\lambda \cdot \tau_{zul}} \quad (8.2\text{-}29)$$

where

$C = 0.433$

$\Omega = 2.0$

$\tau_{zul} = 0.5 \cdot \sigma_{zul}$.

For σ_{zul} the value S_m or S which depends on the test group and is permissible for the general primary membrane stress P_m (see **Table 6.7-1**) shall be taken.

8.2.6.5 Tubesheets for straight-tube heat exchangers

8.2.6.5.1 Minimum wall thickness of tubesheets

The minimum tubesheet wall thickness is the greater of the values obtained from the calculations against bending or shear:

$$e_{Bieg} = \max\left[\frac{FD_1}{2}\left(\frac{p_1'}{\sigma_{zul}}\right)^{1/2}; \frac{FD_2}{2}\left(\frac{p_2'}{\sigma_{zul}}\right)^{1/2}\right] \quad (8.2\text{-}30)$$

The F-values shall be determined in acc. with **Figure 8.2-17**.

$$e_{Schub} = \max\left[\frac{0.155 \cdot D_o \cdot p_1'}{\lambda \cdot \tau_{zul}}; \frac{0.155 \cdot D_o \cdot p_2'}{\lambda \cdot \tau_{zul}}\right] \quad (8.2\text{-}31)$$

(σ_{zul}, τ_{zul} see clause 8.2.6.3)

with

$$p_1' = \max\left(\frac{p_s' - p_e}{2}; p_s'; \frac{p_e}{2}\right) \quad (8.2\text{-}32)$$

and

$$p_2' = \max\left(\frac{p_t' + p_e}{2}; p_t'\right), \text{ if } p_s' > 0 \quad (8.2\text{-}33)$$

or

$$p_2' = \max\left(\frac{p_t' - p_s' + p_e}{2}; p_t' - p_s'\right), \text{ if } p_s' < 0 \quad (8.2\text{-}34)$$

There

$$p_s' = p_1 \cdot \frac{0.4 \cdot J \cdot [1.5 + K \cdot (1.5 + f_s)] - \left[\left(\frac{1-J}{2}\right) \cdot \left(\frac{D_J^2}{D_1^2} - 1\right)\right]}{(1 + J \cdot K \cdot F_q)} \quad (8.2\text{-}35)$$

$$p_t' = p_2 \cdot \frac{1 + 0.4 \cdot J \cdot K \cdot (1.5 + f_t)}{(1 + J \cdot K \cdot F_q)} \quad (8.2\text{-}36)$$

and

$$p_e = \frac{4 \cdot J \cdot E_s \cdot e_s \cdot (\alpha_s \cdot \Theta_s - \alpha_t \cdot \Theta_t)}{(D - 3e_s) \cdot (1 + J \cdot K \cdot F_q)} \quad (8.2\text{-}37)$$

$$\text{with } K = \frac{E_s \cdot e_s \cdot (D - e_s)}{E_t \cdot e_t \cdot N \cdot (d - e_t)} \quad (8.3\text{-}38)$$

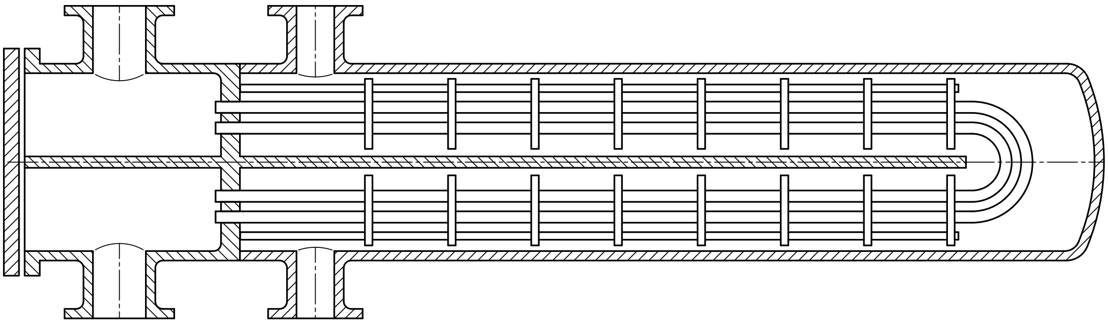
$$F_q = \max\left\{1.0; 0.25 + (F - 0.6) \cdot \left[\frac{300 \cdot e_s \cdot E_s}{K \cdot L \cdot E} \cdot \left(\frac{D_1}{e}\right)^3\right]^{1/4}\right\} \quad (8.2\text{-}39)$$

$$f_s = 1 - N \cdot \left(\frac{d}{D_1}\right)^2 \quad (8.2\text{-}40)$$

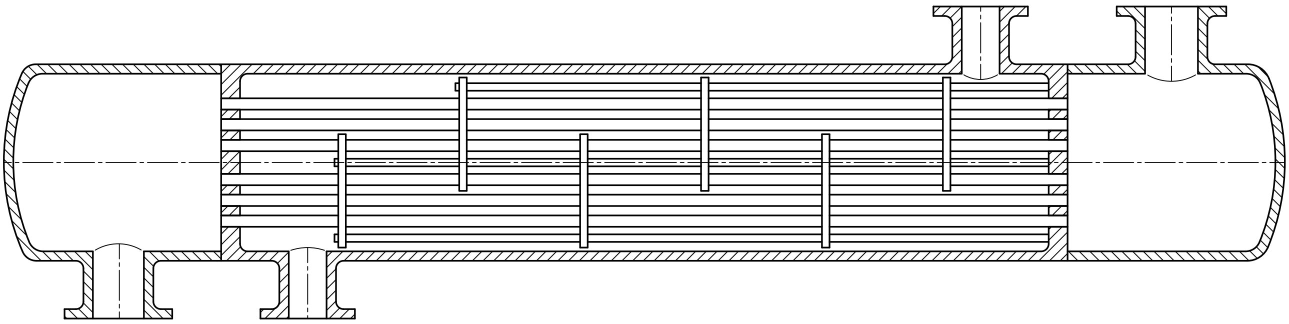
$$f_t = 1 - N \cdot \left[\frac{(d - 2 \cdot e_t)}{D_2}\right]^2 \quad (8.2\text{-}41)$$

Note:

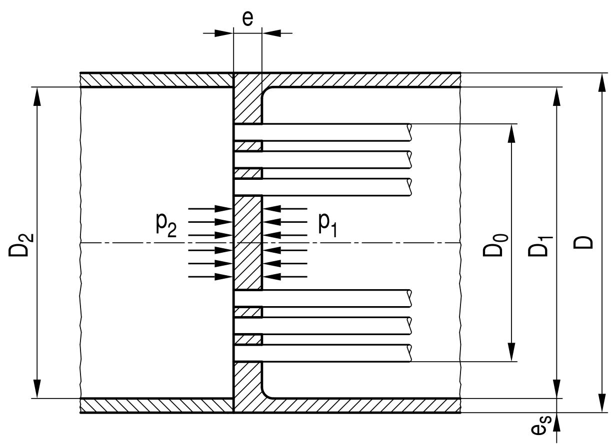
The above formulae may lead to increasingly conservative tubesheet thicknesses for shell diameters D_1 > 1500 mm.



Type a



Type b
with or without expansion joint



Representation of geometric relationship
in the heat exchanger tubesheet area

Figure 8.2-14: Examples for usual heat exchanger types (schematic representation)

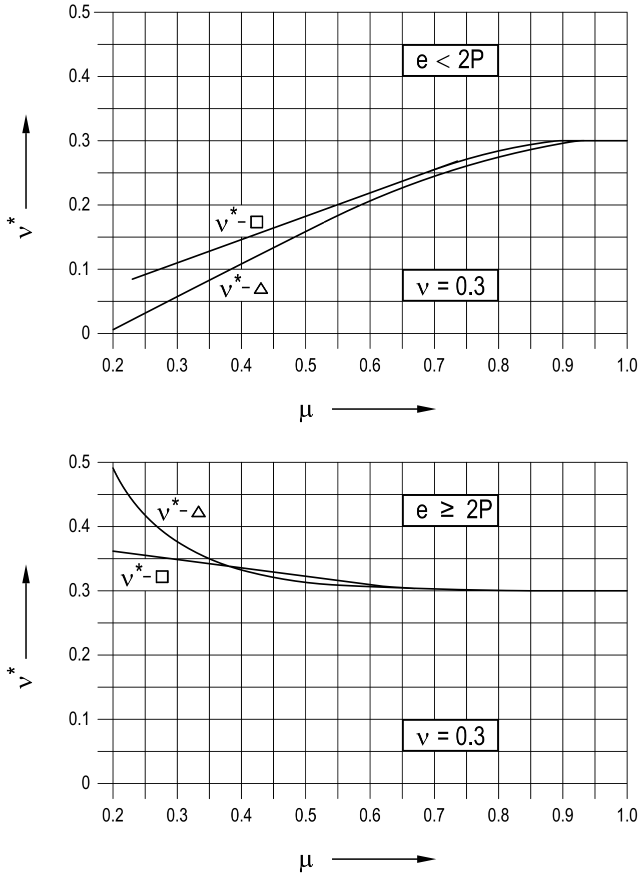


Figure 8.2-15: Effective elastic constants of perforated plate

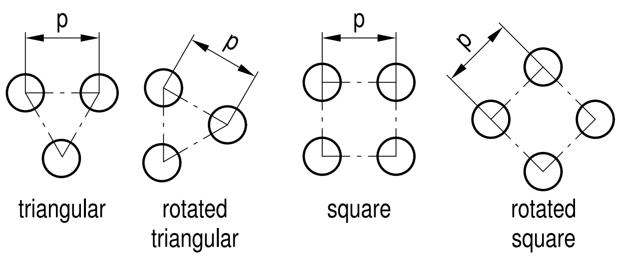


Figure 8.2-16: Tube pattern in tubesheet

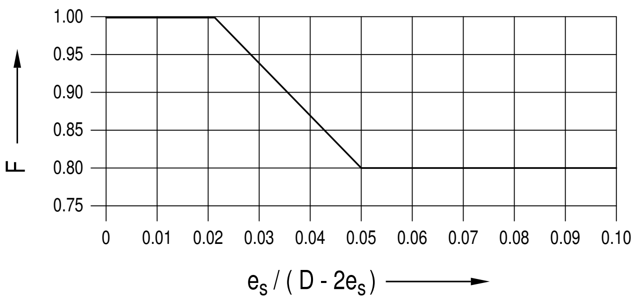


Figure 8.2-17: Determination of F-values

8.2.6.5.2 Longitudinal stress in heat exchanger tubes

For the considered load cases the tube longitudinal stress shall be determined by

$$\sigma_L = \frac{F_q \cdot p_t^x \cdot D_2^2}{4 \cdot N \cdot e_t \cdot (d - e_t)} \tag{8.2-42}$$

where p_t^x is the greatest positive value (resulting in tensile stresses) and the smallest negative value (resulting in compressive stresses) to be determined by means of the following terms:

a) $z \cdot (p_4 + p_e - p_5)$

b) $z \cdot p_e$

c) $p_4 - p_5$

d) $z\,(p_4 + p_e)$

e) p_4

f) $- p_5$

g) $z\,(p_e - p_5)$

with

z = 1.0 if the algebraic sign of p_t^x is negative

z = 0.5 if the algebraic sign of p_t^x is positive

and

$$p_4 = \left(p_t{}' - \frac{f_t}{F_q} \cdot p_2\right) \tag{8.2-43}$$

$$p_5 = \left(p_s{}' - \frac{f_s}{F_q} \cdot p_1\right) \tag{8.2-44}$$

8.2.6.6 Limitation of tube longitudinal stresses

Longitudinal tensile stresses shall be limited in accordance with **Table 6.7-1**.

To avoid buckling of tubes in straight-tube heat exchangers the compressive tube longitudinal stresses shall be limited as follows:

$$|\sigma_L| \le \frac{\pi^2 \cdot r^2 \cdot E_t}{2 \cdot L_K{}^2}, \text{ if } C \le \frac{L_K}{r} \tag{8.2-45}$$

or

$$|\sigma_L| \le \frac{S_m \cdot y}{2} \cdot \left(1 - \frac{L_K}{2 \cdot r \cdot C}\right), \text{ if } C > \frac{L_K}{r} \tag{8.2-46}$$

where

$r = 0.25 \cdot \sqrt{d^2 + (d - 2 \cdot e_t)^2}$

y = 1.4 for ferritic tubes

y = 1.1 for austenitic tubes

$C = \sqrt{\frac{2 \cdot \pi^2 \cdot E_t}{y \cdot S_m}}$

L_K = critical buckling length, see **Figure 8.2-18**

S_m = in acc. with **Table 6.6-1**

8.2.6.7 Allowable tube joint end loads

The most frequently used tube-to-tubesheet joints are shown in **Table 8.2-5**. To each joint a reliability factor F_r is assigned. The tube longitudinal stresses occurring shall satisfy the following conditions:

a) Joints a, b and c

$$\sigma_L \le \sigma_{zul} \cdot F_r \tag{8.2-47}$$

b) Joints d, e and f

$$\sigma_L \le \sigma_{zul} \cdot F_e \cdot F_r \cdot F_y \tag{8.2-48}$$

with σ_{zul} as P_m value in acc. with **Table 6.7-1**, however not to exceed $R_{p0.2T}$.

F_r factor acc. to **Table 8.2-5**

F_e expansion factor

F_e = 1 for grooved holes or for explosion welded tube ends

otherwise:

$$F_e = \frac{\text{expanded tube length}}{\text{tube outside diameters}} \le 1$$

$$F_y = \frac{R_{pT} \text{ of tube sheet material}}{R_{pT} \text{ of tube material}}$$

in which case $F_y \leq 1$ shall apply.

Details of the given design procedures may be taken from the literature referred to hereinafter:

a) TEMA [5]

b) PD 5500: 2000 [4], chapter 3.9

c) ASME Code, Section VIII, Division 1, Appendix AA [6].

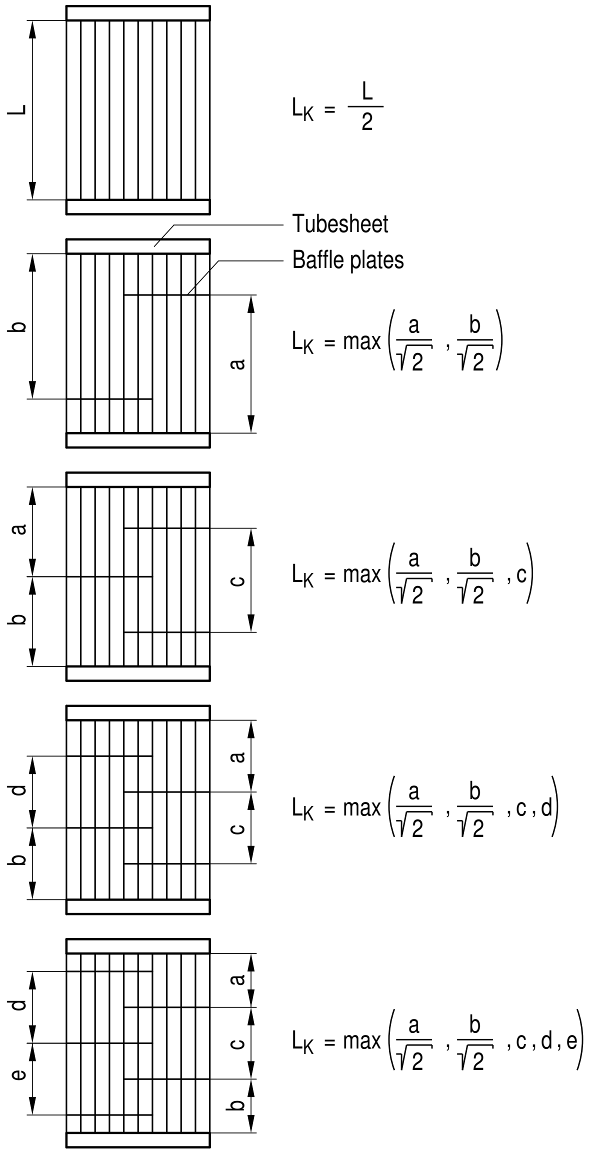


Figure 8.2-18: Determination of buckling length L_K

Joint tube/tubesheet	F_r
a) welded only weld throat $\geq$ tube thickness	0.80
b) welded only weld throat < tube thickness	0.55
c) expanded and welded weld throat $\leq$ tube thickness	0.80
d) expanded and welded weld throat < tube thickness	0.55
e) expanded only	0.50
f) explosion expanded/welded	0.80

Table 8.2-5: F_r values for typical tube joints

8.2.7 Consideration of external forces and moments in pressure vessel walls

8.2.7.1 General

External forces and moments in pressure vessel walls shall be taken into account by applying the methods of Section 6.1.

8.2.7.2 Horizontal vessels on saddles

The loadings in vessel longitudinal and circumferential direction shall be determined.

Note:

Suitable methods to calculate these loadings are e.g. given in British Standard 5500 (2000) „Specification for unfired fusion welded pressure vessels", Annex G, and in AD 2000-Merkblatt S3/2.

8.2.7.3 Vessels with bracket supports

8.2.7.3.1 General

This clause serves to calculate vertical cylindrical vessels the walls of which are subject to local loadings due to bracket supports. The loadings due to radial forces F, transverse forces Q, longitudinal moments M_1 and circumferential moments M_u (**Figure 8.2-19**) are to be evaluated. These loadings shall be distributed proportionally on the connecting plate.

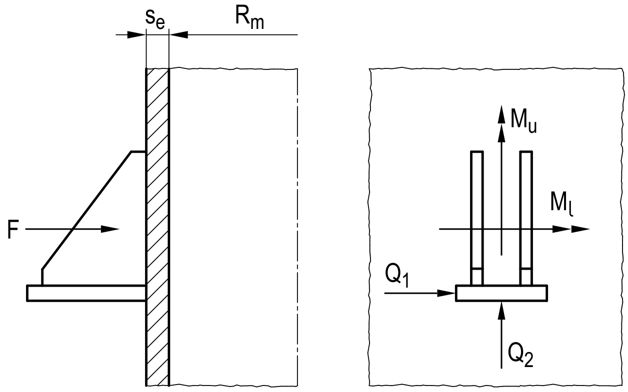


Figure 8.2-19: Vessels with support brackets

8.2.7.3.2 Calculation methods

(1) Suitable methods for consideration of unit shear forces and unit moments can be taken from:

a) WRC Bulletin 297 [2]

and if required, from

b) WRC Bulletin 107 [3] and

c) PD 5500:2000 [4], Annex G

d) AD 2000-Merkblatt S3/4

(2) The classification and limitation of stresses shall be made in accordance with Section 7.7 and 7.8, respectively.

8.3 Pumps

8.3.1 Design values and units relating to Section 8.3

Notation	Design value	Unit
a	outside radius	mm
b	inside radius	mm
r	radius	mm
t	plate thickness	mm
w	deflection of middle plane of plate	mm

Notation	Design value	Unit
A_j, A_0, A_1, A_2, A_3	auxiliary values	—
D	plate stiffness	Nmm
E	modulus of elasticity	N/mm^2
K_i, K_1, K_2, K_3, K_4	integration constants	—
M	bending moment	Nmm
M_r	radial bending moment per unit length	Nmm/mm
M_t	tangential bending moment per unit length	Nmm/mm
M_{rt}	twisting moment per unit length	Nmm/mm
Q_r	radial shear force	N/mm
Q_t	tangential shear force	N/mm
α	angle of tangent at midplane of plate	degree
ν	Poisson's ratio	—
ρ	radius of circle = r/a	—
σ_r	radial bending stress	N/mm^2
σ_t	tangential bending stress	N/mm^2
τ_r	shear stress due to Q_r	N/mm^2
τ_{rt}	shear stress due to M_{rt}	N/mm^2
τ_t	shear stress due to Q_t	N/mm^2
ϕ	angle	degree
Ψ	geometry constant Ψ = b/a	—

8.3.2 Calculation of stresses and strains due to external bending moments on circular flat plates

8.3.2.1 General

External bending moments on circular plates, here covers or heads, that can be applied e.g. by connected piping result in additional stresses and lead to unsymmetrical deflections from the plane of the plate.

The calculation method described in this section is intended to verify the dimensioning of covers or heads that has already been effected for internal pressure and axial forces, if any, in accordance with the pertinent rules. The results obtained by the calculation shall be superposed on the results obtained for internal pressure and axial force.

8.3.2.2 Calculation

8.3.2.2.1 General

The following equations are independent of the type of edge support of the plate. For the edge conditions "clamped" and "simply supported" the pertinent constants K_i of **Table 8.3-1** are given.

Table 8.3-1 also contains equations making a recursive calculation of the constants $K_i = f(\Psi)$ from the auxiliary values A_i possible. They may be taken for complete programming of the formulae.

The equations hereinafter contain the Poisson's ratio as constant $\nu = 0.3$ and apply to geometries within the range a - b < 3t. The course of edge force due to bending moment M is shown in **Figure 8.3-1**.

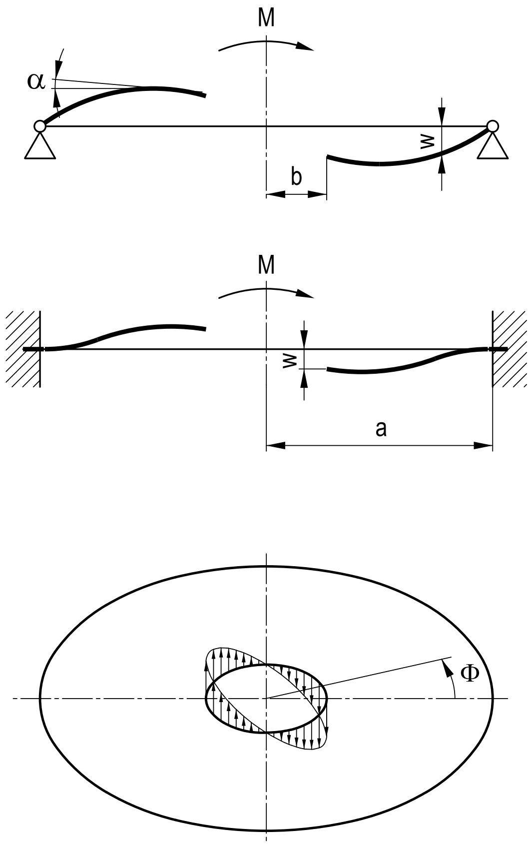


Figure 8.3-1: Course of edge force due to bending moment M; Design values

8.3.2.2.2 Calculation of unit moments and stresses

(1) The forces and moments per unit length can be determined for any point of the circular flat plate by means of the following equations:

a) Radial bending moment

$$M_r = -\frac{D}{a^2} \cdot \left(6.6 \cdot K_1 \cdot \rho + 1.3 \cdot K_3 \cdot \frac{1}{\rho} + 1.4 \cdot K_4 \cdot \frac{1}{\rho^3} \right) \cdot A_0 \cdot \cos\phi \tag{8.3-1}$$

b) Tangential bending moment

$$M_t = -\frac{D}{a^2} \cdot \left(3.8 \cdot K_1 \cdot \rho + 1.3 \cdot K_3 \cdot \frac{1}{\rho} + 1.4 \cdot K_4 \cdot \frac{1}{\rho^3} \right) \cdot A_0 \cdot \cos\phi \tag{8.3-2}$$

c) Twisting moment

$$M_{rt} = \frac{D}{a^2} \cdot \left(1.46 \cdot \left(K_1 \cdot \rho - K_4 \cdot \frac{1}{\rho^3} \right) + 0.7 \cdot K_3 \cdot \frac{1}{\rho} \right) \cdot A_0 \cdot \sin\phi \tag{8.3-3}$$

d) Radial shear force

$$Q_r = \frac{D}{a^3} \cdot \left(8 \cdot K_1 - 2 \cdot K_3 \cdot \frac{1}{\rho^2} \right) \cdot A_0 \cdot \cos\phi \tag{8.3-4}$$

e) Tangential shear force

$$Q_t = \frac{D}{a^3} \cdot \left(8 \cdot K_1 + 2 \cdot K_3 \cdot \frac{1}{\rho^2} \right) \cdot A_0 \cdot \sin\phi \tag{8.3-5}$$

with the plate stiffness

$$D = \frac{E \cdot t^3}{12 \cdot \left(1 - \upsilon^2\right)} \tag{8.3-6}$$

and the auxiliary value

$$A_0 = -\frac{a \cdot M}{\pi \cdot D \cdot \psi^2} \qquad (8.3\text{-}7)$$

(2) These forces and moments result in the following stresses:

a) Bending stress due to M_r

$$\sigma_r = \frac{6 \cdot M_r}{t^2} \qquad (8.3\text{-}8)$$

b) Bending stress due to M_t

$$\sigma_t = \frac{6 \cdot M_t}{t^2} \qquad (8.3\text{-}9)$$

c) Shear stress due to M_{rt}

$$\tau_{rt} = \frac{6 \cdot M_{rt}}{t^2} \qquad (8.3\text{-}10)$$

d) Shear stress due to Q_r

$$\tau_r = \frac{6 \cdot Q_r}{t} \qquad (8.3\text{-}11)$$

e) Shear stress due to Q_t

$$\tau_t = \frac{6 \cdot Q_t}{t} \qquad (8.3\text{-}12)$$

(3) The stress intensities shall be determined at the points with Φ = 0° and Φ = 90° based on the stress theory of von Mises or alternately on the theory of Tresca.

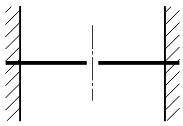
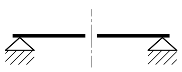
Clamping	K_i/A_i	Formula	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
	K_1	$-(K_2 + K_4)$	0.001273	0.005331	0.012650	0.023150	0.035340	0.046610	0.054870	0.059740
	K_2	$\frac{1}{2} \cdot K_3 - 2 \cdot K_4$	- 0.001295	- 0.005662	- 0.014040	- 0.026300	- 0.039440	- 0.048220	- 0.048500	- 0.039470
	K_3	$\frac{A_2}{A_1 \cdot A_3}$	- 0.002500	- 0.010000	- 0.022500	- 0.040000	- 0.062500	- 0.090000	- 0.122500	- 0.160000
	K_4	$-\frac{1}{A_3}$	0.000023	0.000331	0.001397	0.003149	0.004095	0.001609	- 0.006375	- 0.020260
		$A_1 = -3.3 \cdot \psi + \frac{1.3}{\psi}$			$A_2 = 6.6 \cdot \psi + \frac{1.4}{\psi^3}$			$A_3 = -\frac{A_2}{A_1} \cdot \left(3.3 + \frac{2.7}{\psi^2}\right) - 6.6 - \frac{1.4}{\psi^4}$		
	K_1	$-(K_2 + K_4)$	0.000488	0.001894	0.004066	0.006792	0.009848	0.013030	0.016190	0.019220
	K_2	$\frac{1}{6.6} \cdot (1.3 \cdot K_3 - 5.2 \cdot K_4)$	- 0.000511	- 0.002251	- 0.005791	- 0.011920	- 0.021460	- 0.035160	- 0.053600	- 0.077200
	K_3	$\frac{A_2}{A_1 \cdot A_3}$	- 0.002500	- 0.010000	- 0.022500	- 0.040000	- 0.062500	- 0.090000	- 0.122500	- 0.160000
	K_4	$-\frac{1}{A_3}$	0.000023	0.000357	0.001725	0.005123	0.011610	0.022120	- 0.037410	- 0.057980
		$A_1 = -1.3 \cdot \left(\psi - \frac{1}{\psi}\right)$		$A_2 = -1.4 \cdot \left(\psi - \frac{1}{\psi^3}\right)$			$A_3 = -\frac{A_2}{A_1} \cdot \left(-1.3 - \frac{2.7}{\psi^2}\right) + 1.4 \cdot \left(1 - \frac{1}{\psi^4}\right)$			

Table 8.3-1: Formulae and values for the constants K_i and the auxiliary values A_i

8.3.2.2.3 Calculation of deflections

The deflections can be determined for any point of the circular flat plate by means of the following equations:

a) The deflection of the plate midplane

$$w = \left(K_1 \cdot \rho^3 + K_2 \cdot \rho + K_3 \cdot \rho \cdot \ln\rho + K_4 \cdot \frac{1}{\rho}\right) \cdot A_0 \cdot \cos\phi \qquad (8.3\text{-}13)$$

b) The angle of slope of the tangent to the plate midplane

$$\alpha = \frac{1}{a} \cdot \left(3 \cdot K_1 \cdot \rho^2 + K_2 + K_3 (1 + \ln\rho) - K_4 \cdot \frac{1}{\rho^2}\right) \cdot A_0 \cdot \cos\phi \qquad (8.3\text{-}14)$$

with D and A_0 from equations (8.3-6) and (8.3-7) as well as K_1, K_2, K_3 and K_4 from **Table 8.3-1**.

8.3.3 Furnishing proof of functional capability

8.3.3.1 General

Note:

The proof of functional capability of the total system shall also comprise verifications for internal parts, such as shafts, gaskets and bearings as well as auxiliary and supply systems as well as verifications of hydraulic and mechanical performance data. Interactions with the pressure parts shall be taken into account. Such verifications do not fall under the scope of this safety standard and therefore shall be made in addition to the verifications for pressure-retaining walls.

(1) This clause contains the requirements for pressure-retaining casings which may be met regarding the functional capability of safety-related pumps.

(2) The proof of functional capability of pressure-retaining casings may be furnished, depending on the necessity and the evaluable results obtained, by way of calculation or experimental analysis or a combination of both methods.

(3) For pump casings this proof is usually furnished by functional tests at the manufacturer's works or in the plant (commissioning). If required, verifications by calculation or analogous considerations shall be made.

8.3.3.2 Verification by calculation

8.3.3.2.1 General

Verifications of functional capability by way of calculation shall be made by means of a stress and/or strain analysis and, if required, a stability analysis.

8.3.3.2.2 Functional capability of the pump during the load case

For those parts of the pump casing which are relevant to functional capability the primary stresses resulting from the loadings during the load case shall be limited according to the respective loading level, however, not beyond Level B. It shall be checked whether a strain analysis is to be performed in individual cases. The limitation of stresses may be omitted if it is proved that deformations of casing parts resulting from the loadings of the load case do not impair the functional capability. This is met if e.g. sufficient clearances or cross-sections are still available. In the case of fit-together parts between which there is relative movement (e.g. casing wear ring), short-time contact between the parts can be tolerated if wear-resistant non-fretting combination of materials have been selected.

8.3.3.2.3 Functional capability of the pump after occurrence of the load case

For those pump casing parts being relevant to functional capability the primary stresses resulting from the loadings during the load case shall be limited according to the respective loading level, however, not beyond Level C. Where consistent deformations (permanent set) occur it shall be verified that they do not inadmissibly affect the functional capability.

8.3.3.3 Verification by experimental analysis

Verifications of functional capability by experimental analysis shall, among other things, be made by a performance test on the test bench or during commissioning of the plant under special test conditions.

Prerequisite to the verification is that the results of the test conditions can be fully transferred to the functional requirements, loadings and other boundary conditions occurring during operation and accidents. Where not every boundary condition is realised to a sufficiently exact extent the verification by experimental analysis shall be supplemented by an analytical analysis.

8.4 Valve bodies

8.4.1 Design values and units relating to Section 8.4

Notation	Design value	Unit
d_{aA}	nominal outside diameter of valve in Section A-A, excluding allowances	mm
d_{aR}	nominal outside diameter of connected piping, excluding allowances	mm
d_i	nominal inside diameter as per Figure 8.4-1	mm
d_{iA}	nominal inside diameter of valve in Section A-A, excluding tolerances	mm
d_{iG}	valve body inside diameter as per Figure 8.4-5	mm
d_{iR}	nominal inside diameter of connected piping, excluding tolerances	mm
e	effective length	mm
f	factor acc. to Table 8.4-4	—
h	height according to Figure 8.4-3	mm
m, n	material parameters according to Table 7.8-1	—
p	design pressure for design loading level 0 or the respective internal pressure for loading levels A and B	MPa
p_B	internal pressure at the respective load case	MPa
Δp_{fi}	full range of pressure fluctuations from normal operating to the considered condition	MPa
$p_{f(max)}$	maximum range of pressure fluctuations Δp_{fi}	MPa
r	mean radius in Section A-A according to Figure 8.4-7	mm
r_2, r_4	fillet radius according to Figure 8.4-2	mm
r_3	radius according to Figure 8.4-3	mm
r_t	fillet radius according to Figure 8.4-7	mm
s_A	wall thickness of branch	mm
s_{An}	wall thickness according to Figure 8.4-7	mm
s_{Hn}	wall thickness according to Figure 8.4-7	mm
s_n	wall thickness of valve (acc. to cl. 7.1.4) in Section A-A according to Figures 8.4-4 and 8.4-5	mm
s_{ne}	wall thickness according to Figure 8.4-5	mm
s_R	wall thickness of connected piping according to Figure 8.4-4	mm
A	cross-sectional area of valve in Section A-A acc. to Figures 8.4-4 and 8.4-5	mm^2
A_p	pressure loaded area	mm^2
A_σ	effective cross-sectional area	mm^2
C_a	stress index for oblique valves acc. to equation (8.4-14)	—
C_b	stress index for bending stress acc. to equation (8.4-11)	—
C_2	stress index for secondary thermal stresses due to structural discontinuity in acc. with Figure 8.4-10	—
C_3	stress index for secondary stresses at locations of structural discontinuity due to changes in fluid temperature in acc. with Figure 8.4-8	—
C_4	factor acc. to Figure 8.4-11	—
C_5	stress index for thermal fatigue stress component acc. to Figure 8.4-12	—
C_6	stress index for thermal stresses acc. to equation (8.4-30)	$N \cdot mm^4$
D	usage factor	—
D_{e1}	diameter of the largest circle that can be drawn entirely within the wall at the crotch region, as shown in Figure 8.4-7	mm
D_{e2}	diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch	mm
E	modulus of elasticity at design temperature	N/mm^2
F_{ax}	axial force	N
F'_{ax}	axial force obtained from connected piping	N
M_b	bending moment	Nmm
M'_b	bending moment obtained from connected piping	Nmm
M_R	resulting moment	Nmm
M_t	torsional moment	Nmm

Notation	Design value	Unit
M_t'	torsional moment obtained from connected piping	Nmm
N_i	allowable number of cycles	—
N_{ri}	specified number of cycles	—
P_b	primary bending stress according to Table 7.7-5	N/mm²
P_{eb}	secondary stress from pipe reactions	N/mm²
$P_{eb\ max}$	secondary stress from pipe loadings with full utilization of the allowable stress	N/mm²
P_{lp}	local membrane stress due to internal pressure acc. to equation (8.4-5)	N/mm²
P_m	general primary membrane stress acc. to Table 7.7-5	N/mm²
Q	resulting transverse force	N
Q´	transverse force from connected piping	N
Q_p	sum of primary plus secondary stresses resulting from internal pressure acc. to equation (8.4-13)	N/mm²
Q_{T1}	thermal stress component from through-wall temperature gradient associated with a fluid temperature change rate ≤ 55 °K / hr	N/mm²
Q_{T3}	thermal secondary stress resulting from structural discontinuity according to equation (8.4-15)	N/mm²
R_{mT}	minimum tensile stress of connected piping at design temperature	N/mm²
$R_{p0.2T}$	0.2% proof stress of connected piping at design temperature	N/mm²
S	design stress intensity acc. to Table 6.6-1	N/mm²
S_a	one-half the value of cyclic stress range	N/mm²
$S_{Armatur}$	design stress intensity S or S_m for the valve body material at design temperature in acc. with Table 6.6-1	N/mm²
S_i	peak stress	N/mm²
S_m	design stress intensity according to clause 7.7.3.4	N/mm²
S_n	sum of primary plus secondary stress intensities for one load cycle	N/mm²
$S_{n(max)}$	maximum range of primary plus secondary stresses according to equation (8.4-32)	N/mm²
S_{p1}	general stress intensity at inside surface (crotch region) of body	N/mm²
S_{p2}	general stress intensity at outside surface (crotch region) of body	N/mm²
S_R	design stress intensity acc. to Table 8.4-1	N/mm²
S_{Rohr}	design stress intensity S or S_m for material for connected piping at design temperature in acc. with Table 6.6-1	N/mm²
T	design temperature	K
T_{De1}	temperature acc. to Figure 8.4-6	K
T_{sn}	temperature acc. to Figure 8.4-6	K
$\Delta T'$	maximum magnitude of the difference in wall temperatures for walls of thicknesses (D_{e1}, s_n) resulting from 55 °K/hr fluid temperature change rate acc. to Figure 8.4-9	K
ΔT_f	fluid temperature change	K
ΔT_{fi}	fluid temperature change in Section i	K
$\Delta T_{f(max)}$	maximum change in fluid temperature	K

Notation	Design value	Unit
ΔT_{f1}, ΔT_{f2}, ΔT_{f3}, ΔT_1, ΔT_2, ΔT_3	change in fluid temperature (range of temperature cycles)	K
$W_{Armatur}$	axial section modulus at valve body nominal dimension referring to Section A-A in Figures 8.4-4 and 8.4-5 acc. to equation (8.4-8)	mm³
W_{Rohr}	axial section modulus of connected piping referring to the nominal dimension acc. to equation (8.4-7)	mm³
W_t	valve body section torsional modulus in Section A-A acc. to Figures 8.4-4 and 8.4-5 ($W_t = 2 \cdot W_A$ for circular cross-section with constant wall thickness)	mm³
α	linear coefficient of thermal expansion at design temperature	1/K
α_1	acute angle between flow passage centre lines and bonnet (stem, cone) acc. to Figure 8.4-4	grd
σ_b	stress resulting from bending moments	N/mm²
σ_L	stress from loadings in direction of pipe axis	N/mm²
σ_V	stress intensity	N/mm²
$\tau_{a\ max}$	stress resulting from transverse forces	N/mm²
τ_t	stress resulting from torsional moment	N/mm²

8.4.2 General

(1) For valves meeting all the requirements of this clause, the most highly stressed portions of the body under internal pressure is at the neck to flow passage junction and is characterized by circumferential tension normal to the plane of centre lines, with the maximum value at the inside surface. The rules of clause 8.4.3 are intended to control the general primary membrane stress in the crotch region.

(2) In the crotch region, the maximum primary membrane stress is to be determined by the pressure area method in accordance with the rules of clause 8.4.3. The procedure is illustrated in **Figure 8.4-1**.

(3) The P_m value calculated in accordance with clause 8.4.3 will normally be the highest value of body general primary membrane stress for all normal valve types with typical wall proportioning, whereas in regions other than the crotch unusual body configurations shall be reviewed for possible higher stress regions. Suspected regions are to be checked by the pressure area method applied to the particular local body contours.

(4) The use of the methods of component-specific stress analysis described in clauses 8.4.4 and 8.4.5 necessitates that the requirements set forth in clause 8.4.3 regarding the evaluation of primary membrane stress due to internal pressure are satisfied.

(5) The stress analysis of valve bodies usually is performed in accordance with the methods of clause 8.4.4 or 8.4.5. Loadings resulting from connected pipe are to be generally considered (i.e. by using the maximum possible bending moment of the connected piping).

(6) Clause 8.4.6 may be applied alternately or if the conditions of clause 8.4.4 or 8.4.5 have not been satisfied.

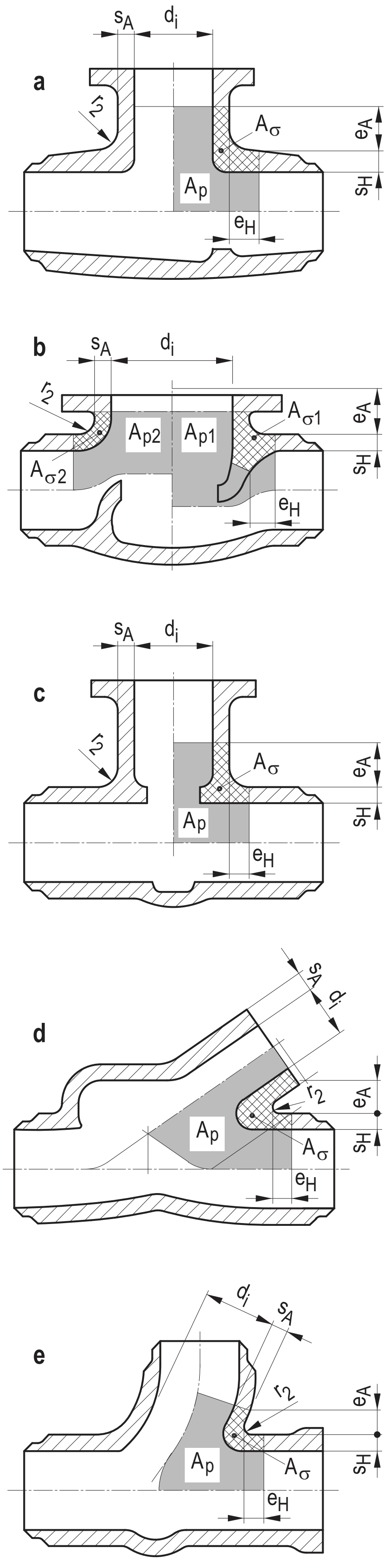


Figure 8.4-1: Pressure area method

8.4.3 Primary membrane stress due to internal pressure

(1) From a drawing to scale of the valve body, depicting the finished section of the crotch region in the mutual plane of the bonnet and flow passage centre lines, determine the fluid (load-bearing) area A_p and the effective cross-sectional (metal) area A_σ. A_p and A_σ are to be based on the internal surface of the body after complete loss of metal assigned to corrosion allowance.

(2) Calculate the crotch general membrane stress intensity as follows:

$$P_m = \left(A_p / A_\sigma + 0.5\right) \cdot p \le S_m \qquad (8.4\text{-}1)$$

The design stress intensity S_m shall be taken from **Table 6.6-1**.

(3) The distances e_H and e_A which provide bounds on the fluid and metal areas are determined as follows; see **Figure 8.4-1**:

$$e_H = \max.\{0.5 \cdot d_i - s_A; s_H\} \qquad (8.4\text{-}2)$$

$$e_A = 0.5 \cdot r_2 + 0.354 \cdot \sqrt{s_A \cdot (d_i + s_A)} \qquad (8.4\text{-}3)$$

In establishing appropriate values for the above parameters, some judgement may be required if the valve body is irregular as it is for globe valves and others with nonsymmetric shapes. In such cases, the internal boundaries of A_p shall be the lines that trace the greatest widths of internal wetted surfaces perpendicular to the plane of the stem and pipe ends (see **Figure 8.4-1**, sketches b, d and e).

(4) If the calculated boundaries for A_p and A_σ, as defined by e_A and e_H, fall beyond the valve body (**Figure 8.4-1**, sketch b, see also **Figure A 4.1-8**), the body surface becomes the proper boundary for establishing A_p and A_σ. No credit is to be taken for any area of connected piping which may be included within the limits of e_A and e_H. If the flange is included in A_σ, no credit will be taken for the flange area, too.

(5) Web or fin-like extensions of the valve body are to be credited to A_σ only to an effective length from the wall equal to the average thickness of the credited portion. The remaining web area is to be added to A_p (**Figure 8.4-1**, sketch b). In addition, the web area credited to A_σ shall satisfy the following condition: A line perpendicular to the plane of the stem and pipe ends from any points in A_σ does not break out of the wetted surface but passes through a continuum of metal until it breaks through the outer surface of the body.

(6) In the case of normal valve body configurations, it is expected that the portions defined by A_σ in the illustrations of **Figure 8.4-1** will be most highly stressed. However, in the case of highly irregular valve bodies, it is recommended that all sections of the crotch be checked to ensure that the largest value of P_m has been established considering both open and fully closed conditions.

8.4.4 General stress analysis for test group A1

(1) This method shall only be applied if the following geometric conditions are satisfied:

a) radius $r_2 \ge 0.3 \cdot s_n$

b) radius $r_3 \ge \max. \begin{cases} 0.05 \cdot s_n \\ 0.1 \cdot h \end{cases}$

c) radius $r_4 < r_2$ is permitted

d) the edges must be chamfered or trimmed.

The radii r_2 and r_4 are shown in **Figure 8.4-2** for the various types of fillet radii. r_3 and h are explained in **Figure 8.4-3**. s_n is the nominal wall thickness according to clause 7.1.4 and **Figure 8.4-4**.

(2) It shall be checked by means of equation (8.4-4) whether the range of allowable primary membrane plus bending stresses in loading levels A and B is not exceeded.

$$P_{lp} + P_{eb} \le 1.5 \cdot S_m \qquad (8.4\text{-}4)$$

$$P_{lp} = 1.5 \cdot \left(\frac{d_{iA}}{2 \cdot s_n} + 0.5\right) \cdot p \cdot C_a \qquad (8.4\text{-}5)$$

with

C_a acc. to equation (8.4-14)

P_{eb} acc. to equation (8.4-6).

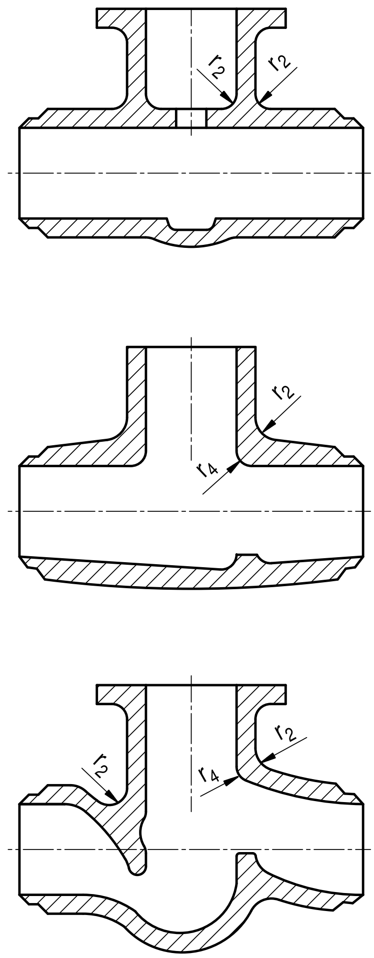


Figure 8.4-2: Fillets and corners

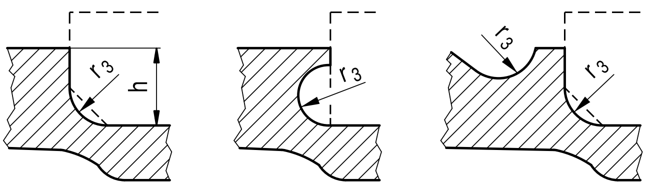


Figure 8.4-3: Acceptable ring grooves

(3) For the purpose of verifying the stress portions resulting from unit shear forces and unit moments of the connected piping, bending stresses in the governing sections acc. to **Figures 8.4-4** and **8.4-5** shall be evaluated as essential stress components.

(4) The bending stresses are determined from:

$$P_{eb} = \frac{C_b \cdot W_{Rohr} \cdot S_R}{W_{Armatur}} \qquad (8.4\text{-}6)$$

with

$$W_{Rohr} = \frac{\pi \cdot \left(d_{aR}{}^4 - d_{iR}{}^4\right)}{32 \cdot d_{aR}} \qquad (8.4\text{-}7)$$

$$W_{Armatur} = \frac{\pi \cdot \left(d_{aA}{}^4 - d_{iA}{}^4\right)}{32 \cdot d_{aA}} \qquad (8.4\text{-}8)$$

where the following condition must be satisfied:

$$W_{Armatur} \geq W_{Rohr} \qquad (8.4\text{-}9)$$

(5) For valve bodies with conical hub acc. to **Figure 8.4-5** the Section A-A shall be taken in consideration of the die-out length e. Here, the following applies:

$$e = 0.5 \cdot \sqrt{\frac{d_{iA} \cdot s_{ne}}{2}} \qquad (8.4\text{-}10)$$

with d_{iA} and s_{ne} according to **Figure 8.4-5**.

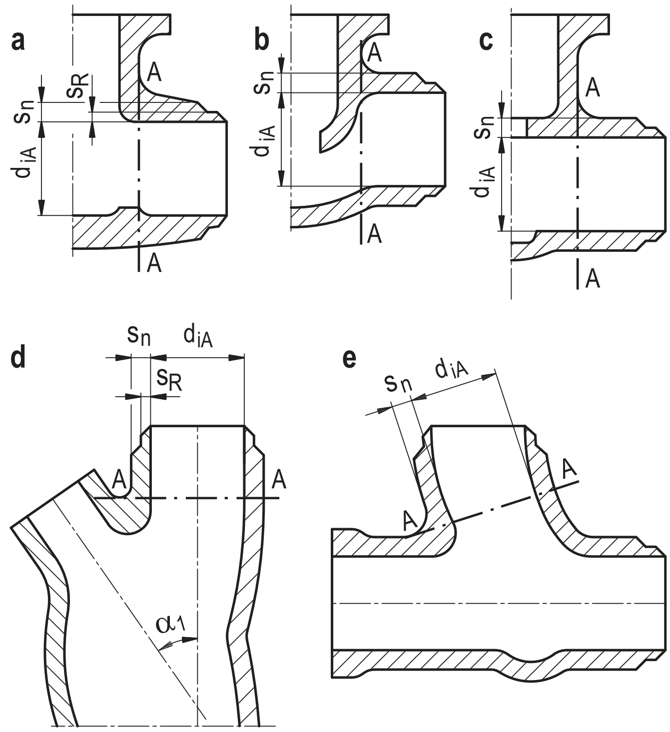
(6) The stress index value C_b is determined as follows:

$$C_b = \max \left\{ 0.335 \cdot \left(\frac{r}{s_n} \right)^{\frac{2}{3}} ; 1.0 \right\} \qquad (8.4\text{-}11)$$

with r and s_n according to **Figures 8.4-4** and **8.4-5**.

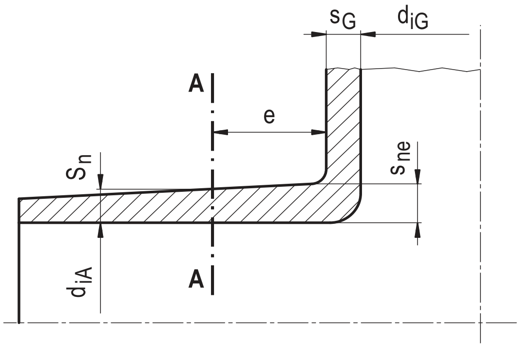
(7) The S_R value in equation (8.4-6) refers to the material of the connected piping. The values of **Table 8.4-1** shall be taken.

(8) No greater loadings on the valve shall be considered than are allowed by the stress intensity level in the piping system. Provided that the same pipe materials, same diameters and section moduli of the valve are considered by the design and the valve itself does not constitute an anchor, the valve body side with the smallest section modulus of the connected piping shall govern the maximum loading of the valve. Otherwise, both sides of the valve body shall be assessed to determine the maximum possible loading.



$$r = \frac{d_{iA} + s_n}{2}$$

Figure 8.4-4: Critical sections of valve bodies



$$r = \frac{d_{iA} + s_n}{2}$$

Figure 8.4-5: Critical section at conical valve bodies

Loading level	Composite materials			
	Pipe	Valve	Pipe	Valve
	Ferrite Ferrite Ferrite Ferrite	Ferritic steel forging Ferritic cast steel Austenitic steel forging Austenitic cast steel	Austenite Austenite Austenite Austenite	Austenitic steel forging Ferritic steel forging Austenitic cast steel Ferritic cast steel
	S_R		S_R	
0	$R_{p0.2T}$		$1.35 \cdot R_{p0.2T}$	
A	$R_{p0.2T}$		$1.35 \cdot R_{p0.2T}$	
B	$R_{p0.2T}$		$1.35 \cdot R_{p0.2T}$	
C	$1.2 \cdot R_{p0.2T}$		$1.62 \cdot R_{p0.2T}$	
D	$\min.\begin{cases}1.6 \cdot R_{p0.2T}\\ R_{mT}\end{cases}$		$\min.\begin{cases}2.16 \cdot R_{p0.2T}\\ R_{mT}\end{cases}$	
$R_{p0.2T}$, R_{mT} : design strength values of connected piping at design temperature				

Table 8.4-1: List of limit values for S_R to be used in the analysis (equation 8.4-6) of the connected piping for composite materials of piping and valve

(9) For equation (8.4-6) the allowable stresses in the various loading levels acc. to **Table 8.4-2** shall be adhered to. When using Table 8.4-2, the following design requirements apply:

a) $d_{iA} \le d_{iG}$ (see **Figure 8.4-5**)

b) $s_n \le s_G$

c) In the case of corner valves it shall be verified that the nozzles do not influence each other; this verification is not required for prismatic body geometries.

The design stress intensity S_m shall be taken from **Table 6.6-1**.

Loading level	Allowable value for P_{eb}
A	$1.5 \cdot S_m$
B	$1.5 \cdot S_m$
C	$1.8 \cdot S_m$
D	$2.4 \cdot S_m$

Table 8.4-2: Allowable stress in the body resulting from pipe loadings

(10) For the calculation of the sum of primary and secondary stresses in Levels A and B the following applies:

$$S_n = Q_P + P_{eb} + 2\, Q_{T3} \qquad (8.4\text{-}12)$$

$$Q_p = 3.0 \cdot \left(\frac{d_{iA}}{2 \cdot s_n} + 0.5\right) \cdot p \cdot C_a \qquad (8.4\text{-}13)$$

where

$$C_a = 0.2 + \frac{0.8}{\sin\alpha_1} \qquad (8.4\text{-}14)$$

α_1 angle between flow passage centre lines in valve body and bonnet (spindle, cone) acc. to **Figure 8.4-4**

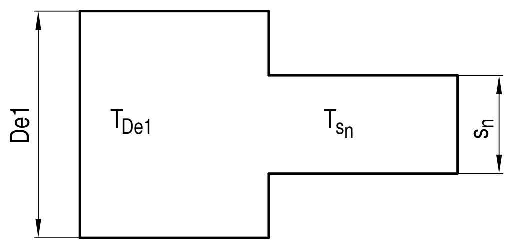
P_{eb} shall be inserted acc. to equation (8.4-6).

d_{iA} and s_n shall be taken from **Figures 8.4-4** and **8.4-5**.

Q_{T3} is determined as follows:

$$Q_{T3} = E \cdot \alpha \cdot C_3 \cdot \Delta T' \qquad (8.4\text{-}15)$$

D_{e1} and D_{e2} shall be determined by means of a detail sketch with reference to the original drawing at a suitable scale.



$\Delta T' = (T_{De1} - T_{sn})$

Figure 8.4-6: Determination of $\Delta T'$

(11) For the loading Levels C and D the following applies:

$$S_n = P_{lp} + P_{eb} \qquad (8.4\text{-}16)$$

P_{lp} is determined from equation (8.4-5); for p the respective internal pressure of Level C or D shall be used.

(12) In the individual loading levels the stress intensity values acc. to **Table 8.4-3** shall not be exceeded in equations (8.4-12) and (8.4-16). The design stress intensity S_m shall be taken from **Table 6.6-1**.

Loading level	Allowable S_n value	
	Forged steel	Cast steel
A	$3 \cdot S_m$	$4 \cdot S_m$
B	$3 \cdot S_m$	$4 \cdot S_m$
C	$2.25 \cdot S_m$	$3 \cdot S_m$
D	$3 \cdot S_m$	$4 \cdot S_m$

Table 8.4-3: Allowable stress intensities for the sum of primary plus secondary stresses in the valve body

(13) The verification for loading levels C and D shall only be made if the respective requirement has been fixed in the component-specific documents.

(14) Valve and piping system may be classified into different loading levels for specific load cases (see component-specific document). In such a case the S_R value for equation (8.4-6) shall be taken with respect to the loading level of the system (see **Table 8.4-1**).

(15) The verification with the equations (8.4-1) to (8.4-16) is only permitted if for all load cases the allowable stress intensity level is not exceeded in the connected piping.

(16) Where pipe rupture is assumed and no anchor is provided between valve and location of rupture, the calculation of the valve body shall be made with the effective or with conservatively assumed pipe unit shear forces and unit moments if valve integrity or functional capability is required by the component-specific document.

8.4.5 General stress analysis for test groups A2 and A3

(1) If the following conditions are satisfied, the stress is deemed to be verified:

$$W_{Armatur} \geq 1.1 \cdot \frac{S_{Rohr}}{S_{Armatur}} \cdot W_{Rohr} \tag{8.4-17}$$

and

$$W_{Armatur} \geq 1.5 \cdot W_{Rohr} \tag{8.4-18}$$

(2) Equations (8.4-17) and (8.4-18) shall apply to cover all loading levels. Where the valve and piping system are classified into differing loading levels in the case of specific load cases (see component-specific documents), W_{Rohr} shall be multiplied with the factor f in equation (8.4-17) in accordance with **Table 8.4-4**.

Loading level		f
Pipe	Valve	
D	B	1.6
D	C	1.33
C	B	1.2

Table 8.4-4: Determination of factor f

(3) Where pipe rupture is assumed and no anchor is provided between valve and location of rupture, equation (8.4-23) shall be satisfied by adherence to the stress intensity limit for $P_m + P_b$ according to **Table 6.7-1** allowable in this case if valve integrity or functional capability is required by the component-specific document.

In addition, the stress S_n of equation (8.4-16) according to **Table 8.4-3** shall be limited for this case, and the value of σ_V according to equation (8.4-23) shall be taken for P_{eb} in equation (8.4-16).

The allowable stress intensities shall be determined by using S instead of S_m.

(4) The stress analysis may also be made in accordance with clause 8.4.4 alternatively to the method described here in which case S shall be further used in lieu of S_m.

8.4.6 Detailed stress analysis with unit shear forces and unit moments obtained from the calculated connected piping

(1) The verification according to this clause is only required if, in the general stress analysis to clauses 8.4.3 or 8.4.4, the allowable stress limit is exceeded or the required condition cannot be satisfied in any case. It is applicable to valves of test groups A1, A2 and A3 in which case the geometric conditions in accordance with clause 8.4.4 (1) and the design requirements to sub-clause 8.4.4 (9) shall also be satisfied. Load cases and superposition of loads shall be taken from the component-specific documents.

(2) From the calculation of the connected piping the following forces and moments are obtained which act on the two points of attachment of the valve for the various load cases:

a) axial forces F'_{ax}

b) transverse forces Q'

c) bending moments M'_b

d) torsional moments M'_t

In accordance with the superposition rule F_{ax}, Q, M_b and M_t shall be determined for each loading level and the stress components shall be calculated from the unit shear forces and unit moments from the connected piping as follows:

Stress resulting from loadings in the direction of pipe axis:

$$\sigma_L = \frac{d_{aA} \cdot p_B}{4 \cdot s_n} + \frac{F_{ax}}{A} \tag{8.4-19}$$

Stress resulting from transverse forces:

$$\tau_{a\,max} = \frac{2 \cdot Q}{A} \tag{8.4-20}$$

Stress resulting from bending moments:

$$\sigma_b = \frac{M_b}{W_A} \cdot C_b \tag{8.4-21}$$

Stress resulting from torsional moment:

$$\tau_t = \frac{M_t}{W_t} \tag{8.4-22}$$

When determining A, W_A and W_t it shall be taken into account that the wall thickness at the valve body inside is to be reduced by the wear allowance.

(3) These individual stresses are simplified to form a stress intensity on the assumption that the maximum stresses all occur simultaneously:

$$\sigma_V = \sqrt{(\sigma_L + \sigma_b)^2 + 3 \cdot \left(\tau_{a_{max}} + \tau_t\right)^2} \tag{8.4-23}$$

(4) For equation (8.4-23) the stress intensity limit values for $P_m + P_b$ according to **Table 6.7-1** shall be adhered to in the various loading levels and test groups.

The design stress intensities S_m and S shall be determined in accordance with **Table 6.6-1**.

(5) The primary and secondary stresses shall be determined in accordance with clause 8.4.4 if the valve has been classified into test group A1.

Here the stress intensity σ_V determined according to equation (8.4-23) shall be taken for P_{eb} in equations (8.4-12) and (8.4-16).

For S_n the allowable stress intensity values according to **Table 8.4-3** then apply.

(6) Where at the time of calculation the valve design has already been made and the unit shear forces and unit moments obtained from the calculation of the connected piping are not yet available they may be fixed as follows:

a) From equations (8.4-12) or (8.4-16) for S_n a value $P_{eb\,max}$ is obtained for each individual loading level if the allowable stress is fully utilized.

b) Where this value ($P_{eb\,max}$) exceeds the allowable stress intensity for equation (8.4-23), $P_{eb\,max}$ shall be reduced to obtain this value.

c) Taking:

$$\sigma_L = \sigma_b = 2 \cdot (\tau_{a\,max} + \tau_t) \tag{8.4-24}$$

and

$$\tau_{amax} = \tau_t = \frac{\sigma_b}{4} \tag{8.4-25}$$

and

$$\sigma_V \leq P_{eb\,max} \tag{8.4-26}$$

the following is obtained:

$$\sigma_b = \sigma_L = \frac{P_{eb\,max}}{\sqrt{5}} \tag{8.4-27}$$

d) With these values the stress intensity σ_V according to equation (8.3-23) shall be determined, and the reliability of this value shall be checked.

e) Where the allowable stress intensity value is adhered to, F_{ax}, Q, M_b and M_t can be determined directly from the values in subclause c). Otherwise, the individual stresses in subclause c) shall be reduced uniformly until the allowable stress intensity value is no more exceeded.

These unit shear forces and unit moments then shall not be exceeded within the calculation of the connected piping or varied only such that they do not lead to a higher loading of the valves. In addition, it shall be taken into account whether, with respect to the classification of the valve according to the component-specific documents, a classification into another loading level and thus a reclassification of the unit shear forces and unit moments may be required to perform a verification of the functional capability by way of calculation.

8.4.7 Fatigue analysis

8.4.7.1 General

A fatigue analysis shall be made for all valves exceeding DN 50 in test groups A1 and A2 with the specified number of load cycles - to be at least 1000 - . A fatigue analysis for test group A3 shall be waived.

Note:

The fatigue analysis methods described hereinafter are so conservative that stress intensifications for valve bodies with multiple external contours are covered by the examination of the critical section according to ***Figure 8.4-7****.*

8.4.7.2 General fatigue evaluation

General fatigue evaluation shall be made for loading Levels A and B in accordance with the methods described hereinafter and shall replace the fatigue analysis according to clause 8.4.7.3 or Section 7.8 if the resulting number of load cycles is greater than the specified number of cycles, however, is greater than 2000.

The maximum total stresses S_{p1} on the body inside and S_{p2} on the body outside can be determined by assuming a fluid temperature change rate not exceeding 55 K/hr as follows:

$$S_{p1} = \frac{2}{3} \cdot Q_p + \frac{P_{eb}}{2} + Q_{T3} + 1{,}3 \cdot Q_{T1} \tag{8.4-28}$$

$$S_{p2} = 0.4 \cdot Q_p + P_{eb} + 2 \cdot Q_{T3} \tag{8.4-29}$$

with

$$Q_{T1} = C_6 \cdot (D_{e1})^2 \tag{8.4-30}$$

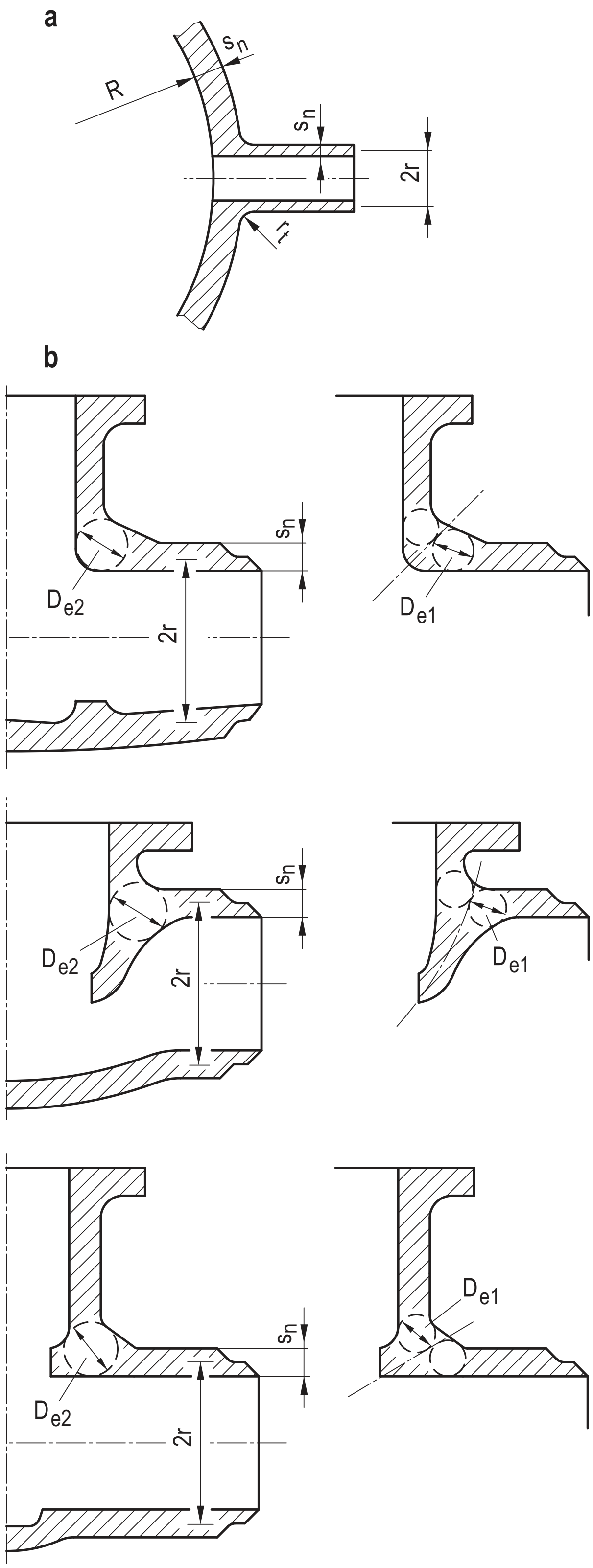
$1.3 \cdot Q_{T1}$ stress component from non-linear temperature distribution

C_6 stress index for thermal stresses

$4.06 \cdot 10^{-3}$ N/mm^4 for austenitic material

$1.07 \,.\, 10^{-3}$ N/mm^4 for ferritic material

With the larger value of S_{p1} and S_{p2} taken as S_a the allowable number of load cycles is obtained from the fatigue curves according to **Figures 7.8-1**, **7.8-2** or **7.8-3** where it shall be taken into account that the difference between the elastic modulus from the curves and that of the valve materials at design temperature is to be considered. The S_a value shall be multiplied with the ratio of E (curve)/E (valve) at design temperature.



D_{e1} = diameter of the largest circle which can be drawn entirely within the wall at the crotch region

D_{e2} = diameter of the largest circle which can be drawn in an area of the crotch on either side of a line bisecting the crotch

For $D_{e1} < s_n$ the following applies: $D_{e1} = s_n$

Figure 8.4-7: Model for determining secondary stresses in valve bodies (crotch region)

8.4.7.3 Detailed fatigue analysis

Note:

The procedure outlined hereinafter can lead to non-conservative results at temperature change rates exceeding 10 K/min.

(1) To perform a detailed fatigue analysis the pressure changes Δp_{fi} and temperature changes ΔT_{fi} with the pertinent number N_{ri} shall be determined for all specified load cycles resulting from operational loadings.

(2) If both heating or cooling effects are expected at fluid temperature change rates exceeding 55 K/hr, the temperature range associated with the pertinent number of cycles per load case each shall be determined assuming e.g. the following variations:

Example:

20 variations ΔT_1 = 250 K heating

10 variations ΔT_2 = 150 K cooling

100 variations ΔT_3 = 100 K cooling

Lump the ranges of variation so as to produce the greatest temperature differences possible:

10 cycles T_{f1} = 150 K + 250 K = 400 K

10 cycles T_{f2} = 250 K + 100 K = 350 K

90 cycles T_{f3} = 100 K

(3) Pressure fluctuations not excluded by the condition in subclause a) hereinafter are to be included in the calculation of the peak stresses. The full range of pressure fluctuations from normal operating condition to the condition under consideration shall be represented by Δp_{fi}.

During the fatigue analysis the following load variations or load cycles need not be considered:

a) pressure variations less than 1/3 of the design pressure for ferritic materials,

pressure variations less than 1/2 of the design pressure for austenitic materials,

b) temperature variations less than 17 K,

c) accident or maloperation cycles expected to occur less than five times (total) during the expected valve life,

d) start-up and shutdown cycles with temperature change rates not exceeding 55 K/hr at a number of load cycles n not exceeding 2000.

(4) For the greatest pressure fluctuations max $\Delta p_{fi} = \Delta p_{f(max)}$ and temperature changes max $\Delta T_{fi} = \Delta T_{f(max)}$ the following equation must be satisfied:

$$Q_p \cdot \frac{p_{f(max)}}{p} + E \cdot \alpha \cdot C_2 \cdot C_4 \cdot \Delta T_{f\,(max)} \begin{cases} \leq 3 \cdot S_m & \text{for forging steel} \\ \leq 4 \cdot S_m & \text{for cast steel} \end{cases} \qquad (8.4\text{-}31)$$

where Q_p shall be determined by equation (8.4-13).

The factors C_2 and C_4 shall be taken from **Figures 8.4-10** and **8.4-11**, respectively. The design stress intensity S_m shall be determined according to **Table 6.6-1**.

(5) $S_{n(max)}$ shall be determined as follows:

$$S_{n(max)} = Q_p \cdot \frac{\Delta p_{f(max)}}{p} + E \cdot \alpha \cdot C_3 \cdot C_4 \cdot \Delta T_{f(max)} \qquad (8.4\text{-}32)$$

Stress index C_3 shall be taken from **Figure 8.4-8**.

Equation (8.4-32) for $S_{n(max)}$ can be calculated separately for each load cycle. Here Δp_{fi} and ΔT_{fi} are then inserted.

(6) The peak stresses S_i shall be calculated as follows:

$$S_i = \frac{4}{3} \cdot Q_p \cdot \frac{\Delta p_{fi}}{p} + E \cdot \alpha \cdot (C_3 \cdot C_4 + C_5) \cdot \Delta T_{fi} \qquad (8.4\text{-}33)$$

C_5 shall be taken from **Figure 8.4-12**.

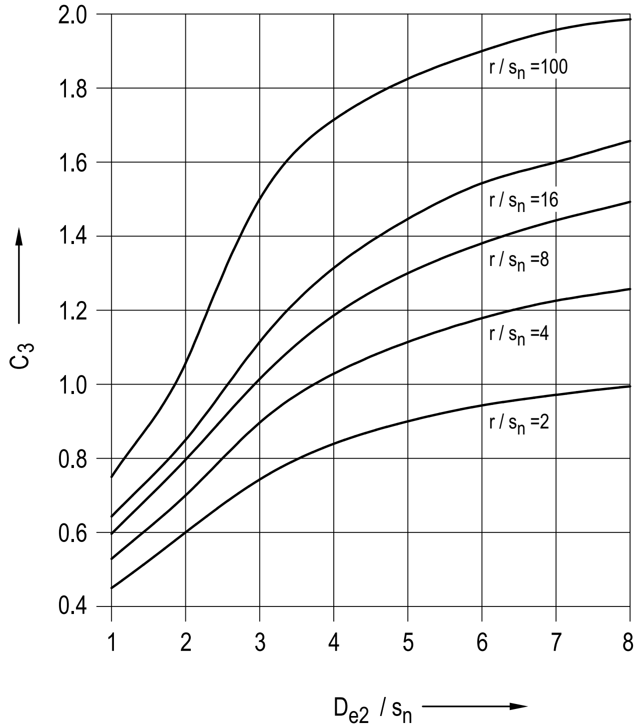


Figure 8.4-8: Stress index for secondary stresses resulting from structural discontinuity due to fluid temperature changes

(7) The half-value of the cyclic stress range S_a for determining the allowable number of cycles N_i shall be calculated as follows:

a) for $S_{n(max)} \leq 3 \cdot S_m$

$$S_a = \frac{S_i}{2} \qquad (8.4\text{-}34)$$

b) for $3 \cdot S_m < S_{n(max)} \leq 3 \cdot m \cdot S_m$

$$S_a = \left[1 + \frac{1-n}{n(m-1)} \cdot \left(\frac{S_n}{3 \cdot S_m} - 1\right)\right] \cdot \frac{S_i}{2} \qquad (8.4\text{-}35)$$

Here, the value of $S_{n(max)}$ or the value S_n determined separately for each load cycle may be used in lieu of S_n. Where in individual load cycles S_n does not exceed $3 \cdot S_m$, the method of subclause a) shall be applied. The material parameters m and n shall be taken from **Table 7.8-2.**

c) for $S_{n(max)} > 3 \cdot m \cdot S_m$

$$S_a = \frac{1}{n} \cdot \frac{S_i}{2} \qquad (8.4\text{-}36)$$

For cast steel the value of $3 \cdot S_m$ shall be substituted by $4 \cdot S_m$ in the conditions of subclauses a) to c).

The allowable numbers of load cycles N_i shall be taken from the fatigue curves in **Figures 7.8-1**, **7.8-2** or **7.8-3** where it shall be taken into account that the difference between the elastic modulus from the curves and that of the valve material at allowable operating temperature (design temperature) are considered.

The S_a value shall be multiplied with the ratio E (curve)/ E (valve) at allowable operating temperature (design temperature).

(8) The fatigue usage (usage factor) D shall be determined as follows:

$$D = \sum \frac{N_{ri}}{N_i} \leq 1.0 \qquad (8.4\text{-}37)$$

where N_i is the allowable number of load cycles and N_{ri} the specified number of cycles according to the component-specific documents.

Where a reduction of fatigue strength due to fluid effects cannot be excluded, then the following measures shall be taken at a threshold for cumulative damage of D = 0.4 to ensure consideration of fluid influence on the fatigue behaviour:

a) the components considered shall be included in a monitoring program to KTA 3211.4, or

b) experiments simulating operating conditions shall be performed, or

c) verifications by calculation shall be made in due consideration of fluid-effected reduction factors and realistic boundary conditions.

8.4.8 Other methods of stress and fatigue analysis

Where the allowable limit values are exceeded when applying the clauses 8.4.4 to 8.4.7 the verification may also be made in accordance with Section 7.7 and 7.8, if required.

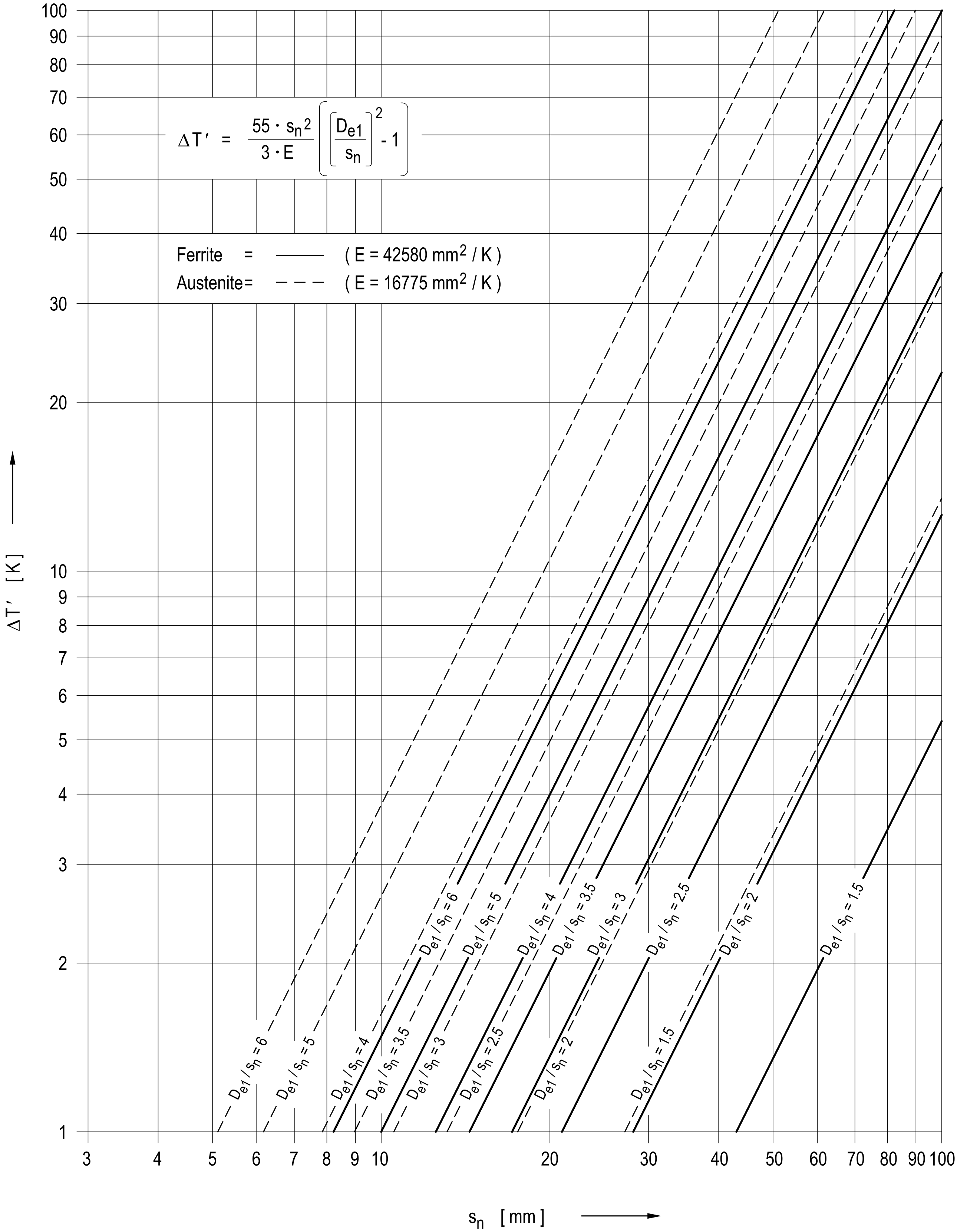


Figure 8.4-9: Maximum temperature difference in valve body (area D_{e1}/s_n), associated with a fluid temperature change rate of 55 K/hr

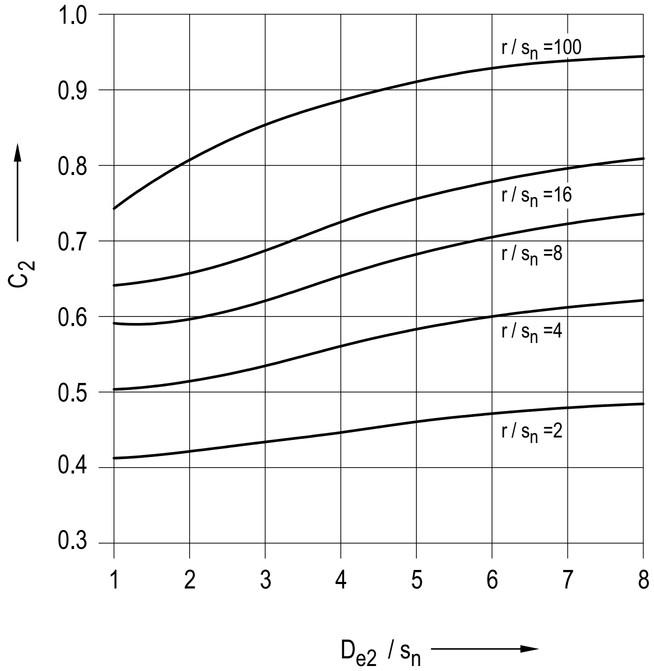


Figure 8.4-10: Stress index C_2 for secondary thermal stresses resulting from structural discontinuity

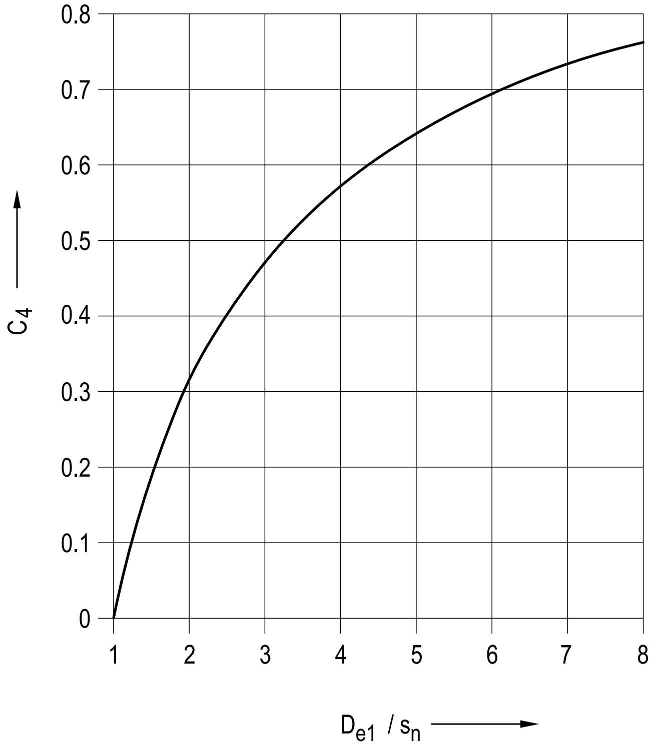


Figure 8.4-11: Maximum magnitude C_4 of the difference in average wall temperatures for wall thicknesses D_{e1} and s_n, resulting from a step change in fluid temperature ΔT_f

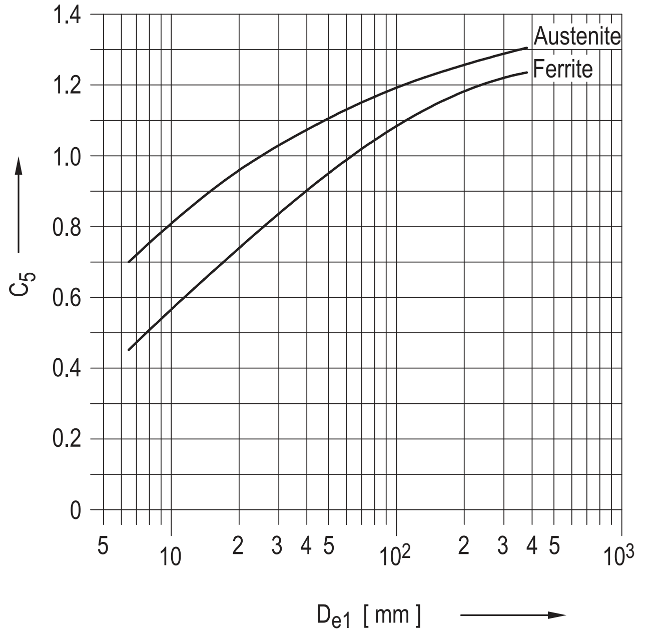


Figure 8.4-12: Stress index C_5 for consideration of thermal fatigue stresses resulting from through-wall temperature gradients caused by step change in fluid temperature

8.5 Piping systems

8.5.1 General

(1) Prerequisite to the application of the component-specific stress and fatigue analysis outlined hereinafter is the design of piping components as per clause 5.3.5 and the dimensioning of the piping components in accordance with Annex A 5.

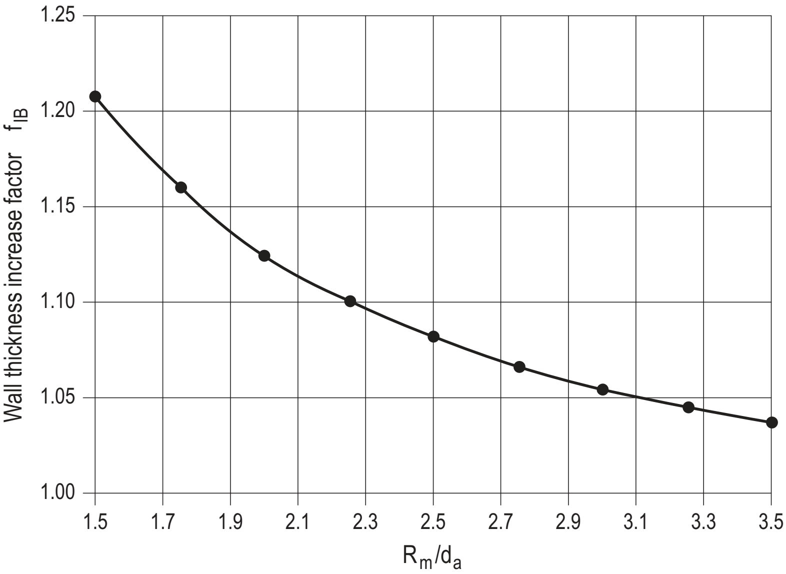
(2) For piping of test group A1 the stress and fatigue analysis according to clause 8.5.2 shall be made. For piping of test group A2 and A3 the stress and fatigue analysis according to clause 8.5.3 shall be made.

(3) The range of application extends to the tube-side effective length e_a (see **Figure A 5-14**) of the reinforced or unreinforced nozzle. This limit is not relevant to the modelling of the system analysis according to clause 7.6.2.

(4) The analysis of the mechanical behaviour of the total system shall be used to determine the directional components of forces and moments of the system in which case the thermal expansion load cases shall be calculated with the modulus of elasticity for the operating temperature. The thermal expansion stresses shall be converted to form a ratio E_k/E_w which is then used to evaluate the various piping elements. When determining stresses the internal pressure shall also be considered in addition to the forces and moments obtained from the analysis of the mechanical behaviour, and additionally for test group A1, the axial and radial temperature distributions.

(5) In lieu of the verifications by calculation for piping systems also standardized pipe laying procedures meeting the requirements of this Safety Standard may be used.

(6) For induction bends meeting the dimensional requirements of KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bend), the design wall thickness for induction bends, $s_{c,IB}$, which considers the notch (wall thickness increase at bend intrados) is derived from the relation $s_{c,IB} = s_c \cdot f_{IB}$, where the factor f_{IB} is to be determined as a function of R_m/d_a from **Figure 8.5-1**. Where the wall thickness ratios R_m/d_a exceeds 3.5, the influence of notches may be negligible if the specifications of Figure 9-1 of KTA 3211.3 are satisfied.



$$f_{IB} = \sqrt{f_{IB,i} \cdot f_{IB,a}}$$

For induction bends to KTA 3211.3, Figure 9-1 the following applies:

$$f_{IB,i} = 0.9091 + 1.202 \cdot \left(\frac{R_m}{d_a}\right)^{-1.24}$$

$$f_{IB,a} = \left[\begin{array}{c} \text{for}: 1.5 \le \dfrac{R_m}{d_a} < 2 \Rightarrow 0.8925 \\ \text{for}: 2 \le \dfrac{R_m}{d_a} < 3.5 \Rightarrow 0.021 \cdot \dfrac{R_m}{d_a} + 0.8505 \end{array}\right]$$

$f_{IB,i}$: wall thickness increase factor at intrados

$f_{IB,a}$: wall thickness reduction factor at extrados

Approximation equation:

$$f_{IB} = -0.0197 \cdot \left(\frac{R_m}{d_a}\right)^3 + 0.1892 \cdot \left(\frac{R_m}{d_a}\right)^2 - 0.6434 \cdot \left(\frac{R_m}{d_a}\right) + 1.8134$$

Figure 8.5-1: Wall thickness increase factor for f_{IB} for standard induction bends

8.5.2 Piping of test group A1

8.5.2.1 Design values and units relating to clause 8.5.2

Notation	Design value	Unit
c_2	wall thickness reduction due to chemical or mechanical wear	mm
d_1	outside diameter at large end of reducer acc. to Figure 8.5-5	mm
d_2	outside diameter at small end of reducer acc. to Figure 8.5-5	mm
d_a	pipe outside diameter	mm
$\hat{d}_a$	maximum outside diameter of cross-section	mm
$\breve{d}_a$	minimum outside diameter of cross-section	mm
d_{an}	nominal outside diameter of pipe	mm
d_i	nominal inside diameter of pipe	mm
d_{il}	inside diameter at location l	mm
d_{in}	nominal inside diameter of pipe	mm
d_{ir}	inside diameter at location r	mm
d_m	diameter acc. to cl. 8.5.2.8.3.4.4 (4)	mm
d_{Aa}	outside diameter of branch	mm
d_{Ai}	inside diameter of branch	mm
d_{Am}	mean diameter of branch	mm
d_{Ha}	outside diameter of run pipe	mm
d_{Hi}	inside diameter of run pipe	mm
d_{Hm}	mean diameter of run pipe	mm
d_{Ra}	outside diameter of branch pipe	mm
d_{Ri}	inside diameter of branch pipe	mm
d_{Rm}	mean diameter of branch pipe	mm
h	design value acc. to equations 8.5-34 and 8.5-37	—
i	stress indices	—
i_1, i_2, i_3, i_4	stress indices for pipe bends under internal pressure	—
i_{amy}, i_{amz}, i_{tby}, i_{tbz}, i_{tmz}	stress indices for pipe bends under moment loading	—
k	flexibility factor	—
k_N	flexibility factor for deformation due to normal force	—
k_Q	flexibility factor for deformation due to transverse force	—
k_T	flexibility factor for deformation due to torsional moment	—
k_x	flexibility factor for bending along axis x	—
k_z	flexibility factor for bending along axis z	—
I	second moment of area	mm^4
m,n	material parameters acc. to Table 7.8-2	—
p	maximum value of pressure in the load cycle under consideration	MPa
p_0	range of operating pressure fluctuations	MPa
r	radius	mm
r_1, r_2, r_3	transition radii in accordance with Figures 8.5-5 and 8.5-7	mm
r_m	mean radius	mm
$\hat{s}_a$	maximum wall thickness of transitional zone for wall thickness transitions	mm
s_1	wall thickness at large end of reducer	mm
s_2	wall thickness at small end of reducer	mm
s_{01}, s_{02}	minimum wall thickness for the straight pipe acc. to Figure 8.5-5	mm
s_l	wall thickness at location l	mm
s_r	wall thickness at location r	mm
s_m	wall thickness acc. to cl. 8.5.2.8.3.4.4(4)	mm

Notation	Design value	Unit
s_c	wall thickness without cladding acc. to clause 7.1.4 or measured wall thickness minus corrosion allowance and cladding; In the case of pipe bends with wall thickness increase at the intrados exceeding 15 %, credit shall be taken for the material notch by using the average value, and in the case of induction bends, the geometric average from the smallest and greatest wall thickness at the centre of bend as design wall thickness s_c. In the case of induction bends meeting the dimensional requirements of KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bends), the requirements of sub-clause 8.5.1 (6) shall be met.	mm
S_n	design value for wall thickness	mm
s_{Ac}	wall thickness of branch	mm
s_{Hc}	wall thickness of run pipe	mm
s_A^*	equivalent wall thickness for branch connection	mm
s_H^*	equivalent wall thickness for run connection	mm
x	general design value	—
x_1	design value acc. to equation (8.5-62)	—
x_2	design value acc. to equation (8.5-63)	—
x_3	design value acc. to equation (8.5-64)	—
x_4	design value acc. to equation (8.5-65)	—
x_K	design value acc. to equation (8.5-73)	—
y	general design value	—
B	primary stress index	—
B_1	primary stress index for internal pressure loading	—
B_2	primary stress index for moment loading	—
B_{2A}	primary stress index for branch due to moment loading	—
B_{2H}	primary stress index for run pipe due to moment loading	—
C	primary plus secondary stress index	—
C_1	primary plus secondary stress index due to internal pressure loading	—
C_2	primary plus secondary stress index due to moment loading	—
C_3	stress index for thermal loading	—
C_4	stress index for thermal loading	—
C_5	stress index acc. to equation (8.5-5)	—
C_{2A}	primary plus secondary stress index for branch due to moment loading	—
C_{2H}	primary plus secondary stress index for run pipe due to moment loading	—
C_x	stiffness regarding branch bending moment along axis x	N/mm^2
C_z	stiffness regarding branch bending moment along axis z	N/mm^2
D	allowable usage factor	—
E	modulus of elasticity	N/mm^2
E_k	modulus of elasticity for cold condition	N/mm^2
E_{rl}	average modulus of elasticity of the two sides r and l of a structural discontinuity or material discontinuity at room temperature	N/mm^2
E_w	modulus of elasticity for hot condition	N/mm^2

Notation	Design value	Unit
F_{1a}	factor for considering out-of-round cross-sections at stress index K_1	—
F_{1b}	factor for considering local out-of-roundness at stress index K_1	—
I_R	moment of inertia of branch pipe	mm^4
K	peak stress index	—
K_1	peak stress index for internal pressure loading	—
K_2	peak stress index for moment loading	—
K_3	peak stress index for thermal loading	—
K_e	plastification factor	—
K_{2A}	peak stress index for branch due to moment loading	—
K_{2H}	peak stress index for run pipe due to moment loading	—
L_1	length of cylindrical portion at the large end of a reducer	mm
L_2	length of cylindrical portion at the small end of a reducer	mm
L_m	length according to cl. 8.5.2.8.3.4.4 (4)	mm
M	material factor in equation (8.5-17)	—
M_1, M_2, M_3	range of moment loading components in directions 1, 2, 3 resulting from the load case combinations under consideration	Nmm
M_{1A}, M_{2A}, M_{3A}	moments on branch	Nmm
M_{1H}, M_{2H}, M_{3H}	moments on run pipe	Nmm
M_A	resulting moment on the branch	Nmm
M_H	resulting moment on the run pipe	Nmm
M_b	bending moment	Nmm
M_{iI}	resulting moment due to mechanical loads in equation (8.5-1)	Nmm
M_{iII}	maximum range of resulting moments in equation (8.5-2)	Nmm
M_{iIII}	maximum range of resulting moments in equation (8.5-3)	Nmm
M_{iIV}	maximum range of moments due to restrained thermal expansion and cyclic thermal anchor and intermediate anchor movements in equation (8.5-4)	Nmm
M_{iV}	maximum range of moments accounting for moments M_{iI} and M_{iII} without M_{iIV} in equation (8.5-6)	Nmm
M_t	torsional moment	Nmm
M_x	torsional moment acc. to Figure 8.5-8	Nmm
M_y	bending moment acc. to Figure 8.5-8	Nmm
M_z	bending moment acc. to Figure Bild 8.5-8	Nmm
$R_{p0.2T}$	0.2 % proof stress at temperature	N/mm^2
$R_{p0.2PT}$	0.2 % proof stress at test temperature	N/mm^2
S_a	one-half the allowable equivalent stress intensity range	N/mm^2
S_m	design stress intensity value acc. to Table 6.6-1	N/mm^2
T	temperature	K
$\breve{T}$	minimum temperature at the considered load cycle	K

Notation	Design value	Unit
$\hat{T}$	maximum temperature at the considered load cycle	K
$T_k(y)$	temperature at distance y from midthickness to point of time t = k	K
$T_j(y)$	temperature at distance y from midthickness to the point of time t = j	K
T_{mj}	average temperature through the wall thickness s_c to the point of time t = j	K
T_{mk}	average temperature through the wall thickness s_c to the point of time t = k	K
T_{mlj}, T_{mlk}	average wall temperature on the side l of a structural discontinuity or material discontinuity to the point of time t = j,k	K
T_{mrj}, T_{mrk}	average wall temperature on the side r of a structural discontinuity or material discontinuity to the point of time t = j,k	K
ΔT	range of temperature difference	K
ΔT_1	range of temperature difference between the temperature of the outside surface and the temperature on the inside surface of the piping product assuming moment generating linear temperature distribution	K
ΔT_{1k}, ΔT_{1j}	portion of T_1 at point of time t = j,k	K
ΔT_2	range of temperature difference for the portion of non-linear temperature distributions	K
ΔT_{2a}, ΔT_{2i}	range of non-linear portion of temperature distributions on the outside/inside	K
$\Delta T(y)$	temperature distribution range for location y	K
ΔT_a, ΔT_i	temperature distribution range on the outside/inside	K
ΔT_m	average temperature distribution range as difference between the average temperatures T_{mk} and T_{mi}	K
ΔT_{ml}	range of average temperature on side l of gross structural discontinuity or material discontinuity	K
ΔT_{mr}	range of average temperature on side r of gross structural discontinuity or material discontinuity	K
W	section modulus	mm^3
Z_A, Z_H	auxiliary values in equations (8.5-40) to (8.5-42)	mm^3
α	linear coefficient of thermal expansion at room temperature	1/K
α_r, α_l	linear coefficient of thermal expansion on side r, l of a structural discontinuity or material discontinuity at room temperature	1/K
δ	allowable average misalignment of butt welds acc. to Figure 8.5-3	mm
δ_1	offset at large end of a reducer	mm
δ_2	offset at small end of a reducer	mm
λ	auxiliary value acc. to equation (8.5-60)	—
ν	Poisson's ratio	—
σ	nominal stress due to loading	N/mm^2
σ_a	stress component in axial direction	N/mm^2
σ_e	ideally elastic stress, stress intensity or equivalent stress range due to loading	N/mm^2
σ_t	stress component in circumferential direction	N/mm^2
σ_r	stress component in radial direction	N/mm^2

Notation	Design value	Unit
σ_N	nominal stress	N/mm^2
$\sigma_N (M_b)$	nominal stress due to bending moment loading M_b	N/mm^2
σ_N (p)	nominal stress due to internal pressure loading p	N/mm^2
σ_I	primary stress intensity	N/mm^2
σ_{II}	equivalent stress range resulting from primary and secondary stresses	N/mm^2
σ_{III}	equivalent stress range resulting from primary and secondary stresses as well as peak stresses	N/mm^2
σ_{IV}	equivalent stress range resulting from secondary stress	N/mm^2
σ_V	equivalent stress range resulting from primary and secondary membrane and bending stresses	N/mm^2
σ_{VI}	equivalent stress range acc. to equation (8.5-7)	N/mm^2
τ	shear stress	N/mm^2
τ_{at}, τ_{ta}	shear stress components in circumferential and axial direction	N/mm^2
τ_N	nominal shear stress	N/mm^2
$\tau_N (M_t)$	nominal shear stress due to torsional moment loading	N/mm^2
φ	angle at circumference acc. to Figure 8.5-8	degree
ψ	auxiliary value acc. to equation (8.5-61)	—

8.5.2.2 General

(1) When applying the component-specific design method in accordance with this clause, clause 7.7.2.3 shall be taken into account with regard to the classification of stresses from restrained thermal expansions.

(2) Where the design stress intensity or allowable usage factor is exceeded when applying the component-specific method according to clause 8.5.2, it is additionally permitted to perform a detailed stress analysis in accordance with Section 7.7 or, if required, a fatigue analysis in accordance with Section 7.8.

Note:

The stress values σ_I to σ_{VI}, given in Section 8.5 as stress intensity or equivalent stress range do no exactly correspond to the respective definitions of clause 7.7.3, but are conservative evaluations of the respective stress intensity or equivalent stress range.

8.5.2.3 Design condition (Level 0)

Except for a single straight pipe, the following conditions apply to the determination and limitation of the primary stress intensity:

$$\sigma_I = B_1 \cdot \frac{d_a \cdot p}{2 \cdot s_c} + B_2 \cdot \frac{d_a}{2 \cdot I} \cdot M_{iI} \le 1.5 \cdot S_m \qquad (8.5\text{-}1)$$

where

σ_I	primary stress intensity	N/mm^2
B_1, B_2	stress indices, see clause 8.5.2.8	—
S_m	design stress intensity acc. to Table 6.6-1 at design temperature	N/mm^2
p	design pressure	MPa
d_a	pipe outside diameter where either $d_a = d_{an}$ or $d_a = d_{in} + 2\ s_c + 2\ c_2$ shall be taken (see Section 6.5)	mm

s_c	wall thickness without cladding acc. to clause 7.1.4 or measured wall thickness minus corrosion allowance and cladding; in the case of pipe bends with wall thickness increase at the intrados exceeding 15 %, credit shall be taken for the material notch by using the average value, and in the case of induction bends, the geometric average from the smallest and greatest wall thickness at the centre of bend as design wall thickness s_c. In the case of induction bends meeting the dimensional requirements of KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bends), the requirements of sub-clause 8.5.1 (6) shall be met.	mm
I	plane moment of inertia	mm^4
M_{iI}	resulting moment due to design mechanical loads; in the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment (moments resulting from different load cases that cannot occur simultaneously need not be used in calculating the resultant moment). If the method of analysis of dynamic loads is such that only magnitudes with relative algebraic signs are obtained, that combination of directional moment components shall be used leading to the greatest resultant moment.	Nmm

8.5.2.4 Level A and B

8.5.2.4.1 General

(1) For each load case, directional moment components shall be determined which always refer to a reference condition. The same applies to load cases under internal pressure and temperature differences.

(2) Where a verification of primary stresses according clause 3.3.3.3 is required for Level B, the primary stress intensity shall be determined according to equation (8.5-1) and be limited to the smaller value of $1.8 \cdot S_m$ and $1.5 \cdot R_{p0,2T}$ in which case p is the operating pressure of the respective load case. If the maximum internal pressure exceeds 1.1 times the design pressure, the primary stress intensity resulting from the circumferential stress due to internal pressure p shall be limited according to **Table 7.7-4** by means of the formulae of Annex A in due consideration of the pertinent design stress intensity to Level B.

8.5.2.4.2 Determination and limitation of the primary plus secondary stress intensity range

The application of the equations given in this clause results in the equivalent stress intensity range where the stresses are caused by operational transients occurring due to changes in mechanical or thermal loadings. Cold-spring, if any, need not be considered. The following condition shall normally be satisfied, otherwise clause 8.5.2.4.4 shall apply:

$$\sigma_{II} = C_1 \cdot \frac{d_a \cdot p_0}{2 \cdot s_c} + C_2 \cdot \frac{d_a}{2 \cdot I} \cdot M_{iII} + C_3 \cdot E_{rl} \cdot \left|\alpha_r \cdot \Delta T_{mr} - \alpha_l \cdot \Delta T_{ml}\right| \le 3 \cdot S_m \quad (8.5\text{-}2)$$

where

σ_{II}	primary plus secondary stress intensity range	N/mm^2
d_a, s_c	see clause 8.5.2.3	
I	second moment of area	mm^4
C_1, C_2, C_3	stress indices, see clause 8.5.2.8	—
p_0	range of operating pressure fluctuations	MPa
E_{rl}	average modulus of elasticity of the two sides r and l of a gross structural discontinuity or a material discontinuity at room temperature	N/mm^2
α_r, α_l	linear coefficient of thermal expansion on side r (l) of a gross structural discontinuity or a material discontinuity at room temperature	1/K
M_{iII}	resultant range of moments In the combination of moments from load sets, all directional moment components in the same direction shall be combined before determining the resultant moment. Here that combination of plant service conditions of Level A and B shall be selected resulting in the greatest values of M_{iII}. If a combination of loadings includes the effects of dynamic loads it shall be based on that range of the two following ranges of moments which results in higher values for M_{iII}: - the resultant range of moments due to the combination of all loads of two service conditions of Level A and B, where one-half range of the dynamic loads shall be considered - the resultant range of dynamic loads alone in which case credit shall be taken for portions of the moments resulting from restraints due to different movement of buildings which may impair the pipe run. Loadings resulting from thermal stratification shall also be considered. Weight effects need not be considered in equation (8.5-2) since they are non-cyclic in character.	Nmm
S_m	design stress intensity according to Table 6.6-1 at the temperature: $T = 0.25 \cdot \breve{T} + 0.75 \cdot \hat{T}$	N/mm^2
ΔT_{mr} (ΔT_{ml})	range of average temperature on side r (l) of gross structural discontinuity or material discontinuity (see clause 8.5.2.4.6).	

8.5.2.4.3 Determination of primary plus secondary plus peak stress intensity range

The stress intensity range σ_{III} resulting from primary plus secondary plus peak stresses shall be calculated according to equation (8.5-3) and is intended to determine the stress intensity range σ_{VI} according to equation (8.5-7). Credit shall also be taken in a suitable manner for loadings resulting from thermal stratification.

Note:

Reference literature [7] contains a proposal for considering thermal stratification.

$$\sigma_{III} = K_1 \cdot C_1 \cdot \frac{d_a \cdot p_0}{2 \cdot s_c} + K_2 \cdot C_2 \cdot \frac{d_a}{2 \cdot I} \cdot M_{iIII} + \frac{1}{2 \cdot (1-\nu)} \cdot K_3 \cdot E \cdot \alpha \cdot \left|\Delta T_1\right| + K_3 \cdot C_3 \cdot E_{rl} \cdot \left|\alpha_r \cdot \Delta T_{mr} - \alpha_l \cdot \Delta T_{ml}\right| + \frac{1}{1-\nu} \cdot E \cdot \alpha \cdot \left|\Delta T_2\right| \quad (8.5\text{-}3)$$

where

σ_{III}	stress intensity range resulting from primary plus secondary stresses and peak stresses	N/mm²
$d_a, S_c, l, p_0,$ $E_{rl}, \alpha_r(\alpha_l)$ $\Delta T_{mr}(\Delta T_{ml})$	see clause 8.5.2.4.2	
M_{iIII}=M_{iII}	see clause 8.5.2.4.2	
C_1, C_2, C_3 K_1, K_2, K_3	see clause 8.5.2.8	
ΔT_1, ΔT_2	see clause 8.5.2.4.6	
α	linear coefficient of thermal expansion at room temperature	1/K
E	modulus of elasticity at room temperature	N/mm²
ν	Poisson's ratio (= 0.3)	—

8.5.2.4.4 Simplified elastic-plastic analysis

8.5.2.4.4.1 Conditions

Where the limitation of the stress intensity range given in equation (8.5-2) cannot be satisfied for one or several pairs of load sets, the alternative conditions of a), b) and c) hereinafter shall be satisfied:

a) Limit of secondary stress intensity range:

$$\sigma_{IV} = C_2 \cdot \frac{d_a}{2 \cdot l} \cdot M_{iIV} \leq 3 \cdot S_m \qquad (8.5\text{-}4)$$

where

σ_{IV}	secondary stress intensity range	N/mm²
C_2, d_a. l	see clause 8.5.2.4.2	
M_{iIV}	greatest range of moments due to loadings resulting from restraint to thermal expansion and cyclic thermal anchor and intermediate anchor movement; credit shall also be taken for loadings resulting from thermal stratification	N/mm
S_m	design stress intensity according to Table 6.6-1 at the temperature: $T = 0.25 \cdot \breve{T} + 0.75 \cdot \hat{T}$	N/mm²

b) Limitation of thermal stress ratcheting

The temperature difference ΔT_1 according to clause 8.5.2.4.6 shall satisfy the following relation:

$$\Delta T_1 \leq \frac{y \cdot R_{p0.2T}}{0.7 \cdot E \cdot \alpha} \cdot C_5 \qquad (8.5\text{-}5)$$

Here, in dependence of

$$x = \frac{p \cdot d_a}{2 \cdot s_c \cdot R_{p0.2T}}$$

the following values for y apply:

x	y
0.3	3.33
0.5	2.0
0.7	1.2
0.8	0.8
Intermediate values shall be subject to straight interpolation.	

where

p	maximum pressure for the set of operating conditions under consideration	MPa
C_5	= 1.1 for ferritic steels, 1.3 for austenitic steels	—
α, E	as defined for equation (8.5-2)	
$R_{p0.2T}$	0.2 % proof stress at average fluid temperature of the transients under consideration	N/mm²

c) Limitation of stress intensity range resulting from primary plus secondary membrane and bending stresses:

The stress intensity range resulting from primary plus secondary membrane and bending stresses without stress components from moments due to restrained thermal expansion in the system shall be limited according to equation (8.5-6).

$$\sigma_v = C_1 \cdot \frac{p_0 \cdot d_a}{2 \cdot s_c} + C_2 \cdot \frac{d_a \cdot M_{iV}}{2 \cdot l} + C_4 \cdot E_{rl} \cdot$$

$$\left|\alpha_r \cdot \Delta T_{mr} - \alpha l \Delta T_{ml}\right| \leq 3 \cdot S_m \qquad (8.5\text{-}6)$$

where

σ_V	stress intensity range resulting from primary plus secondary membrane and bending stresses	N/mm²
C_1, C_2, C_4	see clause 8.5.2.8	
P_0, d_a $S_c, l,$ E_{rl}, α_r (α_l) ΔT_{mr} (ΔT_{ml})	see clause 8.5.2.4.2	
M_{iV}	Range of moments M_{iII} without M_{iIV} for the considered operating conditions; if M_{iII} was formed as the range of moments of the dynamic loads of one operating condition, half the range of the dynamic load portion of M_{iII} shall be taken to form M_{iV}	Nmm
S_m	design stress intensity according to Table 6.6-1 at the temperature: $T = 0.25 \cdot \breve{T} + 0.75 \cdot \hat{T}$	N/mm²

8.5.2.4.4.2 Stress intensity range σ_{VI}

With the primary plus secondary plus peak stress intensity range calculated according to equation (8.5-3) for all pairs of load sets an increased stress intensity range σ_{VI} compared to σ_{III} can be determined:

$$\sigma_{VI} = K_e \cdot \sigma_{III} \qquad (8.5\text{-}7)$$

where

σ_{VI}	equivalent stress intensity range	N/mm²
K_e	plastification factor	—

The magnitude of K_e depends on the value of the stress intensity range σ_{II} according to equation (8.5-2) and is obtained, e.g. by means of the following relationship:

a) $\sigma_{II} \leq 3 \cdot S_m$ $\qquad K_e = 1$

b) $3 \cdot S_m < \sigma_{II} < 3 \cdot m \cdot S_m$ $\qquad K_e = 1 + \left(\frac{(1-n)}{n \cdot (m-1)}\right) \cdot \left(\frac{\sigma_{II}}{3 \cdot S_m} - 1\right)$

c) $\sigma_{II} \geq 3 \cdot m \cdot S_m$ $\qquad K_e = \frac{1}{n}$

where the material parameters m and n can be used up to the temperature T (see **Table 7.8-2**).

K_e values not determined according to b) or c) shall be substantiated by way of calculation or by experimental analysis or be taken from literature. Its applicability shall be proved.

8.5.2.4.5 Fatigue analysis

8.5.2.4.5.1 Detailed determination of the calculative usage factor

The stress intensity ranges σ_{III} obtained from equation (8.5-3) or the stress intensity ranges σ_{VI} obtained from equation (8.5-7) shall be used for the determination of the usage factor according to Section 7.8, where S_a equals $\sigma_{III}/2$ or $\sigma_{VI}/2$ (S_a = one-half the stress intensity). For this purpose, the fatigue curves from **Figures 7.8-1** to **7.8-3** shall be used as basis.

8.5.2.4.5.2 Conservative determination of the usage factor

(1) Within the component-specific method for the determination and evaluation of stresses the fatigue analysis may be performed in accordance with the following procedure. This method shall be used for a conservative evaluation of a component. Where upon application of this method the allowable usage factor D is not exceeded, no detailed fatigue analysis need be performed.

(2) The stress intensity range $2 \cdot S_a = \sigma_{III}$ or σ_{VI} (see clause 8.5.2.4.3 or 8.5.2.4.4) shall be determined by means of equation (8.5-3) if the stress intensity defined hereinafter is used for the respective loadings:

a) As stress intensity range for internal pressure the respective greatest pressure differences of the load case combinations under consideration shall be taken.

b) As stress intensity range of the directional moment components M_{iIII} the greatest range of resulting moments of the load case combinations under consideration shall be taken.

Here, M_{iIII} shall be determined as follows:

$$M_{iIII} = \sqrt{M_1^2 + M_2^2 + M_3^2} \qquad (8.5\text{-}8)$$

$M_{1,2,3}$ range of moments of directions 1, 2, 3 from the load case combinations under consideration

c) As stress intensity range of the stresses resulting from temperature differences (ΔT_1, $\Delta T_{mr} - \Delta T_{ml}$, ΔT_2) the difference of the largest and smallest values (considering the relative algebraic signs) shall be taken for the load case combination under consideration. This also applies to stresses resulting from the absolute value of the difference of the products $\left|\alpha_r \cdot \Delta T_{mr} - \alpha_l \cdot \Delta T_{ml}\right|$,

d) As a conservative approach the number of all load cycles shall be accumulated (cumulative damage) to define the number of load cycles to be used. The allowable number of load cycles can be determined by means of **Figures 7.8-1** to **7.8-3**.

(3) The cumulative usage factor D is found to be the ratio of the actual number of cycles to the allowable number of cycles thus determined. Where the usage factor is less than 1, this location of the piping system need not be evaluated further.

Where a reduction of fatigue strength due to fluid effects cannot be excluded, then the following measures shall be taken at a threshold for cumulative damage of D = 0.4 to ensure consideration of fluid influence on the fatigue behaviour:

a) the components considered shall be included in a monitoring program to KTA 3211.4, or

b) experiments simulating operating conditions shall be performed, or

c) verifications by calculation shall be made in due consideration of fluid-effected reduction factors and realistic boundary conditions.

8.5.2.4.6 Determination of the ranges of temperature differences

(1) The determination of the ranges of temperature differences ΔT_m, ΔT_1 and ΔT_2 shall be based on the actual temperature distribution through the wall thickness s_c to the relevant points of time under consideration. They may be subject to time and location-dependent considerations.

(2) The range of temperature distribution $\Delta T(y)$ for location y is found to read:

$$\Delta T(y) = T_k(y) - T_j(y) \qquad (8.5\text{-}9)$$

with

y radial position in the wall, measured positive outward from the mid-thickness position

$-s_c/2 \le y \le s_c/2$

$T_j(y)$ temperature, as a function of radial position y from mid-thickness to point of time where t = j

$T_k(y)$ temperature, as a function of radial position y from mid-thickness to point of time where t = k

(3) The full temperature distribution range is composed of three parts as shown in **Figure 8.5-2**. Index a refers to the outside and index i to the inside.

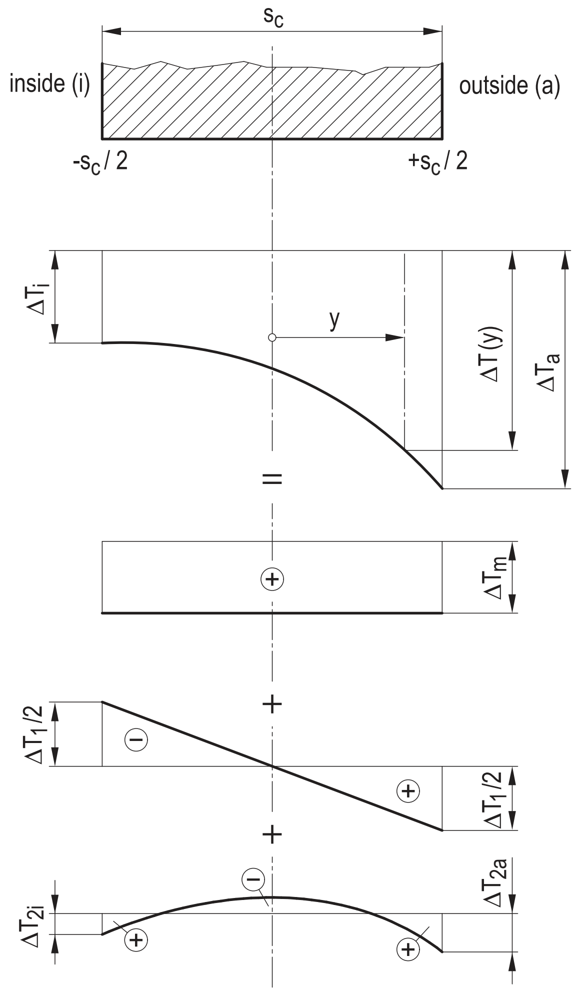


Figure 8.5-2: Decomposition of temperature distribution range

(4) For the determination of the pertinent stress ranges the following relationships apply:

a) Average range ΔT_m as temperature difference between the average temperatures T_{mk} and T_{mj}

$$\Delta T_m = \frac{1}{s_c} \int_{-s_c/2}^{s_c/2} \left[T_k(y) - T_j(y) \right] dy$$

$$= \frac{1}{s_c} \int_{-s_c/2}^{s_c/2} \Delta T(y) dy = T_{mk} - T_{mj} \tag{8.5-10}$$

with

T_{mj}, T_{mk} average value of temperature through wall thickness s_c at point of time where t = j, k

ΔT_m may be used to determine the range of moments M_i resulting from restraint to thermal expansion in the system.

The relationship (8.5-10) with the respective indices also applies to the ranges of average wall temperatures on sides r, l of a structural discontinuity or material discontinuity.

$$\left.\begin{aligned} \Delta T_{mr} &= T_{mrk} - T_{mrj} \\ \Delta T_{ml} &= T_{mlk} - T_{mlj} \end{aligned}\right\} \text{at point of time where t = j, k}$$

These magnitudes may be inserted in equations (8.5-2) and (8.5-3). For cylindrical shapes T_{mrj}, T_{mrk} at point of time where t = j, k shall normally be averaged over a length of $(d_{ir} \cdot s_r)^{1/2}$ and T_{mlj}, T_{mlk} over a length of $(d_{il} \cdot s_l)^{1/2}$.

Here, d_{ir} (d_{il}) is the inside diameter on side r(l) of a structural discontinuity or material discontinuity, and s_r (s_l) the average wall thickness on a length of $(d_{ir} \cdot s_r)^{1/2}$ or $(d_{il} \cdot s_l)^{1/2}$.

b) Range ΔT_1 of the temperature difference between the temperature on the outside surface and the temperature on the inside surface, assuming moment generating equivalent linear temperature distribution

$$\Delta T_1 = \frac{12}{s_c^2} \cdot \int_{-s_c/2}^{s_c/2} y \cdot \left[T_k(y) - T_j(y) \right] dy \tag{8.5-11}$$

c) Range ΔT_2 for that portion of the non-linear thermal gradient through the wall thickness

$$\Delta T_2 = \max . \begin{Bmatrix} \Delta T_{2a} = \left| \Delta T_a - \Delta T_m \right| - \dfrac{\left| \Delta T_1 \right|}{2} \\ \Delta T_{2i} = \left| \Delta T_i - \Delta T_m \right| - \dfrac{\left| \Delta T_1 \right|}{2} \\ 0 \end{Bmatrix} \tag{8.5-12}$$

8.5.2.5 Level P

(1) The test conditions for Level P loadings shall be evaluated in correspondence with the requirements of clause 3.3.3.6.

(2) The stresses shall be determined by means of equation (8.5-1) and limited to 1.35 · $R_{p0.2PT}$. Only if the load cycles exceed the number of ten, the stresses shall be determined by means of equation (8.5-3), and credit shall be taken of the pertinent load cycles as portion of the total accumulative damage of the material in the fatigue analysis.

8.5.2.6 Levels C and D service limits

(1) For the evaluation of Level C and D loadings the requirements of clauses 3.3.3.4 and 3.3.3.5 respectively apply.

(2) For Level C the primary stresses are calculated by means of equation (8.5-1), but are safeguarded with 2.25 · S_m, and shall not exceed 1.8 · $R_{p0.2T}$. Here, for p the respective pressure shall be taken. Where the maximum internal pressure exceeds 1.5 times the design pressure, the primary intensity stress, which is due to the circumferential stress caused by the internal pressure p, shall be limited in accordance with **Table 7.7-4** by means of the formulae of Annex A in due consideration of the pertinent design stress intensity to Level C.

(3) For Level D the primary stresses are calculated by means of (8.5-1), but are safeguarded with the smaller value of 3 · S_m and 2 · $R_{p0.2T}$. Here, for p the respective pressure shall be taken. Where the maximum internal pressure exceeds 2 times the design pressure, the primary stress intensity, which is due to the circumferential stress caused by the internal pressure, shall be limited in accordance with **Table 7.7-4** by means of the formulae of Annex A in due consideration of the pertinent design stress intensity to Level D.

8.5.2.7 Loading levels of special load cases

When performing strength calculations Section 3.1 shall be considered. The primary stresses according to equation (8.5-1) shall be limited such that the piping and components are not damaged.

8.5.2.8 Stress indices

8.5.2.8.1 General

(1) The applicable stress indices (B, C and K values) to be used in equations (8.5-1) to (8.5-4) and (8.5-6) of this Section are indicated in **Table 8.5-1**.

(2) Table 8.5-1 contains stress indices for some commonly used piping products and joints. Where specific data exist, lower stress indices than those given in Table 8.5-1 may be used.

(3) For piping products not covered by **Table 8.5-1** or for which the given requirements are not met, stress indices shall be established by experimental analysis or theoretical analysis.

(4) Stress indices may also be established by means of other rules, guidelines and standards.

8.5.2.8.2 Definition of stress indices

(1) The general definition of a stress index for mechanical load is

$$B, C, K = \frac{\sigma_e}{\sigma} \tag{8.5-13}$$

where

σ_e	ideally elastic stress, stress intensity, or stress intensity range due to mechanical load	N/mm²
σ	nominal stress due to mechanical loading	N/mm²

(2) The B values were derived from limit load calculations. For the C and K values σ_e is the maximum stress intensity or stress intensity range due to loading of the component. The nominal stress σ is shown in equations (8.5-1) to (8.5-4) and (8.5-6), respectively.

(3) The general term for a stress index due to thermal load is:

$$C, K = \frac{\sigma_e}{E \cdot \alpha \cdot \Delta T} \tag{8.5-14}$$

where

σ_e	highest stress intensity due to temperature gradient or temperature range ΔT	N/mm²
E	modulus of elasticity	N/mm²

α	linear coefficient of thermal expansion	1/K
ΔT	temperature gradient or temperature range	K

8.5.2.8.3 Conditions for using stress indices

8.5.2.8.3.1 General

(1) The stress indices given herein and in **Table 8.5-1** including the restrictions specified hereinafter shall be used with the conditions of clauses 8.5.2.2 to 8.5.2.7.

(2) For the calculation of the numerical values of the stress indices and the stresses in accordance with equations (8.5-1) to (8.5-7) the nominal dimensions shall be used in which case between outside and inside diameter the relationship

$$d_i = d_a - 2 \cdot s_c \qquad (8.5\text{-}15)$$

where

s_c	pipe wall thickness according to clause 8.5.2.3	mm

shall be taken into account.

For welded-in parts the nominal dimensions of the equivalent pipe shall be used.

(3) For pipe fittings such as reducers and tapered-wall transitions, the nominal dimensions of the large or small end, whichever gives the larger value of d_a/s_c shall normally be used.

(4) Loadings for which stress indices are given include internal pressure, bending and torsional moments, and temperature differences. The indices are intended to be sufficiently conservative to account also for the effects of transverse forces normally encountered in flexible piping systems. If, however, thrust or transverse forces account for a significant portion of the loading on a given piping component, the effect of these forces shall normally be included in the design analysis. The values of the forces and moments shall normally be obtained from an analysis of the piping system.

(5) The stress indices for welds are not applicable if the radial weld shrinkage exceeds $0.25 \cdot s_c$.

(6) The stress indices given in **Table 8.5-1** only apply to butt girth welds between two items for which the wall thickness is between $0.875 \cdot s_c$ and $1.1 \cdot s_c$ for an axial distance of $\sqrt{d_a \cdot s_c}$ from the welding ends.

(7) For components with longitudinal butt welds, the K_1, K_2 and K_3 indices shown shall be multiplied by 1.1 for flush welds or by 1.3 for as-welded welds. At the intersection of a longitudinal butt weld in straight pipe with a girth butt weld or girth fillet weld, the C_1, K_1, C_2, K_2 and K_3 indices shall be taken as the product of the respective indices.

(8) In general and unless otherwise specified, it is not required to take the product of stress indices for two piping components (e.g. a tee and a reducer, a tee and a girth butt weld) when welded together. The piping component and the weld shall be qualified separately.

(9) For curved pipe or butt welding elbows welded together or joined by a piece of straight pipe less than one pipe diameter long, the stress indices shall be taken as the product of the indices for the elbow or curved pipe and the indices for the girth butt weld, except for stress indices B_1 and C_4 which are exempted.

(10) The stress indices given in **Table 8.5-1** are applicable for components and welds with out-of-roundness not greater than $0.08 \cdot s_c$ where out-of-roundness is defined as $\hat{d}_a - \breve{d}_a$. For straight pipe, curved pipe, longitudinal butt welds in straight pipe, girth butt welds, and wall thickness transitions not meeting this requirement, the stress indices shall be modified as specified below:

a) If the cross-section is out-of-round but with no discontinuity in radius (e.g. an elliptical cross-section), an acceptable value of K_1 may be obtained by multiplying the tabulated values of K_1 with the factor F_{1a}:

$$F_{1a} = 1 + \frac{\hat{d}_a - \breve{d}_a}{s_c} \cdot \left[\frac{1{,}5}{1 + 0.455\,(d_a / s_c)^3\,(p/E)} \right] \qquad (8.5\text{-}16)$$

b) If there are discontinuities in radius, e.g. a flat spot, and if $\hat{d}_a - \breve{d}_a$ is not greater than $0.08 \cdot d_a$, an acceptable value of K_1 may be obtained by multiplying the tabulated values of K_1 with the factor F_{1b}:

$$F_{1b} = 1 + \frac{2\, s_c \cdot M \cdot \breve{R}_{p0.2T}}{d_a \cdot p} \qquad (8.5\text{-}17)$$

where

M=2 for ferritic steels and nonferrous metals except nickel based alloys

M=2.7 for austenitic steels and nickel based alloys

8.5.2.8.3.2 Connecting welds

(1) The stress indices given in **Table 8.5-1** are applicable for longitudinal butt joints in straight pipe, girth butt welds joining items with identical nominal wall thicknesses except as modified hereinafter.

(2) Connecting welds are termed to be either flush welds or as-welded ones, if the requirements in a) or b) are met, respectively.

a) Welds are considered to be flush welds if they meet the following requirements:

The total thickness (both inside and outside) of the reinforcement shall not exceed $0.1 \cdot s_c$.

There shall be no concavity on either the interior or exterior surfaces.

The finished contour shall not have any slope greater than 10 degree (see **Figure 8.5-3**).

b) Welds are considered to be as-welded if they do not meet the requirements for flush welds.

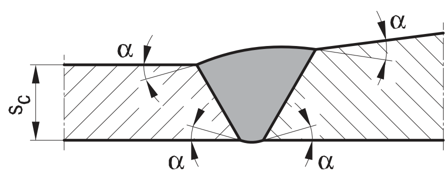


Figure 8.5-3: Allowable weld contour

(3) For as-welded welds joining items with nominal wall thicknesses less than 6 mm, the C_2 index shall be taken as:

$$C_2 = 1.0 + 3\,(\delta/s_c) \qquad (8.5\text{-}18)$$

but not greater than 2.1

where

δ	allowable average misalignment according to **Figure 8.5-4**. A smaller value than 0.8 mm may be used for δ if a smaller value is specified for fabrication. The measured misalignment may also be used. For flush welds $\delta = 0$ may be taken.	mm

Piping products and joints	Internal pressure			Moment loading			Thermal loading		
	B_1	C_1	K_1	B_2	C_2	K_2	C_3	K_3	C_4
Straight pipe, remote from welds or other discontinuities 1)	0.5	1.0	1.0	1.0	1.0	1.0	0.6	1.0	0.5
Butt girth welds between straight pipes or pipe and butt-welded item									
a) flush 1)	0.5	1.0	1.1	1.0	1.0	1.1	0.6	1.1	0.6
b) as-welded 2)	0.5	1.0	1.2	1.0	1.0 2)	1.8	0.6	1.7	0.6
Longitudinal butt welds in straight pipe 1) 2)									
a) flush	0.5	1.0	1.1	1.0	1.0	1.1	1.0	1.1	0.5
b) as-welded $s_c > 5$ mm	0.5	1.1	1.2	1.0	1.2	1.3	1.0	1.2	0.5
c) as-welded $s_c \leq 5$ mm	0.5	1.4	2.5	1.0	1.2	1.3	1.0	1.2	0.5
Transitions 1)									
a) flush or no circumferential weld closer than $(d_{Rm}/2 \cdot s_{Rc})^{1/2}$	0.5	3)	1.2	1.0	3)	1.1	3)	1.1	1.0
b) as-welded	0.5	3)	1.2	1.0	3)	1.8	3)	1.7	1.0
Butt welding reducers 1) to **Figure 8.5-5**	1.0 4)	4)	4)	1.0	4)	4)	1.0	1.0	0.5
Curved pipe or elbows 1)	5)	5)	1.0	5)	5)	1.0	1.0	1.0	0.5
Branch connections 1) 6) to Annex A 2.8	0.5	7)	2.0	7)	7)	7)	1.8	1.7	1.0
Butt welding tees 1) 6) to Annex A 5.2.4	0.5	1.5	4.0	8)	8)	8)	1.0	1.0	0,5

Stress indices shall only be used if the dimensioning requirements of Annex A have been met.
In addition, B values can only be used if $d_a/s_c \leq 50$, C and K values only if $d_a/s_c \leq 100$. For $50 < d_a/s_c \leq 100$ the B_1 values remain valid, the B_2 values shall be multiplied with the factor $1/(X \cdot Y)$ where

$X = 1.3 - 0.006 \cdot (d_a/s_c)$ and

$Y = 1.0224 - 0.000594 \cdot T$ with $Y \leq 1.0$ for ferritic material and $Y = 1.0$ for other materials.

T : design temperature in °C

1) see clause 8.5.2.8.3.1
2) see clause 8.5.2.8.3.2
3) see clause 8.5.2.8.3.3
4) see clause 8.5.2.8.3.4
5) see clause 8.5.2.8.3.5
6) see clause 8.5.2.8.3.6
7) see clause 8.5.2.8.3.6.2
8) see clause 8.5.2.8.3.6.3

Table 8.5-1: Stress indices for use with equations (8.5-1) to (8.5-4) and (8.5-6)

8.5.2.8.3.3 Welded transitions

(1) The stress indices given in **Table 8.4-1** are applicable to butt girth welds between a pipe for which the wall thickness is between $0.875 \cdot s_c$ and $1.1 \cdot s_c$ for an axial distance of $\sqrt{d_a \cdot s_c}$ from the welding end and the transition to a cylindrical component (pipe, attached nozzle, flange) with a greater thickness and a greater or an equal outside diameter and a smaller or an equal inside diameter.

(2) For transitions which on an axial distance of at least $1.5 \cdot s_c$ from the welding end have a taper not exceeding 30 degrees, and on an axial distance of at least $0.5 \cdot s_c$ have a taper not exceeding 45 degrees, and on the inside on an axial distance of $2 \cdot s_c$ from the welding end have a slope not greater than 1:3, the following applies for indices C_1, C_2, C_3:

$$C_1 = 0.5 + 0.33\,(d_a/s_c)^{0.3} + 1.5 \cdot (\delta/s_c) \qquad (8.5\text{-}19)$$

but not greater than 1.8

$$C_2 = 1.7 + 3.0 \cdot (\delta/s_c) \qquad (8.5\text{-}20)$$

but not greater than 2.1

$$C_3 = 1.0 + 0.03 \cdot (d_a/s_c) \qquad (8.5\text{-}21)$$

but not greater than 2.0.

(3) For transitions which on the outside, inside or on both sides, on an axial distance of $\sqrt{d_a \cdot s_c}$ from the welding end, have a slope not greater than 1:3, the following applies for indices C_1, C_2, C_3:

$$C_1 = 1.0 + 1.5 \cdot (\delta/s_c) \qquad (8.5\text{-}22)$$

but not greater than 1.8

$$C_2 = \hat{s}/s_c + 3 \cdot (\delta/s_c) \qquad (8.5\text{-}23)$$

but not greater than the smaller value

$[1.33 + 0.04\,\sqrt{d_a/s_c} + 3\,(\delta/s_c)]$ and 2.1

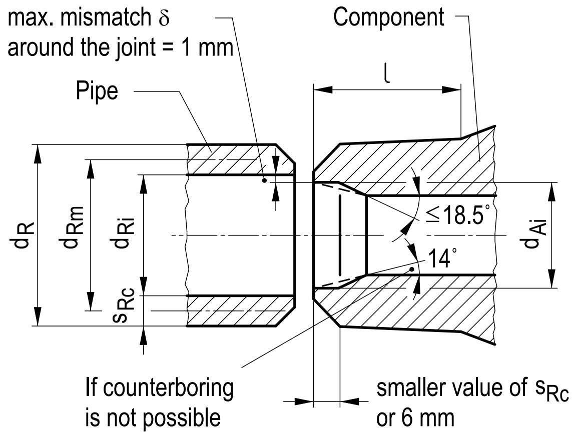
$$C_3 = 0.35\,(\hat{s}/s_c) + 0{,}25 \qquad (8.5\text{-}24)$$

but not greater than 2.0.

(4) For the transitions according to this Section δ shall be selected in accordance with **Figure 8.5-4**. For flush welds and as-welded welds between components with wall thicknesses s_c greater than 6 mm $\delta = 0$ may be taken.

(5) $\hat{s}$ is the maximum wall thickness within the transitional zone. If $\hat{s}/s_c$ does not exceed 1.1, the indices for circumferential welds may be used.

a) Concentric centre lines



b) offset centre lines

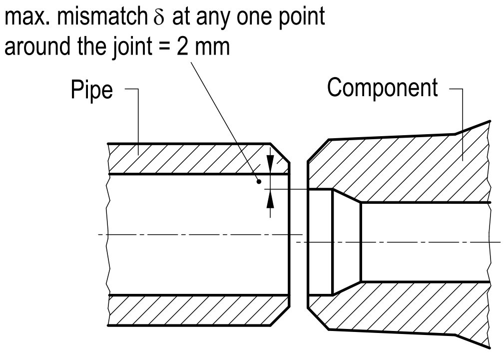


Figure 8.5-4: Butt weld alignment and mismatch tolerances for unequal inside diameter and outside diameter when fairing or back welding on the inside is not possible

8.5.2.8.3.4 Reducers

8.5.2.8.3.4.1 General

The stress indices given in **Table 8.5-1** are applicable for concentric reducers if the following restrictions are considered:

a) α does not exceed 60° (cone angle)

b) the wall thickness is not less than s_{01} throughout the body of the reducer, except in and immediately adjacent to the cylindrical portion on the small end where the thickness shall not be less than s_{02}. The wall thicknesses s_{01} and s_{02} are the minimum wall thicknesses for the straight pipe at the large end and small end, respectively.

8.5.2.8.3.4.2 Primary stress indices

$B_1 = 0.5$ for $\alpha \leq 30°$

$B_1 = 1$ for $30° < \alpha \leq 60°$

8.5.2.8.3.4.3 Primary plus secondary stress indices

(1) For reducers with r_1 and r_2 equal to or greater than $0.1 \cdot d_1$:

$$C_1 = 1.0 + 0.0058 \cdot \alpha \cdot \sqrt{d_n / s_n} \qquad (8.5\text{-}25)$$

$$C_2 = 1.0 + 0.36 \cdot \alpha^{0.4} \cdot (d_n/s_n)^{0.4\,(d_2/d_1\,-\,0.5)} \qquad (8.5\text{-}26)$$

(2) For reducers with r_1 or r_2 smaller than $0.1 \cdot d_1$:

$$C_1 = 1.0 + 0.00465 \cdot \alpha^{1.285} \cdot (d_n/s_n)^{0.39} \qquad (8.5\text{-}27)$$

$$C_2 = 1.0 + 0.0185 \cdot \alpha \cdot \sqrt{d_n / s_n} \qquad (8.5\text{-}28)$$

(3) Here d_n/s_n is the larger value of d_1/s_1 and d_2/s_2 and α is the cone angle according to **Figure 8.5-5**.

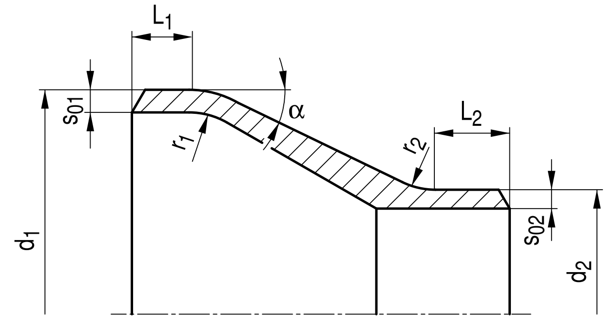


Figure 8.5-5: Concentric reducer

8.5.2.8.3.4.4 Peak stress indices

(1) The K_1 and K_2 indices given hereinafter shall normally be used depending on the type of connecting weld, extent of mismatch and thickness dimensions.

(2) For reducers connected to pipe with flush girth welds (see clause 8.5.2.8.3.2):

$$K_1 = 1.1 - 0.1 \cdot L_m / \sqrt{d_m \cdot s_m} \qquad (8.5\text{-}29)$$

but at least 1.0

$K_2 = K_1$

(3) For reducers connected to pipe with as-welded girth butt welds (see clause 8.5.2.8.3.2), where s_1 or s_2 exceeds 5 mm and δ_1/s_1 or δ_2/s_2 does not exceed 0.1:

$$K_1 = 1.2 - 0.2 \cdot L_m / \sqrt{d_m \cdot s_m} \qquad (8.5\text{-}30)$$

but at least 1.0

$$K_2 = 1.8 - 0.8 \cdot L_m / \sqrt{d_m \cdot s_m} \qquad (8.5\text{-}31)$$

but at least 1.0.

(4) For reducers connected to pipe with as-welded girth butt welds (see clause 8.5.2.8.3.2) where s_1 or s_2 does not exceed 5 mm or δ_1/s_1 or δ_2/s_2 is greater than 0.1:

$$K_1 = 1.2 - 0.2 \cdot L_m / \sqrt{d_m \bullet s_m} \qquad (8.5\text{-}32)$$

but at least 1.0

$$K_2 = 2.5 - 1.5 \cdot L_m / \sqrt{d_m \bullet s_m} \qquad (8.5\text{-}33)$$

but at least 1.0.

$L_m / \sqrt{d_m \cdot s_m}$ is the smaller value of $L_1 / \sqrt{d_1 \cdot s_1}$ or $L_2 / \sqrt{d_2 \cdot s_2}$.

δ_1, δ_2 is the offset at the large end or small end of the reducer (see clause 8.5.2.8.3.2 and **Figure 8.5-4**).

8.5.2.8.3.5 Butt welding elbows and curved pipes

The stress indices given in **Table 8.5-1**, except as added to and modified herein, are applicable to butt welding elbows or curved pipe:

a) Primary stress index

$$B_1 = -0.1 + 0.4 \cdot h \qquad (8.4\text{-}34)$$

but neither less than zero, nor greater than 0.5

$$B_2 = 1.3/h^{2/3} \text{ but at least } 1.0 \qquad (8.5\text{-}35)$$

b) Primary plus secondary stress indices

$$C_1 = \frac{(2 \cdot R - r_m)}{2 \cdot (R - r_m)} \qquad (8.5\text{-}36)$$

$$C_2 = \frac{1.95}{h^{2/3}} \text{ but at least } 1.5 \qquad (8.5\text{-}37)$$

where

$r_m = d_m/2$

$d_m = d_a - s_c$

R = bending radius

$$h = \frac{4 \cdot s_c \cdot R}{d_m^2}$$

8.5.2.8.3.6 Branch connections and butt welding tees

8.5.2.8.3.6.1 General

(1) When determining the stress intensities in accordance with equations (8.5-1) to (8.5-4) and (8.5-6), the following conditions shall be satisfied for branch connections.

(2) The moments are to be calculated at the intersection of the run and branch centre lines

for M_A:

$$M_A = \left(M_{x3}^2 + M_{y3}^2 + M_{z3}^2\right)^{1/2} = \text{resulting moment on branch} \qquad (8.5\text{-}38)$$

for M_H:

$$M_H = \left(M_x^2 + M_y^2 + M_z^2\right)^{1/2} = \text{resulting moment on run} \qquad (8.5\text{-}39)$$

where M_x, M_y and M_z are calculated as follows:

Where the directional moment components of the run M_x, M_y or M_z have the same algebraic signs at intersections 1 and 2 as the moment of the branch as per **Figure 8.5-6** which are in the same direction, then the respective components shall be used to determine the resultant moment loading M_H according to equation (8.5-39) which then equals zero. Otherwise the smaller of the absolute values at the intersections 1 and 2 shall be used to determine M_H.

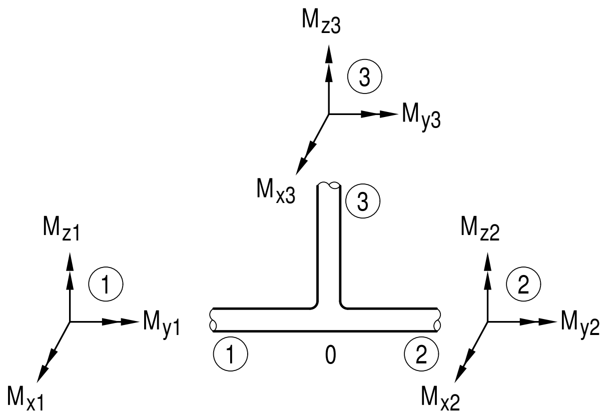


Figure 8.5-6: Designation of moments on branch connection

(3) For branches the M_i terms shall be replaced by the following pairs of terms in equations (8.5-1), (8.5-2), (8.5-3), (8.5-4), and (8.5-6):

a) in equation (8.5-1):

$$B_{2A} \cdot \frac{M_A}{Z_A} + B_{2H} \cdot \frac{M_H}{Z_H} \qquad (8.5\text{-}40)$$

b) in equation (8.5-2), (8.5-4) and (8.5-6):

$$C_{2A} \cdot \frac{M_A}{Z_A} + C_{2H} \cdot \frac{M_H}{Z_H} \qquad (8.5\text{-}41)$$

c) in equation (8.5-3):

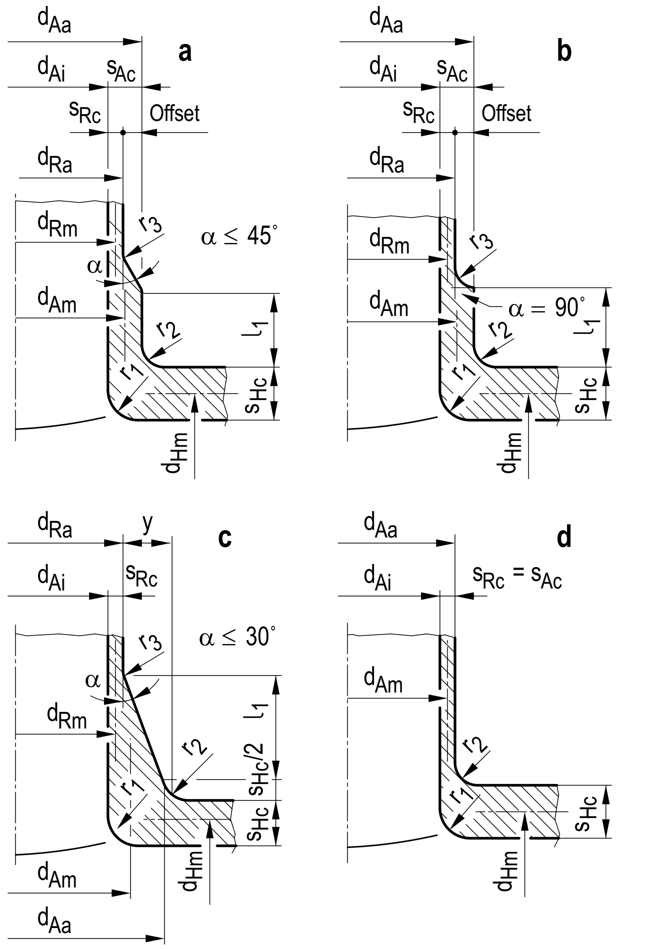
$$C_{2A} \cdot K_{2A} \cdot \frac{M_A}{Z_A} + C_{2H} \cdot K_{2H} \cdot \frac{M_H}{Z_H} \qquad (8.5\text{-}42)$$

where

$$Z_A = \frac{\pi}{4} \cdot d_{Rm}^2 \cdot s_{Rc}$$

$$Z_H = \frac{\pi}{4} \cdot d_{Hm}^2 \cdot s_{Hc}$$

(4) For branches according to Annex A 2.8: d_{Rm}, s_{Rc}, d_{Hm} and s_{Hc} are given in **Figure 8.5-7**.



$s_{Ac} = s_{Rc} + 0.667 \cdot y$

If $l_1 \geq 0.5 \sqrt{\frac{d_{Ai}}{2} \cdot s_{Ac}}$, then d_{Rm} can be taken as the radius to the centre of s_{Ac}.

Figure 8.5-7: Branch connection nomenclature

8.5.2.8.3.6.2 Stress indices for branches complying with Annex A 5.2.5

(1) Applicability of indices

The stress indices indicated are applicable for branch connections if the following conditions a) to h) are satisfied:

a) The branch-to-run radius ratio is

$d_{Am}/d_{Hm} \leq 0.5$

b) The run pipe radius-to-thickness ratio is limited as follows:

$d_{Hm}/s_{Hc} \leq 50$

c) The axis of the branch connection is normal to the run pipe surface.

d) The requirements for reinforcement of areas according to Section A 5.2.5 have been met.

e) The inside corner radius r_1 (see **Figure 8.5-7**) shall be between 0.1 and $0.5 \cdot s_{Hc}$.

f) The branch-to-run fillet radius r_2 (see **Figure 8.5-7**) is not less than the larger of $s_{Ac}/2$ or $(s_{Ac} + y)/2$ (see **Figure 8.5-6 c**) and $s_{Hc}/2$.

g) The branch-to-fillet radius r_3 (see **Figure 8.5-7**) is not less than the larger of $0.002 \cdot \alpha \cdot d_{Aa}$ or $2 \cdot (\sin \alpha)^3$ times the offset as shown in **Figures 8.5-7 a** and **8.5-7 b**.

h) For several branch connections in a pipe, the arc distance measured between the centres of adjacent branches along the outside surface of the run pipe is not less than 1.5 times the sum of the two adjacent branch inside radii in the longitudinal direction, or is not less than the sum of the two adjacent branch radii along the circumference of the run pipe.

(2) Primary stress indices

$$B_{2A} = 0.5 \cdot C_{2A} \geq 1.0 \qquad (8.5\text{-}43)$$

$$B_{2H} = 0.75 \cdot C_{2H} \geq 1.0 \qquad (8.5\text{-}44)$$

(3) Primary plus secondary stress indices

The C_1, C_{2A} and C_{2H} indices can be determined using the following relationships:

$$C_1 = 1.4 \left(\frac{d_{Hm}}{s_{Hc}}\right)^{0.182} \cdot \left(\frac{d_{Rm}}{d_{Hm}}\right)^{0.367} \cdot \left(\frac{s_{Hc}}{s_{Rc}}\right)^{0.382} \cdot \left(\frac{s_{Rc}}{r_2}\right)^{0.148} \qquad (8.5\text{-}45)$$

but at least 1.2.

If r_2/s_{Rc} exceeds 12, use $r_2/s_{Rc} = 12$ for computing C_1.

$$C_{2A} = 3 \left(\frac{d_{Hm}}{2s_{Hc}}\right)^{2/3} \cdot \left(\frac{d_{Rm}}{d_{Hm}}\right)^{1/2} \cdot \left(\frac{s_{Rc}}{s_{Hc}}\right) \cdot \left(\frac{d_{Rm}}{d_{Aa}}\right) \qquad (8.5\text{-}46)$$

but at least 1.5.

$$C_{2H} = 1.15 \left[\left(\frac{d_{Hm}}{2s_{Hc}}\right) \cdot \left(\frac{d_{Rm}}{d_{Hm}}\right) \cdot \left(\frac{s_{Hc}}{s_{Rc}}\right)\right]^{1/4} \qquad (8.5\text{-}47)$$

but at least 1.5.

(4) Peak stress indices

The peak stress indices K_{2A} and K_{2H} for moment loadings may be taken as:

$K_{2A} = 1.0$

$K_{2H} = 1.75$

and $K_{2H} \cdot C_{2H}$ normally shall not be smaller than 2.65.

8.5.2.8.3.6.3 Stress indices for butt welding tees

(1) The stress indices given in **Table 8.5-1** as well as the indices given hereinafter are applicable to butt welding tees if they meet the requirements of clause A 5.2.4.1 or A 5.2.4.2.

(2) To determine the stresses resulting from internal pressure and moments as well as the stress indices the diameters (d_{Ha}, d_{Aa}) and the equivalent wall thicknesses (s_H^+, s_A^+) of the run and branch to be connected shall be used in compliance with clause A 5.2.4.1.5 or A 5.2.4.2.4.

(3) Primary stress indices

The primary stress indices B_{2A} and B_{2H} may be taken as:

$$B_{2A} = 0.4 \cdot \left(\frac{d_{Ha}}{2 \cdot s_H^+}\right)^{2/3} \qquad (8.5\text{-}48)$$

but at least 1.0

$$B_{2H} = 0.5 \cdot \left(\frac{d_{Ha}}{2 \cdot s_H^+}\right)^{2/3} \qquad (8.5\text{-}49)$$

but at least 1.0.

(4) Primary plus secondary stress indices

The C_{2A} and C_{2H} indices for moment loadings shall be taken as follows:

$$C_{2A} = 0.67 \cdot \left(\frac{d_{Ha}}{2 \cdot s_H^+}\right)^{2/3} \qquad (8.5\text{-}50)$$

but at least 2.0

$$C_{2H} = C_{2A} \qquad (8.5\text{-}51)$$

(5) Peak stress indices

The peak stress indices K_{2A} and K_{2H} shall be taken as:

$$K_{2A} = K_{2H} = 1 \qquad (8.5\text{-}52)$$

8.5.2.9 Detailed stress analysis

8.5.2.9.1 General

(1) In lieu of the stress analysis according to clauses 8.5.2.3 to 8.5.2.6 a detailed stress analysis in accordance with this clause may be made.

(2) To determine a normal stress σ the following relation with σ_N as nominal stress and i as stress index applies:

$\sigma = i \cdot \sigma_N$

Accordingly the following applies to shear stresses:

$\tau = i \cdot \tau_N$

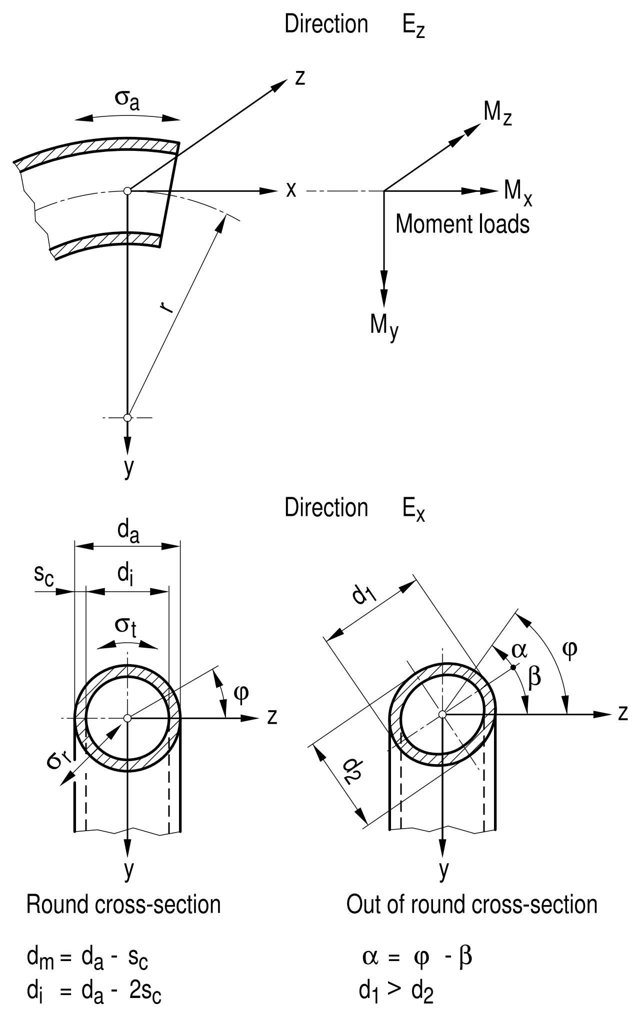


Figure 8.5-8: Pipe elbow nomenclature for detailed stress analysis

(3) The following definitions apply to the nominal stresses in this clause:

for loading due to internal pressure p

$\sigma_N(p) = p \cdot d_i / (2 \cdot s_c)$ (8.5-53)

for loading due to bending moment M_b

$\sigma_N(M_b) = M_b/W$ (8.5-54)

for loading due to torsional moment M_t

$\tau_N(M_t) = M_t/(2 \cdot W)$ (8.5-55)

(4) For the stress components on the pipe section the following definitions apply in compliance with clause 8.2.2 and **Figure 8.5-8**:

σ_a = stress component in axial direction (in the plane of the section under consideration and parallel to the boundary of the section)

σ_t = stress component in circumferential direction (normal to the plane of the section)

σ_r = stress component in radial direction (normal to the boundary of the section)

$\tau_{at} = \tau_{ta}$ = shear stress components in circumferential and axial direction

(5) With these stress components the stress intensities for the investigation points shall be determined and be limited in accordance with **Table 7.7-4**.

8.5.2.9.2 Welding elbows and curved pipes

(1) The stress indices given in **Tables 8.5-2** and **8.5-3** are applicable to elbows and curved pipes provided that the points under investigation are sufficiently remote from girth or longitudinal welds or other local discontinuities. Otherwise, additional theoretical or experimental analyses are required. The applicability of the stress indices for bends with notches (wall thickness increase at intrados) exceeding 15 %, referred to the nominal wall thickness, shall be verified in each individual case.

(2) The nomenclature used for the stress indices can be taken from **Figure 8.5-8** where the directional moment components are defined as follows:

M_x : torsional moment

M_y : bending moment for out-of-plane E_z displacement

M_z : bending moment for in-plane E_y displacement.

(3) The stress indices of **Table 8.5-2** for internal pressure loading have the following magnitudes:

$$i_1 = \frac{r + 0.25 \cdot d_i \cdot \sin\varphi}{r + 0.5 \cdot d_m \cdot \sin\varphi} \qquad (8.5\text{-}56)$$

$$i_2 = 0.5 \cdot d_i/d_m \qquad (8.5\text{-}57)$$

$$i_3 = \frac{d_1 - d_2}{s_c} \cdot \frac{1.5}{1 + 0.5 \cdot (1-\nu^2) \cdot (d_m/s_c)^3 \cdot p/E} \cdot \cos 2\alpha \qquad (8.5\text{-}58)$$

$$i_4 = \frac{2 \cdot s_c}{d_i} \qquad (8.5\text{-}59)$$

(4) The stress indices of **Table 8.4-3** for moment loading, with

$$\lambda = 4 \cdot r \cdot s_c / \left(d_m^2 \cdot \sqrt{1-\nu^2}\right) \qquad (8.5\text{-}60)$$

$$\psi = 2 \cdot p \cdot r^2 / (E \cdot d_m \cdot s_c) \qquad (8.5\text{-}61)$$

$$x_1 = 5 + 6 \cdot \lambda^2 + 24 \cdot \psi \qquad (8.5\text{-}62)$$

$$x_2 = 17 + 600 \cdot \lambda^2 + 480 \cdot \psi \qquad (8.5\text{-}63)$$

$$x_3 = x_1 \cdot x_2 - 6.25 \qquad (8.5\text{-}64)$$

$$x_4 = (1-\nu^2) \cdot (x_3 - 4.5 \cdot x_2) \qquad (8.5\text{-}65)$$

have the following magnitudes and only apply if $\lambda \geq 0.2$.

In the equation for ψ not more than the respective value of the internal pressure p shall be inserted.

The following applies to the bending moment M_y:

$$i_{amy} = \cos\varphi + [(1.5 \cdot x_2 - 18.75) \cdot \cos 3\varphi + 11.25 \cdot \cos 5\varphi]/x_4 \qquad (8.4\text{-}66)$$

$$i_{tby} = -\lambda \cdot (9 \cdot x_2 \cdot \sin 2\varphi + 225 \cdot \sin 4\varphi)/x_4 \qquad (8.4\text{-}67)$$

For the bending moment M_z the following applies:

$$i_{amz} = \sin\varphi + [(1.5 \cdot x_2 - 18.75) \cdot \sin 3\varphi + 11.25 \cdot \sin 5\varphi]/x_4 \qquad (8.4\text{-}68)$$

$$i_{tbz} = \lambda \cdot (9 \cdot x_2 \cdot \cos 2\varphi + 225 \cdot \cos 4\varphi)/x_4 \qquad (8.4\text{-}69)$$

$$i_{tmz} = -0.5 \cdot (d_m/r) \cdot \cos\varphi \cdot \{\cos\varphi + [(0.5 \cdot x_2 - 6.25) \cdot \cos 3\varphi + 2.25 \cdot \cos 5\varphi]/x_4\} \qquad (8.4\text{-}70)$$

Location	Surface	Stress direction	Stress index
Round cross-section			
φ	outside	σ_t	$i_1 - 0.5 \cdot i_4$
	mid		i_1
	inside		$i_1 + 0.5 \cdot i_4$
Any	outside	σ_a	i_2
	mid		i_2
	inside		i_2
Out-of-round cross-section			
φ	outside	σ_t	$i_1 - i_3 - 0.5 \cdot i_4$
	mid		i_1
	inside		$i_1 + i_3 + 0.5 \cdot i_4$
	outside	σ_a	i_2 - 0,3 $\cdot i_3$
	mid		i_2
	inside		$i_2 + 0.3 \cdot i_3$
Round and out-of-round cross-section			
Any	outside	σ_r	0
	mid		$-0.5 \cdot i_4$
	inside		$-i_4$

Table 8.5-2: Stress indices for curved pipe or welding elbows under internal pressure

Location	Surface	Stress direction	Stress index
for torsional moment M_x			
Any	outside	τ_{at}	1
	mid		1
	inside		1
for bending moments M_y			
φ	outside	σ_t	i_{tby}
	mid		0
	inside		$-i_{tby}$
	outside	σ_a	$i_{amy} + \nu \cdot i_{tby}$
	mid		i_{amy}
	inside		$i_{amy} - \nu \cdot i_{tby}$
for bending moments M_z			
φ	outside	σ_t	$i_{tmz} + i_{tbz}$
	mid		i_{tmz}
	inside		$i_{tmz} - i_{tbz}$
	outside	σ_a	$i_{amz} + \nu \cdot i_{tbz}$
	mid		i_{amz}
	inside		$i_{amz} - \nu \cdot i_{tbz}$

Table 8.5-3: Stress indices for curved pipe or welding elbows under moment loading

(5) **Table 8.5-4** applies to the classification as per clause 7.7.2 into stress categories of the stresses determined by the stress indices given here.

Origin of stress	Type of stress 1)	Classification
Internal pressure	Membrane stresses	P_m
	Bending stresses	Q
Moments due to external loads	Membrane and torsional stresses	P_l
	75 % of bending stresses	P_b
	25 % of bending stresses	Q
Moments due to restrained thermal expansion and free end displacements	Membrane, bending and torsional stresses	Q
1) Referred to through wall stresses		

Table 8.5-4: Classification of stresses for curved pipe or elbows in case of detailed stress analysis

8.5.2.9.3 Branches complying with Annex A 5.2.5

For branches complying with Annex A 5.2.5 the stresses due to internal pressure may be determined according to clause 8.2.2.3 and the stresses due to forces and moments according to clause 8.2.2.4 if the pertinent geometric conditions are satisfied.

8.5.2.10 Flexibility factors and stress intensification factors

8.5.2.10.1 General

(1) Compared to straight pipes individual piping components show an increased flexibility when subjected to bending on account of the ovalization of the pipe cross-section causing an increase of stresses.

(2) Where the system analysis for the piping is made to conform to the theory of beams (straight beam with circular cross-section), this increased flexibility shall be taken into account by k values not less than 1 for flexibility factors and C not less than 1 for stress intensification factors.

(3) Compared to the straight pipe, torsional moments as well as normal and transverse forces do neither lead to an increased flexibility nor to an increase of stresses.

8.5.2.10.2 Straight pipes

(1) For the determination of the deflection of straight pipes by bending and torsional moments as well as normal and transverse forces the beam theory applies.

(2) For the analysis of straight pipes all flexibility factors shall be taken as k = 1 and the stress intensification factors as C = 1.

8.5.2.10.3 Pipe elbows and curved pipes

(1) For the curved section of elbows and curved pipes the deflections which according to the theory of beams result from bending moments (M_y and M_z according to **Figure 8.5-9**), shall be multiplied with the flexibility factors k_y or k_z in which case the system analysis can either be made with average values or values for the point under investigation to obtain the flexibility factors.

(2) For the determination of deformations due to torsional moments as well as normal and transverse forces the conventional theory of beams applies.

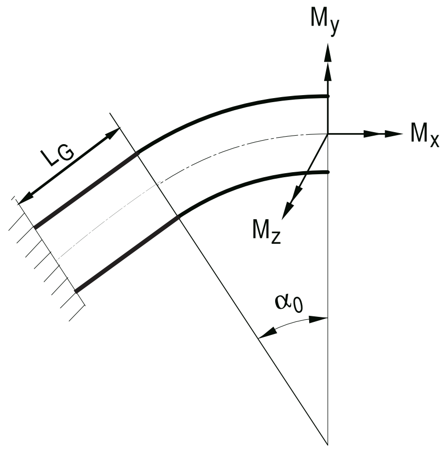


Figure 8.5-9: Direction of moments

(3) The value given hereinafter for the mean flexibility factor $k_m = k_y = k_z$ not less than 1.0 applies if the following conditions for pipe elbows and curved pipes are satisfied:

a) r/d_m not less than 0.85

b) arc length not less than d_m

c) neither at commencement nor end of curvature there are no flanges or similar stiffeners within a distance L_G not exceeding 2 x d_m.

$$k_m = k_p \cdot \frac{1.65}{h} \text{ ; but at least} \geq 1 \tag{8.5-71}$$

with
$$k_p = \frac{1}{1 + \dfrac{p \cdot d_m \cdot X_k}{2 \cdot E \cdot s_c}} \tag{8.5-72}$$

$$X_k = 6 \cdot \left(\frac{d_m}{2 \cdot s_c}\right)^{4/3} \cdot \left(\frac{2 \cdot r}{d_m}\right)^{1/3} \tag{8.5-73}$$

$$h = \frac{4 \cdot r \cdot s_c}{{d_m}^2} \tag{8.5-74}$$

(4) Where flanges or similar stiffeners are located at a distance L_G less than or equal to $d_m/2$ from the commencement or end of curvature, for such bends and bent pipes $k_m = k_y = k_z = 1.0$ or k´ as per footnote 5 of **Table 8.5-5** shall be used.

(5) Where flanges or similar stiffeners are located at a distance L_G less than or equal to 2 x d_m from the commencement or end of curvature, for such bends and bent pipes linear interpolation shall be made between $k_m = k_y = k_z = 1.0$ or k´ as per footnote 5 of **Table 8.5-5** and the result of equation (8.5-71) in dependence of the ratio L_G/d_m.

(6) In the case of system analyses using mean flexibility factors the mean stress indices C_2 shall be taken in accordance with clause 8.5.2.8.3.5.

(7) In the stress analysis using equations (8.5-1) to (8.5-6) the bending stress due to a resulting moment on account of bending and torsional moments is determined to obtain the mean stress index.

(8) The values given hereinafter for flexibility factors at certain points under investigation $k_x \neq k_y \neq k_z$ apply to pipe elbows and curved pipe sections which at both ends are connected to straight pipes showing the dimension of the curved section and the distance of which to the next curved section is at least two times the outside diameter:

$$k_x = 1.0 \tag{8.5-75}$$

$$k_y = k_p \cdot \frac{1.25}{h} \text{ ; but at least} \geq 1 \tag{8.5-76}$$

$$k_z = k_p \cdot \frac{k_\alpha}{h}\text{; but at least} \geq 1 \qquad (8.5\text{-}77)$$

with k_p according to equation (8.5-72)

h according to equation (8.5-74)

k_α = 1.65 for $\alpha_0 \geq 180°$

k_α = 1.30 for $\alpha_0 = 90°$

k_α = 1.10 for $\alpha_0 = 45°$

k_α = h for $\alpha_0 = 0°$

The values for k_z may be subject to linear interpolation between 180° and 0°.

(9) In the case of system analyses using flexibility factors at certain points under investigation the following stress indices C_{2m} related to certain points under investigation and moments shall be used:

$$C_{2x} = 1.0 \qquad (8.5\text{-}78)$$

$$C_{2y} = 1.71/h^{0.53} \text{ but at least} \geq 1 \qquad (8.5\text{-}79)$$

$$C_{2z} = 1.95/h^{2/3} \quad \text{for } \alpha_0 \geq 90° \qquad (8.5\text{-}80)$$

$$= 1.75/h^{0.58} \quad \text{for } \alpha_0 = 45° \qquad (8.5\text{-}81)$$

$$= 1.0 \quad \text{for } \alpha_0 = 0° \qquad (8.5\text{-}82)$$

The values for C_{2z} may be subject to linear interpolation between 90° and 0°, however no value of α_0 smaller than 30° shall be used; C_{2z} shall never be less than 1.

(10) Where flanges or similar stiffeners are located at a distance L_G less than or equal to 2 x d_a from the commencement or end of curvature, for such bends and bent pipes linear interpolation shall be made between k_y and k_z of equations (8.5-76) and (8.5-77) and $k''_{y,z}$ as per sub-clause 11 in dependence of the ratio L_G/d_a.

(11) Bends and bent pipes, where flanges or similar stiffeners are located at a distance L_G less than or equal to $d_a/2$ from the commencement or end of curvature, k_y shall be replaced by k''_y and k_z by k''_z, where the following applies:

$k''_y = c \cdot k_y$, however ≥ 1

$k''_z = c \cdot k_z$, however ≥ 1,

where

$c = h^{1/6}$ if stiffened on one side

$c = h^{1/3}$ if stiffened on both sides.

(12) In the case of system analyses using flexibility factors at certain points under consideration, where the stress analysis is based on equations (8.5-1) to (8.5-6), the bending stress resulting from bending or torsional moments may be determined using the stress indices related to certain points under consideration and moments. Here, the resulting values shall be substituted as follows:

- instead of $B_2 \cdot M_{iI}$ now use

$$\max\left\{1.0 \cdot M_{iI}\,;\, 0.67 \cdot \sqrt{(C_{2x} \cdot M_x)^2 + (C_{2y} \cdot M_y)^2 + (C_{2z} \cdot M_z)^2}\right\} \qquad (8.5\text{-}83)$$

- instead $C_2 \cdot M_{i(II\text{-}V)}$ now use

$$\max\left\{1.5 \cdot M_{i(II-V)}\,;\, 1.0 \cdot \sqrt{(C_{2x} \cdot M_x)^2 + (C_{2y} \cdot M_y)^2 + (C_{2z} \cdot M_z)^2}\right\} \qquad (8.5\text{-}84)$$

8.5.2.10.4 Branches complying with Annex A 5.2.5 with $d_{Ai}/d_{Hi} \leq 0.5$

(1) The deflection behaviour of branch connections complying with Annex A 5.2.5 with d_{Ai}/d_{Hi} not exceeding 0.5 can be modelled according to **Figure 8.5-10** as follows:

a) beam in direction of pipe run axis having pipe run dimensions and extending to the intersection of the run pipe centre line with the branch pipe centre line,

b) assumption of rigid juncture at intersection of pipe run and branch axes,

c) assumption of rigid beam on a branch pipe length of $0.5 \cdot d_{Ha}$ from the juncture (intersection of axes) to the run pipe surface,

d) Assumption of element with local flexibility at the juncture of branch pipe axis and run pipe surface.

(2) The flexibilities (unit of moment per radians) of the flexible element with regard to the branch pipe bending moments can be determined by approximation as follows:

a) for bending along axis x

$$C_x = \frac{E \cdot I_R}{k_x \cdot d_{Ra}} \qquad (8.5\text{-}85)$$

with

$$k_x = 0.1 \cdot \left(\frac{d_{Ha}}{s_{Hc}}\right)^{1.5} \cdot \left(\frac{s_{Hc}}{s_n} \cdot \frac{d_{Ra}}{d_{Ha}}\right)^{0.5} \cdot \frac{s_{Rc}}{s_{Hc}} \qquad (8.5\text{-}86)$$

b) for bending along axis z

$$C_z = \frac{E \cdot I_R}{k_z \cdot d_{Ra}} \qquad (8.5\text{-}87)$$

with

$$k_z = 0.2 \cdot \frac{d_{Ha}}{s_{Hc}} \cdot \left(\frac{s_{Hc}}{s_n} \cdot \frac{d_{Ra}}{d_{Ha}}\right)^{0.5} \cdot \frac{s_{Rc}}{s_{Hc}} \qquad (8.5\text{-}88)$$

Regarding the notations **Figure 8.5-7** applies with the additional definitions

I_R moment of inertia of the branch pipe,

$$I_R = \pi \cdot \left(d_{Ra}{}^4 - d_{Ri}{}^4\right)/\,64 \qquad (8.5\text{-}89)$$

s_n value for nozzle wall thickness, i.e.:

for designs a and b of **Figure 8.5-7**:

$s_n = s_{Ac}$, if $L_1 \geq 0.5 \cdot \sqrt{(d_{Ai} + s_A) \cdot s_A}$

$s_n = s_{Rc}$, if $L_1 < 0.5 \cdot \sqrt{(d_{Ai} + s_A) \cdot s_A}$

for design c of **Figure 8.5-7**:

$s_n = s_{Rc} + (2/3) \cdot y$, if $\alpha \leq 30°$

$s_n = s_{Rc} + 0.385 \cdot L_1$, if $\alpha > 30°$

for design d of **Figure 8.5-7**: $s_n = s_{Rc}$

(3) With regard to the deflection due to torsional, normal and transverse forces the flexible element shall be considered to be rigid.

8.5.2.10.5 Branch connections with $d_{Ai}/d_{Hi} > 0.5$ and butt welding tees

Branch connections with d_{Ai}/d_{Hi} exceeding 0.5 and butt welding tees shall also be modelled in accordance with clause 8.5.2.10.4 and **Figure 8.5-10** where, however, the flexible element shall be omitted.

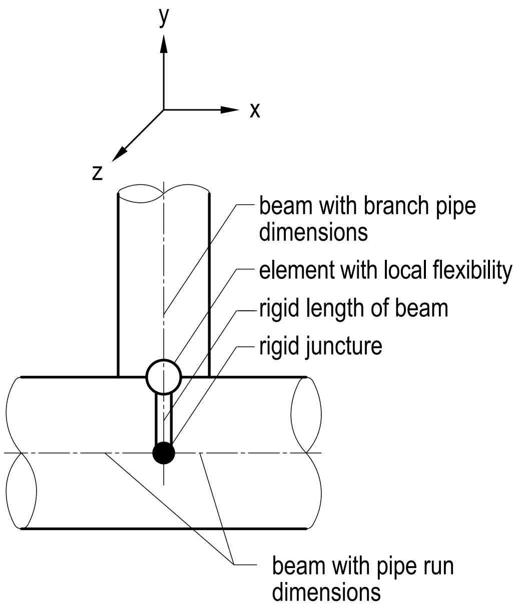


Figure 8.5-10: Modelling of branch connections in straight pipe

8.5.3 Piping systems of test groups A2 and A3

8.5.3.1 Design values and units relating to clause 8.5.3

Notation	Design value	Unit
c_2	wall thickness reduction due to chemical or mechanical wear	mm
d_a	pipe outside diameter	mm
d_i	pipe inside diameter	mm
d_{in}	pipe nominal inside diameter	mm
d_m	mean diameter	mm
d_{Am}	mean diameter of branch connection	mm
d_{Rm}	mean diameter of branch pipe	mm
f	stress range reduction factor for cyclic conditions in accordance with Table 8.5-5	—
h	flexibility characteristic according to Table 8.5-5	—
i	stress intensification factor according to Table 8.5-5	—
k	flexibility factor	—
l	length of segment	mm
p	design pressure	MPa
p_{max}	maximum operating pressure	MPa
r	radius	mm
r_1, r_2, r_3	fillet radii according to Table 8.5-5	mm
s	wall thickness	mm
s_1	wall thickness at large end of reducer	mm
s_2	wall thickness of conical portion of reducer	mm
s_3	wall thickness at small end of reducer	mm
s_x, s_{x1}, s_{x2}	wall thickness according to Table 8.5-5	mm
s_A	wall thickness of branch connection	mm
s_R	wall thickness of branch pipe	mm
s_c	wall thickness without cladding acc. to clause 7.1.4 or measured wall thickness minus corrosion allowance and cladding; In the case of pipe bends with wall thickness increase at the intrados exceeding 15 %, credit shall be taken for the material notch by using the average value, and in the case of induction bends, the geometric average from the smallest and greatest wall thickness at the centre of bend as design wall thickness s_c. In the case of induction bends meeting the dimensional requirements of KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bends), the requirements of sub-clause 8.5.1 (6) shall be met.	mm
y	general design value	—
B_1	primary stress intensity due to internal pressure loading (see clause 8.5.2.8)	—
B_2	primary stress intensity due to moment loading (see clause 8.5.2.8)	—
C_2	primary plus secondary stress intensity due to moment loading	—
E_{20}	modulus of elasticity at 20 °C	N/mm^2
E	modulus of elasticity at temperature	N/mm^2
K_2	peak stress index for moment loading	—
M_{iI}	resultant moment due to dead weight and other sustained loads	Nmm
M_{iII}	resultant moment due to occasional loads such as pressure thrusts, earthquake, aircraft crash, etc.	Nmm
M_{iIII}	resultant moment due to loading from restraint to thermal expansion	Nmm
M_{iIV}	resultant moment ifrom any single nonrepeated anchor movement	Nmm
N	number of load cycles referring to the maximum range of moments	—
N_0	number of load cycles for maximum temperature difference	—
N_1, N_2, to N_n	number of load cycles with smaller temperature differences than at N_0	—
P_l	local primary membrane stress	N/mm^2
P_m	general primary membrane stress	N/mm^2
Q	secondary membrane or bending stress	N/mm^2
S	design stress intensity according to Table 6.6-1	N/mm^2
S_{RT}	design stress intensity according to Table 6.6-1 at room temperature	N/mm^2
S_A	allowable stress intensity range for thermal expansion stresses (see clause 8.5.3.7)	N/mm^2
W	section modulus	mm^3
α	angle according to Table 8.5-5	
δ	allowable mean value for mismatch of butt welds in accordance with Table 8.5-5	mm
σ_I	primary stress intensity acc. to cl. 8.5.3.3	N/mm^2
σ_{II}	primary plus secondary stress intensity acc. to cl. 8.5.3.4	N/mm^2
σ_{III}	stress intensity resulting from restraint to thermal expansion	N/mm^2

Notation	Design value	Unit
σ_{IV}	stress intensity resulting from one single anchor displacement	N/mm^2
σ_V	equivalent stress range resulting from internal pressure, dead weight, other sustained loads, and restraint to thermal expansion	N/mm^2

8.5.3.2 General

(1) When applying the component-specific design method in accordance with this clause, clause 7.7.2.3 shall be taken into account with regard to the classification of stresses from restrained thermal expansions.

Note:

The stress values σ_I to σ_{VI}, given in Section 8.5 as stress intensity or equivalent stress range do no exactly correspond to the respective definitions of clause 7.7.3, but are conservative evaluations of the respective stress intensity or equivalent stress range.

8.5.3.3 Design condition (Level 0)

Except for a single straight pipe, the following conditions apply to the determination and limitation of the primary stress intensity:

$$\sigma_I = B_1 \cdot \frac{d_a \cdot p}{2 \cdot s_c} + B_2 \frac{M_{iI}}{W} \le 1.5 \cdot S \qquad (8.5\text{-}90)$$

where

σ_I	primary stress intensity	N/mm^2
B_1, B_2	stress indices, see clause 8.5.2.8	N/mm^2
S	design stress intensity acc. to Table 6.6-1	N/mm^2
p	design pressure	MPa
d_a	pipe outside diameter, if required $d_a = d_{in} + 2 \cdot s_c + 2 \cdot c_2$, see Section 6.5	mm
s_c	wall thickness without cladding acc. to clause 7.1.4 or measured wall thickness minus corrosion allowance c_2 (see Section 6.4) and cladding; in the case of pipe bends with wall thickness increase at the intrados exceeding 15 %, credit shall be taken for the material notch by using the average value, and in the case of induction bends, the geometric average from the smallest and greatest wall thickness at the centre of bend as design wall thickness s_c. In the case of induction bends meeting the dimensional requirements of KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bends), the requirements of sub-clause 8.5.1 (6) shall be met.	mm
W	pipe section modulus	mm^3
M_{iI}	resulting moment due to dead weight and other sustained loads. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment (moments resulting from different load cases that cannot occur simultaneously need not be considered in calculating the resultant moment). If the method of analysis of dynamic loads is such that only magnitudes without relative algebraic signs are obtained, that combination of directional moment components shall be used leading to the greatest resultant moment.	N/mm^2

8.5.3.4 Level A and B

8.5.3.4.1 General

For each load case the directional moment components shall be determined which always refer to a reference condition.

8.5.3.4.2 Determination and limitation of the primary stress intensity

For the determination of the stress intensity resulting from loadings due to internal pressure, dead weight, other sustained and occacional loads including external events the following condition applies:

$$\sigma_{II} = B_1 \cdot \frac{d_a \cdot p_{max}}{2 \cdot s_c} + B_2 \frac{M_{iI} + M_{iII}}{W} \le 1.8 \cdot S \qquad (8.5\text{-}91)$$

but not exceed $1.5 \cdot R_{p0.2T}$

where

σ_{II}	primary stress intensity	N/mm^2
S	design stress intensity acc. to Table 6.6-1 at the related temperature T	N/mm^2
p_{max}	maximum operating pressure	MPa
M_{iII}	resultant moment due to occasional loadings, such as pressure thrusts, earthquake, aircraft crash, etc. For dynamic loadings only half the range shall be used. Loadings from dynamic effects of anchor displacement may be omitted here, but shall then be considered in equations (8.5-92) and (8.5-94).	N/mm^2

All other notations can be found in 8.5.3.3.

8.5.3.4.3 Determination and limitation of the secondary stress intensity range

(1) Equation (8.5-92) or equation (8.5-94) shall be satisfied.

a) Restraint to thermal expansion

$$\sigma_{III} = i \cdot \frac{M_{iIII}}{W} \le S_A \qquad (8.5\text{-}92)$$

σ_{III}	stress intensity due to restraint to thermal expansion	N/mm^2
M_{iIII}	range of resultant moments due to restrained thermal expansion; also include moment effects of anchor displacements if the load case "dynamic anchor displacement effects" was omitted from equation (8.5-91)	Nmm
S_A	allowable stress range, see clause 8.5.3.7 for thermal expansion stresses	N/mm^2
i	stress intensification factor according to **Table 8.5-5**	—

b) Consideration of the effects of any single nonrepeated anchor movement (e.g. building settlement)

$$\sigma_{IV} = i \cdot \frac{M_{iIV}}{W} \le 3.0 \cdot S \qquad (8.5\text{-}93)$$

M_{iIV}	resultant moment due to any single anchor movement	Nmm
S	design stress intensity at room temperature	N/mm^2
i	stress intensification factor according to **Table 8.5-5**	—

c) The stresses due to internal pressure, dead weight, other sustained loads and restrained thermal expansion shall meet the requirements of the following equation:

$$\sigma_V = \frac{p \cdot d_a}{4 \cdot s_c} + 0.75 \cdot i \cdot \frac{M_{iI}}{W} + i \cdot \frac{M_{iIII}}{W} \le S + S_A \qquad (8.5\text{-}94)$$

i stress intensification factor according to **Table 8.5-5** —

The product 0.75 · i shall not be less than 1.0.

(2) If the requirements of equations (8.5-91) and (8.5-93) have been met, equation (8.5-94) need not be verified.

8.5.3.5 Level C Service limits

(1) For Level C the primary stresses shall be determined by means of equation (8.5-91).

In calculating the resultant moment loading the dynamic effects of anchor displacement due to earthquake or other secondary effects need not be included.

(2) For equation (8.5-91) the allowable design stress intensity to be used for this condition is 2.25 · S, but not greater than 1.8 · $R_{p0.2T}$.

8.5.3.6 Level D service limits

(1) For Level D the primary stress shall be determined by means of equation (8.5-91).

In calculating the resultant moment, the dynamic effects of anchor displacement due to earthquake or other secondary effects need not be included.

(2) For equation (8.5-91) the allowable design stress intensity to be used for this condition is 3.0 · S, but not greater than 2.0 · $R_{p0.2T}$.

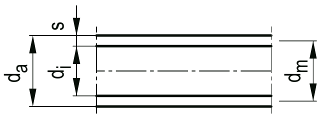
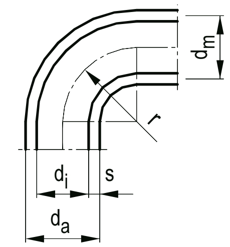
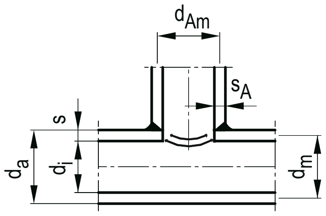
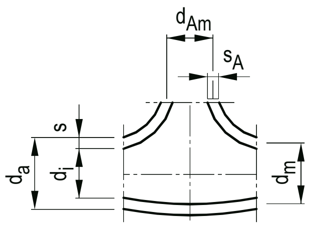
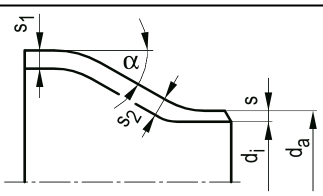
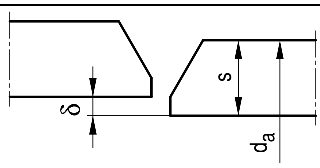
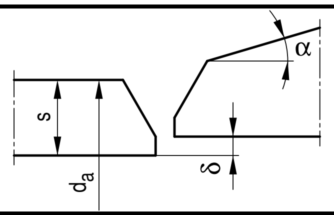
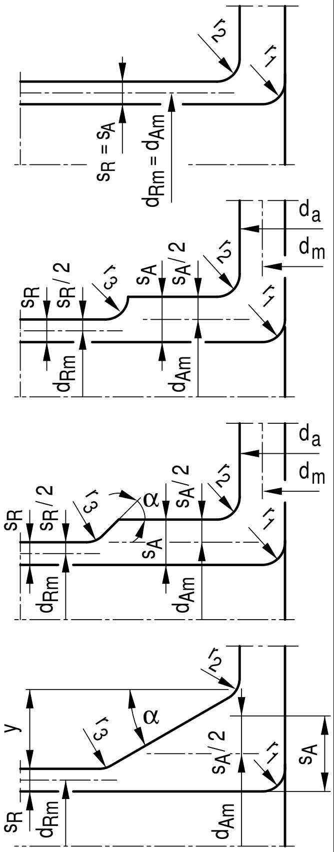
Description	Sketch	Shape factor h and configuration requirements 1)	Flexibility factor k (k ≥ 1) 2)	Stress intensification factor i (i ≥ 1) 3)	Section modulus 4)
1. Straight pipe		—	1	1	$\frac{\pi}{32} \cdot \frac{d_a^4 - d_i^4}{d_a}$
2. Welding elbow or pipe bend 5)		$\frac{4 \cdot r \cdot s}{d_m^2}$	$\frac{1.65}{h}$	$\frac{0.9}{h^{2/3}}$	
3. Tee with set-on, set-in or extruded nozzle		$\frac{2 \cdot s}{d_m}$	1	$\frac{0.9}{h^{2/3}}$ min. 2.1	for run pipe: $\frac{\pi}{32} \cdot \frac{d_a^4 - d_i^4}{d_a}$ for nozzle: $\frac{\pi}{4} \cdot d_{Am}^2 \cdot s_x$ with s_x taken as the smaller value of s_{x1} = s and $s_{x2} = i \cdot s_A$
4. Fabricated welding tee with wall thickness s and s_A for connection		$\frac{8.8 \cdot s}{d_m}$	1	$\frac{0.9}{h^{2/3}}$	
5. Reducer		configuration requirements α ≤ 60 degrees s ≥ d_a/100 $s_2 \ge s_1$	1	$0.5 + \frac{\alpha}{100} \cdot \sqrt{\frac{d_a}{s}}$ max 2.0 (α in degrees)	$\frac{\pi}{32} \cdot \frac{d_a^4 - d_i^4}{d_a}$
6. Butt weld		s ≥ 5 mm and δ ≤ 0.1·s	1	1.0	$\frac{\pi}{32} \cdot \frac{d_a^4 - d_i^4}{d_a}$
		s < 5 mm or δ > 0.1·s	1	flush: 1.0 as-welded: 1.8	

Table 8.5-5: Flexibility characteristics and factors, stress intensification factors and section moduli

Description	Sketch	Shape factor h and configuration requirements [1]	Flexibility factor k ($k \geq 1$) [2]	Stress intensification factor i ($i \geq 1$) [3]	Section modulus [4]
7. Welded transition		$\alpha \leq 30°$	1	$1.3 + 0.0036 \cdot \frac{d_a}{s} + 3.6 \cdot \frac{\delta}{s}$ max. 1.9	$\frac{\pi}{32} \cdot \frac{d_a^4 - d_i^4}{d_a}$
8. Tee with special shape conditions [6]	 $s_A = s_R + 2y/3$ $d_{Aa} = d_{Am} + s_A$	$\frac{d_{Rm}}{d_m} \leq 0.5$ [7] $\frac{d_m}{s} \leq 100$ $0.1 \cdot s \leq r_1 \leq 0.5 \cdot s$ $r_2 \geq \max\left\{\frac{s_A}{2}, \frac{s}{2}, \frac{s_R + y}{2}\right\}$ $\alpha \leq 45°$ $r_3 \geq \max\left\{\alpha \cdot \frac{d_{Rm} + s_R}{500}, 2\sin^3\alpha \cdot (d_{Am} + s_A - d_{Rm} - s_R)\right\}$ For nozzles smaller than DN 100 the above condition for r_1 can be omitted. For r_3 conditions α shall be taken in degrees.	1	for run pipe [7]: $i = 0.4 \cdot \left(\frac{d_m}{2 \cdot s}\right)^{2/3} \cdot \frac{d_{Rm}}{d_m}$ but at least i = 1.5 for nozzle: $i = 1.5 \cdot \left(\frac{d_m}{2 \cdot s}\right)^{2/3} \cdot \left(\frac{d_{Rm}}{d_m}\right)^{1/2} \cdot \frac{s_R}{s} \cdot \frac{d_{Rm}}{d_{Am} + s_A}$	for run pipe: $\frac{\pi}{4} \cdot d_m^2 \cdot s$ for nozzle: $\frac{\pi}{4} \cdot d_{Rm}^2 \cdot s_R$

1) The flexibility factors k to be used in the system analysis and the stress intensification factors i to be used in the stress analysis depend on the shape factor h and only apply if the configuration requirements given are satisfied.

2) The flexibility factor k indicates how the actual deformation of a piping component due to bending is related to the deflection obtained according to the theory of straight or slightly curved beams. In no case shall k be less than 1. **Table 8.5-5** refers to k values ≠ 1 only for elbows/bends. These k values apply over the effective arc length for in-plane and out-of-plane bending. For deformation due to torsional, normal and transverse forces the normal theory of beam shall further apply. The local flexibility of branch connections on tees may be covered by elements with local flexibility at the run pipe surface i.e. at the beginning of the branch, if this can be verified accordingly.

3) The stress intensification factor i is used to correct the stress intensity resulting from moment loading on straight pipe to obtain component-specific values for comparison with the stress intensity limits given in clauses 8.5.3.3 to 8.5.3.6. In no case, i shall be less than 1. Prerequisite to the use of stress intensification factors i is that the components have been adequately dimensioned in accordance with Section A 5.

4) This column contains the section moduli to be used in connection with the stress intensification factor i.

5) For curved pipes or elbows stiffened by a flange or similar stiffeners within a distance less than $d_m/2$ from either end of the curved section of the pipe or from the ends of the elbows, k and i shall be replaced by $k' = c \cdot k$ and $i' = c \cdot i$ in which case the following applies:
$c = h^{1/6}$ if stiffened at one end
$c = h^{1/3}$ if stiffened at both ends as well as
$k', i' \geq 1$.

6) Four different nozzle designs are shown (a, b, c, d); see **Figure 8.5-7.**

7) Where the conditions for r_2 are not satisfied, twice the stress intensification factor obtained from the respective formulae shall be in the calculation for run pipe and branch, however, at least i = 2.1 shall be taken.

Table 8.5-5: Flexibility characteristics and factors, stress intensification factors and section moduli (continued)

8.5.3.7 Determination of the allowable thermal expansion stress range S_A

(1) The allowable stress intensity range S_A shall be

$$S_A = f \cdot (1.25 \cdot S_{RT} + 0.25 \cdot S) \quad (8.5\text{-}95)$$

f stress range reduction factor for cyclic conditions

f refers to the total number of cycles of full temperature cycles over the total expected years service life of the piping system.

f shall be taken from the following Table.

N	f
up to 7 000	1.0
over 7 000 up to 14 000	0.9
over 14 000 up to 22 000	0.8
over 22 000 up to 45 000	0.7
over 45 000 up to 100 000	0.6
over 100 000	0.5
N : number of full temperature cycles	

Where load cycles with smaller temperature differences occur, N is calculated as follows:

$$N = N_0 + \left(\frac{M_{1III}}{M_{0III}}\right)^5 \cdot N_1 + \left(\frac{M_{2III}}{M_{0III}}\right)^5 \cdot N_2 + .. + \left(\frac{M_{nIII}}{M_{0III}}\right)^5 \cdot N_n \quad (8.5\text{-}96)$$

N_0 number of cycles for maximum temperature differences

$N_1, N_2, ..., N_n$ number of cycles with smaller temperature differences than for N_0

(2) The stress range reduction factor f applies for non-corroding operation and corrosion-resistant materials.

8.5.3.8 System analysis

(1) The requirements for system analysis shall be taken from Section 7.6.

(2) The following especially applies to piping systems:

a) The system geometry shall comprise all components and parts which significantly influence the system behaviour. The supporting conditions for supporting elements shall be taken into account.

b) Flexibility factors and stress intensification factors are shown in **Table 8.5-5**. For components not shown in Table 8.5-5, the stress intensification factor may be taken as follows:

$$i = \frac{C_2 \cdot K_2}{2} \quad (8.5\text{-}97)$$

where C_2 and K_2 are stress indices for components shown in **Table 8.5-1**.

c) Simplifying assumptions may be used if it is ensured that forces, moments and stresses including the effects of stress intensification are not underestimated

d) The dimensional properties of pipe and pipe fittings shall be based on the average wall thickness s_c (see clause 7.1.4).

e) Cold springing

Where a piping system is cold sprung, the effects of cold springing shall be evaluated principally for all load cases, however, always for the final assembly condition and the normal operating conditions. Here, the following shall be considered:

- Cold springing has no influence on the extent of the stress range σ_{III} according to clause 8.5.3.4.3.
- The cold springing loads determined for the cold-sprung condition with E_{20} will change at a ratio E_ϑ / E_{20} for the operational load cases.

Where a comprehensive calculation of the effects of cold springing is required the calculation shall be adapted to the assembly conditions. For this purpose, each assembly condition shall be covered which causes distortions and thus stresses in the total system or initially, in system sections. The following shall be taken into account.

- location of non-positive connections,
- size and direction of gaps, clearances and mismatches,
- points of application, direction, extent, and type of the loadings causing cold springing (forces, dead weight),
- free-end displacements.

By approximation cold springing may be calculated using cold spring factors and negative percentages of thermal expansions including external movements transmitted by terminal and intermediate attachments on the designated total system.

Due to the uncertainties arising from this calculation credit shall be taken of not more than 2/3 of the releasing effect of cold springing (e.g. in full-load operation). The loading effects, however, shall be fully considered (e.g. in the final assembly condition).

8.5.3.9 Local overstrain

The piping system analysis according to Section 8.5 assumes elastic behaviour of the entire piping system. This assumption is sufficiently accurate for systems in which plastic straining occurs at many points or over relatively wide regions (e.g. due to P_m + Q or P_l + Q exceeding $3 \cdot S_m$), but fails to reflect the actual strain distribution in unbalanced systems in which only a small portion of the piping undergoes plastic strain.

Unbalance can be produced:

a) by use of small pipe runs in series with larger or stiffer pipe, with the small lines relatively highly stressed,

b) by local reduction in pipe cross-section, or local use of a weaker material,

c) in a system of uniform size and materials, by use of a line configuration for which the neutral axis or thrust line is situated close to the major portion of the line itself, with only a very small offset portion of the line absorbing most of the expansion strain.

In these cases, the weaker or higher stressed portions will be subjected to strain concentrations due to elastic follow-up of the stiffer or lower stressed portions. Such strain concentrations can be avoided by suitable line configurations with a respective supporting concept, taking the aforementioned points (a) to (c) into account. Supporting measures may be the use of higher strength material at locations of possible strain concentrations and, if required, by the application of cold spring. The measures to be taken to avoid strain concentrations are especially important for piping systems of austenitic materials which show a non-linear stress-strain behaviour.

Note:
For literature on the problems of „elastic follow up“ see e.g. F.V. Naugle "Design Guidance for Elastic Follow-up", Journal of Pressure Vessel Technology, Transactions of the ASME Vol. 106, No. 1, February 1984.

8.5.3.10 Determination of resulting moments

The resulting moments M_{iI} to M_{iIV} in equations (8.5-90) to (8.5-93) are calculated as follows:

$$M_i = \sqrt{M_{xi}^2 + M_{yi}^2 + M_{zi}^2} \quad (8.5\text{-}98)$$

For tees the resulting moment of each leg (run pipe and nozzle) shall be calculated separately, with the moments taken at the junction point of the legs (see **Figure 8.5-11**).

For tees with a diameter ratio d_{Rm}^+/d_{Hm} less than 0.5 the branch moments at the outside surface of the run pipe may be used for the branch leg.

All other notations see **Figure 8.5-11**.

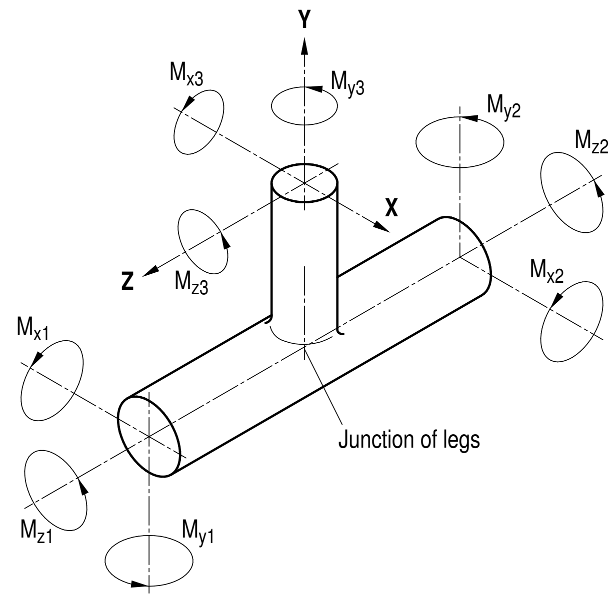


Figure 8.5-11: Moments on tee

8.6 Integral areas of component support structures

8.6.1 General

This section applies to the calculation of the integral areas of component support structures which are intended to accommodate loadings.

The integral areas of component support structures are attached to the pressure-retaining area by welding, forging, casting or fabricated from the solid.

Therefore, the portion of the support structure directly adjacent to the component wall interacts with the component (area of influence).

8.6.2 Limitation of integral area

(1) The limitation of the integral area of component support structures is shown in **Figure 8.6-1**. The distance l is calculated as follows:

a) Shells (e.g. skirts, tubular nozzles)

$$l = 0.5 \cdot \sqrt{r \cdot s_c} \qquad (8.6\text{-}1)$$

where

r mean radius of shell of support structure

s_c thickness of support structure shell in accordance with clause 7.1.4

b) bars or sections

$$l = 0.5 \cdot \sqrt{r^2 / 2} \qquad (8.6\text{-}2)$$

where

r radius of bar of-one-half the maximum cross-sectional dimension of the section

c) other shapes

$$l = 0.5 \cdot \sqrt{r \cdot s_c} \qquad (8.6\text{-}3)$$

where

r on-half the maximum dimension of a flange, tee-section, plat or round section or one-half the maximum leg width of an angle section

s_c flange thickness of sections or plate thickness according to clause 7.1.4

(2) Where, however, a detachable connection is provided within a distance l, the limit between the integral and non-integral area shall be set at this location.

8.6.3 Design

(1) Integral areas of component support structures are to be considered part of the supporting component. All simultaneously occurring loads shall be taken into account. For component support structures the following forces and moments shall be determined:

a) normal force F_N,

b) transverse force F_Q,

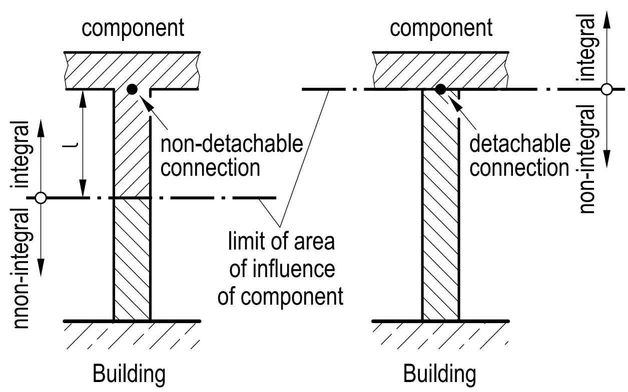
c) torsional moment M_t,

d) bending moment M_b.

(2) The effects of external forces and moments on the component wall shall be considered in accordance with Section 7 or for vessels in accordance with clause 8.2.7.

(3) Accordingly, the stresses shall be evaluated in accordance with Section 7, or for vessels, with clause 8.2.7. The stability behaviour shall be analysed.

Component support structure



l : die-out length

Figure 8.6-1: Type of attachment of component support

9 Type and extent of verification of strength and pertinent documents to be submitted

(1) For the design approval to be made by the authorized inspector in accordance with § 20 AtG (Atomic Energy Act) the following verifications of strength for pressure and activity-retaining components of systems outside the primary circuit shall be carried out and be submitted in form of a report:

a) dimensioning for test group A1, A2, A3 components,

b) analysis of the mechanical behaviour of test group A1 components.

(2) The design, report and inspection shall be based on the pertinent Sections of KTA 3211 Safety Standards.

(3) Each report on design and calculation shall normally contain the following information at the extent required for review of the strength verifications:

a) explanation of design and calculation procedures, especially of assumptions made,

b) indication of calculation procedures, theoretical bases and programmes used,

c) load data, combination of loads and their classification,

d) geometric data,

e) characteristic values (mechanical properties) of the materials used,

f) input data,

g) results obtained including fatigue usage factors,

h) evaluation of results and comparison with allowable values,

i) conclusions drawn from the results obtained,

j) references, bibliography and literature.

Annex A

Dimensioning

A 1 General

(1) The design rules hereinafter apply to the dimensioning of components in accordance with Section 6 and their parts subject to design pressure and additional design mechanical loads at design temperature.

(2) Regarding stress limitation for test group A1 the design stress intensity S_m, and for test groups A2 and A3 the design stress intensity S shall apply (see **Table 6.6-1**).

(3) For the purpose of simplification only the S_m value is taken in the equations.

(4) The design values and units are given separately for each Section.

(5) The confirmatory calculation of parts with nominal wall thickness s_n shall be made within this Annex with the wall thickness $s_{0n} = s_n - c_1 - c_2$ with $s_n \geq s_0 + c_1 + c_2$. Regarding allowances Section 6.4 applies.

(6) The figures contained in this Annex do not include allowances.

(7) The requirements laid down in Annex A 2 for general parts of the pressure retaining wall are also applicable, in consideration of the respective requirement, to specific parts of pumps, valves and piping complying with A 3 to A 5 unless other requirements have been fixed in these Annexes.

(8) Design values and units

Notation	Design value	Unit
b	width	mm
c	wall thickness allowance	mm
d	diameter	mm
h	height	mm
l	length	mm
p	design pressure	MPa
p’	test pressure	MPa
r, R	radii	mm
s	wall thickness	mm
s_0	calculated wall thickness according to Figure 7.1-1	mm

A 2 Dimensioning of parts of the pressure retaining wall

A 2.1 General

The equations given in Section A 2.2 to A 2.10 for dimensioning only apply to the determination of the required wall thickness of the individual parts under internal or external pressure, however, without consideration of the elastic relationship of the entire structure. The loadings on pressure vessel walls resulting from external forces and moments are covered by clause 8.2.7.

A 2.2 Cylindrical shells

A 2.2.1 Design values and units relating to Section A 2.2

Notation	Design value	Unit
d_a	outside diameter of cylindrical shell	mm
d_i	inside diameter of cylindrical shell	mm
f_k	safety factor against elastic instability	—
f_v	additional safety factor against gross plastic deformation	—
l	unsupported length	mm
n	number of lobes	—
p	design pressure	MPa
$p_{zul.}$	allowable pressure	MPa
s_0	calculated wall thickness according to Figure 7.1-1	mm
s_{0n}	nominal wall thickness of the shell excluding allowances according to Section 6.5	mm
Z	design value: $Z = 0.5 \cdot \pi \cdot d_a/l$	—
E	modulus of elasticity	N/mm^2
S_m	design stress intensity for components of test group A1	N/mm^2
U	ovality	%
ν	Poisson's ratio	—
σ_a	stress in axial direction	N/mm^2
$\overline{\sigma}_V$	average equivalent stress	N/mm^2

A 2.2.2 Cylindrical shells under internal pressure

A 2.2.2.1 Scope

The calculation method hereinafter applies to cylindrical shells under internal pressure, where the ratio d_a/d_i does not exceed 1.7. Diameter ratios d_a/d_i not exceeding 2 are permitted if the wall thickness s_{0n} does not exceed 80. Reinforcements of openings in cylindrical shells under internal pressure shall be calculated in accordance with Section 2.8.

A 2.2.2.2 Calculation

(1) For the calculation of the required wall thickness of the shell the following applies:

$$s_0 = \frac{d_a \cdot p}{2 \cdot S_m + p} \qquad \text{(A 2.2-1)}$$

or

$$s_0 = \frac{d_i \cdot p}{2 \cdot S_m - p} \qquad \text{(A 2.2-2)}$$

(2) For the recalculation at given wall thickness the following applies:

$$\overline{\sigma}_V = p \cdot \left(\frac{d_i}{2 \cdot s_{0n}} + 0.5 \right) \leq S_m \qquad \text{(A 2.2-3)}$$

A 2.2.3 Cylindrical shells under external pressure

A 2.2.3.1 Scope

The calculation method hereinafter applies to cylindrical shells under external pressure where the ratio d_a/d_i does not exceed 1.7.

A 2.2.3.2 Safety factors

(1) The additional safety factor against gross plastic deformation shall be taken as f_v = 1.2 irrespective of the material used.

(2) The safety factor against elastic instability shall be taken as f_k = 3.0 irrespective of the material used. Where a higher test pressure as 1.3 · p is required, f_k shall be at least 2.2.

A 2.2.3.3 Calculation

A 2.2.3.3.1 General

(1) It shall be verified by calculation that there is sufficient safety against elastic instability and plastic deformation. The following equations shall be used. The smallest calculated value of p_{zul} shall govern.

(2) The buckling length is the length of the shell. For vessels with dished heads the buckling length begins at the juncture of cylindrical flange (skirt) to knuckle.

A 2.2.3.3.2 Calculation against elastic instability

(1) The calculation shall be made according to:

$$p_{zul} = \frac{E}{f_k} \cdot \left[\frac{2}{\left(n^2 - 1\right) \cdot \left[1 + \left(\frac{n}{z}\right)^2\right]^2} \cdot \frac{s_{0n}}{d_a} + \frac{2}{3 \cdot \left(1 - \nu^2\right)} \cdot \left(n^2 - 1 + \frac{2 \cdot n^2 - 1 - \nu}{1 + \left(\frac{n}{z}\right)^2} \right) \cdot \left(\frac{s_{0n}}{d_a} \right)^3 \right] \quad \text{(A 2.2-4)}$$

where for Z = 0.5 · π · d_a/l shall be taken; n is a full number and shall satisfy the conditions n ≥ 2 and n > Z and shall be selected such that p becomes the smallest value. n means the number of lobes (circumferential waves) which may occur over the circumference in case of instability.

The number of lobes shall be calculated by approximation as follows:

$$n = 1.63 \cdot \sqrt[4]{\frac{d_a^{\,3}}{l^2 \cdot s_{0n}}} \quad \text{(A 2.2-5)}$$

(2) The required wall thickness s_{0n} may be determined in accordance with **Figure A 2.2-1** for usual dimensions. This figure applies to a Poisson's ratio of ν = 0.3. Where the Poisson' s ratio extremely differs from 0.3, equation (A 2.2-4) shall be taken.

A 2.2.3.3.3 Calculation against gross plastic deformation

(1) For d_a/l > 5 the following applies:

$$p_{zul} = \frac{2 \cdot S_m}{f_v} \cdot \frac{s_{0n}}{d_a} \cdot \frac{1}{1 + \frac{1.5 \cdot U \cdot \left(1 - 0.2 \cdot d_a / l\right) \cdot d_a}{100 \cdot s_{0n}}} \quad \text{(A 2.2-6)}$$

The required wall thickness s_{0n} may be determined directly in accordance with **Figure A 2.2-2** for usual dimensions and with U = 1.5 %.

(2) For d_a/l > 5 the larger value of the pressure determined by the two equations hereinafter shall govern the determination of the allowable external pressure:

$$p_{zul} = \frac{2 \cdot S_m}{f_v} \cdot \frac{s_{0n}}{d_a} \geq p \quad \text{(A 2.2-7)}$$

$$p_{zul} = \frac{3 \cdot S_m}{f_v} \cdot \left(\frac{s_{0n}}{l}\right)^2 \geq p \quad \text{(A 2.2-8)}$$

(3) Equation (A 2.2-8) primarily applies to small unsupported lengths. Equations (A 2.2-6) to (A 2.2-8) only apply if no positive primary longitudinal stresses σ_a occur. In equations (A 2.2-6) to (A 2.2-8) S_m shall be replaced by (S_m - σ_a) if σ_a > 0.

A 2.3 Spherical shells

A 2.3.1 Design values and units relating to Section A 2.3

Notation	Design value	Unit
d_a	outside diameter of spherical shell	mm
d_i	inside diameter of spherical shell	mm
f_k	safety factor against elastic instability	—
f_k'	safety factor against elastic instability at increased test pressure	—
f_v	additional safety factor against gross plastic deformation	—
l	unsupported length	mm
n	number of lobes	—
p	design pressure	MPa
p_{zul}	allowable pressure	MPa
s_0	calculated wall thickness according to Figure 7.1-1	mm
s_{0n}	nominal shell wall thickness minus allowances, in accordance with Section 6.5	mm
Z	design value: Z = 0.5 · π · d_a/l	—
C_k	factor according to equation (A 2.3-3)	—
E	modulus of elasticity	N/mm^2
S_m	design stress intensity value for test group A1 components	N/mm^2
U	ovality	%
ν	Poisson's ratio	—
σ_a	axial stress	N/mm^2
σ_k	stress in confirmatory calculation against elastic instability	N/mm^2
σ_v	stress intensity	N/mm^2
$\overline{\sigma}_V$	mean stress intensity	N/mm^2

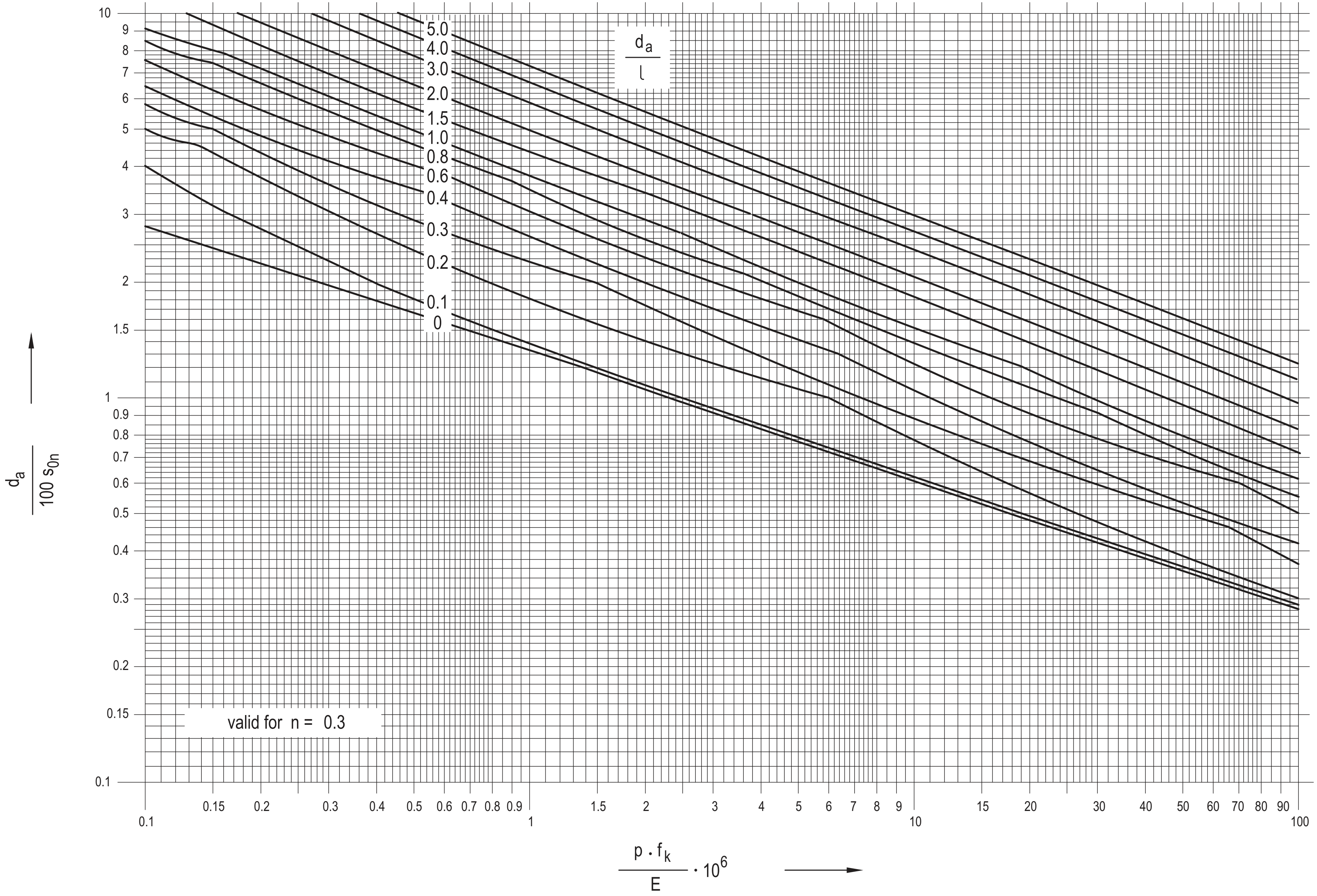


Figure A 2.2-1: Required wall thickness s_{0n} for calculation against elastic instability

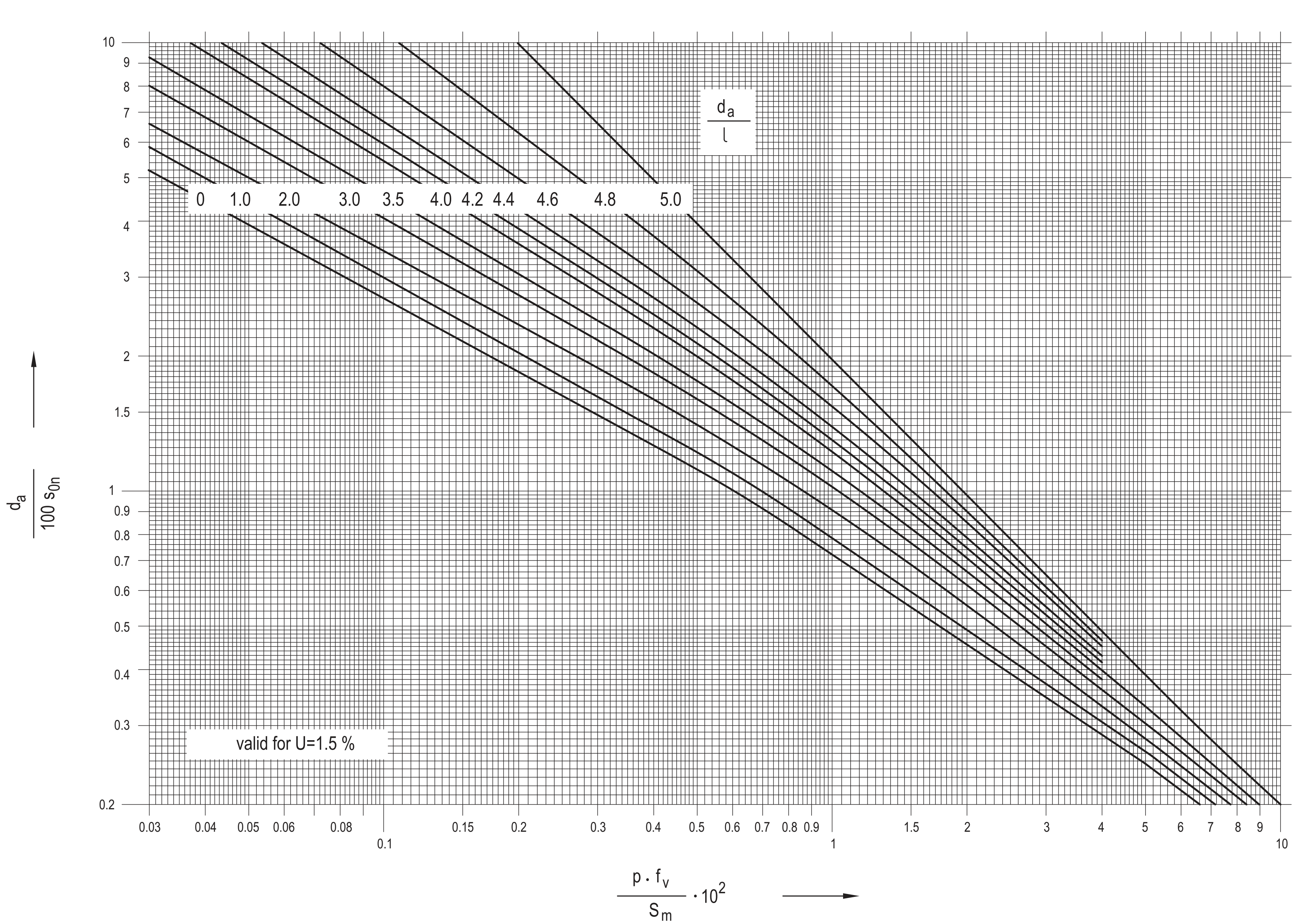


Figure A 2.2-2: Required wall thickness s_{0n} for calculation against gross plastic deformation

A 2.3.2 Spherical shells under internal pressure

A 2.3.2.1 Scope

The calculation hereinafter applies to unpierced spherical shells under internal pressure where the ratio $d_a/d_i \le 1.5$. The calculation of pierced spherical shells under internal pressure shall be made in accordance with Section A 2.8.

A 2.3.2.2 Calculation

(1) For the calculation of the required wall thickness s_0 of spherical shells with a ratio s_{0n}/d_i greater than 0.05 one of the following equations applies:

$$s_0 = \frac{d_a}{2} \cdot \frac{C_k - 1}{C_k} \qquad \text{(A 2.3-1)}$$

or

$$s_0 = \frac{d_i}{2} \cdot (C_k - 1) \qquad \text{(A 2.3-2)}$$

with

$$C_k = \sqrt{1 + \frac{2 \cdot p}{2 \cdot S_m - p}} \qquad \text{(A 2.3-3)}$$

(2) For the calculation of the required wall thickness of thin-walled spherical shells with a ratio s_{0n}/d_i not exceeding 0.05 the following applies:

$$s_0 = \frac{d_a \cdot p}{4 \cdot S_m} \qquad \text{(A 2.3-4)}$$

or

$$s_0 = \frac{d_i \cdot p}{4 \cdot S_m - 2 \cdot p} \qquad \text{(A 2.3-5)}$$

(3) For the confirmatory calculation at a given wall thickness the following applies:

$$\sigma_V = p \cdot \left[\frac{d_i^2}{4 \cdot (d_i + s_{0n}) \cdot s_{0n}} + 0.5 \right] \le S_m \qquad \text{(A 2.3-6)}$$

A 2.3.3 Spherical shells under external pressure

A 2.3.3.1 Scope

The calculation hereinafter applies to spherical shells under external pressure where the ratio d_a/d_i does not exceed 1.5.

A 2.3.3.2 Safety factors

(1) The additional safety factor against gross plastic deformation shall be $f_v = 1.2$ irrespective of the material used.

(2) The safety factor against elastic instability shall be taken from **Table A 2.3-1** irrespective of the material. Where a test pressure higher than $1.3 \cdot p$ is required then the test pressure shall be additionally verified with f_k' from **Table A 2.3-1**.

$\frac{2 \cdot s_0}{d_i}$	f_k	f_k'
0.001	5.5	4.0
0.003	4.0	2.9
0.005	3.7	2.7
0.010	3.5	2.6
$\ge$ 0.1	3.0	2.2
Intermediate values shall be subject to straight interpolation.		

Table A 2.3-1: Safety factors against elastic instability

A 2.3.3.3 Calculation

(1) General

It shall be verified by calculation that there is sufficient safety against elastic instability and plastic deformation. The equations given hereinafter shall be used. The highest value of s_0 obtained from subparagraphs 2 and 3 shall be determining.

(2) Calculation against elastic instability

The required wall thickness is obtained from the following equation:

$$s_0 = d_a \cdot \sqrt{\frac{p \cdot f_k}{1.464 \cdot E}} \qquad \text{(A 2.3-7)}$$

For the confirmatory calculation at a given wall thickness the following applies:

$$\sigma_k = \frac{p}{1.464} \cdot \left(\frac{d_a}{s_{0n}} \right) \le \frac{E}{f_k} \qquad \text{(A 2.3-8)}$$

(3) Calculation against plastic deformation

The required wall thickness is obtained from:

$$s_0 = \frac{d_a}{2} \cdot \left(1 - \sqrt{1 - \frac{2 \cdot p \cdot f_v}{2 \cdot S_m + p \cdot f_v}} \right) \qquad \text{(A 2.3-9)}$$

For spherical shells with a ratio $s_0/d_a \le 0.05$ the required wall thickness may be calculated by approximation from

$$s_0 = \frac{p \cdot d_a \cdot f_v}{4 \cdot S_m} \qquad \text{(A 2.3-10)}$$

For the confirmatory calculation at a given wall thickness the following applies:

$$\overline{\sigma}_V = p \cdot \left[\frac{d_a^2}{4 \cdot (d_a - s_{0n}) \cdot s_{0n}} - 0.5 \right] \le \frac{S_m}{f_v} \qquad \text{(A 2.3-11)}$$

A 2.3.4 Wall thickness of unpenetrated spherical shell and the transition of flange to spherical shell under internal pressure

(1) The wall thickness s_0 of the unpenetrated spherical shell is obtained from equations (A 2.6-3) to (A 2.6-6).

(2) For the wall thickness s_e' at the transition of flange to spherical shell the following applies:

$$s_e \ge s_e' = s_0 \cdot \beta \qquad \text{(A 2.3-12)}$$

The shape factor β takes into account that for a large portion of bending stresses an increased support capability can be expected in case of plastic straining. Where the strain ratio δ of dished heads is assumed, which characterises the support capability, $\beta = 3.5$ may be taken for flanges with inside bolt circle gasket in accordance with **Figures A 2.6-1** and **A 2.6-2**, a value which is obtained by approximation of $\beta = \alpha/\delta$ from **Figure A 2.3-1** .

A 2.3.5 Reinforcement of opening at gland packing space of valves under internal pressure

The reinforcement shall be calculated like for heads with openings according to the area replacement approach method. The strength condition then is:

$$p \cdot \left(\frac{A_p}{A_\sigma} + \frac{1}{2} \right) \le S_m \qquad \text{(A 2.3-13)}$$

The effective lengths are:

$$l_0 = \sqrt{(2 \cdot r + s_0') \cdot s_0'} \qquad \text{(A 2.3-14)}$$

$$l_1 = \sqrt{(d_A + s_A) \cdot s_A} \qquad \text{(A 2.3-15)}$$

with s_0' as actual wall thickness in spherical portion minus allowances c.

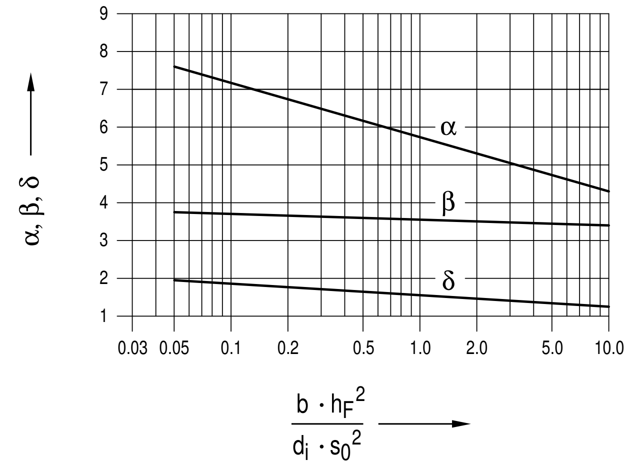


Figure A 2.3-1: Shape factor ß for the transition flange/spherical shell

A 2.4 Conical shells

A 2.4.1 Design values and units relating to Section A 2.4

Notation	Design value	Unit
d_a	outside diameter of conical shell	mm
d_{a1}	outside diameter at large end of cone	mm
d_{a2}	outside diameter at small end of cone	mm
d_i	inside diameter of conical shell	mm
d_{i1}	inside diameter at large end of cone	mm
d_{i2}	inside diameter at small end of cone	mm
e_1	die-out length at large end of cone	mm
e_2	die-out length at small end of cone	mm
e	die-out length according to Fig. A 2.4-3	mm
p	design pressure	MPa
r	transition radius	mm
s	wall thickness	mm
s_0	calculated wall thickness	mm
s_{0n}	nominal wall thickness of shell minus corrosion allowance, according to Section 6.5	mm
s_1	wall thickness at large end of cone	mm
s_2	wall thickness at small end of cone	mm
A_p	pressure-loaded area	mm^2
A_σ	effective cross-sectional area	mm^2
S_m	design stress intensity value for test group A1 components	N/mm^2
β	shape factor in accordance with Table A 2.4-1	—
φ	semi-angle of the apex of the conical section	degree
φ_1	semi-angle of the apex at the large end of the cone	degree
φ_2	semi-angle of the apex at the small end of the cone	degree
Ψ	absolute difference between the semi-apex angles φ_1 and φ_2	degree
σ_1	longitudinal stress	N/mm^2
σ_v	stress intensity	N/mm^2
$\overline{\sigma}_v$	mean stress intensity	N/mm^2

A 2.4.2 Conical shells under internal pressure

A 2.4.2.1 Scope

The calculation hereinafter applies to unpierced conical shells under internal pressure where at the large end of the cone the condition $0.005 \le s_{0n}/d_a \le 0.2$ is satisfied. The calculation of penetrated shells under internal pressure shall be effected in accordance with Section A 2.8.

Note:
For $d_a - d_i = 2 \cdot s_{0n}$ the value $d_a/d_i = 1.67$ corresponds to $s_{0n}/d_a = 0.2$.

A 2.4.2.2 General

(1) Conical shell with corner welds

Conical shells may be welded to each other or to cylindrical shells or sections without knuckle in accordance with clause 5.2.3.

(2) Die-out length

For conical shells with inwardly curved transitions the wall thickness required in accordance with clause A 2.4.2.3, subparagraphs (2) or (4) shall be provided over the knuckle area limited by the die-out length e (see **Figure A 2.4-1**).

The following applies:

$$e_1, e_2 = (r + s_{0n}) \cdot \tan\frac{\Psi}{2} + 0.8 \cdot \sqrt{d_a \cdot s_{0n}} \qquad \text{(A 2.4-1)}$$

In the case of change in wall thickness within the die-out length the respective wall thickness at run-out of curvature shall govern the determination of the lengths e_1 and e_2 according to equation (A 2.4-1).

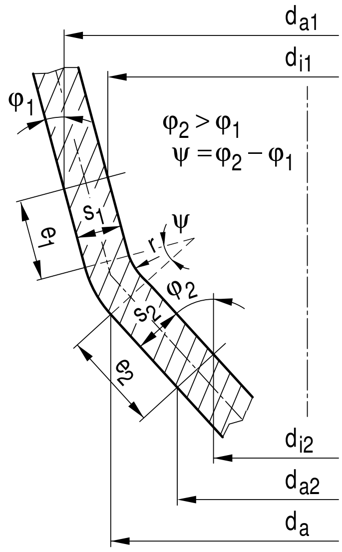


Figure A 2.4-1: Die-out lengths e_1 and e_2

A 2.4.2.3 Calculation

(1) Wall thickness calculation for area without discontinuity of a conical shell with $\varphi \leq 70°$

The required wall thickness of the area without discontinuity of a conical shell (see **Figure A 2.4-2**) is obtained from either

$$s_0 = \frac{d_a \cdot p}{(2 \cdot S_m + p) \cdot \cos\varphi} \quad \text{(A 2.4-2)}$$

or

$$s_0 = \frac{d_i \cdot p}{(2 \cdot S_m - p) \cdot \cos\varphi} \quad \text{(A 2.4-3)}$$

For the confirmatory calculation at a given wall thickness the following applies:

$$\overline{\sigma}_V = p \cdot \left(\frac{d_i}{2 \cdot s_{0n} \cdot \cos\varphi} + 0.5 \right) \leq S_m \quad \text{(A 2.4-4)}$$

For d_a and d_i the diameters at the large end of the area without discontinuity of the conical shell shall be taken in equations (A 2.4-2) to (A 2.4-4).

For d_a and d_i there is the relation:

$$d_i = d_a - 2 \cdot s_{0n} \cdot \cos\varphi \quad \text{(A 2.4-5)}$$

In the case of several consecutive conical shells with the same apex angle all shells shall be calculated in accordance with (A 2.4-2) or (A 2.4-3).

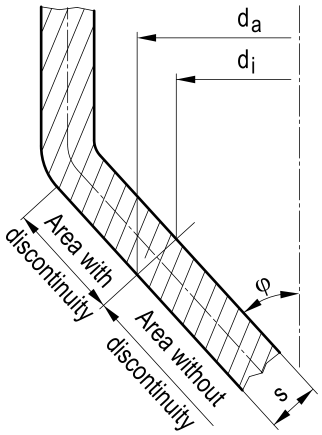


Figure A 2.4-2: Area of shell without discontinuity

(2) Calculation of wall thickness of the area with discontinuity of inwardly curved conical shells and $\varphi \leq 70°$

The wall thickness shall be dimensioned separately with respect to

a) circumferential loading in external knuckle portion,

b) circumferential loading in internal knuckle portion

and

c) loading along the generating line of shell section.

The largest wall thickness obtained from a), b) and c) shall govern the dimensioning.

Regarding the circumferential stress for inwardly curved transitions (**Figure A 2.4-1**) the required wall thickness shall be determined by means of equations (A 2.4-2) or (A 2.4-3) for both sides of the transition.

Regarding the longitudinal stresses the wall thickness can be obtained from:

$$s_0 = \frac{d_a \cdot p \cdot \beta}{4 \cdot S_m} \quad \text{(A 2.4-6)}$$

where the shape factor β shall be taken from **Table A 2.4-1** in dependence of the angle ψ and the ratio r/d_a. Intermediate values may be subject to straight interpolation.

The largest value obtained from equation (A 2.4-2) or (A 2.4-3) and (A 2.4-6) shall be decisive. For the confirmatory calculation at a given wall thickness the following applies:

$$\sigma_l = \frac{d_a \cdot p \cdot \beta}{4 \cdot s_{0n}} \leq S_m \quad \text{(A 2.4-7)}$$

The angle ψ is the absolute difference of half the apex angles φ_1 and φ_2:

$$\psi = |\varphi_1 - \varphi_2| \quad \text{(A 2.4-8)}$$

Where the wall thickness changes within the die-out length (e.g. forgings, profiles) the wall thickness at run-out of curvature shall govern the determination of the lengths e_1 and e_2 according to equation (A 2.4-1).

(3) Wall thickness calculation for the area without discontinuity of conical shells with outwardly curved transitions and $\varphi \leq 70°$

In the case of outwardly curved transitions (**Figure A 2.4-3**) basically all conditions and relationships apply as for inwardly curved transitions.

In addition, the following condition shall be satisfied due to the increased circumferential stress:

$$\sigma_V = p \cdot \left(\frac{A_p}{A_\sigma} + 0.5 \right) \leq S_m \quad \text{(A 2.4-9)}$$

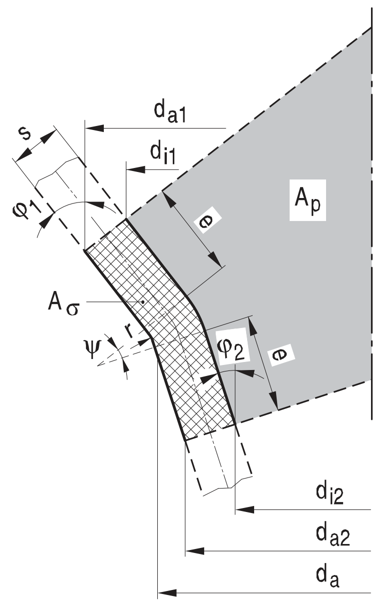


Figure A 2.4-3: Conical shell with outwardly curved transition

(4) Wall thickness calculation for the area with discontinuity of flat conical shells with knuckle and $\varphi > 70°$

In the case of extremely flat cones whose angle inclination to the vessel axis is $\varphi > 70°$, the wall thickness may be calculated in accordance with equation (A 2.4-10) even if a smaller wall thickness than that calculated according to equations (A 2.4-2), (A 2.4-3) or (A 2.4-6) is obtained:

$$s_0 = 0.3 \cdot (d_a - r) \frac{\varphi}{90°} \cdot \sqrt{\frac{p}{S_m}} \quad \text{(A 2.4-10)}$$

A 2.4.3 Conical shells under external pressure

For cones subject to external pressure the calculation shall be made in accordance with clause A 2.4.2.3. For $d_a > 50$ mm the allowanceg c_4 shall be added to the wall thickness obtained from equation (A 2.4-6).

$$c_4 = 2 \cdot \left(1 - \frac{50\ \text{mm}}{d_a} \right) \cdot \text{mm} \quad \text{(A 2.4-11)}$$

where for d_a a numerical value in mm shall be taken. For conical shells with φ not exceeding 45° it shall be additionally verified whether the cone is safe against elastic instability. This verification shall be made in accordance with clause A 2.2.3.3.2 in which case the cone shall be considered to be equal to a cylinder the diameter of which is determined as follows:

$$d_a = \frac{d_{a1} + d_{a2}}{2 \cdot \cos\varphi} \qquad \text{(A 2.4-12)}$$

where

d_{a1} diameter at large end of cone,

d_{a2} diameter at small end of cone.

The axial length of the cone and the adjacent cylindrical sections, if any, shall be taken unless the cylinder is sufficiently reinforced at the juncture in accordance with clause A 2.2.3.

Ψ	r/d_a											
	≤ 0.01	0.02	0.03	0.04	0.06	0.08	0.10	0.15	0.20	0.30	0.40	0.50
0	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
10	1.4	1.3	1.2	1.2	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
20	2.0	1.8	1.7	1.6	1.4	1.3	1.2	1.1	1.1	1.1	1.1	1.1
30	2.7	2.4	2.2	2.0	1.8	1.7	1.6	1.4	1.3	1.1	1.1	1.1
45	4.1	3.7	3.3	3.0	2.6	2.4	2.2	1.9	1.8	1.4	1.1	1.1
60	6.4	5.7	5.1	4.7	4.0	3.5	3.2	2.8	2.5	2.0	1.4	1.1
70	10.0	9.0	8.0	7.2	6.0	5.3	4.9	4.2	3.7	2.7	1.7	1.1
75	13.6	11.7	10.7	9.5	7.7	7.0	6.3	5.4	4.8	3.1	2.0	1.1

Table A 2.4-1: Shape factor β in dependence of the ratio r/d_a and ψ

A 2.5 Dished heads (domend ends)

A 2.5.1 Design values and units relating to Section A 2.5

Notation	Design value	Unit
d_a	outside diameter of dished head	mm
d_i	inside diameter of dished head	mm
d_{Ai}	inside diameter of opening	mm
f_k	safety factor against elastic instability	—
f_k'	safety factor against elastic instability at increased test pressure	—
h_1	height of cylindrical skirt	mm
h_2	height of dished head	mm
r	radius	mm
s_0	calculated wall thickness according to Figure 7.1-1	mm
s_{0n}	nominal wall thickness of the shell excluding allowances according to Section 6.5	mm
p	design pressure	MPa
p'	test pressure	MPa
p_B	elastic instability pressure	MPa
x	distance of weld to knuckle	mm
E	modulus of elasticity	N/mm²
R	radius of dishing	mm
S	design stress intensity according to Section Table 6.6-1	N/mm²
S_m	design stress intensity according to Section Table 6.6-1	N/mm²
β	shape factor	—

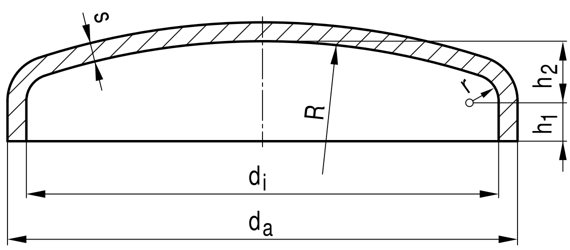


Figure A 2.5-1: Dished unpierced head

A 2.5.2 Dished heads under internal pressure

A 2.5.2.1 Scope

The calculation hereinafter applies to dished heads, i.e. torispherical, semi-ellipsoidal and hemispherical heads under internal pressure if the following relationships and limits are adhered to (see **Figure A 2.5-1**):

a) Torispherical heads

$R = d_a$

$r = 0.1 \cdot d_a$

$h_2 = 0.1935 \cdot d_a - 0.455 \cdot s_{0n}$

$$0.001 \le \frac{s_{0n}}{d_a} \le 0.1$$

b) Semi-ellipsoidal heads

$R = 0.8 \cdot d_a$

$r = 0.154 \cdot d_a$

$h_2 = 0.255 \cdot d_a - 0.635 \cdot s_{0n}$

$$0.001 \le \frac{s_{0n}}{d_a} \le 0.1$$

c) Hemispherical heads

$d_a/d_i \le 1.5$

d) Dished heads of valves

General conditions for dished heads

$R \le d_a$

$r \ge 0.1\ d_a$

$0.005 \le s_v/d_a \le 0.10$

A 2.5.2.2 General

(1) Height of cylindrical skirt

For torispherical heads the height of the cylindrical skirt shall basically be $h_1 \ge 3.5 \cdot s_{0n}$, and for semi-ellipsoidal heads

$h_1 \geq 3.0 \cdot s_{0n}$, however, need not exceed the following dimensions:

Wall thickness s_{0n}, mm	Height of cylindrical skirt h_1, mm
$s_{0n} \leq 50$	150
$50 < s_{0n} \leq 80$	120
$80 < s_{0n} \leq 100$	100
$100 < s_{0n} \leq 120$	75
$120 < s_{0n}$	50

For hemispherical heads no cylindrical skirt is required.

(2) Where a dished head is made of a crown section and a knuckle welded together the connecting weld shall have a sufficient distance x from the knuckle which shall be (see **Figure A 2.5-2**):

a) in case of differing wall thickness of crown section and knuckle:

$x = 0.5 \cdot \sqrt{R \cdot s_{0n}}$

where s_{0n} is the nominal wall thickness of the knuckle excluding allowances.

b) in case of same wall thickness of crown section and knuckle:

$x = 3.5 \cdot s_{0n}$ for torispherical heads,

$x = 3.0 \cdot s_{0n}$ for semi-ellipsoidal heads.

However, the distance l shall normally be at least 100 mm.

c) The determination of the transition from knuckle to crown section shall be based on the inside diameter. For thin-walled torispherical heads to DIN 28011 the transition shall be approximately $0.89 \cdot d_i$ and $0.86 \cdot d_i$ for thin-walled semi-ellipsoidal heads to DIN 28013. These factors are reduced with an increase in wall thickness.

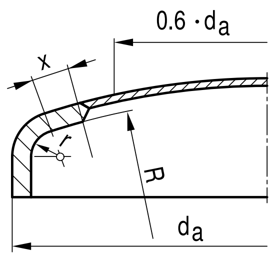


Figure A 2.5-2: Head with differing wall thickness of knuckle and crown section

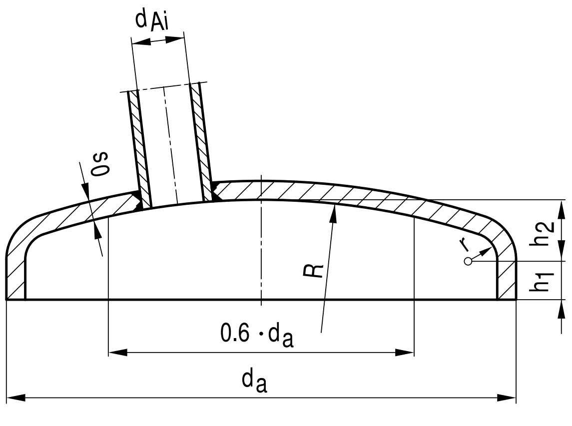


Figure A 2.5-3: Dished head with nozzle

A 2.5.2.3 Reinforcement of openings

(1) Openings in the crown section $0.6 \cdot d_a$ of torispherical and semi-ellipsoidal heads and over the full spherical portion of hemispherical heads shall be examined in accordance with Section A 2.8 for sufficient reinforcement without consideration of the ß values.

(2) Openings outside $0.6 \cdot d_a$ are taken into account by increasing the shape factor ß in accordance with **Figures A 2.5-4** and **A 2.5-5**, respectively.

(3) Openings in the knuckle area are only permitted in exceptional cases in which case dimensioning shall be based on an analysis of the mechanical behaviour.

(4) If the ligament between adjacent openings is not fully within $0.6 \cdot d_a$, this ligament width shall be at least equal to the same of one-half the opening diameters measured on a line between the centres of the openings.

A 2.5.2.4 Calculation

(1) Calculation of the required wall thickness of the knuckle under internal pressure

For the calculation of the required knuckle wall thickness the following applies:

$$s_0 = \frac{d_a \cdot p \cdot \beta}{4 \cdot S_m} \qquad \text{(A 2.5-1)}$$

The wall thickness of the crown section shall be determined in accordance with clause A 2.3.2.2.

(2) The shape factors ß for dished heads (unpierced and pierced heads) shall be taken

a) for torispherical heads from **Figure A 2.5-4**,

b) for semi-ellipsoidal heads from **Figure A 2.5-5**

in dependence of s_0/d_a and d_{Ai}/d_a.

The shape factors ß are based on the stress theory of von Mises.

Here, the curves with $d_{Ai}/d_a > 0$ apply to unreinforced openings for the full area of crown and knuckle.

For unpierced hemispherical heads a shape factor ß = 1.1 applies irrespective of the wall thickness over the distance

$$x = 0.5 \cdot \sqrt{R \cdot s_0} \qquad \text{(A 2.5-2)}$$

from the connecting weld.

(3) Valve-specific designs

For the valve-specific designs according to **Figures A 2.5-6** to **A 2.5-8** the weakening of the main shell may be compensated by the following measures:

a) by an increase of the wall thickness compared to the unpierced head, with an increased wall thickness to extend at least over the length $l_0 = \sqrt{(2 \cdot R_i + s_{V0}) \cdot s_{V0}}$ (see **Figure A 2.5-6**).

b) by tubular reinforcements without or with an increase in wall thickness of the main shell. Credit of an internal projection of a branch can only be taken with the portion of the length $l_{A2} \leq 0.5 \cdot l_1 \leq 0.5 \cdot \sqrt{(d_1 + s_{V1}) \cdot s_{V1}}$ to be contributing to the reinforcement. The wall thickness ratio s_{V1}/s_{V0} shall not to essentially exceed the value 1 (see **Figure A 2.5-7**).

c) by a flanged-out opening (extrusion) in connection with an increased wall thickness of the main shell. Where the pressure-loaded areas A_p and the effective cross-sectional areas A_σ are determined like for tubular reinforcements, i.e. without taking account of the extrusion radii and the cross-section losses, then for A_σ the value $A''_\sigma = 0.9 \cdot A_\sigma$ shall be used in the calculation (see **Figure A 2.5-8**).

A 2.5.3 Dished heads under external pressure

For the calculation of the required wall thickness of the knuckle under external pressure the requirements of clause A 2.5.2 with the additional requirements given hereinafter shall apply:

The required wall thickness s_0 of the knuckle shall be computed by means of equation (A 2.5-1). When computing the required wall thickness s_0 the allowable stress intensity S_m or S depending on the test group shall be reduced by 20 %.

In addition, it shall be verified that the head has been adequately dimensioned against elastic instability in the crown section.

This is the case if

$$p \leq 0.366 \cdot \frac{E}{f_k} \cdot \left(\frac{s_{0n}}{R}\right)^2 \qquad \text{(A 2.5-3)}$$

The safety factor f_k shall be taken from **Table A 2.5-1**. Where a test pressure in excess of p' = 1.3 · p is required, a separate verification of strength against elastic instability shall be made. In this case the safety factor f_k' at test pressure shall not be less than the value given in **Table A 2.5-1**.

$\frac{s_{0n}}{R}$	f_k	f_k'
0.001	5.5	4.0
0.003	4.0	2.9
0.005	3.7	2.7
0.010	3.5	2.6
0.1	3.0	2.2
Intermediate values shall be subject to straight interpolation.		

Table A 2.5-1: Safety factors against elastic instability under external pressure

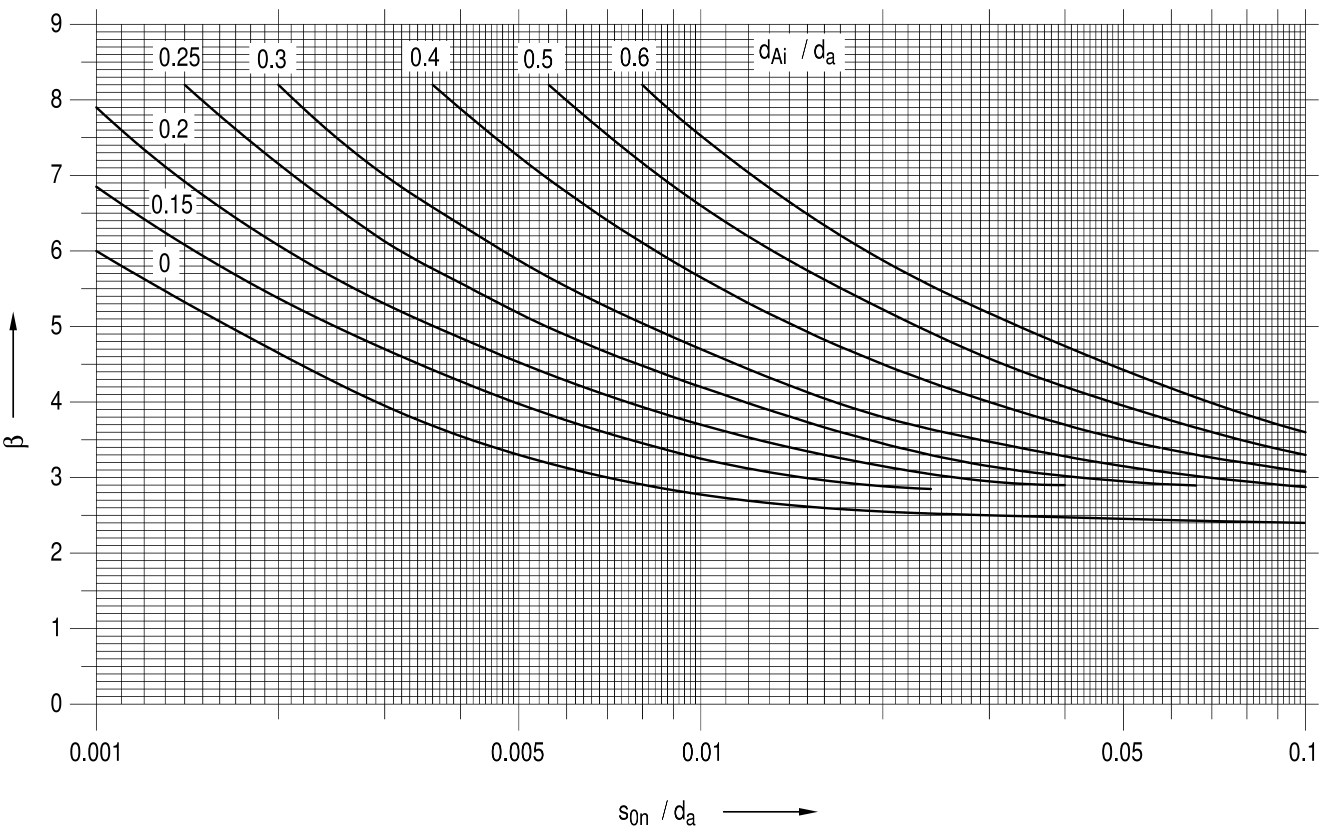


Figure A 2.5-4: Shape factors ß for torispherical heads

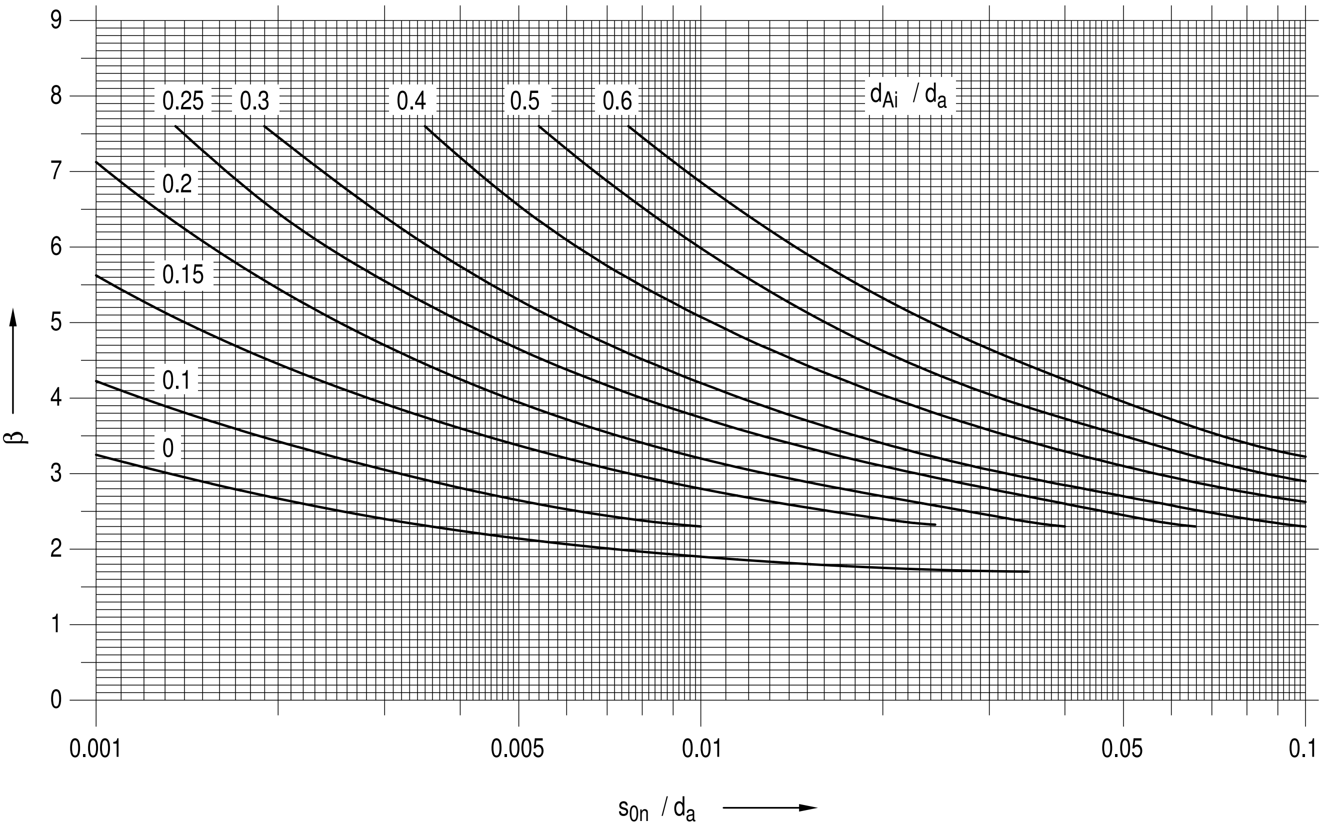


Figure A 2.5-5: Shape factors ß for semi-ellipsoidal heads

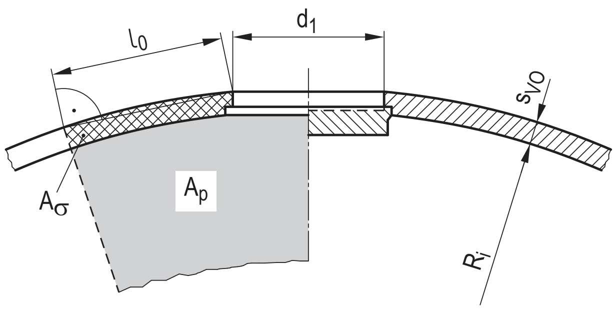


Figure A 2.5-6: Dished head with opening

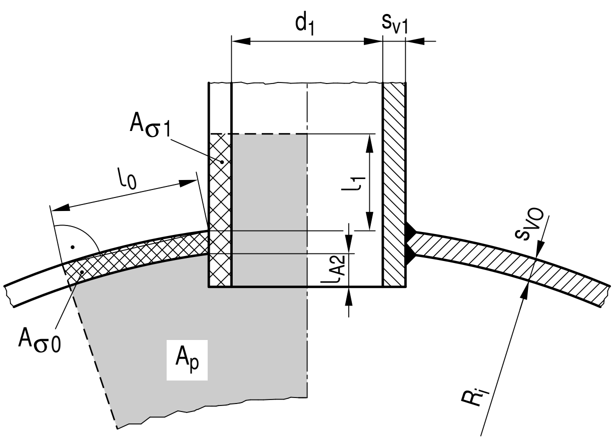


Figure A 2.5-7: Dished head with branch (welded-in tubular reinforcement)

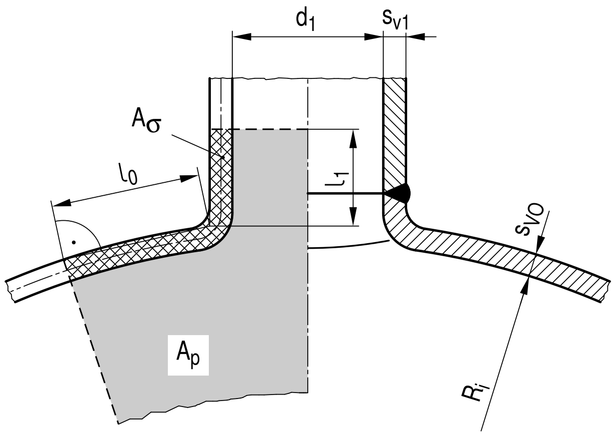


Figure A 2.5-8: Dished head with flanged-out opening

A 2.6 Spherically dished heads with bolting flanges

A 2.6.1 Design values and units relating to Section A 2.6

Notation	Design value	Unit
a_1. a_2. a_D a_F. a_H. a_S. a_V	lever arms in acc. with Figure A 2.6-1	mm
b	effective width of flange	mm
c_1	wall thickness allowance for consideration of fabrication tolerances	mm
c_2	wall thickness allowance for consideration of wall thickness reduction due to chemical or mechanical wear	mm
d_1	diameter at intersection of flange ring and spherical section	mm

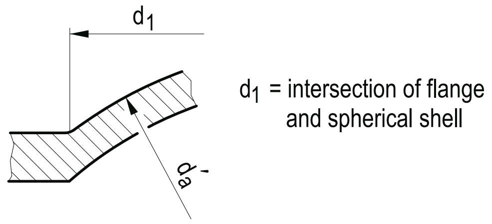
Notation	Design value	Unit
d_a	outside diameter of flange	mm
d_a'	outside diameter of spherical crown section	mm
d_D	mean diameter or diameter of gasket contact circle	mm
d_i	inside diameter of flange	mm
d_i'	inside diameter of spherical crown section	mm
d_L	bolt hole diameter	mm
d_L'	calculated diameter of bolt hole	mm
d_p	centroid of flange when subject to twisting	mm
d_t	bolt circle diameter	mm
h_F	thickness of flange ring	mm
r_a'	outside radius of curvature of spherical crown section	mm
r_i'	inside radius of curvature of spherical crown section	mm
s_n	nominal wall thickness	mm
s_0	wall thickness of spherical crown section	mm
F_D	compression load on gasket	N
F_{DB}	compression load on gasket to ensure tight joint (gasket load difference between design bolt load and total hydrostatic end force)	N
$F_{DBU/L}$	required gasket load at operating condition of floating type flanged joints	N
F_{DV}	gasket seating load	N
F_F	difference between total hydrostatic end force and the hydrostatic end force on area inside flange	N
F_H	horizontal force	N
F_S	bolt load	N
$F_{SBU/L}$	minimum value of bolt load at operating condition of floating type flanged joints	N
F_{S0}	bolt load for gasket seating condition	N
F_V	vertical force	N
M_a	moment of external forces	Nmm
M_{aB}	moment of external forces for operating conditions	Nmm
M_{a0}	moment of external forces for gasket seating condition	Nmm
M_b	bending moment	Nmm
M_t	torsional moment	Nmm
S_m	design stress intensity according to Section Table 6.6-1	N/mm^2
Q	transverse force	N
σ_{BO}	upper limit value of gasket bearing surface load for operating conditions	N/mm^2
σ_{VO}	upper limit value of gasket bearing surface load for gasket seating condition	N/mm^2
σ_{VU}	lower limit value of gasket bearing surface load for gasket seating conditions	N/mm^2
$\sigma_{VU/L}$	minimum gasket contact surface load at bolting-up condition acc. to Section A 2.11	N/mm^2
μ	friction factor	—

A 2.6.2 General

(1) Spherically dished heads with bolting flanges which e.g. are used to cover valve bodies consist of a shallow or deep-dished spherical shell and a bolting flange. Therefore, the strength calculation comprises the calculation of the flange ring and the spherical shell.

(2) According to the geometric relationships distinction is made between type I to **Figure A 2.6-1** as shallow-dished spherical shell (y > 0) and type II to **Figure A 2.6-2** as deep-dished spherical shell (y = 0).

Determination of d_1 :



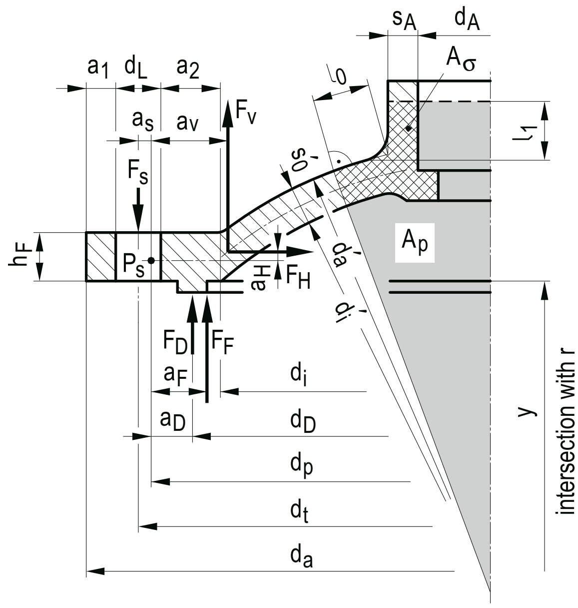


Figure A 2.6-1: Spherically dished head with shallow-dished spherical shell (type I, y > 0)

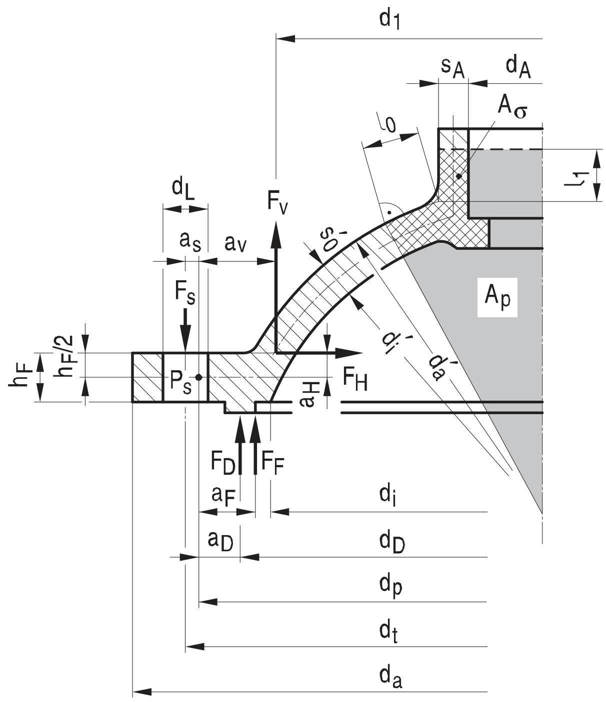


Figure A 2.6-2: Spherically dished head with deep-dished spherical shell (type II, y = 0)

A 2.6.3 Calculation of the flange ring

(1) The strength conditions for the flange ring are:

$$\frac{F_H}{2 \cdot \pi \cdot b \cdot h_f} \le S_m \qquad \text{(A 2.6-1)}$$

$$\frac{M_a}{2 \cdot \pi \cdot \left[\frac{b}{4} h_F{}^2 + \frac{d_1}{8}\left(s_e{}^2 - s_0{}^2\right)\right]} + \frac{F_H}{3\pi \cdot b \cdot h_F} \le S_m \qquad \text{(A 2.6-2)}$$

with

$s_e = s_n - c_1 - c_2$

The wall thickness s_0 of the spherical shell without allowances shall be, at a diameter ratio $d'_a / d'_i \le 1.2$, as follows:

$$s_0 = \frac{r'_i \cdot p}{2 \cdot S_m - p} \qquad \text{(A 2.6-3)}$$

or

$$s_0 = \frac{r'_a \cdot p}{2 \cdot S_m} \qquad \text{(A 2.6-4)}$$

with $d'_a = 2 \cdot r'_a$ and $d'_i = 2 \cdot r'_i$

For $1.2 \le d'_a / d'_i \le 1.5$ the following equations shall be used for calculating the wall thickness s_0 of the spherical shell:

$$s_0 = r'_i \cdot \left(\sqrt{1 + \frac{2p}{2 \cdot S_m - p}} - 1\right) \qquad \text{(A 2.6-5)}$$

$$s_0 = r'_a \cdot \frac{\sqrt{1 + \frac{2 \cdot p}{2 \cdot S_m - p}} - 1}{\sqrt{1 + \frac{2 \cdot p}{2 \cdot S_m - p}}} \qquad \text{(A 2.6-6)}$$

The equations (A 2.6-3) to (A 2.6-6) lead to the same results if $r'_i = r'_a - s_0$.

(2) The moment M_a resulting from external forces referred to the centroid of flange P_S shall be for the operating condition:

$$M_{aB} = F_S \cdot a_S + \left(F_V + F_{ax} + \frac{4 \cdot M_B}{d_1}\right) \cdot a_V + F_F \cdot a_F + F_D \cdot a_D + F_H \cdot a_H \qquad \text{(A 2.6-7)}$$

The compression load F_D on the gasket, in the case of application of a transverse force due to friction at a certain value shall be determined by:

$$F_D = \max\left(\frac{Q}{\mu} + \frac{2 \cdot M_t}{\mu \cdot d_D} - \frac{2 \cdot M_b}{d_D}; F_{DB}\right) \qquad \text{(A 2.6-8)}$$

The compression load F_D on the gasket, in the case of application of a transverse force due to infinite friction shall be determined by:

$$F_D = \max\left[\frac{2 \cdot M_t}{\mu \cdot d_D} - \max\left(\frac{2 \cdot M_b}{d_D}; \frac{4 \cdot M_b}{d_t}\right); F_{DB}\right] \qquad \text{(A 2.6-9)}$$

The moment M_a for the bolting-up condition shall be:

$$M_{a0} = F_{S0}\,(a_S + a_D) \qquad \text{(A 2.6-10)}$$

The moments applied clockwise shall be inserted with negative signs in equations (A 2.6-7) and (A 2.6-10). The strength condition in equation (A 2.6-2) shall be calculated with both moments M_{aB} and M_{a0} where for the bolting-up condition $s_0 = 0$ shall be taken.

(3) The forces are obtained from the following equations:

a) Operating bolt load

$$F_{SBU/L} = F_V + F_F + F_{DBU/L} \cdot S_D + F_{ax} + \frac{4 \cdot M_b}{d_D} + F_Z \qquad \text{(A 2.6-11)}$$

For S_D a value of at least 1.2 shall be taken.

In the verification of bolt stresses the bolt circle diameter d_t may be used instead of the gasket diameter d_D.

In the case of application of a transverse force due to friction at a certain value F_Z shall be determined by:

$$F_Z = \max\left\{0; \frac{Q}{\mu} + \frac{2 \cdot M_t}{\mu \cdot d_D} - F_{DBU/L} - \frac{2 \cdot M_b}{d_D}\right\} \qquad \text{(A 2.6-12)}$$

In the case of application of a transverse force due to infinite friction F_Z shall be determined by:

$$F_Z = \max\left\{0; \frac{2 \cdot M_t}{\mu \cdot d_D} - F_{DBU/L} - \max\left(\frac{2 \cdot M_b}{d_D}; \frac{4 \cdot M_b}{d_t}\right)\right\} \qquad \text{(A 2.6-13)}$$

b) Vertical component of force on head

$$F_V = p \cdot \frac{\pi}{4} \cdot d_i^2 \qquad \text{(A 2.6-14)}$$

c) Difference between total hydrostatic end force and the hydrostatic end force on area inside flange

$$F_F = p \cdot \frac{\pi}{4} \cdot \left(d_D{}^2 - d_i{}^2\right) \qquad \text{(A 2.6-15)}$$

d) gasket load at operating condition

$$F_{DBU/L} = \pi \cdot d_D \cdot b_D \cdot \sigma_{BU/L} \qquad \text{(A 2.6-16)}$$

The allowable (maximum bearable) gasket load reaction at operating condition shall be:

$F_{DBO} = \pi \cdot d_D \cdot b_D \cdot \sigma_{BO}$

with

b_D, $\sigma_{BU/L}$ and σ_{BO} acc. to Section A 2.11.

e) Horizontal component of force on head

$$F_H = p \cdot \frac{\pi}{2} \cdot d_1 \cdot \sqrt{r^2 - \frac{d_i^2}{4}} \qquad \text{(A 2.6-17)}$$

with

$$r = \frac{d'_i}{2}$$

For the gasket seating condition the following bolt load F_{S0U} applies:

$$F_{SOU} = \max.\ \left\{F_{DVU/L} \cdot S_D; F_{SBU/L} \cdot 1.1\right\} \qquad \text{(A 2.6-18)}$$

with

$F_{DVU} = \pi \cdot d_D \cdot b_D \cdot \sigma_{VU/L}$

S_D at least 1.2

In the gasket seating condition the gasket shall be loaded with a maximum of:

$F_{DVO} = \pi \cdot d_D \cdot b_D \cdot \sigma_{VO}$

$\sigma_{VU/L}$ and σ_{VO} acc. to Section A 2.11.

(4) The lever arms of the forces in the equations (A 2.6-7) and (A 2.6-10) used for determining the moments are obtained from **Table A 2.6-1**.

Lever arm	Spherically dished head	
	Type I	Type II
a_S	$0.5\ (d_t - d_p)$	
a_V	$0.5\ (d_p - d_1)$	
a_D	$0.5\ (d_p - d_D)$	
a_H	determine graphically	$0.5 \cdot h_F$
a_F	$a_D + 0.5\ (d_D - d_i)$	

Table A 2.6-1: Lever arms for equations (A 2.6-7) and (A 2.6-10)

(5) The effective width of the flange shall be:

$$b = 0.5 \cdot (d_a - d_i - 2 \cdot d'_L) \qquad \text{(A 2.6-19)}$$

with $d'_L = v \cdot d_L$

For inside diameters d_i equal to or greater than 500 mm v = 0.5 and for d_i less than 500 mm $v = 1 - 0.001 \cdot d_i$ (d_i in mm).

(6) The centroid of flange d_p is obtained from:

$$d_p = d_a - 2 \cdot S_a \qquad \text{(A 2.6-20)}$$

with

$$S_a = \frac{0.5 \cdot a_1^2 + a_2 \cdot (a_1 + d_L + 0.5 \cdot a_2)}{a_1 + a_2} \qquad \text{(A 2.6-21)}$$

and

$$a_1 = 0.5 \cdot (d_a - d_t - d_L) \qquad \text{(A 2.6-22)}$$

$$a_2 = 0.5 \cdot (d_t - d_i - d_L) \qquad \text{(A 2.6-23)}$$

A 2.6.4 Calculation of the spherical shell under internal pressure

The calculation shall be made in accordance with clause A 2.3.4.

A 2.7 Flat plates

A 2.7.1 Design values and units relating to Section A 2.7

Notation	Design value	Unit
a_D	gasket moment arm	mm
d_A	opening diameter	mm
d_D	mean diameter or diameter of gasket contact face	mm
d_i	inside diameter	mm
d_t	bolt circle diameter	mm
p	internal pressure	MPa
S_m	design stress intensity according to Section Table 6.6-1	N/mm^2
s_0	calculated wall thickness	mm
s_{0n}	nominal plate wall thickness minus allowances	mm
s_{RO}	required plate thickness at edge	mm
E	modulus of elasticity	N/mm^2
F_D	maximum gasket seating stress in consideration of the unequal distribution of bolt loads	N

The design values and further notations will be explained with the pertinent equations.

A 2.7.2 Scope

The calculation rules given hereinafter apply to flat plates with and without edge moment under pressure load for the range

$$0.543 \cdot \sqrt[4]{\frac{p}{E}} \le \frac{s_{0n,Pl}}{d_i} \le \frac{1}{3}$$

A 2.7.3 Calculation

A 2.7.3.1 Circular flat plate integral with cylindrical section

(1) In case of a plate integral with a cylindrical section as shown in **Figure A 2.7-1** the plate and cylinder shall be considered a unit.

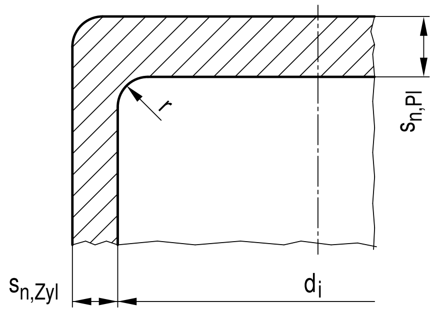


Figure A 2.7-1: Flat plate integral with cylindrical section (for design see Figure 5.2-7)

(2) According to footnote 1) of **Table 7.7-1** there are two possibilities of dimensioning the juncture between flat plate/cylindrical shell.

Note:

Compared to alternative 1, alternative 2 allows for thinner flat plates at greater wall thickness of the cylindrical shell.

a) Alternative 1:

Predimensioning of the plate

$$s_{0,Pl} = 0.45 \cdot d_i \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-1)}$$

Predimensioning of the cylindrical shell in accordance with Section A 2.2.

Check of stresses in cylindrical shell:

$$3 \cdot S_m \ge p \cdot \left[6 \cdot B_1^2 \cdot \frac{0.82 + 0.85 \cdot \frac{B_2}{B_3^2} \cdot \sqrt{B_1}}{6.56 + 3.31 \cdot \frac{B_2^2}{B_3} \cdot \sqrt{B_1}} + \frac{1}{2} \cdot B_1 + 1 \right] \qquad \text{(A 2.7-2)}$$

$$\text{with} \quad B_1 = \frac{d_i + s_{0n,Zyl}}{2 \cdot s_{0n,Zyl}} \qquad \text{(A 2.7-3)}$$

$$B_2 = \frac{s_{0n,Pl}}{s_{0n,Zyl}} \qquad \text{(A 2.7-4)}$$

$$B_3 = \frac{d_i + s_{0n,Zyl}}{2 \cdot s_{0n,Pl}} \qquad \text{(A 2.7-5)}$$

b) Alternative 2:

Predimensioning of the plate

$$s_{0,Pl} = \left(0.45 - 0.1 \cdot \frac{s_{0n,Zyl}}{s_{0n,Pl}}\right) \cdot d_i \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-6)}$$

Predimensioning of the cylinder in accordance with Section A 2.2.

Check of stresses in the cylinder:

$$1.5 \cdot S_m \ge p \cdot \left[6 \cdot B_1^2 \cdot \frac{0.82 + 0.85 \cdot \frac{B_2}{B_3^2} \cdot \sqrt{B_1}}{6.56 + 3.31 \cdot \frac{B_2^2}{B_3} \cdot \sqrt{B_1}} + \frac{1}{2} \cdot B_1 + 1 \right] \qquad \text{(A 2.7-7)}$$

$$\text{with} \quad B_1 = \frac{d_i + s_{0n,Zyl}}{2 \cdot s_{0n,Zyl}} \qquad \text{(A 2.7-8)}$$

$$B_2 = \frac{s_{0n,Pl}}{s_{0n,Zyl}} \qquad \text{(A 2.7-9)}$$

$$B_3 = \frac{d_i + s_{0n,Zyl}}{2 \cdot s_{0n,Pl}} \qquad \text{(A 2.7-10)}$$

For both alternatives it may be required to increase the wall thicknesses obtained from predimensioning for plate and cylindrical shell and to repeat the check of the stresses in the shell at the transition to the plate in accordance with equation (A 2.7-2) or (A 2.7-7).

(3) Openings in flat plates as shown in **Figure A 2.7-1** shall be reinforced in accordance with clause A 2.8.2.3. Alternatively, the procedure of clause A 3.3.5 with equation (A 3.3-4) may be used.

(4) Off-centre openings may be treated like central openings.

(5) At a diameter ratio $d_A/d_i > 0.7$ the plate shall be calculated as flange in accordance with Section A 2.10.

A 2.7.3.2 Unstayed circular plates with additional edge moment

(1) For flat plates provided with a gasket and bolted at the edge the deformation shall also be taken into account by using equation (A 2.7-11) in addition to the strength calculation in accordance with equation (A 2.7-14), so that the tightness of the joint is ensured in which case the bolting-up, test and operating conditions shall be considered.

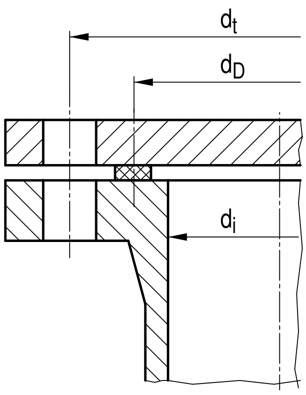


Figure A 2.7-2: Circular flat plate with additional edge moment

(2) The required wall thickness s_0 of unstayed flat circular plates with additional edge moment in same direction in accordance with **Figure A 2.7-2** will be:

$$s_0 = C \cdot d_D \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-11)}$$

The C value shall be taken from **Figure A 2.7-3** in dependence of the ratio d_t/d_D and the δ value in which case the ratio of the required bolt load to hydrostatic end force on inside of flange is

$$\delta = 1 + 4 \cdot \frac{\frac{\sigma_{BU/L}}{p} \cdot b_D \cdot S_D}{d_D} \qquad \text{(A 2.7-12)}$$

where, as a rule S_D = 1.2 is inserted. b_D is the gasket width according to Section A 2.11.

The equation given hereinafter leads to the same C value as **Figure A 2.7-3**:

$$C = \left\{ 0.063 \cdot \left(\frac{0.7}{d_t / d_D} + 2.6 \right) + 0.125 \cdot \delta \cdot \left[0.7 \cdot \left(1 - \frac{1}{d_t / d_D} \right) + 2.6 \cdot \ln(d_t / d_D) \right] \right\}^{1/2} \qquad \text{(A 2.7-13)}$$

The deflection of the plate with wall thickness s_0 in accordance with equation (A 2.7-11) should be checked with respect to the tightness requirements by use of equation (A 2.7-14).

Where the deflection is limited e.g. to $w = 0.001 \cdot d_D$, $x = 0.001$ shall be inserted in equation (A 2.7-14).

$$s_0 \geq \sqrt[3]{\frac{0.0435 \cdot p \cdot d_D^3}{x \cdot E} + \frac{1.05 \cdot F_D \cdot a_D}{\pi \cdot x \cdot E}} \qquad \text{(A 2.7-14)}$$

with the compression load on gasket F_D according to Section A 2.9 and the gasket moment arm

$$a_D = \frac{d_1 - d_D}{2} \qquad \text{(A 2.7-15)}$$

A 2.7.3.3 Unstayed circular plate with opening and additional edge moment

The equations hereinafter apply to the inside pressure p and an edge moment acting in the same direction and resulting from the pertinent bold load. The equations shall apply to the design and test condition. The maximum value of the required plate thickness s_0 obtained shall govern dimensioning.

The required wall thickness shall be:

$$s_0 = C \cdot C_{A1} \cdot d_D \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-16)}$$

with

C	factor as given in Figure A 2.7-3	—
C_{A1}	factor relating to opening as given in Figure A 2.7-4	—
d_D	gasket diameter	mm
p	internal pressure, either use p_A (design pressure) or p_P (test pressure)	MPa

The required wall thickness at the edge then is:

$$s_{R0} = 0.7 \cdot s_0 \qquad \text{(A 2.7-17)}$$

The factor C shall be taken from **Figure A 2.7-3** in dependence of d_t/d_D and

$$\delta = 1 + 4 \cdot \frac{\frac{\sigma_{BU/L}}{p} \cdot b_D \cdot S_D}{d_D} \qquad \text{(A 2.7-18)}$$

with

b_D	gasket width according to Section A 2.11	mm
S_D	safety factor for the gasket, either as SDP = 1.0 for the test condition (Level P) or as SDA = 1.2 for the design condition (Level 0)	—
d_t	bolt circle diameter	mm
d_D	gasket diameter	mm
$\sigma_{BU/L}$	minimum gasket contact surface load at operating condition acc. to Section A 2.11	N/mm^2

The factor C_{A1} is derived from **Figure A 2.7-4** in dependence of d_i/d_D and d_t/d_D as well as the designs with or without tubular reinforcement.

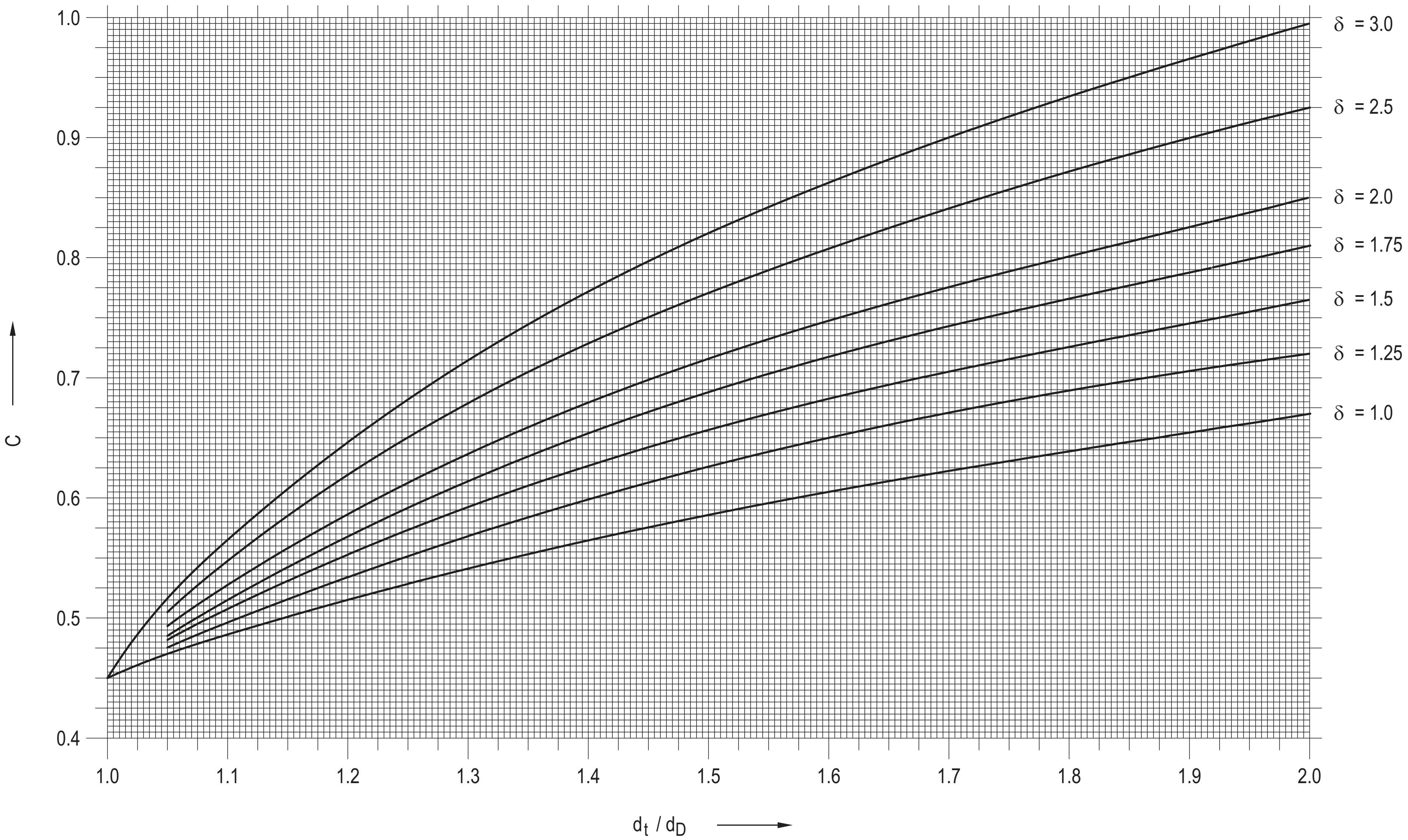


Figure A 2.7-3: Factor C of flat circular plates with additional edge moment acting in same direction (acc. to equation A 2.7-13)

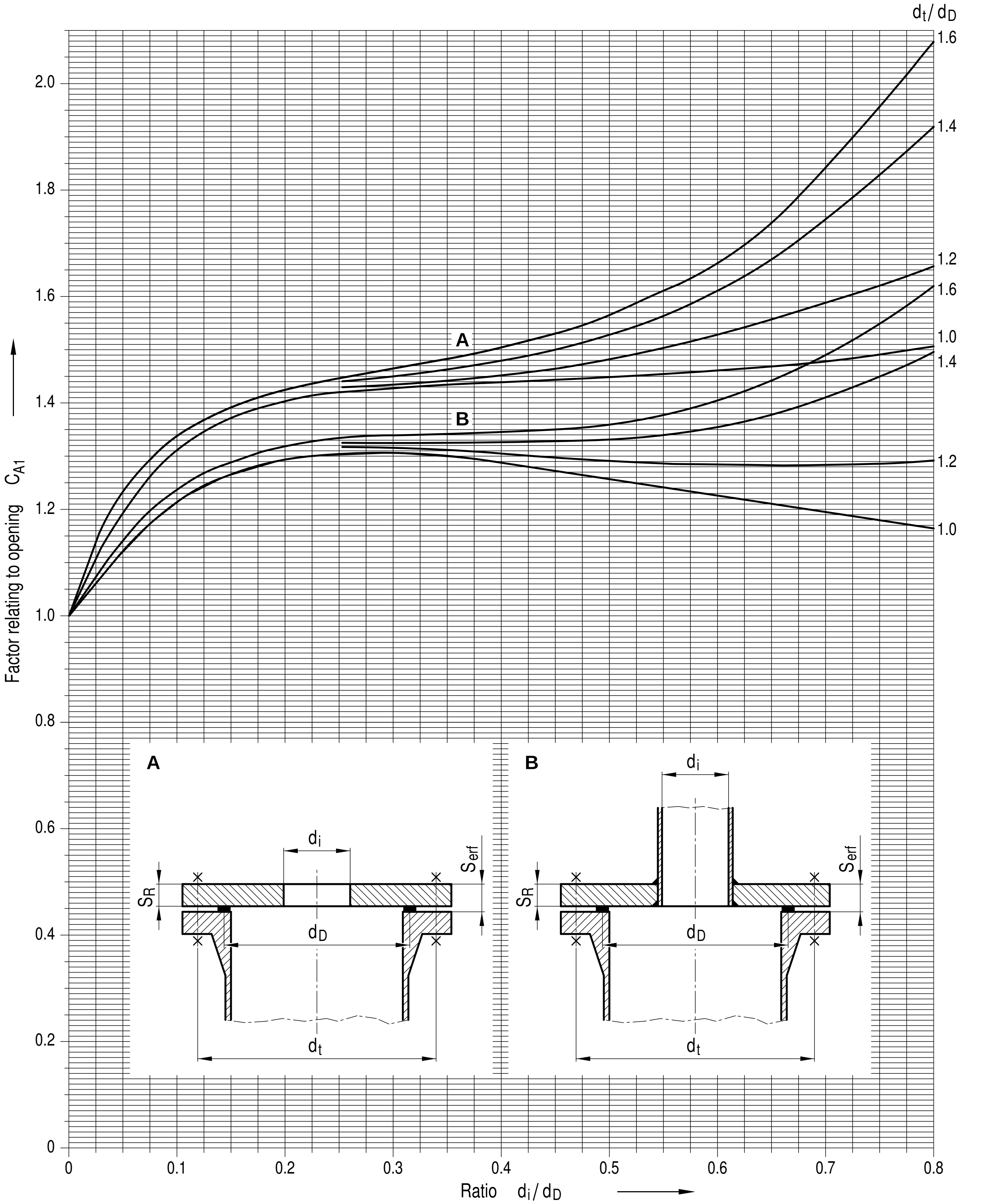


Figure A 2.7-4: Factor C_{A1} for unstayed circular plates with opening and additional edge moment

A 2.7.3.4 Inside circular plate with opening

The equation hereinafter applies to the internal pressure p and shall be applied to the design and test condition. The maximum value of the required plate thickness s_0 obtained shall govern dimensioning.

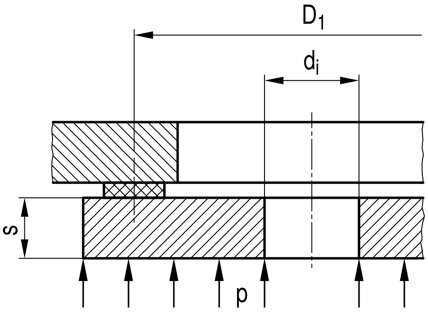


Figure A 2.7-5: Inside circular plate with opening

The required wall thickness shall be:

$$s_0 = C \cdot C_A \cdot D_1 \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-19)}$$

with

C	factor = 0.4	—
C_A	factor relating to opening as given in Figure A 2.7-4	—
D_1	gasket diameter	mm
p	internal pressure, either use p_A (design pressure) or p_P (test pressure)	MPa

Alternately, the factor C_A may be determined from **Figure A 2.7-6** in dependence of d_i/D_1 for case A.

with

d_i	inside diameter of opening	mm
D_1	diameter of plate	mm

A 2.7.3.5 Bolted rectangular and elliptical plates with opening

The equations hereinafter apply to internal pressure p and an edge moment acting in the same direction and resulting from the pertinent bolt load. The equations shall apply to the design and test condition. The maximum value of the required plate thickness s_0 obtained shall govern dimensioning.

The required wall thickness shall be:

$$s_0 = C \cdot C_{A1} \cdot C_E \cdot f \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 2.7-20)}$$

with

C	factor as given in Figure A 2.7-3	—
C_{A1}	factor relating to opening as given in Figure A 2.7-4	—
C_E	factor as given in Figure A 2.7-7	—
f	smallest gasket diameter	mm
e	greatest gasket diameter	mm
p	internal pressure, either use p_A (design pressure) or p_P (test pressure)	MPa

The required wall thickness at the edge then is:

$$s_{R0} = 0.7 \cdot s_0 \qquad \text{(A 2.7-21)}$$

The factor C shall be taken from **Figure A 2.7-3** in dependence of d_t/d_D and

$$\delta = 1 + 4 \cdot \frac{\frac{\sigma_{VU/L}}{p} \cdot b_D \cdot S_D}{f} \qquad \text{(A 2.7-22)}$$

with

b_D	gasket width according to Section A 2.11	mm
S_D	safety factor for the gasket, either as SDP = 1.0 for the test condition (Level P) or as SDA = 1.2 for the design condition level 0)	—
$\sigma_{VU/L}$	minimum gasket contact surface load at bolting-up condition acc. to Section A 2.11	N/mm^2
d_t	smallest inscribed bolt circle diameter	mm

The factor C_{A1} is derived from **Figure A 2.7-4** in dependence of d_i/f in lieu of D_i/d_D and d_t/f in lieu of d_t/d_D as well as the designs with or without tubular reinforcement.

For rectangular and elliptical plates the factor C_E may be determined by means of **Figure A 2.7-7** in dependence of the rectangular or elliptical gasket.

A 2.7.3.6 Clamped flat heads

The equations hereinafter apply to the internal pressure p and shall be applied to the design and test condition. The greatest stress occurring is a radial bending stress at the edge:

$$\sigma_r = \frac{3 \cdot p \cdot r_a^2}{4 \cdot s_0^2} \leq 1.5 \cdot S_m \qquad \text{(A 2.7-23)}$$

The maximum value obtained of the required plate thickness s_0 shall govern dimensioning:

$$s_0 = \frac{r_a}{2} \cdot \sqrt{\frac{3 \cdot p}{1.5 \cdot S_m}} \qquad \text{(A 2.7-24)}$$

with

r_a	radius of plate	mm
p	internal pressure, either use p_A (design pressure) or p_P (test pressure)	N/mm^2

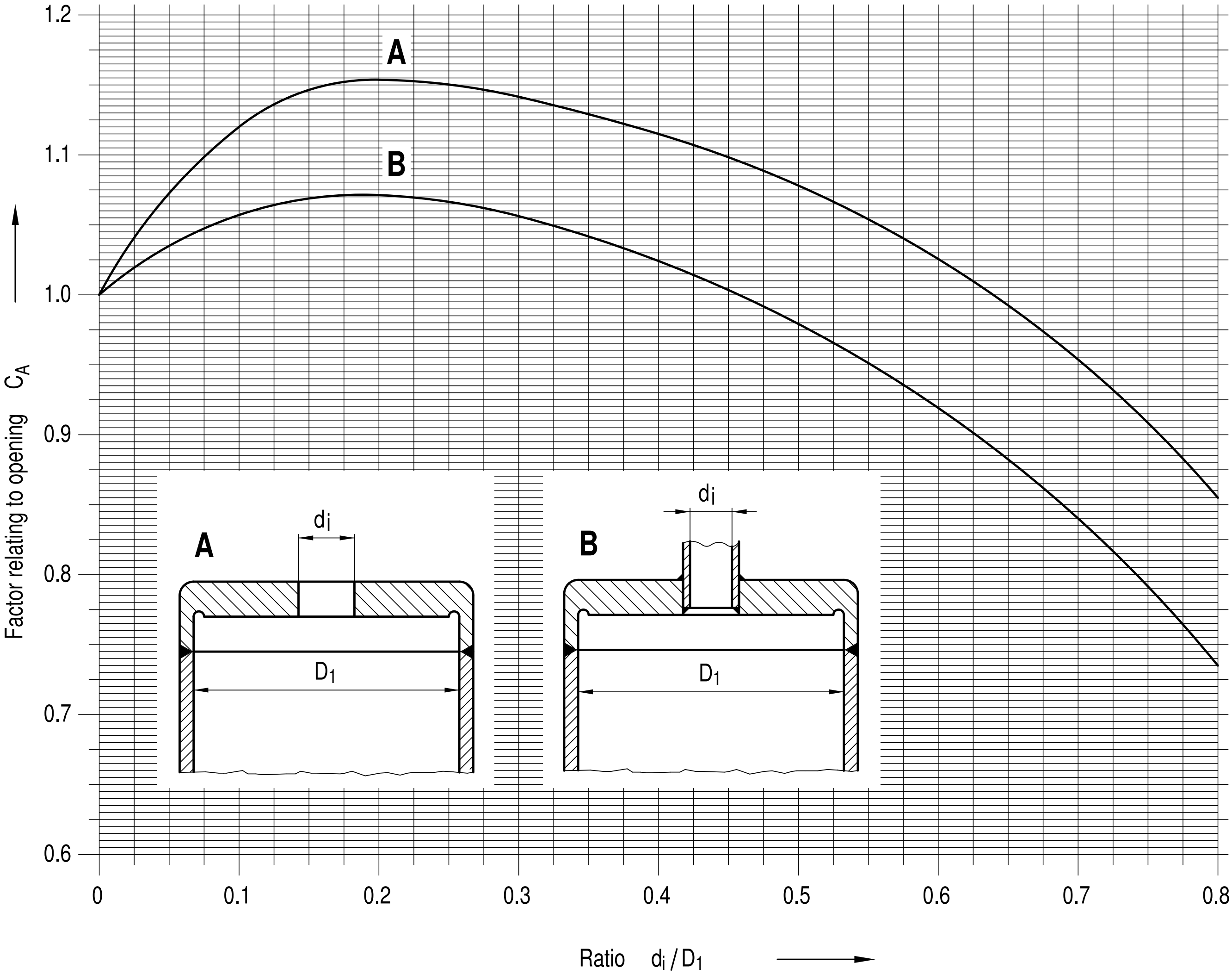


Figure A 2.7-6: Factor C_A relating to opening for stayed circular plate with opening

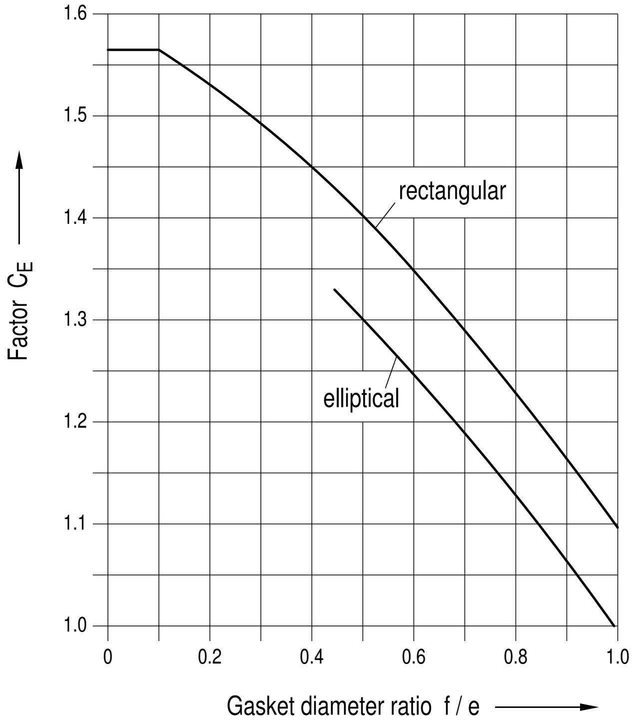


Figure A 2.7-7: Factor C_E for rectangular and elliptical plates

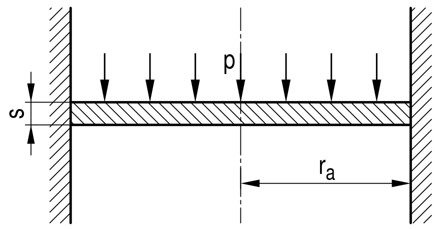


Figure A 2.7-8: Clamped flat head

A 2.7.3.7 Body cover plate with opening for stem and reinforcement of opening, for valves

(1) The cover plate shown in **Figure A 2.7-9** is shown as simply supported circular plate with an outside diameter d_t and an inside diameter d_i. The inside edge is assumed to be guided.

(2) The loading due to internal pressure p is determined by means of the superposition in accordance with **Figure A 2.7-10**. In addition, the loads $F_{s2} = F_{sp} + F_E$ and the compression load on the gasket F_D are applied which are assumed to be uniformly distributed over circular rings.

(3) The stress due to internal pressure is determined by:

$$\sigma_p = \frac{6}{s^2} \cdot M_p \qquad \text{(A 2.7-25)}$$

From the superposition as shown in **Figure A 2.7-10** follows

$$M_p = M_{d_t} - M_{(d_t - d_D)} \quad \text{(A 2.7-26)}$$

Note:

The equations given hereinafter for M_{d_t} and $M_{(d_t-d_D)}$ were taken from "Formulas for Stress and Strain", 5th edition, R.J. Roark, W.C. Young, McGraw-Hill, New York, 1975, Case 2 on p. 339.

where

$$M_{d_t} = \frac{p \cdot d_t^2}{4 \cdot C_8} \cdot L'_{17} \quad \text{(A 2.7-27)}$$

$$M_{(d_t-d_D)} = \frac{p \cdot d_t^2}{4 \cdot C_8} \cdot L''_{17} \quad \text{(A 2.7-28)}$$

$$C_8 = \frac{1}{2} \cdot \left[1 + \nu + (1-\nu) \cdot \left(\frac{d_i}{d_t} \right)^2 \right] \quad \text{(A 2.7-29)}$$

$$L'_{17} = \frac{1}{4} \cdot \left(1 - \frac{1-\nu}{4} \cdot \left[1 - \left(\frac{d_i}{d_t} \right)^4 \right] - \left(\frac{d_i}{d_t} \right)^2 \cdot \left[1 + (1+\nu) \cdot \ln \frac{d_t}{d_i} \right] \right) \quad \text{(A 2.7-30)}$$

$$L''_{17} = \frac{1}{4} \cdot \left(1 - \frac{1-\nu}{4} \cdot \left[1 - \left(\frac{d_D}{d_t} \right)^4 \right] - \left(\frac{d_D}{d_t} \right)^2 \cdot \left[1 + (1+\nu) \cdot \ln \frac{d_t}{d_D} \right] \right) \quad \text{(A 2.7-31)}$$

where

d_t	bolt circle diameter	mm
d_d	gasket diameter	mm
d_i	inside diameter for opening for stem	mm
p	internal pressure, either use as p_A (design pressure) or p_P (test pressure)	MPa
ν	Poisson's ratio	—

(4) The stresses due to the force F_{s2} applied on the circular ring with the diameter k and of the force F_D applied on the circular ring with the diameter d_D were determined as follows:

$$\sigma_{F_{s2}} = \frac{6}{s^2} \cdot M_K \quad \text{(A 2.7-32)}$$

$$\sigma_{F_D} = \frac{6}{s^2} \cdot M_{d_D} \quad \text{(A 2.7-33)}$$

where

Note:

The equations given hereinafter for M_K and M_{d_D} were taken from "Formulas for Stress and Strain", 5th edition, R. J. Roark, W.C. Young, McGraw-Hill, New York 1975, Case 1b, p. 335.

$$M_K = \frac{F_{s2} \cdot d_t \cdot L'_9}{2 \cdot \pi \cdot k \cdot C_8} \quad \text{(A 2.7-34)}$$

$$L'_9 = \frac{k}{d_t} \cdot \left(\frac{1+\nu}{2} \cdot \ln \frac{d_t}{k} + \frac{1-\nu}{4} \cdot \left[1 - \left(\frac{k}{d_t} \right)^2 \right] \right)$$

$$M_{d_D} = \frac{F_D \cdot d_t \cdot L'_9}{2 \cdot \pi \cdot d_D \cdot C_8} \quad \text{(A 2.7-35)}$$

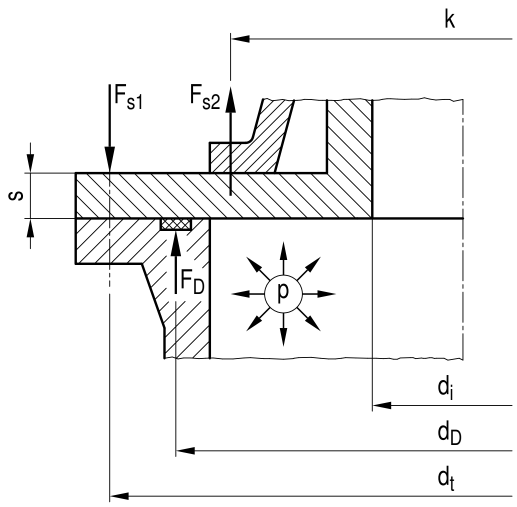
$$L'_9 = \frac{d_D}{d_t} \cdot \left(\frac{1+\nu}{2} \cdot \ln \frac{d_t}{d_D} + \frac{1-\nu}{4} \cdot \left[1 - \left(\frac{d_D}{d_t} \right)^2 \right] \right)$$

where

F_{s2}	resulting force from F_{sp} (actuating force) and F_E (external force, e.g. resulting from earthquake)	N
F_D	compression load on gasket (= F_{DB})	N
d_t	bolt circle diameter	mm
d_d	gasket diameter	mm
k	diameter subject to application of external forces	mm
ν	Poisson's ratio	—

(5) The strength condition for the body cover is as follows:

$$\sigma_{ges} = \sigma_p + \sigma_F + \sigma_{FD} \leq 1.5\, S_m \quad \text{(A 2.7-36)}$$



$F_{s1} = F_D + F_p + F_{s2}$

$F_{s2} = F_{sp} + F_E$

Figure A 2.7-9: Body cover with opening for stem and reinforcement of opening

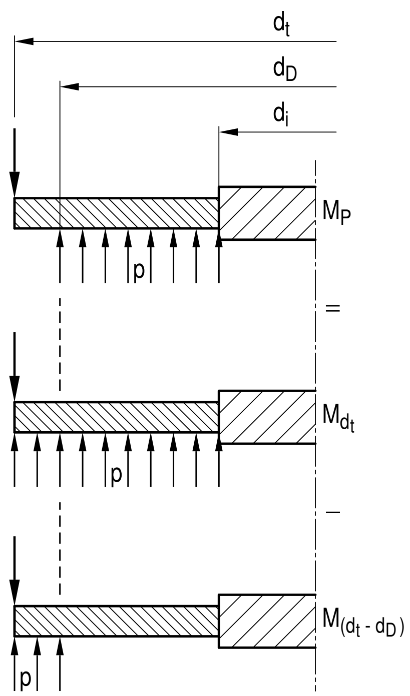
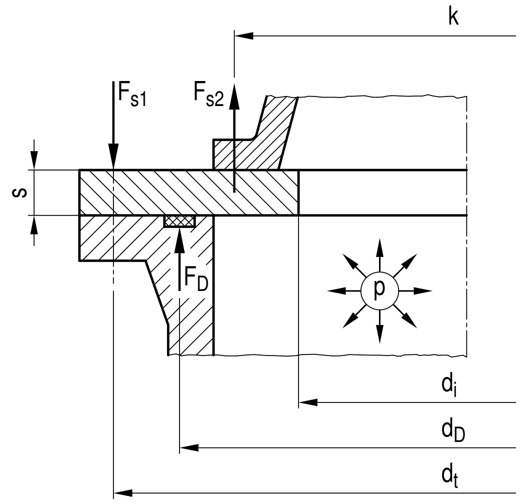


Figure A 2.7-10: Superposition for internal pressure

A 2.7.3.8 Body covers with opening for stem, without tubular reinforcement of opening

(1) The cover plate shown in **Figure A 2.7-11** is shown as simply supported circular plate with an outside diameter d_t and an inside diameter d_i. The inside edge is assumed to be guided.



$F_{s1} = F_D + F_p + F_{s2}$

$F_{s2} = F_{sp} + F_E$

Figure A 2.7-11: Body cover with opening for stem, without reinforcement of opening

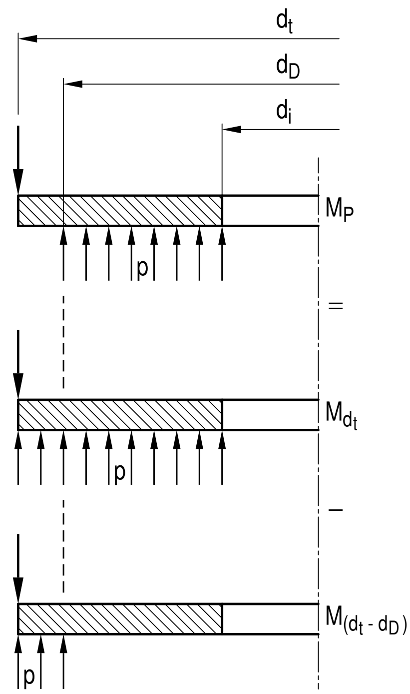


Figure A 2.7-12: Superposition for internal pressure

(2) The loading due to internal pressure p is determined by means of the superposition in accordance with **Figure A 2.7-12**. In addition, the loads $F_{s2} = F_{sp} + F_E$ and the compression load on the gasket F_D are applied which are assumed to be uniformly distributed over circular rings.

(3) The stress due to internal pressure is determined by:

$$\sigma_p = \frac{6}{s^2} \cdot M_p \qquad \text{(A 2.7-37)}$$

From the superposition as shown in **Figure A 2.7-12** follows

$$M_p = M_{d_t} - M_{(d_t - d_D)} \qquad \text{(A 2.7-38)}$$

Note:

The equations given hereinafter for

M_{d_t} *and* $M_{(d_t-d_D)}$

were taken from "Formulas for Stress and Strain", 5th edition, R.J. Roark, W.C. Young, McGraw-Hill, New York, 1975, Case 2a on p. 339.

$$M_{d_t} = \frac{p \cdot d_t^3 \cdot \left(1-\nu^2\right)}{4 \cdot d_i \cdot C_7} \cdot L'_{17} \qquad \text{(A 2.7-39)}$$

$$M_{(d_t-d_D)} = \frac{p \cdot d_t^3 \cdot \left(1-\nu^2\right)}{4 \cdot d_i \cdot C_7} \cdot L'_{17} \qquad \text{(A 2.7-40)}$$

$$C_7 = \frac{1}{2} \cdot \left(1-\nu^2\right) \cdot \left(\frac{d_t}{d_i} - \frac{d_i}{d_t}\right) \qquad \text{(A 2.7-41)}$$

$$L'_{17} = \frac{1}{4} \cdot \left(1 - \frac{1-\nu}{4}\left[1-\left(\frac{d_i}{d_t}\right)^4\right] - \left(\frac{d_i}{d_t}\right)^2 \cdot \left[1+\left(1+\nu\right)\cdot \ln\frac{d_t}{d_i}\right]\right) \qquad \text{(A 2.7-42)}$$

$$L''_{17} = \frac{1}{4} \cdot \left(1 - \frac{1-n}{4} \cdot \left[1-\left(\frac{d_D}{d_t}\right)^4\right] - \left(\frac{d_D}{d_t}\right)^2 \cdot \left[1+\left(1+n\right)\cdot \ln\frac{d_t}{d_D}\right]\right) \qquad \text{(A 2.7-43)}$$

where

d_t	bolt circle diameter	mm
d_d	gasket diameter	mm
d_i	inside diameter for opening for stem	mm
p	internal pressure, either use as p_A (design pressure) or p_P (test pressure)	MPa
ν	Poisson's ratio	—

(4) The stresses due to the force F_D applied on the circular ring with the diameter d_D and of the force F_{s2} applied on the circular ring with the diameter k were determined as follows:

$$\sigma_{F_{s2}} = \frac{6}{s^2} \cdot M_K \qquad \text{(A 2.7-44)}$$

$$\sigma_{F_D} = \frac{6}{s^2} \cdot M_{d_D} \qquad \text{(A 2.7-45)}$$

with

Note:

The equations given hereinafter for M_K and M_{d_D} were taken from "Formulas for Stress and Strain", 5th edition, R. J. Roark, W.C. Young, McGraw-Hill, New York 1975, Case 1a, p. 335.

$$M_K = \frac{F_{s2} \cdot d_t \cdot L'_9}{2 \cdot \pi \cdot k \cdot d_i} \cdot \left(\frac{1-\nu^2}{C_7}\right) \qquad \text{(A 2.7-46)}$$

$$L'_9 = \frac{k}{d_t} \cdot \left\{\frac{1+\nu}{2} \cdot \ln\frac{d_t}{k} + \frac{1-\nu}{4} \cdot \left[1-\left(\frac{k}{d_t}\right)^2\right]\right\}$$

$$M_{d_D} = \frac{F_D \cdot d_t^2 \cdot}{2 \cdot \pi \cdot d_D \cdot d_i} \cdot \frac{L_9''}{C_7} \cdot \left(1 - \nu^2\right) \qquad \text{(A 2.7-47)}$$

$$L_9'' = \frac{d_D}{d_t} \cdot \left\{ \frac{1+\nu}{2} \cdot \ln \frac{d_t}{d_D} + \frac{1-\nu}{4} \cdot \left[1 - \left(\frac{d_D}{d_t} \right)^2 \right] \right\}$$

where

F_{s2}	resulting force from F_{sp} (actuating force) and F_E (external force, e.g. resulting from earthquake)	N
F_D	compression load on gasket (= F_{DB})	N
d_t	bolt circle diameter	mm
d_D	gasket diameter	mm
k	diameter subject to application of external forces	mm
ν	Poisson's ratio	—

(5) The maximum stress is obtained as tangential bending stress:

$$\sigma_{ges} = \sigma_p + \sigma_F + \sigma_{FD} \leq 1.5\ S_m \qquad \text{(A 2.7-48)}$$

A 2.7.3.9 Sealing plate without opening, pressure loaded from one side, without external edge moment

The sealing plate is shown to be a simply supported circular plate with a diameter d_D (see **Figure A 2.7-13**). The load is the pressure p uniformly distributed over the entire plate.

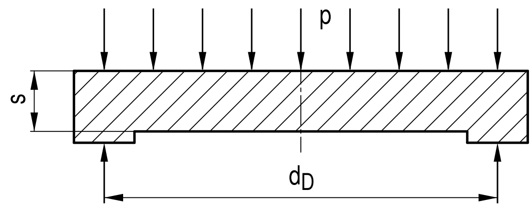


Figure A 2.7-13: Circular flat plate

The maximum loading in the centre of the plate is

$$\sigma_r = \sigma_t = \frac{3 \cdot d_D^2 \cdot p}{32 \cdot s^2} (3 + \nu) \qquad \text{(A 2.7-49)}$$

with ν = 0.3 the strength condition follows

$$\sigma_r = \sigma_t = 0.31 \cdot p \cdot \left(\frac{d_D}{s} \right)^2 \leq 1.5 \cdot S_m \qquad \text{(A 2.7-50)}$$

where

d_D	gasket diameter	mm
s	plate thickness	mm

A 2.7.3.10 Valve sealing disks in the form of a circular plate

The disk is shown as a simply supported circular plate with a diameter d_D (see **Figure A 2.7-14**). The load applied is the pressure p uniformly distributed over the entire plate and the load F distributed uniformly over a circular ring with a diameter d_s. The maximum stress in the centre of the plate is

$$\sigma_r = \sigma_t = \sigma_1 + \sigma_2 \qquad \text{(A 2.7-51)}$$

where the maximum stress results from the internal pressure p as follows

$$\sigma_1 = \frac{3 \cdot d_2^2 \cdot p}{32 \cdot s^2} \cdot (3 + \nu) \qquad \text{(A 2.7-52)}$$

With ν = 0.3 it follows

$$\sigma_1 = 0.31 \cdot p \cdot \left(\frac{d_D}{s} \right)^2 \qquad \text{(A 2.7-53)}$$

The maximum stress due to F is:

$$\sigma_2 = \frac{3 \cdot F}{2 \cdot \pi \cdot s^2} \cdot B_1 \qquad \text{(A 2.7-54)}$$

with

$$B_1 = 0.5 \cdot (1-\nu) + (1+\nu) \cdot \ln \frac{d_D}{d_s} - (1-\nu) \cdot \frac{d_s^2}{2 \cdot d_D^2} \qquad \text{(A 2.7-55)}$$

where

d_D	gasket diameter	mm
s	plate thickness	mm
d_s	effective diameter subject to applied actuating forces	mm

The strength condition is

$$\sigma = \sigma_1 + \sigma_2 \leq 1.5 \cdot S_m \qquad \text{(A 2.7-56)}$$

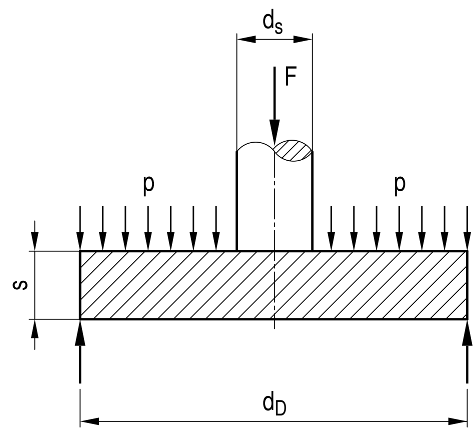


Figure A 2.7-14: Valve sealing disk in the form of a circular plate

A 2.7.3.11 Valve sealing disk in the form of a circular ring plate

The valve disk is shown as a simply supported circular ring plate with free inside edge. The outside diameter is d_D, the inside diameter d_s (see **Figure A 2.7-15**). The load applied is the pressure p uniformly distributed over the ring plate and the load F uniformly distributed at the inner edge of the ring plate.

The maximum bending stress at the inner edge is obtained as circumferential stress as follows:

$$\sigma_t = \sigma_1 + \sigma_2 \qquad \text{(A 2.7-57)}$$

in which case the maximum circumferential stress resulting from the pressure p is

$$\sigma_1 = \frac{3 \cdot p}{\left(d_D^2 - d_s^2\right) \cdot s^2} \cdot B_1 \qquad \text{(A 2.7-58)}$$

$$B_1 = \left(\frac{d_D}{2}\right)^4 \cdot (1+\nu) + \left(\frac{d_s}{2}\right)^4 \cdot (1-\nu) - 4 \cdot \left(\frac{d_D}{2}\right)^2 \cdot \left(\frac{d_s}{2}\right)^2 - $$
$$- 4 \cdot (1-\nu) \cdot \left(\frac{d_D}{2}\right)^2 \cdot \left(\frac{d_s}{2}\right)^2 \cdot \ln \frac{d_D}{d_s} \qquad \text{(A 2.7-59)}$$

(simply supported circular ring plate with free inner edge) and the maximum circumferential stress due to the load F is

$$\sigma_2 = \frac{3 \cdot F}{2 \cdot \pi \cdot \cdot s^2} \cdot B_2 \qquad \text{(A 2.7-60)}$$

$$B_2 = \frac{2 \cdot d_D^2 \cdot (1+\nu)}{d_D^2 - d_s^2} \cdot \ln\frac{d_D}{d_s} + (1-\nu) \qquad \text{(A 2.7-61)}$$

(simply supported circular ring plate with free inner edge and load distributed over the ring)

The strength condition is

$$\sigma_t = \sigma_1 + \sigma_2 \leq 1.5 \cdot S_m \qquad \text{(A 2.7-62)}$$

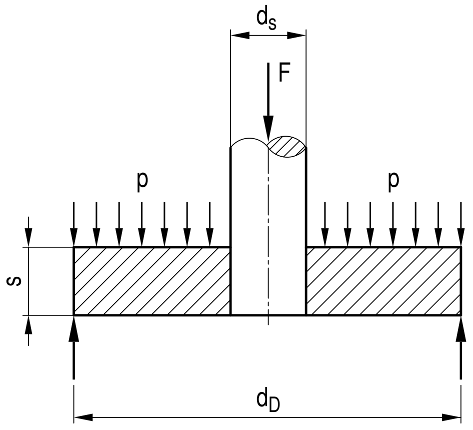


Figure A 2.7-15: Valve disk in the form of a circular ring plate

A 2.8 Reinforcement of openings

A 2.8.1 Design values and units relating to Section A 2.8

Notation	Design value	Unit
d_{Aa}	outside diameter of branch	mm
d_{Ae}	inside diameter of opening plus twice the corrosion allowance c_2	mm
d_{Ai}	inside diameter of opening reinforcement plus twice the corrosion allowance c_2	mm
d_{Am}	mean diameter of nozzle	mm
d_{Hi}	inside diameter of basic shell	mm
d_{Hm}	mean diameter of basic shell at location of opening	mm
e_A	limit of reinforcement, measured normal to the basic shell wall	mm
e_H	half-width of the reinforcement zone measured along the midsurface of the basic shell	mm
e'_H	half-width of the zone in which two thirds of compensation must be placed	mm
l	(see Figure A 2.8-10)	mm
r_1. r_2. r_3	fillet radii	mm
s_A	nominal nozzle wall thickness including the reinforcement, but minus allowances c_1 and c_2	mm
s_{A0}	minimum required nozzles wall thickness	mm
s_H	nominal wall thickness of vessel shell or head at the location of opening including the reinforcement, but minus allowances c_1 and c_2	mm
s_{H0}	minimum required wall thickness of basic shell	mm
s_R	nominal wall thickness of connected piping minus allowances c_1 and c_2	mm
x	slope offset distance	mm
A_e	cross-sectional area of the required reinforcement of opening	mm^2
A_1. A_2. A_3	metal area available for reinforcement	mm^2
F	correction factor acc. to Figure A 2.8-1	—
β	angle between axes of branch and run pipe	degree
δ_5	elongation at fracture	—
φ	angle between vertical and slope	degree

A 2.8.2 Dimensioning of reinforcements of openings in vessels

A 2.8.2.1 Scope

(1) (1) The scope of the design rules given thereinafter correspond to the scopes mentioned in Sections A 2.2 to A 2.5, A 2.7 and A 5.2.4.

(2) The design rules only consider the loadings resulting from internal pressure. Additional forces and moments shall be considered separately.

(3) The angle β between the axes of branch and run pipe shall be equal to or greater than 60°.

A 2.8.2.2 General

(1) Openings shall normally be circular or elliptical. Further requirements will have to be met if the stress index method in accordance with clause 8.2.2.3 is applied. In this case in addition to the scope given in the aforementioned clause the design requirements for the stress index method according to clause 5.2.6 shall be met.

(2) Openings in the basic shell shall be reinforced as follows:

a) by selecting a greater wall thickness for the basic shell than is required for the unpierced basic shell. This wall thickness may be considered to be contributing to the reinforcement on a length e_H measured from the axis of opening,

b) by nozzles which, on a length e_A measured from the outside surface of the basic shell, have a greater wall thickness than is required for internal pressure loading. The metal available for reinforcement shall be distributed uniformly over the periphery of the nozzle,

c) by combining the measures in a) and b) above.

Regarding a favourable shape not leading to increased loadings/stresses subclause c) shall be complied with.

(3) When an opening is to be reinforced the following diameter and wall thickness ratios shall be adhered to:

A wall thickness ratio s_A/s_H up to a maximum of 2 is permitted for d_{Ai} not exceeding 50 mm. This also applies to nozzles with d_{Ai} greater than 50 mm if the diameter ratio d_{Ai}/d_{Hi} does not exceed 0.2.

For nozzles with a diameter ratio d_{Ai}/d_{Hi} greater than 0.2 the ratio s_A/s_H shall basically not exceed 1.3. Higher values are permitted if

a) the additional nozzle wall thickness exceeding the aforementioned wall thickness ratio is not credited for reinforcement of the opening, but is selected for design reasons, or

b) the nozzle is constructed with a reinforcement zone reduced in length (e.g. nozzles which are conical for reasons of improving testing possibilities) in which case the lacking metal area for reinforcement due to the reduced influence length must be compensated by adding metal to the reduced influence length.

Nozzles with inside diameters not less than 120 mm shall be designed with at least two times the wall thickness of the connected piping in which case the factor refers to the calculated pipe wall thickness. Referred to the actual wall thickness the factor shall be at least 1.5.

(4) Openings need not be provided with reinforcement and no verification need be made for openings to A 2.8.3.2 if

a) a single opening has a diameter equal to or less than $0.2 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ or, if there are two or more openings within any circle of diameter $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, but the sum of the diameters of such unreinforced openings shall not exceed $0.25 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, and

b) no two unreinforced openings have their centres closer to each other, measured on the inside wall of the basic shell, than 1.5 times the sum of their diameters, and

c) no unreinforced opening has its centre closer than $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ to the edge of any other locally stressed area (structural discontinuity).

(5) Where nozzle and basic shell are made of materials with differing design stress intensities, the design stress intensity of the basic shell material, if less than that of the nozzle, shall govern the calculation of the entire design.

Where the nozzle material has a lower design stress intensity, the reinforcement zones to be located in areas provided by such material shall be multiplied by the ratio of the design stress intensity values of the reinforcement material and the basic shell material.

Where the materials of the basic shell and the nozzle differ in their specific coefficients of thermal expansion, this difference shall not exceed 15 % of the coefficient of thermal expansion of the run pipe metal.

A 2.8.2.3 Calculation

A 2.8.2.3.1 Required reinforcement

(1) The total cross-sectional area A of the required reinforcement of any opening in cylindrical, spherical and conical shells as well as dished heads under internal pressure shall satisfy the following condition:

$$A \geq d_{Ae} \cdot s_{H0} \cdot F \qquad \text{(A 2.8-1)}$$

where the correction factor F applies to rectangular nozzles and shall have a value of 1 for all planes required for dimensioning. For cylindrical or conical shells F shall be taken from **Figure A 2.8-1** for a plane not required for dimensioning in dependence of its angle to the plane under consideration.

(2) Openings in flat circular heads not exceeding one-half the head diameter shall have an area of reinforcement of at least

$$A \geq 0.5 \cdot d_{Ae} \cdot s_{H0} \qquad \text{(A 2.8-2)}$$

A 2.8.2.3.2 Effective lengths

(1) Credit may be taken for radii or tapers at nozzle-to-basic shell transitions according to clause 5.2.6 in equation (A 2.8-7) determining the effective length.

(2) The effective length of the basic shell shall be determined as follows:

$$e_H = d_{Ae} \qquad \text{(A 2.8-3)}$$

or

$$e_H = 0.5 \cdot d_{Ae} + s_H + s_A \qquad \text{(A 2.8-4)}$$

The calculation shall be based on the greater of the two values. In addition two thirds of the area of reinforcement shall be within the length $2 \cdot e'_H$ (**Figures A 2.8-8** to **A 2.8-10**), where e'_H is the greater value of either

$$e'_H = 0.5 \cdot \left(d_{Ae} + \sqrt{0.5 \cdot d_{Hm} \cdot s_H}\right) \qquad \text{(A 2.8-5)}$$

or

$$e'_H = 0.5 \cdot d_{Ae} + 2/3 \cdot (s_H + s_A) \qquad \text{(A 2.8-6)}$$

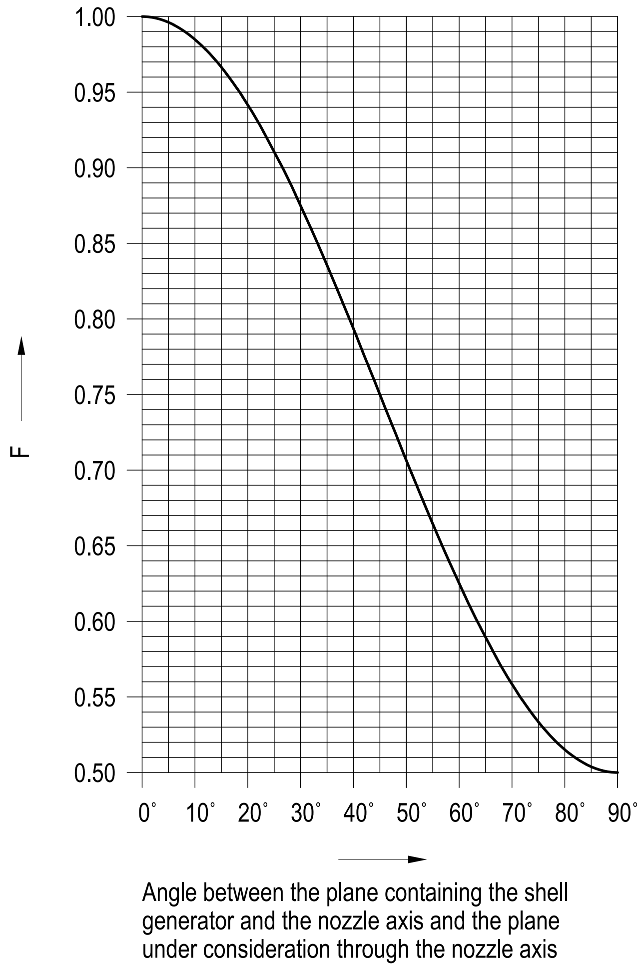


Figure A 2.8-1: Chart for determining the correction factor F for nozzles normal to cylindrical or conical shells

(3) The effective length for nozzles according to **Figures A 2.8-2, A 2.8-3, A 2.8-5, A 2.8-6** shall be determined as follows:

$$e_A = 0.5 \cdot \left(\sqrt{0.5 \cdot d_{Am} \cdot s_A} + r_2\right) \qquad \text{(A 2.8-7)}$$

where

$$d_{Am} = d_{Ai} + s_A \qquad \text{(A 2.8-8)}$$

In the case of a nozzle with tapered inside diameter according to **Figure A 2.8-6** the effective length shall be obtained by using d_{Ai} and s_A values at the nominal outside diameters of the basic shell.

(4) The effective length for nozzles according to **Figures A 2.8-4** and **A 2.8-7** shall be determined as follows:

$$e_A = 0.5 \cdot \sqrt{0.5 \cdot d_{Am} \cdot s_A} \qquad \text{(A 2.8-9)}$$

where

$$d_{Am} = d_{Ai} + s_A \qquad \text{(A 2.8-10)}$$

and additionally for reinforced openings to **Figure A 2.8-4**

$$s_A = s_R + 0.667 \cdot x \qquad \text{(A 2.8-11)}$$

In the case of a nozzle with a tapered inside diameter according to **Figure A 2.7-7** the limit of reinforcement area shall be obtained using d_{Ai} and s_A values at the centre of gravity of nozzle reinforcement area. These values shall be determined, if required, by a trial and error procedure.

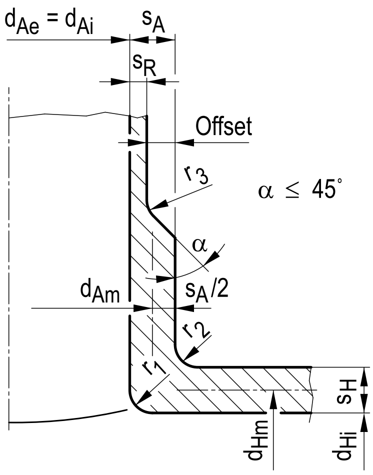


Figure A 2.8-2

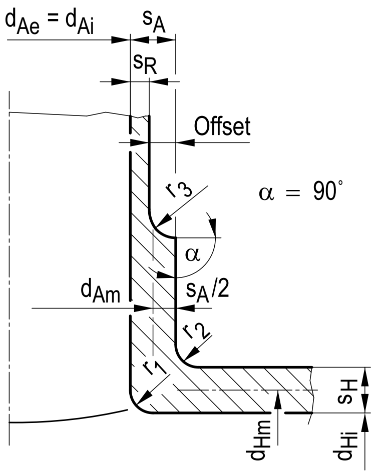


Figure A 2.8-3

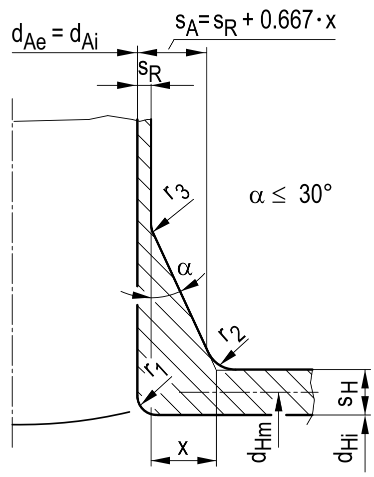


Figure A 2.8-4

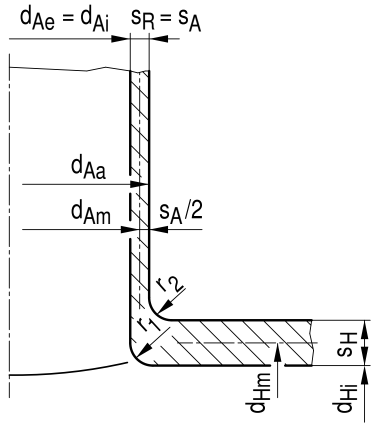


Figure A 2.8-5

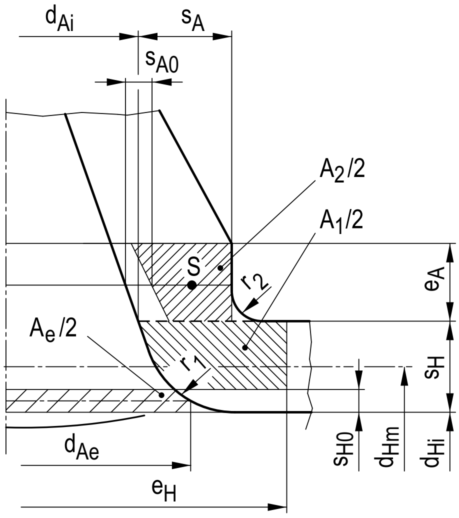


Figure A 2.8-6

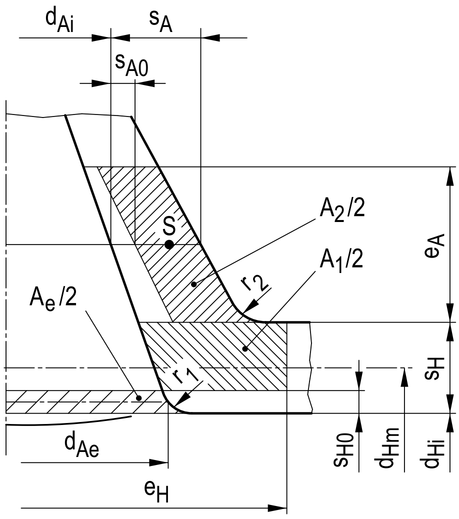


Figure A 2.8-7

Figures A 2.8-2 to **A 2.8-7:** Allowable nozzle configurations

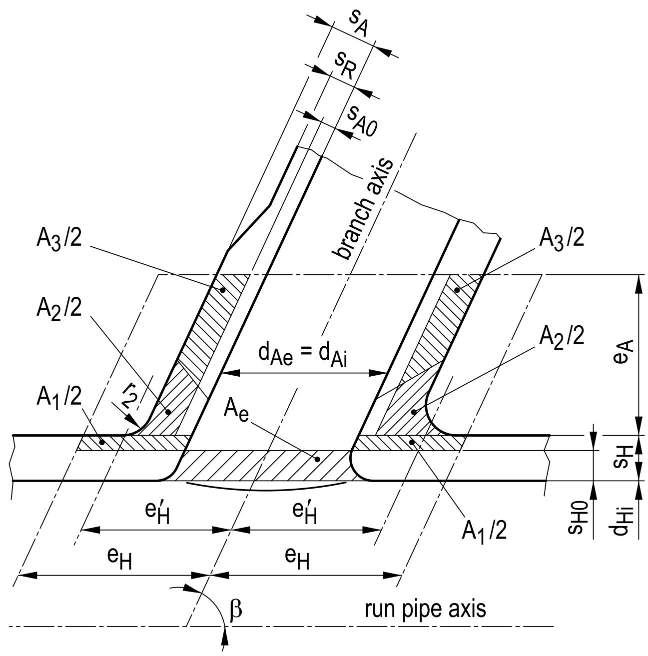


Figure A 2.8-8: Oblique cylindrical branch

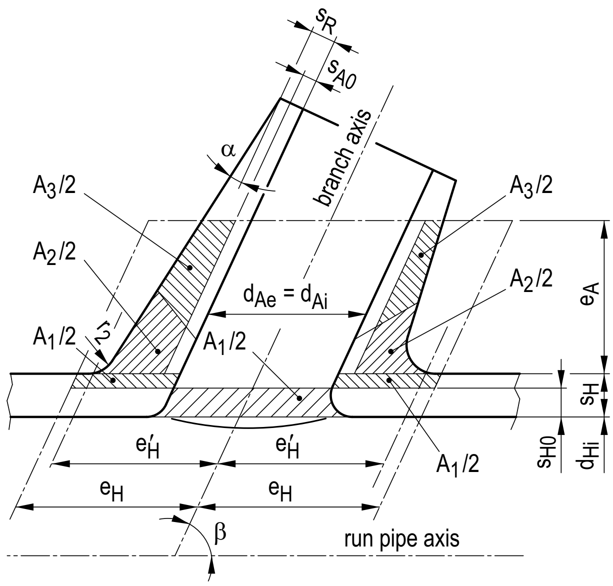


Figure A 2.8-9: Oblique conical branch

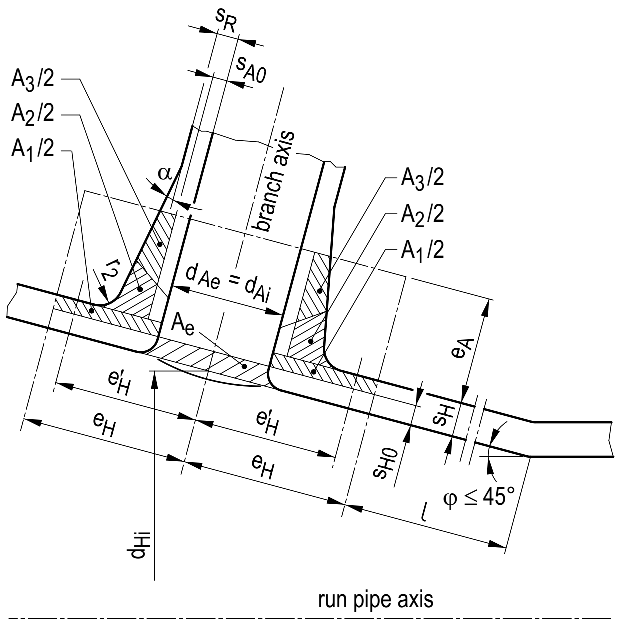


Figure A 2.8-10: Conical shell with reinforced opening

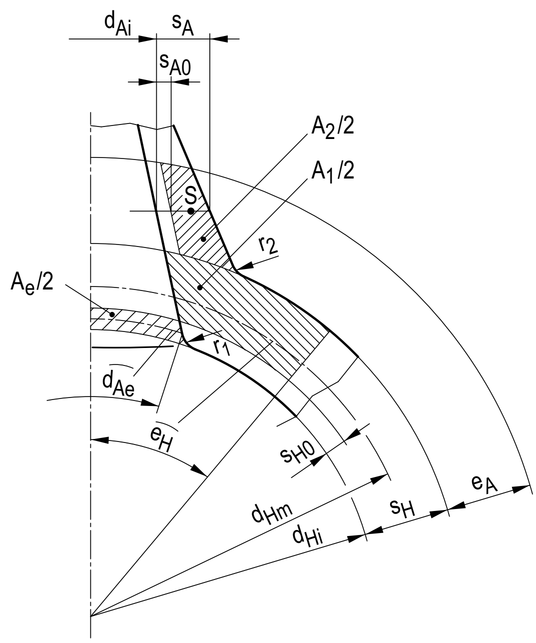


Figure A 2.8-11: Conical branch in spherical shell

A 2.8.2.3.3 Loading scheme for metal areas available for reinforcement

(1) The metal areas A_1, A_2, A_3 available for reinforcement used to satisfy equation (A 2.8-1) are shown in **Figures A 2.8-8** to **A 2.8-11** and shall satisfy the condition $A_1 + A_2 + A_3$ equal to or greater than A.

(2) Interaction between nozzle opening and cone to cylinder transition shall only be taken into account if

$$l < 2.5 \cdot \sqrt{(d_{Hm}/2) \cdot s_H} \quad \text{(A 2.8-12)}$$

where

$$d_{Hm} = d_{Hi} + s_H \quad \text{(A 2.8-13)}$$

A 2.9 Bolted joints

A 2.9.1 Design values and units relating to Section A 2.8.

Notation	Design value	Unit
a. b. c	geometric values for bolt and nut thread in accordance with Figures A 2.9-3 and A 2.9-4	mm
b_D	gasket seating width acc. to Sec. A 2.11	mm
c	design allowance	mm
d	bolt diameter = thread outside diameter	mm
d_2	pitch diameter of thread	mm
d_i	pipe (shell) inside diameter	mm
d_{iL}	diameter of internal bore of bolt	mm
d_D	mean gasket diameter	mm
d_{D1}. d_{D2}	mean gasket diameter for metal-O-ring gaskets	mm
d_k	root diameter of thread	mm
d_M	outer diameter of flange face contact area of metal-to-metal contact type flanged joints	mm
d_s	shank diameter of reduced shank bolt	mm
d_t	bolt circle diameter	mm
$k_1^*, k_{11}^*, k_{12}^*$	gasket factors for metal-O-ring gaskets	N/mm
l	effective thread engagement length or nut thickness	mm
l_B	length of fabricated tapered nut thread end	mm
l_{eff}	(Figure A 2.9-5), compare "l"	mm
l_{ges}	total engagement length or nut thickness	mm
n	number of bolt holes	—
p	design pressure	MPa
p'	test pressure	MPa
A_0	cross-sectional area of shank	mm^2
A_S	section under stress	mm^2
$A_{SG\ Bolzen}$	shear area of bolt thread	mm^2
$A_{SG\ Bi}$	plane of bolt shear area sections	mm^2
$A_{SG\ Mutter}$	shear area of nut thread	mm^2
$A_{SG\ Mi}$	plane of nut shear area sections	mm^2
$A_{SG\ Sackloch}$	shear area of blind hole	mm^2
C_1, C_2, C_3	strength reduction factors	—
D	outside diameter of nut/blind hole thread	mm
D_1	root diameter of nut/blind hole thread	mm
D_2	pitch diameter of nut/blind hole thread	mm
D_c	inside diameter of nut bearing surface, diameter of chamfer	mm
D_m	mean diameter of tapered nut thread end	mm
D_{max}	maximum diameter of tapered nut thread end	mm
$D_{1\ max}$	(see Figure A 2.9-4)	mm

Notation	Design value	Unit
F_{DBO}	allowable gasket load reaction at operating condition of floating type flanged joints	N
$F_{DBU/L}$	required gasket load at operating condition of floating type flanged joints	N
F_{DKU}	gasket load required for obtaining metal-to-metal contact of flange blades	N
F_{DKUx}	gasket load required for obtaining metal-to-metal contact of flange blades in consideration of operating temperature and gasket seating	N
F_{DVO}	allowable gasket load reaction for bolting-up condition of floating type flanged joints	N
$F_{DVU/L}$	gasket seating load	N
F_F	difference between total hydrostatic end force and the hydrostatic end force on area inside flange for design condition	N
F'_F	difference between total hydrostatic end force and the hydrostatic end force on area inside flange for test condition	N
$F_{max\ Bolzen}$	ultimate breaking strength of free loaded bolt thread or shank	N
$F_{max\ G\ Bolzen}$	ultimate breaking strength of engaged bolt thread	N
$F_{max\ G\ Mutter}$	ultimate breaking strength of engaged nut thread	N
F_R	total hydrostatic end force	N
F_{RM}	additional pipe force resulting from pipe moment	N
F_{RM0}	additional pipe force resulting from pipe moment for the bolting-up condition	N
F'_{RM}	additional pipe force resulting from pipe moment for the test condition	N
F_{RP}	hydrostatic end force due to internal pressure	N
F_{RZ}	additional pipe longitudinal force	N
F_{RZ0}	additional pipe longitudinal force for the bolting-up condition	N
F'_{RZ}	additional pipe longitudinal force for the test condition	N
F_{R0}	pipe force effective in piping system at bolting-up condition	N
F_S	operating bolt load (general)	N
F_{S0U}	bolt load for bolting-up condition (lower limit)	N
F_{SB}	bolt load at operating condition of metal-to-metal contact type flanged joints	N
F_{SBU}	bolt load at operating condition of metal-to-metal contact type flanged joints (lower limit)	N
$F_{SBU/L}$	minimum value of bolt load at operating condition of floating type flanged joints	N
F_{SBX}	bolt load at operating condition of floating type flanged joints	N
F_{SKU}	minimum value of bolt load for obtaining metal-to-metal contact of flange blades for metal-to-metal contact type flanged joints	N
F_{S0}	bolt load at bolting-up condition	N
F'_S	bolt load for test condition	N

Notation	Design value	Unit
F_{SPU}	bolt load at test condition (lower limit)	N
F_{Zx}	additional axial force for transfer of transverse forces and torsional moments due to friction at a certain value, for operating condition	N
F_{Z0}	additional axial load for transfer of transverse forces and torsional moments due to friction at a certain value, for bolting-up condition	N
F'_Z	additional axial force for transfer of transverse forces and torsional moments due to friction at a certain value, for test condition	N
M_B	bending moment on pipe	N·mm
M_t	torsional moment on pipe	N·mm
P	thread pitch	mm
Q	transverse force on pipe	N
R_{mB}	tensile strength of bolt material	N/mm²
R_{mM}	tensile strength of nut material	N/mm²
R_{mS}	tensile strength of blind hole material	N/mm²
$R_{p0.2T}$	0.2 % proof stress at operating or test temperature	N/mm²
$R_{p0.2RT}$	0.2 % proof stress at room temperature	N/mm²
R_S	strength ratio	—
S_D	safety factor	—
SW	width across flats	mm
α	pitch angle	Grad
μ_D	gasket friction factor	—
μ_M	friction factor of metal-to-metal contact faces	—
σ_{DB}	average gasket contact surface load for operating condition	N/mm²
σ_{BO}	upper limit value for σ_B	N/mm²
σ_{BU}	lower limit value for σ_B	N/mm²
$\sigma_{BU/L}$	minimum gasket contact surface load at operating condition of floating type flanged joints	N/mm²
σ_{KNS}	minimum gasket contact surface load of metal-to-metal contact type flanged joints	N/mm²
σ_V	gasket contact surface load for gasket seating condition	N/mm²
σ_{VO}	upper limit value for σ_V	N/mm²
σ_{VU}	lower limit value for σ_V	N/mm²
$\sigma_{VU/L}$	minimum gasket contact surface load at bolting-up condition of floating type flanged joints	N/mm²
σ_{zul}	allowable stress as per Table 6.7-2	N/mm²

A 2.9.2 Scope

The calculation rules hereinafter apply to bolts with circular and equi-distant pitch as friction-type connecting elements of pressure-retaining parts. The loads required for the respective gasket seating conditions (bolt load, gasket seating load) are determined for floating type (KHS) and metal-to-metal contact type (KNS) flanged joints (see **Figure A 2.9-1** and clause A 2.9.4). Sufficient stiffness and thus limited flange deflection is prerequisite to the use of metal-to-metal contact type flanged joints. The calculation rules primarily consider static tensile loading. Shear and bending stresses in the bolts resulting e.g. from deflections of flanges and covers, thermal effects (e.g. local or time-dependent temperature gradients, different coefficients of thermal expansion) are not covered by this Section.

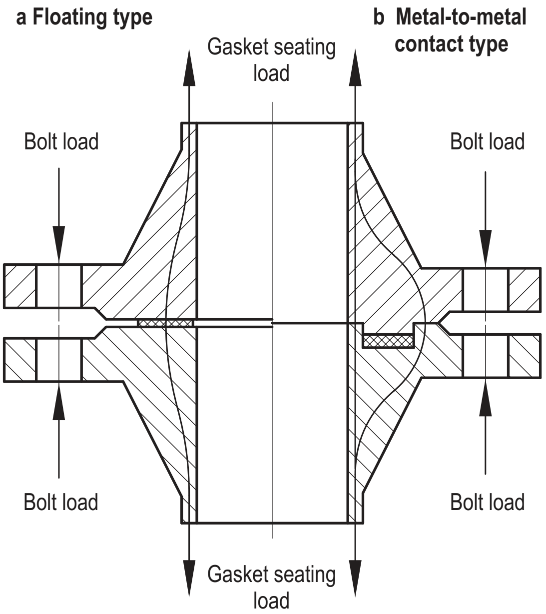


Figure A 2.9-1: Presentation of floating type and metal-to-metal contact type flanged joint (schematic)

A 2.9.3 General

(1) For bolted flange connections proof of tightness and strength shall be rendered (see flow diagram in **Figure A 2.9-2**). Within leak tightness proof the magnitude of initial bolt pre-stress shall basically be determined which is required to ensure tightness of the joint during operating and test conditions. With the proof of strength it shall be verified that the allowable stresses for flanges, bolts and gaskets are not exceeded.

(2) The first step is to select the components of the flanged joint for which the simplified methods indicated in clauses A 2.9.4, A 2.10.4 and 2.10.5 are suited. These methods are used to determine the required dimensions as well as initial bolt pre-stress from specified loadings, the gasket selected (e.g. dimensions, tightness class, gasket factors) and from the allowable stresses of the flanged joint components.

(3) The second step consists in proving the tightness and strength and in verifying the compensation of internal forces and moments (also transverse force and torsional moment). The bolt tightening procedure (e.g. tightening factor) shall be credited in the verification of strength of flanges and bolts.

The tightness shall be proved using the minimum design bolt load. Deviating herefrom, the proof may be based on the average design bolt load in the case of metal-to-metal contact type flanged joints with a number of bolts n equal to or exceeding 8.

The strength of flange and, in the case of floating type flanged joints, of the gasket at bolting-up condition shall be verified taking credit of the maximum design bolt load. The proof of strength at operating condition may be based on the average design bolt load.

For the proof of strength of the bolts the maximum bolt load shall be used.

(4) Where proofs of tightness and strength cannot be rendered, iteration of the process shall be made to repeat all proofs until the conditions have been met.

(5) Such bolts are deemed to be reduced-shank bolts the shank diameter of which does not exceed 0.9 times the root diameter and the shank length of which is at least two times, but should be four times the shank diameter, or such bolts the dimensions of which correspond to DIN 2510-1 to DIN 2510-4. Shank bolts with extended shank length and a shank diameter equal to or less than the root diameter may be used as reduced-shank bolts if their yielding regarding bolt elongation and elastic behaviour regarding bending under the given boundary conditions corresponds to the elastic behaviour of a reduced-shank bolt as defined above with same root diameter and minimum shaft length as specified above.

For bolted joints to DIN EN ISO 898-1, DIN EN ISO 898-2, DIN EN ISO 3506-1, DIN EN ISO 3506-2, DIN EN ISO 3506-3, DIN 267-13 and DIN 2510-1 to DIN 2510-4 a recalculation of the thread loading can be waived if the given nut thickness or thread engagement lengths are adhered to.

Otherwise, the calculation shall be made in accordance with A 2.9.4 or VDI 2230.

A 2.9.4 Dimensioning of bolts

A 2.9.4.1 Bolt load for floating type flanged joints

The bolt load (F_S) shall be determined at operating condition (F_{SBx}), at test condition (F'_S) and at bolting-up condition (F_{S0}).

a) Required bolt load at operating condition

$$F_{SBU/L} = F_R + F_{DBU/L} + F_F + F_Z \qquad \text{(A 2.9-1)}$$

The hydrostatic end force F_R is the force transmitted from the pipe or shell on the flange. This force is obtained for unpierced pipes or shells from the following equation:

$$F_R = F_{RP} + F_{RZ} + F_{RM} \qquad \text{(A 2.9-2)}$$

where

$$F_{RP} = \frac{d_i^2 \cdot \pi \cdot p}{4} \qquad \text{(A 2.9-3)}$$

The additional pipe forces F_{RZ} and F_{RM} consider pipe longitudinal forces F_{RZ} and pipe bending moments M_B, where

$$F_{RM} = \frac{4 \cdot M_B}{d_D} \qquad \text{(A 2.9-4)}$$

On the basis of the prevailing stiffness ratios the effective gasket diameter may be taken instead of the mean gasket diameter d_D.

In the calculation of bolt stresses the bolt circle diameter d_t may be used instead of the mean gasket diameter d_D.

If required F_{RZ} and M_B shall be taken from the static or dynamic piping system analysis.

F_{RZ} and M_B are equal to zero for flanged joints in vessels and pipings to which no piping or only pipings without additional longitudinal force F_{RZ} and without additional pipe bending moment M_B are connected.

The required bolt load at operating condition ($F_{DBU/L}$) is obtained from:

$$F_{DBU/L} = \pi \cdot d_D \cdot b_D \cdot \sigma_{BU/L} \cdot S_D \qquad \text{(A 2.9-5)}$$

For S_D a value of at least 1.2 shall be taken.

The required compression load on the gasket at operating condition $F_{DBU/L}$ is required to ensure tight joint during operation (tightness class L). The gasket factors can be found in Section A 2.11.

For weld lip seals an axial compression force shall be maintained on the flange blade faces to ensure positional stability. For $F_{DBU/L}$ at least a value of 0.15 (F_{RP}+F_F) shall be taken.

The allowable (maximum bearable) compression load on the gasket at operating condition shall be

$$F_{DBO} = \pi \cdot d_D \cdot b_D \cdot \sigma_{BO} \qquad \text{(A 2.9-6)}$$

The difference between total hydrostatic end force and the hydrostatic end force on area inside flange F_F shall be

$$F_F = \frac{\pi}{4} \cdot \left(d_D{}^2 - d_i{}^2\right) \cdot p \qquad \text{(A 2.9-7)}$$

This force F_F is caused by the internal pressure p and is applied on the annular area inside the flange bounded by the gasket diameter d_D and the inside diameter d_i. The mean gasket diameter shall be taken as gasket diameter d_D. For weld lip seals the mean diameter of the weld shall be taken. For concentric double gaskets the mean diameter of the outer gasket shall be taken.

If required, an additional force F_Z shall be applied on the gasket to make possible transfer of a transverse force Q (normal to pipe axis) and a torsional moment M_t due to friction at a certain value in the flanged joint.

F_Z shall be:

aa) for laterally displaceable flanges where transverse forces can only be transferred due to friction at a certain value

$$F_Z = \max\left\{0; \frac{Q}{\mu_D} + \frac{2 \cdot M_t}{\mu_D \cdot d_D} - F_{DBU/L} - \frac{2 \cdot M_B}{d_D}\right\} \qquad \text{(A 2.9-8)}$$

ab) for laterally non-displaceable flanges where transverse forces can be transferred due to infinite friction

$$F_Z = \max\left\{0; \frac{2 \cdot M_t}{\mu_D \cdot d_D} - F_{DBU/L} - F_{RM}\right\} \qquad \text{(A 2.9-9)}$$

Where no other test results have been obtained the friction factors shall be taken as follows:

μ_D = 0.05 for PTFE based gaskets

μ_D = 0.1 for graphite-reinforced gaskets

μ_D = 0.15 for metallic flat contact faces

μ_D = 0.25 for uncoated gaskets on fibre basis

b) Required bolt load at test condition

$$F_{SPU} = \frac{p'}{p} \cdot \left(F_{RP} + \frac{F_{DBU/L}}{S_D} + F_F\right) + F'_{RZ} + F'_{RM} + F'_Z \qquad \text{(A 2.9-10)}$$

The values F'_{RZ} and F'_{RM} correspond to the additional pipe forces at test condition. F'_Z shall be determined by means of equations (A 2.9-8) and (A 2.9-9) in consideration of the test condition.

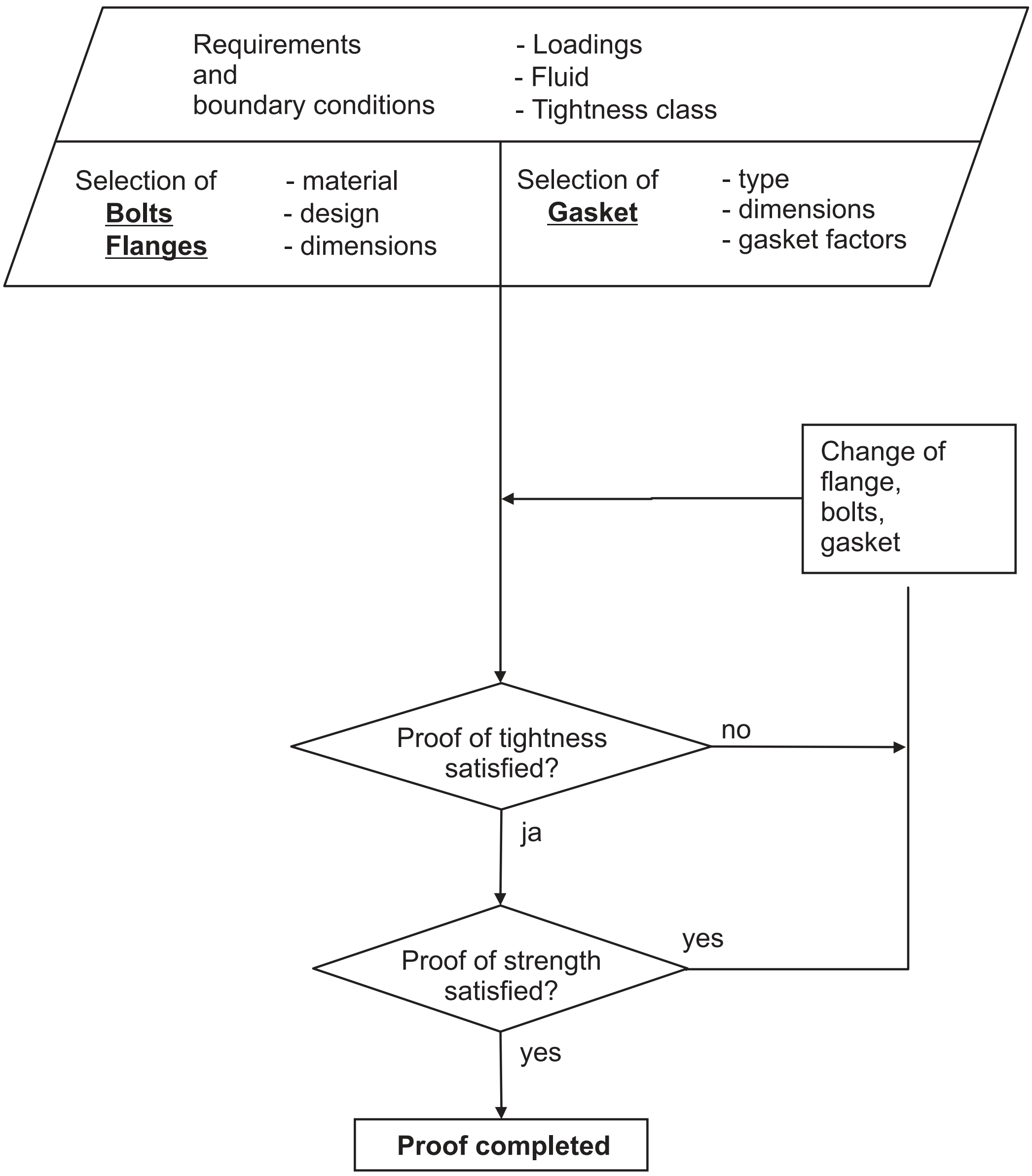


Figure A 2.9-2: General flow diagram for flange design

c) Required bolt load at bolting-up condition (gasket seating)

The bolts shall be so tightened that the required gasket seating is obtained and the bolted joint remains leak tight at the test and operating conditions, and pipe forces F_{R0}, if any, are absorbed.

To satisfy these conditions the following must be met:

$$F_{S0U} \geq F_{DVU/L} + F_{RZ0} + F_{RM0} \quad \text{(A 2.9-11)}$$

but at least

for the test condition

$$F_{S0U} \geq F'_S \quad \text{(A 2.9-12)}$$

and for the operating condition

$$F_{S0U} \geq F_{SBU/L} \quad \text{(A 2.9-13)}$$

Here, $F_{DVU/L}$ is the gasket seating load required to obtain sufficient contact (tightness class L) between gasket and flange facing.

$$F_{DVU/L} = \pi \cdot d_D \cdot b_D \cdot \sigma_{VU/L} \quad \text{(A 2.9-14)}$$

At bolting-up condition the gasket shall only be loaded with

$$F_{DVO} = \pi \cdot d_D \cdot b_D \cdot \sigma_{VO} \quad \text{(A 2.9-15)}$$

A 2.9.4.2 Bolt load for metal-to-metal contact type flanged joints

The bolt load (F_S) shall be determined for obtaining metal-to-metal contact of flange blades (F_{SKU}), for the test condition (F'_S) and the operating condition (F_{SB}). Equation (A 2.10-25) is used to check indirectly if metal-to-metal contact between flange blade faces has been obtained.

a) Required compression load on gasket for obtaining metal-to-metal contact of flange blades

$$F_{SKU} = F_{DKU} + F_{R0} \quad \text{(A 2.9-16)}$$

For gaskets, except for metal O-rings, the required compression load on gasket for obtaining metal-to-metal contact of flange blades is derived from:

$$F_{DKU} = \pi \cdot d_D \cdot b_D \cdot \sigma_{KNS} \quad \text{(A 2.9-17)}$$

for simple metal O-ring gaskets from:

$$F_{DKU} = \pi \cdot d_D \cdot k_1^* \quad \text{(A 2.9-18)}$$

and for double metal O-ring gaskets from:

$$F_{DKU} = \pi \cdot (d_{D1} \cdot k_{11}^* + d_{D2} \cdot k_{12}^*) \quad \text{(A 2.9-19)}$$

For simple metal O-ring gaskets the gasket factor k_1^* and for double metal O-rings the gasket factors k_{11}^* and k_{12}^* shall be taken from the manufacturer's documents.

b) Required bolt load for maintaining metal-to-metal contact of flange blades at op-erating condition

$$F_{SBU} = F_{DKU} + F_R + F_F + F_Z \quad \text{(A 2.9-20)}$$

The pipe force F_R is considered to be the force transmitted from the pipe or shell on the flanged joint. The pipe force for unpierced pipes and shells is obtained from the following equation:

$$F_R = F_{RP} + F_{RZ} + F_{RM} \quad \text{(A 2.9-21)}$$

where

$$F_{RP} = \frac{d_i^2 \cdot \pi \cdot p}{4} \quad \text{(A 2.9-22)}$$

The additional pipe forces F_{RZ} and F_{RM} consider pipe longitudinal forces F_{RZ} and pipe bending moments M_B, where

$$F_{RM} = \frac{4 \cdot M_B}{\frac{(2 \cdot d_M + d_d + b_D)}{3}} \quad \text{(A 2.9-23)}$$

F_Z will be:

aa) for laterally displaceable flanges where transverse forces can only be transferred due to friction at a certain value

$$F_Z = \max\left\{\begin{array}{l} 0; \dfrac{Q}{\mu_M} + \dfrac{2 \cdot M_t}{\mu_M \cdot \left[\dfrac{(2 \cdot d_M + d_d + b_D)}{3}\right]} - \dfrac{\mu_D}{\mu_M} F_{DKU} \\ - \dfrac{2 \cdot M_B}{\dfrac{(2 \cdot d_M + d_d + b_D)}{3}} \end{array}\right\} \quad \text{(A 2.9-24)}$$

ab) for laterally non-displaceable flanges where transverse forces can only be transferred due to infinite friction

$$F_Z = \max\left\{0; \frac{2 \cdot M_t}{\mu_M \cdot \dfrac{(2 \cdot d_M + d_D + b_D)}{3}} - \frac{\mu_D}{\mu_M} F_{DKU} - F_{RM}\right\} \quad \text{(A 2.9-25)}$$

Where no other test results have been obtained, the friction factors shall be taken as follows:

μ_D = 0.10 for graphite-coated gaskets

μ_M = 0.15 for metallic flat contact faces

c) Required bolt load for maintaining metal-to-metal contact of flange blades at test condition

$$F_{SPU} = \frac{p'}{p} \cdot (F_{RP} + F_F) + F_{DKU} + F'_{RZ} + F'_{RM} + F'_Z \quad \text{(A 2.9-26)}$$

The values F'_{RZ} and F'_{RM} correspond to the additional pipe forces at test condition. F'_Z shall be determined by means of equations (A 2.9-24) and (A 2.9-25) in consideration of the test condition.

d) Required bolt load for gasket seating condition

The bolts shall be so tightened that the required bolt load for gasket seating is applied to obtain metal-to-metal contact between flange blade faces at bolting-up condition and to maintain this metal-to-metal contact both at test and operating condition.

To satisfy these requirements, the following is required:

$$F_{S0U} \geq \max\,(F_{SKU} + F_{Z0};\ F_{SBU};\ F_{SPU}) \quad \text{(A 2.9-27)}$$

In this case, F_{Z0} is the additional axial load required at bolting-up to transfer a transverse force or torsional moment. This additional load shall be determined to equations (A 2.9-24) and (A 2.9-25) in which case the forces and moments are to be taken for the bolting-up condition.

A 2.9.4.3 Pre-stressing of bolts

(1) The initial bolt prestress shall be applied in a controlled manner. Depending on the bolt tightening procedure this control e.g. applies to the bolting torque, the bolt elongation or temperature difference between bolt and flange. Here - in dependence of the tightening procedure - e.g. the following influence factors shall be taken into account: friction factor, surface finish, greased condition, gasket seating.

(2) Where the bolts are tightened by means of torque wrench, the bolting torque shall be determined by a suitable calculation or experimental analysis, e.g. VDI 2230, Sheet 1.

A 2.9.4.4 Bolt diameter

(1) The required root diameter of thread d_k of a full-shank bolt or the shank diameter d_s of a reduced shank bolt (with or without internal bore) in a bolted connection with a number n of bolts shall be calculated by means of the following equation:

$$d_k \text{ or } d_s = \sqrt{\frac{4 \cdot F_S}{\pi \cdot n \cdot \sigma_{zul}} + d_{iL}^{\,2}} + c \qquad \text{(A 2.9-28)}$$

with σ_{zul} according to **Table 6.7-2**.

(2) Here, the following load cases shall be considered:

a) the load cases of loading levels 0, A, B, C, D according to lines 1 and 2 of **Table 6.7-2**,

b) the load case of loading level P according to line 3 of **Table 6.7-2**,

c) the bolting-up conditions according to line 4 of **Table 6.7-2** (to consider the scattered range of forces applied depending on the tightening procedure, the respective requirements of VDI 2230, Sheet 1 shall be taken into account).

(3) A design allowance c = 0 mm shall be used for reduced-shank bolts, and for full-shank bolts the following applies for the load cases of loading level 0 according to lines 1 and 2 of **Table 6.7-2**:

$$c = 3 \text{ mm, if } \sqrt{\frac{4 \cdot F_S}{\pi \cdot n \cdot \sigma_{zul}}} \le 20\,\text{mm} \qquad \text{(A 2.9-29)}$$

or

$$c = 1 \text{ mm, if } \sqrt{\frac{4 \cdot F_S}{\pi \cdot n \cdot \sigma_{zul}}} \ge 50\,\text{mm} \qquad \text{(A 2.9-30)}$$

Intermediate values shall be subject to straight interpolation with respect to

$$c = \frac{65 - \sqrt{\dfrac{4 \cdot F_S}{\pi \cdot n \cdot \sigma_{zul}}}}{15} \qquad \text{(A 2.9-31)}$$

For the load cases of the other loading levels c = 0 mm shall be taken.

A 2.9.4.5 Required thread engagement length

A 2.9.4.5.1 General

(1) When determining the required thread engagement length in a cylindrical nut or blind hole it shall normally be assumed that the limit load based on the threadstripping resistance of both the bolt thread and female thread is greater than the load-bearing capacity based on the tensile strength of the free loaded portion of the thread or of the shank in the case of reduced-shank bolts. The load-bearing capacity of the various sections is calculated as follows:

Free loaded thread:

$$F_{max\ Bolzen} = R_{m\ Bolzen} \cdot A_S \qquad \text{(A 2.9-32)}$$

Reduced shank:

$$F_{max\ Bolzen} = R_{m\ Bolzen} \cdot A_0 \qquad \text{(A 2.9-33)}$$

Engaged bolt thread:

$$F_{max\ G\ Bolzen} = R_{m\ Bolzen} \cdot A_{SG\ Bolzen} \cdot C_1 \cdot C_2 \cdot 0.6 \qquad \text{(A 2.9-34)}$$

Engaged nut thread:

$$F_{max\ G\ Mutter} = R_{m\ Mutter} \cdot A_{SG\ Mutter} \cdot C_1 \cdot C_3 \cdot 0.6 \qquad \text{(A 2.9-35)}$$

(2) The calculation of the thread engagement length shall be made for the case with the smallest overlap of flanks in accordance with the clauses hereinafter. To this end, the smallest bolt sizes and greatest nut sizes (thread tolerances) shall be used in the calculation of the effective cross-sections.

(3) At a given thread engagement length or nut thickness it shall be proved that the load-bearing capacity of the free loaded thread portion or reduced shank is smaller than that of the number of engaging bolt or nut threads. Where less credit of bolt strength is taken, the bolt load F_S to clauses A 2.9.4.1 or A 2.9.4.2 may be used. The verification of the required thread engagement length shall then be made to clause A 2.9.4.5.5.

(4) Standard bolts are exempted from the calculation of the thread engagement length in accordance with the following clauses. The calculation of the engagement length in the clauses hereinafter including clause A 2.9.4.5.5 does not apply to bolts with saw-tooth or tapered threads.

(5) Where, in representative tests, thread engagement lengths smaller than that calculated in the following clauses are obtained, these lengths may be used.

A 2.9.4.5.2 Bolted joints with blind hole or cylindrical nut without chamfered inside

The required engagement length l_{ges} for bolted joints with blind hole or cylindrical nut shall be the maximum value obtained from the equations given hereinafter:

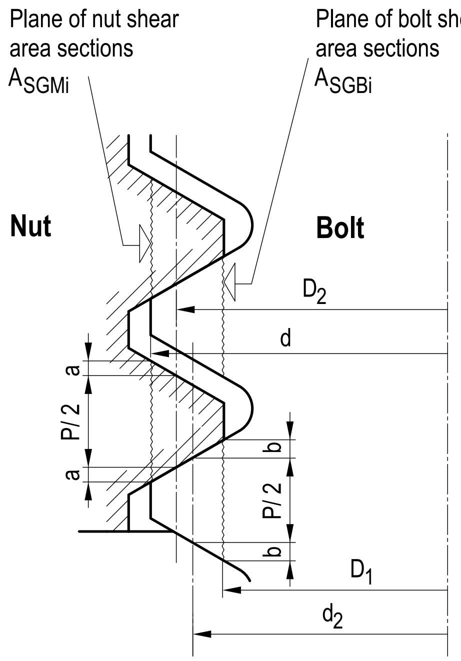
a) The requirement for threadstripping resistance of the bolt thread leads to the condition (see **Figure A 2.9-3**):

$$l_{ges} \ge \frac{A_S \cdot P}{0.6 \cdot C_1 \cdot C_2 \cdot \pi \cdot D_1 \cdot \left[\frac{P}{2} + (d_2 - D_1) \cdot \tan\frac{\alpha}{2}\right]} + 2.0 \cdot P \qquad \text{(A 2.9-36)}$$

In the case of reduced-shank bolts the cross-sectional area of shank A_0 may be inserted instead of the section under stress A_S.

For tapered threads with a thread angle α = 60°

$$\tan\frac{\alpha}{2} = \frac{1}{\sqrt{3}}$$



$$a = \frac{d - D_2}{2} \cdot \frac{1}{\sqrt{3}} \qquad b = \frac{d_2 - D_1}{2} \cdot \frac{1}{\sqrt{3}}$$

Figure A 2.9-3: Representation of design values for bolt and female thread

b) The requirement for threadstripping resistance of the nut or blind hole thread leads to the condition (see **Figure A 2.9-3**):

$$l_{ges} \ge \frac{R_{mB} \cdot A_S \cdot P}{R_{mM} \cdot 0.6 \cdot C_1 \cdot C_3 \cdot \pi \cdot d \cdot \left[\frac{P}{2} + (d - D_2) \cdot \tan\frac{\alpha}{2}\right]} + 2.0 \cdot P \qquad \text{(A 2.9-37)}$$

In the case of a blind hole the tensile strength R_{mS} shall be inserted in lieu of R_{mM}.

c) In addition, the following condition shall be satisfied:

$$l_{ges} \geq 0.8 \cdot d \qquad \text{(A 2.9-38)}$$

The values C_1, C_2 and C_3 shall be determined in accordance with A 2.9.4.5.4.

A 2.9.4.5.3 Bolted joint with tapered thread area without chamfer

The required engagement length l_{ges} for bolted joints with tapered thread area of nut shall be determined as the maximum value obtained from the equations hereinafter.

a) The requirement for threadstripping resistance of the bolt thread leads to the condition (see **Figures 2.9-4** and **A 2.9-5**):

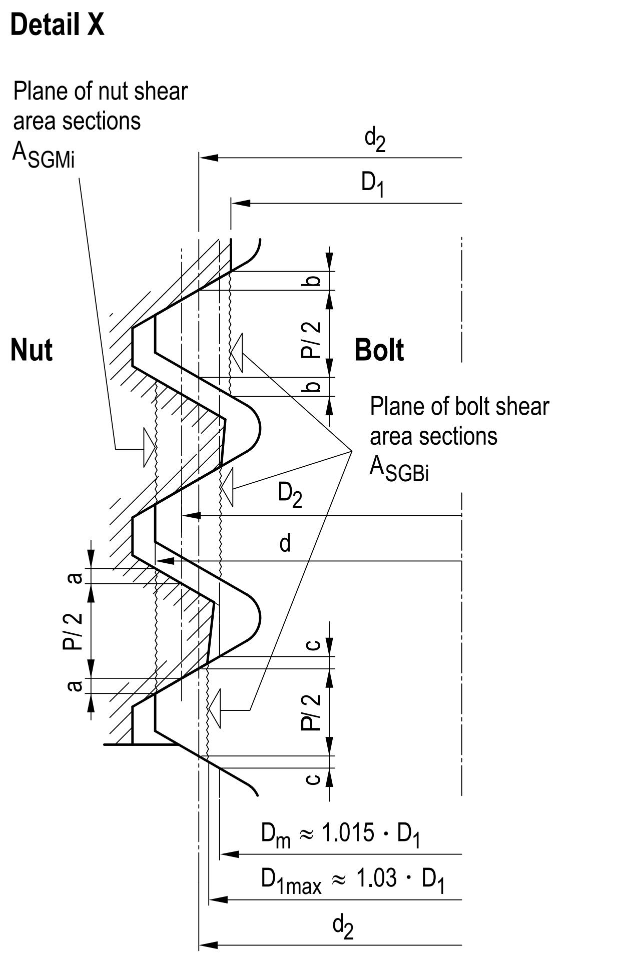
$$l_{ges} \geq l_B + \frac{A_S \cdot P - 0.6 \cdot C_1 \cdot C_2 \cdot l_B \cdot \pi \cdot D_m \cdot \left[\frac{P}{2} + (d_2 - D_m) \cdot \tan\frac{\alpha}{2}\right]}{0.6 \cdot C_1 \cdot C_2 \cdot \pi \cdot D_1 \cdot \left[\frac{P}{2} + (d_2 - D_1) \cdot \tan\frac{\alpha}{2}\right]} + 2.0 \cdot P$$

(A 2.9-39)

b) The requirement for threadstripping resistance of the nut thread leads to the required engagement length l_{ges} (see **Figures A 2.9-4** and **A 2.9-5**) according to equation (A 2.9-37).

c) The thread engagement length l shall satisfy equation (A 2.9-38).

The values C_1, C_2 and C_3 shall be determined in accordance with A 2.9.4.5.4.



$$a = \frac{d - D_2}{2} \cdot \frac{1}{\sqrt{3}} \qquad b = \frac{d_2 - D_1}{2} \cdot \frac{1}{\sqrt{3}} \qquad c = \frac{d_2 - D_m}{2} \cdot \frac{1}{\sqrt{3}}$$

Figure A 2.9-4: Representation of design values for bolt and female thread (tapered female thread)

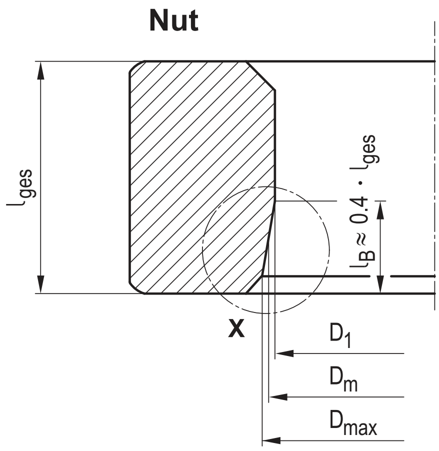


Figure A 2.9-5: Representation of design values for the nut (with tapered portion)

A 2.9.4.5.4 Factors C_1. C_2. C_3

(1) The factor C_1 shall be determined by means of the following equation

$$C_1 = \left[-\left(\frac{SW}{d}\right)^2 + 3.8 \cdot \left(\frac{SW}{d}\right) - 2.61\right] \qquad \text{(A 2.9-40)}$$

for $1.4 \leq \frac{SW}{d} \leq 1.9$

or in accordance with **Figure A 2.9-6**.

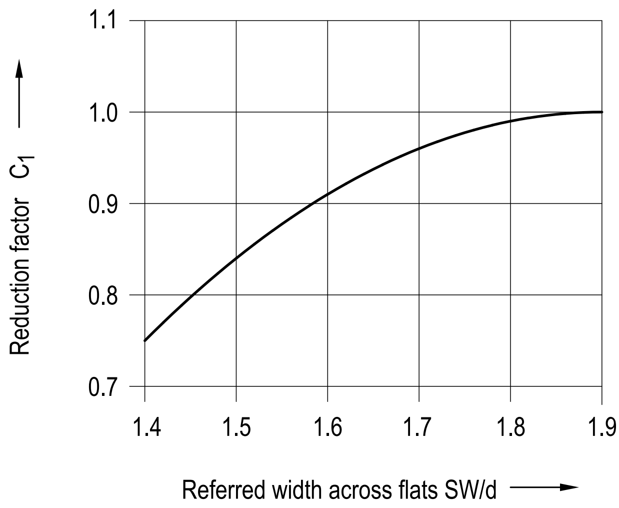


Figure A 2.9-6: Factor C_1 for the reduction of threadstripping resistance of bolt and nut thread due to nut extension

In the case of serrated nuts the width across flats SW shall be replaced by an equivalent value.

(2) The factor C_2 can be determined by means of equation (A 2.9-46) or according to **Figure A 2.9-7**.

The required values are computed as follows:

Strength ratio R_S

$$R_S = \frac{(R_m \cdot A_{SG})_{Mutter/Sackloch}}{(R_m \cdot A_{SG})_{Bolzen}} \qquad \text{(A 2.9-41)}$$

Note:

When determining the strength ratio the quotient of the shear areas $A_{SG\ Mutter/Sackloch}$ and $A_{SG\ Bolzen}$ shall be formed so that the engagement length l can be obtained.

The shear area A_{SG} of the nut or blind hole thread is

$$A_{SG\,Mutter/Sackloch} = \frac{l}{P} \cdot \pi \cdot d \cdot \left[\frac{P}{2} + (d - D_2) \cdot \tan\frac{\alpha}{2} \right] \qquad \text{(A 2.9-42)}$$

The size of the shear area $A_{SG\ Bolzen}$ depends on whether a bolted joint with blind hole or nut with straight thread or a bolted joint with a nut having a tapered threaded portion is concerned.

Therefore, the equation of the shear area $A_{SG\ Bolzen}$ for bolted joints with blind hole or straight nut is:

$$A_{SG\,Bolzen} = \frac{l}{P} \cdot \pi \cdot D_1 \cdot \left[\frac{P}{2} + (d_2 - D_1) \cdot \tan\frac{\alpha}{2} \right] \qquad \text{(A 2.9-43)}$$

The size of the shear area $A_{SG\ Bolzen}$ of a bolt for bolted joints with a nut having a tapered threaded portion as shown in **Figure A 2.9-5** and in consideration of the relationship $l_B = 0.4 \cdot l$ shall be:

$$A_{SG\,Bolzen} = \frac{0.6 \cdot l}{P} \cdot \pi \cdot D_1 \cdot \left[\frac{P}{2} + (d_2 - D_1) \cdot \tan\frac{\alpha}{2} \right] + \\ + \frac{l_B}{P} \cdot \pi \cdot D_m \cdot \left[\frac{P}{2} + (d_2 - D_m) \cdot \tan\frac{\alpha}{2} \right] \qquad \text{(A 2.9-44)}$$

D_m is obtained from $D_m = 1.015 \cdot D_1$ (A 2.9-45)

C_2 is obtained for $1 < R_S \leq 2.2$ from equation

$$C_2 = 5.594 - 13.682\,R_S + 14.107\,R_S^2 - 6.057\,R_S^3 + 0.9353\,R_S^4 \qquad \text{(A 2.9-46)}$$

and for $R_S \leq 1$ to $C_2 = 0.897$.

C_2 may also be determined by means of **Figure A 2.9-7**.

(3) The factor C_3 is obtained for $0.4 \leq R_S < 1$ from the equation

$$C_3 = 0.728 + 1.769\,R_S - 2.896\,R_S^2 + 1.296\,R_S^3 \qquad \text{(A 2.8-47)}$$

and for $R_S \geq 1$ to $C_3 = 0.897$.

C_3 may also be determined by means of **Figure A 2.9-7**.

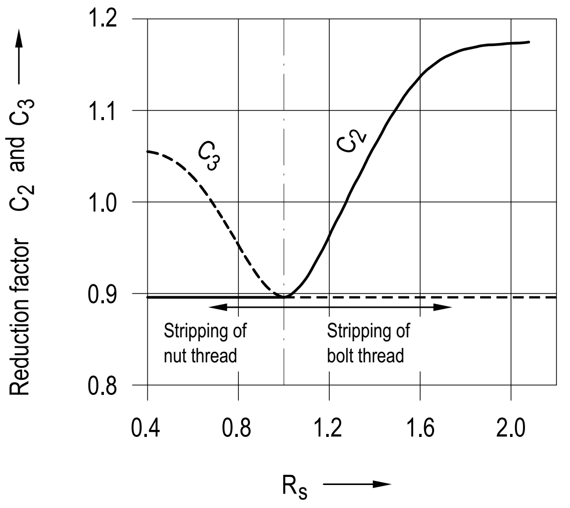


Figure A 2.9-7: Factor for reduction of threadstripping resistance of bolt and nut thread due to plastic deformation of thread

A 2.9.4.5.5 Required engagement length for valve bodies

(1) Alternately to the procedure given in clauses A 2.9.4.5.1 to A 2.9.4.5.4 the engagement length may be checked as follows for valve bodies. Proof is deemed to be furnished if the following conditions are satisfied:

a) $l \geq 0.8 \cdot d$ (A 2.9-48)

and

b) $l \geq \dfrac{2 \cdot F_{max}}{n \cdot \pi \cdot d_2 \cdot S_m}$ (A 2.9-49)

where

- l engagement length
- n number of bolts
- d, d_2 in accordance with **Figure A 2.9-8**

S_m (S in the case of test group A2 or test group A3 valve bodies) is the smaller of the design stress intensity values according to **Table 6.6-1** of the materials to be bolted.

(2) The bolting-up condition and the operating conditions shall be verified separately.

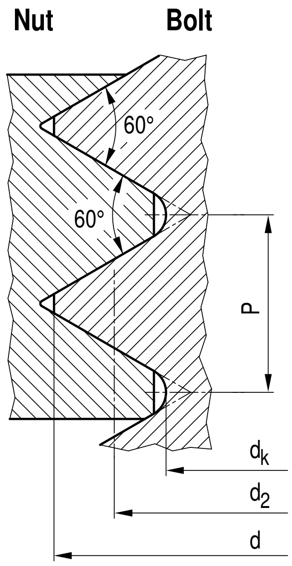


Figure A 2.9-8: Thread dimensions

A 2.10 Flanges

A 2.10.1 Design values and units relating to Section A 2.10

Notation	Design value	Unit
a	moment arm, general	mm
a_1	distance between bolt hole centre and intersection C-C	mm
a_D	distance between bolt hole centre and point of application of compression load on gasket F_D	mm
a_F	distance between bolt hole centre and point of application of force F_F	mm
a_M	distance between bolt hole centre and outer point of contact of the two flange blades	mm
a_R	distance from bolt centre to point of application of total hydrostatic end force F_R	mm
a_{Reib}	$a_{Reib} = 0.5 \cdot [d_t - 0.5 \cdot (d_{Fa} + d_D + b_D)]$	mm
b	radial width of flange ring	mm
b_D	gasket width according to Section A 2.11	mm
c_B	spring stiffness of blank	N/mm
c_D	spring stiffness of gasket	N/mm
$c_{D.KNS}$	spring stiffness of gasket in metal-to-metal contact type flanged joints (in case of spring-back)	N/mm
c_S	spring stiffness of bolts	N/mm
d_1	loose flange ring I. D.	mm
d_2	loose flange ring O. D.	mm
d_D	mean diameter or diameter of gasket contact face	mm
d_{D1}. d_{D2}	mean diameter for double O-ring gasket	mm

Notation	Design value	Unit
d_F	flange or stub-end outside diameter	mm
d_{FA}	outer diameter of flange surface	mm
$d_{Kontakt}$	diameter of area of force application ($F_{Kontakt}$) for metal-to-metal contact type flanged joints	mm
d_M	outer diameter of flange face contact area of metal-to-metal contact type flanged joints	mm
d_L	bolt hole diameter	mm
d'_L	bolt circle design diameter	mm
d_i	inside diameter of pipe, shell, or flange ring	mm
d_t	bolt circle diameter	mm
d_t^*	fictitious bearing surface diameter of loose flanges on stub ends (see figures A 2.10-3, A 2.10-5, A 2.10-6)	mm
e_1. e_2	distance to centroid of flange	mm
f	height of flange facing	mm
g_{KNS}	relaxation factor	—
h	flange thickness	mm
h_A	height of tapered hub	mm
h_B	effective portion of flange skirt on the stiffness of flanged connection	mm
h_D	height of gasket	mm
h_F	effective flange thickness	mm
h_L	thickness of loose flange ring	mm
h_S	flange thickness required to withstand shear stress in section C-C	mm
n	number of bolt holes	—
p	design pressure	MPa
p'	test pressure	MPa
$p_{KNS/L}$	sealable pressure for metal-to-metal contact type flanged joints	MPa
r. r_1	transition radius, see clause 5.2.4 (3)	mm
s_1	required pipe or shell wall thickness for longitudinal force	mm
s_F	thickness of hub at transition to flange	mm
s_R	pipe or shell wall thickness	mm
s_x	wall thickness at section X-X	mm
t	bolt pitch	mm
x_S	bolt elongation	mm
A	cross-sectional area	mm^2
A_1. A_2	partial cross-sectional areas according to **Figure A 2.10-1**	mm^2
C_F	torsional stiffness of flange	$\frac{N \cdot mm}{rad}$
E_B, E_D, E_F, E_S	modulus of elasticity of blank, gasket, flange and bolt materials respectively	N/mm^2
E_{FT}	modulus of elasticity of flange material at temperature	N/mm^2
F_{BZ}	additional force on the blank	N
F_D	compression load on gasket	N
F_{DB}	compression load on gasket for operating condition	N
F_F	difference between total hydrostatic end force and the hydrostatic end force on area inside flange	N
F_i	hydrostatic end force	N
$F_{Kontakt}$	force applied on metal-to-metal contact area for metal-to-metal contact type flanged joints	N
F_R	total hydrostatic end force	N
F_{RP}	hydrostatic end force due to internal pressure	N
F_S	bolt load	N
F_{S0}	bolt load for bolting-up condition	N
F_{S0max}	maximum bolt load possible on account of tightening procedure for gasket seating condition; determined e.g. to VDI 2230, Sheet 1	N
F_{S0U}	bolt load for gasket seating condition (lower limit)	N
F_{SB}	bolt load at operating condition assigned to the respective loading level	N
K. L	factors	—
M	external twisting moment at load case considered	Nmm
M_E	external twisting moment for metal-to-metal contact type flanged joints at bolting-up condition	Nmm
S, S_m	design stress intensity factor in acc. with Table 6.6-1	N/mm^2
S_{P1}. S_{P2}	centroids of partial cross-sectional area, $A_1 = A_2$	—
W	flange section modulus	mm^3
W_A	flange section modulus for section A-A	mm^3
W_B	flange section modulus for section B-B	mm^3
W_{erf}	required flange section modulus	mm^3
W_{vorh}	available flange section modulus	mm^3
W_X	flange section modulus for section X-X	mm^3
α	coefficient of thermal expansion	$°C^{-1}$
γ_{zul}	allowable flange blade angle of inclination to the plane vertical to flange axis	degree
λ	specific leakage rate	$mg/(s \cdot m)$
σ_{Vx}	gasket contact surface load	N/mm^2
σ_{zul}	allowable stress acc. to **Table A 2.10-1**	N/mm^2
Δh	allowable spring-back from full metal-to-metal contact position for the respective pressure rating and tightness class in acc. with **Form A 2.11-2**	mm
$\Delta s_{1.2}$	portion of gap increase (flange blade 1 and 2) due to inclination of flange blade, for metal-to-metal contact type flanged joints	mm

The index “0” refers to the bolting-up/gasket seating condition, and the index “x” to the respective condition under consideration (operating condition, test condition).

A 2.10.2 General

(1) The calculation hereinafter applies to the dimensioning and proof of strength of steel flanges which as friction-type flanged joints of the floating type (KHS) and metal-to-metal contact type (KNS) are subject to internal pressure. Prerequisite to the use of metal-to-metal contact type flange joints is their sufficient stiffness and thus limited gap height within the gasket area. The flanges hereinafter comprise welding-neck flanges, welding stubs, welded flanges and stubs as well as lap-joint flanges and cover flanges.

(2) The tightness shall be proved using the minimum design bolt load. Deviating here from, the proof may be based on the average design bolt load in the case of metal-to-metal contact type flanged joints with a number of bolts n equal to or exceeding 8. In the case of metal-to-metal contact type flanged joints with a number of bolts n less than 8 the gap increase at the gasket shall be verified using the maximum bolt load.

The strength of flange and gasket at bolting-up condition shall be verified taking credit of the maximum design bolt load. The proof of strength at operating condition may be based on the average design bolt load.

(3) Where proof of adequate leak tightness is required for loading levels C and D, it shall be made by substantiating, by way of calculation, the strength and deformation conditions in conformance with clause A 2.10.6 or A 2.10.7.

A 2.10.3 Construction and welding

(1) Vessel flanges may be forged or rolled without seam.

(2) Welding and heat treatment, if required, shall be based on the component specifications.

(3) For flanged joints on nozzles and piping welding-neck flanges to DIN standards or welding-neck flanges with standard dimensions shall be used. The strength calculation of bolted joints for standard pipe flanges to DIN EN 1092-1 may be waived if for pipe flanges with PN ≤ 25 the following measures are taken:

a) under internal pressure loading alone the next higher nominal pressure rating shall be chosen,
b) for loading under internal pressure and external forces the shall be chosen.

A 2.10.4 Dimensioning of flanges for floating type flanged joints

A 2.10.4.1 General

(1) The calculation consists of the dimensioning and proof of tightness and strength to clause A 2.10.6. The flanged joint shall be so dimensioned that the forces during assembly (gasket seating condition), pressure testing, operation and start-up and shutdown operations and incidents, if any, can be withstood.

Where the test pressure $p' > p \cdot \frac{\sigma_{zul\ test\ condition}}{\sigma_{zul\ operating\ condition}}$

the calculation shall also be made for this load case. The condition shall be checked for both the flange and bolt materials.

(2) The flanges shall be calculated using the equations given in the paragraphs hereinafter. The effects of external forces and moments shall be considered and verified.

(3) The flange thickness h_F or h_L on which the calculation is based shall be provided on the fabricated component. Grooves for normal tongue or groove or ring joint facings need not be considered.

(4) The required flange section modulus W_{erf} shall govern the flange design.

(5) For the determination of the required section modulus for the operating condition of flanges as per clauses A 2.10.4.2 and A 2.10.4.3 in Sections A-A and B-B and for flanges as per clause A 2.10.4.4 in section A-A the following applies:

$$W_{erf} = \frac{(F_{DBU/L} + F_Z) \cdot a_D + F_R \cdot a_R + F_F \cdot a_F}{\sigma_{zul}} \qquad \text{(A 2.10-1)}$$

For the mentioned flanges in section C-C the following applies:

$$W_{erf} = \frac{F_{SBU/L} \cdot a_1}{\sigma_{zul}} \qquad \text{(A 2.10-2)}$$

For the flanges as per clause A 2.10.4.5 the following applies:

$$W_{erf} = \frac{F_{SBU/L} \cdot a_D}{\sigma_{zul}} \qquad \text{(A 2.10-3)}$$

For the bolting-up condition the following applies to flanges as per clauses A 2.10.4.2 to A 2.10.4.5 irrespective of the sections:

$$W_{erf} = \frac{F_{S0U} \cdot a_D}{\sigma_{zul}} \qquad \text{(A 2.10-4)}$$

where σ_{zul} is the allowable stress as per **Table A 2.10-1**.

Note:

*The maximum bolt assembly load F_{S0max} shall be considered within the proof of strength, see **Table A 2.10-1** ser. no. 3.*

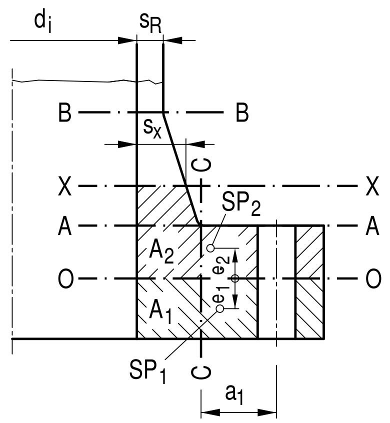


Figure A 2.10-1: Flange cross-section

The equations (A 2.10-1) to (A 2.10-3) may be applied accordingly for the test condition.

The forces F shall be determined in accordance with Section A 2.9.

The moment arms for gaskets in floating-type flanged joints shall be:

$$a_D = \frac{d_t - d_D}{2} \qquad \text{(A 2.10-5)}$$

$$a_R = \frac{d_t - d_i - s_R}{2} \qquad \text{(A 2.10-6)}$$

$$a_F = \frac{2 \cdot d_t - d_D - d_i}{4} \qquad \text{(A 2.10-7)}$$

For stubs d_t shall be inserted as bolt circle diameter d_t^* (see **Figure A 2.10-3** and **A 2.10-5**).

For lap-joint flanges the following applies:

$$a = a_D = \frac{d_t - d_t^*}{2} \qquad \text{(A 2.10-8)}$$

The use of $d_t^* = \frac{d_1 + 2 \cdot r + d_F}{2}$ for calculating the flange and of $d_t^* = d_F$ for calculating the hub are conservative approaches.

d_t^* may be adapted to the actual conditions in dependence of the hub and flange stiffness ratios.

(6) The flange section modulus shall meet the general condition for any arbitrary section X-X (**Figure A 2.10-1**)

$$W_x = 2 \cdot \pi \cdot \left[A_1 \cdot (e_1 + e_2) + \frac{1}{8} \cdot (d_i + s_x) \cdot \left(s_x^{\,2} - s_1^{\,2}\right) \right] \qquad \text{(A 2.10-9)}$$

Here, s_1 is the wall thickness required due to the longitudinal forces in the flange hub, and is calculated by means of the following equation:

$$s_1 = \frac{F_R}{\pi \cdot (d_i + s_R) \cdot \sigma_{zul}} \qquad \text{(A 2.10-10)}$$

σ_{zul} shall be determined in acc. with **Table A 2.10-1**. The factor Φ may be omitted in equation A 2.10-10.

With e_1 and e_2 the centroids of the partial cross-sectional areas $A_1 = A_2$ (shown in **Figure A 2.10-1** as differing hatched areas) adjacent to the neutral line 0-0 are meant, with this neutral line being applicable to the fully plastic condition assumed. The weakening of the flange by the bolt holes shall be considered in the calculation by means of the design diameter d_L' in the following equation:

For flanges with $d_i \geq 500$ mm

$$d_L' = d_L/2 \qquad \text{(A 2.10-11)}$$

and for flanges with $d_i < 500$ mm

$$d_L' = d_L \cdot (1 - d_i/1000) \qquad \text{(A 2.10-12)}$$

A 2.10.4.2 Welding-neck flanges with gasket inside bolt circle and tapered hub according to Fig. A 2.10-2

The flange shall be checked with regard to the sections A-A, B-B and C-C where the smallest flange section modulus shall govern the strength behaviour.

The flange section modulus available in section A-A is obtained from:

$$W_A = \frac{\pi}{4} \cdot \left[\left(d_F - d_i - 2 \cdot d_L' \right) \cdot h_F^2 + (d_i + s_F) \cdot \left(s_F^2 - s_1^2 \right) \right] \geq W_{erf} \qquad \text{(A 2.10-13)}$$

Equation (A 2.10-13) may also be used for the determination of h_F.

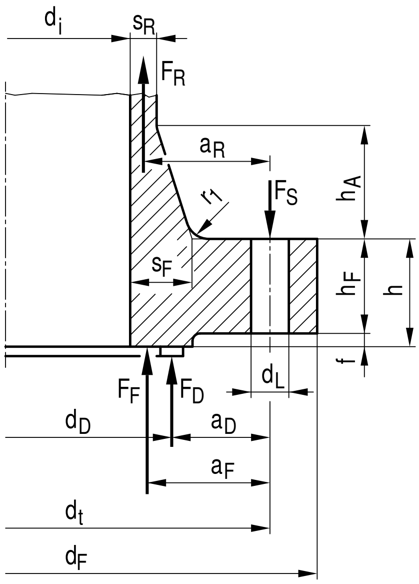


Figure A 2.10-2: Welding-neck flange with tapered hub

The flange section modulus available in section B-B is obtained from:

$$W_B = \pi \cdot \left[2 \cdot \left(d_F - d_i - 2 \cdot d_L' \right) \cdot e_1 \cdot (e_1 + e_2) + \frac{1}{4} \cdot (d_i + s_R) \cdot \right.$$
$$\left. \cdot \left(s_R^2 - s_1^2 \right) \right] \geq W_{erf} \qquad \text{(A 2.10-14)}$$

The centroids e_1 and e_2 for flanges with tapered hub are:

$$e_1 = \frac{1}{4} \cdot \left(h_F + \frac{h_A \cdot (s_F + s_R)}{d_F - d_i - 2 \cdot d_L'} \right) \qquad \text{(A 2.10-15)}$$

$$e_2 = \frac{K}{L} \qquad \text{(A 2.10-16)}$$

where

$$K = 0.5 \cdot \left(d_F - d_i - 2 \cdot d_L' \right) \cdot (h_F - 2 \cdot e_1)^2 +$$
$$+ h_A \cdot (h_F - 2 \cdot e_1) \cdot (s_F + s_R) + \frac{h_A^2}{3} \cdot (s_F + 2 \cdot s_R) \qquad \text{(A 2.10-17)}$$

$$L = (d_F - d_i - 2 \cdot d_L')\,(h_F - 2 \cdot e_1) + h_A \cdot (s_F + s_R) \qquad \text{(A 2.10-18)}$$

The flange thickness h_S required to absorb the shear stress is obtained as follows:

for the bolting-up condition

$$h_{S0} = \frac{2 \cdot F_{S0}}{\pi \cdot (d_i + 2 \cdot s_F) \cdot \sigma_{zul}} \qquad \text{(A 2.10-19)}$$

for the operating condition

$$h_{SB} = \frac{2 \cdot F_{SB}}{\pi \cdot (d_i + 2 \cdot s_F) \cdot \sigma_{zul}} \qquad \text{(A 2.10-20)}$$

where σ_{zul} is the allowable stress as per **Table 2.10-1**.

The flange section modulus in section C-C is obtained from:

$$W_C = \frac{\pi}{4} \cdot \left[h_F^2 \cdot \left(d_F - 2 \cdot d_L' \right) - h_S^2 \cdot (d_i + 2 \cdot s_F) \right] \qquad \text{(A 2.10-21)}$$

In this case, the external moment shall be

$$M_C = F_S \cdot a_1 \qquad \text{(A 2.10-22)}$$

with $F_S = F_{S0U}$ at gasket seating condition
$F_S = F_{SBU/L}$ at operating condition.

The application of equation A 2.10-21 may lead to strongly conservative results, e.g. in the case of $d_D > (d_i + 2 \cdot s_F)$. Detailed examinations to consider lever arm and geometry conditions are permitted.

A 2.10.4.3 Welding stubs with tapered hub according to Figure A 2.10-3

The calculation shall be made in accordance with clause A 2.10.4.2 with $d_L' = 0$.

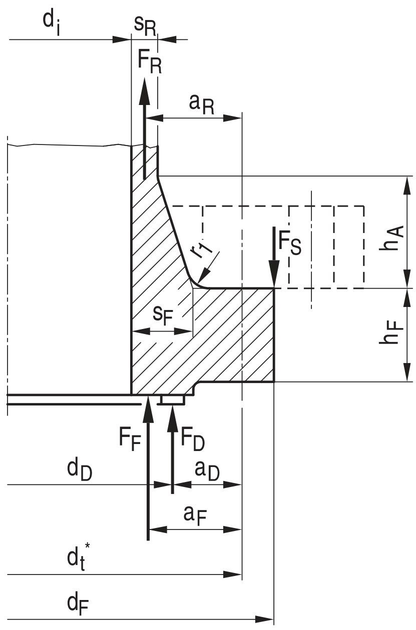


Figure A 2.10-3: Welding stub with tapered hub

A 2.10.4.4 Flanges and stubs with gasket inside bolt circle and cylindrical hub in accordance with Figure A 2.10-4 and Figure A 2.10-5

The flange shall be checked with regard to sections A-A and C-C. The flange section modulus available in section A-A is obtained from:

$$W_A = \frac{\pi}{4}\cdot\left[\left(d_F - d_i - 2\cdot d_L'\right)\cdot h_F^2 + \left(d_i + s_R\right)\cdot\left(s_R^2 - s_1^2\right)\right] \geq W_{erf} \qquad \text{(A 2.10-23)}$$

The flange section modulus available in section C-C is obtained in accordance with clause A 2.9.4.2.

For the calculation of welding stubs d_L' = 0 shall be taken.

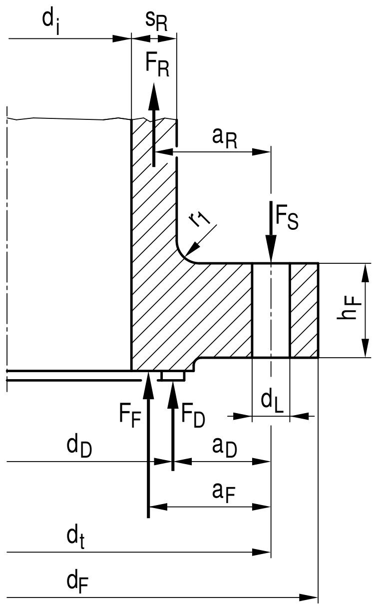


Figure A 2.10-4: Welding-neck flange with cylindrical hub

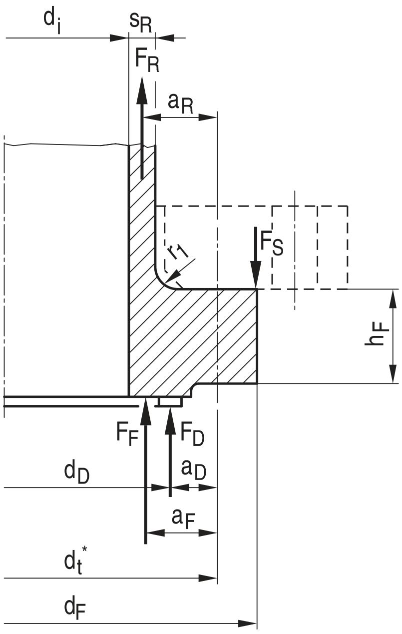


Figure A 2.10-5: Welding stub with cylindrical hub

A 2.10.4.5 Lap-joint flanges to Figure A 2.10-6

The required flange thickness shall be

$$h_L = \sqrt{\frac{4\cdot W_{erf}}{\pi\cdot\left(d_2 - d_1 - 2\cdot d_L'\right)}} \qquad \text{(A 2.10-24)}$$

with W_{erf} obtained from equation (A 2.10-3).

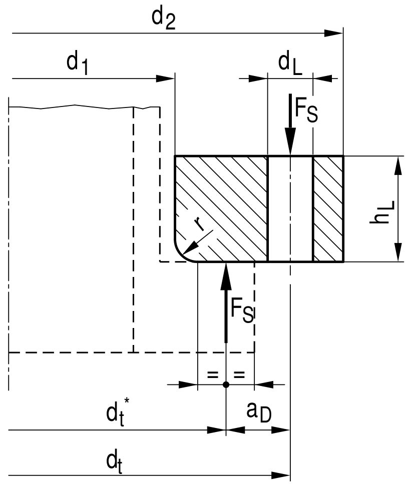


Figure A 2.10-6: Lap-joint flange

A 2.10.5 Dimensioning of flanges of metal-to-metal contact type flanges

(1) In the case of metal-to-metal contact type flanges adequate stiffness and thus limited gap height in the gasket area is required.

(2) The flange section modulus required to provide adequate stiffness is calculated as follows

$$W_{erf} = \frac{0.75\cdot M\cdot\left(d_F + d_i\right)}{E_{FT}\cdot\left(h_F + h_B\right)\cdot\gamma_{zul}}\cdot\frac{180°}{\pi}\cdot\frac{1}{f_{CF}} \qquad \text{(A 2.10-25)}$$

where

$$\gamma_{zul} = \frac{\Delta s_{1,2}\cdot 180°}{\left(a_D - a_M\right)\cdot\pi} \qquad \text{(A 2.10-26)}$$

f_{CF}: ratio of effective flange torsional rigidity to the torsional stiffness determined by calculation to equation A 2.10-37

Where no other values are available, the following values shall be taken for f_{CF}:

f_{CF} = 0.8 for flanges with cylindrical hub

f_{CF} = 0.9 for flanges with tapered hub

The twisting moment M for the cases to be considered is determined as follows:

a) Gasket seating condition

$M = M_E = F_{DKU}\cdot a_D + F_{R0}\cdot a_R + F_{Z0}\cdot a_{Reib}$ (A 2.10-27)

b) Normal and anomalous as well as test condition

$M = F_{DKU}\cdot g_{KNS}\cdot a_D + F_R\cdot a_R + F_F\cdot a_F + F_Z\cdot a_{Reib}$ (A 2.10-28)

The sum of maximum gap increase values of both flange blades $\Delta s_1 + \Delta s_2$ shall be less than the allowable spring-back from full metal-to-metal contact position Δh as indicated by the manufacturer in **Form A 2.11-2** for the respective tightness class.

For tapered-hub flanges the available flange section modulus $W = W_A$ shall be determined to equation (A 2.10-13). In addition, the following applies:

$$h_B = 0.58\cdot\left(\frac{d_i}{s_F}\right)^{0.29}\cdot h_A \qquad \text{(A 2.10-29)}$$

For welding-neck flanges where the pipe or shell attach to the flange without tapered hub, the available flange section modulus $W = W_A$ shall be determined to equation (A 2.10-23). In addition, the following applies:

$$h_B = 0.9\cdot\sqrt{\left(d_i + s_R\right)\cdot s_R} \qquad \text{(A 2.10-30)}$$

(3) The flange section modulus to provide adequate strength is calculated to obtain:

$$W_{erf} = \frac{M}{\sigma_{zul}} \qquad \text{(A 2.10-31)}$$

A 2.10.6 Proof of tightness and strength for floating type flange joints

A 2.10.6.1 General

(1) During start-up and shutdown, the relation between bolt load, pressure load and gasket load in the flange changes due to internal pressure, additional forces and moments independent of operation, temperature-dependent change of elastic moduli, differential thermal expansion, seating of the gasket, especially of non-metallic gaskets.

(2) Based on the selected initial bolt stress and in consideration of the elastic deflection characteristics of the flanged joint with consistent bolt elongation the bolt load and the residual gasket load shall be evaluated in consideration of torsional moments and transverse forces to be transferred for each governing load case.

In the case of identical flange pairs, consistent bolt elongation means the sum of the deflections of the flange $2 \cdot \Delta F$, the bolts ΔS and the gasket ΔD, in case of temperature effects, of the differential thermal expansion in the flange and the bolt ΔW as well as, in the case of seating of the gasket, in the bolted joint and in the gasket ΔV. Taking these magnitudes into account, the bolt elongation in the bolting-up condition E will be consistent for each operating condition x:

$$2 \cdot \Delta F_E + \Delta S_E + \Delta D_E = 2 \cdot \Delta F_x + \Delta S_x + \Delta D_x + \Delta W_x + \Delta V_x \quad \text{(A 2.10-32)}$$

In the case of non-identical flange pairs, $2 \cdot \Delta F$ is substituted by the sum of deflections of the individual flanges $\Delta F_1 + \Delta F_2$, in the case of flange-cover joints $2 \cdot \Delta F$ is substituted by the sum of deflections of the flange and the cover $\Delta F + \Delta B$.

In the case of flanged joints with extension sleeves the stiffness of the extension sleeves shall also be taken into account.

(3) By means of the bolt and gasket loads resulting from the verification by calculation of the strength and deformation conditions for the governing load cases the evaluation of strength of the total flanged joint (flange, blank, bolts and gasket) shall be controlled.

(4) The allowable stresses for flanges shall be taken from **Table A 2.10-1** ser. no. 4. Apart from these allowable stresses, the determination of the flange section moduli shall be based on the force F_{Rx} when using equation A 2.10-10 and on the force F_{SBx} when using equation A 2.10-20.

(5) A more general method for proof of tightness and strength of floating type flanged joints is shown in **Figure A 2.10-8**.

(6) The initial bolt stress required to calculate the operational loadings shall first be determined to clause A 2.9.4.1 ($F_{S0} = F_{S0U}$) even if no dimensioning is required.

(7) For the initial bolt stress the gasket contact surface load σ_V shall be calculated with which the minimum gasket seating load $\sigma_{BU/L}$ at operating condition for the required tightness class is determined, see clause A 2.11.2.

(8) Where the individual conditions in **Figure 2.10-8** are not satisfied, an iterative process shall be applied.

A 2.10.6.2 Simplified procedure for verification by calculation of the strength and deformation conditions in flanged joints

A 2.10.6.2.1 General

(1) For some cases where internal pressure, additional forces and moments, temperature-dependent changes in elastic moduli, different thermal expansion in the flange and the bolts, as well seating of the gasket occurs, equations are given in the following clauses to determine the bolt loads F_S, the compression loads on the gasket F_D as well as the deflections ΔF, ΔS and ΔD for the respective conditions.

(2) Alternatively, an approximate calculation for verifying the strength and deformation conditions may be made by other procedures for a detailed evaluation of the

a) torsional rigidity of flanges,

b) radial internal pressure,

c) effective bolt circle diameter,

d) effective gasket diameter and effective gasket width.

A 2.10.6.2.2 Calculation of spring stiffnesses

A 2.10.6.2.2.1 Bolts

The elastic elongation of bolts can be calculated from

$$\Delta S = \frac{F_S}{c_S} \quad \text{(A 2.10-33)}$$

For full-shank bolts the following applies approximately

$$c_S = \frac{n \cdot \pi \cdot E_S \cdot d_N^2}{4 \cdot (l + 0.8 \cdot d_N)} \quad \text{(A 2.10-34)}$$

For reduced-shank bolts the following applies

$$c_S = \frac{n \cdot \pi \cdot E_S}{4} \cdot \frac{d_K^2 \cdot d_S^2}{d_K^2 \cdot l_S + d_S^2 \cdot (l' + l'' + 0.8 \cdot d_N)} \quad \text{(A 2.10-35)}$$

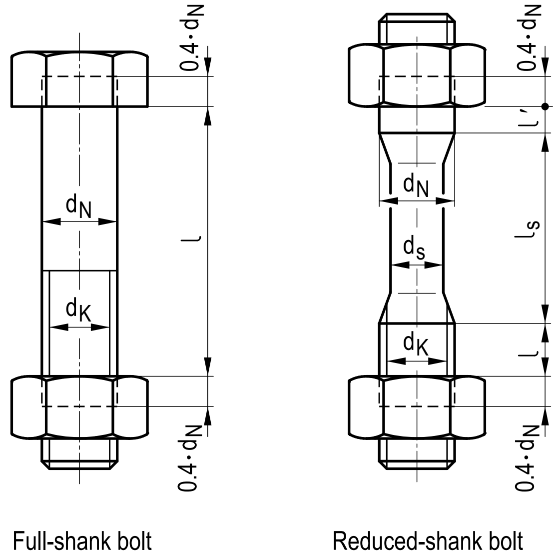


Figure A 2.10-7: Bolts

A 2.10.6.2.2.2 Flanges

The deflection ΔF of the individual flange in the bolt circle is

$$\Delta F = \frac{M \cdot a_D}{C_F} \quad \text{(A 2.10-36)}$$

When determining the relation between bolt load, pressure load and gasket load of a pair of identical flanges, twice the value of ΔF shall always be taken.

$$C_F = \frac{4 \cdot E_F \cdot (h_F + h_B) \cdot W}{3 \cdot (d_F + d_i)} \quad \text{(A 2.10-37)}$$

For flanges with tapered hub $W = W_A$ according to equation (A 2.10-13).

Note:

A tapered hub is assumed to be present, if the following conditions are met:

$$0.2 \le \frac{s_F - s_R}{h_A} \le 0.5$$

and

$$\frac{h_A}{h_F} \ge 0.5$$

In addition, the following applies

$$h_B = 0.58 \cdot \left(\frac{d_i}{s_F}\right)^{0.29} \cdot h_A \qquad \text{(A 2.10-38)}$$

For welded flanges where the pipe or shell is connected to the flange without tapered hub, the following applies

$$W = \frac{\pi}{4} \cdot \left[\left(d_F - d_i - 2 \cdot d'_L\right) \cdot h_F^2 + \left(d_i + s_R\right) \cdot \left(s_R^2 - s_1^2\right)\right] \qquad \text{(A 2.10-39)}$$

In addition, the following applies

$$h_B = 0.9 \cdot \sqrt{\left(d_i + s_R\right) \cdot s_R} \qquad \text{(A 2.10-40)}$$

For lap joint flanges the following applies

$$W = \frac{\pi}{4} \cdot \left(d_2 - d_1 - 2 \cdot d'_L\right) \cdot h_L^2 \qquad \text{(A 2.10-41)}$$

and $h_B = 0$.

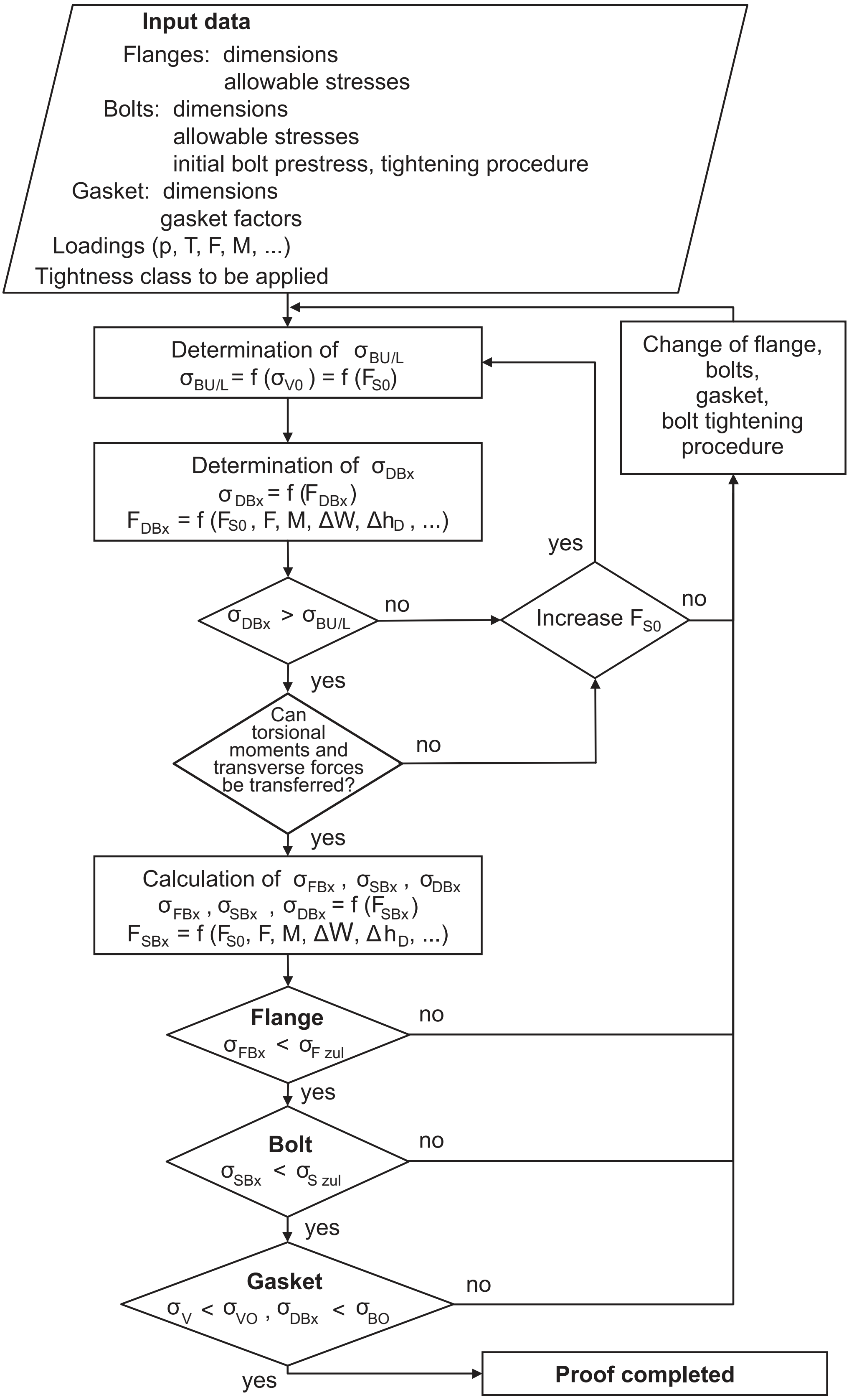


Figure A 2.10-8: Schematic procedural steps for verification of strength of floating type flanged joints

A 2.10.6.2.2.3 Blanks

The deflection ΔB of the blank in the bolt circle for the bolting-up condition (condition 0) shall be:

$$\Delta B_0 = \frac{F_{S0}}{c_{B0}} \qquad \text{(A 2.9-42)}$$

with F_{S0} = F_{D0} bolt load for bolting-up condition

and c_{B0} spring stiffness for bolting-up condition

and for and for the operating condition (condition x):

$$\Delta B_x = \frac{p \cdot \frac{d_D^2 \cdot \pi}{4} + F_{BZ}}{c_{Bxp}} + \frac{F_{Dx}}{c_{BxF_D}} \qquad \text{(A 2.9-43)}$$

where the force F_{Bx} on the cover shall be

$$F_{Bx} = p \cdot \frac{d_D^2 \cdot \pi}{4} + F_{BZ} = F_{RP} + F_F + F_{RZ} \qquad \text{(A 2.9-44)}$$

and

c_{Bxp} = spring stiffness for the loading due to force on cover

and

$c_{BxF_D} = c_{B0} \cdot \frac{E_{BT}}{E_{BRT}}$ = spring stiffness for the loading due to compression load on the gasket F_{Dx}

The spring stiffness for the various types of loading may e.g. be taken from

a) Markus [8]

b) Warren C. Young, case 2a, p. 339 [9]

c) Kantorowitsch [10]

or be determined by suitable methods.

A 2.10.6.2.2.4 Gaskets

The elastic portion of compression (spring-back) of the gasket ΔD can be assumed to be, for flat gaskets

$$\Delta D = \frac{F_D}{c_D} \qquad \text{(A 2.10-45)}$$

where

$$c_D = \frac{E_D \cdot \pi \cdot d_D \cdot b_D}{h_D} \qquad \text{(A 2.10-46)}$$

Depending on the load case E_D is the elastic modulus of the gasket material at bolting-up condition or operating temperature.

For metal gaskets of any type the springback of the gasket is so low in comparison with the flange deflection that it can be neglected.

A 2.10.6.2.2.5 Differential thermal expansion and additional

The equations for calculating the bolt loads and gasket compression loads according to clause A 2.10.6.2.3 may also consider differential thermal expansions between flange, blank, bolts, and gasket as well as time-dependent seating:

$$\Delta W_x = l_k \cdot \alpha_S \cdot (T_{Sx} - 20°) - h_{F1} \cdot \alpha_{F1} \cdot (T_{F1x} - 20°) - h_{F2} \cdot \alpha_{F2} \cdot (T_{F2x} - 20°) - h_D \cdot \alpha_D \cdot (T_{Dx} - 20°) \qquad \text{(A 2.10-47)}$$

where

$(\Delta W)_x$ = differential thermal expansion of flange, blank, bolt, and gasket. The indices 1 and 2 refer to the flange and the mating flange or blank

l_k = grip length (distance between idealized points of effective bolt elongation)

$(\Delta h_D)_x$ = time-dependent gasket seating (to be considered for non-metallic gaskets and combined seals only in which case the manufacturer's data shall be taken as a basis).

A 2.10.6.2.3 Calculation of bolt loads and compression loads on the gasket

A 2.10.6.2.3.1 Case of identical flange pairs

For identical flange pairs the following applies:

$$F_{DBx} = \frac{1}{\frac{1}{c_{Sx}} + \frac{2 \cdot a_D^2}{C_{Fx}} + \frac{1}{c_{Dx}}} \cdot \left[F_{S0} \cdot \left(\frac{1}{c_{S0}} + \frac{2 \cdot a_D^2}{C_{F0}} + \frac{1}{c_{D0}} \right) - F_{Rx} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_R}{a_D} \cdot \frac{2 \cdot a_D^2}{C_{Fx}} \right) - F_{Fx} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_F}{a_D} \cdot \frac{2 \cdot a_D^2}{C_{Fx}} \right) - \Delta W_x - \Delta h_{DX} \right] \qquad \text{(A 2.10-48)}$$

and

$$F_{SBx} = F_{DBx} + F_{Rx} + F_{Fx} \qquad \text{(A 2.10-49)}$$

A 2.10.6.2.3.2 Case of non-identical flange pairs

For flanged joints with non-identical flanges 1 and 2 the following applies:

$$F_{DBx} = \frac{1}{\frac{1}{c_{Sx}} + \frac{a_D^2}{C_{F1x}} + \frac{a_D^2}{C_{F2x}} + \frac{1}{c_{Dx}}} \cdot \left[F_{S0} \cdot \left(\frac{1}{c_{S0}} + \frac{a_D^2}{C_{F10}} + \frac{a_D^2}{C_{F20}} + \frac{1}{c_{D0}} \right) - F_{R1x} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_{R1}}{a_D} \cdot \frac{a_D^2}{C_{F1x}} \right) - F_{R2x} \cdot \frac{a_{R2}}{a_D} \cdot \frac{a_D^2}{C_{F2x}} - F_{F1x} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_{F1}}{a_D} \cdot \frac{a_D^2}{C_{F1x}} \right) - F_{F2x} \cdot \frac{a_{F2}}{a_D} \cdot \frac{a_D^2}{C_{F2x}} - \Delta W_x - \Delta h_{DX} \right] \qquad \text{(A 2.10-50)}$$

$$F_{SBx} = F_{DBx} + F_{Rx} + F_{Fx} \qquad \text{(A 2.10-51)}$$

and

$$F_{R1x} + F_{F1x} = F_{R2x} + F_{F2x} \qquad \text{(A 2.10-52)}$$

A 2.10.6.2.3.3 Flange-blank combination

For flanged joints consisting of a flange and a blank the following applies:

$$F_{DBx} = \frac{1}{\frac{1}{c_{Sx}} + \frac{a_D^2}{C_{Fx}} + \frac{1}{c_{BxFD}} + \frac{1}{c_{Dx}}} \cdot \left[F_{S0} \cdot \left(\frac{1}{c_{S0}} + \frac{a_D^2}{C_{F0}} + \frac{1}{c_{B0}} + \frac{1}{c_{D0}} \right) - F_{Rx} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_R}{a_D} \cdot \frac{a_D^2}{C_{Fx}} \right) - F_{Fx} \cdot \left(\frac{1}{c_{Sx}} + \frac{a_F}{a_D} \cdot \frac{a_D^2}{C_{Fx}} \right) - F_{Bx} \cdot \frac{1}{c_{Bxp}} - \Delta W_x - \Delta h_{DX} \right] \qquad \text{(A 2.10-53)}$$

and

$$F_{Sx} = F_{DBx} + F_{Rx} + F_{Fx} \qquad \text{(A 2.10-54)}$$

A 2.10.7 Proof of tightness and strength of metal-to-metal contact-type flanged joints

A 2.10.7.1 General

(1) The full metal-to-metal contact of flange blade faces ($F_{Kontakt} \geq 0$) shall be maintained at any relevant loading to ensure that the required properties of the metal-to-metal contact type flange joint are satisfied.

(2) The gasket will only transfer a portion of the initial bolt stress.

(3) Depending of the geometry the gap in the gasket area may increase in between the period of time where full metal-to-metal contact and the respective operating conditions are obtained. This increase in gap shall be compensated by the gasket spring-back capability Δh (see **Form A 2.11-2**).

Note:

The increase in gap shall be evaluated on the basis of a spring-back curve representative for the selected type of gasket. The gap increase at the time between gasket seating (obtaining full metal-to-metal contact) and the operating conditions is usually limited to 0.1 mm in case of gasket thicknesses of 4.5 mm and groove depths of 3.3 mm (unless other data are contained in ***Form A 2.11-2****), as the sealing behaviour of spiral gaskets and graphite profile rings is only slightly impaired in case of gasket spring-back to a value of 0.1 mm (see literature [11] and [12]).*

(4) Where the gap dimension changes (e.g. due to differing thermal expansion of the rigged flange components or due to piping loads), the gasket may be subject to relaxation. The change in leakage rate thus caused shall be determined by the relaxation calculated to **Figure A 2.11-5** and the data contained in **Form A 2.11-2**.

(5) **Figure A 2.10-9** shows a general method as to the performance of proofs of tightness and strength for metal-to-metal contact type flanged joints.

(6) The leakage rate of the flanged joint shall be determined by means of the gasket contact surface load σ_{DB} calculated to **Figure A 2.11-1**. The leakage rate shall be less than that required by tightness class L, and the already performed steps (verification of stiffness, determination of bolt load) shall be repeated.

(7) Where the individual conditions of **Figure A 2.10-9** are not satisfied, an iterative process shall be applied.

A 2.10.7.2 Simplified procedure for verification by calculation of the strength and deformation conditions

A 2.10.7.2.1 General

(1) For some cases where internal pressure, additional forces and moments, temperature-dependent changes in elastic moduli, different thermal expansion in the flange and in the bolts, as well as seating of the gasket occur, equations are given in the following clauses to approximately determine the bolt loads F_S, the compression loads on the gasket F_D, the flange moments M as well as the gap increase Δs for the respective conditions.

(2) Alternatively, an approximate calculation for verifying the strength and deformation conditions may be made by other procedures for a detailed evaluation of the

a) torsional rigidity of flanges,

b) radial internal pressure,

c) effective bolt circle diameter.

A 2.10.7.2.2 Input values

(1) The determination of the differences in thermal expansion ΔW_x (except for the gasket) as well as the determination of the spring stiffness for bolts and blanks shall be made in accordance with A 2.10.6.2.2. The flange spring stiffnesses shall be calculated in accordance with clause A 2.10.6.2.2 in consideration of the reduction factors f_{C_F} (see sub-clause A 2.10.5 (2).

(2) The gasket load required to obtain full metal-to-metal contact F_{DKU} shall be determined according to equation (A 2.9-17).

(3) The spring stiffness of metal-to-metal contact type flanged joints shall be derived from the gasket spring-back curve or **Form A 2.11-2** as follows:

$$c_{D,KNS} = \pi \cdot d_D \cdot b_D \cdot E_{D,KNS}/h_D \qquad \text{(A 2.10-55)}$$

Here, depending on the load case, $E_{D,KNS}$ is the elastic modulus of the gasket material at assembly or operating temperature.

(4) The distance of bolt hole centre to outer point of contact between the two flange blades, a_M, shall be taken as lever arm of the contact forces $a_{Kontakt}$. In the case of loose-type flanges this is the distance from bolt centre to stub-end outside diameter d_F. A more exact calculation to the following equation is permitted (iterative procedure with the initial value $a_{Kontakt} = a_M$ when determining the force $F_{Kontakt,0}$):

$$a_{Kontakt} = a_M + \frac{F_{Kontakt\,0}}{2 \cdot d_M \cdot \pi \cdot R_{mRT}} \qquad \text{(A 2.10-56)}$$

In the case of dissimilar flange ring materials the tensile strength R_{mRT} of the weaker flange ring material shall be taken.

A 2.10.7.2.3 Case of identical flange pairs

A 2.10.7.2.3.1 Gasket seating condition

$$x_{S,\,0} = F_{S0} / c_{S0} \qquad \text{(A 2.10-57)}$$

$$\gamma_{F,\,KNS} = F_{DKU} \cdot a_D / C_{F,0} \qquad \text{(A 2.10-58)}$$

$$F_{Kontakt\,0} = \frac{F_{S0} - F_{DKU} + (a_D - a_{Kontakt}) \cdot a_D \cdot c_{DKNS_0} \cdot \left\{\frac{2 \cdot (F_{S0} - F_{DKU})}{C_{F0}}\right\}}{1 + (a_D - a_{Kontakt})^2 \cdot c_{DKNS_0} \cdot \left(\frac{2}{C_{F0}}\right)} \qquad \text{(A 2.10-59)}$$

$$\gamma_{F,0} = \frac{F_{S0} \cdot a_D}{C_{F0}} - \frac{F_{Kontakt\,0}}{C_{F0}} \cdot (a_D - a_{Kontakt}) \qquad \text{(A 2.10-60)}$$

$$F_{D_0} = F_{DKU} - \{2 \cdot (\gamma_{F_0} - \gamma_{F_{KNS}})\}\,(a_D - a_{Kontakt}) \cdot c_{D_{KNS_0}} \qquad \text{(A 2.10-61)}$$

Gap increase at gasket diameter d_D:

$$\Delta s_0 = 2 \cdot (a_D - a_{Kontakt}) \cdot (\gamma_{F_0} - \gamma_{F_{KNS}}) \qquad \text{(A 2.10-62)}$$

Flange moment:

$$M_0 = \gamma_{F_0} \cdot C_{F_0} \qquad \text{(A 2.10-63)}$$

A 2.10.7.2.3.2 Operating condition

$$\gamma_{F_x} = \frac{(a_2 \cdot d_1 - d_2)}{(a_2 \cdot b_1 - b_2)} \qquad \text{(A 2.10-64)}$$

$$F_{Kontakt_x} = -\gamma_{F_x} \cdot b_1 + d_1 \qquad \text{(A 2.10-65)}$$

with the coefficients:

$$b_1 = 2 \cdot c_{S_x} \cdot a_{Kontakt} - 2 \cdot (a_D - a_{Kontakt}) \cdot c_{D_{KNS_x}} \qquad \text{(A 2.10-66)}$$

$$d_1 = 2 \cdot c_{S_x} \cdot a_{Kontakt} \cdot \gamma_{F0} - g_{KNS} \cdot F_{DKU} - 2 \cdot \gamma_{F_{KNS}} \cdot (a_D - a_{Kontakt}) \cdot c_{D_{KNS_x}} - F_{R1} - F_{F1} + c_{S_x} \cdot (x_{S0} - \Delta W_x) \qquad \text{(A 2.10-67)}$$

$$a_2 = a_D - a_{Kontakt} \qquad \text{(A 2.10-68)}$$

$$b_2 = C_{F1_x} + 2 \cdot c_{S_x} \cdot a_{Kontakt} \cdot a_D \qquad \text{(A 2.10-69)}$$

$$d_2 = c_{S_x} \cdot 2 \cdot \gamma_{F_0} \cdot a_{Kontakt} \cdot a_D + F_R \cdot (a_R - a_D) + F_F \cdot (a_F - a_D) + c_{S_x} \cdot (x_{S_0} - \Delta W_x) \cdot a_D \qquad \text{(A 2.10-70)}$$

$$F_{D_x} = g_{KNS} \cdot F_{DKU} - \{2 \cdot (\gamma_{F_x} - \gamma_{F_{KNS}})\} \cdot (a_D - a_{Kontakt}) \cdot c_{D_{KNS_x}} \qquad \text{(A 2.10-71)}$$

$$F_{S_x} = F_{Kontakt_x} + F_{D_x} + F_R + F_F \qquad \text{(A 2.10-72)}$$

Gap increase at mean gasket diameter d_D:

$$\Delta s_x = 2 \cdot (a_D - a_{Kontakt}) \cdot (\gamma_{F_x} - \gamma_{F_{KNS}}) \qquad \text{(A 2.10-73)}$$

Flange moment:

$$M_x = \gamma_{F_x} \cdot C_{F_x} \quad \text{(A 2.10-74)}$$

A 2.10.7.2.4 Case of non-identical flange pairs

A 2.10.7.2.4.1 Gasket seating condition

$$x_{S_0} = F_{S0} / c_{S_0} \quad \text{(A 2.10-75)}$$

$$\gamma_{F1_{KNS}} = F_{DKU} \cdot a_D / C_{F1_0} \quad \text{(A 2.10-76)}$$

$$\gamma_{F2_{KNS}} = F_{DKU} \cdot a_D / C_{F2_0} \quad \text{(A 2.10-77)}$$

$$F_{Kontakt\,0} = \frac{F_{S0} - F_{DKU} + (a_D - a_{Kontakt})a_D \cdot c_{DKNS0}\left\{\frac{F_{S0} - F_{DKU}}{C_{F1_0}} + \frac{F_{S0} - F_{DKU}}{C_{F2_0}}\right\}}{1 + (a_D - a_{Kontakt})^2 \cdot c_{DKNS0}\left(\frac{1}{C_{F1_0}} + \frac{1}{C_{F2_0}}\right)} \quad \text{(A 2.10-78)}$$

$$\gamma_{F1_0} = \frac{F_{S0} \cdot a_D}{C_{F1_0}} - \frac{F_{Kontakt\,0}}{C_{F1_0}}(a_D - a_{Kontakt}) \quad \text{(A 2.10-79)}$$

$$\gamma_{F2_0} = \frac{F_{S0} \cdot a_D}{C_{F2_0}} - \frac{F_{Kontakt\,0}}{C_{F2_0}}(a_D - a_{Kontakt}) \quad \text{(A 2.10-80)}$$

$$F_{D0} = F_{DKU} - \{(\gamma_{F1_0} - \gamma_{F1_{KNS}}) + (\gamma_{F2_0} - \gamma_{F2_{KNS}})\} \cdot (a_D - a_{Kontakt}) \cdot c_{DKNS_0} \quad \text{(A 2.10-81)}$$

Gap increase at gasket diameter d_D:

$$\Delta s_0 = (a_D - a_{Kontakt}) \cdot \{(\gamma_{F1_0} - \gamma_{F1_{KNS}}) + (\gamma_{F2_0} - \gamma_{F2_{KNS}})\} \quad \text{(A 2.10-82)}$$

Flange moments:

$$M_{1_0} = \gamma_{F1_0} \cdot C_{F1_0} \quad \text{(A 2.10-83)}$$

$$M_{2_0} = \gamma_{F2_0} \cdot C_{F2_0} \quad \text{(A 2.10-84)}$$

A 2.10.7.2.4.2 Operating condition

$$\gamma_{F2_x} = \frac{(b_2 - c_2)\cdot(a_2 \cdot d_1 - d_2) - (a_2 \cdot b_1 - b_2)\cdot(d_2 - d_3)}{(b_2 - c_2)\cdot(a_2 \cdot b_1 - c_2) - (a_2 \cdot b_1 - b_2)\cdot(c_2 - c_3)} \quad \text{(A 2.10-85)}$$

$$\gamma_{F1_x} = -\gamma_{F2_x} \cdot \frac{(a_2 \cdot b_1 - c_2)}{(a_2 \cdot b_1 - b_2)} + \frac{(a_2 \cdot d_1 - d_2)}{(a_2 \cdot b_1 - b_2)} \quad \text{(A 2.10-86)}$$

$$F_{Kontakt_x} = -\gamma_{F1_x} \cdot b_1 - \gamma_{F2_x} \cdot b_1 + d_1 \quad \text{(A 2.10-87)}$$

with the coefficients:

$$b_1 = c_{S_x} \cdot a_{Kontakt} - (a_D - a_{Kontakt}) \cdot c_{DKNS_x} \quad \text{(A 2.10-88)}$$

$$d_1 = c_{S_x} \cdot a_{Kontakt}\,(\gamma_{F1_0} + \gamma_{F2_0}) - g_{KNS} \cdot F_{DKU} - \gamma_{F1_{KNS}} \cdot (a_D - a_{Kontakt}) \cdot c_{DKNS_x} - \gamma_{F2_{KNS}} \cdot (a_D - a_{Kontakt}) \cdot c_{DKNS_x} - F_{R1} - F_{F1} + c_{S_x} \cdot (x_{S_0} - \Delta W_x) \quad \text{(A 2.10-89)}$$

$$a_2 = a_D - a_{Kontakt} \quad \text{(A 2.10-90)}$$

$$b_2 = C_{F1_x} + c_{S_x} \cdot a_{Kontakt} \cdot a_D \quad \text{(A 2.10-91)}$$

$$c_2 = c_{S_x} \cdot a_{Kontakt} \cdot a_D \quad \text{(A 2.10-92)}$$

$$d_2 = c_{S_x} \cdot (\gamma_{F1_0} + \gamma_{F2_0}) \cdot a_{Kontakt} \cdot a_D + F_{R1} \cdot (a_{R1} - a_D) + F_{F1} \cdot (a_{F1} - a_D) + c_{S_x} \cdot (x_{S_0} - \Delta W_x) \cdot a_D \quad \text{(A 2.10-93)}$$

$$c_3 = C_{F2_x} + c_{S_x} \cdot a_{Kontakt} \cdot a_D \quad \text{(A 2.10-94)}$$

$$d_3 = c_{S_x} \cdot (\gamma_{F1_0} + \gamma_{F2_0}) \cdot a_{Kontakt} \cdot a_D + F_{R2} \cdot (a_{R2} - a_D) + F_{F2} \cdot (a_{F2} - a_D) + c_{S_x} \cdot (x_{S_0} - \Delta W_x) \cdot a_D \quad \text{(A 2.10-95)}$$

$$F_{D_x} = g_{KNS} \cdot F_{DKU} - \{(\gamma_{F1_x} - \gamma_{F1_{KNS}}) + (\gamma_{F2_x} - \gamma_{F2_{KNS}})\} \cdot (a_D - a_{Kontakt}) \cdot c_{DKNS_x} \quad \text{(A 2.10-96)}$$

$$F_{S_x} = F_{Kontakt_x} + F_{D_x} + F_{R1} + F_{F1} \quad \text{(A 2.10-97)}$$

Gap increase at gasket diameter d_D:

$$\Delta s_x = (a_D - a_{Kontakt}) \cdot \{(\gamma_{F1_x} - \gamma_{F1_{KNS}}) + (\gamma_{F2_x} - \gamma_{F2_{KNS}})\} \quad \text{(A 2.10-98)}$$

Flange moments:

$$M_{1_x} = \gamma_{F1_x} \cdot C_{F1_x} \quad \text{(A 2.10-99)}$$

$$M_{2_x} = \gamma_{F2_x} \cdot C_{F2_x} \quad \text{(A 2.10-100)}$$

A 2.10.7.2.5 Flange-blank combination

The equations for non-identical flange pairs (except for the equations to determine the flange moments on second flange) to clause A 2.10.7.2.4 apply with the following substitute values for modelling the blank as second flange.

$$C_{F2_0} = c_{B_0} \cdot a_D^2 \quad \text{(A 2.10-101)}$$

$$C_{F2,x} = (E_\vartheta / E_{20}) \cdot c_{B_0} \cdot a_D^2 \quad \text{(A 2.10-102)}$$

$$F_{R2} = p \cdot ¼ \cdot \pi \cdot d_D^2 + F_{BZ} \quad \text{(A 2.10-103)}$$

$$a_{R2} = \frac{a_D \cdot c_{BxFD}}{c_{Bxp}} \quad \text{(A 2.10-104)}$$

$$F_{F2} = 0 \quad \text{(A 2.10-105)}$$

The verification of strength of the blank shall be made with the loads F_{D_0} and $F_{Kontakt_0}$ at gasket seating condition and with $F_{Kontakt_x}$, F_{D_x}, p and F_{BZ} at operating condition.

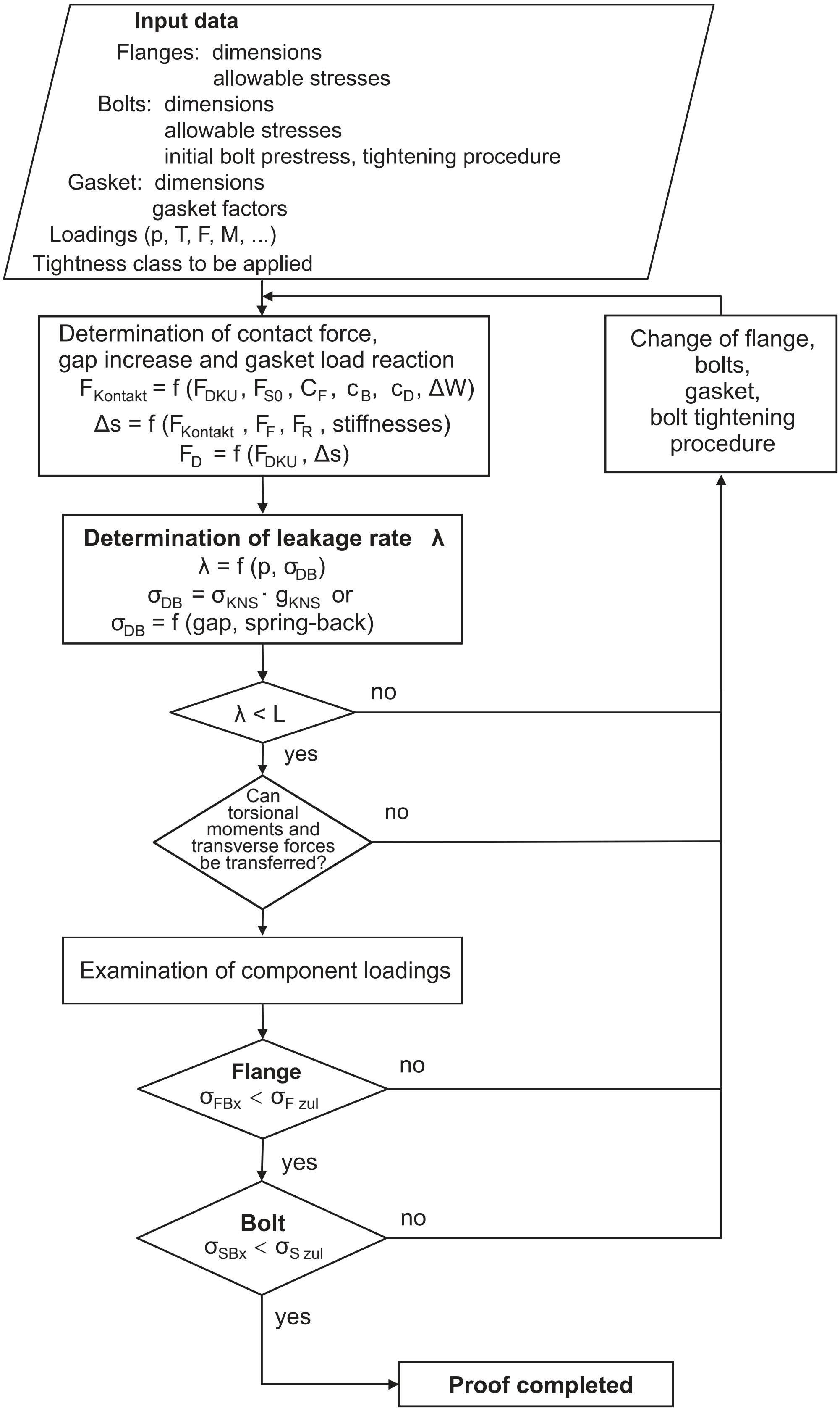


Figure A 2.10-9: Schematic procedural steps for verification of strength of metal-to-metal contact type flanged joints

Ser. no.	Type of stress [1]	Bolting-up condition	Loading levels			
			0	A, B	P	C, D
1	Stress resulting from internal pressure, required gasket load reaction and external loads [2] $F_S = F_{RP} + F_F + F_{DB} + F_{RZ} + F_{RM}$	—	S_m or S [3]	S_m or S [3]	—	$\frac{1}{1.1} \cdot R_{p0.2T}$ [8][9]
2	Stress at test condition [2] $F_{SP} = F'_{RP} + F'_{RZ} + F'_{RM} + F'_F + F'_{DB}$	—	—	—	$\frac{1}{1.1} \cdot R_{p0.2T}$ [8]	—
3	Stress at bolting-up condition [4] [5] F_{S0}	$\frac{1}{1.1} \cdot R_{p0.2RT}$ [8]	—	—	—	—
4	Stress due to internal pressure, external loads, residual gasket load and differential thermal expansion [6], if any, taking the relation between bolt load and residual gasket load at the respective pressure condition into consideration [5] [7]	—	—	$\frac{1}{1.1} \cdot R_{p0.2T}$ [8]	—	—

For diameter ratios $d_F/d_i > 2$ all stress intensity limits shall be reduced by the factor $\Phi = 0.6 + \frac{1}{\sqrt{5.25 + \left(\frac{d_F}{d_i} - 1\right)^2}}$.

1) See clause A 2.10.1 for definition of notations used.

2) If equations (A 2.10-1) to (A 2.10-3) are used.

3) S_m for test group A1 components, S for test group A2 and A3 components.

4) If equation (A 2.10-4) is used, within dimensioning F_{S0U} and within the verification of strength F_{S0max} shall be taken.

5) In consideration of the requirements of sub-clause A 2.9.3 (3).

6) Consideration of differential thermal expansion at a design temperature > 120 °C. This temperature limit does not apply to combinations of austenitic and ferrritic materials for flange and bolts.

7) In the case of calculation as per clause A 2.10.6.

8) For cast steel 0.75 · $R_{p0.2T}$ instead of $R_{p0.2T}/1.1$.

9) Where proof of tightness is required for loading levels C and D, the same procedure as for levels A and B to ser. No. 4 shall apply.

Table A 2.10-1: Allowable stresses σ_{zul} for pressure-loaded flanged joints made of steel

A 2.11 Gaskets

A 2.11.1 General

(1) For the notations and units the requirements of Sections A 2.9.1 and A 2.10.1 apply.

(2) The gasket factors shall be provided by means of **Forms A 2.11-1** and **A 2.11-2**.

Note:

Procedures for determining the gasket factors are contained in [13].

A 2.11.2 Gasket factors for design of floating type flanged joints

Note:

See DIN 28090-1 (1995-09) and DIN EN 13555 (2005-02) for definition of gasket factors.

A 2.11.2.1 Lower limit value for gasket seating at bolting-up condition $\sigma_{VU/L}$

The minimum gasket contact surface load at bolting-up condition $\sigma_{VU/L}$ is the contact surface load that shall be applied on the effective gasket surface (compressed gasket surface) $A_D = \pi \cdot d_D \cdot b_D$ by the bolt load for gasket seating condition F_{S0} to obtain the required tightness at operating condition by adaptation to the flange surface roughness and decrease of inner cavities. **Figure A 2.11-1** shows an example for the determination of the gasket factors for evaluating the sealing properties ($\sigma_{VU/L}$, $\sigma_{BU/L}$).

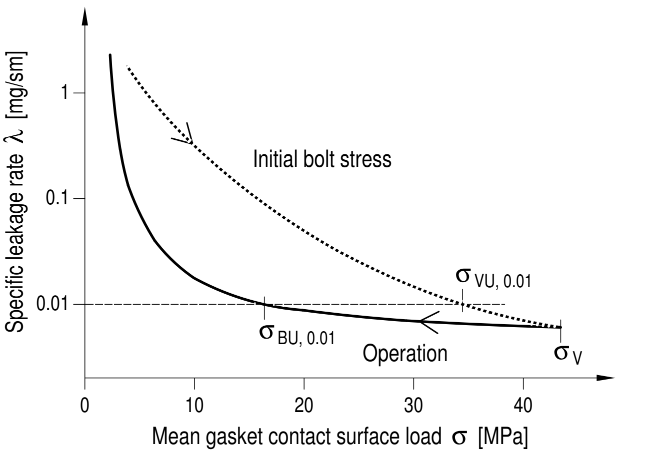
The tightness class relating to the gasket factor $\sigma_{VU/L}$ is indexed, e.g. $\sigma_{VU/0.1}$ for tightness class $L_{0.1}$ with a specific leakage rate $\lambda \leq 0.1$ mg/(s · m).

$\sigma_{VU/L}$ therefore will govern the required minimum gasket seating force for bolting-up condition $F_{DVU} = A_D \cdot \sigma_{VU/L}$ for a specific tightness class L. **Table A 2.11-1** shows possible assignments of tightness classes to the fluid used.

Note:

*(1) See **Figures A 2.11-2** and **A 2.11-3** as regards the determination of the effective gasket seating surface.*

*(2) The gasket width b_D of curved surface metal gaskets to **Figure A 2.11-3** shall be determined to the calculation approaches of DIN EN 1591-1 (2009-10) “Flanges and their joints. Design rules for gasketed circular flange connections. Part 1: Calculation method; German version of EN 1591-2:2001 + A1:2009“, to DIN 2696 (1999-08) “Flange joints with lens gasket” or to manufacturer’s date where the gasket factors pertinent to the respective calculation procedure shall be taken.*



σ_{BU} : minimum gasket contact surface load for operation
σ_{VU} : minimum gasket contact surface load at bolting-up condition
σ_V : effective gasket contact surface load at bolting-up condition
L : tightness class, max. allowable value for λ (here: λ = 0.01)

Figure A 2.11-1: Determination of the gasket factors for evaluating the sealing properties (schematically shown)

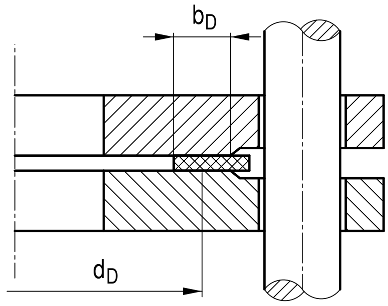


Figure A 2.11-2: Gasket width b_D

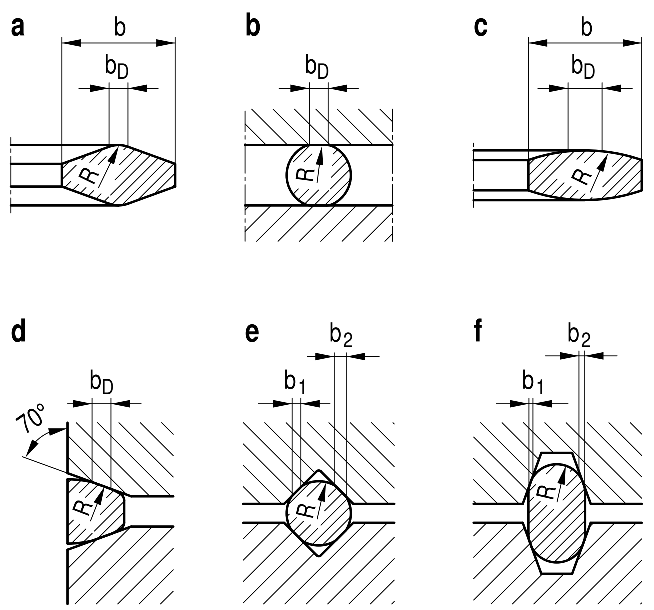


Figure A 2.11-3: Gasket profiles for metallic gaskets with curved surfaces

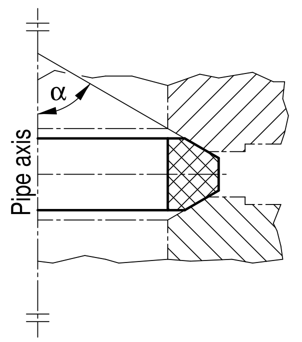


Figure A 2.11-4: Angle α shown with the example of a lens gasket

A 2.11.2.2 Maximum gasket contact surface load at bolting-up condition σ_{VO}

The maximum gasket contact surface load at bolting-up condition σ_{VO} is the maximum contact surface load that may be applied on the effective gasket surface $A_D = \pi \cdot d_D \cdot b_D$ by the bolt load for gasket seating condition in order to avoid inadmissible loosening of the gasketed joint by destruction (compressive load testing) or yielding or creep (compression stress testing) of the gasket. It shall govern the maximum allowable gasket load reaction for bolting-up condition $F_{DVO} = A_D \cdot \sigma_{VO}$ at ambient temperature.

A 2.11.2.3 Minimum gasket contact surface load at operating condition $\sigma_{BU/L}$

The minimum gasket contact surface load at operating condition $\sigma_{BU/L}$ is the contact surface load that shall be applied on the effective gasket surface $A_D = \pi \cdot d_D \cdot b_D$ in order to obtain the requested tightness class for a given fluid, internal pressure and a given temperature.

The characteristic value $\sigma_{BU/L}$ shall be determined in dependence of the gasket contact surface load at bolting-up condition.

The tightness class on which the characteristic value $\sigma_{BU/L}$ is based, is indicated by the index, e.g. $\sigma_{BU/0.1}$ for tightness class $L_{0.1}$ with a specific leakage rate $\lambda \le 0.1$ mg/(s · m).

$\sigma_{BU/L}$ thus determines the required minimum gasket load at operating condition $F_{DBU} = A_D \cdot \sigma_{BU/L}$ for a specified tightness class.

A 2.11.2.4 Maximum gasket contact surface load at operating condition σ_{BO}

The maximum gasket contact surface load at operating condition σ_{BO} is the maximum contact surface load that may be applied on the effective gasket surface $A_D = \pi \cdot d_D \cdot b_D$ at any possible operating condition in order to avoid inadmissible loosening of the gasketed joint by structural damage or creep of the gasket. σ_{BO} governs the maximum allowable gasket load reaction $F_{DBO} = A_D \cdot \sigma_{BO}$ at operating temperature.

A 2.11.2.5 Load compression characteristic Δh_D and gasket factor P_{QR}

(1) The load compression characteristic Δh_D refers to the change in a gasket height under operating condition upon completion of assembly.

Note:

Where the stiffness of the rigged system is known, the loss of seating force can be determined by means of Δh_D.

(2) The gasket factor P_{QR} is a factor used for crediting the influence of relaxation on gasket compression upon bolt tightening and of the long-term effect of the operating temperature.

(3) For the purpose of verifying the of calculation of strength and deformation conditions as per Section A 2.10.6 the gasket characteristic P_{QR} shall be converted to obtain a load compression characteristic value Δh_D in accordance with Section 8.6 of DIN EN 13555.

A 2.11.2.6 Substitute elastic modulus E_D

The substitute elastic modulus E_D describes the elastic recovery behaviour of the gasket. For gaskets with non-linear recovery E_D is defined as the secant modulus of the recovery curve. The values used in the calculation for the substitute elastic modulus E_D shall refer to the initial gasket height (as required by DIN 28090-1).

A 2.11.3 Design values for metal-to-metal contact type joints

A 2.11.3.1 Minimum gasket contact surface load at metal-to-metal contact

The minimum gasket contact surface load σ_{KNS} is the gasket surface load to be exerted by the bolt at bolting-up condition to obtain metal-to-metal contact.

A 2.11.3.2 Sealable pressure at metal-to-metal contact

The sealable pressure $p_{KNS/L}$ is the internal pressure that can be sealed at metal-to-metal contact of flange blades without exceeding a leakage rate to be specified.

A 2.11.3.3 Relaxation factor at metal-to-metal contact

The relaxation factor at metal-to-metal contact g_{KNS} indicates the percentage value by which the gasket contact surface load at metal-to-metal contact decreases at the given operating temperature and over a period of time representing the operating time.

Note:
*See also **Figure A 2.11-5**.*

A 2.11.3.4 Substitute elastic modulus $E_{D,KNS}$

The substitute elastic modulus $E_{D,KNS}$ describes the gasket elastic recovery behaviour for various spring-back conditions from full metal-to-metal contact. $E_{D,KNS}$ is defined as the secant modulus of the recovery curve. The values used in the calculation for the substitute elastic modulus $E_{D,KNS}$ shall refer to the initial gasket height.

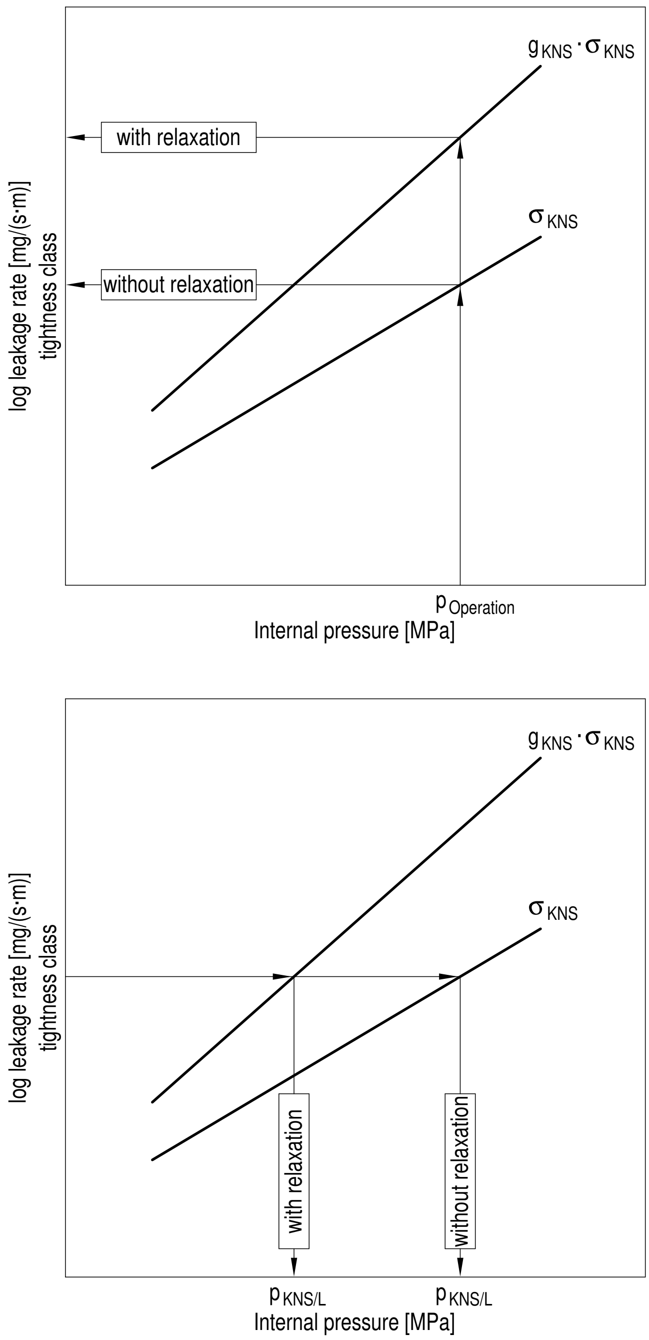


Figure A 2.11-5: Determination of leakage rate (top) and of the sealable pressure (bottom) for metal-to-metal contact type flanged joints (schematically shown)

Tightness class L	Leakage rate at leak test with test fluids He or N_2 mg/(m·s)	Fluid
$L_{1.0}$	1	Water without activity
$L_{0.1}$	10^{-1}	a) Water with activity b) Water vapour without activity c) Pressurised air
$L_{0.01}$	10^{-2}	Water vapour with activity

Table A 2.11-1: Examples for assignment of tightness classes and fluids

Gasket factors [1] for floating type flanged joints

Manufacturer:	Designation:

Sealing properties ($\sigma_{VU/L}$, $\sigma_{BU/L}$)

Dimensions of gasketed flange test connections:

Test fluid [2]	. . . .		. . . .		. . . .		. . . .	
Tightness class	. . . .		. . . .		. . . .		. . . .	
Internal pressure, MPa [3]	. . . .		. . . .		. . . .		. . . .	
Gasket factors, MPa [4]	$\sigma_{VU/L}$; σ_V	$\sigma_{BU/L}$	$\sigma_{VU/L}$; σ_V	$\sigma_{BU/L}$	$\sigma_{VU/L}$; σ_V	$\sigma_{BU/L}$	$\sigma_{VU/L}$; σ_V	$\sigma_{BU/L}$

Deformation properties (σ_{VO}, σ_{BO}, E_D, Δh_D)

Dimensions of gasketed flange test connections:

		RT	100 °C	200 °C	300 °C	400 °C
σ_{VO} or σ_{BO} in MPa [5]						
E_D (σ_V = ... MPa)						
E_D (σ_V = ... MPa)						
E_D (σ_V = ... MPa)						
E_D (σ_V = ... MPa)						

Dimensions of gasketed flange test connections:

		RT		100 °C		200 °C		300 °C		400 °C	
	σ in MPa [6]	C_1	C_2	C_1	C_2	C_1	C_2	C_1	C_2	C_1	C_2
Δh_D in mm											

C: Stiffness of compression stress test equipment C_1 = kN/mm C_2 = kN/mm

1) For gasket platens data on the influence of gasket dimensions (h_D, b_D) are additionally required.

2) The test fluid selected shall be nitrogen or helium. The tightness class and internal pressure stage shall be selected to meet the user's requirements.

3) The internal pressure stages to be used shall preferably be 1, 2, 4, 8 and 16 MPa. In the case of intermediate values the gasket factors of the next higher pressure stage shall always be taken.

4) $\sigma_{BU/L}$ shall be taken in dependence of $\sigma_V \geq \sigma_{VU/L}$. Alternatively graphic representations may be given.

5) In the case of gaskets where creep relaxation has considerable influence on the gasket, these factors can only be considered in connection with Δh_D.

6) Initial gasket contact surface load.

Form A 2.11-1: Summary of gasket factors

Gasket factors for metal-to-metal contact type flanged joints

Manufacturer:	Designation:

Sealing properties ($p_{KNS/L}$)

Dimensions of gasketed flange test connections:	Groove dimensions:
Test fluid:	

Internal pressure 1), MPa	Leakage rate λ, mg/(m·s)			
	σ_{KNS} = ... MPa and Δh = ... mm	0.8 • σ_{KNS} = ... MPa and Δh = ... mm	... • σ_{KNS} = ... MPa and Δh = ... mm	. . .
1				
2				
4				
8				
16				

Deformation properties (σ_{KNS}, g_{KNS}, $E_{D,KNS}$)

Dimensions of gasketed flange test connections:	Groove dimensions:

	RT
σ_{KNS}, MPa	

Dimensions of gasketed flange test connections:	Groove dimensions:

	RT	100 °C	200 °C	300 °C	400 °C
g_{KNS}					

Dimensions of gasketed flange test connections:	Groove dimensions:

Spring-back Δh, mm	E_{DKNS} (RT), MPa	E_{DKNS} (100 °C), MPa	E_{DKNS} (200 °C), MPa	E_{DKNS} (300 °C), MPa	E_{DKNS} (400 °C), MPa

Δh : Spring-back from full metal-to-metal contact

1) The gasket factors of the next higher pressure stage shall be taken.

Form A 2.11-2: Summary of gasket factors

A 3 Pumps

A 3.1 Design values and units relating to Section A 3

Notation	Design value	Unit
b	effective length	mm
c_1, c_2	allowances (see Section 6.4)	mm
d	diameter of casing bore	mm
d_A	diameter of opening in flat circular head	mm
d_i	inside diameter of opening	mm
d_{i1}, d_{i2}	inside diameter of adjacent opening	mm
d_t	bolt circle diameter	mm
e	edge distance to opening	mm
l	nominal width of ligament	mm
p	design pressure	MPa
s	wall thickness of unpierced cylinder	mm
s_A	required wall thickness at edge of opening	mm
s_e	actual wall thickness at edge of opening	mm
s_{pl}	wall thickness of flat circular head	mm
t	thickness of discharge cover	mm
A_p	pressure loaded area	mm^2
A_σ, $A_{\sigma 0}$, $A_{\sigma 1}$	pressure-retaining cross-sectional area	mm^2
B_1, B_2, B_3	auxiliary values	—
C_A	factor for openings	—
C_1	auxiliary value	1/mm
C_2, C_3, C_4	auxiliary values	—
D_a	outside diameter of cylinder	mm
D_i	inside diameter of cylinder	mm
F'_R	pipe force referred to bolt circle diameter d_t	N/mm^2
S	design stress intensity acc. to Section 6.6	N/mm^2
S_m	design stress intensity acc. to Section 6.6	N/mm^2
σ_1	local membrane stress	N/mm^2
σ_b	bending stress	N/mm^2
β	design value acc. to equation (A 3.2-9)	—

A 3.2 General

Figures A 3.2-1, **A 3.2-2** and **A 3.2-3** show the casing types and components covered by the component-specific analysis for pumps for some typical cases. **Figure A 3.2-1** shows an example of a single-stage annular casing pump with a forged casing, **Figure A 3.2-2** with a cast casing. **Figure A 3.2-3** shows the example of a multi-stage barrel-type pump with a forged cylinder. Characteristic for forged casings is the over-dimensioned unpierced cylinder to withstand internal pressure. In these cases, the flattened areas for the pressure (outlet) or suction (inlet) flange govern the dimensioning, for which design principle are given in Section A 3.3.

For the design type of a cast annular casing pump shown in **Figure A 3.2-2** specific dimensioning procedures cannot be given. The calculation may e.g. be based on the theory of bending of shells of revolution.

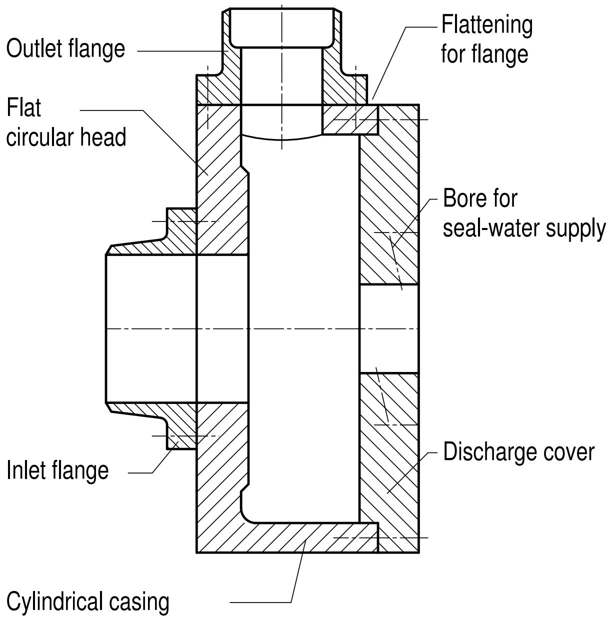


Figure A 3.2-1: Example of a single-stage annular casing pump with forged casing

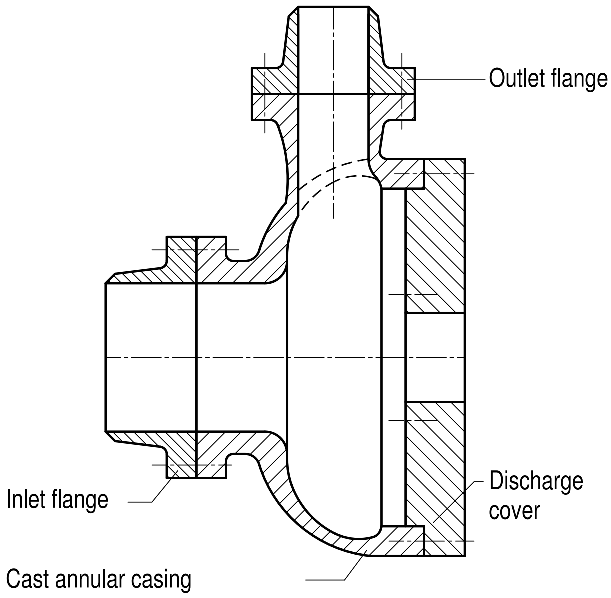


Figure A 3.2-2: Example of a single-stage annular casing pump with cast

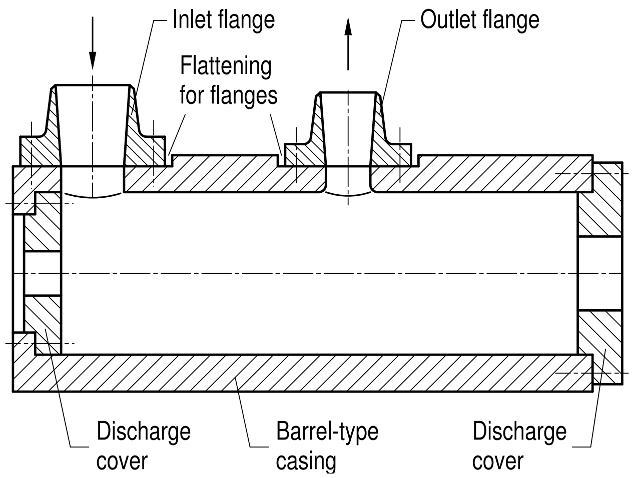


Figure A 3.2-3: Example of a multi-stage barrel-type pump with forged casing

A 3.3 Dimensioning of openings in cylindrical pump bodies with flattening

A 3.3.1 Scope

(1) Only such types of openings on pump bodies are treated as do not fall under the scope of the rules in clause A 2.8.2.

(2) The calculation rules hereinafter apply to cylindrical pump bodies provided with a flattening at the opening (see **Figure A 3.3-1** to **A 3.3-5**) within the following limits:

$$0.002 \le \frac{s_e - c_1 - c_2}{D_a} \le 0.1$$

and

$D_a/D_i \le 1.7$

The diameter ratio d_i/D_i is limited to ≤ 0.6.

When applying the equations of clause A 3.3.5 the following limits shall be adhered to for the circular plate:

$s_{pl}/D_i \le 1/3$ and $d_A/D_i \le 0.7$.

(3) Additional external forces and moments are not covered by this rule and shall be considered separately, if required. One possibility of calculation is, e.g. the consideration of a sufficiently increased internal pressure.

A 3.3.2 Types of reinforcement

(1) The procedure assumes that within the flattened area the actual wall thickness s_e is great enough to compensate for the weakening due to the flattening and opening.

(2) Therefore, for forged casings a greater thickness shall be selected for the cylindrical body than is required for the weakened cylindrical shell, or an eccentric casing shape shall be selected.

(3) For cast casings cast-integral nozzles or local reinforcements for compensation of area may be used.

A 3.3.3 Design strength values

Depending on test group A1, A2 or A3 the design stress intensity S_m or S from **Table 6.6-1** shall be taken.

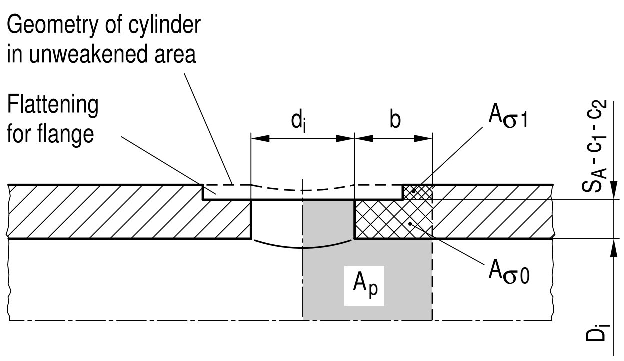


Figure A 3.3-1: Design sketch for barrel-type casings and adjacent openings in longitudinal direction of cylinder

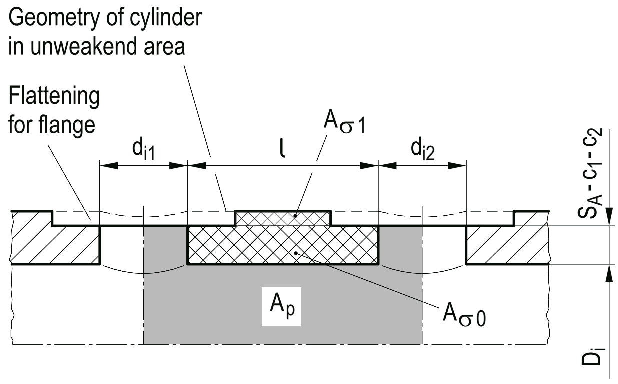


Figure A 3.3-2: Design sketch for barrel-type casing with flattening and adjacent openings in longitudinal direction of cylinder

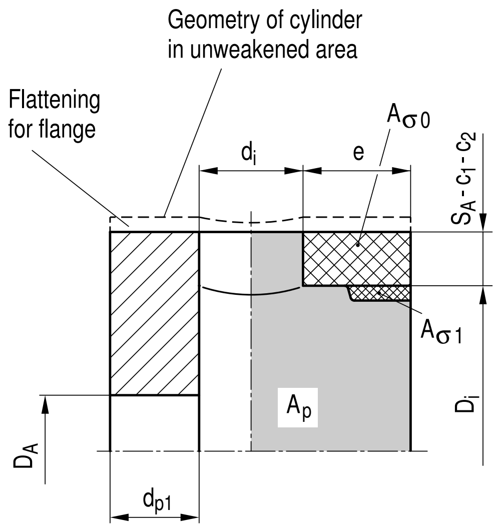


Figure A 3.3-3: Design sketch for annular casings with flattening and opening

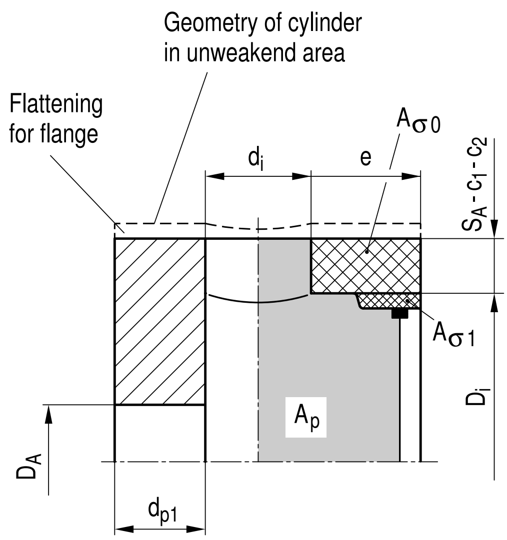


Figure A 3.3-4: Design sketch for an annular casing with flattening and opening; the gasket lies on the inside of the cylinder

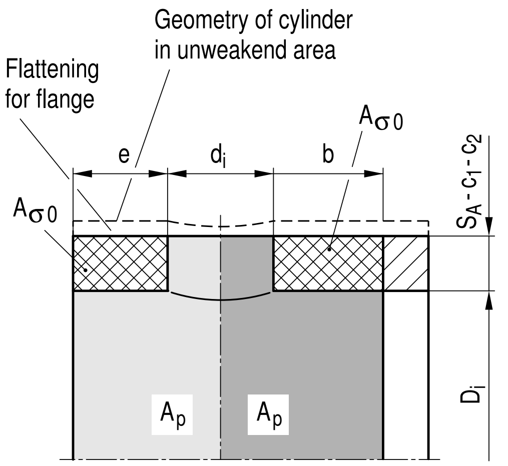


Figure A 3.3-5: Design sketch for pipe casing with flattening and opening

A 3.3.4 Dimensioning

A 3.3.4.1 Required wall thickness

The calculation of the required wall thickness s_A shall be made with the relationship

$$p \cdot \left(\frac{A_p}{A_\sigma} + \frac{1}{2} \right) \le S_m \text{ or } S, \qquad \text{(A 3.3-1)}$$

based on a consideration of equilibrium between the pressure-loaded area and the effective cross-sectional area. The pressure-loaded area A_p to be inserted in equation (A 3.3-1) as well as the pressure-retaining cross-sectional area $A_\sigma = A_{\sigma 0} + A_{\sigma 1}$ are obtained from **Figures A 3.3-1** to **A 3.3-5**.

A 3.3.4.2 Effective length

Not more than b according to equation (A 3.3-2) shall be used as effective length:

$$b = \sqrt{(D_i + s_A - c_1 - c_2) \cdot (s_A - c_1 - c_2)} \qquad \text{(A 3.3-2)}$$

Where an opening is located so close to the cylinder end that the edge distance e to the opening is smaller than b, only the actual length shall be taken.

A 3.3.4.3 Interaction of openings

(1) Interaction of openings may be neglected if the distance is

$$l \ge 2 \cdot \sqrt{(D_i + s_A - c_1 - c_2) \cdot (s_A - c_1 - c_2)}\,. \qquad \text{(A 3.3-3)}$$

(2) Where the distance l does not satisfy equation (A 3.3-3), it shall be checked whether the remaining cross-section between the edges of the openings is able to withstand the loading exerted on it. This is the case if equation (A 3.3-1) is satisfied.

A 3.3.4.4 Influence of boreholes

Where in the section under consideration or within an angle of 22.5° boreholes are provided, they shall be deducted from the pressure-retaining area A_σ in accordance with **Figure A 3.3-6**.

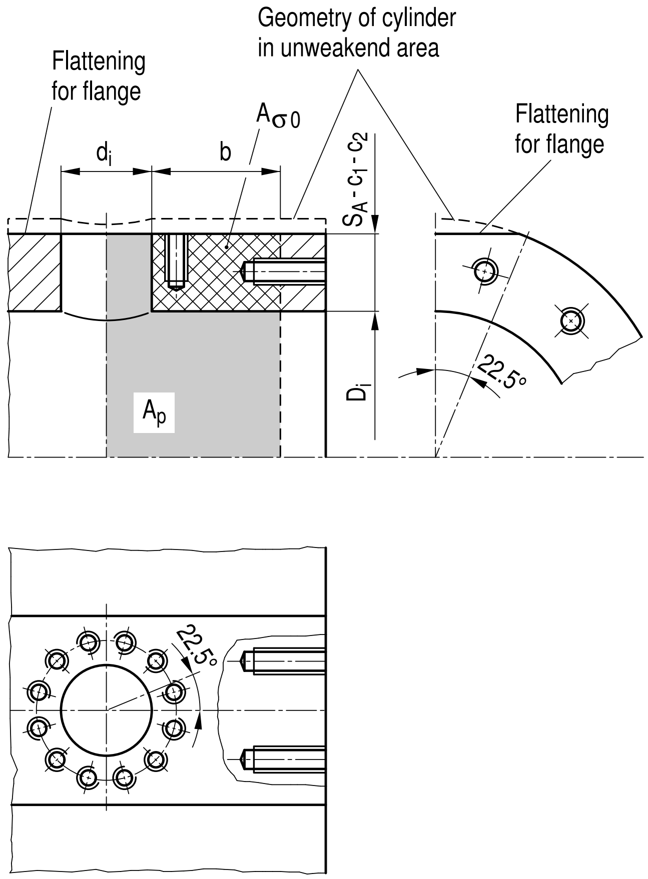


Figure A 3.3-6: Design sketch for consideration of boreholes

A 3.3.5 Influence of flat heads on cylindrical casings

(1) For annular casings, as shown in **Figures A 3.3-3** and **A 3.3-4**, a stress analysis is not required for the transition of cylinder to flat head within the area of flattening and opening for nozzle if

a) the wall thickness of the flat circular head is

$$s_{pl} = 0.4 \cdot C_A \cdot D_i \cdot \sqrt{\frac{p}{S_m}} \qquad \text{(A 3.3-4)}$$

The opening factor C_A shall be taken from **Figure A 3.3-7**.

b) the local membrane and bending stresses in the unweakened cylinder at the transition to the flat circular head satisfy the condition

$$(\sigma_1 + \sigma_b) \le 1.5 \cdot S_m. \qquad \text{(A 3.3-5)}$$

c) the ratio of the wall thickness s in the unweakened area of the cylinder to the wall thickness s_e in the flattened area is

$s/s_e \le 2$.

Where s/s_e exceeds 2 it shall be proved that the local membrane and bending stresses in the unweakened cylindfer at the transition to the flat circular head satisfy the following condition:

$$\frac{s}{s_e} \cdot (\sigma_1 + \sigma_b) \le 3 \cdot S_m \qquad \text{(A 3.3-6)}$$

(2) The determination of the local membrane and bending stresses in the unweakened cylinder may be done analytically by means of the freebody or finite element method (see Section 8.3).

To determine σ_l and σ_b by analysis the following relationships may be used (the geometric dimensions are shown in **Figure A 3.3-8**):

$$\sigma_l = \frac{p \cdot \left(D_i^2 - d_a^2\right)}{(D_i + s) \cdot 4 \cdot s} + \frac{F_R' \cdot d}{(D_i + s) \cdot s} \qquad \text{(A 3.3-7)}$$

$$\sigma_b = \frac{16.85 \cdot p \cdot B_1^3 + 6.3 \cdot B_3^3 \cdot (10.4 \cdot F_R' \cdot C_1 + p \cdot C_2) \cdot \left(2.57 \cdot \sqrt{B_1^3} + B_3 \cdot C_3 \cdot C_4\right)}{10.92 \cdot B_1^2 + \frac{B_3 \cdot C_4}{B_2^2} \cdot \left(30.85 \cdot \frac{\sqrt{B_1^3}}{1-\beta^2} + 8.5 \cdot \sqrt{B_1} \cdot C_3 \cdot B_2^2 + 12 \cdot \frac{B_3 \cdot C_3 \cdot C_4}{1-\beta^2}\right)} \qquad \text{(A 3.3-8)}$$

with

$$\beta = \frac{d_A}{D_i + s} \qquad \text{(A 3.3-9)}$$

$$B_1 = \frac{D_i + s}{2 \cdot s} \qquad \text{(A 3.3-10)}$$

$$B_2 = \frac{s_{pl}}{s} \qquad \text{(A 3.3-11)}$$

$$B_3 = \frac{D_i + s}{2 \cdot s_{pl}} \qquad \text{(A 3.3-12)}$$

$$C_1 = \frac{d_t}{2 \cdot (D_i + s)^2 \cdot \left[(D_i + s)^2 - d_A^2\right]} \cdot \left(\left[(D_i + s)^2 + 1.86 \cdot d_A^2\right] \cdot 2 \cdot \ln\frac{D_i + s}{d_t} + \left(1 - \frac{d_t^2}{(D_i + s)^2}\right) \cdot \left[(D_i + s)^2 \cdot 0.54 + d_A^2\right] - \frac{d_t}{2 \cdot (D_i + s)^2} \cdot \left(\frac{d_t^2}{(D_i + s)^2} + 2 \cdot \ln\frac{D_i + s}{d_t} - 1\right) \right) \qquad \text{(A 3.3-13)}$$

$$C_2 = 1 - 2 \cdot \beta^2 + \beta^2 \cdot \left(4.71 + 7.43 \cdot \frac{\beta^2 \cdot \ln\beta}{1-\beta^2}\right) \qquad \text{(A 3.3-14)}$$

$$C_3 = \frac{(D_i + s)^2}{(D_i + s)^2 - d_A^2} \qquad \text{(A 3.3-15)}$$

$$C_4 = 0.7 + 1.3 \cdot \beta^2 \qquad \text{(A 3.3-16)}$$

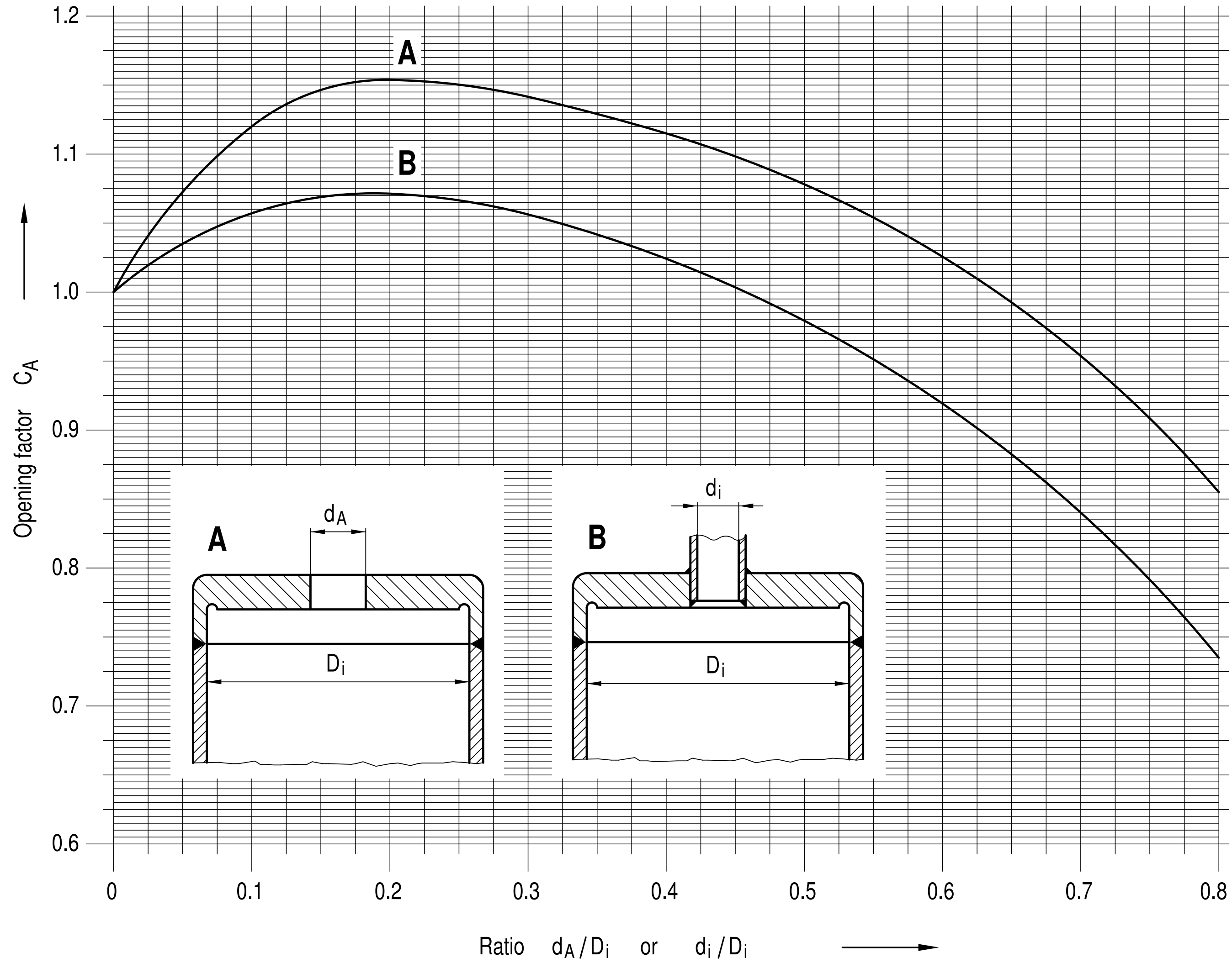


Figure A 3.3-7: Opening factor C_A

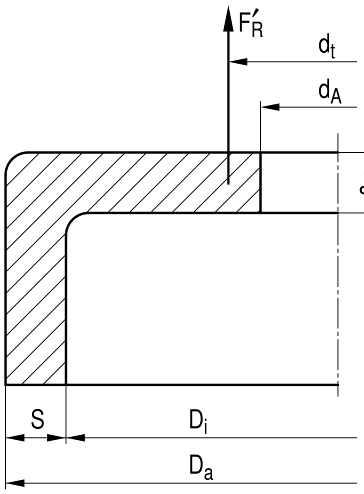


Figure A 3.3-8: Transitional area from cylinder to flat head

A 3.4 Weakening by boreholes

A 3.4.1 Boreholes in casing covers

Service boreholes for mechanical seals, water-lubricated bearings etc. shall be taken into account. Normally, several boreholes are distributed over the cover.

If the borehole diameter is less than 20 % of the minimum wall thickness and is not more than one borehole located in meridional direction, the weakening need not be considered in the strength calculation.

If the borehole diameter exceeds 20 % of the minimum wall thickness or if several boreholes are located in one meridional plane, this location may be dimensioned by using $1.5 \cdot S_m$.

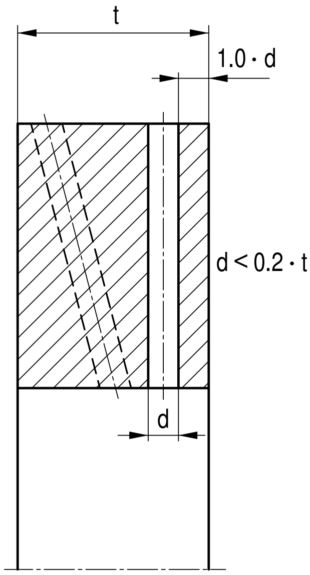


Figure A 3.4-1: Boreholes in casing cover

A 3.4.2 Boreholes in the cylindrical shell

Where the casing shell is designed with 50 N/mm^2 criterion, weakening by an axial borehole (e.g. balance water bore) is permitted.

This location shall be considered in the strength calculation with the following equation:

$$P_m \geq \frac{p \cdot D_i}{2 \cdot s - d} < S_m \qquad \text{(A 3.4-1)}$$

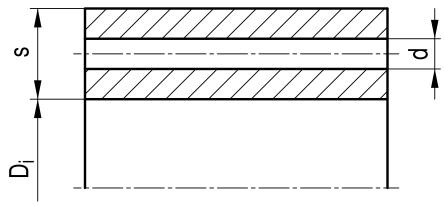


Figure A 3.4-2: Borehole in cylindrical shell

A 3.5 Required engagement length

The requirements of clause A 2.9.4.3 apply.

A 3.6 Washers and extension sleeves for high-strength bolts

A 3.6.1 Washers

If the bolted joint corresponds to the DIN dimensions, also DIN standard washers shall be used. The strength of the washer shall be compatible with the strength of the nut and bolt materials. If this is not the case, a separate verification is required.

Where corrosion protection is required, the washers shall be galvanised and be chromated in accordance with DIN EN ISO 2081 Code D.

A 3.6.2 Extension sleeves

Extension sleeves shall be made in accordance with DIN 2510-7. The use of the bolt material is recommended for extension sleeves (DIN 267-13). If necessary, the same corrosion protection as for washers shall be provided.

A 4 Valves

A 4.1 Valve bodies

A 4.1.1 Design values and units relating to Section A 4.1

Notation	Design value	Unit
a, a_1, a_2	distance	mm
b, b_2	clear width of non-circular cross sections	mm
c_1, c_2	wall thickness allowances	mm
d_{Ai}	inside diameter of branch	mm
d_{Hi}	inside diameter of main body	mm
l	length of transition from circular to elliptical cross-section	mm
e, l'	die-out length	mm
e_A	effective length of branch	mm
e_H	effective length in main body	mm
s_0	calculated wall thickness without allowances	mm
s_{A0}	calculated wall thickness of branch without allowances	mm
s_{An}	nominal wall thickness of branch	mm
s_{H0}	calculated wall thickness of main body excluding allowances	mm
s_{Hn}	nominal wall thickness of main body	mm
s'_H	wall thickness at transition of flange to spherical shell	mm

Notation	Design value	Unit
s_n	nominal wall thickness	mm
s_{Rn}	nominal wall thickness of pipe	mm
y	cylindrical portion in oval bodies	mm
A_p	pressure-loaded area	mm^2
A_σ	effective cross-sectional area	mm^2
B_n	factor for oval cross-sections	—
C_K	factor	—
C	effectiveness of edge reinforcement	—
α	angle between axis of main body and branch axis	degree

Subscripts			
b	bending	u	circumference
l	longitudinal	m	mean/average
r	radial	B	operating condition
t	torsion	0	as-installed condition

A 4.1.2 Scope

The calculation hereinafter applies to valve bodies subject to internal pressure. Loadings resulting from external forces and moments shall be taken into account in accordance with Section 8.4.

A 4.1.3 Calculation of valve bodies at predominantly static loading due to internal pressure

A 4.1.3.1 General

(1) The valve bodies may be considered to be a main body with a determined geometry with openings or branches and branch penetrations. The calculation of the wall thickness therefore comprises the main body lying outside the area influenced by the opening and the opening itself. The main body is considered to be that part of the valve body having the greater diameter so that the following applies:

$d_{Hi} \geq d_{Ai}$ or $b_2 \geq d_{Ai}$.

(2) The transitions between differing wall thicknesses shall not show any sharp fillets or breaks to minimize discontinuity stresses and show a good deformation behaviour. Depending on the chosen stress and fatigue analysis additional design conditions shall be satisfied, e.g. with regard to the transition radii (see Section 8.4).

The main body wall thickness s_{Hn} and the branch thickness s_{An} shall be tapered to the connected pipe wall thickness s_{Rn} on a length of at least $2 \cdot s_{Hn}$ or $2 \cdot s_{An}$, respectively. In addition, the condition of clause 5.1.4.2 regarding the transitional area shall be taken into account.

(3) For the total wall thickness including allowances the following applies:

$$s_{Hn} \geq s_{H0} + c_1 + c_2 \qquad \text{(A 4.1-1)}$$

and

$$s_{An} \geq s_{A0} + c_1 + c_2 \qquad \text{(A 4.1-2)}$$

where s_{Hn} and s_{H0} apply to the main body and s_{An} and s_{A0} to the branches.

(4) For the recalculation of as-built components the following applies:

$$s_{H0} \leq s_{Hn} - c_1 - c_2 \qquad \text{(A 4.1-3)}$$

and

$$s_{A0} \leq s_{An} - c_1 - c_2. \qquad \text{(A 4.1-4)}$$

A 4.1.3.2 Calculation of the main body outside the opening or branch area and without any influences at the boundary

A 4.1.3.2.1 General

The geometric configuration of the main body of valve bodies may be cylindrical, spherical, conical or oval. Accordingly, the wall thicknesses can be determined within body areas remote from discontinuities.

A 4.1.3.2.2 Determination of the required wall thickness s_0 of cylindrical main bodies

The required wall thickness s_0 of cylindrical main bodies shall be determined in accordance with clause A 2.2.2.

A 4.1.3.2.3 Determination of the required wall thickness s_0 of spherical main bodies

The required wall thickness s_0 of spherical main bodies shall be determined in accordance with clause A 2.3.2.

A 4.1.3.2.4 Determination of the required wall thickness s_0 of conical main bodies

The required wall thickness s_0 of conical main bodies shall be determined in accordance with clause A 2.4.2.

A 4.1.3.2.5 Determination of the required wall thickness s_0 of oval main bodies

(1) In the case of oval-shaped cross-sections (**Figure A 4.1-1**) the additional bending loads in the walls shall be considered.

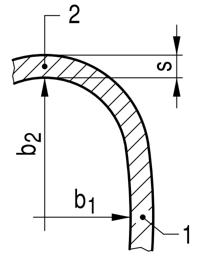


Figure A 4.1-1: Oval-shaped valve body

(2) The theoretical minimum wall thickness for such bodies subject to internal pressure is obtained as follows:

$$s'_0 = \frac{p \cdot b_2}{2 \cdot s_m} \sqrt{B_o^{\,2} + \frac{4 \cdot S_m}{p} \cdot B_n} \qquad \text{(A 4.1-5)}$$

(3) The wall thickness shall be calculated at the locations 1 and 2 shown in **Figure A 4.1-1** for oval cross-sections, since here the bending moments obtain maximum values and thus have essential influence on the strength behaviour.

(4) The factor B_0 depending on the normal forces shall be

for location 1: $B_0 = b_1/b_2$

for location 2: $B_0 = 1$.

(5) B_n shall be taken from **Figure A 4.1-2**.

(6) The factors B_n depending on the bending moments are shown in **Figure A 4.1-2** for oval cross-sections at locations 1 and 2 in dependence of b_1/b_2. The curves satisfy the following equations:

$$B_1 = \frac{1 - k_E^{\,2}}{6} \cdot \frac{K'}{E'} - \frac{1 - 2 \cdot k_E^{\,2}}{6} \qquad \text{(A 4.1-6)}$$

$$B_2 = \frac{1+k_E{}^2}{6} - \frac{1-k_E{}^2}{6} \cdot \frac{K'}{E'} \qquad \text{(A 4.1-7)}$$

$$\text{with } k_E{}^2 = 1 - \left(\frac{b_1}{b_2}\right)^2 \qquad \text{(A 4.1-8)}$$

Note:
K´ and E´ are the full elliptical integrals whose values can be taken in dependence of the module of the integral k_E from Table books such as „Hütte I, Theoretische Grundlagen, 28 th edition, Publishers: W. Ernst u. Sohn, Berlin“.

(7) For the factors relating to $b_1/b_2 \geq 0.5$ the following approximate equations may be used:

$$B_1 = \left(1 - \frac{b_1}{b_2}\right) \cdot \left[0.625 - 0.435 \cdot \sqrt{1 - \frac{b_1}{b_2}}\right] \qquad \text{(A 4.1-9)}$$

$$B_2 = \left(1 - \frac{b_1}{b_2}\right) \cdot \left[0.5 - 0.125 \cdot \left(1 - \frac{b_1}{b_2}\right)\right] \qquad \text{(A 4.1-10)}$$

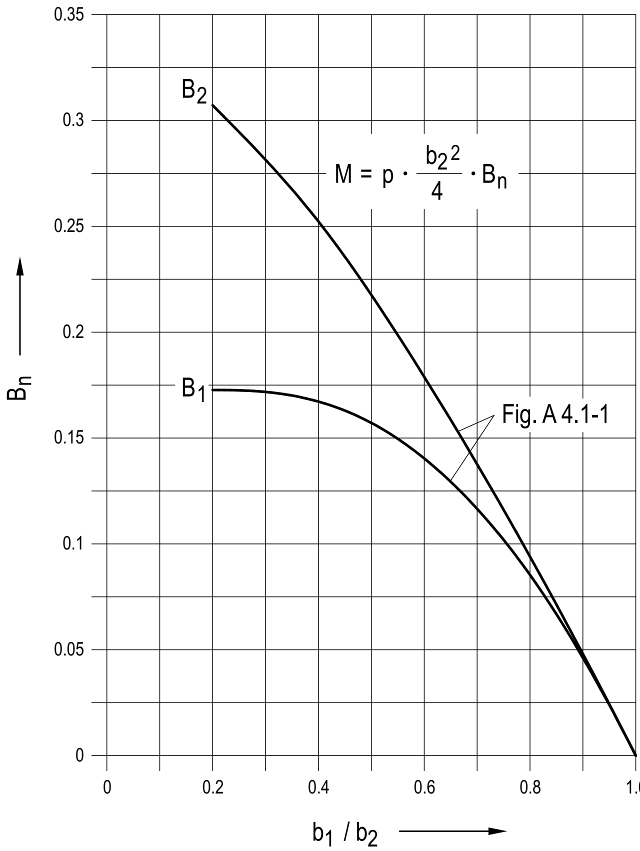


Figure A 4.1-2: Factor B_n for oval cross-sections

(8) The factors also apply to changes in cross-section in oval main bodies, e.g. for gate valves according to **Figure A 4.1-3**, design a and b where the side length b_1 from the crown of the inlet nozzles (flattened oval shape) increases over the length l to obtain b_2 (circular shape). The value b_1 in section B-B at 1/2 shall govern the determination of B_n where l is obtained from

$$l = H - y - \left(\frac{d_{Hi}}{2} + s_H\right) - l' \qquad \text{(A 4.1-11)}$$

with H being a design dimension as per **Figure A 4.1-3**.

For the length l' influenced by the inlet nozzle the following applies:

$$l' = 1.25 \cdot \sqrt{d'_m \cdot s_n} \qquad \text{(A 4.1-12)}$$

$$\text{where } d'_m = \frac{b'_1 + b_2}{2} \qquad \text{(A 4.1-13)}$$

in which case b'_1 and b_2 shall be determined at section A-A on a length l' from the inlet nozzle. s_n is the wall thickness available for l'. In general, b'_1 and l' shall be determined by iteration.

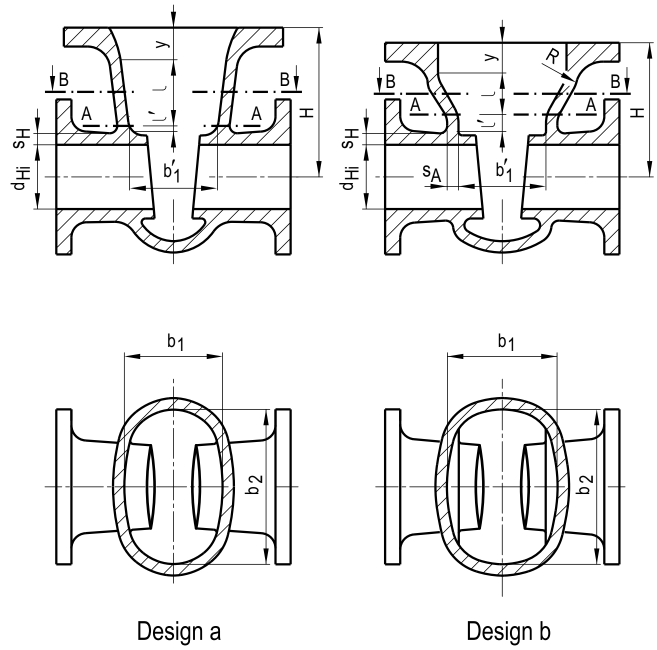


Figure A 4.1-3: Examples for changes in cross-section of oval bodies

(9) For short bodies (e.g. **Figure A 4.1-3**, design a or design b) with the length l remote from discontinuity, corresponding to the design geometry, the supporting effect of the components connected at the end of the body (e.g. flanges, heads, covers) may be credited. Thus, the required minimum wall thickness is obtained by using equation (A 4.1-5) to become:

$$s_0 = s'_0 \cdot k \qquad \text{(A 4.1-14)}$$

(10) The correction factor k shall be obtained, in correspondence to the damping behaviour of the loadings in cylindrical shells, in consideration of experimental test results from non-circular bodies as follows:

$$k = 0.48 \cdot \sqrt[3]{\frac{l^2}{d_m \cdot s'_0}} \qquad \text{(A 4.1-15)}$$

with $0.6 \leq k \leq 1$

The function is shown in **Figure A 4.1-4** in dependence of $\frac{l^2}{d_m \cdot s'_0}$.

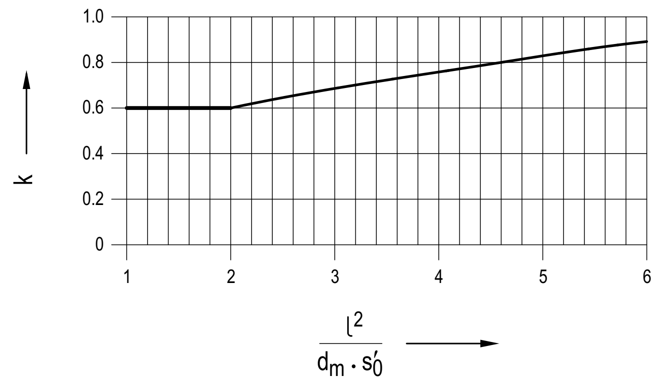


Figure A 4.1-4: Correction factor k for short bodies

(11) $d_m = (b_1 + b_2)/2$ shall be taken for d_m, and s'_0 corresponds to equation (A 4.1-5). For changes in cross-section over a length l, e.g. according to **Figure A 4.1-3**, design a or b, the dimensions b_1 and b_2 shall be taken from Section B-B (at l/2). Local deviations from the body shape irrespective whether they are convex or concave, shall, as a rule, be neglected.

(12) The strength criterion is satisfied if the required wall thickness is locally available provided that the wall thickness transitions are smooth.

A 4.1.3.3 Valve bodies with branch

(1) The strength of the body containing a branch shall be calculated considering the equilibrium of external and internal forces for the highly loaded areas which are the transitions of the cylindrical, spherical or non-circular main body to the branch. The diameter d_H and the wall thickness s_H refer to the main body, and the diameter d_A and the wall thickness s_A to the branch. The following shall apply: $d_{Hi} > d_{Ai}$.

(2) In the case of cylindrical main bodies, see **Figure A 4.1-5**, the section I located in the longitudinal section through the main axis as a rule is subject to the greatest loading with the average main stress component $\overline{\sigma}_I$. In the case of nozzle to main body ratios $\geq$ 0.7, however, the bending stresses occurring in the cross-sectional area to the main axis (Section II) cannot be neglected anymore, i.e. this direction has also be taken into account.

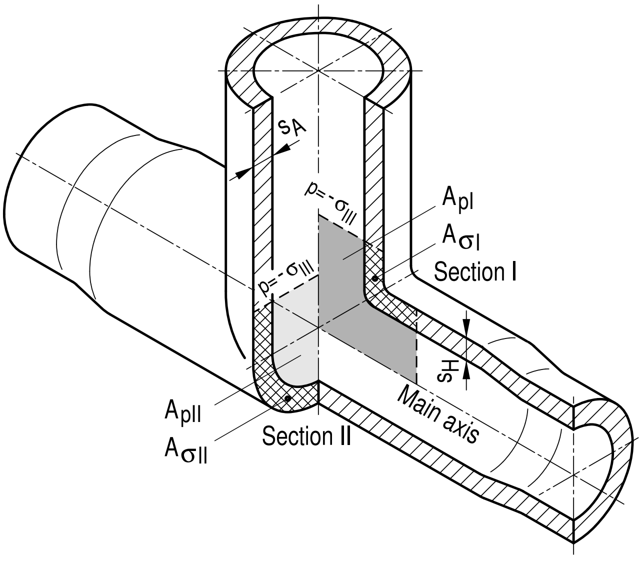


Figure A 4.1-5: Calculated sections for valve bodies with branch

(3) A recalculation of section II can be omitted if the wall thickness differences within the die-out length of this section and compared to section I do not exceed 10 %.

(4) In the case of non-circular bodies with branches and generally in the event of additional forces acting in the direction of the main axis the greatest loading may be obtained in the section with the average main stress component σ_{II} (section II).

(5) In these cases, the calculation shall be effected for both section I and II.

(6) The calculation procedure hereinafter applies to valve bodies with vertical branch, see **Figures A 4.1-6** to **A 4.1-12** as well as with oblique branch if the angle α is not less than 45°, see **Figure A 4.1-14**, provided that s_A does not exceed s_H. Where these conditions cannot be satisfied by certain designs, only the smallest wall thickness s_H can be used in the calculation of the effective length and effective cross-sectional area A_σ.

Note:

*In **Figures A 3.1-5** to **A 3.1-14** the wall thickness shown is the nominal wall thickness minus the allowances c_1 and c_2.*

(7) For the equilibrium of forces in the longitudinal section according to Figures A 3.1-6 to A 3.1-12 the following relationship applies

$$p \cdot A_{pI} = \overline{\sigma}_I \cdot A_{\sigma I} \qquad \text{(A 4.1-16)}$$

where $p \cdot A_{pI}$ is the total external force acting upon the pressure-loaded area A_{pI} (dotted) whereas the internal force $\overline{\sigma}_I \cdot A_{\sigma I}$ is the force acting in the most highly loaded zone of the wall with the cross-sectional area $A_{\sigma I}$ (cross-hatched) and in the cross-section the average main stress $\overline{\sigma}_I$.

(8) The strength condition to be satisfied in accordance with Tresca´s shear stress theory is:

$$\sigma_{VI} = \overline{\sigma}_I - \overline{\sigma}_{III} = p \cdot \frac{A_{pI}}{A_{\sigma I}} + \frac{p}{2} \leq S_m \qquad \text{(A 4.1-17)}$$

(9) In the case of non-circular bodies with branches the following strength condition shall be satisfied to consider those bending stresses exceeding the bending stresses already covered by the calculation of the wall thicknesses according to equations (A 4.1-5) or (A 4.1-14):

$$\sigma_{VI} = \overline{\sigma}_I - \overline{\sigma}_{III} = p \cdot \frac{A_{pI}}{A_{\sigma I}} + \frac{p}{2} \leq \frac{S_m}{1.2} \qquad \text{(A 4.1-18)}$$

(10) In equations (A 4.1-17) and (A 4.1-18) the stress σ_{III} acting normal to wall is considered to be the smallest main stress component which on the pressure-loaded side is σ_{III} = - p and on the unpressurized side is σ_{III} = 0, that is a mean value $\overline{\sigma}_{III}$ = - p/2.

Accordingly, the following applies to the equilibrium of forces in section II (see **Figure A 4.1-6**)

$$p \cdot A_{pII} = \overline{\sigma}_{II} \cdot A_{\sigma II} \qquad \text{(A 4.1-19)}$$

The strength condition in this case is

$$\sigma_{VII} = \overline{\sigma}_{II} - \overline{\sigma}_{III} = p \cdot \frac{A_{pII}}{A_{\sigma II}} + \frac{p}{2} \leq S_m \qquad \text{(A 4.1-20)}$$

and for non-circular bodies

$$\sigma_{VII} \leq \frac{S_m}{1.2} \qquad \text{(A 4.1-21)}$$

(11) For cylindrical valve bodies with $d_{Ai}/d_{Hi} \geq 0.7$ and simultaneously $s_{A0}/s_{H0} < d_A/d_H$ the following condition shall be satisfied in section II:

$$p \cdot \left[\frac{d_{Hi} + s_{H0}}{2 \cdot s_{H0}} + 0.2 \cdot \frac{d_{Ai} + s_{A0}}{s_{A0}} \cdot \sqrt{\frac{d_{Hi} + s_{H0}}{s_{H0}}} \right] \leq 1.5 \cdot S_m \qquad \text{(A 4.1-22)}$$

(12) For non-circular valve bodies the condition shall be:

$$p \cdot \left[\frac{b_2 + s_{H0}}{2 \cdot s_{H0}} + 0.25 \cdot \frac{d_{Ai} + s_{A0}}{s_{A0}} \cdot \sqrt{\frac{b_2 + s_{H0}}{s_{H0}}} \right] \leq 1.5 \cdot S_m \qquad \text{(A 4.1-23)}$$

(13) For the cases shown in **Figures A 4.1-7** to **A 4.1-14** the general strength condition applies:

$$\sigma = p \cdot \left(\frac{A_p}{A_\sigma} + 0.5 \right) \leq S_m \qquad \text{(A 4.1-24)}$$

The pressure-loaded areas A_p and the effective cross-sectional areas A_σ are determined by calculation or a drawing to scale (true to size).

The effective length of the considered cross-sectional areas A_p and A_σ shall be determined as follows (except for spherical bodies to **Figure A 4.1-11** and branches with oblique nozzles to **Figure A 4.1-14**):

$$e_H = \sqrt{(d_{Hi} + s_{H0}) \cdot s_{H0}} \quad \text{(A 4.1-25)}$$

$$e_A = 1.25 \cdot \sqrt{(d_{Ai} + s_{A0}) \cdot s_{A0}} \quad \text{(A 4.1-26)}$$

(14) For the design shown in **Figure A 4.1-6**, section I the following applies:

$$e_H = \sqrt{(b_1 + s_{H0}) \cdot s_{H0}} \quad \text{(A 4.1-27)}$$

$$e_{A1} = 1.25 \cdot \sqrt{(d_{Ai} + s_{A0}) \cdot s_{A0}} \quad \text{(A 4.1-28)}$$

e_{A2} in accordance with subclause (21).

For section II applies:

$$e'_H = \sqrt{(b_2 + s_{H0}) \cdot s_{H0}} \quad \text{(A 4.1-29)}$$

$$e_{A3} = 1.25 \cdot \sqrt{(b_2 + s_{A0}) \cdot s_{A0}} \quad \text{(A 4.1-30)}$$

(15) At a ratio of nozzle opening to main body opening exceeding 0.8 the factor ahead of the root is omitted in equations (A 4.1-26), (A 4.1-28) and (A 4.1-30).

(16) For branches in spherical main bodies with a ratio d_{Ai1}/d_{Hi} or $d_{Ai2}/d_{Hi} \leq 0.5$ the effective length in the spherical portion according to **Figure A 4.1-11**, design a, can be taken to be:

$$e_H = \sqrt{(d_{Hi} + s_{H0}) \cdot s_{H0}} \quad \text{(A 4.1-31)}$$

however, shall not exceed the value obtained by the bisecting line between the centrelines of both nozzles.

For the effective length the following applies:

$$e_A = \sqrt{(d_{Ai} + s_{A0}) \cdot s_{A0}} \quad \text{(A 4.1-32)}$$

At ratios of d_{Ai1}/d_{Hi} or d_{Ai2}/d_{Hi} exceeding 0.5 the effective length shall be determined in accordance with **Figure A 3.1-11**, design b, where e_{A1} or e_{A2} shall be determined in accordance with equation (A 4.1-32).

(17) Valve bodies with oblique nozzles ($\alpha \geq 45°$) may also be calculated by means of equation (A 4.1-17) in which case the pressure-loaded area (dotted) and the pressure-loaded cross-sectional area (cross-hatched) are distributed in accordance with **Figure A 4.1-14**.

Here, the effective length shall be determined as follows:

$$e_H = \sqrt{(d_{Hi} + s_{H0}) \cdot s_{H0}} \quad \text{(A 4.1-33)}$$

$$e_A = \left(1 + 0.25 \cdot \frac{\alpha}{90°}\right) \cdot \sqrt{(d_{Ai} + s_{A0}) \cdot s_{A0}} \quad \text{(A 4.1-34)}$$

In the case of oblique branches the area shall be limited to the pressure-loaded area bounded by the flow passage centre lines. At a ratio of branch opening to main body opening exceeding 0.8 the factor ahead of the root shall be omitted in equation (A 4.1-34).

(18) Where flanges or parts thereof are located within the calculated effective length they shall be considered not to be contributing to the reinforcement, as shown in **Figures A 4.1-6, A 4.1-7, A 4.1-9, A 4.1-12**.

(19) Where effective lengths of reinforcements of openings extend into the tapered portion of the flange hub, only the cylindrical portion shall be considered for the determination of the area of the opening contributing to the reinforcement.

(20) Where within the boundary of the effective cross-sectional area A_σ or within the area of influence of 22.5° to the sectional area boreholes (bolt holes) are provided, these cross-sectional areas shall be deducted from A_σ.

(21) Metal extending to the inside shall be credited to the effective cross-sectional area A_σ up to a maximum length of $e_H/2$ or $e_A/2$.

(22) In the case of a design to **Figure A 4.1-13** where a gasket is arranged such that the pressure-retaining area A_p is smaller than the area obtained from the die-out lengths e_H or e_A, the centre of the gasket may be used to set the boundaries for the area A_p whereas the metal area A_σ is limited by the calculated length e_H or e_A.

In the case of designs with pressure-retaining cover plates where the split segmental ring is located within the die-out length, e_H or e_A may be used for the determination of the effective cross-sectional area A_σ but only up to the centre of the segmental ring in order to limit the radial forces induced by the gasket and the bending stresses at the bottom of the groove.

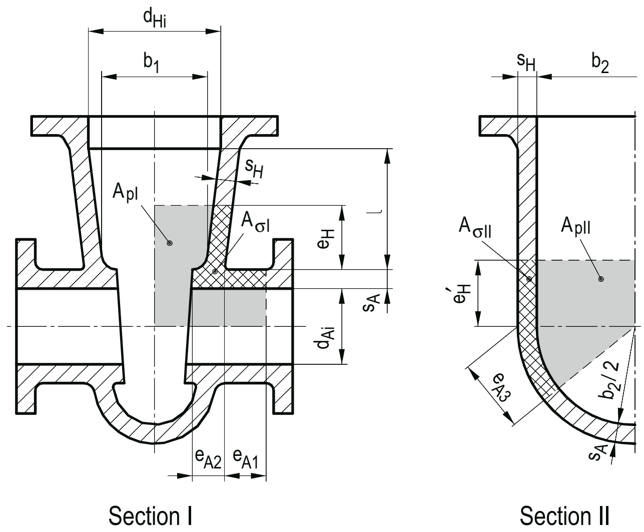


Figure A 4.1-6: Valve bodies

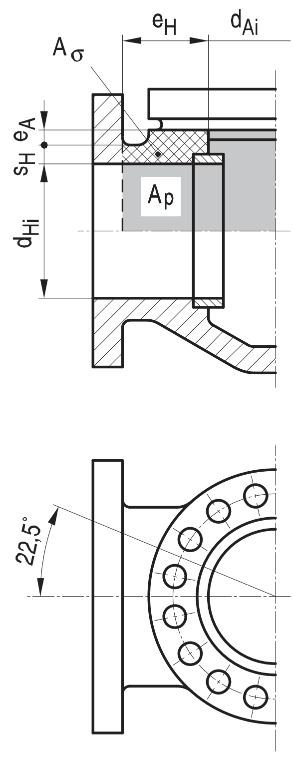


Figure A 4.1-7: Valve body

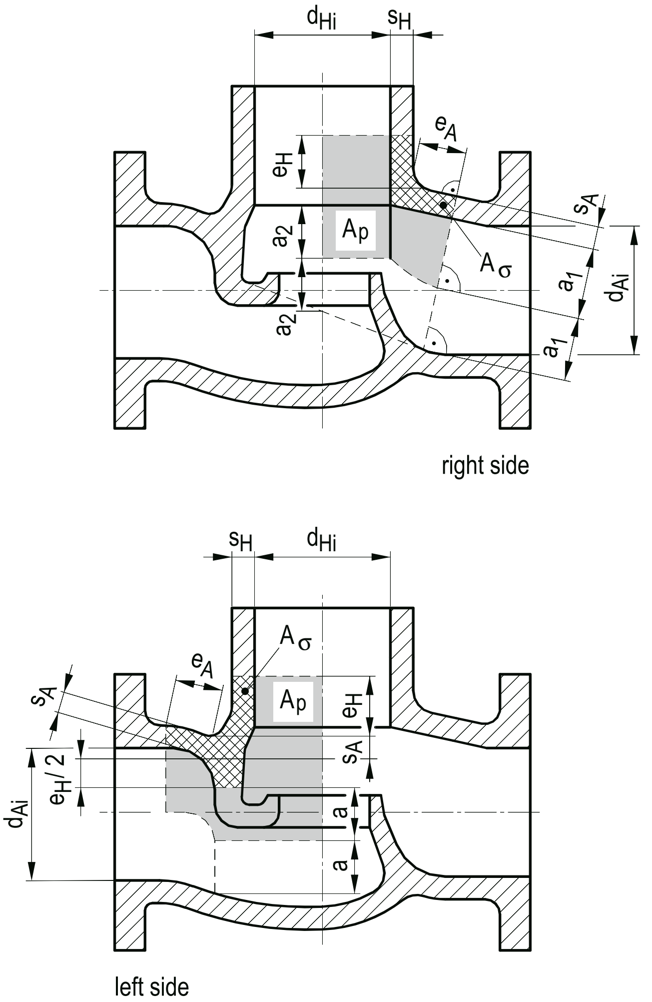


Figure A 4.1-8: Cylindrical valve body

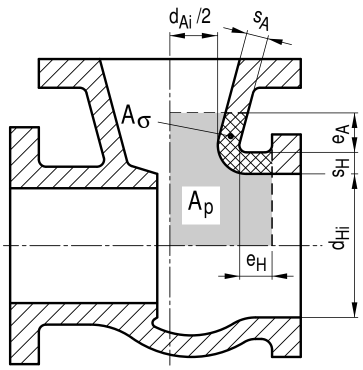


Figure A 4.1-9: Angular-type body

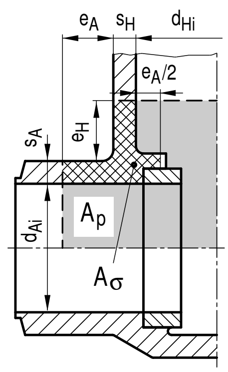
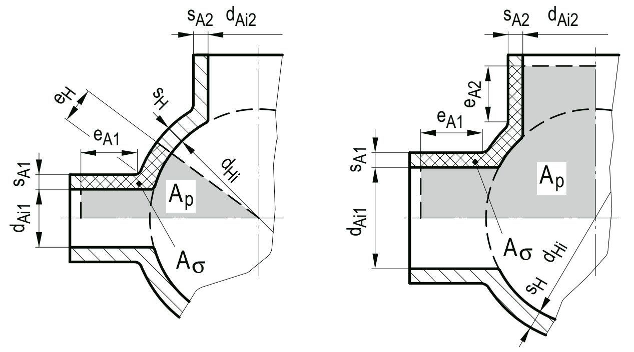


Figure A 4.1-10: Valve body



Design a
Branch in spherical body
with d_{Ai1}/d_{Hi} or $d_{Ai2}/d_{Hi} \leq 0.5$

Design b
Branch in spherical body
with d_{Ai1}/d_{Hi} or $d_{Ai2}/d_{Hi} > 0.5$

Figure A 4.1-11: Spherical bodies

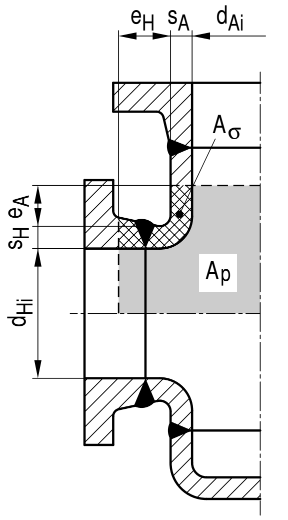


Figure A 4.1-12: Valve body

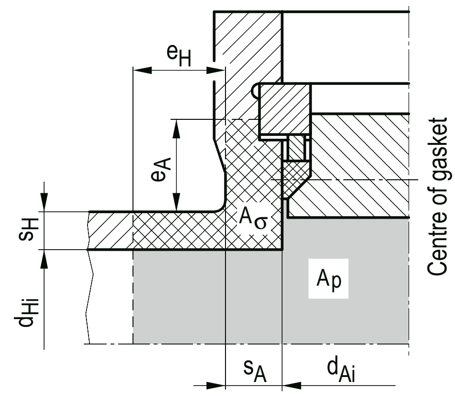


Figure A 4.1-13: Example for cover

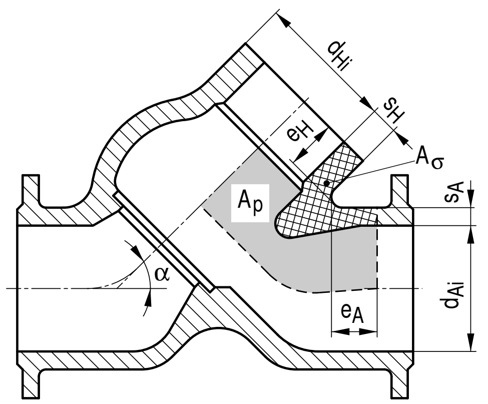


Figure A 4.1-14: Cylindrical body with oblique branch

A 4.2 Valve body closures

A 4.2.1 Spherically dished heads with bolting flanges

The calculation of spherically dished heads with bolted flange shall be made in accordance with Section A 2.6.

A 4.2.2 Dished heads

The calculation of dished heads shall be made in accordance with Section A 2.5.

A 4.2.3 Flat plates

A 4.2.3.1 General

Closures designed as flat plates are often used as external or internal covers of valve bodies. Here, primarily flat circular plates or annular ring plates are concerned as shown in clauses A 2.7.3.2 to A 2.7.3.7. Other plate types (e.g. rectangular or elliptical to clause A 2.7.3.5) are special cases to be referred to in the pertinent literature. In the case of valves, a superposition of load cases may occur resulting from internal pressure loading and additional forces. The load cases then can be considered to originate from individual loadings, as was done before, and be covered by a summation of moments. In this case, however, it shall be taken into account that the maximum moments of the individual loadings will not result in the maximum total moment in any case. In this case, the location and size of the maximum shall be determined considering the course of the load cases.

The strength condition is either contained in the wall thickness formulae or is written explicitly as follows:

$$\sigma_r, \sigma_t = \frac{6 \cdot M_{max}}{s^2} \leq 1.5 \cdot S_m \qquad \text{(A 4.2-1)}$$

The dimensioning of flat plates shall be made in accordance with Section A 2.7.

A 4.3 Bolts for valves

Bolts for valves shall be calculated according to Section A 2.9.

A 4.4 Self-sealing cover plates

(1) Design values and units relating to Section A 4.4

Notation	Design value	Unit
a	width of bearing	mm
b	width of spacer	mm
b_D	width of raised facing	mm
d_a	outside diameter of body	mm
d_0	inside diameter of body	mm
d_1	inside diameter of ring groove	mm
d_2	diameter of cover plate	mm
h_0	minimum height of bearing surface	mm
h_D	minimum height of facing	mm
h_v	thickness of cover plate	mm
h_1	thickness of lap ring R	mm
s_1	body wall thickness at location of ring groove	mm
F_{ax}	axial force	N
F_B	axial force distributed uniformly over the circumference	N
F_Z	additional axial force	N
M_b	bending moment	Nmm
S_m	design stress intensity	N/mm²

(2) The strength calculation is intended to examine the weakest section (section I-I or II-II in **Figure A 4.4-1**). At the same time, the most important dimensions of the cover plate shall be calculated by elementary procedure, e.g. the ring R inserted in the groove. In the event of dimensions deviating from the geometric conditions shown in **Figure A 4.4-1** the formulae given hereinafter may be applied accordingly.

(3) The axial force distributed uniformly over the circumference is calculated as follows:

$$F_B = p \cdot \frac{\pi}{4} \cdot d_0^2 + F_Z \qquad \text{(A 4.4-1)}$$

F_Z is an additional axial force acting in the same direction (equation A 4.4-3 to A 4.4-8: force applied over cover; equation A 4.4-9 and A 4.4-10: additional loadings applied over the body, e.g. axial force, bending moment). In the case of a bending moment and an axial force, F_Z is determined as follows:

$$F_Z = F_{ax} + \frac{4 \cdot M_B}{d_1 + s_1} \qquad \text{(A 4.4-2)}$$

(4) The minimum width of the pressure-retaining areas on the bearing surface and on the spacer are obtained considering frictional conditions and sealing requirements:

$$a, b \geq \frac{F_B}{1.5 \cdot \pi \cdot d_0 \cdot S_m} \qquad \text{(A 4.4-3)}$$

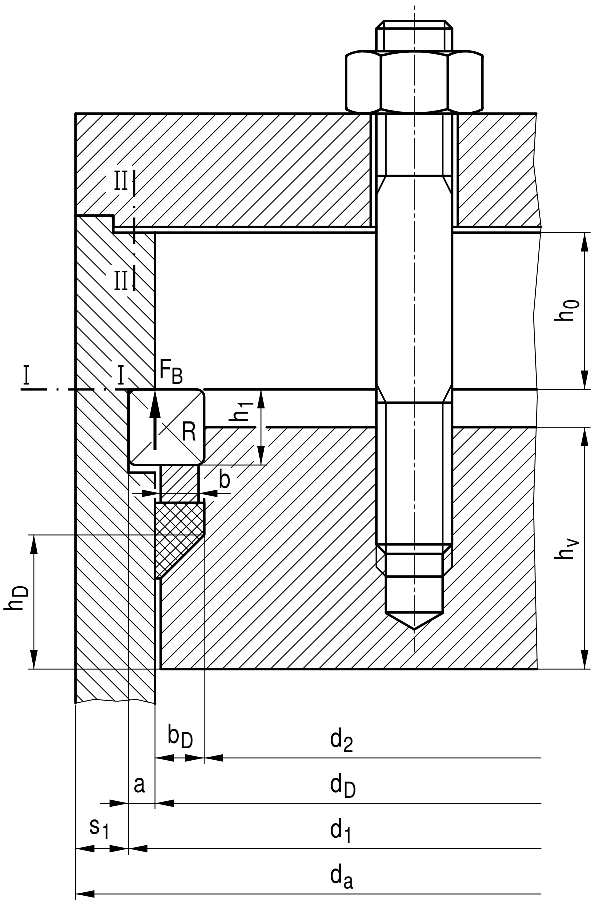


Figure A 4.4-1: Self-sealing cover plates

(5) The minimum thickness of the lap ring R is obtained from the calculation against shear and bending, and the maximum value obtained shall be inserted.

Regarding shear the following applies:

$$h_1 \geq \frac{2 \cdot F_B}{\pi \cdot d_0 \cdot S_m} \qquad \text{(A 4.4-4)}$$

Regarding bending the following applies:

$$h_1 \geq 1.38 \cdot \sqrt{\frac{F_B \cdot (a + b_D)/2}{d_0 \cdot S_m}} \qquad \text{(A 4.4-5)}$$

(6) The minimum height of the bearing surface (section II-II) is obtained from the design against shear:

$$h_0 \geq \frac{2 \cdot F_B}{\pi \cdot d_1 \cdot S_m} \qquad \text{(A 4.4-6)}$$

and against bending

$$h_0 \geq 1.13 \cdot \sqrt{\frac{F_B \cdot a}{d_1 \cdot S_m}} \quad \text{with } a = \frac{d_1 - d_0}{2} \qquad \text{(A 4.4-7)}$$

(7) For the minimum thickness of the raised face the following applies:

$$h_D \geq 1.13 \cdot \sqrt{\frac{F_B \cdot \frac{b_D}{2}}{d_2 \cdot S_m}} \qquad \text{(A 4.4-8)}$$

(8) The minimum thickness h_V of the cover plate can be determined by assuming an idealized, simply supported circular plate or annular ring plate (case 1, case 7 or case 8 from Table 5 of DIN EN 12516-2).

(9) Strength condition for section I-I

$$F_B \cdot \left(a + \frac{s_1}{2}\right) \leq \frac{\pi}{4} \cdot \left[h_0^2 \cdot (d_a - d_0) + (d_a - s_1) \cdot \left(s_1^2 - s_2^2\right)\right] \cdot S_m \qquad \text{(A 4.4-9)}$$

and

$$s_2 = \frac{F_B}{\pi \cdot (d_a - s_1) \cdot S_m} \leq s_1 \qquad \text{(A 4.4-10)}$$

A 4.5 Valve flanges

Valve flanges shall be calculated according to Section A 2.10.

A 5 Piping systems

A 5.1 General

(1) The design rules hereinafter apply to the dimensioning of individual piping components subject to internal pressure loading where the internal pressure is derived from the design pressure. Additional loadings, e.g. external forces and moments, shall be considered separately in which case the rules contained in Section 8.5 may apply to the piping components.

(2) The rules for dimensioning are comprised, in dependence of the test group, in Section 5.2 for test group A1 as well as in Section A 5.3 for test groups A2 and A3. The general design values and units are given in (5). Further design values and units are contained in the individual Sections or clauses.

(3) Where within dimensioning a recalculation is made of components with actual nominal wall thickness s_n, the wall thickness $s_{0n} = s_n - c_1 - c_2$ shall be used in the calculation in this Annex A5.

(4) The figures in this Annex do not show allowances.

(5) Design values and units

Notation	Design value	Unit
b	width	mm
c	wall thickness allowance	mm
d	diameter	mm
h	height	mm
l	length	mm
p	design pressure	MPa
p’	test pressure	MPa
r, R	radii	mm
s	wall thickness	mm
s_0	calculated wall thickness according to Figure 7.1-1	mm
s_{0n}	nominal wall thickness minus allowances c_1 and c_2 according to Figure 7.1-1	mm
s_n	nominal wall thickness according to Figure 7.1-1	mm
v	efficiency	—
A	area	mm^2
E	modulus of elasticity	N/mm^2
F	force	N
I	second moment of area	mm^4
M	moment	N/mm
S	safety factor	—
φ	angle	Grad
q	flattening	mm
W	section modulus	mm^3
U	ovality	%
T	temperature	°C
ν	Poisson's ratio = 0.3 for steel	—
σ	stress	N/mm^2
σ_l	longitudinal stress	N/mm^2
σ_r	radial stress	N/mm^2
σ_u	circumferential stress	N/mm^2
σ_V	stress intensity	N/mm^2
S_m	design stress intensity	N/mm^2
τ	shear stress	N/mm^2

Signs	Meaning
Indicator at head ˆ	maximum value e.g. $\hat{p}$
Indicator at head ˘	minimum value e.g. $\breve{p}$
Indicator at head ¯	mean value e.g. $\bar{\sigma}$
Indicator at head ~	fluctuating, e.g. $\tilde{\sigma}$
Indicator at head ´	belonging to pressure test, e.g. p´
Subscript	numerical index, e.g. n_i

A 5.2 Test group A1

A 5.2.1 Cylindrical shells under internal pressure

The calculation shall be made in acc. with clause A 2.2.2.

A 5.2.2 Bends and curved pipes under internal pressure

A 5.2.2.1 Scope

The calculation hereinafter applies to bends and curved pipes subject to internal pressure where the ratio $d_a/d_i \leq 1.7$. Diameter ratios $d_a/d_i \leq 2$ are permitted if the wall thickness $s_{0n} \leq 80$ mm.

A 5.2.2.2 Allowable wrinkling

Wrinkles the dimensions of which meet the requirements hereinafter, need not be recalculated:

a) Depth of wrinkling

$$h_m = \frac{d_{a2} + d_{a4}}{2} - d_{a3} \le 0.03 \cdot d_m \qquad \text{(A 5-1)}$$

b) Ratio of distance a to depth h_m of wrinkle

$$\frac{a}{h_m} \ge 12 \qquad \text{(A 5-2)}$$

A 5.2.2.3 Design values and units

Notation	Design value	Unit
d_m	mean diameter (see **Figure A 5-1**)	mm
d_i	inside diameter	mm
d_a	outside diameter	mm
r, R	radii	mm
s	wall thickness	mm
s_{0i}	calculated wall thickness at intrados	mm
s_{0a}	calculated wall thickness at extrados	mm
B_i	factor for determining the wall thickness at the intrados	—
B_a	factor for determining the wall thickness at the extrados	—
$\overline{\sigma}_i$	mean stress at intrados	N/mm²
$\overline{\sigma}_a$	mean stress at extrados	N/mm²
h_m	depth of wrinkle	mm
a	distance between any two adjacent wrinkles	mm

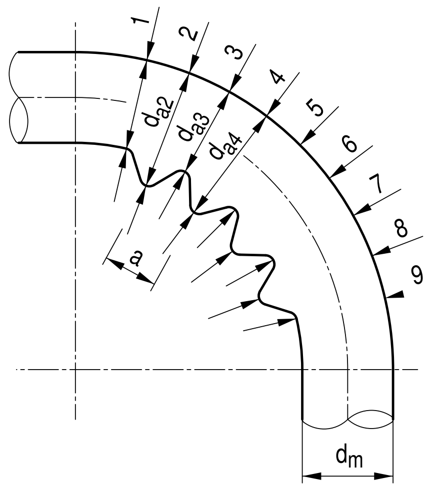


Figure A 5-1: Wrinkles on pipe bend

Note:
*The wrinkles in **Figure A 5-1** are shown excessively for clarity´s sake.*

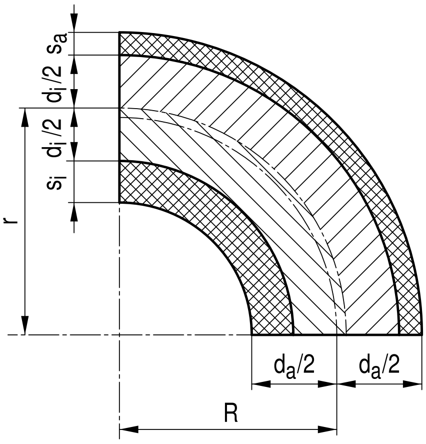


Figure A 5-2: Notations used for pipe bend

A 5.2.2.4 Calculation

(1) For the calculation of the wall thickness of bends or curved pipes under internal pressure the requirements of clause A 2.2.2 apply in which case it shall be taken into account that the loading at the intrados is greater by the factor B_i and at the extrados is smaller by B_a than at straight cylindrical shells.

(2) The calculated wall thickness at the intrados is obtained from:

$$s_{0i} = s_0 \cdot B_i \qquad \text{(A 5-3)}$$

(3) The calculated wall thickness at the extrados is obtained from:

$$s_{0a} = s_0 \cdot B_a \qquad \text{(A 5-4)}$$

(4) Determination of the factor B_i

For bends and curved pipes with given inside diameters the following applies:

$$B_i = \frac{r}{s_0} - \frac{d_i}{2 \cdot s_0} - \sqrt{\left(\frac{r}{s_0} - \frac{d_i}{2 \cdot s_0}\right)^2 - 2 \cdot \frac{r}{s_0} + \frac{d_i}{2 \cdot s_0}} \qquad \text{(A 5-5)}$$

The factor B_i may also be taken from **Figure A 5-3** in dependence of r/d_i and s_0/d_i.

For bends and curved pipes with given outside diameter the following applies:

$$B_i = \frac{d_a}{2 \cdot s_0} + \frac{r}{s_0} - \left(\frac{d_a}{2 \cdot s_0} + \frac{r}{s_0} - 1\right) \cdot \sqrt{\frac{\left(\frac{r}{s_0}\right)^2 - \left(\frac{d_a}{2 \cdot s_0}\right)^2}{\left(\frac{r}{s_0}\right)^2 - \frac{d_a}{2 \cdot s_0} \cdot \left(\frac{d_a}{2 \cdot s_0} - 1\right)}} \qquad \text{(A 5-6)}$$

The factor B_i may also be taken from **Figure A 5-4** in dependence of R/d_a and s_0/d_a.

(5) Determination of the factor B_a

For bends and curved pipes with given inside diameter the following applies:

$$B_a = \sqrt{\left(\frac{r}{s_0} - \frac{d_i}{2 \cdot s_0}\right)^2 + 2 \cdot \frac{r}{s_0} + \frac{d_i}{2 \cdot s_0}} - \frac{d_i}{2 \cdot s_0} - \frac{r}{s_0} \qquad \text{(A 5-7)}$$

The factor B_a may also be taken from **Figure A 5-5** in dependence of r/d_i and s_0/d_i.

For bends and curved pipes with given outside diameters the following applies:

$$B_a = \frac{d_a}{2 \cdot s_0} - \frac{r}{s_0} - \left(\frac{d_a}{2 \cdot s_0} - \frac{r}{s_0} - 1 \right) \cdot \sqrt{\frac{\left(\frac{r}{s_0}\right)^2 - \left(\frac{d_a}{2 \cdot s_0}\right)^2}{\left(\frac{r}{s_0}\right)^2 - \frac{d_a}{2 \cdot s_0} \cdot \left(\frac{d_a}{2 \cdot s_0} - 1\right)}} \tag{A 5-8}$$

The factor B_a can be taken from **Figure A 5-6** in dependence of R/d_a and s_0/d_a.

(6) Calculation of stresses

In the equations (A 5-9) to (A 5-12) either the nominal diameters d_{an} and d_{in} in in connection with the wall thicknesses s_{0na} and s_{0ni}, respectively or actual diameters in connection with actual wall thicknesses minus allowances c_1 and c_2 shall be used.

The strength condition for the intrados at given inside diameter shall be:

$$\overline{\sigma}_i = \frac{p \cdot d_i}{2 \cdot s_{0i}} \cdot \frac{2 \cdot r - 0.5 \cdot d_i}{2 \cdot r - d_i - s_{0i}} + \frac{p}{2} \le S_m \tag{A 5-9}$$

The strength condition for the intrados at given outside diameter shall be:

$$\overline{\sigma}_i = \frac{p \cdot (d_a - s_{0i} - s_{0a})}{2 \cdot s_{0i}} \cdot \frac{2 \cdot R - 0.5 \cdot d_a + 1.5 \cdot s_{0i} - 0.5 \cdot s_{0a}}{2 \cdot R - d_a + s_{0i}} + \frac{p}{2} \le S_m \tag{A 5-10}$$

The strength condition for the extrados at given inside diameter shall be:

$$\overline{\sigma}_a = \frac{p \cdot d_i}{2 \cdot s_{0a}} \cdot \frac{2 \cdot r + 0.5 \cdot d_i}{2 \cdot r + d_i - s_{0a}} + \frac{p}{2} \le S_m \tag{A 5-11}$$

The strength condition for the extrados at given outside diameter shall be:

$$\overline{\sigma}_a = \frac{p \cdot (d_a - s_{0i} - s_{0a})}{2 \cdot s_{0i}} \cdot \frac{2 \cdot R + 0.5 \cdot d_a + 1.5 \cdot s_{0i} - 1.5 \cdot s_{0a}}{2 \cdot R + d_a - s_{0a}} + \frac{p}{2} \le S_m \tag{A 5-12}$$

A 5.2.3 Reducers

Reducers shall be calculated in accordance with the requirements of clause A 2.4.2.

A 5.2.4 Butt welding tees

A 5.2.4.1 Butt welding tees forged from solid

A 5.2.4.1.1 Scope

(1) These calculation rules apply to butt welding tees forged from the solid as well as bored and turned butt welding tees with nominal diameter not exceeding DN 100. They only consider loadings resulting from internal pressure. Additional forces and moments shall be considered separately.

(2) The dimensions “a” and “b” shall not be less than the values given in DIN EN 10253-2 and DIN EN 10253-4 for „F“ and „G“.

(3) The external transition radius r_2 shall be at least $0.1 \cdot d_{Aa}$.

(4) A wall thickness ratio s_A/s_H not exceeding 2 is permitted for d_{Ai} not exceeding 50 mm. This also applies to nozzles with d_{Ai} greater than 50 mm, provided that the diameter radio d_{Ai}/d_{Hi} does not exceed 0.2. For branches with a diameter radio d_{Ai}/d_{Hi} greater than 0.2 the ratio s_A/s_H shall basically not exceed 1.3. Higher values are permitted if

a) the additional nozzle wall thickness exceeding the aforementioned wall thickness ratio is not credited for reinforcement of the nozzle opening but is selected for design reasons

or

b) the nozzle is fabricated with reinforcement area reduced in length (e.g. nozzles which are conical to improve test conditions for the connecting pipe) in which case the lacking metal area for reinforcement due to the reduced influence length may be compensated by adding metal to the reduced influence length

or

c) the ratio of nozzle diameter to run pipe diameter does not exceed 1 : 10.

A 5.2.4.1.2 General

The weakening of the run pipe may be compensated by an increase of the wall thickness in the highly loaded zone at the opening (see **Figure A 5-7**) which can be obtained by forging or machining.

A 5.2.4.1.3 Design values and units

See clause A 5.2.5.3 and **Figure A 5-7** with respect to the design values and units. In addition, the following applies:

Notation	Design value	Unit
d_{Ha}	nominal outside diameter of run pipe at outlet	mm
d_{Aa}	nominal outside diameter for branch connection	mm
s_1	nominal wall thickness of run pipe at outlet	mm
s_2	nominal wall thickness for branch connection	mm
s_A^+	equivalent wall thickness for branch connection	mm
s_H^+	equivalent wall thickness for run pipe at outlet	mm
p^+	allowable internal pressure in tee	N/mm^2

A 5.2.4.1.4 Calculation

(1) For the calculation of the effective lengths of the run and the branch clause A 5.2.5.4.2 shall apply.

(2) The required area of reinforcement shall be determined according to clause A 5.2.5.4.1.

A 5.2.4.1.5 Equivalent wall thicknesses for connection at branch and run pipe outlet

The wall thicknesses s_H^+ and s_A^+ required by Section 8.4 for stress analysis are those wall thicknesses obtained for pipes with the outside diameters d_{Ha} and d_{Aa} if they are dimensioned with the allowable internal pressure p^+ for tees. Then, the following applies:

$$s_H^+ = \frac{p^+ \cdot d_{Ha}}{2 \cdot S_m + p} \tag{A 5-13}$$

$$s_A^+ = s_H^+ \cdot d_{Aa} / d_{Ha} \tag{A 5-14}$$

For simplification $p^+ = p$ can be taken.

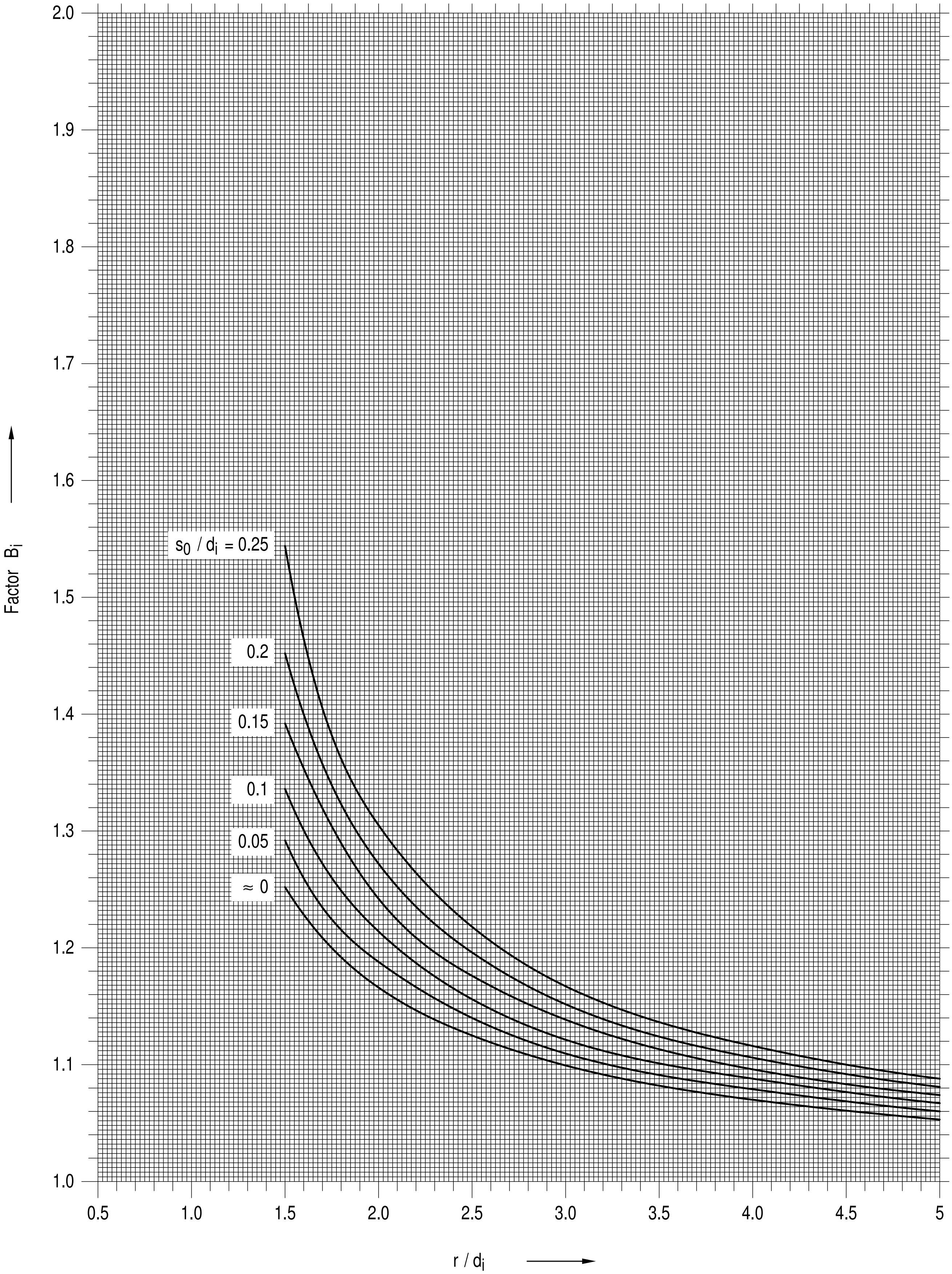


Figure A 5-3: Factor B_i for the intrados at given inside diameter

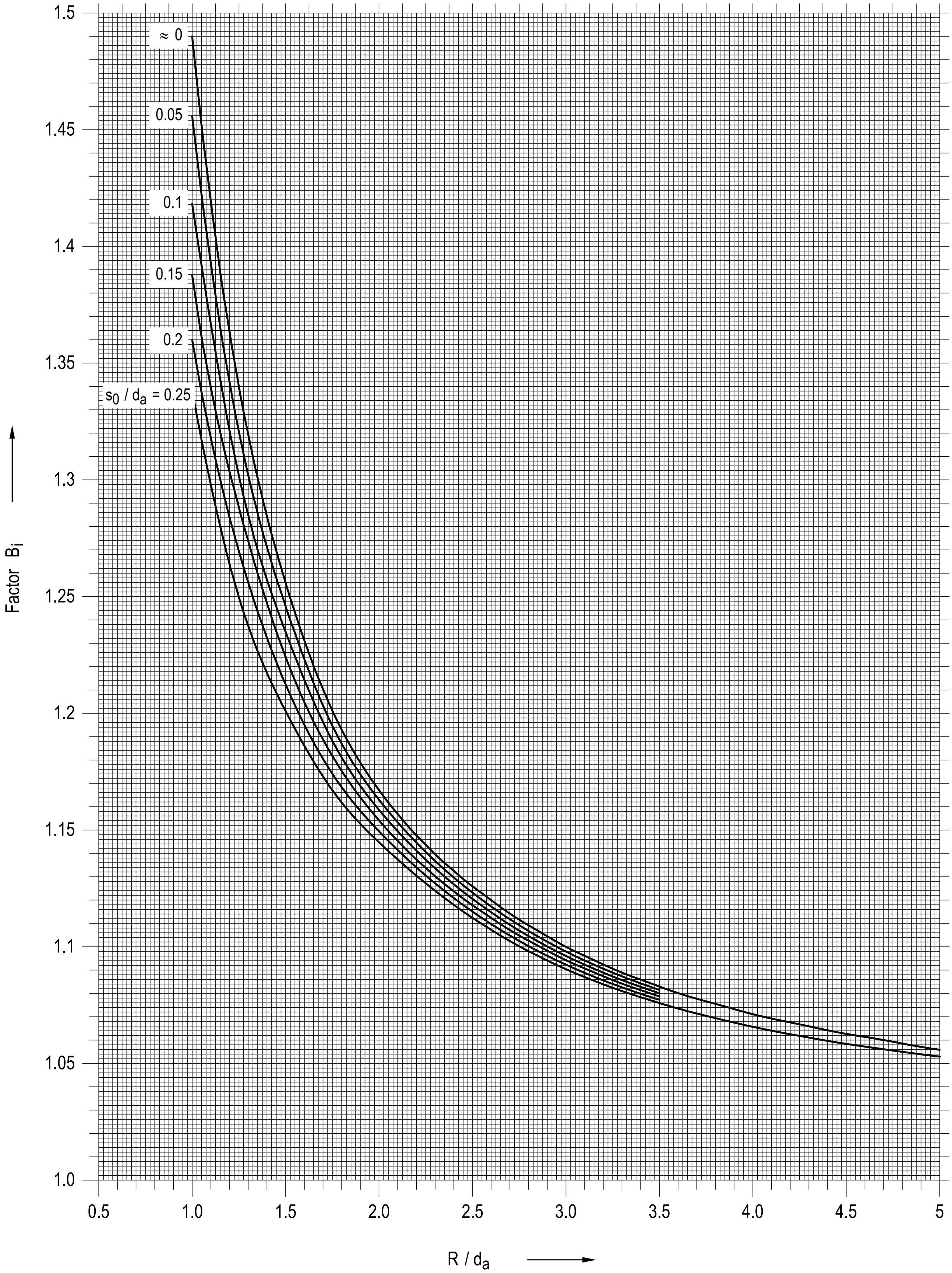


Figure A 5-4: Factor B_i for the intrados at given outside diameter

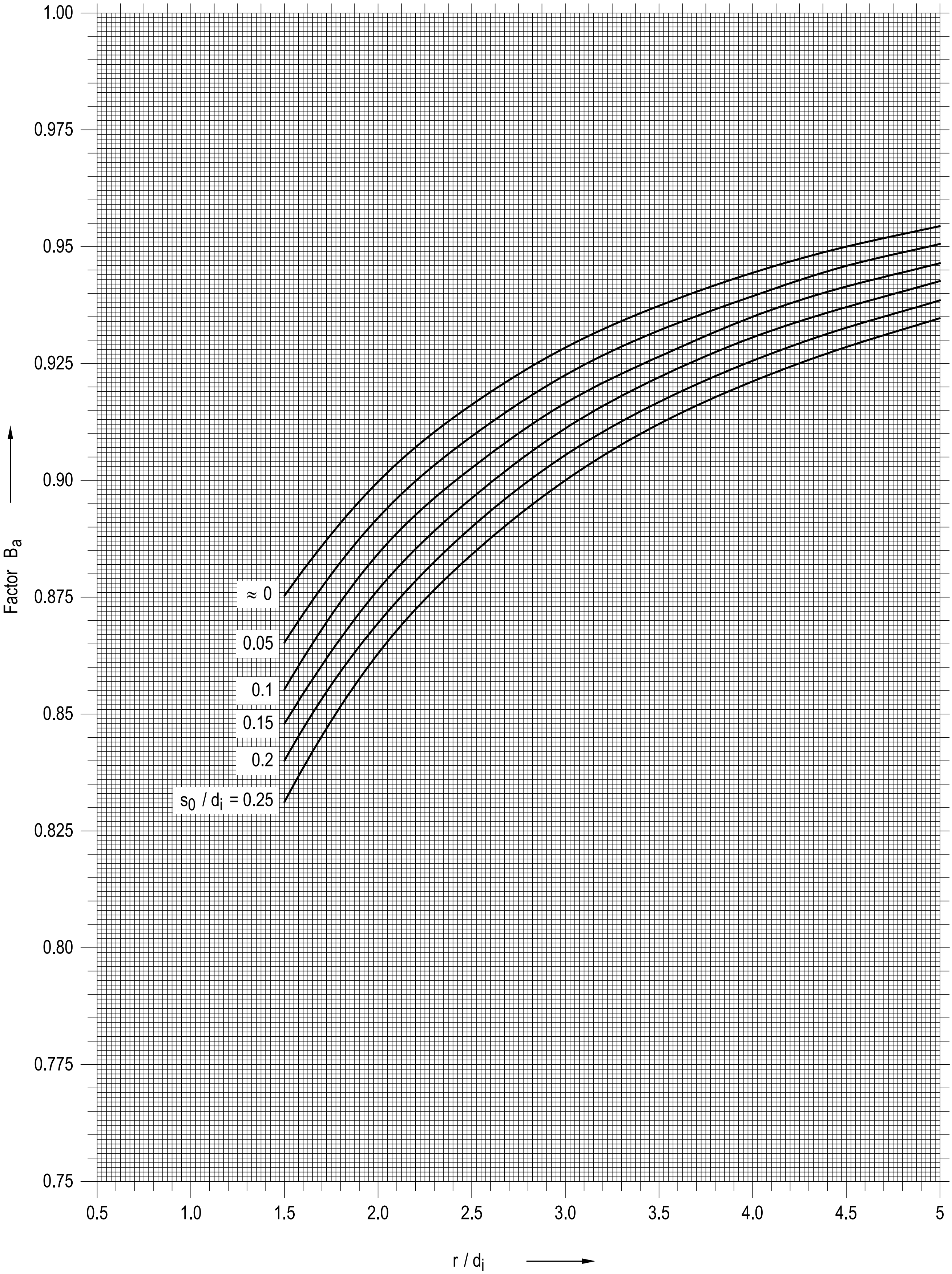


Figure A 5-5: Factor B_a for the extrados at given inside diameter

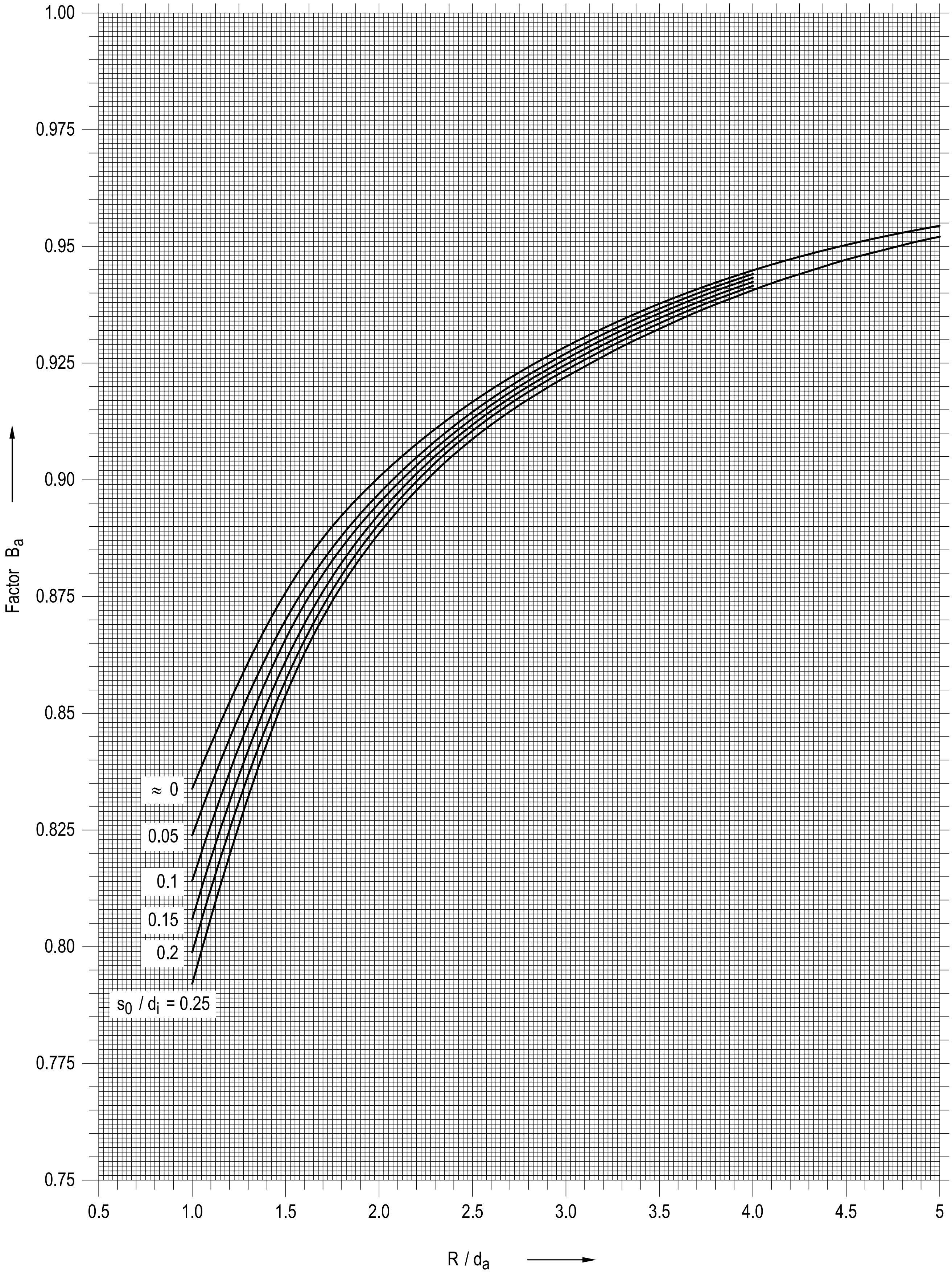


Figure A 5-6: Factor B_a for the extrados at given outside diameter

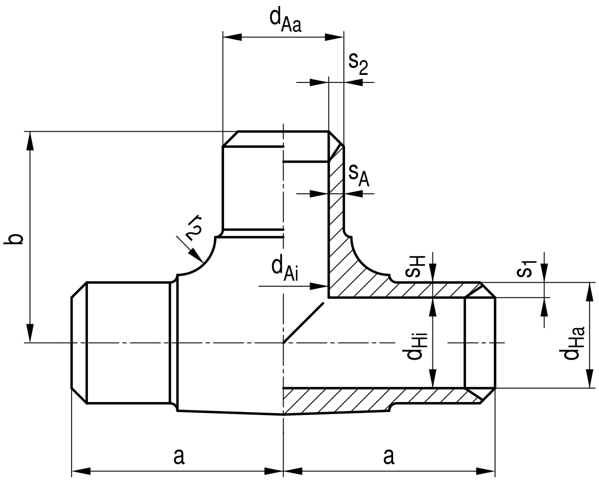


Figure A 5-7: Branch forged from solid, bored or turned

A 5.2.4.2 Die-formed butt welding tees

A 5.2.4.2.1 Scope

(1) These calculation rules apply to seamless tees fabricated by die-forming from seamless, rolled or forged pipes (see **Figure A 5-8**).

(2) The dimensions "a" and "b" shall not exceed the values given in DIN EN 10253-2 and DIN EN 10253-4 for "F" and "G". For tees with nominal diameters exceeding DN 300 the following equations apply for the dimensions "a" and "b":

$$a \geq 0.75\ d_{Ha} \qquad \text{(A 5-15)}$$

and

$$b \geq 0.5\ d_{Ha} + 0.25\ d_{Aa} \qquad \text{(A 5-16)}$$

(3) The external transition radius r_2 shall be at least $0.1 \cdot d_{Aa}$.

(4) At no location shall the wall thickness of the tee be more than twice and not less than 0.875 times the connecting wall thickness s_1. Only at the branch outlet the wall thickness may be reduced to $0.875 \cdot s_2$ on a maximum length of $2 \cdot s_2$.

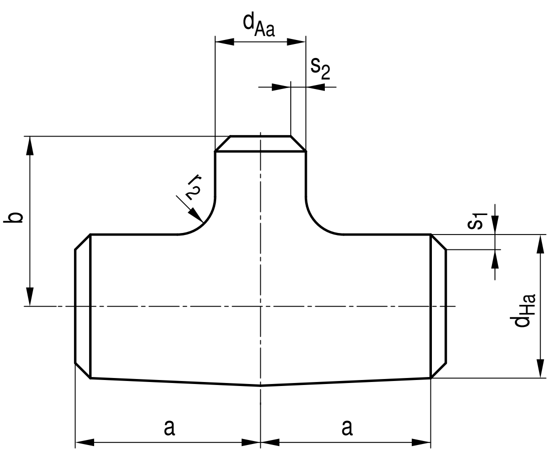


Figure A 5-8: Die-formed butt-welding tee

A 5.2.4.2.2 Design values and units

See clause A 5.2.5.3 and **Figure A 5-8** regarding the design values and units. In addition, the following applies:

Notation	Design value	Unit
A_p	pressure loaded area according to **Figure A 5-9**	mm²
A_σ	effective cross-sectional areas acc. to **Figure A 5-9** upon deduction of wall thickness	mm²
d_{Ha}	nominal outside diameter of run pipe at outlet	mm
d_{Aa}	nominal outside diameter of branch connection	mm
s_H^+	equivalent wall thickness of run pipe at outlet	mm
s_A^+	equivalent wall thickness for branch connection	mm
s_1	nominal wall thickness for run pipe at outlet	mm
s_2	nominal wall thickness for branch connection	mm
α	angle to correspond to **Figure A 5-9**	degree

A 5.2.4.2.3 Calculation

(1) With e_H as maximum value of

$$e_H = d_{Ai} \qquad \text{(A 5-17)}$$

$$e_H = 0.5 \cdot d_{Ai} + s_H + s_A \qquad \text{(A 5-18)}$$

$$e_H = 0.5 \cdot d_{Ai} + s_A + r_2 \cdot (1 - \sin\alpha) \qquad \text{(A 5-19)}$$

however, not to exceed e_H = a, and with e_A as the greater value of

$$e_A = 0.5 \cdot \left(\sqrt{0.5 \cdot d_{Am} \cdot s_A} + r_2\right) \qquad \text{(A 5-20)}$$

$$e_A = r_2 \cdot \cos\alpha, \qquad \text{(A 5-21)}$$

however not to exceed

$$e_A = b - (r_2 + s_H) \cdot \cos\alpha - 0.5 \cdot d_{Hi} \qquad \text{(A 4-22)}$$

the following condition shall be satisfied

$$\sigma_V \leq p \cdot \left(\frac{A_{p1}/\cos\alpha + A_{p2} + A_{p3} + A_{p4}}{A_\sigma} + 0{,}5\right) \leq S_m \qquad \text{(A 5-23)}$$

(2) With e'_H as maximum value of

$$e'_H = 0.5 \cdot \left(d_{Ai} + \sqrt{0.5 \cdot d_{Hm} \cdot s_H}\right) \qquad \text{(A 5-24)}$$

$$e'_H = 0.5 \cdot d_{Ai} + \frac{2}{3} \cdot \left(s_H + s_A\right) \qquad \text{(A 5-25)}$$

$$e'_H = 0.5 \cdot d_{Ai} + s_A + r_2 \cdot (1 - \sin\alpha), \qquad \text{(A 5-26)}$$

however, not to exceed e'_H = a, and with e_A as computed above the following condition shall be satisfied additionally

$$\sigma'_V \leq p \cdot \left(\frac{A'_{p1}/\cos\alpha + \frac{2}{3} \cdot A_{p2} + A_{p3} + A_{p4}}{A'_\sigma} + 0.5\right) \leq S_m$$

(A 5-27)

The areas A_p and A_σ are shown in **Figure A 5-9**.

A 5.2.4.2.4 Equivalent wall thickness for connection of run pipe and branch outlet

(1) The connecting wall thicknesses s_H^+ and s_A^+ required by Section 8.5 for stress analysis then lead to a value S being the greater value obtained from σ_V and σ'_V (see clause A 5.2.4.2.3) to become

$$s_H^+ = \frac{p \cdot d_{Ha}}{2 \cdot S + p} \qquad \text{(A 5-28)}$$

$$s_A^+ = \frac{p \cdot d_{Aa}}{2 \cdot S + p} = s_H^+ \cdot d_{Aa} / d_{Ha} \qquad \text{(A 5-29)}$$

(2) As $S \leq S_m$ must be satisfied, s_H^+ and s_A^+ can also be determined with S_m instead of S.

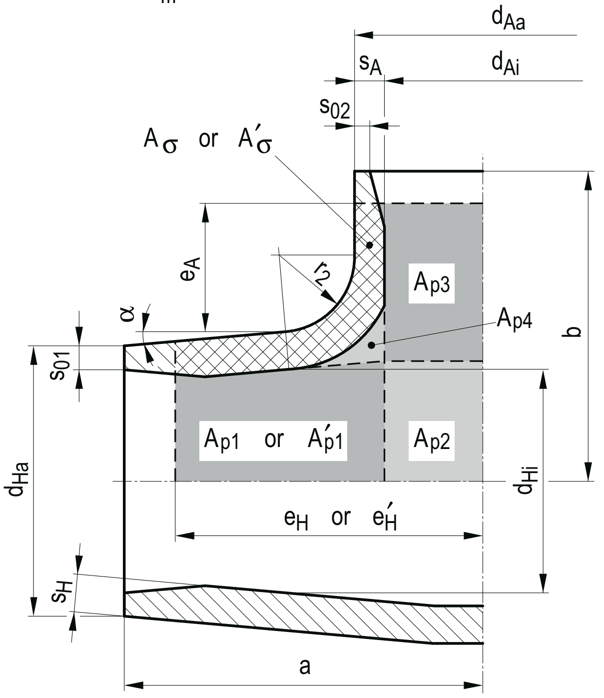


Figure A 5-9: Reinforcement area dimensions for butt welding tees

A 5.2.5 Reinforcement of openings in pipe run

A 5.2.5.1 Scope

(1) The scope of the calculation rules hereinafter is given in clause A 2.2.2.1.

(2) The rules consider the loadings resulting from internal pressure. Additional forces and moments shall be considered separately.

A 5.2.5.2 General

(1) Openings shall normally be circular or elliptical. Further requirements are to be met when using the stress intensity values according to Section 8.5.

(2) The angle β (see **Figure A 2.8-8**) between nozzle axis and run pipe axis shall normally not be less than 60°, but shall normally not exceed 120°.

(3) Openings in a run pipe may be reinforced as follows:

a) by selecting a greater wall thickness for the run pipe than is required for an unpierced run. This wall thickness shall be provided at least up to a length e_H measured from the axis of the opening,

b) by branches which, on a length e_A measured from the surface of the run, have a greater wall thicknesses than is required for internal pressure loading. The material required for reinforcement shall be distributed uniformly over the periphery of the branch,

c) by a combination of the measures shown in a) and b) above.

Regarding a favourable shape not leading to increased loadings/stresses subclause c) shall preferably be used.

(4) In the case of several adjacent openings the conditions for the area of reinforcement shall be satisfied for be planes through the centre of the opening and normal to the surface of the run pipe.

(5) When an opening is to be reinforced the following diameter and wall thickness ratios shall be adhered to:

A wall thickness ratio s_A/s_H not exceeding 2 is permitted for d_{Ai} not exceeding 50 mm. This also applies to branches with d_{Ai} greater than 50 mm, provided that the diameter radio d_{Ai}/d_{Hi} does not exceed 0.2. For branches with a diameter radio d_{Ai}/d_{Hi} greater than 0.2 the ratio s_A/s_H shall basically not exceed 1.3. Higher values are permitted if

a) the additional branch wall thickness exceeding the aforementioned wall thickness ratio is not credited for reinforcement of the nozzle opening, but is selected for design reasons or

b) the branch is fabricated with reinforcement area reduced in length (e.g. branches which are conical to improve NDT conditions for the connecting pipe) where the lacking metal area for reinforcement due to the reduced influence length may be compensated by adding metal to the reduced influence length or

c) the ratio of branch diameter to run pipe diameter does not exceed 1 : 10.

(6) Openings need not be provided with reinforcement and no verification need be made for openings to A 5.2.5.4 if

a) a single opening has a diameter not exceeding $0.2 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, or, if there are two or more openings within any circle of diameter $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, but the sum of the diameters of such unreinforced openings shall not exceed $0.25 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ and

b) no two unreinforced openings shall have their centres closer to each other, measured on the inside wall of the run pipe, than the sum of their diameters, and

c) no unreinforced opening shall have its edge closer than $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ to the centre of any other locally stressed area (structural discontinuity).

Note:

See clause 7.7.2.2 for definition of locally stressed area.

(7) Where run pipe and branch are made of materials with differing design stress intensities, the stress intensity of the run pipe material, if less than that of the branch, shall govern the calculation of the entire design

Where the branch material has a lower design stress intensity, the reinforcement zones to be located in areas provided by such material shall be multiplied by the ratio of the design stress intensity values of the reinforcement material and the run pipe material.

Where the materials of the run pipe and the branch differ in their specific coefficients of thermal expansion, this difference shall not exceed 15 % of the coefficient of thermal expansion of the run pipe metal.

A 5.2.5.3 Design values and units

(See also Figures A 2.8-2 to A 2.8-11 and A 5-10 to A 5-13)

Notation	Design value	Unit
d_{Ai}	inside diameter of opening plus twice the corrosion allowance c_2	mm
d_{Am}	mean diameter of branch	mm
d_{Hi}	inside diameter of run pipe	mm
d_{Hm}	mean diameter of run pipe	mm
d_n	nominal diameter of tapered branch	mm
r_1	inside radius of branch pipe	
r_2	minimum radius acc. to clause 5.2.6	mm

Notation	Design value	Unit
s_A	nominal wall thickness of branch including reinforcement, but minus allowances c_1 and c_2	mm
s_{A0}	calculated wall thickness of branch	mm
s_H	nominal wall thickness of run pipe including the reinforcement, but minus allowances c_1 and c_2	mm
s_{H0}	calculated wall thickness of run pipe	mm
s_R	nominal wall thickness of branch pipe minus allowances c_1 and c_2	mm
s_{R0}	calculated wall thickness of branch pipe	mm
y	slope offset distance	mm
α	angle between vertical and slope (see also Figures A 5-10, A 5-11 and A 5-13)	degree

The following notations can be taken from **Figures A 2.8-8** and **A 2.8-9**:

Notation	Design value	Unit
A_1, A_2, A_3	metal areas available for reinforcement	mm^2
e_A	limit of reinforcement measured normal to the run pipe wall	mm
e_H	half-width of the reinforcement zone measured along the midsurface of the run pipe	mm
e'_H	half-width of the zone in which two thirds of compensation must be placed	mm
β	angle between axes of branch and run pipe	degree

A 5.2.5.4 Calculation

A 5.2.5.4.1 Required reinforcement

(1) The total cross-sectional area A of the reinforcement required in any given plane for a pipe under internal pressure shall satisfy the following condition:

$$A \geq d_{Ai} \cdot s_{H0} \cdot (2 - \sin\beta) \quad \text{(A 5-30)}$$

(2) The required reinforcing material shall be uniformly distributed around the periphery of the branch.

A 5.2.5.4.2 Effective lengths

(1) Credit may be taken for radii or tapers at nozzle-to-basic shell transitions according to clause 5.2.6 in equation (A 5-35) determining the effective length.

(2) The effective length of the basic shell shall be determined as follows:

$$e_H = d_{Ai} \quad \text{(A 5-31)}$$

or

$$e_H = 0.5 \cdot d_{Ai} + s_H + s_A \quad \text{(A 5-32)}$$

The calculation shall be based on the greater of the two values. In addition two thirds of the area of reinforcement shall be within the length $2 \cdot e'_H$ (**Figure A 2.8-8** and **A 2.8-9**), where

e'_H is the greater value of either

$$e'_H = 0.5 \cdot [d_{Ai} + (0.5 \cdot d_{Hm} \cdot s_H)^{1/2}] \quad \text{(A 5-33)}$$

and

$$e'_H = 0.5 \cdot d_{Ai} + \frac{s_A}{\sin\beta} + s_H \quad \text{(A 5-34)}$$

(2) The effective length of a cylindrical branch shall be determined as follows:

$$e_A = 0.5 \cdot [(0.5 \cdot d_{Am} \cdot s_A)^{1/2} + r_2] \quad \text{(A 5-35)}$$

where

$$d_{Am} = d_{Ai} + s_A \quad \text{(A 5-36)}$$

See also **Figures A 5-10, A 5-11, A 5-12**.

(3) The effective length of a tapered branch shall be determined as follows:

$$e_A = 0.5 \cdot (0.5 \cdot d_n \cdot s_A)^{1/2} \quad \text{(A 5-37)}$$

where

$$d_n = d_{Ai} + s_R + y \cdot \cos\alpha \quad \text{(A 5-38)}$$

See also **Figure A 5-13**.

For branches with tapered inside diameter d_n shall be determined by trial and error procedure.

A 5.2.5.4.3 Loading scheme for metal areas available for reinforcement

The metal areas A_1, A_2, A_3 available for reinforcement used to satisfy equation (A 5-30) are shown in **Figures A 2.8-8** and **A 2.8-9**, and shall satisfy the condition $A_1 + A_2 + A_3$ equal to or greater than A.

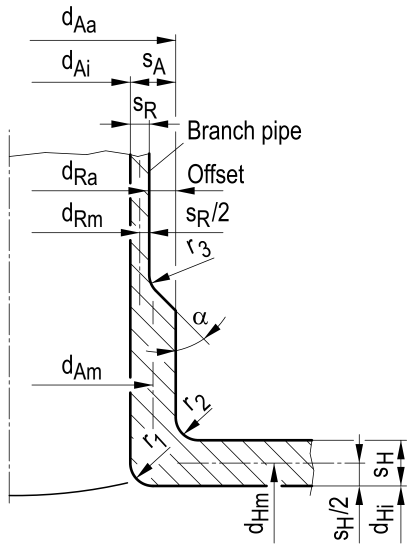


Figure A 5-10: Branch

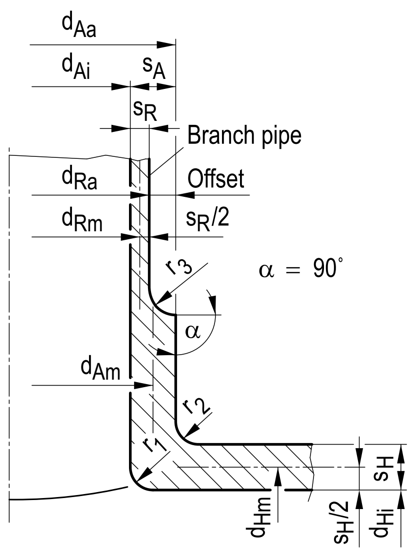


Figure A 5-11: Branch

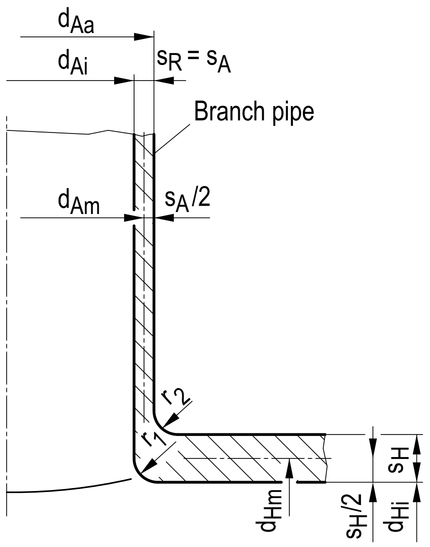


Figure A 5-12: Branch

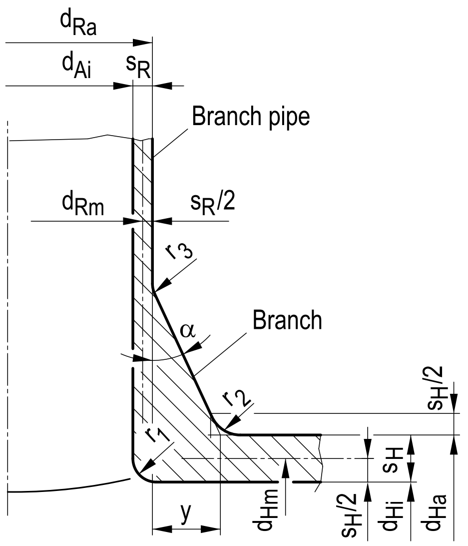


Figure A 5-13: Branch

A 5.2.6 Pipe flanges

Pipe flanges of test group A1 shall be dimensioned in accordance with Section A 2.10.

A 5.3 Test groups A2 and A3

A 5.3.1 Cylindrical shells under internal pressure

The calculation shall be made in accordance with the rules of clause A 5.2.1 in which case the design stress intensity value S acc. to Section 6.7 shall be taken in lieu of S_m in the equations.

A 5.3.2 Bends and curved pipes under internal pressure

The calculation shall be made in accordance with the rules of clause A 5.2.2, in which case the design stress intensity value S acc. to Section 6.7 shall be taken in lieu of S_m in the equations.

A 5.3.3 Reducers

The calculation shall be made in accordance with the rules of Section A 2.4, in which case the design stress intensity value S acc. to Section 6.7 shall be taken in lieu of S_m in the equations.

A 5.3.4 Forged fittings

The requirements of clause A 5.2.4 also apply to test groups A2 and A3.

A 5.3.5 Reinforcement of openings in pipe run

A 5.3.5.1 Scope

(1) The calculation method hereinafter applies to cylindrical shells under internal pressure where the ratio d_a/d_i does not exceed 1.7. Diameter ratios d_a/d_i not exceeding 2 are permitted if the wall thickness s_0 does not exceed 80 mm.

(2) The calculation rules consider the loadings resulting from internal pressure. Additional forces and moments shall be considered separately.

A 5.3.5.2 General

(1) Openings shall be circular or elliptical. For elliptical openings it is assumed that the ratio of the greater to the smaller axis does not exceed 1.5.

That diameter shall be used which lies in the direction of the shell generator. Further requirements are to be met when using the stress intensity values according to clause 8.5.2.8.

(2) The angle β (see **Figure A 2.8-8**) between nozzle axis and run pipe axis shall normally not be less than 60°, but shall normally not exceed 120°.

(3) Openings in a run pipe may be reinforced as follows:

a) by selecting a greater wall thickness for the run pipe than is required for an unpierced run. This wall thickness shall be provided at least up to a length e_H measured from the axis of the opening,

b) by branches which, on a length e_A measured from the surface of the run, have a greater wall thicknesses than is required for internal pressure loading. The material required for reinforcement shall be distributed uniformly over the periphery of the branch,

c) by a combination of the measures shown in a) and b) above.

Regarding a favourable shape not leading to increased loadings/stresses subclause c) shall preferably be used.

(4) In the case of several adjacent openings the conditions for the area of reinforcement shall be satisfied for be planes through the centre of the opening and normal to the surface of the run pipe.

(5) When an opening is to be reinforced the following diameter and wall thickness ratios shall be adhered to:

A wall thickness ratio s_A/s_H not exceeding 2 is permitted for d_{Ai} not exceeding 50 mm. This also applies to branches with d_{Ai} greater than 50 mm, provided that the diameter radio d_{Ai}/d_{Hi} does not exceed 0.2. For branches with a diameter radio d_{Ai}/d_{Hi} greater than 0.2 the ratio s_A/s_H shall basically not exceed 1.0. Higher values are permitted if

a) the additional branch wall thickness exceeding the aforementioned wall thickness ratio is not credited for reinforcement of the nozzle opening, but is selected for design reasons or

b) the branch is fabricated with reinforcement area reduced in length (e.g. branches which are conical to improve NDT conditions for the connecting pipe) where the lacking metal area for reinforcement due to the reduced influence length may be compensated by adding metal to the reduced influence length or

c) the ratio of branch diameter to run pipe diameter does not exceed 1 : 10.

(6) Openings need not be provided with reinforcement and no verification need be made for openings to A 5.3.5.4 if

a) a single opening has a diameter not exceeding $0.2 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, or, if there are two or more openings within any circle of diameter $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$, but the sum of the diameters of such unreinforced openings shall not exceed $0.25 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ and

b) no two unreinforced openings shall have their centres closer to each other, measured on the inside wall of the run pipe, than the sum of their diameters, and

c) no unreinforced opening shall have its edge closer than $2.5 \cdot \sqrt{0.5 \cdot d_{Hm} \cdot s_H}$ to the centre of any other locally stressed area (structural discontinuity).

(7) Where run pipe and branch are made of materials with differing design stress intensities, the stress intensity of the run pipe material, if less than that of the branch, shall govern the calculation of the entire design. Where the branch material has a lower design stress intensity, the reinforcement zones to be located in areas provided by such material shall be multiplied by the ratio of the design stress intensity values of the reinforcement material and the run pipe material.

Where the materials of the run pipe and the branch differ in their specific coefficients of thermal expansion, this difference shall not exceed 15 % of the coefficient of thermal expansion of the run pipe metal.

A 5.3.5.3 Design values and units

(See also **Figures A 2.8-8** and **A 5-10** to **A 5-13**)

Notation	Design value	Unit
d_{Ai}	inside diameter of opening plus twice the corrosion allowance c_2	mm
d_{Am}	mean diameter of branch	mm
d_{Hi}	inside diameter of run pipe	mm
d_{Hm}	mean diameter of run pipe	mm
d_n	nominal diameter of tapered branch	mm
r_1	inside radius of branch pipe	mm
r_2	minimum radius acc. to clause 5.2.6	
s_A	nominal wall thickness of branch including reinforcement, but minus allowances c_1 and c_2	mm
s_H	nominal wall thickness of run pipe including the reinforcement, but minus allowances c_1 and c_2	mm
s_R	nominal wall thickness of branch pipe minus allowances c_1 and c_2	mm
y	slope offset distance	mm
α	angle between vertical and slope (see also Figures A 4-10, A 4-11 and A 4-13)	degree

The following notations can be taken from **Figures A 2.8-8** and **A 2.8-9**:

Notation	Design value	Unit
A_1, A_2, A_3	metal areas available for reinforcement	mm^2
e_A	limit of reinforcement measured normal to the run pipe wall	mm
e_H	half-width of the reinforcement zone measured along the midsurface of the run pipe	mm
e'_H	half-width of the zone in which two thirds of compensation must be placed	mm
β	angle between axes of branch and run pipe	degree

A 5.3.5.4 Calculation

A 5.3.5.4.1 General

A direct calculation of the run pipe wall thickness is not possible for the general case of an oblique branch without or with additional reinforcement due to the various influence factors. At first, the wall thickness s_H shall be a value assumed on the basis of experience gained, and then the correctness of the assumed value shall be checked.

A 5.3.5.4.2 Strength calculation

With the pressure-loaded area A_p (hatched) and the effective cross-sectional areas A_σ (cross-hatched) the strength condition for zone (**Figure A 5-14**) shall be

$$\bar{\sigma} \le p \cdot \left(\frac{A_{pI}}{A_{\sigma I} + A'_{\sigma I}} + \frac{1}{2} \right) \le S_m \qquad \text{(A 5-39)}$$

and for zone II (**Figure A 5-14**)

$$\bar{\sigma} \le p \cdot \left(\frac{A_{pII}}{A_{\sigma II} + A'_{\sigma II}} + \frac{1}{2} \right) \le S_m \qquad \text{(A 5-40)}$$

The effective lengths shall only be inserted for the run pipe with a maximum

$$e_H = \sqrt{(d_{Hi} + s_H) \cdot s_H} \qquad \text{(A 5-41)}$$

and for the nozzle with

$$e_A = \left(1 + 0.25 \cdot \frac{\Psi_A}{90} \right) \cdot \sqrt{(d_{Ai} + s_A) \cdot s_A} \qquad \text{(A 5-42)}$$

Where one nozzle portion projects inside the shell, only the portion $l_{A2} \le 0.5 \bullet e_A$ shall be considered to be contributing to the reinforcement and be used in the calculation. Where the material of the branch has a lower stress intensity as the run pipe with S, the dimensioning can be made on the basis of the strength condition for zone I

$$\left(S - \frac{p}{2} \right) \cdot A_{\sigma I} + \left(S' - \frac{p}{2} \right) \cdot A'_{\sigma I} \ge p \cdot A_{pI} \qquad \text{(A 5-43)}$$

and for zone II accordingly.

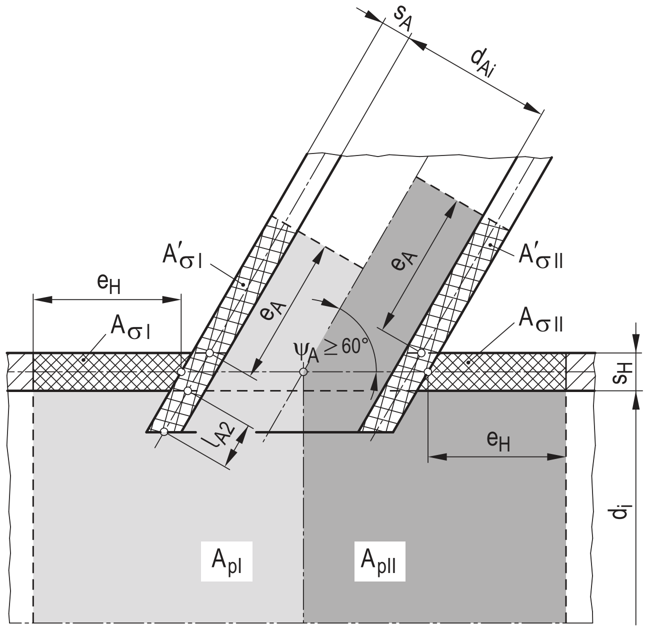


Figure A 5-14: Loading scheme for a cylindrical shell with oblique branch

A 5.3.5.4.3 Branches with $d_{Ai}/d_{Hi} \geq 0.7$

For branches with $d_{Ai}/d_{Hi} \geq 0.7$ and $s_A/s_H < d_{Ai}/d_{Hi}$ the following condition shall be satisfied in the section normal to the run pipe axis at the transition of run pipe to branch:

$$\frac{p}{1.5} \cdot \left(\frac{d_{Hi} + s_H}{2 \cdot s_H} + 0.2 \cdot \frac{d_{Ai} + s_A}{s_A} \sqrt{\frac{d_{Hi} + s_H}{s_H}} \right) \leq S \qquad \text{(A 5-44)}$$

A 5.3.5.4.4 Cylindrical shells with several openings and branches

Adjacent openings of branches shall be treated like single openings or branches if the centre-to-centre distance t_φ according to **Figures A 5-15** and **A 5-16** comply with the following relationship:

$$t_\varphi \leq \left(\frac{d_{Ai1}}{2} + s_{A1} \right) + \left(\frac{d_{Ai2}}{2} + s_{A2} \right) + 2 \cdot \sqrt{(d_{Hi} + s_H) \cdot s_H}$$

(A 5-45)

If this is not the case, the strength consideration for the section passing through the adjacent openings or branches under the respective angle to the shell generator acc. to **Figures A 5-15** and **A 5-16** shall be made in which case the following strength condition applies:

$$\overline{\sigma}_\varphi = \frac{p}{2} \cdot \frac{A_{p0} \cdot \left(1 + \cos^2 \varphi_A\right) + 2 \cdot A_{p1} + 2 \cdot A_{p2}}{A_{\sigma 0} + A_{\sigma 1} + A_{\sigma 2}} + \frac{p}{2} \leq S$$

(A 5-46)

The calculation of oblique or circumferential pitches is made like for a longitudinal pitch in which case in the strength condition of equation (A 5-46) the pressure area $2 \cdot A_{p0}$ has to be corrected by the following factor:

$$\frac{1 + \cos^2 \varphi_A}{2}$$

If the stress intensity of the branches is lower than that of the run pipe, the following applies:

$$\left(S - \frac{p}{2}\right) \cdot A_{\sigma 0} + \left(S' - \frac{p}{2}\right) \cdot A_{\sigma 1} + \left(S'' - \frac{p}{2}\right) \cdot A_{\sigma 2} = \\ = \frac{p}{2} \cdot \left[A_{p0} \cdot \left(1 + \cos^2 \varphi_A\right) + 2 \cdot A_{p1} + 2 \cdot A_{p2}\right] \qquad \text{(A 5-47)}$$

Compare with **Figure A 5-15** and **Figure A 5-16**.

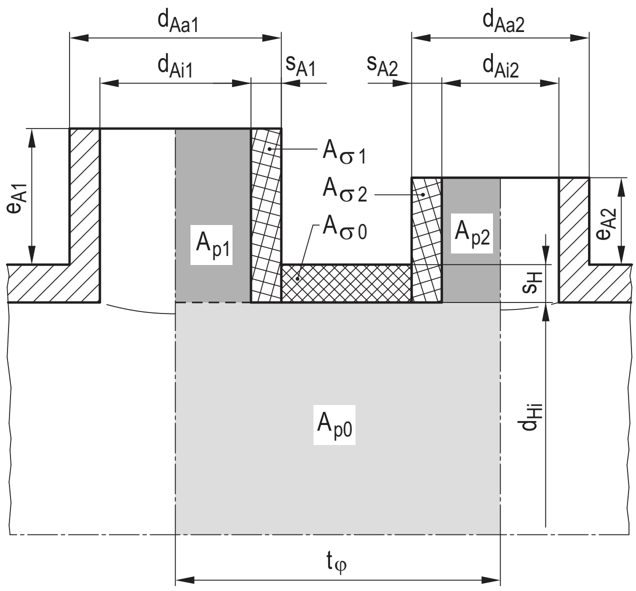


Figure A 5-15: Adjacent openings with differing branch diameters (shown for $\varphi_A = 0$)

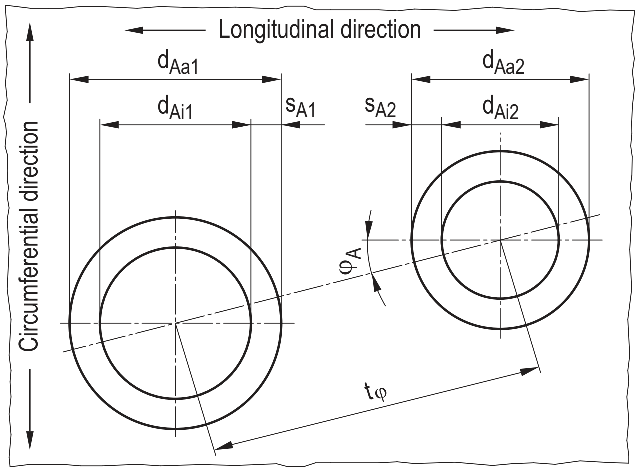


Figure A 5-16: Adjacent openings with differing branch diameters (shown as developed length)

In the case of unequal pitches $t_{\varphi 1}$ and $t_{\varphi 2}$ to **Figure A 5-17** the highest ligament stress shall govern the dimensioning of the run pipe. The run pipe wall thickness shall be provided on both sides of the opening up to a length e_H to equation (A 5-41), measured from the edge of the openings.

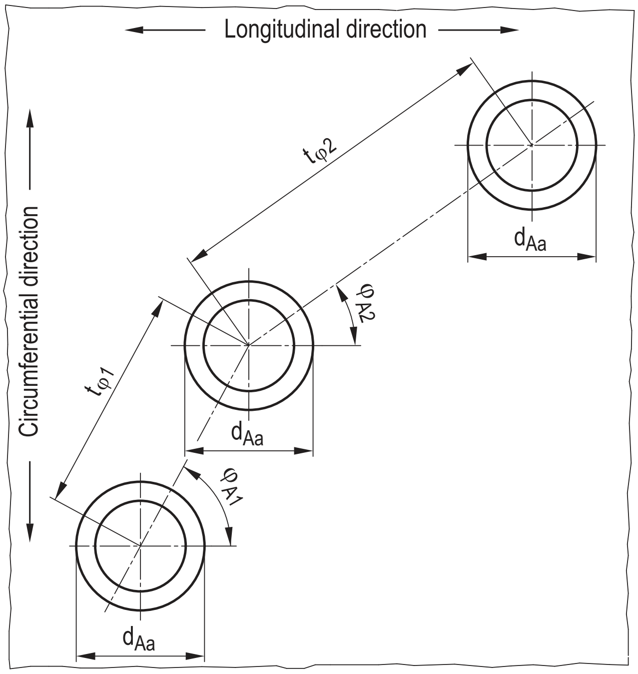


Figure A 5-17: Adjacent openings with different centre-to-centre distances (shown as developed length)

A 5.3.5.4.5 Cylindrical shells with non-radial branch

(1) For cylindrical shells **Figure A 5-18**, sketches a and b, where the branch is non-radial, but is fitted with an angle Ψ_{A1} to the tangent at the run pipe, the higher stress may occur in the cross-section, **Figure A 5-18**, sketch a, or in the longitudinal section, **Figure A 5-18**, sketch b. In both cases the strength condition acc. to equations (A 5-39) and (A 5-40) shall apply, in which case the areas A_p and A_σ, respectively, as shown in the figures, shall be inserted.

(2) The effective lengths shall only be inserted for the run pipe in accordance with equation (A 5-42) and for the branch in accordance with equation (A 5-43) where $\Psi_A = \Psi_{A1}$ shall be taken.

(3) The branch wall thickness s_A shall not exceed the run pipe wall thickness s_H. The weld connecting run pipe and branch shall be a full strength weld as indicated in **Figure A 5-18**.

A 5.3.6 Pipe flanges of test group A2/A3

The dimensioning of pipe flanges of test group A2/A3 shall be made in accordance with Section A 2.10.

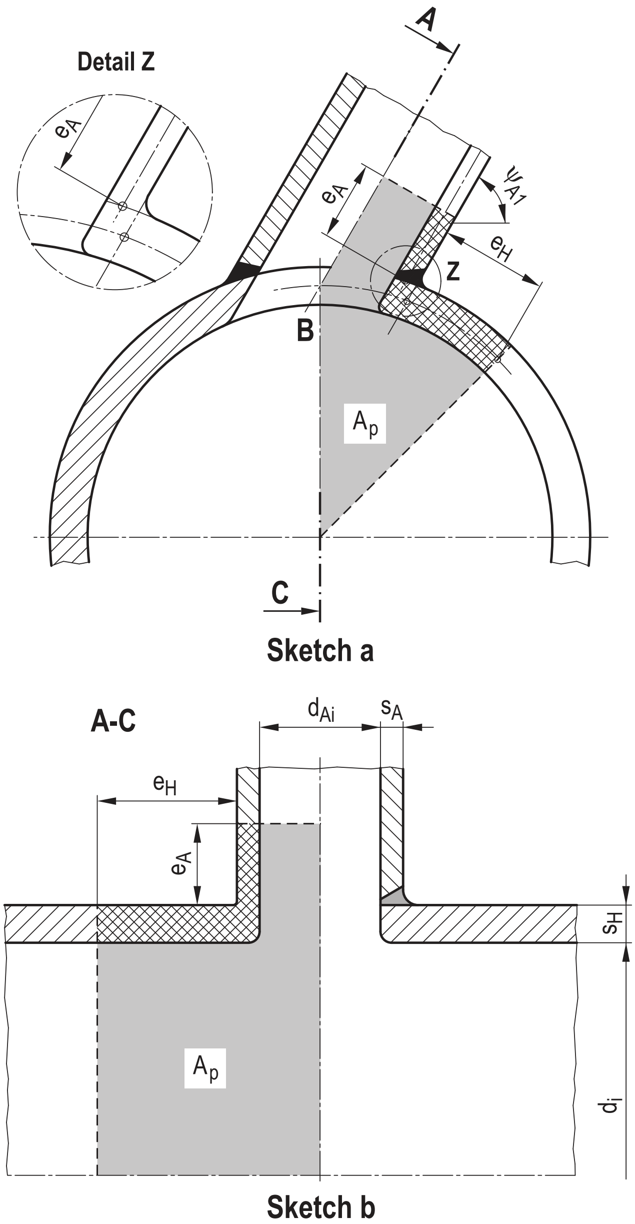


Figure A 5-18: Loading scheme for a cylindrical shell with non-radial branch

Annex B

Requirements as to the primary stress analysis in case of numerical reassessments

B 1 General

(1) This Annex qualitatively and methodically describes an alternate verification procedure as to the numerical reassessment of primary stresses under the prerequisites of Section B2 hereafter if the design requirements based on design loading level (level 0) are not met. This Annex is not applicable to primary stress analyses of new systems and components.

Note:

For the purpose of a transparent verification procedure the technical reasons for the necessity of applying this Annex (e.g. change of safety standard, new knowledge on effects) are indicated in the supporting documentation.

(2) The determination of the general primary membrane stresses shall basically be made to Section 6 in case of numerical reassessments.

(3) On the basis of the actual knowledge on possibly occurring load cases the values for pressure, temperature and additional loads used in reassessment are determined more exactly. The applicability of these values shall be justified.

Note:

Depending on the knowledge on possibly occurring load cases several data sets for the pressure, temperature and additional loads values may be determined for the reassessment.

(4) In case of a reassessment of primary stresses by analysis the verification procedure to Section B 3 may be used.

(5) If the verification procedure to Section B 3 is applied, the reassessment and the loads used shall be documented in the plant documentation in due consideration of the requirements of KTA safety standard 1404 so that at a later date no loads exceeding the verified values can be considered to be acceptable.

B 2 Prerequisites

(1) The component to be reassessed satisfies the principles of Basis Safety.

(2) The safety valves and other safety equipment are adjusted such that the pressure during specified normal operation exceeds the reassessed pressure only for a short period of time and the loading levels of level B are adhered to.

(3) If allowances are omitted with regard to the design according to design approval documents 1 (e.g. if the maximum pressure of a specified load case is covered by the design pressure) this is justified for safety reasons.

(4) When actual dimension are used the measurement and evaluation methods shall be indicated within the reassessment procedure.

B 3 Reassessment procedure

(1) The load case data consist of the values fixed according to B 1(3) for reassessment pressure, temperature and additional loads to be considered.

(2) The reassessment pressure for a component or part shall be at least the greatest pressure difference between the pressure loaded areas according to loading level A.

(3) The reassessment temperature is intended to determine the strength values. It shall be at least equal to the wall temperature to be expected at the point considered for the governing mechanical load case as per (2) and (4).

(4) In case of superposition with the reassessment pressure, the additional reassessment loads shall be at least so high that they cover the simultaneously acting unfavourable primary loadings of level A.

(5) The primary stress analysis shall be made

a) on the basis of the effects as per (2) and (4) in correspondence with Section 6 in compliance with level 0 loading limits and

b) for loading levels B, C, D and P by adherence to the loading limits of the respective level

as per **Table 7.7-4**, in which case the actual geometric dimensions of the parts (e.g. wall thickness) may be used.

(6) The analysis of the mechanical behaviour may either be verified by a general analysis as per Section 7 or by a component-specific analysis as per Section 8.

Annex C

Regulations and Literature referred to in this Safety Standard

(The references exclusively refer to the version given in this annex. Quotations of regulations referred to therein refer to the version available when the individual reference below was established or issued.)

Atomic Energy Act (AtG)		Act on the Peaceful Utilization of Atomic Energy and the Protection against its Hazards (Atomic Energy Act) of December 23, 1959 (BGbl. I, p. 814) as Amended and Promulgated on July 15, 1985 (BGBl. I, p. 1565), last Amendment by article 2 of the Law dated 23rd July 2013 (BGbl. I, 2013, no. 41, p. 2553)
StrlSchV		Ordinance on the Protection against Damage and Injuries Caused by Ionizing Radiation (Radiation Protection Ordinance) dated 20th July 2001 (BGBl. I 2001, No. 38, p. 1714), at last amended by article 5 para. 7 of the Law dated 24th February 2012 (BGbl. I, p. 212)
Safety Criteria	(1977-10)	Nuclear Power Plant Safety Criteria Promulgation as of 21 October 1977 (BAnz. no. 206, 3rd November 1977)
SiAnf		Safety Requirements for Nuclear Power Plants of November 22, 2012 (BAnz AT of January 24th, 2013)
Incident Guidelines	(1983-10)	Guidelines for the Assessment of the Design of PWR Nuclear Power Plants against Incidents pursuant to Sec. 28, para (3) of the Radiological Protection Ordinance (Incident Guidelines 1983) (Annex of BAnz. no. 245, 31st December 1983)
RSK-LL	(1981-10)	Reactor Safety Committee Guidelines for Pressurized Water Reactors, 3rd edition, 14 October 1981) BAnz. No. 69 dated 14 April 1982
KTA 1404	(2013-11)	Documentation during the construction and operation of nuclear power plants
KTA 2201.4	(2012-11)	Design of nuclear power plants against seismic events; Part 4: Components
KTA 3201.2	(2013-11)	Components of the Reactor Coolant Pressure Boundary of Light Water Reactors; Part 2: Design and analysis
KTA 3205.1	(2002-06)	Component Support Structures with Non-Integral Connections; Part 1: Component Support Structures with Non-Integral Connections for Components of the Reactor Coolant Pressure Boundary of Light Water Reactors
KTA 3211.1	(2000-06)	Pressure and Activity Retaining Components of Systems outside the Primary Circuit; Part 1: Materials
KTA 3211.3	(2012-11)	Pressure- and Activity Retaining Components of Systems outside the Primary Circuit; Part 3: Manufacture
KTA 3211.4	(2012-11)	Pressure and Activity Retaining Components of Systems outside the Primary Circuit; Part 4: Inservice Inspections and Operational Monitoring
DIN 267-13	(2007-05)	Fasteners - Technical specifications - Part 13: Parts for bolted connections with specific mechanical properties for use at temperatures ranging from -200 °C to +700 °C
DIN EN ISO 898-1	(2013-05)	Mechanical properties of fasteners made of carbon steel and alloy steel - Part 1: Bolts, screws and studs with specified property classes - Coarse thread and fine pitch thread (ISO 898-1:2013); German version EN ISO 898-1:2013
DIN EN ISO 898-2	(2012-08)	Mechanical properties of fasteners made of carbon steel and alloy steel - Part 2: Nuts with specified property classes - Coarse thread and fine pitch thread (ISO 898-2:2012); German version EN ISO 898-2:2012
DIN EN 1092-1	(2013-04)	Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories, PN designated - Part 1: Steel flanges; German version EN 1092-1:2007+A1:2013
DIN EN ISO 2081	(2009-05)	Metallic and other inorganic coatings - Electroplated coatings of zinc with supplementary treatments on iron or steel (ISO 2081:2008); German version EN ISO 2081:2008
DIN 2510-1	(1974-09)	Bolted connections with reduced shank, survey and installation; studies relating to the calculation of bolted connections
DIN 2510-2	(1971-08)	Bolted Connections with Reduced Shank; Metric Thread with Large Clearence, Nominal Dimensions and Limits
DIN 2510-3	(1971-08)	Bolted Connections with Reduced Shank; Stud-bolts
DIN 2510-4	(1971-08)	Bolted Connections with Reduced Shank; Studs
DIN 2510-7	(1971-08)	Bolted Connections with Reduced Shank; Extension Sleeves
DIN EN ISO 3506-1	(2010-04)	Mechanical properties of corrosion-resistant stainless steel fasteners - Part 1: Bolts, screws and studs (ISO 3506-1:2009); German version EN ISO 3506-1:2009

DIN EN ISO 3506-2	(2010-04)	Mechanical properties of corrosion-resistant stainless steel fasteners - Part 2: Nuts (ISO 3506-2:2009); German version EN ISO 3506-2:2009
DIN EN ISO 3506-3	(2010-04)	Mechanical properties of corrosion-resistant stainless steel fasteners - Part 3: Set screws and similar fasteners not under tensile stress (ISO 3506-3:2009); German version EN ISO 3506-3:2009
DIN EN 10253-2	(2008-09)	Butt-welding pipe fittings - Part 2: Non alloy and ferritic alloy steels with specific inspection requirements; German version EN 10253-2:2007
DIN EN 10253-4	(2008-06)	Butt-welding pipe fittings - Part 4: Wrought austenitic and austenitic-ferritic (duplex) stainless steels with specific inspection requirements; German version EN 10253-4:2008
DIN EN 12516-2	(2004-10)	Industrial valves - Shell design strength - Part 2: Calculation method for steel valve shells; German version EN 12516-2:2004
DIN EN 13555	(2005-02)	Flanges and their joints - Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connections; German version EN 13555:2004
DIN 28011	(2012-06)	Torispherical heads
DIN 28013	(2012-06)	Ellipsoidal heads
VDI 2230 Blatt 1	(2003-02)	Systematic calculation of high duty bolted joints - Joints with one cylindrical bolt
AD 2000-MB A 5	(2000-10)	Openings, closures and closure elements
AD 2000-MB B 13	(2012-07)	Single-ply bellows expansion joints
AD 2000-MB S 3/2	(2004-02)	General verification of stability for pressure vessels - Verification of load-carrying capacity for horizontal vessels on saddle supports
AD 2000-MB S 3/4	(2001-09)	General verification of stability for pressure vessels - Vessels with support brackets
AD 2000-MB W 0	(2006-07)	General principles for materials

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(Stress intensification factor K_e for the determination of plastic strains obtained from elastic design),
Technische Überwachung 35 (1994), No. 6, pp. 268 - 278)

[2] WRC Bulletin 297 (September 1987)
Local Stresses in Cylindrical Shells due to External Loadings on Nozzles-Supplement to WRC Bulletin No. 107 (Revision I)

[3] WRC Bulletin 107 (August 1965, Revision März 1979)
Local Stresses in Spherical and Cylindrical Shells due to External Loadings

[4] British Standard PD 5500:2000, Annex G
Specification for unfired fusion welded pressure vessels

[5] TEMA
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[6] ASME Boiler and Pressure Vessel Code
Section VIII, Division 1, Appendix AA, 2001

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(Sealing properties of gaskets for bolted flange connections; 28th MPA-Seminar, 10th and 11th October 2002, Stuttgart)

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(Determination of properties for flange connections of floating type and metal-to-metal contact type; 27th MPA-Seminar, 4th and 5th October 2001, Stuttgart)

Annex D (informative)

Changes with respect to the edition 1992-06 and explanations

To Section 1 “Scope”

The scope was put more precisely as regards piping and valves with ≤ DN 50.

To Section 2 “General principles and definitions”

(1) At several locations, Section 2 “General principles and definitions” was actualised in conformance with KTA 3201.2 (1996-06).

(2) The definitions of terms were comprised in a separate section. The related requirements were put more precisely.

(3) Sub-clause (12) in KTA 3211.2 (1992-06) was deleted as sub-clause 7.7.1 (6) contains adequate rules.

(4) Within the procedures for changing KTA Safety Standards 3211.2 and 3211.3 Table 2-1 was changed with respect to the classification criteria of test group A3 in order to eliminate the differences existing between the general specification for basic safety, the convoy specifications and the requirements of the KTA 3211 series. The formulation used up to now “allowable materials outside the scope of KTA 3211.1 for plants liable to supervision” was replaced by the more distinct limitation “materials within the scope of AD 2000-Merkblatt W 0”. The former stipulations for the use of austenitic materials with s ≤ 16 mm and ≤ DN 150 were no more taken over since KTA 3211.1 (2000-06) contains respective requirements.

To Section 3 “Load case classes as well as design, service and test loadings and limits of components”

(1) The text in clause 3.2.4.1 was adapted to the formulations of the document “Safety Criteria for Nuclear Power Plants” and KTA 3201.2 (1996-06).

(2) In Section 3.3 notes and information were added to loading levels 0, A, B, C, and D to describe the objectives of verification of the various loading levels.

(3) Clause 3.3.3.2 was adapted to comply with the supplemented text considering primary stress limits in Table 7.7-4 for loading level A.

(4) The stipulations for consideration of level C load cases in the fatigue analysis were put more precisely in clause 3.3.3.4 and in Table 7.7-4. Any cycle occurred due to level C events with respect to its contribution to component fatigue shall be considered within operational monitoring (see KTA 3211.4, Section 8.1).

(5) The requirement of clause 3.3.3.6 was changed such that any pressure testing is to be considered in the fatigue analysis if the number of pressure tests exceeds 10.

To Section 4 “Effects of the components due to mechanical and thermal loadings and fluid effects”

A new formulation was included to require that the fluid effects on component fatigue are to be considered according to the state of science and technology. The following changes were made:

a) At any pertinent location “corrosion and erosion” was replaced by the more general formulation “fluid effects”.
b) Sub-clause 3 was supplemented to say that fluid effects may reduce the fatigue strength.
c) In Section 4.5, a new sub-clause was added to cover, in connection with clause 7.8.3, requirements for the case where uncertainty as to the fluid effect on component integrity exists.

To Section 5 “Design”

(1) At several locations, Section 5 was editorially changed such that the formulations represent design requirements (e.g. clause 5.1.4.2, clause 5.3.2.3). The requirements as to favourable conditions for component service loadings were supplemented to include loading by thermal stratification.

(2) Clause 5.2.4 was put more precisely to cover the arrangement of bolts in flanges in compliance with the conventional flange design rules.

(3) Figure 5.2-2 was editorially revised. As regards the test length notations, Figures 5.2-5, 5.2-7, 5.3-1, 5.3-2, 5.3-3, and 5.3-4 were adapted to KTA 3211.3.

(4) In Section 5.2.5 the reference was extended to sheets 1 to 4 of DIN 2510 since all sheets contain design requirements for reduced shank bolts. By including sub-cause 5 it was pointed out that the design of threaded connections shall ensure a mainly tensile loading of the bolts.

(5) The maximum allowable wall thickness ratio of nozzle to shell (Section 5.2.6, sub-clause 3, Table 5.2-1) $s_A/s_H \le 1.3$ is a conservative rule and is subject to basic safety. It shall ensure both a favourable distribution of stresses and wall thickness reserves in the shell to withstand nozzle forces. Detail analyses performed show that the principles of basic safety can be adhered to even if the wall thickness ratio is $s_A/s_H \le 1.5$. For this reason, an individual rule was taken over in KTA 3211.2 to require that a general analysis of the mechanical behaviour to Section 7 has to be performed in any case.

(6) Clause 5.3.2.4 in conjunction with Section 8.2.5 was adapted to the current edition of AD 2000-Merkblatt B 13.

To Section 6 “Dimensioning”

(1) The formulations in Section 6.1 were put more precisely and supplemented

a) on account of a formal contradiction between the former sub-clauses 2 and 5 which was noted by the Federal Ministry for the Environment,
b) upon evaluation of the Reactor Safety Committee statement dated 24th July 2008 (410th meeting) “Strength hypotheses in the range of application of KTA safety standards when re-evaluating components and systems; evaluation of safety aspects regarding the question of optional application of the von Mises or Tresca yield criterion in KTA safety standards”,
c) upon evaluation of the sub-committee “Program and Principle Questions for Comprehension of KTA rules” (UA-PG), 33rd UA-PG meeting dated 10th March 2010,
d) for inclusion of stipulations regarding limit analysis.

This aims at clearly fixing dimensioning requirements and excluding mal-interpretation of requirements, where possible.

(2) The requirements for claddings in Section 6.3 were put more precisely.

(3) The stress limit term $R_{p0,2T}/1.5$ was included in Table 6.6-1 (dimensioning to Annex A for rolled and forged austenitic steels) which deviates from the ASME Code terms (min $\{R_{p0,2RT}/1.5$; $R_{p0,2T}/1.1$; $R_{mRT}/3{,}0$; $R_{mT}/3.0\}$) to corre-

spond to the dimensional equations not taken over from the ASME Code, but from German technical rules to take credit for the stress limitations laid down in these rules. This is especially of importance for materials where, on account of the material properties, the strain limit R_p will govern dimensioning, and strain limitation would no more suffice due to the calculation procedure on which dimensioning is based. One example for this is the deformation behaviour of dished heads.

As the stress intensity limitation concept in the analyses of the mechanical behaviour on the basis of linear-elastic stress analyses, where limit load factors at non-uniform stress distribution are used, principally will not be distinguished from the stress intensity limitation concept in conventional design rules, it is considered suitable to base dimensioning on the same stress intensity limits if dimensioning is performed to verify primary stresses.

Taking the aforementioned additional stress intensity limit term into account, the allowable stress level in primary stress verification will be less than the stress intensity limit required by the ASME Code for austenitic materials. This also applies to other configurations where an adaptation is not required (e.g. for cylindrical shells under internal pressure). The proposed method for calculating the equivalent stress intensity for dimensioning purposes thus is a simplification to additionally contain some conservatism.

(4) Footnote 3 was added to Table 6.7-1 to ensure that in no case will the allowable stresses for components of test groups A2 and A3 exceed the allowable stresses of test group A1.

(5) Table 6.7-2 was put more precisely and extended to cover the line “Total Stress” and a stress intensity limit for taking credit of the torsional moment applied during bolt assembly by means of a torque wrench. The table was changed such that the assembly condition is shown under a separate column and the allowable stresses for Level P are only indicated for the test condition.

To Sections 7.1 to 7.7

(1) In clause 7.1.2 it was made clear that welds are to be included in fatigue analyses.

(2) As elastic-plastic analyses cannot always be based on actual stress-strain relationships and no concrete requirements have been laid down up to now, the last sentence in sub-clause 7.3 (1) was changed.

(3) Within the verification it shall be evaluated how far the various factors of influence may affect the results during modelling and to what extent they have to be included in the model. If the evaluation shows that direct inclusion into the model is not required, the requirement to take credit of such influences is sufficiently satisfied. Therefore, sub-clauses (3) and (5) of clause 7.6.2.3 were uniformly changed to read “shall be considered”.

(4) Clause 7.6.2.6 was adapted to KTA 3201.2 (1996-06).

(5) Within the process of changing this KTA Safety Standard the possibility was checked of including procedural requirements for the analysis of damping behaviour with respect to the service loadings. It was found out that concrete requirements have to be subject to individual considerations. The following procedure is principally to be followed:

The essential parameters of a dynamic piping analysis are the

- calculation methods to the “modal analysis” or “direct integration”,
- selection of the damping degree for operational events or incidents,
- consideration of the so-called “frequency shift”.

When performing dynamic piping analysis methods distinction is made between the modal analysis and the direct integration (time history method). Both methods are considered to be equivalent. The damping parameters on which the analysis is based, either are the constant damping for all frequencies or a frequency-dependent Raleigh damping with the two parameters α and β. From the user’s point of view, the more simple application of constant damping for all frequency ranges of Raleigh damping are to be preferred since here the damping hyperbola has to be drawn up between two significant frequencies. As regards the determination of these frequency points – e.g. first natural frequency and a higher frequency subject to engineering judgment to an extent of approximately 60 Hz to 80 Hz – extensive engineering or calculus of variations may be necessary.

In the American ASME Code (Section III, Division 1, Appendices, Table N-1230-1) a damping value of 5 % is recommended for piping systems both for the “Operational Basis Earthquake (OBE)” and the “Safe Shutdown Earthquake (SSE)”. This value applies independently of the frequency. The proposed damping value has been increased compared to former editions of the ASME code to prevent optimum piping system design from being impaired by too stringent requirements for earthquake resistance. Here, the experience made was considered that comparably flexibly routed piping systems in fossil-fired power plants and chemical plants will not fail in the event of earthquake.

The selection of the damping value in dynamic piping system analyses is primarily based on KTA 2201.4 where the usually used damping value of 4 % is derived for external events. Operational loadings are also often based on this damping value of 4 %.

In addition, the results of dynamic piping system analyses are considerably influenced by the selection of the so-called frequency shift. By this means, inaccuracies in the application of the system masses and geometric lengths shall be considered such that resonance effects, if any, between natural and excitation frequency in the piping system can be verified. In practice, a frequency shift of ± 2 % (for both frequency types) is often used as indication for resonance effects, if any.

VDI 3842 “Vibration in piping systems” (2004-06) contains explanations to vibration calculations.

(6) To avoid mal-interpretations, the definition of local primary stresses in clause 7.7.2.2 was revised in correspondence with the ASME Code.

(7) At several locations, clause 7.7.3 “Superposition and evaluation of stresses” was put more precisely to clarify the requirements.

(8) Section 7.7.4 “Limit analysis” was taken over from KTA 3201.2 (1996-06) for test group A1 and adapted to test groups A2/A3 in conformance with the ratio of S_m/S values and the required limit load.

(9) In Table 7.7-4 a footnote was included for loading level 0 to refer to the new normative Annex B which describes the procedure for numerical reassessments of a component.

To Section 7.8 “Fatigue Analysis”:

(1) At several locations the formulation was put more precisely.

(2) Within the process of changing this KTA safety standard, the national and international knowledge on fatigue curves to be applied within the fatigue analysis for ferritic and austenitic materials was discussed in detail and evaluated. Here, the influence exerted by the environment was treated in detail.

(3) In safety standard KTA 3211.2 (edition 1992-06) the ASME design fatigue curve (“Langer curve” [1]) dating from the

1960ies which considered various moduli of elasticity, formed the basis for fatigue analysis. The evaluation of later more comprehensive test results obtained in the USA and in Japan for austenitic materials shows that the average value curve (under air), which was the basis of the original (old) ASME design curve, can lead to non-conservative results starting at 10^4 load cycles [2] [NUREG/CR-6909].

For this reason, a new design fatigue curve for austenitic materials was introduced in ASME Section III, Appendix 1, edition 2009b. Compared to the old curve, the new design fatigue curve shows the following changes:

a) in the short-term range 10^1 to $5 \bullet 10^2$ load cycles higher allowable stress amplitudes or at a given stress amplitude a higher permissible number of load cycles,

b) in the range 10^3 to 10^5 load cycles lower allowable stress amplitudes or at a given stress amplitude a lower permissible number of load cycles,

c) inclusion of a high-cycle fatigue range up to 10^{11} load cycles,

d) omission of the curves A – C.

Within several research projects [3] to [7] studies were made to find out in how far the new ASME design fatigue curve under air is transferrable and applicable to the stabilised austenitic materials 1.4550 and 1.4541 used in German nuclear power plants. By means of the results obtained independent average design fatigue curves for room temperature and temperatures exceeding 80 °C and, based on these results, new design fatigue curves were derived [8] to [10]. Contrary to the evaluation of NUREG/CR-6909 [2], where no influence of temperature on the average value curve is assumed, the results in [3], [5] and [7] show a non-negligible temperature influence in the range starting with 10^4 load cycles. For this reason, average value and design fatigue curves are explicitly derived at room temperature and temperatures exceeding 80 °C for the stabilised austenitic materials 1.4550 and 1.4541 [8]. In the USA the influence of the temperature on the average value curve under air is not assessed to be significant (NUREG/CR-6909 [2], chapter 5.3.1). Therefore, no differentiation of the design fatigue curve is made in the ASME Code as regards the temperature influence. When evaluating the influence of the environment using F_{en} factors, the temperature is explicitly taken into account [2]. For the other austenitic steels the design fatigue curves of the ASME Code starting with the edition ASME 2009b were taken over. For ferritic materials the current design fatigue curves remain applicable.

(4) With regard to a quantitative evaluation of the influence by the environment on the fatigue strength tests were performed under the research projects [11] to [14] and evaluated by comparison with the "Environmental Factor" F_{en} presented in NUREG/CR-6909 [2]. Therefore, evaluation criteria to cover the environmental influence on fatigue behaviour are available on account of the laboratory studies, however, the results are evaluated to be conservative. At present, there is no uniform procedure in technical rules on international level to cover a possible environmental influence on fatigue strength. This is especially made clear in the reports [15] and [16] dealing with the calculation of F_{en}.

On international technical level, several calculation procedures exist to cover the environmental influence. Besides the NUREG/CR 6909 procedure further numerical approaches exist. Among these approaches is the approximation of Argonne National Laboratory (ANL) published in the report ANL-LWRS47 as well as a procedure discussed on the ASME Code Meeting "Section III Subgroup on Fatigue Strength", Nashville TN, May 15, 2012. A proposal for a detailed guideline for a numerical procedure can be found in [16]. A further unification of the American procedure can be expected [17] if at the end of 2014 the revisions of NUREG/CR 6909 [2] and of Reg. Guide 1.207 [18] have been published.

The laboratory tests up to now were nearly exclusively performed under constant uniaxial loading conditions. New knowledge gained suggests the service-life favouring influence of near-realistic long holding times as regards loading conditions (holding time effect). This applies to both air environment and LWR environmental conditions, see e.g. [19] to [22]. Consideration of this effect will be envisaged upon provision of secured experimental tests, if required.

At present, there are only insufficient test results available for a statistically confirmed quantification as regards the distribution of proportional shares of temperature end environmental influences, respectively. First examinations, however, show the temperature influence under various ambient conditions (vacuum, air and environment) [23]. It is shown that the service-life limiting environmental influence currently proved by experimental investigations already contains a non-quantifiable share of the temperature influence which depends on the strain amplitude and the allowable number of cycles, respectively [24].

The available test results obtained by AREVA-SAS as regards the environmental influence with realistic transient strain histories show that the original (old) ASME design fatigue curve for austenite both under air and under moderate environmental conditions (based on a factor $F_{en} = 3$) further leads to conservative results [25] to [27].

Where fatigue strength determination is based on the actually occurred loading events relevant to fatigue, the reducing effect of the environment, especially if the environmental influence can no more be considered to be moderate, cannot basically be neglected for the purpose of damage prevention.

Sub-clause 7.8.3 (2) contains respective measures to consider the environmental influence. Evaluations in due consideration of the factors of influence to be used according to the existing procedures applicable to the conditions (material, temperature, oxygen concentration, strain rate) prevailing in German nuclear power plants (PWR and BWR) proved that he application of attention thresholds is justified. A level of attention value D = 0.4 was established for both ferritic and austenitic materials. Where the measures necessary according to 7.8.3 (2) are determined by means of fatigue evaluations established on the basis of the design fatigue curve in KTA safety standard 3211.2 (1992-96), a value of D = 0.2 instead of D = 0.4 is considered to be justified for austenitic materials.

The calculation to take account of the environmental influence shall be made on the basis of the design curves under air (see e.g. [2], [16]).

Alternatively, the environmental influence on fatigue may be taken into account by detailed proofs in consideration of the temperature, oxygen content and strain rate, e.g. by application of the aforementioned procedures or experimental verifications.

The aforementioned procedure is principally applicable to take account of the environmental influence. The tests, however, indicate that the F_{en} factors to be derived in most cases have a conservative character. Laboratory tests show that, when considering the interaction between surface and environmental effects as well as realistic loading signals, considerable reserves can be proved experimentally [24] to [28]. Therefore, a quantification of conservativities is possible by experimental studies. This makes possible the partial covering of this environmental influence by the existing design curves and the derivation of allowable F_{en} factors "$F_{en,allowable}$" (see also the procedure described in [28] the implementation of which into the French RCC M-Code is under discussion). Investigations made by MPA- Stuttgart show same approaches [9].

Due to the conservative design, the operational measures taken and preventive maintenance operational experience in Germany shows that the environment has no significant influence on the component service life.

For the purpose of a conservative procedure within the establishment of safety standards the definition of attention thresholds takes the fact into account that on the basis of laboratory tests, the environmental influence on the fatigue strength is evidenced. Should this state of knowledge develop further in the future, then decision shall be made on account of this new basis on the level of attention thresholds in KTA 3211.2 or, if required, their omission be decided upon. The definition of attention thresholds reflects the present state of knowledge of environmental influences and represents a pragmatic procedure transferrable into operational practice, which has a unique status in the international set of technical rules and standards.

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<u>To Section 7.11 “Stress, strain and fatigue analyses for flange joints”</u>

In Section 7.1.1 which up to now has only dealt with stress and fatigue analyses of bolts, the performance of stress, strain and fatigue analyses for flanged joints in correspondence with KTA 3201.2 (1996-06) was determined anew. This included the adaptation to the actualised requirements of Sections A2.9 to A2.11 which also cover the stress analyses for flanges and bolts and always require verified data to be supplied by the gasket manufacturer.

<u>To Section 7.12 “Avoidance of thermal stress ratcheting for components of test group A1”</u>

(1) Section 7.12 was taken over by KTA 3201.2 (1996-06) in which case its use was limited to test group A1 components and the equations for determining the plastic strain increments $\Delta\varepsilon$ in the case of the simplified verification were changed, on a case-by-case distinction made, such that no negative strain portions will occur anymore. These equations now are clearly formulated for the user.

According to this, conservative estimation procedures are prescribed and the formerly only general requirement to verify that failure due to thermal stress ratcheting is to be avoided is specified more precisely. The estimation procedures are based on the procedures given by the ASME Code (NB 3228 and Code case N 47). Their use is limited to test group A1 as the required detailed verifications are available for this test group.

(2) A possible alternative for both the fatigue analysis and the verification of occurrence of ratcheting is the application of the simplified theory of plastic zones (Zarka’ shakedown method) (see also e.g. H. HÜBEL: Vereinfachte Fließzonentheorie, Bauingenieur, Vol. 73, 1989, No. 11, pp. 492-502).

<u>To Section 8.1 “Component-specific analysis of the mechanical behaviour; General”</u>

(1) In sub-clause (1) it was pointed out that all component-specific analyses and strength calculations are recognised current calculation procedures and that, if several procedures are indicated, all indicated procedures are considered to be equivalent and thus are all permitted.

(2) Sub-clause (4) was supplemented to require that the influences of welds that were not dressed or dressed from one side only on the fatigue strength are to be considered where fatigue analyses are performed.

<u>To Section 8.2 “ Vessels”</u>

(1) As dimensioning of nozzles subject to internal pressure need not be compulsorily performed to the equations of Annex A 2.8, the formulation in clauses 8.2.1.2 and 8.2.1.3 was put more precisely.

(2) Section 8.2.5 was adapted to the current AD rules. As the dimensioning of bellows expansion joints is to be made in conformance with the AD rules and all dimensioning requirements were taken over from these AD rules into KTA 3211.2, the former reference to Section 6.1 contained in clause 8.2.5.3.1 could be deleted without replacement.

The changes in equations 8.2-6, 8.2-7 and 8.2-12 were made due to differing dimensional units contained in KTA 3211.2 and AD 2000-Merkblatt B 13.

(3) In Section 8.2.6 “Heat exchanger tubesheets” sub-clause 8.2.6 (4) was added to require that, where the tube-side pressure is higher than double the shell-side pressure ($p_2 > 2 \cdot p_1$), it shall be verified that the shell is capable of withstanding the resulting axial force (adaptation to the current edition of AD 2000-Merkblatt B 5).

<u>To Section 8.3 “Pumps”</u>

Sub-clause 8.3.2.2.2 (3) was added to say that the determination of stress intensities can be based on the stress theory of von Mises or alternatively on the theory of Tresca.

<u>To Section 8.4 “Valve bodies”</u>

(1) At several locations, more precise formulations were taken over and the equations were adapted to the information shown in the figures.

(2) The verification of primary stresses for valve bodies taken over (from the ASME code) in Section 8.4.3 is considered necessary since the simplified component-specific analysis procedure for verifying stress and fatigue was taken over from the ASME code as a self-contained procedure. The dimensioning of valve bodies to Annex A4 is, however, based on the standard DIN EN 12516-2. The simplified stress and fatigue analysis to Section 8.4.7.2 cannot be applied without further modification on valve bodies dimensioned in accordance with Annex A4.

The requirements in sub-clause 8.4.3 (4) and in Figure 8.4-1 (sketch b) represent more stringent requirements than those of the ASME Code. They are considered necessary as the effective flange blade area cannot be credited both in the dimensioning of the valve body and the dimensioning of the flange.

(3) In the general stress analysis (Section 8.4.4) the local membrane stress portion due to internal pressure P_{lp} was taken over (equation 8.4-4) for loading levels 0, A and B in addition to the secondary stress portion from pipe reactions P_{eb} to conform with the procedure in the French RCC-M Code. The design requirements for the performance of the general stress analysis were put more precisely by inserting the respective dimensional units in Figure 8.4-5 (the diameter and wall thickness of the valve body shall be greater than those of the connected piping).

(4) Upon evaluation of the experience made with the application of KTA 3201.2 (1996-06) it was found out that the gen-

eral stress analysis may also be applied to corner valves unless the nozzles do not influence each other. According to current experience from the evaluation of finite element calculations the mutual influence of prismatic bodies is negligible. The design requirements regarding the use of Table 8.4-2 were changed accordingly.

(5) As only the primary membrane stress is evaluated at Level 0, the requirement for Level 0 was deleted in sub-clause 8.4.4 (2) and in Table 8.4-2.

To Section 8.5 "Piping systems"

(1) In Section 8.5.1 the requirements for the use of Section 8.5 were put more precisely. The aim of the stipulations in Section 8.5.1 is to ensure that, in any case, the piping components are designed to withstand internal pressure (circumferential stress). The verifications of Section 8.5 provided by equations (8.5-1) to (8.5-6) essentially deal with the longitudinal stress portion due to internal pressure in connection with additional stresses arising from thermal and mechanical loadings. The applicability of these equations depends on the satisfaction of the respective geometry conditions contained in Section 8.5. The compulsory connection with a certain dimensioning procedure for determining the wall thickness of the piping components in excess of the requirements of Section 6 is not necessary as the ratio of the stress intensification factor to the basic stress is completely covered by the stipulations of Section 8.5 and thus it is ensured that the allowable stresses are not exceeded when applying the respective equations.

(2) The requirements in sub-clause 8.5.1 (6) and in Figure 8.5-1 were extended to cover induction bends so that wall thickness increases (at intrados) and decreases (at extrados) on induction bends can be considered in the component-specific analysis of the mechanical behaviour. These requirements are prerequisite to the fact that the dimensions to KTA 3211.3, sub-clause 9.3.3.4 (5) a) (standard induction bends) are not exceeded. Where induction bends do not meet this requirement, the notch (wall thickness increase at intrados) shall be considered in compliance with the extended definition of wall thickness s_c in clauses 8.5.2.1 and 8.5.3.1.

(3) As Section 8.5 was drawn up for piping systems, it was made clear in clauses 8.5.2.3 and 8.5.3.3 that the primary stress intensity for the - originally not intended - application of equation (8.5-1) to a single straight pipe is to be modified.

(4) In clause 8.5.2.4.2 the requirement for consideration of

a) moment portions resulting from restraints due to different movement of buildings which may impair the pipe run,
b) loadings resulting from thermal stratification

were included. The procedure shown in referenced literature [9] presents a simplified and, under certain circumstances, very conservative method. Where stratified layer heights and widths are known, a detailed analysis may be performed instead of the simplified procedure to result in more exact stress intensities.

(5) In clause 8.5.2.4.6 "Determination of the ranges of temperature differences" it was made clear that time and location-dependent considerations are permitted. Figure 8.5-2 was changed to be more precise.

(6) In sub-clause 8.5.2.9.1 it was pointed out that the stress intensities are to be limited in accordance with Table 7.7-4.

(7) Clause 8.5.2.9.2 was supplemented to require that in individual cases the applicability of the stress indices for bends with notches (wall thickness increase at intrados) exceeding 15 % (induction bends), referred to the nominal wall thickness, are to be verified. It is intended to verify the applicability of the stress indices for induction bends by comparative calculations and to precise the requirements based on such calculations.

(8) The stress index C_3 in line 1 of Table 8.5-1 (straight pipe remote from welds or other discontinuities) was adapted to line 2 (butt girth welds) and fixed to a value of 0.6. This value was fixed to deviate from the ASME Code upon consultation of the relevant ASME committee, since it correctly represents the physical behaviour.

The stress index B_1 for pipe bends or curved pipes was determined to correspond with the current ASME Code.

The correction factors for the B_2 indices indicated in Table 8.5-1 for piping with $50 < d_a/s_c \leq 100$ were taken over from the ASME Code, Section III, NC-3673.2 (b-1), Note 3, and remedy the lack of B_2 indices not having been available up to now for thin-walled pipes with $d_a/s_c > 50$. This largely clarifies the question as to the usability of stress intensification factors "i" for the verification of primary stresses for piping with $50 < d_a/s_c \leq 100$. The corrections presented take credit of the deviating damage behaviour of thin-walled pipes with $d_a/s_c > 50$ compared to that of pipes with $d_a/s_c \leq 50$. Numerous experimental and theoretical investigations have proved that pipes with $d_a/s_c < 30$ reach the plastic moment in which case a large plastic plateau representing the redistribution of moments and only slight ovalisations occur. The pipes finally fail to show clear buckling. In the case of d_a/s_c ratios between 30 and 70 considerable ovalisations and only little plastification will occur. In the case of d_a/s_c ratios > 70 the pipes fail to show wrinkling prior to reaching yield strength.

(9) Clause 8.5.2.10 "Flexibility factors for system analysis" was replaced by Section 8.4.9 "Flexibility factors and stress intensification factors" from KTA 3201.2 (1996-06) to cover the following actualisation:

a) To avoid large differences in K factors occurring in some cases (in the transitional areas of the equations for determining the K factors) when applying the former rules, it was determined, upon evaluation of extensive finite element analyses to determine the K factors by linear interpolation for these transitional areas.
b) The primary stresses shall not be less than those of the straight pipe remote from discontinuities. In the case of low stress indices C_{2m} a stress less than that for the straight pipe can be calculated by applying a factor of 0.67. Therefore, equations 8.5-82 and 8.5-83 were supplemented so that at least the value of $1.0 \cdot M_{il}$ is to be used.

(10) In Section 8.5.3 the primary stress verifications based on stress intensification factors "i" for piping in test groups A2 and A3 have been omitted (equations 8.5-90 and 8.5-92 in the edition 1992-06 of this KTA Safety Standard). The verification of primary stresses can be completely made on the basis of stress indices. Primary stress verifications based on stress intensification factors "i" are considered not to be correctly applicable for thin-walled piping.

To Section A 2.5 "Dished heads (domed ends)"

Sub-clause A 2.5.2.4 (3) along with Figure A 2.5-6 were deleted as the adopted verification procedure to AD-Merkblatt B3 has proved to be unsuitable and within the range of application of KTA 3201.2 using $S_{0n}/d_a \geq 0.001$ no elastic instability need be expected in the knuckle subject to internal pressure (cf. H. Hey: Questions of knuckle instability, TÜ 1988, No. 12, pp. 408-413). Meanwhile, the required verification in AD Specification Sheet B3 was removed accordingly.

To Section A 2.9 "Bolted Joints" and A 2.10 "Flanges"

(1) Sections A 2.9 and A 2.10 were supplemented to the current state of knowledge to cover

a) requirements for the design of metal-to-metal type flanged joints,

b) flow diagrams showing the principle procedural steps for the verification of metal-to-metal contact and floating type flanged joints.

These procedures basically are not new, but include the current practice in consideration of DIN EN standards effective to date.

(2) In Sections A 2.9.3 and A 2.10.2 “General” the required procedural steps for proof of strength are laid down to correspond to the newly included flow diagrams.

The requirement of sub-clause A 2.9.3 (3) to base the various steps on the bolt load serves to ensure that the verification procedure is performed as required. In the case of metal-to-metal contact-type flanged joints the stipulations consider that at a number of bolts greater than or equal to 8 the flange shows a more uniform deformation behaviour.

(3) Upon evaluation of the VDI Report 1903 “Design, Dimensioning, Application” (VDI Verlag Dresden 2005) Section A 2.9.3 was changed to permit the alternative application of the proof of strength and the equations in Section A 2.9.4.5.2 were adapted accordingly so that no contradictions exists to the current requirements of VDI 2230. As a consequence of the adaptation of the procedure of VDI 2230 the calculation of the total engagement length in consideration of the thread chamfer could be omitted (clause A 2.9.4.3.5 of the 1992-06 edition of this KTA Safety Standard).

(4) In the equations A 2.9-1, A 2.9-5, A 2.9-10, A 2.9-11, and A 2.9-16 as well as A 2.10-1 the safety factor S_D was removed or added such that it takes credit of the actually pertinent bolt load (clarification, no change of content).

(5) In equation A 2.9-9 the request for consideration of the maximum force obtained from pipe bending moment has been omitted, as for the transferability of friction forces for compensation of torsional moments the friction effect on the gasket is governing and thus the gasket diameter d_D becomes effective.

(6) Sub-clause A 2.9.4.1 d) “Pre-stressing of bolts” was omitted as essential parts of the content were taken over as principal requirements in clause A 2.9.3.1.

(7) In the equations A 2.9-23, A 2.9-24 and A 2.9-25 used for the determination of bolt loads for metal-to-metal contact-type flanged joints in order to maintain full metal-to-metal contact of flange blades at operating condition it is laid down that the friction forces are transmitted over the metal contact surface between gasket and flange edges in which case a conservative assumption is made of linear increase in friction force obtained from flange rotation and extending from the gasket to the flange edge.

(8) Equation A 2.9-27 used for determining the bolt load for metal-to-metal contact-type flanged joints in order to obtain full metal-to-metal contact of flange blades at gasket seating condition was supplemented to include that friction forces are also considered to transfer torsional moments and transverse forces.

(9) A new requirement was added to sub-clause A 2.9.4.5.1 (3) to state that for the determination of the required thread engagement length the actual bold load may be used instead of the bold load referring to the tensile strength. This stipulation may e.g. be valid in case of retrofitting measures if bolts with a greater thread diameter are used.

(10) For the shear strength a uniform value of 0.6 shall be taken. In equations A 2.9-36, A 2.9-37 and A 2.9-39 the value of 0.55 used up to now was changed to 0.6 and thus adapted to equations A 2.9-34 and A 2.9-35.

(11) In Figures A 2.10-3, A 2.10-5 and A 2.10-6 the application of bolt load was represented to correspond to the conventional design rules. Due to the changed application of bolt load a conservative stipulation was made to correspond to the design requirements of clause 5.2.4.

(12) New equation A 2.10-46 was included for the dimensioning of metal-to-metal contact-type flanged joints. This equation serves to determine the allowable flange rotation with which adequate tightness of the flanged joint is still obtained. The spring-back gap at the gasket $\Delta s_{1,2}$ shall govern which shall be taken from standards or the manufacturer’s data in dependence of the type of gasket. Corresponding data shall be available in accordance with Form A 2.11-2.

The proofs shall be made for the bolting-up condition, for normal and anomalous operation and for the test condition.

(13) The proofs of tightness and strength for metal-to-metal contact-type flanged joints were supplemented on the basis of the proofs established for floating-type flanged joints. This new algorithm is essentially based on the calculation model on which the verification procedure for floating-type flanged joints is based. Here, the flanges were idealised as twisting bodies or linear torsion springs and the bolts as longitudinal springs. In the case of metal-to-metal contact-type flanged joints the twisting point (inversion centre) of the flanges bolted together will shift, during assembly upon reaching full metal-to-metal contact between flange blades, towards the bearing surfaces of the two flange blades. The load-dependent flange rotations thus obtained will cause a gap increase in the gasket area which has to be compensated - in case of an idealised linear-elastic gasket curve. The contact force on the bearing surface which is effective in addition to the gasket load reaction is an extension of the rules compared to the mechanical behaviour of the complete flanged connection of floating-type joints with the extension being considered in the deduction of the algorithm introduced. On the basis of this algorithm a verification by calculation of the strength and deformation conditions can be made analogously to the verifications used for floating-type flanged joints. The opening of the gap due to the twisting flange blades may be determined for the bolting-up and the operating condition and be evaluated on the basis of the gasket data sheet.

(14) In clause A 2.10.6.2.2.2 the delimitation of DIN 2505 (draft, edition 1991) between flanges with tapered hub and flanges without tapered hub was taken over.

(15) Table A 2.10-1 was changed to contain a separate column for the bolting-up condition and to state allowable stresses only for loading level P at test condition. The factor Φ was simultaneously adapted to correspond to DIN EN 1591 from which it was taken. In addition, the bolt loads on which the various verification steps are to be based were put more precisely.

(16) Further literature on stress and distortion ratios of metal-to-metal contact flanged joints can be found in [1] to [3].

[1] G. Müller: Überprüfung der Kraft- und Verformungsverhältnisse bei Flanschverbindungen mit Dichtungen im Kraftnebenschluss, Dichtungstechnik, Edition 01/2011, Vulkan-Verlag Essen

[2] G. Müller: Vereinfachtes rechnerisches Verfahren zur Überprüfung der Kraft- und Verformungsverhältnisse bei Flanschverbindungen mit Dichtungen im Kraftnebenschluss, Sonderdruck, March 2011, TÜV NORD EnSys Hannover GmbH & Co. KG

[3] Forschungsbericht „Experimentelle Ermittlung der zulässigen Belastungen von Rohrleitungsflanschverbindungen DN100 mit der Dichtung im Kraftnebenschluss (KNS)“, SA-AT 19/08, December 2010, Materialprüfungsanstalt Universität Stuttgart

To Section A 2.11 “Gaskets”

(1) Section A 2.11 “Gaskets” was fundamentally revised and adapted to the current state of knowledge. It was considered

purposeful to change over to the gasket factors determined to DIN 28090-1 (1995-09) “Static gaskets for flanged joints - Part 1: Gasket factors and test procedures”. It is assumed that in the future a list of characteristic values to correspond to the respective forms specified will be established.

(2) The definition of the characteristic values for floating-type flanged joints was taken over from DIN 28090-1 (1995-09).

(3) Former Table A 2.11-1 was deleted and replaced by a sample form summarising characteristic values (Forms A 2.11-1 and A 2.11-2).

(4) Further explanations to the use of Section A 2.11 can e.g. be found in literature referenced hereafter:

a) H. Kockelmann, J. Bartonicek, E. Roos: Characteristics of gaskets for bolted flanged connections - present state of the art, The 1998 ASME/JSME Joint Pressure Vessel and Piping Conference, San Diego, California; July 26-30, 1998, PVP-Vol. 367, pp. 1/10
b) H. Kockelmann: Leckageraten von Dichtungen für Flanschverbindungen, Rohrleitungstechnik, 7. Ausgabe (1998), S. 194/216, Vulkan-Verlag Essen
   H. Kockelmann: Leakage rates of gaskets for flanged joints, Rohrleitungstechnik, 7th edition, pp. 194-216, Vulkan-Verlag (in German)
c) H. Kockelmann, R. Hahn, J. Bartonicek, H. Golub, M. Trobitz, F. Schöckle: Kennwerte von Dichtungen für Flanschverbindungen, 25. MPA-Seminar, 7. und 8. Oktober 1999, Stuttgart
   H. Kockelmann, R. Hahn, J. Bartonicek, H. Golub, M. Trobitz, F. Schöckle: Characterisitc values of gaskets for flanged joints, 25th MPA-Seminar, Stuttgart, 7th and 8th October 1999 (in German)
d) H. Kockelmann, J. Bartonicek, R. Hahn, M. Schaaf: Design of Bolted Flanged Connections of Metal-to-Metal Contact Type, ASME PVP Conference 2000, July 23-27, 2000, Seattle, USA
e) H. Kockelmann, Y. Birembaut: Asbestos Free Materials for Gaskets for Bolted Flanged Connections, Synthesis Report of the Brite Euram Project BE 5191 Focusing on Gasket Factors and Associated Gasket Testing Procedures, 4th International Symposium on Fluid Sealing, 17-19 Septembre 1996, Mandelieu/France

<u>To Section A 4 “Valves”</u>

Equation A 4.1-23 was changed to conform to the respective equation in DIN EN 12516-2.

<u>To Annex B “Requirements as to the primary stress analysis in case of numerical reassessments”</u>

(1) In practice the KTA safety standards 3201.2 and 3211.2 valid for new constructions are also applied to already existing components for which verifications by calculation have been made to prove that damage prevention required to correspond to the state of science and technology is adhered to. The necessity of performing numerical reassessments is given e.g. if:

a) the existing component is within the range where replacement measures are taken and the admissibility of changed attachment loads, e.g. due to the changed dead weight of the replaced component, has to be verified,
b) grinding work has been done on the existing component (e.g. to establish the testability for in-service inspections) which leads to a less-than-normal wall thickness,
c) new knowledge is gained on the loads to be verified or on the loading capability of the component to be verified.

(2) Adaptations required compared to the state of licencing, which result from updated requirements set by the current state of science and technology with regard to “analysis of mechanical behaviour” verifications can be relatively simply realised in practice by limited plant changes (e.g. optimisation of the supporting system or adaptions of the mode of operation or water chemistry). However, adaptations due to updated design and dimensioning requirements in most cases are only possible by completely replacing the respective components.

(3) With the dimensioning to Section 6 (requirements for loading level 0) the minimum requirements for component dimensions are determined. Additional dimensional requirements may be possible due to the effects of loading levels A, B, C, D or P. By means of loading level 0 it is possible, within the erection of a plant, to stagger the totality of strength verifications over a certain period of time. A first step will be to determine component dimensioning with the level 0 and P data, which generally is prerequisite for the release of manufacturing. A second step will be the stress analysis based on level A, B, C and D data which should be available prior to pressure testing. To simplify and minimise the calculation expenditure, as a rule only longitudinal stresses are credited for in levels A, B, C and D. Circumferential stresses shall have already been limited in level 0 calculations.

(4) Given the fact that in practice the dimensioning step is taken with conservative assumptions during the erection phase which, inter alia, are intended to cover planning uncertainties, the dimensioning requirements to Section 6 need not necessarily be met in case of reassessment of primary stresses to ensure damage prevention according to the state of science and technology. Therefore, other steps are permitted to verify sufficient dimensioning in case of numerical reassessments where the planning uncertainties of the first plant design are no more given. This means that the primary stress analysis may be made to follow the mode described in Annex B in due consideration of the state of knowledge at the point in time of reassessment such that sufficient dimensioning required by the state of science and technology can be verified. Here, it shall be taken into account that the design pressure and temperature have to cover the loading level A conditions, i.e. that the loads of level 0 and A may be identical in limited cases. This condition is identical to the requirements contained in ASME BPVC 2010, Section III, Division 1, Subsection NC, Article NC-3112 and Subsection NCA, Article 2142.1. This means that no additional safety is quantifiable when applying level 0 compared to the application of level A.

(5) To make corresponding evaluation criteria available, new qualitative and methodical requirements as to the primary stress analysis were taken over in Annex B for numerical reassessments.