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Cryogenic vessels — Static vacuum-insulated vessels —

Part 1:
Design, fabrication, inspection and tests

Récipients cryogéniques — Récipients isolés sous vide statiques —

Partie 1: Exigences de conception de fabrication, d'inspection, et d'essais

Reference number
ISO 21009-1:2008(E)

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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

ISO 21009-1 was prepared by Technical Committee ISO/TC 220, *Cryogenic vessels*.

ISO 21009 consists of the following parts, under the general title *Cryogenic vessels — Static vacuum-insulated vessels*:

— *Part 1: Design, fabrication, inspection and tests*

— *Part 2: Operational requirements:*

This corrected version incorporates the following corrections:

— a single safety factor is given for the knuckle-region;

— the straight flange length requirement is expressed in terms of s;

— the formulae specifying cones which come under the field of application have been corrected;

— the cone angle is specified for internal pressure calculation;

— the formulae used for internal pressure calculation have been corrected;

— the formulae used for external pressure calculation have been corrected;

— the symbols used to denote wall thickness in Figure 7 have been changed;

— the Greek symbols used in Figures 10.1 to 10.8 (with the exception of φ) have been replaced by Latin symbols;

— the relationship to the pressure vessel code has specified with regard to calculations made for austenitic stainless steels;

— the cross-references in Annex G have been corrected;

— the formula for calculating moment of inertia, I, in relation to stiffening rings has been corrected;

— the formulae for calculating limits of reinforcement normal to the vessel wall by increased nozzle thickness have been corrected.

Cryogenic vessels — Static vacuum-insulated vessels —

Part 1: **Design, fabrication, inspection and tests**

1 Scope

This part of ISO 21009 specifies requirements for the design, fabrication, inspection and testing of static vacuum-insulated cryogenic vessels designed for a maximum allowable pressure of more than 0,5 bar.

This part of ISO 21009 applies to static vacuum-insulated cryogenic vessels for fluids as specified in 3.4 and does not apply to vessels designed for toxic fluids.

For static vacuum-insulated cryogenic vessels designed for a maximum allowable pressure of not more than 0,5 bar this International Standard may be used as a guide.

2 Normative references

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

ISO 4126-2, *Safety devices for protection against excessive pressure — Part 2: Bursting disc safety devices*

ISO 4136, *Destructive tests on welds in metallic materials — Transverse tensile test*

ISO 9016, *Destructive tests on welds in metallic materials — Impact tests — Test specimen location, notch orientation and examination*

ISO 9606-1, *Approval testing of welders — Fusion welding — Part 1: Steels*

ISO 9606-2, *Qualification test of welders — Fusion welding — Part 2: Aluminium and aluminium alloys*

ISO 9712, *Non-destructive testing — Qualification and certification of personnel*

ISO 10474, *Steel and steel products — Inspection documents*

ISO 14732, *Welding personnel — Approval testing of welding operators for fusion welding and of resistance weld setters for fully mechanized and automatic welding of metallic materials*

ISO 15607, *Specification and qualification of welding procedures for metallic materials — General rules*

ISO 15613, *Specification and qualification of welding procedures for metallic materials — Qualification based on pre-production welding test*

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

ISO 15614-2, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 2: Arc welding of aluminium and its alloys*

ISO 17636, *Non-destructive testing of welds — Radiographic testing of fusion-welded joints*

ISO 21010, *Cryogenic vessels — Gas/materials compatibility*

ISO 21013-3, *Cryogenic vessels — Pressure-relief accessories for cryogenic service — Part 3: Sizing and capacity determination*

ISO 21028-1 *Cryogenic vessels — Toughness requirements for materials at cryogenic temperature — Part 1: Temperatures below -80 °C*

ISO 21028-2 *Cryogenic vessels — Toughness requirements for materials at cryogenic temperature — Part 2: Temperatures between -80 °C and -20 °C*

ISO 23208, *Cryogenic vessels — Cleanliness for cryogenic service*

ISO 21009-2, *Cryogenic vessels — Static vacuum insulated vessels — Part 2: Operational requirements*

ISO 21011, *Cryogenic vessels — Valves for cryogenic service*

EN 10028-7, *Flat products made of steels for pressure purposes — Part 7: Stainless steels*

EN 13068-3, *Non-destructive testing – Radioscopic testing — Part 3: General principles of radioscopic testing of metallic materials by X- and gamma rays*

ASME Boiler and Pressure Vessel Code, *Section V: Nondestructive Examination*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

3.1
accessories
service equipment which has a safety related function with respect to pressure containment and/or control

EXAMPLE Accessories include protective or limiting devices, controlling and monitoring devices, valves and indicators.

3.2
automatic welding
welding in which the parameters are automatically controlled

NOTE Some of these parameters may be adjusted to a limited extent, either manually or automatically, during welding to maintain the specified welding conditions.

3.3
bursting disc device
non-reclosing pressure relief device ruptured by differential pressure

NOTE The bursting disc device is the complete assembly of installed components including, where appropriate, the bursting disc holder.

3.4
cryogenic fluid
refrigerated liquefied gas
gas which is partially liquid because of its low temperature

NOTE This includes totally evaporated liquids and supercritical fluids.

EXAMPLE In ISO 21009, the (refrigerated, but) non-toxic gases, and mixtures of them, shown in Table 1, are referred to as cryogenic fluids.

Table 1 — Refrigerated but non toxic gases

classification code	Identification number, name and description	
3° A	Asphyxiant gases	
	1913	Neon, refrigerated liquid
	1951	Argon, refrigerated liquid
	1963	Helium, refrigerated liquid
	1970	Krypton, refrigerated liquid
	1977	Nitrogen, refrigerated liquid
	2187	Carbon dioxide, refrigerated liquid
	2591	Xenon, refrigerated liquid
	3136	Trifluoromethane, refrigerated liquid
	3158	Gas, refrigerated liquid, not otherwise specified (NOS)
3° O	Oxidizing gases	
	1003	Air, refrigerated liquid
	1073	Oxygen, refrigerated liquid
	2201	Nitrous oxide, refrigerated liquid, oxidizing
	3311	Gas, refrigerated liquid, oxidizing, NOS
3° F	Flammable gases	
	1038	Ethylene, refrigerated liquid
	1961	Ethane, refrigerated liquid
	1966	Hydrogen, refrigerated liquid
	1972	Methane, refrigerated liquid or natural gas, refrigerated liquid, with high methane content
	3138	Ethylene, acetylene and propylene mixture, refrigerated liquid, containing at least 71,5 % ethylene with not more than 22,5 % acetylene and not more than 6 % propylene
	3312	Gas, refrigerated liquid, flammable, NOS
The flammable gases and mixtures of them may be mixed with: helium, neon, nitrogen, argon, carbon dioxide. Oxidizing and flammable gases may not be mixed. NOTE The classification code, identification number, name and description are according to UN codes.		

3.5
documentation
technical documents delivered by the manufacturer to the owner consisting of:

— all certificates establishing the conformity with this part of ISO 21009 (e.g. material, pressure test, cleanliness, safety devices);

— a short description of the vessel (including characteristic data, etc.);

— a list of fluids and their net mass for which the cryogenic vessel is designed;

— an operating manual (for the user) that contains

    — a short description of the vessel (including characteristic data, etc.),

    — a statement that the vessel is in conformity with this part of ISO 21009, and

    — the instructions for normal operation.

3.6
gross volume of the inner vessel
internal volume of the inner vessel , excluding nozzles, pipes etc. determined at minimum design temperature and atmospheric pressure

3.7
handling loads
loads exerted on the static cryogenic vessel in all normal transport operations including loading, unloading, pressure loading during transportation, installation, etc.

3.8
inner vessel
pressure vessel intended to contain the **cryogenic fluid** to be stored

3.9
manufacturer of the static cryogenic vessel
company that carries out the final assembly, including the final acceptance test, of the static cryogenic vessel

3.10
maximum allowable pressure
maximum pressure permissible at the top of the vessel in its normal operating position

3.11
net volume of the inner vessel
volume of the inner vessel, below the inlet to the relief devices, excluding nozzles, pipes etc., determined at minimum design temperature and atmospheric pressure

3.12
normal operation
intended operation of the vessel either up to the **maximum allowable pressure** or subjected to **handling loads**

3.13
outer jacket
gas-tight enclosure which contains the **inner vessel** and enables the vacuum to be established

3.14
piping system
tubes, pipes and associated components which can come in contact with **cryogenic fluids** including valves, fittings, pressure relief devices, and their supports

3.15
pressure
gauge pressure
pressure relative to atmospheric pressure

3.16
relief plate
plate retained by atmospheric pressure which allows relief of excess internal pressure, generally from the vacuum jacket

3.17
relief plug
plug retained by atmospheric pressure which allows relief of excess internal pressure, generally from the vacuum jacket

3.18
service equipment
measuring instruments, filling, discharge, venting, safety, pressurizing, cooling and thermal insulation devices

3.19
static cryogenic vessels
thermally insulated vessel intended for use with one or more **cryogenic fluids** in a stationary condition

NOTE Static cryogenic vessels consist of inner vessel(s), an outer jacket and the piping system.

3.20
thermal insulation
vacuum inter-space between the inner vessel and the outer jacket

NOTE The space may or may not be filled with material to reduce the heat transfer between the inner vessel and the outer jacket.

3.21
year built
date of the final acceptance test of the final assembled cryogenic vessel at the manufacturer

4 Symbols

For the purposes of this document, the following symbols apply:

c	allowances for corrosion	mm
d_i	diameter of opening	mm
d_a	outside diameter of tube or nozzle	mm
f	narrow side of rectangular or elliptical plate	mm
l_b	buckling length	mm
n	number	—
p	design pressure as defined by 10.2.3.2.1 and 10.3.3.2	bar
p_e	allowable external pressure limited by elastic buckling	bar
p_k	strengthening pressure	bar

p_p	allowable external pressure limited by plastic deformation	bar
p_s	maximum allowable gauge pressure	bar
p_T	test pressure [see 10.2.3.2.3]	bar
r	radius e.g. inside knuckle radius of dished end and cones	mm
s	minimum wall thickness	mm
s_e	actual wall thickness	mm
v	factor indicative of the utilisation of the permissible design stress in joints or factor allowing for weakenings	—
x	(decay-length zone) distance over which governing stress is assumed to act	mm
A	cross sectional area of reinforcing element	mm^2
A_s	elongation at fracture	%
C_β	design factors	—
D	shell diameter	mm
D_a	outside diameter e.g. of a cylindrical shell	mm
D_i	internal diameter e.g. of a cylindrical shell	mm
E	Young's modulus	N/mm^2
H	Safety coefficient for pressure test	—
I	moment of inertia of reinforcing element	mm^4
K	material property used for design (see 10.3.2.3.1)	N/mm^2
Kt	material property at t °C used for design (e.g. K_{20} for material property at 20 °C) (see 10.3.2.3.2)	N/mm^2
R	radius of curvature e.g. inside crown radius of dished end	mm
S	safety factor at design pressure	—
S_k	safety factor against elastic buckling at design pressure	—
S_p	safety factor against plastic deformation at design pressure	—
S_T	safety factor against plastic deformation at proof test pressure	—
Z	auxiliary value	—
ν	Poisson ratio	—
u	out-of-roundness	—
σ_k	design stress value	N/mm^2

5 General requirements

5.1 The static cryogenic vessel shall safely withstand the mechanical and thermal loads and the chemical effects encountered during pressure test and normal operation. These requirements are deemed to be satisfied if Clauses 6 to 11 are fulfilled. The vessel shall be tested in accordance with Clause 12, marked in accordance with Clause 13, and operated in accordance with ISO 21009-2.

5.2 Static cryogenic vessels shall be equipped with valves, pressure relief devices, etc. configured and installed in such a way that the vessel can be operated safely. The number of openings in the inner vessel for this equipment shall be kept to a minimum.

5.3 The static cryogenic vessel shall be clean for the intended service in accordance with ISO 23208.

5.4 The manufacturer shall retain the documents referred to in 3.5, and all supporting documentation (including that from his subcontractors if any), for a period required by regulation(s) (e.g. product liability). In addition the manufacturer shall retain all supporting and background documentation (including that from his subcontractors if any) which establishes that the vessel conforms to this part of ISO 21009.

6 Mechanical loads

6.1 General

The static cryogenic vessel shall resist the mechanical loads mentioned in Clause 6 without such deformation which could affect safety and which could lead to leakage.

The mechanical loads to be considered are:

- loads exerted during the pressure test as specified in 6.2;
- loads imposed during installation and removal of the vessel;
- dynamic loads during transport of the vessel.

The following loads shall be considered to act in combination where relevant:

- a pressure equal to the maximum allowable pressure in the inner vessel and pipework;
- the pressure exerted by the liquid when filled to capacity;
- loads produced by the thermal movement of the inner vessel, outer jacket and inter-space piping;
- full vacuum in the outer jacket;
- a pressure in the outer jacket equal to the set pressure of the relief device protecting the outer jacket;
- wind loads and other site conditions (e.g. seismic loads, thermal loads) to the vessel when filled to capacity.

6.2 Load during the pressure test

The load exerted during the pressure test used for calculation shall be:

$$p_\mathrm{T} \geqslant H\,(p_\mathrm{s} + 1)$$

where

p_T is the test pressure (in bar);

H is 1,43 in Europe and 1,3 in North America and for other parts of the world, a value consistent with the applicable pressure vessel code;

p_S is the maximum allowable gauge pressure (in bar);

$+ 1$ is the allowance for external vacuum (in bar).

7 Chemical effects

Due to operating temperatures and the materials of construction, the possibility of chemical action on the inner surfaces in contact with the cryogenic fluids can be discounted.

Due to the fact that the inner vessel is inside an evacuated outer jacket, neither external corrosion of the inner vessel, nor corrosion on the inner surfaces of the outer jacket will occur. Therefore inspection openings are not required in the inner vessel or the outer jacket.

Corrosion allowance is also not required on surfaces in contact with the operating fluid or exposed to the vacuum inter-space between the inner vessel and the outer jacket.

The material and the protection for the surfaces exposed to the atmosphere shall be suitable for intended use (e.g. resistant to industrial and marine atmospheres).

8 Thermal conditions

The following thermal conditions shall be taken into account:

a) for the inner vessel and its associated equipment the full range of temperatures expected;

b) for the outer jacket and equipment thereof [other equipment than covered by a)]:

- a minimum working temperature of –20 °C, unless otherwise specified and marked in accordance with Clause 13;

- a maximum working temperature of 50 °C.

9 Material

The materials used to manufacture the inner vessels and associated equipment shall meet the requirements defined in 9.1 to 9.2.

9.1 Selection of materials

9.1.1 Materials which are or might be in contact with cryogenic fluids shall be in accordance with ISO 21010.

9.1.2 Materials used at low temperatures shall follow the requirements of the relevant ISO 21028; for non-metallic materials low temperature suitability shall be validated by an experimental method, taking into account operating temperatures.

9.1.3 The base materials, listed in Annex K, subject to meeting the extra requirements given in the main body of this part of ISO 21009, are suitable for and may be employed in the manufacture of the cryogenic vessels conforming to ISO 21009-1.

9.2 Inspection certificate

9.2.1 The head and shell material shall be according to ISO 21028-1 or ISO 21028-2 and shall be declared by an inspection certificate 3.1.B in accordance with ISO 10474.

9.2.2 The material manufactured to a recognised international standard shall meet the testing requirements according to ISO 21028-1 or ISO 21028-2 and be declared by an inspection certificate 3.1.B in accordance with ISO 10474.

9.3 Materials for outer jackets and service equipment

The outer jacket and the service equipment not subjected to cryogenic temperature shall be manufactured from material suitable for the intended service.

10 Design

10.1 Design options

10.1.1 General

The design shall be carried out in accordance with one of the options given in 10.1.2, 10.1.3 or 10.1.4.

In the case of 9 % Ni steel, the additional requirements of Annex B shall be satisfied.

For metallic materials used at cryogenic temperatures the requirements of ISO 21028-1 and ISO 21028-2 shall be satisfied.

When further use of cold properties is allowed the requirements of Annex E shall be satisfied.

10.1.2 Design by calculation

Calculation of all pressure and load bearing components shall be carried out. The pressure part thicknesses of the inner vessel and outer jacket shall not be less than required by 10.3. Additional calculations may be required to ensure the design is satisfactory for the operating conditions including an allowance for external loads (e.g. seismic).

10.1.3 Design by calculation when adopting pressure strengthening (if allowed)

The pressure retaining capability of inner vessels manufactured from austenitic stainless steel, strengthened by pressure, shall be calculated in accordance with Annex C. In some cases, designs adopting pressure strengthening might not be allowed by the applicable authorities where the vessel is to be operated.

10.1.4 Design of components by calculation supplemented with experimental methods

Where it is not possible to design non-inner-vessel components by calculation alone, planned and controlled experimental means may be used, provided that the results confirm the safety factors required in 10.3. An example would be the application of strain gauges to assess stress levels.

10.2 Common design requirements

10.2.1 General

The requirements of 10.2.2 to 10.2.8 are applicable to all vessels irrespective of the design option used.

In the event of an increase in any one of the following parameters, the initial design process shall be repeated:

- maximum allowable pressure;
- specific mass (density) of the densest gas for which the vessel is designed;
- maximum tare weight of the inner vessel;
- nominal length and/or diameter of the inner shell;

or, in the event of any change relative to

- the type of material or grade (e.g. stainless steel to aluminium or change of stainless steel grade),
- the fundamental shape,
- the decrease in the minimum mechanical properties of the material being used, or
- the modification of the design of an assembly method concerning any part under stress, particularly as far as the support systems between the inner vessel and the outer jacket or the inner vessel itself or the protective frame, if any, are concerned.

10.2.2 Design specification and documentation

To enable the design to be prepared, the following information shall be available:

- maximum allowable pressure;
- fluids intended to be contained;
- gross volume of the inner vessel;
- configuration;
- location of fastening points and loads allowable on these points;
- method of handling and securing during transit and site erection;
- site conditions (ambient temperatures, seismic, etc.);
- shipping modes (road, rail, water, etc.) of the empty vessel;
- filling and emptying rates;
- range of ambient temperatures, if different from 8b);
- gross mass;
- details of fastenings.

A design document in the form of drawings with text if any shall be prepared. It shall contain the information given above plus the following where applicable:

- definition of which components are designed by calculation, by pressure strengthening, by experiment and by satisfactory in-service experience;
- drawings with dimensions and thicknesses of load bearing components;
- specification of all load bearing materials including grade, class, temper, testing etc. as relevant;

— applicable material test certificates;

— location and details of welds and other joints, welding and other joining procedures, filler, joining materials etc. as relevant;

— calculations to verify compliance with this International Standard;

— design test programme;

— non-destructive testing requirements;

— pressure test requirements;

— piping configuration including type, size and location of all valves and relief devices;

— details of lifting points and lifting procedure;

— calculations for wind and seismic loads.

10.2.3 Design loads

10.2.3.1 General

Under normal operating conditions, static vessels are not expected to see pressure variations.

If the static vessel is specifically intended for more than 4 000 pressure cycles, fatigue life shall be calculated in accordance with an internationally recognized standard.

NOTE A pressure cycle is defined as a pressure variation more than 50 % of the design pressure for austenitic stainless steels and 20 % for the other materials.

The static cryogenic vessel shall be able to safely withstand the mechanical and thermal loads encountered during normal operation, transportation and pressure test, as specified in 10.2.3.2 to 10.2.3.7.

10.2.3.2 Inner vessel

10.2.3.2.1 The following loads shall be considered to act in the combinations specified in 10.2.3.2.2:

a) pressure during operation when the vessel contains cryogenic liquid product

$$p_{\mathrm{cL}} = p_{\mathrm{s}} + p_{\mathrm{L}} + 1\,\mathrm{bar}$$

where

p_{s} is the maximum allowable gauge pressure (bar);

p_{L} is the pressure (bar) exerted by the weight of the liquid contents when the vessel is filled to capacity with either

i) boiling liquid at atmospheric pressure, or

ii) cryogenic fluid at its equilibrium triple point or melting point temperature at atmospheric pressure

[p_{L} is neglected if less than 5 % of (p_{s} + 1). If p_{L} is greater than 5 % of (p_{s} + 1), it is allowed to reduce the value by 5 % of (p_{s} + 1)];

b) pressure during operation when the vessel contains only gaseous product at 20 °C

$$p_{cG} = p_s + 1\,\text{bar}$$

NOTE 1 This equation applies only if Annex E is used.

c) reactions at the support points of the inner vessel during operation when the vessel contains cryogenic liquid product. The reactions shall be determined by the weight of the inner vessel, the weight of the maximum contents of the cryogenic liquid and vapour and seismic loadings where appropriate. The seismic loadings shall include any forces exerted on the vessel by the insulation;

d) reactions at the support points of the inner vessel during operation when the vessel contains only gaseous product at 20 °C. The reactions shall be determined by the weight of the inner vessel, its contents and seismic loadings where appropriate. The seismic loadings shall include any forces exerted on the vessel by the insulation;

NOTE 2 This condition applies only if Annex E is used.

e) load imposed by the piping due to the differential thermal movement of the inner vessel, the piping and the outer jacket, where the following cases shall be considered:

— cooldown (inner vessel warm - piping cold);

— filling and withdrawal (inner vessel cold - piping cold);

— storage (inner vessel cold - piping warm);

f) load imposed on the inner vessel at its support points when cooling from ambient to operating temperature;

g) loads imposed during transit and site erection;

NOTE 3 The static cryogenic vessel is not intended to be transported filled. It may be transported empty or containing marginal residues of cryogenic fluid from one location to another.

h) load imposed by pressure in annular space equal to the set pressure of the outer jacket relief device and atmospheric pressure in inner vessel.

10.2.3.2.2 The vessel shall be capable of withstanding the following combinations of loadings from 10.2.3.2.1. The design pressure, p, is equal to pressure specified therein, in each combination 1, 2 and 3:

1) operation at maximum allowable working pressure when vessel is filled with cryogenic liquid: 10.2.3.2.1 a) + c) + e) + f);

2) operation at maximum allowable working pressure when vessel is filled with gas at 20 °C: b) + d);

3) pressure test: see 10.2.3.2.3;

4) shipping and lifting: 10.2.3.2.1 g);

5) vessel subject to external pressure developed in the vacuum jacket: 10.2.3.2.1 h).

The inner vessel shall, in addition, be capable of holding the pressure test fluid without gross plastic deformation.

10.2.3.2.3 The design shall be evaluated for the following conditions:

pressure test: the value used for design purposes shall be the higher of:

$p_T = H\ (p_s + 1)$ or see 12.5.1 or

$p_T = 1{,}25\ (p_s + p_L + 1)\dfrac{K_{20}}{K_t}$ bar

NOTE 1 H is equal to 1,43 in Europe and to 1,3 in North America.

NOTE 2 When cold properties are used, see Annex E where K_{design} is used instead of K_t.

considered for each element of the vessel, e.g. shell, courses, head.

The 1 bar is added to allow for the external vacuum.

10.2.3.3 Outer jacket

The following loads shall be considered to act in combination where relevant:

a) an external pressure of 1 bar;

b) an internal pressure equal to the set pressure of the outer jacket pressure relief device;

c) load imposed by the supporting systems in the outer jacket taking into consideration site conditions, e.g. wind and seismic loadings;

d) load imposed by piping as defined in 10.2.3.2.1 e);

e) load imposed at the inner vessel support points in the outer jacket when the inner vessel cools from ambient to operating temperature and during operation;

f) loads imposed during transit and site erection;

g) external loads from e.g. wind, seismic or other site conditions;

h) gross mass.

10.2.3.4 Inner vessel supports

The inner vessel supports shall be designed for the load specified in 10.2.3.2.1 c) and f) to a maximum allowable stress value which is equal to 0,75 K_{20}. Additionally this maximum stress value shall not be exceeded during shipping with loads of 1,7 g down, 1 g upwards and laterally and 2 g in the direction of the travel based on an empty vessel.

10.2.3.5 Outer jacket supports

The outer jacket supports shall be suitable for the load defined in 10.2.3.3 to a maximum allowable stress value equal to 0,75 K_{20}.

10.2.3.6 Lifting points

Lifting points shall be suitable for lifting the static cryogenic vessel when empty and lifted in accordance with the specified procedure to a maximum allowable stress value equal to 0,75 K_{20}.

10.2.3.7 Piping and accessories

Piping and accessories shall be designed such that their lowest natural frequency is higher than 30 cycles per second. Piping including valves, fittings and supports shall be designed for the following loads. The following loads shall be considered to act in combination where relevant:

a) pressure during operation: not less than the set pressure of the system pressure relief devices, e.g. set pressure of the thermal relief device;

b) thermal loads defined in 10.2.3.2.1 f);

c) loads generated during pressure relief discharge;

d) a design pressure not less than the maximum allowable pressure, p_s, of the inner vessel plus any appropriate liquid head. For piping inside the vacuum jacket a further 1 bar shall be added.

10.2.4 Inspection openings

Inspection openings are not required in the inner vessel or the outer jacket, provided that the requirements of ISO/DIS 21009-2 are followed.

NOTE 1 Due to the combination of materials of construction and operating fluids, internal corrosion cannot occur.

NOTE 2 The inner vessel is inside the evacuated outer jacket and hence external corrosion of the inner vessel cannot occur.

NOTE 3 The elimination of inspection openings also assists in maintaining the integrity of the vacuum in the interspace.

10.2.5 Pressure relief

10.2.5.1 General

Relief devices for the inner vessel shall be in accordance with ISO 21013-3;

Relief devices for the outer jacket shall be in accordance with Annex I.

10.2.5.2 Inner vessel

The inner vessel shall be provided with a pressure limiting system to protect the vessel against excessive pressure. Examples of current practice are shown in Annex D. The system shall

— be designed so that it is fit for purpose,

— be independent of other functions, unless its safety function is not affected by such other functions,

— limit the vessel pressure to 110 % maximum allowable pressure in all emergency cases except fire engulfment[1)],

— fail safely,

— contain redundant features, and

— contain non-common-mode failure mechanisms (diversity).

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1) Where required, to protect the vessel against fire engulfment, a bursting disc can be used which is set at the test pressure of the vessel.

The capacity of the protection system shall be established by considering all of the probable conditions contributing towards internal excess pressure. For example:

a) normal vessel heat leak;

b) heat leak with loss of vacuum;

c) failure in the open position of the pressure build-up regulator;

d) flow capacity of any other valve in a line connecting a high pressure source to the inner vessel;

e) recycling from any possible combination of pumps;

f) flash gas, plus liquid, from maximum capacity of filling system fed into a tank which is at operating temperature;

g) external fire condition with the loss of vacuum shall be considered if required (normally not required for directly buried underground installations).

The excess pressure created by any combination of conditions a) to f) shall be limited to not more than 110 % of maximum allowable pressure by at least one re-closable device. The required capacity of this re-closable device may be calculated in accordance with ISO 21013-3.

NOTE Where, in addition, a non re-closable, fail safe device is fitted, its operating pressure should be chosen such that its ability to retain pressure is unaffected by the operation of the re-closable device at 110 % of maximum allowable pressure. The required capacity of any device provided for redundancy shall be equal to the required capacity of the primary device at vessel test pressure.

Shut off valves or equivalent may be installed upstream of pressure relief devices, provided that interlocks are fitted to ensure that the vessel has sufficient relief capacity at all times.

The relief valve system piping shall be sized such that the pressure drops during discharge are fully taken into account so that the vessel pressure is not excessive and also so that the valve does not reseat instantly, i.e. chatter.

The maximum pressure drop of the pipework to the pressure relief device should not exceed that specified in ISO 21013-3.

10.2.5.3 Outer jacket

A pressure relief device shall be fitted to the outer jacket. The device shall be set to open at a pressure which prevents collapse of the inner vessel and is not more than 0,5 bar.

The discharge area of the pressure relief device(s) should not be less than 0,34 mm^2/l capacity of the inner vessel for small vessels up to 15 000 l. However, normally the size of this device need not exceed 5 000 mm^2.

10.2.5.4 Piping

Any section of pipework containing cryogenic fluid which can be isolated shall be protected by a relief valve or other suitable relief device.

10.2.6 Valves

10.2.6.1 General

Valves shall conform to ISO 21011.

10.2.6.2 Isolating valves

To prevent any large spillage of liquid, a secondary means of isolation shall be provided for those lines emanating from below the liquid level that are

— greater than 13 mm bore and exhausting to atmosphere, or

— greater than 50 mm bore when forming part of a closed system.

The secondary means of isolation may be within the user installation and shall provide an equivalent level of protection.

The secondary means of isolation, where provided, may be achieved, for example, by the installation of a second valve, positioned so that it can be operated safely in emergency, an automatic fail-closed valve or a non-return valve or fixed or removable cap on the open end of the pipe.

10.2.7 Filling ratio

Means shall be provided to ensure that the vessel is not filled to more than 95 % of its total volume with liquid at the filling condition.

10.2.8 Electrical continuity

For all static vessels designed to store flammable fluids, means shall be provided to assure electrical continuity.

10.3 Design by calculation

10.3.1 General

When design is by calculation in accordance with 10.1.2, the dimensions of the inner vessel and outer jacket shall not be less than that determined in accordance with 10.3.

10.3.2 Inner vessel

10.3.2.1 General

The information in 10.3.2.2 to 10.3.2.6 shall be used to determine the pressure part thicknesses in conjunction with the calculation formulae of 10.3.6.

10.3.2.2 Design loads and allowable stresses

a) In accordance with 10.2.3.2.1 a), c), e), f) and 10.2.3.2.2, 1), material properties determined either in accordance with 10.3.2.3.2 or 10.3.2.3.3 shall be used if allowed by the applicable authorities where the vessel is to be operated.

b) In accordance with 10.2.3.2.1 b), d), g), h), and 10.2.3.2.2, 2), 3), 4), and 5).

Material properties determined in accordance with 10.3.2.3.2 shall be adopted.

10.3.2.3 Material property, K

10.3.2.3.1 General

The material property, K, to be used in the calculations shall be as follows:

— for austenitic stainless steel and unalloyed aluminium, 1 % proof strength;

— for all other metals the yield strength, and if not available 0,2 % proof strength.

NOTE Upper yield strength may be used.

10.3.2.3.2 K_{20}

R_e and R_m shall be the minimum guaranteed values at 20 °C taken from the material standard.

In the case of austenitic stainless steels, the specified minimum values may be exceeded by up to 15 % for carrying all loads listed in 10.2.3.2 for the design pressure, p, specified under 10.2.3.2.1 a) if the pressure vessel code does not allow it.

The 15 % higher values of K_{20} may be used provided this higher value is attested in the inspection certificate and the following conditions are met:

— the increased properties are verified by testing each cast (production lot);

— the welding procedures are suitably qualified.

Ratios of R_e/R_m exceeding 0,85 are not allowed for steels in the construction of welded tanks. In determining the ratio, R_e/R_m, the minimum specified value of R_e and R_m in the material inspection certificate shall be used.

K shall be the minimum value at 20 °C taken from the material standard (see Annex J).

10.3.2.3.3 K_t

The permissible value of K shall be determined for the material at the operating temperature corresponding to the saturation temperature, at the maximum allowable pressure of the vessel, of the contained cryogenic fluid. The value of K and E shall be determined from the material standard (see EN 10028-7 Annex F for austenitic stainless steels) or shall be guaranteed by the material manufacturer.

10.3.2.4 Safety factors, S, S_T, S_p, and S_k

Safety factors, the ratio of material property, K, over the maximum allowable stress, are a) or b):

a) internal pressure (pressure on the concave surface):

  — at vessel maximum allowable pressure

  $$S = 1{,}5$$

  — at vessel test pressure

  $$S_T = 1{,}05$$

b) external pressure (pressure on the convex surface):

— cylinders and cones

$S_p = 1{,}6$

$S_k = 3{,}0$

— spherical region

$S_p = 2{,}4$

$S_k = 3{,}0 + 0{,}002\ R/s$

— knuckle region

$S_p = 1{,}8$

10.3.2.5 Weld joint factors, v

For internal pressure (pressure on the concave surface)

v = 0,85 or 1,0 (see Table 7)

For external pressure (pressure on the convex surface)

v = 1,0

10.3.2.6 Allowances for corrosion, c

$c = 0$.

10.3.3 Outer jacket

10.3.3.1 General

The following shall be used to determine the pressure part thickness in conjunction with the calculation formulae of 10.3.6.

10.3.3.2 Design pressure, p

The internal design pressure, p, shall be equal to the set pressure of the outer jacket pressure relief device.

The external design pressure, p, shall be 1 bar.

10.3.3.3 Material property, K

The material property, K, to be used in the calculations shall be at 20 °C, as defined in 10.3.2.3.

10.3.3.4 Safety factors, S, S_p, and S_k

Internal pressure (pressure on the concave surface)

$S = 1{,}1$

External pressure (pressure on the convex surface)

— cylinders and cones

$S_p = 1{,}1$

$S_k = 2{,}0$

NOTE For well proven designs, a factor of safety, S_k, equal to 1,5 is acceptable provided that

— D is not more than 2 300 mm,

— l_b is not more than 10 200 mm, and

— the annular space is perlite insulated.

— spherical region

$S_p = 1{,}6$

$S_k = 2{,}0 + 0{,}0014\ R/s$

— knuckle region

$S_p = 1{,}2$

10.3.3.5 Weld joint factors, v

For internal pressure (pressure on the concave surface)

$v = 0{,}7$

For external pressure (pressure on the convex surface)

$v = 1{,}0$

10.3.3.6 Allowances for corrosion, c

No allowance is required.

$c = 0$

NOTE External surfaces should be adequately protected against corrosion.

10.3.4 Supports and lifting points

The supports and lifting points shall be designed for the loads defined in 10.2, using established structural design methods and safety factors specified in 10.3.2.4 and 10.3.2.5.

When designing the inner vessel the temperature and corresponding mechanical properties of the structural attachment attached to the inner vessel may be those of the component in question when the inner vessel is filled to capacity with cryogenic fluid at a temperature not lower than the saturation temperature at pressure, p_s. However, it shall be checked whether the stresses are acceptable in warm conditions (i.e. vessel empty).

10.3.5 Piping and accessories

Piping shall be designed for the loads defined in 10.2.3.7 using established piping design methods and safety factors specified in 10.3.2.4.

10.3.6 Calculation formulae

10.3.6.1 Cylinders and spheres subject to internal pressure (pressure on the concave surface)

10.3.6.1.1 Field of application

Cylinders and spheres where:

$$D_\mathrm{a} / D_\mathrm{i} \leqslant 1{,}2$$

10.3.6.1.2 Openings

For reinforcement of openings see 10.3.6.7.

10.3.6.1.3 Calculation

The required minimum wall thickness, s, is

for cylinders

$$s = \frac{D_\mathrm{a} p}{20(K/S)\,\mathrm{v} + p} + c$$

for spheres

$$s = \frac{D_\mathrm{a} p}{40(K/S)\,\mathrm{v} + p} + c$$

10.3.6.2 Cylinders subject to external pressure (pressure on the convex surface)

10.3.6.2.1 Field of application

Cylinders where

$$D_\mathrm{a} / D_\mathrm{i} \leqslant 1{,}2$$

10.3.6.2.2 Openings

Openings shall be calculated in accordance with 10.3.6.7, using for the pressure in the formula a value equal to the external pressure as though it were internally applied.

10.3.6.2.3 Calculation

Annex L gives two alternative calculation methods. Both methods give comparable results and shall be equally accepted.

10.3.6.3 Spheres subject to external pressure (pressure on the convex surface)

Spheres subject to external pressure shall be evaluated in accordance with Annex L, Clauses L.1.2 and L.2.2.

10.3.6.4 Dished ends

10.3.6.4.1 Field of application

The following dished ends may be utilised:

a) hemispherical ends where $D_a/D_i \leqslant 1{,}2$;

b) torispherical ends where $0{,}5\ D_a \leqslant R \leqslant D_a$ and $0{,}5\ D_a \geqslant r \geqslant 0{,}06 D_a$ (for $r/D_a \leqslant 15\ \%$, rules shall be applied for $0{,}002 \leqslant (s-c)/D_a \leqslant 0{,}1$ and for $r/D_a > 15\ \%$, rules shall be applied for $0{,}001 \leqslant (s-c)/D_a \leqslant 0{,}1$);

c) 2:1 elliptical ends where $R = 0{,}9\ D_a$ and $r = 0{,}170\ D_a$.

NOTE In the case of elliptical ends $0{,}001 \leqslant (s-c)/D_a \leqslant 0{,}1$).

Dished ends of vacuum jackets are not required to meet the above restrictions on R and r, when r is greater or equal to $3s$.

10.3.6.4.2 Straight flange

The straight flange length, h_1 [Figure 4a)], shall be not less than 3 s for all ends.

The straight flange may be shorter providing that in the case of inner vessels the circumferential joint between the dished end and the cylinder is non-destructively tested as required for a weld joint factor of 1,0.

NOTE Other flange/weld configurations may be used provided that suitable calculations are carried out.

10.3.6.4.3 Intermediate heads

Heads, without limit to thickness, may be installed in accordance with Figure F.2. The outside diameter of the head skirt shall be a close fit inside the ends of the adjacent sections of the cylinder.

The butt weld and fillet weld shall be adequately sized to jointly resist any relevant pressure, mechanical and thermal loads. This may be achieved by accurate detailed stress analysis and by adopting the criteria for acceptable stresses of Annex A.

Where only pressure stresses are present, a simplified approach may be adopted such that the butt weld and fillet weld are sized to resist in shear a load equivalent to 1,5 times the maximum differential pressure across the head multiplied by the cross sectional area of the shell.

The allowable shear stress in this simplified case should not exceed $K/3$ where the area of the butt weld in shear is the width at the root of the weld multiplied by the circumferential length of the weld and the area of the fillet weld is the throat thickness multiplied by the circumferential length of the weld.

Where the stresses in the attachment are fully analysed and assessed in accordance with Annex A, the fillet weld may be omitted. In other cases the fillet weld must be continuous.

10.3.6.4.4 Internal pressure calculation (pressure on concave surface)

10.3.6.4.4.1 Crown and hemisphere thickness

The wall thickness of the crown region of dished ends and of hemispherical ends shall be determined using 10.3.6.1.3 for spheres with $D_a = 2\ (R + s)$.

Openings within the crown area of 0,6 D_a of torispherical ends [see Figure 4b)], and in hemispherical ends shall be reinforced in accordance with 10.3.6.7. When pad type reinforcement is used the edge of the pad shall not extend beyond the area of 0,8 D_a for 10 % torispherical ends or 0,7 D_a for 2:1 elliptical ends.

10.3.6.4.4.2 If the ligament on the connecting line between adjacent openings is not entirely within the 0,6 D_a region the ligament shall not be less than half the sum of the opening diameters.

10.3.6.4.5 External pressure calculations (pressure on the convex surface)

See Annex L.

10.3.6.5 Cones subject to internal or external pressure

10.3.6.5.1 Symbols and units

For the purposes of 10.3.6.5, the following symbols apply in addition to those given in Clause 4:

A	area of reinforcing ring	mm^2
D_{a1}	outside diameter of connected cylinder (see Figure 7)	mm
D_{a2}	outside diameter at effective stiffening (see Figure 9)	mm
D_k	design diameter (see Figure 7)	mm
D_s	shell diameter at nozzle (see Figure 8)	mm
I	moment of inertia about the axis parallel to the shell	mm^4
l	cone length between effective stiffenings (see Figure 9)	mm
s_g	required wall thickness outside corner area	mm
s_l	required wall thickness within corner area	mm
x_i	characteristic lengths (i = 1,2,3) to define corner area [Figures 7 a) and b) and 10.3.6.5.5]	mm
φ	cone angle	°
r	inside radius of knuckle	mm

10.3.6.5.2 Field of application

Cones according to Figure 7 where:

$$0{,}001 \leqslant \frac{s_g - c}{D_{a1}} \leqslant 0{,}1$$

and

$$0{,}001 \leqslant \frac{s_l - c}{D_{a1}} \leqslant 0{,}1$$

Small ends with a knuckle can be safely assessed and verified as a small end with a corner joint.

For external pressure $|\varphi| \leqslant 70°$.

Other cone angles may be used provided that suitable calculations are carried out.

10.3.6.5.3 Openings

Openings outside of the corner area (Figure 8) shall be designed as follows.

If $|\varphi| < 70°$ design according to 10.3.6.7 using an equivalent cylinder diameter of:

$$D_\mathrm{i} = \frac{D_\mathrm{s} + d_\mathrm{i}|\sin\varphi|}{\cos\varphi}$$

$|\varphi| \geqslant 70°$ design according to 10.3.6.5.7.

10.3.6.5.4 Non-destructive testing

All corner joints shall be subject to the examination required for a weld joint factor of 1,0. See Table 7.

10.3.6.5.5 Corner area

The corner area is that part of the cone where the dominant stresses are bending stresses in the longitudinal direction.

The corner area is defined in Figures 7 a) and b) by x_1, x_2, x_3 calculated from the following equations:

$$x_1 = \sqrt{D_{\mathrm{a}1}(s_\mathrm{l} - c)}$$

$$x_2 = 0{,}7\sqrt{\frac{D_{\mathrm{a}1}(s_\mathrm{l} - c)}{\cos\varphi}}$$

$$x_3 = 0{,}5x_1$$

10.3.6.5.6 Internal pressure calculation (pressure on concave surface) $|\varphi| \leqslant 70°$

a) within corner area

The required wall thickness (s_1) within the corner area is calculated from Figures 10.1 to 10.7 for the large end and Figure 10.8 for the small end of a cone using the following variables:

$$\varphi,\ \frac{pS}{15K\nu}, \text{ and } \frac{r}{D_\mathrm{a}\,l}$$

For a corner joint use the curve for $\frac{r}{D_{\mathrm{a}1}} = 0$.

For intermediate cone angles use linear interpolation. The wall thickness, s_l, in the corner area shall not be less than the required thickness, s_g, outside of the corner area as calculated in 10.3.6.5.6 b).

b) outside corner area

The required wall thickness, s_g, outside the corner area is calculated from:

$$s_g = \frac{D_k\, p}{20\frac{K}{S}\nu - p} \div \frac{1}{\cos\varphi} + c$$

where

for the large end, $D_k = D_{a1} - 2\left[s_1 + r\left(1 - \cos\varphi\right) + x_2 \sin\varphi\right]$.

For the small end, D_k is the maximum diameter of the cone, where the wall thickness is s_g.

10.3.6.5.7 Internal pressure calculation (pressure on the concave surface) $|\varphi| > 70°$

If $r \geqslant 0{,}01\, D_{a1}$ the required wall thickness is

$$s_l = s_g = 0{,}3\left(D_a\, l - r\right) \times \frac{|\varphi|}{90} \times \sqrt{\frac{p}{10\left(\frac{k}{S}\right)\nu}} + c$$

10.3.6.5.8 External pressure calculation (pressure on the convex surface)

Stability against elastic buckling and plastic deformation shall be verified using 10.3.6.2 and an equivalent cylinder.

For the example shown in Figure 9 the equivalent cylinder diameter between the knuckle and the stiffener is:

$$D_a = \frac{D_{a1} + D_{a2}}{2\cos|\varphi|}$$

and the equivalent cylinder length is:

$$l = \frac{D_{a1} - D_{a2}}{2\sin|\varphi|}$$

Depending on the relevant boundary conditions the equivalent length between two effective stiffening sections shall be reliably estimated within the context of 10.3.6.2.

When $\varphi \geqslant 10°$ the corner area of a large end can be considered as effective stiffening.

For small ends the thickness in the corner area shall not be less than 2,5 times the required thickness of the conical shell with the same angle $|\varphi|$ or a stiffener shall be fitted with the following properties:

$$I \geqslant \frac{p\left(D_{a1}\right)^4}{960\left(\frac{E}{S_k}\right)} \tan|\varphi|$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted value of I is not less than that determined at the test pressure with a safety factor of 0,74 S_k.

$$A \geqslant \frac{p\left(D_{\mathrm{a1}}\right)^2}{80\left(\dfrac{K}{S_{\mathrm{p}}}\right)} \tan\left|\varphi\right|$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted value of A is not less than that determined at the test pressure with a safety factor of 0,74 S_{p}.

S_{k} (cylinder) is the safety factor to prevent elastic buckling from 10.3.2.4 or 10.3.3.4.

S_{p} (cylinder) is the safety factor to prevent plastic deformation from 10.3.2.4 or 10.3.3.4.

D_{a1} is the diameter according to Figure 7 b).

The shell over a width of $0{,}5\sqrt{D_{\mathrm{a1}} s_1}$ can be used to calculate the moment of inertia and the area.

In addition the corner joint should not be regarded as a classical boundary condition i.e. the overall length should be formed from the individual meridional length of the cone and cylinder.

In addition, the cone shall be verified using 10.3.6.5.6 and the safety factors S_{p} for cylinders from 10.3.2.4. or 10.3.3.4. If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at the test pressure with a safety factor of 0,74 S_{k}. For thickness calculations in the corner area, v shall be the value applicable for internal pressure.

10.3.6.6 Flat ends

10.3.6.6.1 Symbols

For the purposes of 10.3.6.6, the following symbols apply in addition to those given in Clause 4:

— d_1, d_2 etc. opening diameters in mm;

— D_1, D_2 etc. flat end diameters in mm.

10.3.6.6.2 Field of application

Welded or solid flat ends where Poisson ratio is approximately 0,3 and

$$\frac{(s-c)}{D} \geqslant \sqrt[4]{\frac{0{,}0087\,p}{E}}$$

and

$$\frac{(s-c)}{D} \leqslant \frac{1}{3}$$

10.3.6.6.3 Openings

Openings are calculated in accordance with 10.3.6.6.4 but with the C factor multiplied by C_{A}, where C_{A} is given in Figure 11.

10.3.6.6.4 Calculation

The required minimum wall thickness of a circular flat end is:

$$s = CD_1\sqrt{\frac{0{,}1pS}{K}} + c$$

C and D_1 are taken from Figure 12.

The required minimum wall thickness of a rectangular or elliptical flat end is

$$s = CC_E f\sqrt{\frac{0{,}1pS}{K}} + c$$

where C_E is taken from Figure 13.

10.3.6.7 Openings in cylinders, spheres and cones

10.3.6.7.1 Reinforcement methods

Openings may be reinforced by one or more of the following typical but not exclusive methods:

— increase of shell thickness, see Figures 14 and 15;

— set-in or set-on ring reinforcement, see Figures 16 and 17;

— pad reinforcement, see Figure 18;

— increase of nozzle thickness, see Figures 19 and 20;

— pad and nozzle reinforcement, see Figure 21.

Where ring or pad reinforcement is used on the inner vessel, the space between the two fillet welds shall be vented into the vacuum inter-space.

10.3.6.7.2 Design of openings

All nozzles shall be attached to the vessel wall with a full penetration weld unless the attachment weld is maintained at atmospheric temperatures at all times or the weld is not subjected to thermal cycling.

The fillet weld on a reinforcing pad shall have a minimum throat thickness of half of the pad thickness.

The throat thickness of a fillet weld of each nozzle to shell weld shall be not less than the required thickness of the thinner part.

Where the strength of the reinforcing material is lower than the strength of the shell material an allowance in accordance with 10.3.6.7.3 shall be made in the design calculations. If the strength of the reinforcing material is higher than the strength of the shell material, no allowance for the increased strength is permitted.

The design rules for non-perpendicular nozzles shall be based on a perpendicular nozzle, using the dimension of the major elliptical axis.

10.3.6.7.3 Calculation

Annex M gives two alternative calculation methods. Both methods give comparable results and shall be equally accepted.

10.3.7 Calculations for operating loads

Unless the design has been validated by experiment, calculations in addition to those in 10.3.6 may be required to ensure that stresses due to operating loads are within acceptable limits. All load conditions expected during service shall be considered (see 10.2.3).

In these calculations, static loads are substituted for static plus dynamic loads.

The analysis shall take account of gross structural discontinuities, but need not consider local stress concentrations.

Annex A or ASME, section VIII, Division 2 provides terminology and acceptable stress limits when an elastic stress analysis is performed.

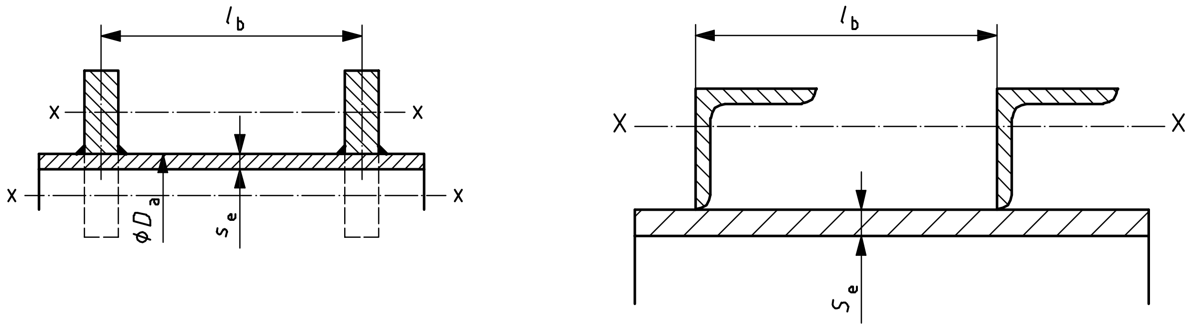
Acceptable calculation methods include:

— finite element;

— finite difference;

— boundary element;

— recognised text books, codes and standards.



a) b)

Figure 1 — Stiffening rings

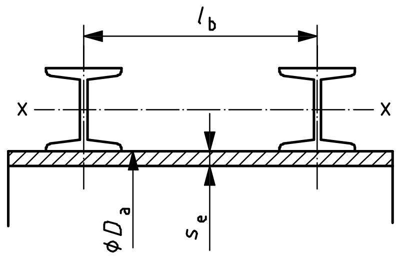


Figure 2 — Sectional materials stiffeners

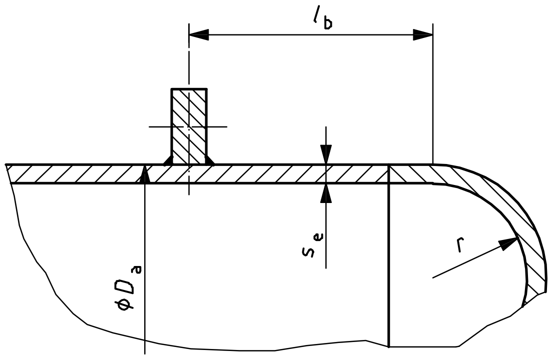
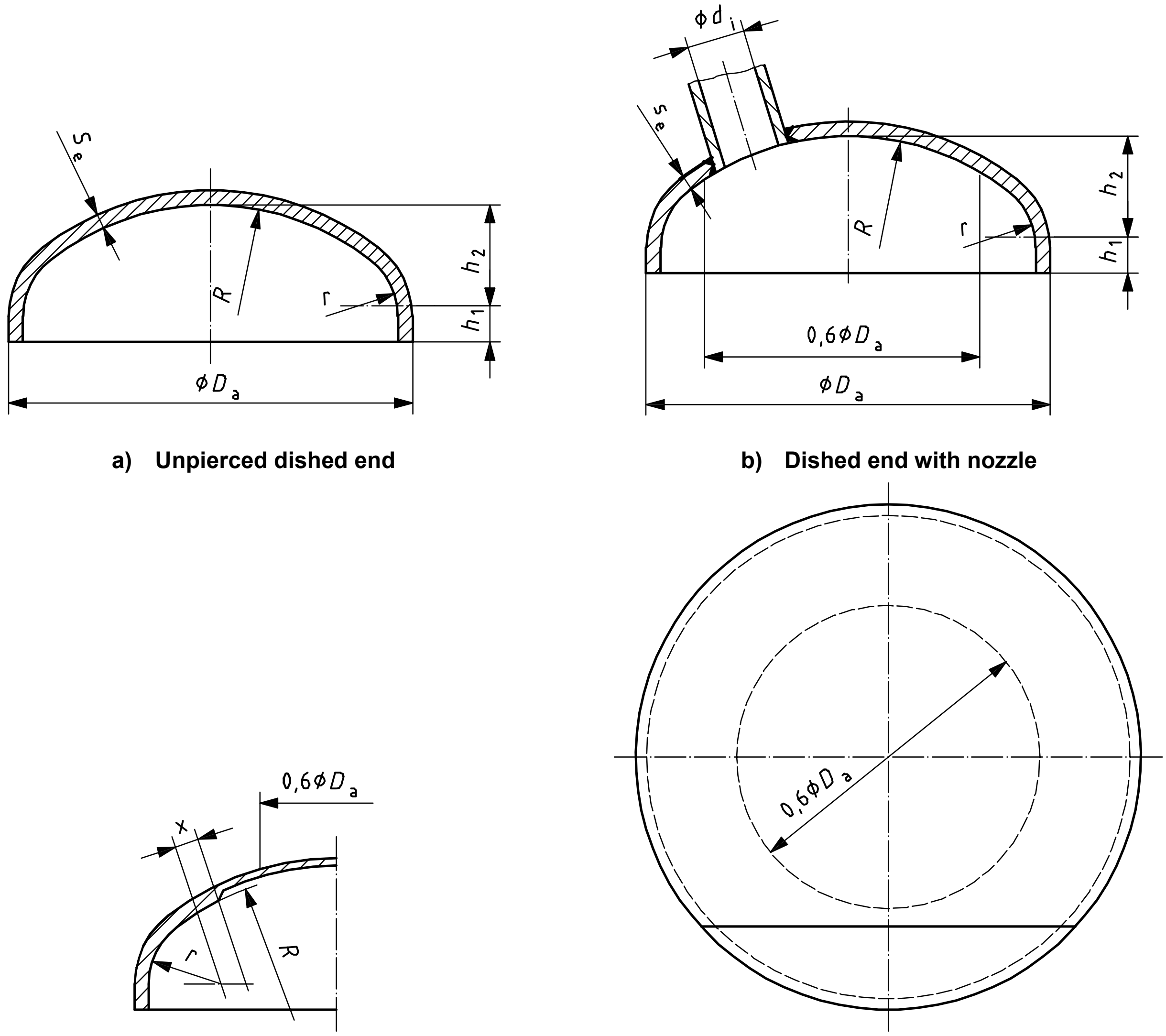


Figure 3 — Dished ends



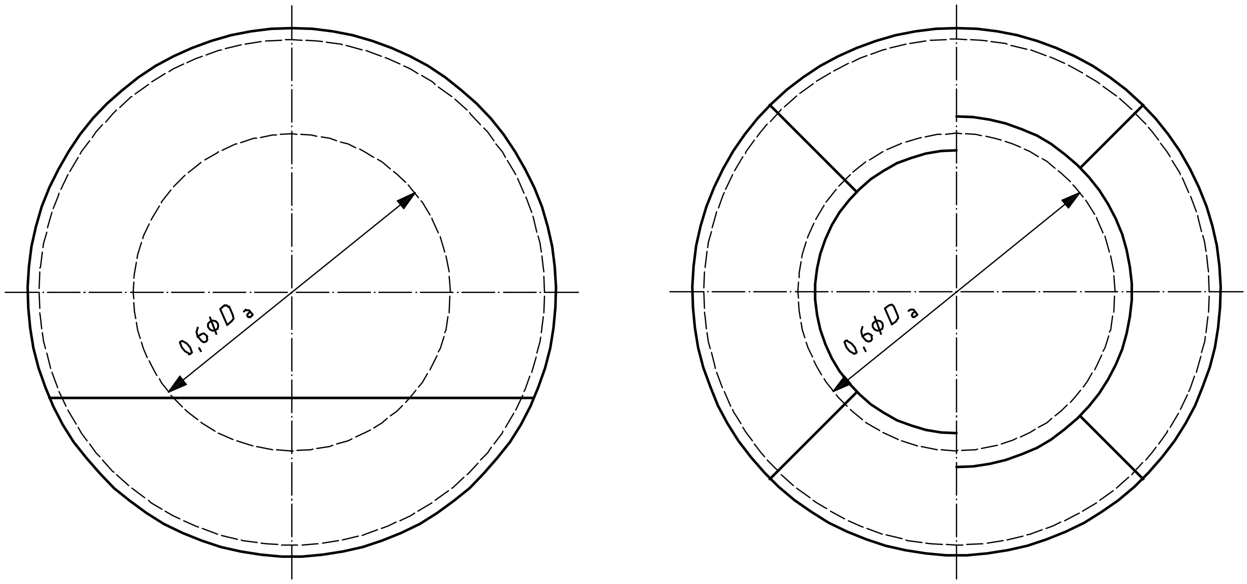
a) Unpierced dished end

b) Dished end with nozzle

c) End with knuckle and crown of unequal wall thickness

d) Weld outside 0,6, D_a

Figure 4 — Vessel ends and weld positions



e) Weld inside 0,6, D_a

f) End welded together from round plate and segments

v = 1,0   v = 0,85 or 1,0

Figure 4 (*continued*)

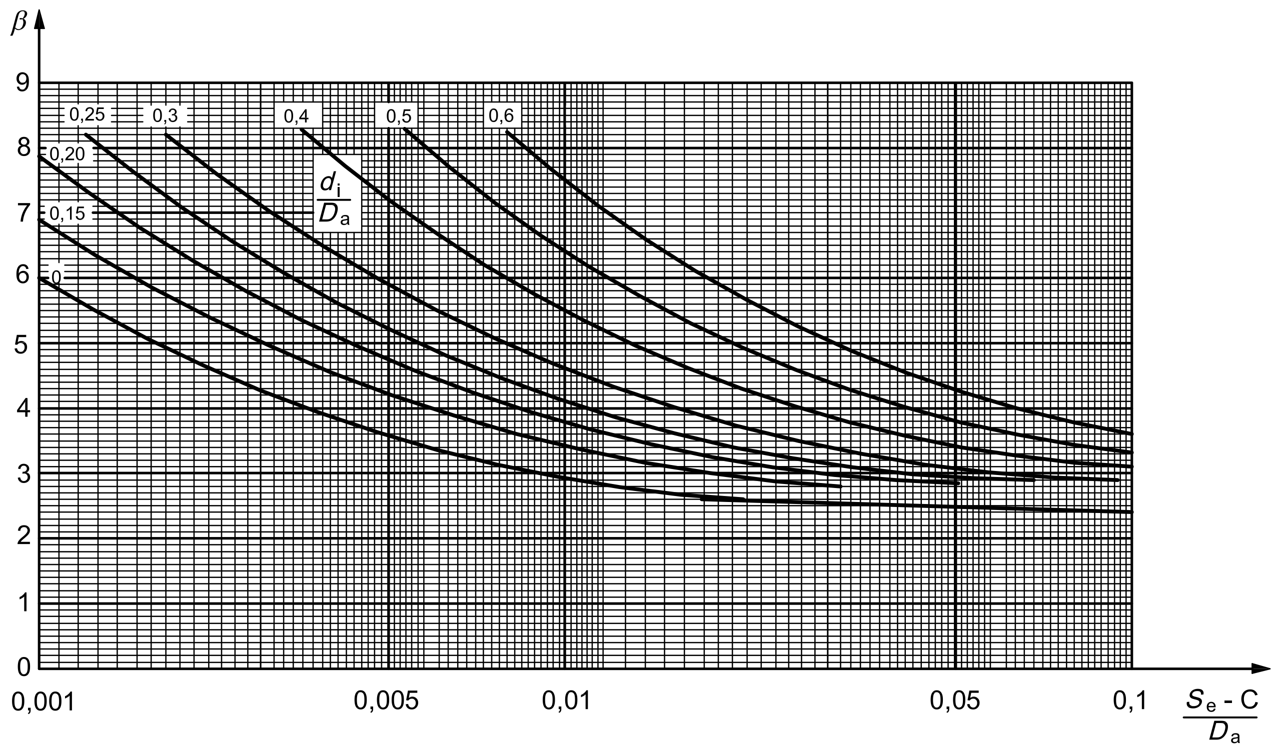


Figure 5 — Design factors, β, for 10 % torispherical dished ends

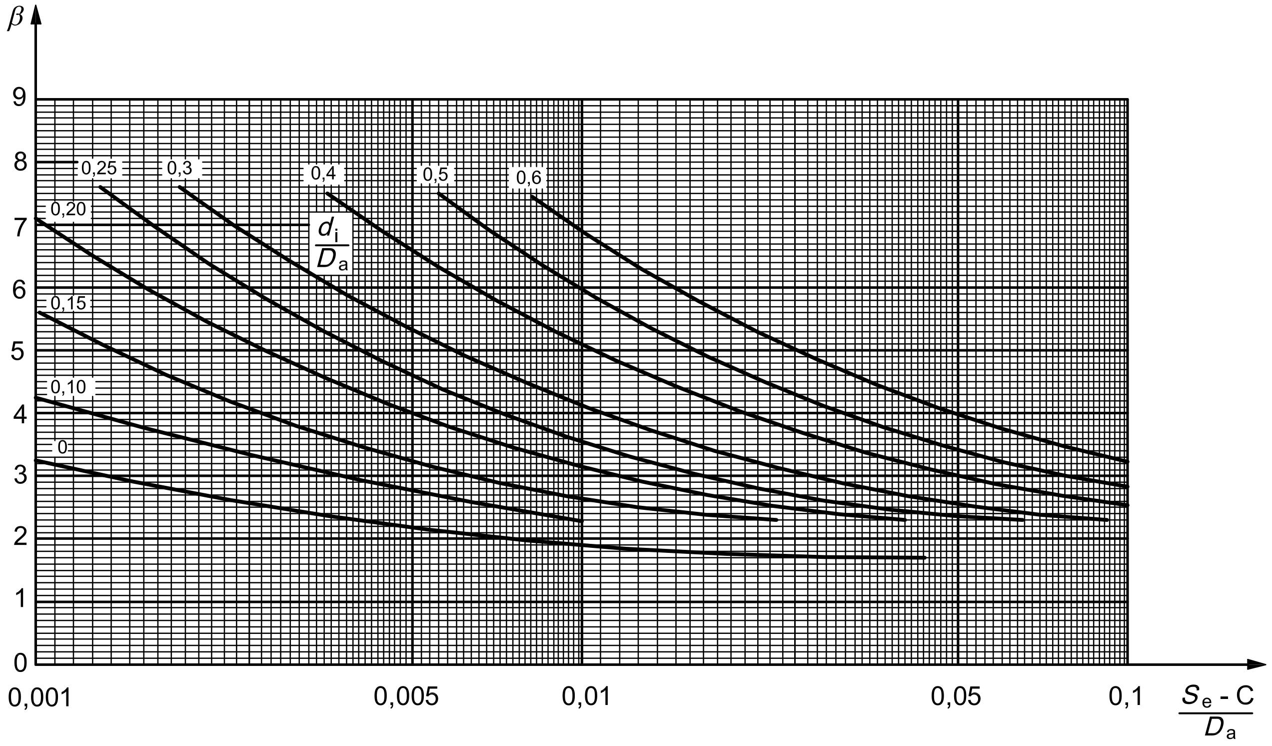
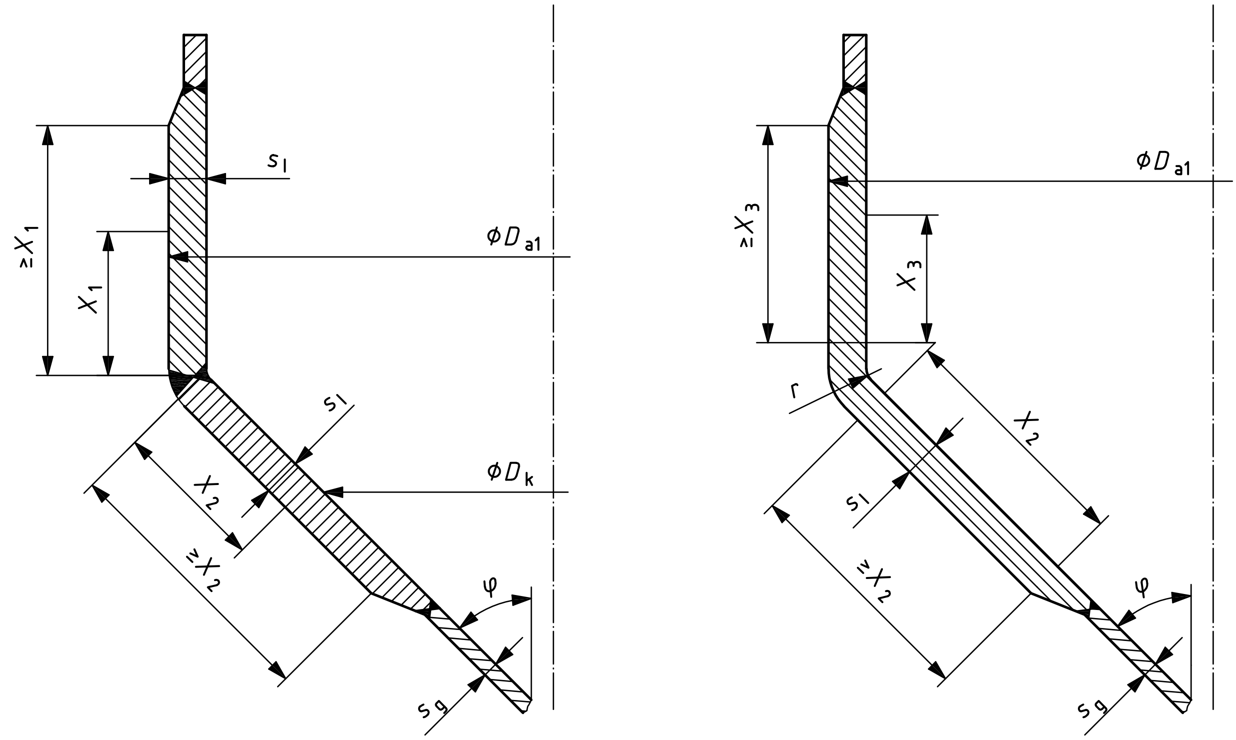
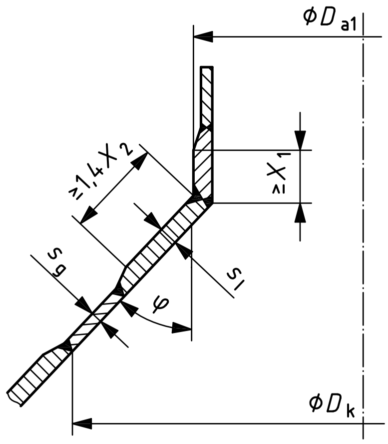


Figure 6 — Design factors, β, for 2:1 torispherical dished ends



a) Geometry of convergent conical shells



b) Geometry of a divergent conical shell

Figure 7

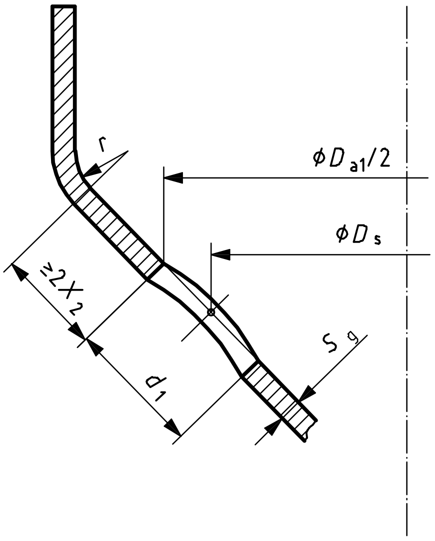


Figure 8 — Geometry of a cone opening

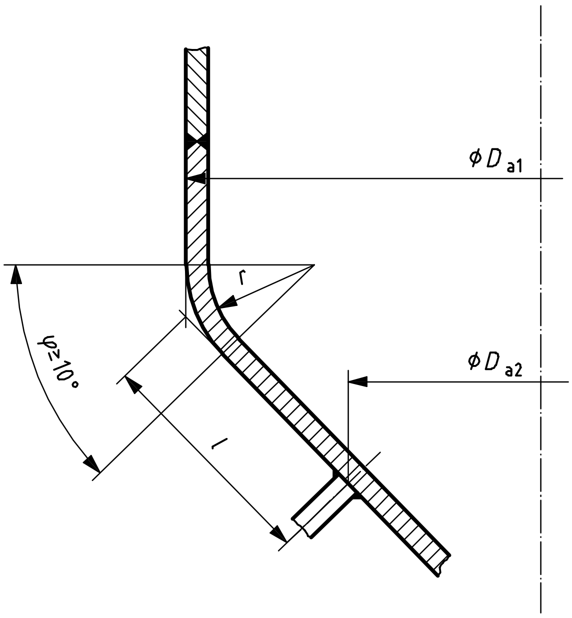


Figure 9 — Geometrical quantities in the case of loading by external pressure

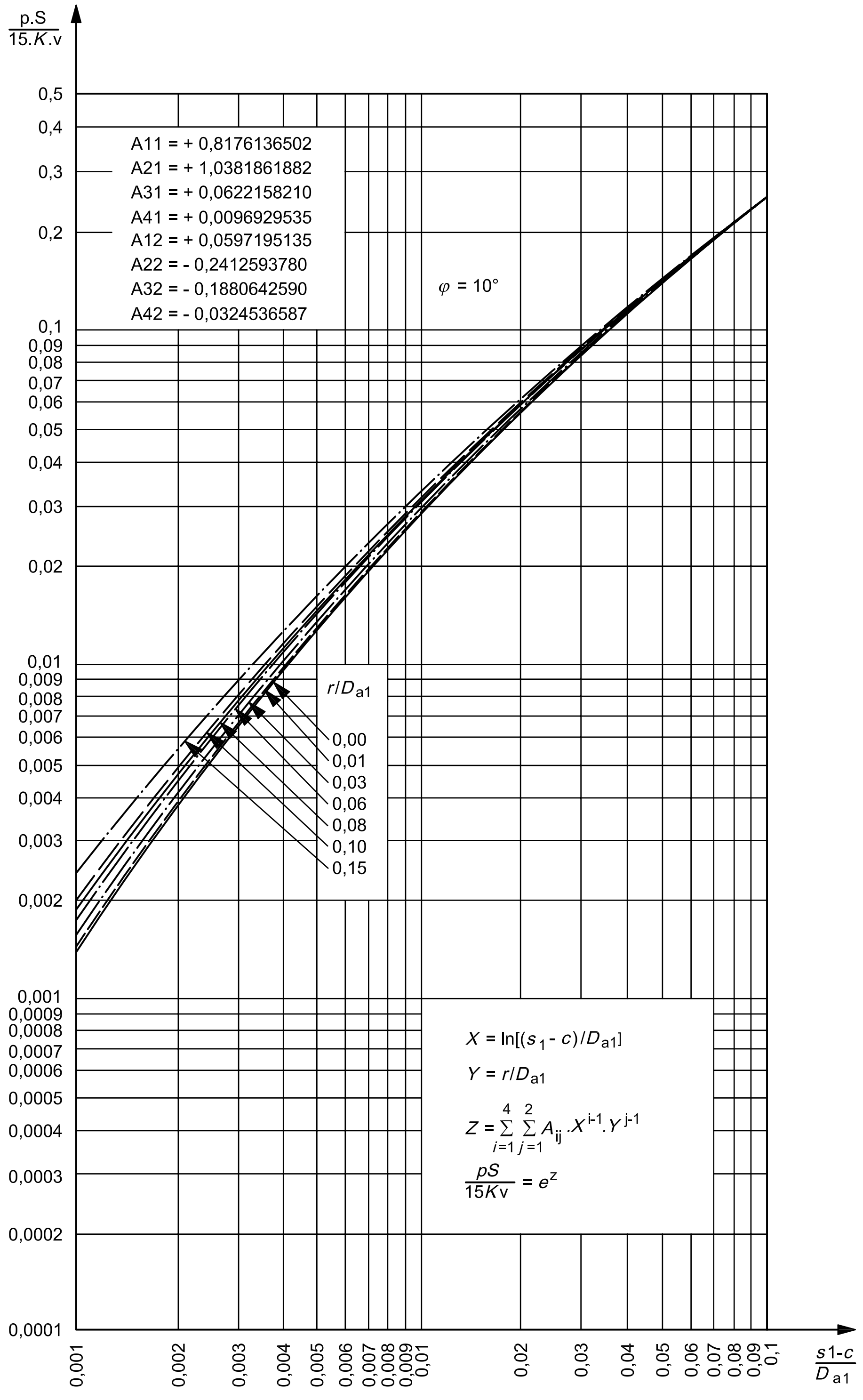


Figure 10.1 — Permissible value, $\frac{pS}{15K\mathrm{v}}$, for convergent cone with an opening angle $\varphi = 10°$

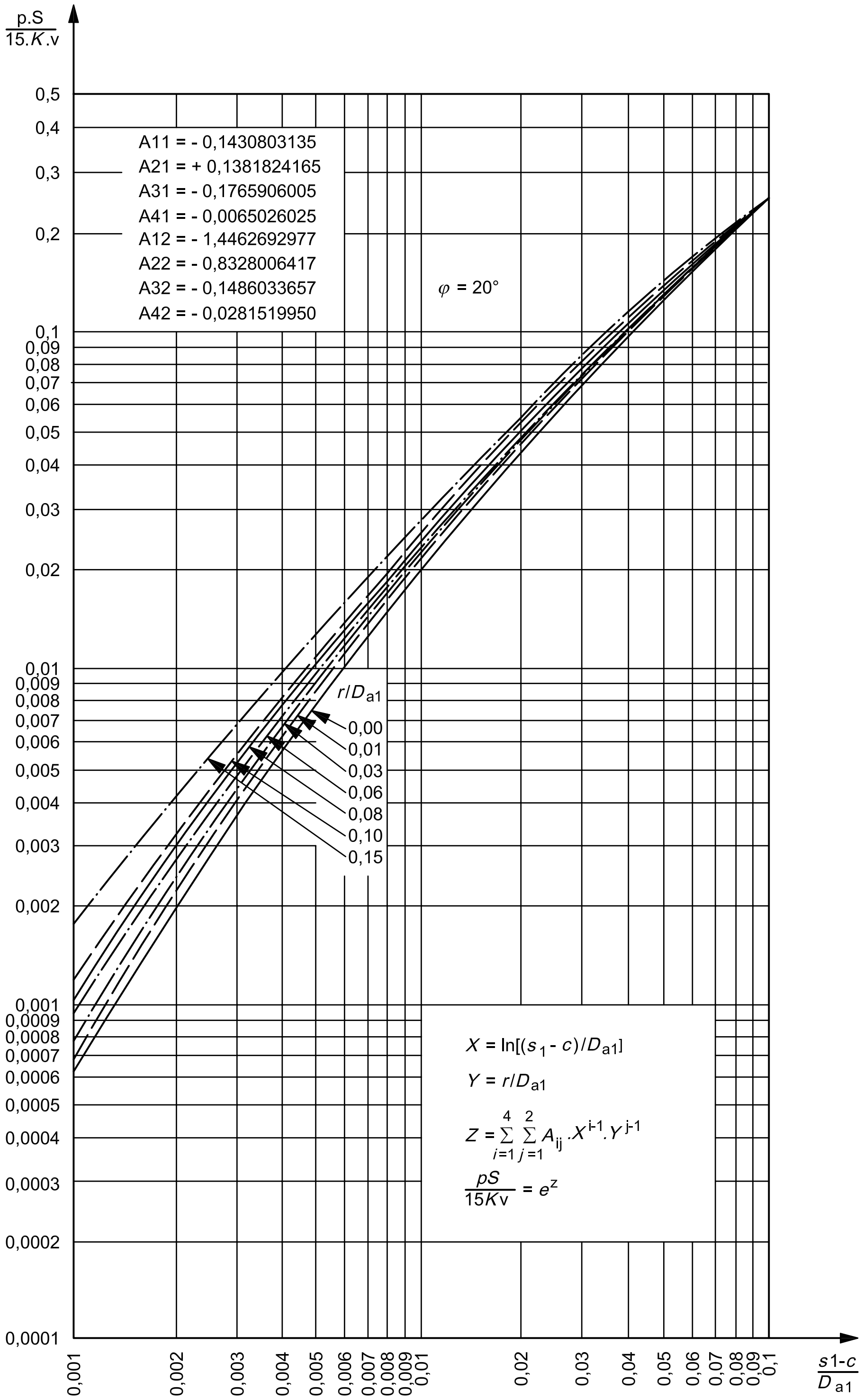


Figure 10.2 — Permissible value, $\frac{pS}{15K\text{v}}$, for convergent cone with an opening angle $\varphi = 20°$

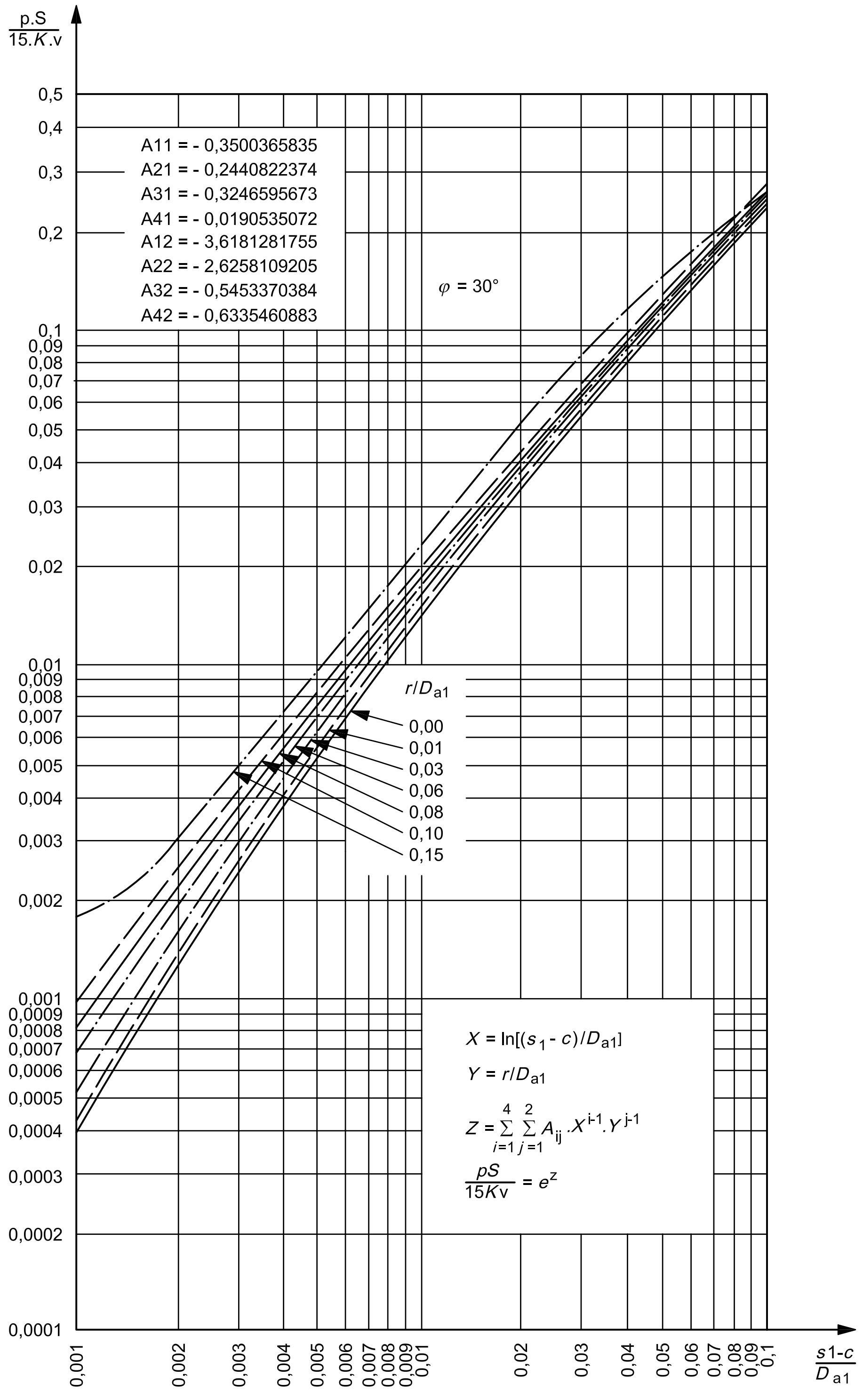


Figure 10.3 — Permissible value, $\frac{pS}{15K\text{v}}$, for convergent cone with an opening angle $\varphi = 30°$

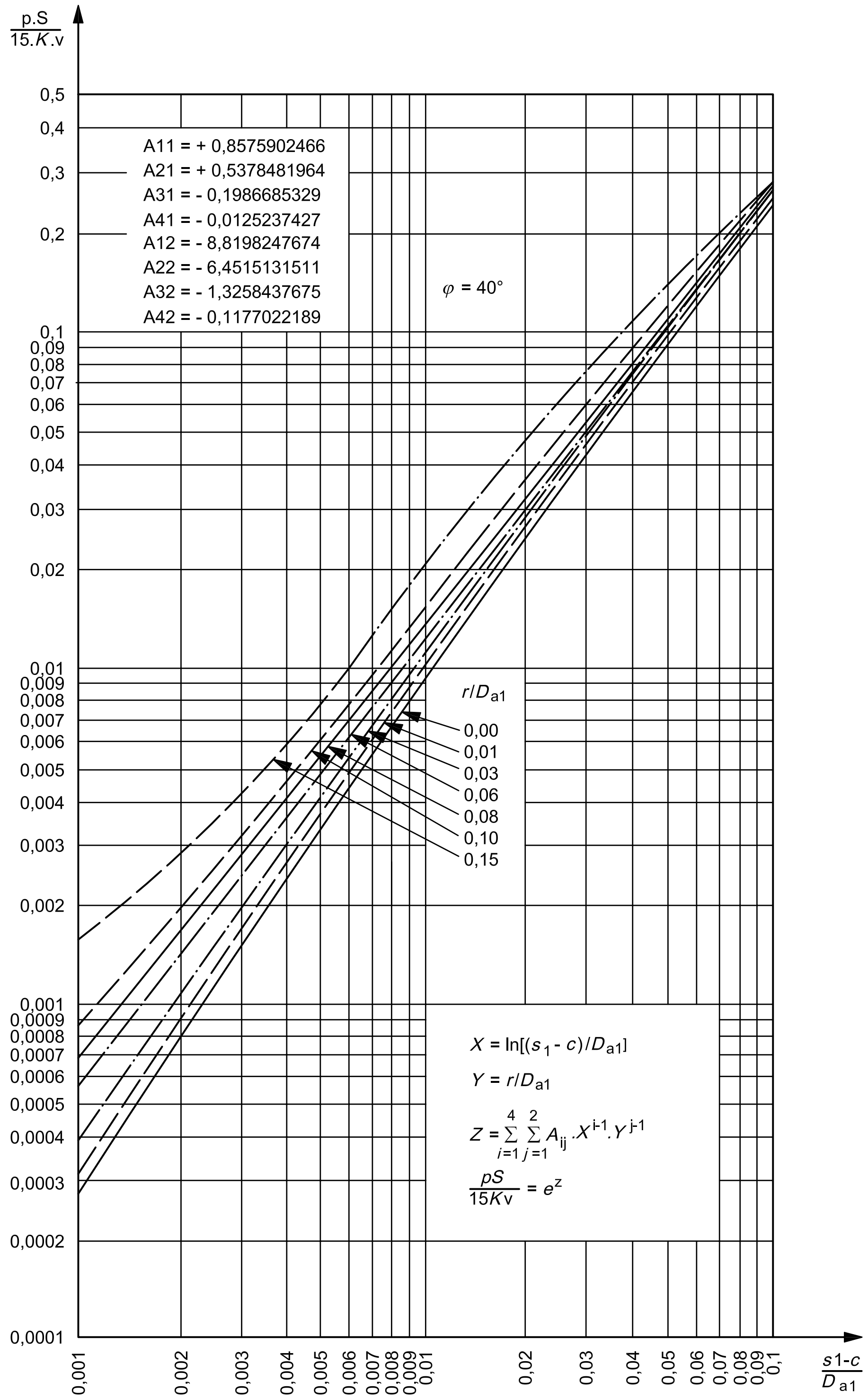


Figure 10.4 — Permissible value, $\frac{pS}{15K\text{v}}$, for convergent cone with an opening angle $\varphi = 40°$

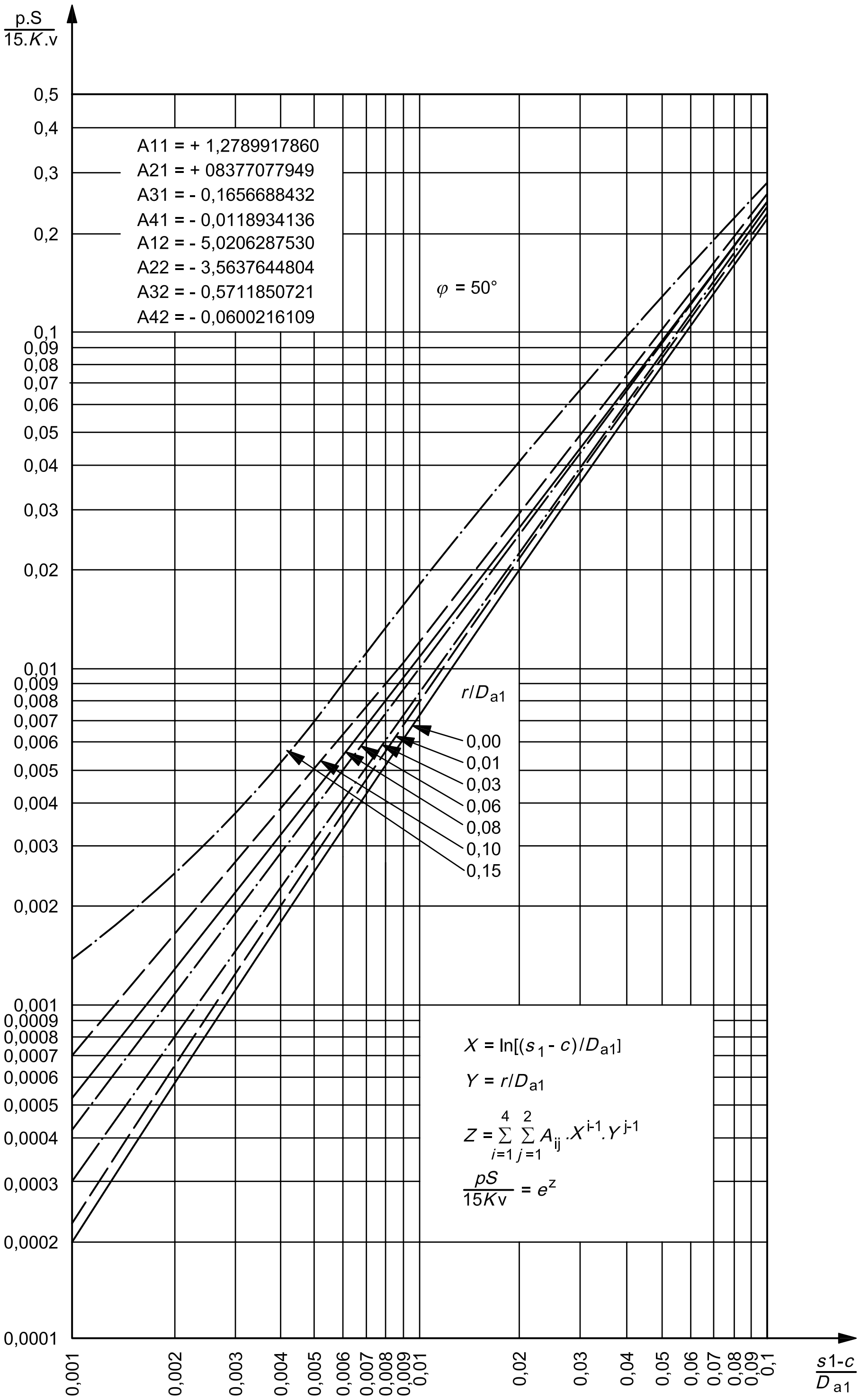


Figure 10.5 — Permissible value, $\frac{pS}{15K\mathrm{v}}$, for convergent cone with an opening angle $\varphi = 50°$

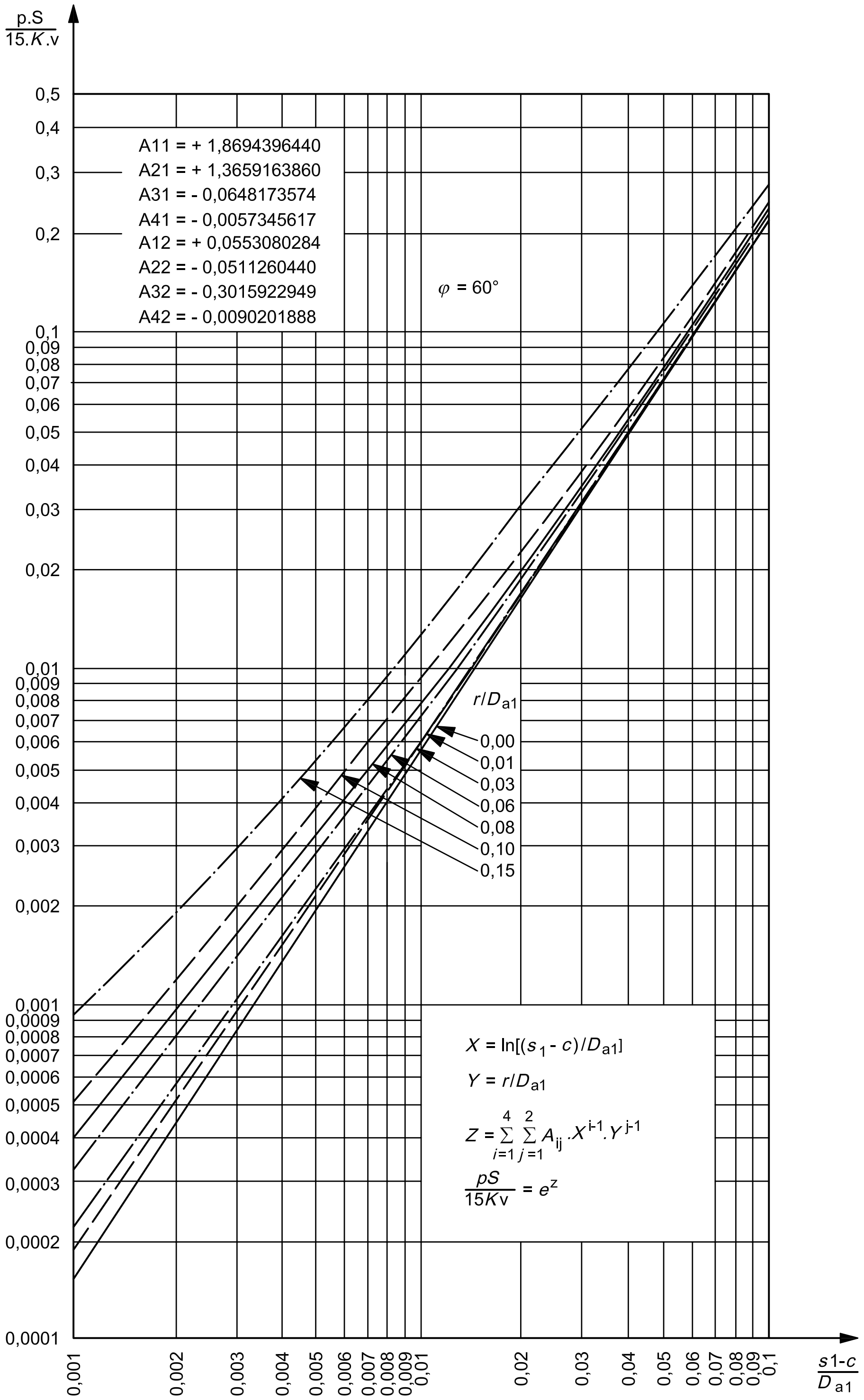


Figure 10.6 — Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle $\varphi = 60°$

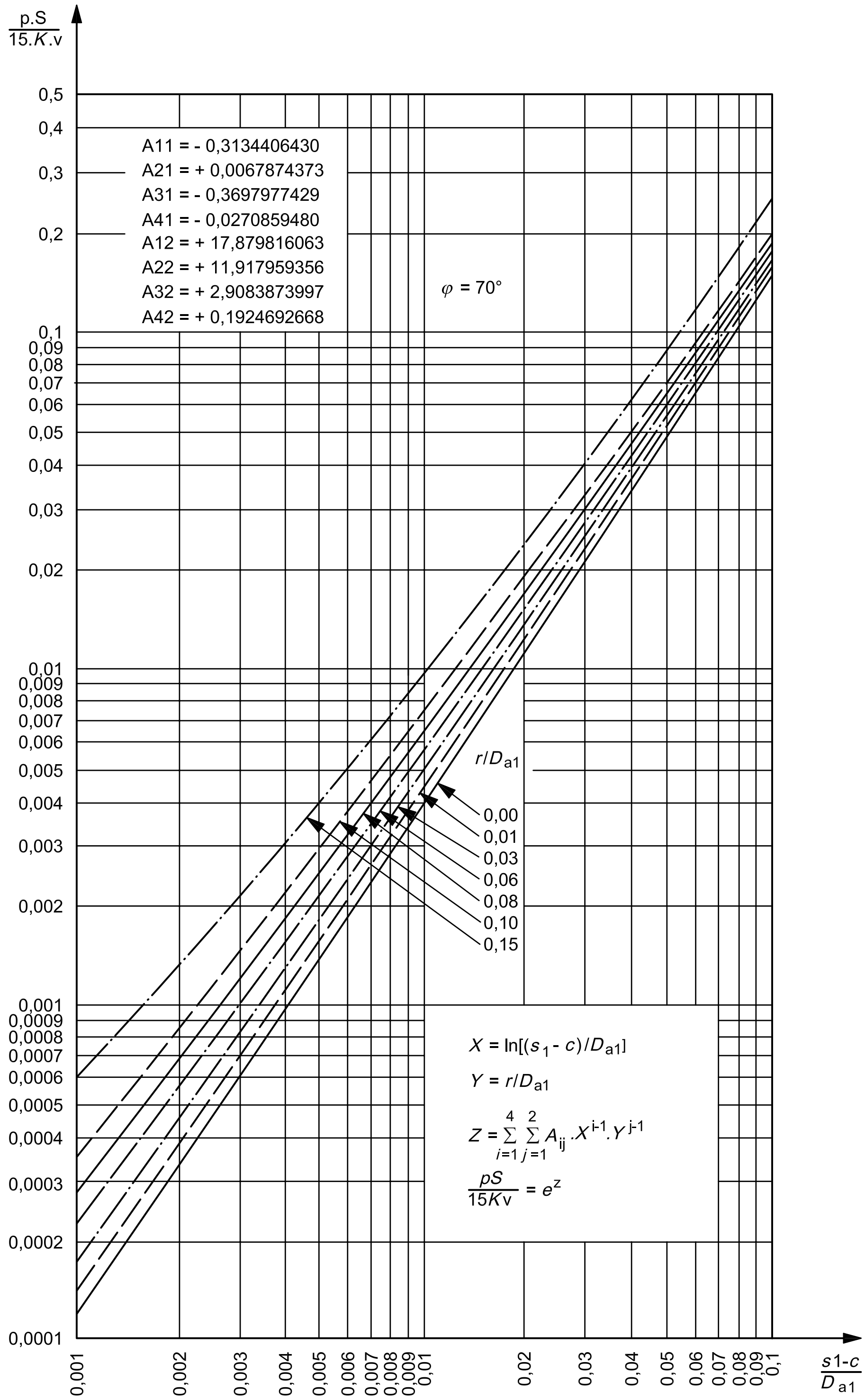


Figure 10.7 — Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle $\varphi = 70°$

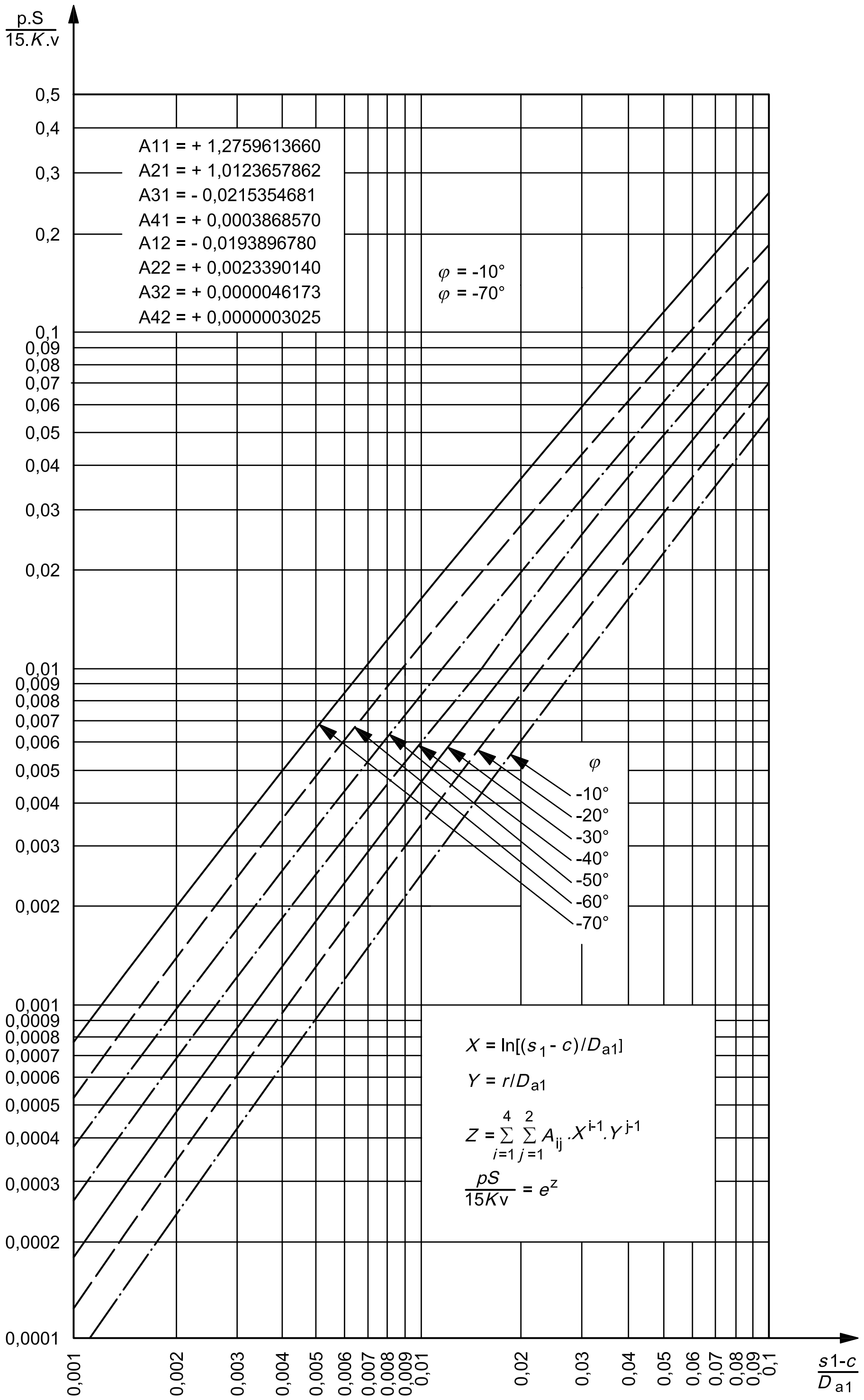
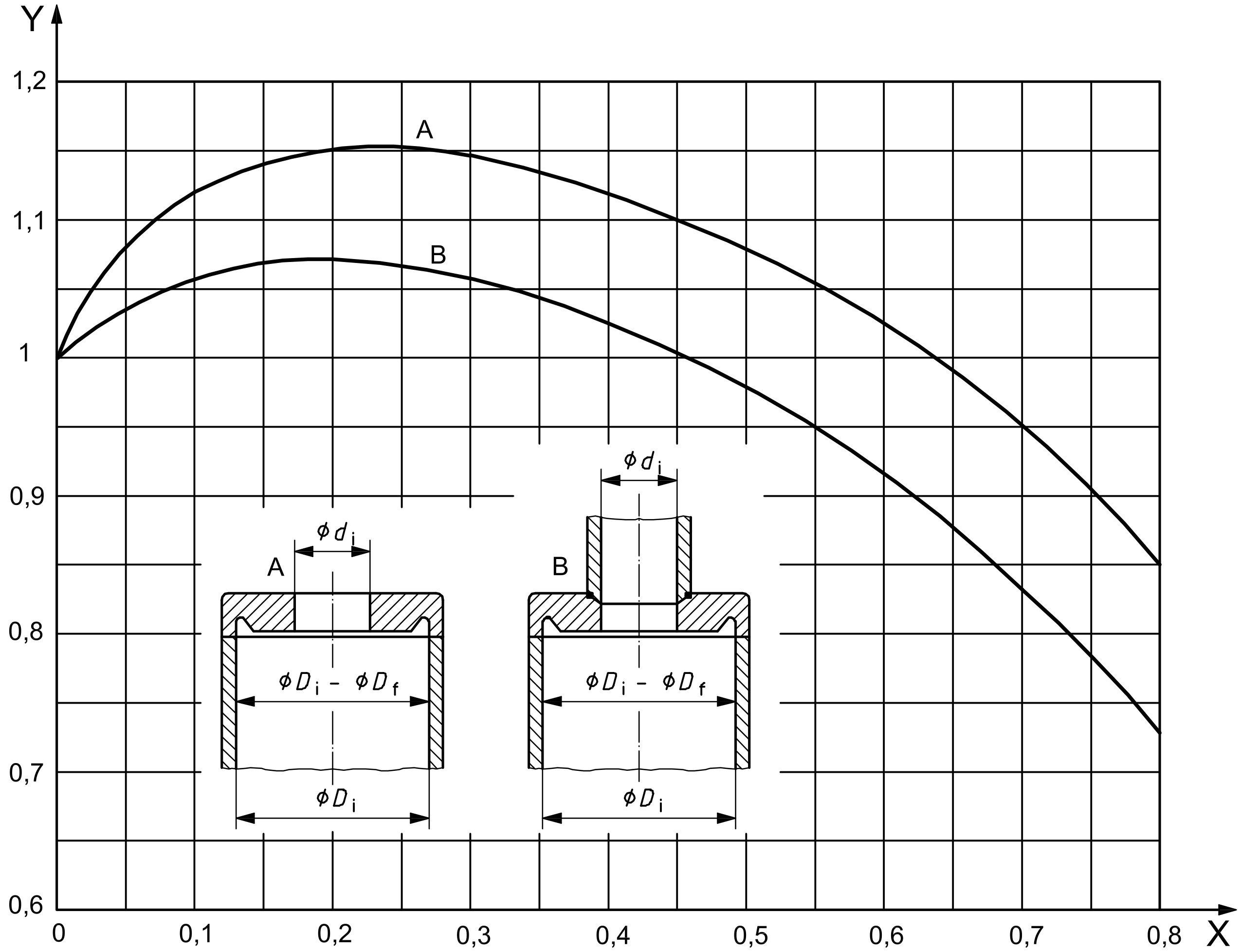


Figure 10.8 — Permissible value, $\frac{pS}{15K\text{v}}$, for convergent cone (corner joint) with an opening angle φ = –10° to –70°



Key

Y Opening factor, C_A

X Ratio d_i/D_i resp. d_i/f

Type A

d = inside diameter of opening

D_i = design diameter

f = short side of elliptical end

$$C_A = \begin{cases} \left| \sum_{i=1}^{6} A_i \left(\frac{d}{D_i} \right)^{i-1} \right| & 0 < \left(\frac{d}{D_i} \right) \leqslant 0{,}8 \\ \left| \sum_{i=1}^{6} A_i \left(\frac{d}{f} \right)^{i-1} \right| & 0 < \left(\frac{d}{f} \right) \leqslant 0{,}8 \end{cases}$$

A_1 = 0,999 034 20

A_2 = 1,980 626 00

A_3 = 9,018 554 00

A_4 = 18,632 830 00

A_5 = 19,497 590 00

A_6 = 7,612 568 00

Type B

d = inside diameter of opening

D_i = design diameter

f = short side of elliptical end

$$C_A = \begin{cases} \left| \sum_{i=1}^{6} A_i \left(\frac{d}{D_i} \right)^{i-1} \right| & 0 < \left(\frac{d}{D_i} \right) \leqslant 0{,}8 \\ \left| \sum_{i=1}^{6} A_i \left(\frac{d}{f} \right)^{i-1} \right| & 0 < \left(\frac{d}{f} \right) \leqslant 0{,}8 \end{cases}$$

A_1 = 1,001 003 44

A_2 = 0,944 284 68

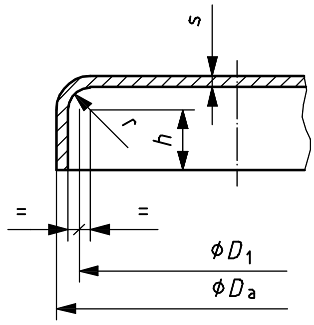
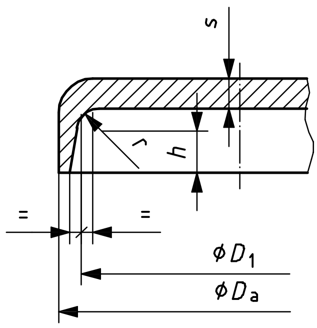
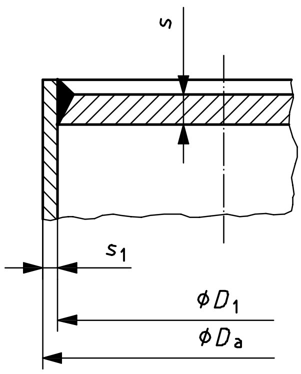
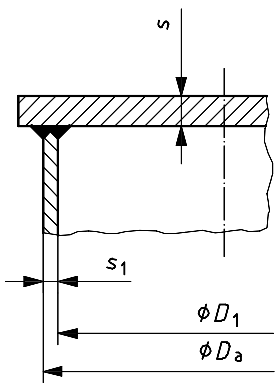
A_3 = 4,312 102 00

A_4 = 8,389 435 00

A_5 = 9,206 283 84

A_6 = 3,694 941 96

Figure 11 — Opening factor, C_A, for flat ends and plates without additional marginal moment

<table>
<tr><th>Type of flat end design
(principle only)</th><th>Conditions</th><th>Design factor, C</th></tr>
<tr><td>a) flat end
</td><td>1. knuckle radius:
<table>
<tr><td>D_a</td><td></td><td>r_{min}</td></tr>
<tr><td>up to 500</td><td></td><td>30</td></tr>
<tr><td>over 500</td><td>up to 1400</td><td>35</td></tr>
<tr><td>over 1400</td><td>up to 1600</td><td>40</td></tr>
<tr><td>over 1600</td><td>up to 1900</td><td>45</td></tr>
<tr><td>over 1900</td><td></td><td>50</td></tr>
</table>
and $r \geqslant 1{,}3\ s$
2. cylindrical part: $h \geqslant 3{,}5 \times s$</td><td>0,30</td></tr>
<tr><td>b) forged or pressed flat end
</td><td>1. knuckle radius:
$r \geqslant \frac{s}{3}$, however at least 8 mm
2. cylindrical part: $h \geqslant s$</td><td>0,35</td></tr>
<tr><td>c) flat plate welded into the shell from one side only
</td><td>plate thickness:
$s \leqslant 3\ s_1$
$s > 3\ s_1$</td><td>
0,45
0,50</td></tr>
<tr><td>d) plate welded into the shell with welds at both sides of the latter
</td><td>plate thickness:
$s \leqslant 3\ s_1$
$s > 3\ s_1$
Only killed steels may be utilised. When plate material is employed, over an area of at least 3 s_1 in the weld zone there shall be no evidence of material discontinuities in the plate.</td><td>
0,40
0,45</td></tr>
</table>

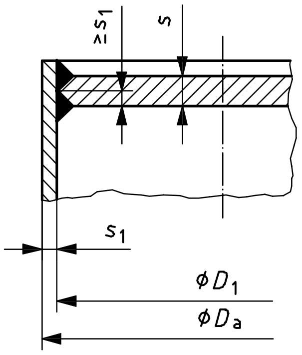
Type of flat end design (principle only)	**Conditions**	**Design factor**, C
e) flat plate welded into the shell from both sides 	plate thickness: $s \leqslant 3\ s_1$ $s > 3\ s_1$	 0,35 0,40

Figure 12 — Design factors for unstayed circular flat ends and plates

11 Fabrication

11.1 General

11.1.1 The manufacturer, or their sub-contractor, shall have equipment available to ensure manufacture and testing in accordance with the design.

11.1.2 The manufacturer shall maintain:

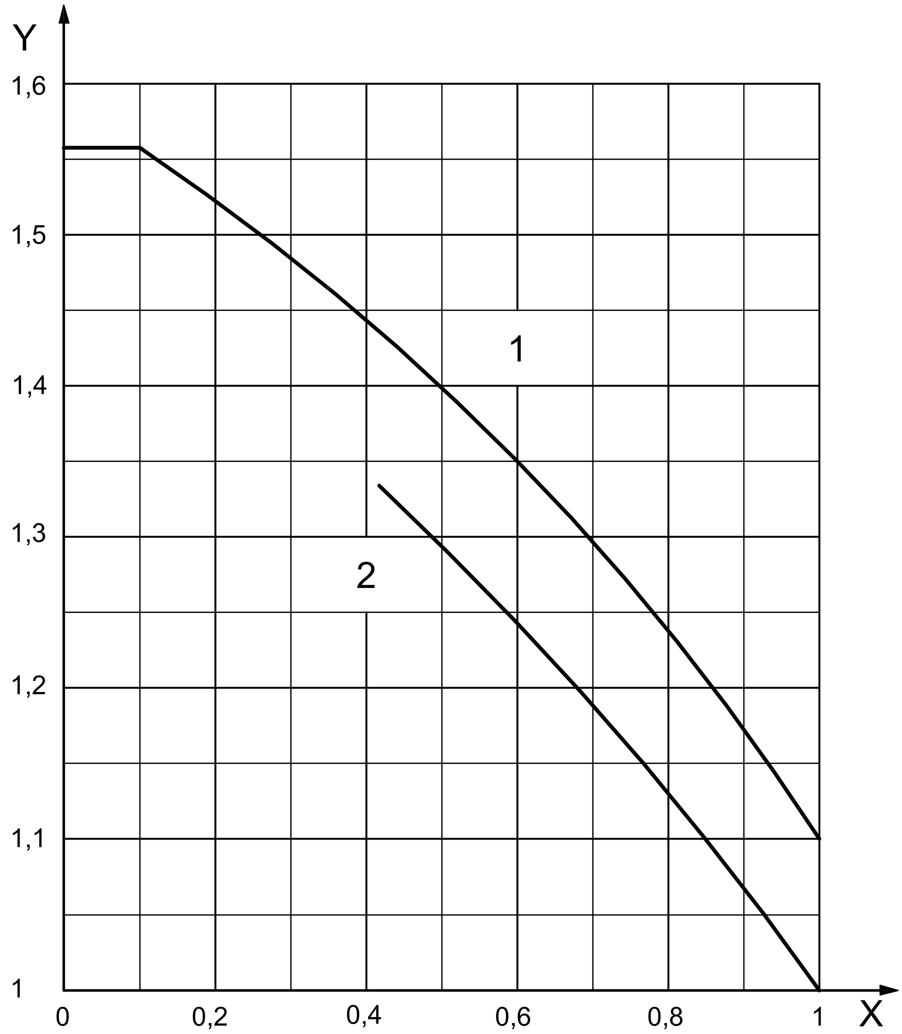
— a system of material traceability for pressure bearing parts used in the construction of the inner vessel;

— design dimensions within specified tolerances;

— necessary cleanliness of the inner vessel, associated piping and other equipment which could come in contact with the cryogenic fluid.

11.2 Cutting

Material may be cut to size and shape by thermal cutting, machining, cold shearing or other appropriate method. Thermally cut material shall be dressed back by machining or grinding.



Key

X Ratio factor, f/e

Y Design factor, C_e

1 Rectangular plate

2 Elliptical plate

Rectangular plates

f = short side of the rectangular plate

e = long side of the rectangular plate

$$C_e = \begin{cases} \displaystyle\sum_{i=1}^{4} A_i \left(\frac{f}{e}\right)^{i-1} & 0,1 < \left(\frac{f}{e}\right) \leqslant 1,0 \\ 1,562 & 0 < \left(\frac{f}{e}\right) \leqslant 0,1 \end{cases}$$

$A_1 = 1,589\ 146\ 00$

$A_2 = -0,239\ 349\ 90$

$A_3 = -0,335\ 179\ 80$

$A_4 = 0,085\ 211\ 76$

Elliptical plates

f = short side of the elliptical plate

e = long side of the elliptical plate

$$C_A = \begin{cases} \displaystyle\sum_{i=1}^{6} A_i \left(\frac{d}{D_i}\right)^{i-1} & 0 < \left(\frac{d}{D_i}\right) \leqslant 0,8 \\ \displaystyle\sum_{i=1}^{6} A_i \left(\frac{d}{f}\right)^{i-1} & 0 < \left(\frac{d}{f}\right) \leqslant 0,8 \end{cases}$$

$A_1 = 1,489\ 146\ 00$

$A_2 = -0,239\ 349\ 90$

$A_3 = -0,335\ 179\ 80$

$A_4 = 0,085\ 211\ 76$

Figure 13 — Design factor, C_E, for rectangular or elliptical flat plates

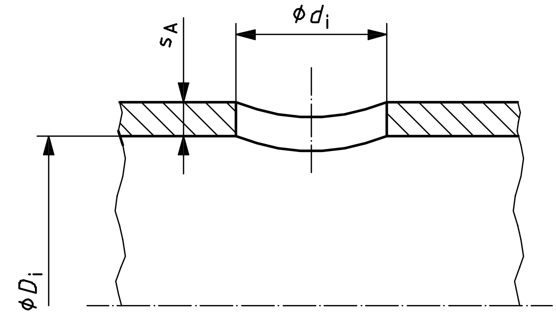


Figure 14 — Increased thickness of a cylindrical shell

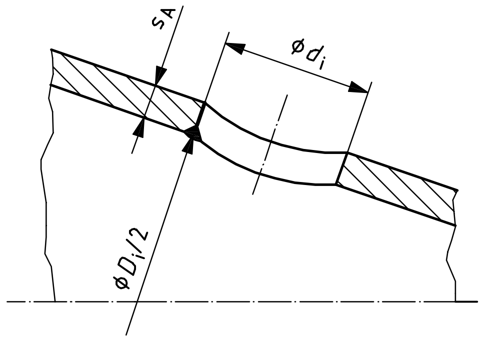


Figure 15 — Increased thickness of a conical shell

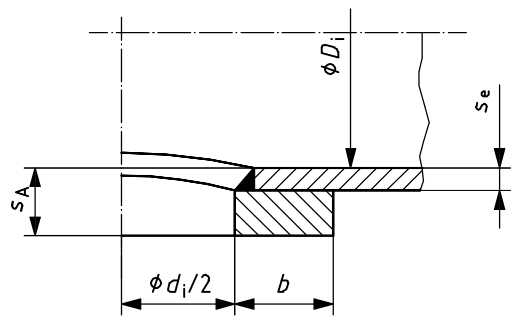


Figure 16 — Set-on reinforcement ring

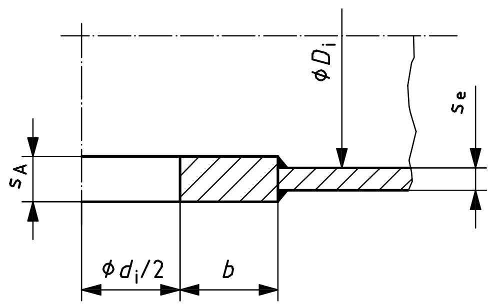


Figure 17 — Set-in reinforcement ring

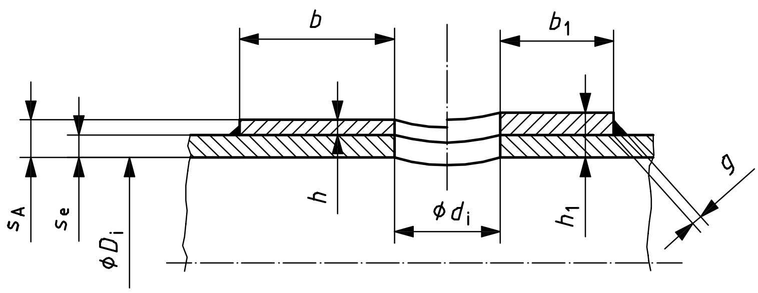


Figure 18 — Pad reinforcement

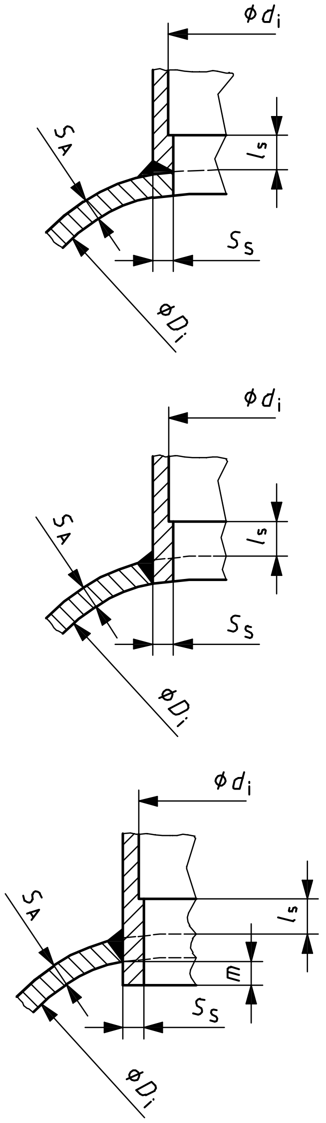


Figure 19 — Nozzle reinforcement

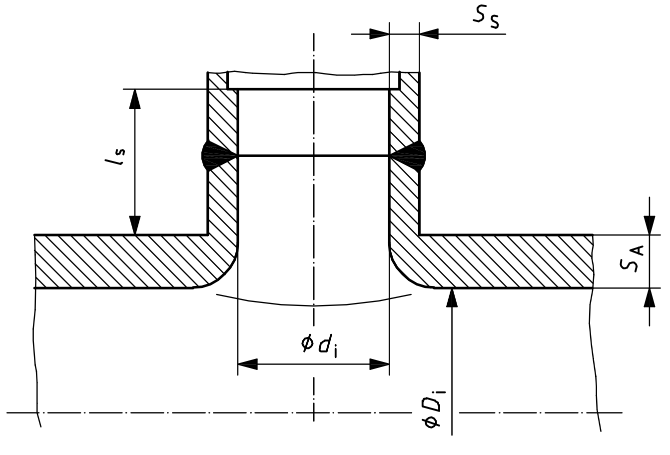
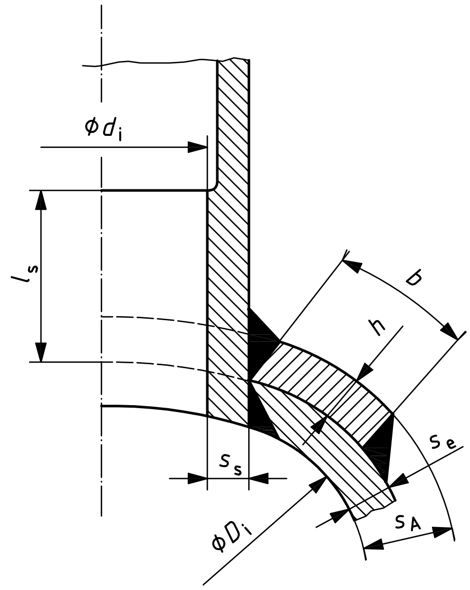
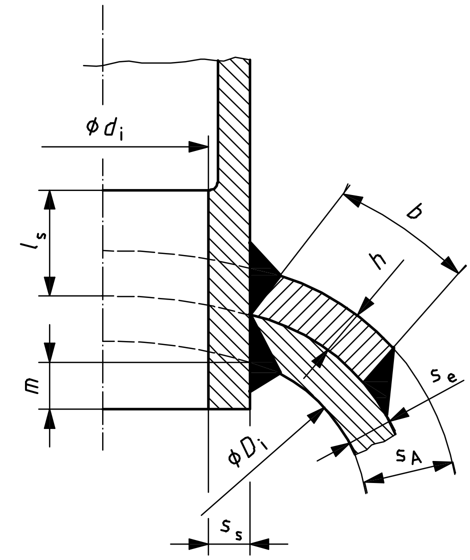


Figure 20 — Necked out opening



Type a)



Type b)

Figure 21 — Pad

11.3 Cold forming

11.3.1 Austenitic stainless steel

Heat treatment after cold forming is not required in any of the following cases.

a) For operating temperatures down to −196 °C

1) the test certificate for the base material shows an elongation at fracture A_5 of not less than 30 % and either the cold forming deformation is not more than 15 % or it is checked that the residual elongation of the material from the formed end in the maximum deformation zone is not less than 15 %;

2) the cold forming deformation is greater than or equal to 15 % and it is demonstrated that the residual elongation (elongation at fracture minus cold forming deformation) is not less than 15 %;

Cold forming deformation may be calculated according to:

$$F = 100 \ln \frac{D_{\mathrm{b(x)}}}{D_{\mathrm{e}} - 2e}$$

where

e is the thickness of the initial product;

$D_{\mathrm{b(x)}}$ is the diameter of the initial product;

D_{e} is the external diameter of the final product;

ln is the natural logarithm.

b) For operating temperatures below –196 °C, the test certificate for the base material shows an elongation at fracture A_5 of not less than 30 % and the cold forming deformation is not more than 10 %.

c) For formed heads, except for inner vessels for hydrogen or mixtures of hydrogen, the test certificate for the base material shows an elongation at fracture A_5:

— not less than 30 % in the case of wall thicknesses not more than 15 mm at any design temperatures;

— not less than 35 % in the case of wall thicknesses more than 15 mm at any design temperatures.

Where heat treatment is required this shall be carried out in accordance with the material standard.

NOTE For the hydrogen vessels to avoid failure by hydrogen embrittlement, stable stainless steel or higher ductility may be required (see ISO 21010).

11.3.2 Ferritic steel

Requirements for post-forming heat treatment are:

a) material for the outer jacket, including cold formed ends with or without joggled joints, does not require post-forming heat treatment;

b) 9 % Ni steel requires post-forming heat treatment where cold-forming deformation exceeds 5 %. Fully certified quenched and tempered or double normalised and tempered 9 % Ni steel shall be stress relieved at 560 °C to 580 °C. Forming and stress relieving may be performed in several stages. A test piece taken from the parent material that accompanies the formed part through all stages of heat treatment shall be tested after all heat treatment is complete to demonstrate that the material mechanical properties conform to the requirements of the material standard;

c) for the following ferritic steels used for the inner vessel, post-forming heat treatment is not required where the forming deformation is not more than 5 %:

1) nickel alloyed steels suitable for low temperature use;

2) carbon and carbon-manganese steels:

— where $R_m \leqslant 530$ N/mm^2

— or where $530 < R_m \leqslant 650$ N/mm^2 and $R_{0,002} \leqslant 360$ N/mm^2.

When heat treatment is required, suitable heat treatments after cold forming are normalising, normalising (double) plus tempering, quenching plus tempering or solution annealing.

Parameters given by the base material manufacturer in the test certificate shall be taken as an indication or recommendation for heat treatments except that other heat treatments may be applied if the procedure is qualified and the product or a test piece representing the product is tested after forming and heat treatment.

11.3.3 Aluminium or aluminium alloy

Cold formed ends made from aluminium or aluminium alloy do not normally require post-forming heat treatment, unless there is a risk of stress corrosion in service. Treatment shall be carried out in accordance with the material standard.

11.4 Hot forming

11.4.1 General

Forming shall be carried out in accordance with a written qualified procedure. The forming procedure shall specify the heating rate, the holding temperature, the temperature range and time for which the forming takes place and shall give details of any heat treatment to be given to the formed part.

11.4.2 Austenitic stainless steel

Material shall be heated uniformly in an appropriate atmosphere without flame impingement, to a temperature not exceeding the recommended hot forming temperature of the material. When forming is carried out after the temperature of the material has fallen below 900 °C the requirements of 11.3.1 apply.

11.4.3 Ferritic steel

Requirements for post-forming heat treatment are:

a) 9 % Ni steel that is hot formed shall be double normalised and tempered or quenched and tempered in accordance with the material standard to establish the material properties specified therein. Test piece(s) shall be provided and tested in accordance with the material standard;

b) ferritic steel that is hot formed shall be heat treated in accordance with the material standard to establish the material properties specified therein:

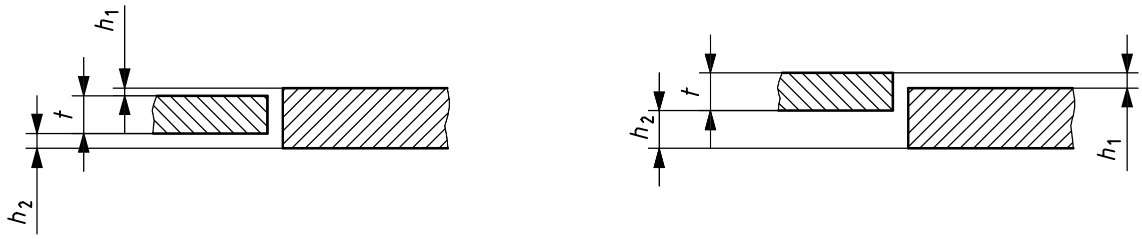
   — air quenched steels shall be tempered subsequently;

   — test pieces shall be provided and tested in accordance with the material standard;

   — for normalised steels a post-forming heat treatment is not necessary if the hot forming is done within the temperature range specified in the material standard; further test pieces are not required.

11.4.4 Aluminium or aluminium alloy

Post-forming heat treatment may be omitted if evidence in the form of a procedure qualification can be provided showing that the elongation at fracture A_5 of the formed material is not less than 10 %.



Nomenclature

h, h_1, h_2 = surface misalignments

t = thickness of the thinner plate

e = distance from the surface of the thicker plate to the centreline of the thinner plate

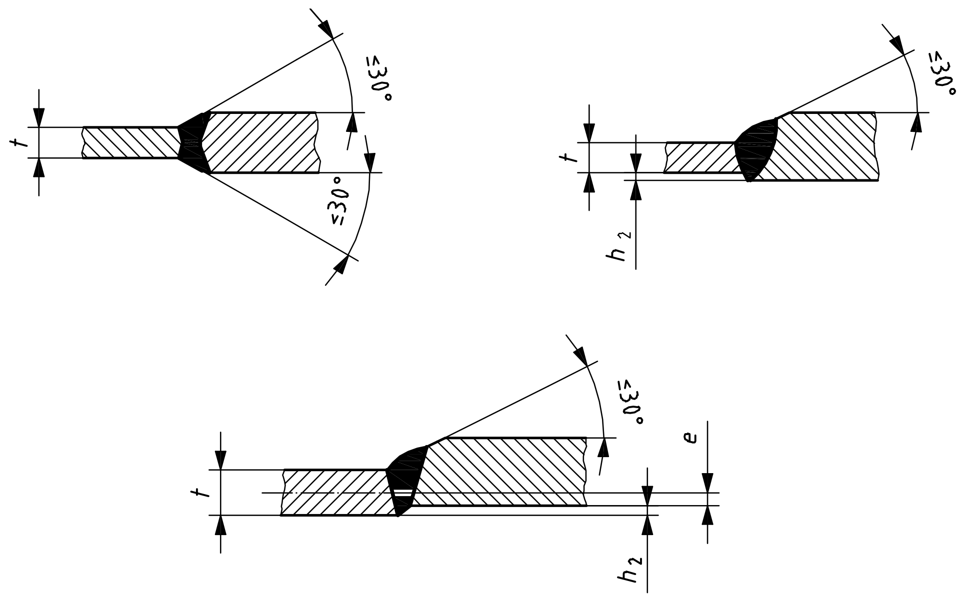
For longitudinal seams:

$h_1 \leqslant 0{,}15\ t$ and $h_2 \leqslant 0{,}15\ t$

For circumferential seams:

$h_1 \leqslant 0{,}25\ t$ and $h_2 \leqslant 0{,}25\ t$

a) Seams which do not require a taper



For longitudinal seams:

$h \leqslant 0{,}15\ t$ and

$$e = \frac{t}{2} - h \geqslant 0{,}35\ t$$

For circumferential seams:

$h_2 \leqslant 0{,}25\ t$ and

$$e = \frac{t}{2} - h \geqslant 0{,}25\ t$$

b) Seams which do require a taper

Figure 22 — Plate alignment

11.5 Manufacturing tolerances

11.5.1 Plate alignment

Except where a tapered transition is provided, misalignment of the surfaces of adjacent plates at welded seams shall be:

— for longitudinal seams, not more than 25 % of the thickness of the thinner plate up to a maximum of 3 mm;

— for circumferential seams, not more than 25 % of the thickness of the thinner plate up to a maximum of 3 mm.

Where a taper is provided between the surfaces, this shall have a slope of not more than 18,4°. The taper may include the width of the weld, the lower surface being built up with added weld metal if necessary. Where material is removed from a plate to provide a taper, the thickness of either plate shall not be reduced below that required for the design.

The distance between either surface of the thicker plate and the centre line of the thinner plate of tapered seams shall be:

— for longitudinal seams, not less than 35 % of the thickness of the thinner plate;

— for circumferential seams, not less than 35 % of the thickness of the thinner plate.

In no case shall the surface of any plate lie between the centre lines of the two plates.

These requirements are illustrated in Figure 22.

11.5.2 Thickness

The thickness of the vessel shall not be less than the design thickness. This shall be taken as the thickness of the vessel after manufacture and any variations in thickness shall be gradual.

11.5.3 Dished ends

The knuckle radius shall not be less than specified and any variation of the crown radius shall not be abrupt and shall adhere to the following tolerances:

$$^{+0{,}625}_{-1{,}25}\ \%$$

11.5.4 Cylinders

11.5.4.1 The actual circumference shall not deviate from the circumference calculated from the specified diameter by more than 1,5 %.

11.5.4.2 The out-of-roundness, u, calculated from the expression

$$\text{out-of-roundness, } u = \frac{200\left(D_{\text{max}} - D_{\text{min}}\right)}{D_{\text{max}} + D_{\text{min}}}\%$$

shall be not more than the values shown in Table 2.

Table 2 — Permitted out-of-roundness

Wall thickness to diameter ratio	Permitted out-of-roundness for	
	internal pressure	external pressure
$s/D \leqslant 0{,}01$	2,0 %	1,5 %
$s/D > 0{,}01$	1,5 %	1,5 %

The determination of the out-of-roundness need not consider the elastic deformation due to the dead-weight of the pressure vessel. At nozzle positions, a greater out-of-roundness may be permitted if it can be justified by calculation or strain gauge measurement. Single dents or knuckles shall be within the tolerances. Dents shall be smooth and their depth which is the deviation from the generatrix of the shell shall not exceed 1 % of their length or 2 % of their width respectively. Greater dents and knuckles are permissible provided they have been proven admissible by calculation or by strain measurements.

Irregularities in profile (checked by a 20 ° gauge) shall not exceed 2 % of the gauge length. This maximum value may be increased by 25 % if the length of the irregularities does not exceed one quarter of the length of the shell part between two circumferential seams with a maximum of 1 m. Greater irregularities require proof by calculation or strain-gauge measurement that the stresses are permissible.

Furthermore, where irregularity in the profile occurs at the welded seam and is associated with “flats” adjacent to the weld, the irregularity in profile or “peaking” shall not exceed the values given in Table 3.

A conservative method of measurement (covering peaking and ovality) shall be by means of a 20 ° profile gauge (or template).

The use of such a profile gauge is illustrated in Figure 23. Two readings shall be taken, P_1 and P_2 on each side of the seam, at any particular location, the maximum peaking is taken as being equivalent to 0,25 $(P_1 + P_2)$.

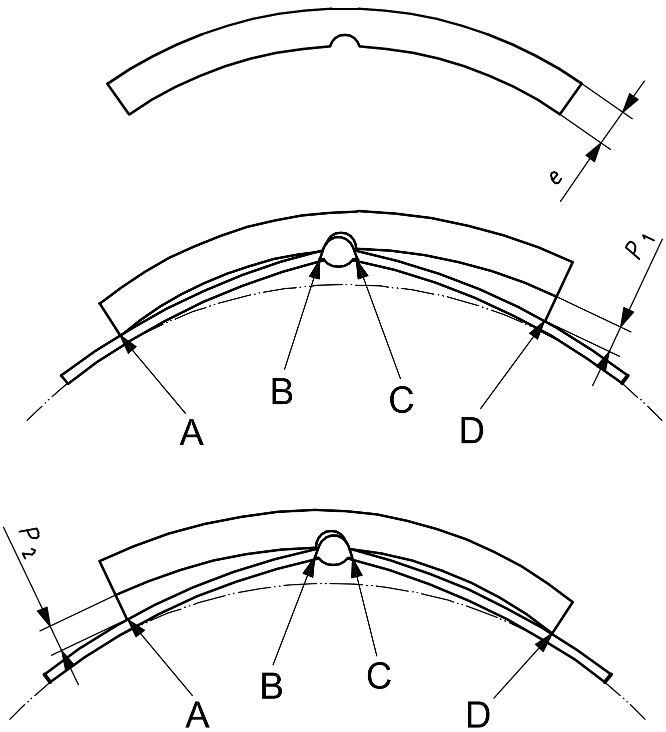


Figure 23 — Gauge details

Measurements should be taken at approximately 250 mm intervals on longitudinal seams to determine the location with the maximum peaking value. Use of other types of gauges such as bridge gauges or needle gauges is not prohibited. The maximum peaking value permitted is given in Table 3.

Table 3 — Maximum permitted peaking

Dimensions in millimetres

Vessel ratio wall thickness *s* to diameter *D*	Maximum permitted peaking
$s/D \leqslant 0{,}025$	5
$s/D > 0{,}025$	10

For all ratios, a maximum permitted peaking is e.

For cylinders subject to external pressure and where the circumference has a flattened portion, it shall be demonstrated that the shell has sufficient strength to avoid plastic deformation where the depth of flattening is more than 0,4 % of the outside diameter of the cylinder. The depth of flattening shall be measured as a deviation from the normal curvature or from the line of the cylindrical shell. Adequate strength may be determined by calculation in accordance with the formula given in L.1.1.2, using a value of u determined as follows:

$$u = \frac{400}{D_\mathrm{a}} q$$

where

q is the depth of flattening, in millimetres;

D_a is the external diameter of the cylinder, in millimetres.

11.5.4.3 Departure of the cylinder axis from a straight line shall be not more than 0,5 % of the cylindrical length, except where required by the design.

11.6 Welding

11.6.1 General

This part of ISO 21009 requires that the welding method be appropriate and be carried out by qualified welders and/or operators, that the materials be compatible and that there is verification by a welding procedure test.

11.6.2 Qualification

Welding procedures shall be approved in accordance with ISO 15614-1, ISO 15614-2 or ISO 15613 as applicable.

Welders and welding operators shall be qualified in accordance with ISO 9606-1 or ISO 9606-2 or ISO 14732 as applicable.

11.6.3 Temporary attachments

Temporary attachments welded to pressure bearing parts shall be kept to a practical minimum.

Temporary attachments welded directly to pressure bearing parts shall be compatible with the immediately adjacent material.

It is permissible to weld dissimilar metal attachments to intermediate components, such as pads, which are connected permanently to the pressure containing part. Compatible welding materials shall be used for dissimilar metal joints.

Temporary attachments shall be removed from the inner vessel prior to the first pressurization. The removal technique shall avoid impairing the integrity of the inner vessel and shall be by chipping or grinding. Any rectification necessary by welding of damaged regions shall be undertaken in accordance with an approved welding procedure.

The area of the inner vessel from where the temporary attachments have been removed shall be dressed smooth and examined by appropriate non-destructive testing.

Any attachments on the outer jackets may be removed by thermal cutting as well as by the methods described above.

11.6.4 Welded joints

11.6.4.1 Some specific weld details appropriate to vessels conforming to this part of ISO 21009 are given in Annex F. These details show sound and currently accepted practice. It is not intended that these are mandatory nor should they restrict the development of welding technology in any way.

The manufacturer, in selecting an appropriate weld detail, shall consider:

— the method of manufacture;

— the service conditions;

— the ability to carry out necessary non-destructive testing.

Weld details may be used provided their suitability is proven by procedure approval according to ISO 15614-1, ISO 15614-2 or ISO 15613 as applicable.

To avoid sub-standard welding of ferritic steels excess residual magnetism shall be avoided.

11.6.4.2 Where any part of a vessel is made in two or more courses, the longitudinal weld seams of adjacent courses shall be staggered. A minimum of 100 mm is recommended. Joggled joints may be used in stainless steels for circumferential welds only and plate thickness up to 8 mm. Backing strips may be used for circumferential welds only with no thickness restriction. When forming the joggled joints, reduction in toughness shall be considered for low temperature.

11.6.4.3 As the mechanical characteristics of work-hardened austenitic stainless steels can be adversely affected if the material is not welded properly, the additional requirements below shall be applied:

— the heat input during welding shall be not more than 1,5 kJ/mm per bead to be verified in the procedure qualification test;

— the material shall cool down to a temperature of not more than 200 °C between passes;

— the material shall not be heat treated after welding.

If post heat treatment is required it shall be demonstrated that the required material (mechanical, corrosion resistance, etc.) will not be adversely affected.

See also B.2.7, B.2.8, B.2.10 and B.2.11.

11.7 Non-welded permanent joints

Where non-welded joints are made between metallic materials and/or non-metallic materials, procedures shall be established in a manner similar to that used in establishing welding procedures, and these procedures shall be followed for all joints. Similarly, operators shall be qualified in such procedures and only qualified personnel shall then carry out these procedures.

Brazing procedures and brazing approvals can be found in EN 13133 and EN 13134 or ASME, Section VIII, Division 1 or any equivalent standard.

12 Inspection and testing

12.1 Quality plan

A quality plan shall include as a minimum, the inspection and testing stages listed in 12.1.1.

12.1.1 Inspection stages during manufacture of an inner vessel

The following inspection stages shall be conducted during the manufacture of an inner vessel:

- verification of material test certificates and correlation with materials;
- approval of weld procedure qualification records;
- approval of welders qualification records;
- examination of material cut edges;
- examination of set up of seams for welding including dimensional check;
- examination of weld preparations, tack welds;
- visual examination of welds;
- verification of non-destructive testing;
- testing production-control test-plates for welds and, where required, for formed parts after heat treatment;
- verification of cleaning of inside surface of vessel;
- examination of completed vessel including dimensional check;
- pressure test.

12.1.2 Additional inspection stages during manufacture of a static cryogenic vessel

The following inspection stages shall be conducted during the manufacture of static cryogenic vessels:

- verification of cleanliness and dryness of static cryogenic vessel;
- visual examination of welds not covered by 12.1.1;
- leak proofness tests ensuring the integrity of vacuum, and leak testing of external piping when it is connected to the inner vessel;
- ensure integrity of vacuum;
- leak test of external piping;
- check documentation and installation of pressure relief device(s);
- check installation of vacuum space relief device;
- check name plate and any other specified markings;
- examination of completed vessel including dimensional check.

12.2 Production control test plates

12.2.1 Requirements

Production control test plates shall be produced and tested for the inner vessel as follows:

a) one test plate per vessel for each welding procedure on longitudinal joints except as specified in b);

b) after 10 sequential test plates to the same procedure have successfully passed the tests, testing may be reduced to one test plate per 50 m of longitudinal joint for 9 % Ni and ferritic steels and to one test plate per 130 m for other metals.

Production control test plates are not required for the outer jacket.

The results of the tests shall be as follows:

— weld tensile test (T): R_{et}, R_m and A_5 of the test specimens shall normally not be less than the corresponding specified minimum values for the parent metal, or the agreed values of the welding procedure approved;

— impact test (IW, IH): this test shall be performed in accordance with the appropriate part of ISO 21028;

— bend test (BF, BR, BS): the testing and the test requirements shall comply with 7.4.2 of ISO 15614-1 for steels and with 7.4.2 of ISO 15614-2 for aluminium and its alloys;

— macro etch (Ma): the macro etch shall show sound build-up of beads and sound penetration.

12.2.2 Extent of testing

The number and type of test specimens to be taken from the test plate is dependent on material and thickness and shall be in accordance with the requirements in Tables 5 and 6 for the particular material and thickness applicable.

NOTE The symbols for Tables 5 and 6 are given in Table 4.

The test plate shall be of sufficient size to allow for the required specimens including an allowance for retests.

Table 4 — Test specimens

Designation	Symbol
Face bend test to ISO 5173	BF
Root bend test to ISO 5173	BR
Side bend test to ISO 5173	BS
Tensile test to ISO 4136	T
Impact test; weld deposit to ISO 9016	IW
Impact test; HAZ to ISO 9016	IH
Macro etch	Ma

Table 5 — Testing of production test plates for steels

Group	e in mm	Test specimens
Fine grain steels normalised or thermo mechanically treated	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e \leqslant 35$	3 IW, 3 IH, 1 T, 1 Ma
Ni steels up to 9 % Ni	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 3 IH, 1 T, 1 Ma
Austenitic stainless steels	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 1 T, 1 Ma

Table 6 — Testing of production test plates for aluminium

Group	e in mm	Test specimens
Pure aluminium and aluminium with up to 1,5 % impurities or alloy content	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	2 BS, 1 T, 1 Ma

12.3 Non-destructive testing

12.3.1 General

Non-destructive testing personnel shall be qualified for the duties according to ISO 9712.

Radiographic examination shall be carried out in accordance with ISO 17636. Radioscopy may also be used and shall be carried out in accordance with EN 13068-3 or ASME Section V.

Non-destructive testing for welding imperfections is not required on the outer jacket of static cryogenic vessels.

12.3.2 Extent of examination for surface imperfections

Visual examination (if necessary aided by 5 x magnification lens) shall be carried out on all weld deposits (see Table 8). If any doubt arises, this examination shall be supplemented by surface crack detection.

Arc strike contact points and areas from which temporary attachments have been removed shall be ground smooth and subjected to surface crack detection.

12.3.3 Extent of examination for weld imperfections

Examination of the inner vessel for inner vessel weld seams shall be by radiographic examination unless a special case is made to justify ultra-sonic or other methods. The extent of examination of main seams on the inner vessel shall be in accordance with Table 7. See Table 9 for acceptance levels.

When hemispherical ends without a straight flange are welded together or to a cylinder, the weld shall be tested as a longitudinal weld. Any welds within a hemispherical end shall also be tested as longitudinal welds.

Table 7 — Extent of radiographic examination for welded seams

Weld joint factor	Radiographic examination		
	Longitudinal seams	T junctions	Circumferential seams
1,0	100 % [a]	100 %	25 % [a]
0,85	2 % [b]	10 % or minimum of 1 joint per vessel	2 % [b]

NOTE 1 When a butt weld occurs less than 3 times the weld thickness (min. 50 mm) from a nozzle cut out, it is necessary to take additional radiographic film(s) local to the nozzle where the original film(s) have not included this location.

NOTE 2 For additional requirements for 9 % Ni steel see Annex B.

NOTE 3 For corner joints of cones and areas of high bending stress treat the circumferential seam as a longitudinal seam with joint factor 1.

NOTE 4 Additional testing might be required when pneumatic proof testing is used.

[a] The level of radiographic examination may be reduced to 10 % of each seam of each vessel if 25 vessels have been successfully built using the same welding procedure, provided:

— the welding procedure is unaltered;
— the welding experience has been retained in the workshop;
— the testing methods are the same;
— the results of non-destructive testing have not revealed any unacceptable systematic defects.

[b] The 2 % level of radiographic examination may be carried out on a batch of vessels provided they are built within a period of 3 months. The number of vessels included in a test batch should not be more than 5. The 2 % should not be included in the film length of the T junctions examined.

12.3.4 Acceptance levels

12.3.4.1 Acceptance levels for surface imperfections

Table 8 shows the acceptance criteria for surface imperfections.

Table 8 — Acceptance levels for surface imperfections

Imperfection	Limit for acceptable imperfection
Lack of penetration	Not permitted
Undercut	Where the thickness is less than 3 mm no visible undercut is permitted. Where the thickness is not less than 3 mm, slight and intermittent undercut is acceptable, provided that it is not sharp and is not more than 0,5 mm.
Shrinkage groove	As undercut
Root concavity	As undercut
Excessive penetration	Where the thickness is less than 5 mm, excessive penetration shall be not more than 2 mm. Where the thickness is not less than 5 mm, excessive penetration shall not be more than 3 mm.
Excess weld material	Where the thickness is less than 5 mm, excess weld metal shall not be greater than 2 mm and the weld shall blend smoothly. Where the thickness is 5 mm or greater, the maximum excess weld metal shall not exceed 3 mm and the weld shall blend smoothly.
Irregular surface Sagging Incompletely filled groove Irregular width Poor restart	Reinforcement to be of continuous and regular shape with complete filling of groove.

Table 8 (*continued*)

Imperfection	Limit for acceptable imperfection
Overlap	Not permitted
Linear misalignment	See 11.5.1
Arc strike Spatter Tungsten spatter Torn surface Grinding mark Chipping mark	Grind smooth, acceptable subject to thickness measurement and surface crack detection test.
Surface cracks	Not permitted

12.3.4.2 Acceptance levels for weld imperfections of the inner vessels

Table 9 shows the acceptance criteria for internal weld imperfections detected by radiographic examination.

Table 9 — Acceptance levels for weld imperfections of the inner vessels

Imperfection	Limit for acceptable imperfection
Cracks and lack of sidewall fusion	Not permitted
Incomplete root fusion	Not permitted
Flat root concavity	Acceptable
Inclusions (including oxide in aluminium welds). Strings of pores, worm holes parallel to the surface and strings of tungsten.	30 % of thickness The maximum length shall be the greater of 7 mm or 2/3 t.
Interrun fusion defects and root defects in multipass weld	As inclusions
Multiple in-line inclusions	Collectively in any radiographed length equal to six times the material thickness, the total length of inclusion shall not be greater than the material thickness.
Area of general porosity visible on a film	Acceptable if less than 2 % of projected area of weld.
Individual pores	Acceptable if diameter is less than 25 % of the thickness with a maximum of 4 mm.
Worm holes perpendicular to the surface	Where the thickness is less than 10 mm, worm holes are not permitted. Where the thickness is not less than 10 mm, isolated examples are acceptable provided the depth is estimated to be not more than 30 % of the thickness.
Tungsten inclusions	Where the thickness is less than 12 mm, tungsten inclusions are acceptable provided the length is not more than 3 mm. Where the thickness is not less than 12 mm, tungsten inclusions are acceptable provided the length is not more than 25 % of the thickness.

12.3.4.3 Extent of examination of non-welded joints

Where non-welded joints are used between metallic materials and/or non-metallic materials, the quality plan referred to in 12.1 shall include reference to an adequate technical specification.

This technical specification shall include the description of the requirements for inspection and testing, together with the criteria necessary to allow for the repair of any imperfections.

12.4 Rectification

12.4.1 General

Although unacceptable volumetric or surface imperfections may be repaired by removing the imperfections and rewelding, 100 % of all repaired welds shall be examined to the original acceptance standards.

12.4.2 Manually welded seams

When repairs to welds are carried out as a result of radiographic examination which is less than 100 %, in addition to the full radiography of repair, a radiographic image using a film (200 mm) shall be taken of either side of the repair to ensure the imperfection was isolated and not systematic. Where the imperfections are systematic and characterised by recurrence of the same imperfection, the extent of examination shall be increased to 100 % until the cause of the imperfections has been found and eliminated.

12.4.3 Seams produced using automatic welding processes

If any unacceptable imperfections are found by radiographic examination, all main weld seams shall be 100 % radiographically examined on all vessels produced with the same welding machine and welding procedure from the start of the production period or from the last accepted non-destructive test.

12.5 Pressure testing

12.5.1 Every inner vessel shall be subjected to a pressure test and its leak tightness shall be demonstrated. This leak tightness may be demonstrated during the establishment of the vacuum or by a separate leak test at pressures up to 90 % of design pressure.

The test pressure shall not be less than the higher of:

$H\,(p_\mathrm{S} + 1)$ bar hydrostatic or 1,25 $(p_\mathrm{S} + 1)$ bar pneumatic

NOTE 1 H is equal to 1,43 in Europe and to 1,3 in North America.

$1{,}25\left(p_\mathrm{S} + p_\mathrm{L} + 1\right)\dfrac{K_{20}}{K_t}$ bar, considered for each element of the vessel e.g. shell, courses, head.

NOTE 2 If the inner vessel is enclosed in vacuum (less than 1 mm Hg pressure reading) during pressure testing, the test pressure can be calculated as $P_T = H\left(P_\mathrm{S} + 1\right) - 1$ bar.

NOTE 3 When cold properties are used, see Annex E where K_design is used instead of K_t.

Where the test is carried out hydraulically, the pressure shall be raised gradually to the test pressure, holding it there for 30 min. Then the pressure shall be reduced to the design pressure so that a visual examination of all surfaces and joints can be made. The vessel shall not show any sign of gross plastic deformation or leakage. The test may be carried out pneumatically on a similar basis. As pneumatic testing employs substantially greater stored energy than hydraulic testing, it shall normally be carried out where adequate facilities and procedures are employed to assure the safety of inspectors, employees and the public.

12.5.2 Vessels which have been repaired subsequent to the pressure test shall be re-subjected to the specified pressure test after completion of the repairs.

12.5.3 Where austenitic stainless steel comes into contact with water, the chloride content of the water and time of exposure shall be controlled so as to avoid stress corrosion cracking.

12.5.4 The piping system shall be subjected to a pressure test at a pressure not less than 1,1 times the design pressure [10.2.3.7 d)] for the appropriate section of pipework. It is not necessary to strength-test mechanical joints and fittings that have demonstrated satisfactory in-service experience.

13 Marking and labelling

The static cryogenic vessel shall bear the following markings in clearly legible and durable characters:

a) on the inner vessel:

1) name and address, or other means of identification of the manufacturer of the inner vessel;

2) serial number of the inner vessel;

3) mark confirming successful final acceptance tests of the inner vessel;

b) on the outer jacket:

1) "ISO 21009" to show that the static cryogenic vessel is in conformity with this part of ISO 21009;

2) name and address, or other means of identification of the manufacturer of the static cryogenic vessel;

3) serial number of the inner vessel;

4) maximum allowable working pressure (p_S in bar) of the static cryogenic vessel;

5) test pressure (p_T in bar) of the static cryogenic vessel;

6) volume of the inner vessel (in litres);

7) year of manufacture;

8) date (year followed by the month) of the final test;

9) the identification of those cryogenic fluids for which the static cryogenic vessel is approved (chemical symbols may be used);

10) minimum design temperature of the jacket if lower than –20 °C;

11) optional marking: maximum gross weight of the product to be contained (this marking can be found in the instructions of use);

12) information marked on the inner vessel [see a)] shall be repeated on the data plate, mounted or permanently attached to the outer jacket;

c) prior to filling:

1) a flow sheet with operation instructions;

2) an unshortened identification (see 3.4) of the fluid which is stored in the static cryogenic vessel;

3) danger label(s) associated with the fluid;

4) name of the fluid supplier.

The marks as described under a) and b) shall be permanently affixed, e.g. stamped, either on a reinforced part of the static cryogenic vessel, or on a data plate.

The technique employed for marking and attaching shall not adversely affect the integrity of the static cryogenic vessel.

Marks described under c) can either be stamped or indicated on a durable information disc or label attached to the static cryogenic vessel or indicated in an adherent and clearly visible manner such as painting or by an equivalent process.

Additional markings are permitted, provided that they do not obscure or create confusion with specified markings called for in this International Standard.

14 Final assessment

When all necessary tests specified in Clause 12 are carried out successfully and the documentation (see 3.5) is completed, the final assessment is terminated.

15 Periodic inspection

Appropriate periodic inspection procedures are described in ISO/DIS 21009-2.

Annex A
(normative)

Elastic stress analysis

A.1 General

This annex provides rules to be followed if an elastic stress analysis is used to evaluate components of a static cryogenic vessel for operating conditions. The loads to be considered are those defined in 10.2.3.

A.4 and A.5 give alternative criteria for demonstrating the acceptability of design on the basis of elastic analysis. The criteria in A.5 apply only to local stresses in the vicinity of attachments, supports, nozzles, etc.

The calculated stresses in the area under consideration are grouped into the following stress categories:

— general primary membrane stress;

— local primary membrane stress;

— primary bending stress;

— secondary stress.

Stress intensities, f_m, f_L, f_b, and f_g, can be determined from the principle stresses, f_1, f_2, and f_3, in each category using the maximum shear stress theory of failure, see A.2.1.

The stress intensities determined in this way shall be less than the allowable values given in A.3 and A.4 or A.5.

Peak stresses need not be considered as they are only relevant when evaluating designs for cyclic service. Static cryogenic vessels within the scope of this International Standard are not considered to be in cyclic service.

Figure A.1 and Table A.1 have been included as guidance, where A.4 is used for evaluation, in establishing stress categories for some typical cases and stress intensity limits for combinations of stress categories. There will be instances when references to definitions of stresses will be necessary to classify a specific stress condition to a stress category. A.4.5 explains the reason for separating them into two categories “general” and “secondary” in the case of thermal stresses.

A.2 Terminology

A.2.1 Stress intensity

The stress intensity is twice the maximum shear stress, i.e. the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tension stresses are considered positive and compression stresses are considered negative.

The principal stresses, f_1, and f_2, acting tangentially to the surface at the point under consideration should be calculated from the following equations:

$$f_1 = 0{,}5 \times \left(\sigma_1 + \sigma_2 + \sqrt{(\sigma_1 - \sigma_2)^2 + 4 \times \tau^2} \right)$$

$$f_2 = 0{,}5 \times \left(\sigma_1 + \sigma_2 - \sqrt{(\sigma_1 - \sigma_2)^2 + 4 \times \tau^2} \right)$$

where:

σ_1 is the circumferential stress;

σ_2 is the meridional stress (longitudinal in a cylindrical shell);

τ is the shear stress.

A.2.2 Gross structural discontinuity

A gross structural discontinuity is a source of stress or strain intensification that affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole.

Examples of gross structural discontinuities are:

EXAMPLE 1 End-to-shell junctions.

EXAMPLE 2 Junctions between shells of different diameters or thicknesses.

EXAMPLE 3 Nozzles.

A.2.3 Local structural discontinuity

A local structural discontinuity is a source of stress or strain intensification that affects a relatively small volume of material and does not have a significant effect on the overall stress or strain pattern or on the structure as a whole.

EXAMPLE 1 Small fillet radii.

EXAMPLE 2 Small attachments.

EXAMPLE 3 Partial penetration welds.

A.2.4 Normal stress

The normal stress is the component of stress normal to the plane of reference; this is also referred to as direct stress.

Usually, the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to be made up in turn of two components, one of which is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration, and the other of which varies with the location across the thickness.

A.2.5 Shear stress

The shear stress is the component of stress acting in the plane of reference.

A.2.6 Membrane stress

The membrane stress is the component of stress that is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.

A.2.7 Primary stress

A primary stress is a stress produced by mechanical loadings only and so distributed in the structure so that no redistribution of load occurs as a result of yielding. A normal stress, or a shear stress developed by the imposed loading, is necessary to satisfy the simple laws of equilibrium of external and internal forces and moments. The basic characteristic of this stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure, or at least in gross distortion. A thermal stress is not classified as a primary stress. Primary stress is divided into “general” and “local” categories. The local primary stress is defined in A.2.8.

Examples of general primary stress are:

EXAMPLE 1 The stress in a cylindrical or a spherical shell due to internal pressure or to distributed live loads.

EXAMPLE 2 The bending stress in the central portion of a flat head due to pressure.

A.2.8 Primary local membrane stress

Cases arise in which a membrane stress produced by pressure or other mechanical loading, and associated with a primary and/or a discontinuity effect, produces excessive distortion in the transfer of load to other portions of the structure.

Conservatism requires that such a stress be classified as a primary local membrane stress even though it has some characteristics of a secondary stress. A stressed region may be considered as local if the distance over which the stress intensity exceeds 110 % of the allowable general primary membrane stress, does not extend in the meridional direction more than $0{,}5\sqrt{Rs}$, and if it is not closer in the meridional direction than $2{,}5\sqrt{Rs}$ to another region where the limits of general primary membrane stress are exceeded. Where R and s are respectively the radius and thickness of the component.

An example of a primary local stress is the membrane stress in a shell produced by external load and moment at a permanent support or at a nozzle connection.

A.2.9 Secondary stress

A secondary stress is a normal stress or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected.

An example of secondary stress is the bending stress at a gross structural discontinuity.

A.2.10 Peak stress

The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack. A stress that is not highly localised falls into this category if it is of a type that cannot cause noticeable distortion.

EXAMPLE 1 The surface stresses in the wall of a vessel or pipe produced by thermal shock.

EXAMPLE 2 The stress at a local structural discontinuity.

A.3 Limit for longitudinal compressive general membrane stress

The longitudinal compressive stress shall not exceed 0,93 ΔK for ferritic steels and 0,73 ΔK for austenitic stainless steel and aluminium alloys.

Where Δ is obtained from Figure A.2 or A.3 in terms of p_e/p_{yss} and where:

$$p_e = 1{,}21E\left(\frac{S}{R}\right)^2$$

and

$$p_{yss} = 1{,}86K\left(\frac{S}{R}\right) \quad \text{for ferritic steel}$$

and

$$p_{yss} = 1{,}46K\left(\frac{S}{R}\right) \quad \text{for austenitic stainless steel and aluminium alloys.}$$

A.4 Stress categories and stress limits for general application

A.4.1 General

A calculated stress depending upon the type of loading and/or the distribution of such stress will fall within one of the five basic stress categories defined in A.4.2 to A.4.6. For each category, a stress intensity value is derived for a specific condition of design. To satisfy the analysis this stress intensity shall fall within the limit detailed for each category.

A.4.2 General primary membrane stress category

The stresses falling within the general primary membrane stress category are those defined as general primary stresses in A.2.7 and are produced by pressure and other mechanical loads, but excluding all secondary and peak stresses. The value of the membrane stress intensity is obtained by averaging these stresses across the thickness of the section under consideration. The limiting value of this stress intensity, f_m, is the allowable stress value 2 K/3.

A.4.3 Local primary membrane stress category

The stresses falling within the local primary membrane stress category are those defined in A.2.8 and are produced by pressure and other mechanical loads, but excluding all thermal and peak stresses. The stress intensity, f_L, is the average value of these stresses across the thickness of the section under consideration and is limited to K.

A.4.4 General or local primary membrane plus primary bending stress category

The stresses falling within the general or local primary membrane plus primary bending stress category are those defined in A.2.7 but the stress intensity value, f_b, $(f_m + f_b)$ or $(f_L + f_b)$, is the highest value of those stresses acting across the section under consideration excluding secondary and peak stresses. f_b is the primary bending stress intensity, which means the component of primary stress proportional to the distance from centroid of solid section. The stress intensity, f_b, $(f_m + f_b)$ or $(f_L + f_b)$, is not to exceed K.

A.4.5 Primary plus secondary stress category

The stresses falling within the primary plus secondary stress category are those defined in A.2.7 plus those of A.2.9 produced by pressure, other mechanical loads and general thermal effects. The effects of gross structural discontinuities, but not of local structural discontinuities (stress concentrations), should be included. The stress intensity value, $(f_m + f_b + f_g)$ or $(f_L + f_b + f_g)$, is the highest value of these stresses acting across the section under consideration and shall be limited to 2 K.

A.4.6 Thermal stress

Thermal stress is a self-balancing stress produced by a non-uniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature.

For the purpose of establishing allowable stresses, the following two types of thermal stress are recognised, depending on the volume or area in which distortion takes place:

a) general thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds 2 K the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. This type is therefore classified as secondary stress in Table A.1 and Figure A.1;

Examples of general thermal stress are:

EXAMPLE 1 The stress produced by an axial thermal gradient in a cylindrical shell.

EXAMPLE 2 The stress produced by the temperature difference between a nozzle and the shell to which it is attached.

b) local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses are only considered from the fatigue standpoint.

EXAMPLE 3 A small cold spot in a vessel wall.

A.5 Specific criteria, stress categories and stress limits for limited application

A.5.1 General

The criteria and stress limits for particular stress categories for elastically calculated stresses adjacent to attachments and supports and to nozzles and openings which are subject to the combined effects of pressure and externally applied loads are specified in A.5.2. to A.5.4.

The minimum separation in the meridional direction between adjacent loaded attachments, pads, nozzles or openings or other stress concentrating features shall not be less than $2{,}5\sqrt{Rs}$.

R and s are respectively the radius and thickness of the component. The criteria of A.2.8 are not applicable to this section.

If design acceptability is demonstrated by A.5 then the use of A.4 is not required.

A.5.2 Attachments and supports

The dimension in the circumferential direction of the loaded area shall not exceed one third of the shell circumference. The stresses adjacent to the loaded area due to pressure acting in the shell may be taken as the shell pressure stresses without any concentrating effects due to the attachment.

Under the design combined load the following stress limits apply:

— the primary membrane stress intensity shall not exceed 0,8 K;

— the stress intensity due to the sum of primary membrane and primary bending stresses shall not exceed 4 K/3;

— the stress intensity due to the sum of primary membrane stresses, primary bending stresses and thermal stresses shall not exceed 2 K.

A.5.3 Nozzles and openings

The nozzle or opening shall be reinforced in accordance with 10.3.6.7.

Under the design combined load the following stress limits apply:

— the primary membrane stress intensity should not exceed 0,8 K;

— the stress intensity due to the sum of primary membrane stresses and primary bending stresses shall not exceed 1,5 K;

— the stress intensity due to the sum of primary membrane stresses, primary bending stresses and thermal stresses shall not exceed 2 K.

A.5.4 Additional stress limits

Where significant compressive membrane stresses are present the possibility of buckling shall be investigated and the design modified if necessary (see A.3). In cases where the external load is highly concentrated, an acceptable procedure would be to limit the sum of membrane and bending stresses (total compressive stress) in any direction at that point to 0,9 K.

Where shear stress is present alone, it shall not exceed K/3. The maximum permissible bearing stresses should not exceed K. Where there are tri-axial stresses, the largest of the stresses shall not exceed K.

Table A.1 — Classification of stresses for some typical cases

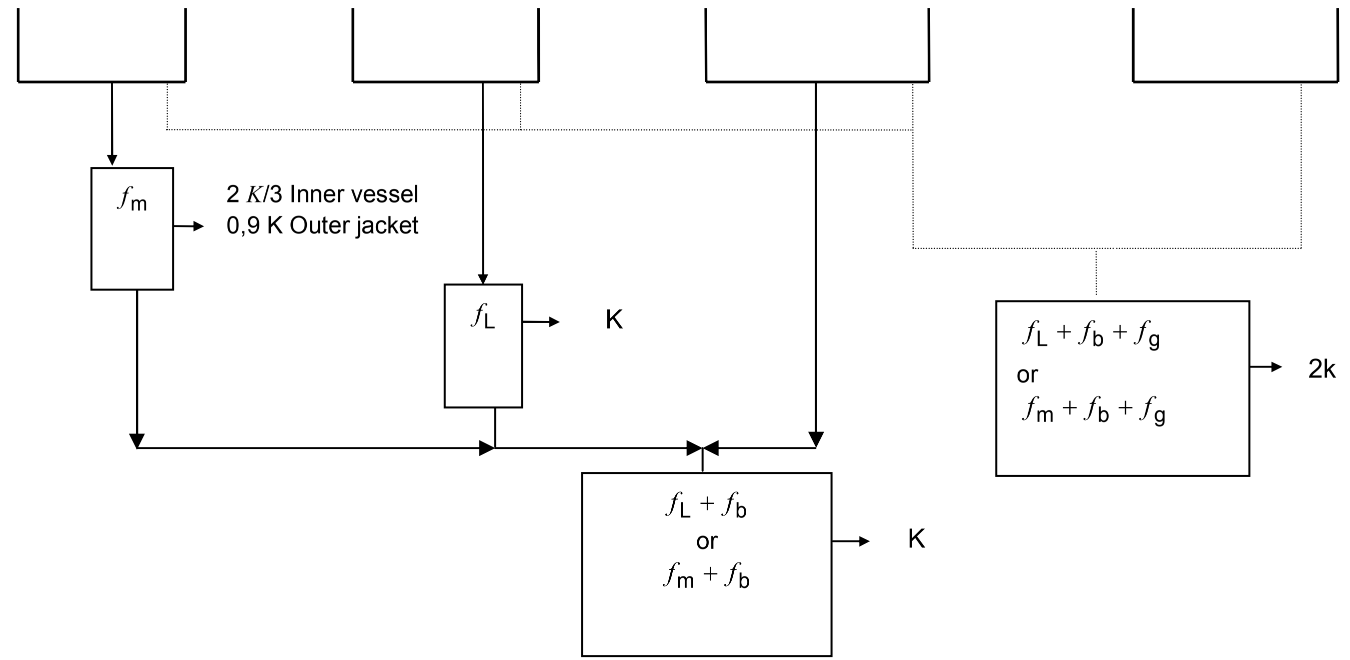
<table>
<tr><th>Vessel component</th><th>Location</th><th>Origin of stress</th><th>Type of stress</th><th>Classification</th></tr>
<tr><td rowspan="6">Cylindrical or spherical shell</td><td rowspan="4">Shell plate remote from discontinuities</td><td rowspan="2">Internal pressure</td><td>General membrane</td><td>f_m</td></tr>
<tr><td>Gradient through plate thickness</td><td>f_g</td></tr>
<tr><td rowspan="2">Axial thermal gradient</td><td>Membrane</td><td>f_g</td></tr>
<tr><td>Bending</td><td>f_g</td></tr>
<tr><td rowspan="2">Junction with head or flange</td><td rowspan="2">Internal pressure</td><td>Membrane</td><td>f_L</td></tr>
<tr><td>Bending</td><td>f_g</td></tr>
<tr><td rowspan="6">Any shell or end</td><td rowspan="2">Any section across entire vessel</td><td>External load or moment, or internal pressure</td><td>General membrane averaged across full section.
Stress component perpendicular to cross section</td><td>f_m</td></tr>
<tr><td>External load or moment</td><td>Bending across full section.
Stress component perpendicular to cross section</td><td>f_m</td></tr>
<tr><td rowspan="2">Near nozzle or other opening</td><td rowspan="2">External load or moment, or internal pressure</td><td>Local membrane</td><td>f_L</td></tr>
<tr><td>Bending</td><td>f_g</td></tr>
<tr><td rowspan="2">Any location</td><td rowspan="2">Temperature difference between shell and end</td><td>Membrane</td><td>f_g</td></tr>
<tr><td>Bending</td><td>f_g</td></tr>
</table>

Table A.1 (*continued*)

Vessel component	Location	Origin of stress	Type of stress	Classification
Dished end or conical end	Crown	Internal pressure	Membrane	f_{m}
			Bending	f_{b}
	Knuckle or junction to shell	Internal pressure	Membrane	f_{L} [a]
			Bending	f_{g}
Flat end	Centre region	Internal pressure	Membrane	f_{m}
			Bending	f_{b}
	Junction to shell	Internal pressure	Membrane	f_L
			Bending	f_{g}
Perforated end or shell	Typical ligament in a uniform pattern	Pressure	Membrane (average through cross section)	f_{m}
			Bending (average through width of ligament, but gradient through plate)	f_{b}
	Isolated or atypical ligament	Pressure	Membrane	f_{g}
			Bending	f_{g}
Nozzle	Cross section perpendicular to nozzle axis	Internal pressure or external load or moment	General membrane (average across full section). Stress component perpendicular to section	f_{m}
		External load or moment	Bending across nozzle section	f_{m}
	Nozzle wall	Internal pressure	General membrane	f_{m}
			Local membrane	f_{L}
			Bending	f_{g}
		Differential expansion	Membrane	f_{g}
			Bending	f_{g}

[a] Consideration should also be given to the possibility of buckling and excessive deformation in vessels with large diameter-to-thickness ratio.

Stress category	Primary General	Local	Bending	Secondary
Description (for examples see Table A.1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or differential thermal expansion. Excludes local stress concentrations.
Symbol (see NOTE 2)	f_m	f_L	f_b	f_g
Combination of stress components and allowable limits of stress intensities				



NOTE 1 The stresses in category f_g are those parts of the total stress that are produced by thermal gradients, structural discontinuities, etc., and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of f_m (or f_L) + $f_b + f_g$ and not f_g alone.

NOTE 2 The symbols f_m, f_L, f_b and f_g do not represent single quantities but rather sets of six quantities representing the six stress components.

Figure A.1 — Stress categories and limits of stress intensity

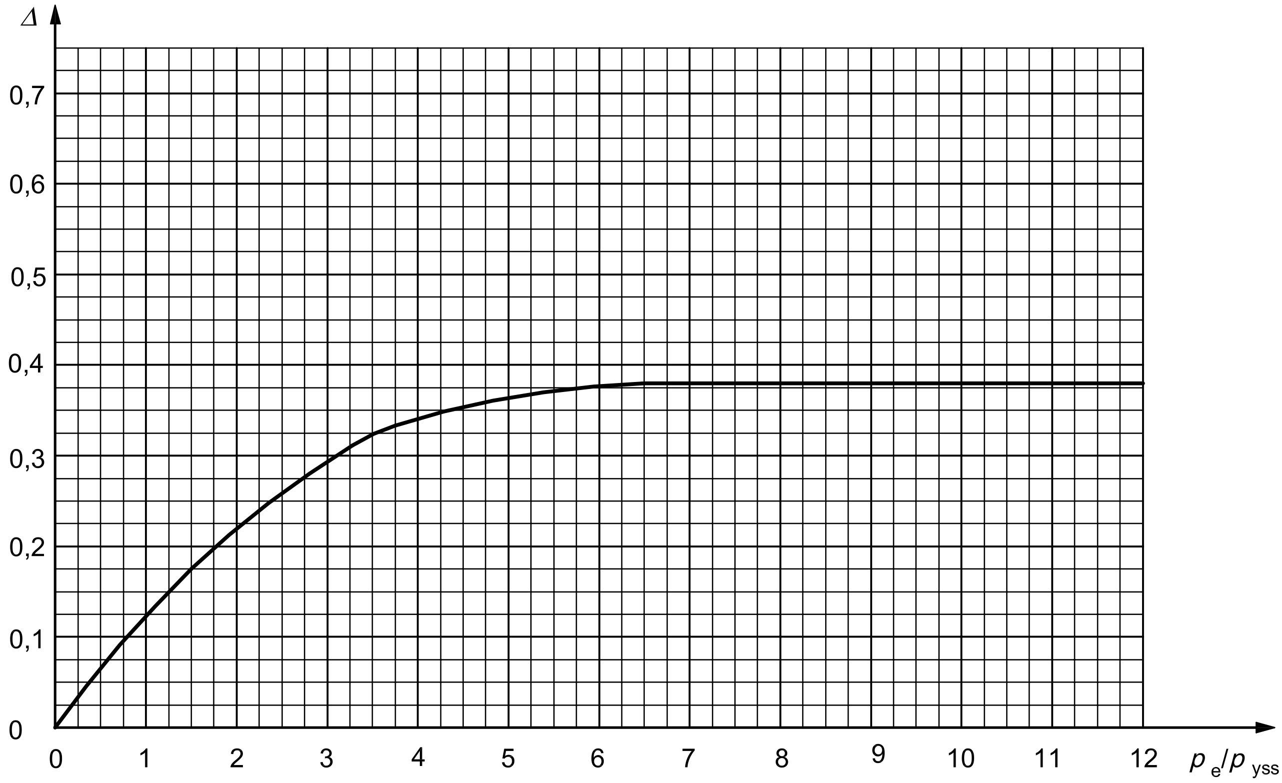


Figure A.2 — For vessels subject to external pressure

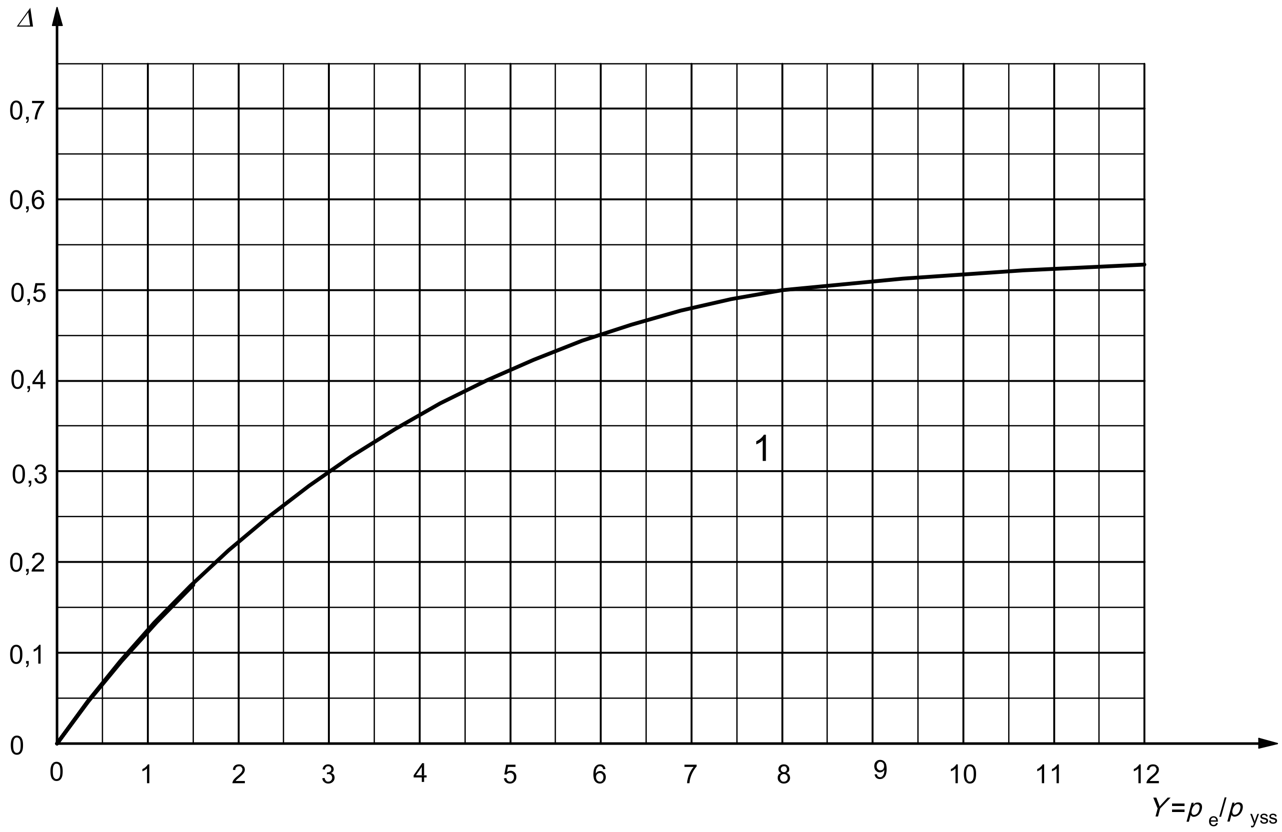


Figure A.3 — For vessels not subject to external pressure

Annex B
(normative)

Additional requirements for 9 % Ni steel

B.1 Introduction

Vessels constructed of 9 % Ni steels are normally welded using an austenitic or modified austenitic consumable. The 1 % or 0,2 % proof strength of the parent plate material normally exceeds that of an all weld metal sample. These weld metals exhibit excellent ductility and work hardening characteristics. After work hardening, the enhanced proof strength of the weld metal is maintained within an entirely elastic regime.

The value of K to be adopted in the calculation formulae of 10.3.6 is that of the parent 9 % Ni shell material.

During the first proof pressure test after fabrication, the welds plastically strain by a small, but sufficient amount such that their strength increases to create equilibrium with the applied loads. Thereafter the vessel behaves elastically when subjected to the maximum allowable working pressure.

B.2 Specific requirements

B.2.1 The minimum design temperature of vessels constructed of 9 % Ni steel shall not be less than −196 °C.

B.2.2 The maximum design temperature shall be 50 °C and a maximum temperature of 200 °C shall not be exceeded, when defrosting or drying the vessel at low pressure.

B.2.3 The maximum thickness of the vessel at the weld edge preparation shall not exceed 30 mm. A high nickel austenitic weld wire shall be used when the thickness of the vessel at the weld edge preparation exceeds 20 mm.

B.2.4 The full length of all butt joints shall be examined by radiographic or ultrasonic methods before the first proof pressure test. Defects that are unacceptable to this International Standard shall be repaired and re-examined to demonstrate compliance.

B.2.5 The full length of all branch attachment welds shall be examined by dye penetrant before the first proof pressure test. Imperfections that are unacceptable to this International Standard shall be repaired and re-examined to demonstrate compliance.

B.2.6 The vessel and all welds shall be examined visually after the proof pressure test to ensure that there is no evidence of gross deformation.

B.2.7[2)] The weld procedure qualification and production control transverse tensile test specimens shall:

— show no gross deformation when subjected to a tensile stress equal to the minimum specified material property, K, of the parent plate. Some small reduction in area is acceptable due to the expected plastic deformation associated with strain hardening. The measured 1 % proof stress of the transverse tensile test piece when using a 50 mm gauge length shall not be less than the minimum specified material property, K, of the parent plate;

— demonstrate a rupture strength not less than the minimum specified ultimate strength of the parent plate.

B.2.8 Longitudinal bend tests, as permitted by ISO 15607 shall be used rather than side bend tests when qualifying weld procedures or testing production control test plates.

B.2.9 The heat affected zone and weld metal at the weld fusion boundary shall be demonstrated to attain an ISO V-notch impact strength of 50 J at −196 °C, as an average of 3 test pieces, during weld procedure qualification and production control plate testing. The test piece shall be a transverse specimen.

B.2.10[2)] Openings shall not be located with their centre lines closer to principal seams than twice their diameter.

B.2.11[2)] Butt welds shall not be located where they are subject to high bending stresses which can result in plastic cycling and incremental collapse.

B.2.12 9 % Ni vessels may be fitted with nozzles of stainless steel. Where the outside diameter of the nozzle exceeds 75 mm, the stresses in the shell and nozzle due to pressure, mechanical loads and thermal expansion shall be assessed and shown to comply with the requirements of Annex A and to provide an adequate fatigue life for the intended application of the vessel.

B.2.13 Filler wires shall be selected from austenitic, modified austenitic or high nickel austenitic materials.

B.2.14 9 % Ni material conforming to EN 10028-4 or DIN 17280 is suitable for the construction of cryogenic vessels conforming to this International Standard. Other materials may be suitable.

2) These items also apply to work hardened austenitic stainless steel.

Annex C
(normative)

Pressure strengthening of vessels from austenitic stainless steels

C.1 Introduction

Austenitic stainless steel exhibits stress/strain characteristics (Figure C.2), different from that of carbon steel (Figure C.1), that enable stainless steel to accept strain as a means of increasing its proof strength. Plastic deformation of 10 % is possible with steels having an elongation at fracture of at least 35 % in the solution heat treated condition.

Austenitic stainless steel that has been strained to a higher proof strength will retain and even increase its enhanced strength advantage at cryogenic temperatures.

For instance, when austenitic stainless steel is loaded in tension to a stress, σ_k , above its proof strength and then unloaded a permanent plastic elongation will result. When this steel is loaded again it will remain elastic up to this higher stress which is then the new proof strength; only when the stress exceeds σ_k will the deformation follow the original stress/strain curve.

When the strengthening stress, σ_k , has been chosen the minimum wall thickness of parts of the vessel can be calculated from the design operating stress to be equal to or less than two thirds of σ_k (which is equal to the new proof strength).

In practice the strengthening is produced by pressurizing the finished vessel to a pressure, p_k, known to produce the required stress which in turn gives the required amount of plastic deformation to withstand the pressure load.

This technology primarily applies to vessels (or parts of vessels) of non-complex “balloon-type” design, i.e. structures where the pressure induced membrane stresses are dominant. Other parts of the vessel are normally designed based on conventional design stress values following Clause 10 and the relevant annexes of this International Standard.

NOTE This method is also known as *Cold-Stretching*. However, using the word *Cold* in connection with cryogenic vessels may be misleading since the strengthening pressure is applied at room temperature. Also, the *Stretching* will be slight, if any, when using shell material in the work-hardened condition. On the other hand, applying a pressure in excess of the normal test pressure effectively demonstrates the strength and pressure bearing capability of all parts of the complete vessel.

C.2 Field of application

This annex applies to cryogenic pressure vessels made from austenitic stainless steel of a wall thickness of not more than 30 mm, strengthened by pressurization at room temperature after being completed and intended for a maximum operating temperature of less than 50 °C.

IMPORTANT — This method shall be used only if accepted by the applicable regulations. If used, all requirements of this annex shall be applied.

C.3 Definitions and units of measurement

Definitions, symbols and units of measurement given in Clauses 3 and 4 apply to this annex, with the following addition:

pressure strengthened vessel
pressure vessel, which has been subjected to a calculated and controlled internal pressure (strengthening pressure) after completion

NOTE 1 The wall thickness of such a vessel is calculated on the basis of the stress at the strengthening pressure and not on the basis of the conventional design stress value of the material used.

NOTE 2 Pressure vessels made from solution heat treated material will be subject to a controlled plastic deformation during the strengthening operation as its yield point is raised. Pressure vessels made from work-hardened material will be subject to little or no plastic deformation.

C.4 Materials

C.4.1 Accepted materials of construction that have already been proven suitable for pressure strengthening for operating temperatures of not less than −196 °C are the austenitic stainless steels specified in Table C.1. Requirements regarding these materials are found in EN 10028-7.

When material is delivered in a work-hardened condition, the material shall have an elongation at fracture A_5 of not less than 35 %.

Table C.1 — Austenitic stainless steels accepted for pressure strengthening of cryogenic vessels for operating temperatures of not less than –196 °C

Steel designation		Solution heat treated material		Pressure strengthened vessel
Name	Number	$R_{p0.2}$ N/mm^2 min	$R_{p1.0}$ N/mm^2 min	σ_k N/mm^2 max
X5CrNi18-10	1.4301	210	250	410
X2CrNi19-11	1.4306	200	240	400
X2CrNiN18-10	1.4311	270	310	470
X6CrNiTi18-10	1.4541	200	240	400
X6CrNiNb18-10	1.4550	200	240	400
X5CrNiN19-09	1.4315	270	310	470
SA/A-240 340	S 30400	—	—	410
SA/A-240 304L	S 30403	—	—	385
SA/A-240 304N	S 30451	—	—	470
SA/A-240 316	S 31600	—	—	410
SA/A-240 316L	S 31603	—	—	385
SA/A-240 316N	S 31651	—	—	470
SA/A-240 316LN	S 31653	—	—	410

C.4.2 In case-stable or metastable austenitic steels according to Clause 9 other than those listed in Table C.1 are to be qualified for pressure strengthening, or the vessel operating temperature will be below –196 °C, steel quality and welding procedure shall be validated by the type approval test detailed below. This test shall be carried out in addition to the tests required by 9.1 and 11.6.1.

A welded test plate shall be subjected to a tensile stress across the weld equal to the anticipated value of σ_k. From this test plate specimens shall be tested as follows:

a) to test the base material: two tensile tests along the direction of the applied stress and one set of impact tests across the direction of the applied stress;

b) to test the weld: two tensile tests across the weld and one set of impact tests of the weld metal according to 4.5.3 of ISO 21028-1:2004.

One tensile test and the impact tests shall be carried out at the lowest operating temperature, the other tensile test shall be carried out at 20 °C.

The base material and the weld shall comply with:

$R_{p0,2} \geqslant \sigma_k$; $A_5 \geqslant 25\ \%$; a_k ISO-V $\geqslant 50\ \mathrm{J/cm^2}$

C.5 Design

C.5.1 General

C.5.1.1 Wall thicknesses calculated according to C.5.2.3 refer to thicknesses before strengthening.

C.5.1.2 Nominal diameters may be used in the design calculations. No allowance is necessary for the possible increase in diameter due to strengthening.

C.5.1.3 Maximum design stress value is limited to 200 N/mm^2 above $R_{p0,2}$ for the material in the solution heat treated condition.

C.5.1.4 The weld joint factor 1,0 may be used for the calculation of all pressure strengthened parts of the vessel (longitudinal welds in cylinder, cone or end).

C.5.1.5 Pressure strengthening applies to vessels (or parts of vessels) where the pressure induced membrane stresses are dominant. Other parts of the vessel shall be designed in accordance with Clause 10 and the relevant annexes of this International Standard. This requirement shall not preclude utilisation of the strengthening process, provided that the manufacturer can show that it does not cause deformations that impair the integrity of the vessel.

C.5.2 Design for internal pressure

C.5.2.1 Design stress values

The design stress value, σ_k, at 20 °C can be selected freely up to the highest allowable design stress value, σ_{kmax}, according to Table C.1. This highest allowable design stress value is the same whether the material used is in the solution heat treated or work-hardened condition.

C.5.2.2 Calculation of the strengthening pressure

The required strengthening pressure, p_k, is calculated according to the formula

$$p_k = 1{,}5\,p$$

NOTE Strained material is also known to increase its strength when cooled to cryogenic temperatures. However, the effect on strengthening pressure (analogous to the effect on test pressure as in 10.2.3.2.1 g) of this document) is not taken into account in this annex.

C.5.2.3 Calculation of wall thicknesses

C.5.2.3.1 General

The wall thickness of the various parts of the pressure vessel shall be calculated according to applicable subclauses of this International Standard with the modifications shown in Table C.2.

Table C.2 — Modification of formulae for the design of pressure strengthened vessels

Subclause of this International Standard		Modification, see subclause in this annex
10.3.6.1	Cylinders and spheres subject to internal pressure	C.5.2.3.3
10.3.6.4	Dished ends subject to internal or external pressure	
	10.3.6.4.4 Internal pressure calculation (pressure on the concave surface)	C.5.2.3.4
10.3.6.5	Cones subject to internal or external pressure	
	10.3.6.5.6 Internal pressure calculation (pressure on the concave surface) $\lvert\varphi\rvert \leqslant 70°$	C.5.2.3.4
	10.3.6.5.7 Internal pressure calculation (pressure on the concave surface) $\lvert\varphi\rvert > 70°$	C.5.2.3.2
10.3.6.6	Flat ends	C.5.2.3.2
10.3.6.7	Openings in cylinders, spheres and cones	C.5.2.3.5

C.5.2.3.2 Parts where bending stresses are dominant and large deformations cannot be accepted, like flat cones according to 10.3.6.5.7 and flat ends according to 10.3.6.6, shall be calculated in the normal way using the design pressure, p, and design stress values according to 10.3.2.3. That is, the effect of the strengthening may not be utilised in such designs.

Additionally, the capability to pass the strengthening without plastic deformation shall be checked by repeating the calculations using the strengthening pressure (taking the mass of contents into account) for the test pressure, p_T, and the design stress value at 20 °C from 10.3.2.2 a).

C.5.2.3.3 When designing parts according to 10.3.6.1.3 insert into the applicable formulae the following:

— design stress value, σ_{k} ;

— weld joint factor 1,0.

C.5.2.3.4 Parts according to 10.3.6.4.4 and 10.3.6.5.6 shall be designed with the same modifications as in C.5.2.3.2. Additionally the shape factor, β, for dished ends may be reduced to:

— for 10 % torispherical ends, 2,93;

— for 2:1 torispherical ends, 1,91.

However, it shall be demonstrated by calculation or experiment that the strain during strengthening will not cause excessive deformation in regions subject to bending stresses. In cases where the deformation will lead to a better shape (e.g. deeply dished ends turning hemispherical) the method may be used even with large bending stresses.

Also the risk of buckling in regions where compressive stresses occur (i.e. the knuckle of dished ends and corner area of cones) shall be paid special attention. But, since buckling is heavily dependent on initial imperfections and work-hardening of the material before pressurization, there is no substitute for experience. However, the stretching process in itself will reveal any such tendencies (see C.6.1).

C.5.2.3.5 For reinforcements of openings the stiffness of the attachment shall be considered so that over-dimensioned reinforcements are avoided. Preferably openings without reinforcement should be used. Unreinforced openings in this context includes openings having reinforcement not complying with 10.3.6.7.5.

For openings where the hole diameter exceeds that given below, calculation of the reinforcement shall be made according to 10.3.6.7 with the same modifications as in C.5.2.3.3.

When using external plate reinforcement or other kinds of reinforcements that are not welded with full penetration, the risk of overloading of the welds during strengthening shall be observed.

When ligament efficiency is less than 1, stresses due to strengthening shall be analysed according to 10.3.7.

Largest allowed opening of unreinforced single holes

In the case of holes joining a nozzle etc. to the shell, the inside diameter of the nozzle shall not exceed d_{max}.

$$d_{\text{max}} = 0{,}4\sqrt{D_{\text{y}}s} + C$$

where

d_{max} is the diameter of largest allowed opening (major axis for oval holes), mm;

D_{y} is the outside diameter of shell, mm;

R is the inside crown radius of end, mm;

s_0 is the wall thickness of unpierced shell, mm;

s is the true wall thickness of shell, mm;

μ $= s_0/s$;

C $= 60\sqrt{2(1-\mu)}$ with a maximum of 60 mm.

The value of d_{max} calculated may be rounded up to the nearest higher even 10 mm. d_{max} shall however meet the conditions:

$$d_{\text{max}} \leqslant 150 \quad \text{mm}$$

$$d_{\text{max}} \leqslant 0{,}2D_{\text{y}}$$

The wall thickness of an unpierced cylinder is calculated from

$$s_0 = \frac{pD_{\text{y}}}{20\dfrac{\sigma_{\text{k}}}{1{,}5} + 2p}$$

The wall thickness of the crown region of an unpierced dished end is calculated from

$$s_0 = \frac{pR}{20\frac{\sigma_k}{1{,}5}}$$

C.5.3 Design for external pressure

C.5.3.1 If a pressure strengthened vessel normally operating under internal pressure can be subject to external pressure, the vessel shall also be designed to withstand external pressure according to the applicable subclauses of Clause 10.

By these calculations the design stress value shall be taken from 10.3.2.3. If the pressure strengthened vessel is made from solution heat treated material the safety factors, S_k, given in 10.3.2.4 may be replaced by S_k/1,5.

NOTE This modification is a consequence of the improved shape of the pressure vessel produced by the straining so that a lower factor of safety can be accepted.

In the case of vessels having large nozzles in the shell or when this improvement of the shape is otherwise doubtful, the above modification may be utilised only if measurements after strengthening show that the vessel is not significantly out of round.

C.5.3.2 If a vessel is shaped such that it is subject to an external pressure during the strengthening operation, it shall be calculated using the strengthening pressure (taking the mass of contents into account) as a test pressure, p_T, and the material properties at 20 °C from 10.3.2.3.2.

C.6 Manufacturing and inspection

C.6.1 Strengthening procedure

C.6.1.1 The strengthening operation, which is a step in the production of the finished vessel, shall be made following written instructions. These instructions shall include the steps described in C.6.1.2 to C.6.1.6.

When vessels under pressure require inspection and measurement, adequate facilities and procedures shall be employed to assure the safety of inspectors, employees and the public.

C.6.1.2 Fill the vessel with liquid. Before the vessel is closed, wait for at least 15 min to let any air dissolved in the liquid escape. Then top up and seal the vessel.

C.6.1.3 The circumference of all courses shall be measured (e.g. with steel tapes) where the largest increase in cross-section is expected. The strain rate during the strengthening operation shall be calculated over the full circumference.

C.6.1.4 The strengthening is normally carried out as follows: the pressure is raised to the strengthening pressure and maintained until the strain rate has dropped to less than 0,1 %/h. The time under pressure shall be not less than one hour (see however C.6.1.5). The strain rate shall be checked by repeated measurements of the circumference according to C.6.1.3. The requirement of 0,1 %/h shall be met during the last half hour.

NOTE The total time under pressure can be long. This can be reduced if a 5 % higher pressure is applied during the first 0,5 h to 1 h of the operation.

C.6.1.5 For pressure vessels having a diameter not more than 2 000 mm the time under pressure may be reduced to 30 min and the requirement of 0,1 %/h be met during the last 15 min.

C.6.1.6 The strengthening operation replaces the initial pressure testing of the vessel. Should later pressure testing be required, only the normal test pressure shall be used. If the vessel requires to be repaired, this repair and pressure testing or possibly renewed strengthening shall be carried out in accordance with C.6.3.4.

C.6.2 Procedure record

There shall be a written record of the operation, containing at least the following information:

- pressurizing sequence specifying pressure readings and time;
- circumference measurements before, during and after pressurization;
- strain rate calculations from circumference measurements;
- any significant changes of shape and size relevant to the functioning of the vessel;
- any requirement for renewed strengthening (according to C.6.1.6 and C.6.3.4).

C.6.3 Welding

C.6.3.1 The strengthening method presumes high quality welding. The same rules apply as for conventionally produced cryogenic vessels, except that production control test plates need not be taken.

C.6.3.2 Non-destructive testing shall be carried out before the strengthening to the extent stipulated in 6.3 for the weld joint factor 1,0. Where high local stress and strain concentrations can be expected during the strengthening operation, examination with liquid penetrant shall also be carried out e.g. at changes in wall-thickness or at welded nozzles.

C.6.3.3 After the strengthening operation and reducing the pressure to the design, pressure welds shall be visually examined externally for their full lengths. Places which have been examined with liquid penetrant according to C.6.3.2 shall also if possible be tested at random using a volumetric method (preferably by radiographic examination).

C.6.3.4 Renewed strengthening shall be carried out if pressure strengthened parts of the vessel have been significantly affected by post strengthening welding. Exceptions are permitted for tack-welding of attachments carrying low loads only (e.g. insulation supports) and welding of nozzles not more than 10 % of the vessel inner diameter (with a maximum of 100 mm) or minor weld repairs with comparable effect on the construction. Such welds shall be examined according to C.6.3.2 and C.6.3.3.

Unless renewed pressure strengthening is carried out there shall be a normal pressure test as required by 12.5.2 after all welding on pressure retaining parts.

C.6.4 Pressure vessel drawing

C.6.4.1 In addition to the information required by 10.2.2, the drawing shall bear the following information:

- the vessel is manufactured according to Annex C;
- strengthening pressure in bars;
- thicknesses and diameters shown apply before strengthening.

C.6.4.2 Details to be welded in place after the strengthening shall be marked on the drawing.

C.6.5 Data plate

The data plate shall in addition to the information according to Clause 13 bear the text “PRESSURE STRENGTHENED”.

C.7 Comments

C.7.1 Strengthening theory

Austenitic stainless steels exhibit considerable work-hardening upon deformation while retaining the characteristics of the material. The stress required for further deformation increases continuously as the deformation increases. Thus, a stress/strain curve for austenitic steel does not have the flow region typical of carbon and low-alloy steels. Compare the stress/strain curves in Figure C.1 and C.2.

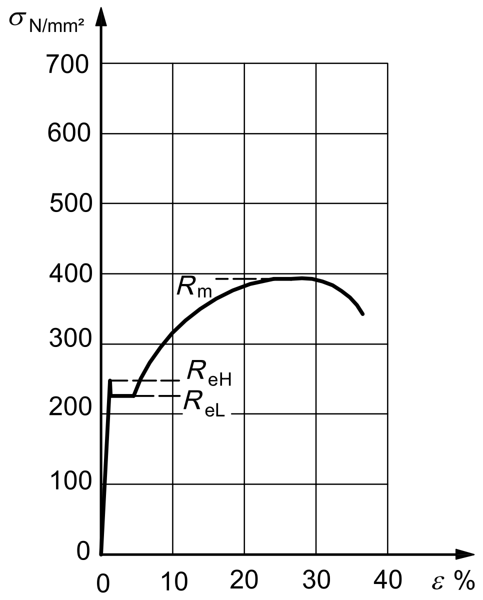


Figure C.1 — Stress/strain curve for carbon steel

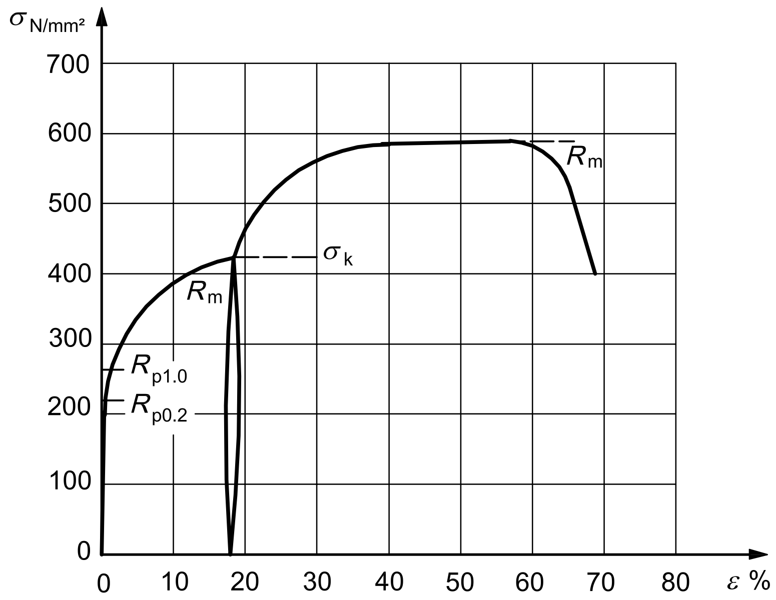


Figure C.2 — Stress/strain curve for austenitic stainless steel

If a tensile test piece of solution heat treated austenitic stainless steel is loaded to a strengthening stress, σ_k , and then unloaded, a permanent plastic elongation will be found. When the same test piece is loaded again the deformation will remain elastic up to a higher stress level than before. Only when the stress, σ_k , is exceeded will the plastic deformation continue along the original curve.

A test piece which has been loaded to the strengthening stress, σ_k , can be regarded as a new test piece with:

$$R_{p0,2} = \sigma_k$$

An austenitic stainless steel that has been stretched at room temperature to a higher proof strength also exhibits higher proof strength stress at all other temperatures.

The toughness of the material after stretching to 10 % (nominal strain) will still be satisfactory, since austenitic steels in the solution heat treated condition has an elongation at fracture not less than 35 %.

The plastic deformation required is achieved by subjecting the finished pressure vessel to a strengthening pressure, p_k. This pressure is calculated so that there is sufficient safety margin with respect to plastic deformation from stresses caused by a pressure equal to the design pressure, p.

Minimum wall thicknesses for the different parts of the vessel are calculated after establishing a suitable design stress value, σ_k .

During the strengthening of the finished vessel, the material reaches a strengthening stress (σ_k) that is at least 1,5 times the design operating stress.

C.7.2 Work-hardened material

C.7.2.1 The term work-hardened material shall be applied to material that has had its proof strength raised through cold rolling, roll straightening, uniaxial stretching in a stretching machine or other types of cold work.

C.7.2.2 Work-hardened material can be used in order to reduce or eliminate the deformation due to strengthening of the pressure vessel. It is primarily used in cylinders for internal pressure.

C.7.2.3 The increase in the proof strength of a work-hardened material is about the same in all directions. The proof strength of work-hardened plate shall be determined on samples taken across the direction of rolling or stretching respectively.

C.7.2.4 The structure of work-hardened material differs from solution heat treated material only in that the number of dislocations is higher. Material that has been subject to a homogeneous deformation is free from residual stresses. Work-hardening does not significantly affect the resistance to general corrosion.

Welding of work-hardened material gives rise to a heat-affected zone (HAZ), the width of which depends on the welding method. In arc welding with coated electrodes, the width of the zone is about equal to the thickness of the material.

The proof strength in the zone may be reduced, but the subsequent strengthening restores it to about the same level as that of the surrounding material.

Impact toughness and corrosion resistance in the zone depend primarily on the initial material condition (analysis, well annealed structure) and the welding method (extent of heating) but only slightly on the degree of strengthening.

Strengthening of a pressure vessel generally decreases local residual stresses introduced into the vessel during the manufacturing process.

C.7.3 Derivation of formulae

C.7.3.1 Consider a cylinder of middle diameter, D, and design pressure, p, which has been strengthened to a design stress value, σ_k. Its wall thickness should comply with the formula for cylinders in 10.3.6.1.3:

$$s = \frac{pDs_\mathrm{F}}{20\sigma_\mathrm{k}z}$$

The strengthening shall be carried out in such a way that the shell is subjected to the stress, σ_k. The stress in a cylinder is:

$$\sigma = \frac{pD}{20s}$$

and the strengthening pressure, p_k, will therefore be:

$$p_\mathrm{k} = \frac{20s\sigma_\mathrm{k}}{D}$$

If s according to the formula given in C.5.2.3.5 is substituted:

$$p_\mathrm{k} = p\frac{s_\mathrm{F}}{z}$$

Since $S_\mathrm{F} = 1{,}5$ and $z = 1{,}0$ this corresponds to the formula given in C.5.2.2. Cylinders can be calculated from the formula in 4.3.6.1.3, if σ_k is inserted as the design stress value and 1 as the weld joint factor.

NOTE If a weld joint factor, z, less than 1,0 is applied to any single main seam an increase in strengthening pressure is required according to the formula given in C.5.2.3.5. To sustain this higher pressure the thickness of all parts of the vessel would then need to be increased.

C.7.3.2 If a shell consists of several courses and one of them is made thicker than the others, it will have a lower σ_k than the other courses after strengthening.

The thicker course then needs a higher strengthening pressure than the others. Since this is impossible, this course will fail to satisfy the above formula (not “strengthened enough”), as the anticipated proof strength, σ_k, will not be reached.

In order to achieve the full theoretical effect throughout the vessel, it would be necessary to decrease the thickness of the thicker course. Since this would hardly increase the safety of the vessel it is allowed to use greater thickness in some parts, e.g. where required by external loads, even if this is not theoretically correct.

Correspondingly, constant wall thickness is allowed in conical ends, even though the strengthening theory strictly speaking requires the thickness to be decreased in proportion to the radius. Similarly, the spherical part of a dished end will in some cases be “insufficiently pressure strengthened”.

C.7.3.3 The derivation of formulae in C.7.3.1 applies to parts free from bending stresses, i.e. cylinders, spheres and hemispherical ends.

Utilisation of the strengthening effect is generally not permitted for parts subject to primary bending stresses. For such parts, it is necessary to investigate the stresses during strengthening (see C.5.2.3.2) and normal operation.

Certain pressure vessel parts, such as dished and conical ends, contain so-called secondary bending stresses (see Annex A). It is permissible to use the strengthening effect in such parts, but the magnitude of the secondary bending stresses must be investigated and should normally not exceed $2\sigma_k$.

Excepted from this requirement of investigation are 2:1 torispherical ends, where experience has shown the bending stresses to be moderate.

C.7.3.4 Experience has shown that it is possible to use design stress values for pressure strengthened material when dimensioning reinforcement pads according to 10.3.6.7.

C.7.3.5 This annex does not preclude utilisation of the strengthening effect, provided that the manufacturer can show that it does not cause harmful deformation or other problems.

C.7.4 Deformations at strengthening

C.7.4.1 The highest allowable design stress value, $\sigma_{k\,\mathrm{max}}$, for the different steels has consistently been set 200 N/mm^2 higher than $R_{p0,2}$ for the solution heat treated material.

In conventional tensile testing, this maximum stress produces less than 10 % elongation.

C.7.4.2 The strengthening process can be simulated in tensile testing by allowing extra time under load. This increases the elongation under maximum stress by another 1 % to 2 %.

After simulated strengthening, the proof strength, $R_{p0,2}$, of the material (calculated on the basis of the cross sectional area before the strengthening) is about 30 N/mm^2 higher than the strengthening stress, σ_k, used.

C.7.4.3 A multi-axial stress state results in other elongation values than tensile testing. These elongation values can be assessed according to a graph of the deformation hardening of the material as applied to the effective values of stress, σ, and elongation, ε.

$$\sigma = \sqrt{\frac{1}{2}\left[\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2\right]}$$

$$\varepsilon = \sqrt{\frac{2}{9}\left[\left(\varepsilon_1 - \varepsilon_2\right)^2 + \left(\varepsilon_2 - \varepsilon_3\right)^2 + \left(\varepsilon_3 - \varepsilon_1\right)^2\right]}$$

If the effective values are set = 1, the principal stresses and elongations obtained for the simplest stress conditions are given in Table C.3:

Table C.3 — Stresses and elongations for different load cases

	True stress				True elongation			
	σ_1	σ_2	σ_3	$\underline{\sigma}$	ε_1	ε_2	ε_3	$\underline{\varepsilon}$
Tensile test	1	0	0	1	1	−0,5	−0,5	1
Cylinder	1,15	0,58	0	1	0,87	0	−0,87	1
Sphere	1	1	0	1	0,5	0,5	−1	1

Among other things, Table C.3 expresses the fact that a tensile test sample contracts in two dimensions, while a cylinder decreases only in thickness by an amount corresponding to the increased circumference.

Table C.3 shows that a certain effective stress σ produces different elongation in the principal stress direction, ε_1, for the different load cases. The same effective stress that produces a strain of 10 % in a tensile test ($\varepsilon_1 = 1{,}0$) produces a circumferential strain 8,7 % ($\varepsilon_1 = 0{,}87$) in a cylinder shell and 5 % ($\varepsilon_1 = 0{,}5$) in a sphere.

The true stresses, σ_1, σ_2, σ_3, and $\underline{\sigma}$, are calculated on the basis of the cross-sectional area of the material after deformation. If instead the nominal stresses are used, calculated on the basis of the original cross-sectional area of the material, the comparison of strains will be different.

The following example gives an indication of the difference.

EXAMPLE Values from a typical deformation hardening curve of austenitic stainless steel are used, i.e. 0,2 %/280 N/mm^2 and 10 %/420 N/mm^2. If equal nominal principal stresses, $\sigma_{1\,\mathrm{nom}}$, are applied to this material, the principal strain, ε_1, for the cylinder is altered from 0,87 to 0,66 and for the sphere from 0,5 to 0,58.

The strain at bursting pressure is half of the maximum homogeneous strain at tensile testing for a cylinder and one third for a sphere.

C.7.4.4 In practice, the maximum circumferential strain of cylinders is usually 3 % to 5 % when using solution heat-treated plate, less in the spherical part of the ends. The following factors contribute to the measured values being lower than the theoretically calculated maximum value:

— the proof strength, $R_{\mathrm{p0,2}}$, is higher than the specified minimum for the material;

— the plate thickness is greater than nominal;

— there are reinforcing effects of ends, nozzles, etc.

C.7.4.5 It should be observed that strengthening of pressure vessels of solution heat treated material can affect the position, direction and roundness of nozzles. This does not entail any reduction of the safety of the vessel, but may in certain cases be a nuisance to the user.

NOTE One way to minimise these changes is to weld the nozzles in place after the strengthening, whereupon the vessel could require renewed strengthening (see C.6.3.4). This second strengthening generally leads to much smaller deformations.

C.7.4.6 When a welded tube is used for nozzles in a cylinder (or cone), the longitudinal weld of the tube should be located in the direction where the stresses are lowest, i.e. in a plane perpendicular to the longitudinal axis of the cylinder (or cone).

Annex D
(informative)

Pressure limiting systems

In designing the pressure limiting systems, the manufacturer is required to assess the hazards that apply to the pressure equipment being manufactured. The equipment shall then be designed and constructed taking account of the assessment.

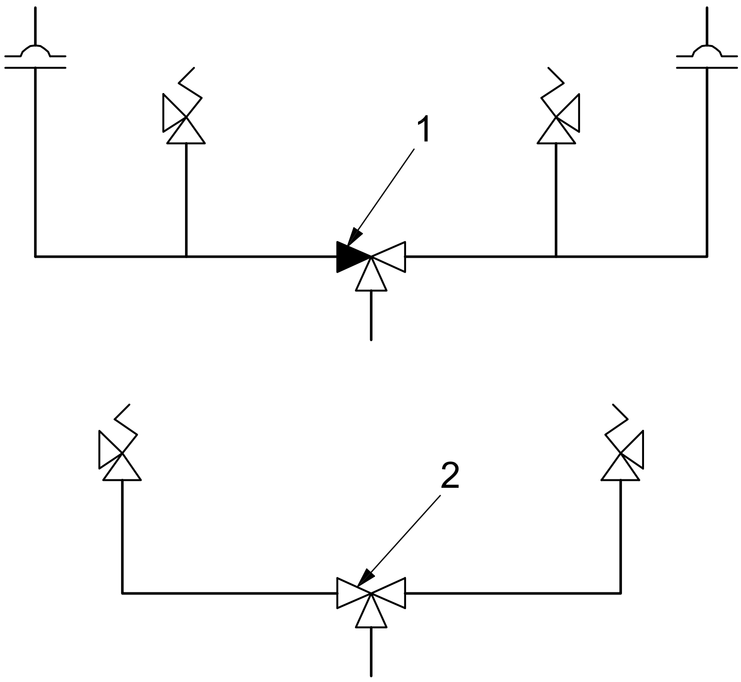
In selecting the most appropriate solutions, the manufacturer shall:

— eliminate or reduce hazards as far as is reasonably possible; and

— take the necessary protection measures against hazards which cannot be reasonably eliminated.

The selection of numbers, type and arrangement of the devices in the pressure limiting system is complex and requires the designer to consider carefully quality, reliability, service, application and maintenance.

In this International Standard no specific system of excess pressure protection is recommended, but two examples of relief systems currently in use are shown in Figure D.1:



Key

1 changeover valve

2 3-way valve

Figure D.1 — Examples of relief systems

Annex E
(normative)

Further use of the material cold properties to resist pressure loads

E.1 General

There are significant benefits to be gained from taking advantage of the enhanced properties of some materials such as stainless steels, 9 % Ni steels and aluminium alloys at cryogenic temperature and there are examples of these phenomena currently used in pressurized systems. This annex deals with a theoretical design method for further use of the material cold properties to resist pressure loads.

E.2 Introduction

This annex concerns a method of calculation of the wall thickness of the inner vessel of vacuum insulated cryogenic vessels permanently containing liquids. Cold vessels designed for air gases (nitrogen, oxygen, argon) which are refilled systematically when the level of cryogenic liquid drops below 25 %, can be taken as an example.

This calculation is based on the following considerations.

As long as there is cryogenic liquid - even in very low quantities - in a static vacuum-insulated cryogenic vessel storage, the temperature of the “hottest” point of the wall of the inner vessel does not exceed a temperature, t, which we can consider as being the maximum allowable temperature for normal operating conditions.

This temperature, t, can be determined experimentally for each type of cryogenic vessel, taking into account all likely operating conditions. An example of calculation with $t = -80$ °C is given at the end of this annex.

The calculation of the wall thickness of the inner vessel can be performed on the basis of the material property at the temperature, t. In such a case, an additional protection system activated by either low liquid level or the direct temperature, t, is fitted.

The additional protection system shall operate, in order to prevent excessive stresses in a vessel, at ambient temperature.

For the initial pressure test $p_T = p_S + 1$ bar can be considered for the “exceptional design condition”.

E.3 Field of application

This annex applies to cryogenic pressure vessels manufactured from materials following the requirements of ISO 21028-1.

E.4 General requirements

When the method described in this annex is followed, all the requirements included in the main part of this International Standard shall be followed with some exceptions concerning the calculation method (see 10.2 and 10.3) as indicated in E.5.

E.5 Specific calculation methods

In this specific calculation method the modifications to the general requirements are:

10.2.3.2.1 b) replaced by:

pressure during operation when the vessel contains only gaseous product at t °C.

$$p_{\mathrm{cG}} = p_{\mathrm{S}} + 1 \text{ bar}$$

10.2.3.2.1 d) replaced by:

reactions at the support points of the inner vessel during operation when the vessel contains only gaseous product at t °C. The reactions shall be determined by the weight of the inner vessel, its contents and seismic loadings where appropriate. The seismic loadings shall consider any forces exerted on the vessel by the insulation.

10.2.3.2.2 replaced by:

pressure test: The value used for validation purposes shall be the highest of:

$$p_{\mathrm{T}} = 1{,}25\left(p_{\mathrm{S}} + p_{\mathrm{L}} + 1\right)\frac{K_{20}}{K_{\text{design}}} \quad \text{(normal design condition, full vessel)}$$

K_{design} is the material property at a temperature specified by the manufacturer for a particular design case;

$$p_{\mathrm{T}} = 1{,}43\left(p_{\mathrm{S}} + 1\right)\frac{K_{20}}{K_{\mathrm{t}}} \quad \text{(normal design condition, nearly empty vessel);}$$

$$p_{\mathrm{T}} = p_{\mathrm{S}} + 1 \quad \text{(exceptional design condition);}$$

considered for each element of the vessel e.g. shell, courses, head, etc.;

the 1 bar is added to allow for the external vacuum.

10.2.3.2.2, 2) replaced by:

operation at maximum allowable working pressure when the vessel is filled with gas at t °C: b) + d);

10.2.6.2 to add at the end:

— in addition, the inner vessel shall be fitted with an additional protection system operating under pressure, p'_{S}, so that:

$$p'_{\mathrm{S}} = \left(p_{\mathrm{S}} + 1\right)\frac{K_{20}}{K_{\mathrm{t}}} - 1$$

— when the level of liquid drops below a minimum level, in no case lower than 5 % or when the temperature exceeds the predetermined design temperature, t. This system shall be agreed by the purchaser, the manufacturer and the notified body and shall be at least as reliable as that of the pressure limiting system.

10.2.3.2.2, 2) replaced by:

— in accordance with 10.2.3.2.1 1), 3), 4) and 5);

— material properties determined in accordance with 10.3.2.3.2 shall be adopted.

10.2.3.2.2, 2) to add a new c):

— in accordance with the modified 10.2.3.2.1 1) 2) (see above);

— material properties determined in accordance with the new following 10.3.2.3.4 shall be adopted.

add a new 10.3.2.3.4

— at maximum allowable temperature, t, for normal operating conditions:

— this temperature, t, is the maximum temperature of the wall of the inner vessel taking into account all foreseeable operating conditions. These conditions shall be determined and agreed with a notified body. The temperature used by the designer shall not be lower than the proven maximum temperature;

— the K_t value of K at t temperature shall be determined from the material standard (see EN 10028-7, Annex C for austenitic stainless steel) or shall be guaranteed by the material manufacturer.

EXAMPLE **Calculations of the thickness of the cylindrical part of the inner vessel of a cold converter 11000/20 bar**

p_s = maximum operating pressure = 20 bar
p_L = hydrostatic pressure = 0,82 bar
D_i = inside diameter = 1 480 mm
t = maximum allowable temperature for normal operation conditions

Material: X2CrNiN 18-10 (304LN)
K_{+20} = 310 MPa
$K_t = K_{-80}$ = 420 MPa
$K_{design} = K_{-140}$ = 531 MPa

Type of calculation	Calculation according to the main part of this International Standard ($t = +20$ °C)			Calculation according to Annex E ($t = -80$ °C)		
		v = 1	v = 0,85		v = 1	v = 0,85
1) According to 10.2.3.2.2 1) (c, e and f neglected in first approximation)	$e_1 = \dfrac{(p_s + p_L + 1)Di}{20\dfrac{K_{-140}}{1,5}\text{v}}$	4,56 mm	5,37 mm	$e_1 = 4.2.3.2$	4,56 mm	5,37 mm
2) According to 10.2.3.2.2 2) and E.4 d)(d neglected in first approximation)	$e_2 = \dfrac{(p_s + 1)Di}{20\dfrac{K_{20}}{1,5}\text{v}}$	7,52 mm	8,85 mm	$e_2 = \dfrac{(p_s + 1)D_i}{20\dfrac{K_{-80}}{1,5}\text{v}}$	5,55 mm	6,53 mm
3) According to 10.2.3.2.1 and E.4 c)	$p_T = \text{Max}\begin{cases} 1,43(p_s+1) \\ 1,25(p_s+p_L+1)\dfrac{K_{20}}{K_{-140}} \end{cases}$	30,03 bar 15,92 bar		$p_T = \text{Max}\begin{cases} 1,43(p_s+1)\dfrac{K_{20}}{K_{-80}} \\ 1,25(p_s+P_L+1)\dfrac{K_{20}}{K_{-140}} \\ p_s+1 \end{cases}$	22,16 bar 15,92 bar 21 bar	
4) According to 10.2.3.2.2 3)	$e_3 = \dfrac{p_T Di}{20\dfrac{K_{20}}{1,05}}$		7,52 mm	$e_3 = \dfrac{p_T Di}{20\dfrac{K_{20}}{1,05}}$		5,55 mm
Thickness to be used increase in thickness		7,52 mm	8,85 mm		5,55 mm 26 %	6,53 mm 26 %

Annex F
(informative)

Specific weld details

F.1 Field of application

Specific weld details given in F.2 are currently in common usage in cryogenic vessels and are appropriate to this service. Although the scope of EN 1708-1 or ASME VIII, does not specifically consider the application of weld details to cryogenic vessels, the manufacturer may consult it for guidance.

F.2 Specific weld detail

In general the welds are to be adequate to carry the expected loads and need not be designed on the basis of joint wall thickness.

F.2.1 Joggle joint, see Figure F.1

This joint may be used for cylinder-to-cylinder and end-to-cylinder (excluding cone to cylinder) connections provided that:

a) when the flanged section of a dished end is joggled, the joggle is sufficiently clear of the knuckle radius to ensure that the edge of the circumferential seam is at least 12 mm clear of the knuckle (see 10.3.6.4.2 for the dimensions);

b) when a cylinder with a longitudinal seam is joggled

- the welds are ground flush internally and externally for a distance of approximately 50 mm prior to joggling with no reduction of plate thickness below the required minimum, and
- on completion of joggling, the area of the weld is subjected to dye penetrant examination and is proven to be free of cracks;

c) the offset section which forms the weld backing is a close fit within its mating section at the weld round the entire circumference;

d) the profile of the offset is a smooth radius without sharp corners;

e) on completion of welding the weld fills the groove smoothly to the full thickness of the plate edges being joined;

f) the junction of the longitudinal and circumferential seams are examined radiographically and found to be free from significant defects.

F.2.2 Intermediate ends, see Figure F.2 and 10.3.6.4.3.

F.2.3 Backing strip, see Figure F.3.

May be used only for circumferential seams in cylinders, ends, nozzles and interspace pipes and for seams in ends, when the second side is inaccessible for welding and provided that non-destructive testing can be satisfactorily carried out where applicable.

F.2.4 End plate closure, see Figure F.4 for two examples of the many ways of welding flat plates. See also Figure 12.

F.2.5 Non full-penetration nozzle weld, see Figure F.5.

May be used to attach set in nozzles to ends and cylinders provided that the strength of the attachment welds can be demonstrated to be sufficient to contain the design nozzle loadings.

F.2.6 Non continuous fillet-weld on attachments.

May be used for all attachments to main pressure components provided that the following criteria are met:

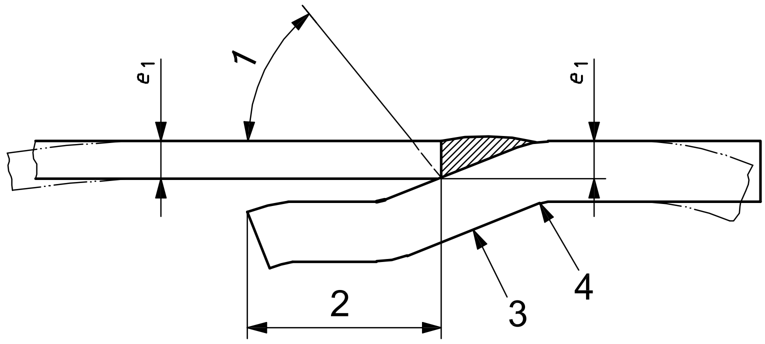
— strength is adequate for design loadings;

— crevices between attached component and main pressure envelope can be demonstrated not to conflict with F.3.

F.3 Oxygen service requirements

The need for cleanliness of equipment in liquid oxygen and other oxidising liquid service is described in ISO 21010 and ISO 23208.

The internal weld details shall be such that debris, contaminants, hydrocarbons or degreasants cannot accumulate to such a quantity so as to cause a fire risk in future operation.



Key

1 bevel optional

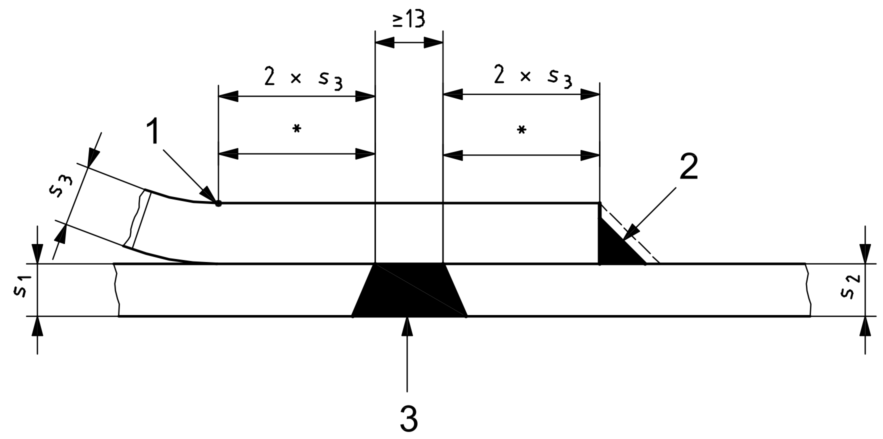
2 as desired

3 depth of offset $= e_1$

4 avoid sharp break

Figure F.1 — Joggle joint

Dimensions in millimetres



Key

1 tangent point

2 continuous fillet weld

3 butt weld

s_1 Cylinder thickness.

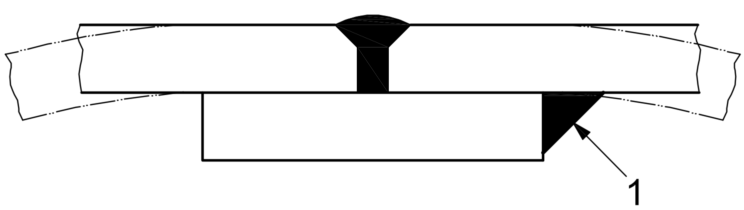
s_2 Cylinder thickness.

s_3 End thickness.

* Need not exceed 25 mm.

NOTE Cylinder thickness, s_1, and s_2, may vary.

Figure F.2 — Intermediate end



Key

1 Intermittent or continuous fillet weld

Figure F.3 — Backing strip

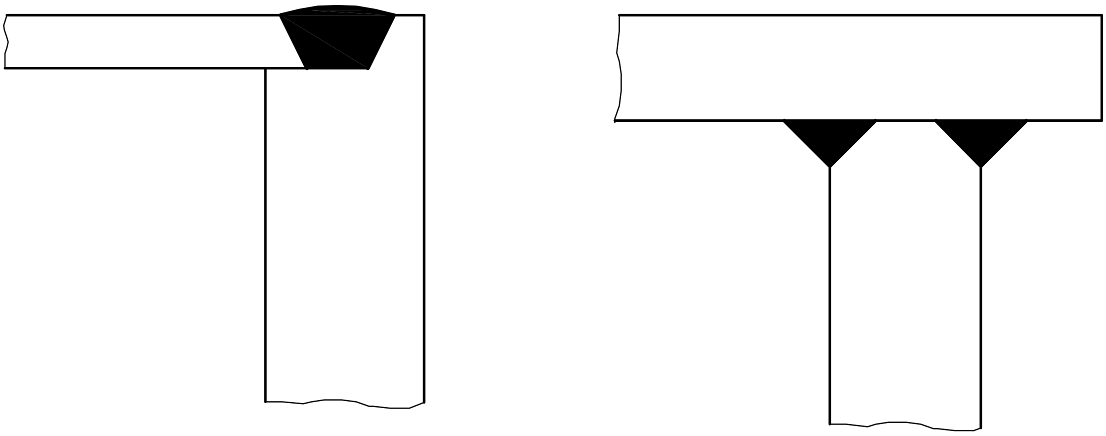


Figure F.4 — End plate closure (examples)

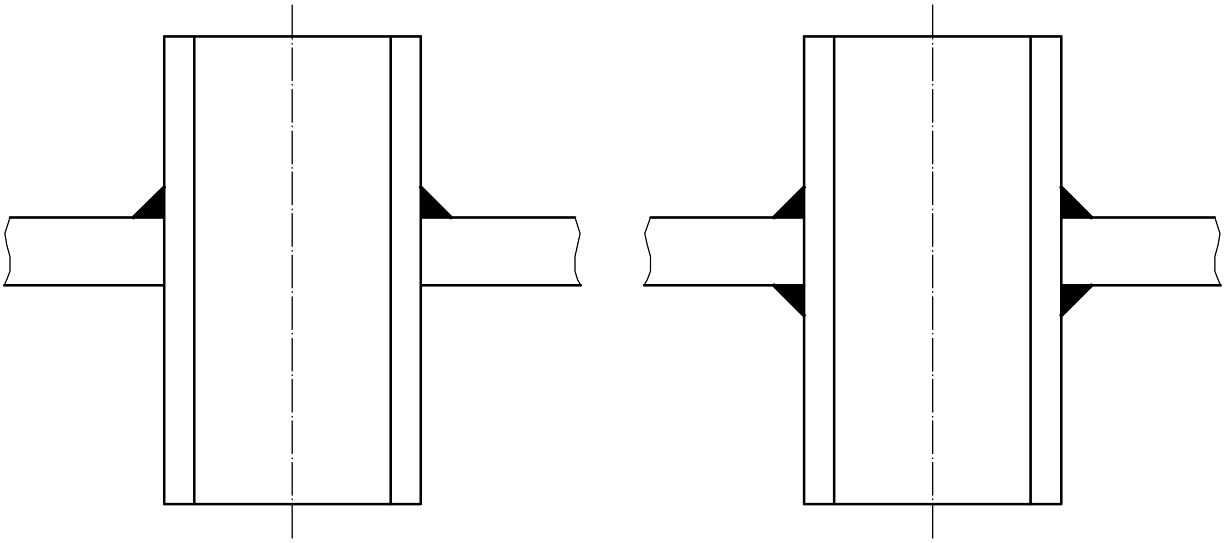


Figure F.5 — Non full penetration nozzle welds

Annex G
(normative)

Additional requirements for flammable fluids

G.1 In addition to the requirements of Clauses 10, 11 and 12, static vacuum-insulated vessels designed for use with the gases listed in Table 1 shall comply with the additional items given in G.2 to G.10.

G.2 Means shall be provided to ensure that the vessel is not filled to more than 95 % of its total volume, with liquid at the filling condition.

G.3 The selection and use of materials and joining procedures shall be carefully considered in the design of the installation to avoid secondary failure in the event of external fire.

G.4 For vessels of not more than 5 t capacity, the first valve of the supply line shall be close to the vessel and capable of being safely operated in an emergency.

G.5 For vessels of more than 50 t capacity a remotely controlled shut off valve, with a mechanical, pneumatic or electrical position indicator shall be fitted before or after the first manual locking shut off valve connected to the liquid phase of the filling and supply pipes. The remotely controlled valve shall operate in a fall safe mode. The fittings must be designed so that they continue to function to the necessary extent at the temperatures to be expected in the event of a self produced fire.

G.6 For vessels of more than 5 t and not more than 50 t capacity a remotely controlled shut-off valve shall be fitted before or after the first manual shut-off valve connected to the liquid phase of the supply pipes.

G.7 For vessels of more than 50 t capacity the first shut off fitting in the filling and supply pipe for the liquid phase shall be designed as a welded outer fitting of fire-safe quality or as an inner fitting.

G.8 The secondary means of isolation may be within the user installation and shall provide an equivalent level of protection.

G.9 Vessels shall be equipped with safety devices against overfilling (level limiter). Vessels with a capacity of more than 50 t shall be equipped with two independent safety devices protecting against overfilling, whereby one such safety device may be incorporated in the level indicator. The two devices protecting against overfilling should operate with different measuring methods.

G.10 Because of the risk of fire and explosion, consideration shall be given in the design of the installation to the provision of:

a) upward venting stacks, means of preventing water blockage or freezing and duplicate stacks;

b) leak-tight piping and equipment.

Annex H
(informative)

Relief devices

H.1 All relief devices, blowndown and purge valves should be connected to a venting system that discharges the contents safely.

H.2 All materials used should be compatible with the specific flammable fluid under consideration.

H.3 All valves and equipment should be suitable for use with the specific cryogenic flammable fluid under consideration.

H.4 The design of the static vessel and its installation should ensure, by the provision of suitable vents, that flammable gas cannot accumulate in cabinets, etc.

H.5 All metallic components of the static vessels should be electrically continuous. The whole installation should be provided with earthing devices so that the resistance to earth is less than 10 Ω.

H.6 In the particular case of liquid hydrogen the possibility of air condensing on uninsulated coldparts should be considered.

H.7 The liquid fill line secondary isolation valve should be either a non-return valve or fail-closed automatic shut off valve.

H.8 Arrangements allowing the vessel (initially) and the loading/filling pipework system to be purged with a non-flammable/non-oxidising gas.

Annex I
(normative)

Outer jacket relief devices

I.1 Field of application

This annex covers the requirements for design, manufacture and testing of pressure protection devices required on outer jackets of vacuum insulated cryogenic vessels in order to reduce any accidental accumulation of pressure.

I.2 Requirements

I.2.1 General

The device shall be either a relief plate/plug or a bursting disc.

Bursting disc devices shall be in accordance with ISO 4126-2.

I.2.2 Design

The pressure protection device shall be capable of withstanding full vacuum and all demands of normal vessel operation including its own mass acceleration during transportation.

The set pressure and the open relieving area are specified in 10.2.6.2. Consideration shall be given to prevention of blocking of the device by insulation materials during operation.

The plate or plug of a relief plate/plug type device shall be designed and installed such that it cannot harm personnel when ejected.

I.2.3 Materials

The pressure protection devices shall be resistant to normal atmospheric corrosion. The materials of construction shall be suitable for the range of ambient temperatures expected in service.

I.2.4 Testing

Relief plate/plug type relieving devices shall not require testing other than a prototype test to verify the set pressure.

Burst disc assemblies shall be tested in accordance with ISO 4126-2.

I.2.5 Inspection

Relief plate/plug type devices shall be subjected to an inspection programme that ensures compliance with the drawings or specification.

Bursting discs shall be inspected in accordance with ISO 4126-2.

I.2.6 Marking

Bursting discs shall be certified and marked in accordance with ISO 4126-2.

Other pressure protection devices shall be marked with this International Standard, i.e. ISO 21009-1.

Annex J
(informative)

Increased material property for austenitic stainless steel

The European Union pressure equipment directive (PED) stipulates three methods for ensuring that a material is suitable for pressure equipment. However, within the European agreement for transport of dangerous goods by road (ADR), for austenitic steels, the specified minimum values according to the material standard may be exceeded by up to 15 % if these higher values are attested in the inspection certificate. This method has been used successfully for a number of years.

K is the minimum value at 20 °C taken from the material standard.

Higher values of K may be used provided that the following conditions are met:

— the material manufacturer should guarantee compliance with this higher value, in writing, when accepting the order;

— the increased properties are verified by testing each rolled plate or coil of the material to be delivered;

— the increased properties are attested in the inspection certificate.

In the case of austenitic stainless steels the specified minimum value may be exceeded by up to 15 % provided this higher value is attested in the inspection certificate.

In addition, for austenitic stainless steel a strength value obtained in work hardened material may be used in the design provided this value and the requirements of 11.3.1 are maintained in the finished component. Requirements for welding of work hardened austenitic stainless steels are given in 11.6.4.3.

The value of E (Young's modulus) at 20 °C should be used in calculation.

Annex K
(normative)

Base materials

Table K.1 — Pressure vessels

Specification No	Material Grade	Material Number
EN 10028-4	X8Ni9	1.5662 (HT 640 & HT 680)
EN 10222-3	X8Ni9	1.5662
DIN 17457 – DIN 17458	X6CrNiTi 18-10	1.4541
DIN 17457 – DIN 17458	X6CrNiNb 18-10	1.4550
DIN 17457 – DIN 17458	X6CrNiMoTi 17-12-2	1.4571
DIN 17457 – DIN 17458	X2CrNiMo 18-14-3	1.4435
DIN 17457 – DIN 17458	X2CrNiMo 17-13-3	1.4436
DIN 17457 – DIN 17458	X2CrNiMoN 17-13-5	1.4439
EN 10028-3	P275NL1	1.0488
EN 10028-3	P275NL2	1.1104
EN 10028-3	P355NL1	1.0566
EN 10028-3	P355NL2	1.1106
EN 10028-3	P460NL1	1.8915
EN 10028-3	P460NL2	1.8916
EN 10028-5	P 355ML1	1.8832
EN 10028-5	P355ML2	1.8833
EN 10028-5	P420ML1	1.8835
EN 10028-5	P420ML2	1.8828
EN 10028-5	P460ML1	1.8837
EN 10028-5	P460ML2	1.8831
EN 10028-7	X2CrNi18-9	1.4307
EN 10028-7	X2CrNi19-11	1.4306
EN 10028-7	X2CrNIN18-10	1.4311
EN 10028-7	X5CrNiN19-9	1.4315
EN 10028-7	X5CrNi18-10	1.4301
EN 10028-7	X2CrNIN23-4	1,4362
EN 10028-7	X3CrNiMo17-12-2	1.4401
EN 10028-7	X2CrNiMo17-12-2	1.4404
EN 10028-7	X2CrNiMoN17-11-2	1.4406
EN 10028-7	X2CrNiMoN17-13-3	1.4429
EN 10028-7	X2CrNiMoN22-5-3	1.4462
EN 10028-7	X2CrNiMoCuN25-6-3	1.4507

Table K.1 (*continued*)

Specification No	Material Grade	Material Number
EN 10028-7	X2CrNiMoN25-7-4	1.4410
EN 10028-7	X2CrNiMoCuWN25-7-4	1.4501
EN 10222-5	X2CrNi18-9	1.4307
EN 10222-5	X5CrNi18-10	1.4301
EN 10222-5	X5CrNiMo17-12-2	1.4401
EN 10222-5	X2CrNiMo17-12-2	1.4404
EN 10088-3	X2CrNi19-11	1.4306
EN 10088-3	X2CrNiN18-10	1.4311
EN 10088-3	X5CrNi18-10	1.4301
EN 10088-3	X5CrNiMo17-12-2	1.4401
EN 10088-3	X2CrNiMo17-13-2	1.4404
EN 10088-3	X2CrNiMoN17-12-2	1.4406
EN 10088-3	X2CrNiMo17-13-3	1.4429
JIS G 3127	SL9N520	1.5662
JIS G 4303-4305	SUS304	1.4301
JIS G4303-4305	SUS304L	1.4307
JIS G4303-4305	SUS316	1.4401
JIS G4303-4305	SUS316L	1.4404
JIS G 4317-4320	SUS321	1.4541
JIS G 4317-4320	SUS347	1.4550
JIS G 4317-4320	SUS316	1.4401
JIS G 4317-4320	SUS316L	1.4404
JIS G 4317-4320	SUS317L	1.4439
JIS G 4313-4315	SUS304	1.4301
JIS G 4313-4315	SUS304L	1.4307
JIS G 4313-4315	SUS304N1	1.4406
JIS G4313-4315	SUS304LN	1,4311
JIS G4317-4320	SUS316L	1.4406
JIS G4315-4315	SUS316LN	1.4429
SA/A-240	304LN	S 30453
SA/A-240	304N	S 30451
SA/A-240	316LN	S 31653
SA/A-240	316N	S 31651
SA/A-240	201LN	S 20153
SA/A-240	201-1	S 20100
SA/A-666	201-1	S 20100
SA/A-240	201-2	S 20100
SA/A-666	201-2	S 20100

Table K.1 (*continued*)

Specification No	Material Grade	Material Number
SA/A-240	201L	S 20103
SA/A-666	201L	S 20103
SA/A-479	316LN	S 31653
SA/A-479	316N	S 31651
SA/A-240	XM-29	S 24000
SA/A-479	XM-29	S 24000
SA/A-479	304	S 30400
SA/A-479	304L	S 30403
SA/A-240	304	S 30400
SA/A-240	304L	S 30403
SA/A-479	304LN	S 30453
SA/A-479	304N	S 30451
SA/A-240	XM-19	S 20910
SA/A-479	XM-19	S 20910
SA/A-479	—	S 21800
SA/A-353	—	K 81340
SA/A-553	I	K 81340
SA/A-522	I	K 81340
SA/A-553	2	K 71340
SA/A-351	—	—
SA/A-516	55	K 01800
SA/A-516	60	K 02100
SA/A-516	65	K 02403
SA/A-516	70	K 02700
SA/A-517	E	K 21604
SA/A-517	F	K 11576
SA/A-612	—	K 02900
A-276	201LN	S 20153
A-276	304	S 30400
A-276	304L	S 30403
A-276	304LN	S 30453
A-276	304N	S 30451
A-276	316	S 31600
A-276	316L	S 31603
A-276	316LN	S 31653
A-276	316N	S 31651
A-276	201	S 20100
A-276	XM-29	S 24000

Table K.1 (*continued*)

Specification No	Material Grade	Material Number
A-276	XM-19	S 20910
A-276	—	S 21800
SB/B-209	5083	A 95083
SB/B-221	5083	A 95083
SB/B-209	6061	A 96061
SB/B-221	6061	A 96061
SB/B-211	6061	A 96061
SB/B-308	6061	A 96061
SB/B-209	3003	A 93003
SB/B-209	5052	A 95052
SB/B-211	3003	A 93003
SB/B-221	3003	A 93003
B-221	6063	A 96063
B-133	—	—
B-16	—	—

NOTE SA/SB prefix to specification number refers to ASME specifications; A/B prefix to specification number refers to ASTM specifications.

When materials to ASME specifications are unavailable, materials to the same specification number with a prefix of A/B (ASTM) may be used.

All ASME/ASTM specification numbers listed below are equally acceptable with a suffix M.

Table K.2 — Piping and pipe fittings

NOTE Piping and pipe fittings to ASTM standards are seamless.

Specification N°	Material grade	Material number
SA/A-312	TP 316L	S 31603
SA/A-358	TP 316L	S 31603
SA/A-249	TP 316L	S 31603
SA/A-409	TP 316L	S 31603
SA/A-688	TP 316L	S 31603
SA/A-813	TP 316L	S 31603
SA/A-814	TP 316L	S 31603
SA/A-249	TP 316	S 31600
SA/A-312	TP 316	S 31600
SA/A-358	TP 316	S 31600
SA/A-409	TP 316	S 31600
SA/A-688	TP 316	S 31600
SA/A-813	TP 316	S 31600
SA/A-814	TP 316	S 31600
SA/A-249	TP 316LN	S 31653
SA/A-312	TP 316LN	S 31653
SA/A-358	TP 316LN	S 31653
SA/A-688	TP 316LN	S 31653
SA/A-249	TP 316N	S 31651
SA/A-312	TP 316N	S 31651
SA/A-358	TP 316N	S 31651
SA/A-688	TP 316N	S 31651
SA/A-813	TP 316N	S 31651
SA/A-814	TP 316N	S 31651
SA/A-249	TPXM-29	S 24000
SA/A-312	TPXM-29	S 24000
SA/A-688	TPXM-29	S 24000
SA/A-249	TP 304L	S 30403
SA/A-312	TP 304L	S 30403
SA/A-358	TP 304L	S 30403
SA/A-409	TP 304L	S 30403
SA/A-688	TP 304L	S 30403
SA/A-813	TP 304L	S 30403
SA/A-814	TP 304L	S 30403
SA/A-249	TP 304	S 30400
SA/A-334	8	K 81340

Table K.2 (*continued*)

Specification N°	Material grade	Material number
SA/A-333	8	K 81340
SA/A-312	TP 304	S 30400
SA/A-358	TP 304	S 30400
SA/A-409	TP 304	S 30400
SA/A-688	TP 304	S 30400
SA/A-813	TP 304	S 30400
SA/A-814	TP 304	S 30400
SA/A-249	TP 304LN	S 30453
SA/A-312	TP 304LN	S 30453
SA/A-358	TP 304LN	S 30453
SA/A-688	TP 304LN	S 30453
SA/A-813	TP 304LN	S 30453
SA/A-814	TP 304LN	S 30453
SA/A-249	TP 304N	S 30451
SA/A-312	TP 304N	S 30451
SA/A-358	TP 304N	S 30451
SA/A-688	TP 304N	S 30451
SA/A-813	TP 304N	S 30451
SA/A-814	TP 304N	S 30451
SA/A-312	TP 321	S 32100
SA/A-249	TP 321	S 32100
SA/A-358	TP 321	S 32100
SA/A-409	TP 321	S 32100
SA/A-813	TP 321	S 32100
SA/A-814	TP 321	S 32100
SA/A-213	TP 316L	S 31603
SA/A-312	TP 316L	S 31603
SA/A-430	FP 316	S 31600
SA/A-213	TP 316	S 31600
SA/A-312	TP 316	S 31600
SA/A-376	TP 316	S 31600
SA/A-213	TP 316LN	S 31653
SA/A-312	TP 316LN	S 31653
SA/A-376	TP 316LN	S 31653
SA/A-430	FP 316N	S 31651
SA/A-213	TP 316N	S 31651
SA/A-312	TP 316N	S 31651
SA/A-376	TP 316N	S 31651

Table K.2 (*continued*)

Specification N°	Material grade	Material number
SA/A-182	F 316L	S 31603
SA/A-336	F 316L	S 31603
SA/A-403	316L	S 31603
SA/A-182	F 316	S 31600
SA/A-336	F 316L	S 31600
SA/A-403	316	S 31600
SA/A-182	F 316LN	S 31653
SA/A-336	F 316LN	S 31653
SA/A-403	316LN	S 31653
SA/A-182	F 316N	S 31651
SA/A-336	F 316N	S 31651
SA/A-403	316N	S 31651
SA/A-182	F 304L	S 30403
SA/A-336	F 304L	S 30403
SA/A-403	304L	S 30403
SA/A-182	F 304	S 30400
SA/A-336	F 304	S 30400
SA/A-403	304	S 30400
SA/A-182	F 304LN	S 30453
SA/A-336	F 304LN	S 30453
SA/A-403	304LN	S 30453
SA/A-182	F 304N	S 30451
SA/A-336	F 304N	S 30451
SA/A-403	304N	S 30451
SA/A-552	I	K 81340
SA/A-420	WPL8	K 81340
SA/A-213	TP 304L	S 30403
SA/A-213	TP 304	S 30400
SA/A-213	TP 304LN	S 30453
SA/A-213	TP 304N	S 30451
SA/A-430	FP 304	S 30400
SA/A-430	FP 304N	S 30451
SA/A-376	TP 304LN	S 30453
SA/A-376	TP 304	S 30400
SA/A-376	TP 304N	S 30451
SA/A-376	TP 316LN	S 31653
SA/A-376	TP 316	S 31600
SA/A-376	TP 316N	S 31651

Table K.2 (*continued*)

Specification N°	Material grade	Material number
SA/A-213	TP 321	S 32100
SA/A-333	1	K 03008
SA/A-333	2	—
SA/A-333	3	K 31918
SA/A-333	4	K 11267
SA/A-333	5	—
SA/A-333	6	K 03006
SA/A-333	7	K 21903
SA/A-333	8	K 81340
SA/A-333	9	K 22035
SA/A-333	10	—
SA/A-333	11	—
SA/A-105	—	K 03504
SA/A-350	LF-1	K 03504
SA/A-350	LF 2	K 03009
SA/A-350	LF 3	K 03011
SA/A-350	LF 5	K 13050
SA/A-350	LF 6	—
SA/A-350	LF 9	K 22036
SA/A-350	LF 787	—
SA/A-106	A	K 02501
SA/A-106	B	K 03006
SA/A-106	C	K 03501
SB/B-247	5083	A 95083
SB/B-247	6061	A 96061
SB/B-241	5083	A 95083
SB/B-241	6061	A 96061
SB/B-210	6061	A 96061
SB/B-241	3003	A93003
SB/B-75	—	—
A-511	TP 304	S 30400
A-511	TP 304L	S 30403
A-511	TP 316L	S 31603
A-511	TP 316	S 31600
A-511	TP 321	S 32100
A-351	304	S 30400
A-351	316	S 31600
A-351	304L	S 30403

Table K.2 (*continued*)

Specification N°	Material grade	Material number
A-351	316L	S 31603
A-269	TP 316L	S 31603
A-269	TP 316	S 31600
A-269	TP 316LN	S 31653
A-269	TP 316 N	S 31651
A-269	TP 304L	S 30403
A-269	TP 304	S 30400
A-269	TP 304LN	S 30453
A-269	TP 304N	S 30451
A-269	TP 321	S 32100
A-269	TPXM 29	S 24000
A-632	TP 304L	S 30403
A-632	TP 304	S 30400
A-632	TP 316L	S 31603
A-632	TP 316	S 31600
A-632	TP 321	S 32100
A-733	TP 304L	S 30403
A-733	TP 304	S 30400
A-733	TP 316L	S 31603
A-733	TP 316	S 31600
NF A 49-117	TUZ2CN18-10	—
NF A 49-147	TUZ2CN18-10	—

Annex L
(normative)

Cylindrical shells subject to external pressure (pressure on the convex surface) — Calculation

This annex gives two calculation methods which are equally recognised and which give comparable results.

L.1 Method 1

Calculations are performed for elastic buckling and for plastic deformation. The lowest calculated pressure, p_e, or p_p, shall not be less than the external design pressure.

NOTE 1 The buckling length, l_b, is the maximum length of the shell between two reinforcing elements (see Figure 2).

NOTE 2 For vessels with dished ends, the buckling length starts at a distance of 1/3 the head depth from the tangent line.

L.1.1 Cylindrical shells

L.1.1.1 Elastic buckling

Calculations are performed using the following formula:

$$p_e = \frac{E}{S_k}\left\{\frac{20}{\left(n^2-1\right)\times\left[1+\left(n/Z\right)^2\right]^2}\times\frac{s-c}{D_a}+\frac{80}{12\times\left(1-\nu^2\right)}\times\left[n^2-1+\frac{2n^2-1-\nu}{1+\left(n/Z\right)^2}\right]\times\left[\frac{s-c}{D_a}\right]^3\right\}$$

where $Z = 0{,}5\ \pi\ D_a/l_b$ and n is an integer equal to or greater than 2 and greater than Z, so determined that the value for P_e is a minimum. n denotes the number of lobes produced by the buckling process which may occur at the circumference in the event of failure. The number of lobes can be estimated using the following approximation equation:

$$n = 1{,}63\times\left(\frac{D^3{}_a}{l_b{}^2\left(s-c\right)}\right)^{0{,}25}$$

For tubes and pipes, calculations may be performed using the following simplified formula:

$$p_e = \frac{E}{S_k}\times\frac{20}{1-\nu^2}\times\left(\frac{s-c}{D_a}\right)^3$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at test pressure with a safety factor of 0,74 S_k.

L.1.1.2 Plastic deformation

When $D_a/l_b \leqslant 5$

$$p_p = \frac{20K}{S_p} \times \frac{s-c}{D_a} \times \frac{1}{1 + \frac{1.5u\left(1-0,2D_a/l_b\right)D_a}{100(s-c)}}$$

When $D_a/l_b > 5$

The higher pressure obtained using the equations below shall not be less than the external design pressure.

$$p_p = \frac{20K}{S_p} \frac{(s-c)}{D_a}$$

$$p_p = \frac{30K}{S_p} \times \left(\frac{s-c}{l_b}\right)^2$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at test pressure with a safety factor of 0,74 S_k.

L.1.1.3 Stiffening rings

For the inner and outer vessels:

$$I \geqslant 0{,}124 \times \frac{p \times D_a^3 \times \sqrt{D_a \times (s-c)}}{10E}$$

$$A \geqslant \frac{0{,}75\, pD_a}{10K} \sqrt{D_a(s-c)}$$

The moment of inertia I is relative to the neutral axis of the reinforcing elements cross-section parallel to the shell axis (see axis X - X in Figures 1 and 2). Narrow and high reinforcing elements of the kind shown in Figure 1 may undergo severe buckling. Where the height of the element is greater than 8 times its width, a more accurate calculation shall be made.

Where stiffening rings are joined to the shell by means of intermittent welds, the fillet welds at each side shall cover at least one third of the shell circumference and the number of weld discontinuities must be at least 2n. The number of buckling lobes n is obtained as indicated in L.1.1.1.

If a test pressure, p_T, higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted values of I and A are not less than those determined by above formulae at a pressure of 0,74 p_T.

L.1.2 Dished ends

L.1.2.1 Elastic buckling

There is adequate resistance to elastic buckling when:

$$p \leqslant 3{,}66 \frac{E}{S_k} \left(\frac{s\text{-}c}{R}\right)^2$$

L.1.2.2 Plastic deformation

Resistance to plastic deformation is determined by using L.1.1.2 with the appropriate safety factor, S_p, defined in 10.3.2.4 and 10.3.3.4.

L.2 Method 2

L.2.1 Cylindrical shells

The calculated critical collapsing pressure, p_cr, shall not be less than the external design pressure.

$$p_\mathrm{cr} = \frac{13E\left(\dfrac{s}{D_\mathrm{a}}\right)^{2,5}}{\left(\dfrac{l_\mathrm{b}}{D_\mathrm{a}}\right) - 0,45\sqrt{\dfrac{s}{D_\mathrm{a}}}}$$

NOTE 1 The buckling length, l_b, is the length of the unsupported cylindrical shell (see Figures 1 and 2). Other forms of stiffening may be used.

NOTE 2 For vessels with dished ends, the buckling length starts at the junction of the cylinder and the knuckle region of the dished end (see Figure 3).

L.2.1.1 Elastic buckling

Cylindrical shells

If $\dfrac{l_\mathrm{b}}{D_\mathrm{a}} > 1{,}537\dfrac{(1-\nu^2)^{0,25}}{\left(\dfrac{s}{D_\mathrm{a}}\right)^{0,5}}$

$$pe = \frac{E}{S_\mathrm{k}}\left(\frac{20}{1-\nu^2}\right)\left(\frac{s-c}{D_\mathrm{a}}\right)^3$$

If $\dfrac{l_\mathrm{b}}{D_\mathrm{a}} \leqslant 1{,}537\dfrac{(1-\nu^2)^{0,25}}{\left(\dfrac{s}{D_\mathrm{a}}\right)^{0,5}}$

$$pe = \frac{24-2E\left(\dfrac{s}{D_\mathrm{a}}\right)^{2.5}}{S_\mathrm{k}\left(1-\nu^2\right)^{0.75}\left[\left(\dfrac{l_\mathrm{b}}{D_\mathrm{a}}\right)-0,45\left(\dfrac{s}{D_\mathrm{a}}\right)^{0,5}\right]}$$

L.2.1.2 Stiffening Rings

Each stiffening ring shall have a minimum moment of inertia as determined by either of the following formulae:

$$I = \frac{S_{\mathrm{k}} p D_{\mathrm{a}}^{3} l_{\mathrm{b}}}{280E}$$

or

$$I' = \frac{S_{\mathrm{k}} p D_{\mathrm{a}}^{3} l_{\mathrm{b}}}{218E}$$

where

$I =$ Required moment of inertia of the stiffening ring cross section about its neutral axis parallel to the axis of the shell

$I' =$ Required moment of inertia of the combined ring-shell cross section about its neutral axis parallel to the axis of the shell

The required moment of inertia of the combined ring-shell section shall be maintained completely around the circumference of the cylinder unless the adequacy of the shell to carry the required critical collapse pressure is demonstrated through a finite element analysis method verified with scale model tests of each type of design.

L.2.1.2.1 If stiffening rings are used in designing the cylindrical portion (shell) of the inner vessel or vacuum jacket for external pressure, each ring must be attached to the shell by fillet welds. Stiffening ring attachment welds on the outside of the vacuum jacket must be continuous. All other ring attachment welds may be intermittent. Care should be taken in the design of ring attachments to minimize localized areas of buckling. Where intermittent welds are used, the total length of welds on each side of the ring must be at least one-third of the shell circumference, or if welded on one side, two-thirds of the shell circumference. The intermittent attachment welds shall be uniformly distributed and if welded on both sides, shall be staggered.

A portion of the shell may be included when calculating the moment of inertia of the ring. The effective width of shell plate, x, on each side of the attachment to the ring is given by the formula:

$$x = 0{,}78\left(\frac{D_{\mathrm{a}}(s-c)}{2}\right)^{0.5}$$

L.2.1.2.2 Where a stiffening ring consists of a closed section having two webs attached to the shell, the shell plate between the webs shall be included up to the limit of twice the value defined in L.2.1.2.1. The flange of the section, if not a standard structural shape, is subject to the same limitation, with x based on D_{a} and s of the shell. The closed section between the ring and shell must be provided with means to equalize pressure to the space occupied by the ring.

L.2.1.2.3 Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring or parts (webs) of one stiffening ring. If the stiffeners or webs of stiffeners should be so located that the maximum permissible effective shell sections overlap on either or both sides of a stiffener or web, the effective shell section for that stiffener or web shall be shortened by one-half of each overlap.

L.2.1.2.4 Length of the attachment weld segments shall not be less than 50 mm and shall have a maximum clear spacing between toes of adjacent weld segments of $8s$ for external rings and $12s$ for internal rings.

The number of intermittent attachment welds on each ring shall be at least $2n$ where n, the number of buckling lobes is given by

$$n = 1{,}63\left(\frac{D_a^{\,3}}{l_b^{\,2}\left(s-c\right)}\right)^{0{,}25}$$

Size of the fillet weld leg size shall be not less than the smallest of the following:

a) 6 mm;

b) s, shell thickness;

c) b, web thickness of the stiffener ring.

L.2.2 Dished ends

The calculated pressure p_c shall not be less than the external design pressure.

$$p_c = 2{,}5E\left(\frac{s}{R}\right)^2$$

Plastic deformation

p_p, the pressure derived from the following formula shall be higher than p obtained for elastic buckling using the above formula.

$$p_p = \frac{20K_{20}\left(s-c\right)}{S_p\left(R+s\right)}$$

For ellipsoidal ends $(R + s)$ may be taken as B_oD_a. Where B_o is obtained from Table L.1.

Table L.1 — Values of spherical radius factor B_o for ellipsoidal end with pressure on convex side

$\frac{D_a}{2h_o}$	3.0	2.8	2.6	2.4	2.2	2.0	1.8	1.6	1.4	1.2	1.0
B_o	1.36	1.27	1.18	1.08	0.99	0.90	0.81	0.73	0.65	0.57	0.50
NOTE Interpolation permitted for intermediate values.											

Annex M
(normative)

Design of openings in cylinders, spheres and cones — Calculation

M.1 General

This annex gives two calculation methods which are equally recognised and which give comparable results.

M.2 Method 1

Where the material property, K, of the reinforcement is lower than that of the shell, the cross section of pad reinforcement and the thickness of nozzle reinforcement shall be reduced by the ratio of K values. In the case of a shell subjected only to internal pressure, with a row of nozzles joined to the shell by fully penetrating welds, it is not necessary to calculate the individual reinforcement required for each nozzle. However the thickness of the shell to resist internal pressure shall be calculated using the least value of the weakening factor of either v_A obtained from the following equation or v.

Openings shall also be reinforced according to the following relationship:

$$\frac{p}{10}\left(\frac{A_p}{A\sigma}+\frac{1}{2}\right)\leqslant\frac{K}{S}$$

which is based on equilibrium between the pressurized area, A_p, and the load bearing cross sectional area, $A\sigma$. The wall thickness obtained from this relationship shall be not less than the thickness of the unpierced shell.

The pressurized area, A_p, and the load bearing cross sectional area, $A\sigma$, which equals $A\sigma_0 + A\sigma_1 + A\sigma_2$, are obtained from Figures M.1 to M.5.

M.2.1 Symbols and units

For the purposes of 10.3.6.7, the following symbols apply in addition to those given in Clause 4:

b	width of pad, ring or shell reinforcement	mm
h	thickness of pad-reinforcement	mm
l	ligament (web) between two nozzles	mm
l'_s	length of nozzle reinforcement outstandings	mm
m	protruding length of nozzle	mm
s	length of nozzle reinforcement in stand	mm
s_A	required wall thickness at opening edge	mm
s_S	wall thickness of nozzle	mm
t	In this context: centre-to-centre distance between two nozzles	mm

M.2.2 Field of application

Round openings and the reinforcement of round openings in cylinders, spheres and cones within the following limits:

$$0{,}002 \leqslant \frac{(s-c)}{D_a} \leqslant 0{,}1$$

$$\frac{(s-c)}{D_a} < 0{,}002 \text{ is acceptable if } \frac{d_i}{D_a} \leqslant \frac{1}{3}$$

These rules only apply to cones if the wall thickness is determined by the circumferential stress.

NOTE 1 Additional external forces and moments are not covered by this subclause and are to be considered separately where necessary.

NOTE 2 These design rules permit plastic deformations of up to 1 % at highly stressed local areas during pressure test. Openings should therefore be carefully designed to avoid abrupt changes in geometry.

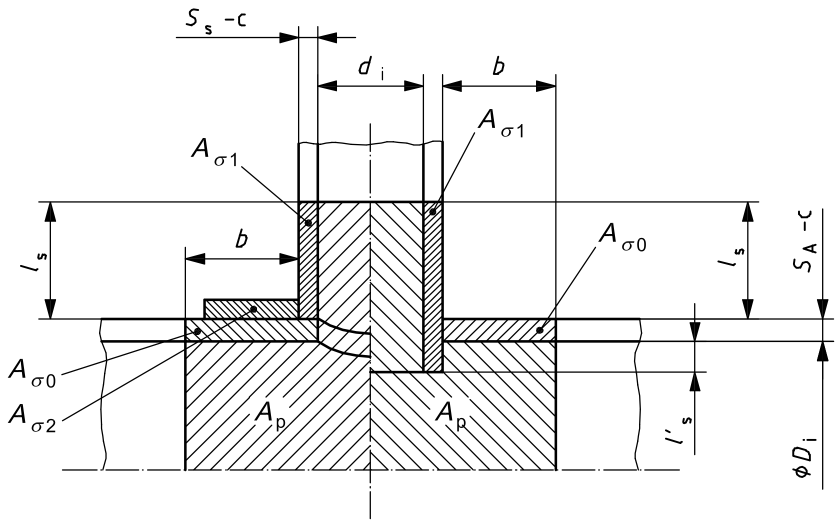


Figure M.1 — Calculation scheme for cylindrical shells

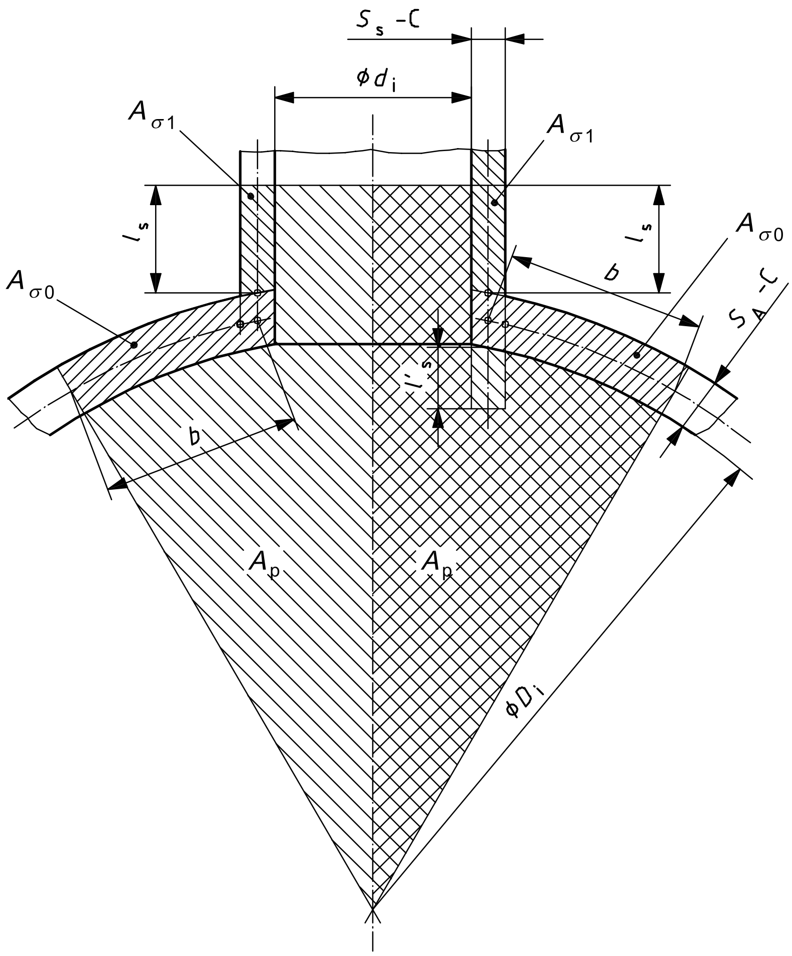


Figure M.2 — Calculation scheme for spherical shells

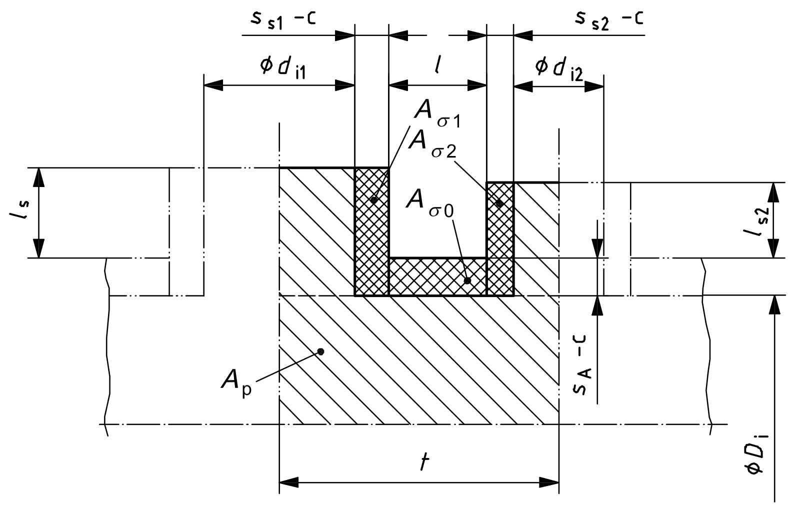


Figure M.3 — Calculation scheme for adjacent nozzles in a sphere or in a longitudinal direction of a cylinder

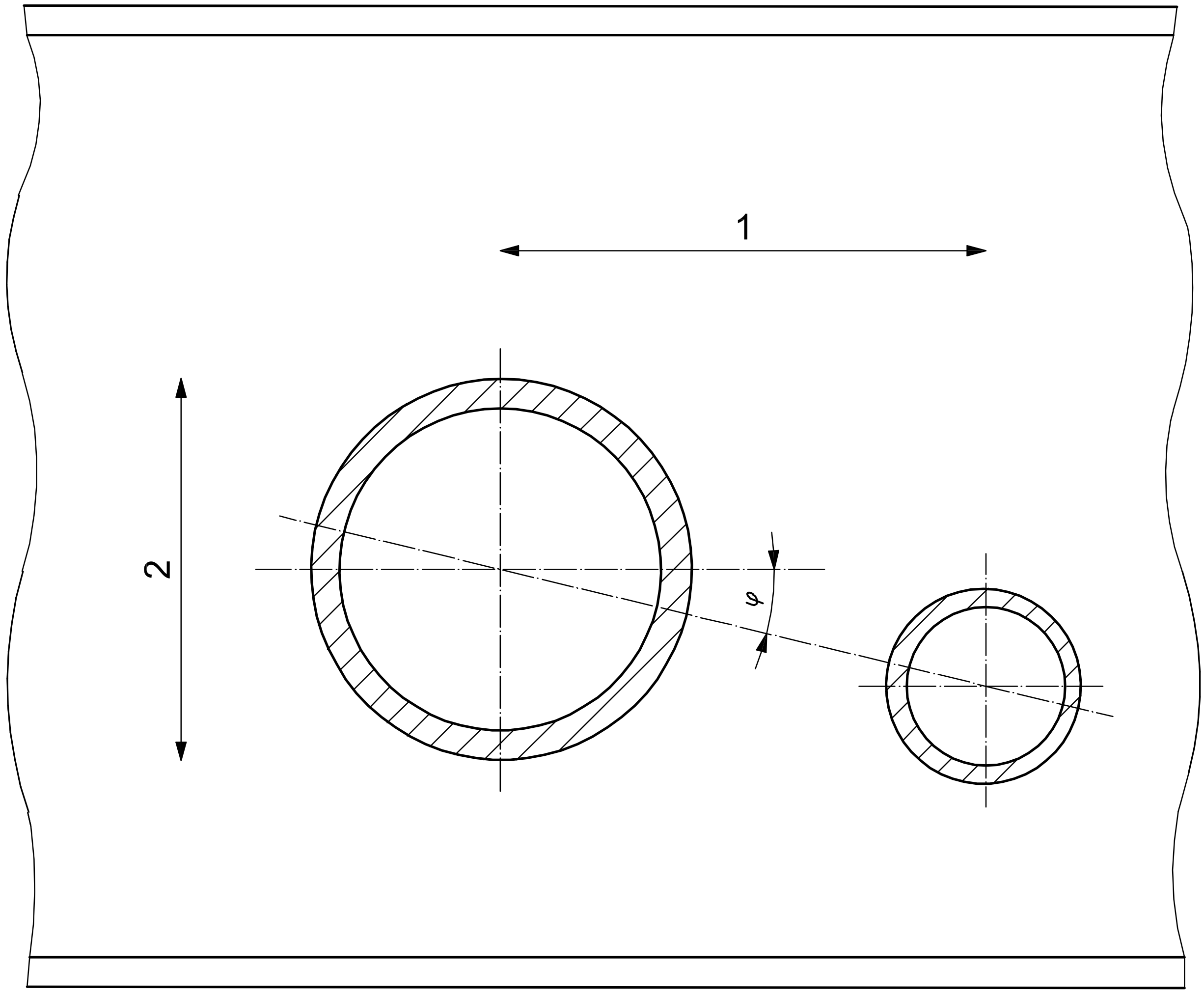


Figure M.4 — Openings between longitudinal and circumferential directions

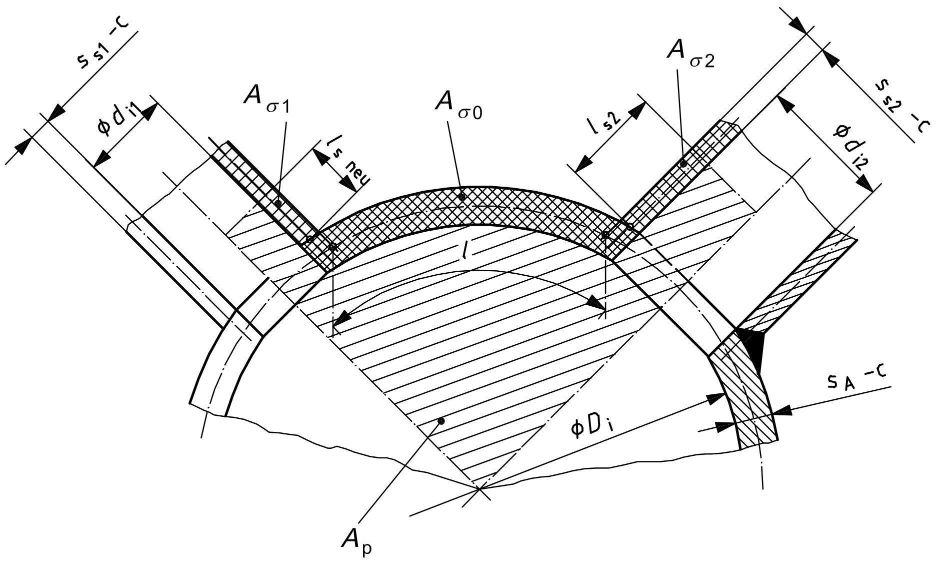


Figure M.5 — Calculation scheme for adjacent nozzles in a sphere or in a circumferential direction of a cylinder

The maximum extent of the load bearing cross sectional area shall be not more than b as defined in the formula below for shells and l_S as defined in the formulae for nozzles, as appropriate.

The protrusion of nozzles, l_S, may be included as a load bearing cross sectional area up to a maximum length of

$$l'_S = 0{,}5\ l_S$$

The restrictions of 10.3.6.7.7 and 10.3.6.7.8 shall be observed.

If the material property, K_1, K_2, etc. of the reinforcing material is lower than that of the shell the dimensions shall comply with:

$$\left(\frac{K}{S} - \frac{p}{20}\right) A_{\sigma 0} + \left(\frac{K_1}{S} - \frac{p}{20}\right) A_{\sigma 1} + \left(\frac{K_2}{S} - \frac{p}{20}\right) A_{\sigma 2} \geqslant \frac{p}{10} A_{\mathrm{p}}$$

$$b = \left[\left(D_{\mathrm{i}} + S_{\mathrm{A}} - c \right) \left(S_{\mathrm{A}} - c \right) \right]^{0{,}5}$$

with a minimum of 3 S_A (see Figures 17, 18 and 19).

For calculation purposes S_A shall be limited to not more than twice the actual wall thickness.

The thickness of pad reinforcement in accordance to Figure 19 preferably shall not be more than the actual wall thickness to which the pad is attached.

Internal pad reinforcement is not allowed.

The width of the pad reinforcement may be reduced to b_1 provided the pad thickness is increased to h_1 according to:

$$b_1 \times h_1 \geqslant b \times h$$

and the limits given above are observed.

M.3 Method 2

The symbols used in this clause are defined as follows:

Subscript n refers to nozzle and v refers to vessel

A_{r} = total cross-sectional area of reinforcement required in the plane under consideration, mm^2 (see Figure M.6) (includes consideration of nozzle area through shell if $\frac{K_{20n}}{K_{20v}} < 1{,}0$)

A_1 = area in excess thickness in the vessel wall available for reinforcement, mm^2 (see Figure M.6) (includes consideration of nozzle area through shell if $\frac{K_{20n}}{K_{20v}} < 1{,}0$)

A_2 = area in excess thickness in the nozzle wall available for reinforcement, mm^2 (see Figure M.6)

A_3 = area available for reinforcement when the nozzle extends inside vessel wall, mm^2 (see Figure M.6)
A_{41}, A_{42}

A_{43} = cross-sectional area of various welds available for reinforcement, mm^2 (see Figure M.6)

A_5 = cross-sectional area of material added as reinforcement, mm^2 (see Figure M.6)

c = corrosion allowance, mm

D_p = outside diameter of reinforcing element, mm (actual size of reinforcing element may exceed the limits of reinforcement; however, credit cannot be taken for any material outside these limits)

d = finished diameter of circular opening or finished dimension (chord length at mid-surface of thickness excluding excess thickness available for reinforcement) of nonradial opening in the plane under consideration, mm [see Figure M.6]

v = 1 (see definitions for s_r and s_n)

v_1 = 1 when an opening is in the solid plate or in a full penetration butt joint; or

= joint efficiency when any part of the opening passes through any other welded joint

h_i = distance nozzle projects beyond the inner surface of the vessel wall, mm. Extension of the nozzle beyond the inside surface of the vessel wall is not limited; however, for reinforcement calculations, credit shall not be taken for material outside the limits of reinforcement

R_n = inside radius of the nozzle under consideration, mm

$\frac{K_{20}}{S}$ = allowable stress value in tension, N/mm^2

$\frac{K_{20n}}{S}$ = allowable stress in nozzle, N/mm^2

$\frac{K_{20v}}{S}$ = allowable stress in vessel, N/mm^2

$\frac{K_{20p}}{S}$ = allowable stress in reinforcing element, N/mm^2

f_r = strength reduction factor, not greater than 1.0

$f_{r1} = \frac{K_{20n}}{K_{20v}}$ for nozzle wall inserted through the vessel wall

f_{r1} = 1.0 for nozzle wall abutting the vessel wall

f_{r3} = lesser of K_{20n} or K_{20p}/K_{20v}

$f_{r4} = K_{20p}/K_{20v}$

s = specified vessel wall thickness in the corroded condition, (not including forming allowances), mm. For pipe it is the nominal thickness less manufacturing under-tolerance allowed in the pipe specification.

s_p = thickness or height of reinforcing element, mm

s_i = nominal thickness of internal projection of nozzle wall, mm

s_r = required thickness, mm, of a seamless shell based on the circumferential stress, or of a formed end, for the designated pressure using $v = 1$.

Reinforcement shall be provided in amount and distribution such that the area requirements for reinforcement are satisfied for all plains through the centre of the opening and normal to the vessel surface. For a circular opening in a cylindrical shell, the plane containing the axis of the shell is the plane of greatest loading due to pressure. Not less than half the required reinforcement shall be on each side of the centre line of single openings.

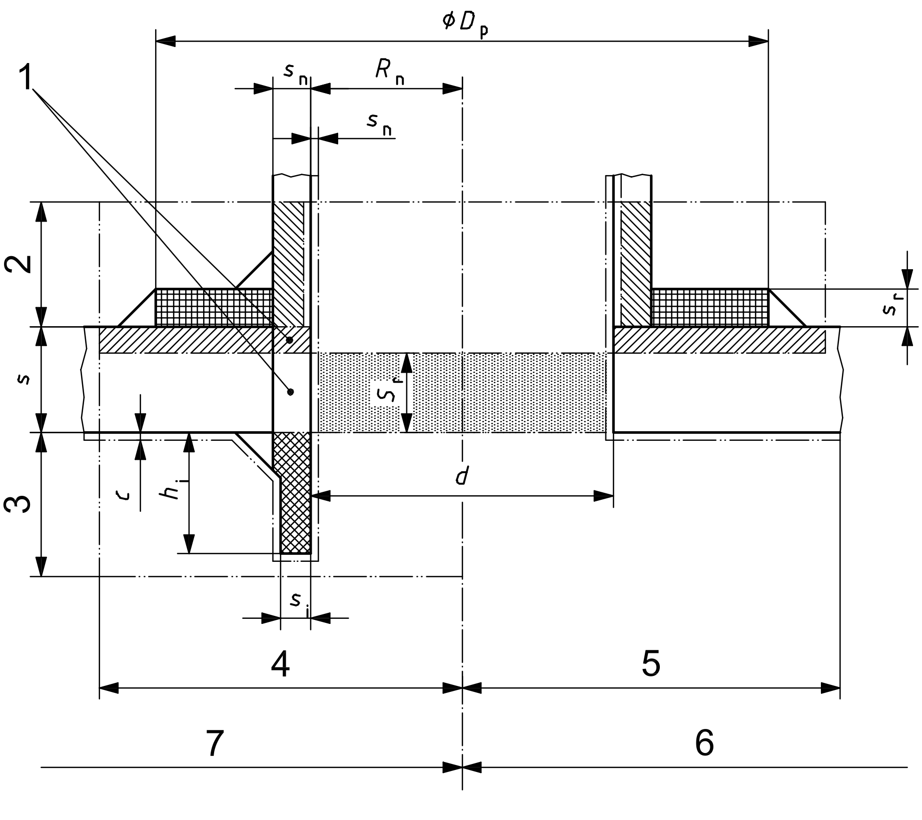
The total cross-sectional area of reinforcement, A_r, required in any given plane through the opening for a shell or dished end under internal pressure shall not be less than

$$A_r = d\, s_r + 2\, s_n\, s_r\, (1 - f_{r1})$$

The reinforcement required for openings in vessels under external pressure need be only 50 % of that required for the above formula.

When two openings are spaced so that their limits of reinforcement overlap, the two openings shall be reinforced in the plane connecting the centres with a combined reinforcement that has an area not less than the sum of the areas required for each opening. No portion of the cross section is to be considered as applying to more than one opening, nor to be considered more than once in a combined area.

The limits of reinforcement and the details of calculations for the reinforcement area to be provided are shown in Figure M.1.



Key

1 GENERAL NOTE: includes consideration of areas if $K_{2on}/K_{2ov} < 1.0$ (both sides of C_L)

2 $0.78\sqrt{R_n s_n}$

3 $0.78\sqrt{R_n s_i , h_i}$ use smaller value

4 $\underbrace{d \text{ or } n + s_n + s}_{\text{use larger value}}$

5 $\underbrace{d \text{ or } n + s_n + s}_{\text{use larger value}}$

6 For nozzle wall abutting the vessel wall

7 For nozzle wall inserted through the vessel wall

without reinforcing element

	$= A_r = ds_r + 2s_n s_r (1 - f_{r1})$	Area required
	$= A_1 = d(v_1 s - s_r) - 2s_n(v_1 s - s_r)(1 - f_{r1})$ $= 2(s + s_n)(v_1 s - s_r) - 2s_n(v_1 s - s_r)(1 - f_{n1})$	Area available in shell; use larger value
	$= A_2 = 1.56\sqrt{R_n s_n (s_n - s_m) f_{r1}}$	Area available in nozzle projecting outward
	$= A_3 = 1.56\sqrt{R_n s_n (s_i - f_{r1})}$ $= 2h_i s_i f_{r1}$	Area available in inward nozzle; use smaller value
	$= A_{41} =$ outward nozzle weld $= (leg)^2 f_{r2}$ $= A_{43} =$ inward nozzle weld $= (leg)^2 f_{r2}$ if $A_1 + A_2 + A_3 + A_{41} + A_{43} + > A_r$ if $A_1 + A_2 + A_3 + A_{41} + A_{43} + < A_r$	Area available in outward weld Area available in inward weld; opening is adequately reinforced; opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

with reinforcing element added

	$A =$ same as A, above $A_1 =$ same as A_1, above $A_2 =$ same as A_2, above $A_3 =$ same as A_3, above	Area required Area available Area available in nozzle projecting outward Area available in inward weld
	$= A_{41} =$ outward nozzle weld $= (leg)^2 f_{r3}$	Area available in outward weld
	$= A_{42} =$ outer element weld $= (leg)^2 f_{r4}$ $= A_{43} =$ inward nozzle weld $= (leg)^2 f_{r2}$	Area available in outer weld Area available in inward weld
	$= A_5 = (D_p - d - 2s_n) s_p f_{r4}$ if $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 > A_r$	Area available in element Opening is adequately reinforced

Figure M.6 — Nomenclature and formulas for reinforced openings

M.4 Ring or pad reinforcement or increased shell thickness

If the actual wall thickness of the cylinder or sphere is less than the required thickness, s_A, at the opening, the opening is adequately reinforced if the wall thickness, s_A, is available around the opening over a width of:

$$b = \sqrt{(D_i + s_A - c)(s_A - c)}$$

with a minimum of 3 s_A (see Figure M.7).

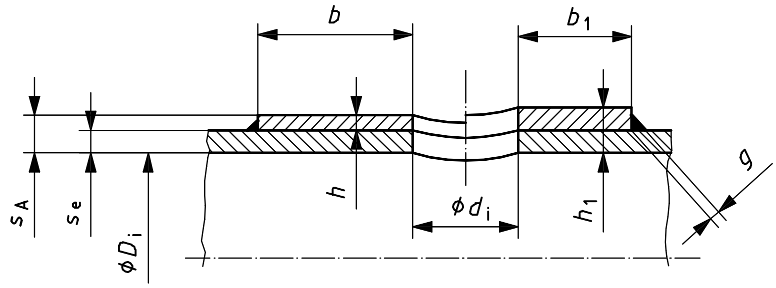


Figure M.7

For calculation purposes s_A shall be limited to not more than twice the actual wall thickness.

The thickness of pad reinforcement in accordance with Figure 18 preferably shall be not more than the actual wall thickness to which the pad is attached.

Internal pad reinforcement is not allowed.

The width of the pad reinforcement may be reduced to b_1 provided the pad thickness is increased to h_1 according to:

$$b_1 \times h_1 \geqslant b \times h$$

and the limits given above are observed.

M.5 Reinforcement by increased nozzle thickness

For calculation purposes s_s shall be not more than twice the actual wall thickness.

The thickness of the nozzle shall preferably be not greater than twice the actual shell thickness.

The wall thickness, s_A, at the opening shall extend over a width, b, in accordance with the formula with a minimum of 3 s_A.

The limits of reinforcement normal to the vessel wall are:

— for cylinders and cones, $l_s = 1{,}25\sqrt{(d_i + s_s - c)(s_s - c)}$

— for spheres, $l_s = \sqrt{(d_i + s_s - c)(s_s - c)}$

The length, l_s, may be reduced to l_{s1} provided that the thickness, s_s, is increased to s_{s1} according to the following:

$$l_{s1} \times s_{s1} \geqslant l_s \times s_s$$

and the limits given above are observed.

M.5.1 Reinforcement by a combination of increased shell and nozzle thicknesses

Shell and nozzle thicknesses may be increased in combination for the reinforcement of openings (Figure 20).

For the calculation of reinforcement M.2 and M.3 shall be applied together. The increase in shell thickness may be achieved by an actual increase in shell thickness or the addition of a pad.

M.5.2 Multiple openings

Multiple openings are regarded as single openings provided the distance l between two adjacent openings, Figures M.3 and M.5, complies with:

$$l \geqslant 2\sqrt{(D_i + s_A - c)(s_A - c)}$$

If l is less than required by this above formula a check shall be made to determine whether the cross section between openings is able to withstand the load acting on it. Adequate reinforcement is available if the requirement of the formulae given in M.2 as appropriate is met.

Where adjacent openings in a cylinder are arranged intermediately between the longitudinal and circumferential direction the calculation scheme for the longitudinal direction (Figure M.3) shall be applied, but the part of the pressurized area corresponding to the unpierced cylinder $\left(\frac{tD_i}{2}\right)$ may be reduced with an arrangement factor $=$ 0,5 (1 $+$ cos^2 φ).

See Figure M.4 for angle, φ.

Nozzles joined to the shell in line by full penetration welds with the wall thickness calculated for internal pressure only may be designed with a weakening factor:

$$v_A = \frac{(t - d_i)}{t}$$

If the nozzles are not attached by full penetration welds, D_a shall be used in the above formula.

Annex N
(normative)

Design of ends for internal pressure

N.1 Torispherical end knuckle thickness and hemispherical end to shell junction thickness

N.1.1 Thickness requirement

The required thickness of the knuckle region and hemispherical end junction shall be:

$$s = \frac{D_a p \beta}{40\left(\frac{K}{S}\right) v} + c$$

For hemispherical ends a β value of 1,1 shall be applied within the distance x from the tangent line joining the end to the cylinder,

where $x = 0{,}5\sqrt{R(s-c)}$

β is taken from Figure 5 for 10 % torispherical ends and from Figure 6 for 2:1 torispherical ends as a function of $\frac{(s-c)}{D_a}$. Iteration is necessary.

When there are openings outside the area 0,6 D_a the required thickness is found using β from Figures 5 and 6 using the appropriate curve for the relevant value of $\frac{d_i}{D_a}$.

The β factor is derived from the lower curves of Figures 5 and 6 when there are no openings outside the area 0,6 D_a.

D_a is the diameter of the end as shown in Figures 4 a) and b).

N.1.2 Alternative thickness requirement

The required thickness of the knuckle region and hemispherical end junction shall be:

$$s = \frac{pRM}{20\frac{K_{20}\nu}{S} - 0{,}2p}$$

where

$$M = 0{,}25\left(3 + \sqrt{\frac{R}{r}}\right)$$

R is the inside crown radius;

r is the inside knuckle radius.

Dished ends designed for normal operation under internal pressure (pressure on concave side shall have $R \leqslant D_a$ and $r \geqslant 0{,}06\ D_a$ but in no case less than 3 s. Dished ends of vacuum jackets are not required to meet the above restrictions on R and r except $r \geqslant 3\ s$.

N.1.3 Requirement for domed ends welded together from crown and knuckle components

If a domed end is welded together from crown and knuckle components, the joint shall be at a sufficient distance, x, from the knuckle.):

x shall be the larger of the following:

— 100 mm;

— $0{,}78\sqrt{R(s)}$ if the crown and knuckle are of different thickness, where s is the thickness of the knuckle component;

— 3,5 s is applicable in case crown and knuckle are of equal thickness.

v = 1,0 may be used, if the scope of testing corresponds to that specified for a design stress level equal to the permissible design stress level or in the case of one-piece ends.

N.2 Elliptical ends

The required thickness at the thinnest point after forming of elliptical ends under pressure on the concave side shall be determined by:

$$s = \frac{pD_{\mathrm{i}}B}{20\dfrac{K_{20}v}{S} - 0{,}2p}$$

where $B = \dfrac{1}{6}\left[2 + \left(\dfrac{D_{\mathrm{i}}}{2h}\right)^2\right]$

and h is equal to one-half of the length of the minor axis of the ellipsoidal end, or the inside depth of the ellipsoidal end measured from the tangent line (end-bend line), mm.

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