PD CEN/TR 13445-102:2015



BSI Standards Publication

Unfired pressure vessels

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



National foreword

This Published Document is the UK implementation of CEN/TR 13445-102:2015.

The UK participation in its preparation was entrusted to Technical Committee PVE/1, Pressure Vessels.

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

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

© The British Standards Institution 2015. Published by BSI Standards Limited 2015

ISBN 978 0 580 87643 1

ICS 23.020.30

Compliance with a British Standard cannot confer immunity from legal obligations.

This Published Document was published under the authority of the Standards Policy and Strategy Committee on 30 September 2015.

Amendments issued since publication

Date Text affected

TECHNICAL REPORT RAPPORT TECHNIQUE TECHNISCHER BERICHT

CEN/TR 13445-102

May 2015

ICS 23.020.30

English Version

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

Unbefeuerte Druckbehälter - Beispiel 2: Stehende Behälter mit Tragpratzen

This Technical Report was approved by CEN on 10 February 2015. It has been drawn up by the Technical Committee CEN/TC 54.

CEN members are the national standards bodies of Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, Former Yugoslav Republic of Macedonia, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and United Kingdom.



EUROPEAN COMMITTEE FOR STANDARDIZATION COMITÉ EUROPÉEN DE NORMALISATION EUROPÄISCHES KOMITEE FÜR NORMUNG

CEN-CENELEC Management Centre: Avenue Marnix 17, B-1000 Brussels

Contents Page

Forev	vord	3
Introd	luction	4
3.1	Drawing of the vessel	
3.2	Calculation model	6
3.3	Operating conditions	7
3.4	Comments on the operating conditions provided by the User	7
4.1	General	8
4.2	Is EN13445 applicable to the vessel?	8
4.3	Warning of Annex A of reference [1]	8
4.4	Prerequisites of Annex A of reference [1]	
5.1	Permitted materials	10
5.2	Requirements given in 4.2 of reference [2]	11
5.3	Requirements given in 4.3 of reference [2]	
5.4	Requirements given in 4.4 of reference [2]	11
5.5	Materials selected for the vessel example 2	12
6.1	General	
6.2	Basic design	15
6.3	Fatigue calculations	
6.4	Determination of test pressures of the vessel in Annex C	17
6.5	Determination of the deformation according to EN 13445-4 reference [4], Clause 9 in	
	Annex C	19
6.6	Data used in example 2	
6.7	Conditions of applicability of calculations	20
7.1	General	20
7.2	Material traceability	20
7.3	Manufacturing tolerances	
7.5	Welding, as in 8 of reference [4]	
7.6	Manufacture and testing of welds – Production test, as in 8 of reference [4]	
7.7	Forming of pressure parts, as in 9 of reference [4]	
7.8	Post weld heat treatment (PWHT), as in 10 of reference [4]	
8.1	Generality	
8.2	Non destructive testing, as in 4.3 of reference [5]	
8.3	Determination of extent of non-destructive testing, as in 6.6.2 of reference [5]	28
Anne	x A (informative) Drawing of example 2	29
Anne	x B (informative) Nameplate of example 2	30
Δnne	x C (informative) Design calculation of example 2	31

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.

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

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.

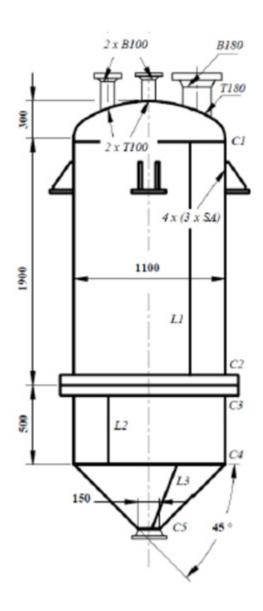


Figure 1 — Briefed lay-out

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

 $\underline{\text{https://osha.europa.eu/nl/legislation/directives/exposure-to-chemical-agents-and-chemical-safety/osh-related-aspects/58}$

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;

PD CEN/TR 13445-102:2015

CEN/TR 13445-102:2015 (E)

- 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);
- 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):
- I) 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).

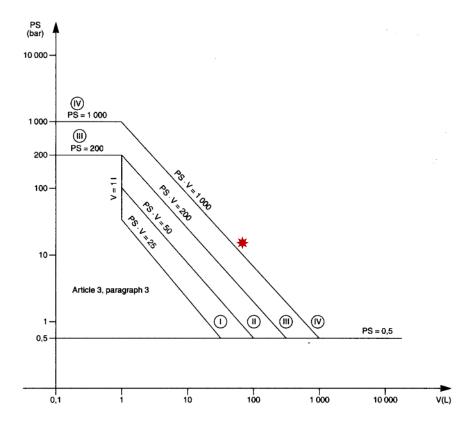


Figure 3 — Vessel category

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.

CEN/TR 13445-102:2015 (E)

Table 1 — Materials and main material characteristics in example 2

Vessel part	Material	Material	EN 13445-2	Dimensions	Material	Main mate	rial characteristics			
		designation	reference	(mm) (see also Annex C)	group to - CR ISO 15608	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(fastene rs) main flange:	25CrMo4(+Q T)	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	— а)					

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

Vessel part	Material group	Material designation		Dimensions (mm) (see also Annex C)	group to CR ISO	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 $e_n=15,\text{base}$ 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

 $T = 260 \, ^{\circ}\text{C}$

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 *P*max 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, *P*max 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 $\Delta \sigma$

 $\Delta \sigma$ shall be calculated from the pressure range ΔP as follows:

$$\Delta \sigma = \frac{\Delta P}{P \max} \cdot \eta \cdot f \tag{17.6-1}$$

where

- Pmax 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 Pmax 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:

maximum structural stress in detail under consideration under pressure P_{max}

nominal design stress at calculation pressure

Where $\Delta \sigma > 3 f$, $\Delta \sigma$ 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: $C_{\rm e}$ and $C_{\rm T}$ at a welded joint or vessel part and also the effective stress concentration factor $K_{\rm f}$ for notch effect at an unwelded part. These factors are defined in 17.6.2

6.3.5.2 At a welded joint

$$\Delta \sigma^* = (\frac{\Delta \sigma}{C_{\rm e} \cdot C_{\rm T}}) \tag{17.6-9}$$

6.3.5.3 At a unwelded region

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

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)

$$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \text{etc...} = \sum_{i=1}^{k} \frac{n_i}{N_i}$$
 (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:

$$P_{t} = 1,25 \cdot P_{d} \cdot \frac{f_{a}}{f_{T_{d}}}$$
(10.2.3.3.1-1)

or

$$P_{\rm t} = 1{,}43 \cdot P_{\rm s} \tag{10.2.3.3.1-2}$$

where:

 $P_{\rm t}$ is the test pressure measured at the highest point of the chamber of the vessel in the test position;

PD CEN/TR 13445-102:2015

CEN/TR 13445-102:2015 (E)

 $P_{
m d}$ and $T_{
m d}$ are the coincident design pressure and design temperature values for the maximum pressure load case:

 $P_{\rm s}$ is the maximum allowable pressure of the vessel;

 $f_{\rm a}$ is the nominal design stress for normal operating load cases of the material of the part under consideration at the test temperature;

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

Since the ratio $\frac{f_{\rm a}}{f_{T_{\rm d}}}$ depends on the material of the part under consideration, the value $\frac{f_{\rm a}}{f_{T_{\rm d}}}$ to be used for

calculation of $P_{\rm t}$ 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 $P_{\rm t_1}$ using the formula $P_{\rm t_1} = 1,25 \cdot P_{\rm d} \cdot \frac{f_{\rm a}}{f_{T_{\rm d}}}$

6.4.2.3 Step 3

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

6.4.2.4 Step 4

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

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:

PD CEN/TR 13445-102:2015 **CEN/TR 13445-102:2015 (E)**

- The pressure range given in the data of example 2;
- The maximum permissible pressure as defined in 3.16, using the analysis thickness $e_a = e_0 c$;
- 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**.

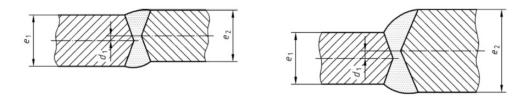
These tolerances can be inferior to those for predominantly non cyclic loads. This is the case for peaking on longitudinal butt welds.

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 $e_1 \le e_2$

Figure 5.2-1 — Middle line alignment d₁

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):

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

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 $e/D \ge 0,01$.

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 $\it e$	Maximum permitted peaking P
<i>e</i> ≤ 3	1,5
3 ≤ e < 6	2,5
6 ≤ <i>e</i> < 9	3,0
9 ≤ <i>e</i>	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 conditions (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).

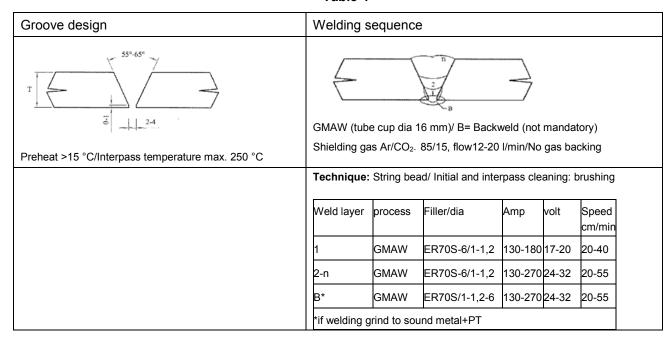
Table 3 — Excerpt of Table A-1 of reference [3] — Pressure bearing welds - Longitudinal welds in cylinders and cones, welds in spheres and dished ends

Ref.	Type of joints	Design requirements	Applicable weld testing group	Fatigue class 1)	Lamellar tearing susceptibility	Corrosion 3)	EN 1708 -1:1998
M 1	e_1 e_1		1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.2	A	N	1.1.4
M 10	e_1 e_1	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

- 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



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	Corrosion 3)	EN 1708- 1:1998
C 1	e, e,		1, 2, 3, 4	see Table 18-4 details n° 1.1 and 1.2	A	N	1.1.4
C 11	e_1 e_1	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	$\alpha > 30^{\circ}$	in case of unequal thicknesses, limited to: $e_2 - e_1 \leq \mathrm{Min} \left[0.3 e_1 \; ; 4 \right]$ — calculation of stresses — round the weld inside by grinding	1, 2, 3, 4	see Table 18-4 detail n° 1.4	A	N	-

PD CEN/TR 13445-102:2015 **CEN/TR 13445-102:2015 (E)**

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.

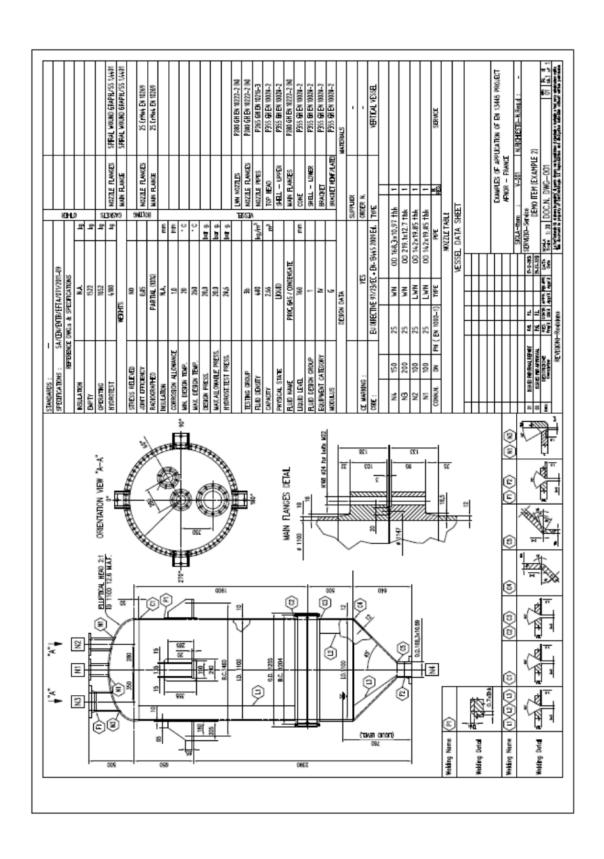
The Non-destructive methods included in the Table are:

For example 2, UT and PT will be used.

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

Annex A (informative)

Drawing of example 2



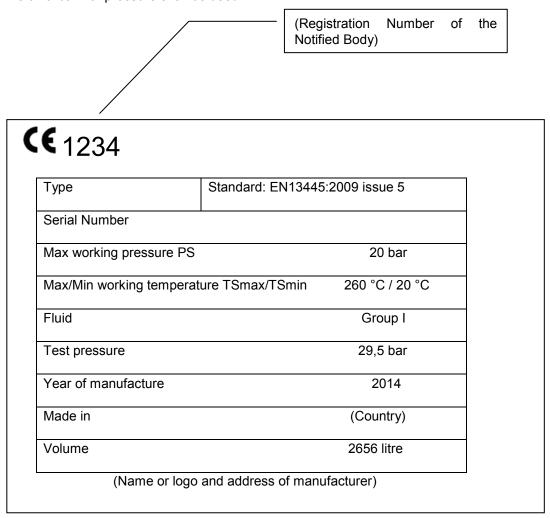
Annex B (informative)

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.



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	<i>T</i> =	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

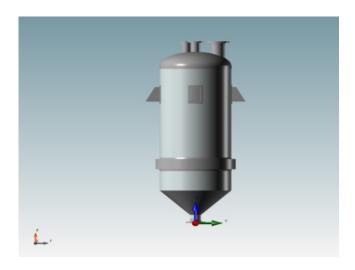


Figure C.1

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

		1		ii pressures iii wpa
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

32

C.5 Bill of materials

Component	Dimensions	Material
-	Id = 1 100,00 mm, Od = 1 124,00 mm,	P355GH (EN 10028-2:2009)
SHELL_ LOWER PORTION	Tk = 12,00 mm, L = 377,25 mm	t ≤ 16,00 mm - Plate
MAIN FLANGE_LOWER SIDE - Flange	ld = 1100,00 mm, Od = 1255,00 mm, Tk = 95,00 mm	P280GH (NT,QT) (EN 10222-2:2001) 50,001 \leq t \leq 160 - Forging
MAIN FLANGE_LOWER SIDE - Gasket	Spiral-wound metal asbestos or mineral f	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 = 1724,25 mm	P355GH (EN 10028-2:2009) t ≤ 16,00 mm - Plate
MAIN FLANGE UPPER SIDE - Flange	ld = 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	ld = 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 f	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	ld = 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 f	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	ld = 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 f	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	ld = 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 f	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		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

Design data Internal design temperature Internal design pressure Joint efficiency	Ti = Pi = z =	
Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/2.0}}{2.4}) =$	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{\text{p0.2/20}}}{15}; \frac{R_{m/20}}{24}) =$	212,50 MPa
Nominal design stress in test condition	$f_{\text{test}} = (\frac{R_{\text{p0.2}/\text{Ttest}}}{105})^{=}$	338,10 MPa
Geometry Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance	Di = De = L = en = c = ce = δ =	1 100,00 mm 1 124,00 mm 377,25 mm 12,00 mm 1,00 mm 0 mm 0 mm
Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness	$Ph = P=Pi+Ph = Di'=Di+2\delta+2c = e = \frac{P \cdot D'_i}{2f \cdot z - P} + c + ce + \delta = e$	2,00 MPa
Maximum allowable pressures (at the top of the vessel) Maximum allowable test pressure Maximum allowable design pressure	= =	7,27 MPa 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 Test pressure as per EN13445-5 Formula (10.2.3.3.1-1) Test pressure as per EN13445-5 Formula (10.2.3.3.1-2) Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness	$ \text{Item f0/f} = \\ \text{Pt1=1,25} \cdot \text{Pe} \cdot (\text{Item f0/f}) = \\ \text{Pt2 = 1,43} \cdot \text{Pe} = \\ \text{Pt = max(Pt1,Pt2)} = \\ \text{Pht = } \\ \text{Pc = Pt+Pht = } \\ \text{Di' = Di+2\delta} = \\ e = \frac{P \cdot D_i^t}{2f \cdot z - P} + \delta = \\ \end{aligned}$	1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,03 MPa 2,98 MPa 1 100,00 mm 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 = T =	260 °C 260 °C
Design temperature Number of required fatigue cycles	Nreg =	1200
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 Peeking or flat	δο = δpf =	0,60 mm 2,00 mm
Ovality	u =	0,75 %
Partial stress factor	η1=(3·δο)/en =	0,15000
Partial stress factor	η2=1,5·u·(Di/en) =	1,03125
Partial stress factor	η 4=6· δ pf/en =	1,00000
Stress factor	$\eta = (1 + \eta 1 + \eta 2 + \eta 4) \cdot z =$	2,70406
Pseudo-elastic stress range	Δσ=(ΔP/Pmax)·η·f = Neg =	281,70 MPa 300,21128
Equivalent number of full pressure cycles 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	$\Delta \sigma D = \Delta \sigma $	46,43 MPa
Cut-off limit Fictitious stress range for insertion into the fatigue design curves	Δσcut = Δσ*=[Δσ/Ce·CT] =	25,52 MPa 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 Offset	z = δ =	0,85000
Partial stress factor	η1=δ/(2·en) =	0,60 mm 0,02500
Stress factor	$\eta = (1+\eta 1) \cdot z =$	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 Temperature correction factor	T*= 0,75·Tmax+0,25·Tmin = CT =	200 °C 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	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	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(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 \leq t \leq 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	Flange - f	Hub - fH	Bolting - fB
stresses			
Design condition	121,33 MPa / 17 597,9 psi	165,33 MPa / 23 979,6 psi	128,73 MPa / 18671,2 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 = Gmean =	95,00 mm 1 147,00 mm
Mean gasket diameter Hub length	h =	35,00 mm
Thickness of hub at back of flange	q1 =	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
This knows of his at small cha (contouct)	90	11,00 111111
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	$b = 2.52 \cdot \sqrt{(b0)} =$	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 mm ²
Distance between centre lines of adjacent bolts	δb =	55,62 mm
Bolt outside diameter	db =	22,00 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{A} =$	2 081 217 N
Compression load on gasket to ensure tight joint	$HG = 2\pi \cdot G \cdot b \cdot m \cdot P =$	345 804 N
Minimum required bolt load for operating condition	Wop = H + HG =	2 427 022 N
Millimum required boil load for operating condition	VVOP = 11 + 11G =	3 099 036 N
	$H_t = \frac{G^2 \pi P_t}{4} =$	3 099 030 N
Minimum required bolt load for the test condition	Wt = Ht + $2b \cdot \pi \cdot G \cdot m \cdot Pt$ =	3 613 956 N
Minimum required bolt load for assembly condition	$WA = \pi b \cdot G \cdot y =$	1 988 375 N
Total required cross-sectional area of bolts	W_A , W_{op} , W_t =	18 853,1 mm ²
·	$A_{B,min} = \max \left[\frac{W_A}{f_{E,A}}, \frac{W_{op}}{f_B}, \frac{W_t}{f_{E,t}} \right] =$	
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	19 176,0 mm ²
Maximum bolts area for gasket crush	$A \cdot \text{max} = \frac{2\pi \cdot y \cdot G \cdot N}{1 + 2\pi \cdot y \cdot G \cdot N} = \frac{2\pi \cdot y \cdot G \cdot N}{1 + 2\pi \cdot y} = 2\pi \cdot $	68 049,7 mm²
	AB =	0.700.000
Design bolt load for assembly condition	$W = 0.5 (A_{B,\min} + A_B) J_{B,A} =$	2 788 800 N
		B ≥ AB,min: Ok

PD CEN/TR 13445-102:2015

Flange constants Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{\delta_\theta}{m + 0.5}}}; 1\right] =$	1,00000
Ratio of the flange diameters Length parameter	$\begin{bmatrix} \sqrt{2d_b} + \frac{60}{m+0.5} \end{bmatrix}$ $K = A / B^* = 10$ $l_0 = \sqrt{B^* \cdot g_0^*} = 1$	1,13884 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 Radial distance from bolt circle to circle on which HD acts Radial distance from gasket load reaction to bolt circle Radial distance from bolt circle to circle on which HT acts	HT = H - HD = hD = (C - B* - g1*) / 2 = hG = (C - G) / 2 = hT = (2C - B* - G) / =	173 636 N 42,25 mm 26,47 mm 38,73 mm
Flange stress factor	$\beta_T = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448 K^2)(K - 1)} = $	1,86292
Flange stress factor	$\beta_U = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{136136(K^2 - 1)(K - 1)} =$	16,44915
Flange stress factor $oldsymbol{eta}_{_{Y}}$ =	$= \frac{1}{K-1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right] =$	14,96876
Hub stress correction factor	$\beta F = \beta V = \phi =$	0,87349 0,34643 1,24196 2,29678
Flange moments		
Total moment acting upon flange for assembly condition Total moment acting upon flange for operating condition Moment factor used to design split rings Moment exerted on the flange per unit of length (operating)	$\begin{aligned} \text{MA} &= \text{W} \cdot \text{hG} = \\ M_{\text{op}} &= H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G = \\ & \text{Fs} = \\ M &= F_s \cdot M_{\text{op}} \frac{C_F}{B^*} = \\ M &= F_s \cdot M_A \frac{C_F}{B^*} = \end{aligned}$	73 816,6 N·m 96 474,1 N·m 1,00 87 5 N·m
Moment exerted on the flange per unit of length (assembly)	$M = F_s \cdot M_{\text{op}} \frac{\delta}{B^*}$ $M = F_s \cdot M_s \frac{C_F}{D^*} =$	67,0 N·m
	. "B"	
Flange stresses - operating condition Longitudinal stress in hub	$\sigma_{H} = \frac{\varphi M}{\lambda g_{1}^{*2}} =$ $\sigma_{r} = \frac{\left(1333e \cdot \beta_{F} + l_{0}\right)M}{\lambda \cdot e^{2} \cdot l_{0}} =$	154,58 MPa
Radial stress in flange	$\sigma = \frac{(1333e \cdot \beta_F + l_0)M}{(1333e \cdot \beta_F + l_0)M} =$	8,47 MPa
Tangential stress in flange	$\sigma_{\theta} = \frac{\beta_{\underline{Y}} \cdot M}{e^2} - \sigma_{\underline{K}^2 - 1} = \sigma_{\theta} = \frac{\beta_{\underline{Y}} \cdot M}{e^2} - \sigma_{\underline{K}^2 - 1} = \sigma_{\theta} = \sigma_{\theta}$	79,71 MPa
Stress factor	0,5k	1,03400 1,5min(f;fH): Ok $k \cdot \sigma r \le f$: Ok $k \cdot \sigma \theta \le f$: Ok $(\sigma H + \sigma r) \le f$: Ok $\sigma H + \sigma \theta \ge f$: Ok
	0,5%	5.1 · 50/ = 1. OK
Flange stresses - seating condition Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda_{\mathcal{O}}^{*2}} =$	118,27 MPa
Radial stress in flange	$\sigma_{H} = \frac{\varphi M}{\lambda g_{1}^{*2}} = $ $\sigma_{r} = \frac{\left(1333e \cdot \beta_{F} + l_{0}\right) M}{\lambda \cdot e^{2} \cdot l_{0}} = $	6,48 MPa
Tangential stress in flange	$\sigma_{\theta} = \frac{\beta_{\gamma} \cdot M}{e^2} - \sigma_{r} \frac{K^2 + 1}{K^2 - 1} =$	60,99 MPa
	0,5ki	1,5min(f;fH): Ok k·σr ≤ f: Ok k·σθ ≤ f: Ok (σH + σr) ≤ f: Ok σH + σθ) ≤ f: Ok

Maximum allowable massaures (at the top of the vessell		
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	$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m + 0.5}}}; 1 \right] =$	1,00000
Ratio of the flange diameters		1,14091
Length parameter	$K = A / B = I_0 = \sqrt{B \cdot g_0^*} = H_D = \frac{\pi}{4} (B^2 P) =$	114,89 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4} (B^2 P) =$	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	$\beta_T = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} =$	1,86214
Flance stress factor	$V^2/1 + 855246 \log V = 1 = 1$	16 22655
Flange stress factor	$\beta_U = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)} =$	16,22655
Flange stress factor	$\beta_{v} = \frac{1}{K - 1} \left[0.66845 + 5.7169 \left(\frac{K^{2} \log_{10} K}{V^{2} - 1} \right) \right] =$	14,76619
3	· (
	βF =	0,87709
	βV =	0,36048
Hub stress correction factor	$\varphi = \begin{pmatrix} aR + L & a^3R \end{pmatrix}$	1,20489
	$\lambda = \left(\frac{e\beta_F + I_0}{\beta_{P}I_0} + \frac{e^3\beta_V}{\beta_{P}I_0g^{*2}}\right) =$	2,07775
	(11 7 7 000 7	
Flange moments	$M = \{m, L + m, L + m, L + m, L \}$	440.004.7.1
Total moment acting upon flange for operating condition	$M_{\text{op}} = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G =$	140 691,7 N·m
Moment factor used to design split rings Moment exerted on the flange per unit of length (operating)	Fs = C_{π} _	1,00 127,9 N·m
Moment exerted on the hange per unit or length (operating)	$M = F_s \cdot M_{op} \frac{C_F}{B} =$	127,9 N·III
Flange stresses - operating condition		
Longitudinal stress in hub	$\sigma_H = \frac{\phi M}{10^{*2}} =$	216,71 MPa
Radial stress in flange	$(1333e^{\beta_F + l_0})^{M} =$	13,41 MPa
	$\sigma_r = \frac{\lambda \cdot e^2 \cdot l_0}{\lambda \cdot e^2 \cdot l_0}$,
Tangential stress in flange	$\sigma_{H} = \frac{\varphi M}{\lambda g_{1}^{*2}} =$ $\sigma_{r} = \frac{\left(1333e \cdot \beta_{F} + l_{0}\right) M}{\lambda \cdot e^{2} \cdot l_{0}} =$ $\sigma_{\theta} = \frac{\beta_{Y} \cdot M}{e^{2}} - \sigma_{r} \frac{K^{2} + 1}{K^{2} - 1} =$	106,92 MPa
Stress factor	k =	1,03333
	k·σH≤	1,5min(f;fH): Ok
		k·σr ≤ f: Ok
	0.56	$k \cdot \sigma \theta \le f$: Ok
		$(\sigma H + \sigma r) \le f$: Ok $(\sigma H + \sigma \theta) \le f$: Ok
	0,5%	(3.1 · 30) = 1. OK

Simplified fatigue assessment according to EN13445-3 Clause 17

Load condition 1, load details	_	0.00.145
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	$\Delta \sigma = (\Delta P / P max) \cdot \eta \cdot f =$	158,98 MPa
Equivalent number of full pressure cycles	Neq =	799,82023
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T*=0,75\cdot Tmax+0,25\cdot Tmin =$	200 °C
Temperature correction factor	CT =	0,94000
Weld class	C =	63
Endurance limit	$\Delta \sigma D =$	46,43 MPa
Cut-off limit	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	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	$\Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f =$	158,98 MPa
Equivalent number of full pressure cycles	Neq =	799,82023
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T^*=0,75\cdot Tmax+0,25\cdot Tmin =$	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	$\Delta \sigma D =$	175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{15(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{\Delta \sigma}{\Delta \sigma_0})} =$	1,36693
Cut-off limit	Δσcut =	116,70 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] \cdot \text{Kf} =$	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
- · · · · · · · · · · · · · · · · · · ·	=	0,00472 0,00472
Load 1, partial damage index for Hub to plate junction	= =	0,00472

C.10 Cylindrical shell - SHELL_ UPPER PORTION

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

Number of required fatigue cycles

Nominal design stress at design temperature

Ultimate tensile strength at room temperature

Design data Internal design temperature Internal design pressure Joint efficiency	Ti = Pi = z =	260 °C 2,00 MPa 0,85
Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate Nominal design stress at internal design temperature	$f = \min(\frac{R_{\text{p0.2}/T}}{1.5}; \frac{R_{\text{m/20}}}{2.4}) =$	165,33 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{\text{p0.2/20}}}{R_{\text{p0.2/20}}} \cdot \frac{R_{m/20}}{R_{m/20}}) =$	212,50 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{\text{p0.2/20}}}{1.5}; \frac{R_{\text{m/20}}}{2.4}) = f_{\text{test}} = (\frac{R_{\text{p0.2/Ttest}}}{1.05}) = f_{\text{test}}$	338,10 MPa
Geometry Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance	Di = De = L = en = c = ce = δ =	1 100,00 mm 1 120,00 mm 1 724,25 mm 10,00 mm 1,00 mm 0 mm 0,30 mm
Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness	$Ph = P=Pi+Ph = Di'=Di+2\delta+2c = e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta = e$	0 MPa 2,00 MPa 1 102,60 mm 9,20 mm en ≥ e: Ok
Maximum allowable pressures (at the top of the vessel) Maximum allowable test pressure Maximum allowable design pressure	= =	5,88 MPa 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 Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause	$ \text{Item f0/f} = \\ \text{Pt1=1,25} \cdot \text{Pe} \cdot (\text{Item f0/f}) = \\ \text{Pt2=1,43} \cdot \text{Pe} = \\ \text{Pt=max}(\text{Pt1,Pt2}) = \\ \text{Pht} = \\ \text{Pc=Pt+Pht} = \\ \text{Di'=Di+2} \delta = \\ e = \frac{P \cdot D_i!}{2f \cdot z - P} + \delta = \\ \\ \textbf{17}$	1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,02 MPa 2,98 MPa 1 100,60 mm 5,16 mm en ≥ e: Ok
Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle Design temperature Number of required fatigue cycles	P = ΔP = Tmin = Tmax = T = Nreg =	2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C

1 200

165,33 MPa

510,00 MPa

Nreq =

Rm =

f =

Yield strength at design temperature	Rp0,2/T =	248,00 MPa
Load condition 1, Longitudinal butt weld Maximum allowable pressure (component) Nominal thickness	Pmax = en =	2,20 MPa 10,00 mm
Inside diameter Offset	Di = δο =	1 100,00 mm 0,50 mm
Peeking or flat Ovality	δpf = u =	1,67 mm 0,75 %
Partial stress factor Partial stress factor	η1=(3·δο)/en = η2=1,5·u·(Di/en) =	0,15000 1,23750
Partial stress factor Stress factor	η4=6·δpf/en = η=(1+η1+η2+η4)·z =	1,00020 2,87955
Pseudo-elastic stress range Equivalent number of full pressure cycles	$\Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f = Neq =$	378,63 MPa 603,65734
Thickness correction factor Assumed mean cycle temperature	Ce = T*=0,75·Tmax+0,25·Tmin =	1,00000 200 °C
Temperature correction factor Weld class	CT = C =	0,94000 63
Endurance limit Cut-off limit	$\Delta \sigma D = \Delta \sigma cut = \Delta \sigma cu$	46,43 MPa 25,52 MPa
Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	Δσ*=[Δσ/Ce·CT] = N = D=Nreq/N =	402,80 MPa 7 658 0,15670
Load condition 1, Circumferential butt weld	В-тисулу -	0,10070
Joint efficiency	Z =	0,85000
Maximum allowable pressure (component) Calculation thickness	Pmax = en =	2,20 MPa 10,00 mm
Joint efficiency Partial stress factor Offset	z = η0 = δ =	0,85000 0,10000
Partial stress factor	η1=δ/(2·en) =	0,50 mm 0,02500
Stress factor Pseudo-elastic stress range Equivalent number of full programs evalue	$ η=(1+η0+η1)\cdot z = Δσ=(ΔP/Pmax)\cdot η\cdot f = Nog = $	0,95625 125,74 MPa
Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature	Neq = Ce = T*=0,75·Tmax+0,25·Tmin =	603,65734 1,00000 200 °C
Temperature correction factor Weld class	CT = C =	0,94000 63
Endurance limit Cut-off limit	ΔσD = Δσcut =	46,43 MPa 25,52 MPa
Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] = N =$	133,76 MPa 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 Load 1, partial damage index for Circumferential butt weld	= =	0,15670 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) $50,001 \le t \le 160$ - Forging

Shell material P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate **Bolting material** 25CrMo4 (EN 10269:2009) t ≤ 100,00 mm- Bolting

Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel) Gasket

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

Test condition	204,76 MPa / 29 698,2 psi	338,10 MPa / 49 036,6 psi	220,00 MPa / 3	31 908,3 psi
Internal pressure			Pd =	2,00 MPa
Overpressure due to	o static head		Ph =	0 MPa
Calculation pressure	e		P =	2,00 MPa
Calculation tempera	ature		T =	260 °C
Corrosion allowance	е		C =	1,00 mm
Flange external diar	meter		A =	1 255,00 mm
Inside diameter			B =	1 100,00 mm
Inside diameter (cor	rroded)		$B^* = B + 2c =$	1 102,00 mm
Bolt circle			C =	1 204,00 mm
Flange thickness			e =	103,00 mm
Mean gasket diame	ter		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 paramete	ers			
Gasket factor	- -		m =	3,00
Minimum gasket se	ating pressure		y =	69,00 MPa
Gasket contact widt	0.1		w =	20,00 mm
Basic gasket seating			b0 = w / 2 =	10,00 mm
Effective gasket sea	_		$b = 2.52 \cdot \sqrt{(b0)} =$	7,97 mm
Diameter of gasket	•	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 bo	alt		_	282,0 mm ²
TOOL area of one bu	л		_	202,0 11111

oads

201110440		
Number of bolts	=	68
Bolt type	=	ISO M22 x 2,50
Root area of one bolt	=	282,0 mm ²
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	$HG = 2\pi \cdot G \cdot b \cdot m \cdot P =$	345 804 N
Minimum required bolt load for operating condition	Wop = H + HG =	2 427 022 N
	$G^2\pi P_{\star} =$	3 097 684 N

Minimum required bolt load for the test condition (MAIN FLANGE LOWER SIDE) 3613956 N Minimum required bolt load for assembly condition 1988375 N Total required cross-sectional area of bolts 18 853,1 mm²

Total cross-sectional area of bolts at the section of least bolt diameter Maximum bolts area for gasket crush

 $A_{b} \max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}} = W = 0.5 (A_{B,min} + A_{B}) f_{B,A} =$ 2788800 N

Design bolt load for assembly condition

43

19 176,0 mm²

68 049,7 mm²

AB ≥ AB,min: Ok

PD CEN/TR 13445-102:2015

CEN/TR 13445-102:2015 (E)

Bolt pitch correction factor	$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6\varepsilon}{m+0.5}}}; 1 \right] =$	1,00000
Ratio of the flange diameters Length parameter	$K = A / B^* = I = \sqrt{B^* \cdot a^*} = I$	1,13884 99,59 mm
Hydrostatic end force applied via shell to flange	$K = A / B^{*} = I_{0} = \sqrt{B^{*} \cdot g_{0}^{*}} = I_{D} = \frac{\pi}{4} (B^{*2}P) = I_{D} = I_{D}$	1 907 581 N
Hydrostatic end force due to pressure on flange face Radial distance from bolt circle to circle on which HD acts Radial distance from gasket load reaction to bolt circle	HT = H - HD = hD = (C - B* - g1*) / 2 = hG = (C - G) / 2 =	26,47 mm
Radial distance from bolt circle to circle on which HT acts Flange stress factor	4	38,73 mm 1,86292
Flange stress factor	$\beta_T = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 8548 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 8588 \log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} = \frac{K^2 (1 + 8588 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log_{10} K)} = \frac{K^2 (1 + 868 \log_{10} K) - 1}{(10472 + 1948 \log$	
rialige stress lactor	$\beta_U = \frac{K^2 (1 + 855246 \log_{10} K) - 1}{136136 (K^2 - 1)(K - 1)} =$	10,44913
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 = βV =	0,87093 0,33259
Hub stress correction factor	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3 \beta_V}{\beta_U l_0 g_0^{*2}}\right) =$	1,35490 3,75921

Flange moments

Moment exerted on the flange per unit of length (assembly)

MA = W·hG =	73 816,6 N·m
$M_{\text{op}} = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G =$	98 858,6 N·m
Fs =	1,00
$M = F_s \cdot M_{op} \frac{C_F}{B^*} =$	89,7 N·m
$M = F_s \cdot M_A \frac{C_F}{B^*} =$	67,0 N·m

Flange stresses - operating condition Longitudinal stress in hub

zongwamar ou ood m mab	
Radial stress in flange	

Tangential stress in flange

Stress factor

$\sigma_H = \frac{\varphi M}{\lambda \alpha^{*2}} =$	143,70 MPa
$\sigma_0 = \frac{\left(1333e \cdot \beta_F + l_0\right)M}{\left(1333e \cdot \beta_F + l_0\right)M} =$	4,95 MPa
$\sigma_{\theta} = \frac{\lambda \cdot e^2 \cdot l_0}{e^2} = \sigma_{\theta} \frac{K^2 + 1}{e^2} = \sigma_{\theta} \frac{K^2 + 1}{K^2 - 1}$	88,28 MPa
k =	1,03400 ,5min(f;fH): Ok

k·or ≤ f: Ok $k \cdot \sigma \theta \le f$: Ok $0.5k(\sigma H + \sigma r) \le f$: Ok

 $0.5k(\sigma H + \sigma \theta) \le f$: Ok

Flange stresses - seating condition

Longitudinal stress in hub

Radial stress in flange

Tangential stress in flange

$$\sigma_{H} = \frac{\varphi M}{\lambda g_{1}^{*2}} = 107,30 \text{ MPa}$$

$$\sigma_{r} = \frac{\left(1333e^{*}\beta_{F} + l_{0}\right)M}{\lambda^{*}e^{2} \cdot l_{0}} = 3,70 \text{ MPa}$$

$$\sigma_{\theta} = \frac{\beta_{Y} \cdot M}{e^{2}} - \sigma_{r} \frac{K^{2} + 1}{K^{2} - 1} = 65,92 \text{ MPa}$$

 $k \cdot \sigma H \le 1,5 \min(f;fH)$: Ok k·σr ≤ f: Ok $k \cdot \sigma \theta \le f$: Ok $0.5k(\sigma H + \sigma r) \le f$: Ok $0.5k(\sigma H + \sigma \theta) \le f$: 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
Usadrootatio toot		
Hydrostatic test	Pt =	2,95 MPa
Item hydrostatic test pressure Overpressure due to static head	Ph =	0,02 MPa
Calculation pressure	P = Pt + Ph =	2,98 MPa
Calculation pressure	1 -1(1111-	2,50 Wii a
Flange constants		
Bolt pitch correction factor	$\left[\begin{array}{cc} \delta_b \end{array}\right] =$	1,00000
	$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{\delta_e}{m + 0.5}}}; 1 \right] =$	
Ratio of the flange diameters	K=A/B=	1,14091
Length parameter	$I = \sqrt{R_{1\alpha} *} =$	104,88 mm
	$K = A / B = l_0 = \sqrt{B \cdot g_0^*} = H_D = \frac{\pi}{4} (B^2 P) =$	
Hydrostatic end force applied via shell to flange	$H_D = \frac{1}{4}(B^2P) =$	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 Radial distance from bolt circle to circle on which HT acts	hG = (C - G) / 2 = hT = (2C - B - G) / 4 =	26,47 mm 39,23 mm
Flange stress factor	$V^2/1 + 855246\log V = 1 = 0$	1,86214
Trange stress ractor	$\beta_T = \frac{K^2 (1 + 855246\log_{10} K) - 1}{(10472 + 19448K^2)(K - 1)} =$	1,00214
	$T = (10472 + 19448K^2)(K-1)$	40.00055
Flange stress factor	$R = \frac{K^2(1+855246\log_{10}K)-1}{10}$	16,22655
	$\beta_U = \frac{K^2 (1+855246\log_{10} K) - 1}{136136(K^2 - 1)(K - 1)} = \frac{1}{K - 1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right] = \frac{1}{10000000000000000000000000000000000$	
Flange stress factor	$\left[\left(K^2 \log_{10} K \right) \right] =$	14,76619
$\beta_{_{Y}} = 0$	$\frac{1}{K-1} \left[\frac{0.66845 + 5.7169}{K^2 - 1} \right]$	
	βF = βV =	0,87542
	βV =	0,34994
Hub stress correction factor	$\phi = \langle aB + I aB \rangle - \langle aB \rangle$	1,29900 3,24563
	$\lambda = \left(\frac{e\beta_F + I_0}{\beta_T I_0} + \frac{e^3\beta_V}{\beta_U I_0 g_0^{*2}}\right) = 0$	3,24303
_		
Flange moments	$M = H \cdot l \cdot + H \cdot l \cdot + H \cdot l \cdot + =$	1111170 F N
Total moment acting upon flange for operating condition Moment factor used to design split rings	$M_{\text{op}} = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G =$	144 170,5 N·m 1,00
Moment exerted on the flange per unit of length (operating)	C _π =	131,1 N·m
moment exerted on the named per anit or longth (operating)	Fs = $M = F_s \cdot M_{\text{op}} \frac{C_F}{B} =$	101,111
Flange stresses - operating condition	nM.	004 04 MD-
Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{10^{-k^2}} =$	204,91 MPa
Dadial stress in flames	$(1333a_1B_1 + I_1)M_1 =$	0.47 MDa
Radial stress in flange	$\sigma_r = \frac{\left(\frac{15556}{1621} \rho_F + i_0\right)M}{\frac{1621}{1621}} =$	8,17 IVIPa
Tangential stress in flange	$\sigma_{H} = \frac{\varphi M}{\lambda g_{1}^{*2}} =$ $\sigma_{r} = \frac{\left(1333e \cdot \beta_{F} + l_{0}\right) M}{\lambda \cdot e^{2} \cdot l_{0}} =$ $\sigma_{\theta} = \frac{\beta_{Y} \cdot M}{e^{2}} - \sigma_{r} \frac{K^{2} + 1}{K^{2} - 1} =$	120.10 MPa
an garaan an aas ar nan ga	$\sigma_{\theta} = \frac{r}{e^2} - \sigma_r \frac{K + 1}{K^2 - 1}$,
Stress factor	K -	1,00000
	k·σH	≤ 1,5min(f;fH): Ok
		k·σr ≤ f: Ok
	0.5	k·σθ ≤ f: Ok
		$k(\sigma H + \sigma r) \le f$: Ok $k(\sigma H + \sigma \theta) \le f$: Ok
Simplified fatigue assessment according to EN13445-		K(011 + 00) ≥ 1. OK
-		
Load condition 1, load details		
Design pressure	P =	,
Pressure range	ΔP =	•
Minimum operating temperature during cycle	Tmin =	20 °C
Maximum operating temperature during cycle Design temperature	Tmax = T =	260 °C 260 °C
Design temperature	I —	200 0

Design temperature

20 °C 260 °C 260 °C

PD CEN/TR 13445-102:2015

CEN/TR 13445-102:2015 (E)

Nominal design stress at design temperature $R = 121,33 \mathrm{MPa}$ Vield strength at design temperature $R = 120,00 \mathrm{MPa}$ Vield strength at design temperature $R = 120,00 \mathrm{MPa}$ Vield strength at design temperature $R = 120,00 \mathrm{MPa}$ 182.00 MPa 182.00 MP	Number of required fatigue cycles	Nreq =	1 200
Vield strength at design temperature Rp0,2T = 182,00 MPa			
Load condition 1, Junction to shell (of thickness es) Maximum allowable pressure (flange) Pmax = 2.02 MPa Calculation thickness n = 1,50000 1,000 mm			
Maximum allowable pressure (flange)	field Strength at design temperature	κρυ,2/1 –	102,00 IVIPa
Maximum allowable pressure (flange) Calculation thickness Calculation thickness Stress factor Stress factor Pseudo-elastic stress range Qu'elastic qu'el	Load condition 1. Junction to shell (of thickness es)		
Stress factor $\eta = 1,50000$ Pseudo-elastic stress range $\Delta \sigma = (\Delta P/P max) + \eta = 157.42 MPa$ Equivalent number of full pressure cycles $\lambda \sigma = (\Delta P/P max) + \eta = 157.42 MPa$ Equivalent number of full pressure cycles $\lambda \sigma = (\Delta P/P max) + \eta = 157.42 MPa$ Thickness correction factor $\lambda \sigma = 1.00000$ C Temperature correction factor $\lambda \sigma = 1.00000$ C Temperature correction factor $\lambda \sigma = 1.00000$ C Temperature correction factor $\lambda \sigma = 1.000000$ C Temperature correction factor $\lambda \sigma = 1.000000$ C Temperature correction factor $\lambda \sigma = 1.0000000$ C C $\lambda \sigma = 1.00000000$ C C $\lambda \sigma = 1.0000000000$ C C $\lambda \sigma = 1.00000000000000000000000000000000000$		Pmax =	2,02 MPa
Pseudo-elastic stress range $\Delta \sigma = (\Delta P/P \text{max}) \cdot \eta^{-1} = 157.42 \text{ MPa}$ and the Equivalent number of full pressure cycles N = 2776,56702 T76,000 T76 T76 T76 T776,000 T76	Calculation thickness	en =	10,00 mm
Equivalent number of full pressure cycles Thickness correction factor Ce = 1.00000 Assumed mean cycle temperature T*=0,75-Tmax+0,25-Tmin = 200 °C Temperature correction factor Weld class Endurance limit Cut-off limit ADD = 46,433 MPa Ficitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index Load condition 1, Hub to plate junction Maximum allowable pressure (flange) Required lastic stress range Stress factor Pseudo-elastic stress range Pseudo-elastic stress corection factor Pseudo-elastic stress concentration Pseudo-elastic stress concentration Pseudo-elastic stress range Pseudo-elastic stres			•
Thickness correction factor $Ce = 1,00000$ Assumed mean cycle temperature $T^*=0,75 \cdot Tmax+0,25 \cdot Tmin = 200 ^{\circ}C$ Temperature correction factor $CT = 0,94000$ Wolf class $CT = 0,94000$ Add $CT = 0,94000$ Assumed mean cycle temperature $CT = 0,94000$ Assumed mean cycle temperature $CT = 0,94000$ Assumed an encycle temperature $CT = 0,94000$ Add $CT $			
Assumed mean cycle temperature $ \begin{array}{c} \text{T*=0,75-Tmax+0,25-Tmin} = & 200 ^{\circ}\text{C} \\ \text{Temperature correction factor} \\ \text{Veid class} \\ \text{C = } \\ \text{63} \\ \text{Endurance limit} \\ \text{AdD} \\ \text{Cut-off limit} \\ \text{ADD} \\ \text{Endurance limit} \\ \text{Bounder of allowable fatigue cycles} \\ \text{Partial fatigue damage index} \\ \text{D=Nreq/N} \\ \text{Endurance limit} \\ Endurance limi$			•
Temperature correction factor $CT = 0.94000$ $CC = 6.3$			
Weld class Endurance limit A DD = 46,43 MPa Endurance limit A DD = 46,43 MPa Cut-off limit A DD = 46,43 MPa Acocut = 25,52 MPa Fictitious stress range for insertion into the fatigue design curves N = 106,553 Partial fatigue damage index Load condition 1, Hub to plate junction Maximum allowable pressure (flange) Calculation thickness Partial fatigue damage index D = Nreq/N = 2,02 MPa Calculation thickness Partial fatigue damage index D = Nreq/N = 16,00 mm Stress factor Passure-clastic stress range Equivalent number of full pressure cycles Passure daman cycle temperature Passure correction factor T = 0,75-Tmax+0,25-Tmin = 200 °C T = 0,94000 Transition radius Theoretical stress concentration factor Temperature correction factor Trenderical stress concentration factor The damage index summary Effective stress concentration factor Cut-off limit Fatigue cycles and damage index summary Load 1, partial damage index summary Load 1, partial damage index summary Load 1, partial damage index for Hub to plate junction Cut-12 Ellipsoidal head - TOP HEAD Design data Internal design temperature T = 260 °C Internal design temperature T = 260 °C Internal design pressure P = 2,00 MPa			
Cut-off limit $Accut = 2.5.52 MPa$ Fictitious stress range for insertion into the fatigue design curves $Aco^* = [\Delta a/Ce CT] = 167,47 MPa$ $Accut = 2.5.52 MPa$ $Accut = 167,47 MPa$ $Accut = 167,4$		C =	,
Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles $N = 106.553$ $N = 106.00$		$\Delta \sigma D =$	•
Number of allowable fatigue cycles Partial fatigue damage index $D=Nreq/N=0.01126$ Dearmage index $D=Nreq/N=0.01126$ Load condition 1, Hub to plate junction Maximum allowable pressure (flange) Pmax = 2.02 MPa Calculation thickness Passure (flange) Pmax = 16.00 mm Stress factor Passure of this pressure cycles Passure of the pressure cycles Passure of the pressure cycles Passure or P			•
Partial fatigue damage index			
Load condition 1, Hub to plate junction Maximum allowable pressure (flange) Pmax = 16,00 mm 2,02 MPa Calculation thickness en = 16,00 mm 1,50000 n = 15,0000 1,50000 n = 176,66702 15,0000 n = 176,66702 157,42 MPa 157,42 MPa r = 16,00 mm 157,42 MPa r = 157,42 MPa 200°C r = 157,66702 r = 10,0000 n = 157,42 MPa r = 157,66702 r = 1,50000 r = 1,50000 r = 1,50000 n = 157,42 MPa r = 1,50000 n = 157,42 MPa r = 1,50000 r = 1,50000 n = 157,42 MPa n = 157,42 MPa n = 157,42 MPa n = 157,42 MPa n = 150,0000 mm r = 13,000 mm r = 13,000 mm n = 15,200 mm n = 13,000 mm n = 15,200 MPa n = 13,000 mm n = 15,200 mm n = 15,200 mm n = 15,200 mm n = 15,200 MPa n =			
Maximum allowable pressure (flange)	Partial fatigue damage index	D=Nreq/N =	0,01126
Maximum allowable pressure (flange)	Load condition 1. Hub to plate junction		
Calculation thickness $\begin{array}{c} \text{Calculation thickness} \\ \text{Stress factor} \\ \text{Pseudo-elastic stress range} \\ \text{Equivalent number of full pressure cycles} \\ \text{Thickness correction factor} \\ Thicknes$		Pmax =	2.02 MPa
Pseudo-elastic stress range Equivalent number of full pressure cycles Neq = 776,56702 Thickness correction factor $C = 1,00000$ Assumed mean cycle temperature $C = 1,00000$ Assumed mean cycle temperature $C = 1,00000$ Thickness correction factor $C = 1,00000$ Transition radius $C = 1,000000$ Transition radius $C = 1,0000000$ Transition radius $C = 1,0000000$ Transition radius $C = 1,0000000$ Transition radius $C = 1,00000000$ Transition radius $C = 1,00000000000000000000000000000000000$	· · · · · · · · · · · · · · · · · · ·		,
Equivalent number of full pressure cycles Thickness correction factor Ce = 1,00000 Assumed mean cycle temperature T*=0,75·Tmax+0,25·Tmin = 200 °C Temperature correction factor Transition radius Transition radius Theoretical stress concentration factor Endurance limit Effective stress concentration factor Cut-off limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles N = 266 197 Partial fatigue damage index Fatigue cycles and damage index summary Load 1, partial damage index for Junction to shell (of thickness es) Total damage index: Junction to shell (of thickness es) Cut-off ladmage index: Junction to shell (of thickness es) Cotal damage index: Junction to shell (of thickness es) Cotal damage index: Junction to shell (of thickness es) Fatigue cycles and damage index for Junction to shell (of thickness es) Cotal damage index: Junction to shell (of thickness es) Fatigue cycles and damage index for Junction to shell (of thickness es) Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycles and damage index for Hub to plate junction Fatigue cycle cycle in the fatigue design cycle in the fatigue design of the fatigue cycle in the fatigue design of the fatigue cycle in the	Stress factor	η =	•
Thickness correction factor $C = 1,00000$ Assumed mean cycle temperature $T^*=0,75 \cdot Tmax+0,25 \cdot Tmin = 200 ^{\circ}C$ Temperature correction factor $C = 0,94000$ Transition radius $C = 0,940$			
Assumed mean cycle temperature T*=0,75·Tmax+0,25·Tmin = 0.00 °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 $\Delta DD = 175,20$ MPa Effective stress concentration factor $Kt=1,4$ (r≥en/4) = $1,40000$ $175,20$ MPa $1,40000$ Endurance limit $\Delta DD = 175,20$ MPa $1,40000$ $1,400$		•	
Temperature correction factor Transition radius r = 0,94000 Transition radius r = 13,00 mm Theoretical stress concentration factor $Kt=1,4$ (r2en/4) = 1,40000 Endurance limit ADD = 175,20 MPa Effective stress concentration factor $K_f = 1 + \frac{1.5(K_f - 1)}{1 + 0.5 \max(1, K_f \frac{\Delta \sigma}{d \sigma_b})} = 1,36833$ Cut-off limit ACC Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles N = 229,16 MPa Number of allowable fatigue cycles N = 266 197 Partial fatigue damage index Summary DeNreq/N = 0,00451 Fatigue cycles and damage index summary Load 1, partial damage index for 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 T			
Transition radius $r = 13,00 \text{ mm}$ Theoretical stress concentration factor $r = 13,00 \text{ mm}$ Theoretical stress concentration factor $r = 10,0000 \text{ mm}$			
Theoretical stress concentration factor Endurance limit $ \Delta \sigma D = 175,20 \text{ MPa} $ Effective stress concentration factor $ K_f = 1 + \frac{1}{1 + 0.5 \cdot \max(1, K_f - 1)} = 1,36833 $ $ K_f = 1 + \frac{1}{1 + 0.5 \cdot \max(1, K_f - 1)} = 1,36833 $ Cut-off limit $ \Delta \sigma cut = 116,70 \text{ MPa} $ Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles $ N = 266 \cdot 197 $ Partial fatigue damage index $ D = N r eq/N $			
Endurance limit Effective stress concentration factor $K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1, K_t \frac{\Delta \sigma}{\Delta \sigma})} = 175,20 \text{ MPa}$ Cut-off limit $\Delta \sigma \text{ interpolation} = 1,36833$ $\Delta \sigma interpolation$		•	,
Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Number of allowable fatigue cycles Number of allowable fatigue cycles N = 266 197 Partial fatigue damage index 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) Total damage index: Junction to shell (of thickness es) Summary Load 1, partial damage index for Hub to plate junction Summary Load 1, partial damage index: Junction to shell (of thickness es) Summary Summar	Endurance limit	ΛσD =	
Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Number of allowable fatigue cycles Number of allowable fatigue cycles N = 266 197 Partial fatigue damage index 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) Total damage index: Junction to shell (of thickness es) Summary Load 1, partial damage index for Hub to plate junction Summary Load 1, partial damage index: Junction to shell (of thickness es) Summary Summar	Effective stress concentration factor	$K_c = 1 + \frac{1.5(K_t - 1)}{1.5(K_t - 1)} = 1.5(K_t - 1)$	1,36833
Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Number of allowable fatigue cycles Number of allowable fatigue cycles N = 266 197 Partial fatigue damage index 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) Total damage index: Junction to shell (of thickness es) Summary Load 1, partial damage index for Hub to plate junction Summary Load 1, partial damage index: Junction to shell (of thickness es) Summary Summar		$1 + 0.5 \cdot \max(1, K_t \frac{\Delta \sigma}{\Delta \sigma_0})$	
Number of allowable fatigue cycles Partial fatigue damage index Fatigue cycles and damage index summary Load 1, partial damage index for Junction to shell (of thickness es) Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction		Δσcut =	116,70 MPa
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) Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction End (1)<1: Ok TDI(1)<1: Ok TDI(2)<1: Ok TDI(2			
Load 1, partial damage index for Junction to shell (of thickness es) Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction Ellipsoidal head - TOP HEAD According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa	Partial fatigue damage index	D=Nreq/N =	0,00451
Load 1, partial damage index for Junction to shell (of thickness es) Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction Ellipsoidal head - TOP HEAD According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa	Estimus avales and demans index summens		
Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction Total damage index: Hub to plate junction Ellipsoidal head - TOP HEAD According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa		_	0.01126
Load 1, partial damage index for Hub to plate junction 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 Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa			
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		=	-,-
C.12 Ellipsoidal head - TOP HEAD According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa		=	
C.12 Ellipsoidal head - TOP HEAD According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Ti = 2,00 MPa			
According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Pi = 2,00 MPa			TDI(2)<1: Ok
According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8 Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Pi = 2,00 MPa	0.40 EW		
Design data Internal design temperature Internal design pressure Ti = 260 °C Internal design pressure Pi = 2,00 MPa	C.12 Ellipsoidal head - TOP HEAD		
Internal design temperature $Ti = 260 ^{\circ}\text{C}$ Internal design pressure $Pi = 2,00 \text{MPa}$	According to: EN 13445 Ed. 2009 Issue 5, Clause 7 and 8		
Internal design temperature $Ti = 260 ^{\circ}\text{C}$ Internal design pressure $Pi = 2,00 \text{MPa}$			
Internal design temperature $Ti = 260 ^{\circ}\text{C}$ Internal design pressure $Pi = 2,00 \text{MPa}$	Dosign data		
Internal design pressure Pi = 2,00 MPa		т; _	260 °C

165,33 MPa

212,50 MPa

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

Nominal design stress at internal design temperature

Nominal design stress at room temperature

Nominal design stress in test condition	$f_{\text{test}} = \left(\frac{R_{\text{p0.2/Ttest}}}{105}\right) =$	338,10 MPa
Geometry Inside diameter Outside diameter Head outside height Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Straight flange length Straight flange thickness Knuckle thickness	Di = De = H = en = c = ce = δ = I(sf) = en(sf) = en(k) =	1 100,00 mm 1 128,00 mm 339,00 mm 14,00 mm 1,00 mm 0 mm 1,40 mm 50,00 mm 14,00 mm
Internal pressure Overpressure due to static head Calculation pressure Y parameter Z parameter X ratio N parameter β(0,1) parameter β parameter β parameter 0,2 % proof strength at design temperature Design stress for buckling equation Joint efficiency	$\begin{array}{c} \text{Ph} = \\ \text{P=Pi+Ph} = \\ \text{Y=min(ec/R;0,04)} = \\ \text{Z=log10(1/Y)} = \\ \text{X=r/Di} = \\ N = 1006 - \frac{1}{[62 + (90Y)^4]} = \\ \beta_{01} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837) = \\ \beta_{02} = \max[0.95(0.56 - 194Y - 82.5Y^2), 0.5] = \\ \beta = 10[(0.2 - X)\beta_{01} + (X - 0.1)\beta_{02}] = \\ \text{Rp02t} = \\ \text{fb=Rp02t/1,5} = \\ \text{z} = \end{array}$	0 MPa 2,00 MPa 0,01080 1,96652 0,17143 0,86502 0,78982 0,50295 0,58490 248,00 MPa 165,33 MPa 1,00000
Head ratio Inside diameter Equivalent crown radius Equivalent knuckle radius Required thickness of end to limit membrane stress in Required thickness of knuckle to avoid axisymmetric y	$\frac{c_s}{21z} - 0.5P$	2,00000 1 104,80 mm 992,40 mm 189,40 mm 8,42 mm 9,60 mm
Minimum required thickness Straight flange thickness Straight flange minimum required thickness	e=max(ey;es) = e(sf) = e =	9,60 mm 14,00 mm 9,12 mm en(sf) ≥ e(sf): Ok
Knuckle check due to encroaching nozzle Largest inside diameter of nozzles encroaching knuckl V parameter	$V = \log_{10}(1000\frac{P}{f})$	193,70 mm 1,08267
A parameter B parameter Weakening factor due to the presence of nozzle	$A = 0.54 + 0.41V - 0.044V^3 = B = 7.77 - 4.53V + 0.744V^2 = B = max(A + B\frac{d_i}{D_e}; 1 + 0.5B\frac{d_i}{D_e}) = 0$ avoid $e_{y(k)} = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + c + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \frac{\beta P \beta_k (0.75R + 0.2D_i^*)}{f} + ce + \delta = \beta P \beta_k (0.75R + 0.2D_i^$	0,92806 3,73760 1,56988
Required thickness of knuckle encroached by nozzle to axisymmetric yielding Minimum knuckle thickness	o avoid $e_{y(k)} = \frac{\beta P \beta_k (0.75R + 0.2D_i^r)}{f} + c + ce + \delta = e(k) = \max[ey(k);es] = 0$	13,12 mm 13,12 mm en(k) ≥ e(k): Ok en ≥ e: Ok
Maximum allowable pressures (at the top of the Maximum allowable test pressure Maximum allowable design pressure	ne vessel) = = =	7,66 MPa 3,43 MPa
Deformation according to EN13445-4 Clause 9 Manufactured in one piece Spherical part Knuckle segments	$F(1)=100\cdot \ln[(1,21\cdot De)/(De-2\cdot en)] = F(2)=100\cdot \ln\{2\cdot R\cdot asin[(0,4\cdot De/R)/(0,8\cdot De-2\cdot en)]\} = F(3)=(100\cdot en)/(r+en/2) = F($	21,576 % 6,913 % 7,216 %

II donatatia taat		
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 = Pc=Pt+Pht =	0,005 MPa 2,96 MPa
Calculation pressure Y parameter	Y=min(ec/R;0,04) =	0,00829
Z parameter	Z = log 10(1/Y) =	2,08141
X ratio	X=r/Di =	0,17084
N parameter	$N = 1006 - \frac{1}{[62 + (90Y)^4]} =$	0,85239
$\beta(0,1)$ parameter $\beta_{0,1} = N(-0.1)$	$ 833Z^{3} + 10383Z^{2} - 12943Z + 0.837 = $ $ 833Z^{3} + 10383Z^{2} - 12943Z + 0.837 = $ $ 833Z^{3} + 10383Z^{2} - 12943Z + 0.837 = $ $ 833Z^{3} + 10383Z^{2} - 12943Z + 0.837 = $ $ 62 + (901) $	0,84247
$\beta(0,2)$ parameter $\beta_{0,2} = 1$	$\max[0.95(0.56 - 194Y - 82.5Y^2), 0.5] =$	0,51133
β parameter	$\beta = 10[(0.2 - X)\beta_{01} + (X - 0.1)\beta_{02}] =$	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 Head ratio	z = K=Di/2h =	1,00000 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	$e_{s} = \frac{PR'}{2fz - 0.5P} + \delta = e_{y} = \frac{\beta P(0.75R + 0.2D'_{i})}{f} + \delta = e_{y} = \frac{\beta P(0.75R + 0.2D'_{i})}{f} + \delta = e_{y} = \frac{(0.75R + 0.5D'_{i})(\frac{P}{111f_{b}}(\frac{D'_{i}}{r'_{i}}))}{f} + \delta$	6,80 mm
Required thickness of knuckle to avoid plastic buckling	$D_{1}^{\prime} = 0.825 \left(\frac{1}{13}\right) = 0.825 \left(\frac{1}{13}\right)$	6,08 mm
$e_b = ($	$(0.75R + 0.5D_i)(\frac{P}{111f}(\frac{D_i}{r'})) + \delta$	
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	$V = \log_{10}(1000 \frac{P}{f}) = 0$	0,94204
A parameter		0,88945
B parameter	$B = 7.77 - 4.53V + 0.744V^2 =$	4,16281
Weakening factor due to the presence of nozzle	$A = 0.54 + 0.41V - 0.044V^{2} = B = 7.77 - 4.53V + 0.744V^{2} = \beta_{k} = \max(A + B\frac{d_{i}}{D_{e}}; 1 + 0.5B\frac{d_{i}}{D_{e}}) = e_{y(k)} = \frac{\beta P \beta_{k} (0.75R + 0.2D_{i}^{t})}{f} + \delta$	1,60429
Required thickness of knuckle encroached by nozzle to avoid	$\beta P \beta_k (0.75 R + 0.2 D_i^t)$ =	9,63 mm
axisymmetric yielding	$e_{y(k)} = \frac{f}{f}$	
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	enze. ok
Load condition 1, load details	_	0.00145
Design pressure Pressure range	P = ΔP =	2,00 MPa 1,75 MPa
Minimum operating temperature during cycle	Tmin =	1,73 Wil a
Maximum operating temperature during cycle	Tmax =	260 °C
Design temperature	T =	260 °C
Number of required fatigue cycles	Nreq =	1200
Nominal design stress at design temperature Ultimate tensile strength at room temperature	f = Rm =	165,33 MPa 510,00 MPa
Yield strength at design temperature	Rp0,2/T =	248,00 MPa
Load condition 1, all butt welds	_	
Maximum allowable pressure (component) Nominal thickness	Pmax =	3,43 MPa
Joint efficiency	en = z =	14,00 mm 1,00000
Butt weld offset	δ =	0,70 mm
Partial stress factor	$\eta 1 = (3 \cdot \delta)/en =$	0,15000
Butt weld angular misalignment	Θ = Dm=(Di+Do)/2 =	2,50 °
Partial stress factor	Dm=(Di+Do)/2 =	1 114,00 mm

Partial stress factor	η3=(Θ/50)·√[Dm/(2·en)] =	0,31538
Stress factor	$\eta = (1 + \eta 1 + \eta 3) \cdot z =$	1,46538
Pseudo-elastic stress range	$\Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f =$	123,50 MPa
Equivalent number of full pressure cycles	Neg =	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] =	131,39 MPa
Number of allowable fatigue cycles	N =	220 675
Partial fatigue damage index	D=Nreq/N =	0,00544
Tartial latigue damage much	B-Mcq/M-	0,00044
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	Δσ=(ΔP/Pmax)·η·f =	210,70 MPa
Equivalent number of full pressure cycles	Neg =	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	Δσcut =	25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	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	I - 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 165,33 MPa fs =

(N) (EN 10222-2:2001) $t \le 35,00 \text{ mm}$ - Forging Nozzle material: P280GH fb = Nominal design stress of the nozzle material 134,00 MPa

134,00 MPa fob = min(fs, fb) =

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 c = 1,00 mm Corrosion allowance External corrosion allowance ce = 0 mm

Undertolerance Maximum width of the opening on shell without nozzle Internal diameter External diameter of the nozzle Analysis thickness of the nozzle Length of nozzle extending outside the shell Effective thickness of the nozzle Effective length of nozzle outside the shell for reinforcement Stress loaded cross-sectional area effective as reinforcement - welds		0 mm 142,00 mm 104,30 mm 104,30 mm 19,85 mm 177,50 mm 18,85 mm 48,18 mm 48,18 mm 48,18 mm 0 mm² eb ≤ e_ab: Ok a,b / ea,s ≤ 3: Ok
Pad geometry Effective width of reinforcing plate for reinforcement Stress loaded cross-sectional area effective as reinforcement - pad	$Ip' = \min(l_so, l_p) = Af_p = e_p \cdot l_p' =$	0 mm 0 mm²
Shell geometry Analysis thickness of shell wall Analysis thickness of shell wall Shell internal diameter Inside height of the dished end Inside radius of curvature of the shell at the opening centre	$e_{a\beta} = t_{\text{shell}} - c_{\text{shell}} - \delta_{\text{shell}} = e_{\text{CS}} = 0$ Di = Id_head + 2(cs + cs') = h = H_head - e_head + cs + = cs' $r_{\text{is}} = \frac{0.44D_i^2}{2h} + 0.02D_i = 0$	11,60 mm 11,60 mm 1104,80 mm 277,40 mm
Check of distance from shell discontinuity Maximum length of shell contributing to opening reinforcement Head's analysis thickness Head's knuckle radius Distance between nozzle edge and knuckle-shell tangent line as per in 7 Limit distance between nozzle edge and knuckle-shell tangent line as per in 7.7.2 Distance between an opening and a shell discontinuity Length of shell from the edge of the opening to a shell discontinuity Minimum value for w which has no influence on Is from shell discontinuitin Required minimum value for w	$w_{min}(7.7.2)=2,5 \cdot \sqrt{(ek \cdot rk)} =$ $w = $ $l_s = w =$	152,00 mm 11,60 mm 189,40 mm 814,74 mm 117,18 mm 493,00 mm 493,00 mm 152,00 mm 112,80 mm
Internal pressure Overpressure due to static head Calculation pressure	Ph = P = Pi + Ph =	0 MPa 2,00 MPa
Transverse section	$\delta = \frac{d_{eb}}{2r_{ms}} = $ $a = \frac{d_{eb}}{2r_{ms}} = $	4,08 ° 71,06 mm
Mean radius of curvature Maximum length of shell contributing to opening reinforcement Effective length of shell for opening reinforcement Pressure loaded area - shell	$r_{\text{ms}} = r_{\text{is}} + 0.5e_{a,s} = l_{\text{so}} = \sqrt{(2r_{\text{is}} + e_{c,s})e_{c,s}} = l_{s} = \sqrt{(2r_{\text{is}} + e_{c,s})e_{c,s}} = l_{s} = -1$ $Ap_{s} = 0.5r^{2} = \frac{l_{s} + a}{1 + a} = -1$	995,91 mm 152,00 mm 152,00 mm 109 786,4 mm ²
Length of penetration into shell wall Stress loaded cross-sectional area effective as reinforcement - nozzle Stress loaded cross-sectional area - shell Pressure loaded area - nozzle Additional area due to obliquity of the nozzle Stress loaded cross-sectional area effective as reinforcement - pad Reactive force $F_r = (\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f_s - 0.5P \right) + \operatorname{Af}_p(\operatorname{Af}_s + \operatorname{Af}_w) \left(f$	e s = e_a,s = Af _b = $e_b \cdot (l_b' + l_{bi}' + e_{s'})$ = Af _s = $(l_s' + e_{ab} - c_{shell})e_{c,s}$ = Ap _b = $0.5d_{ib} \cdot (l_b' + e_{a,s})$ = Ap ϕ = Af ϕ =	11,60 mm 1 126,9 mm ² 1 763,2 mm ² 3 117,6 mm ² 0 mm ²

 $P_{\max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_{\phi}) + 0.5(Af_s + Af_w + Af_b + Af_p)} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob}}{(Ap_s + Ap_b + 0.5Ap_{\phi}) + 0.5(Af_s + Af_w + Af_b + Af_p)} = \frac{(Af_s + Af_w) \cdot f_s}{(Af_s + Af_w + Af_b + Af_p)} = \frac{(Af_s + Af_w) \cdot f_s}{(Af_s + Af_w + Af_b + Af_p)} = \frac{(Af_s + Af_w) \cdot f_s}{(Af_s + Af_w) \cdot f_s} = \frac{(Af_s$ Pressure load 225 808 N 3,87 MPa Maximum allowable pressure 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 303,98 mm 161,98 mm Ligament length $L_{\min} = \max \left[3e_{\alpha \hat{\wp}}, 0.2\sqrt{(2r_{is} + e_{\mathcal{C}\beta})e_{\mathcal{C}\beta}} \right] = d1 =$ Mean shell radius at the centres of adjacent nozzles 990,11 mm Minimum required ligament length 34,80 mm 142,00 mm 142,00 mm L≥Lmin: Ok $d1+d2 \le 0,2\sqrt{(2r_is+e_c,s)\cdot e_c,s]}$: Not satisfied, ligament check required Ligament check Inside radius of curvature of the shell at the opening centre $r_{\rm is} = \frac{D_{\it e}}{2} - e_{\it a\it p} = \\ {\rm Angle~between~the~centre-to-centre~line~of~openings~and~the~generatrix~of~the~shell} \\ {\rm Ap~of~the~shell~for~the~length~Lb} \\ {\rm Ap}_{\rm Ls} = \frac{0.5r^2_{\rm is} \cdot L_b(1+\cos(\phi))}{r_{\rm is} + 0.5e_{\it a\it p} \cdot \sin(\phi)} = \\ {\rm Ap~of~the~shell~openings~and~the~generatrix~of~the~shell~openings~and~the~generatri$ 990.11 mm 90.00° 149 609,7 mm² 142,00 mm 142,00 mm $\phi_{-}e1 = \phi_{-}e2 = 0$ $r_{os1} = \frac{r_{is1}}{\sin^{2}(\varphi)} + 0.5e_{a\beta} = 0$ $r_{os2} = \frac{r_{is2}}{\sin^{2}(\varphi)} + 0.5e_{a\beta} = 0$ $\delta_{1} = \frac{d_{cb1}}{2r_{os1}} = 0$ $\delta_{2} = \frac{d_{cb2}}{2r_{os2}} = 0$ $a_{1} = r_{os1} \left[\arcsin\left(\delta_{1} + \sin(\varphi_{c1})\right) - \varphi_{c1} \right] = 0$ $a_{2} = r_{os2} \left[\arcsin\left(\delta_{2} + \sin(\varphi_{c2})\right) - \varphi_{c2} \right] = 0$ $Af_{Ls} = (L_{b} - a_{1} - a_{2}) \cdot e_{c\beta} = 0$ $Ap\phi = 0$ 0° 0 ° 995,91 mm 995,91 mm 4,08° 4,08° 71,06 mm 71,06 mm Af of the shell contained along the length Lb 1877.5 mm² Āρφ1 = 0 mm² Αρφ2 = 0 mm² Apb1 = Apb2 = Afb1 = 3 117,6 mm² 3 117,6 mm² 1 126,9 mm² Afb2 =1 126,9 mm² Afp1 = Afp2 = 0 mm² 0 mm² fob1 = 134.00 MPa fob2 =134,00 MPa fop1 =0 MPa $F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{bl}(f_{obl} - 0.5P) + Af_{pl}(f_{opl} - 0.5P) + = + Af_{col}(f_{obl} - 0.5P) + Af_{col}($ 0 MPa Reactive force 608 288 N ${}^{+} {\rm Af}_{\rm b2} \big(f_{\rm ob2} - 0.5P \, \big) + {\rm Af}_{\rm p2} \big(f_{\rm op2} - 0.5P \, \big) \\ F_{\rm req} = P \, \big({\rm Ap}_{\rm Ls} + {\rm Ap}_{\rm b1} + 0.5 {\rm Ap}_{\varphi 1} + {\rm Ap}_{\rm b2} + 0.5 {\rm Ap}_{\varphi 2} \, \big) = 0.5 \, {\rm Ap}_{\varphi 2} \, + 0.5 \,$ Pressure load 311690 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 406.09 mm Lb = 225.54 mm Ligament length $L_{\min} = \max \left[3e_{a\varsigma}, 0.2\sqrt{(2r_{is} + e_{c\varsigma})e_{c\varsigma}} \right] =$ Mean shell radius at the centres of adjacent nozzles 990,11 mm 34,80 mm Minimum required ligament length 142.00 mm 219,10 mm L≥Lmin: Ok $d1+d2 \le 0,2\sqrt{(2r_is+e_c,s)\cdot e_c,s]}$: Not satisfied, ligament check required

Ligament check		
Inside radius of curvature of the shell at the opening centre	$r_{\rm is} = \frac{D_e}{2} - e_{a\beta} =$	990,11 mm
Angle between the centre-to-centre line of openings and the graph of the shell for the length Lb	eneratrix of the shell $\Phi = Ap_{Ls} = \frac{0.5r_{1s}^2 L_b (1 + \cos(\phi))}{r_{1s} + 0.5e_{as} \sin(\phi)} = 0$	90,00 ° 199864,9 mm²
	$r_{is} + 0.5e_{\alpha\beta} \cdot \sin(\phi)$ deb1 =	142,00 mm
	deb2 =	219,10 mm
	φ_e1 = φ e2 =	0 °
	$r_{\text{os1}} = \frac{r_{\text{is1}}}{\sin^2(\varphi)} + 0.5e_{\alpha,\beta} = $	995,91 mm
	$r_{os2} = \frac{r_{is2}}{\sin^2(\varphi)} + 0.5e_{as} =$	995,91 mm
	$\delta_1 = \frac{d_{\text{cbl}}}{2r}$	4,08 °
	$\delta_2 = \frac{\frac{d_{\text{ob2}}}{d_{\text{eb2}}}}{2r_{\text{os2}}} =$	6,30 °
	$a_1 = r_{os1} \left[\arcsin \left(\delta_1 + \sin \left(\varphi_{e1} \right) \right) - \varphi_{e1} \right] = a_2 = r_{os2} \left[\arcsin \left(\delta_2 + \sin \left(\varphi_{e2} \right) \right) - \varphi_{e2} \right] =$	71,06 mm
Af of the color of	$a_2 = r_{os2} \left[\arcsin \left(\delta_2 + \sin \left(\varphi_{e2} \right) \right) - \varphi_{e2} \right] =$	109,77 mm
Af of the shell contained along the length Lb	$Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{c,s} = Ap\phi 1 =$	2 612,9 mm² 0 mm²
	Apφ2 =	0 mm²
	Apb1 =	3 117,6 mm ²
	Apb2 = Afb1 =	5 724,8 mm ² 1 126,9 mm ²
	Afb2 =	582,2 mm ²
	Afp1 =	0 mm²
	Afp2 =	0 mm ²
	fob1 = fob2 =	134,00 MPa 111,73 MPa
	fop1 =	0 MPa
Reactive force $F = (Af_{Ls} + Af_w)(f_s - 0.5P)$	$+ Af_{b1}(f_{ob1} - 0.5P) + Af_{p1}(f_{op1} - 0.5P) + =$	0 MPa = 643 734 N
$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{ob2})$	$_{\text{op2}} = 0.5P$)	
	$(Ap_{Ls} + Ap_{h1} + 0.5Ap_{\omega 1} + Ap_{h2} + 0.5Ap_{\omega 2}) =$	417 415 N
		F≥Freq: Ok
Hydrostatic test		
Item hydrostatic test pressure Overpressure due to static head	Pt = Ph =	2,95 MPa 0,002 MPa
Calculation pressure	P = Pt + Ph =	2,96 MPa
Transverse section		
11411070130 30011011	$\delta = \frac{d_{eb}}{2r_{ms}} =$ $a = \frac{d_{eb}}{2r_{ms}} =$ $t = \frac{I_{eb}}{I_{eb}} =$ $r_{ms} = r_{is} + 0.5e_{a,s} =$ $I_{so} = \sqrt{(2r_{is} + e_{c,s})e_{c,s}} =$ $I_{s'} = \min[I_{s}, I_{so}] =$ $Ap_{s} = 0.5r^{2}i_{s}\sqrt{.5e_{a,s} + r_{is}} =$	4,08 °
	$a = \frac{d_{\text{eb}}^{\text{ms}}}{2r_{\text{ms}}} =$	71,06 mm
Mean radius of curvature Maximum length of shell contributing to opening reinforcement	$r_{\text{ms}} = r_{\text{is}} + 0.5e_{a\beta} = 1$ t $l_{co} = \sqrt{(2r_{i.} + e_{ab})e_{ab}} = 1$	996,36 mm 158,46 mm
Effective length of shell for opening reinforcement	$l_{s} = \min[l_{s}, l_{so}] =$	158,46 mm
Pressure loaded area - shell	$Ap_s = 0.5r^2 \frac{I_s^7 + a}{0.5e_{as} + r_{is}} =$	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 - Stress loaded cross-sectional area - shell	- nozzle $Af_b = e_b \cdot (l_b + l_{bi} + e_s') = Af = (l' + e_b - e_b - e_b) e_b - e_b$	1 227,5 mm ² 1 996,5 mm ²
Pressure loaded area - nozzle	- nozzle $ \begin{array}{c} \mathbf{e's} = \mathbf{e_a,s} = \\ \mathbf{Af_b} = e_b \cdot (l_b + l_{bi} + e_s') = \\ \mathbf{Af_s} = (l_s' + e_{ab} - c_{shell}) e_{c,s} = \\ \mathbf{Ap_b} = 0.5 d_{ib} \cdot (l_b + e_{a,s}) = \\ \end{array} $	3 163,2 mm ²
Additional area due to obliquity of the nozzle	Αρφ =	0 mm²
Stress loaded cross-sectional area effective as reinforcement	- pad Afp = SP) + $Af_p(f_{op} - 0.5P)$ + $Af_b(f_{ob} - 0.5P)$ =	0 mm²
Reactive force $F_r = (Af_s + Af_w)(f_s - 0.5)$	$F = P(\Lambda_{op} + \Lambda_{b}) + AI_{b}(J_{ob} - USP) = P(\Lambda_{ob} + \Lambda_{b}) + O(\Lambda_{ob}) = P(\Lambda_{ob} + \Lambda_{b}) + O(\Lambda_{ob}) = P(\Lambda_{ob} + \Lambda_{b}) + O(\Lambda_{ob}) = P(\Lambda_{ob} + \Lambda_{ob}) = P(\Lambda_{ob} + \Lambda_{ob}) + O(\Lambda_{ob}) = P(\Lambda_{ob} + \Lambda_{ob}) = P($	985 913 N
Pressure load Maximum allowable pressure (A	$Fp - F(Aps + Apo + 0.5Ap\phi) = Af_s + Af_w) \cdot f_s + Af_b \cdot f_s + Af_w \cdot f_{sm} =$	343 011 N 8,42 MPa
$P_{\max} = \frac{1}{(Ap_s + Ap)}$		-, <u>-</u> u

Fr≥Fp: Ok

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

Design data		
Design data Internal design temperature	Ti =	260 °C
Internal design pressure	Pi =	
Joint efficiency	z =	1,00
Shell material: P355GH (EN 10028-2:2009) t ≤ 10	6,00 mm - Plate	
Nominal design stress of shell material	fs =	165,33 MPa
Nozzle material: P280GH (N) (EN 10222-2	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 geometry Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a crit		Yes
Offset from shell border	= =	0 mm 60,00°
Angular offset 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 Internal diameter	$d = d_{ib} = Id + 2(c + \delta) =$	142,00 mm 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	I_b =	215,80 mm
Effective thickness of the nozzle	e_b = e_ab - c - ce - =	18,85 mm
	$I_{\text{bo(max)}} = \sqrt{(d_{\text{cb}} - e_b)e_b} = I_{\text{bo}} = \min(I_b, I_{\text{bo(max)}}) = I_b = I_b$	48,18 mm
	$h_{bo}(\max) = V(a_{cb} e_b)e_b$ $h_{bo} = \min(h_{bo}(h_{bo}(\max))) =$	48,18 mm
Effective length of nozzle outside the shell for reinforcer	ment $l_b' = \min(l_{be}, l_b) =$	48,18 mm
Stress loaded cross-sectional area effective as reinforce		0 mm²
		eb ≤ e_ab: Ok
		ea,b / ea,s ≤ 3: Ok
Pad geometry		
Effective width of reinforcing plate for reinforcement	$ p' = \min(l_so, l_p) = \\ \text{ement - pad} \qquad \qquad \text{Af}_p = e_p \cdot l_p' = \\ $	0 mm
Stress loaded cross-sectional area effective as reinforce	ement - pad $Af_p = e_p \cdot l_p' =$	0 mm²
Shall gramating		
Shell geometry Analysis thickness of shell wall	$e_{as} = t_{\text{shell}} - c_{\text{shell}} - \delta_{\text{shell}} =$	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 + =	277,40 mm
Inside radius of curvature of the shell at the opening cer	ntre $0.44D^2$ =	990,11 mm
g	ntre $r_{is} = \frac{0.44D_i^2}{2h} + 0.02D_i =$,
Observation of distance from the University 19		
Check of distance from shell discontinuity	cement	152 00 mm
Maximum length of shell contributing to opening reinford	cement $l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = ek =$	152,00 mm
Head's analysis thickness Head's knuckle radius	ek = rk =	11,60 mm 189,40 mm
Distance between nozzle edge and knuckle-shell tange	ent line as per in 7.7.2 $w(7,7,2) =$	507,07 mm
Limit distance between nozzle edge and knuckle-shell t		117,18 mm
per in 7.7.2		

Distance between an opening and a shell discontinuity Length of shell from the edge of the opening to a shell disconti Minimum value for w which has no influence on Is from shell d Required minimum value for w		213,00 mm 213,00 mm 152,00 mm 112,80 mm
Internal pressure Overpressure due to static head Calculation pressure	Ph = P = Pi + Ph =	
Transverse section	.1	4.00 %
	$\delta = \frac{d_{\text{eb}}}{2r_{\text{ms}}} = $ $a = \frac{d_{\text{eb}}}{2r_{\text{ms}}} = $	4,08 °
	$a = \frac{d_{cb}}{2r_{mc}} =$	71,06 mm
Mean radius of curvature Maximum length of shell contributing to opening reinforcement	$r_{\text{ms}} = r_{\text{is}} + 0.5e_{a,s} = 1$ $l_{\text{so}} = \sqrt{(2r_{\text{is}} + e_{c,s})e_{c,s}} = 1$	995,91 mm 152,00 mm
Effective length of shell for opening reinforcement Pressure loaded area - shell	$r_{ms} = r_{is} + 0.5e_{a,\beta} = 1$ $I_{so} = \sqrt{(2r_{is} + e_{c,\beta})e_{c,\beta}} = 1$ $I_{s}' = \min[I_{s}, I_{so}] = 1$ $Ap_{s} = 0.5r^{2} \frac{I_{s}' + a}{0.5e_{a,\beta} + r_{is}} = 1$	152,00 mm 109 786,4 mm²
Length of penetration into shell wall Stress loaded cross-sectional area effective as reinforcement Stress loaded cross-sectional area - shell Pressure loaded area - nozzle	E S - E_a,S -	11,60 mm 1 126,9 mm ² 1 763,2 mm ² 3 117,6 mm ²
Additional area due to obliquity of the nozzle	Αρφ =	0 mm²
Stress loaded cross-sectional area effective as reinforcement Reactive force $F_r = (Af_s + Af_w)(f_s - 0.5)$	- pad Afp = $(P_{op} - 0.5P) + Af_b (f_{ob} - 0.5P) = 0.5P$	0 mm² 439 633 N
Pressure load Maximum allowable pressure	Fp = P(Aps + Apb + 0,5Ap ϕ) = Af _s + Af _w)· f_s + Af _b · f_{ob} + Af _p · f_{op} = f_s + 0.5Ap ϕ) + 0.5(Af _s + Af _w + Af _b + Af _p)	225 808 N 3,87 MPa
$^{\prime\prime}$ max $^{-}$ (Ap _s + Ap	$(p_b + 0.5Ap_g) + 0.5(Af_s + Af_w + Af_b + Af_p)$	Fr≥Fp: Ok
Adjacent openings		
LWN Flange N2 - DN100 PN25 - LWN Flange N1 - DN Centre-to-centre distance taken on the mean surface of the sh		303,98 mm
Ligament length Mean shell radius at the centres of adjacent nozzles	L =	161,98 mm 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 =	142,00 mm
	d2 =	142,00 mm L ≥ Lmin: Ok
d1+d2 ≤ 0,2 v	[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligame	
Ligament check Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as} =$	990,11 mm
Angle between the centre-to-centre line of openings and the g		90,00°
Angle between the centre-to-centre line of openings and the graph of the shell for the length Lb	$Ap_{Ls} = \frac{0.5r_{is}^2 L_b (1 + \cos(\phi))}{r_{is} + 0.5e_{a\beta} \sin(\phi)} =$	149 609,7 mm ²
	deb1 = deb2 =	142,00 mm 142,00 mm
	φ e1 =	0 °
	φ_e2 = r _{is1}	0 ° 995,91 mm
	$r_{os1} = \frac{r_{is1}}{\sin^2(\varphi)} + 0.5e_{a,\varsigma} = r_{os2} = \frac{r_{is2}}{\sin^2(\varphi)} + 0.5e_{a,\varsigma} = r_{os2} = r_{os2$	995,91 mm
		4,08 °
	$\delta_1 = \frac{d_{\text{eb1}}}{2r_{\text{os1}}} = $ $\delta_2 = \frac{d_{\text{eb2}}}{2r_{\text{os2}}} = $	4,08 °
	$a_1 = r_{os1} \left[\arcsin \left(\delta_1 + \sin \left(\varphi_{-1} \right) \right) - \varphi_{-1} \right] =$	71,06 mm
Af af the shall contain a tale of the trace (1.1)	$a_1 = r_{os1} \left[\arcsin \left(\delta_1 + \sin \left(\varphi_{e1} \right) \right) - \varphi_{e1} \right] = a_2 = r_{os2} \left[\arcsin \left(\delta_2 + \sin \left(\varphi_{e2} \right) \right) - \varphi_{e2} \right] = a_1 = a_2 $	71,06 mm
Af of the shell contained along the length Lb	$Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{c,s} = Ap\phi 1 =$	1 877,5 mm² 0 mm²

		Αρφ2 =	0 mm ²
		Apb1 = Apb2 =	3 117,6 mm ² 3 117,6 mm ²
		Apb2 = Afb1 =	1 126,9 mm ²
		Afb2 =	1 126,9 mm ²
		Afp1 =	0 mm²
		Afp2 =	0 mm²
		fob1 =	134,00 MPa
		fob2 =	134,00 MPa
		fop1 =	0 MPa
Reactive force	$F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{bl}(j_s)$	fop2 = $f_{\text{old}} - 0.5P + Af_{\text{pl}}(f_{\text{old}} - 0.5P) + =$	0 MPa 608 288 N
	$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op2} - 0.5P) + F_{req} = P(Ap_{Ls} + P_{req}) + Af_{p2}(f_{ob2} - 0.5P)$)	
Pressure load	$P_{\text{req}} = P \left(Ap_{Ls} + \right)$	$-Ap_{b1} + 0.5Ap_{\varphi 1} + Ap_{b2} + 0.5Ap_{\varphi 2}) =$	311 690 N
			F≥Freq: Ok
Hydrostatic test		5 ,	0.05.140
Item hydrostatic test pressure		Pt =	2,95 MPa
Overpressure due to static he	ead	Ph = P = Pt + Ph =	0,003 MPa
Calculation pressure		P = P(+ PII =	2,96 MPa
Transverse section			
Transverse section		$d_{ch} =$	4,08 °
		$\delta = \frac{\sigma}{2r_{\rm ms}}$,
		$\delta = \frac{d_{eb}}{2r_{ms}} =$ $a = \frac{d_{eb}}{2r_{ms}} =$ $r_{ms} = r_{is} + 0.5e_{a,\varsigma} =$ $l_{so} = \sqrt{(2r_{is} + e_{c,\varsigma})e_{c,\varsigma}} =$	71,06 mm
Mean radius of curvature		$r_{\rm ms} = r_{\rm is} + 0.5e_{\alpha s} =$	996,36 mm
Maximum length of shell con	tributing to opening reinforcement	$I_{-2} = \sqrt{(2r_{-} + e_{-2})e_{-2}} =$	158,46 mm
Effective length of shell for or	- · · · · ·	$I_0' = \min[I_0, I_{\infty}] =$	158,46 mm
Pressure loaded area - shell	yermig reimereement	$l_s + a =$	112 899,3 mm ²
		$I_{s}' = \min [I_{s}, I_{so}] = Ap_{s} = 0.5r^{2} \frac{I_{s}' + a}{i_{s}0.5e_{a.s} + r_{is}} = 0.5r^{2} \frac{I_{s}' + a}{i_{s}0.5e_{a.s} + r_{is}}$, .
Length of penetration into she	ell wall	$e's = e_a, s = Af_b = e_b \cdot (l_b' + l_{bi}' + e_s') =$	12,60 mm
Stress loaded cross-sectiona	Il area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l_b' + l_{bi}' + e_{s'}) =$	1 227,5 mm ²
Stress loaded cross-sectiona		$Af_s = (I_s + e_{ab} - c_{shell})e_{c,s} =$	1 996,5 mm²
Pressure loaded area - nozzl		$Ap_b = 0.5d_{ib} \cdot (I_b + e_{a,s}) =$	3 163,2 mm ²
Additional area due to obliqui	ty of the nozzle	Apφ =	0 mm²
Stress loaded cross-sectiona	If area effective as reinforcement - pad $F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_w$	Afp =	0 mm²
	$F_r = (AI_s + AI_w)(f_s = 0.5F) + AI$	$I_p(f_{op} - 0.5P) + AI_b(f_{ob} - 0.5P) =$	985 912 N
Pressure load	$(\Delta f + \Delta f)$	$Fp = P(Aps + Apb + 0.5Ap\phi) =$	343 104 N
Maximum allowable pressure	$P_{\text{max}} = \frac{(AI_S + AI_1)}{(AD_1 + AD_1 + AD_2)}$	$\frac{f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{f_{ob} + Af_s \cdot Af_b + Af_b} = \frac{f_{ob} + Af_b \cdot Af_b}{f_{ob} + Af_b \cdot Af_b}$	8,42 MPa
	$(Ap_s + Ap_b + 0.5A)$	$(P_{\varphi})^{+0.5}(AI_s+AI_w+AI_b+AI_p)$	F=> F=+ O!-
			Fr≥Fp: Ok

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

Design date		
Design data Internal design temperature	Ti =	260 °C
Internal design temperature Internal design pressure	11 – Pi =	
Joint efficiency	z =	
,		
Shell material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm - PI	ate	
Nominal design stress of shell material	fs =	165,33 MPa
Nozzle material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm - 5		
Nominal design stress of the nozzle material	fb =	,
	fob = min(fs, fb) =	111,73 MPa
Nozzlo goomotry		
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 =	,
External corrosion allowance	ce =	
Undertolerance	δ =	1,59 mm
Maximum width of the opening on shell without nozzle	d =	219,10 mm
Internal diameter	$d_{ib} = Id + 2(c + \delta) =$	198,88 mm
External diameter of the nozzle Nominal thickness of the nozzle	d_eb = Od - 2ce = e ab =	219,10 mm 12,70 mm
Length of nozzle extending outside the shell	e_ab = b =	
Effective thickness of the nozzle	e_b = e_ab - c - ce - =	
Ellocate anothers of the horris	δ	
	$I_{\text{bo(max)}} = \sqrt{(d_{\text{eb}} - e_b)e_b} = I_{\text{bo}} = \min(I_b I_{\text{bo(max)}}) = I_b I_{\text{bo}}$	45,97 mm
	$l_{\text{bo}} = \min(l_{\text{bo}} l_{\text{bo}} (max)) =$	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 =	
		eb ≤ e_ab: Ok
		ea,b / ea,s \leq 3: Ok
Pad geometry		
Effective width of reinforcing plate for reinforcement	$lp' = min(l_so, l_p) = Af_p = e_{p} \cdot l_p' =$: 0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	$AI_p = e_p I_p =$: 0 mm²
Chall are a materia		
Shell geometry	a = t · · · - δ · · · - δ · · · -	. 11.60 mm
Analysis thickness of shell wall Analysis thickness of shell wall	$e_{as} = t_{\text{shell}} - c_{\text{shell}} - \delta_{\text{shell}} = $ e cs =	
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 + =	
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
	$r_{is} = \frac{2h}{2h} + 0.02D_i$	
Check of distance from shell discontinuity		
Maximum length of shell contributing to opening reinforcement	$l_{\rm so} = \sqrt{(2r_{\rm is} + e_{c,s})e_{c,s}} =$	
Head's analysis thickness	ek =	,
Head's knuckle radius	rk =	, -
Distance between nozzle edge and knuckle-shell tangent line as per in Limit distance between nozzle edge and knuckle-shell tangent line as	w(7,7,2) = 0 w min(7.7.2)=2,5·√(ek·rk) =	348,78 mm 117,18 mm
per in 7.7.2	w_iiiii(1.1.2)-2,5' ((ek'ik) =	111,10111111
POI III 1.1.2		

	given in 9.7.2.4 and Figure 9.5.4,		w = take this nozzle into	104,45 mm account when
	knuckle thickness, as defined in of the opening to a shell disconting		l c = w =	104,45 mm
	as no influence on is from shell dis		l_s = w = w_p = l_so =	
Required minimum value for v			w_pco w min =	
·			_	·
Head's knuckle check du	e to encroaching nozzle			
Knuckle thickness			enk(head) =	14,00 mm
Minimum knuckle thickness			ek(head) =	
Adiacent eneminas			enk(nead	l) ≥ ek(head): Ok
Adjacent openings	VN Flance N4 DN400 DN25			
	VN Flange N1 - DN100 PN25 en on the mean surface of the she		Lb =	406,09 mm
Ligament length	sir on the mean surface of the sire		i =	225 E4 mm
Mean shell radius at the centre	es of adjacent nozzles		r is =	990,11 mm
Minimum required ligament le	ngth	$L_{\min} = \max \left[3e_{\alpha\beta}, 0.2 \sqrt{\frac{1}{2}} \right]$	$\left[(2r_{is} + e_{cs})e_{cs} \right] =$	34,80 mm
			d1 =	219,10 mm
			d2 =	142,00 mm
	14 . 10 . 1 0 0 /	· · · · · · · · · · · · · · · · · · ·	NI. C C. C. C. I. P	L ≥ Lmin: Ok
	01+02 ≤ 0,2√j	[(2r_is+e_c,s)·e_c,s]:	Not satisfied, ligame	nt cneck required
Ligament check				
Inside radius of curvature of the	ne shell at the opening centre		D _e =	990,11 mm
inside radius of curvature of the	ne shell at the opening centre centre line of openings and the geb		$r_{is} = \frac{-\epsilon}{2} - e_{as}$	
Angle between the centre-to-c	entre line of openings and the ge	eneratrix of the shell	$\Phi = \begin{pmatrix} 1 + \cos(A) \end{pmatrix}$	90,00 °
Ap of the shell for the length L	.D	$Ap_{1s} = \frac{0.5r_{1s}^2 L}{r_1 + 6}$	$0.5e^{-1\sin(\phi)}$	199 864,9 mm²
		-Ls /is	deb1 =	219,10 mm
			deb2 =	142,00 mm
			φ_e1 =	0 °
			φ_e2 =	0°
		$r_{osl} = -$	$ \frac{r_{\text{is1}}}{\sin^2(\varphi)} + 0.5e_{q,\varphi} = $	995,91 mm
		$r_{os2} = \frac{s}{s}$ $a_1 = r_{os1} \left[\arcsin \left(\delta_1 + s \right) \right]$ $a_2 = r_{os2} \left[\arcsin \left(\delta_2 + s \right) \right]$ $Af_{r_0} = f_{r_0}$	$r_{is2} = $	995,91 mm
		$r_{os2} = -$	$\frac{1}{\sin^2(\varphi)} + 0.5e_{a,s}$	000,01 111111
			$s = \frac{d_{\text{eb1}}}{d_{\text{eb1}}} =$	6,30 °
			$o_1 - \frac{2r_{os1}}{2r_{os1}}$	
			$\delta_2 = \frac{d_{eb2}}{2} =$	4,08 °
		$a = r$. [arcsin ($\delta_{r} + s$	$\sin(\alpha) = 2r_{os2}$	109,77 mm
		$a_1 = r_{os1}$ arcsin ($\delta_1 + s$	$\sin(\varphi_{e1})$ φ_{e1} =	71,06 mm
Af of the shell contained along	the length Lb	$Af_{T_0} = (1)$	$L_b - a_1 - a_2$) $e_{c,s} =$	2 612,9 mm ²
, oo oo ooaoo a.og	, a.ooga. =2	Is (-	Ãρφ1 =	0 mm ²
			Apφ2 =	0 mm²
			Apb1 =	5 724,8 mm ²
			Apb2 = Afb1 =	3 117,6 mm ²
			Afb2 =	582,2 mm² 1 126,9 mm²
			Afp1 =	0 mm ²
			Afp2 =	0 mm²
			fob1 =	111,73 MPa
			fob2 =	134,00 MPa
			fop1 = fop2 =	0 MPa 0 MPa
Reactive force	$F = (Af_{Ls} + Af_w)(f_s - 0.5P) +$	$Af_{b1}(f_{ab1} = 0.5P) + A$	$f_{p1}(f_{op1} - 0.5P) + =$	651 951 N
	$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op})$		- vpi /	
Pressure load	$F_{mn} = P$	$\left(\operatorname{Ap}_{L_{s}}^{2} + \operatorname{Ap}_{b1}^{1} + 0.5\operatorname{Ap}_{\varphi 1}^{1}\right)$	+Ap +0.5Ap \ -	417 415 N
1 1033410 1044	2 1cq 2 1	(rLsrb1 σωrrp _{φ1}	$-\mathbf{r}_{b2}$ $-\mathbf{r}_{\varphi 2}$	F≥Freq: Ok
				. = 1.10q. OK

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

Design data Internal design temperal Internal design pressure Joint efficiency		Ti = Pi = z =	260 °C 2,00 MPa 1,00
Flange material Shell material Bolting material Gasket	P280GH (N) (EN 102 25CrMo4 (EN 10269:2009)	222-2:2001) t ≤ 35,00 mm- Forging 222-2:2001) t ≤ 35,00 mm- Forging) t ≤ 100,00 mm- Bolting stos or mineral fibre filled (Stainless steel or Mo	nel)
Flange standard / specific Flange rating Nominal size Number of bolts Bolt type Material group Calculation temperature Internal pressure Overpressure due to state Calculation pressure Maximum pressure at temperature		= = = = = = = = = = = = = = = = = = =	EN 1092-1:2007 25 100 8 ISO M20 x 2,50 3E1 260 °C 2,00 MPa 0 MPa 2,00 MPa 2,39 MPa
Maximum allowable New & cold (flange) Hot & corroded (flange) New & cold (bolts) Hot & corroded (bolts) Maximum allowable test Maximum allowable des		the vessel) = = = = = = = = = = = = = = = = = = =	2,50 MPa 2,39 MPa 2,50 MPa 2,39 MPa 8,41 MPa 3,87 MPa
Hydrostatic test Item hydrostatic test pre Overpressure due to sta Calculation pressure Maximum pressure at te		Pt = Ph = P = Pt + Ph = ecifications Pmax =	2,95 MPa 0,002 MPa 2,96 MPa 2,50 MPa
Simplified fatigue as	sessment according to	EN13445-3 Clause 17	
Load condition 1, load Design pressure Pressure range Minimum operating temp Maximum operature Design temperature Number of required fatign Nominal design stress a Ultimate tensile strength Yield strength at design	perature during cycle perature during cycle ue cycles t design temperature at room temperature	P = ΔP = Tmin = Tmax = T = Nreq = f = Rm = Rp0,2/T =	2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C 1 200 134,00 MPa 460,00 MPa 201,00 MPa
Load condition 1, Hu Maximum allowable pred Calculation thickness Stress factor Pseudo-elastic stress ra Equivalent number of fu Thickness correction fact Assumed mean cycle te Temperature correction Transition radius	nge Il pressure cycles tor mperature	$Pmax = en = n = n = n = \Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f = Neq = Ce = T*=0,75 \cdot Tmax+0,25 \cdot Tmin = CT = r = r = n$	2,39 MPa 19,85 mm 1,50000 147,05 MPa 469,90695 1,00000 200 °C 0,94000 5,00 mm

Theoretical stress concentration factor Endurance limit Effective stress concentration factor Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	$Kt=1,4 \text{ (r}\geq\text{en}/4) = \\ \Delta\sigma D = \\ 1.5(K_t-1) = \\ \frac{1.5(K_t-1)}{1+0.5\cdot\text{max}(1;K_t\frac{\Delta\sigma}{\Delta\sigma_D})} = \\ \Delta\sigma\text{cut} = \\ \Delta\sigma^*=[\Delta\sigma/\text{Ce}\cdot\text{CT}]\cdot\text{Kf} = \\ N = \\ D=\text{Nreq/N} = $	1,40000 175,20 MPa 1,37794 116,70 MPa 215,56 MPa 370 582 0,00324
Load condition 1, Nozzle without pad weld Maximum allowable pressure (opening) Calculation thickness Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	Pmax = en = n = n = n = n = n = n = n = n =	3,86 MPa 19,85 mm 3,00000 182,26 MPa 111,83229 1,00000 200 °C 0,94000 63 46,43 MPa 25,52 MPa 193,89 MPa 68 662 0,01748
Fatigue cycles and damage index summary Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction Load 1, partial damage index for Nozzle without pad weld Total damage index: Nozzle without pad weld	= = = =	0,00324 0,00324 0,01748 0,01748 TDI(1)<1: Ok TDI(2)<1: Ok
C.17 Cylindrical shell		
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance	Di = De = L = en = c = ce = δ =	102,30 mm 142,00 mm 211,35 mm 19,85 mm 1,00 mm 0 mm 0 mm
Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness	$Ph = P=Pi+Ph = Di'=Di+2\delta+2c = e = \frac{P \cdot D'_i}{2f \cdot z - P} + c + ce + \delta = e$	0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok
Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness	$ \text{Item f0/f} = \\ \text{Pt1=1,25} \cdot \text{Pe} \cdot (\text{Item f0/f}) = \\ \text{Pt2=1,43} \cdot \text{Pe} = \\ \text{Pt=max}(\text{Pt1,Pt2}) = \\ \text{Pht} = \\ \text{Pc=Pt+Pht} = \\ \text{Di'=Di+2} \delta = \\ e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta = \\ \\ \text{Potation of the points} $	1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,002 MPa 2,96 MPa 102,30 mm 0,59 mm en ≥ e: Ok

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

Design data Internal design temperate Internal design pressure Joint efficiency Flange material Shell material Bolting material Gasket	P280GH (N) (EN 10222-2:2001) t P280GH (N) (EN 10222-2:2001) t 25CrMo4 (EN 10269:2009) t ≤ 100,00 m Spiral-wound metal asbestos or mineral fi	≤ 35,00 mm- Forging nm- Bolting	260 °C 2,00 MPa 1,00
Flange standard / specific Flange rating Nominal size Number of bolts Bolt type Material group	cation	= = = = =	EN 1092-1:2007 25 100 8 ISO M20 x 2,50 3E1
·	nperature allowed by the specifications	T = Pd = Ph = P = Pi + Ph = Pmax =	260 °C 2,00 MPa 0 MPa 2,00 MPa 2,39 MPa
New & cold (flange) Hot & corroded (flange) New & cold (bolts) Hot & corroded (bolts) Maximum allowable test Maximum allowable design		= = = = =	2,50 MPa 2,39 MPa 2,50 MPa 2,39 MPa 8,41 MPa 3,87 MPa
Hydrostatic test Item hydrostatic test pres Overpressure due to stat Calculation pressure Maximum pressure at ter		Pt = Ph = P = Pt + Ph = Pmax =	2,95 MPa 0,003 MPa 2,96 MPa 2,50 MPa
	sessment according to EN13445-3 Clau	use 17	
Load condition 1, load Design pressure Pressure range Minimum operating temp Maximum operating temp Design temperature Number of required fatigute Nominal design stress at Ultimate tensile strength Yield strength at design to the pressure of the pressure	erature during cycle perature during cycle ue cycles design temperature at room temperature	P = ΔP = Tmin = Tmax = T = Nreq = f = Rm = Rp0,2/T =	2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C 1 200 134,00 MPa 460,00 MPa 201,00 MPa
Load condition 1, Hull Maximum allowable pres Calculation thickness Stress factor Pseudo-elastic stress ran Equivalent number of full Thickness correction fact Assumed mean cycle ten Temperature correction fact	sure (flange) ge pressure cycles or nperature	$Pmax = \\ en = \\ \eta = \\ \Delta\sigma = (\Delta P/Pmax) \cdot \eta \cdot f = \\ Neq = \\ Ce = \\ T^* = 0.75 \cdot Tmax + 0.25 \cdot Tmin = \\ CT = $	2,39 MPa 19,85 mm 1,50000 147,05 MPa 469,90695 1,00000 200 °C 0,94000

Transition radius	r =	5,00 mm
Theoretical stress concentration factor Endurance limit	Kt=1,4 (r≥en/4) = ΔσD =	1,40000 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 Number of allowable fatigue cycles	Δσ*=[Δσ/Ce·CT]·Kf = N =	215,56 MPa 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 Pseudo-elastic stress range	η = $\Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f =$	3,00000 182,26 MPa
Equivalent number of full pressure cycles	Neq =	111,83229
Thickness correction factor Assumed mean cycle temperature	Ce = T*=0,75·Tmax+0,25·Tmin =	1,00000 200 °C
Temperature correction factor Weld class	CT = C =	0,94000 63
Endurance limit Cut-off limit	ΔσD = Δσcut =	46,43 MPa 25,52 MPa
Fictitious stress range for insertion into the fatigue design curves	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	193,89 MPa
Number of allowable fatigue cycles Partial fatigue damage index	N = D=Nreq/N =	68 662 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 Load 1, partial damage index for Nozzle without pad weld	= =	0,00324 0,01748
Total damage index: Nozzle without pad weld	=	0,01748 TDI(1)<1: Ok
		TDI(2)<1: Ok
C.19 Cylindrical shell		TDI(2)<1: Ok
C.19 Cylindrical shell Inside diameter	Di =	TDI(2)<1: Ok
Inside diameter Outside diameter	De =	102,30 mm 142,00 mm
Inside diameter Outside diameter Length Nominal thickness	De = L = en =	102,30 mm 142,00 mm 249,65 mm 19,85 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance	De = L = en = c = ce =	102,30 mm 142,00 mm 249,65 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance	De = L = en = c =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure	De = L = en = c = ce = δ =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure	De = L = en = c = ce = δ = Ph = P=Pi+Ph =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter	De = L = en = c = ce = δ = Ph = P=Pi+Ph = Di'=Di+2δ+2c =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 2,00 MPa 2,00 MPa 104,30 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure	De = L = en = c = ce = δ = Ph = P=Pi+Ph =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness	De = L = en = c = ce = δ = Ph = P=Pi+Ph = Di'=Di+2δ+2c =	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 2,00 MPa 2,00 MPa 104,30 mm 1,78 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio	$De = L = en = c = en = c = ce = \delta = $ $Ph = P = P = P + P + e = D = P + D + c + ce + \delta = $ $Item f0/f = $	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	$De = L = en = c = en = c = ce = \delta = $ $Ph = P = P = P + P + e = D = P + D + c + ce + \delta = $ $e = \frac{P \cdot D!}{2f \cdot z - P} + c + ce + \delta = $	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure	$De = L = en = c = en = c = ce = \delta = de = de = de = de = de = de = de$	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok 1,18132 2,95 MPa 2,86 MPa 2,95 MPa
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure	$De = L = en = C = en = C = ce = \delta = Ce = \delta = Ce = Ce = Ce = Ce = Ce $	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,003 MPa 2,96 MPa
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter	$De = L = en = c = en = c = ce = \delta = de = de = de = de = de = de = de$	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,003 MPa 102,30 mm
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Internal pressure Overpressure due to static head Calculation pressure Inside diameter Minimum required thickness Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure	$De = L = en = C = en = C = ce = \delta = Ce = \delta = Ce = Ce = Ce = Ce = Ce $	102,30 mm 142,00 mm 249,65 mm 19,85 mm 1,00 mm 0 mm 0 mm 0 MPa 2,00 MPa 104,30 mm 1,78 mm en ≥ e: Ok 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,003 MPa 2,96 MPa

C.20 Nozzle - NOZZLE N3 (DN 200)

Design data		
Design data Internal design temperature	Ti =	260 °C
Internal design pressure	71 – Pi =	2,00 MPa
Joint efficiency	z =	1,00
•	L =	
Material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTul Nominal design stress at internal design temperature		111,73 MPa
Nonlinal design stress at internal design temperature	$f = \min(\frac{R_{\text{p0.2}/T}}{15}; \frac{R_{m/20}}{24})$	TTT,73 WIFa
Nominal design stress at room temperature	$f = \min(\frac{R_{\text{p0.2/20}}}{15}; \frac{R_{\text{m/20}}}{2.4})$	170,83 MPa
	$f = \min(\frac{15}{15}, \frac{24}{24})$	
Nominal design stress in test condition	$f_{\text{test}} = \left(\frac{R_{\text{p02/Ttest}}}{1.05}\right) = \frac{1.5 \cdot 2.4}{1.05}$	252,38 MPa
	test 105	
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+20+20 = P(D)! =	198,88 mm
Minimum required thickness	Di'=Di+2 δ +2c = $e = \frac{P \cdot D'_i}{2f \cdot z - P} + c + ce + \delta$	4,38 mm
	2, 2 1	en≥e: Ok
		5 = 5 G.K
Maximum allowable pressures (at the top of the vessel)		
Maximum allowable test pressure	=	5.61 MDo
		3,0 i ivira
Maximum allowable design pressure	=	5,61 MPa 2,44 MPa
Deformation according to EN 13445-4 Clause 9	=	2,44 MPa
Deformation according to EN 13445-4 Clause 9 Deformation	=	2,44 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test	= F=50·en/(Di/2+en/2) =	2,44 MPa 6,153 %
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio	= F=50·en/(Di/2+en/2) = Item f0/f =	2,44 MPa 6,153 % 1,18132
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1	$F=50\cdot en/(Di/2+en/2)=$ $Item f0/f =$ $Pt1=1,25\cdot Pe\cdot (Item f0/f) =$	2,44 MPa 6,153 % 1,18132 2,95 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2δ =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2δ =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter	= F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2δ =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness	F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2 δ = e = $\frac{P \cdot D_i^t}{2f \cdot z - P} + \delta$ =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause	F=50·en/(Di/2+en/2) = Item f0/f = Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2 δ = e = $\frac{P \cdot D_i^t}{2f \cdot z - P} + \delta$ =	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ 17	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ 17	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure Pressure range	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ $P = \Delta P =$	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok 2,00 MPa 1,75 MPa
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ $P = \Delta P =$ $Tmin =$	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok 2,00 MPa 1,75 MPa 20 °C
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ $P = \Delta P =$ $Tmin =$ $Tmax =$	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok 2,00 MPa 1,75 MPa 20 °C 260 °C
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle Design temperature	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ $P = \Delta P =$ $Tmin =$ $Tmax =$ $T =$	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok 2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C
Deformation according to EN 13445-4 Clause 9 Deformation Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter Minimum required thickness Simplified fatigue assessment according to EN 13445-3 Clause Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle	$F=50 \cdot en/(Di/2+en/2) =$ $Item f0/f =$ $Pt1=1,25 \cdot Pe \cdot (Item f0/f) =$ $Pt2=1,43 \cdot Pe =$ $Pt=max(Pt1,Pt2) =$ $Pht =$ $Pc=Pt+Pht =$ $Di'=Di+2\delta =$ $e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$ $P = \Delta P =$ $Tmin =$ $Tmax =$	2,44 MPa 6,153 % 1,18132 2,95 MPa 2,95 MPa 0,004 MPa 2,96 MPa 196,88 mm 2,75 mm en ≥ e: Ok 2,00 MPa 1,75 MPa 20 °C 260 °C

Ultimate tensile strength at room temperature Yield strength at design temperature	Rm = Rp0,2/T =	410,00 MPa 167,60 MPa
Load condition 1, Circumferential butt weld Maximum allowable pressure (cylinder) Calculation thickness Joint efficiency Offset Partial stress factor Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	$Pmax = en = z = \delta = \delta = \eta 1 = \delta/(2 \cdot en) = \eta = (1 + \eta 1) \cdot z = \Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f = Neq = Ce = T*=0,75 \cdot Tmax + 0,25 \cdot Tmin = CT = C = \Delta \sigma D = \Delta \sigma cut = \Delta \sigma = \Delta \sigma cut = N = D = Nreg/N = D = Nreg/N = Social = Soc$	10,80 MPa 12,70 mm 1,00000 1,27 mm 0,01969 1,01969 18,46 MPa 5,10094 1,00000 200 °C 0,94000 40 29,48 MPa 16,20 MPa 19,63 MPa 38 155 258 0,00003
	D-ME4/M -	0,00003
Load condition 1, Nozzle without pad weld Maximum allowable pressure (head) Calculation thickness Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	Pmax = en = n = n = n = n = n = n = n = n =	3,43 MPa 12,70 mm 3,00000 170,87 MPa 158,95492 1,00000 200 °C 0,94000 63 46,43 MPa 25,52 MPa 181,78 MPa 83 325 0,01440
Fatigue cycles and damage index summary Load 1, partial damage index for Circumferential butt weld Total damage index: Circumferential butt weld Load 1, partial damage index for Nozzle without pad weld Total damage index: Nozzle without pad weld Validation warnings:	= = = =	0,00003 0,00003 0,01440 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) $t \le 35,00$ mm- Forging Shell material P265GH (EN 10216-2:2008) $t \le 16,00$ mm- SeamlessTube 25CrMo4 (EN 10269:2009) $t \le 100,00$ 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 Number of bolts Bolt type Material group	= = = =	200 12 ISO M24 x 3,00 3E1
Calculation temperature Internal pressure Overpressure due to static head Calculation pressure Maximum pressure at temperature allowed by the specifications	T = Pd = Ph = P = Pmax =	260 °C 2,00 MPa 0 MPa 2,00 MPa 2,39 MPa
Maximum allowable pressures (at the top of the vessel) New & cold (flange) Hot & corroded (flange) New & cold (bolts) Hot & corroded (bolts)	= = = =	2,50 MPa 2,39 MPa 2,50 MPa 2,39 MPa
Hydrostatic test Item hydrostatic test pressure Overpressure due to static head Calculation pressure Maximum pressure at temperature allowed by the specifications	Pt = Ph = P = (Pt + Ph) / 1,43 = Pmax =	2,95 MPa 0,0009 MPa 2,07 MPa 2,50 MPa
Simplified fatigue assessment according to EN 13445-3 Claus Load condition 1, load details	s e 17	2.00 MPa
Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle Design temperature Number of required fatigue cycles Nominal design stress at design temperature Ultimate tensile strength at room temperature Yield strength at design temperature	P = ΔP = Tmin = Tmax = T = Nreq = f = Rm = Rp0,2/T =	2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C 1200 134,00 MPa 460,00 MPa 201,00 MPa
Load condition 1, Junction to shell (of thickness es) Maximum allowable pressure (flange) Calculation thickness Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	Pmax = en = n = n = n = n = n = n = n = n =	2,39 MPa 12,70 mm 1,50000 147,05 MPa 469,90695 1,00000 200 °C 0,94000 63 46,43 MPa 25,52 MPa 156,44 MPa 130 725 0,00918
Load condition 1, Hub to plate junction Maximum allowable pressure (flange) Calculation thickness Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Transition radius Theoretical stress concentration factor Endurance limit	$Pmax = en = n$ $q = 0$ $\Delta \sigma = (\Delta P / Pmax) \cdot \eta \cdot f = 0$ $Neq = 0$ $Ce = 0$ $T^* = 0.75 \cdot Tmax + 0.25 \cdot Tmin = 0$ $CT = 0$ $r = 0$ $Kt = 1.4 (r \ge en/4) = 0$ $\Delta \sigma D = 0$	2,39 MPa 16,30 mm 1,50000 147,05 MPa 469,90695 1,00000 200 °C 0,94000 5,00 mm 1,40000 175,20 MPa

Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index Fatigue cycles and damage index summary Load 1, partial damage index for Junction to shell (of thickness es Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction	$K_f = 1 + \frac{15(K_t - 1)}{1 + 0.5 \cdot \max(1; K_t \frac{d\sigma}{d\sigma_D})} = \frac{\Delta \sigma \cot t}{\Delta \sigma^* = [\Delta \sigma / \text{Ce-CT}] \cdot \text{Kf}} = \frac{N}{N} = \frac{D = \text{Nreq/N}}{D = N}$	370 582 0,00324 0,00918
C.22 Conical shell - CONICAL SHELL		
According to: EN 13445 Ed. 2009 Issue 5, Clauses 7 and 8		
Design data Internal design temperature Internal design pressure Joint efficiency	Ti = Pi = z =	
Material: P355GH (EN 10028-2:2009) t ≤ 16,00 mm- Plate Nominal design stress at internal design temperature Nominal design stress at room temperature Nominal design stress in test condition	$f = \min(\frac{R_{\text{p0.2}/T}}{15}; \frac{R_{\text{m/20}}}{24}) = f = \min(\frac{R_{\text{p0.2}/20}}{15}; \frac{R_{\text{m/20}}}{24}) = f_{\text{test}} = (\frac{R_{\text{p0.2}/\text{Ttest}}}{105}) = f_{\text{test}} = \frac{R_{\text{p0.2}/\text{Ttest}}}{105}$	165,33 MPa 212,50 MPa 338,10 MPa
Geometry Length Nominal thickness Corrosion allowance External corrosion allowance Undertolerance Maximum Inside Diameter Maximum Outside Diameter Minimum Inside Diameter Minimum Outside Diameter Minimum Outside Diameter Half-apex angle Cone thickness at large end Cone thickness at small end Nominal thickness of cylinder at large end Minimum thickness of cylinder at small end Minimum thickness of cylinder at small end	L = en = c = ce = δ = Di = De = di = de = α = e2nL = e2nS = e1nL = e1ns = e1s =	12,00 mm 1,00 mm 0 mm 0,30 mm
Internal pressure Overpressure due to static head Calculation pressure Mean diameter of the cone at large end Calculation diameter Minimum required cone thickness Maximum allowable pressure of conical section at large end	$Ph = P = P = P = P + Ph = Dc = Di + e + nL + c + ce + \delta = DK = Dc - e + L - 2r[1 - cos(\alpha)] - l2 \cdot sin(\alpha) = e + e + e + e + e + e + e + e + e + e$	0,003 MPa 2,00 MPa 1 113,30 mm 1 012,43 mm 11,51 mm 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	$1,4\cdot 12L=1,4\cdot \sqrt{((Dc^*e2L)/cos(\alpha))}=$	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
,	- (- 3-/	e1nL≥ej: Ok
		e2nL≥ej: Ok
Ower Hand Street Law		
Small end junction Mean diameter of the cone	$dc=di+e1ns+c+ce+\delta=$	162,27 mm
Minimum length along cylinder	I1s=√(dc*e1s') =	37,35 mm
Minimum length along cone	$12s = \sqrt{((dc^*e2s')/\cos(\alpha))} =$	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	$F=50 \cdot en/(di/2+en/2) =$	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 Item hydrostatic test pressure	Pt2=1,43·Pe = Pt=max(Pt1,Pt2) =	2,86 MPa 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)]-I2·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
		CHECZ. OR
Large end junction (without knuckle)		
Minimum length along cylinder	$1,4\cdot 11L=1,4\cdot \sqrt{(Dc*e1L)} =$	139,29 mm
Minimum length along cone	$1,4\cdot l2L=1,4\cdot \sqrt{((Dc*e2L)/cos(\alpha))} =$	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		8,35 mm
Maximum allowable pressure of junction at large end	Pmax(large) =	4,34 MPa e1nL≥ej: Ok
		e2nL≥ej: Ok
		02/12=0j. OK
Small end junction		
Mean diameter of the cone	$dc=di+e1ns+c+ce+\delta =$	161,27 mm
Minimum length along cylinder	I1s=√(dc*e1s') =	42,06 mm
Minimum length along cone	$12s = \sqrt{((dc \cdot e2s')/cos(\alpha))} =$	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 = 7.6-25 =	2,27 mm
βH factor defined in 7.6.8 Minimum required thickness at the junction at the small end of the		1,18 mm 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
Observation of Section 2	0.01 47	
Simplified fatigue assessment according to EN 13445-3	Guause 17	
Load condition 1, load details	D -	2 00 MD=
Design pressure Pressure range	P = ΔP =	2,00 MPa 1,75 MPa
Minimum operating temperature during cycle	ΔP – Tmin =	1,75 MFa 20 °C
Maximum operating temperature during cycle	Tmax =	260 °C
1		

Design temperature Number of required fatigue cycles Nominal design stress at design temperature Ultimate tensile strength at room temperature Yield strength at design temperature	T = Nreq = f = Rm = Rp0,2/T =	260 °C 1 200 165,33 MPa 510,00 MPa 248,00 MPa
Load condition 1, Longitudinal butt weld Maximum allowable pressure (component) Nominal thickness Inside diameter Offset Peeking or flat Ovality Partial stress factor Partial stress factor Partial stress factor Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	$Pmax = \\ en = \\ Di = \\ \delta o = \\ \delta pf = \\ u = \\ \eta 1 = (3 \cdot \delta o)/en = \\ \eta 2 = 1,5 \cdot u \cdot (Di/en) = \\ \eta 4 = 6 \cdot \delta pf/en = \\ \eta = (1 + \eta 1 + \eta 2 + \eta 4) \cdot z = \\ \Delta \sigma = (\Delta P/Pmax) \cdot \eta \cdot f = \\ Neq = \\ Ce = \\ T^* = 0,75 \cdot Tmax + 0,25 \cdot Tmin = \\ CT = \\ C = \\ \Delta \sigma D = \\ \Delta \sigma cut = \\ \Delta \sigma cut = \\ \Delta \sigma = [\Delta \sigma/Ce \cdot CT] = \\ N = \\ D = Nreq/N = \\ Co = \\ C = (\Delta \sigma/Ce \cdot CT) = \\ N = \\ D = Nreq/N = \\ Co = (\Delta \sigma/Ce \cdot CT) = \\ C = (\Delta \sigma/Ce \cdot CT) $	2,08 MPa 12,00 mm 1100,00 mm 0,60 mm 2,00 mm 0,75% 0,15000 1,03125 1,00000 2,70406 375,73 MPa 712,30418 1,00000 200 °C 0,94000 63 46,43 MPa 25,52 MPa 399,71 MPa 7 837 0,15312
Load condition 1, Circumferential butt weld		,
Joint efficiency Maximum allowable pressure (component) Calculation thickness Joint efficiency Partial stress factor Offset Partial stress factor Stress factor Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor Assumed mean cycle temperature Temperature correction factor Weld class Endurance limit Cut-off limit Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	z = Pmax = en = z = q0 = d = d = d = d = d = d = d = d = d =	0,85000 2,08 MPa 12,00 mm 0,85000 0,10000 0,60 mm 0,02500 0,95625 132,87 MPa 712,30418 1,00000 200 °C 0,94000 63 46,43 MPa 25,52 MPa 141,35 MPa 177 214 0,00677
Fatigue cycles and damage index summary Load 1, partial damage index for Longitudinal butt weld Total damage index: Longitudinal butt weld Load 1, partial damage index for Circumferential butt weld Total damage index: Circumferential butt weld	= = = =	0,15312 0,15312 0,00677 0,00677 TDI(1)<1: Ok TDI(2)<1: Ok

C.23 Cylindrical shell - NOZZLE N4 (DN150)

Joint efficiency: Material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTube Nominal design stress at internal design temperature Nominal design stress at room temperature Nominal design stress at room temperature $ f = \min_{\substack{N \neq 220 \\ N \neq 220 \\ N \neq 220}} \frac{R_{po217}}{R_{po20}} = 170,83 \text{ MP} $ Nominal design stress in test condition $ f = \min_{\substack{N \neq 220 \\ N \neq 220 \\ N \neq 20}} \frac{R_{po217}}{R_{po217}} \frac{R_{po20}}{R_{po217}} = 170,83 \text{ MP} $ Nominal design stress in test condition $ f = \min_{\substack{N \neq 220 \\ N \neq 20}} \frac{R_{po217}}{R_{po217}} \frac{R_{po20}}{R_{po217}} = 170,83 \text{ MP} $ Nominal these in test condition $ f = \min_{\substack{N \neq 220 \\ N \neq 20}} \frac{R_{po217}}{R_{po217}} \frac{R_{po20}}{R_{po217}} = 170,83 \text{ MP} $ Nominal thickness $ f = \min_{\substack{N \neq 220 \\ N \neq 20}} \frac{R_{po217}}{R_{po217}} \frac{R_{po20}}{R_{po217}} = 170,83 \text{ MP} $ Corrosion allowance $ f = n = 10,97 \text{ m} $ Nominal thickness $ f = n = 10,97 \text{ m} $ Internal pressure Overpressure due to static head Overpressure due to static head Ph = DiPi+Ph = 2,00 MP Calculation pressure Naminum allowable pressures (at the top of the vessel) Maximum allowable test pressure Maximum allowable design pressure $ f = n = 10,002 \text{ MP} $ Maximum allowable design pressure Maximum allowable statio tem minimum allowables ratio tem minimum pressure Perany(Pt,Pe) = 2,95 MP Overpressure as per EN13445-5 Formula 10.2.3.3.1-1 Pt1=1,25-Pe·(Item f0/f) = 1,1813 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Pt2=1,43-Pe = 2,86 MP Pt=max(Pt,Pe) = 2,95 MP Overpressure due to static head in test condition Perany(Pt,Pe) = 2,95 MP Overpressure due to static head in test condition Perany(Pt,Pe) = 2,95 MP Overpressure ange Perpliphicable Perpliphicable Perpliphicable Perpliphicable Perpliphicable Perpl	Design data Internal design temperature Internal design pressure	Ti = Pi =	260 °C 2,00 MPa
Nominal design stress at internal design temperature $f = \frac{R_{RA2} T_{1}}{I_{1}} \frac{R_{R220}}{15} = 111,73 \text{ MP}$ Nominal design stress at room temperature $f = \min(\frac{R_{RA2} T_{1}}{15}, \frac{R_{R220}}{24}) = 170,83 \text{ MP}$ Nominal design stress in test condition $\frac{R_{RA2} T_{1}}{I_{1}} = \frac{R_{RA2} T_{1}}{105} = 252,38 \text{ MP}$ Nominal design stress in test condition $\frac{R_{RA2} T_{1}}{I_{1}} = \frac{R_{RA2} T_{1}}{105} = 252,38 \text{ MP}$ Secondary $\frac{R_{RA2} T_{1}}{I_{1}} = \frac{R_{RA2} T_{1}}{I_{1}} = 242$ Secondary $\frac{R_{RA2} T_{1}}$			1,00
Nominal design stress at room temperature $f = \frac{1}{p} \frac{1}{p$	Material: P265GH (EN 10216-2:2008) t ≤ 16,00 mm- SeamlessTub	oe e	
Nominal design stress at room temperature $f = \min \left(\frac{N_{pa220}}{1.5}, \frac{N_{pa20}}{2.1} \right) = 170,83 \text{MP}$ Nominal design stress in test condition $f = \min \left(\frac{N_{pa20}}{1.05}, \frac{N_{pa20}}{1.05} \right) = 252,38 \text{MP}$ Geometry Inside diameter $Di = 16,30 \text{mr}$ Inside diameter $De = 168,30 \text{mr}$ Length $De = 168,30 \text{mr}$ Nominal thickness $en = 10,97 \text{mr}$ Corrosion allowance $en = 10,97 \text{mr}$ Corrosion allowance $en = 10,97 \text{mr}$ Undertolerance $\delta = 10,00 \text{mr}$ Internal pressure Overpressure due to static head $Ph = 10,003 \text{MP}$ Calculation pressure due to static head $Ph = 10,003 \text{MP}$ Calculation pressure due to static head $Ph = 10,003 \text{MP}$ Calculation pressure due to static head $Ph = 10,003 \text{MP}$ Calculation pressure $e^{\frac{P+D_1}{2}+P+C} + c \cdot e + \delta = \frac{P+D_1}{2} + c \cdot e +$	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
Geometry Inside diameter Di = 146,36 mm Outside diameter De = 186,30 mm Length L = 75,15 mm Nominal thickness en = 10,97 mm Corrosion allowance c = 1,09 mm External corrosion allowance c = 10,97 mm Undertolerance δ = 0,7 mm Internal pressure P=Pi+Ph = 0,003 MP Calculation pressure due to static head P=Pi+Ph = 0,003 MP Calculation pressure P=Di=Di+26+2c = 151,10 mm Inside diameter Di=Di+26+2c = 151,10 mm Maximum ellowable pressures (at the top of the vessel) = $\frac{P\cdot D_i^2}{2f\cdot z - P} + c + ce + \delta$ = 3,74 mm Maximum allowable test pressure = 30,49 MP Maximum allowable design pressure = 30,49 MP Deformation according to EN 13445-4 Clause 9 Endown the complex of the vessel	Nominal design stress at room temperature	$f = \min(\frac{R_{\text{p0.2/20}}}{R_{\text{p0.2/20}}} \cdot \frac{R_{\text{m/20}}}{R_{\text{m/20}}}) =$	170,83 MPa
Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance Overpressure Overpressure due to static head Corrosion allowance Ph = 0,003 MP Calculation pressure Overpressure due to static head Ph = 0,003 MP Calculation pressure Ninimum required thickness P=Pi+Ph = 2,000 MP Raximum allowable pressures (at the top of the vessel) Maximum allowable test pressure Maximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Raximum allowable test pressure P=00 Pi=Di+2δ+2c = 151,10 mm Pi=1,25-Pe-(litem f0/f) = 1,1813 Pi=1,25-Pe-(litem f0/f) = 2,95 MP Pi=max(PTI-Pi=2) = 2,86 MP Pi=max(PTI-Pi=2) = 2,86 MP Pi=Di+2δ = 2,86 MP Pi=Di+2δ = 2,86 MP Pi=Di+2δ = 2,86 MP Pi=Di+2δ = 2,95 MP Pi=Di+2δ = 2,95 MP Pi=Di+2δ = 1,75 MP Riximum required thickness P=00 Pi-Pi+Pt = 0,03 MP Pi=Di+2δ = 1,75 MP Riximum operating temperature during cycle Minimum required thickness P= 2,00 MP Pressure ange P= 2,00 MP Pressure ange P= 1,75 MP Riximum operating temperature during cycle Minimum operating temperature during cycle Nomber of required fatigue cycles Nreq = 120	Nominal design stress in test condition	$f_{\text{test}} = (\frac{R_{\text{p02}}/\text{Ttest}}{105}) =$	252,38 MPa
Overpressure due to static head $Ph = 0,003 \text{ MP} \\ \text{Calculation pressure} \\ \text{Inside diameter} \\ \text{Minimum required thickness} \\ Phi+Ph = 151,10 \text{ mr} \\ \text{Inside diameter} \\ \text{Minimum required thickness} \\ Phi-Phi+Ph = 151,10 \text{ mr} \\ \text{Inside diameter} \\ \text{Minimum required thickness} \\ Phi-Phi+Phi = 151,10 \text{ mr} \\ \text{Inside diameter} \\ Phi-Phi-Phi = 151,10 \text{ mr} \\ \text{Inside diameter} \\ \text{Maximum allowable pressures (at the top of the vessel)} \\ \text{Maximum allowable test pressure} \\ \text{Maximum allowable design pressure} \\ \text{Paramation according to EN 13445-4 Clause 9} \\ \text{Deformation according to EN 13445-4 Clause 9} \\ \text{Deformation according to EN 13445-5 Formula 10.2.3.3.1-1} \\ \text{Pti} = 1,25 \cdot \text{Pe} \cdot \text{(Item f0/f)} = 1,1813 \\ \text{Test pressure as per EN13445-5 Formula 10.2.3.3.1-2} \\ \text{Item profits pressure as per EN13445-5 Formula 10.2.3.3.1-2} \\ \text{Ptem prypressure due to static head in test condition} \\ \text{Ptem prypressure due to static head in test condition} \\ \text{Ptemax(Pt1,Pt2)} = 2,95 \text{ MP} \\ \text{Overpressure due to static head in test condition} \\ \text{Plnt} = 0,03 \text{ MP} \\ \text{Calculation pressure} \\ \text{Pc} = Pt-Phi = 2,99 \text{ MP} \\ \text{Inside diameter} \\ \text{Minimum required thickness} \\ P = 2,20 \text{ MP} \\ \text{Ensign pressure} \\ \text{Pe} = \frac{P \cdot D_i}{2} + \delta = 2,20 \text{ MP} \\ \text{Pressure range} \\ \text{AP} = 1,75 \text{ MP} \\ \text{Pressure range} \\ \text{Maximum operating temperature during cycle} \\ \text{Tmin} = 20^{\circ} \\ \text{Maximum operating temperature during cycle} \\ \text{Tmin} = 20^{\circ} \\ \text{Number of required fatigue cycles} \\ \text{Nreq} = 120 \\ \text{Nreq} = 12$	Inside diameter Outside diameter Length Nominal thickness Corrosion allowance External corrosion allowance	De = L = en = c = ce =	146,36 mm 168,30 mm 75,15 mm 10,97 mm 1,00 mm 0 mm 1,37 mm
Maximum allowable test pressure Maximum allowable design pressure Deformation according to EN 13445-4 Clause 9 Deformation F=50·en/(Di/2+en/2) = 6,973 $^{\circ}$ Hydrostatic test Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Test pressure due to static head in test condition Calculation pressure Pc=Pt+Pht = 0,03 MP Calculation pressure Pc=Pt+Pht = 0,03 MP Inside diameter Minimum required thickness Pi=Di' ₁ − δ = 149,10 mr Allowable fatigue assessment according to EN 13445-3 Clause 17 Load condition 1, load details Design pressure P = 2,00 MP Timin = 20 $^{\circ}$ Maximum operating temperature during cycle Timin = 20 $^{\circ}$ Maximum operating temperature during cycle Timax = 260 $^{\circ}$ Design temperature T = 200 $^{\circ}$ Number of required fatigue cycles Nreq = 120	Overpressure due to static head Calculation pressure Inside diameter	P=Pi+Ph =	0,003 MPa 2,00 MPa 151,10 mm 3,74 mm en ≥ e: Ok
Deformation F=50·en/(Di/2+en/2) = 6,973 Structure Hydrostatic test Item minimum allowables ratio Item f0/f = 1,1813 Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Pt1=1,25·Pe·(ltem f0/f) = 2,95 MP Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Pt2=1,43·Pe = 2,86 MP Item hydrostatic test pressure Pt2=1,43·Pe = 2,95 MP Overpressure due to static head in test condition Pht = 0,03 MP Calculation pressure Pc=Pt+Pht = 2,99 MP Inside diameter Di'=Di+20 = 149,10 mr Minimum required thickness e P·D' ₁ 2,26 mr Minimum required thickness P·D' ₁ 2,26 mr Ensign pressure P = 2,00 MP Vacad condition 1, load details Design pressure P = 2,00 MP Pressure range ΔP = 1,75 MP Minimum operating temperature during cycle Tmin = 20°° Maximum operating temperature during cycle Tmin = 20°° Number of required fatigue cycles Nreq = 120	Maximum allowable test pressure	=	30,49 MPa 12,03 MPa
Item minimum allowables ratioItem f0/f =1,1813Test pressure as per EN13445-5 Formula 10.2.3.3.1-1Pt1=1,25·Pe·(Item f0/f) =2,95 MPTest pressure as per EN13445-5 Formula 10.2.3.3.1-2Pt2=1,43·Pe =2,86 MPItem hydrostatic test pressurePt=max(Pt1,Pt2) =2,95 MPOverpressure due to static head in test conditionPt =0,03 MPCalculation pressurePc=Pt+Pht =2,99 MPInside diameterDi'=Di+2δ =149,10 mrMinimum required thickness $e = \frac{P \cdot D_i^+}{2f \cdot z - P} + \delta =$ 2,26 mrSimplified fatigue assessment according to EN 13445-3 Clause 17Load condition 1, load detailsP =2,00 MPPressure rangeAP =1,75 MPMinimum operating temperature during cycleTmin =20 °Maximum operating temperature during cycleTmax =260 °Number of required fatigue cyclesNreq =1 20		F=50·en/(Di/2+en/2) =	6,973 %
Load condition 1, load detailsDesign pressureP = 2,00 MPPressure range ΔP = 1,75 MPMinimum operating temperature during cycleTmin = 20 ° 0Maximum operating temperature during cycleTmax = 260 ° 0Design temperatureT = 260 ° 0Number of required fatigue cyclesNreq = 120	Item minimum allowables ratio Test pressure as per EN13445-5 Formula 10.2.3.3.1-1 Test pressure as per EN13445-5 Formula 10.2.3.3.1-2 Item hydrostatic test pressure Overpressure due to static head in test condition Calculation pressure Inside diameter	Pt1=1,25·Pe·(Item f0/f) = Pt2=1,43·Pe = Pt=max(Pt1,Pt2) = Pht = Pc=Pt+Pht = Di'=Di+2δ =	1,18132 2,95 MPa 2,86 MPa 2,95 MPa 0,03 MPa 2,99 MPa 149,10 mm 2,26 mm en ≥ e: Ok
Design pressureP =2,00 MPPressure range ΔP =1,75 MPMinimum operating temperature during cycleTmin =20 °Maximum operating temperature during cycleTmax =260 °Design temperatureT =260 °Number of required fatigue cyclesNreq =1 20		17	
	Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle Design temperature Number of required fatigue cycles Nominal design stress at design temperature	ΔP = Tmin = Tmax = T = Nreq = f =	2,00 MPa 1,75 MPa 20 °C 260 °C 260 °C 1 200 111,73 MPa 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	$\eta = (1+\eta 1) \cdot z =$	1,02735
Pseudo-elastic stress range	$\Delta \sigma = (\Delta P / P max) \cdot \eta \cdot f =$	16,70 MPa
Equivalent number of full pressure cycles	Neq =	3,69475
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T*=0,75\cdot Tmax+0,25\cdot Tmin =$	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	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	17,77 MPa
Number of allowable fatigue cycles	N =	Unlimited
Since $\Delta \sigma^* < \Delta \sigma$ cut, 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	=	Ō
•		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) $t \le 35,00$ mm- Forging Shell material P265GH (EN 10216-2:2008) $t \le 16,00$ mm- SeamlessTube 25CrMo4 (EN 10269:2009) $t \le 100,00$ mm- Bolting

Gasket Spiral-wound metal asbestos or mineral fibre filled (Stainless steel or Monel)

Flange standard / specification Flange rating Nominal size Number of bolts Bolt type Material group	= = = = =	EN 1092-1:2007 25 150 8 ISO M24 x 3,00 3E1
Calculation temperature Internal pressure Overpressure due to static head Calculation pressure Maximum pressure at temperature allowed by the specifications	T = Pd = Ph = P = Pmax =	260 °C 2,00 MPa 0,003 MPa 2,00 MPa 2,39 MPa
Maximum allowable pressures (at the top of the vessel) New & cold (flange) Hot & corroded (flange) New & cold (bolts) Hot & corroded (bolts)	= = =	2,50 MPa 2,39 MPa 2,50 MPa 2,39 MPa

Hydrostatic test Item hydrostatic test pressure Overpressure due to static head Calculation pressure Maximum pressure at temperature allowed by the specifications	Pt = Ph = P = (Pt + Ph) / 1,43 = Pmax =	2,95 MPa 0,03 MPa 2,09 MPa 2,50 MPa
Simplified fatigue assessment according to EN 13445-3 Cla		2,50 WII a
	ause 17	
Load condition 1, load details Design pressure Pressure range Minimum operating temperature during cycle Maximum operating temperature during cycle	P = ΔP = Tmin = Tmax =	2,00 MPa 1,75 MPa 20 °C 260 °C
Design temperature Number of required fatigue cycles	T = Nreq =	260 °C 1 200
Nominal design stress at design temperature Ultimate tensile strength at room temperature Yield strength at design temperature	f = Rm = Rp0,2/T =	134,00 MPa 460,00 MPa 201,00 MPa
Load condition 1, Junction to shell (of thickness es)		0.00.140
Maximum allowable pressure (flange) Calculation thickness Stress factor	Pmax = en = η =	2,39 MPa 10,97 mm 1,50000
Pseudo-elastic stress range Equivalent number of full pressure cycles Thickness correction factor	Δσ=(ΔP/Pmax)·η·f = Neq = Ce =	147,26 MPa 471,85718 1,00000
Assumed mean cycle temperature Temperature correction factor Weld class	T*=0,75·Tmax+0,25·Tmin = CT = C =	200 °C 0,94000 63
Endurance limit Cut-off limit	$\Delta \sigma D = \Delta \sigma cut = \Delta \sigma cut$	46,43 MPa 25,52 MPa
Fictitious stress range for insertion into the fatigue design curves Number of allowable fatigue cycles Partial fatigue damage index	Δσ*=[Δσ/Ce·CT] = N = D=Nreq/N =	156,66 MPa 130 185 0,00922
Load condition 1, Hub to plate junction		
Maximum allowable pressure (flange) Calculation thickness Stress factor	Pmax = en =	2,39 MPa 14,50 mm 1,50000
Pseudo-elastic stress range Equivalent number of full pressure cycles	η = Δσ=(Δ,P/Pmax)·η·f = Neg =	147,26 MPa 471,85718
Thickness correction factor Assumed mean cycle temperature Temperature correction factor	Ce = T*=0,75·Tmax+0,25·Tmin = CT =	1,00000 200 °C 0,94000
Transition radius Theoretical stress concentration factor Endurance limit	r = Kt=1,4 (r≥en/4) = .σD =	3,63 mm 1,40000 175,20 MPa
Effective stress concentration factor	$K_f = 1 + \frac{1.5(K_t - 1)}{1 + 0.5 \cdot \max(1, K_t \frac{\Delta \sigma}{\Delta \sigma_D})} = 0$	1,37775
Cut-off limit Fictitious stress range for insertion into the fatigue design curves	$\Delta \sigma$ = $\Delta \sigma^* = [, \sigma/\text{Ce} \cdot \text{CT}] \cdot \text{Kf} = 0$	116,70 MPa 215,83 MPa
Number of allowable fatigue cycles Partial fatigue damage index	N = D=Nreq/N =	367 972 0,00326
Fatigue cycles and damage index summary Load 1, partial damage index for Junction to shell (of thickness es) Total damage index: Junction to shell (of thickness es) Load 1, partial damage index for Hub to plate junction Total damage index: Hub to plate junction	= = = =	0,00922 0,00922 0,00326 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) $t \le 16,00 \text{ mm}$ - Plate Bracket material: P355GH (EN 10028-2:2009) $t \le 16,00 \text{ mm}$ - 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)	=	Α
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 m ²
Wind pressure	Wp =	0 MPa
Horizontal seismic acceleration	Sh =	0 g
Vertical seismic acceleration	Sv =	0 q

	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
$F_{AB} = \frac{F}{AB} + \frac{4M_A}{AB}$	3 729 N	10 240 N	4 050 N
$F_{\text{Vi}} = \frac{F_n}{n} + \frac{\pi N_A}{n[D_i + 2(a_1 + e_a + e_a)]}$			
$F_{\rm Hi} = \frac{F_H}{n}$	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
$\sqrt{D_{\rm eq}e_a}$			
$K_{17} = \frac{1}{\sqrt{0.36 + 0.5\lambda + 0.5\lambda^2}}$	0,34973	0,34973	0,33421
$\sigma_m = \frac{P \cdot D_{eq}}{2e_a}$	0 MPa	168,71 MPa	126,74 MPa
$v_1 = \min[0.08\lambda; 0.40]$	0,27881	0,27881	0,29405
$v_2 = \frac{\sigma_m}{K_2 f}$	0	0,47525	0,61324
$K_{*} = \frac{1 - v_{2}^{2}}{}$	1,30226	0,78013	0,57912
$K_1 = \frac{1 - v_2^2}{\left(\frac{1}{3} + v_1 v_2\right) + \sqrt{\left(\frac{1}{3} + v_1 v_2\right)^2 + (1 - v_2^2)v_1^2}}$			
r\Z	1,25	1,05	1,25
$\sigma_{b,\text{all}} = K_1 K_2 f$	345,91 MPa	276,95 MPa	119,69 MPa
$\sigma_{b,\text{all}} = K_1 K_2 f$ $a_{1,\text{eq}} = a_1 + e_2 + \frac{F_{\text{Hi}} \cdot h}{F_{\text{Vi}}}$	170,00 mm	170,00 mm	170,00 mm
$F_{i,\text{max}} = \left(\frac{\sigma_{b,\text{all}} \cdot e_a^2 \cdot b_3}{K_{17} \cdot a_{1,\text{eq}}}\right)$	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

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	$\Delta \sigma = (\Delta P / P max) \cdot \eta \cdot f =$	223,54 MPa
Equivalent number of full pressure cycles	Neq =	603,65734
Thickness correction factor	Ce =	1,00000
Assumed mean cycle temperature	$T*=0,75\cdot Tmax+0,25\cdot Tmin =$	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	$\Delta \sigma^* = [\Delta \sigma / \text{Ce} \cdot \text{CT}] =$	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

72



British Standards Institution (BSI)

BSI is the national body responsible for preparing British Standards and other standards-related publications, information and services.

BSI is incorporated by Royal Charter. British Standards and other standardization products are published by BSI Standards Limited.

About us

We bring together business, industry, government, consumers, innovators and others to shape their combined experience and expertise into standards -based solutions.

The knowledge embodied in our standards has been carefully assembled in a dependable format and refined through our open consultation process. Organizations of all sizes and across all sectors choose standards to help them achieve their goals.

Information on standards

We can provide you with the knowledge that your organization needs to succeed. Find out more about British Standards by visiting our website at bsigroup.com/standards or contacting our Customer Services team or Knowledge Centre.

Buying standards

You can buy and download PDF versions of BSI publications, including British and adopted European and international standards, through our website at bsigroup.com/shop, where hard copies can also be purchased.

If you need international and foreign standards from other Standards Development Organizations, hard copies can be ordered from our Customer Services team.

Subscriptions

Our range of subscription services are designed to make using standards easier for you. For further information on our subscription products go to bsigroup.com/subscriptions.

With **British Standards Online (BSOL)** you'll have instant access to over 55,000 British and adopted European and international standards from your desktop. It's available 24/7 and is refreshed daily so you'll always be up to date.

You can keep in touch with standards developments and receive substantial discounts on the purchase price of standards, both in single copy and subscription format, by becoming a **BSI Subscribing Member**.

PLUS is an updating service exclusive to BSI Subscribing Members. You will automatically receive the latest hard copy of your standards when they're revised or replaced.

To find out more about becoming a BSI Subscribing Member and the benefits of membership, please visit bsigroup.com/shop.

With a **Multi-User Network Licence (MUNL)** you are able to host standards publications on your intranet. Licences can cover as few or as many users as you wish. With updates supplied as soon as they're available, you can be sure your documentation is current. For further information, email bsmusales@bsigroup.com.

BSI Group Headquarters

389 Chiswick High Road London W4 4AL UK

Revisions

Our British Standards and other publications are updated by amendment or revision.

We continually improve the quality of our products and services to benefit your business. If you find an inaccuracy or ambiguity within a British Standard or other BSI publication please inform the Knowledge Centre.

Copyright

All the data, software and documentation set out in all British Standards and other BSI publications are the property of and copyrighted by BSI, or some person or entity that owns copyright in the information used (such as the international standardization bodies) and has formally licensed such information to BSI for commercial publication and use. Except as permitted under the Copyright, Designs and Patents Act 1988 no extract may be reproduced, stored in a retrieval system or transmitted in any form or by any means – electronic, photocopying, recording or otherwise – without prior written permission from BSI. Details and advice can be obtained from the Copyright & Licensing Department.

Useful Contacts:

Customer Services

Tel: +44 845 086 9001

Email (orders): orders@bsigroup.com
Email (enquiries): cservices@bsigroup.com

Subscriptions

Tel: +44 845 086 9001

Email: subscriptions@bsigroup.com

Knowledge Centre

Tel: +44 20 8996 7004

Email: knowledgecentre@bsigroup.com

Copyright & Licensing

Tel: +44 20 8996 7070 Email: copyright@bsigroup.com

