

Gas-loaded accumulators for fluid power applications

ICS 17.040.30

National foreword

This British Standard is the UK implementation of EN 14359:2006+A1:2010. It supersedes BS EN 14359:2006 which is withdrawn.

The start and finish of text introduced or altered by amendment is indicated in the text by tags. Tags indicating changes to CEN text carry the number of the CEN amendment. For example, text altered by CEN amendment A1 is indicated by **A1** **A1**.

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Gas-loaded accumulators for fluid power applications

Accumulateurs hydropneumatiques pour transmissions
hydrauliques

Hydrospeicher für Hydraulikanwendungen

This European Standard was approved by CEN on 18 September 2006 and includes Amendment 1 approved by CEN on 16 November 2010.

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Foreword

This document (EN 14359:2006+A1:2010) has been prepared by Technical Committee CEN/TC 54 “Unfired pressure vessels”, the secretariat of which is held by BSI.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by June 2011, and conflicting national standards shall be withdrawn at the latest by June 2011.

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

This document includes Amendment 1, approved by CEN on 2010-11-16.

This document supersedes EN 14359:2006.

The start and finish of text introduced or altered by amendment is indicated in the text by tags **A1** **A1**.

Where appropriate, equations and techniques are consistent with the requirements of EN 13445-3:2002 but this European Standard is presumed to satisfy the essential requirements of the Pressure Equipment Directive 97/23/EC in its own right.

NOTE If any matter of interpretation or doubt arises as to the meaning or effect of any normative part of this European Standard, or as to whether anything should be done or has been omitted to be done, in order that this European Standard should be complied with in full, the matter needs to be referred to the CEN/TC 54 Committee.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive(s).

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

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

1 Scope

1.1 This European Standard specifies the requirements for materials, design, manufacture, testing inspection, safety systems and documentation (including instructions for first operation), for commonly-used types of gas-loaded accumulators and gas bottles for fluid power applications (see 1.2).

1.2 This European Standard applies to the following types of components, defined as the pressure-containing envelope of gas-loaded accumulators:

- bladder type;
- diaphragm type;
- piston type;
- transfer type;
- gas bottles used to provide additional gas capacity.

They consist of one or several parts joined together by a variety of mechanical means and by welding.

1.3 This European Standard applies to gas-loaded accumulators which operate with the following conditions:

- subject to an internal gauge pressure greater than 0,5 bar;
- working temperature of not lower than $-50\text{ }^{\circ}\text{C}$ and not higher than $+200\text{ }^{\circ}\text{C}$;
- containing Group 2 liquids and gases as defined in the Pressure Equipment Directive 97/23/EC.

It does not apply to:

- accumulators for use with dangerous fluids (see NOTE 1).

NOTE 1 Fluid power applications utilize non-dangerous fluids as categorized in ISO 6743-4 in addition to an inert gas (e.g. nitrogen) which is used as the pre-charging medium.

NOTE 2 There are no design limits to the volume of the accumulator.

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.

EN 1043-1, *Destructive tests on welds in metallic materials — Hardness testing — Part 1: Hardness test on arc welded joints*

EN 1968:2002, *Transportable gas cylinders — Periodic inspection and testing of seamless steel gas cylinders*

EN 10002-1, *Metallic materials — Tensile testing — Part 1: Method of test at ambient temperature*

EN 10045-1, *Metallic materials — Charpy impact test — Part 1: Test method*

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

EN 13018, *Non-destructive testing — Visual testing — General principles*

EN 13445-2:2002, *Unfired pressure vessels — Part 2: Materials*

EN 13445-3:2002, *Unfired pressure vessels — Part 3: Design*

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

EN ISO 898-1:1999, *Mechanical properties of fasteners made of carbon steel and alloy steel — Part 1: Bolts, screws and studs (ISO 898-1:1999)*

EN ISO 6506-1, *Metallic materials — Brinell hardness test — Part 1: Test method (ISO 6506-1:2005)*

EN ISO 6506-2, *Metallic materials — Brinell hardness test — Part 2: Verification and calibration of testing machines (ISO 6506-2:2005)*

EN ISO 6506-3, *Metallic materials — Brinell hardness test — Part 3: Calibration of reference blocks (ISO 6506-3:2005)*

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

ISO 262, *ISO general-purpose metric screw threads — Selected sizes for screws, bolts and nuts*

ISO 9110-1, *Hydraulic fluid power — Measurement techniques — Part 1: General measurement principles*

ISO 9110-2, *Hydraulic fluid power — Measurement techniques — Part 2: Measurement of average steady-state pressure in a closed conduit*

ISO 10771-1, *Hydraulic fluid power — Fatigue pressure testing of metal pressure-containing envelopes — Part 1: Test method*

3 Terms, definitions, symbols, units and abbreviated terms

3.1 Terms and definitions

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

3.1.1

gas-loaded accumulator

hydraulic accumulator with separator between liquid and gas where the liquid is pressurized using the compressibility of an inert gas (for example nitrogen). The separator can be a bladder, a diaphragm or a piston.

Gas-loaded accumulators have shells, which can consist of cylinders, dished ends and flat plates. Openings are always isolated, located on the axis centre line and positioned at both ends of the accumulator. It is assumed that such vessels are axis-symmetrical

3.1.2

bladder accumulator

gas-loaded accumulator consisting of pressure-retaining shell, either spun-forged from seamless tube, hammer-forged from hollow bar or of welded construction, in which the liquid and gas are separated by a flexible bag or bladder normally retained at one end of the shell

3.1.3

diaphragm accumulator

gas-loaded accumulator consisting of pressure-retaining shell assembly, in which the construction can either be screwed or welded, with integral ports in which the liquid and gas are separated by a flexible membrane normally retained at its largest diameter to the shell

3.1.4

piston accumulator

gas-loaded accumulator consisting of cylinder body and end cap assemblies in which the liquid and gas are separated by a rigid sliding piston

3.1.5

transfer type accumulator

gas-loaded accumulator with a port for connecting additional gas capacity from one or more gas bottle(s)

3.1.6

gas bottle

inter-connected pressure vessels consisting of body and, depending upon construction, port assemblies used to provide additional gas capacity and communicating with the gas chamber of the accumulator by means of a pipe connection

3.2 Symbols, units and abbreviated terms

3.2.1 General

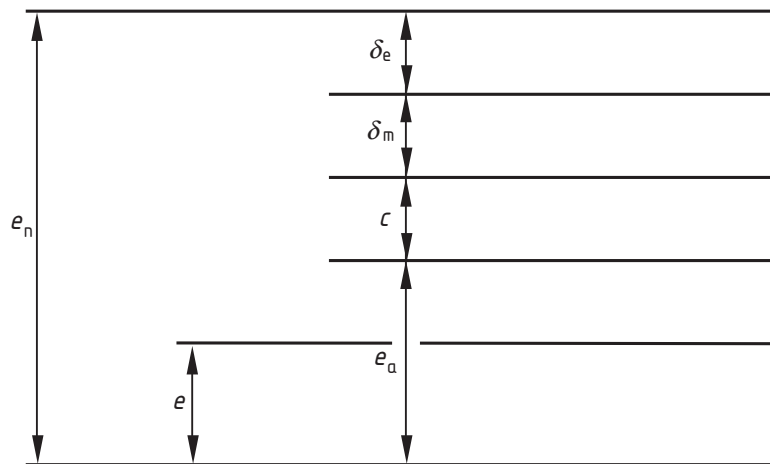
For the purposes of this document, the following symbols, units and abbreviated terms apply.

Table 3.2-1 — Symbols, characteristics and units

Symbol	Characteristics	Unit
e	Required thickness of the component, when obtained by direct calculation or the assumed thickness of the component, when obtained by non-direct calculation	mm ^a
e_a	Analysis thickness of the component = $e_n - \delta_e - \delta_m - c$	mm ^a
e_n	Nominal thickness of the component, as specified on the manufacturing detail drawings	mm ^a
c	Corrosion allowance	mm ^a
δ_e	Absolute value of the negative tolerance taken from the material standard of the component	mm ^a
δ_m	Possible thinning during manufacturing process of the component	mm ^a
N	Number of pressure cycles	
P_0	Pre-charging pressure; the gas pressure in the accumulator when the hydraulic circuit is not under pressure (initial state) at a temperature of $(20 \pm 5)^\circ\text{C}$	MPa ^b
P_1	Minimum working pressure of the hydraulic circuit	MPa ^b
P_2	Maximum working pressure of the hydraulic circuit	MPa ^b
P_3	Set pressure of the safety accessory for the accumulator, if one is fitted	MPa ^b
PS	Maximum allowable pressure, the pressure for which the accumulator has been designed and/or qualified by test	MPa ^b
PT	Test pressure	MPa ^b
P_2/P_0	Allowable pressure ratio below which the accumulator type can be used	
R_{eH}	Minimum upper yield strength	MPa ^b
R_m	Minimum tensile strength	MPa ^b
$R_{m/t}$	Minimum tensile strength at design temperature $t^\circ\text{C}$	MPa ^b
$R_{p0.2}$	Minimum 0,2 % - proof strength	MPa ^b
$R_{p0.2/t}$	Minimum 0,2 % - proof strength at design temperature $t^\circ\text{C}$	MPa ^b
$R_{p1.0}$	Minimum 1,0 % - proof strength	MPa ^b
$R_{p1.0/t}$	Minimum 1,0 % - proof strength at design temperature $t^\circ\text{C}$	MPa ^b
$TS_{\min}$	Minimum operating temperature of the hydraulic fluid or of the environment, whichever is lower or equal	$^\circ\text{C}$
$TS_{\max}$	Maximum operating temperature of the hydraulic fluid or of the environment, whichever is higher or equal	$^\circ\text{C}$
V	Internal volume of the gas chamber	litre
V_0	Gas volume at pressure P_0	litre
V_1, V_2	Volumes occupied by the gas contained in the accumulator and the additional chambers, if any, at pressures P_1 and P_2 at their respective temperatures	litre
z	Weld joint coefficient	
^a The relationships between the defined thicknesses are shown in Figure 3.1.		
^b MPa for calculation purposes only, otherwise the unit should be bar (1 MPa = 10 bar).		

3.2.2 Inter-relation of thickness definitions

The inter-relation of the various definitions of thickness is shown in Figure 3.1.



Key

- e required thickness
- e_n nominal thickness
- e_a analysis thickness ($e_n - \delta_e - \delta_m - c$)
- c corrosion allowance
- δ_e absolute value of the negative tolerance taken from the material standard of the component
- δ_m possible thinning during manufacturing process of the component

Figure 3.1 — Relationship of thickness definitions

4 Materials

4.1 Requirements for metallic materials

The pressure containing envelope of gas-loaded accumulators shall be constructed of either:

- harmonised materials used for the manufacture of unfired pressure vessels and meeting the requirements of EN 13445-2:2002;
- materials other than those specified in EN 13445-2:2002 provided that they have been accepted by a particular material appraisal;
- materials covered by a European approval for materials in accordance with Article 11 of Pressure Equipment Directive 97/23/EC.

4.2 Material certificates for components of the pressure containing envelope

Components used in the manufacture of the pressure containing envelope of gas-loaded accumulators to category II, III and IV according to Annex II of the Pressure Equipment Directive 97/23/EC, shall be accompanied by an inspection document in accordance with EN 10204:2004 type 3.1 – see Annex B.

5 Basic design and calculation criteria

5.1 General

The requirements of Clause 5 shall apply when the materials and welds are not subject to localised corrosion in the presence of either products which the gas-loaded accumulator is to contain, or the environment in which it is located.

5.2 Corrosion

A corrosion allowance is not normally required for accumulators covered by this European Standard. Where there is a risk of corrosion, a protection method and/or corrosion allowance shall be applied to the affected surfaces of the accumulator.

5.3 Qualification by similarity

Accumulators are often serially produced and it is possible to qualify a range of accumulators based upon the design, calculation and testing of one model within the range provided that other accumulators are similar. Two accumulators are similar provided that:

- they are made of the same material of the same form and origin;
- they are identical with the exception of length;
- the internal length of the cylindrical portion is not less than three times its external diameter.

If the length of the cylindrical portion is less than three times its external diameter, then a detailed stress analysis shall be undertaken.

5.4 Design methods

5.4.1 General

This European Standard specifies methods for the design by equations of accumulators or accumulator components. Satisfactory application of such equations alone shall be sufficient to demonstrate conformity to this European Standard provided the accumulator will be subjected to less than 500 pressure cycles between P_2 and P_1 during its lifetime.

Where the accumulator will be subjected to more than 500 pressure cycles between P_2 and P_1 , the manufacturer shall make an assessment for the effects of fatigue, either by analysis or test. This assessment shall form part of the Technical Documentation File.

Clauses 17 and 18 of EN 13445-3:2002 shall be used as the basis for a fatigue analysis and 7.6 of this European Standard shall be used as the method for conducting a fatigue cycling test.

Figure 5.1 shows the design process to be adopted.

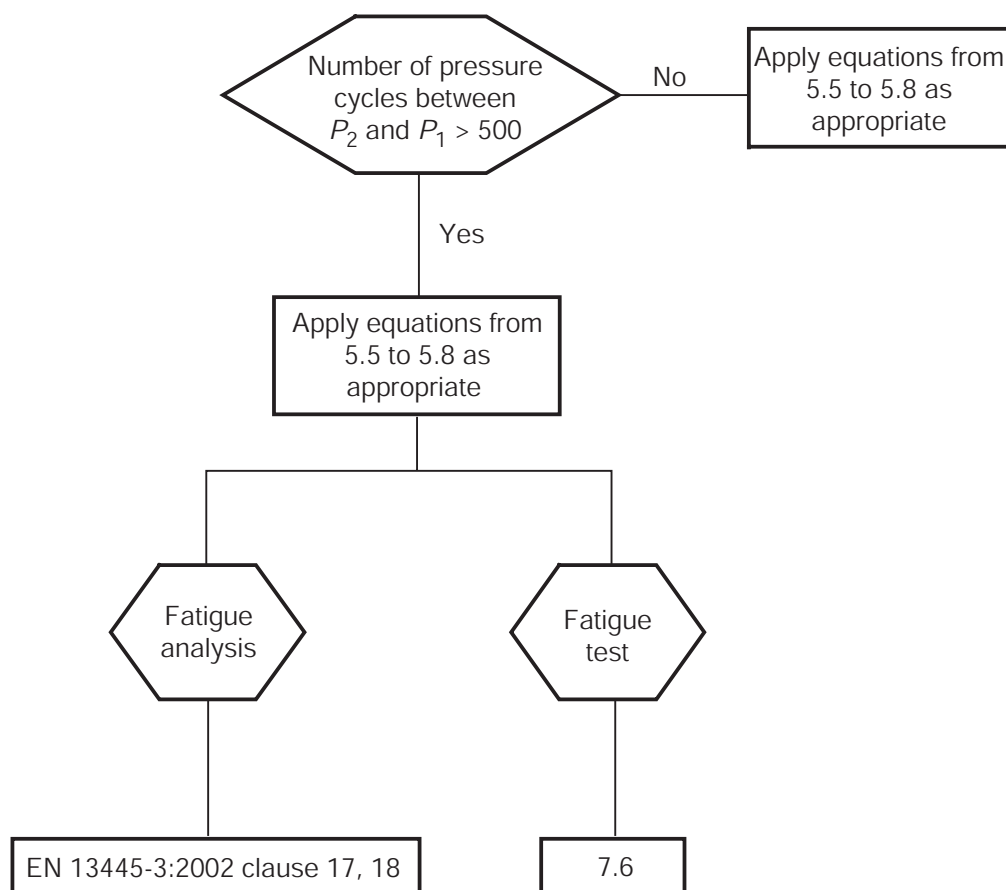


Figure 5.1 — Design process

5.4.2 Basic symbols, units and description

Table 5.4-1 lists basic symbols and units. Further and modified notation is provided in the individual sections.

Table 5.4-1 — Basic symbols and units

Symbols	Characteristics	Units
D_e	Outside diameter of the shell	mm
D_i	Inside diameter of the shell	mm
e_s	Required thickness of end to limit membrane stress in central part of shell	mm
e_y	Required thickness of knuckle to avoid axis-symmetric yielding	mm
f	Nominal design stress at design temperature	MPa ^a
f_a	Nominal design stress at test temperature	MPa ^a
f_{shear}	Nominal design shear stress at design temperature = $f \times 0,8$	MPa ^a
f_{test}	Nominal design stress for testing conditions	MPa ^a
h	Internal height of dished end measured from cylindrical part	mm
r	Inside radius of curvature of a knuckle	mm
R	Inside spherical radius of central part of torispherical end	mm
^a MPa for calculation purposes only, otherwise the unit should be bar (1 MPa = 10 bar).		

5.4.3 Maximum allowable values for the nominal design stress for pressure bearing parts

This subclause specifies maximum allowable values for the nominal design stress for pressure parts other than bolts and physical properties of steels.

Maximum values for the nominal design stress at operating temperatures are given in Table 5.4-2.

Table 5.4-2 — Maximum allowable values of the nominal design stress for pressure parts other than bolts

	Normal	Under test conditions
Steels other than austenitic with $A < 30\%$	$f = \min \left(\frac{R_{p0,2/t}}{1,5}; \frac{R_{m/20}}{2,4} \right)$	$f_{\text{test}} = \frac{R_{p0,2/t_{\text{test}}}}{1,05}$
Austenitic steels with $30\% < A \leq 35\%$	$f = \frac{R_{p1,0/t}}{1,5}$	$f_{\text{test}} = \frac{R_{p1,0/t_{\text{test}}}}{1,05}$
Austenitic steels with $A > 35\%$	$f = \max \left[\left(\frac{R_{p1,0/t}}{1,5} \right); \min \left(\frac{R_{p1,0/t}}{1,2}; \frac{R_{m/t}}{3} \right) \right]$	$f_{\text{test}} = \max \left(\frac{R_{p1,0/t_{\text{test}}}}{1,05}; \frac{R_{m/t_{\text{test}}}}{2} \right)$
Cast steels	$f = \min \left(\frac{R_{p0,2/t}}{1,9}; \frac{R_{m/20}}{3} \right)$	$f_{\text{test}} = \frac{R_{p0,2/t_{\text{test}}}}{1,33}$

NOTE Only $f = \frac{R_{p0,2/t}}{1,5}$ is valid for fine-grained steels and heat-treated steels.

5.5 Design and calculation methods common to all accumulator types

5.5.1 General

All applicable equations shall be used in order to demonstrate conformity with this European Standard. The maximum allowable pressure PS can be replaced by the test pressure PT when calculating for test conditions.

5.5.2 Specific definitions

5.5.2.1

Cylinder

right circular cylinder

5.5.2.2

Torispherical end

dished end, made up of a spherical cap, a toroidal knuckle and a cylindrical shell, the three components having common tangents where they meet

5.5.2.3

Klöpfer-type

torispherical end for which $R/D_e = 1,0$ and $r/D_e = 0,1$

5.5.2.4

Korbbogen-type

torispherical end for which $R/D_e = 0,8$ and $r/D_e = 0,154$

5.5.3 Cylindrical shells

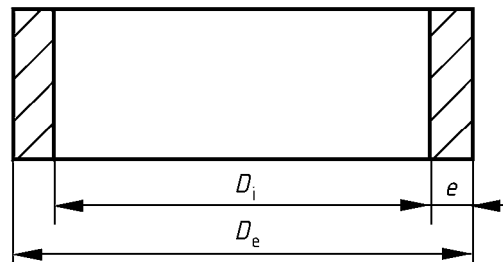


Figure 5.2 — Geometry for cylindrical shells

The required wall thickness e is given by:

$$e = \frac{PS \cdot D_i}{2f \cdot z - PS} \quad (5.5-1)$$

or

$$e = \frac{PS \cdot D_e}{2f \cdot z + PS} \quad (5.5-2)$$

For a given geometry:

$$PS = \frac{4f \cdot z \cdot e_a}{D_e + D_i} \quad (5.5-3)$$

NOTE The requirements of Equations 5.5-2 and 5.5-3 are valid for e/D_e not greater than 0,16.

5.5.4 Dished ends under internal pressure

5.5.4.1 Hemispherical ends

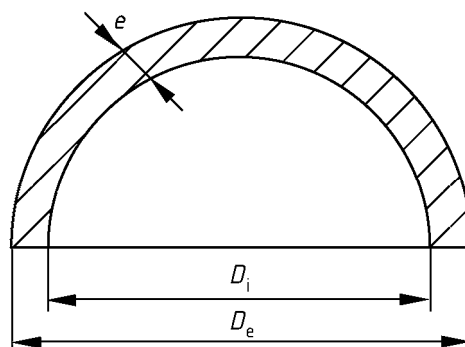


Figure 5.3 — Hemispherical end

The required wall thickness e of a hemispherical end (or spherical shell) shall be calculated from one of the two following equations:

$$e = \frac{PS \cdot D_i}{4 \cdot f \cdot z - PS} \quad (5.5-4)$$

or

$$e = \frac{PS \cdot D_e}{4 \cdot f \cdot z + PS} \quad (5.5-5)$$

Shell thickness may be increased at junctions with other parts such as cylindrical shells or the torus of a torispherical end. Increased thickness may also be necessary to provide reinforcement at isolated openings. The thickness of the cylinder up to the tangent line shall be kept at or above the minimum of the required cylinder wall thickness.

5.5.4.2 Torispherical ends

5.5.4.2.1 General

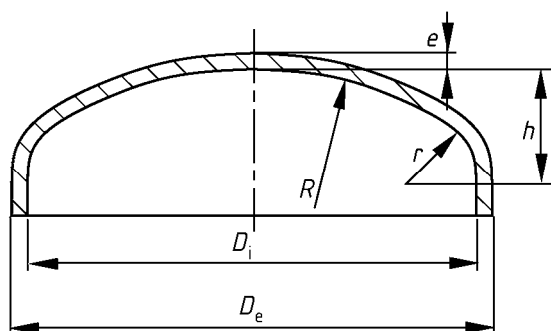


Figure 5.4 — Torispherical end

5.5.4.2.2 Conditions of applicability

The following requirements are limited in application to ends for which all of the following conditions are met:

- a) $r \leq 0,2 D_i$
- b) $r \geq 0,06 D_i$
- c) $r \geq 2e$
- d) $e \leq 0,08 D_e$
- e) $e_a \geq 0,005 D_e$
- f) $R \leq D_e$

The required thickness e shall be the greatest of e_s and e_y where

$$e_s = \frac{PS \cdot R}{2 \cdot f \cdot z - 0,5 \cdot PS} \quad (5.5-6)$$

$$e_y = \frac{\beta \cdot PS (0,75 \cdot R + 0,2 \cdot D_i)}{f} \quad (5.5-7)$$

where β is found from the following equations:

$$Y = \min(e/R; 0,4) \quad (5.5-8)$$

$$Z = \log_{10}(1/Y) \quad (5.5-9)$$

$$X = r/D_i \quad (5.5-10)$$

$$N = 1,006 - \frac{1}{6,2 + (90 \cdot Y)^4} \quad (5.5-11)$$

For $X = 0,06$

$$\beta_{0,06} = N(-0,3635Z^3 + 2,2124Z^2 - 3,2937Z + 1,8873)$$

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

$$\beta = 25\{(0,1 - X)\beta_{0,06} + (X - 0,06)\beta_{0,1}\}$$

For $X = 0,1$

$$\beta_{0,1} = N(-0,1833Z^3 + 1,0383Z^2 - 1,2943Z + 0,837)$$

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

$$\beta = 10\{(0,2 - X)\beta_{0,1} + (X - 0,1)\beta_{0,2}\}$$

For $X = 0,2$

$$\beta_{0,2} = \max\{0,95(0,56 - 1,94Y - 82,5Y^2); 0,5\} \quad (5.5-12)$$

NOTE The above equations for β lead to an iterative calculation for which a computer procedure is recommended.

5.5.4.2.3 Special conditions for Klöpper-type ends

$$R = D_e$$

$$r = 0,1 D_e$$

$$h = 0,1935 D_e - 0,455 e_a$$

$$0,001 \leq \frac{e_a}{D_e} \leq 0,1 \quad (5.5-13)$$

5.5.4.2.4 Special conditions for Korbogen-type ends

$$R = 0,8 D_e$$

$$r = 0,154 D_e$$

$$h = 0,255 D_e - 0,635 e_a$$

$$0,001 \leq \frac{e_a}{D_a} \leq 0,1 \quad (5.5-14)$$

5.5.5 Isolated openings and nozzles in spherical shells and spherical centre areas of dished ends

5.5.5.1 Specific symbols and units

Table 5.5-1 lists further specific symbols and units.

Table 5.5-1 — Specific symbols and units

Symbol	Characteristics	Unit
a	Distance taken along the average wall surface on the section where the reinforcement of an opening has to be calculated, between the opening centre and the external edge of a nozzle; if no nozzle is present, a is the distance between the centre and the internal edge of the opening	mm
A_f	Stress-loaded cross-sectional area effective as compensation	mm ²
A_{fb}	A_f of a nozzle (b =branch)	mm ²
A_{fs}	A_f of shell wall (main body)	mm ²
A_{fw}	Area of weld between nozzle and shell (only if outside the shell and nozzle shape, fillet weld)	mm ²
A_p	Pressure-loaded area	mm ²
A_{ps}	A_p of shell (main body)	mm ²
A_{pb}	A_p of nozzle	mm ²
d	Diameter (or maximum width) of opening, or inside diameter of nozzle	mm
d_{eb}	Outside diameter of nozzle	mm
d_{ib}	Inside diameter of nozzle	mm
e_{bk}	Required thickness of knuckle to avoid plastic buckling	mm
e_b	Required thickness of nozzle (or mean thickness within the length l_{bo} or l_{bio})	mm
e'_b	Effective thickness of nozzle useful for reinforcement	mm
e'_s	Length of penetration of nozzle into shell wall for set-in nozzles with partial penetration	mm
e_y	Required thickness of knuckle to avoid axis-symmetric yielding	mm
f_b	f of nozzle	MPa ^a
f_{bk}	Nominal design stress for buckling equation	MPa ^a
f_s	f of shell material (main body)	MPa ^a
l_b	Length of nozzle extending outside the shell	mm
l'_b	Effective length of nozzle outside the shell, useful for reinforcement	mm
l_{bi}	Length of nozzle extending inside the shell	mm
l'_{bi}	Effective length of nozzle inside the shell, useful for reinforcement	mm
l_{bo}	Maximum length of nozzle outside the shell useful for reinforcement	mm
l_{bio}	Maximum length of nozzle inside the shell, useful for reinforcement	mm
l'_s	Effective length of shell, useful for opening reinforcement	mm
l_{so}	Maximum length of shell contributing to opening reinforcement taken on the mean radius of curvature of the shell wall	mm
r_{is}	Inside radius of curvature of the shell at the opening centre	mm
r_{es}	Outside radius of curvature of the shell at the opening centre	mm
r_{ms}	Mean radius of curvature of the shell at the opening centre	mm

^a MPa for calculation purposes only, otherwise the unit should be bar (1 MPa = 10 bar).

5.5.5.2 General

All openings shall be isolated and circular with centrelines coaxial with the axis. For spherical shells and hemispherical or torispherical ends the following conditions shall be met.

$$\frac{d}{2 \cdot r_{is}} \leq 0,5 ; \frac{d}{D_e} \leq 0,6 \quad (5.5-15)$$

NOTE Reinforcement is only permitted by increasing the wall thickness of the shell and/or a nozzle. Nozzles may be 'set-on' or 'set-in' (see Figures 5.6 and 5.8).

5.5.5.3 Small openings

If an isolated opening has a diameter d meeting the following condition:

$$d \leq 0,14 \cdot \sqrt{e_s \cdot (2 \cdot r_{is} + e_s)} \quad (5.5-16)$$

then the opening shall be considered a "small opening" and it needs no reinforcement. No further calculation is necessary.

5.5.5.4 Openings and nozzles used in accumulators

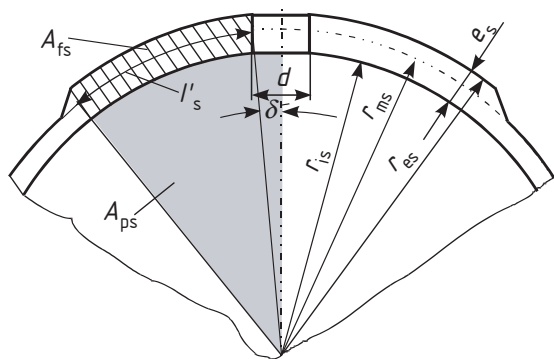


Figure 5.5 — Reinforcement by wall thickness

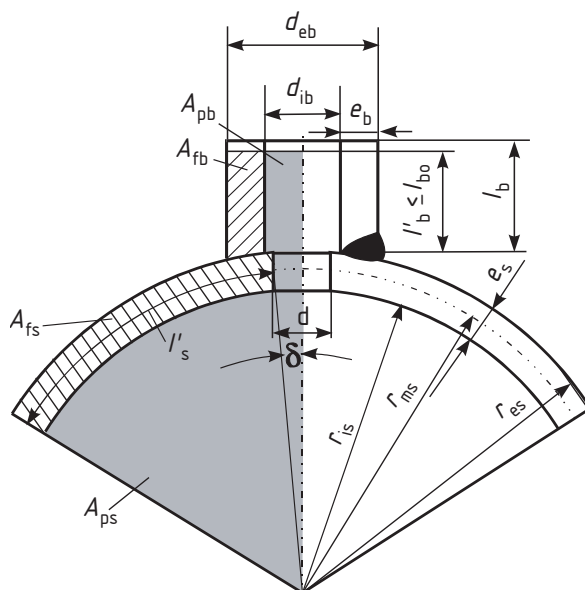


Figure 5.6— Reinforcement by Set-on nozzle (and wall thickness)

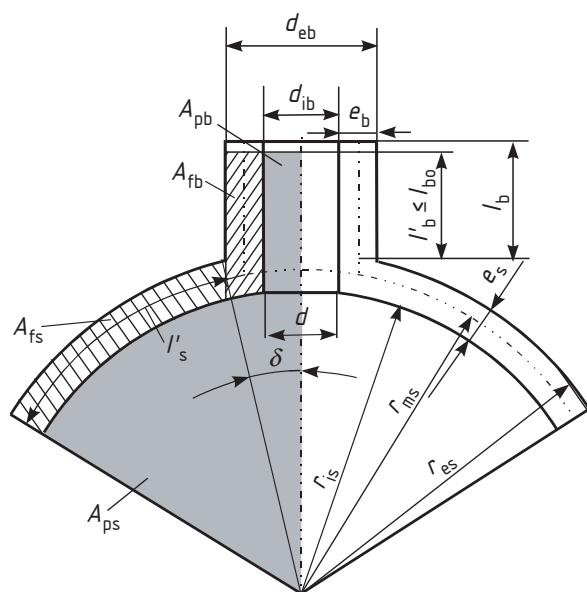


Figure 5.7 — Reinforcement by forged nozzle (and wall thickness)

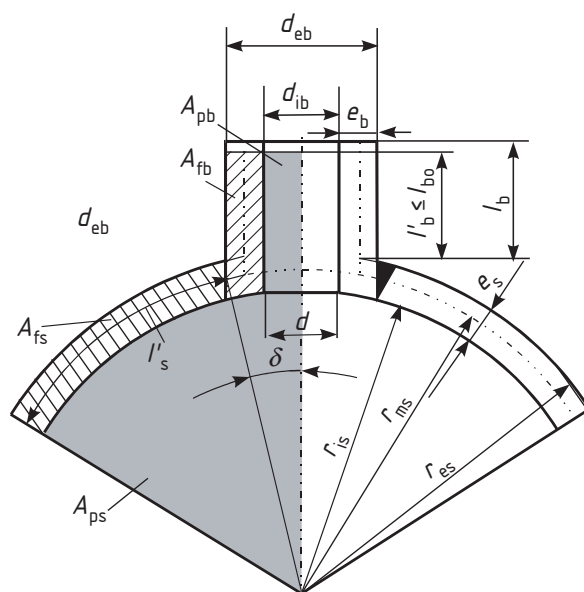


Figure 5.8 — Reinforcement by Set-in nozzle (and wall thickness)

5.5.5.5 Calculation procedure for openings in shells

5.5.5.5.1 Purpose

The design method specified in this clause is applicable for a spherical shell or the spherical part of a torispherical end.

The following conditions shall be satisfied:

$$(A_{fs} + A_{fw}) \cdot (f_s - 0,5 \cdot PS) + A_{fb} \cdot (f_{ob} - 0,5 \cdot PS) \geq PS \cdot (A_{ps} + A_{pb}) \quad (5.5-17)$$

$$f_{ob} = \text{MIN}(f_s ; f_b) \quad (5.5-18)$$

$$\text{For spherical shells: } r_{is} = \frac{D_e}{2} - e_s = \frac{D_i}{2} \quad (5.5-19)$$

$$\text{For hemispherical or torispherical ends: } r_{is} = R \quad (5.5-20)$$

$$\text{For dished ends and spherical shells: } A_{ps} = 0,5 \cdot r_{is}^2 \cdot \frac{l'_s + a}{0,5 \cdot e_s + r_{is}} \quad (5.5-21)$$

With nozzle axis perpendicular to the shell wall the value a is given by the following equation:

$$a = r_{ms} \cdot \arcsin(\delta) \quad (5.5-22)$$

where

$$r_{ms} = r_{is} + 0,5 \cdot e_s \quad (5.5-23)$$

$$\delta = \frac{d_{eb}}{2 \cdot r_{ms}} \quad (5.5-24)$$

Moreover:

$$\text{for set-in nozzles: } Af_s = e_s \cdot l'_s \quad (5.5-25)$$

$$\text{for set-on nozzles: } Af_s = e_s \cdot (e_b + l'_s) \quad (5.5-26)$$

5.5.5.5.2 Reinforcement by increased wall thickness of the shell

The length of shell l'_s , contributing to reinforcement of the opening, taken from the edge of the opening or from the external diameter of a nozzle and along the mean surface of the shell, shall not be greater than l_{so} .

$$l'_s \leq l_{so} = \sqrt{[(2 \cdot r_{is} + e_s) \cdot e_s]} \quad (5.5-27)$$

5.5.5.5.3 Reinforcement by increased nozzle thickness

The reinforcement of an opening can be obtained by increasing the wall thickness of the nozzle above the minimum thickness required in order to withstand the internal pressure (see 5.5.3). This possibility is independent of any reinforcement provided by increasing the wall thickness of the shell. The length contributing to the reinforcement shall not be greater than l_{so} for the shell and not more than l_{bo} for the nozzle, with:

$$l_{bo} \leq \sqrt{[(d_{eb} - e_b) \cdot e_b]} \quad (5.5-28)$$

for set-in nozzles and forged nozzles ($l'_{bi}=0$):

$$Af_b = e'_b \cdot (l'_b + l'_{bi} + e'_s) \quad (5.5-29)$$

for set-on nozzles:

$$Af_b = e'_b \cdot l'_b \quad (5.5-30)$$

where

$$l'_b = \text{MIN} (l_{bo}; l_b);$$

$$l'_{bi} = \text{MIN} (l_{bio}; l_{bi});$$

e_s is the length of penetration (full or part) of set-in nozzle into shell wall ($\leq e_s$);

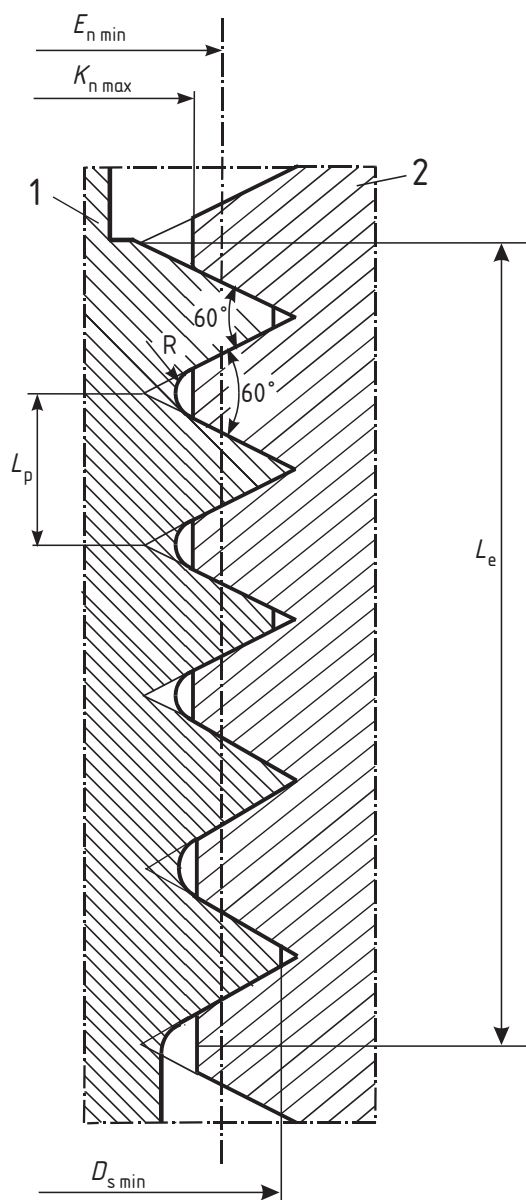
$$A_{pb} = 0,5 \cdot d_i \cdot (l'_b + e_s);$$

A_{fs} and A_{ps} have already been defined in Table 5.5-1.

5.5.6 Thread calculation

5.5.6.1 General

The requirements of 5.5.6 shall apply when assessing the stresses in the threads of gas-loaded accumulators.



Key

- 1 external thread
- 2 internal thread

NOTE Thread geometry (60°) shown as typical only – other thread types are permissible.

Figure 5.9 — Geometry of threads

5.5.6.2 Specific symbols and units

Table 5.5-2 lists further specific symbols and units.

Table 5.5-2 — Specific symbols and units

Symbol	Characteristics	Unit
$E_{n \max}$	Maximum pitch diameter of internal thread	mm
$E_{n \min}$	Minimum pitch diameter of internal thread	mm
$D_{s \min}$	Minimum major diameter of external thread	mm
f_{sh}	Shear stress at calculation pressure	MPa ^a
$K_{n \max}$	Maximum minor diameter of internal thread	mm
L_e	Length of thread engagement at thread pitch diameter	mm
L_p	Thread pitch	mm

^a MPa for calculation purposes only, otherwise the unit should be bar (1 MPa = 10 bar).

5.5.6.3 Shearing stress and length of thread engagement

The shearing stress in threads shall be determined from the following equation:

$$f_{sh} = \frac{D_i^2 \cdot PS}{2 \cdot L_e \cdot E_{n \min}} \quad (5.5-31)$$

The length of thread engagement L_e shall be determined from the following equation:

$$L_e \geq \frac{D_i^2 \cdot PS}{2 \cdot f_{sh} \cdot E_{n \min}} \quad (5.5-32)$$

5.5.6.4 Thread compressive stress

The compressive stress in the thread shall be determined from the following equation:

$$f_{comp} = \frac{D_i^2 \cdot PS \cdot L_p}{L_e \cdot (D_{s \min}^2 - K_{n \max}^2)} \quad (5.5-33)$$

The following shall be satisfied:

$$f_{comp} \leq R_{p0,2/t}$$

And the length of thread engagement L_e shall be determined from the following equation:

$$L_e \geq \frac{D_i^2 \cdot PS \cdot L_p}{f \cdot (D_{s \min}^2 - K_{n \max}^2)} \quad (5.5-34)$$

5.6 Specific design criteria for piston accumulators

5.6.1 Threaded end caps

5.6.1.1 General

The requirements of 5.6.1 shall apply when assessing the stresses in piston accumulators with threaded end caps.

5.6.1.2 Specific symbols and units

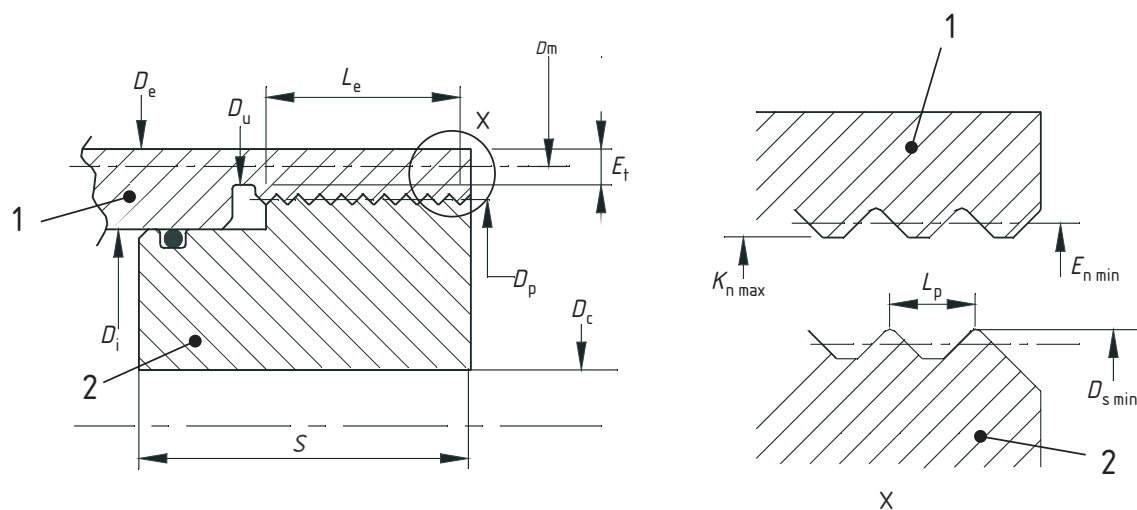
Table 5.6-1 lists further specific symbols and units.

Table 5.6-1 — Specific symbols and units

Symbols	Characteristics	Units
A_U	Cross-sectional area of accumulator body thread relief	mm ²
C	End plate attachment factor	
D_C	End plate connection port diameter	mm
D_p	Nominal pitch diameter of body thread	mm
D_U	Thread relief diameter	mm
E_t	Thickness of shell at thread relief	mm
f_b	Flaring stress	MPa ^a
f_{comp}	Thread compressive stress	MPa ^a
F_e	Applied force (moment per unit length on accumulator body) - Nmm/mm	N
D_m	Mean body diameter at thread relief	mm
S	Thickness of end plates	mm
Y	End plate attachment factor	
Y'	End plate attachment factor	
b	Seal half section diameter (for 'O' ring)	mm
m	Seal coefficient	
G_o	Seal groove outside diameter	mm
G	Seal reaction diameter	mm
α	Factor used for flaring stress calculation	
β	Factor used for flaring stress calculation	mm ⁻¹
ν	Poisson's ratio	

^a MPa for calculation purposes only, otherwise the unit should be bar (1 MPa = 10 bar).

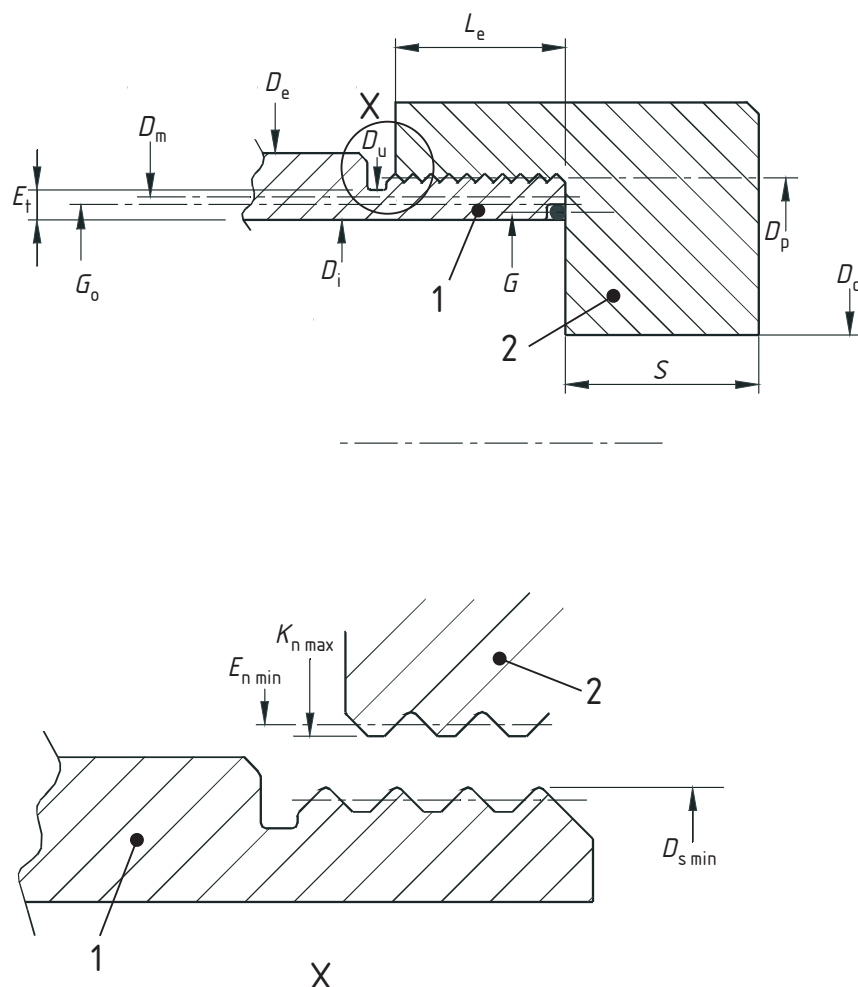
5.6.1.3 Geometry



Key

- 1 internally threaded body
- 2 end plate

Figure 5.10 — End cap geometry – internally threaded body



Key

- 1 externally threaded body
- 2 end plate

Figure 5.11 — End cap geometry – externally threaded body

5.6.1.4 Dimensions for screw threads

Table 5.6-2 shows the recommended thread pitch for a range of thread sizes. This should be used in conjunction with the following equations.

Table 5.6-2 — Screw threads

Nominal pitch diameter of body thread (D_p)	Thread pitch (L_p)
≤ 100 mm	$\geq 1,5$ mm
100 mm to 200 mm	≥ 2 mm
200 mm to 400 mm	≥ 3 mm
> 400 mm	≥ 4 mm

NOTE For thread calculation see 5.5.6.

5.6.1.5 Thickness of flat end plates

5.6.1.5.1 Internally threaded body

The thickness of flat end plates s for an internally threaded body shall be determined from the following equation:

$$S = C \cdot Y \cdot D_i \sqrt{\frac{PS}{f}} \quad (5.6-1)$$

The value of $C = 0,5$ is suitable for threaded end plates.

Y is dependent upon the value of the quotient D_c/D_i .

$$Y = \left\{ \sum_{i=1}^6 A_i \cdot \left(\frac{D_c}{D_i} \right)^{i-1} ; \quad 0 < \left(\frac{D_c}{D_i} \right) \leq 0,8 \right\} \quad (5.6-2)$$

$$A_1 = 0,999\,034\,2$$

$$A_2 = 1,980\,626\,0$$

$$A_3 = -9,018\,554\,0$$

$$A_4 = 18,632\,830\,0$$

$$A_5 = -19,497\,590\,0$$

$$A_6 = 7,612\,568\,0$$

Alternatively, a value of 1,16 may be used for Y which will provide a conservative design.

5.6.1.5.2 Externally threaded body

The thickness of flat end plates s for an externally threaded body shall be determined from the following equation:

$$S = Y' \cdot S_o \quad (5.6-3)$$

where

$$S_o = \sqrt{\left[\frac{3(3+\nu)}{32} G^2 + 3 \left(\frac{G}{4} + 2 \cdot b \cdot m \right) (D_p - G) \right] \frac{PS}{f}} \quad (5.6-4)$$

and

$$Y' = \sqrt{\frac{G}{G - D_c}} ; \quad G = G_o - 2 \cdot b \quad (5.6-5)$$

NOTE G can be approximated by the mean geometrical diameter of the seal and b , to the half width of the seal in contact (b approximates to the half section diameter for an 'O' ring).

where

$m = 0,5$ for elastomeric seals with hardness less than 75 IRDH;

$m = 1$ for elastomeric seals with hardness equal to or greater than 75 IRDH;

for other cases, refer to EN 13445-3:2002, Annex H.

5.6.1.6 Tension and flaring stress in threaded accumulator body

The tension and flaring stress in a threaded accumulator body shall be determined from the following equations:

$$f_b = \frac{\pi \cdot D_i^2 \cdot PS}{4 \cdot A_u} + \frac{6 \cdot F_e \cdot \alpha}{E_t^2} \quad (5.6-6)$$

and

$$F_e = \frac{D_i^2 \cdot PS \cdot |D_m - D_p|}{8 \cdot D_m} \quad (5.6-7)$$

where for an internally threaded body:

$$A_u = \frac{\pi \cdot (D_e^2 - D_u^2)}{4} \quad (5.6-8)$$

and

$$E_t = \frac{D_e - D_u}{2} \quad (5.6-9)$$

and

$$D_m = \frac{D_e + D_u}{2} \quad (5.6-10)$$

And for an externally threaded body:

$$A_u = \frac{\pi \cdot (D_u^2 - D_i^2)}{4} \quad (5.6-11)$$

and

$$E_t = \frac{D_u - D_i}{2} \quad (5.6-12)$$

and

$$D_m = \frac{D_i + D_u}{2} \quad (5.6-13)$$