INTERNATIONAL STANDARD

ISO 21009-1

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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
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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

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

This document was prepared by Technical Committee ISO/TC 220, *Cryogenic vessels*.

This second edition cancels and replaces the first edition (ISO 21009-1:2008), which has been technically revised.

The main changes are as follows:

— correction of the formulae;

— Clauses 11 and 12 have been revised;

— Annex C has been aligned with the modification performed in the other ISO/TC 220 design standards.

A list of all parts in the ISO 21009 series can be found on the ISO website.

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

Cryogenic vessels — Static vacuum-insulated vessels —

Part 1:
Design, fabrication, inspection and tests

1 Scope

This document 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 document applies to static vacuum-insulated cryogenic vessels for fluids and does not apply to vessels designed for toxic fluids.

This document also gives guidance for static vacuum-insulated cryogenic vessels designed for a maximum allowable pressure of not more than 0,5 bar.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 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 9328-4, *Steel flat products for pressure purposes — Technical delivery conditions — Part 4: Nickel-alloy steels with specified low temperature properties*

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

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

ISO 10474:2013, *Steel and steel products — Inspection documents*

ISO 14732, *Welding personnel — Qualification testing of welding operators and weld setters for mechanized and automatic welding of metallic materials*

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

ISO 15614-1:2017, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 1: Arc and gas welding of steels and arc welding of nickel and nickel alloys*

ISO 17636-1, *Non-destructive testing of welds — Radiographic testing of fusion-welded joints — Part 1: X- and gamma-ray techniques with film*

ISO 17636-2, *Non-destructive testing of welds — Radiographic testing of fusion-welded joints — Part 2: X- and gamma-ray techniques with digital detectors*

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

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

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

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 degrees C*

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

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

EN 13445-3, *Unfired pressure vessels — Design*

ASME *Boiler and Pressure Vessel Code, Section VIII, Division 2: Alternative Rules*

3 Terms and definitions

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

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

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

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

3.1
accessories
service equipment which has a safety-related function with respect to either pressure containment or control or both

EXAMPLE Protective or limiting devices, controlling and monitoring devices, valves and indicators.

3.2
automatic welding
welding in which all operations are performed without welding operator intervention during the process

Note 1 to entry: Manual adjustment of welding variables by the welding operator during welding is not possible.

[SOURCE: ISO 14732:2013, 3.1]

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

Note 1 to entry: 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 1 to entry: This includes totally evaporated liquids and supercritical fluids.

Note 2 to entry: In the ISO 21009 series, 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	
	1 913	Neon, refrigerated liquid
	1 951	Argon, refrigerated liquid
	1 963	Helium, refrigerated liquid
	1 970	Krypton, refrigerated liquid
	1 977	Nitrogen, refrigerated liquid
	2 187	Carbon dioxide, refrigerated liquid
	2 591	Xenon, refrigerated liquid
	3 136	Trifluoromethane, refrigerated liquid
	3 158	Gas, refrigerated liquid, not otherwise specified (NOS)
3° O	Oxidizing gases	
	1 003	Air, refrigerated liquid
	1 073	Oxygen, refrigerated liquid
	2 201	Nitrous oxide, refrigerated liquid, oxidizing
	3 311	Gas, refrigerated liquid, oxidizing, NOS
3° F	Flammable gases	
	1 038	Ethylene, refrigerated liquid
	1 961	Ethane, refrigerated liquid
	1 966	Hydrogen, refrigerated liquid
	1 972	Methane, refrigerated liquid or natural gas, refrigerated liquid, with high methane content
	3 138	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
	3 312	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 shall 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

Note 1 to entry: Documentation consists of:

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

— a short description of the vessel (e.g. including characteristic data);

— 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 (e.g. including characteristic data),

  — a statement that the vessel is in conformity with this document, and

  — the instructions for *normal operation* (3.10).

3.6
gross volume
<*inner vessel*> (3.7) internal volume of the inner vessel determined at minimum design temperature and atmospheric pressure

3.7
inner vessel
pressure vessel intended to contain the *cryogenic fluid* (3.4) to be stored

3.8
manufacturer of the static cryogenic vessel
company that carries out the final assembly, including the final acceptance test, of the *static cryogenic vessel* (3.17)

3.9
maximum allowable pressure
maximum *pressure* (3.13) permissible at the top of the vessel in its normal operating position

3.10
normal operation
intended operation of the vessel either up to the *maximum allowable pressure* (3.9) or subjected to handling loads

Note 1 to entry: Handling loads are exerted on the *static cryogenic vessel* (3.17) in all normal transport operations including, e.g. loading, unloading, pressure loading during transportation, installation.

3.11
outer jacket
gas-tight enclosure which contains the *inner vessel* (3.7) and enables the vacuum to be established

3.12
piping system
tubes, pipes and associated components which can come in contact with *cryogenic fluids* (3.4) including valves, fittings, pressure relief devices, and their supports

3.13
pressure
gauge pressure
pressure relative to atmospheric pressure

3.14
pressure strengthened vessel
pressure vessel, which has been subjected to a calculated and controlled internal pressure (strengthening pressure) after completion

Note 1 to entry: 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 to entry: 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.

3.15
relief plate
plate retained by atmospheric pressure which allows relief of excess internal pressure, generally from the vacuum jacket

3.16
service equipment
accessories, equipment or instruments that will be used to measure the level, to fill or discharge the tank, to vent the tank, to protect the tank against overpressure and to raise the tank pressure and its thermal insulation

Note 1 to entry: The thermal insulation is a vacuum inter-space between the *inner vessel* (3.7) and the *outer jacket* (3.11).

3.17
static cryogenic vessel
thermally insulated vessel intended for use with one or more *cryogenic fluids* (3.4) in a stationary condition

Note 1 to entry: Static cryogenic vessels consist of *inner vessel(s)* (3.7), an *outer jacket* (3.11) and the *piping system* (3.12).

4 Symbols

A	cross sectional area of reinforcing element	mm^2
A	area of reinforcing ring	mm^2
A_s	elongation at fracture	%
b	width of pad, ring or shell reinforcement	mm
C_β	design factors	—
c	allowances for corrosion	mm
D	shell diameter	mm
D_a	outside diameter e.g. of a cylindrical shell	mm
D_{a1}	outside diameter of connected cylinder (see Figure 7)	mm
D_{a2}	outside diameter at effective stiffening (see Figure 9)	mm
D_1, D_2	flat end diameters	mm
D_i	internal diameter e.g. of a cylindrical shell	mm
D_k	design diameter (see Figure 7)	mm
D_s	shell diameter at nozzle (see Figure 8)	mm
d_a	outside diameter of tube or nozzle	mm
d_i	diameter of opening	mm
d_1, d_2	opening diameter	mm
E	Young's modulus	N/mm^2
f	narrow side of rectangular or torispherical plate	mm
H	Safety coefficient for pressure test	—
h	thickness of pad-reinforcement	mm

I	moment of inertia of reinforcing element	mm^4
K	material property used for design (see 10.3.2.3.1)	N/mm^2
K_t	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
L	cone length between effective stiffenings (see Figure 9)	mm
l	ligament (web) between two nozzles	mm
l_b	buckling length	mm
l'_s	length of nozzle reinforcement outstandings	mm
l_s	length of nozzle reinforcement in stand	mm
m	protruding length of nozzle	mm
n	number	—
p	design pressure as defined by 10.2.3.2.1 and 10.3.3.2	bar (or MPa)
p_e	external pressure	bar (or MPa)
p_{e1}	allowable external pressure limited by elastic buckling	bar (or MPa)
p_{e2}	allowable external pressure limited by elastic buckling including reinforcement	bar (or MPa)
p_k	strengthening pressure	bar (or MPa)
p_p	allowable external pressure limited by plastic deformation	bar (or MPa)
p_s	maximum allowable gauge pressure	bar (or MPa)
p_T	test pressure [see 10.2.3.2.3]	bar (or MPa)
R	radius of curvature e.g. inside crown radius of dished end	mm
r	inside radius of knuckle	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	—
s	minimum wall thickness	mm
s_A	required wall thickness at opening edge	mm
s_e	actual wall thickness	mm
s_g	required wall thickness outside corner area	mm
s_l	required wall thickness within corner area	mm
s_S	wall thickness of nozzle	mm

T	temperature	°C
t	wall thickness of nozzle	mm
u	out-of-roundness	—
V	factor indicative of the utilization of the permissible design stress in joints or factor allowing for weakenings	—
ν	Poisson ratio	—
x	(decay-length zone) distance over which governing stress is assumed to act	mm
x_i	characteristic lengths (i = 1,2,3) to define corner area [Figures 7 a) and b) and 10.3.6.5.4]	mm
Z	auxiliary value	—
φ	cone angle	°
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 through 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 documentation, and all supporting documents (including those from subcontractors, if any), taking legal compliance into consideration (e.g. product liability). In addition, the manufacturer shall retain all supporting and background documents (including those from subcontractors, if any) which establish that the vessel conforms to this document.

6 Mechanical loads

6.1 General

The static cryogenic vessel shall resist the mechanical loads mentioned in this clause without such deformation which can affect safety and which can 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;

— mass of vessel when filled to capacity;

— 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:

$p_T \geq H(p_s + 1)$ bar or $[p_T \geq H(p_s + 0{,}1) MPa]$

where

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;

+ 1 (in bar) or [+0,1 (in MPa)] is the allowance for external vacuum.

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

9.1 General

The materials used to manufacture the inner vessels and associated equipment shall meet the requirements defined in 9.2 through 9.3.

9.2 Selection of materials

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

9.2.2 Materials used at low temperatures shall follow the requirements of the relevant ISO 21028-1 and ISO 21028-2; non-metallic materials shall be suitable for operating temperatures and the refrigerated gas.

9.2.3 The base materials, listed in Annex K, subject to meeting the extra requirements given in the main body of this document, are suitable for and may be employed in the manufacture of the cryogenic vessels conforming to this document.

9.3 Inspection certificate

9.3.1 The head and shell material shall be declared by an "inspection certificate 3.1", in accordance with ISO 10474:2013, 5.1, or "inspection certificate 3.2", in accordance with ISO 10474:2013, 5.2, if a specific manufacture qualification is not available.

9.3.2 The material manufactured to a recognized document shall be declared by an "inspection certificate 3.1", in accordance with ISO 10474:2013, 5.1, or "inspection certificate 3.2", in accordance with ISO 10474:2013, 5.2, if a specific manufacture qualification is not available.

9.4 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, it is possible that designs adopting pressure strengthening will not be allowed by the competent 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 through 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 or diameter, or both, of the inner shell;

or, in the event of any change relative to:

— the type of material or grade (e.g. stainless steel 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 (e.g. ambient temperatures, seismic);

— shipping modes (e.g. road, rail, water) of the empty vessel;

— filling and emptying rates;

— range of ambient temperatures, if different from 8 b);

— gross mass;

— details of fastenings in combination with the expected loads from the vessel itself.

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, e.g. grade, class, temper, testing, as relevant;

— applicable material test certificates;

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

— calculations to verify conformance to this document;

— 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 1 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 through 10.2.3.7.

If a detailed fatigue life calculation is required, the evaluation shall be conducted according to EN 13445-3, ASME VIII-2 or equivalent standards/codes under consideration of the imperfections under 11.5 and 12.3.4.

Information about the fatigue life shall be reported to the users in the operating manual.

NOTE 2 Fatigue analysis can be satisfied for existing designs through documented evidence of previous long-term satisfactory service under the same operating conditions.

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:

$$p_{CL} = p_s + p_L + 1 \text{ bar or } [\, p_{CL} = p_s + p_L + 0{,}1\ MPa\,]$$

where p_L is the pressure 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 permissible to reduce the value by 5 % of (p_s + 1)].

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

$$p_{CG} = p_s + 1 \text{ bar or } [\, p_{CG} = p_s + 0{,}1\ MPa\,]$$

NOTE 1 This formula 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 can 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:

$p_\mathrm{T} = H\ (p_s + 1)$ bar [or $p_\mathrm{T} = H\ (p_s + 0{,}1)$ MPa] or see 12.5.1 or

$$p_\mathrm{T} = 1{,}25\ (p_s + p_L + 1)\ \frac{K_{20}}{K_t}\ bar\ [\text{or}\ p_\mathrm{T} = 1{,}25\ (p_s + p_L + 0{,}1)\ \frac{K_{20}}{K_t}\ MPa\ ]$$

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

The 1 bar (or 0,1 MPa) is added to allow for the external vacuum.

NOTE 1 *H* is equal to 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.

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

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 (or 0,1 MPa);

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 0,66 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 0,66 K_{20}. Compressive stresses and stability shall be considered, see for example EN 13445-3 and ASME CODE CASE 2286-5 in combination with AISC Manual of Steel Construction 15th edition. The load conditions and combinations shall take into consideration the applicable regulations of the countries of use.

10.2.3.6 Lifting points

Lifting points shall be suitable for lifting loads under consideration of the load direction initiated of the self-load of the static cryogenic vessel and 10.3.4, which are lifted in accordance with the specified procedure under taking into account of a maximum allowable stress value equal to 0,66 K_{20}.

NOTE The maximum self-load corresponds to the weight of the entire non-corroded pressure vessel, including e.g. fittings (floors, packs), insulation, fire protection, pipelines, catwalks and ladders.

10.2.3.7 Piping and accessories

Piping including valves, fittings and supports shall be designed for the following loads, which 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 (or 0,1 MPa) shall be added;

e) The design shall consider the effects of vibration during transportation.

10.2.4 Inspection openings

Inspection openings are not required in the inner vessel or the outer jacket, provided that the requirements of ISO 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;

NOTE 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.

— fail safely;

— contain redundant features; and

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

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.

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 (0,05 MPa).

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 gross volume with liquid at the filling condition.

10.2.8 Electrical continuity

For all static vessels designed to store flammable and oxidising 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 through 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 competent 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 materials to be used for pressure-bearing parts shall meet the following requirements.

Where the behaviour of a material can be affected by manufacturing processes or operating conditions, to an extent that would adversely affect the safety or service life of the pressure vessel, this shall be taken into consideration when specifying materials.

Adverse effects can arise from manufacturing processes, e.g. degree of cold forming and heat treatment; operating conditions, e.g. hydrogen embrittlement, corrosion, scaling and ageing behaviour of the material after cold forming.

Materials for pressure bearing parts shall conform to the technical delivery conditions in 9.2.

Materials shall be selected to be compatible with anticipated fabrication steps and to be suitable for the internal fluid and external environment. Both normal operating conditions and transient conditions occurring during fabrication transport, testing and operation shall be taken into account when specifying the materials.

The materials should be grouped in accordance with ISO/TR 15608 to relate manufacturing and inspection requirements to generic material types.

Materials have been allocated into these groups in accordance with their chemical composition and properties in view of manufacture and heat treatment after welding.

The material property used in the calculations shall be in accordance with 10.3.2.3.2.

10.3.2.3.2 Material property K_{20}

The material property, K_{20}, to be used in the calculations shall be as follows:

Material	Property *K*	Requirements
austenitic stainless steel	1 % yield strength or σ_k according to Table C.1 or 1 % yield strength x 1,15 according to Annex J	Annex C and only for material according to Table C.1 Further requirements [a]
all other metals	0,2 % yield strength (or 1,0 % if applicable)	R_e/R_m not exceeding 0,625, excluding fine-grained steel and specially heat-treated steel

[a] 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). The 15 % higher values of K_{20} may be used provided this higher value is guaranteed and attested in the inspection certificate and the welding procedures are suitably qualified for these higher properties.

NOTE Upper yield strength can be used.

10.3.2.3.3 Material property used for design 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 10028-7:2017, 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 6)

For external pressure (pressure on the convex surface)

$v = 1,0$

10.3.2.6 Allowances for corrosion, c

$c = 0$

NOTE ISO 21009-2 gives requirements for safe operation for such cryogenic vessels.

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 (0,1 MPa).

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

$S_p = 1,1$

$S_k = 2,0$

For well proven designs (evidence of satisfactory service under the same operating conditions), a factor of safety, S_k, equal to 1,5 is acceptable provided that

a) D is not more than 2 300 mm,

b) l_b is not more than 10 200 mm, and

c) the annular space is perlite insulated.

— spherical region

$S_p = 1,6$

$S_k = 2,0 + 0,001\ 4\ R/s$

— 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.

$c = 0$

External surfaces should be adequately protected against corrosion.

10.3.4 Supports and lifting points

10.3.4.1 Supports for operating condition

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. Due to the dynamic nature of lifting loads, they shall be multiplied by 1,5 for the purpose of designing lifting points.

When designing the inner vessel, the temperature and corresponding mechanical properties of the structural attachment that is 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).

10.3.4.2 Lifting points and supports for installation and transport condition

The lifting points and supports shall be designed for the loads based on transport, installation loads which are dynamic nature by multiplying by factor 1,5 under consideration of 10.2.3.4 and 10.2.3.6 using established structural design methods and safety factors specified in 10.3.2.4 and 10.3.2.5.

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.

Welding seams of piping inside the vacuum space shall be full penetration welds.

The ability to carry out necessary non-destructive testing based on acceptance levels in Table 8 ("Acceptance levels for fatigue loaded vessels") shall be considered.

Piping system shall have sufficient inherent flexibility to prevent the following during their design lifetime:

a) Failure of piping or supports from overstress under consideration of paragraph 10.2.3.1;

b) Leakage at any point in the piping;

c) Detrimental stresses or distortion in the piping or in-line equipment (e.g. valves) or in connected equipment resulting from excessive thrusts and moments in the piping.

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:

$$D_a / D_i \leq 1{,}2$$

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

$$s = \frac{D_a p}{20(K/S)\ v + p} + c$$

for spheres

$$s = \frac{D_a p}{40(K/S)\ v + p} + c$$

In case of p in MPa, conversion shall be considered.

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

10.3.6.2.1 Field of application

Cylinders where

$D_a / D_i \leq 1{,}2$

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)

10.3.6.3.1 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.

10.3.6.3.2 Calculation

Spheres subject to external pressure shall be evaluated in accordance with Annex L, in particular subclauses 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 utilized:

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

b) 10 % torispherical ends where $R = D_a$ and $r = 0{,}1 D_a$

c) 2:1 torispherical ends where $R = 0{,}8\ D_a$ and $r = 0{,}154\ D_a$.

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

NOTE Other end shapes can be used provided suitable calculations are carried out.

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 mm 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 can 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 shall 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 crown region of a torispherical head shall be defined as 0,6 D_a (see Figure 4).

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 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 torispherical ends.

10.3.6.4.4.2 Knuckle thickness

The iteration method of Annex N shall be used for determining the wall thickness of the knuckle region of dished ends and of hemispherical ends, considering Figure 5 and 6.

If there are openings in the area outside 0,6 D_a, these are taken into account by increasing the design factor in the ratio $di/D_a > 0$ according to Figures 5 and 6.

10.3.6.4.4.3 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)

10.3.6.4.5.1 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.

10.3.6.4.5.2 Calculation

See Annex L.

10.3.6.5 Cones subject to internal or external pressure

10.3.6.5.1 Field of application

Cones according to Figure 7 where:

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

and

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

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

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

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

10.3.6.5.2 Openings

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

If $|\varphi| \leq 70°$ design according to 10.3.6.5.5 using an equivalent cylinder diameter of:

$$D_i = \frac{D_s + d_1 |\sin \varphi|}{\cos \varphi}$$

$|\varphi| > 70°$ design according to 10.3.6.5.6.

10.3.6.5.3 Non-destructive testing

All corner joints shall be subject to the examination required for a weld joint factor of 1,0. See Table 6.

10.3.6.5.4 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 formulae:

$$x_1 = \sqrt{D_{a1} (s_l - c)}$$

$$x_2 = 0{,}7 \sqrt{\frac{D_{a1} (s_l - c)}{\cos \varphi}}$$

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

10.3.6.5.5 Internal pressure calculation (pressure on concave surface) $|\varphi| \leq 70°$

a) within corner area

The required wall thickness (s_1) within the corner area is calculated from Figures 10 a) to 10 g) for the large end and Figure 10 h) for the small end of a cone using the following variables:

$$\varphi,\ \frac{pS}{15Kv},\ \text{and}\ \frac{r}{D_{a1}}$$

For a corner joint use the curve for $\frac{r}{D_{a1}} = 0$.

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 10.3.6.5.5 b).

b) outside corner area

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

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

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

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

In case of p in MPa, conversion shall be considered.

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

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

$$s_l = s_g = 0{,}3(D_{a1} - r) \times \frac{|\varphi|}{90} \times \sqrt{\frac{p}{10\left(\frac{k}{S}\right)v}} + c$$

In case of p in MPa, conversion shall be considered.

10.3.6.5.7 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 $\varphi \geq 10°$ 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:

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

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.

$$A \geq \frac{p(D_{a1})^2}{80\left(\frac{K}{S_p}\right)} \tan|\varphi|$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted value of 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 $0{,}5\sqrt{D_{a1}s_1}$ 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 10.3.6.5.5 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.

In case of p in MPa, conversion shall be considered.

10.3.6.6 Flat ends

10.3.6.6.1 Field of application

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

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

and

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

10.3.6.6.2 Openings

Openings are calculated in accordance with 10.3.6.6.3 but with the C factor multiplied by C_A, where C_A is given in Figure 11.

10.3.6.6.3 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 torispherical flat end is

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

where C_E is taken from Figure 13.

In case of p in MPa, conversion shall be considered.

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 torispherical axis.

10.3.6.7.3 Calculation

One of the alternative calculation methods presented in Annex M shall be used. 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.

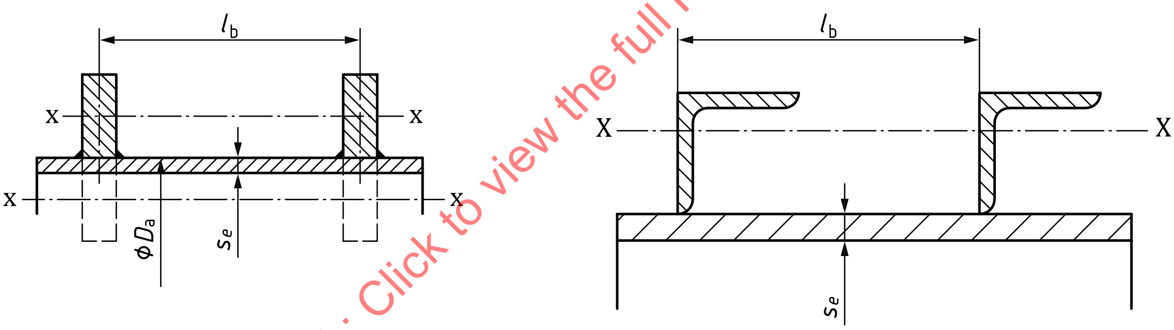
Acceptable calculation methods include:

— finite element;

— finite difference;

— boundary element;

— recognised text books, codes and standards.



a) b)

Figure 1 — Stiffening rings

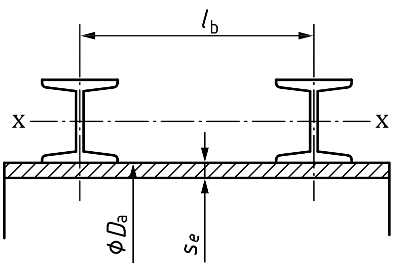


Figure 2 — Sectional materials stiffeners

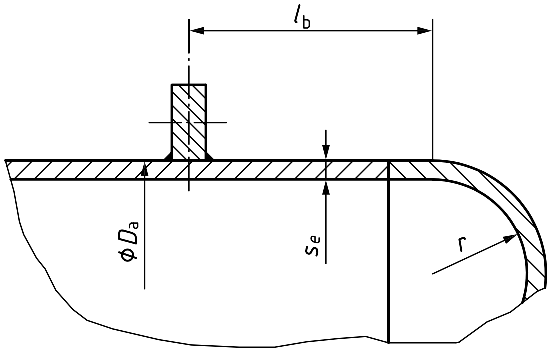
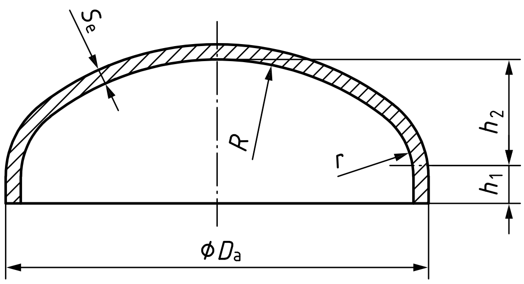
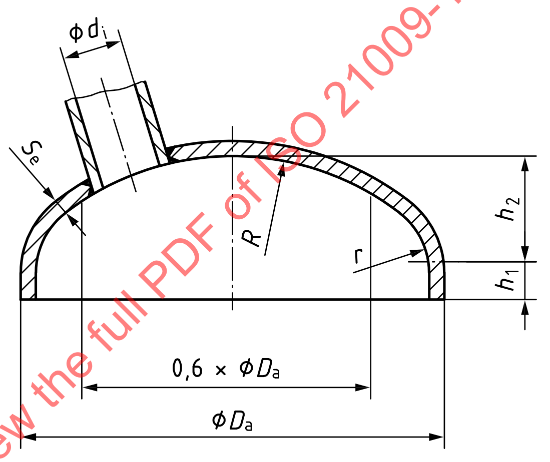


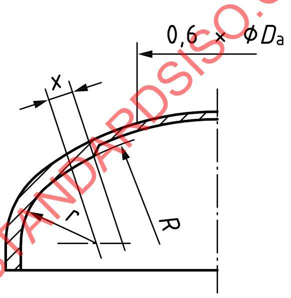
Figure 3 — Dished ends



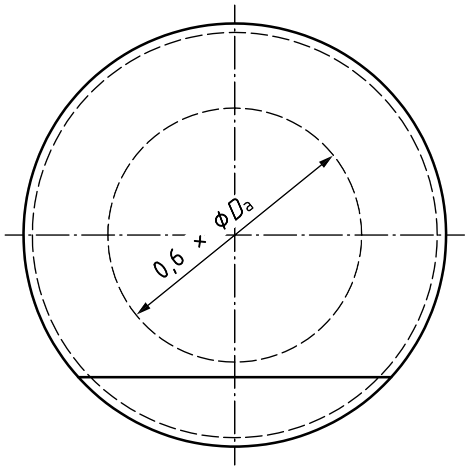
a) Unpierced dished end



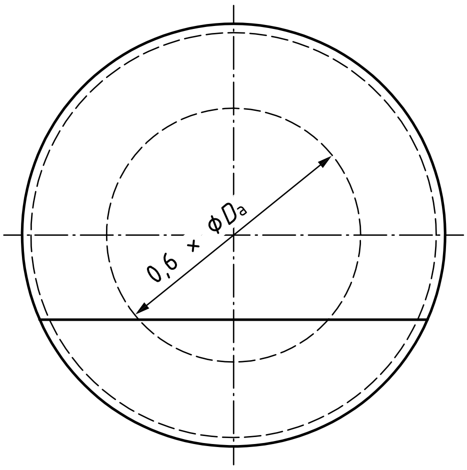
b) Dished end with nozzle



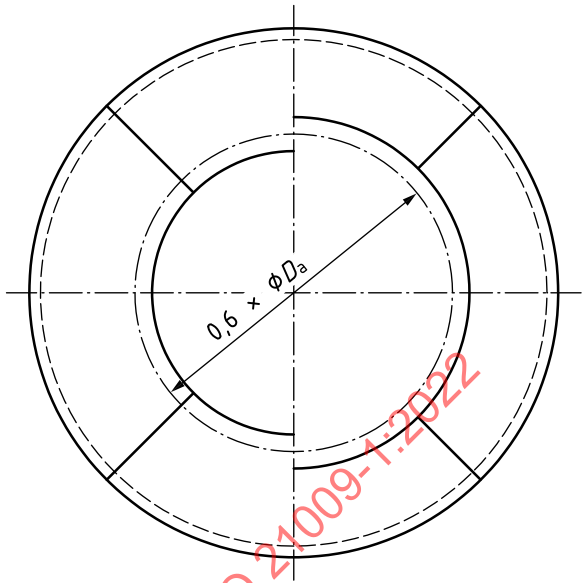
c) End with knuckle and crown of unequal wall thickness



d) Weld outside 0,6, D_a



e) Weld inside 0,6, D_a



f) End welded together from round plate and segments

Figure 4 — Vessel ends and weld positions

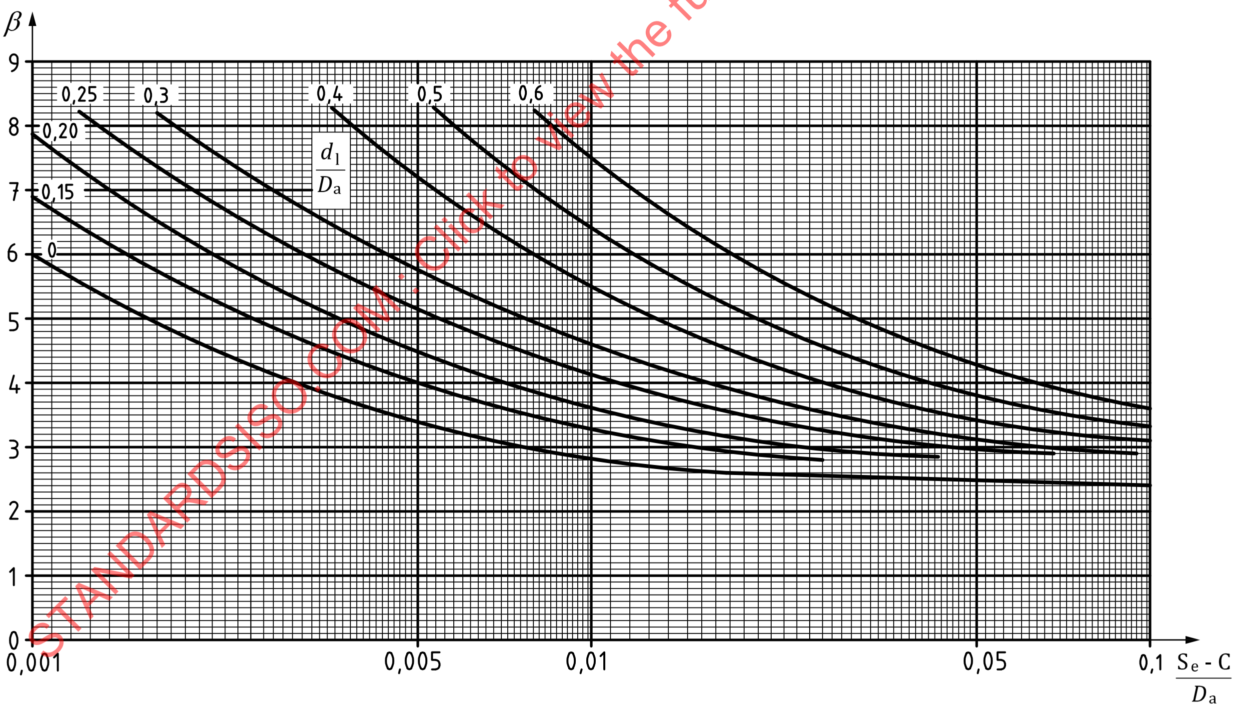


Figure 5 — Design factors, β, for 10 % torispherical dished ends

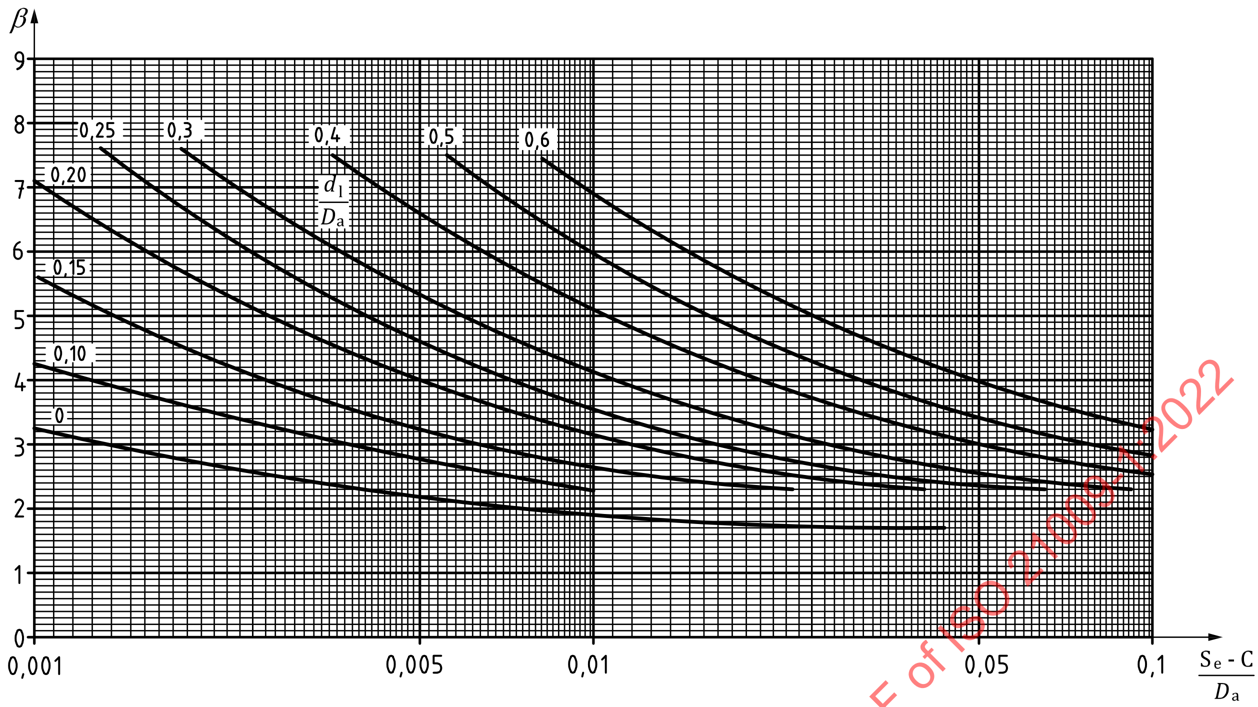
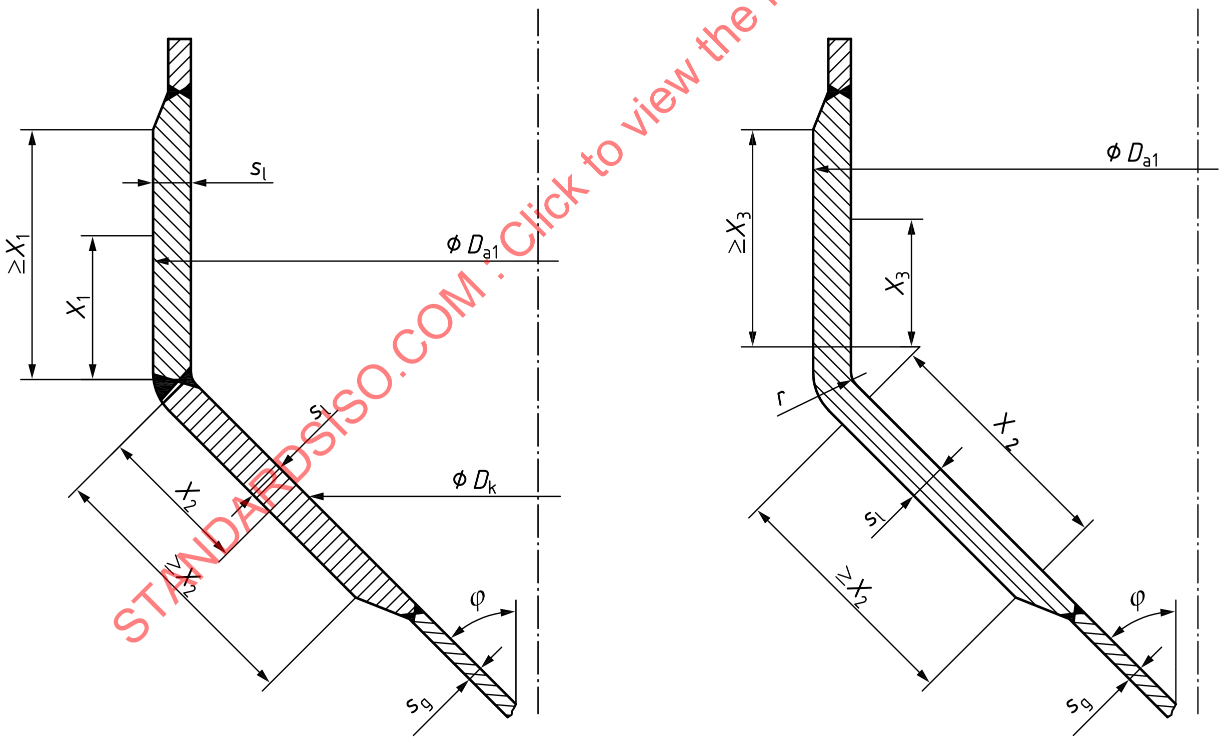
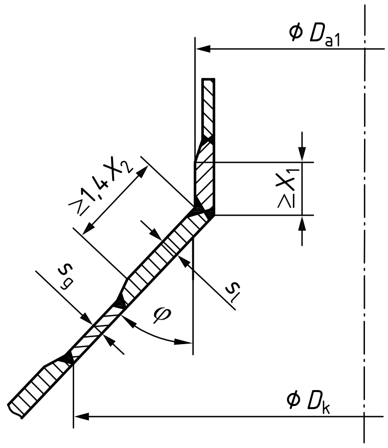


Figure 6 — Design factors, β, for 2:1 torispherical dished ends



a) Geometry of convergent conical shells



b) Geometry of a divergent conical shell

Figure 7 — Geometry of conical shell

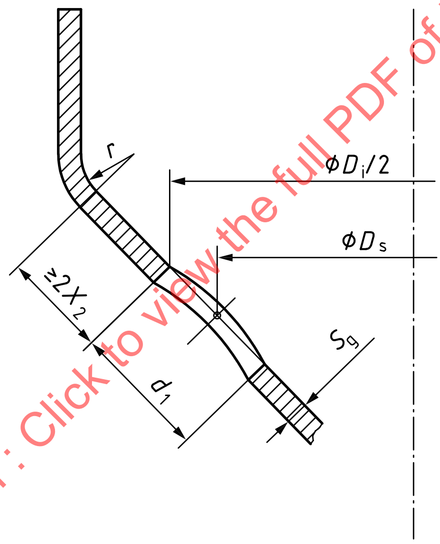


Figure 8 — Geometry of a cone opening

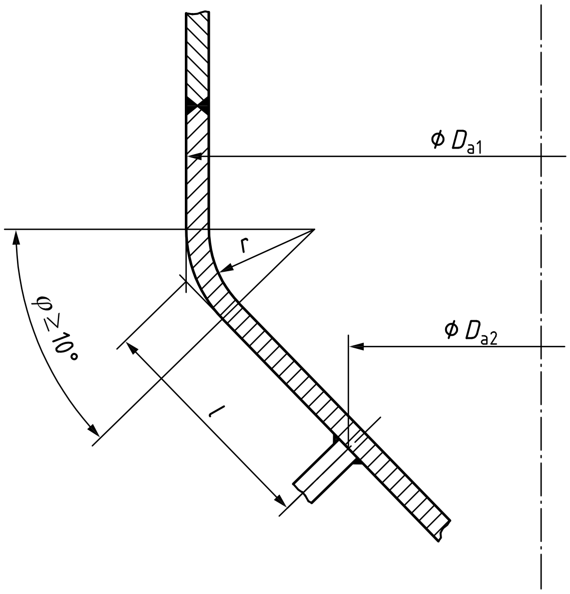
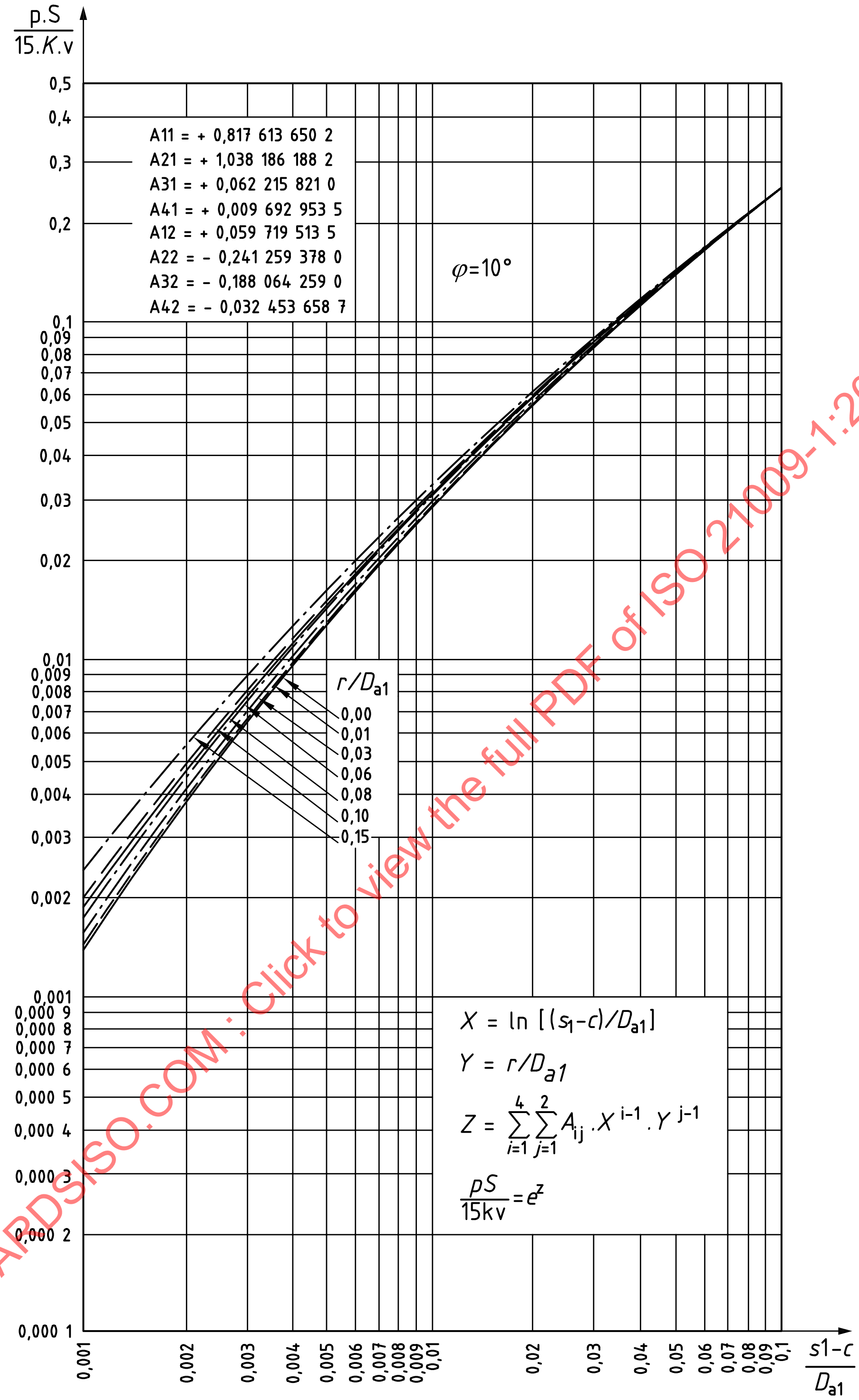
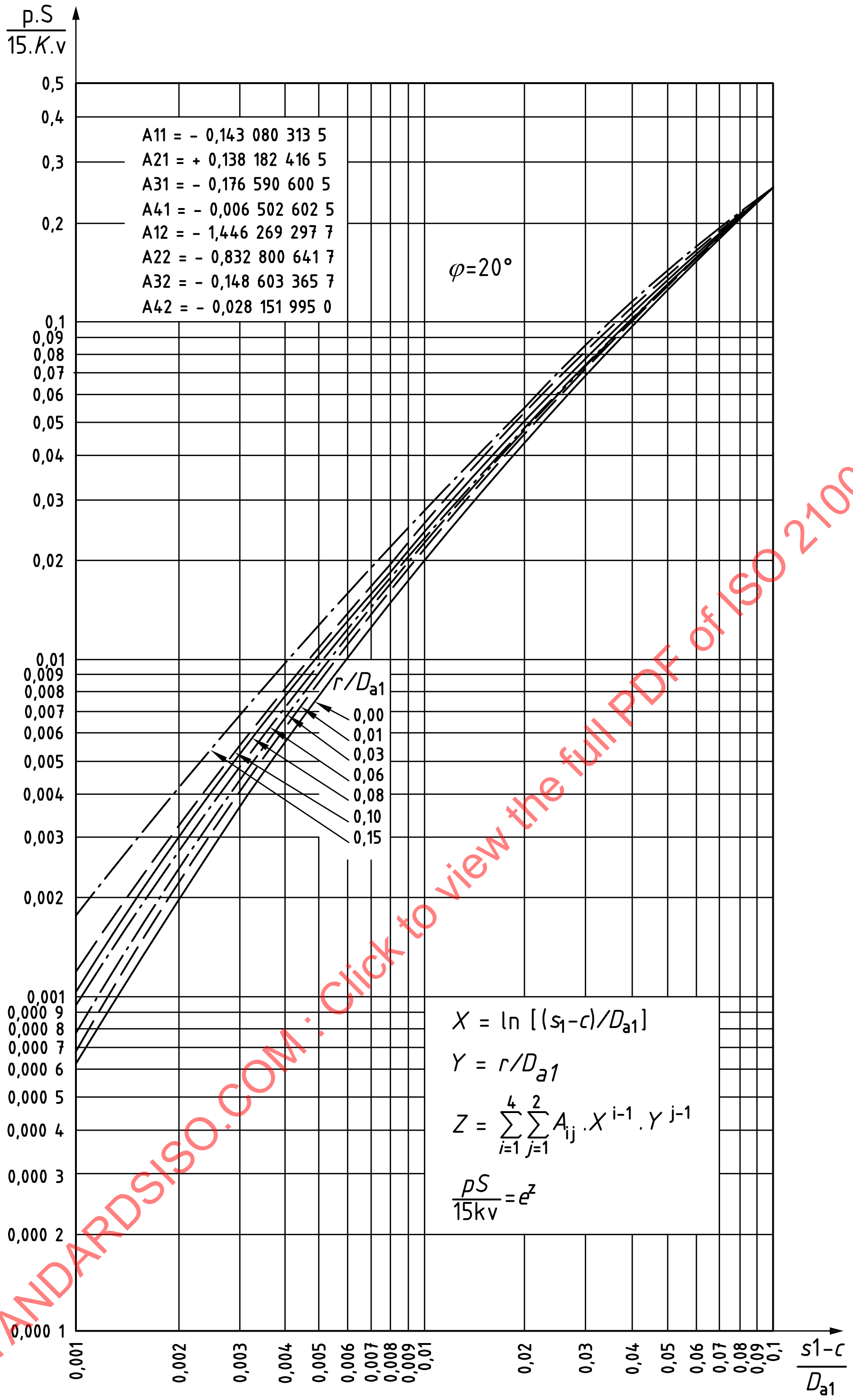


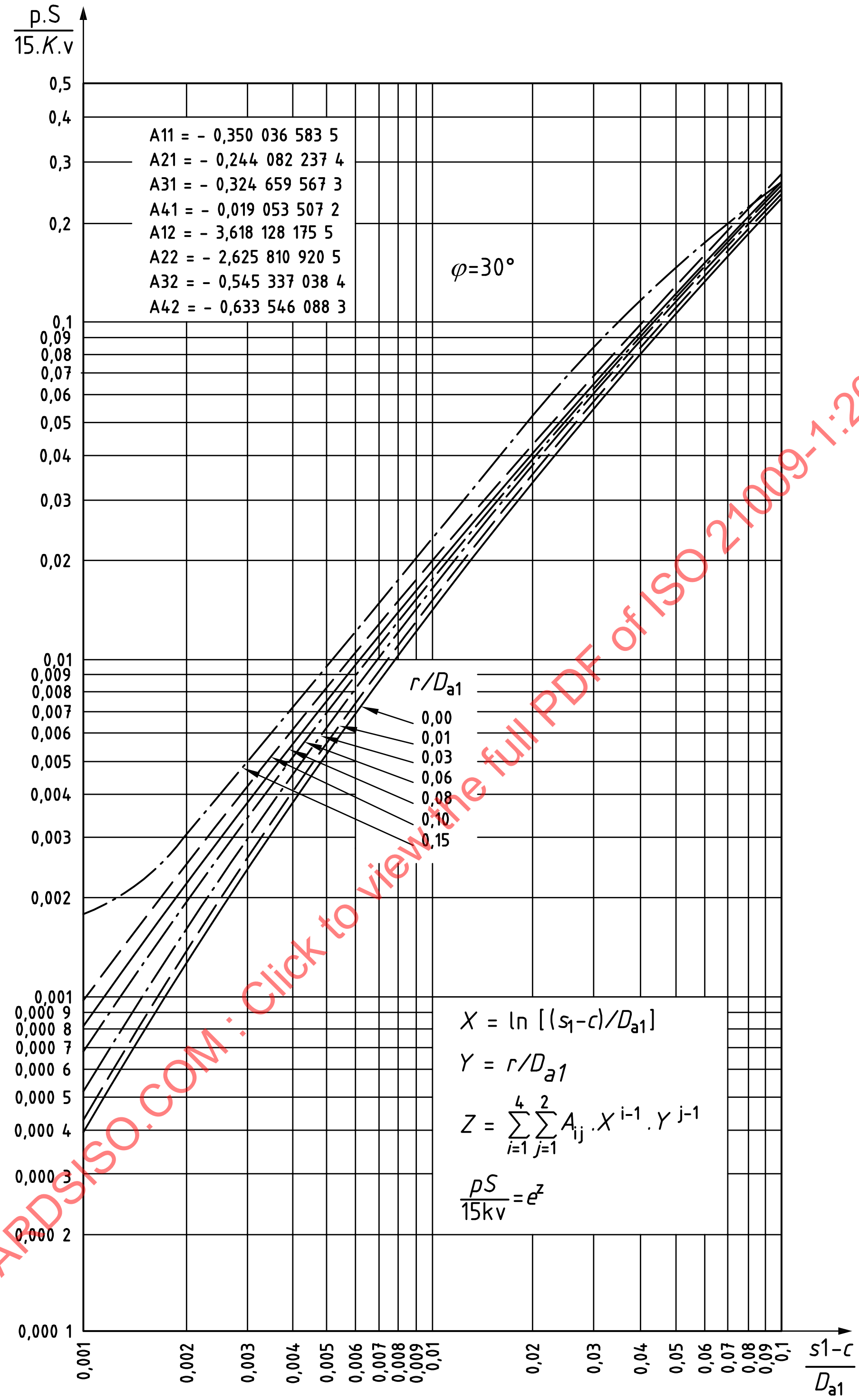
Figure 9 — Geometrical quantities in the case of loading by external pressure



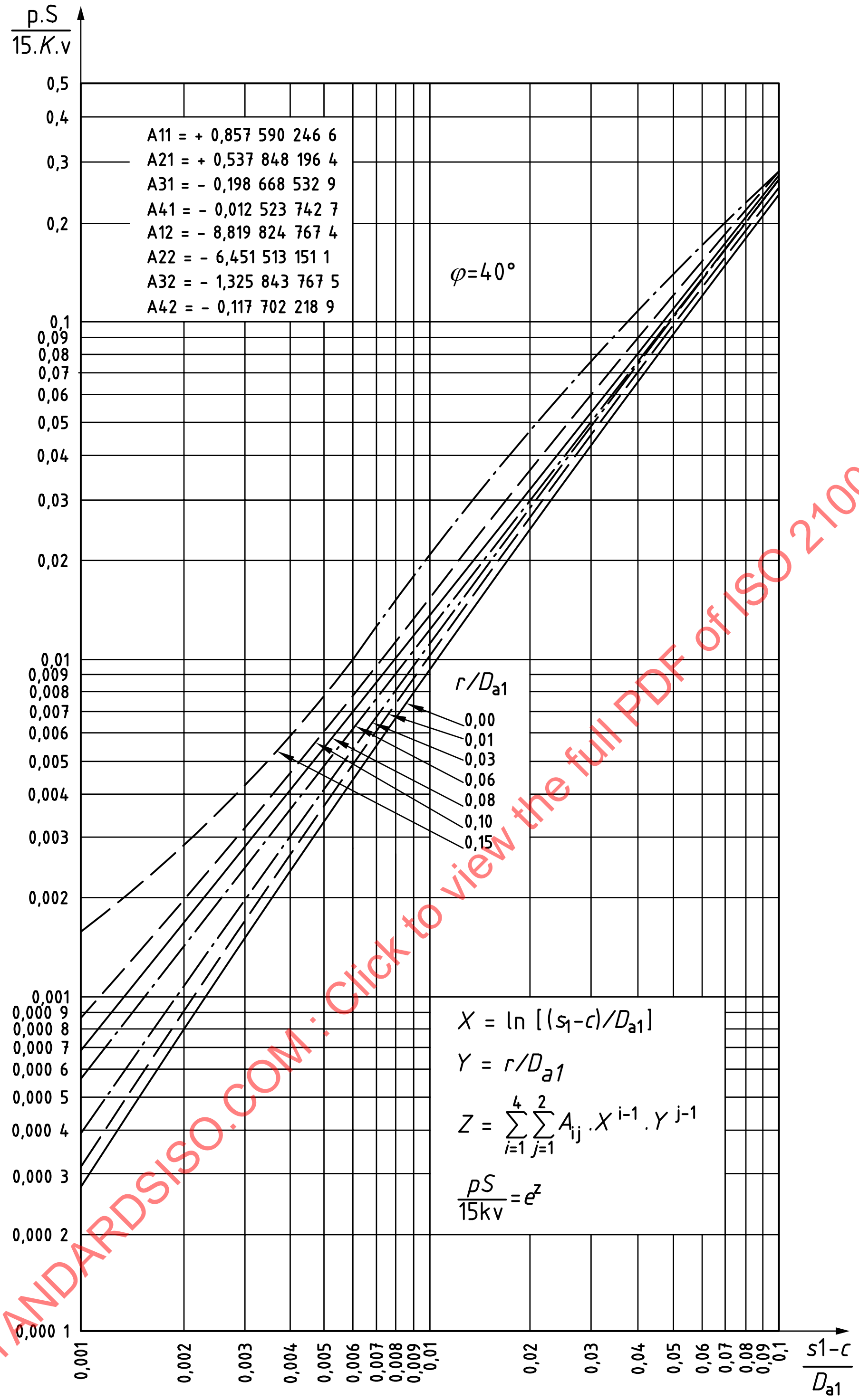
a) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 10°



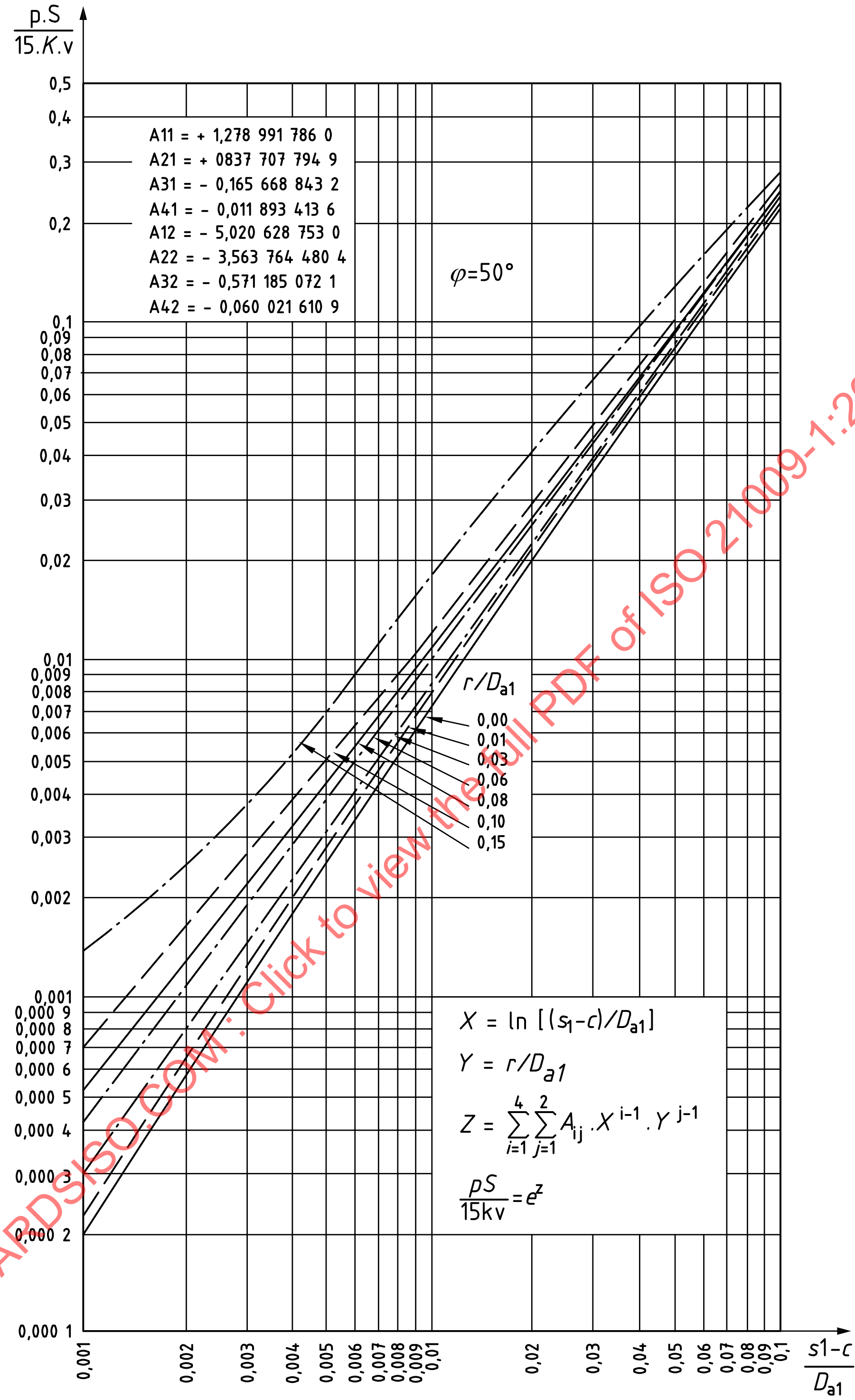
b) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 20°



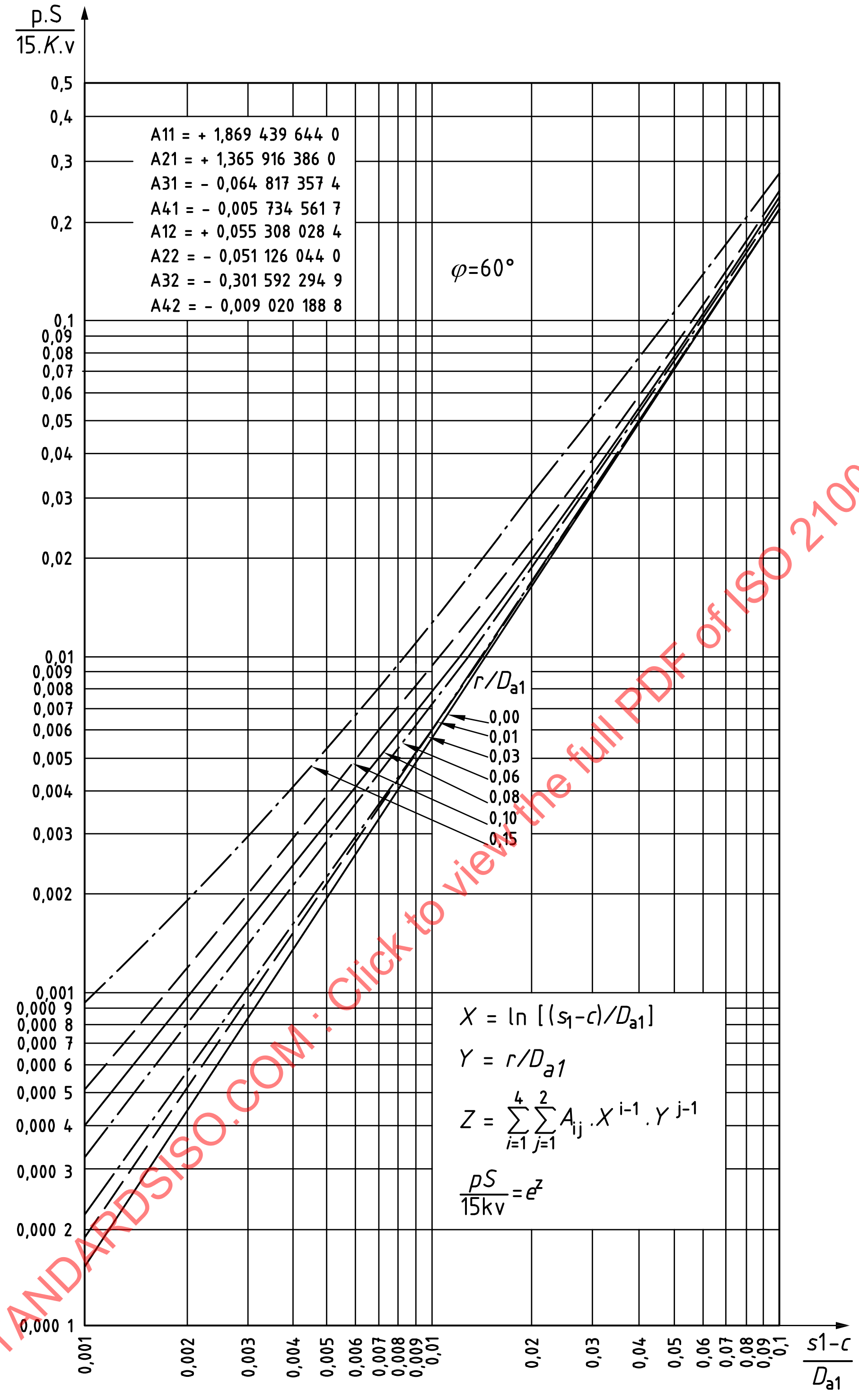
c) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 30°



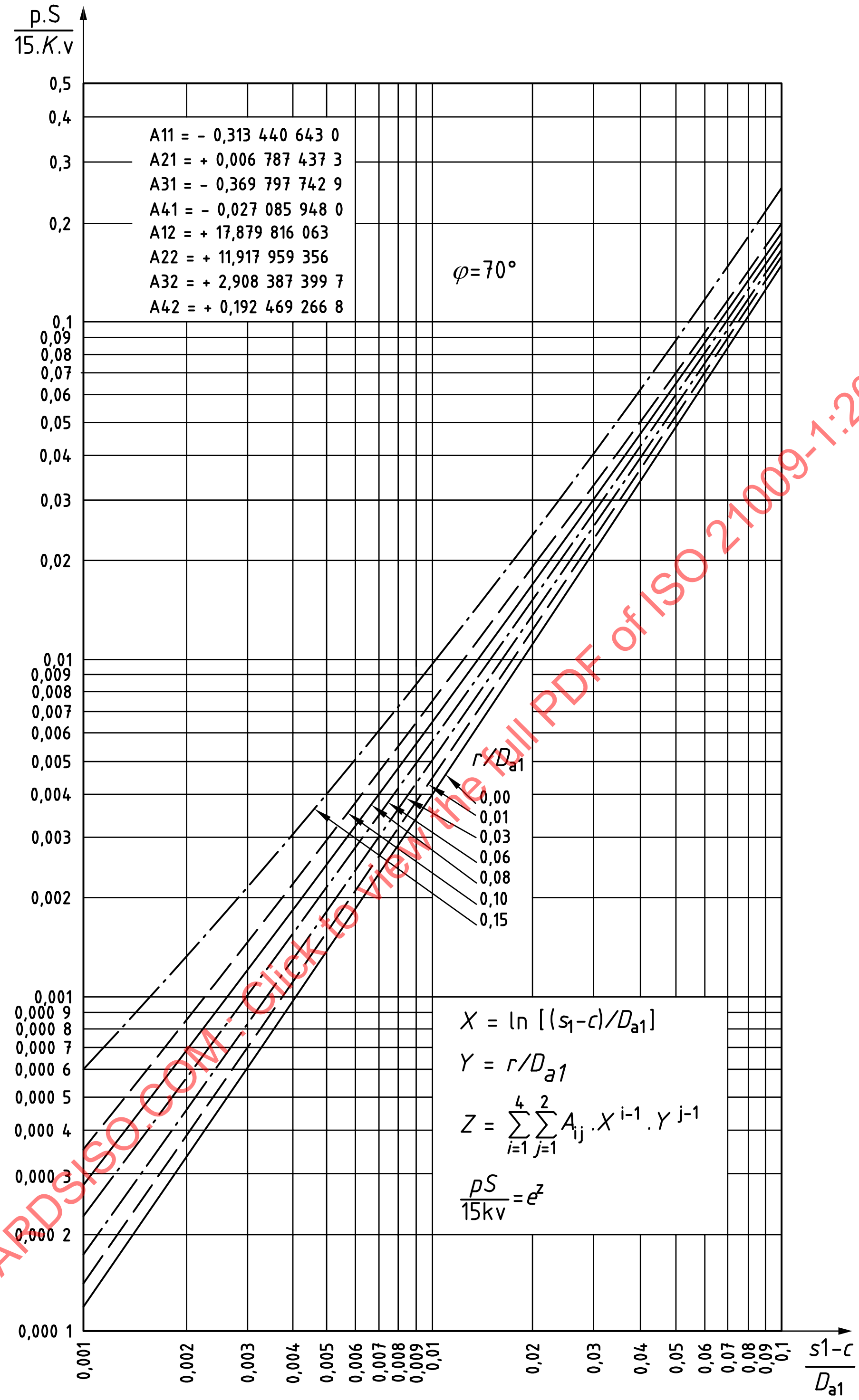
d) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 40°



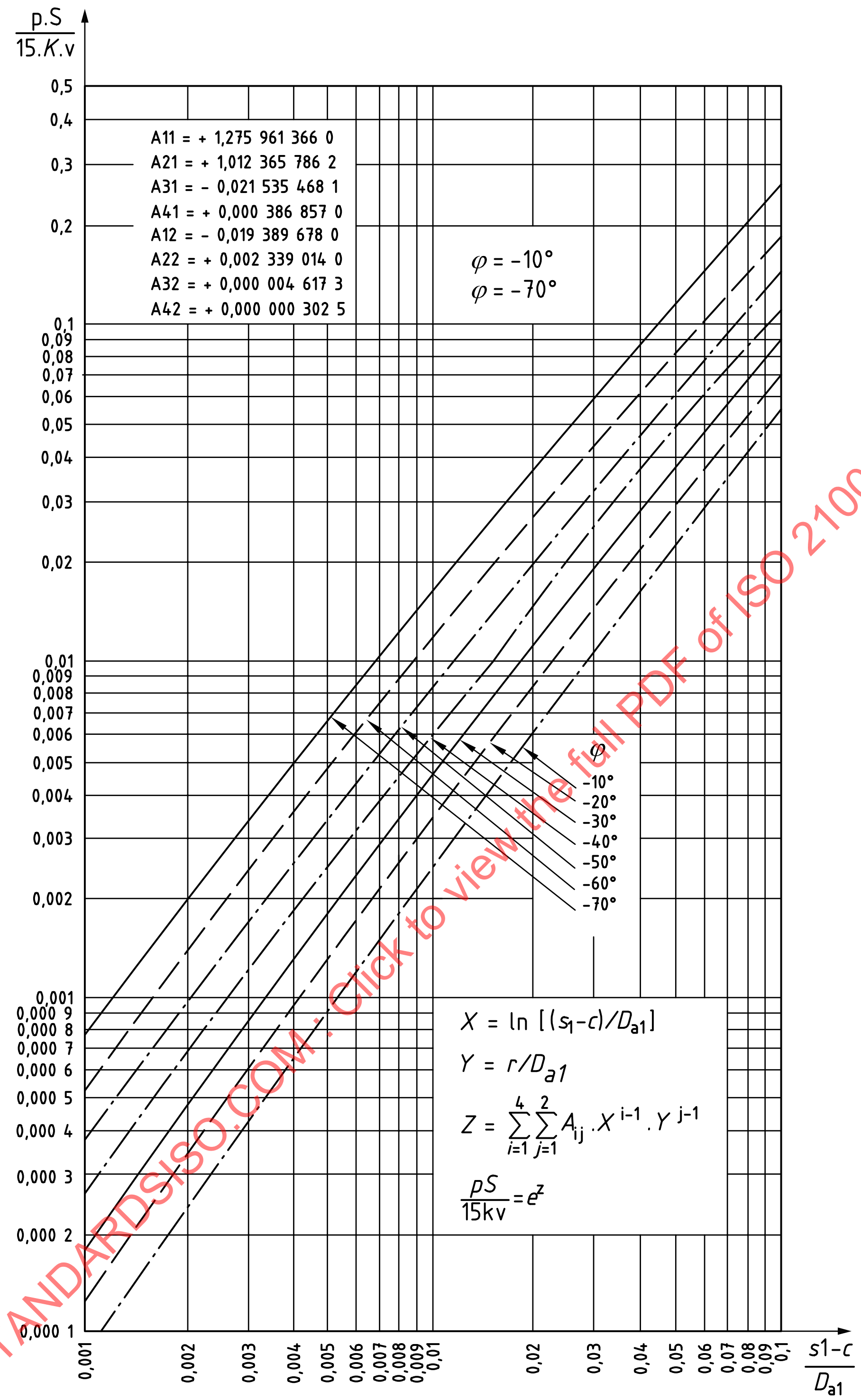
e) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 50°



f) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 60°

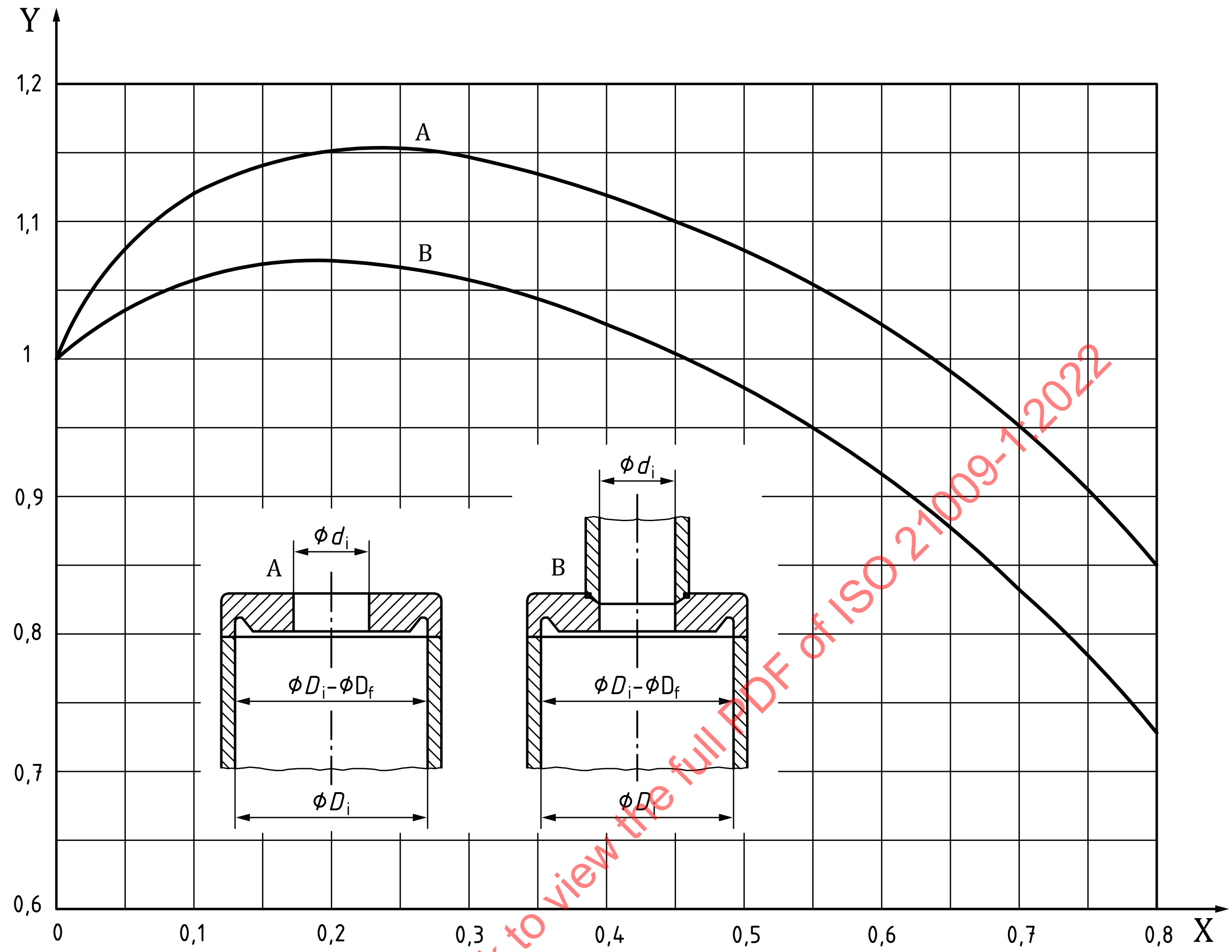


g) Permissible value, $\frac{pS}{15Kv}$, for convergent cone with an opening angle φ = 70°



h) Permissible value, $\frac{pS}{15Kv}$, for convergent cone (corner joint) with an opening angle φ = −10° to −70°

Figure 10 — Permissible values for convergent cone



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 torispherical end

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

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 torispherical end

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

A_1 = 1,001 003 44

A_2 = 0,944 284 68

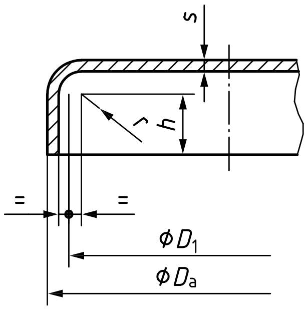
A_3 = 4,312 102 00

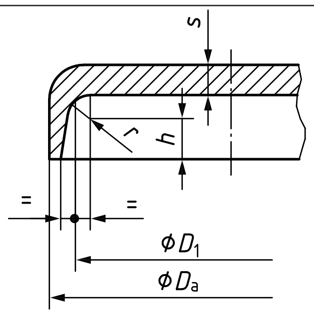
A_4 = 8,389 435 00

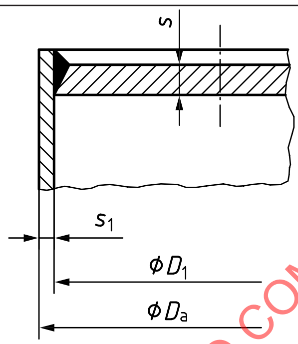
A_5 = 9,206 283 84

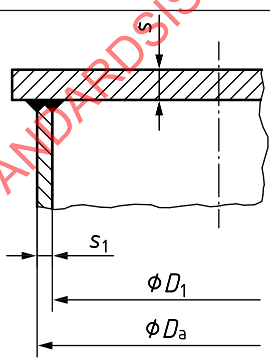
A_6 = 3,694 941 96

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><th>D_a</th><th>r_{min}</th></tr>
<tr><td>up to 500</td><td>30</td></tr>
<tr><td>over 500 up to 1 400</td><td>35</td></tr>
<tr><td>over 1 400 up to 1 600</td><td>40</td></tr>
<tr><td>over 1 600 up to 1 900</td><td>45</td></tr>
<tr><td>over 1 900</td><td>50</td></tr>
</table>
and r W 1,3 s
2. cylindrical part: h W 3,5 × s</td><td>0,30</td></tr>
<tr><td>

b) Forged or pressed flat end</td><td>1. knuckle radius:
r W $\frac{s}{3}$, however at least 8 mm
2. cylindrical part: h W s</td><td>0,35</td></tr>
<tr><td>

c) Flat plate welded into the shell from one side only</td><td>plate thickness:
$s \leq 3\ s_1$
$s > 3\ s_1$</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:
$s \leq 3\ s_1$
$s > 3\ s_1$
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>

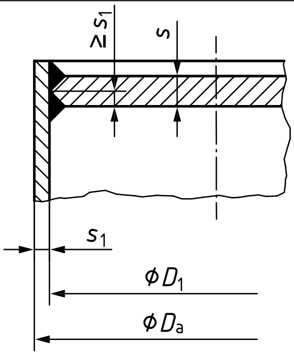
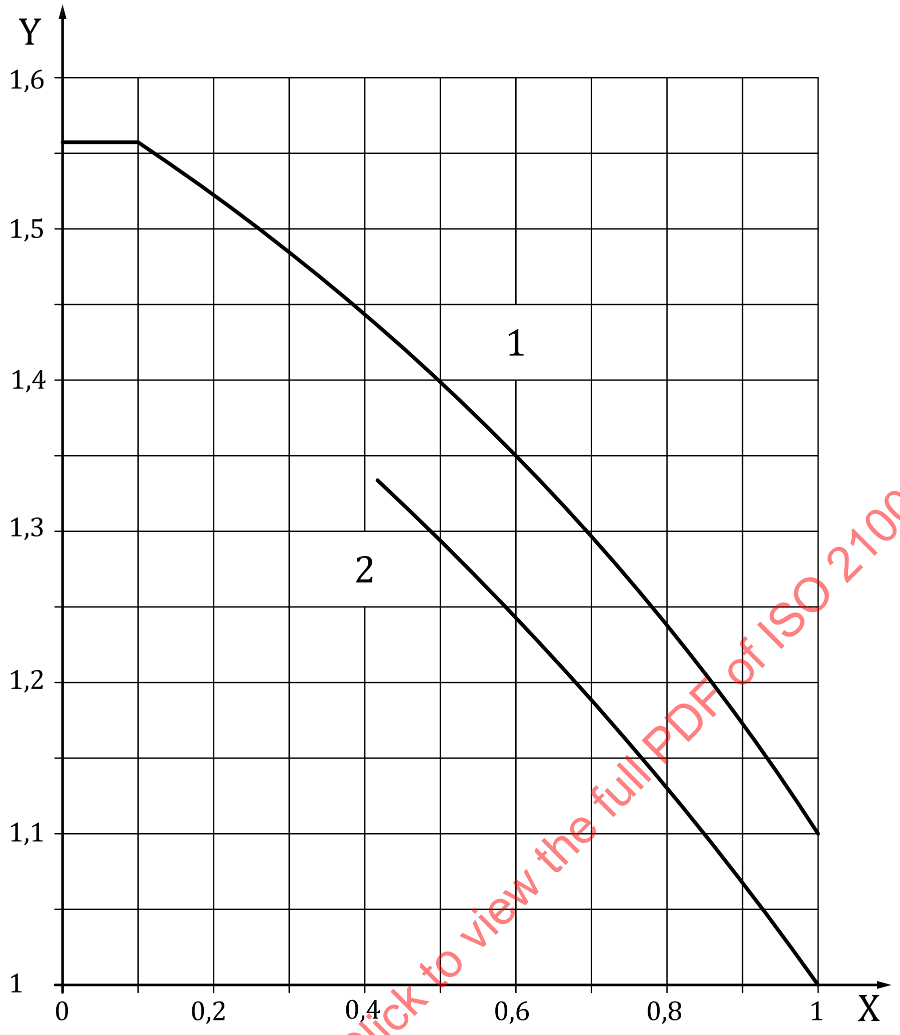
Type of flat end design (principle only)	Conditions	Design factor, C
 **e) Flat plate welded into the shell from both sides**	plate thickness: $s \leq 3\ s_1$ $s > 3\ s_1$	0,35 0,40

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



Key

X ratio factor, *f/e*

Y design factor, C_e

1 rectangular plate

2 torispherical plate

Rectangular plates

f = short side of the rectangular plate

e = long side of the rectangular plate

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

A_1 = 1,589 146 00

A_2 = −0,239 349 90

A_3 = −0,335 179 80

A_4 = 0,085 211 76

Torispherical plates

f = short side of the torispherical plate

e = long side of the torispherical plate

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

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 torispherical flat plates

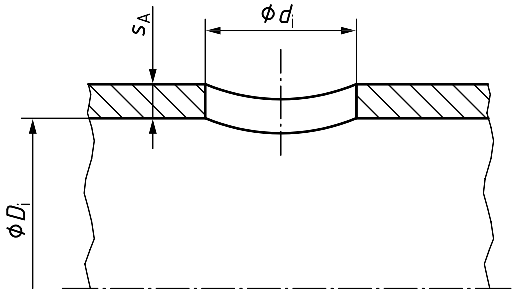


Figure 14 — Increased thickness of a cylindrical shell

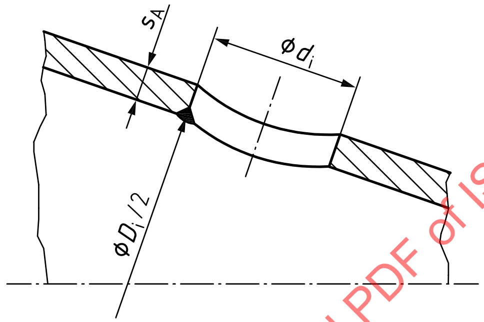


Figure 15 — Increased thickness of a conical shell

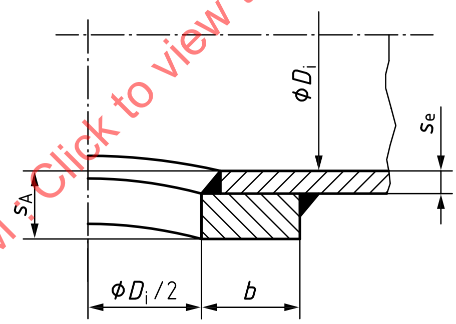


Figure 16 — Set-on reinforcement ring

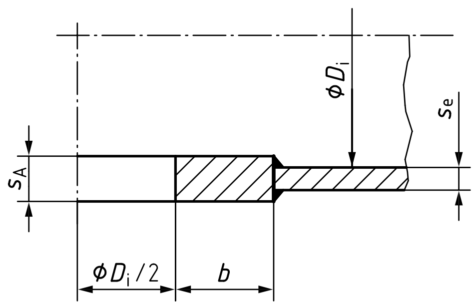


Figure 17 — Set-in reinforcement ring

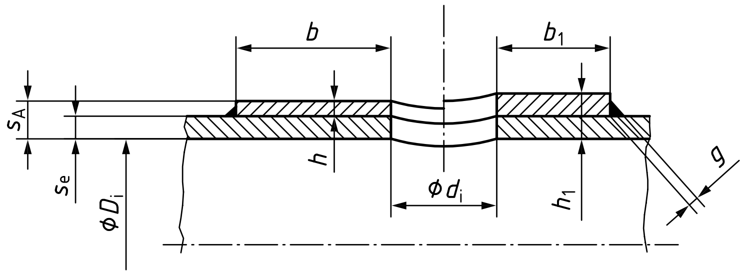


Figure 18 — Pad reinforcement

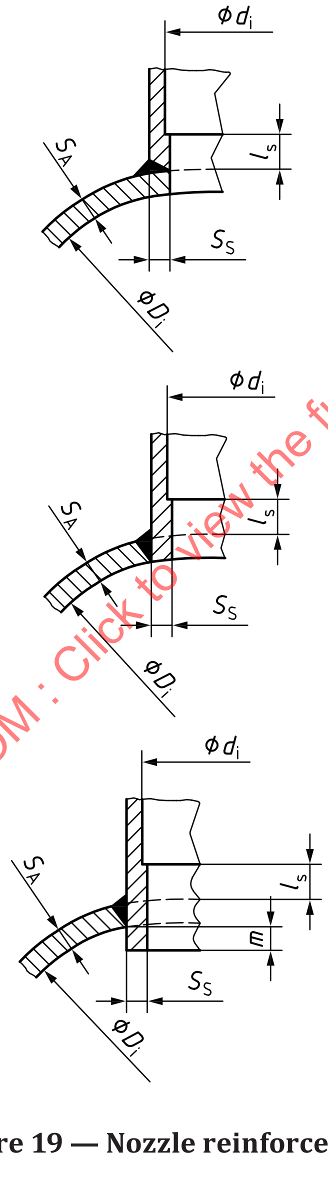


Figure 19 — Nozzle reinforcement

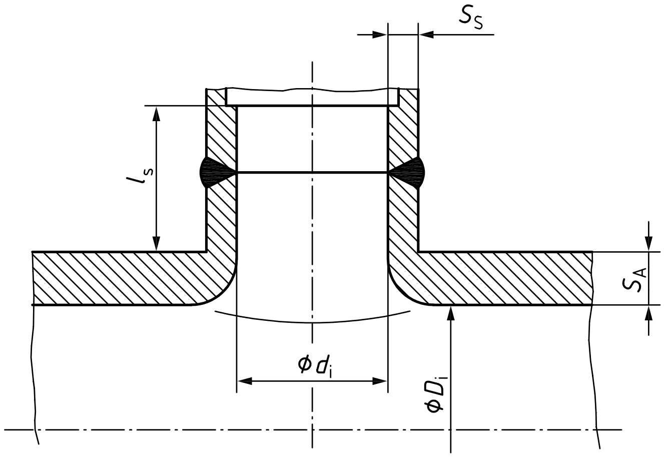
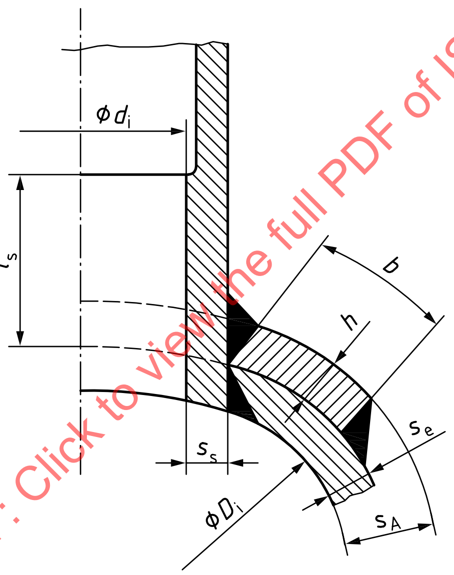
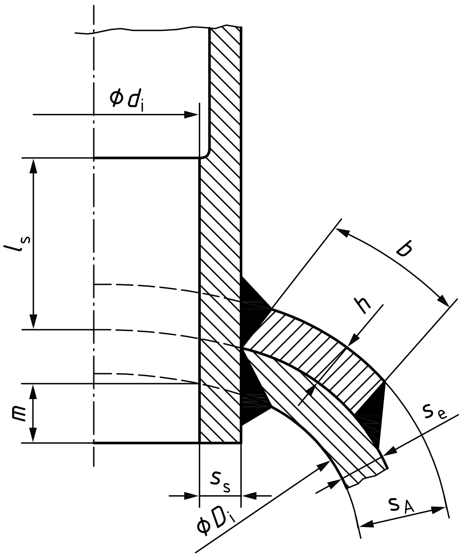


Figure 20 — Necked out opening



a) Type a: Nozzle with non-through tube



b) Type b: Nozzle with through tube

Figure 21 — Pad

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.

For fatigue evaluation see also the relevant limitation of the applied document for fatigue evaluation, e.g. EN 13445, ASME VIII-2.

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 can 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.

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:

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

where

e is the thickness of the initial product;

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

D_e is the external diameter of the final product;

l_n 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.

Heat treatment of cold formed heads should be performed for liquid hydrogen service or for cryogenic gases containing unacceptable levels of H_2S (see ISO 11114-1).

NOTE For the hydrogen vessels to avoid failure by hydrogen embrittlement, stable stainless steel or higher ductility can 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 normalized 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 $R_m \leq 530$ N/mm^2

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

When heat treatment is required, suitable heat treatments after cold forming are normalizing, normalizing (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.

11.4 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 normalized 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 normalized 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.

c) Post weld heat treatment (PWHT) applicable to pressure vessels made of nickel and nickel alloys, the following shall additionally apply:

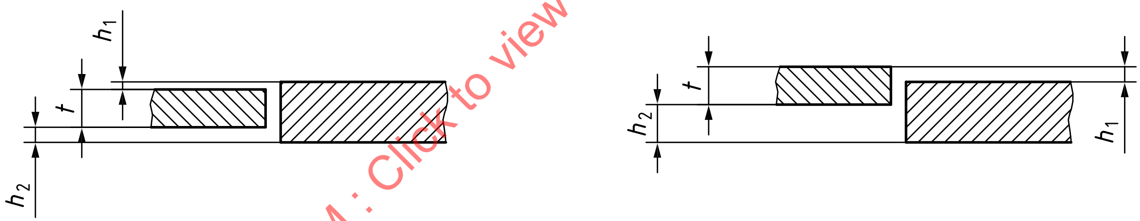
- post weld heat treatment is not normally necessary for welded nickel or nickel alloy pressure vessels. When in service cracking is possible, e.g. vessels in contact with caustic soda, fluorosilicates or some mercury salts, a stress relieving procedure should be considered;
- if post weld heat treatment is required then the annealing heat treatment shall be performed in accordance with a written procedure which describes the parameters required;
- annealing shall be carried out in accordance with the material manufacturer's recommendations;
- precautions shall be taken to avoid contamination and embrittlement (as described in 7.9.2); after annealing the surfaces shall be descaled.
- hot forming of nickel and nickel alloy materials shall be carried out in accordance with the manufacturer's recommendations such that grain boundary liquation and overheating is avoided. The material shall be heated uniformly without flame impingement.

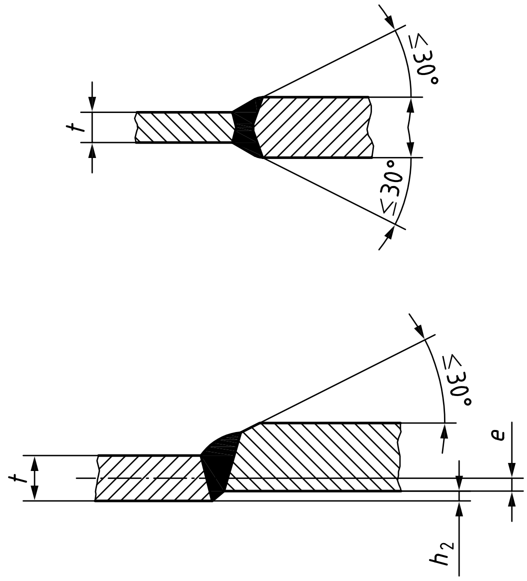
NOTE 1 See also EN 13445-10.

NOTE 2 Most fuels can be used provided that detrimental impurities, e.g. sulphur, are kept at low levels.

Nickel and nickel alloys shall be cleaned before heating.

Embrittlement by low melting point metals such as sulphur, phosphorus, lead, zinc and their alloys can occur from marking materials, die lubricants, pickling liquids, and any waste products encountered during the manufacturing process. Care should be taken to avoid contact with any foreign substances which may be taken into the surface of the material at elevated temperatures.





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:

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

For circumferential seams:

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

a) Seams which do not require a taper

For longitudinal seams:

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

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

For circumferential seams:

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

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

b) Seams which do require a taper

Figure 22 — Plate alignment

11.5 Manufacturing tolerances

11.5.1 General

The recommendations in this subclause are suitable for vessels subjected to predominantly static loads.

For fatigue loads, the manufacturing tolerances shall meet the recommendations of the applied design codes/standards.

To avoid fatigue damage in case of cyclic loading, more severe fabrication, inspection and testing requirements are needed for critical areas of the pressure vessels, see also 12.3.4.2.

For cyclic loaded vessels the absence of surface imperfections and the necessity of smooth transitions are essential. Only smooth transitions are allowed. Similarly, shape imperfections such as peaking are absolutely critical and the maximum permissible peaking of the applied standard/code, or the value permitted in the fatigue analysis, shall not be exceeded.

11.5.2 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 15 % 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 5 mm.

Where a taper is provided between the surfaces, this shall have a slope of not more than 30°. 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 25 % 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 23.

11.5.3 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.

11.5.4 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:

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

11.5.5 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

$$u = \frac{200(D_{max} - D_{min})}{D_{max} + D_{min}}\%$$

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

Table 2 — Permitted out-of-roundness

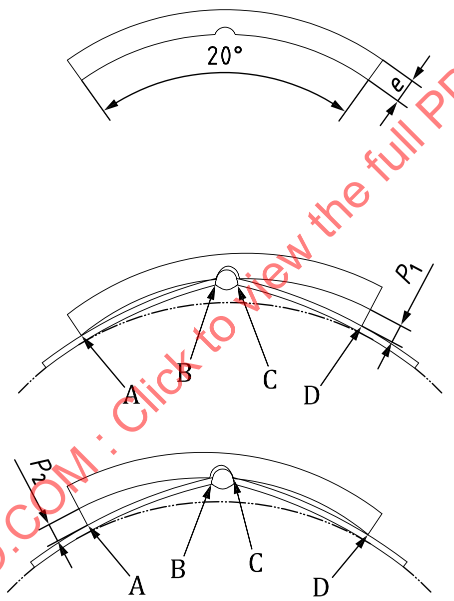
Wall thickness to diameter ratio	Permitted out-of-roundness for	
	Internal pressure	External pressure
$s/D \leq 0,01$	2,0 %	1,5 %
$s/D > 0,01$	1,5 %	1,5 %

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).



Key

A 1st contact point for measuring picking value

B 1st point of measurement

C 2nd point of measurement

D 2nd contact point for measuring picking value

P_1 distance between D and the gauge

P_2 distance between A and the gauge

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
$s/D \leq 0,025$	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:

$$u = \frac{400}{D_a} q$$

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 document requires that the welding method be appropriate and be carried out by qualified welders 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, or ISO 15613 as applicable.

Welders and welding operators shall be qualified in accordance with ISO 9606-1 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 document are given in Annex F or EN 1708-1. These details show sound and currently accepted practice.

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;

— for cold strengthening, see Annex C

— for pressure strengthening of vessels from austenitic stainless steels, see Annex C.

Weld details may be used provided their suitability is proven by procedure approval according to ISO 15614-1 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 (e.g. mechanical, corrosion resistance) 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 or non-metallic materials, or both, 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

12.1.1 General

A quality plan shall include as a minimum, the inspection and testing stages listed in 12.1.2.

12.1.2 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 welder's 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;

— in case of application of Annex C (pressure strengthening of vessels from austenitic stainless steels), inspection scope and requirements shall be adapted as specified in Annex C.

12.1.3 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 12.1.2;

— leak proofness tests ensuring the integrity of vacuum, and leak testing of external piping when it is connected to the inner vessel;

— ensuring integrity of vacuum;

— leak test of external piping;

— checking documentation and installation of pressure relief device(s);

— checking installation of vacuum space relief device;

— checking 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 21028-1 and ISO 21028-2;

— bend test (BF, BR, BS): the testing and the test requirements shall comply with ISO 15614-1:2017, 7.4.2 for steels;

— 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 Table 5 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	$e \leq 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e \leq 35$	3 IW, 3 IH, 1 T, 1 Ma
Ni steels up to 9 % Ni	$e \leq 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 3 IH, 1 T, 1 Ma

Table 5 *(continued)*

Group	*e* in mm	Test specimens
Austenitic stainless steels	$e \leq 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 1 T, 1 Ma

12.3 Non-destructive testing

12.3.1 General

Non-destructive testing personnel shall be qualified for the duties in accordance with ISO 9712.

Non-destructive testing shall be performed according to ISO 17635 and ISO 5817, specifying general rules and standards to be applied to the different types of testing, for either the methodology or the acceptance level for metallic materials.

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

12.3.2 Extent of examination for surface imperfections

All welds shall be visually examined in accordance with ISO 17637 and ISO 5817.

If any doubt arises, this examination shall be supplemented by surface-crack detection, e.g. penetrant testing according to ISO 3452-1 and ISO 23277.

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 carried out by radiographic film examination in accordance with ISO 17636-1 and ISO 10675-1.

Other suitable methods according to ISO 17635 are applicable:

— to use radiographic techniques with digital detectors and processing according ISO 17636-2.

— to use ultrasonic testing in accordance with ISO 17640 and ISO 11666 (and ISO 22825 for austenitic steels), or to use other methods according to ISO 17635.

The extent of radiographic examination of main seams on the inner vessel shall be in accordance with Table 6. See subclause 12.3.4 for acceptance criteria.

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 6 — 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]

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

NOTE 2 For corner joints of cones and areas of high bending stress treat the circumferential seam as a longitudinal seam with joint factor 1.

NOTE 3 Additional testing can 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.

12.3.4 Acceptance levels

12.3.4.1 Acceptance levels for surface imperfections

The results of the weld checks and inspections shall meet quality level C of ISO 5817:2014, Table 1 and the corresponding ISO standards for testing classes and acceptance levels, as defined in Table 7 (from ISO 17635:2016, Annex A).

Table 7 — For predominantly static loaded vessels

NDT method	Testing techniques and levels in accordance with	Acceptance levels in accordance with
Visual testing (VT)	ISO 17637	ISO 5817 Level C
Radiographic testing (RT)	ISO 17636-1 class B[a]	ISO 10675-1 Level 2
Penetrant testing (PT)	ISO 3452-1	ISO 23277 Level 2X

[a] The minimum number of exposures for circumferential weld testing may correspond to the requirements of ISO 17636-1, class A.

Additional requirements for the following imperfections:

— stray arc (601) – removal plus 100 % penetrant testing to ensure no imperfection;

— spatter (602) – weld spatter shall be removed from all pressure parts and load carrying attachment welds.

— torn surface (603), grinding mark (604), chipping mark (605) shall be ground to provide a smooth transition;

— underflushing (606) shall not be permitted. Any local underflushing shall be related to design characteristics (calculated thickness)

12.3.4.2 Acceptance criteria for fatigue loaded vessels

The results of the weld checks and inspections shall meet quality level B of ISO 5817 and the corresponding ISO standards for testing classes and acceptance levels as defined in Table 8 (from ISO 17635:2016, Annex A).

Table 8 — Acceptance levels for fatigue loaded vessels

NDT method	Testing techniques and levels in accordance with	Acceptance levels in accordance with
Visual testing (VT)	ISO 17637	ISO 5817 Level B
Radiographic testing (RT)	ISO 17636-1 class B	ISO 10675-1 Level 1
Penetrant testing (PT)	ISO 3452-1	ISO 23277 Level 2X

Additional requirements for the following imperfections:

— stray arc (601) – removal plus 100 % penetrant testing to ensure no imperfection;

— spatter (602) – weld spatter shall be removed from all pressure parts and load carrying attachment welds.

— torn surface (603), grinding mark (604), chipping mark (605) shall be ground to provide a smooth transition;

— underflushing (606) shall not be permitted. Any local underflushing shall be related to design characteristics.

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:

$H\ (p_s + 1)$ bar [or $H\ (p_s + 0{,}1)$ MPa] hydrostatic or $1{,}25\ (p_s + 1)$ bar [or $1{,}25\ (p_s + 0{,}1)$ MPa] pneumatic

NOTE 1 *H* is equal to 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.

$1{,}25(p_s + p_L + 1)\frac{K_{20}}{K_t}$ bar, or $[1{,}25(p_s + p_L + 0{,}1)\frac{K_{20}}{K_t}$ $MPa,]$ 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 $P_T = H(P_s + 1) - 1$ bar or $[P_T = H(P_s + 0{,}1) - 0{,}1$ MPa].

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) reference to this document, i.e. "ISO 21009-1: 2022" to show that the static cryogenic vessel is in conformity with this document;

   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 or MPa) of the static cryogenic vessel;

   5) test pressure (p_T in bar or MPa) 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 document.

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 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 document 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 formulae:

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

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

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 or a discontinuity effect, or both, 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 $2{,}5\sqrt{Rs}$, 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.

Where Δ is obtained from Figures 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

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

and

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

A.4 Stress categories and stress limits for general application

A.4.1 General

A calculated stress depending upon the type of loading or the distribution of such stress, or both, 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

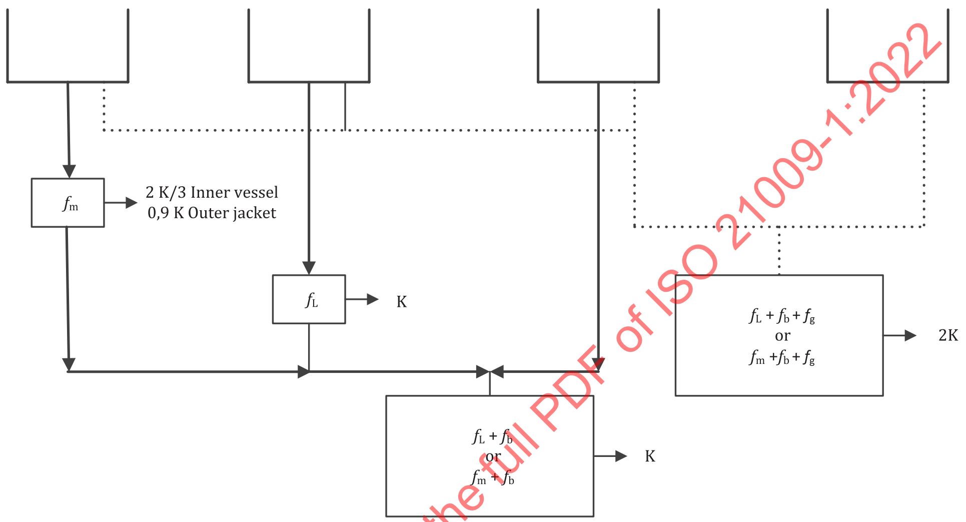
Vessel component	Location	Origin of stress	Type of stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane	f_m
			Gradient through plate thickness	f_g
		Axial thermal gradient	Membrane	f_g
			Bending	f_g
	Junction with head or flange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

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

Table A.1 *(continued)*

Vessel component	Location	Origin of stress	Type of stress	Classification
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			Secondary
	General	Local	Bending	
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, e.g. thermal gradients, structural discontinuities, and do not include primary stresses which can also exist at the same point. It is 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

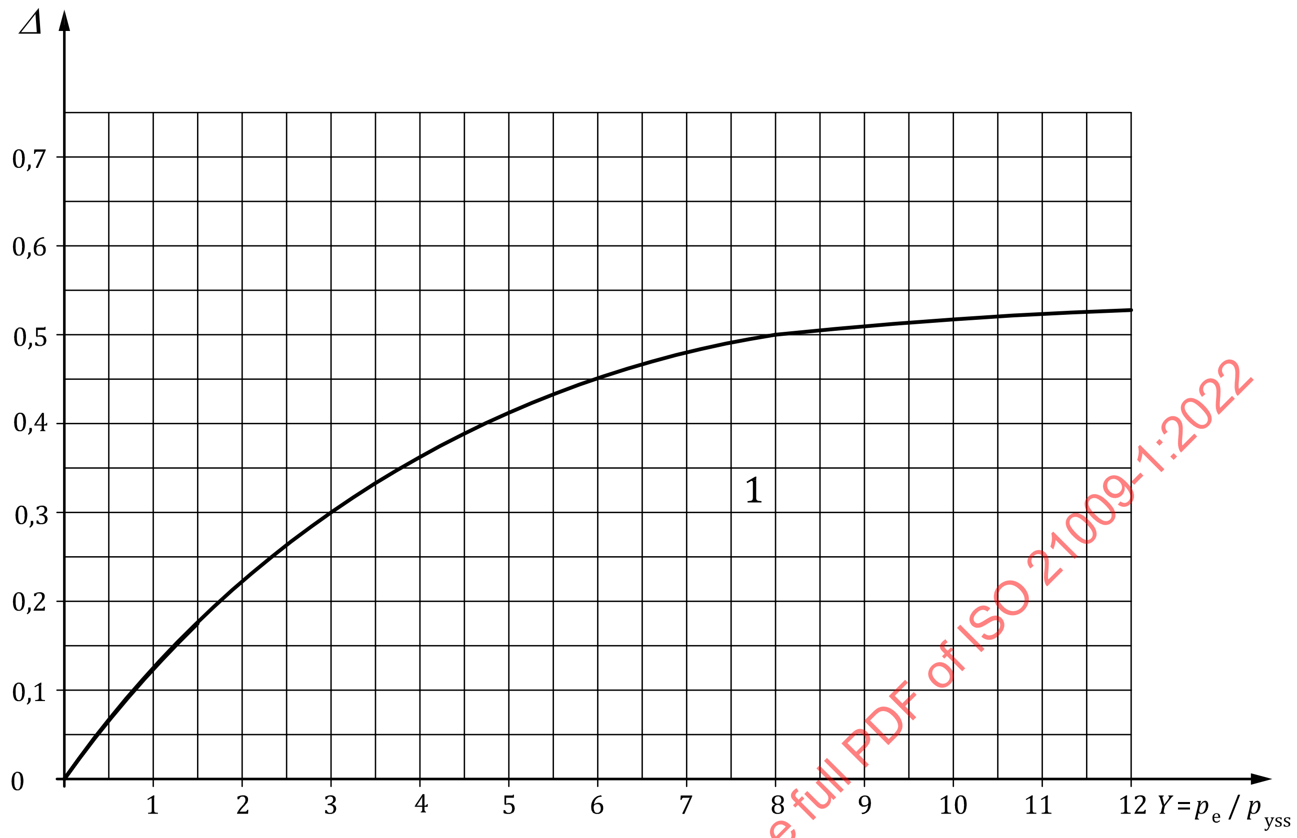


Figure A.2 — For vessels subject to external pressure

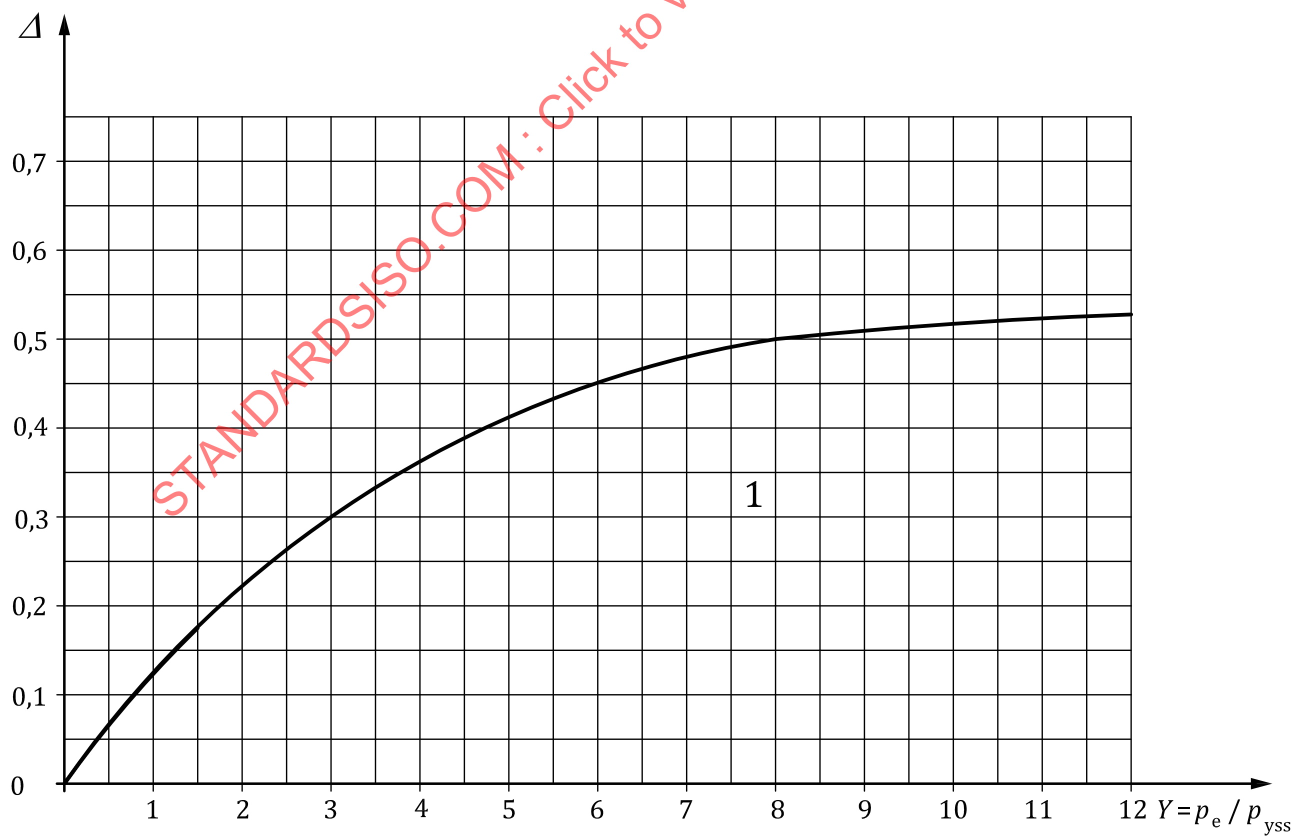


Figure A.3 — For vessels not subject to external pressure

Annex B
(normative)

Additional requirements for 9 % Ni steel

B.1 General

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 document 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 document 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 [1)] 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

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

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 Openings shall not be located with their centre lines closer to principal seams than twice their diameter.

B.2.11 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 ISO 9328-4 is suitable for the construction of cryogenic vessels conforming to this document. Other materials may be suitable provided sufficient test data is made available to demonstrate the suitability of the material.

Annex C
(normative)

Pressure strengthening of vessels from austenitic stainless steels

C.1 General

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 2/3 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 document.

NOTE This method is also known as "cold-stretching". However, using the word "cold" in connection with cryogenic vessels can 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 ≤ 50 °C.

IMPORTANT — The requirements for such cryogenic pressure vessels are given in national legislation. If this method is used, all requirements of this annex shall be applied.

C.3 Materials

C.3.1 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 and ASME Section VIII Div. 1.

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	$Rp0.2$ N/mm² min	$Rp1.0$ N/mm² min	σ_k N/mm² 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	—	—	405
SA/A-240 304L	S 30403	—	—	370
SA/A-240 304N	S 30451	—	—	440
SA/A-240 316	S 31600	—	—	405
SA/A-240 316L	S 31603	—	—	370
SA/A-240 316N	S 31651	—	—	440
SA/A-240 316LN	S 31653	—	—	405

C.3.2 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 maximum anticipated value of σ_k (taking into account bi-dimensional values of 1,15 of σ_k).

Alternatively, a welded test plate may be pre-stretched to an elongation of 10% (see C.6.4.3: 8,7% multiplied with 1,15 for conversion of uniaxial to biaxial stress condition) multiplied with a Safety Factor of minimum 1,2 to simulate cold stretching under consideration of material and thickness tolerances and deviations during fabrication/cold stretching.

From this test plate, specimens shall be tested in accordance with Table C.2.

Table C.2 — Testing of pre-stretched production test plates for steels

Group	Test specimens
Austenitic stainless steels ≤ 12 mm	1 BF, 1 BR, 1 T, 1 Ma
Austenitic stainless steels > 12 mm	1 T, 1 Ma, 3 IW,

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 impact value shall not be less than 0,53 mm lateral expansion.

The base material and the weld shall comply with:

$R_{p0,2} \geq \sigma_k$; $A_5 \geq 25$ %; $a_{kISO\text{-}V} \geq 48$ J/cm^2

The material sheets shall be placed so that the direction of rolling is in the same direction as the vessel's circumference. The difference between the metal sheets used for the vessel shell shall not exceed the following criteria:

$$\left(k_{p0,2(certificate)} \times s_{(measuredvalue)}\right)\max \Big/ \left(k_{p0,2(certificate)} \times s_{(measuredvalue)}\right)\min \leq 1{,}2$$

C.4 Design

C.4.1 General

C.4.1.1 Wall thicknesses calculated according to C.4.2.3 refer to thicknesses before strengthening.

C.4.1.2 Nominal diameters may be used in the design calculations. No allowance is necessary for the possible increase in diameter due to strengthening.

C.4.1.3 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.4.1.4 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.4.1.5 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 document. 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.4.1.6 Fastenings and supports should be preferably attached in non-cold stretched areas otherwise for the simultaneously acting pressure and additional loads the different orientation of material hardening (anisotropic hardening) has been taken into account.

C.4.1.7 Further criteria for Nozzle design:

— For the design of Nozzle connections, the reinforcement effect shall be taken into account;

— $(A_{\text{cut out of shell}}$ x $Re_{\text{ base material}}) / (A_{\text{nozzle}}$x $Re_{\text{ nozzle}}) < 1{,}5$;

— For additional evidence needed if nozzles integrated in the knuckle region of head and for largest allowed opening of unreinforced single holes see C.4.2.3.5.

C.4.2 Design for internal pressure

C.4.2.1 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.4.2.2 Calculation of the strengthening pressure

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

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

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 10.3.2.3.3) is not taken into account in this annex.

C.4.2.3 Calculation of wall thicknesses

C.4.2.3.1 General

The wall thickness of the various parts of the pressure vessel shall be calculated according to applicable subclauses of this document with the modifications shown in Table C.3.

Table C.3 — Modification of formulae for the design of pressure strengthened vessels

Subclause of this document		Modification, see subclause in this annex
10.3.6.1	Cylinders and spheres subject to internal pressure	C.4.2.3.3
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.4.2.3.4
10.3.6.5	Cones subject to internal or external pressure 10.3.6.5.5 Internal pressure calculation (pressure on the concave surface) $|\varphi|$ u 70° 10.3.6.5.6 Internal pressure calculation (pressure on the concave surface) $|\varphi| > 70°$	 C.4.2.3.4 C.4.2.3.2
10.3.6.6	Flat ends	C.4.2.3.2
10.3.6.7	Openings in cylinders, spheres and cones	C.4.2.3.5

C.4.2.3.2 Parts where bending stresses are dominant and large deformations cannot be accepted, i.e. flat cones according to 10.3.6.5.6 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, i.e. 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.4.2.3.3 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.4.2.3.4 Parts according to 10.3.6.4.4 and 10.3.6.5.5 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.5.1).

C.4.2.3.5 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.

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.4.2.3.3.

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.

C.4.2.3.6 Largest allowed opening of unreinforced single holes.

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

$$d_{max} = 0,4\sqrt{D_y \times s} + C$$

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:

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

$$d_{max} \le 0{,}2D_y$$

The wall thickness of an unpierced cylinder is calculated from:

$$s_0 = \frac{pD_y}{20\frac{\sigma_k}{1{,}5} + 2p}$$

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

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

In case of p in MPa, conversion shall be considered.

C.4.3 Design for external pressure

C.4.3.1 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.4.3.2 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.5 Manufacturing and inspection

C.5.1 Strengthening procedure

C.5.1.1 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.5.1.2 to C.5.1.6.

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.

The procedure shall be monitored and verified on a prototype or a demonstration vessel.

C.5.1.2 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.5.1.3 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.5.1.4 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.5.1.5). The strain rate shall be checked by repeated measurements of the circumference according to C.5.1.3. 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.5.1.5 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.5.1.6 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.5.3.4.

C.5.2 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 according to C.5.1.4;

— any significant changes of shape and size relevant to the functioning of the vessel;

— any requirement for renewed strengthening (according to C.5.1.6 and C.6.3.4);

— the direction of rolling of the material sheets (according to C.4)

C.5.3 Welding

C.5.3.1 The strengthening method.

The strengthening method presumes high quality welding. For cold strengthening the following additional requirements shall apply:

— production control test plates shall follow C.3.2;

— Longitudinal welding of cylindrical shell needs to exceed at least 10 %.

C.5.3.2 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.5.3.3 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.5.3.2 shall also if possible be tested at random using a volumetric method (preferably by radiographic examination).

C.5.3.4 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.5.3.2 and C.5.3.3.

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.5.4 Pressure vessel drawing

C.5.4.1 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.5.4.2 Details to be welded in place after the strengthening shall be marked on the drawing.

C.5.5 Inspection and testing

C.5.5.1 General

For testing production control, on a regular basis the welded test plates shall be pre-stretched according to C.3.2, before working out the specimen for destructive testing.

C.5.5.2 Non-destructive testing

C.5.5.2.1 Non-destructive testing shall be carried out before the strengthening to the extent stipulated in 12.3 for the weld joint factor 1,0.

Irregularities shall be inside of the limits accordingly to ISO 5817:2014, Table 1 evaluation group B.

C.5.5.2.2 After the strengthening operation and reducing the pressure to the ambient pressure, welds shall be visually examined externally for their full lengths. 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, T junctions, Attachments and at welded nozzles. Places which have been examined with liquid penetrant shall also if possible be tested at random using a volumetric method (preferably by radiographic examination).

C.5.5.2.3 To ensure proper repairs of pressure strengthened vessels, they should be performed exclusively by the manufacturer. Only repairs to cold stretched components are given below. In the case of repairs, a hydrostatic pressure test shall be carried out at stretching pressure (p_k), and then a radiographic examination and liquid penetration test shall be performed. The repairs and the subsequent test shall be documented.

a) Repairing weld

Defective points in welds metal shall be machined. Welding qualification is required for repairing the weld. The repair weld shall be tested by means of a radiographic examination and liquid penetration test. Strength shall be verified by an extended hydrostatic pressure test with a pressure according to p_k

b) Extension of radiographic examination

When defects except tolerances are revealed during the RX sounding, one or several supplementary films (200 mm mini) will be realized in the continuation of one or several defects, whether it is before the hydraulic test or after the hydraulic test. See 12.4.

c) Installation of none-strain hardening material

In case plates are inserted in the shell of a pressure vessel, as a matter of principle, the same material shall be used. The strength of these plates shall be suitable for their place of installation in the cold stretched vessels.

The welds shall be subjected to a visual examination.

C.5.6 Data plate

The name plate shall, in addition to the information according to Clause 13, bear the text "PRESSURE STRENGTHENED".

C.6 Comments

C.6.1 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 Figures C.1 and C.2.

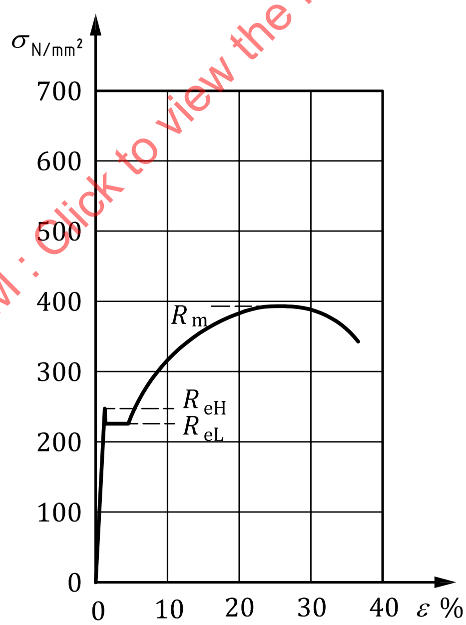


Figure C.1 — Stress/strain curve for carbon steel

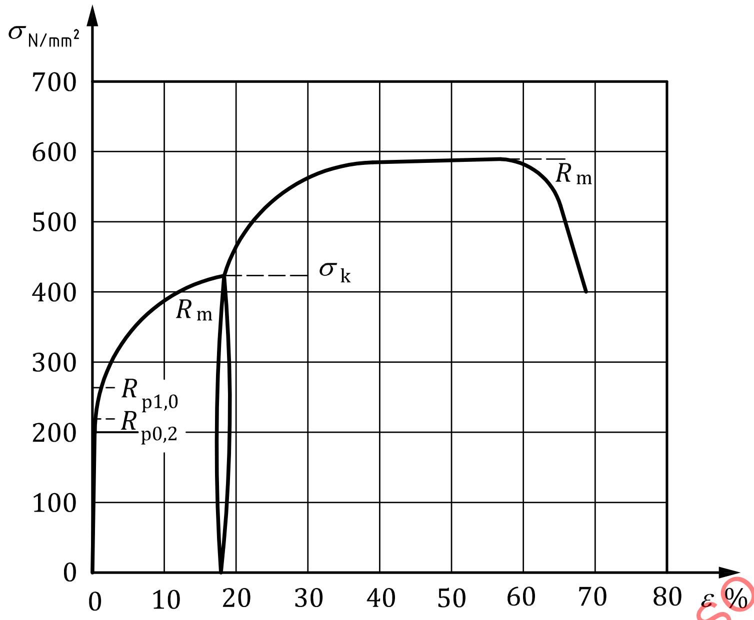


Figure C.2 — Stress/strain curve for austenitic stainless steel

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.6.2 Work-hardened material

C.6.2.1 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.6.2.2 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.6.2.3 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.6.2.4 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.6.3 Derivation of formulae

C.6.3.1 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 be in accordance with the formula for cylinders in 10.3.6.1.3:

$$s = \frac{pDs_F}{20\sigma_k z}$$

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:

$$p_k = \frac{20s\sigma_k}{D}$$

If s according to the formula given above is substituted:

$$p_k = p\frac{s_F}{z}$$

In case of p in MPa, conversion shall be considered.

Since S_F = 1,5 and z = 1,0 this corresponds to the formula given in C.4.2.2. Cylinders can be calculated from the formula in 10.3.6.1.3, if σ_k is inserted as the design stress value and 1 as the weld joint factor.

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 above. To sustain this higher pressure the thickness of all parts of the vessel would then need to be increased.

C.6.3.2 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.6.3.3 The derivation of formulae in C.6.3.1 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.4.2.3.2) 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 shall 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.6.3.4 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.6.3.5 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.6.4 Deformations at strengthening

C.6.4.1 The highest allowable design stress value, $\sigma_{k\,\max}$, 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.6.4.2 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.6.4.3 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.

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

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

If the effective values are set = 1, the principal stresses and elongations obtained for the simplest stress conditions are given in Table C.4:

Table C.4 — 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.4 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.4 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 (ε_1 = 1,0) produces a circumferential strain 8,7 % (ε_1 = 0,87) in a cylinder shell and 5 % (ε_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.6.4.4 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, $R_{p0,2}$, is higher than the specified minimum for the material;

— the plate thickness is greater than nominal;

— there are reinforcing effects of, e.g. ends, nozzles.

C.6.4.5 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 minimize these changes is to weld the nozzles in place after the strengthening, whereupon the vessel could require renewed strengthening (see C.5.3.4). This second strengthening generally leads to much smaller deformations.

C.6.4.6 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.

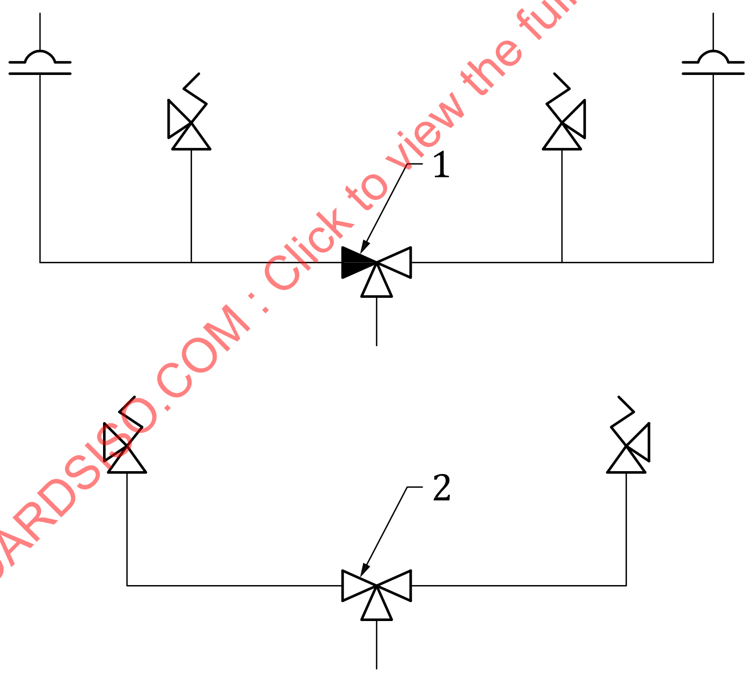
In selecting the most appropriate solutions, the manufacturer shall:

— 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 document, 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 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.

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 can be considered 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 or ($p_T = p_S + 0{,}1$ MPa) can be considered for the “exceptional design condition”.

E.2 Field of application

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

E.3 General requirements

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

E.4 Specific calculation methods

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

Replace 10.2.3.2.1 b) by:

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

$$p_{CG} = p_s + 1 \text{ bar or } [\, p_{CG} = p_s + 0{,}1 \; MPa \,]$$

Replace 10.2.3.2.1 d) 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.

Replace 10.2.3.2.3 by:

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

$$p_T = 1{,}25(p_s + p_L + 1)\frac{K_{20}}{K_{design}} \text{ bar or } \left[p_T = 1{,}25(p_s + p_L + 0{,}1)\frac{K_{20}}{K_{design}} \text{ MPa} \right]$$ (normal design condition, full vessel)

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

$$p_T = \mathrm{H}(p_s + 1)\frac{K_{20}}{K_t} \text{ bar or } \left[p_T = \mathrm{H}(p_s + 0{,}1)\frac{K_{20}}{K_t} \text{ MPa} \right]$$ (normal design condition, nearly empty vessel);

where

p_T is the test pressure

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;

$$p_T = p_s + 1 \text{ bar or } [\, p_T = p_s + 0{,}1 \text{ MPa}]$$ (exceptional design condition);

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

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

Replace 10.2.3.2.2, 2) by:

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

At the end of 10.2.5.2 add:

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

$$p'_s = (p_s + 1)\frac{K_{20}}{K_t} - 1 \text{ bar or } \left[p'_s = (p_s + 0{,}1)\frac{K_{20}}{K_t} - 0{,}1 \text{ MPa} \right]$$

— 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 competent authorities and shall be at least as reliable as that of the pressure limiting system.

Replace 10.3.2.2, b) by:

— in accordance with 10.2.3.2.2 1), 3), 4) and 5);

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

Add a new c) to 10.3.2.2, b):

— in accordance with the modified 10.2.3.2.2 1) 2) (see above);

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

Add a new subclause 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 competent authority. 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 ISO 9328-4 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 11 000/20 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 document (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)	$e_1 = \frac{(p_s + p_L + 1)Di}{20 \frac{K_{-140}}{1,5} v}$	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.3 d)(d neglected in first approximation)	$e_2 = \frac{(p_s + 1)Di}{20 \frac{K_{20}}{1,5} v}$	7,52 mm	8,85 mm	$e_2 = \frac{(p_s + 1)D_i}{20 \frac{K_{-80}}{1,5} v}$	5,55 mm	6,53 mm
3) According to 10.2.3.2.1 and E.3 c)	$p_T = \max \begin{cases} 1,43(p_s + 1) \\ 1,25(p_s + p_L + 1)\frac{K_{20}}{K_{-140}} \end{cases}$	30,03 bar 15,92 bar		$p_T = \max \begin{cases} 1,43(p_s + 1)\frac{K_{20}}{K_{-80}} \\ 1,25(p_s + P_L + 1)\frac{K_{20}}{K_{-140}} \\ p_s + 1 \end{cases}$	22,16 bar 15,92 bar 21 bar	
4) According to 10.2.3.2.2 3)	$e_3 = \frac{p_T Di}{20 \frac{K_{20}}{1,05}}$	7,52 mm		$e_3 = \frac{p_T Di}{20 \frac{K_{20}}{1,05}}$	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.

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

F.2 Specific weld detail

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 junctions of the longitudinal and circumferential seams are examined radiographically and found to be free from significant defects;

g) if Annex C is applied, specific precautions shall be taken regarding the "high notch effect" and the corresponding problem of initiated cracks on the root side, the bending of circumferential welding seam during cold stretching, the volumetric NDT and, if specified, the cycling loading.

Regarding the corresponding problem of initiated cracks on the root side and bending of circumferential welding seam during cold stretching, additionally, a parallel manufactured test specimen should be tested as follows:

— prestretching of 5% of the non-machined specimen;

— Macro-Test with no cracks.

Regarding the volumetric NDT a verification should be done as part of the procedure qualification (see C.5.5.2).

Regarding the cycling loading see h).

h) for fatigue evaluation, see restrictions of the applied standard.

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

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

This strip 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.

This weld 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.

Non-full penetration nozzle welds 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 bear the design nozzle loadings.

For nozzle connections where thermal conditions according to Clause 8 are relevant or Annex C is applied, only full penetrated welding seams are allowed.

F.2.6 Non-continuous fillet-weld on attachments.

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

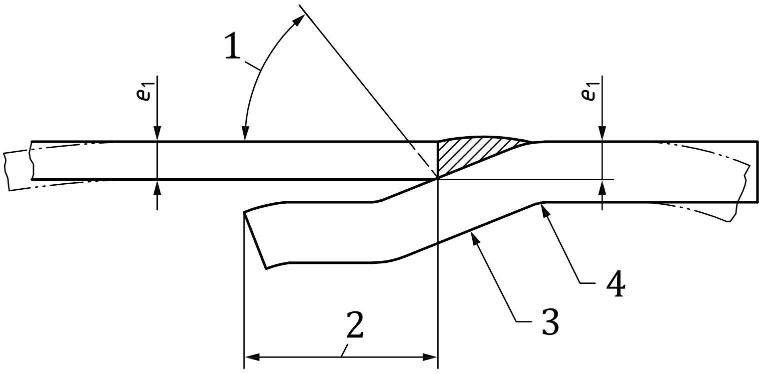
— strength is adequate for design loadings;

— 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 as to cause a fire risk in future operations.



Key

1 bevel optional

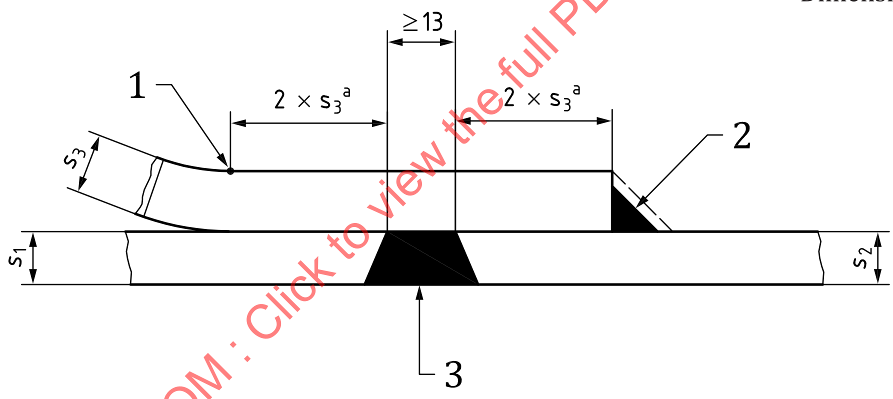
2 as desired

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.

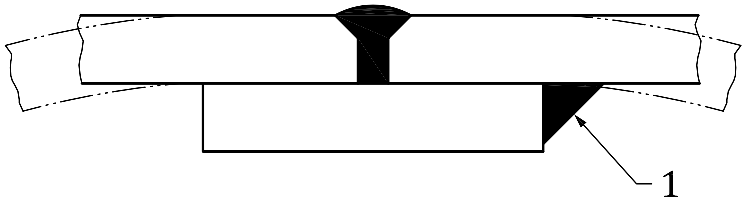
s_2 cylinder thickness.

s_3 end thickness.

[a] Need not exceed 25 mm.

NOTE Cylinder thickness, s_1, and s_2 can vary.

Figure F.2 — Intermediate end



Key

1 intermittent or continuous fillet weld

Figure F.3 — Backing strip

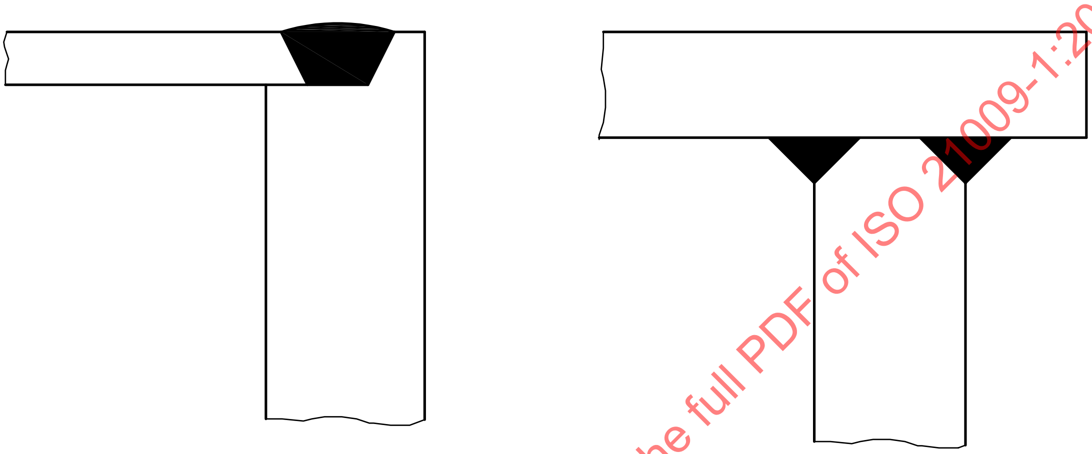


Figure F.4 — End plate closure (examples)

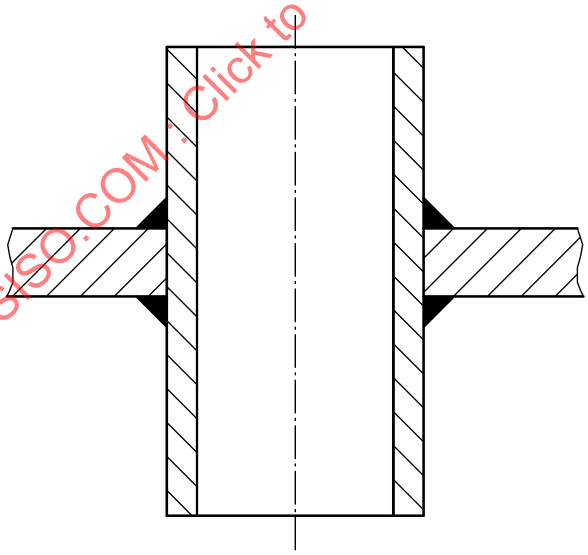


Figure F.5 — Non-full penetration nozzle welds

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 fail-safe mode. The fittings shall 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)

Flammable gas vents and relief systems

H.1 All relief devices, blown down 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 (for more information see IEC 60079-32-2).

H.6 In the particular case of liquid hydrogen, the possibility of air condensing on uninsulated cold parts 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 10.2.5.3. 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 document, i.e. ISO 21009-1: 2022.