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BSI Standards Publication

Unfired pressure vessels

Part 102: Example of application of vertical vessel with bracket supports

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National foreword

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A list of organizations represented on this committee can be obtained on request to its secretary.

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Unfired pressure vessels - Part 102: Example of application of vertical vessel with bracket supports

Unbefeuerte Druckbehälter - Beispiel 2: Stehende Behälter mit Tragpratzen

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Foreword

This document (CEN/TR 13445-102:2015) has been prepared by Technical Committee CEN/TC 54 “Unfired pressure vessels”, the secretariat of which is held by BSI.

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Introduction

Harmonized standards under Pressure Equipment Directive (97/23/EC) have been adopted over the past few years on the basis of mandate M 071. These standards give appropriate solutions for designing and building safe pressure equipment complying with the pressure equipment directives.

Although the main standards for the major product groups are now available, further action is needed to ensure a take-up by industry of these standards.

A recent public consultation on the use of EN Standards in the field of pressure equipment has shown that better knowledge of content and better usability are the more substantial aspects to encourage the use of the harmonized European standards (document CEN/PE/AN N 220).

The Pressure equipment Migration Help Desk, EN 13445/MHD, was created in August 2002 to give to the standard users a central point where raising questions and obtaining authorized answers. From the questions it received, the help desk has identified the publication of examples of application as a key issue and has developed rules of procedure for their publication as CEN deliverables (document CEN/PE/AN N 128).

Examples of application is an efficient way to help the standard user to correctly understand and apply the requirements of the standard and to be aware of the permissible deviations, possible alternatives, use of normative reference documents, etc. It can also assist training organization and software developers.

The project, in its efforts to broaden the application of the European Standards harmonized for PED, will support the actions of the European Commission in the field of safety of pressure equipment.

It will also promote the use of these European Standards on the global market.

1 Scope

This Technical Report details the design, manufacturing, inspection and testing of a steel vessel submitted to pressure cycles, using the EN 13445 series for "Unfired pressure vessels", to guide the user of these standards in sequential decision making, together with some alternative choices.

2 Normative references

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

EN 13445-1:2009_Issue 5, *Unfired pressure vessels – Part 1: General* [1]

EN 13445-2:2009_Issue 5, *Unfired pressure vessels – Part 2: Materials* [2]

EN 13445-3:2009_Issue 5, *Unfired pressure vessels – Part 3: Design* [3]

EN 13445-4:2009_Issue 5, *Unfired pressure vessels – Part 4: Fabrication* [4]

EN 13445-5:2009_Issue 5, *Unfired pressure vessels – Part 5: Inspection and testing* [5]

EN 10028-2:2003, *Flat products made of steels for pressure purposes – Part 2: Non-alloy and alloy steels with specified elevated temperature properties* [6]

3 The vessel and its operating conditions

3.1 Drawing of the vessel

The technical drawing of the vessel and vessel details is represented in Annex A:

A note in the introduction of EN 13445-1, clearly says that "In EN 13445 the term pressure vessel includes the welded attachments up to and including the nozzle flanges, screwed or welded connections".

The briefed lay-out is given as in Figure 1.

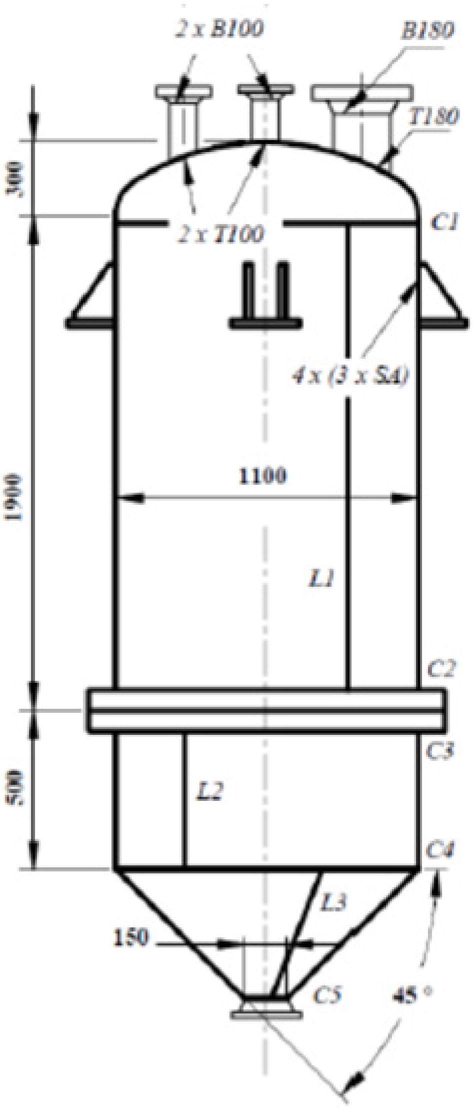


Figure 1 — Briefed lay-out

3.2 Calculation model

The calculation model is presented in 3D in Figure 2.

Figure 2 — Calculation model

3.3 Operating conditions

The general characteristics given by the user are reproduced below:

a) content: gas group 1, density of 0,48;

b) internal pressure: 18 bar / 0,5 bar;

c) temperature: 20 °C/260 °C;

d) number of expected full pressure cycles: 1200.

3.4 Comments on the operating conditions provided by the User

The gas group 1 is a dangerous fluid according to Council Directive 67/548/EEC of 27 June 1967 on the approximation of the laws, regulations and administrative provisions relating to the classification, packaging and labelling of dangerous substances.

See also:

http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=CELEX:31967L0548:en:NOT

and further information on

https://osha.europa.eu/nl/legislation/directives/exposure-to-chemical-agents-and-chemical-safety/osh-related-aspects/58

In the contract 1 200 pressure cycles from 18 bar (1,8 MPa) to 0,5 bar (0,05 MPa) are expected. A design pressure of 1,8 MPa will not be used to avoid a short duration pressure surge at each cycle (See Pressure

Equipment Directive, *Annex I*, clause 2.11.2 *Pressure limiting devices*). Therefore the safety valve will be set at 2 MPa (pressure higher than 1,8 MPa + 10 %) and the design pressure of 2 MPa will be used in design calculations for static loadings.

4 Application of EN 13445-1 [1]

4.1 General

This part contains general information on the scope of the standard as well as terms, definitions, quantities, symbols and units which are applied throughout the standard.

Before designing and manufacturing the vessel according to the standard, the manufacturer shall verify the applicability of the standard EN 13445 and perform a number of prerequisites.

4.2 Is EN13445 applicable to the vessel?

The answer is yes, since the vessel does not belong to the vessels mentioned in Clause 1 of reference [1] which are:

- Vessels of riveted construction;
- Vessels of lamellar cast iron or any material not included in part 2, 6 or 8 of the standard;
- Multilayered, autofrettaged or pre-stressed vessels.

4.3 Warning of Annex A of reference [1]

The standard EN 13445 is harmonized under the Pressure Equipment Directive (97/23/EC). This means that if the vessel meets the requirements of this standard, it can be presumed to conform to those essential safety requirements which are listed in the Annexes ZA of each individual part.

In this connection, it should be understood that the standard is indivisible. The design and manufacturing of the vessel requires application of all relevant parts of the standard, in this case of Part 1 General [1], Part 2 Materials [2], Part 3 Design [3], Part 4 Fabrication [4] and Part 5 Inspection and testing [5], since the vessel is a steel vessel.

Part 7 and Part 9 are not mandatory parts in this sense.

4.4 Prerequisites of Annex A of reference [1]

4.4.1 Operating conditions

Operating conditions provided by the User will be used in the design calculations, but a design pressure of 2 MPa will be used in calculations for static loadings, as it is mentioned in 4.3.

4.4.2 Actions to be considered according to the list in 5.3.1 of EN 13445-3 reference [3]

a) internal pressure;

b) maximum static head of contained fluid;

c) weight of the vessel;

d) maximum weight of contents under operating conditions;

e) weight of water under hydraulic pressure test conditions;

f) wind, snow, and ice loading (not present);

g) earthquake loading (negligible);

h) other loads supported by or reacting on the vessel, including loads during transport and installation (negligible);

i) stresses caused by supporting lugs, ring, girders, saddles, internal structures or connecting piping or intentional offsets of median lines on adjacent components. (Only stresses caused by bracket supports will be considered);

j) shock loads caused by water hammer or surging of the vessel contents (not present);

k) bending moments caused by eccentricity of the centre of the working pressure relative to the neutral axis of the vessel (not present);

l) stresses caused by temperature differences including transient conditions and by differences in coefficients of thermal expansion (Not requested by the User);

m) stresses caused by fluctuations of pressure, temperature and external loads (Stresses caused by fluctuations of pressure and temperature will be considered);

n) stresses caused by the decomposition of unstable fluids (not present).

4.4.3 Classification of load cases

4.4.3.1 Normal load cases

Normal load cases are those acting on the pressure vessel during normal operation, including start-up and shutdown. They result of combination of actions mentioned in 5.3.2.

4.4.3.2 Exceptional load cases

Exceptional load cases are those corresponding to events of very low probability requiring the safe shutdown and inspection of the vessel or plant. No such exceptional load case is expected.

4.4.3.3 Testing load cases

Testing load cases include testing load cases for final assessment and testing load cases in service. Only the hydraulic test for final assessment will be considered.

4.4.4 The Category of the vessel as defined in the Pressure Equipment Directive (PED)

Taking into consideration:

— The maximum allowable pressure PS: 20 bar

— The fluid group: 1

— The volume of the vessel: 2.656 L

— The potential energy content product PS.V = 53.120 bar.L

The vessel category is IV (See Figure 3, Excerpt from Table 1 of *Annex II* of the PED where the case is represented by a red dark dot or see Figure A-1 of CR 13445-7).

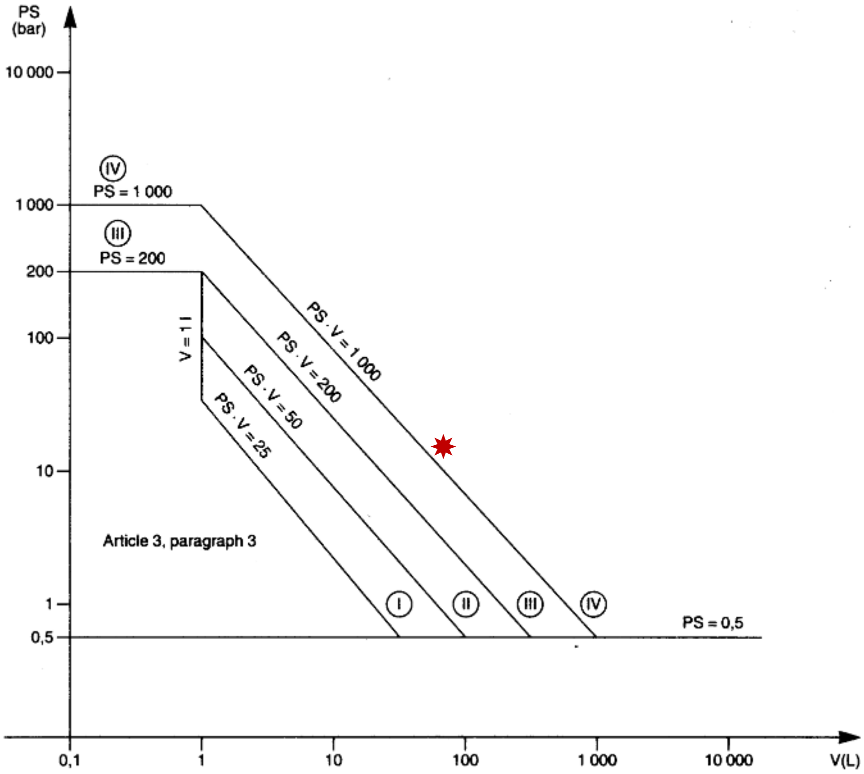


Figure 3 — Vessel category

4.4.5 The Conformity Assessment Module to be used

Applicable modules of Category IV are B+D, B+F, G, H1. Module G is used throughout this example (this is according to the drawing Example 2).

5 Application of EN 13445-2 [2]

5.1 Permitted materials

5.1.1 General

Clause **A.4 Materials** of reference [1] recalls the principles.

Specific requirements apply to materials for pressure-bearing parts. They are given in 4.1, 4.2, 4.3 and 4.4 of reference [2].

5.1.2 Requirements given in 4.1 of reference [2]

- Materials shall be selected to be compatible with anticipated fabrication steps and to be suitable for internal fluid and external environment
- Materials shall be accompanied by inspection documents in accordance with EN 10204:2004

NOTE 1 This is not properly speaking a design requirement, but a means to inspect material properties.

- Materials shall be free from surface and internal defects which can impair their intended usability
- Steels shall have a specified minimum elongation after fracture: 14 %
- Steels shall have a specified minimum impact energy measured on a Charpy V-notch impact test specimen greater or equal to 27 J for ferritic steels, etc.
- The chemical composition of ferritic steels intended for welding and forming shall not exceed 0,25 % C, 0,035 % P, 0,025 % S.

Only materials which are qualified for pressure equipment may be used. Qualification of materials can be made in three different ways:

- Materials from European harmonized Standards, see 4.3.1 of reference [2]. Certain materials supplied in accordance with European material Standards are accepted as qualified for use in pressure-bearing parts. These materials are enumerated in Table E.2-1 of reference [2].
- Materials with a European Approval for materials (EAM), see 4.3.2 of reference [2]. Materials with an EAM, which states that they can be used for products under the PED, are qualified for use in relevant products according to this standard. EAMs are published in the Official Journal, and the European Commission maintains a list of EAMs on their web site.

NOTE 2 This web site is presently accessible under the address

http://ec.europa.eu/enterprise/pressure_equipment/ped/materials/published_en.html.

- Materials with a Particular Material Appraisal (PMA), see 4.3.3 of reference [2]. Materials, which have been subject to a PMA are qualified. This appraisal is carried out by the manufacturer (and in certain cases checked by a Notified Body).

NOTE 3 The European Commission and Member States have in November 2006 agreed on "Guiding Principles for the contents of Particular Materials Appraisals". The document is published on:

http://ec.europa.eu/enterprise/pressure_equipment/ped/materials/index_en.html.

5.2 Requirements given in 4.2 of reference [2]

Materials for example 2 are high temperatures steels for which the requirements of **4.2.2 Design temperature above 20 °C** apply.

In **4.2.5** specific requirements are given for steels for **fasteners** (bolts, nuts, etc.).

5.3 Requirements given in 4.3 of reference [2]

4.3 addresses **Technical delivery conditions** for steels and welding consumables. For example 2, the **European standards** for plates, tubes, and forgings will be used. European standards will also be used for welding consumables.

Table E.2-1 of reference [2] provides an overview on materials for pressure purposes. This Table will be used for example 2.

5.4 Requirements given in 4.4 of reference [2]

4.4 addresses **Marking**. This marking ensures traceability between the product and the inspection documents.

NOTE Marking has no incidence on design calculations.

5.5 Materials selected for the vessel example 2

Characteristics of the steels, fasteners and gaskets selected for the vessel are given in Table 1 and reproduced in Annex C to this report.

P355 GH of the European harmonized standard EN 10028-2 [6] was selected for the shell (upper, lower), dished end and cone elements of the vessel (also support brackets). This steel was preferred to P295GH to have a smaller weight (approximately 15 % or 120 kg for all plate made materials) with a slightly higher price (approximate price difference in Western Europe is 100 € per metric ton).

P280 GH of the European harmonized standard EN 10222-2 was selected for the main flange upper and lower side.

Table 1 — Materials and main material characteristics in example 2

Vessel part	Material	Material designation	EN 13445-2 reference	Dimensions (mm) (see also Annex C)	Material group to CR ISO 15608	Main material characteristics				
						Tensile min/max MPa	**Min yield MPa**	**Min elong A5 at room temp**	**Min impact energy KV, J at - 20 °C**	**Min impact energy KV, J at +20 °C**
Cylindrical shell upper, lower	Ferritic steel plate for high temperature service	EN 10028-2 P355 GH (1.0473)	See Table E.2-1 of EN 13445-2	e < 16 mm e_n = 12 for lower part, e_n = 10 for upper part	1.2	510-650	355	20	27	40
Conical shell	Ferritic steel plate for high temperature service	EN 10028-2 P355 GH (1.0473)	See Table E.2-1 of EN 13445-2	e < 16 mm e_n = 12	1.2	510-650	355	20	27	40
Dished end	Ferritic steel plate for high temperature service	EN 10028-2 P355 GH (1.0473)	See Table E.2-1 of EN 13445-2	e < 16 mm e_n = 14	1.2	510-650	355	20	27	40
Main flange upper and lower side	Forging	EN 10222-2 P280 GH (1.0426)	See Table E.2-1 of EN 13445-2	50,00 < t < 160 mm e_n = 95 for lower part, e_n = 103 for upper part	1.2	490-610	280-305	22		27
Bolts(fasteners) main flange:	25CrMo4(+QT)	EN 10269 (dia.< 100 mm)	See Table E.2-1 of EN 13445-2	Number=68 M22x2,5 M22x2,5	— a)	800-950	Upper 0,2%600	15		27-32
Gasket	Spirally wound mineral filled stainless steel -Monel			Gasket parameters m=3, y=69 MPa	— a)					

Table 1 — Materials and main material characteristics in example 2 (*continued*)

Vessel part	Material group	Material designation	EN 13445-2 reference	Dimensions (mm) (see also Annex C)	Material group to CR ISO 15608	Main material characteristics				
						Tensile min/max MPa	Min yield MPa	Min elong A5 at room temp.	Min impact energy KV, J at -20°C	Min impact energy KV, J at +20°C
Nozzle N3 (DN200)	Standard XS	EN 10216-2 P265GH (1.0425)	See Table E.2-1 of EN 13445-2	Deb=219,10 eb=12,70	1.1	410-570	265[b)]	23	28	—
Nozzle N4 DN150	Standard XS	EN 10216-2 (1.0425)	See Table E.2-1 of EN 13445-2	Deb=168.3 eb=10,97	1.1	410-570	265[b)]	23	28	—
LWN Flange at N1/N2 (DN 100)	PN25	EN 10222-2 P280GH (1.0477)	See Table E.2-1 of EN 13445-2	25 bar rating 235/102,3 eb=19,85	1.2	460-580	280	21		27
Standard flange pos. at N3 (DN 200)	PN25	EN 10222-2 P280GH (1.0477)	See Table E.2-1 of EN 13445-2	25 bar rating 235/102,3 eb = 19,85	1.2	460-580	280	21		27
Standard flange pos. at N4 (DN 150)	PN25	EN 10222-2 P280GH (1.0477)	See Table E.2-1 of EN 13445-2	25 bar rating 235/102,3 eb = 28	1.2	460-580	280	21		27
Brackets and reinforce ment plates	Ferritic steel plate for high tempe-rature service	EN 10028 -2 P355 GH 1.0477 (1.0473)	See Table E.2-1 of EN 13445-2	T < 16 mm web e_n=15,base plate e_n=20 reinforcing plate e_n=10	1.2	510-650	355	20	27	40
a) not applicable b) at 100 °C										

6 Application of EN 13445-3 [3]

6.1 General

General definitions and general requirements are in Clauses 1 to 6 of **reference [3]**. Design requirements for the various components are contained in the relevant clauses of **reference [3]**. Specific design requirements for the simplified fatigue analysis are contained in Clause 17.

In this document, the principles of the calculations are presented. For details, it is recommended to examine the calculation sheets obtained by software. Each calculation sheet follows step by step the paragraphs of the relevant clause of reference [3].

The calculations sheets are gathered in pages 7 to 58 of Annex C to this report. Main results of the calculations are in pages 1 to 6.

6.2 Basic design

6.2.1 Verification of thicknesses

The first step is the verification of the thicknesses of the various components or parts composing the pressure vessel under the design loading:

P = 2 MPa

T = 260 °C

This is done successively for the cylindrical shells and their flanges, the ellipsoidal head and the attached nozzles, the conical shell, the cylindrical shell, the nozzle N4 and the brackets.

6.2.2 Determination of the maximum permissible pressure Pmax

Then the maximum permissible pressure defined in 3.16 is calculated for each component or vessel part using the formula given in the column entitled Maximum permissible pressure Pmax of Table 17-1 of reference [3].

For example, for cylindrical shells Pmax is given in Clause 7 of reference [3] by Formula (7.4.3).

As explained in 17.6.1, Pmax will be used in the fatigue calculations.

6.3 Fatigue calculations

6.3.1 General

Fatigue calculations of example 2 are performed using the formulae of Clause 17 of **reference [3]**. This is done in six steps.

6.3.2 Determination of fatigue sensitive locations

These locations are:

— Welded zones

— Unwelded zones with stress concentration

For guidance, see Table 17-1 of reference [3].

6.3.3 Determination of pseudo-elastic stress range Δσ

Δσ shall be calculated from the pressure range ΔP as follows:

$$\Delta\sigma = \frac{\Delta P}{P\max} \,.\, \eta \,.\, f \qquad (17.6\text{-}1)$$

where

- *P*max is the maximum permissible pressure of the component or vessel part under consideration as defined in Clause 4, except for dished ends where a specific definition of *P*max applies (see NOTE 2 of Table 17-1)
- f is the nominal stress of the component or vessel part under consideration, at calculation temperature.
- The value of η is obtained from Table 17-1 for each weld detail. It is an upper bound of the following ratio:

$$\frac{\text{maximum structural stress in detail under consideration under pressure } P_{\max}}{\text{nominal design stress at calculation pressure}}$$

Where Δσ > 3f , Δσ shall be increased according to the rule given in 18.8 to account for elastic-plastic cyclic conditions.

6.3.4 Stress factors *η* and associated maximum permissible pressures

Stress factors η and associated permissible pressure are given in Table 17-1 for each component or vessel part. Stress factors η depend on shape imperfections.

6.3.5 Fictitious stress range

6.3.5.1 General

The fictitious stress range is used for determination of the allowable number of cycles. It includes the thickness and temperature corrections: C_e and C_T at a welded joint or vessel part and also the effective stress concentration factor K_f for notch effect at an unwelded part. These factors are defined in 17.6.2

6.3.5.2 At a welded joint

$$\Delta\sigma^* = \left(\frac{\Delta\sigma}{C_e \cdot C_T}\right) \qquad (17.6\text{-}9)$$

6.3.5.3 At a unwelded region

$$\Delta\sigma^* = \left(\frac{\Delta\sigma}{C_e \cdot C_T}\right) \cdot K_f \qquad (17.6.10)$$

6.3.6 Determination of the allowable number of cycles

6.3.6.1 General

The allowable number of cycles is obtained by introducing $\Delta\sigma^*$ in the appropriate fatigue design curve among the curves of Figure 17-4 Total fatigue damage index.

6.3.6.2 Classification of welded joints

The welded joints shall be allocated to the classes given in Table 17-4 which are testing group dependent. For example 2, only the column testing group 3 is to consider.

6.3.6.3 Unwelded regions

For unwelded regions, the class UW fatigue design curve in Figure 17-4 applies.

6.3.7 Fatigue results

All fatigue damage index computed in Annex C are acceptable since they are less than 1. The maximum value 0,157 is reached at the longitudinal butt weld of the shell upper portion.

There is no critical area as defined in 17.2.16.

NOTE This point is important for future Non-Destructive Testing. See Clause 8 of this report.

6.3.8 Total fatigue damage index

The total fatigue damage index is calculated using Formula (17.7-1)

$$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \text{etc...} = \sum_1^k \frac{n_i}{N_i} \tag{17.7-1}$$

6.3.9 Fatigue results

All fatigue damage index computed in Annex C are acceptable since they are less than 1. The maximum value 0,157 is reached at the longitudinal butt weld of the shell upper portion.

There is no critical area as defined in 17.2.16.

NOTE This point is important for future Non-Destructive Testing. See Clause 9 of this report.

6.4 Determination of test pressures of the vessel in Annex C

6.4.1 Principle

In **Annex C**, the test pressure of the vessel is computed, as explained in Clause 10 of **EN 13445-5, reference [5].** It is the greater of:

$$P_t = 1{,}25 \cdot P_d \cdot \frac{f_a}{f_{T_d}} \tag{10.2.3.3.1-1}$$

or

$$P_t = 1{,}43 \cdot P_s \tag{10.2.3.3.1-2}$$

where:

P_t is the test pressure measured at the highest point of the chamber of the vessel in the test position;

P_d and T_d are the coincident design pressure and design temperature values for the maximum pressure load case;

P_s is the maximum allowable pressure of the vessel;

f_a is the nominal design stress for normal operating load cases of the material of the part under consideration at the test temperature;

f_{T_d} is the nominal design stress for normal operating load cases of the material of the part under consideration at temperature T_d;

Since the ratio $\frac{f_a}{f_{T_d}}$ depends on the material of the part under consideration, the value $\frac{f_a}{f_{T_d}}$ to be used for calculation of P_t shall not be less than the **smallest ratio** obtained considering the different materials of the main pressure bearing parts (e.g. shells, ends, tubesheets of heat exchangers, tube bundles, main flanges but **ignoring bolting associated to main flanges**). Main pressure bearing parts **do not include** pressure rated standard flanges and bolting designed without calculation according to the rules of 11.4.2 of EN 13445-3:2009.

6.4.2 Procedure followed in Annex C for example 2

The test pressure is determined in five steps:

6.4.2.1 Step 1

List the main components of the vessel using the criteria given above in 7.3.1 (e.g. shells, ends, etc.).

6.4.2.2 Step 2

Compute for each of them the pressure P_{t_1} using the formula $P_{t_1} = 1{,}25 \cdot P_d \cdot \frac{f_a}{f_{T_d}}$

6.4.2.3 Step 3

Compute the pressure $P_{t_2} = 1{,}43 \cdot P_s$

6.4.2.4 Step 4

Compute the test pressure of the component using the formula $P_t = \max(P_{t_1}, P_{t_2})$

6.4.2.5 Step 5

Compute the test pressure of the vessel: it is the smallest of the test pressures of the main components.

6.4.3 Results of the calculations

In page 3 of Annex C the test pressures of the main components are gathered. The test pressure of the vessel is the smallest of the test pressures of the main components.

For example 2, the test pressure of the vessel is equal to 2,95 MPa. It is limited to 2,95 MPa by the test pressure of the main flange which is greater than 2,86 MPa ($1{,}43 \cdot P_s$).

6.5 Determination of the deformation according to EN 13445-4 reference [4], Clause 9 in Annex C

The deformations of the cylindrical and conical shells and the deformation of the ellipsoidal head are computed in Annex C using formulae of Clause 9 of reference [4].

The most important deformation is in the ellipsoidal head manufactured in one piece: 21,6 %. Deformation is limited to 1 % in upper and lower cylindrical shells and to 7,4 % in the conical shell.

These values are important for heat treatment during manufacturing (see Clause 7 of this report).

6.6 Data used in example 2

6.6.1 Materials

Materials have been selected in Clause 5 of this document. For the main parts of the vessel, they belong to the material group 1.2.

A corrosion allowance c = 1 mm has been introduced for the inner wall. No corrosion allowance has been introduced for the outer wall.

6.6.2 Material nominal design stresses

Nominal design stresses have been determined from material data included in material standards using the formulae of Clause 6 of EN 13445-3, reference [3].

6.6.3 Selection of testing group and weld joint factor

Table 6.6.1-1 of EN 13445-5, **reference [5],** shows that the testing group 3B is applicable since the material group is 1.2.

For governing joints (See 5.6 of EN 13445-3), the corresponding weld joint factor is z = 0,85.

6.6.4 Design load case (determination of thicknesses), reference [3]

The following values have been used:

— The calculation pressure as defined in 5.3.10;

— The nominal design stress at calculation pressure as defined in 6.1.3;

— The analysis thickness as defied in 5.2.3;

— The joint efficiency factor z = 0,85.

NOTE As it is conservative, the possibility of using the higher yield stress Re has not been utilized in this example.

6.6.5 Operating load case (simplified fatigue analysis), reference [3]

The following parameters have been used for the calculation of the stress range $\Delta\sigma$ of Formula (17.6.1) of Clause 17 and the calculation of the fictitious stress range:

— The pressure range given in the data of example 2;

— The maximum permissible pressure as defined in 3.16, using the analysis thickness $e_a = e_n - c$;

— The nominal stress of the component at calculation temperature;

— The η factor obtained from the Table 17-1 for the weld detail or vessel part taking in consideration the tolerances which shall not exceed those permitted in **reference [4].**

6.6.6 Testing load case for final assessment, reference [5]

The following parameters have been used for the calculation of the test pressure:

— The nominal design stress of the material at test temperature;

— The nominal thickness (no corrosion allowance);

— The joint efficiency factor z = 1.

6.7 Conditions of applicability of calculations

The validation of the fatigue calculations necessitates the respect of the following limitations:

— Design requirements for welded joints of Annex A of reference [3] are satisfied (see Clause 7 of this document);

— Manufacturing tolerances of Clause 5 of reference [4] are satisfied (see Clause 7 of this document);

— For weld seams, the Manufacturer shall assume certain tolerances and derive the corresponding stress factors to be used for fatigue assessment (See Table 17-1.). Then the assumed tolerances should be checked and guaranteed after manufacturing (see 17.4.6 of reference [3]).

— Testing requirements given in reference [5] are satisfied (see Clause 8 of this document).

7 Application of EN 13445-4, reference [4]

7.1 General

Most of the requirements for manufacturing are to be found in EN 13445-4:2009 for pressure vessels and vessel parts made from steel.

7.2 Material traceability

The vessel manufacturer shall have and maintain an identification system for materials used in fabrication.

NOTE This is without influence on design calculations.

7.3 Manufacturing tolerances

7.3.1 Principle

For example 2, the designer shall verify that the tolerances used in design calculations are less than the maximum manufacturing tolerances permitted by Clause 5 of reference [4] for **dynamic and cyclic loads**.

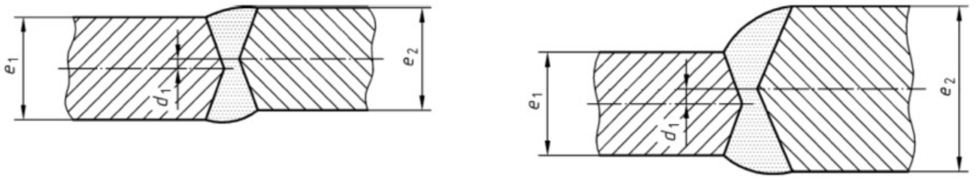
These tolerances can be inferior to those for predominantly non cyclic loads. This is the case for peaking on longitudinal butt welds.

7.3.2 Surface geometry of welds, as in 5.1 of reference [4]

The surface geometry of welded butt and fillet joints shall meet the requirements of EN 13445-5, **reference [5]**.

7.3.3 Middle line alignment, as in 5.2 of reference [4]

Middle line alignment is represented on the Figure 4.



a) Middle line alignment d_1 at equal thickness $e_1 = e_2$ b) Middle line alignment d_1 at different thickness $e_1 \leq e_2$

Figure 5.2-1 — Middle line alignment d_1

Figure 4 — Abstract from EN 13445-5:2009

For longitudinal welds in the cylindrical shells and the conical shell, the maximum misalignment given by Table 5.2.1 is $e_1/10$.

For example 2, this means a maximum misalignment of 1,2 mm for the cylindrical lower shell. The value used in the fatigue calculation was **0,60 mm**.

For circular welds in the cylindrical shells and the conical shell, the maximum misalignment given by Table 5.2.3 is $e_1/10 + 1$.

For example 2, this means a maximum misalignment of 2,2 mm for the cylindrical lower shell. The value used in the fatigue design calculations was **0,60 mm.**

NOTE A similar approach has been applied to the other shells.

7.3.4 Surface alignment, as in 5.3 of reference [4]

The transition across the weld between parts of the same thickness shall be smooth and gradual with a slope of 1 in 4 over the width of the weld.

Where different thicknesses are being joined, a taper shall be produced in accordance with **Annex A** of **EN 13445-3, reference [3]**

NOTE In the calculations, it was supposed that these transitions had been done.

7.3.5 Tolerances for vessels subjected to internal pressure, as in 5.4 of reference [4]

7.3.5.1 External diameter, as in 5.4.1 of reference [4]

For cylindrical shells the mean external diameter derived from the circumference shall not deviate by more than 1,5 % from the specified external diameter.

NOTE In the calculations, it was supposed that this condition was met.

7.3.5.2 Out of roundness, as in 5.4.2 of reference [4]

Out of roundness (O) shall be calculated in accordance with the following Formula (5.4-1):

$$O\,[\%] = \frac{2 \cdot (D_{\max} - D_{\min})}{D_{\max} + D_{\min}} \cdot 100 \qquad (5.4\text{-}1)$$

It shall not exceed the following values:

a) 1,5 % for the ratio of e/D < 0,01;

b) 1,0 % for the ratio of $e/D \geq$ 0,01.

For example 2, we have $e\,/\,D$ = 0,0107 for the lower cylindrical shell. Thus the out of roundness of this cylindrical shell not exceed 1 %. The fatigue calculation was performed with O = 0,75 %.

NOTE A similar approach has been applied to the other shells.

7.3.5.3 Deviation from the longitudinal axis, as in 5.4.3 of reference [4)

The deviation from the longitudinal axis over the length of the cylindrical portion of the pressure vessel shall not exceed 0,5 % of the length of the shell.

Calculations were performed in assuming that the deviation from the longitudinal axis met this condition.

7.3.5.4 Irregularities in profile, as in 5.4.4 of reference [4]

a) Irregularities in profile

Irregularities in profile (e.g. dents, buckling, flats on nozzle positions) shall be smooth and the depth shall be checked by a 20° gauge and shall not exceed the following values:

1) 2 % of the gauge length; or

2) 2,5 % of the gauge length provided that the length of the irregularities does not exceed one quarter of the length (with a maximum of 1 m) of the shell part between two circumferential joints.

Greater irregularities require proof by calculation or strain gauge measurement that the stresses are permissible.

Calculations were performed in assuming that irregularities in profile did not exceed the aforementioned values.

b) Peaking on longitudinal butt welds

When irregularity in the profile occurs at the welded joint and is associated with **"flats"** adjacent to the weld, the irregularity in profile or peaking shall not exceed the values given Tables 5.4-1 and 5.4-2.

For example 2, only Table 5.4-2 **"Maximum permitted peaking P in longitudinal welds for dynamic and cyclic loads"** is to consider, see Table 2.

Table 2 — Table 5.4-2 of reference [4] — Maximum permitted peaking P in longitudinal welds for dynamic and cyclic loads

Dimensions in millimetres

Vessel wall thickness e	Maximum permitted peaking P
$e \leq 3$	1,5
$3 \leq e < 6$	2,5
$6 \leq e < 9$	3,0
$9 \leq e$	the lesser of e/3, or 10 mm

For example 2, This table gives a maximum permitted peaking of **4 mm** for the lower cylindrical shell. A value of **2 mm** was used in the calculations.

NOTE A similar approach has been used for the other shells.

7.3.5.5 Local thinning, as in 5.4.5 of reference [4]

Local thinning means local areas of thickness below the values (e+c) where e is the required thickness and c is the corrosion allowance. Local thinning shall be permissible without further calculation provided all of the conditions of 5.4.5 are fulfilled.

In the calculations for example 2, it was supposed that these conditions were fulfilled.

7.3.5.6 Dished ends, as in 5.4.6 of reference [4]

The dished end of example 2 shall be aligned with the tolerances specified in Table 5.4-3, except that the crown radius shall not be greater than that specified in the design and the knuckle radius shall not be less than the values specified in the design.

In the calculations for example 2, it was supposed that these conditions were fulfilled.

7.3.6 Tolerances for vessels subjected to external pressure, as in 5.5 of reference [4]

This is not applicable to example 2 since the vessel is only subjected to internal pressure.

7.3.7 Structural tolerances, as in 5.6 of reference [4]

Structural tolerances, other than those specified in 5.4 and 5.5 of **reference [4]** should not exceed the values recommended in Annex A of **reference [4]**.

In the calculations for example 2, it was supposed that these conditions were fulfilled.

7.4 Welding, as in 7 of reference [4]

7.4.1 General

Only two paragraphs are applicable to example 2: General and Vessels or parts made of more than one course. Lapped joints, joggle joints, permanent backing strips have not been used.

For reference, see **Annex A** to this document – **Drawing of Example 2**.

7.4.2 General, as in 7.1 of reference [4]

The manufacturer in selecting an appropriate weld detail should give consideration to:

a) the method of manufacture;

b) the service conditions (e.g. corrosion);

c) the ability to carry out the necessary non-destructive testing (NDT) required in accordance with EN 13445-5, **reference [5];**

d) the design requirements given in 5.7 and in **Annex A** of EN 13445-3, **reference [3]** for welds.

7.4.3 Vessels or parts made of more than one course

When a vessel or vessel part is made of two or more courses the longitudinal weld joints of adjacent courses shall be staggered by 4.*e* with 10 mm minimum.

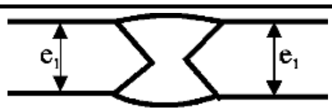
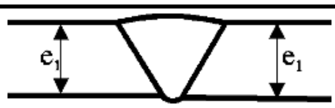
7.4.4 Application to example 2

For example 2, longitudinal welds in cylinders and cones will be full penetration butt welds made from both sides, according to reference M 1 of Table A-1 of Annex A of **reference [3]**. These welds, accessible for NDT, allow fatigue and normal service conditons (e.g. for corrosion). See Table 3 of this report.

Fatigue **class 63** for testing group 3 is given in **Table 18-4 or Table 17-4** for stress range of details n° 1.1 and 1.2. This fatigue class was used in the design calculations.

As represented in **Annex A** to this document, the longitudinal joint of the lower cylindrical shell and the longitudinal joint of the conical shell have been staggered by more than 4.*e* (48 mm).

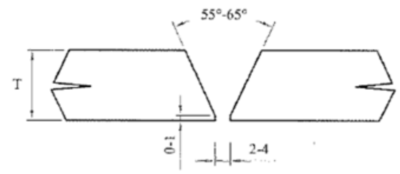
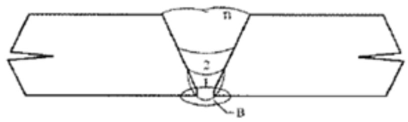
Table 3 — Excerpt of Table A-1 of reference [3] — Pressure bearing welds - Longitudinal welds in cylinders and cones, welds in spheres and dished ends

Ref.	Type of joints	Design requirements	Applicable weld testing group	Fatigue class 1)	Lamellar tearing susceptibility 2)	Corrosion 3)	EN 1708 -1:1998
M 1			1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.2	A	N	1.1.4
M 10		allowed for fatigue only if full penetration can be verified at least by visual inspection	1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.5	A	N	1.1.1

1) Fatigue class: see clauses 17 and 18.
2) Lamellar tearing susceptibility: A = no risk B = possible risk.
3) Corrosion N = normal conditions S = not permitted.

It would be also possible to use full penetration butt welds made from one side, according to reference **M 10** of Table A-1. In this case the corresponding fatigue class given in Table 18-4 or 17-4 would be **40**. But if special manufacturing procedures were used to obtain a full penetration, the fatigue class could be increased up to **63**.

Table 4

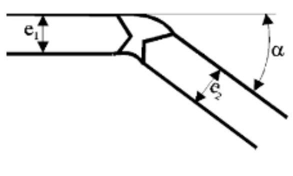
<table>
<tr><td>Groove design</td><td>Welding sequence</td></tr>
<tr><td>
Preheat >15 °C/Interpass temperature max. 250 °C</td><td>
GMAW (tube cup dia 16 mm)/ B= Backweld (not mandatory)
Shielding gas Ar/CO_2 85/15, flow12-20 l/min/No gas backing</td></tr>
<tr><td></td><td>Technique: String bead/ Initial and interpass cleaning: brushing
<table>
<tr><td>Weld layer</td><td>process</td><td>Filler/dia</td><td>Amp</td><td>volt</td><td>Speed cm/min</td></tr>
<tr><td>1</td><td>GMAW</td><td>ER70S-6/1-1,2</td><td>130-180</td><td>17-20</td><td>20-40</td></tr>
<tr><td>2-n</td><td>GMAW</td><td>ER70S-6/1-1,2</td><td>130-270</td><td>24-32</td><td>20-55</td></tr>
<tr><td>B*</td><td>GMAW</td><td>ER70S/1-1,2-6</td><td>130-270</td><td>24-32</td><td>20-55</td></tr>
<tr><td colspan="6">*if welding grind to sound metal+PT</td></tr>
</table></td></tr>
</table>

Circular welds in cylinders and cones, connecting weld between dished end and shell, connecting weld between lower cylindrical shell and conical shell will be also full penetration butt welds. They are of reference C1 or C11 in Table A-2 of Annex A of **reference [3]**, except for the welds at the ends of the conical shell which are of reference C17. See Table 5 of this report.

For C 17 the condition on thicknesses is satisfied.

Relevant fatigue is **63 except for the welds at the ends of the conical shell where it is 56** since the angle α of the cone is equal to 45°.

Table 5 — Excerpt of Table A-2 of reference [3] - Pressure bearing welds - Circumferential welds in cylinders, cones and dished ends

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1)]	Lamellar tearing susceptibility [2)]	Corrosion [3)]	EN 1708-1:1998
C 1			1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.2	A	N	1.1.4
C 11		allowed for fatigue only if full penetration can be verified	1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.5	A	N	1.1.1
C 17	$\alpha > 30°$ 	in case of unequal thicknesses, limited to: $e_2 - e_1 \leq \mathrm{Min}\lfloor 0{,}3e_1 \; ; 4 \rfloor$ — calculation of stresses — round the weld inside by grinding	1, 2, 3, 4	see Table 18-4 detail n° 1.4	A	N	-

For example 2, the same approach has been applied to:

— nozzles;

— welded neck flanges;

— welded flanges;

— welds of brackets.

7.5 Welding, as in 8 of reference [4]

7.5.1 Welding procedure specifications

The manufacturer shall compile welding procedure specifications, in accordance with EN ISO 15609-1:2004 for all welds.

7.5.2 Welding procedure qualification record (WPQR)

Welding procedure specifications to be used in production shall be qualified to an appropriate WPQR.

For example 2 this shall be achieved by performing welding procedure qualification tests in accordance with EN ISO 15614-1:2004.

7.5.3 Qualification of welders and welding operators

Welders and welding operators shall be approved to EN ISO 9606 or EN 1418:1997 respectively.

NOTE Recently EN ISO 9606 replaced EN 267-1.

7.6 Manufacture and testing of welds – Production test, as in 8 of reference [4]

7.6.1 Required number of production tests

The amount of required production tests is specified in **8.2.**

For example 2, with a joint coefficient of 0,85, the amount is one test plate per 200 m of longitudinal welds.

After 10 consecutive test plates have successfully passed the tests, testing may be reduced to the following: one test plate per 1 500 m of longitudinal welds.

7.6.2 Extent of testing

Extent of testing is given in **Table 8.3.1**

For example 2, material group is 1.2 and thickness less or equal to 12 mm. Test specimens are:

— 1 **FB** (One Face Bend test to EN 910:1996)

— 1 **RB** (One Root Bend test to EN 910:1996)

— 1 **MA** (Macro examination to EN 1321:1996)

7.6.3 Performance of tests and acceptance criteria

7.6.3.1 Bend test

The testing and the acceptance criteria shall conform to EN ISO 15614-1:2004.

7.6.3.2 Macro-examination

The testing and the acceptance criteria shall conform to EN ISO 15614-1:2004.

The macro examination shall show sound build-up of beads and sound penetration.

7.7 Forming of pressure parts, as in 9 of reference [4]

7.7.1 Ratio of deformation, as in 9.2 of reference [4]

Formula (9.2-1) has been used for the calculation of the deformation of the elliptic head of example 2.

Formula (9.2-2) has been used for the calculation of the deformation of the cylindrical and conic shells.

Results of calculations are given in Annex C to this report. See also 7.5 of this report.

7.7.2 Cold forming, as in 9.3.1 of reference [4]

For example 2, cold forming of shells may be carried out at temperature below the maximum permissible temperature for stress relieving minus 30 °C.

7.7.3 Hot forming, as in 9.3.2 of reference [4]

Hot forming of the elliptical head is recommended at temperature above stress relieving temperature, usually at normalizing temperature, above 350 °C but below 720 °C.

7.8 Post weld heat treatment (PWHT), as in 10 of reference [4]

For example 2, no PWHT is necessary.

8 Application of EN 13445-5, reference [5]

8.1 Generality

EN 13445-5, reference [5], deals with Inspection and Testing. In this Clause 8 only non-destrctive testing (NDT) requirements will be developed. Inspection requirements will not be mentioned because they have no impact on the design.

8.2 Non destructive testing, as in 4.3 of reference [5]

The type and amount of non-destructive testing of a pressure vessel shall be based upon the testing group or combination of testing groups when permitted in 6.6.1.2 (see Table 6.6.1-1: testing groups for steel pressure vessels and Table 6.6.2-1: extent of non-destructive testing).

For example 2, a unique testing group 3 is used for the whole vessel. More precisely the testing subgroup 3b which is compatible with the steels of the material group 1b will be used. See Table 6.6.1-1.

8.3 Determination of extent of non-destructive testing, as in 6.6.2 of reference [5]

Table 6.6.2.2 of reference [5] applies to all joints of example 2.

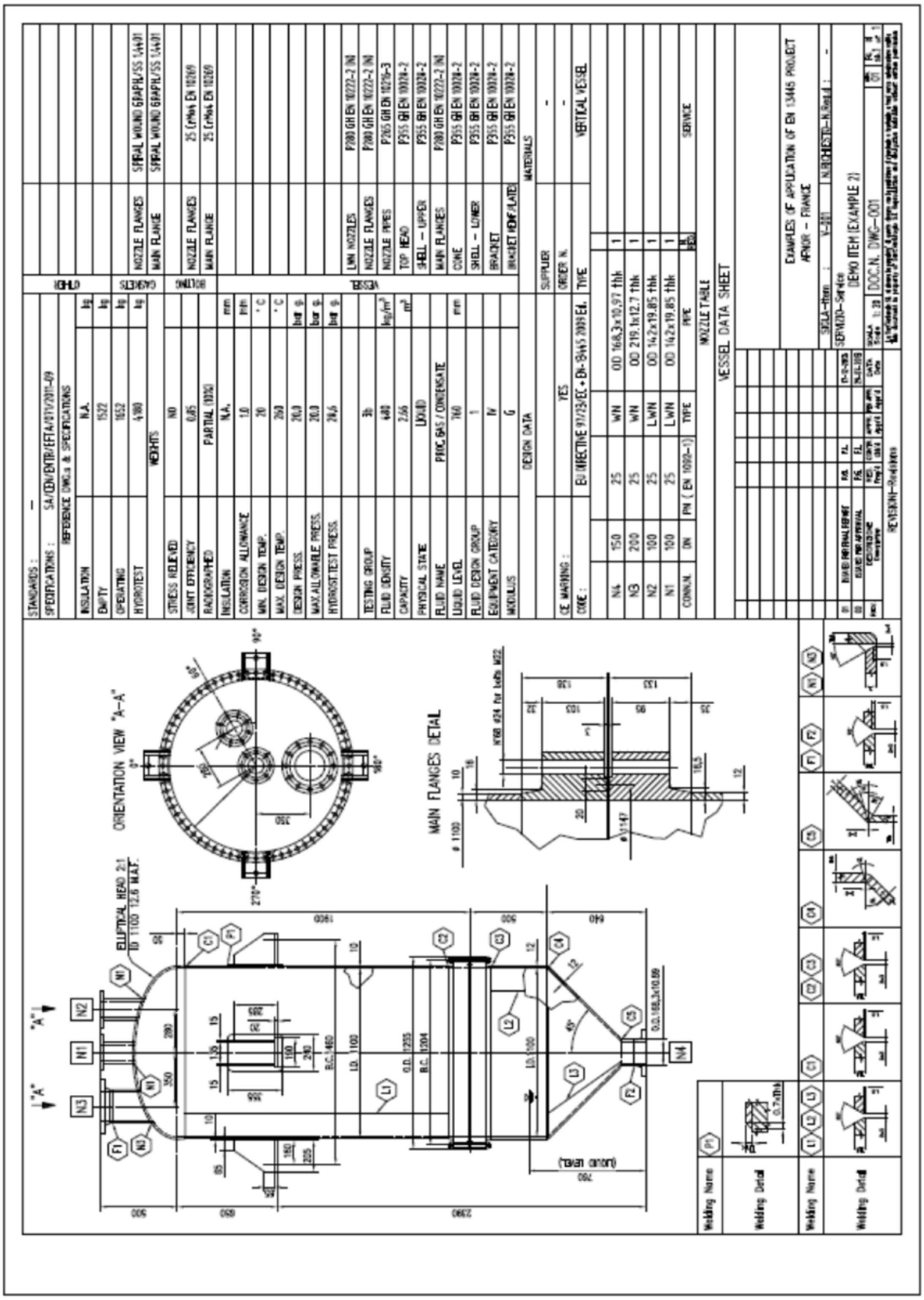
The Non-destructive methods included in the Table are:

RT = Radiographic testing, **UT** = Ultrasonic testing, **MT** = Magentic particle testing, **PT** = Penetrant testing

For example 2, UT and PT will be used.

Annex A
(informative)

Drawing of example 2



Annex B
(informative)

Nameplate of example 2

See EN 13445-5, 10.1.1.

Nameplates of material suitable for the intended service and with a thickness sufficient to withstand distortion due to the application of the marking and be compatible with the method of attachment. The minimum thickness shall be not less than 1 mm. Marking shall be done in characters not less than 5 mm high and shall be produced by casting, etching, embossing, debossing, stamping or engraving, including the identification of EN 13445.The nameplate shall be attached in such a way that removal would require the wilful destruction of the nameplate or its attachment system.The nameplate shall remain visible and legible for the lifetime of the vessel.

The units of measurement used in marking or stamping the equipment and accessories shall follow the SI units. The unit “bar” for pressure shall be used.

(Registration Number of the Notified Body)

CE 1234

Type	Standard: EN13445:2009 issue 5
Serial Number	
Max working pressure PS	20 bar
Max/Min working temperature TSmax/TSmin	260 °C / 20 °C
Fluid	Group I
Test pressure	29,5 bar
Year of manufacture	2014
Made in	(Country)
Volume	2656 litre

(Name or logo and address of manufacturer)

Annex C
(informative)

Design calculation of example 2

C.1 General

Calculation report

EN 13445 Ed. 2009 Issue 5

Internal design pressure	*P* =	2,00 MPa
External design pressure	*P Ext* =	0 MPa
Internal design temperature	*T* =	260 °C
External design temperature	*T Ext* =	20 °C
Internal corrosion allowance	*c* =	1,00 mm
External corrosion allowance	*ce* =	0 mm
Joint efficiency	*z* =	0,85
Minimum design temperature	=	20 °C

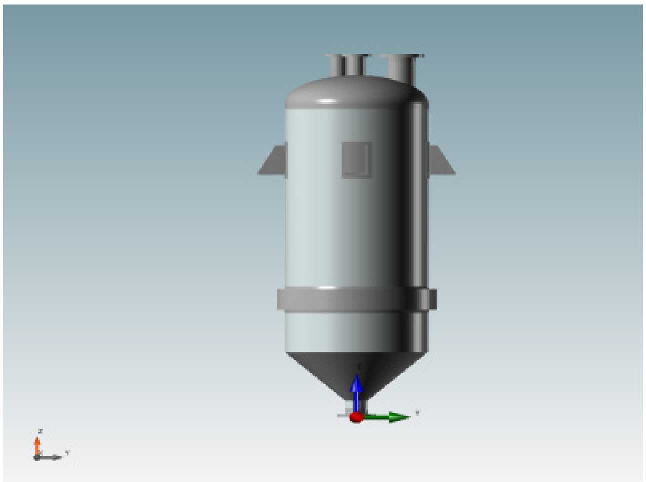


Figure C.1

C.2 Test pressure

All pressures in MPa

Component	*Static head (design)*	*Static head (test)*	*Min f0/f*	*1.25·P·f0/f*	*1.43·P*	*Max test pressure*
SHELL_ LOWER PORTION	0,0005	0,03	1,285	3,21	2,86	3,21
MAIN FLANGE_LOWER SIDE	0	0,02	1,181	2,95	2,86	2,95
SHELL_ UPPER PORTION	0	0,02	1,285	3,21	2,86	3,21
MAIN FLANGE UPPER SIDE	0	0,02	1,181	2,95	2,86	2,95
TOP HEAD	0	0,005	1,285	3,21	2,86	3,21
CONICAL SHELL	0,003	0,03	1,285	3,21	2,86	3,21
NOZZLE N4 (DN150)	0,003	0,03	1,529	3,82	2,86	3,82

Item design pressure P = 2,00 MPa
Item MAWP (Hot & Corroded conditions) = 2,00 MPa (limited by MAIN FLANGE_LOWER SIDE)
Item MAP (New & Cold conditions) = 2,29 MPa (limited by MAIN FLANGE_LOWER SIDE)
Item Lowest Stress Ratio = 1,181
Item test pressure = Pt=max(Pt1,Pt2) = 2,95 MPa

C.3 Maximum Pressures

All pressures in Mpa

Component	*MAP N&C*	*MAWP H&C*	*MAEP N&C*	*MAEWP H&C*
SHELL_ LOWER PORTION	7,27	2,78		
MAIN FLANGE_LOWER SIDE	2,29	2,00		
SHELL_ UPPER PORTION	5,88	2,20		
MAIN FLANGE UPPER SIDE	2,29	2,02		
TOP HEAD	7,66	3,43		
CONICAL SHELL	3,73	2,08		
NOZZLE N4 (DN150)	30,49	12,03		

C.4 Weights

Component	*Dead*	*Live*	*Liquid*	*Full of water*	*Operating*
SHELL_ LOWER PORTION	125 kg	0 kg	51 kg	484 kg	176 kg
MAIN FLANGE_LOWER SIDE	205 kg	0 kg	0 kg	329 kg	205 kg
SHELL_ UPPER PORTION	472 kg	0 kg	0 kg	2 111 kg	472 kg
MAIN FLANGE UPPER SIDE	217 kg	0 kg	0 kg	345 kg	217 kg
TOP HEAD	158 kg	0 kg	0 kg	380 kg	158 kg
LWN Flange N1 - DN100 PN25	19 kg	0 kg	0 kg	19 kg	19 kg
LWN Flange N2 - DN100 PN25	21 kg	0 kg	0 kg	21 kg	21 kg
NOZZLE N3 (DN 200)	19 kg	0 kg	0 kg	24 kg	19 kg
STD Flange N3 - DN200 PN25	20 kg	0 kg	0 kg	23 kg	20 kg
CONICAL SHELL	128 kg	0 kg	78 kg	302 kg	206 kg
NOZZLE N4 (DN150)	4 kg	0 kg	1 kg	5 kg	5 kg
STD Flange N4 - DN150 PN25	14 kg	0 kg	1 kg	15 kg	15 kg
SUPPORT BRACKET	119 kg	0 kg	0 kg	119 kg	119 kg
Totals:	1 521 kg	0 kg	131 kg	4 177 kg	1 652 kg

C.5 Bill of materials

Component	Dimensions	Material
SHELL_ LOWER PORTION	Id = 1 100,00 mm, Od = 1 124,00 mm, Tk = 12,00 mm, L = 377,25 mm	P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate
MAIN FLANGE_LOWER SIDE - Flange	Id = 1 100,00 mm, Od = 1 255,00 mm, Tk = 95,00 mm	P280GH (NT,QT) (EN 10222-2:2001) 50,001 ≤ t ≤ 160 - Forging
MAIN FLANGE_LOWER SIDE - Gasket	Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)	
MAIN FLANGE_LOWER SIDE - Bolts	68 x ISO M22 x 2,50	25CrMo4 (EN 10269:2009) t ≤ 100,00 mm - Bolting
SHELL_ UPPER PORTION	Id = 1 100,00 mm, Od = 1 120,00 mm, Tk = 10,00 mm, L = 1 724,25 mm	P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate
MAIN FLANGE UPPER SIDE - Flange	Id = 1 100,00 mm, Od = 1 255,00 mm, Tk = 103,00 mm	P280GH (NT,QT) (EN 10222-2:2001) 50,001 <= t <= 160 - Forging
TOP HEAD	Id = 1 100,00 mm, Od = 1 128,00 mm, Ratio = 2, Tk = 14,00 mm	P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate
LWN Flange N1 - DN100 PN25 - Flange	Id = 102,30 mm, Od = 235,00 mm, Tk = 19,85 mm	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging
LWN Flange N1 - DN100 PN25 - Gasket	Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)	
LWN Flange N1 - DN100 PN25 - Bolts	8 x ISO M20 x 2,50	25CrMo4 (EN 10269:2009) t ≤ 100,00 mm - Bolting
LWN Flange N2 - DN100 PN25 - Flange	Id = 102,30 mm, Od = 235,00 mm, Tk = 19,85 mm	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging
LWN Flange N2 - DN100 PN25 - Gasket	Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)	
LWN Flange N2 - DN100 PN25 - Bolts	8 x ISO M20 x 2,50	25CrMo4 (EN 10269:2009) t ≤ 100,00 mm - Bolting
NOZZLE N3 (DN 200)	Standard 200 XS pipe	P265GH (EN 10216-2:2008) t ≤ 16,00 mm - Seamless tube
STD Flange N3 - DN200 PN25 - Flange	Id = 193,70 mm, Od = 360,00 mm, Tk = 30,00 mm	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging
STD Flange N3 - DN200 PN25 - Gasket	Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)	
STD Flange N3 - DN200 PN25 - Bolts	12 x ISO M24 x 3,00	25CrMo4 (EN 10269:2009) t ≤ 100,00 mm - Bolting
CONICAL SHELL	Min Id = 150,00 mm, Max Id = 1 100,00 mm, Tk = 12,00 mm, α = 45,00 °, L = 475,00 mm	P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate
NOZZLE N4 (DN150)	Standard 150 XS pipe	P265GH (EN 10216-2:2008) t ≤ 16,00 mm - Seamless tube
STD Flange N4 - DN150 PN25 - Flange	Id = 146,36 mm, Od = 300,00 mm, Tk = 28,00 mm	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging
STD Flange N4 - DN150 PN25 - Gasket	Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)	
STD Flange N4 - DN150 PN25 - Bolts	8 x ISO M24 x 3,00	25CrMo4 (EN 10269:2009) t ≤ 100,00 mm - Bolting
SUPPORT BRACKET - Bolts	4 x ISO_TEMA M20 x 2,50	25CrMo4 (EN 10269:2009) t ≤ 1006,00 mm - Bolting

C.6 Nozzle connections

Name	***Flange***	***Material***	***OD***	***Tk***
LWN Flange N1 - DN100 PN25	100 LWN 25 EN1092_1	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm	142,00 mm	19,85 mm
LWN Flange N2 - DN100 PN25	100 LWN 25 EN1092_1	P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm	142,00 mm	19,85 mm
NOZZLE N3 (DN 200)	200 WN 25 EN1092_1	P265GH (EN 10216-2:2008) t ≤ 16,00 mm	219,10 mm	12,70 mm
NOZZLE N4 (DN150)	150 WN 25 EN1092_1	P265GH (EN 10216-2:2008) t ≤ 16,00 mm	168,30 mm	10,97 mm

C.7 Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition, component, detail	**Required cycles**	**Allowable cycles**	**Damage index**
1, SHELL_ LOWER PORTION, Longitudinal butt weld	1200	18595	0,065
1, SHELL_ LOWER PORTION, Circumferential butt weld	1200	555932	0,002
1, MAIN FLANGE_LOWER SIDE, Junction to shell (of thickness es)	1200	103455	0,012
1, MAIN FLANGE_LOWER SIDE, Hub to plate junction	1200	254488	0,005
1, SHELL_ UPPER PORTION, Longitudinal butt weld	1200	7658	0,157
1, SHELL_ UPPER PORTION, Circumferential butt weld	1200	209109	0,006
1, MAIN FLANGE UPPER SIDE, Junction to shell (of thickness es)	1200	106553	0,011
1, MAIN FLANGE UPPER SIDE, Hub to plate junction	1200	266197	0,005
1, TOP HEAD, all butt welds	1200	220675	0,005
1, TOP HEAD, Knuckle region weld	1200	44440	0,027
1, LWN Flange N1 - DN100 PN25, Hub to plate junction	1200	370582	0,003
1, LWN Flange N1 - DN100 PN25, Nozzle without pad weld	1200	68661	0,017
1, LWN Flange N2 - DN100 PN25, Hub to plate junction	1200	370582	0,003
1, LWN Flange N2 - DN100 PN25, Nozzle without pad weld	1200	68661	0,017
1, NOZZLE N3 (DN 200), Circumferential butt weld	1200	38155257	0,000
1, NOZZLE N3 (DN 200), Nozzle without pad weld	1200	83324	0,014
1, STD Flange N3 - DN200 PN25, Junction to shell (of thickness es)	1200	130725	0,009
1, STD Flange N3 - DN200 PN25, Hub to plate junction	1200	370582	0,003
1, CONICAL SHELL, Longitudinal butt weld	1200	7837	0,153
1, CONICAL SHELL, Circumferential butt weld	1200	177213	0,007
1, NOZZLE N4 (DN150), Circumferential butt weld	1200	Unlimited	0,000
1, STD Flange N4 - DN150 PN25, Junction to shell (of thickness es)	1200	130184	0,009
1, STD Flange N4 - DN150 PN25, Hub to plate junction	1200	367971	0,003
1, SUPPORT BRACKET, Bracket or support weld	1200	53271	0,023

Allowable number of cycles: 7658 (limited by Load condition 1, SHELL_ UPPER PORTION, Longitudinal butt weld)

C.8 Cylindrical shell - SHELL_ LOWER PORTION

According to: EN 13445 Ed. 2009 Issue 5, Clauses 7 and 8

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	0,85

Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	212,50 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	338,10 MPa

Geometry

Inside diameter	Di =	1 100,00 mm
Outside diameter	De =	1 124,00 mm
Length	L =	377,25 mm
Nominal thickness	en =	12,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0,0005 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	1 102,00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	8,90 mm
		en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	7,27 MPa
Maximum allowable design pressure	=	2,78 MPa

Deformation according to EN13445-4 Clause 9

Deformation	F=50·en/(Di/2+en/2) =	1,079 %

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula (10.2.3.3.1-1)	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula (10.2.3.3.1-2)	Pt2 =1 ,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt = max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,03 MPa
Calculation pressure	Pc = Pt+Pht =	2,98 MPa
Inside diameter	Di' = Di+2δ =	1 100,00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	4,87 mm
		en ≥ e: Ok

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	165,33 MPa
Ultimate tensile strength at room temperature	Rm =	510,00 MPa
Yield strength at design temperature	Rp0,2/T =	248,00 MPa

Load condition 1, Longitudinal butt weld

Maximum allowable pressure (component)	Pmax =	2,78 MPa
Nominal thickness	en =	12,00 mm
Inside diameter	Di =	1 100,00 mm
Offset	δo =	0,60 mm
Peeking or flat	δpf =	2,00 mm
Ovality	u =	0,75 %
Partial stress factor	η1=(3·δo)/en =	0,15000
Partial stress factor	η2=1,5·u·(Di/en) =	1,03125
Partial stress factor	η4=6·δpf/en =	1,00000
Stress factor	η=(1+η1+η2+η4)·z =	2,70406
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	281,70 MPa
Equivalent number of full pressure cycles	Neq =	300,21128
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	299,68 MPa
Number of allowable fatigue cycles	N =	18 595
Partial fatigue damage index	D=Nreq/N =	0,06453

Load condition 1, Circumferential butt weld

Joint efficiency	z =	0,85000
Maximum allowable pressure (component)	Pmax =	2,78 MPa
Calculation thickness	en =	12,00 mm
Joint efficiency	z =	0,85000
Offset	δ =	0,60 mm
Partial stress factor	η1=δ/(2·en) =	0,02500
Stress factor	η=(1+η1)·z =	0,87125
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	90,77 MPa
Equivalent number of full pressure cycles	Neq =	300,21128
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*= 0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	96,56 MPa
Number of allowable fatigue cycles	N =	555 933
Partial fatigue damage index	D=Nreq/N =	0,00216

Fatigue cycles and damage index summary

Load 1, partial damage index for Longitudinal butt weld	=	0,06453
Total damage index: Longitudinal butt weld	=	0,06453
Load 1, partial damage index for Circumferential butt weld	=	0,00216
Total damage index: Circumferential butt weld	=	0,00216
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.9 Welding neck flange - MAIN FLANGE_LOWER SIDE

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Flange material	**P280GH (NT,QT) (EN 10222-2:2001) 50,001 ≤ t ≤ 160- Forging**
Shell material	**P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate**
Bolting material	**25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**
Gasket	**Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**

Allowable stresses	**Flange - f**	**Hub - fH**	**Bolting - fB**
Design condition	121,33 MPa / 17 597,9 psi	165,33 MPa / 23 979,6 psi	128,73 MPa / 18 671,2 psi
Seating condition	143,33 MPa / 20 788,7 psi	212,50 MPa / 30 820,5 psi	146,67 MPa / 21 272,2 psi
Test condition	204,76 MPa / 29 698,2 psi	338,10 MPa / 49 036,6 psi	220,00 MPa / 31 908,3 psi

Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	2,00 MPa
Calculation temperature	T =	260 °C
Corrosion allowance	c =	1,00 mm
Flange external diameter	A =	1 255,00 mm
Inside diameter	B =	1 100,00 mm
Inside diameter (corroded)	B* = B + 2c =	1 102,00 mm
Bolt circle	C =	1 204,00 mm
Flange thickness	e =	95,00 mm
Mean gasket diameter	Gmean =	1 147,00 mm
Hub length	h =	35,00 mm
Thickness of hub at back of flange	g1 =	18,50 mm
Thickness of hub at back of flange (corroded)	g1* =	17,50 mm
Thickness of hub at small end	g0 =	12,00 mm
Thickness of hub at small end (corroded)	g0* =	11,00 mm

Gasket parameters

Gasket factor	m =	3,00
Minimum gasket seating pressure	y =	69,00 MPa
Gasket contact width	w =	20,00 mm
Basic gasket seating width	b0 = w / 2 =	10,00 mm
Effective gasket seating width	b = 2,52·√(b0) =	7,97 mm
Diameter of gasket load reaction	G = Gmean + w - 2b =	1 151,06 mm

Bolt loads

Number of bolts	=	68
Bolt type	=	ISO M22 x 2,50
Root area of one bolt	=	282,0 mm²
Distance between centre lines of adjacent bolts	δb =	55,62 mm
Bolt outside diameter	db =	22,00 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	2 081 217 N
Compression load on gasket to ensure tight joint	HG = 2π·G·b·m·P =	345 804 N
Minimum required bolt load for operating condition	Wop = H + HG =	2 427 022 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	3 099 036 N
Minimum required bolt load for the test condition	Wt = Ht + 2b·π·G·m·Pt =	3 613 956 N
Minimum required bolt load for assembly condition	WA = πb·G·y =	1 988 375 N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}}; \frac{W_{op}}{f_B}; \frac{W_t}{f_{B,t}}\right]$ =	18 853,1 mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	19 176,0 mm²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$ =	68 049,7 mm²
Design bolt load for assembly condition	$W = 0{,}5(A_{B,\min} + A_B) f_{B,A}$ =	2 788 800 N
		B ≥ AB,min: Ok

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}};1\right]$	=	1,00000
Ratio of the flange diameters	K = A / B*	=	1,13884
Length parameter	$l_0 = \sqrt{B^* \cdot g_0{}^*}$	=	110,10 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^{*2}P\right)$	=	1 907 581 N
Hydrostatic end force due to pressure on flange face	HT = H - HD	=	173 636 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2	=	42,25 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2	=	26,47 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B* - G) / 4	=	38,73 mm
Flange stress factor	$\beta_T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.0472+1.9448K^2)(K-1)}$	=	1,86292
Flange stress factor	$\beta_U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=	16,44915
Flange stress factor	$\beta_Y = \frac{1}{K-1}\left[0.66845+5.7169\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=	14,96876
	βF	=	0,87349
	βV	=	0,34643
Hub stress correction factor	φ	=	1,24196
	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3\beta_V}{\beta_U l_0 g_0{}^{*2}}\right)$	=	2,29678

Flange moments

Total moment acting upon flange for assembly condition	MA = W·hG	=	73 816,6 N·m
Total moment acting upon flange for operating condition	$M_{op} = \lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \rvert$	=	96 474,1 N·m
Moment factor used to design split rings	Fs	=	1,00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op} \frac{C_F}{B^*}$	=	87,5 N·m
Moment exerted on the flange per unit of length (assembly)	$M = F_s \cdot M_A \frac{C_F}{B^*}$	=	67,0 N·m

Flange stresses - operating condition

Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda g_1{}^{*2}}$	=	154,58 MPa
Radial stress in flange	$\sigma_r = \frac{(1.333e \cdot \beta_F + l_0)M}{\lambda \cdot e^2 \cdot l_0}$	=	8,47 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2+1}{K^2-1}$	=	79,71 MPa
Stress factor	k	=	1,03400
	k·σH ≤ 1,5min(f;fH):		Ok
	k·σr ≤ f:		Ok
	k·σθ ≤ f:		Ok
	0,5k(σH + σr) ≤ f:		Ok
	0,5k(σH + σθ) ≤ f:		Ok

Flange stresses - seating condition

Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda g_1{}^{*2}}$	=	118,27 MPa
Radial stress in flange	$\sigma_r = \frac{(1.333e \cdot \beta_F + l_0)M}{\lambda \cdot e^2 \cdot l_0}$	=	6,48 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2+1}{K^2-1}$	=	60,99 MPa
	k·σH ≤ 1,5min(f;fH):		Ok
	k·σr ≤ f:		Ok
	k·σθ ≤ f:		Ok
	0,5k(σH + σr) ≤ f:		Ok
	0,5k(σH + σθ) ≤ f:		Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,47 MPa
Hot & corroded (flange)	=	2,00 MPa
New & cold (bolts)	=	2,32 MPa
Hot & corroded (bolts)	=	2,03 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,02 MPa
Calculation pressure	P = Pt + Ph =	2,98 MPa

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\dfrac{\delta_b}{2d_b + \frac{6e}{m+0,5}}}\ ;1\right]$ =	1,00000
Ratio of the flange diameters	K = A / B =	1,14091
Length parameter	$l_0 = \sqrt{B \cdot g_0^*}$ =	114,89 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^2 P\right)$ =	2 830 183 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	268 853 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B - g1*) / 2 =	42,75 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2 =	26,47 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B - G) / 4 =	39,23 mm
Flange stress factor	$\beta_T = \dfrac{K^2\left(1+8.55246\log_{10}K\right)-1}{\left(1.0472+1.9448K^2\right)(K-1)}$ =	1,86214
Flange stress factor	$\beta_U = \dfrac{K^2\left(1+8.55246\log_{10}K\right)-1}{1.36136\left(K^2-1\right)(K-1)}$ =	16,22655
Flange stress factor	$\beta_Y = \dfrac{1}{K-1}\left[0.66845+5.7169\left(\dfrac{K^2\log_{10}K}{K^2-1}\right)\right]$ =	14,76619
	βF =	0,87709
	βV =	0,36048
Hub stress correction factor	φ =	1,20489
	$\lambda = \left(\dfrac{e\beta_F + l_0}{\beta_T l_0} + \dfrac{e^3\beta_V}{\beta_U l_0 g_0^{*2}}\right)$ =	2,07775

Flange moments

Total moment acting upon flange for operating condition	$M_{op} = \left\lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \right\rvert$ =	140 691,7 N·m
Moment factor used to design split rings	Fs =	1,00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op} \dfrac{C_F}{B}$ =	127,9 N·m

Flange stresses - operating condition

Longitudinal stress in hub	$\sigma_H = \dfrac{\varphi M}{\lambda g_1^{*2}}$ =	216,71 MPa
Radial stress in flange	$\sigma_r = \dfrac{\left(1.333e \cdot \beta_F + l_0\right)M}{\lambda \cdot e^2 \cdot l_0}$ =	13,41 MPa
Tangential stress in flange	$\sigma_\theta = \dfrac{\beta_Y \cdot M}{e^2} - \sigma_r \dfrac{K^2+1}{K^2-1}$ =	106,92 MPa
Stress factor	k =	1,03333
	k·σH ≤ 1,5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0,5k(σH + σr) ≤ f:	Ok
	0,5k(σH + σθ) ≤ f:	Ok

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	121,33 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	182,00 MPa

Load condition 1, Junction to shell (of thickness es)

Maximum allowable pressure (flange)	Pmax =	2,00 MPa
Calculation thickness	en =	12,00 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	158,98 MPa
Equivalent number of full pressure cycles	Neq =	799,82023
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	169,13 MPa
Number of allowable fatigue cycles	N =	103 455
Partial fatigue damage index	D=Nreq/N =	0,01160

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,00 MPa
Calculation thickness	en =	18,50 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	158,98 MPa
Equivalent number of full pressure cycles	Neq =	799,82023
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Transition radius	r =	13,00 mm
Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	ΔσD =	175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,36693
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT]·Kf =	231,19 MPa
Number of allowable fatigue cycles	N =	254 488
Partial fatigue damage index	D=Nreq/N =	0,00472

Fatigue cycles and damage index summary

Load 1, partial damage index for Junction to shell (of thickness es)	=	0,01160
Total damage index: Junction to shell (of thickness es)	=	0,01160
Load 1, partial damage index for Hub to plate junction	=	0,00472
Total damage index: Hub to plate junction	=	0,00472
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.10 Cylindrical shell - SHELL_ UPPER PORTION

According to: EN 13445 Ed. 2009 Issue 5, Clauses 7 and 8

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	0,85

Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4}) =$	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4}) =$	212,50 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05}) =$	338,10 MPa

Geometry

Inside diameter	Di =	1 100,00 mm
Outside diameter	De =	1 120,00 mm
Length	L =	1 724,25 mm
Nominal thickness	en =	10,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0,30 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	1 102,60 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	9,20 mm
		en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	5,88 MPa
Maximum allowable design pressure	=	2,20 MPa

Deformation according to EN 13445-4 Clause 9

Deformation	F=50·en/(Di/2+en/2) =	0,901 %

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,02 MPa
Calculation pressure	Pc=Pt+Pht =	2,98 MPa
Inside diameter	Di'=Di+2δ =	1 100,60 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	5,16 mm
		en ≥ e: Ok

Simplified fatigue assessment according to EN 13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	165,33 MPa
Ultimate tensile strength at room temperature	Rm =	510,00 MPa

Yield strength at design temperature	Rp0,2/T =	248,00 MPa

Load condition 1, Longitudinal butt weld

Maximum allowable pressure (component)	Pmax =	2,20 MPa
Nominal thickness	en =	10,00 mm
Inside diameter	Di =	1 100,00 mm
Offset	δo =	0,50 mm
Peeking or flat	δpf =	1,67 mm
Ovality	u =	0,75 %
Partial stress factor	η1=(3·δo)/en =	0,15000
Partial stress factor	η2=1,5·u·(Di/en) =	1,23750
Partial stress factor	η4=6·δpf/en =	1,00020
Stress factor	η=(1+η1+η2+η4)·z =	2,87955
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	378,63 MPa
Equivalent number of full pressure cycles	Neq =	603,65734
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	402,80 MPa
Number of allowable fatigue cycles	N =	7 658
Partial fatigue damage index	D=Nreq/N =	0,15670

Load condition 1, Circumferential butt weld

Joint efficiency	z =	0,85000
Maximum allowable pressure (component)	Pmax =	2,20 MPa
Calculation thickness	en =	10,00 mm
Joint efficiency	z =	0,85000
Partial stress factor	η0 =	0,10000
Offset	δ =	0,50 mm
Partial stress factor	η1=δ/(2·en) =	0,02500
Stress factor	η=(1+η0+η1)·z =	0,95625
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	125,74 MPa
Equivalent number of full pressure cycles	Neq =	603,65734
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	133,76 MPa
Number of allowable fatigue cycles	N =	209 109
Partial fatigue damage index	D=Nreq/N =	0,00574

Fatigue cycles and damage index summary

Load 1, partial damage index for Longitudinal butt weld	=	0,15670
Total damage index: Longitudinal butt weld	=	0,15670
Load 1, partial damage index for Circumferential butt weld	=	0,00574
Total damage index: Circumferential butt weld	=	0,00574
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.11 Welding neck flange - MAIN FLANGE UPPER SIDE

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Flange material	**P280GH (NT,QT) (EN 10222-2:2001) 50,001 ≤ t ≤ 160- Forging**
Shell material	**P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate**
Bolting material	**25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**
Gasket	**Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**

Allowable stresses	**Flange - f**	**Hub - fH**	**Bolting - fB**
Design condition	121,33 MPa / 17 597,9 psi	165,33 MPa / 23 979,6 psi	128,73 MPa / 18 671,2 psi
Seating condition	143,33 MPa / 20 788,7 psi	212,50 MPa / 30 820,5 psi	146,67 MPa / 21 272,2 psi
Test condition	204,76 MPa / 29 698,2 psi	338,10 MPa / 49 036,6 psi	220,00 MPa / 31 908,3 psi

Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	2,00 MPa
Calculation temperature	T =	260 °C
Corrosion allowance	c =	1,00 mm
Flange external diameter	A =	1 255,00 mm
Inside diameter	B =	1 100,00 mm
Inside diameter (corroded)	B* = B + 2c =	1 102,00 mm
Bolt circle	C =	1 204,00 mm
Flange thickness	e =	103,00 mm
Mean gasket diameter	Gmean =	1 147,00 mm
Hub length	h =	32,00 mm
Thickness of hub at back of flange	g1 =	16,00 mm
Thickness of hub at back of flange (corroded)	g1* =	15,00 mm
Thickness of hub at small end	g0 =	10,00 mm
Thickness of hub at small end (corroded)	g0* =	9,00 mm

Gasket parameters

Gasket factor	m =	3,00
Minimum gasket seating pressure	y =	69,00 MPa
Gasket contact width	w =	20,00 mm
Basic gasket seating width	b0 = w / 2 =	10,00 mm
Effective gasket seating width	b = 2,52·√(b0) =	7,97 mm
Diameter of gasket load reaction	G = Gmean + w - 2b =	1 151,06 mm

Bolt loads

Number of bolts	=	68
Bolt type	=	ISO M22 x 2,50
Root area of one bolt	=	282,0 mm²
Distance between centre lines of adjacent bolts	δb =	55,62 mm
Bolt outside diameter	db =	22,00 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	2 081 217 N
Compression load on gasket to ensure tight joint	HG = 2π·G·b·m·P =	345 804 N
Minimum required bolt load for operating condition	Wop = H + HG =	2 427 022 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	3 097 684 N
Minimum required bolt load for the test condition (MAIN FLANGE_LOWER SIDE)	Wt =	3 613 956 N
Minimum required bolt load for assembly condition	WA = πb·G·y =	1 988 375 N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}}; \frac{W_{op}}{f_B}; \frac{W_t}{f_{B,t}}\right]$ =	18 853,1 mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	19 176,0 mm²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$ =	68 049,7 mm²
Design bolt load for assembly condition	$W = 0,5(A_{B,\min} + A_B) f_{B,A}$ =	2 788 800 N

AB ≥ AB,min: Ok

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\dfrac{\delta_b}{2d_b + \frac{6e}{m+0,5}}}\ ;1\right]$ =	1,00000
Ratio of the flange diameters	K = A / B* =	1,13884
Length parameter	$l_0 = \sqrt{B^{*} \cdot g_0^{*}}$ =	99,59 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^{*2}P\right)$ =	1 907 581 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	173 636 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2 =	43,50 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2 =	26,47 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B* - G) / 4 =	38,73 mm
Flange stress factor	$\beta_T = \dfrac{K^2\left(1+8.55246\log_{10}K\right)-1}{\left(1.0472+1.9448K^2\right)\left(K-1\right)}$ =	1,86292
Flange stress factor	$\beta_U = \dfrac{K^2\left(1+8.55246\log_{10}K\right)-1}{1.36136\left(K^2-1\right)\left(K-1\right)}$ =	16,44915
Flange stress factor	$\beta_Y = \dfrac{1}{K-1}\left[0.66845+5.7169\left(\dfrac{K^2\log_{10}K}{K^2-1}\right)\right]$ =	14,96876
	βF =	0,87093
	βV =	0,33259
Hub stress correction factor	φ =	1,35490
	$\lambda = \left(\dfrac{e\beta_F + l_0}{\beta_T l_0} + \dfrac{e^3\beta_V}{\beta_U l_0 g_0^{*2}}\right)$ =	3,75921

Flange moments

Total moment acting upon flange for assembly condition	MA = W·hG =	73 816,6 N·m
Total moment acting upon flange for operating condition	$M_{op} = \left\lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \right\rvert$ =	98 858,6 N·m
Moment factor used to design split rings	Fs =	1,00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op}\dfrac{C_F}{B^{*}}$ =	89,7 N·m
Moment exerted on the flange per unit of length (assembly)	$M = F_s \cdot M_A\dfrac{C_F}{B^{*}}$ =	67,0 N·m

Flange stresses - operating condition

Longitudinal stress in hub	$\sigma_H = \dfrac{\varphi M}{\lambda g_1^{*2}}$ =	143,70 MPa
Radial stress in flange	$\sigma_r = \dfrac{\left(1.333e \cdot \beta_F + l_0\right)M}{\lambda \cdot e^2 \cdot l_0}$ =	4,95 MPa
Tangential stress in flange	$\sigma_\theta = \dfrac{\beta_Y \cdot M}{e^2} - \sigma_r\dfrac{K^2+1}{K^2-1}$ =	88,28 MPa
Stress factor	k =	1,03400
	k·σH ≤ 1,5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0,5k(σH + σr) ≤ f:	Ok
	0,5k(σH + σθ) ≤ f:	Ok

Flange stresses - seating condition

Longitudinal stress in hub	$\sigma_H = \dfrac{\varphi M}{\lambda g_1^{*2}}$ =	107,30 MPa
Radial stress in flange	$\sigma_r = \dfrac{\left(1.333e \cdot \beta_F + l_0\right)M}{\lambda \cdot e^2 \cdot l_0}$ =	3,70 MPa
Tangential stress in flange	$\sigma_\theta = \dfrac{\beta_Y \cdot M}{e^2} - \sigma_r\dfrac{K^2+1}{K^2-1}$ =	65,92 MPa
	k·σH ≤ 1,5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0,5k(σH + σr) ≤ f:	Ok
	0,5k(σH + σθ) ≤ f:	Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,46 MPa
Hot & corroded (flange)	=	2,02 MPa
New & cold (bolts)	=	2,32 MPa
Hot & corroded (bolts)	=	2,03 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,02 MPa
Calculation pressure	P = Pt + Ph =	2,98 MPa

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}}\ ;1\right]$ =	1,00000
Ratio of the flange diameters	K = A / B =	1,14091
Length parameter	$l_0 = \sqrt{B \cdot g_0{}^*}$ =	104,88 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^2 P\right)$ =	2 828 948 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	268 736 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B - g1*) / 2 =	44,00 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2 =	26,47 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B - G) / 4 =	39,23 mm
Flange stress factor	$\beta_T = \frac{K^2\left(1+8.55246\log_{10}K\right)-1}{\left(1.0472+1.9448K^2\right)\left(K-1\right)}$ =	1,86214
Flange stress factor	$\beta_U = \frac{K^2\left(1+8.55246\log_{10}K\right)-1}{1.36136\left(K^2-1\right)\left(K-1\right)}$ =	16,22655
Flange stress factor	$\beta_Y = \frac{1}{K-1}\left[0.66845+5.7169\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$ =	14,76619
	βF =	0,87542
	βV =	0,34994
Hub stress correction factor	φ =	1,29900
	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3\beta_V}{\beta_U l_0 g_0{}^{*2}}\right)$ =	3,24563

Flange moments

Total moment acting upon flange for operating condition	$M_{op} = \lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \rvert$ =	144 170,5 N·m
Moment factor used to design split rings	Fs =	1,00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op}\frac{C_F}{B}$ =	131,1 N·m

Flange stresses - operating condition

Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda g_1{}^{*2}}$ =	204,91 MPa
Radial stress in flange	$\sigma_r = \frac{\left(1.333e \cdot \beta_F + l_0\right)M}{\lambda \cdot e^2 \cdot l_0}$ =	8,17 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r\frac{K^2+1}{K^2-1}$ =	120,10 MPa
Stress factor	k =	1,03333
	k·σH ≤ 1,5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0,5k(σH + σr) ≤ f:	Ok
	0,5k(σH + σθ) ≤ f:	Ok

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C

Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	121,33 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	182,00 MPa

Load condition 1, Junction to shell (of thickness es)

Maximum allowable pressure (flange)	Pmax =	2,02 MPa
Calculation thickness	en =	10,00 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	157,42 MPa
Equivalent number of full pressure cycles	Neq =	776,56702
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	167,47 MPa
Number of allowable fatigue cycles	N =	106 553
Partial fatigue damage index	D=Nreq/N =	0,01126

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,02 MPa
Calculation thickness	en =	16,00 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	157,42 MPa
Equivalent number of full pressure cycles	Neq =	776,56702
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Transition radius	r =	13,00 mm
Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	ΔσD =	175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,36833
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT]·Kf =	229,16 MPa
Number of allowable fatigue cycles	N =	266 197
Partial fatigue damage index	D=Nreq/N =	0,00451

Fatigue cycles and damage index summary

Load 1, partial damage index for Junction to shell (of thickness es)	=	0,01126
Total damage index: Junction to shell (of thickness es)	=	0,01126
Load 1, partial damage index for Hub to plate junction	=	0,00451
Total damage index: Hub to plate junction	=	0,00451
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.12 Ellipsoidal head - TOP HEAD

According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	212,50 MPa

Nominal design stress in test condition	$f_{\text{test}} = (\frac{R_{\text{p0,2/Ttest}}}{1,05})$ =	338,10 MPa

Geometry

Inside diameter	Di =	1 100,00 mm
Outside diameter	De =	1 128,00 mm
Head outside height	H =	339,00 mm
Nominal thickness	en =	14,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	1,40 mm
Straight flange length	l(sf) =	50,00 mm
Straight flange thickness	en(sf) =	14,00 mm
Knuckle thickness	en(k) =	14,00 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Y parameter	Y=min(ec/R;0,04) =	0,01080
Z parameter	Z=log10(1/Y) =	1,96652
X ratio	X=r/Di =	0,17143
N parameter	$N = 1,006 - \frac{1}{[6,2+(90Y)^4]}$ =	0,86502
β(0,1) parameter	$\beta_{0,1} = N(-0,1833Z^3 + 1,0383Z^2 - 1,2943Z + 0,837)$ =	0,78982
β(0,2) parameter	$\beta_{0,2} = \max[0,95(0,56 - 1,94Y - 82,5Y^2); 0,5]$ =	0,50295
β parameter	$\beta = 10[(0,2-X)\beta_{0,1} + (X-0,1)\beta_{0,2}]$ =	0,58490
0,2 % proof strength at design temperature	Rp02t =	248,00 MPa
Design stress for buckling equation	fb=Rp02t/1,5 =	165,33 MPa
Joint efficiency	z =	1,00000
Head ratio	K=Di/2h =	2,00000
Inside diameter	Di'=Di+2·c+2·δ =	1 104,80 mm
Equivalent crown radius	R'=Di'(0,44K+0,02) =	992,40 mm
Equivalent knuckle radius	r'=Di'(0,5/K-0,08) =	189,40 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fz - 0,5P} + c + ce + \delta$ =	8,42 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0,75R' + 0,2D_i')}{f} + c + ce + \delta$ =	9,60 mm
Minimum required thickness	e=max(ey;es) =	9,60 mm
Straight flange thickness	e(sf) =	14,00 mm
Straight flange minimum required thickness	e =	9,12 mm
		en(sf) ≥ e(sf): Ok

Knuckle check due to encroaching nozzle

Largest inside diameter of nozzles encroaching knuckle region	di =	193,70 mm
V parameter	$V = \log_{10}(1000\frac{P}{f})$ =	1,08267
A parameter	$A = 0,54 + 0,41V - 0,044V^3$ =	0,92806
B parameter	$B = 7,77 - 4,53V + 0,744V^2$ =	3,73760
Weakening factor due to the presence of nozzle	$\beta_k = \max(A + B\frac{d_i}{D_e}; 1 + 0,5B\frac{d_i}{D_e})$ =	1,56988
Required thickness of knuckle encroached by nozzle to avoid axisymmetric yielding	$e_{y(k)} = \frac{\beta P \beta_k (0,75R' + 0,2D_i')}{f} + c + ce + \delta$ =	13,12 mm
Minimum knuckle thickness	e(k)=max[ey(k);es] =	13,12 mm
		en(k) ≥ e(k): Ok
		en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	7,66 MPa
Maximum allowable design pressure	=	3,43 MPa

Deformation according to EN13445-4 Clause 9

Manufactured in one piece	F(1)=100·ln[(1,21·De)/(De-2·en)] =	21,576 %
Spherical part	F(2)=100·ln{2·R·asin[(0,4·De/R)/(0,8·De-2·en)]} =	6,913 %
Knuckle segments	F(3)=(100·en)/(r+en/2) =	7,216 %

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,005 MPa
Calculation pressure	Pc=Pt+Pht =	2,96 MPa
Y parameter	Y=min(ec/R;0,04) =	0,00829
Z parameter	Z=log10(1/Y) =	2,08141
X ratio	X=r/Di =	0,17084
N parameter	$N = 1.006 - \frac{1}{[6.2 + (90Y)^4]}$ =	0,85239
β(0,1) parameter	$\beta_{0.1} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837)$ =	0,84247
β(0,2) parameter	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2); 0.5]$ =	0,51133
β parameter	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$ =	0,60790
0,2 % proof strength at design temperature	Rp02t =	355,00 MPa
Design stress for buckling equation	fb=Rp02t/1,05 =	338,10 MPa
Joint efficiency	z =	1,00000
Head ratio	K=Di/2h =	2,00000
Inside diameter	Di'=Di+2·δ =	1 102,80 mm
Equivalent crown radius	R'=Di'(0,44K+0,02) =	991,40 mm
Equivalent knuckle radius	r'=Di'(0,5/K-0,08) =	188,40 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fz - 0.5P} + \delta$ =	5,75 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R' + 0.2D_i')}{f} + \delta$ =	6,80 mm
Required thickness of knuckle to avoid plastic buckling	$e_b = (0.75R' + 0.5D_i')\left[\frac{P}{111f_b}\left(\frac{D_i'}{r'}\right)^{0.825}\right]^{\left(\frac{1}{1.5}\right)} + \delta$ =	6,08 mm
Minimum required thickness	e=max(ey;es;eb) =	6,80 mm

Knuckle check due to encroaching nozzle

Largest inside diameter of nozzles encroaching knuckle region	di =	193,70 mm
V parameter	$V = \log_{10}(1000\frac{P}{f})$ =	0,94204
A parameter	$A = 0.54 + 0.41V - 0.044V^3$ =	0,88945
B parameter	$B = 7.77 - 4.53V + 0.744V^2$ =	4,16281
Weakening factor due to the presence of nozzle	$\beta_k = \max(A + B\frac{d_i}{D_e}; 1 + 0.5B\frac{d_i}{D_e})$ =	1,60429
Required thickness of knuckle encroached by nozzle to avoid axisymmetric yielding	$e_{y(k)} = \frac{\beta P \beta_k (0.75R' + 0.2D_i')}{f} + \delta$ =	9,63 mm
Minimum knuckle thickness	e(k)=max[ey(k);es;eb] =	9,63 mm
		en(k) ≥ e(k): Ok
		en ≥ e: Ok

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	165,33 MPa
Ultimate tensile strength at room temperature	Rm =	510,00 MPa
Yield strength at design temperature	Rp0,2/T =	248,00 MPa

Load condition 1, all butt welds

Maximum allowable pressure (component)	Pmax =	3,43 MPa
Nominal thickness	en =	14,00 mm
Joint efficiency	z =	1,00000
Butt weld offset	δ =	0,70 mm
Partial stress factor	η1=(3·δ)/en =	0,15000
Butt weld angular misalignment	Θ =	2,50 °
Partial stress factor	Dm=(Di+Do)/2 =	1 114,00 mm

Partial stress factor	$\eta3=(\Theta/50)\cdot\sqrt{[Dm/(2\cdot en)]}$ =	0,31538
Stress factor	$\eta=(1+\eta1+\eta3)\cdot z$ =	1,46538
Pseudo-elastic stress range	$\Delta\sigma=(\Delta P/Pmax)\cdot\eta\cdot f$ =	123,50 MPa
Equivalent number of full pressure cycles	Neq =	158,95492
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T^*=0,75\cdot Tmax+0,25\cdot Tmin$ =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	$\Delta\sigma D$ =	46,43 MPa
Cut-off limit	$\Delta\sigma cut$ =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta\sigma^*=[\Delta\sigma/Ce\cdot CT]$ =	131,39 MPa
Number of allowable fatigue cycles	N =	220 675
Partial fatigue damage index	D=Nreq/N =	0,00544

Load condition 1, Knuckle region weld

Maximum allowable pressure (component)	Pmax =	3,43 MPa
Calculation thickness	en =	14,00 mm
Stress factor	η =	2,50000
Pseudo-elastic stress range	$\Delta\sigma=(\Delta P/Pmax)\cdot\eta\cdot f$ =	210,70 MPa
Equivalent number of full pressure cycles	Neq =	158,95492
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T^*=0,75\cdot Tmax+0,25\cdot Tmin$ =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	$\Delta\sigma D$ =	46,43 MPa
Cut-off limit	$\Delta\sigma cut$ =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta\sigma^*=[\Delta\sigma/Ce\cdot CT]$ =	224,15 MPa
Number of allowable fatigue cycles	N =	44 441
Partial fatigue damage index	D=Nreq/N =	0,02700

Fatigue cycles and damage index summary

Load 1, partial damage index for all butt welds	=	0,00544
Total damage index: all butt welds	=	0,00544
Load 1, partial damage index for Knuckle region weld	=	0,02700
Total damage index: Knuckle region weld	=	0,02700
		TDI(1)<1: Ok

C.13 Reinforcement of opening - LWN Flange N1 - DN100 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 9

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Shell material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate

Nominal design stress of shell material	fs =	165,33 MPa

Nozzle material: P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging

Nominal design stress of the nozzle material	fb =	134,00 MPa
	fob = min(fs, fb) =	134,00 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a critical area	=	Yes
Offset from shell border	=	0 mm
Angular offset	=	0 °
Offset k between nozzle and shell axis	=	0 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm

Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	142,00 mm
Internal diameter	d_ib = Id + 2(c + δ) =	104,30 mm
External diameter of the nozzle	d_eb = Od - 2ce =	142,00 mm
Analysis thickness of the nozzle	e_ab =	19,85 mm
Length of nozzle extending outside the shell	l_b =	177,50 mm
Effective thickness of the nozzle	e_b = e_ab - c - ce - δ =	18,85 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b)e_b}$ =	48,18 mm
	$l_{bo} = \min(l_b; l_{bo(max)})$ =	48,18 mm
Effective length of nozzle outside the shell for reinforcement	$l_b' = \min(l_{bo}; l_b)$ =	48,18 mm
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm²
		eb ≤ e_ab: Ok
		ea,b / ea,s ≤ 3: Ok

Pad geometry

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	$\mathrm{Af}_p = e_p \cdot l_p'$ =	0 mm²

Shell geometry

Analysis thickness of shell wall	$e_{a,s} = t_{shell} - c_{shell} - \delta_{shell}$ =	11,60 mm
Analysis thickness of shell wall	e_cs =	11,60 mm
Shell internal diameter	Di = Id_head + 2(cs + cs') =	1 104,80 mm
Inside height of the dished end	h = H_head - e_head + cs + cs' =	277,40 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{0.44D_i^2}{2h} + 0.02D_i$ =	990,11 mm

Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ =	152,00 mm
Head's analysis thickness	ek =	11,60 mm
Head's knuckle radius	rk =	189,40 mm
Distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w(7.7.2) =	814,74 mm
Limit distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w_min(7.7.2)=2,5·√(ek·rk) =	117,18 mm
Distance between an opening and a shell discontinuity	w =	493,00 mm
Length of shell from the edge of the opening to a shell discontinuity	l_s = w =	493,00 mm
Minimum value for w which has no influence on ls from shell discontinuities	w_p = l_so =	152,00 mm
Required minimum value for w	w_min =	112,80 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P = Pi + Ph =	2,00 MPa

Transverse section

	$\delta = \frac{d_{eb}}{2r_{ms}}$ =	4,08 °
	$a = \frac{d_{eb}}{2r_{ms}}$ =	71,06 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{a,s}$ =	995,91 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ =	152,00 mm
Effective length of shell for opening reinforcement	$l_s' = \min[l_s, l_{so}]$ =	152,00 mm
Pressure loaded area - shell	$\mathrm{Ap}_s = 0{,}5r_{is}^2\frac{l_s' + a}{0{,}5e_{a,s} + r_{is}}$ =	109 786,4 mm²
Length of penetration into shell wall	e's = e_a,s =	11,60 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$\mathrm{Af}_b = e_b \cdot (l_b' + l_{bi}' + e_s')$ =	1 126,9 mm²
Stress loaded cross-sectional area - shell	$\mathrm{Af}_s = (l_s' + e_{ab} - c_{shell})e_{c,s}$ =	1 763,2 mm²
Pressure loaded area - nozzle	$\mathrm{Ap}_b = 0.5d_{ib} \cdot (l_b' + e_{a,s})$ =	3 117,6 mm²
Additional area due to obliquity of the nozzle	Apφ =	0 mm²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp =	0 mm²
Reactive force	$F_r = (\mathrm{Af}_s + \mathrm{Af}_w)(f_s - 0.5P) + \mathrm{Af}_p(f_{op} - 0.5P) + \mathrm{Af}_b(f_{ob} - 0.5P)$ =	439 633 N

Pressure load Fp = P(Aps + Apb + 0,5Apφ) = 225 808 N

Maximum allowable pressure $P_{\max} = \dfrac{(\mathrm{Af}_s + \mathrm{Af}_w)\cdot f_s + \mathrm{Af}_b\cdot f_{\mathrm{ob}} + \mathrm{Af}_p\cdot f_{\mathrm{op}}}{(\mathrm{Ap}_s + \mathrm{Ap}_b + 0.5\mathrm{Ap}_\varphi) + 0.5(\mathrm{Af}_s + \mathrm{Af}_w + \mathrm{Af}_b + \mathrm{Af}_p)}$ = 3,87 MPa

Fr ≥ Fp: Ok

Adjacent openings

LWN Flange N1 - DN100 PN25 - LWN Flange N2 - DN100 PN25

Centre-to-centre distance taken on the mean surface of the shell Lb = 303,98 mm

Ligament length L = 161,98 mm

Mean shell radius at the centres of adjacent nozzles r_is = 990,11 mm

Minimum required ligament length $L_{\min} = \max\left[3e_{a,s};\ 0.2\sqrt{(2r_{\mathrm{is}} + e_{c,s})e_{c,s}}\right]$ = 34,80 mm

d1 = 142,00 mm

d2 = 142,00 mm

L ≥ Lmin: Ok

d1+d2 ≤ 0,2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required

Ligament check

Inside radius of curvature of the shell at the opening centre $r_{\mathrm{is}} = \dfrac{D_e}{2} - e_{a,s}$ = 990,11 mm

Angle between the centre-to-centre line of openings and the generatrix of the shell Φ = 90,00 °

Ap of the shell for the length Lb $\mathrm{Ap}_{\mathrm{Ls}} = \dfrac{0.5r_{\mathrm{is}}^2\cdot L_b(1+\cos(\phi))}{r_{\mathrm{is}} + 0.5e_{a,s}\cdot\sin(\phi)}$ = 149 609,7 mm²

= 142,00 mm

deb2 = 142,00 mm

ϕ_e1 = 0 °

ϕ_e2 = 0 °

$r_{\mathrm{os1}} = \dfrac{r_{\mathrm{is1}}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$r_{\mathrm{os2}} = \dfrac{r_{\mathrm{is2}}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$\delta_1 = \dfrac{d_{\mathrm{eb1}}}{2r_{\mathrm{os1}}}$ = 4,08 °

$\delta_2 = \dfrac{d_{\mathrm{eb2}}}{2r_{\mathrm{os2}}}$ = 4,08 °

$a_1 = r_{\mathrm{os1}}\left[\arcsin(\delta_1 + \sin(\varphi_{\mathrm{e1}})) - \varphi_{\mathrm{e1}}\right]$ = 71,06 mm

$a_2 = r_{\mathrm{os2}}\left[\arcsin(\delta_2 + \sin(\varphi_{\mathrm{e2}})) - \varphi_{\mathrm{e2}}\right]$ = 71,06 mm

Af of the shell contained along the length Lb $\mathrm{Af}_{\mathrm{Ls}} = (L_b - a_1 - a_2)\cdot e_{c,s}$ = 1 877,5 mm²

Apφ1 = 0 mm²

Apφ2 = 0 mm²

Apb1 = 3 117,6 mm²

Apb2 = 3 117,6 mm²

Afb1 = 1 126,9 mm²

Afb2 = 1 126,9 mm²

Afp1 = 0 mm²

Afp2 = 0 mm²

fob1 = 134,00 MPa

fob2 = 134,00 MPa

fop1 = 0 MPa

fop2 = 0 MPa

Reactive force $F = (\mathrm{Af}_{\mathrm{Ls}} + \mathrm{Af}_w)(f_s - 0.5P) + \mathrm{Af}_{\mathrm{b1}}(f_{\mathrm{ob1}} - 0.5P) + \mathrm{Af}_{\mathrm{p1}}(f_{\mathrm{op1}} - 0.5P) + {}$ $+ \mathrm{Af}_{\mathrm{b2}}(f_{\mathrm{ob2}} - 0.5P) + \mathrm{Af}_{\mathrm{p2}}(f_{\mathrm{op2}} - 0.5P)$ = 608 288 N

Pressure load $F_{\mathrm{req}} = P\left(\mathrm{Ap}_{\mathrm{Ls}} + \mathrm{Ap}_{\mathrm{b1}} + 0.5\mathrm{Ap}_{\varphi1} + \mathrm{Ap}_{\mathrm{b2}} + 0.5\mathrm{Ap}_{\varphi2}\right)$ = 311 690 N

F ≥ Freq: Ok

LWN Flange N1 - DN100 PN25 - NOZZLE N3 (DN 200)

Centre-to-centre distance taken on the mean surface of the shell Lb = 406,09 mm

Ligament length L = 225,54 mm

Mean shell radius at the centres of adjacent nozzles r_is = 990,11 mm

Minimum required ligament length $L_{\min} = \max\left[3e_{a,s};\ 0.2\sqrt{(2r_{\mathrm{is}} + e_{c,s})e_{c,s}}\right]$ = 34,80 mm

d1 = 142,00 mm

d2 = 219,10 mm

L ≥ Lmin: Ok

d1+d2 ≤ 0,2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required

Ligament check

Inside radius of curvature of the shell at the opening centre $r_{is} = \frac{D_e}{2} - e_{a,s}$ = 990,11 mm

Angle between the centre-to-centre line of openings and the generatrix of the shell Φ = 90,00 °

Ap of the shell for the length Lb $Ap_{Ls} = \frac{0.5r^2_{is} \cdot L_b(1+\cos(\phi))}{r_{is} + 0.5e_{a,s} \cdot \sin(\phi)}$ = 199 864,9 mm²

deb1 = 142,00 mm

deb2 = 219,10 mm

ϕ_e1 = 0 °

ϕ_e2 = 0 °

$r_{os1} = \frac{r_{is1}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$r_{os2} = \frac{r_{is2}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$\delta_1 = \frac{d_{eb1}}{2r_{os1}}$ = 4,08 °

$\delta_2 = \frac{d_{eb2}}{2r_{os2}}$ = 6,30 °

$a_1 = r_{os1}[\arcsin(\delta_1 + \sin(\varphi_{e1})) - \varphi_{e1}]$ = 71,06 mm

$a_2 = r_{os2}[\arcsin(\delta_2 + \sin(\varphi_{e2})) - \varphi_{e2}]$ = 109,77 mm

Af of the shell contained along the length Lb $Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{c,s}$ = 2 612,9 mm²

Apφ1 = 0 mm²

Apφ2 = 0 mm²

Apb1 = 3 117,6 mm²

Apb2 = 5 724,8 mm²

Afb1 = 1 126,9 mm²

Afb2 = 582,2 mm²

Afp1 = 0 mm²

Afp2 = 0 mm²

fob1 = 134,00 MPa

fob2 = 111,73 MPa

fop1 = 0 MPa

fop2 = 0 MPa

Reactive force $F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{b1}(f_{ob1} - 0.5P) + Af_{p1}(f_{op1} - 0.5P) + $
$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op2} - 0.5P)$ = 643 734 N

Pressure load $F_{req} = P(Ap_{Ls} + Ap_{b1} + 0.5Ap_{\varphi 1} + Ap_{b2} + 0.5Ap_{\varphi 2})$ = 417 415 N

F ≥ Freq: Ok

Hydrostatic test

Item hydrostatic test pressure Pt = 2,95 MPa

Overpressure due to static head Ph = 0,002 MPa

Calculation pressure P = Pt + Ph = 2,96 MPa

Transverse section

$\delta = \frac{d_{eb}}{2r_{ms}}$ = 4,08 °

$a = \frac{d_{eb}}{2r_{ms}}$ = 71,06 mm

Mean radius of curvature $r_{ms} = r_{is} + 0.5e_{a,s}$ = 996,36 mm

Maximum length of shell contributing to opening reinforcement $l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ = 158,46 mm

Effective length of shell for opening reinforcement $l_s' = \min[l_s, l_{so}]$ = 158,46 mm

Pressure loaded area - shell $Ap_s = 0{,}5r^2_{is}\frac{l_s' + a}{0{,}5e_{a,s} + r_{is}}$ = 112 899,3 mm²

Length of penetration into shell wall e's = e_a,s = 12,60 mm

Stress loaded cross-sectional area effective as reinforcement - nozzle $Af_b = e_b \cdot (l_b' + l_{bi}' + e_s')$ = 1 227,5 mm²

Stress loaded cross-sectional area - shell $Af_s = (l_s' + e_{ab} - c_{shell})e_{c,s}$ = 1 996,5 mm²

Pressure loaded area - nozzle $Ap_b = 0.5d_{ib} \cdot (l_b' + e_{a,s})$ = 3 163,2 mm²

Additional area due to obliquity of the nozzle Apφ = 0 mm²

Stress loaded cross-sectional area effective as reinforcement - pad Afp = 0 mm²

Reactive force $F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$ = 985 913 N

Pressure load Fp = P(Aps + Apb + 0,5Apφ) = 343 011 N

Maximum allowable pressure $P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$ = 8,42 MPa

Fr ≥ Fp: Ok

C.14 Reinforcement of opening - LWN Flange N2 - DN100 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 9

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Shell material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate

Nominal design stress of shell material	fs =	165,33 MPa

Nozzle material: P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm - Forging

Nominal design stress of the nozzle material	fb =	134,00 MPa
	fob = min(fs, fb) =	134,00 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a critical area	=	Yes
Offset from shell border	=	0 mm
Angular offset	=	60,00 °
Offset k between nozzle and shell axis	=	280,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	142,00 mm
Internal diameter	d_ib = Id + 2(c + δ) =	104,30 mm
External diameter of the nozzle	d_eb = Od - 2ce =	142,00 mm
Analysis thickness of the nozzle	e_ab =	19,85 mm
Length of nozzle extending outside the shell	l_b =	215,80 mm
Effective thickness of the nozzle	e_b = e_ab - c - ce - δ =	18,85 mm
	$l_{bo(max)} = \sqrt{(d_{cb} - e_b)e_b}$ =	48,18 mm
	$l_{bo} = \min(l_b; l_{bo(max)})$ =	48,18 mm
Effective length of nozzle outside the shell for reinforcement	$l_b' = \min(l_{bo}; l_b)$ =	48,18 mm
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm²

eb ≤ e_ab: Ok
ea,b / ea,s ≤ 3: Ok

Pad geometry

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	$Af_p = e_p \cdot l_p'$ =	0 mm²

Shell geometry

Analysis thickness of shell wall	$e_{a,s} = t_{shell} - c_{shell} - \delta_{shell}$ =	11,60 mm
Analysis thickness of shell wall	e_cs =	11,60 mm
Shell internal diameter	Di = Id_head + 2(cs + cs') =	1 104,80 mm
Inside height of the dished end	h = H_head - e_head + cs + cs' =	277,40 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{0.44D_i^2}{2h} + 0.02D_i$ =	990,11 mm

Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ =	152,00 mm
Head's analysis thickness	ek =	11,60 mm
Head's knuckle radius	rk =	189,40 mm
Distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w(7,7,2) =	507,07 mm
Limit distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w_min(7.7.2)=2,5·√(ek·rk) =	117,18 mm

Distance between an opening and a shell discontinuity w = 213,00 mm
Length of shell from the edge of the opening to a shell discontinuity l_s = w = 213,00 mm
Minimum value for w which has no influence on ls from shell discontinuities w_p = l_so = 152,00 mm
Required minimum value for w w_min = 112,80 mm

Internal pressure

Overpressure due to static head Ph = 0 MPa
Calculation pressure P = Pi + Ph = 2,00 MPa

Transverse section

$\delta = \frac{d_{eb}}{2r_{ms}}$ = 4,08 °

$a = \frac{d_{eb}}{2r_{ms}}$ = 71,06 mm

Mean radius of curvature $r_{ms} = r_{is} + 0.5e_{a,s}$ = 995,91 mm
Maximum length of shell contributing to opening reinforcement $l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ = 152,00 mm
Effective length of shell for opening reinforcement $l_s' = \min[l_s, l_{so}]$ = 152,00 mm
Pressure loaded area - shell $Ap_s = 0,5r^2_{is}\frac{l_s' + a}{0,5e_{a,s} + r_{is}}$ = 109 786,4 mm²

Length of penetration into shell wall e's = e_a,s = 11,60 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle $Af_b = e_b \cdot (l_b' + l_{bi}' + e_s')$ = 1 126,9 mm²
Stress loaded cross-sectional area - shell $Af_s = (l_s' + e_{ab} - c_{shell})e_{c,s}$ = 1 763,2 mm²
Pressure loaded area - nozzle $Ap_b = 0.5d_{ib} \cdot (l_b' + e_{a,s})$ = 3 117,6 mm²
Additional area due to obliquity of the nozzle Apφ = 0 mm²
Stress loaded cross-sectional area effective as reinforcement - pad Afp = 0 mm²
Reactive force $F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$ = 439 633 N
Pressure load Fp = P(Aps + Apb + 0,5Apφ) = 225 808 N
Maximum allowable pressure $P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$ = 3,87 MPa

Fr ≥ Fp: Ok

Adjacent openings

LWN Flange N2 - DN100 PN25 - LWN Flange N1 - DN100 PN25

Centre-to-centre distance taken on the mean surface of the shell Lb = 303,98 mm
Ligament length L = 161,98 mm
Mean shell radius at the centres of adjacent nozzles r_is = 990,11 mm
Minimum required ligament length $L_{min} = \max[3e_{a,s}; 0.2\sqrt{(2r_{is} + e_{c,s})e_{c,s}}]$ = 34,80 mm
d1 = 142,00 mm
d2 = 142,00 mm
L ≥ Lmin: Ok
d1+d2 ≤ 0,2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required

Ligament check

Inside radius of curvature of the shell at the opening centre $r_{is} = \frac{D_e}{2} - e_{a,s}$ = 990,11 mm
Angle between the centre-to-centre line of openings and the generatrix of the shell Φ = 90,00 °
Ap of the shell for the length Lb $Ap_{Ls} = \frac{0.5r^2_{is} \cdot L_b(1 + \cos(\phi))}{r_{is} + 0.5e_{a,s} \cdot \sin(\phi)}$ = 149 609,7 mm²
deb1 = 142,00 mm
deb2 = 142,00 mm
ϕ_e1 = 0 °
ϕ_e2 = 0 °

$r_{os1} = \frac{r_{is1}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$r_{os2} = \frac{r_{is2}}{\sin^2(\varphi)} + 0.5e_{a,s}$ = 995,91 mm

$\delta_1 = \frac{d_{eb1}}{2r_{os1}}$ = 4,08 °

$\delta_2 = \frac{d_{eb2}}{2r_{os2}}$ = 4,08 °

$a_1 = r_{os1}[\arcsin(\delta_1 + \sin(\varphi_{e1})) - \varphi_{e1}]$ = 71,06 mm
$a_2 = r_{os2}[\arcsin(\delta_2 + \sin(\varphi_{e2})) - \varphi_{e2}]$ = 71,06 mm
Af of the shell contained along the length Lb $Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{c,s}$ = 1 877,5 mm²
Apφ1 = 0 mm²

	$Ap\varphi 2$	=	0 mm²
	Apb1	=	3 117,6 mm²
	Apb2	=	3 117,6 mm²
	Afb1	=	1 126,9 mm²
	Afb2	=	1 126,9 mm²
	Afp1	=	0 mm²
	Afp2	=	0 mm²
	fob1	=	134,00 MPa
	fob2	=	134,00 MPa
	fop1	=	0 MPa
	fop2	=	0 MPa
Reactive force	$F = (\mathrm{Af}_{\mathrm{Ls}} + \mathrm{Af}_w)(f_s - 0.5P) + \mathrm{Af}_{\mathbf{b1}}(f_{\mathbf{ob1}} - 0.5P) + \mathrm{Af}_{\mathbf{p1}}(f_{\mathbf{op1}} - 0.5P) + {}$ $+ \mathrm{Af}_{\mathbf{b2}}(f_{\mathbf{ob2}} - 0.5P) + \mathrm{Af}_{\mathbf{p2}}(f_{\mathbf{op2}} - 0.5P)$	=	608 288 N
Pressure load	$F_{\mathbf{req}} = P(\mathrm{Ap}_{\mathrm{Ls}} + \mathrm{Ap}_{\mathbf{b1}} + 0.5\mathrm{Ap}_{\varphi 1} + \mathrm{Ap}_{\mathbf{b2}} + 0.5\mathrm{Ap}_{\varphi 2})$	=	311 690 N
			F ≥ Freq: Ok

Hydrostatic test

Item hydrostatic test pressure	Pt	=	2,95 MPa
Overpressure due to static head	Ph	=	0,003 MPa
Calculation pressure	P = Pt + Ph	=	2,96 MPa

Transverse section

	$\delta = \dfrac{d_{\mathbf{eb}}}{2r_{\mathbf{ms}}}$	=	4,08 °
	$a = \dfrac{d_{\mathbf{eb}}}{2r_{\mathbf{ms}}}$	=	71,06 mm
Mean radius of curvature	$r_{\mathbf{ms}} = r_{\mathbf{is}} + 0.5e_{a,s}$	=	996,36 mm
Maximum length of shell contributing to opening reinforcement	$l_{\mathbf{so}} = \sqrt{(2r_{\mathbf{is}} + e_{c,s})e_{c,s}}$	=	158,46 mm
Effective length of shell for opening reinforcement	$l_s' = \min[l_s, l_{\mathbf{so}}]$	=	158,46 mm
Pressure loaded area - shell	$\mathrm{Ap}_s = 0,5r^2_{\mathbf{is}} \dfrac{l_s' + a}{0,5e_{a,s} + r_{\mathbf{is}}}$	=	112 899,3 mm²
Length of penetration into shell wall	e's = e_a,s	=	12,60 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$\mathrm{Af}_b = e_b \cdot (l_b' + l_{\mathbf{bi}}' + e_s')$	=	1 227,5 mm²
Stress loaded cross-sectional area - shell	$\mathrm{Af}_s = (l_s' + e_{\mathbf{ab}} - c_{\mathbf{shell}})e_{c,s}$	=	1 996,5 mm²
Pressure loaded area - nozzle	$\mathrm{Ap}_b = 0.5d_{\mathbf{ib}} \cdot (l_b' + e_{a,s})$	=	3 163,2 mm²
Additional area due to obliquity of the nozzle	$Ap\varphi$	=	0 mm²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm²
Reactive force	$F_r = (\mathrm{Af}_s + \mathrm{Af}_w)(f_s - 0.5P) + \mathrm{Af}_p(f_{\mathbf{op}} - 0.5P) + \mathrm{Af}_b(f_{\mathbf{ob}} - 0.5P)$	=	985 912 N
Pressure load	Fp = P(Aps + Apb + 0,5Apφ)	=	343 104 N
Maximum allowable pressure	$P_{\max} = \dfrac{(\mathrm{Af}_s + \mathrm{Af}_w) \cdot f_s + \mathrm{Af}_b \cdot f_{\mathbf{ob}} + \mathrm{Af}_p \cdot f_{\mathbf{op}}}{(\mathrm{Ap}_s + \mathrm{Ap}_b + 0.5\mathrm{Ap}_\varphi) + 0.5(\mathrm{Af}_s + \mathrm{Af}_w + \mathrm{Af}_b + \mathrm{Af}_p)}$	=	8,42 MPa
			Fr ≥ Fp: Ok

C.15 Reinforcement of opening - NOZZLE N3 (DN 200)

According to: EN 13445 Ed. 2009 Issue 5, Clause 9

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1.00

Shell material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate

Nominal design stress of shell material	fs =	165,33 MPa

Nozzle material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm - Seamless tube

Nominal design stress of the nozzle material	fb =	111,73 MPa
	fob = min(fs, fb) =	111,73 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a critical area	=	Yes
Offset from shell border	=	0 mm
Angular offset	=	180,00 °
Offset k between nozzle and shell axis	=	350,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	1,59 mm
Maximum width of the opening on shell without nozzle	d =	219,10 mm
Internal diameter	d_ib = Id + 2(c + δ) =	198,88 mm
External diameter of the nozzle	d_eb = Od - 2ce =	219,10 mm
Nominal thickness of the nozzle	e_ab =	12,70 mm
Length of nozzle extending outside the shell	l_b =	164,22 mm
Effective thickness of the nozzle	e_b = e_ab - c - ce - δ =	10,11 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b)e_b}$ =	45,97 mm
	$l_{bo} = \min(l_b; l_{bo(max)})$ =	45,97 mm
Effective length of nozzle outside the shell for reinforcement	$l_b' = \min(l_{bo}; l_b)$ =	45,97 mm
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	50,0 mm²
		eb ≤ e_ab: Ok
		ea,b / ea,s ≤ 3: Ok

Pad geometry

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	$Af_p = e_p \cdot l_p'$ =	0 mm²

Shell geometry

Analysis thickness of shell wall	$e_{a,s} = t_{shell} - c_{shell} - \delta_{shell}$ =	11,60 mm
Analysis thickness of shell wall	e_cs =	11,60 mm
Shell internal diameter	Di = Id_head + 2(cs + cs') =	1 104,80 mm
Inside height of the dished end	h = H_head - e_head + cs + cs' =	277,40 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \dfrac{0.44D_i^2}{2h} + 0.02D_i$ =	990,11 mm

Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}}$ =	152,00 mm
Head's analysis thickness	ek =	11,60 mm
Head's knuckle radius	rk =	189,40 mm
Distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w(7,7,2) =	348,78 mm
Limit distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2	w_min(7.7.2)=2,5·√(ek·rk) =	117,18 mm

Distance between an opening and a shell discontinuity	w =	104,45 mm

Since nozzle is outside limits given in 9.7.2.4 and Figure 9.5.4, head's calculation will take this nozzle into account when calculating minimum required knuckle thickness, as defined in 7.7

Length of shell from the edge of the opening to a shell discontinuity	l_s = w =	104,45 mm
Minimum value for w which has no influence on ls from shell discontinuities	w_p = l_so =	152,00 mm
Required minimum value for w	w_min =	112,80 mm

Head's knuckle check due to encroaching nozzle

Knuckle thickness	enk(head) =	14,00 mm
Minimum knuckle thickness	ek(head) =	13,12 mm
		enk(head) ≥ ek(head): Ok

Adjacent openings

NOZZLE N3 (DN 200) - LWN Flange N1 - DN100 PN25

Centre-to-centre distance taken on the mean surface of the shell	Lb =	406,09 mm
Ligament length	L =	225,54 mm
Mean shell radius at the centres of adjacent nozzles	r_is =	990,11 mm
Minimum required ligament length	$L_{\min} = \max\left[3e_{a,s}; 0.2\sqrt{(2r_{is} + e_{c,s})e_{c,s}}\right]$ =	34,80 mm
	d1 =	219,10 mm
	d2 =	142,00 mm
		L ≥ Lmin: Ok

d1+d2 ≤ 0,2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required

Ligament check

Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{a,s}$ =	990,11 mm
Angle between the centre-to-centre line of openings and the generatrix of the shell	Φ =	90,00 °
Ap of the shell for the length Lb	$\mathrm{Ap}_{Ls} = \frac{0.5r_{is}^2 \cdot L_b(1+\cos(\phi))}{r_{is} + 0.5e_{a,s} \cdot \sin(\phi)}$ =	199 864,9 mm²
	deb1 =	219,10 mm
	deb2 =	142,00 mm
	ϕ_e1 =	0 °
	ϕ_e2 =	0 °
	$r_{os1} = \frac{r_{is1}}{\sin^2(\varphi)} + 0.5e_{a,s}$ =	995,91 mm
	$r_{os2} = \frac{r_{is2}}{\sin^2(\varphi)} + 0.5e_{a,s}$ =	995,91 mm
	$\delta_1 = \frac{d_{eb1}}{2r_{os1}}$ =	6,30 °
	$\delta_2 = \frac{d_{eb2}}{2r_{os2}}$ =	4,08 °
	$a_1 = r_{os1}\left[\arcsin(\delta_1 + \sin(\varphi_{e1})) - \varphi_{e1}\right]$ =	109,77 mm
	$a_2 = r_{os2}\left[\arcsin(\delta_2 + \sin(\varphi_{e2})) - \varphi_{e2}\right]$ =	71,06 mm
Af of the shell contained along the length Lb	$\mathrm{Af}_{Ls} = (L_b - a_1 - a_2) \cdot e_{c,s}$ =	2 612,9 mm²
	Apφ1 =	0 mm²
	Apφ2 =	0 mm²
	Apb1 =	5 724,8 mm²
	Apb2 =	3 117,6 mm²
	Afb1 =	582,2 mm²
	Afb2 =	1 126,9 mm²
	Afp1 =	0 mm²
	Afp2 =	0 mm²
	fob1 =	111,73 MPa
	fob2 =	134,00 MPa
	fop1 =	0 MPa
	fop2 =	0 MPa
Reactive force	$F = (\mathrm{Af}_{Ls} + \mathrm{Af}_w)(f_s - 0.5P) + \mathrm{Af}_{b1}(f_{ob1} - 0.5P) + \mathrm{Af}_{p1}(f_{op1} - 0.5P) + \mathrm{Af}_{b2}(f_{ob2} - 0.5P) + \mathrm{Af}_{p2}(f_{op2} - 0.5P)$ =	651 951 N
Pressure load	$F_{req} = P(\mathrm{Ap}_{Ls} + \mathrm{Ap}_{b1} + 0.5\mathrm{Ap}_{\varphi1} + \mathrm{Ap}_{b2} + 0.5\mathrm{Ap}_{\varphi2})$ =	417 415 N
		F ≥ Freq: Ok

C.16 Standard Long Welding Neck flange - LWN Flange N1 - DN100 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Flange material	**P280GH**	**(N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Shell material	**P280GH**	**(N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Bolting material	**25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**	
Gasket	**Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**	

Flange standard / specification	=	EN 1092-1:2007
Flange rating	=	25
Nominal size	=	100
Number of bolts	=	8
Bolt type	=	ISO M20 x 2,50
Material group	=	3E1
Calculation temperature	T =	260 °C
Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P = Pi + Ph =	2,00 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,39 MPa

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,50 MPa
Hot & corroded (flange)	=	2,39 MPa
New & cold (bolts)	=	2,50 MPa
Hot & corroded (bolts)	=	2,39 MPa
Maximum allowable test pressure	=	8,41 MPa
Maximum allowable design pressure	=	3,87 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,002 MPa
Calculation pressure	P = Pt + Ph =	2,96 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,50 MPa

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	134,00 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	201,00 MPa

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	19,85 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	147,05 MPa
Equivalent number of full pressure cycles	Neq =	469,90695
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Transition radius	r =	5,00 mm

Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	ΔσD =	175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,37794
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT]·Kf =	215,56 MPa
Number of allowable fatigue cycles	N =	370 582
Partial fatigue damage index	D=Nreq/N =	0,00324

Load condition 1, Nozzle without pad weld

Maximum allowable pressure (opening)	Pmax =	3,86 MPa
Calculation thickness	en =	19,85 mm
Stress factor	η =	3,00000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	182,26 MPa
Equivalent number of full pressure cycles	Neq =	111,83229
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	193,89 MPa
Number of allowable fatigue cycles	N =	68 662
Partial fatigue damage index	D=Nreq/N =	0,01748

Fatigue cycles and damage index summary

Load 1, partial damage index for Hub to plate junction	=	0,00324
Total damage index: Hub to plate junction	=	0,00324
Load 1, partial damage index for Nozzle without pad weld	=	0,01748
Total damage index: Nozzle without pad weld	=	0,01748
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.17 Cylindrical shell

Inside diameter	Di =	102,30 mm
Outside diameter	De =	142,00 mm
Length	L =	211,35 mm
Nominal thickness	en =	19,85 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	104,30 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	1,78 mm
		en ≥ e: Ok

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,002 MPa
Calculation pressure	Pc=Pt+Pht =	2,96 MPa
Inside diameter	Di'=Di+2δ =	102,30 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	0,59 mm
		en ≥ e: Ok

C.18 Standard Long Welding Neck flange - LWN Flange N2 - DN100 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Flange material **P280GH** **(N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Shell material **P280GH** **(N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Bolting material **25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**
Gasket **Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**

Flange standard / specification	=	EN 1092-1:2007
Flange rating	=	25
Nominal size	=	100
Number of bolts	=	8
Bolt type	=	ISO M20 x 2,50
Material group	=	3E1

Calculation temperature	T =	260 °C
Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P = Pi + Ph =	2,00 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,39 MPa

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,50 MPa
Hot & corroded (flange)	=	2,39 MPa
New & cold (bolts)	=	2,50 MPa
Hot & corroded (bolts)	=	2,39 MPa
Maximum allowable test pressure	=	8,41 MPa
Maximum allowable design pressure	=	3,87 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,003 MPa
Calculation pressure	P = Pt + Ph =	2,96 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,50 MPa

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	134,00 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	201,00 MPa

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	19,85 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	147,05 MPa
Equivalent number of full pressure cycles	Neq =	469,90695
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000

Transition radius	r =	5,00 mm
Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	ΔσD =	175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,37794
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT]·Kf =	215,56 MPa
Number of allowable fatigue cycles	N =	370 582
Partial fatigue damage index	D=Nreq/N =	0,00324

Load condition 1, Nozzle without pad weld

Maximum allowable pressure (opening)	Pmax =	3,86 MPa
Calculation thickness	en =	19,85 mm
Stress factor	η =	3,00000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	182,26 MPa
Equivalent number of full pressure cycles	Neq =	111,83229
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	193,89 MPa
Number of allowable fatigue cycles	N =	68 662
Partial fatigue damage index	D=Nreq/N =	0,01748

Fatigue cycles and damage index summary

Load 1, partial damage index for Hub to plate junction	=	0,00324
Total damage index: Hub to plate junction	=	0,00324
Load 1, partial damage index for Nozzle without pad weld	=	0,01748
Total damage index: Nozzle without pad weld	=	0,01748
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.19 Cylindrical shell

Inside diameter	Di =	102,30 mm
Outside diameter	De =	142,00 mm
Length	L =	249,65 mm
Nominal thickness	en =	19,85 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	104,30 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + \mathrm{ce} + \delta$ =	1,78 mm
		en ≥ e: Ok

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,003 MPa
Calculation pressure	Pc=Pt+Pht =	2,96 MPa
Inside diameter	Di'=Di+2δ =	102,30 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	0,59 mm
		en ≥ e: Ok

C.20 Nozzle - NOZZLE N3 (DN 200)

According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00

Material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTube

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	111,73 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	170,83 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	252,38 MPa

Geometry

Inside diameter	Di =	193,70 mm
Outside diameter	De =	219,10 mm
Length	L =	178,22 mm
Nominal thickness	en =	12,70 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	1,59 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	198,88 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	4,38 mm
		en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	5,61 MPa
Maximum allowable design pressure	=	2,44 MPa

Deformation according to EN 13445-4 Clause 9

Deformation	F=50·en/(Di/2+en/2) =	6,153 %

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,004 MPa
Calculation pressure	Pc=Pt+Pht =	2,96 MPa
Inside diameter	Di'=Di+2δ =	196,88 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	2,75 mm
		en ≥ e: Ok

Simplified fatigue assessment according to EN 13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	111,73 MPa

Ultimate tensile strength at room temperature	Rm =	410,00 MPa
Yield strength at design temperature	Rp0,2/T =	167,60 MPa

Load condition 1, Circumferential butt weld

Maximum allowable pressure (cylinder)	Pmax =	10,80 MPa
Calculation thickness	en =	12,70 mm
Joint efficiency	z =	1,00000
Offset	δ =	1,27 mm
Partial stress factor	η1=δ/(2·en) =	0,01969
Stress factor	η=(1+η1)·z =	1,01969
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	18,46 MPa
Equivalent number of full pressure cycles	Neq =	5,10094
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	40
Endurance limit	ΔσD =	29,48 MPa
Cut-off limit	Δσcut =	16,20 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	19,63 MPa
Number of allowable fatigue cycles	N =	38 155 258
Partial fatigue damage index	D=Nreq/N =	0,00003

Load condition 1, Nozzle without pad weld

Maximum allowable pressure (head)	Pmax =	3,43 MPa
Calculation thickness	en =	12,70 mm
Stress factor	η =	3,00000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	170,87 MPa
Equivalent number of full pressure cycles	Neq =	158,95492
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	181,78 MPa
Number of allowable fatigue cycles	N =	83 325
Partial fatigue damage index	D=Nreq/N =	0,01440

Fatigue cycles and damage index summary

Load 1, partial damage index for Circumferential butt weld	=	0,00003
Total damage index: Circumferential butt weld	=	0,00003
Load 1, partial damage index for Nozzle without pad weld	=	0,01440
Total damage index: Nozzle without pad weld	=	0,01440
		TDI(1)<1: Ok
		TDI(2)<1: Ok

Validation warnings:

- Since nozzle is outside limits given in 9.7.2.4 and Figure 9.5.4, head's calculation will take this nozzle into account when calculating minimum required knuckle thickness, as defined in 7.7 Code reference: 7.7

C.21 Standard Welding neck flange - STD Flange N3 - DN200 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Flange material	**P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Shell material	**P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTube**
Bolting material	**25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**
Gasket	**Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**

Flange standard / specification	=	EN 1092-1:2007
Flange rating	=	25

Nominal size	=	200
Number of bolts	=	12
Bolt type	=	ISO M24 x 3,00
Material group	=	3E1

Calculation temperature	T =	260 °C
Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	2,00 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,39 MPa

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,50 MPa
Hot & corroded (flange)	=	2,39 MPa
New & cold (bolts)	=	2,50 MPa
Hot & corroded (bolts)	=	2,39 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,0009 MPa
Calculation pressure	P = (Pt + Ph) / 1,43 =	2,07 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,50 MPa

Simplified fatigue assessment according to EN 13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	134,00 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	201,00 MPa

Load condition 1, Junction to shell (of thickness es)

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	12,70 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	147,05 MPa
Equivalent number of full pressure cycles	Neq =	469,90695
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	156,44 MPa
Number of allowable fatigue cycles	N =	130 725
Partial fatigue damage index	D=Nreq/N =	0,00918

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	16,30 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	147,05 MPa
Equivalent number of full pressure cycles	Neq =	469,90695
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Transition radius	r =	5,00 mm
Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	ΔσD =	175,20 MPa

Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,37794
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT]·Kf =	215,56 MPa
Number of allowable fatigue cycles	N =	370 582
Partial fatigue damage index	D=Nreq/N =	0,00324

Fatigue cycles and damage index summary

Load 1, partial damage index for Junction to shell (of thickness es)	=	0,00918
Total damage index: Junction to shell (of thickness es)	=	0,00918
Load 1, partial damage index for Hub to plate junction	=	0,00324
Total damage index: Hub to plate junction	=	0,00324
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.22 Conical shell - CONICAL SHELL

According to: EN 13445 Ed. 2009 Issue 5, Clauses 7 and 8

Design data

Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	0,85

Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	212,50 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	338,10 MPa

Geometry

Length	L =	475,00 mm
Nominal thickness	en =	12,00 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0,30 mm
Maximum Inside Diameter	Di =	1 100,00 mm
Maximum Outside Diameter	De =	1 124,00 mm
Minimum Inside Diameter	di =	150,00 mm
Minimum Outside Diameter	de =	174,00 mm
Half-apex angle	α =	45,00 °
Cone thickness at large end	e2nL =	12,00 mm
Cone thickness at small end	e2ns =	12,00 mm
Nominal thickness of cylinder at large end	e1nL =	12,00 mm
Minimum thickness of cylinder at large end	e1L =	8,90 mm
Nominal thickness of cylinder at small end	e1ns =	10,97 mm
Minimum thickness of cylinder at small end	e1s =	3,74 mm

Internal pressure

Overpressure due to static head	Ph =	0,003 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Mean diameter of the cone at large end	Dc=Di+e1nL+c+ce+δ =	1 113,30 mm
Calculation diameter	DK=Dc-e1L-2r[1-cos(α)]-l2·sin(α) =	1 012,43 mm
Minimum required cone thickness	e2 =	11,51 mm
Maximum allowable pressure of conical section at large end	Pmax(cone) =	2,08 MPa
		en≥e2: Ok

Large end junction (without knuckle)

Minimum length along cylinder	1,4·l1L=1,4·√(Dc*e1L) =	131,29 mm
Minimum length along cone	1,4·l2L=1,4·√((Dc*e2L)/cos(α)) =	177,52 mm
β factor defined in 7.6.6	7,6,6 =	1,46590
Minimum required thickness at the junction at the large end of the cone	e2L=ej=(P·Dc·β)/2f+c+ce+δ =	11,18 mm
Maximum allowable pressure of junction at large end	Pmax(large) =	2,17 MPa
		e1nL≥ej: Ok
		e2nL≥ej: Ok

Small end junction

Mean diameter of the cone	dc=di+e1ns+c+ce+δ =	162,27 mm
Minimum length along cylinder	l1s=√(dc*e1s') =	37,35 mm
Minimum length along cone	l2s=√((dc*e2s')/cos(α)) =	49,55 mm

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	3,73 MPa
Maximum allowable design pressure	=	2,08 MPa

Deformation according to EN 13445-4 Clause 9

Deformation	F=50·en/(di/2+en/2) =	7,407 %

Hydrostatic test

Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,03 MPa
Calculation pressure	Pc=Pt+Pht =	2,99 MPa
Mean diameter of the cone at large end	Dc=Di+e1nL+δ =	1 112,30 mm
Calculation diameter	DK=Dc-e1L-2r[1-cos(α)]-l2·sin(α) =	1 030,92 mm
Minimum required cone thickness	e2 =	6,74 mm
Maximum allowable pressure of conical section at large end	Pmax(cone) =	5,38 MPa
		en≥e2: Ok

Large end junction (without knuckle)

Minimum length along cylinder	1,4·l1L=1,4·√(Dc*e1L) =	139,29 mm
Minimum length along cone	1,4·l2L=1,4·√((Dc*e2L)/cos(α)) =	140,92 mm
β factor defined in 7.6.6	7.6.6 =	1.63938
Minimum required thickness at the junction at the large end of the cone	e2L=ej=(P·Dc·β)/2f+δ =	8,35 mm
Maximum allowable pressure of junction at large end	Pmax(large) =	4,34 MPa
		e1nL≥ej: Ok
		e2nL≥ej: Ok

Small end junction

Mean diameter of the cone	dc=di+e1ns+c+ce+δ =	161,27 mm
Minimum length along cylinder	l1s=√(dc*e1s') =	42,06 mm
Minimum length along cone	l2s=√((dc*e2s')/cos(α)) =	51,66 mm
s factor defined in 7.6.8	s=e2ns/e1ns =	1,07 mm
τ factor defined in 7.6.8	7.6-24/23 =	2,27 mm
βH factor defined in 7.6.8	7.6-25 =	1,18 mm
Minimum required thickness at the junction at the small end of the cone	e2s =	0,31 mm
Maximum allowable pressure of junction at small end	Pmax(small) =	39,12 MPa
		e1ns≥e2s: Ok
		e2ns≥e2s: Ok
		en ≥ e: Ok

Simplified fatigue assessment according to EN 13445-3 Clause 17
Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C

Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	165,33 MPa
Ultimate tensile strength at room temperature	Rm =	510,00 MPa
Yield strength at design temperature	Rp0,2/T =	248,00 MPa

Load condition 1, Longitudinal butt weld

Maximum allowable pressure (component)	Pmax =	2,08 MPa
Nominal thickness	en =	12,00 mm
Inside diameter	Di =	1 100,00 mm
Offset	δo =	0,60 mm
Peeking or flat	δpf =	2,00 mm
Ovality	u =	0,75%
Partial stress factor	η1=(3·δo)/en =	0,15000
Partial stress factor	η2=1,5·u·(Di/en) =	1,03125
Partial stress factor	η4=6·δpf/en =	1,00000
Stress factor	η=(1+η1+η2+η4)·z =	2,70406
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	375,73 MPa
Equivalent number of full pressure cycles	Neq =	712,30418
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	399,71 MPa
Number of allowable fatigue cycles	N =	7 837
Partial fatigue damage index	D=Nreq/N =	0,15312

Load condition 1, Circumferential butt weld

Joint efficiency	z =	0,85000
Maximum allowable pressure (component)	Pmax =	2,08 MPa
Calculation thickness	en =	12,00 mm
Joint efficiency	z =	0,85000
Partial stress factor	η0 =	0,10000
Offset	δ =	0,60 mm
Partial stress factor	η1=δ/(2·en) =	0,02500
Stress factor	η=(1+η0+η1)·z =	0,95625
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	132,87 MPa
Equivalent number of full pressure cycles	Neq =	712,30418
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	141,35 MPa
Number of allowable fatigue cycles	N =	177 214
Partial fatigue damage index	D=Nreq/N =	0,00677

Fatigue cycles and damage index summary

Load 1, partial damage index for Longitudinal butt weld	=	0,15312
Total damage index: Longitudinal butt weld	=	0,15312
Load 1, partial damage index for Circumferential butt weld	=	0,00677
Total damage index: Circumferential butt weld	=	0,00677
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.23 Cylindrical shell - NOZZLE N4 (DN150)

According to: EN 13445 Ed. 2009 Issue 5, Clauses 7 and 8

Design data		
Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	2,00 MPa
Joint efficiency	z =	1,00
Material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTube		
Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	111,73 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	170,83 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	252,38 MPa
Geometry		
Inside diameter	Di =	146,36 mm
Outside diameter	De =	168,30 mm
Length	L =	75,15 mm
Nominal thickness	en =	10,97 mm
Corrosion allowance	c =	1,00 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	1,37 mm
Internal pressure		
Overpressure due to static head	Ph =	0,003 MPa
Calculation pressure	P=Pi+Ph =	2,00 MPa
Inside diameter	Di'=Di+2δ+2c =	151,10 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	3,74 mm
		en ≥ e: Ok
Maximum allowable pressures (at the top of the vessel)		
Maximum allowable test pressure	=	30,49 MPa
Maximum allowable design pressure	=	12,03 MPa
Deformation according to EN 13445-4 Clause 9		
Deformation	F=50·en/(Di/2+en/2) =	6,973 %
Hydrostatic test		
Item minimum allowables ratio	Item f0/f =	1,18132
Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	Pt1=1,25·Pe·(Item f0/f) =	2,95 MPa
Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	Pt2=1,43·Pe =	2,86 MPa
Item hydrostatic test pressure	Pt=max(Pt1,Pt2) =	2,95 MPa
Overpressure due to static head in test condition	Pht =	0,03 MPa
Calculation pressure	Pc=Pt+Pht =	2,99 MPa
Inside diameter	Di'=Di+2δ =	149,10 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	2,26 mm
		en ≥ e: Ok
Simplified fatigue assessment according to EN 13445-3 Clause 17		
Load condition 1, load details		
Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	111,73 MPa
Ultimate tensile strength at room temperature	Rm =	410,00 MPa

Yield strength at design temperature	Rp0,2/T =	167,60 MPa

Load condition 1, Circumferential butt weld

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	111,73 MPa
Ultimate tensile strength at room temperature	Rm =	410,00 MPa
Yield strength at design temperature	Rp0,2/T =	167,60 MPa
Maximum allowable pressure (component)	Pmax =	12,03 MPa
Calculation thickness	en =	10,97 mm
Joint efficiency	z =	1,00000
Offset	δ =	0,60 mm
Partial stress factor	η1=δ/(2·en) =	0,02735
Stress factor	η=(1+η1)·z =	1,02735
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	16,70 MPa
Equivalent number of full pressure cycles	Neq =	3,69475
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	17,77 MPa
Number of allowable fatigue cycles	N =	Unlimited
Since Δσ* < Δσcut, the fatigue action of the cycles shall be ignored		
Partial fatigue damage index	D=0 =	0

Fatigue cycles and damage index summary

Load 1, partial damage index for Circumferential butt weld	=	0
Total damage index: Circumferential butt weld	=	0
		TDI(1)<1: Ok

C.24 Standard Welding neck flange - STD Flange N4 - DN150 PN25

According to: EN 13445 Ed. 2009 Issue 5, Clause 11

Flange material	**P280GH (N) (EN 10222-2:2001) t ≤ 35,00 mm- Forging**
Shell material	**P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTube**
Bolting material	**25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting**
Gasket	**Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)**

Flange standard / specification	=	EN 1092-1:2007
Flange rating	=	25
Nominal size	=	150
Number of bolts	=	8
Bolt type	=	ISO M24 x 3,00
Material group	=	3E1

Calculation temperature	T =	260 °C
Internal pressure	Pd =	2,00 MPa
Overpressure due to static head	Ph =	0,003 MPa
Calculation pressure	P =	2,00 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,39 MPa

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	2,50 MPa
Hot & corroded (flange)	=	2,39 MPa
New & cold (bolts)	=	2,50 MPa
Hot & corroded (bolts)	=	2,39 MPa

Hydrostatic test

Item hydrostatic test pressure	Pt =	2,95 MPa
Overpressure due to static head	Ph =	0,03 MPa
Calculation pressure	P = (Pt + Ph) / 1,43 =	2,09 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	2,50 MPa

Simplified fatigue assessment according to EN 13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	134,00 MPa
Ultimate tensile strength at room temperature	Rm =	460,00 MPa
Yield strength at design temperature	Rp0,2/T =	201,00 MPa

Load condition 1, Junction to shell (of thickness es)

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	10,97 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	147,26 MPa
Equivalent number of full pressure cycles	Neq =	471,85718
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	ΔσD =	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	156,66 MPa
Number of allowable fatigue cycles	N =	130 185
Partial fatigue damage index	D=Nreq/N =	0,00922

Load condition 1, Hub to plate junction

Maximum allowable pressure (flange)	Pmax =	2,39 MPa
Calculation thickness	en =	14,50 mm
Stress factor	η =	1,50000
Pseudo-elastic stress range	Δσ=(Δ,P/Pmax)·η·f =	147,26 MPa
Equivalent number of full pressure cycles	Neq =	471,85718
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Transition radius	r =	3,63 mm
Theoretical stress concentration factor	Kt=1,4 (r≥en/4) =	1,40000
Endurance limit	,σD =	175,20 MPa
Effective stress concentration factor	$K_f = 1+\frac{1.5(K_t-1)}{1+0.5\cdot\max(1; K_t\frac{\Delta\sigma}{\Delta\sigma_D})}$ =	1,37775
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[,σ/Ce·CT]·Kf =	215,83 MPa
Number of allowable fatigue cycles	N =	367 972
Partial fatigue damage index	D=Nreq/N =	0,00326

Fatigue cycles and damage index summary

Load 1, partial damage index for Junction to shell (of thickness es)	=	0,00922
Total damage index: Junction to shell (of thickness es)	=	0,00922
Load 1, partial damage index for Hub to plate junction	=	0,00326
Total damage index: Hub to plate junction	=	0,00326
		TDI(1)<1: Ok
		TDI(2)<1: Ok

C.25 Brackets - SUPPORT BRACKET

According to: EN 13445 Ed. 2009 Issue 5, Clause 16.10

Shell material:	**P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate**
Bracket material:	**P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate**

Nominal shell thickness	en =	10,00 mm
Corrosion allowance	c =	1,00 mm
Undertolerance	δ =	0,30 mm
Shell inside diameter	Di =	1 100,00 mm
Number of brackets	n =	4
Bracket type (Figure 16.10-1)	=	A
Web thickness	e3 =	15,00 mm
Baseplate thickness	e4 =	20,00 mm
Flange width of bracket	b1 =	190,00 mm
Thickness of reinforcing plate	e2 =	10,00 mm
Width of reinforcing plate	b2 =	240,00 mm
Height of reinforcing plate	b3 =	360,00 mm
Distance from centre of load to shell or reinforcing plate	a1 =	160,00 mm
Distance between webs of bracket	g =	150,00 mm
Height of bracket	h1 =	285,00 mm
Depth of bracket	h2 =	205,00 mm
Vertical distance from the centre of the bracket to the base of the leg	h =	142,50 mm

Loads

Center of geometry	hc =	1 841,11 mm
Center of gravity	hg =	1 792,13 mm
Exposed wind area	A =	3,48 m²
Wind pressure	Wp =	0 MPa
Horizontal seismic acceleration	Sh =	0 g
Vertical seismic acceleration	Sv =	0 g

	Erection	Hydrotest	Operating
Shear (wind)	0 N	0 N	0 N
Shear (earthquake)	0 N	0 N	0 N
Moment (wind)	0 N·m	0 N·m	0 N·m
Moment (earthquake)	0 N·m	0 N·m	0 N·m
Weight	1 521 kg	4 177 kg	1 652 kg
Vertical load	14 916 N	40 959 N	16 198 N

Applied forces

	Erection	Hydrotest	Operating
Vertical force acting at the base of the legs - F	14 916 N	40 959 N	16 198 N
Horizontal force acting at the base of the legs - FH	0 N	0 N	0 N
Moment at the centre-point of the cross section at the base of the legs - MA	0 N·m	0 N·m	0 N·m
Analysis thickness - ea	9,70 mm	9,70 mm	8,70 mm
Equivalent calculation diameter - Deq	1 100,00 mm	1 100,00 mm	1 102,60 mm
$F_{Vi} = \frac{F}{n} + \frac{4M_A}{n[D_i + 2(a_1 + e_a + e_2)]}$	3 729 N	10 240 N	4 050 N
$F_{Hi} = \frac{F_H}{n}$	0 N	0 N	0 N

Load limits of the shell

	Erection	Hydrotest	Operating
Maximum nominal design stress - f	212,50 MPa	338,10 MPa	165,33 MPa
Calculation pressure	0 MPa	2,98 MPa	2,00 MPa
Calculation temperature	20,00 °C	20,00 °C	260,00 °C
$\lambda = \frac{b_3}{\sqrt{D_{eq} e_a}}$	3,48514	3,48514	3,67565
$K_{17} = \frac{1}{\sqrt{0.36 + 0.5\lambda + 0.5\lambda^2}}$	0,34973	0,34973	0,33421
$\sigma_m = \frac{P \cdot D_{eq}}{2e_a}$	0 MPa	168,71 MPa	126,74 MPa
$\upsilon_1 = \min[0.08\lambda; 0.40]$	0,27881	0,27881	0,29405
$\upsilon_2 = \frac{\sigma_m}{K_2 f}$	0	0,47525	0,61324
$K_1 = \frac{1 - \upsilon_2^2}{\left(\frac{1}{3} + \upsilon_1\upsilon_2\right) + \sqrt{\left(\frac{1}{3} + \upsilon_1\upsilon_2\right)^2 + (1 - \upsilon_2^2)\upsilon_1^2}}$	1,30226	0,78013	0,57912
K2	1,25	1,05	1,25
$\sigma_{b,all} = K_1 K_2 f$	345,91 MPa	276,95 MPa	119,69 MPa
$a_{1,eq} = a_1 + e_2 + \frac{F_{Hi} \cdot h}{F_{Vi}}$	170,00 mm	170,00 mm	170,00 mm
$F_{i,max} = \left(\frac{\sigma_{b,all} \cdot e_a{}^2 \cdot b_3}{K_{17} \cdot a_{1,eq}}\right)$	197 072 N	157 780 N	57 401 N
	FVi ≤ Fi,max: OK	FVi ≤ Fi,max: OK	FVi ≤ Fi,max: OK

Simplified fatigue assessment according to EN 13445-3 Clause 17

Load condition 1, load details

Design pressure	P =	2,00 MPa
Pressure range	ΔP =	1,75 MPa
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1 200
Nominal design stress at design temperature	f =	165,33 MPa
Ultimate tensile strength at room temperature	Rm =	510,00 MPa
Yield strength at design temperature	Rp0,2/T =	248,00 MPa

Load condition 1, Bracket or support weld

Maximum allowable pressure (shell)	Pmax =	2,20 MPa
Joint efficiency of shell	z =	0,85
Stress factor	η=2·z =	1,70
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f =	223,54 MPa
Equivalent number of full pressure cycles	Neq =	603,65734
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	T*=0,75·Tmax+0,25·Tmin =	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	71
Endurance limit	ΔσD =	52,33 MPa
Cut-off limit	Δσcut =	28,76 MPa
Fictitious stress range for insertion into the fatigue design curves	Δσ*=[Δσ/Ce·CT] =	237,80 MPa
Number of allowable fatigue cycles	N =	53 271
Partial fatigue damage index	D=Nreq/N =	0,02253

Fatigue cycles and damage index summary

Load 1, partial damage index for Bracket or support weld	=	0,02253
Total damage index: Bracket or support weld	=	0,02253
		TDI(1)<1: Ok

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