INTERNATIONAL STANDARD

ISO 20421-1

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Cryogenic vessels — Large transportable vacuum-insulated vessels —

Part 1: Design, fabrication, inspection and testing

Récipients cryogéniques — Récipients transportables isolés sous vide de grande contenance —

Partie 1: Conception, fabrication, inspection et essais

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Contents

Page

Foreword

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

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

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

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

ISO 20421-1 was prepared by Technical Committee ISO/TC 220, *Cryogenic vessels*.

ISO 20421 consists of the following parts, under the general title *Cryogenic vessels — Large transportable vacuum-insulated vessels*:

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

— *Part 2: Operational requirements*

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Cryogenic vessels — Large transportable vacuum-insulated vessels —

Part 1: Design, fabrication, inspection and testing

1 Scope

This part of ISO 20421 specifies requirements for the design, fabrication, inspection and testing of large transportable vacuum-insulated cryogenic vessels of more than 450 l volume, which are permanently (fixed tanks) or not permanently (demountable tanks and portable tanks) attached to a means of transport, for one or more modes of transport.

This part of ISO 20421 applies to large transportable vacuum-insulated cryogenic vessels for fluids specified in 3.1 and does not apply to vessels designed for toxic fluids.

This part of ISO 20421 does not include the general vehicle requirements, e.g. running gear, brakes, lighting, etc., which are in accordance with the relevant standards/regulations.

This International Standard does not cover specific requirements for refillable liquid-hydrogen tanks that are primarily dedicated as fuel tanks in vehicles. For fuel tanks used in land vehicles, see ISO 13985.

2 Normative references

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

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

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

ISO 5173, *Destructive tests on welds in metallic materials — Bend tests*

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

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

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

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

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

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

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

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

ISO 15614-1, *Specification and qualification of welding procedures for metallic materials — Part 1: Welding procedure tests for the arc welding of steels*

ISO 15614-2, *Specification and approval of welding procedures for metallic materials — Part 2: Welding procedure tests for the arc welding of aluminium and its alloys*

ISO 15614-3, *Specification and approval of welding procedures for metallic materials — Part 3: Welding procedure tests for the arc welding of aluminium and its alloys*

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

ISO 20421-2, *Cryogenic vessels — Large transportable vacuum-insulated vessels — Part 2: Operational requirements*

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

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

ISO 21013-1, *Cryogenic vessels — Safety devices for protection against excessive pressure — Part 1: Reclosable pressure-relief valves*

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*

ASME VIII-2

EN 1708-1, *Welding — Basic weld joint details in steel — Part 1: Pressurized components*

EN 10028-4, *Flat products made of steels for pressure purposes — Part 4: Nickel alloy steels with specified low temperature properties*

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

EN 12300, *Cryogenic vessels — Cleanliness for cryogenic service*

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

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

EN 13445-4, *Unfired pressure vessels — Part 4: Fabrication*

UN Recommendations on the transport of dangerous goods — Model regulations (12th revised edition)

3 Terms and definitions

For the purposes of this part of ISO 20421, the following terms and definitions apply.

3.1
cryogenic fluid
refrigerated liquefied gas
gas which is partially liquid because of its low temperature

NOTE 1 This includes totally evaporated liquids and supercritical fluids.

NOTE 2 In the context of this part of ISO 20421, the refrigerated but non-toxic gases and gas mixtures given in Table 1 are referred to as cryogenic fluids.

Table 1 — Refrigerated but non-toxic gases

Classification code	Identification number, name and description [a]	
	Asphyxiant gases	
3 °A	1913	Neon, refrigerated liquid
	1951	Argon, refrigerated liquid
	1963	Helium, refrigerated liquid
	1970	Krypton, refrigerated liquid
	1977	Nitrogen, refrigerated liquid
	2187	Carbon dioxide, refrigerated liquid
	2591	Xenon, refrigerated liquid
	3136	Trifluoromethane refrigerated liquid
	3158	Gas, refrigerated liquid, N.O.S. (not otherwise specified)
	Oxidizing gases	
3 °O	1003	Air, refrigerated liquid
	1073	Oxygen, refrigerated liquid
	2201	Nitrous oxide, refrigerated liquid, oxidizing
	3311	Gas, refrigerated liquid, oxidizing, N.O.S.
	Flammable gases	
3 °F	1038	Ethylene, refrigerated liquid
	1961	Ethane, refrigerated liquid
	1966	Hydrogen, refrigerated liquid
	1972	Methane, refrigerated liquid or natural gas, refrigerated liquid, with high methane content
	3138	Ethylene, acetylene and propylene mixture, refrigerated liquid, containing at least 71,5 % ethylene with not more than 22,5 % acetylene and not more than 6 % propylene
	3312	Gas, refrigerated liquid, flammable, N.O.S.

[a] Classification codes, identification number, name and description according to the United Nations.

3.2
large transportable cryogenic vessels
tank
thermally insulated vessel of more than 450 l intended for the transport of one or more cryogenic fluids, consisting of an inner vessel, an outer jacket, all of the valves and service equipment together with the structural parts

NOTE The large transportable cryogenic vessel comprises a complete assembly that is ready for service

3.3
thermal insulation
vacuum interspace between the inner vessel and the outer jacket

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

3.4
inner vessel
pressure vessel intended to contain the cryogenic fluid to be transported

3.5
outer jacket
gas-tight enclosure which contains the inner vessel and enables the vacuum to be established

3.6
normal operation
the intended operation of the vessel at a pressure not greater than the maximum allowable working pressure including the **handling loads**

3.7
handling loads
loads exerted on the transportable cryogenic vessel in all normal conditions of transport including loading, unloading, moving and lifting

3.8
documentation
technical documents delivered by the manufacturer to the owner consisting of:

- all certificates establishing the conformity with this part of ISO 20421 (e.g. material, pressure test, cleanliness, safety devices);
- a short description of the vessel (including characteristic data, etc.);
- a list of fluids and their net mass for which the cryogenic vessel is designed;
- an operating manual (for the user) which consists of:
  - a short description of the vessel (including characteristic data, etc.);
  - a statement that the vessel is in conformity with this part of ISO 20421;
  - the instructions for normal operation.

3.9
piping system
all pipes, tubes and associated components which can come in contact with cryogenic fluids including valves, fittings, pressure-relief devices and their supports

3.10
service equipment
measuring instruments and filling, discharge, venting, safety, heating, cooling and insulating devices

3.11
manufacturer of the large transportable cryogenic vessel
company that carries out the final assembly, including the final acceptance test, of the large transportable cryogenic vessel

3.12
gross volume of the inner vessel
internal volume of the inner vessel, excluding nozzles, pipes, etc., determined at minimum design temperature and atmospheric pressure

3.13
tare mass
mass of the empty large transportable cryogenic vessel

3.14
net volume of the inner vessel
volume of the inner vessel, below the inlet to the relief devices, excluding nozzles, pipes, etc., determined at minimum design temperature and atmospheric pressure

3.15
net mass
maximum allowable mass of the cryogenic fluid which may be filled

NOTE 1 The maximum allowable mass is equal to the mass of the cryogenic liquid occupying 98 % of the net volume of the inner vessel under conditions of incipient opening of the relief device with the vessel in a level attitude and the mass of the gas at the same conditions in the remaining volume of the inner vessel.

NOTE 2 Cryogenic liquid helium can occupy 100 % of the volume of the inner vessel at any pressure.

3.16
gross mass
sum of tare mass plus net mass

3.17
pressure
pressure relative to atmospheric pressure, i.e. gauge pressure

3.18
fixed tank
tank vehicle
large transportable vessel permanently attached to a vehicle or to units of running gear

3.19
demountable tank
large transportable vessel non-permanently attached to a vehicle

NOTE When attached to the carrier vehicle, the demountable tank meets the requirements prescribed for a fixed tank. It is designed to be lifted only when empty.

3.20
portable tank
large transportable vessel designed primarily to be loaded onto a transport vehicle or ship

NOTE It can be lifted full and loaded and discharged without removal of structural element.

3.21
maximum allowable pressure
p_s
maximum pressure permissible at the top of the vessel in its normal operating position

3.22
relief plate/plug
plate or plug retained by atmospheric pressure which allows relief of excess internal pressure, generally from the vacuum jacket

3.23
bursting disc device
a non-reclosing pressure-relief device ruptured by differential pressure

NOTE It is the complete assembly of installed components including where appropriate the bursting disc holder.

3.24
pressure-strengthened vessel
pressure vessel which has been subjected to a calculated and controlled internal pressure (strengthening pressure) after completion, the wall thickness of which 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 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.

4 Symbols

For the purposes of this part of ISO 20421, the following symbols apply (units of measurement are in the column at right):

c	allowance for corrosion	mm
d_i	diameter of opening	mm
d_a	outside diameter of tube or nozzle	mm
f	narrow side of rectangular or elliptical plate	mm
l_b, l'_b	buckling length	mm
n	number of lobes	—
p	design pressure as defined in 10.3.2.2	bar
p	calculation pressure	bar
p_e	allowable external pressure limited by elastic buckling	bar
p_k	strengthening pressure	bar
p_p	allowable external pressure limited by plastic deformation	bar
p_s	maximum allowable pressure	bar
p_T	pressure test (see 6.2)	bar

r	radius, e.g. inside knuckle radius of dished end and cones	mm
s	minimum thickness	mm
s_e	actual wall thickness	mm
v	factor indicative of the utilization of the permissible design stress in joints or factor allowing for weakenings	—
x	(decay-length zone) distance over which governing stress is assumed to act	mm
A	cross-sectional area of reinforcing element	mm^2
C, β	design factors	—
D	shell diameter	mm
D_a	outside diameter, e.g. of a cylindrical shell	mm
D_i	internal diameter, e.g. of a cylindrical shell	mm
E	Young's modulus	N/mm^2
I	moment of inertia of reinforcing element	mm^4
R_e	minimum guaranteed yield stress or 0,2 % proof stress at 20 °C (1 % proof stress for austenitic steel)	N/mm^2
R_m	minimum guaranteed tensile strength at 20 °C	N/mm^2
K	material property used for design (see 10.3.2.3)	N/mm^2
K_T	material property at temperature T in °C (e.g. K_{20} for material property at 20 °C (see 10.3.2.3.2)	N/mm^2
R	radius of curvature, e.g. inside crown radius of dished end	mm
S	safety factor at design pressure, in relation with R_e	—
S_k	safety factor against elastic buckling at design pressure	—
S_p	safety factor against plastic deformation	—
Z	auxiliary value	—
ν	Poisson's ratio	—
u	out of roundness (see 11.5.4.2)	—

5 General requirements

5.1 The large transportable cryogenic vessel shall safely withstand the mechanical and thermal loads and the chemical effects encountered during pressure test and normal operation. These requirements are deemed to be satisfied if Clauses 6 to 12 are fulfilled. The vessel shall be marked in accordance with Clause 13, tested in accordance with Clause 14 and operated in accordance with ISO 20421-2.

5.2 Large transportable 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 large transportable cryogenic vessel shall be clean for the intended service in accordance with ISO 23208.

5.4 The manufacturer shall retain the documents referred to in 3.8, and all supporting documentation (including that from his subcontractors, if any), for a period required by regulation(s) (e.g. product liability). In addition, the manufacturer shall retain all supporting and background documentation (including that from his subcontractors, if any) which establishes that the vessel conforms to this part of ISO 20421.

6 Mechanical loads

6.1 General

The large transportable cryogenic vessel shall resist the mechanical loads mentioned in 10.2.3 without such deformation which could affect safety and which could lead to leakage. This requirement can be validated by:

- the calculation;
- the calculation and pressure-strengthening method, if allowed;
- the calculation and experimental method.

6.2 Load during the pressure test

The load exerted during the pressure test shall be:

$$p_T \geqslant 1{,}3(p_s + 1)$$

where

p_T = test pressure (in bar);

p_s = maximum allowable pressure (in bar);

+ 1 = allowance for external vacuum (in bar).

7 Chemical effects

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

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 interspace 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:

For the inner vessel and its associated equipment the full range of temperature expected.

For the outer jacket and equipment thereof (other than equipment covered in the previous paragraph):

— a minimum working temperature of −20 °C;

— a maximum working temperature of 50 °C.

NOTE This does not apply if the jacket is designed for a lower temperature to be marked on the nameplate.

9 Materials

For the materials used to manufacture the transportable cryogenic vessels, the requirements defined in 9.1 to 9.3 shall be met.

9.1 Selection of materials

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

9.1.2 Materials used at low temperatures shall follow the requirements of the relevant parts of ISO 21028-1 and ISO 21028-2; for non-metallic materials. low-temperature suitability shall be validated by an experimental method, taking into account operating temperatures.

9.1.3 The base materials, listed in Annex G, subject to meeting the extra requirements given in the main body of this part of ISO 20421, are suitable for and may be employed in the manufacture of the cryogenic vessels, in conformance with ISO 20421-1.

9.2 Inspection certificates

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

9.2.2 The material manufactured to a recognized International Standard shall meet the testing requirements of ISO 21028-1 and ISO 21028-2 and shall be declared by an inspection certificate 3.1b in accordance with ISO 10474.

9.2.3 The delivery of material which is not manufactured to a recognized International Standard shall be guaranteed by an inspection certificate 3.1a in accordance with ISO 10474 confirming that the material fulfils 9.1 of this part of ISO 20421-1. The material manufacturer shall follow a recognized International Standard for processing and establishing the guaranteed material properties.

9.2.4 The outer jacket and the equipment not subjected to low 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.

Metallic materials used at cryogenic temperatures shall meet the requirements of the relevant sections of ISO 21028-1 and ISO 21028-2.

In the case of 9 % Ni steel, the additional requirements in Annex C 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 dynamic loads.

10.1.3 Design by calculation and 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 D, if allowed by the applicable authorities where the vessel is to be operated.

10.1.4 Design of components by calculation supplemented with experimental methods

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

10.2 Common design requirements

10.2.1 General

The requirements of 10.2.2 to 10.2.7 are applicable to all vessels irrespective of the design option used.

In the event of an increase in at least one of the following parameters, the initial design process shall be repeated to take account of these modifications:

— maximum allowable pressure;

— specific mass (density) of the densest gas for which the vessel is designed;

— maximum tare weight of the inner vessel;

— nominal length and/or diameter of the inner shell;

or, in the event of any change relative:

— to the type of material or grade (e.g. stainless steel to aluminium);

— to the fundamental shape;

— to the decrease in the minimum mechanical properties of the material being used;

- to 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

To enable the design to be prepared, the following information which defines a vessel type shall be available:

- maximum allowable pressure;
- fluids intended to be contained;
- gross volume of the inner vessel;
- dimensions and allowable weight, taking into account characteristics of the vehicle;
- location of fastening points and loads allowable on these points;
- filling and emptying rate;
- range of ambient temperature, if different from Clause 8;
- transportation mode (see Tables 2 and 3).

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

- definition of which components are designed by calculation, by pressure strengthening, by experiment and by satisfactory in-service experience;
- drawings with dimensions and thicknesses of load-bearing components;
- specification of all load-bearing materials including grade, class, temper, testing, etc., as relevant;
- applicable material test certificates;
- location and details of welds and other joints, welding and other joining procedures, filler, joining materials, etc., as relevant;
- calculations to verify compliance with this part of ISO 20421;
- 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 fastenings.

10.2.3 Design loads

10.2.3.1 General

The large transportable cryogenic vessel shall be able to withstand safely the mechanical and thermal loads encountered during a pressure test and normal operation. The static forces used shall be obtained as required in 10.2.3.1.1 and 10.2.3.1.2.

10.2.3.1.1 The inner vessel, its fastenings and supports shall be designed for the static forces obtained by multiplying the load factors applicable for the transportation modes given in Table 2 with the maximum weight imposed on the inner vessel. The maximum weight imposed on the inner vessel shall include the weights of the inner vessel, its fastenings and supports, maximum permissible content, piping, insulation and any other item supported on the inner vessel. Each load case shall be considered separately, but all forces in a load case shall be considered acting simultaneously. The static forces obtained are equivalent to the dynamic loads experienced during normal operation of the transport vessel. The load factors for assessment of fatigue life are given in Table 3.

10.2.3.1.2 The outer jacket, its fastenings and supports shall be designed for the static forces obtained by multiplying the load factors applicable for the transportation modes given in Table 2 with the maximum weight imposed on the outer jacket. The maximum weight imposed on the outer jacket shall include the weights of the outer jacket, with all its enclosures including inner vessel filled to the maximum permissible capacity and the weights of all items fastened to or supported from/to the outer jacket such as piping, controls, cabinets, etc. Each load case shall be considered separately, but all forces in a load case shall be considered acting simultaneously. The static forces obtained are equivalent to the dynamic loads experienced during normal operation of the transport vessel. The load factors for assessment of fatigue life are given in Table 3.

10.2.3.1.3 Fatigue life calculation shall be conducted according to EN 13445-3, ASME VIII-2 or equivalent standards/codes, and shall be conducted for 10^4 cycles or for the highest number of cycles given on the curve, whichever is lowest. Alternatively, the life of the vessel may be specified and marked on the nameplate.

In fatigue evaluation of any item designed to withstand more than one load case, the maximum loadings in each direction from all applicable load cases shall be considered to act simultaneously in determining the magnitude of alternating stresses. A pressure cycle occurs when the pressure variation is more than 50 % of the maximum allowable pressure p_s. The usage factor shall not exceed 1,00.

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

Table 2 — Design load factors for normal operations in specified transportation modes

Transportation modes	Load case	Load factors				
		Forward	Backward	Up	Down	Lateral
Road and water	1	2,0			1,0	
	2		2,0		1,0	
	3			1,0		
	4				2,0	
	5 [a]				1,0	1,0
	5A [a]				1,0	2,0
Rail with cushioning devices [b]	1	2,0			1,0	
	2		2,0		1,0	
	3			2,0		
	4				2,0	
	5				1,0	2,0
Rail without cushioning devices [b]	1	4,0			1,0	
	2		4,0		1,0	
	3			2,0		
	4				2,0	
	5 [a]				1,0	2,0
	5A [a]				1,0	4,0

NOTE For mixed transportation modes, the higher appropriate design factor applies.

[a] Load case 5A should be considered instead of load case 5 if the direction of the travel is not known.

[b] The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 kilometre per hour impact.

Table 3 — Factors for fatigue analysis in specified transportation modes

Transportation modes	Load case	Load factors					
		Forward	Backward	Up	Down		Lateral
		cyclic	cyclic	cyclic	cyclic	steady	cyclic
Road and water	1	0,7				1,0	
	2		0,7			1,0	
	3			1,0			
	4				1,0	1,0	
	5					1,0	0,7
Rail with cushioning devices [b]	1	2,0				1,0	
	2		2,0			1,0	
	3			1,0			
	4				1,0	1,0	
	5 [a]					1,0	1,0
	5A [a]					1,0	2,0
Rail without cushioning devices [b]	1	4,0				1,0	
	2		4,0			1,0	
	3			1,0			
	4				1,0	1,0	
	5 [a]					1,0	1,0
	5A [a]					1,0	4,0

[a] Load case 5A should be considered instead of load case 5 if the direction of the travel is not known.

[b] The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 kilometre per hour impact.

10.2.3.2 Inner vessel

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

a) calculation pressure, p, where

$$p = p_s + p_L + 1 \text{ bar} ; \quad (1)$$

p_L is the pressure, in bar, exerted by the mass of the liquid contents when the vessel is filled to capacity and subject to each load defined in 10.2.3.1, with either:

1) boiling liquid at atmospheric pressure; or

2) cryogenic fluid at its equilibrium triple point or melting-point temperature at atmospheric pressure;

b) loads imposed on the inner vessel due to the mass of the inner vessel and its contents when subject to each of the loads defined in 10.2.3.1;

c) loads imposed by the piping due to the differential thermal movement of the inner vessel, the piping and the outer jacket, in which the following cases shall be considered:

— cool down (inner vessel warm/piping cold);

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

— transport and storage (inner vessel cold/piping warm);

d) reactions at the support points of the inner vessel during operation when the vessel contains cryogenic liquid product. The reactions shall be determined as described in 10.2.3.1.1;

e) 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 as described in 10.2.3.1.1.

10.2.3.2.2 The design shall be evaluated for the following conditions:

Pressure test: the value used for validation purposes shall be:

$$p_T \geqslant 1{,}3(p_s + 1) \text{ bar} \quad (2)$$

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

p_s is the maximum allowable pressure, in bar.

The 1 bar is added to allow for the external vacuum. The primary membrane stress at test pressure shall not exceed the value prescribed in the relevant regulation but in no case the yield stress of the material.

The minimum test pressure of the inner vessel shall be 3 bar. This requirement does not apply to heating or cooling systems and related service equipments.

10.2.3.3 Outer jacket

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

a) an external pressure of 1 bar;

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

c) load imposed by the inner-vessel and its contents at the support points in the outer jacket when subject to the forces specified in 10.2.3.1.1 and 10.2.3.1.2 and Tables 2 and 3.

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

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) reactions at the outer-jacket fastening points when subject to the forces specified in 10.2.3.1.2 and Tables 2 and 3.

10.2.3.4 Self-supporting vessels

In the case of vehicles in which the inner vessel and possibly the outer jacket constitute stressed self-supporting members of the vehicle, these shall be designed to withstand the stresses thus imposed in addition to stresses from other sources [see 10.2.3.2.1 c) and 10.2.3.3 f)].

10.2.3.5 Inner-vessel supports

The inner-vessel supports shall be designed for the loads specified in 10.2.3.1 and 10.2.3.2 to a maximum allowable stress value equal to $\frac{2}{3}K_{20}$.

10.2.3.6 Surge plates

The inner-vessel shall be divided by surge plates to provide stability and limit dynamic loads to the requirements of 10.2.3, unless it is to be filled equal to or more than 80 % of its capacity or nominally empty. The cross-sectional area of the surge plate shall be at least 70 % of the cross-section of the vessel.

The volume between surge plates shall not exceed $\frac{10\,500}{s_g}$ litres where s_g is the specific gravity of the cryogenic fluid at 1 bar saturation.

Surge plates and their attachments to the shell shall be designed to resist the stresses caused by a pressure evenly distributed across the area of the surge plate. The pressure is calculated by considering the mass of liquid between the plates decelerating at 2 g (10.2.3).

10.2.3.7 Outer-jacket supports

The outer-jacket supports shall be suitable for the load defined in 10.2.3.3.

10.2.3.8 Fastening points

Fastening points shall be suitable for fastening the large transportable cryogenic vessel to the vehicle when filled to capacity and subject to each of the loads defined in 10.2.3.

10.2.3.9 Protection of upper fittings

The fittings and accessories mounted on the upper part of the vessel shall be protected in such a way that damage caused by overturning cannot impair operational integrity. This protection may take the form of cylindrical profile of the vessel, of strengthening rings, protective canopies or transverse or longitudinal members so shaped that effective protection is given (e.g. structures of frame such as in ISO 1496-3).

10.2.3.10 Stability

The overall width of the ground-level bearing surface (distance between the outer points of contact with the ground of the right-hand tyre and the left-hand tyre of the same axle) shall be at least equal to 90 % of the height of the centre of gravity of the fully laden tank vehicle. In an articulated vehicle the mass on the axles of the load-carrying unit of the laden semitrailer shall not exceed 60 % of the nominal total laden mass of the complete articulated vehicle. However, applicable regulations where the vessel is to be operated shall apply.

10.2.3.11 Piping and valves

Piping including valves, fittings and supports shall withstand the following loads. With the exception of a), the loads shall be considered to act in combination where relevant.

a) pneumatic pressure test: not less than the allowable working pressure p_s plus 1 bar for piping inside the vacuum jacket;

b) pressure during operation: not less than the set pressure of the system pressure-relief device;

c) thermal loads defined in 10.2.3.2 d);

d) dynamic loads;

e) set pressure of thermal-relief devices where applicable;

f) loads generated during pressure-relief discharge.

Piping and accessories shall be designed such that their lowest natural frequency is equal to or higher than 30 cycles per second.

This equipment shall be protected or positioned so as to be protected against the risk of being wrenched off or damaged during transport.

In the particular case of liquid with a boiling temperature colder than that of liquid nitrogen, the possibility of air condensing on uninsulated parts shall be considered.

The leakproofness of this equipment shall be ensured in the event of overturning of the vehicle. The gaskets shall be made of a material compatible with the fluid carried, in accordance with ISO 21010.

Each bottom-filling or bottom-discharge opening shall be provided with at least two independent shut-off devices in series, the first being a stop valve situated as close as possible to the outer jacket and provided with protection against mechanical damage at least equal to that afforded by the outer jacket.

For flammable fluids only, in order to prevent leaks of flammable fluids, the first stop valve shall be an instant-closing safety device which closes automatically in the event of an unintended movement of the vehicle or of fire during the filling/emptying operation. It shall also be possible to operate the closing device by remote control. All vent pipes, including pressure-relief devices and purge valves, shall be connected to a vent pipe allowing safe discharge. The control cabinet shall be vented so that flammable gas cannot accumulate therein.

10.2.4 Fatigue

The design shall take into account the effect of cyclic stress on the inner vessel, outer jacket and their attachments during normal conditions of operation, including pressure cycles.

When considering the case of fatigue, the common requirement of designing with loads according to 10.2.3 will be such as to accommodate the effects of fatigue. It may be necessary to pay particular attention to specific details in the supports and piping systems to avoid stress raisers.

10.2.5 Corrosion allowance

Corrosion allowance is not required on surfaces in contact with the operating fluid. Corrosion allowance is not required on other surfaces if they are adequately protected against corrosion.

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.

10.2.6 Inspection openings

Inspection openings are not required in the inner vessel or the outer jacket, provided that the requirements of ISO 20421-2 are followed.

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

10.2.7 Pressure relief

Relief systems shall be designed to meet the requirements given in 10.2.7.1 to 10.2.7.3.

10.2.7.1 Inner vessel

10.2.7.1.1 An inner vessel shall be provided with not less than two independent spring-loaded pressure-relief devices. The pressure-relief devices shall open automatically at a pressure not less than the maximum allowable pressure and be fully open at a pressure equal to 110 % of the maximum allowable pressure, in accordance with this part of ISO 20421. These devices shall, after discharge, close at a pressure not less than 10 % below the maximum allowable pressure, and shall remain closed at all lower pressures. The pressure-relief devices shall be of the type that will resist dynamic forces, including surge.

10.2.7.1.2 In the case of the loss of vacuum, an additional reclosing pressure-relief device, set at no more than 110 % of the maximum allowable pressure, may be used, and the combined capacity of all pressure-relief devices installed shall be sufficient so that the pressure (including accumulation) inside the vessel does not exceed 120 % of the maximum allowable pressure, in accordance with this part of ISO 20421. For non-flammable refrigerated liquefied gases (except oxygen) and hydrogen, this capacity may be achieved by the use of bursting discs in parallel with the required safety relief devices. Bursting discs shall rupture at nominal pressure equal to the test pressure.

10.2.7.1.3 Under the circumstances described in 10.2.7.1.1 and 10.2.7.1.2, together with complete fire engulfment, the combined capacity of all pressure-relief devices shall be sufficient to limit the pressure in the vessel to the test pressure.

10.2.7.1.4 The required capacity of the relief devices shall be calculated in accordance with ISO 21013-1.

The pressure-relief system shall be sized so that the pressure drop during discharge does not cause the valve to reseat instantly.

10.2.7.2 Outer jacket

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

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.

10.2.7.3 Piping

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

10.2.8 Valves

Valves shall conform to ISO 21011.

10.2.9 Insulation

For oxygen or gases having a boiling point below –182 °C at atmospheric pressure, the insulation installed on the tank shall be in accordance with ISO 21010.

10.2.10 Degree of filling

The degree of filling of large transportable vacuum-insulated vessels shall remain below the level at which, if the contents were raised to the temperature at which the vapour pressure equalled to opening pressure of the lowest set pressure relief valve or steam, the volume of the liquid would reach 98 % of the vessel’s net volume. Degree of filling for helium may be 100 % of the net volume. Pre-trip inspection shall ensure that the above limits are not exceeded.

10.2.11 Electrical continuity

All metallic components of large transportable vacuum-insulated vessels intended for the carriage of flammable gases shall be electrically continuous. The electrical resistance, as measured by an ohmmeter, between the inner vessel and/or related metallic components to the vehicle chassis shall not exceed 10 ohms.

10.3 Design by calculation

10.3.1 General

The dimensions of the inner vessel and outer jacket shall not be less than that determined in accordance with this subclause.

10.3.2 Inner vessel

10.3.2.1 General

The minimum thickness of the inner vessel shall be the larger of the thickness value from Table 4 or 10.3.6.

Table 4 — Inner-vessel minimum wall thickness

Inner vessel Diameter, D, in millimetres	**Minimum wall thickness, s_r, in millimetres** for reference material [a]
$D \leqslant 1\,800$	3
$D > 1\,800$	4

[a] Reference material is material having a product $R_m \times A_5$ of approximately 10 000, which yields the cube root of $R_m \times A_5 = 21{,}4$.

For other materials, the required minimum thickness of the metal used is:

$$s = \frac{21.4 \times s_r}{\sqrt[3]{R_m \times A_5}}$$

where

s is the required minimum equivalent thickness of the material used, in millimetres;

R_m is the minimum tensile strength of the metal used, in Newton per square millimetre, at a temperature not lower than the saturation temperature of the fluid at pressure P_s, according to national or international standards;

A_5 is the minimum elongation at fracture (in %) of the metal used at a temperature not lower than the saturation temperature of the fluid at pressure P_s, according to national or international standards.

Alternatively, the values of R_m and A_5 up to 15 % higher than guaranteed minimum values at 20 °C, according to national or international standards, may be used, provided this higher value is certified in the material certification for each cast (heat) of material used.

Values of R_m and A_5 at the same temperature shall be used.

Under these conditions, the reference material equivalent thickness of the inner vessel can be determined as follows:

$$s = s_e$$

$$s_r = s_e \frac{\sqrt[3]{R_m \times A_5}}{21{,}4} \tag{3}$$

where

s_e is the actual wall thickness of the inner vessel.

The R_m and A_5 values at a temperature not lower than the saturation temperature of the fluid at pressure p_s shall be determined from the appropriate material standard or shall be guaranteed by the material manufacturer.

10.3.2.2 Design pressure, p

The internal design pressure shall be p as defined in 10.2.3.2.1 a).

The inner vessel shall be designed for a minimum external pressure equal to the set pressure of the outer-jacket pressure-relief device.

10.3.2.3 Material properties, K

10.3.2.3.1 General

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

— for austenitic stainless steels, R_e = 1 % proof strength;

— for carbon steels, aluminium and aluminium alloys, R_e = yield strength and, if not available, 0,2 % proof strength.

For calculation purposes the material property, K, of the inner vessel shall be limited to 0,57 of R_m, the minimum guaranteed tensile strength.

10.3.2.3.2 K_{20}

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

In the case of austenitic stainless steels, the specified minimum values may be exceeded by up to 15 % for carrying all loads listed in 10.2.3.2, except 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 attested in the inspection certificate and the following conditions are met:

— the material manufacturer shall guarantee compliance with this higher value, in writing, when accepting the order;

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

— the welding procedure shall be suitably qualified.

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

10.3.2.3.3 K_t

The permissible values of R_e and R_m shall be determined for the material at the operating temperature corresponding to a temperature not lower than the saturation temperature of the fluid at pressure p_s. The values of R_e, R_m and E shall be determined from the appropriate material standard (see EN 10028-7 Annex F for austenitic stainless steels) or shall be guaranteed by the material manufacturer.

10.3.2.3.4 Brittleness

The material shall not be subject to brittle fracture at its minimum operating temperature (see ISO 21028-1 and ISO 21028-2).

10.3.2.3.5 Elongation

For steel, the elongation at fracture in % shall be not less than

$$\frac{10\,000}{\text{determined tensile strength in N/mm}^2} \text{ at } 20\ ^\circ\text{C} \tag{4}$$

and in any case it shall be not less than 16 % for fine grained steels and not less than 20 % for other steels. For aluminium and aluminium alloys the elongation at fracture shall not be less than 12 %.

Elongation and determined tensile strengths are the actual values indicated in the material certificates.

10.3.2.4 Safety factors S, S_p and S_k

Safety factors are the ratio of material property K, K_{20}, or K_t over the maximum allowable stress.

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

s = 1,73

This may be reduced to 1,5 for road transportation mode, if allowed by the applicable authorities where the vessel is to be operated.

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

— cylindrical shells: S_p = 1,4

S_k = 2,0

— spherical region: S_p = 2,1

S_k = 2,0 + 0,001 8 R/s_e

— knuckle region: S_p = 1,6

10.3.2.5 Weld joint factor, v

v = 1 for all butt double-welded joints and single-welded butt joints with removable backing strips with complete penetration and full fusion and circumferential seams with permanent backing strip and circumferential joggle joints.

10.3.2.6 Corrosion allowances, c

c = 0

No corrosion allowance is required.

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 formula of 10.3.6.

The reference material equivalent thickness of the outer jacket shall be determined as follows:

$$s_r = s_e \frac{\sqrt[3]{R_m \times A_5}}{21{,}4} \tag{5}$$

where

s_r is the reference material equivalent thickness, in mm;

s_e is the actual wall thickness of the outer jacket, in mm;

R_m is the minimum tensile strength, in N/m^2, at 20 °C;

A_5 is the elongation at fracture, in %, at 20 °C.

The aggregate reference material equivalent thickness of the outer-jacket wall and inner-vessel wall shall be not less than 5 mm if the diameter of the inner vessel is not more than 1 800 mm, and not less than 6 mm if this diameter is more than 1 800 mm.

10.3.3.2 Calculation pressure, p

The internal calculation pressure, p, shall be not less than the set pressure of the outer, jacket pressure, relief device.

The external calculation pressure shall be 1 bar.

10.3.3.3 K_{20}

See 10.3.2.3.2.

10.3.3.4 Safety factors S, S_p and S_k in relation to K, K_{20} or K_t

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

   S = 1,73 may be reduced to 1,5 for road transportation mode, if allowed.

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

   — cylindrical shells S_p = 1,1

     S_k = 2,0

   — spherical region S_p = 1,6

     S_k (see applicable method in Annex H)

   — knuckle region S_k = 1,2

10.3.3.5 Plastic deformation

Resistance to plastic deformation is determined by using 10.3.6.1.3 with the appropriate safety factor, S_p, defined in 10.3.2.4 and 10.3.3.4.

10.3.3.6 Weld joint factor, v

For internal pressure (pressure on the concave surface), v = 0,85.

For external pressure (pressure on the convex surface), v = 1,0.

10.3.3.7 Corrosion allowances, c

For austenitic stainless steel, c = 0.

For aluminium alloys, c = 0.

NOTE *UN Recommendations on the transport of dangerous goods* (6.7.4.2.1) do not allow aluminium alloys for the outer jacket.

For carbon steel, c = 1,0 mm.

NOTE c may be reduced to zero if the external surface is adequately protected against corrosion.

10.3.4 Attachments

For those items attached to the inner vessel, other than the inner-vessel supports (see 10.2.3.4), the allowable stress shall not exceed the 0,75 K_{20} or 0,75 K_t as applicable. Other attachments to the jacket shall be designed for the loads defined in 10.2.3.1 using established design methods and allowable stress not exceeding 0,75 K_{20}. See also Annex B for an acceptable method.

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

10.3.5 Piping and accessories

Piping shall be designed for the loads defined in 10.2.3.1 using established piping design methods and safety factors. However, the overall safety factor used on the material property K shall not be less than the values given in 10.3.2.4.

10.3.6 Calculation formula

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

10.3.6.1.1 Field of application

The field of application for cylindrical shells and spheres is:

$D_a/D_i \leqslant 1,2$.

10.3.6.1.2 Openings

For reinforcement of openings, see 10.3.6.7.

10.3.6.1.3 Calculation

The required minimum wall thickness, s, is:

— for cylindrical shells:

$$s = \frac{D_a p}{20 \frac{K_{20}}{S} v + p} + c \tag{6}$$

— for spherical shells:

$$s = \frac{D_a p}{40 \frac{K_{20}}{S} v + p} + c \tag{7}$$

10.3.6.2 Cylindrical shells subject to external pressure (pressure on the convex surface)

10.3.6.2.1 Field of application

The field of application for cylindrical shells is:

$D_a/D_i \leqslant 1{,}2$.

10.3.6.2.2 Openings

Openings shall be calculated in accordance with 10.3.6.7 using the external pressure as an internal pressure.

10.3.6.2.3 Calculation

Annex H 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)

See Annex H.

10.3.6.4 Dished ends

10.3.6.4.1 Field of application

The following dish ends may be used:

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

b) torispherical ends where $0{,}5D_a \leqslant R \leqslant D_a$ and $0{,}5D_a \geqslant r \geqslant 0{,}06D_a$;

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

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

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

10.3.6.4.2 Straight flange

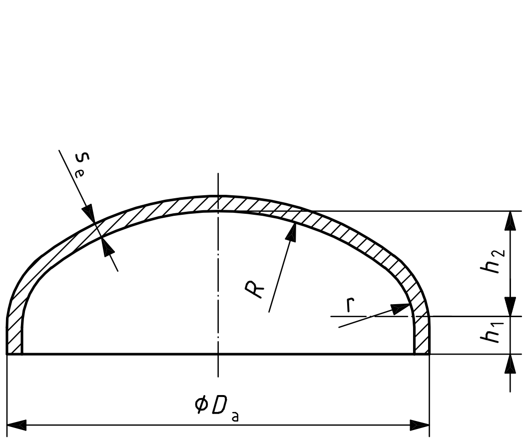
The straight flange length, h_1 [Figure 1a) and b)], shall not be less than:

— for 10 % torispherical ends: 3,0 s;

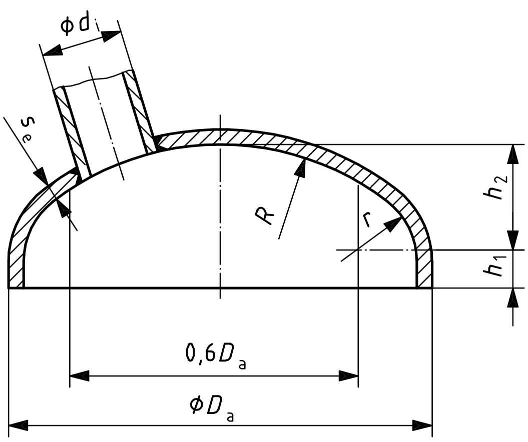
— for 2:1 torispherical ends: 3,0 s;

— for 2:1 elliptical ends: 3,0 s.

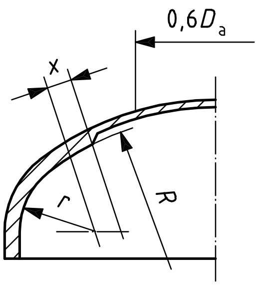
The straight flange may be shorter provided that in the case of inner vessels, the circumferential joint between the dished end and the cylinder is non-destructively tested as required for a weld joint factor of 1,0.



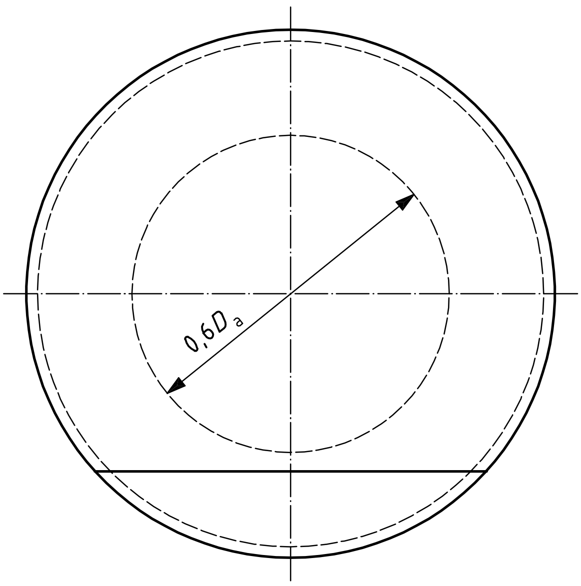
a) Unpierced dished end



b) Dished end with nozzle

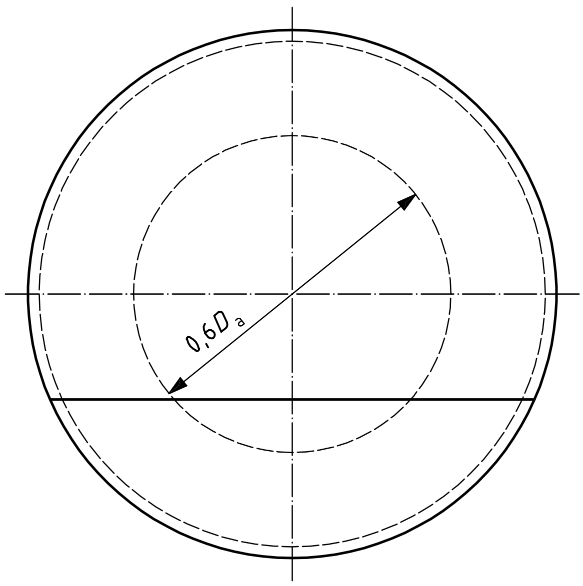


c) End with knuckle and crown of unequal wall thickness



v = 0,85 or 1,0

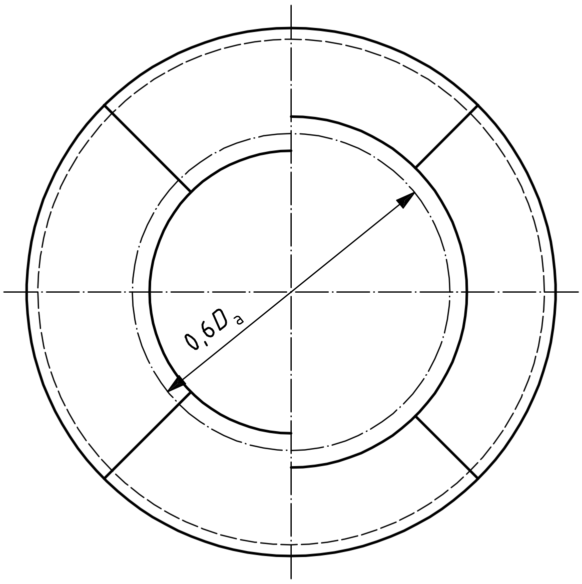
d) Weld outside 0,6 D_a



v = 1,0

e) Weld inside 0,6 D_a

Figure 1 — Examples of dished ends



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

f) End welded together from round plate and segments

Figure 1 (*continued*)

10.3.6.4.3 Internal-pressure calculation (pressure on concave surface)

10.3.6.4.3.1 Crown and hemisphere thickness

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

The opening within the crown area of $0{,}6D_a$ of torispherical ends 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 $0{,}8D_a$.

10.3.6.4.3.2 Torispherical end knuckle thickness and hemispherical end to shell junction thickness

The required thickness of the knuckle region or hemispherical end junction shall be:

$$s = \frac{pRM}{20\dfrac{K_{20}v}{S} - 0{,}2p}$$

where

$$M = 0{,}25\left(3 + \sqrt{\frac{R}{r}}\right)$$

and R = inside crown radius and r = inside knuckle radius.

Dished ends designed for normal operation under internal pressure (pressure on concave side) shall have $R \leqslant D_a$ and $r \geqslant 0{,}06D_a$, but in no case less than 3 s. Dished ends of vacuum jackets are not required to meet the above restrictions on R and r except $r \geqslant 3$ s.

10.3.6.4.3.3 Elliptical ends

The required thickness at the thinnest point after forming of elliptical ends under pressure on the concave side shall be determined by:

$$s = \frac{pD_iB}{20\frac{K_{20}v}{S} - 0{,}2p}$$

where

$$B = \frac{1}{6}\left[2 + \left(\frac{D_i}{2h}\right)^2\right];$$

h is one-half of the length of the minor axis of the ellipsoidal end, or the inside depth of the ellipsoidal end measured from the tangent line (end-bend line), in mm.

10.3.6.4.3.4 If a dished end is welded together from crown and knuckle components, the joint shall be at a sufficient distance x from the knuckle.

x shall be the larger of the following:

— 100 mm;

— $0{,}78\sqrt{R(s)}$ if the crown and knuckle are of different thickness, where s is the thickness of the knuckle component;

— 3,5 s.

v = 1,0 may be used if the scope of testing corresponds to that specified for a design stress level equal to the permissible design stress level or in the case of one-piece ends.

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

10.3.6.5 Cones subject to internal or external pressure

10.3.6.5.1 Symbols and units

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

A	area of reinforcing ring	mm^2
D_{a1}	outside diameter of connected cylinder (see Figure 7)	mm
D_{a2}	outside diameter at effective stiffening (see Figure 9)	mm
D_k	design diameter (see Figure 7)	mm
D_s	shell diameter at nozzle (see Figure 8)	mm
I	moment of inertia about the axis parallel to the shell	mm^4
l	cone length between effective stiffenings (see Figure 9)	mm

s_g	required wall thickness outside corner area	mm
s_l	required wall thickness within corner area	mm
x_i	characteristic lengths (i = 1,2,3) to define corner area [Figures 7 a) and 7 b) and 10.3.6.5.5]	mm
φ	cone angle	°
r	inside radius of knuckle	mm

10.3.6.5.2 Field of application

The field of application for cones is according to Figure 2 where:

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

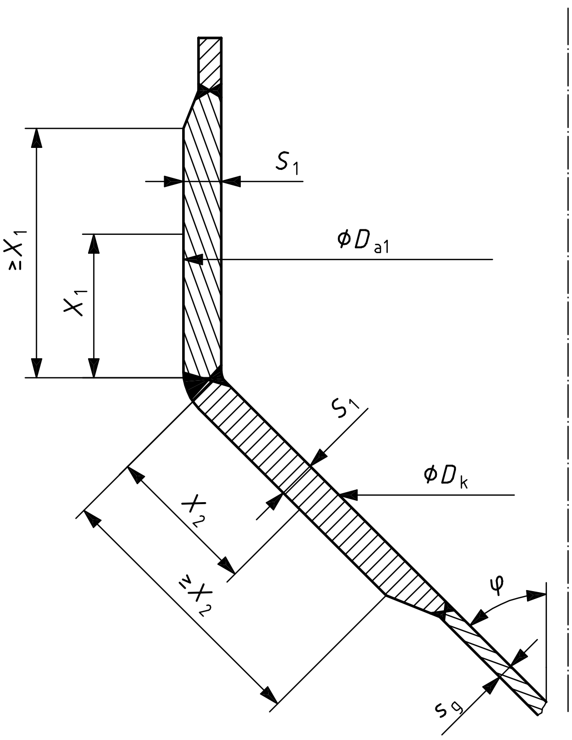
and

$$0{,}001 \leqslant \frac{s_1 - c}{D_{a1}} \leqslant 0{,}1$$

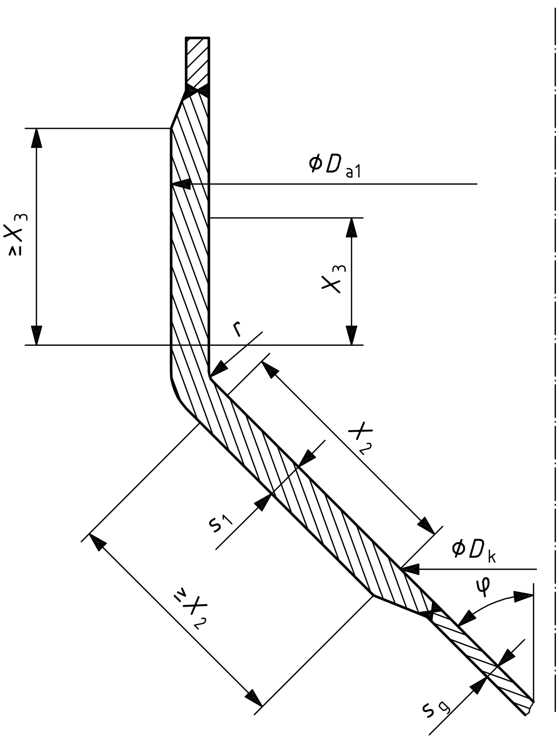
Small ends with a knuckle can be safely assessed and verified as a small end with a corner joint.

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

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



a) **Geometry of convergent conical shells**



b) **Geometry of a divergent conical shell**

Figure 2 — Examples of cones

10.3.6.5.3 Openings

Openings outside of the corner area (Figure 3) shall be designed as follows:

— if $|\varphi| < 70°$ design according to 10.3.6.5.6 using an equivalent cylinder diameter of:

$$D_i = \frac{D_s + d_i\,|\sin\varphi|}{\cos\varphi} \tag{8}$$

— if $|\varphi| \geqslant 70°$ design according to 10.3.6.5.6.

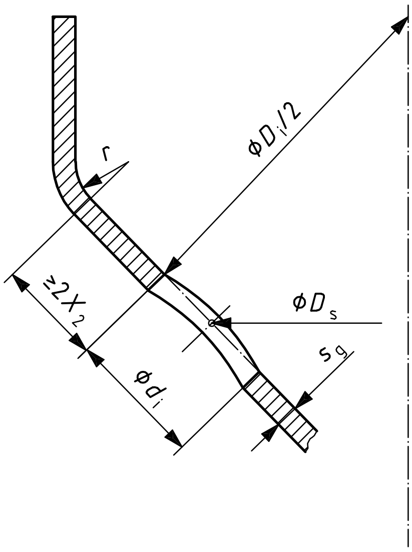


Figure 3 — Geometry of a cone opening

10.3.6.5.4 Non-destructive testing

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

10.3.6.5.5 Corner area

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

The corner area is defined in Figures 2 a) and 2 b) by x_1, x_2, x_3 calculated from the following equations:

$$x_1 = \sqrt{D_{a1}(s_1 - c)} \tag{9}$$

$$x_2 = 0{,}7\sqrt{\frac{D_{a1}(s_1 - c)}{\cos\varphi}} \tag{10}$$

$$x_3 = 0{,}5\,x_1 \tag{11}$$

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

a) Within corner area:

The required wall thickness (s_1) within the corner area is calculated from Figures 4 a) to 4 g) for the large end and Figure 4 h) for the small end of a cone using the following variables:

$$\varphi, \quad \frac{p\,S}{15K\,v} \text{ and } \frac{r}{D_{a_1}}$$

For a corner joint, the curve for $\frac{r}{D_{a1}} = 0$ shall be used.

For intermediate cone angles, linear interpolation shall be used.

b) Outside corner area:

The wall thickness s_1 in the corner area shall not be less than the required thickness s_g outside the corner area as follows:

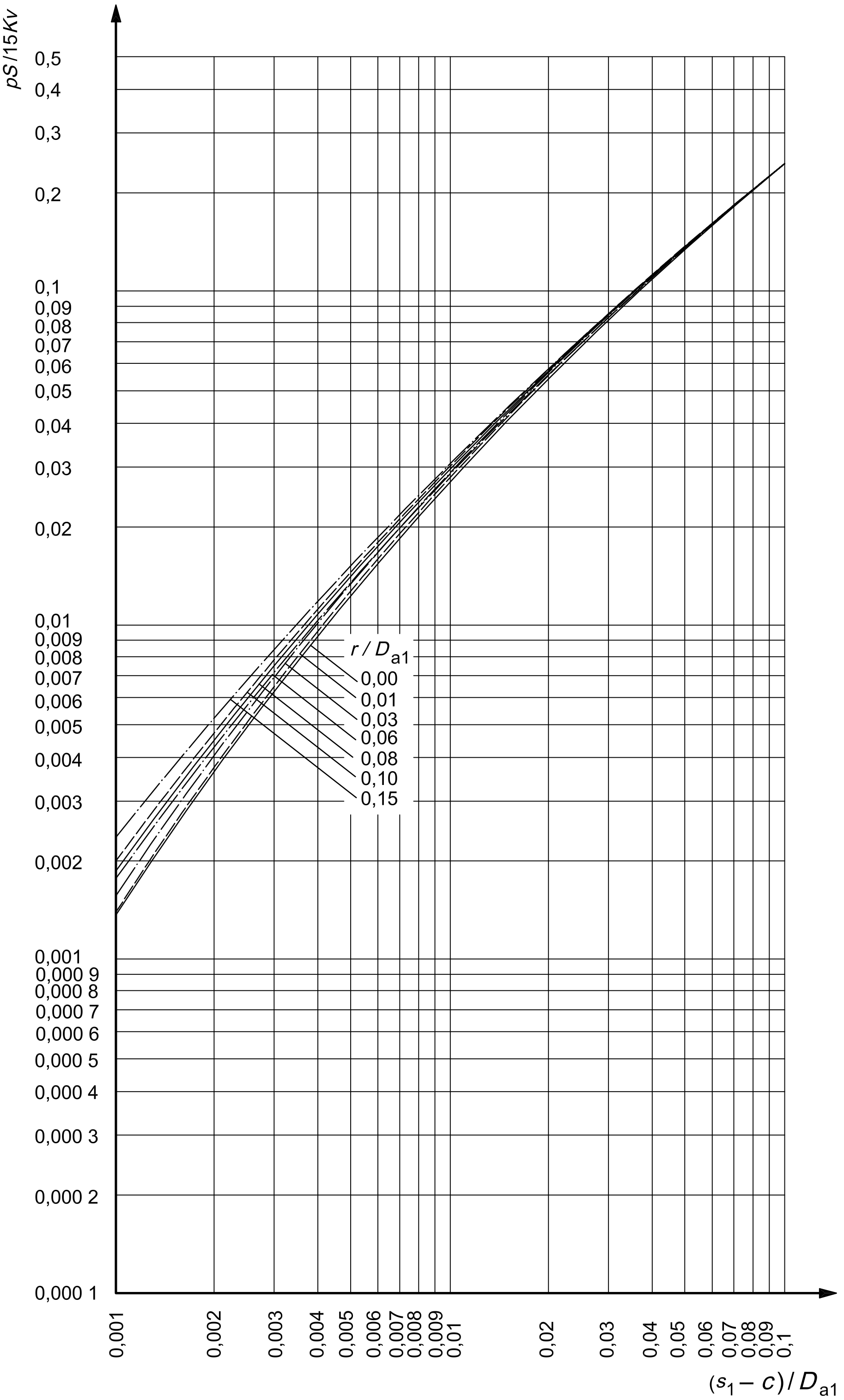
The required wall thickness, s_g, outside the corner area, is calculated from:

$$s_g = \frac{D_k\,p}{20\frac{K}{S}v - p} \times \frac{1}{\cos\varphi} + c \tag{12}$$

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.



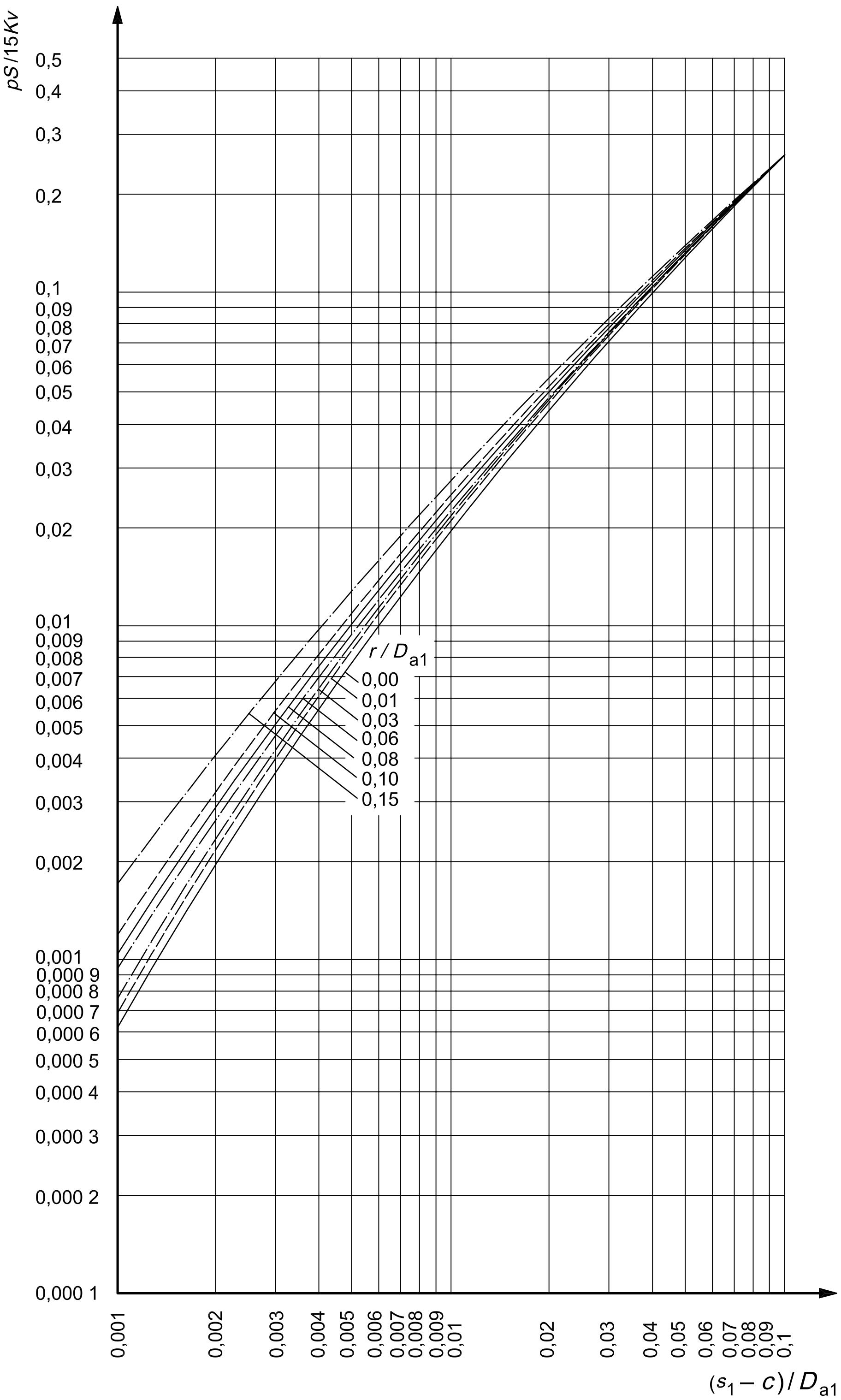
Key

X = ln[$(s_1 - c)/D_{a1}$] $$Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$$

Y = r/D_{a1} $$\frac{pS}{15Kv} = e^Z$$

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

Figure 4



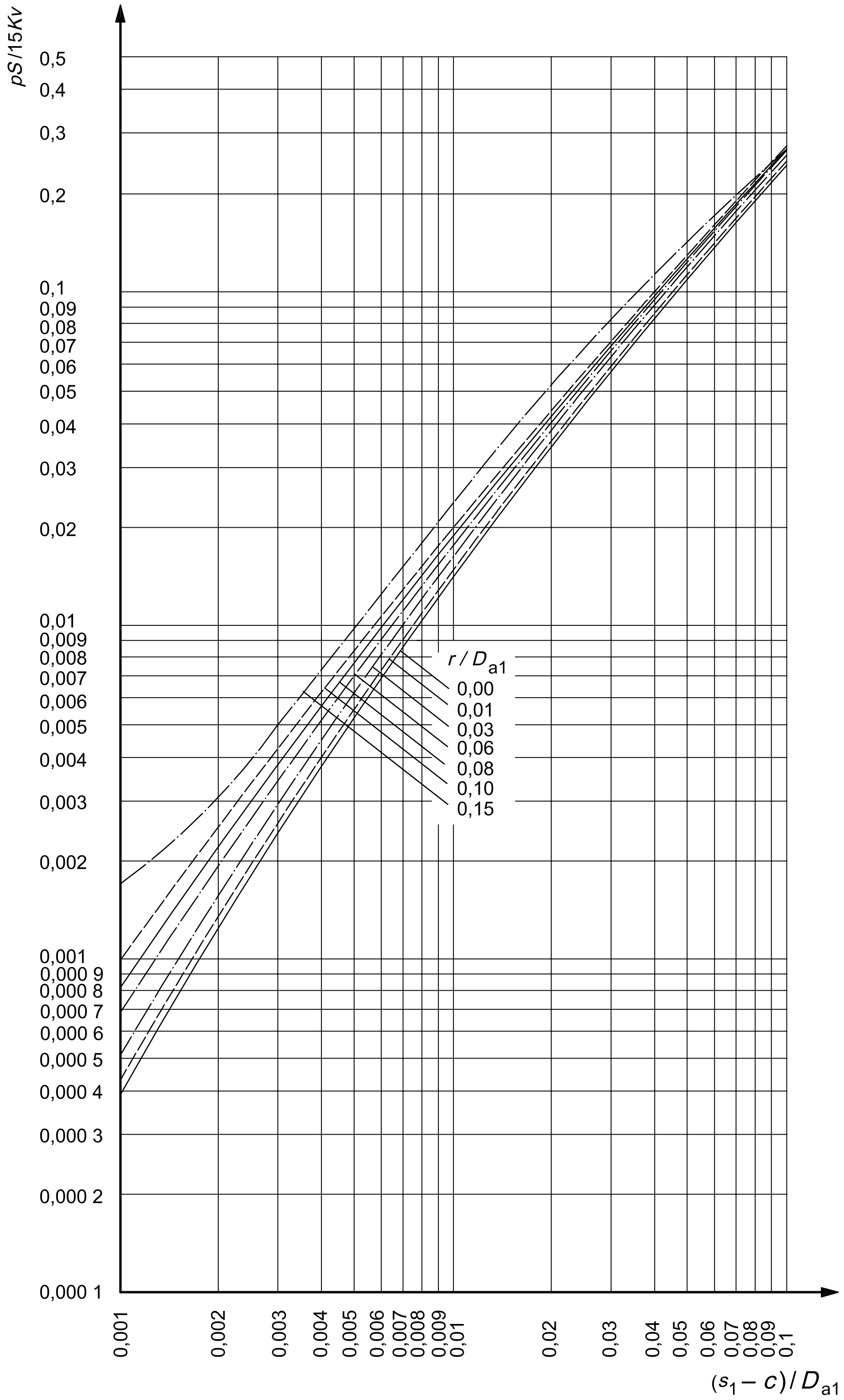
Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \frac{pS}{15Kv} = e^{Z}$

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

Figure 4 (*continued*)



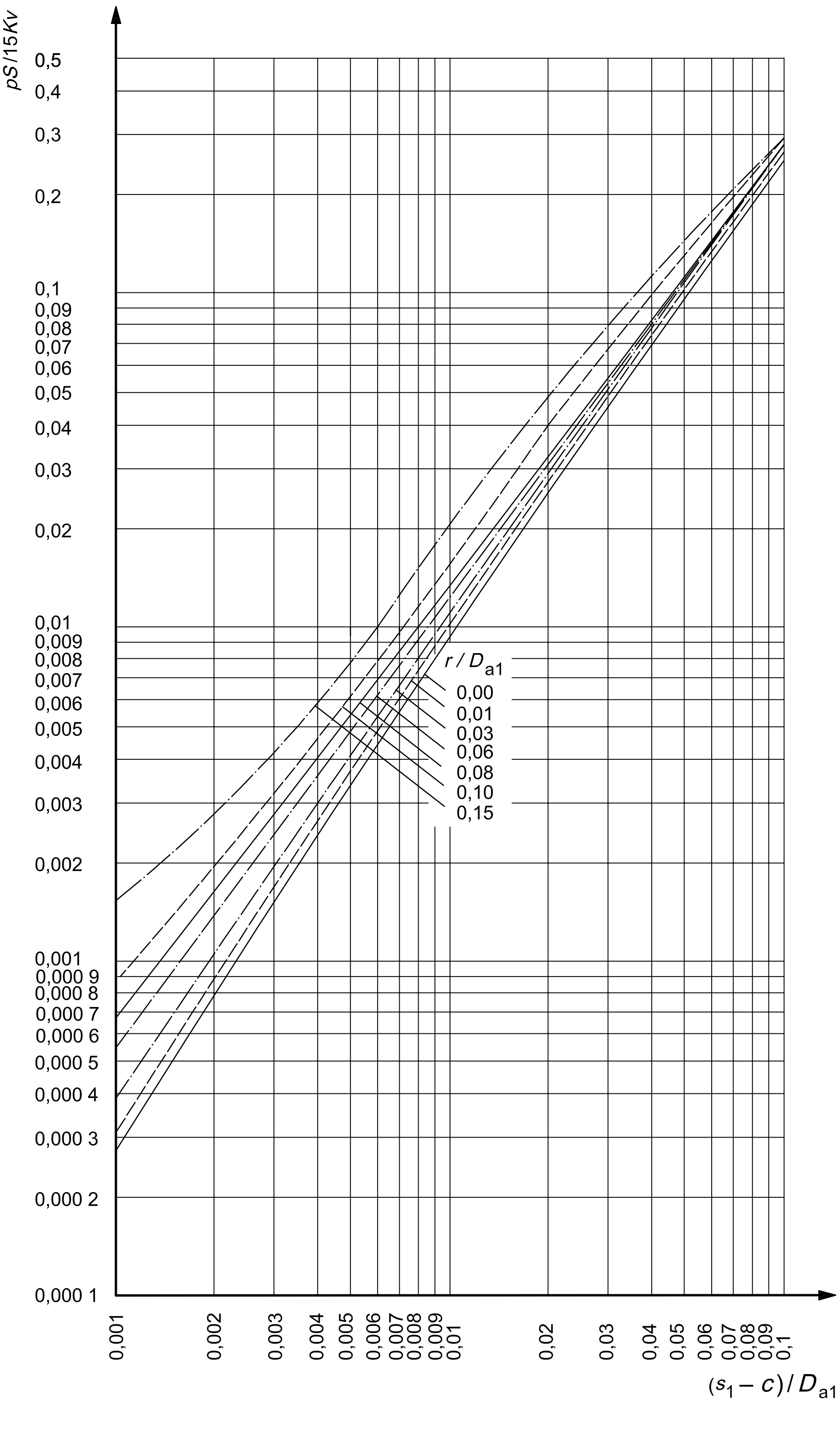
Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \frac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)



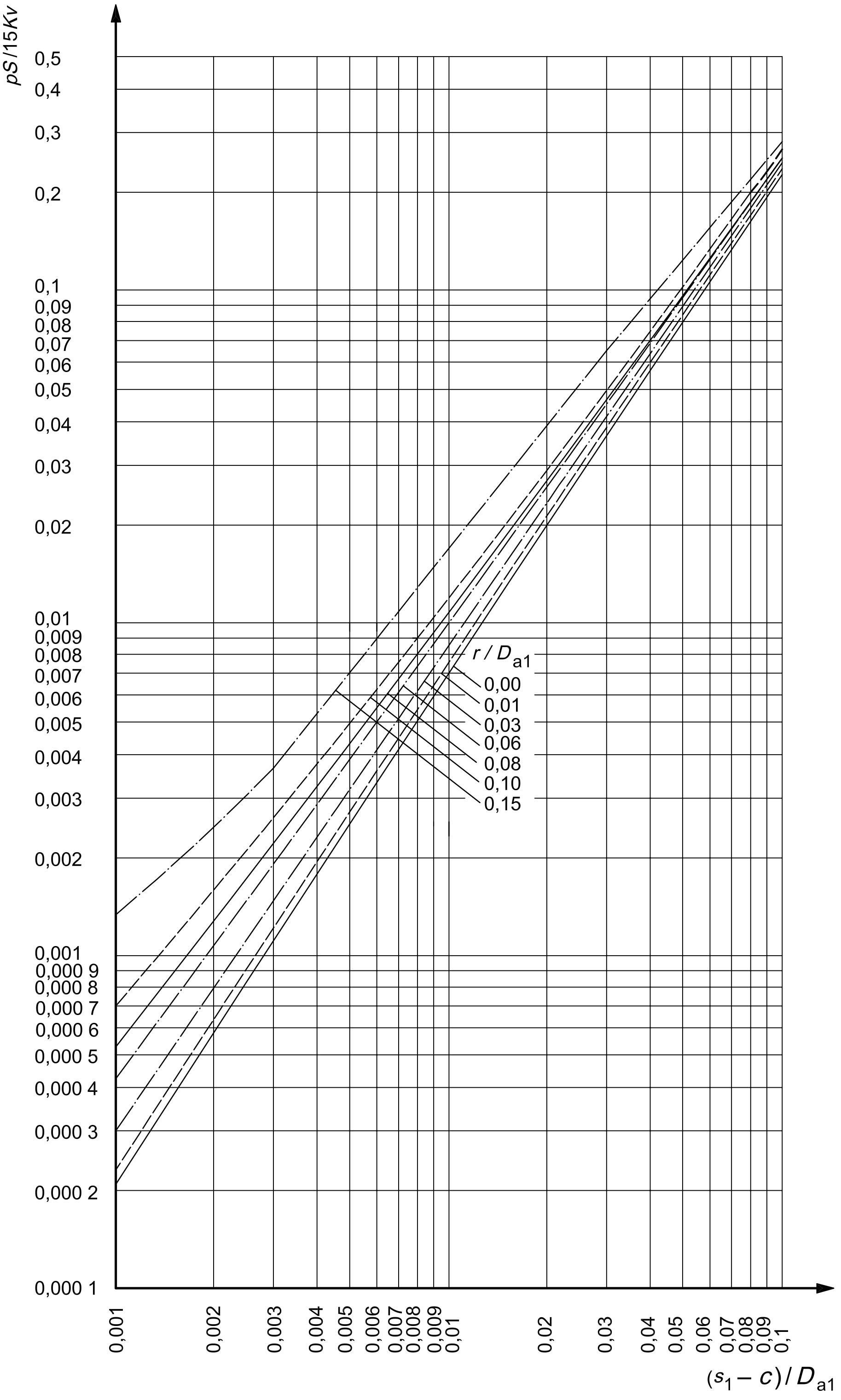
Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \frac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)



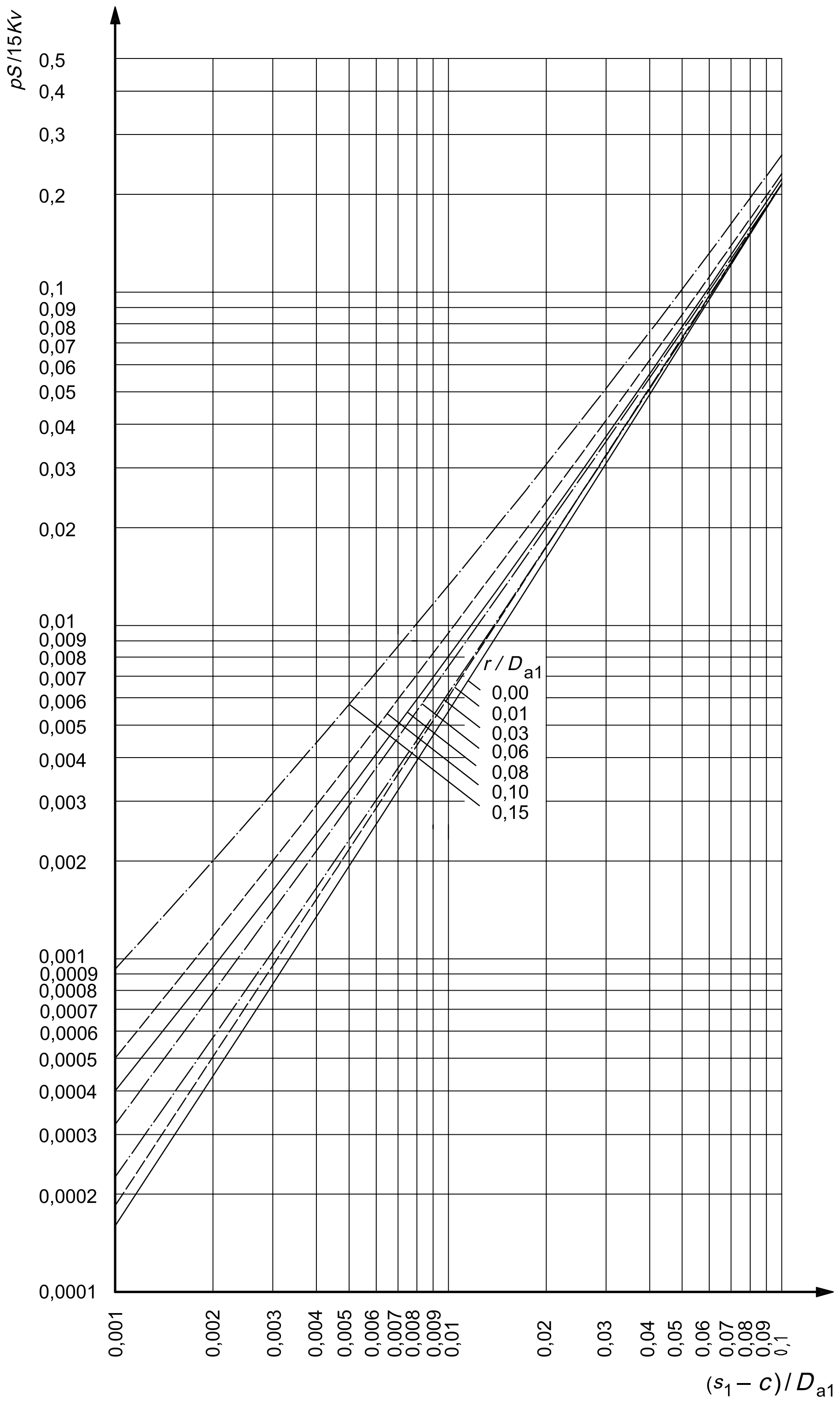
Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \dfrac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)



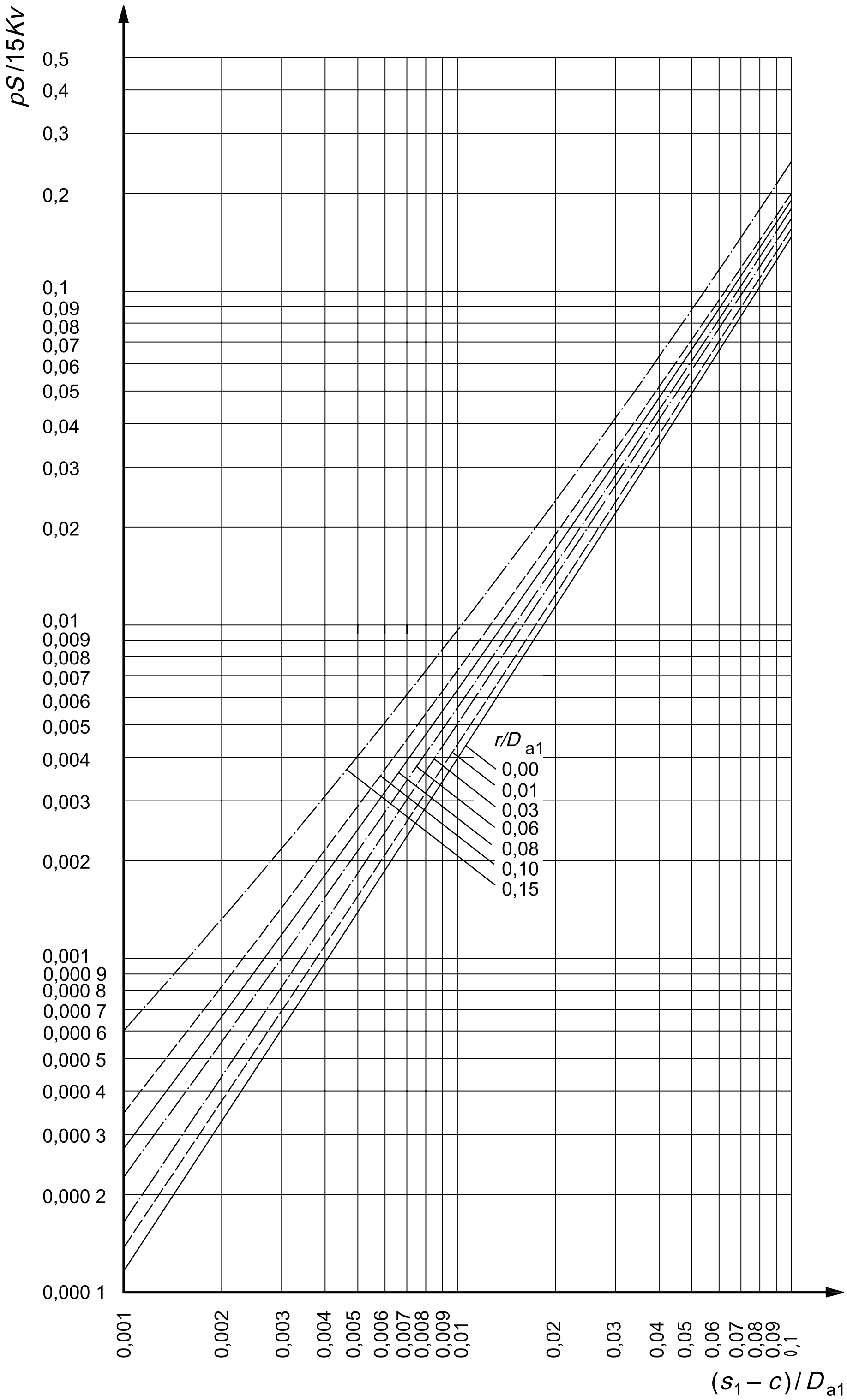
Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \frac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)



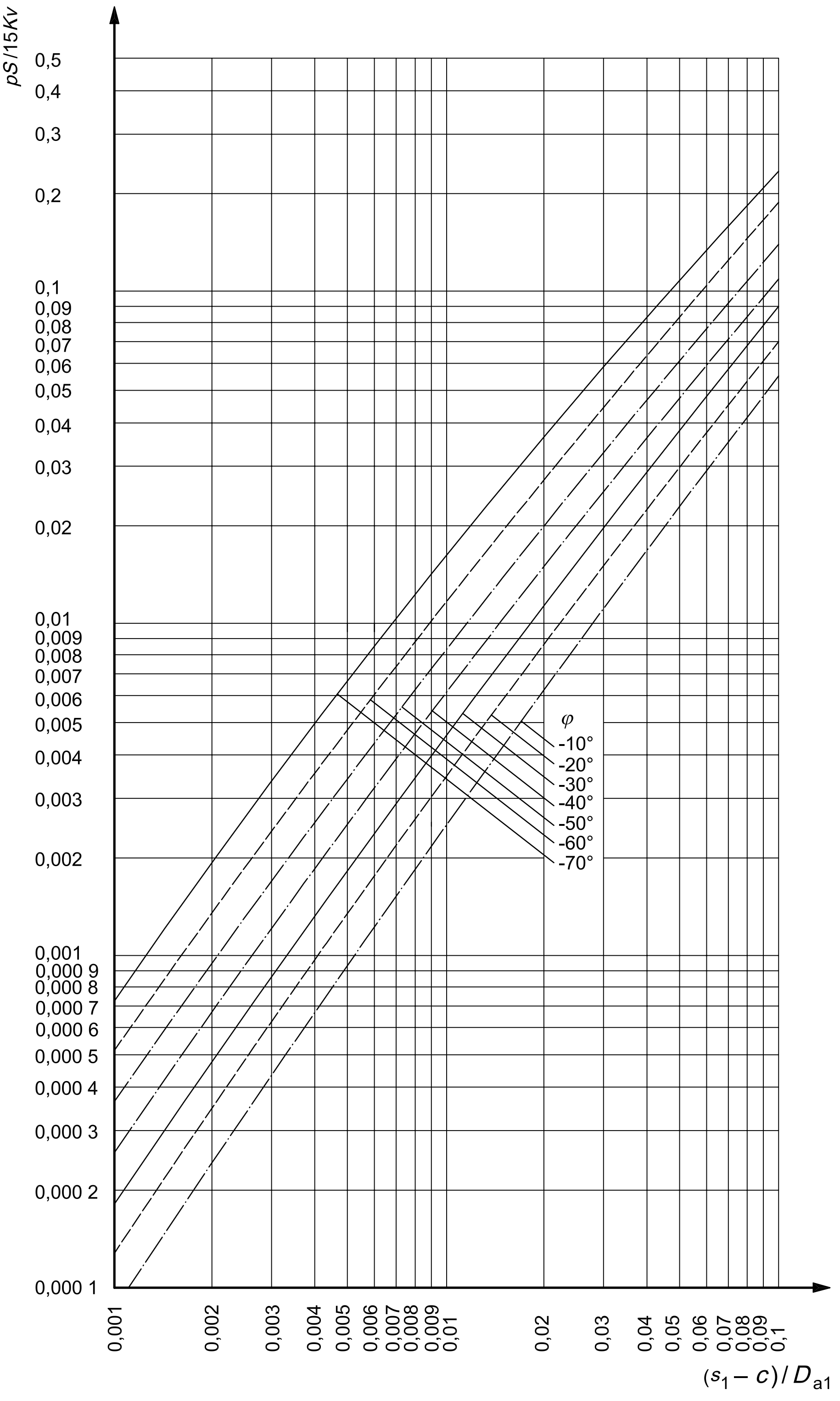
Key

X = ln[$(s_1 - c)/D_{a1}$] $Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\frac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)



Key

X = ln[$(s_1 - c)/D_{a1}$] $\quad Z = \sum_{i=1}^{4} \sum_{j=1}^{2} A_{ij} \cdot X^{i-1} \cdot Y^{j-1}$

Y = r/D_{a1} $\quad \frac{pS}{15Kv} = e^Z$

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

Figure 4 (*continued*)

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

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

$$s_1 = s_g = 0{,}3(D_{a1} - r)\, x \frac{|\varphi|}{90}\, x \sqrt{\frac{p}{10\left(\frac{K}{S}\right) v}} + c \tag{13}$$

10.3.6.5.8 External-pressure calculation (pressure on the convex surface)

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

For the example shown in Figure 5 the equivalent cylinder diameter between the knuckle and the stiffener is:

$$D_a = \frac{D_{a1} + D_{a2}}{2 \cos |\varphi|} \tag{14}$$

and the equivalent cylinder length is:

$$l = \frac{D_{a1} - D_{a2}}{2 \sin |\varphi|} \tag{15}$$

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

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

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

$$I \geqslant 0{,}010\,4\, S_k \frac{p(D_{a1})^4}{10E} \tan|\varphi| \tag{16}$$

$$A \geqslant 0{,}125\, S_p \frac{p(D_{a1})^2}{10K} \tan|\varphi| \tag{17}$$

where

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 10.3.6.5.7 and the safety factors S_p for cylinders from 10.3.2.4 or 10.3.3.4 increased by 20 %. For thickness calculations in the corner area, v shall be the value applicable for internal pressure.

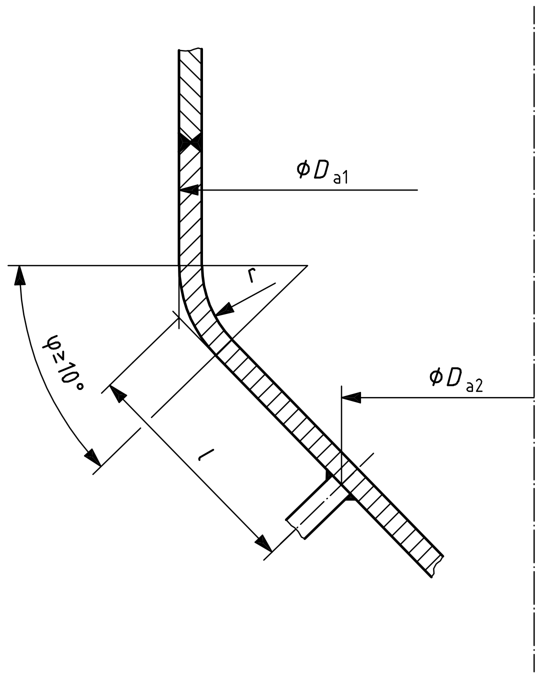


Figure 5 — Geometrical quantities in the case of loading by external pressure

10.3.6.6 Flat ends

10.3.6.6.1 Symbols and units

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

d_1, d_2, etc., opening diameters, in mm;

D_1, flat-end diameters, in mm. A_s is shown in Figure 6.

10.3.6.6.2 Field of application

The field of application includes welded or solid flat ends where Poisson's ratio is approximately 0,3, and:

$$\frac{(s-c)}{D} \geqslant \sqrt[4]{\frac{0{,}008\,7\,p}{E}}$$

and

$$3\frac{(s_e-c)}{D_1} \leqslant 1$$

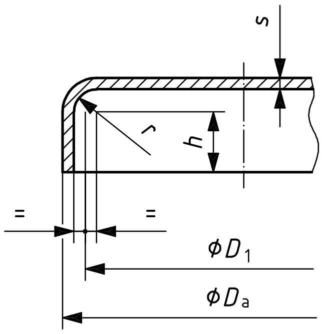
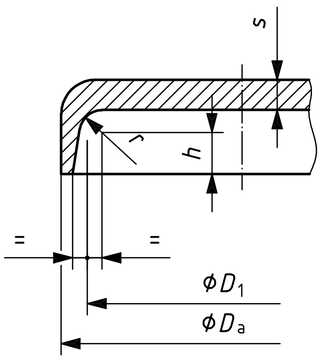
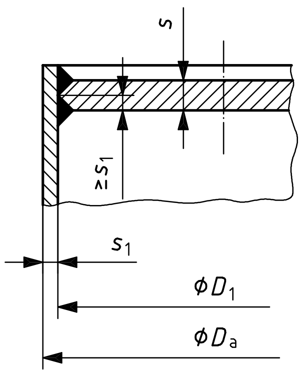
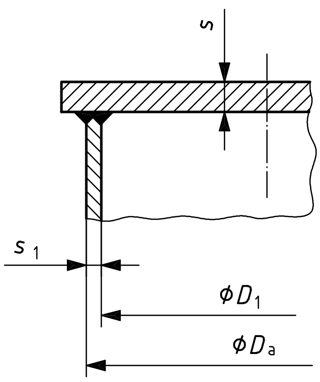
<table>
<tr><th>Type of flat-end design
(principle only)</th><th>Conditions</th><th>Design factor C</th></tr>
<tr><td>a) Flat end
</td><td>1. Knuckle radius :
<table><tr><th>D_a</th><th></th><th>r_{min}</th></tr><tr><td colspan="2">up to 500</td><td>30</td></tr><tr><td>over 500</td><td>up to 1 400</td><td>35</td></tr><tr><td>over 1 400</td><td>up to 1 600</td><td>40</td></tr><tr><td>over 1 600</td><td>up to 1 900</td><td>45</td></tr><tr><td>over 1900</td><td></td><td>50</td></tr></table>and $r \geqslant 1{,}3s$
2. cylindrical part: $h \geqslant 3{,}5 \times s$</td><td>0,31</td></tr>
<tr><td>b) Forged or pressed flat end
</td><td>1. Knuckle radius:
$r \geqslant \frac{s}{3}$, however at least 8 mm
2. Cylindrical part: $h \geqslant s$</td><td>0,41</td></tr>
<tr><td>c) Flat plate welded into the shell from both sides
</td><td>Plate thickness:
$s \leqslant 3s_1$
$s > 3s_1$
$m = \frac{s_{1}r}{s_1}$
$s_1 r$ = minimum required thickness for pressure</td><td>$\sqrt{0{,}33m}$ 0,45 min</td></tr>
<tr><td>d) Plate welded into the shell with welds at both sides of the latter
</td><td>Plate thickness:
$s \leqslant 3s_1$
$s > 3s_1$
Only killed steels may be utilized. When plate material is employed over an area of at least $3s_1$ in the weld zone there shall be no evidence of material discontinuities in the plate.</td><td>$\sqrt{0{,}33m}$ 0,45 min</td></tr>
</table>

Figure 6 — Design factors for unstayed circular flat ends and plates

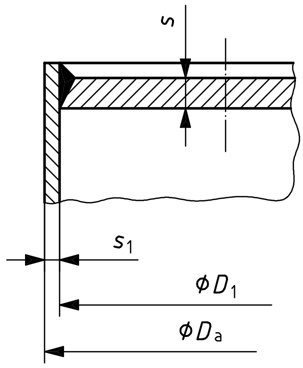
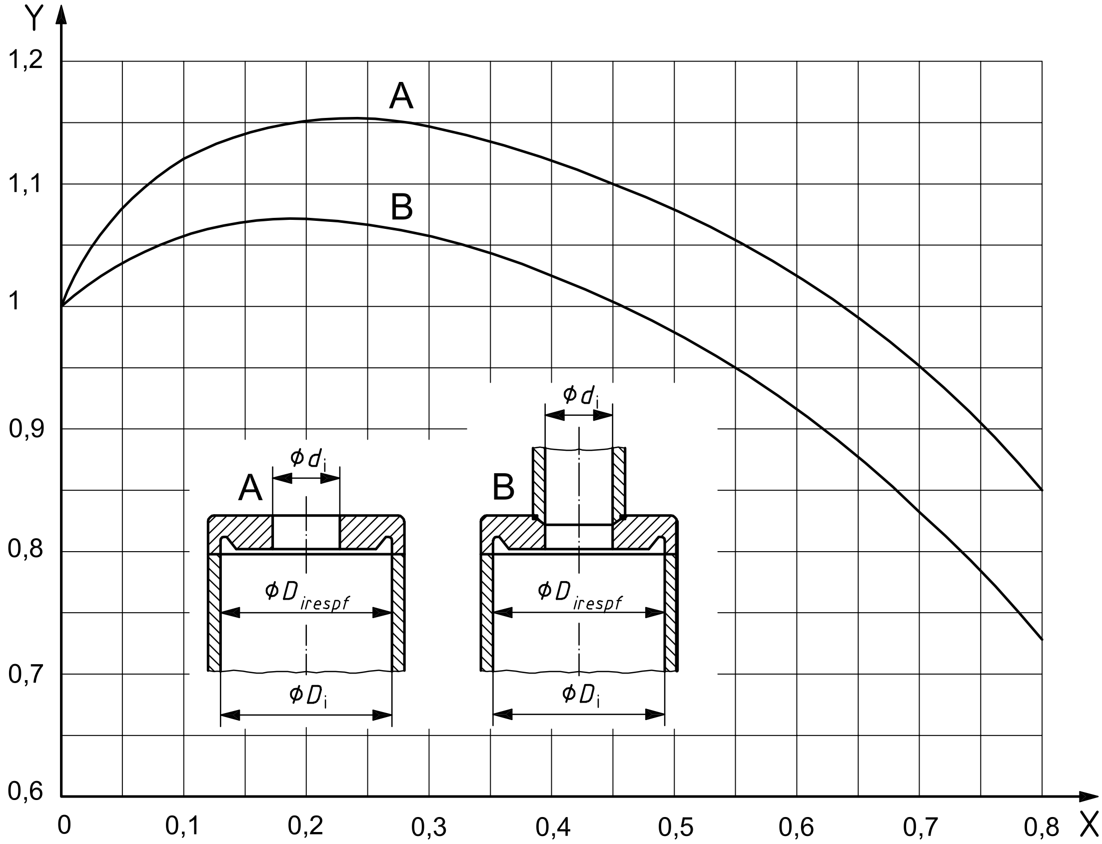
e) Flat plate welded into the shell from one side only 	Plate thickness: $s \leqslant 3s_1$ $s > 3s_1$	$\sqrt{0{,}33m}$ 0,45 min

Figure 6 (*continued*)

10.3.6.6.3 Openings

Openings are calculated in accordance with 10.3.6.6.4 but with the C factor multiplied by C_A, where C_A is given in Figure 7.



Key

Y Opening factor C_A

X Ratio d_i/D_i

Type A

d_i = inside diameter of opening

D_i = design diameter

D_{irespf} = diameter of the short side of the elliptical end

Type B

d_i = inside diameter of opening

D_i = design diameter

D_{irespf} = diameter of the short side of the elliptical end

Figure 7 — Opening factor, C_A, for flat ends and plates without additional marginal moment

f = short side of elliptical end

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

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

f = short side of elliptical end

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

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 7 (*continued*)

10.3.6.6.4 Calculation

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

$$s = CD_1 \sqrt{\frac{0{,}1pS}{K}} + c \tag{18}$$

where C and D_1 are taken from Figure 6.

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

$$s = CC_E \sqrt{\frac{0{,}1pS}{K}} + c \tag{19}$$

where C_E is taken from Figure 8.

10.3.6.7 Openings in cylinders, spheres and cones

10.3.6.7.1 Symbols and units

For the purposes of 10.3.6.7, the following symbols apply in addition to those given in Clause 4:

b	width of pad, ring or shell reinforcement	mm
h	thickness of pad reinforcement	mm
l	ligament (web) between two nozzles	mm
l_s	length of nozzle reinforcement outstanding	mm
s	length of nozzle reinforcement in stand	mm
s_A	required wall thickness at opening edge	mm
s_S	wall thickness of nozzle	mm
t	in this context, centre-to-centre distance between two nozzles	mm

10.3.6.7.2 Field of application

Round openings and the reinforcement of round openings in cylinders, spheres and cones within the following limits:

$$0{,}002 \leqslant \frac{(s-c)}{D_a} \leqslant 0{,}1$$

$$\frac{(s-c)}{D_a} < 0{,}002 \text{ is acceptable if } \frac{d_i}{D_a} \leqslant \frac{1}{3}.$$

These rules only apply to cones if the wall thickness is determined by the circumferential stress.

NOTE 1 Additional external forces and moments are not covered by this subclause and are to be considered separately where necessary.

NOTE 2 These design rules permit plastic deformations of up to 1 % at highly stressed local areas during pressure test. Openings should therefore be carefully designed to avoid abrupt changes in geometry.

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

10.3.6.7.3 Reinforcement methods

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

— increase of shell thickness;

— set-in or set-on ring reinforcement;

— pad reinforcement (see Figure 9);

— increase of nozzle thickness (see Figures 10 and 11);

— pad and nozzle reinforcement.

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

10.3.6.7.4 Design of openings

All nozzles shall be attached to the vessel wall with a not full-penetration weld unless the attachment weld is maintained at atmospheric temperature at all times.

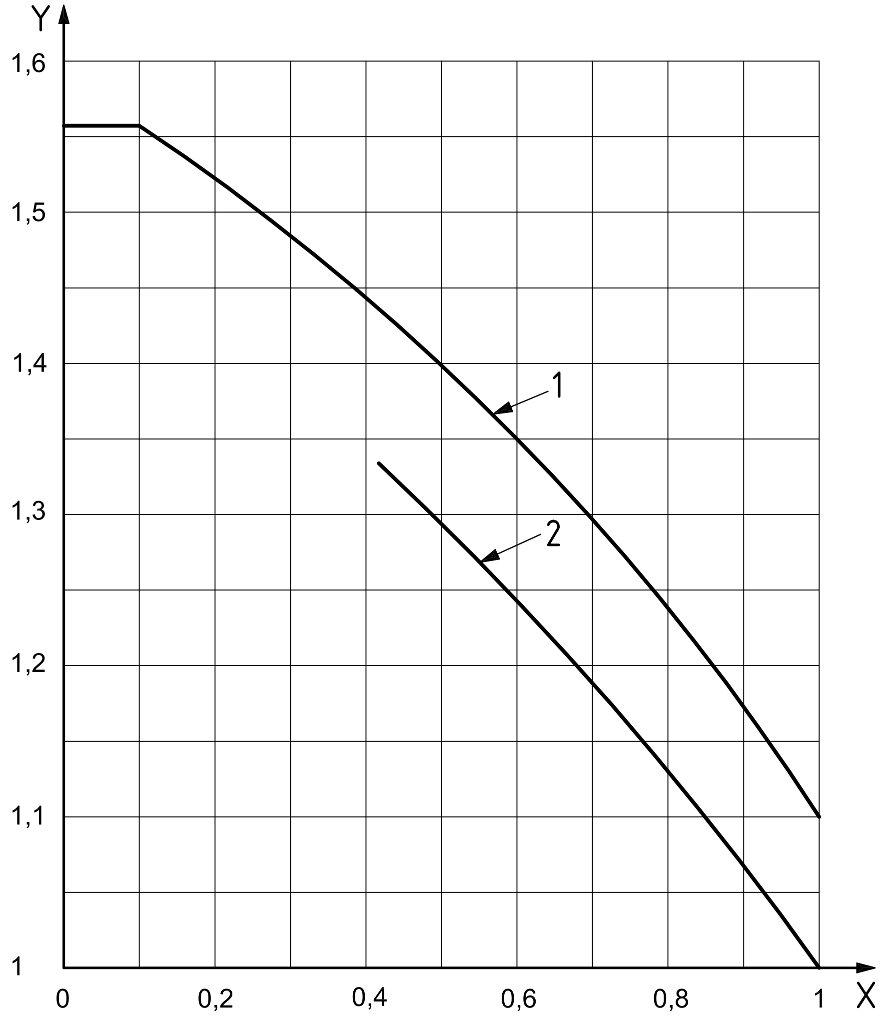
The fillet weld on a reinforcing pad shall have a minimum throat thickness of half of the pad thickness.

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 10.3.6.7.5 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.

10.3.6.7.5 Calculation

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



Key

1 Rectangular plate

2 Elliptical plate

Rectangular plates

f = short side of the rectangular plate

e = long side of the rectangular plate

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

A_1 = 1,589 146 00

A_2 = –0,239 349 90

A_3 = –0,335 179 80

A_4 = 0,085 211 76

Elliptical plates

f = short side of the elliptical plate

e = long side of the elliptical plate

$$C_\mathrm{e} = \left\{ \sum_{i=1}^{4} A_i \left(\frac{f}{e}\right)^{i-1} \quad \middle| \quad 0{,}43 < \left(\frac{f}{e}\right) \leqslant 1{,}0 \right\}$$

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 8 — Design factor, C_e, for rectangular or elliptical flat plates

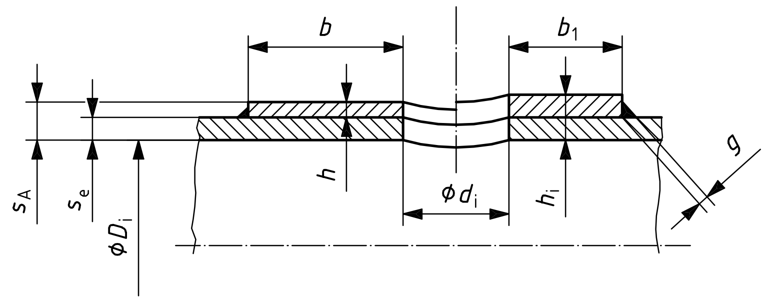
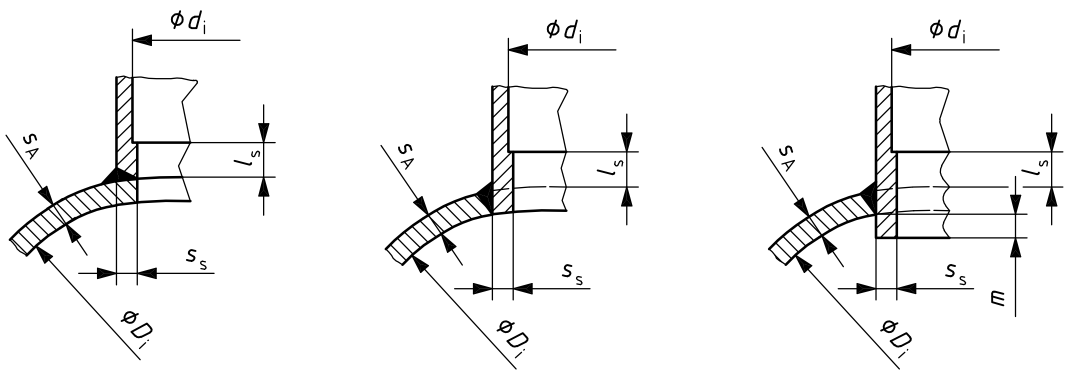


Figure 9 — Pad reinforcement



Type a

Type b

Type c

Figure 10 — Nozzle reinforcement

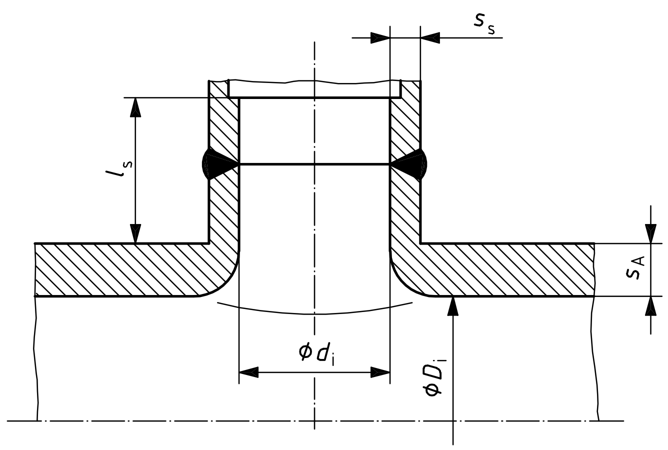


Figure 11 — Necked-out opening

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, equivalent static loads shall be substituted for static plus dynamic loads.

The analysis shall take account of gross structural discontinuities.

Annex B or ASME VIII-2 provides terminology and acceptable stress limits when an elastic stress analysis is performed.

Acceptable calculation methods include:

- finite element;
- finite difference;
- boundary element;
- recognized text books, codes and standards.

Planned and controlled experimental means may be used in order to confirm these calculations, for example by application of strain gauges to verify stress levels.

11 Fabrication

11.1 General

11.1.1 The manufacturer, or his or her subcontractor, shall have equipment available to ensure manufacture and testing in accordance with the design.

11.1.2 The manufacturer shall maintain:

- a system of material traceability for pressure-bearing parts used in the construction of the inner vessel;
- design dimensions within specified tolerances;
- necessary cleanliness of the inner vessel, associated piping and other equipment which could come in contact with the cryogenic fluid.

11.2 Cutting

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

11.3 Cold forming

11.3.1 Austenitic stainless steel

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

a) for operating temperatures down to –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 15 % or it is demonstrable that the residual elongation is not less than 15 %;

b) for operating temperatures below –196 °C: the cold-forming deformation is greater than or equal to 15 % and it is demonstrated that the residual elongation is not less than 15 %;

c) for formed heads, the test certificate for the base material shows an elongation at fracture A_5:

— not less than 40 % in the case of wall thicknesses not more than 15 mm at design temperatures down to –196 °C;

— not less than 45 % in the case of wall thicknesses more than 15 mm at design temperatures down to –196 °C;

— not less than 50 % at design temperatures below –196 °C.

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

Cold-forming deformation can be calculated according to EN 13445-4, Clause 9.

11.3.2 Ferritic steel

The following requirements for post-forming heat treatment shall be observed:

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 $R_m \leqslant 530$ N/mm^2

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

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

11.3.3 Aluminium or aluminium alloy

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

11.4 Hot forming

11.4.1 General

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

11.4.2 Austenitic stainless steel

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

11.4.3 Ferritic steel

The following requirements for post-forming heat treatment shall be observed:

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 specified temperature range, specified in the material standards; further test pieces are not required.

11.4.4 Aluminium or aluminium alloy

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

11.5 Manufacturing tolerances

11.5.1 Plate alignment

Except where a tapered transition is provided, misalignment of the surfaces of adjacent plates at welded seams shall be for longitudinal and circumferential seams, not more than 25 % of the thickness of the thinner plate up to a maximum of 3 mm.

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

The distance between either surface of the thicker plate and the centre line of the thinner plate of tapered seams shall be for longitudinal and circumferential seams, not less than 35 % of the thickness of the thinner plate.

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

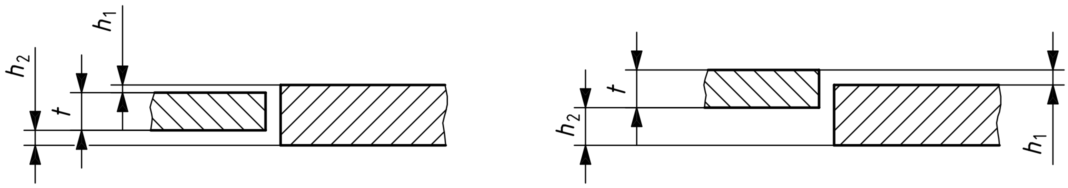
These requirements are illustrated in Figure 12.

The following nomenclature applies:

— h, h_1, h_2 = surface misalignments;

— t = thickness of the thinner plate;

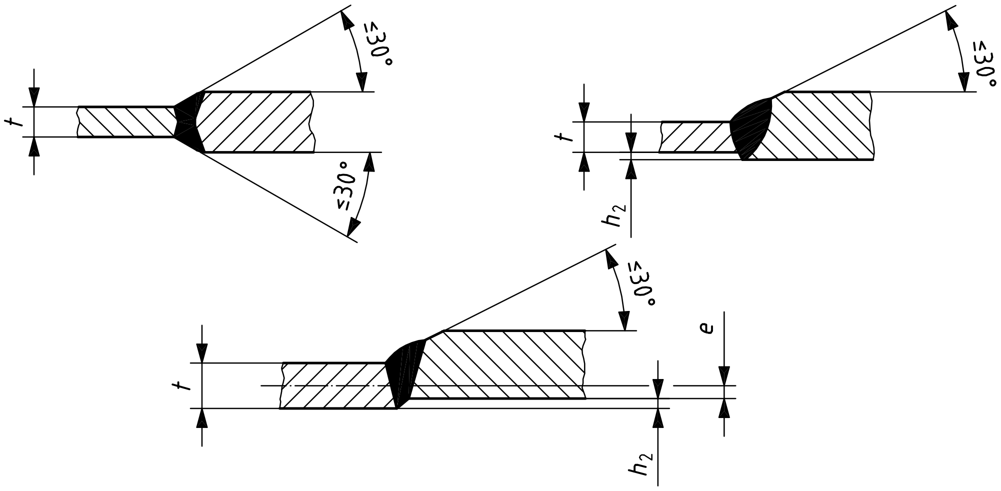
— e = distance from the surface of the thicker plate to the centre line of the thinner plate.



NOTE For longitudinal and circumferential seams:

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

a) Seams which do not require a taper



NOTE For longitudinal and circumferential seams:

$e = \frac{t}{2} - h_2 \geqslant 0{,}35t$.

b) Seams which do require a taper

Figure 12 — Plate alignment

11.5.2 Thickness

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

11.5.3 Dished ends

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

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

11.5.4 Cylinders

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

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

$$\text{out of roundness } \frac{200(D_{\text{max}} - D_{\text{min}})}{D_{\text{max}} + D_{\text{min}}} \ \% \qquad (20)$$

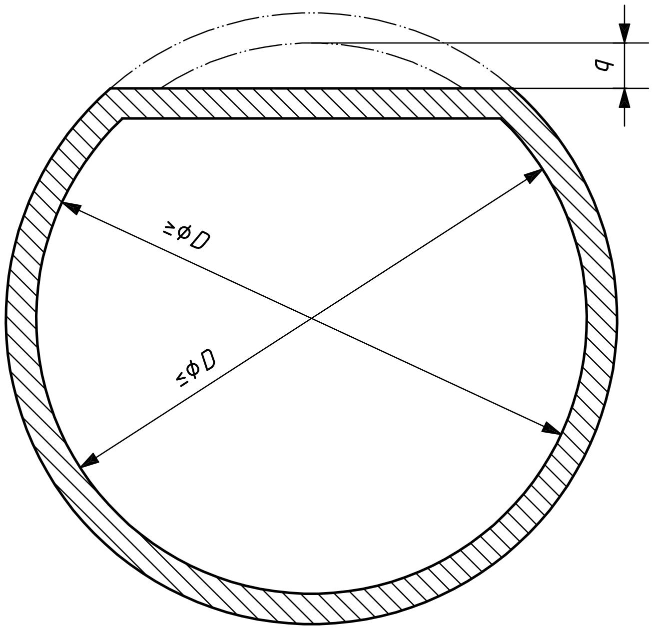
shall be not more than the values shown in Table 5.

Table 5 — Permitted out of roundness

Wall thickness to diameter ratio	Permitted out of roundness for	
	internal pressure	external pressure
s/D = 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 (see Figure 13).



NOTE 1 Definitions:

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

u equivalent to $\frac{4}{D_a} \times q \times 100$

NOTE 2 Limitations:

$u \leqslant 15\ \%$

$q \leqslant 0{,}003\ 75 D_a$

Figure 13 — Allowable shape imperfections

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 6.

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 14. Two readings shall be taken, P_1 and P_2, on each side of the seam, at any particular location. The maximum peaking is taken as being equivalent to 0,25 (P_1 + P_2).

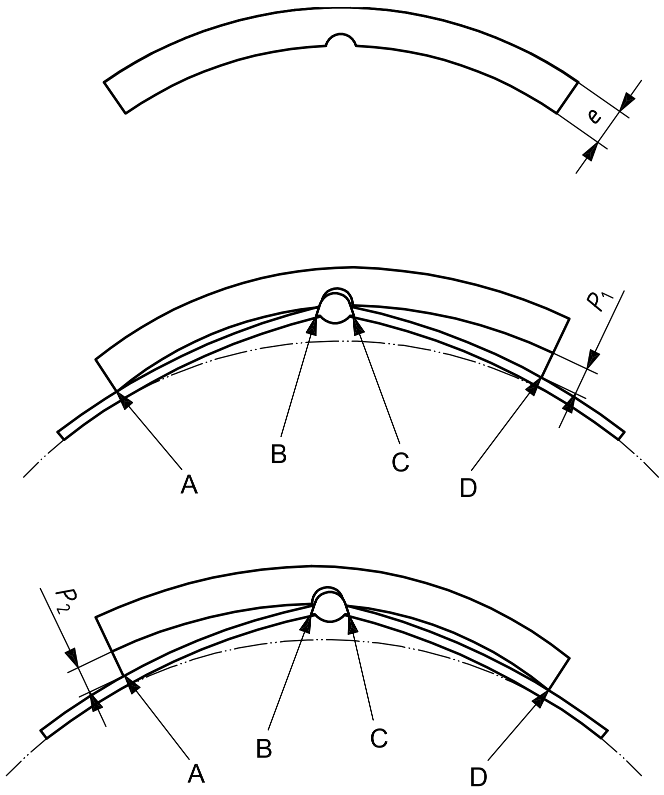


Figure 14 — Gauge details

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 are not prohibited. The maximum peaking value permitted is given in Table 6.

Table 6 — Maximum permitted peaking

Dimensions in millimetres

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

For all ratios, a maximum permitted peaking is e.

For cylinders subject to external pressure and where the circumference has a flattened portion, it shall be demonstrated that the shell has sufficient strength to avoid plastic deformation where the depth of flattening is more than 0,4 % of the outside diameter of the cylinder. The depth of flattening shall be measured as a deviation from the normal curvature or from the line of the cylindrical shell. Adequate strength may be determined by calculation in accordance with formula (11) of 10.3.6.5.5, using a value of u determined as follows:

$$u = \frac{400}{D_a} \times q \tag{21}$$

where

q is the depth of flattening, in mm;

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

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

The welding method shall be appropriate and shall be carried out by qualified welders and/or operators, the materials shall be compatible, and there shall be verification by a welding procedure test.

11.6.2 Qualification

Welding procedures shall be approved in accordance with ISO 15614-2, ISO 15614-3 or with ISO 14732 as applicable, or with equivalent standards.

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

11.6.3 Temporary attachments

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

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

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

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

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

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

11.6.4 Welded joints

11.6.4.1 Some specific weld details appropriate to vessels conforming to this part of ISO 20421 are given in Annex E. These details show sound and currently accepted practice. It is not intended that these be mandatory, nor should they restrict the development of welding technology in any way.

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

— the method of manufacture;

— the service conditions;

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

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

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

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

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

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

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

— the material shall not be heat treated after welding.

See also C.2.7, C.2.8, C.2.9, and C.2.10.

11.7 Non-welded joints

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

12 Inspection and testing

12.1 Quality plan

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

12.1.1 Inspection stages during manufacture of an inner vessel

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

— verification of material test certificates and correlation with materials;

— approval of weld procedure qualification records;

— approval of welders' qualification records;

— examination of material cut edges;

— examination of setup of seams for welding including dimensional check;

— examination of weld preparations, tack welds;

— visual examination of welds;

— verification of non-destructive testing;

— testing production control test plates for welds and, where required, for formed parts after heat treatment;

— verification of cleaning of inside surface of vessel;

— examination of completed vessel including dimensional check;

— pressure test.

12.1.2 Additional inspection stages during manufacture of a large transportable cryogenic vessel

The following inspection stages shall be conducted during the manufacture of a large transportable cryogenic vessel:

— verification of cleanliness and dryness of the cryogenic vessel (see ISO 23208);

— visual examination of welds not covered by 12.1.1;

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

— leak test of external piping;

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

— check of installation of vacuum-space relief device;

— check of nameplate 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 shall be used 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 100 m for other metals, one test plate per 130 m for other metals, provided the joints are made within any three-month period.

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 ISO 21028-1 and ISO 21028-2;

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

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

12.2.2 Extent of testing

The number and type of test specimens to be taken from the test plate is dependent on material and thickness, and shall be in accordance with the requirements in Tables 6 and 7 for the particular material and thickness applicable.

NOTE The symbols for Tables 8 and 9 are given in Table 7.

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

Prior to cutting the test piece, non-destructive testing of the test plate may be applied in order that the test specimens are taken from sound areas.

Table 7 — 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 8 — Testing of production test plates for steels

Group	e in mm	Test specimens
Fine-grain steels normalized or thermo-mechanically treated	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e \leqslant 35$	3 IW, 3 IH, 1 T, 1 Ma
Ni steels up to 9 % Ni	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 3 IH, 1 T, 1 Ma
Austenitic stainless steels	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e$	3 IW, 1 T, 1 Ma
Impact test requirement for steels used below 77 K	$e > 2{,}5$	3 IW, 1 T, 1 Ma

Table 9 — Testing of production test plates for aluminium

Group	e in mm	Test specimens
Pure aluminium and aluminium with up to 1,5 % impurities or alloy content	$e \leqslant 12$	1 BF, 1 BR, 1 T, 1 Ma
	$12 < e \leqslant 35$	2 BS, 1 T, 1 Ma

12.3 Non-destructive testing

12.3.1 General

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

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

Non-destructive testing is not required on the outer jacket of cryogenic vessels.

12.3.2 Extent of examination for surface imperfections

Visual examination (if necessary aided by x5 lens) shall be carried out on all weld deposits. See Table 11 for acceptance levels. If any doubt arises, this examination shall be supplemented by surface-crack detection.

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

12.3.3 Extent of examination for inner-vessel weld seams

Examination of the inner vessel for inner-vessel weld seams shall be by radiographic examination unless a special case is made to justify ultra-sonic or other methods. The extent of examination of main seams on the inner vessel shall be in accordance with Table 10. See Table 12 for acceptance levels.

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

Table 10 — Extent of radiographic examination for welded seams of the inner vessel

Longitudinal seams	T junctions [a]	Circumferential seams [a]
100 %	100 %	100 %

[a] Unless the following conditions are met, in which case no radiographic examination is required, but weld joint factor of 0,7 shall be used for the circumferential seam:

1) Circumferential seam is not a butt joint.

2) Length of the vessel is less than 1 500 mm.

3) Design pressure of the vessel is less than 2 bar.

4) The vessel is not to carry flammable or toxic fluids.

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

NOTE 2 Additional examination may be required when pneumatic proof testing is used.

12.3.4 Acceptance levels

12.3.4.1 Acceptance levels for surface imperfections

Table 11 shows the acceptance criteria for surface imperfections.

Table 11 — Acceptance levels for surface imperfections

Imperfection	Limit for acceptable imperfection
Lack of penetration	Not permitted.
Undercut	Where the thickness is less than 3 mm no visible undercut is permitted.
	Where the thickness is not less than 3 mm, slight and intermittent undercut is acceptable, provided that it is not sharp and is not more than 0,5 mm.
Shrinkage groove	As undercut.
Root concavity	As undercut.
Excessive penetration	Where the thickness is less than 5 mm, excessive penetration shall be not more than 2 mm.
	Where the thickness is not less than 5 mm, excessive penetration shall not be more than 3 mm.
Excess weld material	Where the thickness is less than 5 mm, excess weld metal shall not be greater than 2 mm and the weld shall blend smoothly.
	Where the thickness is 5 mm or greater, the maximum excess weld metal shall not exceed 3 mm and the weld shall blend smoothly.
Irregular surface	Reinforcement to be of continuous and regular shape with complete filling of groove.
Sagging	
Incompletely filled groove	
Irregular width	
Poor restart	
Overlap	Not permitted.
Linear misalignment	See 11.5.1.
Arc strike	Grind smooth, acceptable subject to thickness measurement and surface-crack detection test.
Spatter	
Tungsten spatter	
Torn surface	
Grinding mark	
Chipping mark	
Surface cracks	Not permitted.

12.3.4.2 Acceptance levels for internal volumetric imperfections

Table 12 shows the acceptance criteria for internal volumetric imperfections detected by radiographic examination.

Table 12 — Acceptance levels for internal volumetric imperfections

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

12.3.4.3 Extent of examination of non-welded joints

Where non-welded joints are used between metallic materials and/or non-metallic materials, the quality plan referred to in 12.1 shall include reference to an adequate technical specification. This technical specification shall include the description of the requirements for inspection and testing, together with the criteria necessary to allow for the repair of any imperfections.

12.4 Rectification

The weld 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.

12.5 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 the design pressure.

The test pressure for the inner vessel shall not be less than the highest of:

$1{,}3(p_s + 1)$ bar

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

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 show no gross plastic deformation or leakage (except as in Annex D). 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 only 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 in accordance with 10.2.3.11. It is not necessary to strength test mechanical joints and fittings that have demonstrated satisfactory in-service experience.

12.5.5 Leakproofness tests (see 12.1.2).

13 Marking and labelling

The large transportable cryogenic vessel shall bear the markings and labellings required by the applicable Road/Rail/Sea Regulation. Examples of tank plates (of the complete tank and of the inner vessel) for fixed tanks (tank vehicles), demountable tanks, tank containers and tank swap bodies are given for information in Annex A.

At a minimum, the labelling shall include the size limitations, if any, for operation and transportation mode.

Marking shall be on a corrosion-resistant metal plate, permanently attached to the outer jacket of the cryogenic vessel, in a conspicuous place quickly accessible for inspection.

14 Final acceptance test

The first date and stamp of the expert (marks 14, 16 — see A.1) after the final acceptance test on the large transportable cryogenic vessel confirms that the marking and labelling, and that the vessel itself, meets the requirements of this part of ISO 20421. These marks also confirm that the large transportable cryogenic vessel is ready for putting into service provided that the requirements of ISO 20421-2 are met.

15 Periodic inspection

The large transportable cryogenic vessel has to be inspected periodically in accordance with the relevant Road/Rail/Sea Regulations. By affixing the mark of the inspector and the date on the data plate, the inspector confirms the successfully completed periodic inspection.

The periodic inspection shall be done in accordance with ISO 20421-2.

Annex A
(informative)

Examples of tank plates

A.1 Example 1: Tank plate (of the complete tank) for fixed tanks of road tankers (tank vehicles), demountable tanks, tank containers and tank swap bodies

1	**Manufacturer**				
2	**Approval number**		2a	**Conformity mark**	
3	**Manufacturer's serial number**		4	**Year of manufacture**	
5	**Tank code**		6	**Test pressure**	**bar**
7	**Capacity of the tank**	**litres**	8	**Design temperature**	**°C**
9	**Material and materials standards of**	**inner vessel:**			
		outer jacket:			
10	**Insulation**	**vacuum-insulated** or **thermally insulated by vacuum**			
11	**Maximum allowable working pressure**	**bar**			
12	**The proper shipping name of the gas(es), for whose transport the portable tank is approved**				
13	**Minimum filling temperature for each gas**	**°C**	**°C**	**°C**	
14	**Maximum permissible load mass for each gas**	**kg**	**kg**	**kg**	
15	**Date (month and year) of initial test and most recent periodic test**				
16	**Stamp of the expert who carried out the tests**				
17	**Name of owner or operator**				
18	**Maximum permissible mass**	**kg**			
19	**Unladen mass**	**kg**			

Figure A.1 — Tank plate for the complete tank

The markings of Figure A.1 are explained in Table A.1.

Table A.1 — Explanation of the marking of the tank plate (compete tank)

No.	Content/explanation
1	Manufacturer's name or mark of the complete tank.
2	Approval number given by the competent authority or body designated by this authority .
2a	Conformity mark (π) according to annex VII of the Directive 1999/36/EC, accompanied by the identification number of the notified or approved body.
3	Serial or production number issued by the manufacturer.
4	Year of manufacture.
5	Tank code according to the certificate.
6	Test pressure (gauge) of the shell, in MPa or bar.
7	Water capacity, in litres.
8	Design temperature, in °C.
9	Materials of the shell (and of the ends if different) of the inner vessel and reference to materials standards, if available and materials of the shell (and of the ends if different) of the outer jacket and reference to materials standards, if available.
10	Type of insulation of the tank in words, e.g. "thermally insulated" or "thermally insulated by vacuum", if applicable, in an official language of the country of registration and also, if that language is not English, French or German, in English, French or German, unless any agreements concluded between the countries concerned in the transport operation provide otherwise.
11	Maximum (allowable) working pressure (gauge), in bar or MPa.
12	The name(s) in full of the gas(es), and, in addition for gases classified under an n.o.s. entry, the technical name of the gases, for whose transport the tank is approved, if applicable.
13	Minimum filling temperature for each gas.
14	Maximum allowable (net) mass of each gas according to mark 12, in kg.
15	Month and year of the initial inspection and of each subsequent periodic inspection.
16	Stamp of the inspector (notified or approved body) who carried out the inspection(s) of mark 14 accompanied by the identification number of the notified or approved body.
17	Name of owner or operator.
18	Maximum permissible mass (gross mass of the road tanker (tank vehicle), demountable tank, tank container and tank swap body).
19	Unladen mass (tare mass of the road tanker (tank vehicle), demountable tank, tank container and tank swap body).

A.2 Example 2 :Tank plate for the inner vessel of fixed tanks of road tankers (tank vehicles), demountable tanks, tank containers and tank swap bodies

Table A.2 — Explanation of the marking of the tank plate for the inner vessel

No.	Content/explanation
1	Manufacturer's name or mark of the inner vessel.
2	Kind of allocating to the approval number of Table 2.1.
3	Serial or production number issued by the manufacturer.
4	Year of manufacture.
5	To be dropped.
6	Test pressure (gauge) of the inner vessel, in MPa or bar.
7	Water capacity, in litres.
8	Design temperature, in °C.
9	Materials of the shell (and of the ends if different) of the inner vessel and reference to materials standards, if available.
10	To be dropped.
11	Maximum (allowable) working pressure (gauge), in bar or MPa.
12	To be dropped.
13	
14	
15	Month and year of the initial inspection.
16	Stamp of the inspector who carried out the inspection of mark 14.
17	To be dropped.
18	
19	

Annex B
(normative)

Elastic stress analysis

B.1 General

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

B.4 and B.5 give alternative criteria for demonstrating the acceptability of design on the basis of elastic analysis. The criteria in B.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 principal stresses f_1, f_2 and f_3 in each category, using the maximum shear stress theory of failure (see B.2.1).

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

Peak stresses need not be considered as they are only relevant when evaluating designs for cyclic service. Large vacuum-insulated transportable cryogenic vessels within the scope of this part of ISO 20421 are not considered to be in cyclic service.

Figure B.1 and Table B.1 have been included as guidance, where B.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. B.4.5 explains the reason for separating them into two categories, “general” and “secondary” in the case of thermal stresses.

B.2 Terminology

B.2.1 Stress intensity

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

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

B.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.

B.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.

B.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.

B.2.5 Shear stress

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

B.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.

B.2.7 Primary stress

A primary stress is a stress produced by mechanical loadings only and so distributed in the structure 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 B.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.

B.2.8 Primary local membrane stress

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

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

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

B.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.

B.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 localized 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.

B.3 Limit for longitudinal compressive general membrane stress

The longitudinal compressive stress shall not exceed 0,93ΔK for ferritic steels and 0,73 ΔK for austenitic stainless steel and aluminium alloys, where Δ is obtained from Figure B.2 or B.3 in terms of P_e / P_{yss} and where:

$$P_e = \frac{1{,}21 Es^2}{R^2};$$

and

$$P_{yss} = \frac{1{,}86 Ks}{R} \quad \text{for ferritic steel;}$$

and

$$P_{yss} = \frac{1{,}46 Ks}{R} \quad \text{for austenitic stainless steel and aluminium alloys.}$$

B.4 Stress categories and stress limits for general application

B.4.1 General

A calculated stress depending upon the type of loading and/or the distribution of such stress will fall within one of the five basic stress categories defined in B.4.2 to B.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.

B.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 B.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 $2K/3$.

B.4.3 Local primary membrane stress category

The stresses falling within the local primary membrane stress category are those defined in B.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.

B.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 B.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.

B.4.5 Primary plus secondary stress category

The stresses falling within the primary plus secondary stress category are those defined in B.2.7 plus those of B.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 $2K$.

B.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 recognized, 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 $2K$, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. This type is therefore classified as secondary stress in Table B.1 and Figure B.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 A small cold spot in a vessel wall.

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

B.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 B.5.2 to B.5.4.

The minimum separation 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 B.2.8 are not applicable to this section.

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

B.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 should not exceed $0{,}6K$ or $0{,}3R_{m.}$

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

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

B.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 0,8K.

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

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

B.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 B.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 the point to 0,9K.

Where shear stress is present alone, it shall not exceed K/3. The maximum permissible bearing stresses should not exceed K.

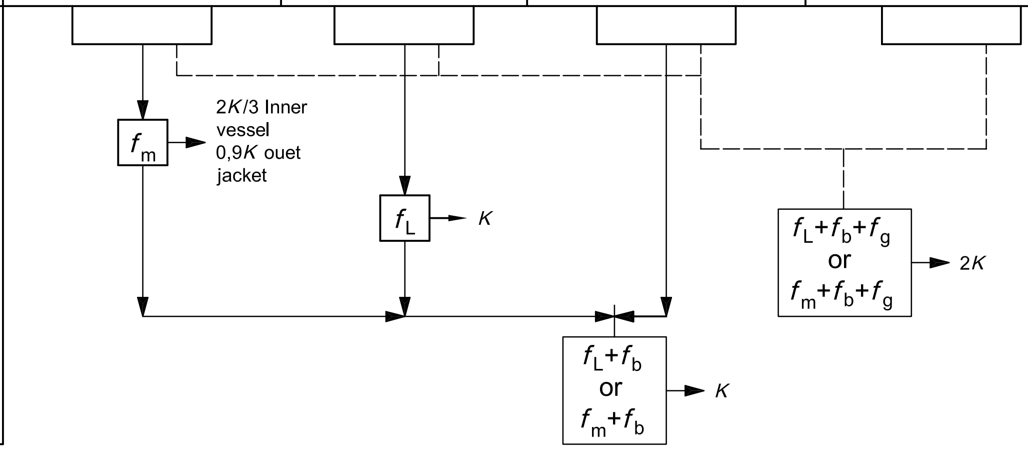
Table B.1 — Classification of stresses for some typical cases

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	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

Table B.1 (*continued*)

Vessel component	Location	Origin of stress	Type of stress	Classification
Dished end or conical end	Crown	Internal pressure	Membrane	f_m
			Bending	f_b
	Knuckle or junction to shell	Internal pressure	Membrane	f_L
			Bending	f_g
Flat end	Centre region	Internal pressure	Membrane	f_m
			Bending	f_b
	Junction to shell	Internal pressure	Membrane	f_L
			Bending	f_g
Perforated end or shell	Typical ligament in a uniform pattern	Pressure	Membrane (average through cross section)	f_m
			Bending (average through width of ligament, but gradient through plate)	f_b
	Isolated or atypical ligament	Pressure	Membrane	f_g
			Bending	f_g
Nozzle	Cross section perpendicular to nozzle axis	Internal pressure or external load or moment	General membrane (average across full section), stress component perpendicular to section)	f_m
		External load or moment	Bending across nozzle section	f_m
	Nozzle wall	Internal pressure	General membrane	f_m
			Local membrane	f_L
			Bending	f_g
		Differential expansion	Membrane	f_g
			Bending	f_g
NOTE 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 B.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 permissible limits of stress intensities				



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

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

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

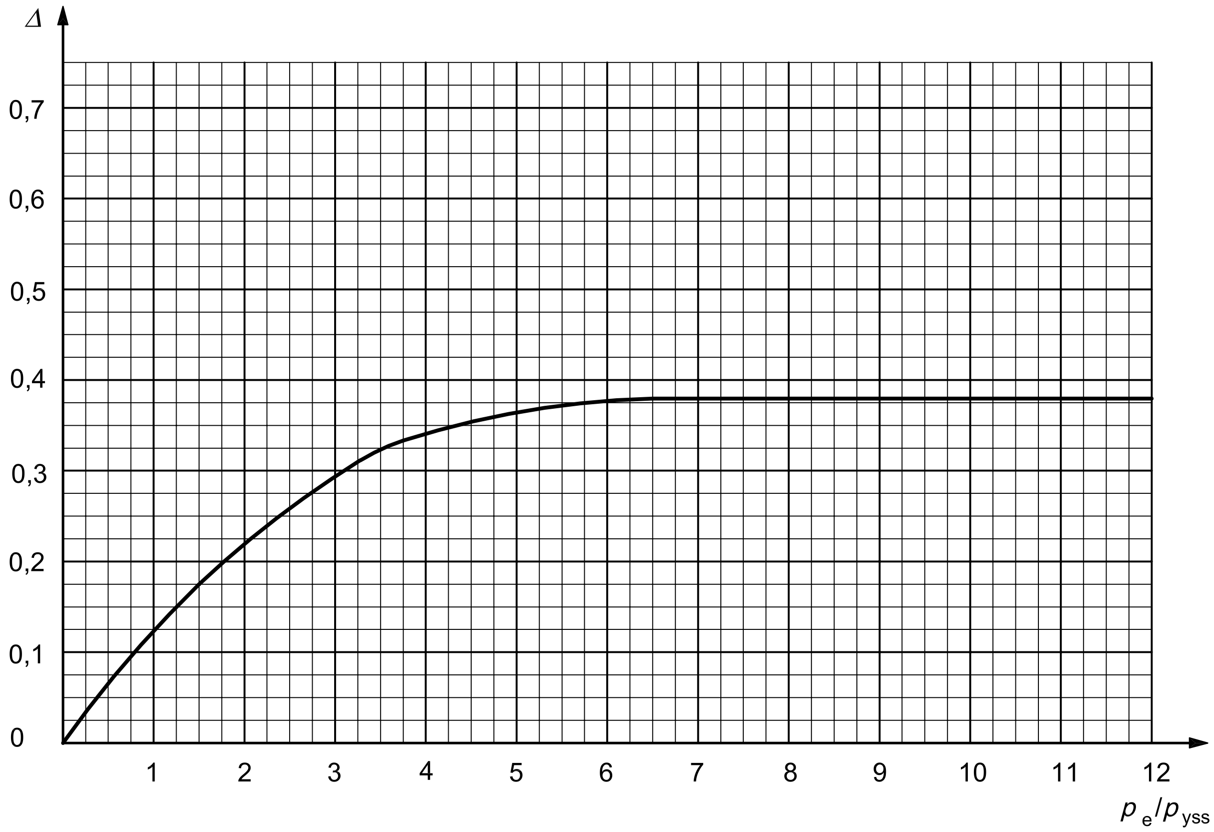


Figure B.2 — For vessels subject to external pressure

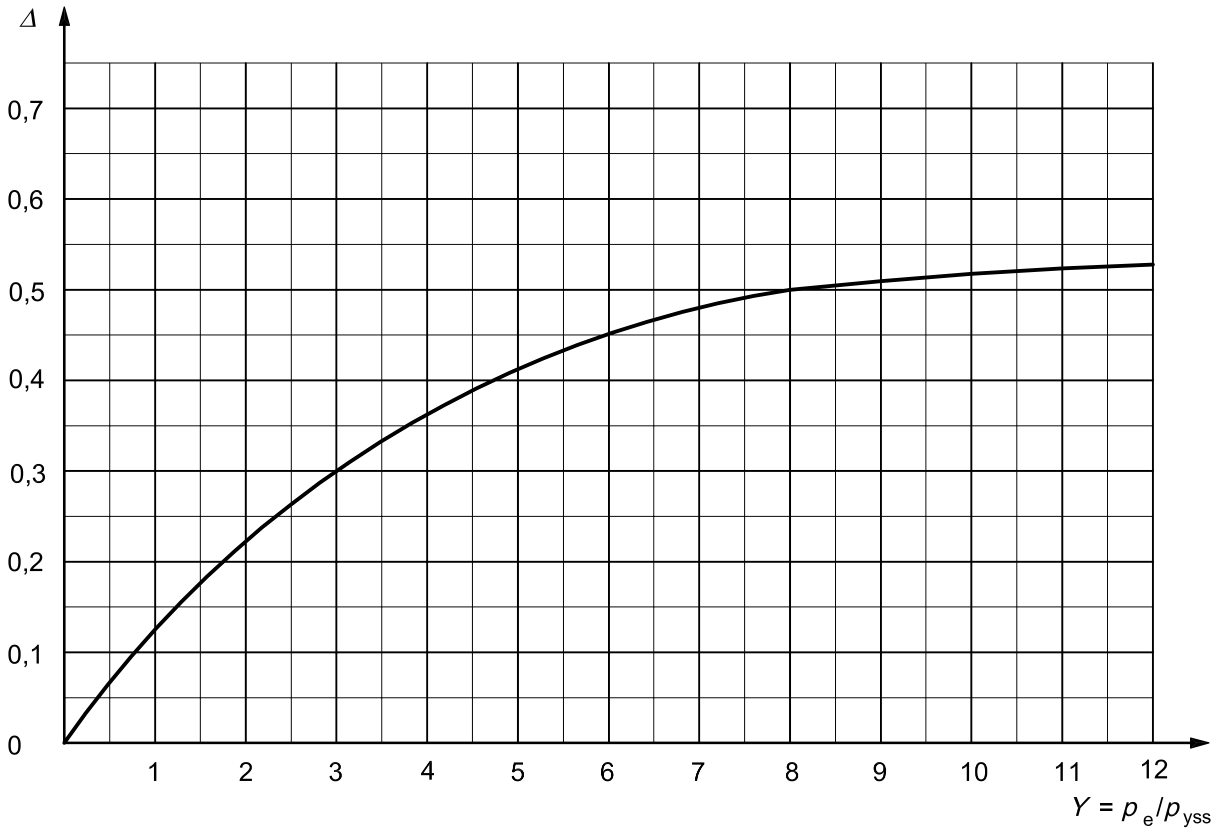


Figure B.3 — For vessels not subject to external pressure

Annex C
(normative)

Additional requirements for 9 % Ni steel

C.1 Introduction

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

The value of K to be adopted in the calculation formula of Table 8 is that of the parent 9 % Ni steel 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.

C.2 Specific requirements

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

C.2.2 The maximum design temperature shall not exceed 50 °C, when defrosting or drying the vessel at low pressure.

C.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.

C.2.4 The full length of all branch attachment welds shall be examined by dye penetrant before the first proof pressure test.

C.2.5 Imperfections that are unacceptable according to this part of ISO 20421 shall be repaired and re-examined to demonstrate compliance.

C.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.

C.2.7 The weld procedure qualification and production control transverse tensile test specimens shall:

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

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

NOTE These items also apply to work-hardened austenitic stainless steel.

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

C.2.9 The heat-affected zone at the weld fusion boundary shall be demonstrated to attain an ISO V-notch impact strength of 50 joules or 0,38 mm for lateral expansion at −196 °C, as an average of three test pieces, during weld procedure qualification and production control plate testing. These tests shall be performed in accordance with ISO 21028-1 and ISO 21028-2.

C.2.10 Openings shall not be located with their centre lines closer to principal seams than twice their diameter.

NOTE These items also apply to work-hardened austenitic stainless steel.

C.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.

C.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.

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

C.2.14 9 % Ni material conforming to EN 10028-4 is suitable for the construction of cryogenic vessels conforming to this part of ISO 20421. Other materials may be suitable provided sufficient test data is made available to demonstrate the suitability of the material.

Annex D
(informative)

Pressure strengthening of vessels from austenitic stainless steels

D.1 General

Austenitic stainless steel exhibits stress/strain characteristics (Figure D.2), different from that of carbon steel (Figure D.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 be plastic, and it will then 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 stress to be equal to or less than three-quarters 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 in D.4 and the relevant annexes of this part of ISO 20421.

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

D.2 Application of this annex

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

D.3 Materials

D.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 D.1. Requirements regarding these materials are found in EN 10028-7.

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

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

Steel designation		Solution heat-treated material		Pressure-strengthened vessel
Name	Number	$R_{p0,2}$ N/mm² min.	$R_{p1,0}$ 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	—	—	410
SA/A-240 304L	S 30403	—	—	385
SA/A-240 304N	S 30451	—	—	470
SA/A-240 316	S 31600	—	—	410
SA/A-240 316L	S 31603	—	—	385
SA/A-240 316N	S 31651	—	—	470
SA/A-240 316LN	S 31653	—	—	410

D.4.2 In case stable or metastable austenitic steels (when the strengthening strain exceeds 5 %) according to Clause 9, other than those listed in Table D.1, are to be qualified for pressure strengthening, or the vessel operating temperature will be below −196 °C, steel quality and welding procedure shall be validated by the type approval test detailed below. This test shall be carried out in addition to the tests required by 9.1 and 11.6.1.

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

— base material: two tensile tests along the direction of the applied stress and one set of impact tests across the direction of the applied stress;

— weld: two tensile tests across the weld and one set of impact tests of the weld metal according to ISO 21028-1 and ISO 21028-2.

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 0,38 mm lateral expansion.

The base material and the weld shall comply with:

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

D.4 Design

D.4.1 General

D.4.1.1 Wall thicknesses calculated according to D.4.3 refers to thicknesses before strengthening.

D.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.

D.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.

D.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).

D.4.1.5 Pressure strengthening applies to vessels (or part 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 part of ISO 20421. This requirement shall not preclude utilization of the strengthening process, provided the manufacturer can show that it does not cause deformations that impair the integrity of the vessel.

D.4.2 Design for internal pressure

D.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 $\sigma_{k\max}$ according to Table D.1. This highest allowable design stress value is the same whether the material used is in the solution heat-treated or work-hardened condition.

D.4.2.2 Calculation of the strengthening pressure

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

$$p_k \geqslant 1{,}33p \qquad \text{(D.1)}$$

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.

D.4.2.3 Calculation of wall thicknesses

D.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 part of ISO 20421 with the modifications shown in Table D.2.

Table D.2 — Modification of formula for the design of pressure-strengthened vessels

Subclause of this part of ISO 20421		Modification, see subclause in this annex
10.3.6.1	Cylinders and spheres subject to internal pressure	D.4.2.3.3
10.3.6.4	Dished ends subject to internal or external pressure, 10.3.6.4.3 — Internal pressure calculation (pressure on the concave surface)	D.4.2.3.4
10.3.6.5	Cones subject to internal or external pressure 10.3.6.5.6 — Internal-pressure calculation (pressure on the concave surface) $\|\varphi\| \leqslant 70°$, 10.3.6.5.7 — Internal-pressure calculation (pressure on the concave surface) $\|\varphi\| > 70°$	D.4.2.3.4 D.4.2.3.2
10.3.6.6	Flat ends	D.4.2.3.2
10.3.6.7	Openings in cylinders, spheres and cones	D.4.2.3.2 D.4.2.3.5

D.4.2.3.2 Parts where bending stresses are dominant and large deformations cannot be accepted, like flat cones according to 10.3.6.5.7 and flat ends according to 10.3.6.6, shall be calculated in the normal way using the design pressure p and design stress values according to 10.3.2.3. That is, the effect of the strengthening may not be used 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 10.3.2.3.

D.4.2.3.3 When designing parts according to 10.3.6.1, insert into the applicable formula the following:

— design stress value σ_k;

— weld joint factor 1,0.

D.4.2.3.4 Parts shall be designed with the same modifications as in D.4.3.3.2. 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. However, 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 D.5.1).

D.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 include openings having reinforcement not complying with 10.3.6.7.5.

For openings, where the hole diameter exceeds that given below, calculation of the reinforcement shall be made according to 10.3.6.7 of this part of ISO 20421 with the same modifications as in D.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 10.3.6.7 of this part of ISO 20421.

4.2.3.6 Largest allowed opening of unreinforced single holes

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

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

D_y = outside diameter of shell, in mm;

R = inside crown radius of end, in mm;

s_0 = wall thickness of unpierced shell, in mm;

s = true wall thickness of shell, in mm;

μ = s_0/s;

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

$$d_{max} = 0{,}4\sqrt{D_y s + C} \tag{D.2}$$

The value of d_{max} calculated according to formula (D.2) may be rounded up to the nearest higher even 10 mm. d_{max}, however, shall meet the conditions:

$$d_{max} \leqslant 150 \text{ mm} \tag{D.3}$$

$$d_{max} \leqslant 0{,}2D_y \tag{D.4}$$

The wall thickness of an unpierced cylinder is calculated from:

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

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

$$s_0 = \frac{pR}{20\dfrac{\sigma_k}{1{,}5}} \tag{D.6}$$

D.4.3 Design for external pressure

D.4.3.1 If a pressure strengthened vessel normally operating under internal pressure could be subject to external pressure, the vessel shall also be designed to withstand external pressure according to the applicable subclauses of Clause 3.

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 used only if measurements after strengthening show that the vessel is not significantly out of round.

D.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.

D.5 Manufacturing and inspection

D.5.1 Strengthening procedure

D.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 D.5.1.2 to D.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.

D.5.1.2 The vessel is filled with liquid. Before the vessel is closed, there should be a wait of at least 15 min to let any air dissolved in the liquid escape. The vessel is then topped up and sealed.

D.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.

D.5.1.4 The strengthening shall be 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 D.5.1.5). The strain rate shall be checked by repeated measurements of the circumference according to D.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 30 min to 1 h of the operation.

D.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.

D.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 needs to be repaired, this repair and pressure testing or possibly renewed strengthening shall be carried out in accordance with D.5.3.4.

D.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;

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

— any requirement for renewed strengthening (according to D.5.1.6 and D.5.3.4).

D.5.3 Welding

D.5.3.1 The strengthening method presumes high-quality welding. The same rules apply as for conventionally produced cryogenic vessels, except that production control test plates need not be taken.

D.5.3.2 Non-destructive testing shall be carried out before the strengthening to the extent stipulated in 12.3 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.

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

D.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 D.5.3.2 and D.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.

D.5.4 Pressure-vessel drawing

D.5.4.1 In addition to the information required by 10.2.2, the drawing shall bear the following text:

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

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

D.5.5 Data plate

The data plate shall, in addition to the information in Clause 14, bear the text “PRESSURE STRENGTHENED”.

D.6 Comments

D.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 Figure D.1 and D.2.

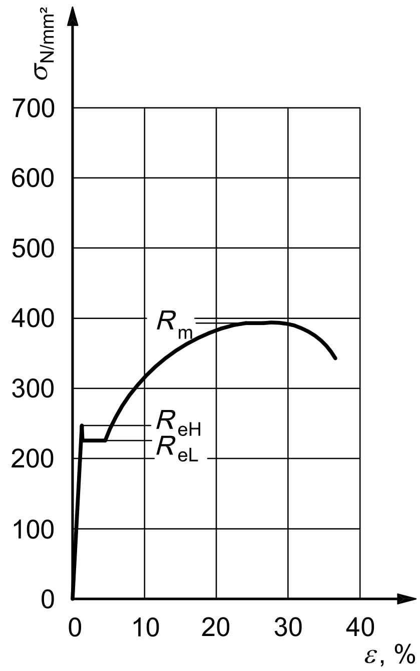


Figure D.1 — Stress/strain curve for carbon steel

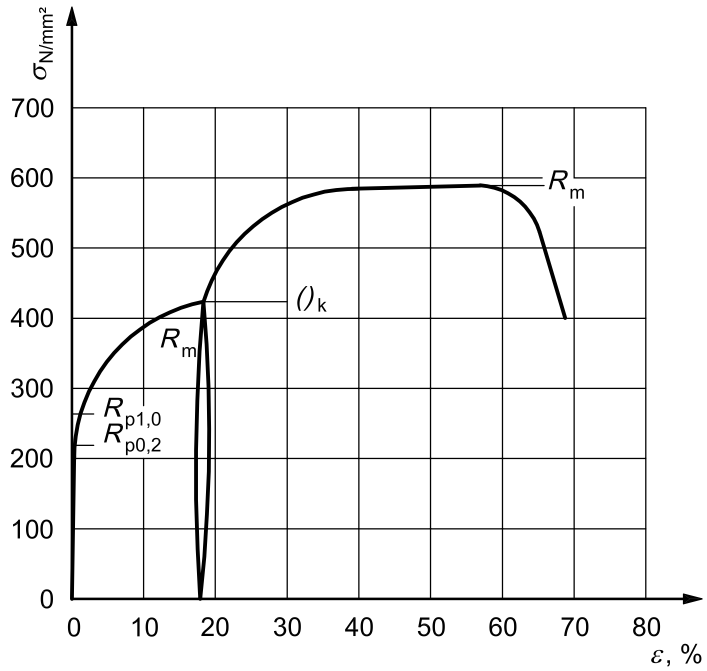


Figure D.2 — Stress/strain curve for austenitic stainless steel

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:

$$R_{p0,2} = \sigma_k \qquad \text{(D.7)}$$

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 have 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 –1,33 times the design stress, p, and 1,73 times the stress at maximum allowable pressure, p_s.

D.6.2 Work-hardened material

D.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.

D.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.

D.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.

D.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.

D.6.3 Derivation of formula

D.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 comply with the formula for cylinders (6) in 10.3.6.1.3:

$$s = \frac{pDs_F}{20\sigma_k z} \qquad \text{(D.8)}$$

NOTE To simplify the equation the middle diameter is used and the possible (corrosion) allowance is discarded.

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:

$$\sigma = \frac{pD}{20s} \qquad \text{(D.9)}$$

and the strengthening pressure p_k will therefore be:

$$p_k = \frac{20s\sigma_k}{D} \qquad \text{(D.10)}$$

If s according to formula (D.2) is substituted:

$$p_k = p\frac{s_F}{z} \qquad \text{(D.11)}$$

Since S_F = 1,33 and υ = 1,0, this corresponds to formula (D.1). Obviously, cylinders can be calculated from the formula in D.6.1 σ_k is inserted as the design stress value and 1,0 as the weld joint factor.

NOTE If a weld joint factor less than 1,0 is applied to any single main seam, an increase in strengthening pressure is required according to formula (D.5). To sustain this higher pressure, the thickness of all parts of the vessel would then need to be increased.

D.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 formula (D.8) (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".

D.6.3.3 The derivation of the formula in D.6.3.1 applies to parts free from bending stresses, i.e. cylinders, spheres and hemispherical ends.

Use 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 D.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 must be investigated and should normally not exceed $2\sigma_k$.

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

D.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.

D.6.3.5 This annex does not preclude the use of the strengthening effect, provided that the manufacturer can show it does not cause harmful deformation or other problems.

D.6.4 Deformations at strengthening

D.6.4.1 The highest allowable design stress value $\sigma_{k\,\mathrm{max}}$ for the different steels has consistently been set 200 N/mm² higher than $R_{\mathrm{p0,2}}$ for the solution heat-treated material.

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

D.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, $R_{\mathrm{p0,2}}$, of the material (calculated on basis of the cross-sectional area before the strengthening) is about 30 N/mm² higher than the strengthening stress σ_k used.

D.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:

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

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

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

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

	True stress				True elongation			
	σ_1	σ_2	σ_3	σ	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 D.3 expresses the fact that a tensile test sample contracts in two dimensions, while a cylinder decreases only in thickness by an amount corresponding to the increased circumference.

Table D.3 shows that a certain effective stress, σ, produces different elongations 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 σ are calculated on 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, $\sigma_{1\text{nom}}$, 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.

D.6.4.4 In practice, the maximum circumferential strain of cylinders is usually 3 % to 5 % when using a 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 $R_{\text{p0,2}}$ is higher than the specified minimum for the material;

— the plate thickness is greater than nominal;

— there are reinforcing effects of ends, nozzles, and D.

D.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 may require renewed strengthening (see D.5.3.4). This second strengthening generally leads to much smaller deformations.

D.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 E
(informative)

Specific weld details

E.1 Field of application

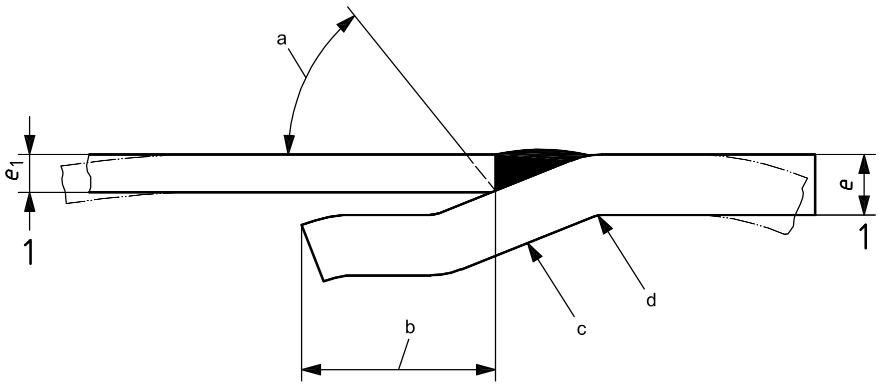
Specific weld details given in E.2 are currently in common usage in cryogenic vessels and are appropriate to this service. Although the scope of EN 1708-1 does not specifically consider the application of weld details to cryogenic vessels, the manufacturer may consult it for guidance.

E.2 Weld detail

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

E.2.1 Joggle joint

See Figure E.1.



Key

[a] bevel optional

[b] as desired

[c] depth of offset = e_1

[d] avoid sharp break

Figure E.1 — Joggle joint

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:

   1) 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

   2) on completion of joggling, the area of the weld is subjected to dye-penetrant examination and is proven to be free of cracks;

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

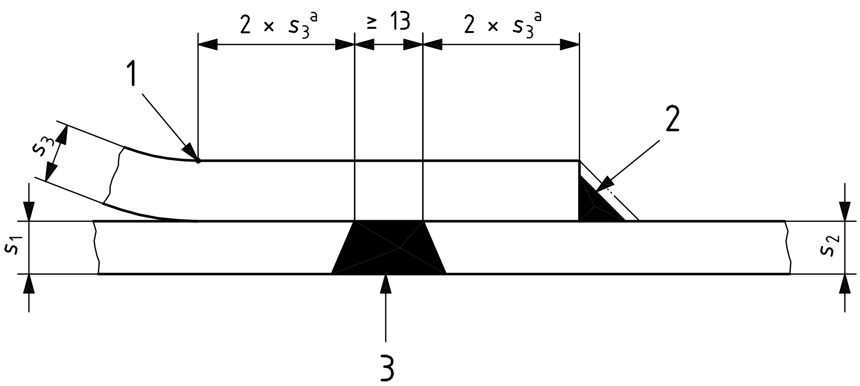
d) the profile of the offset is a smooth radius without sharp corners;

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

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

E.2.2 Intermediate ends

See Figure E.2 and 10.3.6.4.3.



Key

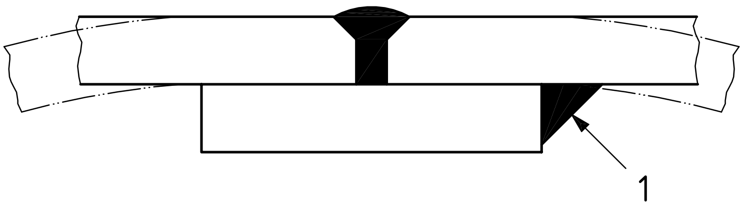
1 tangent point
2 continuous fillet weld
3 butt weld
s_1 cylinder thickness
s_2 cylinder thickness
s_3 end thickness
[a] Not to exceed 25 mm.

NOTE Cylinder thickness s_1 and s_2 may vary.

Figure E.2 — Intermediate end

E.2.3 Backing strip

See Figure E.3.



Key

1 intermittent or continuous fillet weld

Figure E.3 — Backing strip

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

E.2.4 End plate closure

See Figure E.4 for two examples of the many ways of welding flat plates. See also Figure 6.

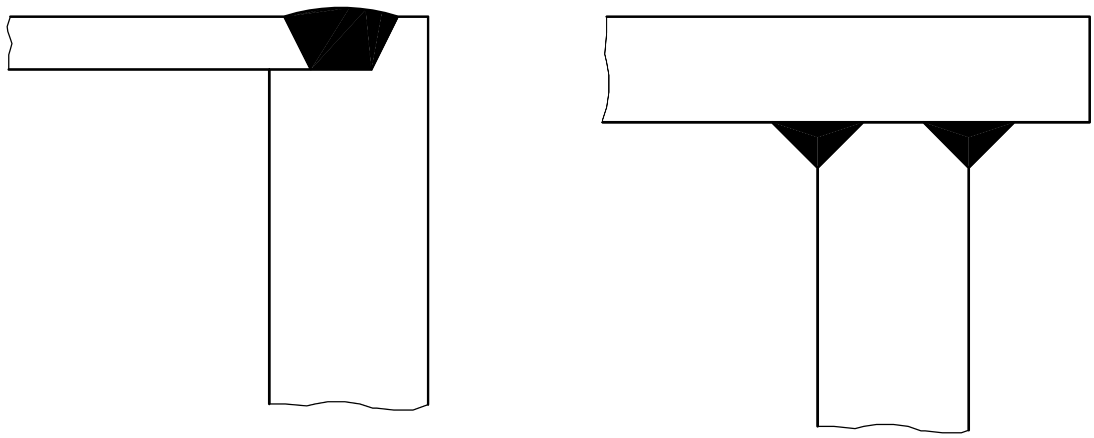


Figure E.4 — End plate closure (examples)

E.2.5 Non-full penetration nozzle weld

See Figure E.5.

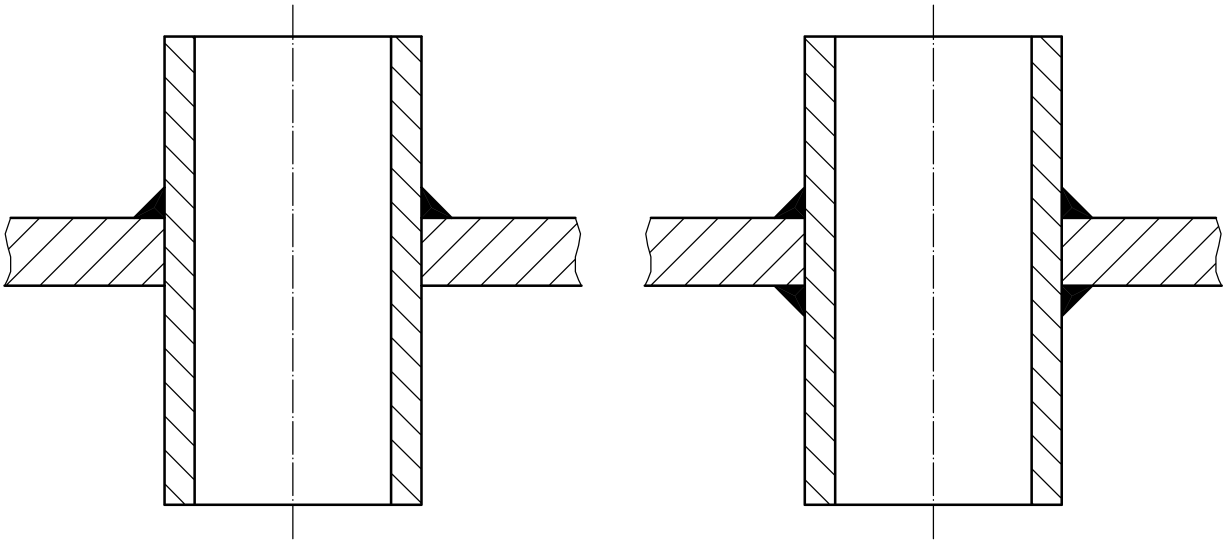


Figure E.5 — Non-full penetration nozzle welds

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 contain the design nozzle loadings.

E.2.6 Non-continuous fillet weld on attachments

Non-continuous fillet welds on attachments may be used for all attachments to main pressure components provided that the following criteria are met:

— strength is adequate for design loadings;

— crevices between attached component and main pressure envelope can be demonstrated not to conflict with E.3.

E.3 Oxygen service requirements

The need for cleanliness of equipment in liquid oxygen and other oxidizing liquid service is described in ISO 21010 and EN 12300:1998.

The internal weld details shall be such that debris, contaminants, hydrocarbons or degreasants can not accumulate so as to cause a fire risk in future operation.

Annex F
(informative)

Outer-jacket relief devices

F.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.

F.2 Requirements

F.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.

F.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 10.2.3.3. 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.

F.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.

F.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.

F.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.

F.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 part of ISO 20421.

Annex G
(informative)

Base materials

Table G.1 — Pressure vessels

Specification No.	Material grade	Material number
EN 10028-4	X8Ni9	1.5662 (HT 640 & HT 680)
EN 10222-3	X8Ni9	1.5662
DIN 17280	X8Ni9	1.5662
DIN 17458	X2CrNi19-11	1.4306
DIN 17457 – DIN 17458	X5CrNi 18-10	1.4301
DIN 17457 – DIN 17458	X2CrNiN 18-10	1.4311
DIN 17457 – DIN 17458	X6CrNiTi 18-10	1.4541
DIN 17457 – DIN 17458	X6CrNiNb 18-10	1.4550
DIN 17457 – DIN 17458	X5CrNiMo 17-12-2	1.4401
DIN 17457 – DIN 17458	X2CrNiMo 17-13-2	1.4404
DIN 17457 – DIN 17458	X6CrNiMoTi 17-12-2	1.4571
DIN 17457 – DIN 17458	X2CrNiMoN 17-13-3	1.4429
DIN 17457 – DIN 17458	X2CrNiMo 18-14-3	1.4435
DIN 17457 – DIN 17458	X2CrNiMo 17-13-3	1.4436
DIN 17457 – DIN 17458	X2CrNiMoN 17-13-5	1.4439
EN 10028-3	P275NL1	1.0488
EN 10028-3	P275NL2	1.1104
EN 10028-3	P355NL1	1.0566
EN 10028-3	P355NL2	1.1106
EN 10028-3	P460NL1	1.8915
EN 10028-3	P460NL2	1.8916
EN 10028-5	P355ML1	1.8832
EN 10028-5	P355ML2	1.8833
EN 10028-5	P420ML1	1.8835
EN 10028-5	P420ML2	1.8828
EN 10028-5	P460ML1	1.8837
EN 10028-5	P460ML2	1.8831
EN 10028-7	X2CrNi18-9	1.4307
EN 10028-7	X2CrNi19-11	1.4306
EN 10028-7	X2CrNiN18-10	1.4311
EN 10028-7	X5CrNiN19-9	1.4315

Table G.1 (*continued*)

Specification No.	Material grade	Material number
EN 10028-7	X5CrNi18-10	1.4301
EN 10028-7	X2CrNiN23-4	1.4362
EN 10028-7	X3CrNiMo17-12-2	1.4401
EN 10028-7	X2CrNiMo17-12-2	1.4404
EN 10028-7	X2CrNiMoN17-11-2	1.4406
EN 10028-7	X2CrNiMoN17-13-3	1.4429
EN 10028-7	X2CrNiMoN22-5-3	1.4462
EN 10028-7	X2CrNiMoCuN25-6-3	1.4507
EN 10028-7	X2CrNiMoN25-7-4	1.4410
EN 10028-7	X2CrNiMoCuWN25-7-4	1.4501
EN 10222-5	X2CrNi18-9	1.4307
EN 10222-5	X5CrNi18-10	1.4301
EN 1022-5	X5CrNiMo17-12-2	1.4401
EN 10222-5	X2CrNiMo17-12-2	1.4404
EN 10088-3	X2CrNi19-11	1.4306
EN 10088-3	X2CrNiN18-10	1.4311
EN 10088-3	X5CrNi18-10	1.4301
EN 10088-3	X5CrNiMo17-12-2	1.4401
EN 10088-3	X2CrNiMo17-13-2	1.4404
EN 10088-3	X2CrNiMoN17-12-2	1.4406
EN 10088-3	X2CrNiMo17-13-3	1.4429
JIS G 3127	SL9N520	1.5662
JIS G 4303-4305	SUS304	1.4301
JIS G 4303-4305	SUS304L	1.4307
JIS G 4303-4305	SUS316	1.4401
JIS G 4303-4305	SUS316L	1.4404
JIS G 4317-4320	SUS321	1.4541
JIS G 4317-4320	SUS347	1.4550
JIS G 4317-4320	SUS316	1.4401
JIS G 4317-4320	SUS316L	1.4404
JIS G 4317-4320	SUS317L	1.4439
JIS G 4313-4315	SUS304	1.4301
JIS G 4313-4315	SUS304L	1.4307
JIS G 4313-4315	SUS304N1	1.4406
JIS G 4313-4315	SUS304LN	1.4311
JIS G 4317-4320	SUS316L	1.4406
JIS G 4315-4315	SUS316LN	1.4429
SA/A-240	304LN	S 30453

Table G.1 (*continued*)

Specification No.	Material grade	Material number
SA/A-240	304N	S 30451
SA/A-240	316LN	S 31653
SA/A-240	316N	S 31651
SA/A-240	201LN	S 20153
SA/A-240	201-1	S 20100
SA/A-666	201-1	S 20100
SA/A-240	201-2	S 20100
SA/A-666	201-2	S 20100
SA/A-240	201L	S 20103
SA/A-666	201L	S 20103
SA/A-479	316LN	S 31653
SA/A-479	316N	S 31651
SA/A-240	XM-29	S 24000
SA/A-479	XM-29	S 24000
SA/A-479	304	S 30400
SA/A-479	304L	S 30403
SA/A-240	304	S 30400
SA/A-240	304L	S 30403
SA/A-479	304LN	S 30453
SA/A-479	304N	S 30451
SA/A-240	XM-19	S 20910
SA/A-479	XM-19	S 20910
SA/A-479	—	S 21800
SA/A-353	—	K 81340
SA/A-553	I	K 81340
SA/A-522	I	K 81340
SA/A-553	2	K 71340
SA/A-351	—	—
SA/A-516	55	K 01800
SA/A-516	60	K 02100
SA/A-516	65	K 02403
SA/A-516	70	K 02700
SA/A-517	E	K 21604
SA/A-517	F	K 11576
SA/A-612	—	K 02900
A-276	201LN	S 20153
A-276	304	S 30400
A-276	304L	S 30403

Table G.1 (*continued*)

Specification No.	Material grade	Material number
A-276	304LN	S 30453
A-276	304N	S 30451
A-276	316	S 31600
A-276	316L	S 31603
A-276	316LN	S 31653
A-276	316N	S 31651
A-276	201	S 20100
A-276	XM-29	S 24000
A-276	XM-19	S 20910
A-276	—	S 21800
SB/B-209	5083	A 95083
SB/B-221	5083	A 95083
SB/B-209	6061	A 96061
SB/B-221	6061	A 96061
SB/B-211	6061	A 96061
SB/B-308	6061	A 96061
SB/B-209	3003	A 93003
SB/B-209	5052	A 95052
SB/B-211	3003	A 93003
SB/B-221	3003	A 93003
B-221	6063	A 96063
B-133	—	—
B-16	—	—

NOTE SA/SB prefix to specification number refers to ASME specifications. A/B prefix to specification number refers to ASTM specifications. When materials to ASME specifications are unavailable, materials to the same specification number with a prefix of A/B (ASTM) may be used. All ASME/ASTM specification numbers listed are equally acceptable with a suffix M.

Table G.2 — Piping and pipe fittings

Specification No.	Material grade	Material number
SA/A-312	TP 316L	S 31603
SA/A-358	TP 316L	S 31603
SA/A-249	TP 316L	S 31603
SA/A-409	TP 316L	S 31603
SA/A-688	TP 316L	S 31603
SA/A-813	TP 316L	S 31603
SA/A-814	TP 316L	S 31603
SA/A-249	TP 316	S 31600
SA/A-312	TP 316	S 31600
SA/A-358	TP 316	S 31600
SA/A-409	TP 316	S 31600
SA/A-688	TP 316	S 31600
SA/A-813	TP 316	S 31600
SA/A-814	TP 316	S 31600
SA/A-249	TP 316LN	S 31653
SA/A-312	TP 316LN	S 31653
SA/A-358	TP 316LN	S 31653
SA/A-688	TP 316LN	S 31653
SA/A-249	TP 316N	S 31651
SA/A-312	TP 316N	S 31651
SA/A-358	TP 316N	S 31651
SA/A-688	TP 316N	S 31651
SA/A-813	TP 316N	S 31651
SA/A-814	TP 316N	S 31651
SA/A-249	TPXM-29	S 24000
SA/A-312	TPXM-29	S 24000
SA/A-688	TPXM-29	S 24000
SA/A-249	TP 304L	S 30403
SA/A-312	TP 304L	S 30403
SA/A-358	TP 304L	S 30403
SA/A-409	TP 304L	S 30403
SA/A-688	TP 304L	S 30403
SA/A-813	TP 304L	S 30403
SA/A-814	TP 304L	S 30403
SA/A-249	TP 304	S 30400
SA/A-334	8	K 81340
SA/A-333	8	K 81340

Table G.2 (*continued*)

Specification No.	Material grade	Material number
SA/A-312	TP 304	S 30400
SA/A-358	TP 304	S 30400
SA/A-409	TP 304	S 30400
SA/A-688	TP 304	S 30400
SA/A-813	TP 304	S 30400
SA/A-814	TP 304	S 30400
SA/A-249	TP 304LN	S 30453
SA/A-312	TP 304LN	S 30453
SA/A-358	TP 304LN	S 30453
SA/A-688	TP 304LN	S 30453
SA/A-813	TP 304LN	S 30453
SA/A-814	TP 304LN	S 30453
SA/A-249	TP 304N	S 30451
SA/A-312	TP 304N	S 30451
SA/A-358	TP 304N	S 30451
SA/A-688	TP 304N	S 30451
SA/A-813	TP 304N	S 30451
SA/A-814	TP 304N	S 30451
SA/A-312	TP 321	S 32100
SA/A-249	TP 321	S 32100
SA/A-358	TP 321	S 32100
SA/A-409	TP 321	S 32100
SA/A-813	TP 321	S 32100
SA/A-814	TP 321	S 32100
SA/A-213	TP 316L	S 31603
SA/A-312	TP 316L	S 31603
SA/A-430	FP 316	S 31600
SA/A-213	TP 316	S 31600
SA/A-312	TP 316	S 31600
SA/A-376	TP 316	S 31600
SA/A-213	TP 316LN	S 31653
SA/A-312	TP 316LN	S 31653
SA/A-376	TP 316LN	S 31653
SA/A-430	FP 316N	S 31651
SA/A-213	TP 316N	S 31651
SA/A-312	TP 316N	S 31651
SA/A-376	TP 316N	S 31651
SA/A-182	F 316L	S 31603

Table G.2 (*continued*)

Specification No.	Material grade	Material number
SA/A-336	F 316L	S 31603
SA/A-403	316L	S 31603
SA/A-182	F 316	S 31600
SA/A-336	F 316L	S 31600
SA/A-403	316	S 31600
SA/A-182	F 316LN	S 31653
SA/A-336	F 316LN	S 31653
SA/A-403	316LN	S 31653
SA/A-182	F 316N	S 31651
SA/A-336	F 316N	S 31651
SA/A-403	316N	S 31651
SA/A-182	F 304L	S 30403
SA/A-336	F 304L	S 30403
SA/A-403	304L	S 30403
SA/A-182	F 304	S 30400
SA/A-336	F 304	S 30400
SA/A-403	304	S 30400
SA/A-182	F 304LN	S 30453
SA/A-336	F 304LN	S 30453
SA/A-403	304LN	S 30453
SA/A-182	F 304N	S 30451
SA/A-336	F 304N	S 30451
SA/A-403	304N	S 30451
SA/A-552	I	K 81340
SA/A-420	WPL8	K 81340
SA/A-213	TP 304L	S 30403
SA/A-213	TP 304	S 30400
SA/A-213	TP 304LN	S 30453
SA/A-213	TP 304N	S 30451
SA/A-430	FP 304	S 30400
SA/A-430	FP 304N	S 30451
SA/A-376	TP 304LN	S 30453
SA/A-376	TP 304	S 30400
SA/A-376	TP 304N	S 30451
SA/A-376	TP 316LN	S 31653
SA/A-376	TP 316	S 31600
SA/A-376	TP 316N	S 31651
SA/A-213	TP 321	S 32100

Table G.2 (*continued*)

Specification No.	Material grade	Material number
SA/A-333	1	K 03008
SA/A-333	2	—
SA/A-333	3	K 31918
SA/A-333	4	K 11267
SA/A-333	5	—
SA/A-333	6	K 03006
SA/A-333	7	K 21903
SA/A-333	8	K 81340
SA/A-333	9	K 22035
SA/A-333	10	—
SA/A-333	11	—
SA/A-105		K 03504
SA/A-350	LF-1	K 03504
SA/A-350	LF 2	K 03009
SA/A-350	LF 3	K 03011
SA/A-350	LF 5	K 13050
SA/A-350	LF 6	—
SA/A-350	LF 9	K 22036
SA/A-350	LF 787	—
SA/A-106	A	K 02501
SA/A-106	B	K 03006
SA/A-106	C	K 03501
SB/B-247	5083	A 95083
SB/B-247	6061	A 96061
SB/B-241	5083	A 95083
SB/B-241	6061	A 96061
SB/B-210	6061	A 96061
SB/B-241	3003	A93003
SB/B-75	—	—
A-511	TP 304	S 30400
A-511	TP 304L	S 30403
A-511	TP 316L	S 31603
A-511	TP 316	S 31600
A-511	TP 321	S 32100
A-351	304	S 30400
A-351	316	S 31600
A-351	304L	S 30403
A-351	316L	S 31603

Table G.2 (*continued*)

Specification No.	Material grade	Material number
A-269	TP 316L	S 31603
A-269	TP 316	S 31600
A-269	TP 316LN	S 31653
A-269	TP 316 N	S 31651
A-269	TP 304L	S 30403
A-269	TP 304	S 30400
A-269	TP 304LN	S 30453
A-269	TP 304N	S 30451
A-269	TP 321	S 32100
A-269	TPXM 29	S 24000
A-632	TP 304L	S 30403
A-632	TP 304	S 30400
A-632	TP 316L	S 31603
A-632	TP 316	S 31600
A-632	TP 321	S 32100
A-733	TP 304L	S 30403
A-733	TP 304	S 30400
A-733	TP 316L	S 31603
A-733	TP 316	S 31600
NF A 49-117	TUZ2CN18-10	—
NF A 49-147	TUZ2CN18-10	—

NOTE SA/SB prefix to specification number refers to ASME specifications. A/B prefix to specification number refers to ASTM specifications. When materials to ASME specifications are unavailable, materials to the same specification number with a prefix of A/B (ASTM) may be used. All ASME/ASTM specification numbers listed are equally acceptable with a suffix M.

Annex H
(normative)

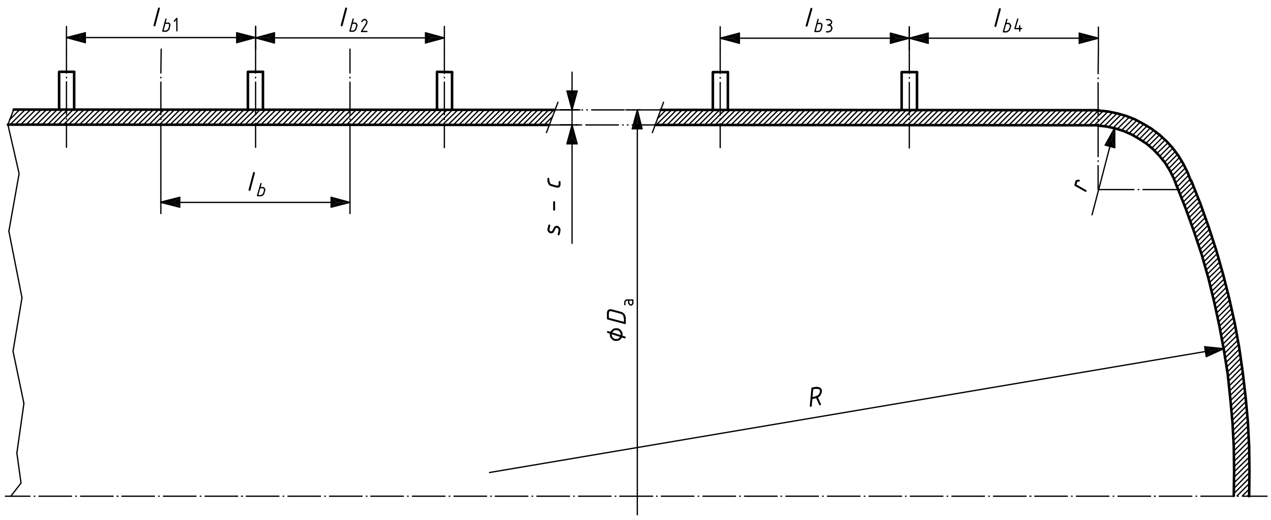
Components subject to external pressure (pressure on the convex surface) — Calculation

H.1 General

This annex gives two calculation methods which are equally recognized and which give comparable results.

Calculations are performed for elastic buckling and for plastic deformation. The lowest calculated pressure, p_e or p_p, shall not be less than the external design pressure.

NOTE The buckling length l_b is the maximum length of the shell supported by a reinforcement ring as defined in Figure H.1.



Key

l_b = maximum of l_b, l_{b2}, l_{b3}, b_{b4} for design of cylindrical shell

$l_b = \frac{l_{b1} + l_{b2}}{2}$ for design of reinforcing elements

Figure H.1 — Determination of buckling length

H.2 Method 1

H.2.1 Cylindrical shells

H.2.1.1 Elastic buckling

Calculations are performed using the following formula:

$$p_e = \frac{E}{S_k}\left\{\frac{20}{\left(n^2-1\right)\left[1+\left(n/Z\right)^2\right]^2}\,\frac{s-c}{D_a}\,\frac{80}{12\left(1-\nu^2\right)}\left[n^2-1+\frac{2n^2-1-\nu}{1+\left(n/Z\right)^2}\right]\left[\frac{s-c}{D_a}\right]^3\right\} \tag{H.1}$$

where $Z = 0{,}5\sqrt{D_a/l_b}$ and n is an integer equal to or greater than 2 and greater than Z, so determined that the value for P_e is a minimum. n denotes the number of lobes produced by the buckling process which may occur at the circumference in the event of failure. The number of lobes can be estimated using the following approximation equation:

$$n = 1{,}63 \times \sqrt[4]{\frac{D_a^3}{l_b^2\left(s-c\right)}} \tag{H.2}$$

For tubes and pipes, calculations may be performed using the following simplified formula:

$$p_e = \frac{E}{S_k} \times \frac{20}{1-\nu^2} \times \left(\frac{s-c}{D_a}\right)^3 \tag{H.3}$$

H.2.1.2 Plastic deformation

When $D_a/l_b \leqslant 5$, the following formula applies:

$$p_p = \frac{20K}{S_p} \times \frac{s-c}{D_a} \times \frac{1}{1+\dfrac{1{,}5\,u\left(1-0{,}2D_a/l_b\right)D_a}{100(s-c)}} \tag{H.4}$$

When $D_a/l_b > 5$, the following formula applies:

The higher pressure obtained using equations (H.5) and (H.6) shall not be less than the external design pressure.

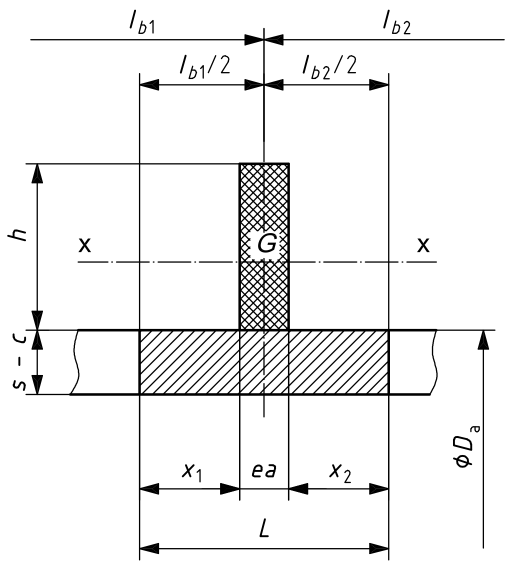
$$p_p = \frac{20K}{S_p}\frac{\left(s-c\right)}{D_a} \tag{H.5}$$

$$p_p = \frac{30K}{S_p} \times \left(\frac{s-c}{l_b}\right)^2 \tag{H.6}$$

H.2.1.3 Stiffening rings

In addition to the ends, effective reinforcing elements may be regarded as including the types of element illustrated in Figure H.2.

$h/ea \leqslant 16$



$$x_1 = 0{,}55\sqrt{D_a(s-c)}$$

$$x_1 \leqslant \frac{l_{b1} - ea}{2}$$

$$x_2 = 0{,}55\sqrt{D_a(s-c)} \leqslant \frac{l_{b2} - ea}{2}$$

a)

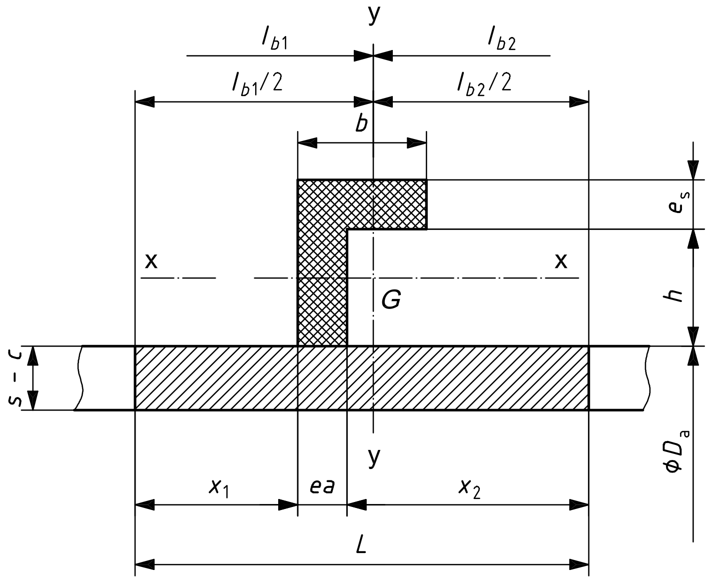
Figure H.2 — Determination of reinforcing elements

$$Min\ of$$

$$h/ea \leqslant 50$$

$$b \leqslant ea + 0{,}55\sqrt{D_a \cdot es}$$

$$b \leqslant ea + 16es$$



$$x_1 = 0{,}55\sqrt{D_a(s-c)} + ea \leqslant \frac{l_{b1}}{2}$$

$$x_2 = 0{,}5\sqrt{D_a(s-c)} \leqslant \frac{l_{b2}}{2}$$

NOTE In Figure H.2 b), *yy* may be taken through the centre line of the vertical leg of reinforcement for simplifying calculations; in this case the equations are similar.

b)

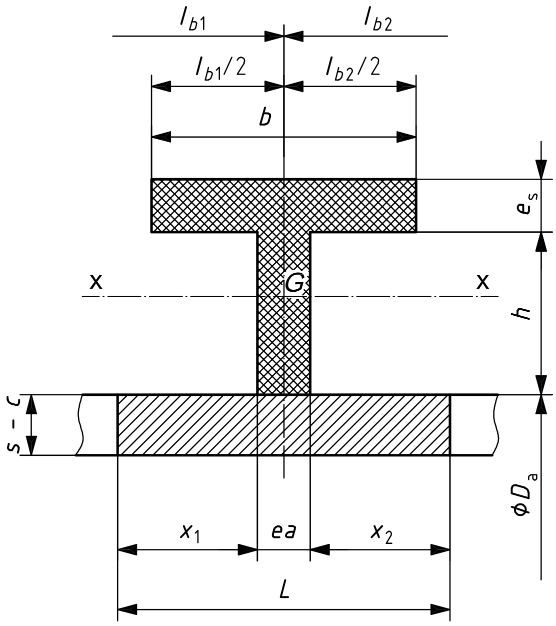
Figure H.2 (*continued*)

Min of

$$h/ea \leqslant 50$$

$$b \leqslant ea + 1{\cdot}1\sqrt{Da \cdot es}$$

$$b \leqslant ea + 32es$$



$$x_1 = 0{,}55\sqrt{D_a(s-c)}$$

$$x_1 \leqslant \frac{l_{b1} - ea}{2}$$

$$x_2 = 0{,}55\sqrt{D_a(s-c)} \leqslant \frac{l_{b2} - ea}{2}$$

c)

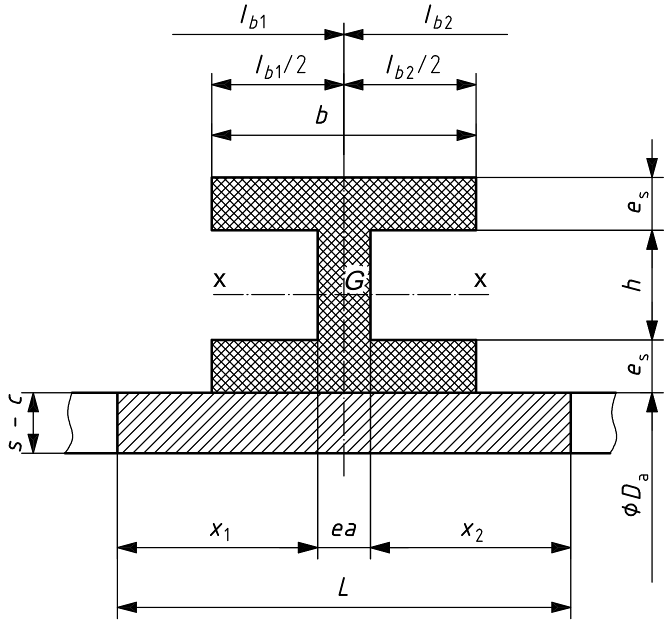
Figure H.2 (*continued*)

Min of

$$h/ea \leqslant 50$$

$$b \leqslant ea + 1 \cdot 1\sqrt{Da \cdot es}$$

$$b \leqslant ea + 32es$$



$$x_1 = 0{,}55\sqrt{D_a(s-c)}$$

$$x_1 \leqslant \frac{l_{b1} - ea}{2}$$

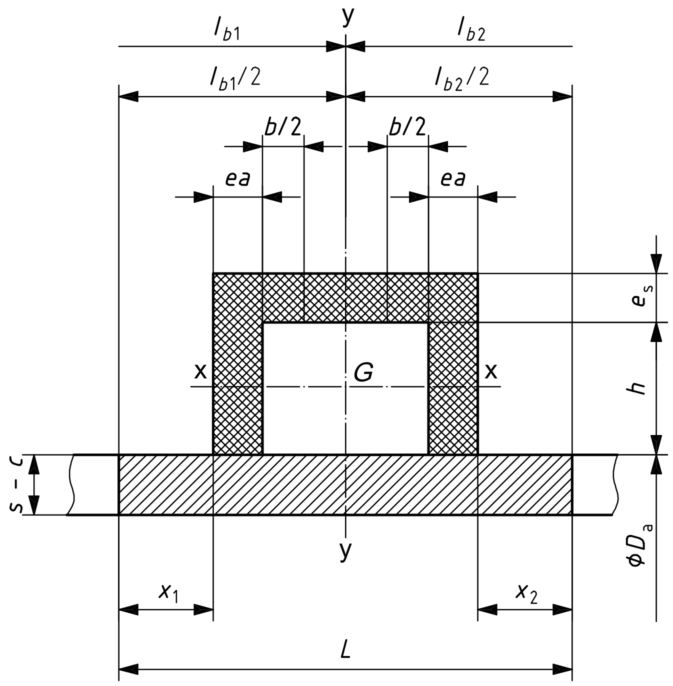
$$x_2 = 0{,}55\sqrt{D_a(s-c)} \leqslant \frac{l_{b2} - ea}{2}$$

d)

Figure H.2 (*continued*)

$$\frac{h}{ea} \leqslant 50 \quad b \leqslant 1 \cdot 1\sqrt{D_a \cdot es}$$

$$b \leqslant 1 \cdot 1\sqrt{D_a(s-c)}$$



e)

$$x_1 = \min\left\{0{,}55\sqrt{D_a(s-c)};\frac{l_{b1}}{2}\right\} \qquad x_2 = \min\left\{0{,}55\sqrt{D_a(s-c)};\frac{l_{b2}}{2}\right\}$$

L is the portion of the shell which acts as part of the reinforcing element and contribute to its effective moment of inertia.

$$L \leqslant 2{,}2 \times \sqrt{D_a(s-c)} + 2ea \qquad L \leqslant 1{,}1\sqrt{D_a - 2y} + 2ea + W_r$$

Figure H.2 (*continued*)

The reinforcing elements, including the stiffening rings welded to the shell and the portion L of the shell (see Figure H.2), shall satisfy the following conditions:

g) for the inner vessel:

$$I \geqslant 0{,}124 \times \frac{p \times D_a{}^3 \times \sqrt{D_a \times s}}{10 \times E} \tag{H.7}$$

h) for the outer jacket:

$$I \geqslant 0{,}042 S_k \frac{pD_a{}^3 l'_b}{10E} \tag{H.8}$$

For demonstrated satisfactory experience, a factor of safety S_k equal to or greater than 1,3 is acceptable.

$$A \geqslant 0{,}5 S_p \frac{pD_a l'_b}{10K} \tag{H.9}$$

The portion of the shell supported by the reinforcing element has a length:

$$l'_b = \sqrt{D_a s} \tag{H.10}$$

The moment of inertia, I, is relative to the neutral axis of the reinforcing element cross-section parallel to the shell axis (see axis xx in Figure H.2).

The flat-bar stiffness and the Γ, T, H or U profile stiffness shall satisfy the conditions given in Figure H.2. Stiffening rings (full or partial) to provide structural integrity shall be securely attached to the outer jacket.

Where stiffening rings are joined to the shell by means of intermittent welds, the fillet welds at each side shall cover at least one-third of the shell circumference, be uniformly distributed (see Figure H.3) and the number of weld discontinuities shall be at least $2n$. The number of buckling lobes, n, is obtained as indicated in H.3.1.2.5.

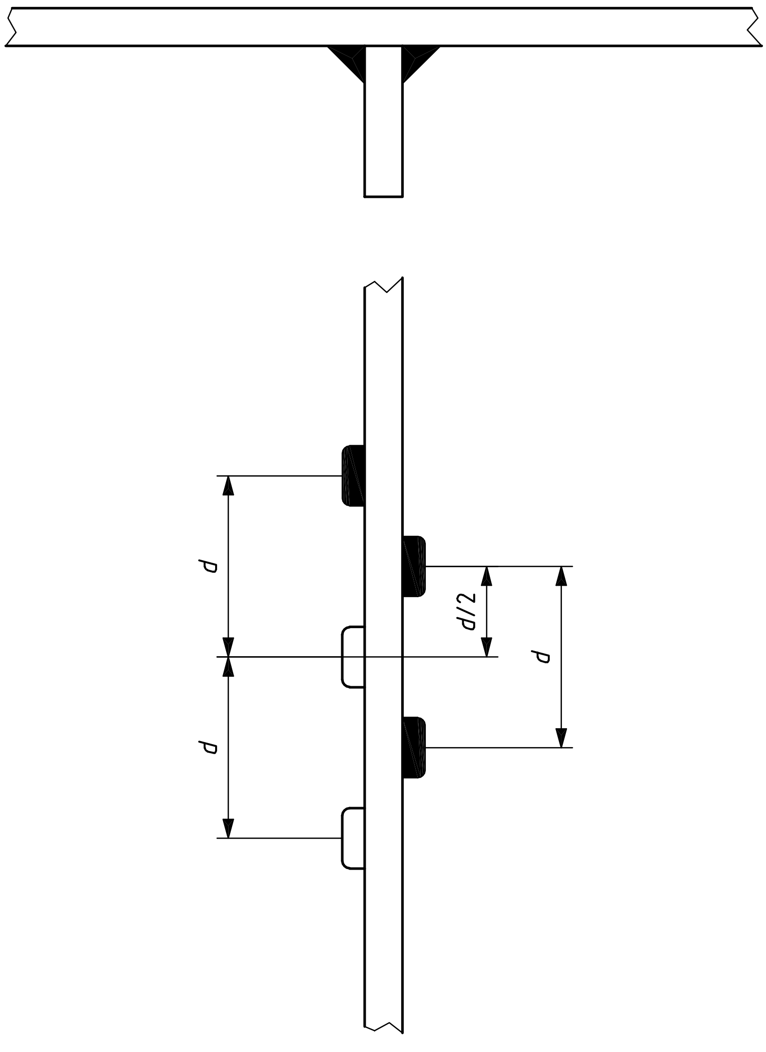


Figure H.3 — Example of joining stiffening ring to shell

H.2.2 Dished ends and spherical shells

H.2.2.1 Elastic buckling

There is adequate resistance to elastic buckling when:

$$p \leqslant 3{,}66\ \frac{E}{S_k}\left(\frac{s-c}{R}\right)^2$$

where S_k = 2,0 + 0,001 4 $R/(s-c)$. (H.11)

H.3 Method 2

H.3.1 Cylindrical shells

H.3.1.1 Elastic buckling

If $\dfrac{l_b}{D_a} > 1{,}537\dfrac{\left(1-\nu^2\right)^{0{,}25}}{\left(\dfrac{s}{D_a}\right)^{0{,}5}}$, then

$$p_e = \frac{E}{S_k}\left(\frac{20}{1-\nu^2}\right)\left(\frac{s-c}{D_a}\right)^3 \qquad \text{(H.12)}$$

If $\dfrac{l_b}{D_a} \leqslant 1{,}537\dfrac{\left(1-\nu^2\right)^{0{,}25}}{\left(\dfrac{s}{D_a}\right)^{0{,}5}}$, then

$$p_e = \frac{24{,}2E\left(\dfrac{s}{D_a}\right)^{2{,}5}}{S_k\left(1-\nu^2\right)^{0{,}75}\left[\left(\dfrac{l_b}{D_a}\right)-0{,}45\left(\dfrac{s}{D_a}\right)^{0{,}5}\right]} \qquad \text{(H.13)}$$

H.3.1.2 Stiffening rings

H.3.1.2.1 Each stiffening ring shall have a minimum moment of inertia as determined by either of the following formula:

$$I = \frac{S_k p {D_a}^3 l_b}{280E} \qquad \text{(H.14)}$$

or

$$I' = \frac{S_k p {D_a}^3 l_b}{218E} \qquad \text{(H.15)}$$

where

I is the required moment of inertia of the stiffening ring cross-section about its neutral axis parallel to the axis of the shell;

I' is the required moment of inertia of the combined ring-shell cross-section about its neutral axis parallel to the axis of the shell.

The required amount of inertia of the combined ring-shell section shall be maintained completely around the circumference of the cylinder unless the adequacy of the shell to carry the required critical collapse pressure is demonstrated through the finite element analysis method verified with scale model tests of each type of design.

H.3.1.2.2 If stiffening rings are used in designing the cylindrical portion (shell) of the inner vessel or vacuum jacket for external pressure, each ring shall be attached to the shell by fillet welds. Stiffening ring attachment welds on the outside of the vacuum jacket shall be continuous. All other ring attachment welds may be intermittent. Care should be taken in the design of ring attachments to minimize localized areas of buckling. Where intermittent welds are used, the total length of welds on each side of the ring must be at least one-third of the shell circumference, or if welded on one side, two-thirds of the shell circumference. The intermittent attachment welds shall be uniformly distributed and if welded on both sides, shall be staggered as shown in Figure H.3.

A portion of the shell may be included when calculating the moment of inertia of the ring. The effective width of shell plate, x, on each side of the attachment to the ring is given by the formula:

$$x = 0{,}78 \left(\frac{D_a \left(s - c \right)}{2} \right)^{0{,}5}$$

H.3.1.2.3 Where a stiffening ring consists of a closed section having two webs attached to the shell, the shell plate between the webs shall be included up to the limit of twice the value of x as defined in H.3.1.2.2. The flange of the section, if not a standard structural shape, is subject to the same limitation, with x based on D_a and s of the shell. The closed section between the ring and shell shall be provided with means to equalize pressure to the space occupied by the ring.

H.3.1.2.4 Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring or parts (webs) of one stiffening ring. If the stiffeners or webs of stiffeners should be so located that the maximum permissible effective shell sections overlap on either or both sides of a stiffener or web, the effective shell section for that stiffener or web shall be shortened by one-half of each overlap.

H.3.1.2.5 Length of the attachment weld segments shall not be less than 50 mm and shall have a maximum clear spacing between toes of adjacent weld segments of $8s$ for external rings and $12s$ for internal rings.

The number of intermittent attachment welds on each ring shall be at least $2n$ where n, the number of buckling lobes, is given by:

$$n = 1{,}63 \left(\frac{D_a{}^3}{l_b 2(s - c)} \right) 0{,}25$$

The size of the fillet weld leg shall be not less than the smallest of the following:

— 6 mm,

— shell thickness s,

— web thickness of the stiffener ring b.

H.3.2 Dished ends and spherical shells

The calculated pressure, p_c, shall not be less than the external design pressure.

For elastic buckling:

$$\rho_e = 1{,}25E\left(\frac{s}{R}\right)^2$$

For plastic deformation:

p_p, the pressure derived from the formula

$$p_p = \frac{20K_{20}\left(s-c\right)}{S_p\left(R+s\right)}$$

shall be higher than p_e obtained for elastic buckling using the above formula.

For ellipsoidal ends, $(R + s)$ may be taken as B_oD_a, where B_o is obtained from Table H.1.

Table H.1 — Values of spherical radius factor B_o for ellipsoidal end with pressure on convex side

$\frac{D_a}{2h_o}$	3,0	2,8	2,6	2,4	2,2	2,0	1,8	1,6	1,4	1,2	1,0
B_o	1,36	1,27	1,18	1,08	0,99	0,90	0,81	0,73	0,65	0,57	0,50
NOTE Interpolation is permitted for intermediate values.											

Annex I
(normative)

Design of openings in cylinders, spheres and cones — Calculation

I.1 General

This annex gives two calculation methods which are equally recognized and which give comparable results.

I.2 Method 1

Where the material property K of the reinforcement is lower than that of the shell, the cross-section of pad reinforcement and the thickness of nozzle reinforcement shall be reduced by the ratio of K values before determining the factor v_A. In the case of a shell subjected only to internal pressure, with a row of nozzles joined to the shell by fully penetrating welds, it is not necessary to calculate the individual reinforcement required for each nozzle. However, the thickness of the shell to resist internal pressure shall be calculated using the least value of weakening factor of either v_A obtained from equation (I.10) or v.

Openings shall also be reinforced according to the following relationship:

$$\frac{\rho}{10}\left(\frac{A_\rho}{A\sigma}+\frac{1}{2}\right) \leqslant \frac{K}{S} \tag{I.1}$$

which is based on equilibrium between the pressurized area A_p and the load-bearing cross-sectional area $A\sigma$. The wall thickness obtained from this relationship shall be not less than the thickness of the unpierced shell.

The pressurized area A_p and the load-bearing cross-sectional area $A\sigma$ which equals $A\sigma_0 + A\sigma_1 + A\sigma_2$ are obtained from Figures I.1 to I.5.

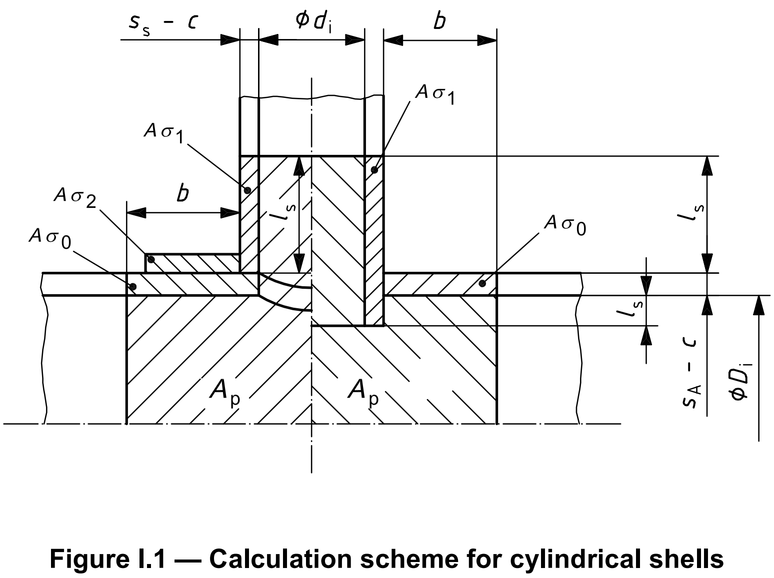


Figure I.1 — Calculation scheme for cylindrical shells

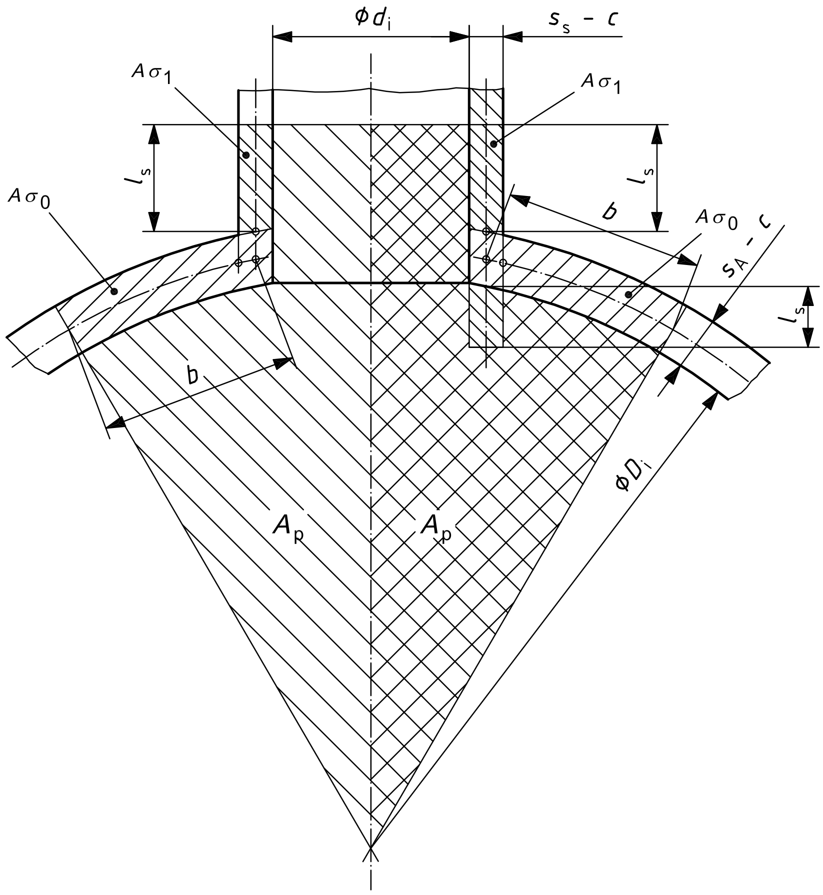


Figure I.2 — Calculation scheme for spherical shells

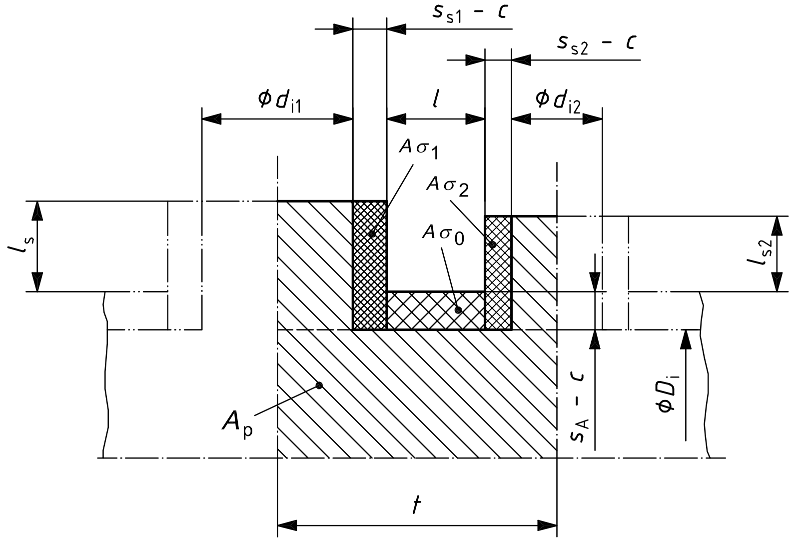


Figure I.3 — Calculation scheme for adjacent nozzles in a sphere or in a longitudinal direction of a cylinder

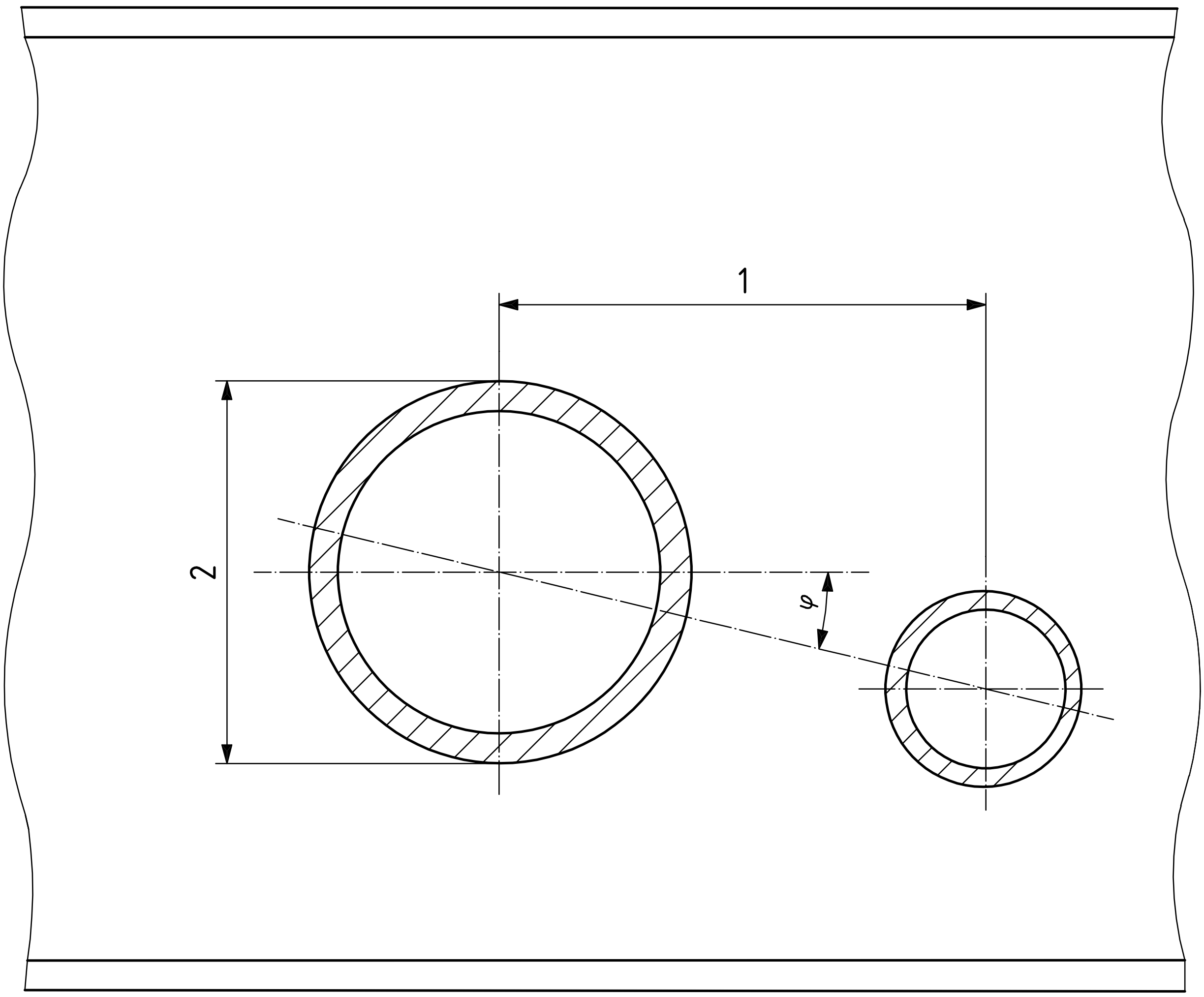


Figure I.4 — Openings between longitudinal and circumferential direction

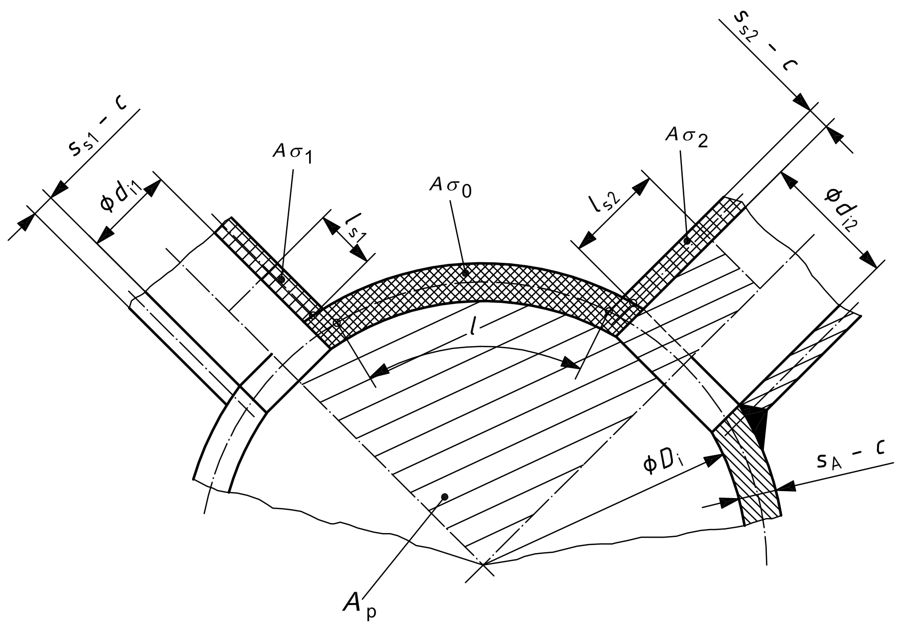


Figure I.5 — Calculation scheme for adjacent nozzles in a sphere or in a circumferential direction of a cylinder

The maximum extent of the load-bearing cross-sectional area shall be not more than b as defined in formula (I.4) for shells and l_S as defined in formula (I.6) or (I.7) for nozzles, as appropriate.

The protrusion of nozzles l_S may be included as load-bearing cross-sectional area up to a maximum length of:

$$l's' = 0{,}5\ ls \tag{I.2}$$

The restrictions given shall be observed.

If the material property K_1, K_2, etc. of the reinforcing material is lower than that of the shell the dimensions shall comply with:

$$\left(\frac{K}{S}-\frac{\rho}{20}\right)A_{\sigma 0}+\left(\frac{K_1}{S}-\frac{\rho}{20}\right)A_{\sigma 1}+\left(\frac{K_2}{S}-\frac{\rho}{20}\right)A_{\sigma 2} \geqslant \frac{\rho}{10}A_{\rho} \tag{I.3}$$

I.3 Method 2

The symbols used in this paragraph are defined as follows:

Subscript n refers to nozzle and v refers to vessel.

A_r total cross-sectional area of reinforcement required in the plane under consideration, in mm^2 (see Figure I.6) [includes consideration of nozzle area through shell if $\frac{K_{20n}}{K_{20v}} < 1{,}0$];

A_1 area in excess thickness in the vessel wall available for reinforcement, in mm^2 (see Figure I.6) [includes consideration of nozzle area through shell if $\frac{K_{20n}}{K_{20v}} < 1{,}0$];

A_2 area in excess thickness in the nozzle wall available for reinforcement, in mm^2 (see Figure I.6);

A_3 area available for reinforcement when the nozzle extends inside vessel wall, in mm^2 (see Figure I.6);

A_{41}, A_{42}, A_{43} cross-sectional area of various welds available for reinforcement, in mm^2 (see Figure I.6);

A_5 cross-sectional area of material added as reinforcement, in mm^2 (see Figure I.6);

c corrosion allowance, in mm;

D_p outside diameter of reinforcing element, in mm (actual size of reinforcing element may exceed the limits of reinforcement; however, credit cannot be taken for any material outside these limits);

d finished diameter of circular opening or finished dimension (chord length at mid-surface of thickness excluding excess thickness available for reinforcement) of non radial opening in the plane under consideration, in mm (see Figure I.6);

v 1 (see definitions for s_r and s_n);

v_1 1, when an opening is in the solid plate or in a full penetration butt joint; or joint efficiency when any part of the opening passes through any other welded joint;

h_i distance nozzle projects beyond the inner surface of the vessel wall, in mm. Extension of the nozzle beyond the inside surface of the vessel wall is not limited; however, for reinforcement calculations, credit shall not be taken for material outside the limits of reinforcement;

R_n inside radius of the nozzle under consideration, in mm;

$\frac{K_{20}}{S}$ allowable stress value in tension, in Newtons/mm^2;

$\frac{K_{20n}}{S}$ allowable stress in nozzle, in Newtons/mm^2;

$\frac{K_{20v}}{S}$ allowable stress in vessel, in Newtons/mm^2;

$\frac{K_{20p}}{S}$ allowable stress in reinforcing element, in Newtons/mm^2;

f_r strength-reduction factor, not greater than 1,0;

f_{r1} $\frac{K_{20n}}{K_{20v}}$ for nozzle wall inserted through the vessel wall;

f_{r1} 1,0 for nozzle wall abutting the vessel wall;

f_{r3} lesser of K_{20n} or K_{20p}/K_{20v};

f_{r4} K_{20p}/K_{20v};

s specified vessel wall thickness in the corroded condition (not including forming allowances), in mm (for pipe it is the nominal thickness less manufacturing under-tolerance allowed in the pipe specification);

s_p thickness or height of reinforcing element, in mm;

s_i nominal thickness of internal projection of nozzle wall, in mm;

s_r required thickness, in mm, of a seamless shell based on the circumferential stress, or of a formed end, for the designated pressure using v = 1.

Reinforcement shall be provided in amount and distribution such that the area requirements for reinforcement are satisfied for all planes through the centre of the opening and normal to the vessel surface. For a circular opening in a cylindrical shell, the plane containing the axis of the shell is the plane of greatest loading due to pressure. Not less than half the required reinforcement shall be on each side of the centre line of single openings.

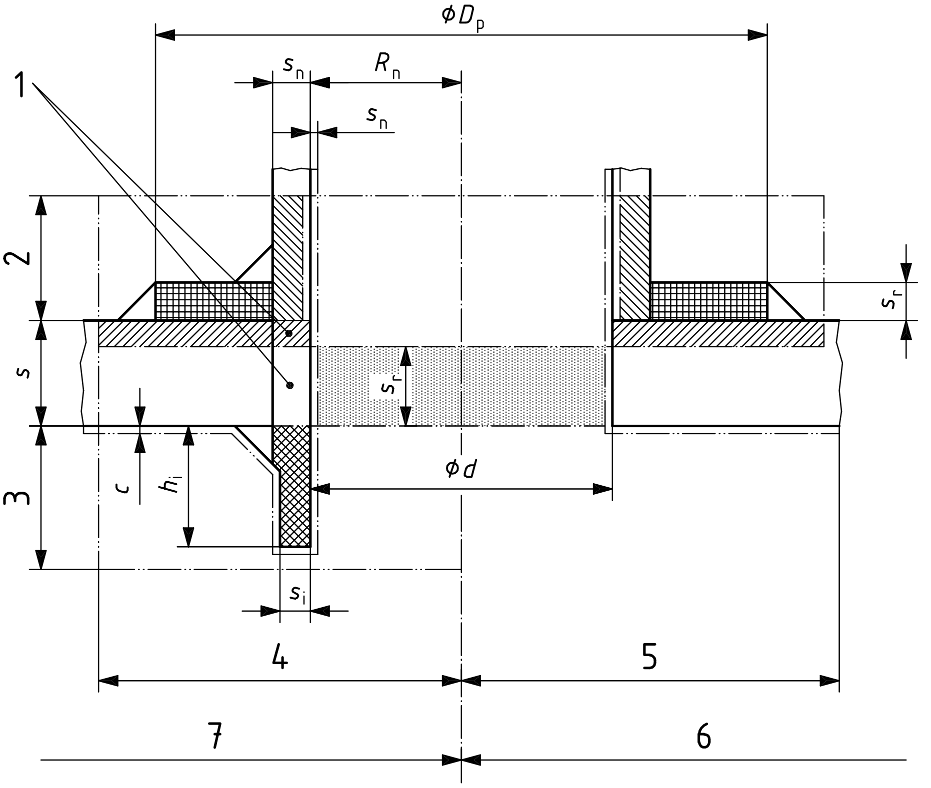
The total cross-sectional area of reinforcement, A_r, required in any given plane through the opening for a shell or dished end under internal pressure shall not be less than:

$$A_r = ds_r + 2s_n\, s_r\,(1 - f_{r1})$$

The reinforcement required for openings in vessels under external pressure need be only 50 % of that required per the above formula.

When two openings are spaced so that their limits of reinforcement overlap, the two openings shall be reinforced in the plane connecting the centres with a combined reinforcement that has an area not less than the sum of the areas required for each opening. No portion of the cross-section is to be considered as applying to more than one opening, nor to be considered more than once in a combined area.

The limits of reinforcement and the details of calculations for the reinforcement area to be provided are shown in Figure I.6.



Key

1 GENERAL NOTE : includes consideration of areas if $K_{2on}/K_{2ov} < 1{,}0$ (both sides of C_L)

2 $0{,}78\sqrt{R_n s_n}$

3 $0{,}78\sqrt{R_n s_i\,, h_i}$ use smaller value

4 $\underbrace{d \text{ or } n + s_n + s}_{\text{use larger value}}$

5 $\underbrace{d \text{ or } n + s_n + s}_{\text{use larger value}}$

6 For nozzle wall abutting the vessel wall

7 For nozzle wall inserted through the vessel wall

Figure I.6 — Nomenclature and formulas for reinforced openings

Without reinforcing element:

$= A_r = ds_r + 2s_n s_r (1 - f_{r1})$	Area required
$= A_1 = d(v_1 s - s_r) - 2s_n (v_1 s - s_r)(1 - f_{r1})$ $= 2(s + s_n)(v_1 s - s_r) - 2s_n (v_1 s - s_r)(1 - f_{n1})$	Area available in shell; use larger value
$= A_2 = 1{,}56\sqrt{R_n s_n (s_n - s_m) f_{r1}}$	Area available in nozzle projecting outward
$= A_3 = 1{,}56\sqrt{R_n s_n (s_i - f_{r1})}$ $= 2h_i s_i f_{r1}$	Area available in inward nozzle; use smaller value
$= A_{41} =$ outward nozzle weld $= (leg)^2 f_{r2}$	Area available in outward weld
$= A_{43} =$ inward nozzle weld $= (leg)^2 f_{r2}$	Area available in inward weld
if $A_1 + A_2 + A_3 + A_{41} + A_{43} + > A_r$	Opening is adequately reinforced
if $A_1 + A_2 + A_3 + A_{41} + A_{43} + < A_r$	Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With reinforcing element added:

$A \;\; =$ same as A above	Area required
$A_1 =$ same as A_1 above	Area available
$A_2 =$ same as A_2 above	Area available in nozzle projecting outward
$A_3 =$ same as A_3 above	Area available in inward weld
$= A_{41} =$ outward nozzle weld $= (leg)^2 f_{r3}$	Area available in outward weld
$= A_{42} =$ outer element weld $= (leg)^2 f_{r4}$	Area available in outer weld
$= A_{43} =$ inward nozzle weld $= (leg)^2 f_{r2}$	Area available in inward weld
$= A_5 = (D_p - d - 2s_n) s_p f_{r4}$	Area available in element
if $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 > A_r$	Opening is adequately reinforced

Figure I.6 (*continued*)

I.4 Ring or pad reinforcement or increased shell thickness

If the actual wall thickness of the cylinder or sphere is less than the required thickness s_A at the opening, the opening is adequately reinforced if the wall thickness s_A is available around the opening over a width of:

$$b = \sqrt{\left(D_i + s_A - c\right)\left(s_A - c\right)} \tag{I.4}$$

with a minimum of $3s_A$ (see Figure I.7).

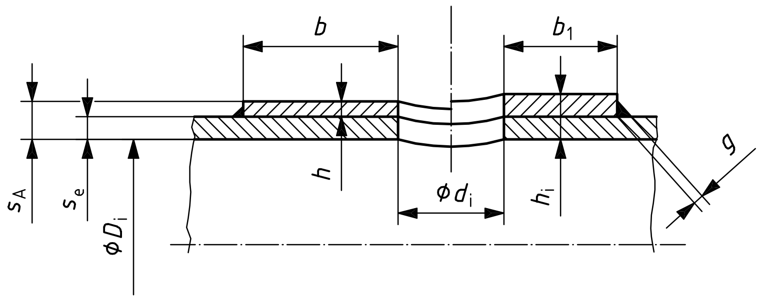


Figure I.7 — Pad reinforcement

For calculation purposes, s_A shall be limited to not more than twice the actual wall thickness.

The thickness of pad reinforcement in accordance with Figure I.7 should be not more than the actual wall thickness to which the pad is attached.

Internal pad reinforcement is not allowed.

The width of the pad reinforcement may be reduced to b_1, provided the pad thickness is increased to h_1, according to:

$$b_1 \times h_1 \geqslant b \times h \tag{I.5}$$

and the limits given above are observed.

I.5 Reinforcement by increased nozzle thickness

For calculation purposes, s_S shall be not more than twice s_A.

The thickness of the nozzle s_S should be not greater than twice s_A.

The wall thickness, s_A, at the opening shall extend over a width b in accordance with formula (I.4) with a minimum of $3s_A$.

The limits of reinforcement normal to the vessel wall are:

— for cylinders and cones, $l_s = 125\sqrt{\left(d_i + s_s - c\right)\left(s_s - c\right)}$; (I.6)

— for spheres, $l_s = l_s = \sqrt{\left(d_i + s_s - c\right)\left(s_s - c\right)}$. (I.7)

The length l_s may be reduced to l_{s1}, provided that the thickness s_s is increased to s_{s1}, according to the following:

$$l_{s1} \times s_{s1} \geqslant l_s \times s_s \tag{I.8}$$

and the limits given above are observed.

I.5.1 Reinforcement by a combination of increased shell and nozzle thicknesses

Shell and nozzle thicknesses may be increased in combination for the reinforcement of openings.

For the calculation of reinforcement, Figures I.2 and I.3 shall be applied together. The increase in shell thickness may be achieved by an actual increase in shell thickness or the addition of a pad.

I.5.2 Multiple openings

Multiple openings are regarded as single openings, provided the distance l between two adjacent openings, Figures I.3 and I.5, complies with:

$$l \geqslant 2\sqrt{(D_i + s_A - c)(s_A - c)} \tag{I.9}$$

If l is less than required by formula (I.9), a check shall be made to determine whether the cross-section between openings is able to withstand the load acting on it. Adequate reinforcement is available if the requirement of formula (I.1) or (I.3), as appropriate, is met.

Where adjacent openings in a cylinder are arranged intermediately between the longitudinal and circumferential direction, the calculation scheme for the longitudinal direction (Figure I.3) shall be applied, but the part of the pressure-loaded area corresponding to the unpierced cylinder $\left(\frac{tD_i}{2}\right)$ may be reduced with an arrangement factor $-0{,}5(1 + \cos^2 \varphi)$.

See Figure I.4 for angle φ.

Nozzles joined to the shell in line by full-penetration welds with the wall thickness calculated for internal pressure only may be designed with a weakening factor:

$$v_A = \frac{(t - d_i)}{2} \tag{I.10}$$

If the nozzles are not attached by full-penetration welds, D_a shall be used in formula (I.10).

Bibliography

[1] ISO 13985, *Liquid hydrogen — Land vehicle fuel tanks*

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