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Calculation report

EN 13445 Ed. 2014 Issue 3

Project: Diplomová práce
Item: Heat exchanger water-natural gas
Customer:
Drawing:
Revision:
Date: 20.05.2019

	Tube side	Shell side
Internal design pressure	10.69 MPa	10.69 MPa
Internal design temperature	37.78 °C	121.11 °C
Corrosion allowance	3.20 mm	3.20 mm
Vacuum?	No	No
	Both sides	
Minimum design temperature	-20.00 °C	
TEMA Class:	TEMA compliant R	



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Test Pressure - Tube side (MPa)

Component	P	Static head (design)	Static head (test)	Stress ratio	1.25·P·f₀/f	1.43·P
Torosferické dno komory	10.69	0	0.03	1	13.36	15.29
Komora	10.69	0	0.03	1	13.36	15.29
Komorová příruba	10.69	0	0.03	1,027	13.72	15.29
Trubkovnice	10.69	0	0.03	1,047	13.98	15.29
Svazek trubek	10.69	0	0.02	1,06	14.17	15.29

All pressures in MPa.

Tube side design pressure P = 10.69 MPa

Tube side MAWP (Hot & Corroded conditions) = 11.29 MPa (limited by Komorová příruba)

Tube side lowest stress ratio = 1.000 (limited by Torosferické dno komory)

Tube side test pressure = $P_t = \max[1.25 \cdot P_d \cdot (\text{Item } f_0/f); 1.43 \cdot P_d] = 15.29 \text{ MPa}$ **Maximum Pressures - Tube side (MPa)**

Component	Internal, test	Internal	External, test	External
Torosferické dno komory	27.85	16.71		
Komora	23.84	14.28		
Komorová příruba	11.89	11.29		
Trubkovnice	19.25	16.67		
Svazek trubek	38.38	24.04	31.89	21.18

All pressures in MPa.

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Test Pressure - Shell side (MPa)

Component	P	Static head (design)	Static head (test)	Stress ratio	1.25·P·f ₀ /f	1.43·P
Plášťová příruba	10.69	0	0.03	1,16	15.50	15.29
Plášť	10.69	0	0.03	1,047	13.99	15.29
Trubkovnice	10.69	0	0.03	1,047	13.98	15.29
Svazek trubek	10.69	0	0.005	1,06	14.17	15.29
Torosferické dno pláště	10.69	0	0.004	1,047	13.99	15.29

All pressures in MPa.

Shell side design pressure P = 10.69 MPa

Shell side MAWP (Hot & Corroded conditions) = 10.97 MPa (limited by Plášťová příruba)

Shell side lowest stress ratio = 1.047 (limited by Trubkovnice)

Shell side test pressure = $P_t = \max[1.25 \cdot P_d \cdot (\text{Item } f_0/f); 1.43 \cdot P_d] = 15.29 \text{ MPa}$

Maximum Pressures - Shell side (MPa)

Component	Internal, test	Internal
Plášťová příruba	11.88	10.97
Plášť	23.84	13.64
Trubkovnice	19.25	16.67
Svazek trubek	31.89	21.18
Torosferické dno pláště	27.87	15.97

All pressures in MPa.

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Weights

Component	Dead	Live	Liquid	Full of water	Operating
Trubka-výpusť komory 1	1 kg	0 kg	0 kg	1 kg	1 kg
Torosferické dno komory	141 kg	0 kg	0 kg	198 kg	141 kg
Příruba-výpusť komory 1	3 kg	0 kg	0 kg	3 kg	3 kg
Trubka-výpusť komory 2	1 kg	0 kg	0 kg	1 kg	1 kg
Příruba-výpusť komory 2	3 kg	0 kg	0 kg	3 kg	3 kg
Komora	187 kg	0 kg	0 kg	350 kg	187 kg
Trubka- vstup plynu	77 kg	0 kg	0 kg	97 kg	77 kg
Příruba-vstup plynu	126 kg	0 kg	0 kg	137 kg	126 kg
Trubka- výstup plynu	77 kg	0 kg	0 kg	97 kg	77 kg
Příruba-výstup plynu	126 kg	0 kg	0 kg	137 kg	126 kg
Komorová příruba	420 kg	0 kg	0 kg	471 kg	420 kg
Plášťová příruba	420 kg	0 kg	0 kg	471 kg	420 kg
Plášť	1 070 kg	0 kg	0 kg	1 172 kg	1 070 kg
Trubka-vstup vody	9 kg	0 kg	0 kg	10 kg	9 kg
Příruba-vstup plynu	10 kg	0 kg	0 kg	10 kg	10 kg
Trubka-výstup vody	9 kg	0 kg	0 kg	10 kg	9 kg
Příruba-výstup plynu	10 kg	0 kg	0 kg	10 kg	10 kg
Trubkovnice	422 kg	0 kg	0 kg	422 kg	422 kg
Svazek trubek	1 261 kg	0 kg	0 kg	1 723 kg	1 261 kg
Torosferické dno pláště	141 kg	0 kg	0 kg	198 kg	141 kg
Trubka-odvzdušnění	1 kg	0 kg	0 kg	1 kg	1 kg
Příruba-odvzdušnění	5 kg	0 kg	0 kg	5 kg	5 kg
Totals:	4 590 kg	0 kg	0 kg	5 600 kg	4 590 kg

Total shell side volume: 0.21410 m³

Total tube side volume: 0.79594 m³

Total volume: 1.01005 m³

Center of gravity (erection): Cx=0 mm, Cy=0 mm, Cz=2 039.40 mm, W = 4 590 kg

Center of gravity (operating): Cx=0 mm, Cy=0 mm, Cz=2 039.40 mm, W = 4 590 kg

Center of gravity (test): Cx=0 mm, Cy=0 mm, Cz=2 035.22 mm, W = 5 600 kg

Definitions

Dead: net, uncorroded weight of component, including additional dead weight

Live: additional live weight on component

Liquid: weight of liquid contained in component in operating conditions (depending on liquid level)

Insulation: weight of insulation on component, when present

Full of water: weight of each component in test condition

Operating: weight of each component in operating condition

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Bill of materials

Component	Dimensions	Material
Trubka-výpusť komory 1	Id = 21.80 mm, Od = 31.80 mm, Tk = 5.00 mm, L = 120.00 mm	10CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7380
Torosferické dno komory	Id = 620.00 mm, Od = 676.00 mm, Crown = 540.80 mm, Knuckle = 104.10 mm, Tk = 28.00 mm	13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335
Příruba-výpusť komory 1 - Flange	Id = 21.80 mm, Od = 140.00 mm, Tk = 22.00 mm	13CrMo4-5 (NT) (EN 10222-2) - Forging (35,001 ≤ t ≤ 60) - No.: 1.7335
Příruba-výpusť komory 1 - Gasket	Rubber with cotton fabric insertion	
Příruba-výpusť komory 1 - Bolts	4 x ISO M16 x 2.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Trubka-výpusť komory 2	Id = 21.80 mm, Od = 31.80 mm, Tk = 5.00 mm, L = 120.00 mm	10CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7380
Příruba-výpusť komory 2 - Flange	Id = 21.80 mm, Od = 140.00 mm, Tk = 22.00 mm	13CrMo4-5 (NT) (EN 10222-2) - Forging (35,001 ≤ t ≤ 60) - No.: 1.7335
Příruba-výpusť komory 2 - Gasket	Rubber with cotton fabric insertion	
Příruba-výpusť komory 2 - Bolts	4 x ISO M16 x 2.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Komora	Id = 620.00 mm, Od = 676.00 mm, Tk = 28.00 mm, L = 540.00 mm	13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335
Trubka- vstup plynu	Id = 291.60 mm, Od = 355.60 mm, Tk = 32.00 mm, L = 300.00 mm	11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383
Příruba-vstup plynu - Flange	Id = 291.60 mm, Od = 585.00 mm, Tk = 74.00 mm	13CrMo4-5 (NT,QT) (EN 10222-2) - Forging (100,001 ≤ t ≤ 150) - No.: 1.7335
Příruba-vstup plynu - Gasket	Rubber with cotton fabric insertion	
Příruba-vstup plynu - Bolts	16 x ISO M39 x 4.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Trubka- výstup plynu	Id = 291.60 mm, Od = 355.60 mm, Tk = 32.00 mm, L = 300.00 mm	11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383
Příruba-výstup plynu - Flange	Id = 291.60 mm, Od = 585.00 mm, Tk = 74.00 mm	13CrMo4-5 (NT,QT) (EN 10222-2) - Forging (100,001 ≤ t ≤ 150) - No.: 1.7335
Příruba-výstup plynu - Gasket	Rubber with cotton fabric insertion	
Příruba-výstup plynu - Bolts	16 x ISO M39 x 4.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Komorová příruba - Flange	Id = 620.00 mm, Od = 1 018.55 mm, Tk = 102.00 mm	11CrMo9-10 (NT) (EN 10222-2) - Forging (t ≤ 200.00 mm) - No.: 1.7383
Komorová příruba - Gasket	Rubber with cotton fabric insertion	
Komorová příruba - Bolts	28 x ISO_TEMA M42 x 4.50	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Plášťová příruba - Flange	Id = 620.00 mm, Od = 1 018.55 mm, Tk = 102.00 mm	11CrMo9-10 (NT) (EN 10222-2) - Forging (t ≤ 200.00 mm) - No.: 1.7383
Plášťová příruba - Gasket	Rubber with cotton fabric insertion	
Plášť	Id = 620.00 mm, Od = 676.00 mm, Tk = 28.00 mm, L = 2 400.00 mm	13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335
Trubka-vstup vody	Id = 76.66 mm, Od = 101.66 mm, Tk = 12.50 mm, L = 300.00 mm	11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383
Příruba-vstup plynu - Flange	Id = 76.66 mm, Od = 230.00 mm, Tk = 33.00 mm	13CrMo4-5 (NT,QT) (EN 10222-2) - Forging (70 ≤ t ≤ 90) - No.: 1.7335
Příruba-vstup plynu - Gasket	Rubber with cotton fabric insertion	

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Příruba-vstup plynu - Bolts	8 x ISO M24 x 3.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Trubka-výstup vody	Id = 76.66 mm, Od = 101.66 mm, Tk = 12.50 mm, L = 300.00 mm	11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383
Příruba-výstup plynu - Flange	Id = 76.66 mm, Od = 230.00 mm, Tk = 33.00 mm	13CrMo4-5 (NT,QT) (EN 10222-2) - Forging (70 ≤ t ≤ 90) - No.: 1.7335
Příruba-výstup plynu - Gasket	Rubber with cotton fabric insertion	
Příruba-výstup plynu - Bolts	8 x ISO M24 x 3.00	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Trubkovnice - Flange	Od = 860.00 mm, Tk = 133.35 mm	13CrMo4-5 (NT,QT) (EN 10222-2) - Forging (100,001 ≤ t ≤ 150) - No.: 1.7335
Svazek trubek	Id = 34.80 mm, Od = 40.00 mm, Tk = 2.60 mm, L = 2 380.00 mm	13CrMo4-5 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7335
Torosferické dno pláště	Id = 620.00 mm, Od = 676.00 mm, Crown = 540.80 mm, Knuckle = 104.10 mm, Tk = 28.00 mm	13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335
Trubka-odvzdušnění	Id = 41.00 mm, Od = 51.00 mm, Tk = 5.00 mm, L = 150.00 mm	11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383
Příruba-odvzdušnění - Flange	Id = 41.00 mm, Od = 170.00 mm, Tk = 25.00 mm	13CrMo4-5 (NT) (EN 10222-2) - Forging (35,001 ≤ t ≤ 60) - No.: 1.7335
Příruba-odvzdušnění - Gasket	Rubber with cotton fabric insertion	
Příruba-odvzdušnění - Bolts	4 x ISO M20 x 2.50	25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218

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Material properties summary

10CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 40.00$ mm) - No.: 1.7380

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	186.67 MPa	280.00 MPa	480.00 MPa	211 092.00 MPa	0.000011500 1/°C
Design	182.07 MPa	273.11 MPa	480.00 MPa	209 718.83 MPa	0.000011646 1/°C
Test	266.67 MPa				

11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	225.00 MPa	355.00 MPa	540.00 MPa	211 092.00 MPa	0.000011500 1/°C
Design	225.00 MPa	347.89 MPa	0 MPa	209 718.83 MPa	0.000011646 1/°C
Test	225.00 MPa				

11CrMo9-10 (NT) (EN 10222-2) - Forging ($t \leq 200.00$ mm) - No.: 1.7383

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	200.00 MPa	300.00 MPa	520.00 MPa	211 092.00 MPa	0.000011500 1/°C
Design	194.81 MPa	292.22 MPa	0 MPa	209 718.83 MPa	0.000011646 1/°C
Test	285.71 MPa				

13CrMo4-5 (EN 10028-2) - Plate ($16,001 \leq t \leq 60$) - No.: 1.7335

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	187.50 MPa	290.00 MPa	450.00 MPa	204 842.00 MPa	0.000011500 1/°C
Design	187.50 MPa	287.04 MPa	450.00 MPa	203 548.34 MPa	0.000011646 1/°C
Test	276.19 MPa				

13CrMo4-5 (EN 10216-2) - Seamless tube ($t \leq 40.00$ mm) - No.: 1.7335

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	183.33 MPa	290.00 MPa	440.00 MPa	204 842.00 MPa	0.000011500 1/°C
Design	172.90 MPa	259.36 MPa	440.00 MPa	198 193.42 MPa	0.000012242 1/°C
Test	276.19 MPa				

13CrMo4-5 (NT) (EN 10222-2) - Forging ($35,001 \leq t \leq 60$) - No.: 1.7335

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	183.33 MPa	275.00 MPa	440.00 MPa	204 842.00 MPa	0.000011500 1/°C
Design	181.11 MPa	271.67 MPa	0 MPa	203 548.34 MPa	0.000011646 1/°C
Test	261.90 MPa				

13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($100,001 \leq t \leq 150$) - No.: 1.7335

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	170.00 MPa	255.00 MPa	440.00 MPa	204 842.00 MPa	0.000011500 1/°C
Design	169.26 MPa	253.89 MPa	0 MPa	203 548.34 MPa	0.000011646 1/°C
Test	242.86 MPa				

13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($70 \leq t \leq 90$) - No.: 1.7335

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	176.67 MPa	265.00 MPa	440.00 MPa	204 842.00 MPa	0.000011500 1/°C

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Design	163.85 MPa	245.78 MPa	0 MPa	198 193.42 MPa	0.000012242 1/°C
Test	252.38 MPa				

25CrMo4 (EN 10269) - Bolting (t ≤ 100.00 mm) - No.: 1.7218

Temp.	Allowable (1)	Yield strength (2)	Tensile strength	Elasticity	Thermal expansion
Room	250.00 MPa	440.00 MPa	600.00 MPa	211 000.00 MPa	0.000011100 1/°C
Design	250.00 MPa	437.04 MPa	0 MPa	209 444.04 MPa	0.000011100 1/°C
Test	419.05 MPa				

Notes
(1) Allowable stress calculation may vary upon component type, conditions and other factors. Refer to each component's calculation page for its allowable stress value
(2) Yield strength shown refers to 0.2% plastic strain

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Nozzle connections

Name	Flange	Material	OD	Tk
Trubka-výpusť komory 1	25 WN 160 EN1092_1	10CrMo9-10 (EN 10216-2)	31.80 mm	5.00 mm
Trubka-výpusť komory 2	25 WN 160 EN1092_1	10CrMo9-10 (EN 10216-2)	31.80 mm	5.00 mm
Trubka- vstup plynu	300 WN 160 EN1092_1	11CrMo9-10 (EN 10216-2)	355.60 mm	32.00 mm
Trubka- výstup plynu	300 WN 160 EN1092_1	11CrMo9-10 (EN 10216-2)	355.60 mm	32.00 mm
Trubka-vstup vody	80 WN 160 EN1092_1	11CrMo9-10 (EN 10216-2)	101.66 mm	12.50 mm
Trubka-výstup vody	80 WN 160 EN1092_1	11CrMo9-10 (EN 10216-2)	101.66 mm	12.50 mm
Trubka-odvzdušnění	40 WN 160 EN1092_1	11CrMo9-10 (EN 10216-2)	51.00 mm	5.00 mm

Nozzle positions

Name	Placed on	Type	Distance from reference	Orientation	Notes
Trubka-výpusť komory 1	Torosferické dno komory	Off center/ Set in	120.00 mm	0 °	
Trubka-výpusť komory 2	Torosferické dno komory	Off center/ Set in	120.00 mm	180.00 °	
Trubka- vstup plynu	Komora	Radial/ Set in	270.00 mm	0 °	
Trubka- výstup plynu	Komora	Radial/ Set in	270.00 mm	180.00 °	
Trubka-vstup vody	Plášť	Radial/ Set in	2 255.00 mm	180.00 °	
Trubka-výstup vody	Plášť	Radial/ Set in	145.00 mm	0 °	
Trubka-odvzdušnění	Torosferické dno pláště	Radial/ Set in	0 mm	0 °	

Nozzle welds

Name	Nozzle to wall	Pad to wall	Shell groove	Pad groove	Inside
Trubka-výpusť komory 1					
Trubka-výpusť komory 2					
Trubka- vstup plynu					
Trubka- výstup plynu					
Trubka-vstup vody					
Trubka-výstup vody					
Trubka-odvzdušnění					

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Nozzle - Trubka-výpust' komory 1*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Access or inspection opening	=	No

Material: 10CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7380

Nominal design stress at internal design temperature	$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_m/20}{2.4}\right) =$	182.07 MPa
Nominal design stress at room temperature	$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_m/20}{2.4}\right) =$	186.67 MPa
Nominal design stress in test condition	$f_{test} = \left(\frac{R_{p0.2/T_{test}}}{1.05}\right) =$	266.67 MPa

Geometry

Inside diameter	Di =	21.80 mm
Outside diameter	De =	31.80 mm
Length	L =	120.00 mm
Nominal thickness	en =	5.00 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	28.20 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	4.05 mm
e/De ≤ 0,16 (0.12700 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	26.00 MPa
Maximum allowable design pressure	=	15.14 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	$F = 50 \cdot en / (Di/2 + en/2) =$	18.657 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.32 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	21.80 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	0.64 mm
e/De ≤ 0,16 (0.02000 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Torispherical head - Torosferické dno komory*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature

Ti = 37.78 °C

Internal design pressure

Pi = 10.69 MPa

Joint efficiency

z = 1.00

Material: 13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335

Nominal design stress at internal design temperature

$$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_{m/20}}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress at room temperature

$$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_{m/20}}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress in test condition

$$f_{\text{test}} = \left(\frac{R_{p0.2/T_{\text{test}}}}{1.05}\right) = 276.19 \text{ MPa}$$

Geometry

Inside diameter

Di = 620.00 mm

Outside diameter

De = 676.00 mm

Head outside height

H = 267.69 mm

Nominal thickness

en = 28.00 mm

Minimum head thickness after forming

t-c' = 28.00 mm

Corrosion allowance

c = 3.20 mm

External corrosion allowance

ce = 0 mm

Undertolerance

δ = 0 mm

Straight flange length

l(sf) = 84.00 mm

Straight flange undertolerance

δ(sf) = 0 mm

Straight flange thickness

en(sf) = 28.00 mm

Straight flange joint efficiency

z(sf) = 1.00000

Knuckle thickness

en(k) = 28.00 mm

Inside spherical radius of central part of torispherical head

R = 540.80 mm

Inside knuckle radius

r = 104.10 mm

Internal pressure

Overpressure due to static head

Ph = 0 MPa

Calculation pressure

P=Pi+Ph = 10.69 MPa

Parameter Y

Y=min(ec/R;0.04) = 0.02987

Parameter Z

Z=log10(1/Y) = 1.52483

Ratio X

X=r/Di = 0.17130

Parameter N

$$N = 1006 - \frac{1}{[62 + (90Y)^4]} = 0.98888$$

Parameter β(0.1)

$$\beta_{0.1} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837) = 0.62072$$

Parameter β(0.2)

$$\beta_{0.2} = \max[0.95(0.56 - 194Y - 82.5Y^2), 0.5] = 0.50000$$

Parameter β

$$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}] = 0.53464$$

Joint efficiency

z = 1.00000

Inside spherical radius of central part of torispherical head

R'=R+c = 544.00 mm

Inside diameter

Di'=Di+2·c = 626.40 mm

Inside knuckle radius

r'=r+c = 107.30 mm

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Required thickness of end to limit membrane stress in central part

$$e_s = \frac{PR'}{2fZ - 0.5P} + c + ce + \delta = 18.93 \text{ mm}$$

Required thickness of knuckle to avoid axisymmetric yielding

$$e_y = \frac{\beta P(0.75R + 0.2D_i)}{f} + c + ce + \delta = 19.46 \text{ mm}$$

Minimum required thickness

$$e = \max(e_y, e_s) = 19.46 \text{ mm}$$

Straight flange minimum required thickness

$$e(sf) = 21.58 \text{ mm}$$

en(sf) ≥ e(sf): Ok**en ≥ e: Ok****Maximum allowable pressures (at the top of the vessel)**

Maximum allowable test pressure

$$= 27.85 \text{ MPa}$$

Maximum allowable design pressure

$$= 16.71 \text{ MPa}$$

Deformation according to EN13445-4 Clause 9

Manufactured in one piece (9.2.1)

$$F(1) = 100 \cdot \ln[(1.11 \cdot De)/(De - 2 \cdot en)] = 19.083 \%$$

Spherical part (9.2.1)

$$F(2) = 100 \cdot \ln\{2 \cdot R \cdot \sin[(0.4 \cdot De/R)/(0.8 \cdot De - 2 \cdot en)]\} = 15.543 \%$$

Segments deformation of multi-sectional torispherical heads or spheres (9.2.5)

$$F(3) = (100 \cdot en)/(r + en/2) = 23.708 \%$$

Hydrostatic test

Item or side minimum allowables ratio

$$\text{Item } f_0/f = 1.00000$$

Coincident design pressure for the maximum pressure load case

$$P_d = 10.69 \text{ MPa}$$

Test pressure as per EN13445-5 formula 10.2.3.3.1-1

$$P_{t1} = 1.25 \cdot P_d \cdot (\text{Item } f_0/f) = 13.36 \text{ MPa}$$

Test pressure as per EN13445-5 formula 10.2.3.3.1-2

$$P_{t2} = 1.43 \cdot P_d = 15.29 \text{ MPa}$$

Item or side hydrostatic test pressure

$$P_t = \max(P_{t1}, P_{t2}) = 15.29 \text{ MPa}$$

Overpressure due to static head in test condition

$$P_{ht} = 0.03 \text{ MPa}$$

Calculation pressure

$$P_c = P_t + P_{ht} = 15.32 \text{ MPa}$$

Joint efficiency

$$Z = 1.00000$$

Parameter Y

$$Y = \min(ec/R; 0.04) = 0.02933$$

Parameter Z

$$Z = \log_{10}(1/Y) = 1.53263$$

Ratio X

$$X = r/D_i = 0.16791$$

Parameter N

$$N = 1006 - \frac{1}{[6.2 + (90Y)^4]} = 0.98775$$

Parameter β_{01}

$$\beta_{01} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837) = 0.62459$$

Parameter β_{02}

$$\beta_{02} = \max[0.95(0.56 - 1.94Y - 82.5Y^2), 0.5] = 0.50000$$

Parameter β

$$\beta = 10[(0.2 - X)\beta_{01} + (X - 0.1)\beta_{02}] = 0.53998$$

Inside spherical radius of central part of torispherical head

$$R' = R = 540.80 \text{ mm}$$

Inside diameter

$$D_i' = D_i = 620.00 \text{ mm}$$

Inside knuckle radius

$$r' = r = 104.10 \text{ mm}$$

Required thickness of end to limit membrane stress in central part

$$e_s = \frac{PR'}{2fZ - 0.5P} + \delta = 15.21 \text{ mm}$$

Required thickness of knuckle to avoid axisymmetric yielding

$$e_y = \frac{\beta P(0.75R + 0.2D_i)}{f} + \delta = 15.86 \text{ mm}$$

Minimum required thickness

$$e = \max(e_y, e_s) = 15.86 \text{ mm}$$

en ≥ e: Ok

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Reinforcement of opening - Trubka-výpust' komory 1*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	187.50 MPa
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Nozzle material: 10CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	182.07 MPa
	fob = min(fs, fb) =	182.07 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset k between nozzle and shell axis	=	120.00 mm
Angular offset	=	0 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	31.80 mm
Internal diameter	d_ib = Id + 2(c + δ) =	28.20 mm
External diameter of the nozzle	d_eb = Od - 2ce =	31.80 mm
Nominal thickness of the nozzle	e_ab =	5.00 mm
Length of nozzle extending outside the shell	l_b =	92.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	1.80 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	7.35 mm
	$l_{bo} = \min(l_b; l_{bo(max)})$ =	7.35 mm
	$l'_b = \min(l_{bo}; l_b)$ =	7.35 mm
	$l'_{bi} = \min(l_{bi}; 0.5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement	Afw =	0 mm²
Effective length of nozzle inside the shell for reinforcement	$\delta^* = \frac{d_{eb}}{2r_{ms}}$ =	0.02858
Stress loaded cross-sectional area effective as reinforcement - welds	=	12.74 °
Obliquity angle in the longitudinal or transversal cross-section	=	≤ arcsin(1-δ*): Ok

Pad geometry

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

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Head inside radius	R =	540.80 mm
Inside radius of curvature of the shell at the opening centre	r_is = R + cs + cs' =	544.00 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

Head's analysis thickness

$$e_k = 24.80 \text{ mm}$$

Head's knuckle radius

$$r_k = 107.30 \text{ mm}$$

Distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w(7.7.2) = 281.07 \text{ mm}$$

Limit distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w_{\min}(7.7.2) = 2.5 \cdot \sqrt{(e_k \cdot r_k)} = 128.97 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 145.08 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 145.08 \text{ mm}$$

Minimum value for w which has no influence on l_s from shell discontinuities

$$w_p = l_{so} = 166.12 \text{ mm}$$

Required minimum value for w

$$w_{\min} = 0 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Transverse section

Distance taken along the mid-thickness of the shell between the centre of the opening and the external edge of the nozzle

$$\delta = \frac{d_{eb}}{2r_{ms}} = 0.02858$$

$$a = 0.5r_{ms} [\arcsin(\delta + \sin(\varphi)) + \arcsin(\delta - \sin(\varphi))] = 16.30 \text{ mm}$$

Mean radius of curvature

$$r_{ms} = r_{is} + 0.5e_{as} = 556.40 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l'_s = \min[l_{so}, l_s] = 145.08 \text{ mm}$$

Pressure loaded area - shell

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 42\,917.2 \text{ mm}^2$$

Length of penetration into shell wall

$$e's = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{f_b} = e_b \cdot (l'_b + l'_{bi} + e'_s) = 57.9 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 3\,597.9 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{p_b} = 0.5d_{ib} \cdot (l'_b + e_{as}) = 453.3 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p\varphi} = 89.9 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{fp} = 0 \text{ mm}^2$$

Reactive force

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{f_p}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 665\,600 \text{ N}$$

Pressure load

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) = 464\,111 \text{ N}$$

Maximum allowable pressure

$$P_{\max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{f_p} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{f_p})} = 15.14 \text{ MPa}$$

Fr ≥ Fp: Ok**Adjacent openings**

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Trubka-výpust' komory 1 - Trubka-výpust' komory 2

Centre-to-centre distance taken on the mean surface of the shell

$$L_b = 241.91 \text{ mm}$$

Ligament length

$$L = 209.30 \text{ mm}$$

Mean shell radius at the centres of adjacent nozzles

$$r_{is} = 544.00 \text{ mm}$$

Minimum required ligament length

$$L_{min} = \max \left[3e_{as}, 0.2\sqrt{(2r_{is} + e_{cs})e_{cs}} \right] = 74.40 \text{ mm}$$

$$d1 = 31.80 \text{ mm}$$

$$d2 = 31.80 \text{ mm}$$

L ≥ Lmin: Ok**d1+d2 ≤ 0.2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required****Ligament check**

Inside radius of curvature of the shell at the opening centre

$$r_{is} = \frac{D_e}{2} - e_{as} = 544.00 \text{ mm}$$

Angle between the centre-to-centre line of openings and the generatrix of the shell

$$\phi = 90.00^\circ$$

Ap of the shell for the length Lb

$$Ap_{Ls} = \frac{0.5r_{is}^2 \cdot L_b (1 + \cos(\phi))}{r_{is} + 0.5e_{as} \sin(\phi)} = 64\,333.6 \text{ mm}^2$$

$$deb1 = 31.80 \text{ mm}$$

$$deb2 = 31.80 \text{ mm}$$

$$_e1 = 0^\circ$$

$$_e2 = 0^\circ$$

$$r_{os1} = \frac{r_{is1}}{\sin^2(\phi)} + 0.5e_{as} = 556.40 \text{ mm}$$

$$r_{os2} = \frac{r_{is2}}{\sin^2(\phi)} + 0.5e_{as} = 556.40 \text{ mm}$$

$$\delta_1 = \frac{d_{eb1}}{2r_{os1}} = 0.02858$$

$$\delta_2 = \frac{d_{eb2}}{2r_{os2}} = 0.02858$$

$$a_1 = r_{os1} [\arcsin(\delta_1 + \sin(\phi_{e1})) - \phi_{e1}] = 15.90 \text{ mm}$$

$$a_2 = r_{os2} [\arcsin(\delta_2 + \sin(\phi_{e2})) - \phi_{e2}] = 15.90 \text{ mm}$$

Af of the shell contained along the length Lb

$$Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{cs} = 5\,210.7 \text{ mm}^2$$

$$Ap\phi1 = 89.9 \text{ mm}^2$$

$$Ap\phi2 = 89.9 \text{ mm}^2$$

$$Apb1 = 453.3 \text{ mm}^2$$

$$Apb2 = 453.3 \text{ mm}^2$$

$$Afb1 = 57.9 \text{ mm}^2$$

$$Afb2 = 57.9 \text{ mm}^2$$

$$Afp1 = 0 \text{ mm}^2$$

$$Afp2 = 0 \text{ mm}^2$$

$$fob1 = 182.07 \text{ MPa}$$

$$fob2 = 182.07 \text{ MPa}$$

$$fop1 = 0 \text{ MPa}$$

$$fop2 = 0 \text{ MPa}$$

Reactive force

$$F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{b1}(f_{ob1} - 0.5P) + Af_{p1}(f_{op1} - 0.5P) +$$

$$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op2} - 0.5P) = 969\,603 \text{ N}$$

Pressure load

$$F_{req} = P(Ap_{Ls} + Ap_{b1} + 0.5Ap_{\phi1} + Ap_{b2} + 0.5Ap_{\phi2}) = 698\,379 \text{ N}$$

F ≥ Freq: Ok

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Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Calculation pressure	P = Pt + Ph =	15.32 MPa

Transverse section

	$\delta = \frac{d_{eb}}{2r_{ms}}$	=	0.02866
Distance taken along the mid-thickness of the shell between the centre of the opening and the external edge of the nozzle	$a = 0,5r_{ms}[\arcsin(\delta + \sin(\varphi)) + \arcsin(\delta - \sin(\varphi))]$	=	16.31 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	554.80 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	176.26 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	145.15 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	42 555.8 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l'_b + l'_{bi} + e'_s)$	=	197.9 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	4 064.1 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib} \cdot (l'_b + e_{as})$	=	431.4 mm ²
Additional area due to obliquity of the nozzle	$Ap\varphi$	=	54.1 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 142 582 N
Pressure load	$Fp = P(Ap_s + Ap_b + 0.5Ap\varphi)$	=	659 008 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	26.03 MPa

Fr ≥ Fp: Ok**Adjacent openings****Trubka-výpust' komory 1 - Trubka-výpust' komory 2**

Centre-to-centre distance taken on the mean surface of the shell	Lb =	241.91 mm
Ligament length	L =	209.30 mm
Mean shell radius at the centres of adjacent nozzles	r_is =	540.80 mm
Minimum required ligament length	$L_{\min} = \max \left[3e_{as}; 0.2\sqrt{(2r_{is} + e_{cs})e_{cs}} \right]$	= 84.00 mm
	d1 =	31.80 mm
	d2 =	31.80 mm

L ≥ Lmin: Ok**d1+d2 ≤ 0.2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required**

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Ligament check

Inside radius of curvature of the shell at the opening centre

$$r_{is} = \frac{D_e}{2} - e_{as} = 540.80 \text{ mm}$$

Angle between the centre-to-centre line of openings and the generatrix of the shell

$$\phi = 90.00^\circ$$

Ap of the shell for the length Lb

$$Ap_{Ls} = \frac{0.5r_{is}^2 \cdot L_b (1 + \cos(\phi))}{r_{is} + 0.5e_{as} \cdot \sin(\phi)} = 63\,762.3 \text{ mm}^2$$

$$deb1 = 31.80 \text{ mm}$$

$$deb2 = 31.80 \text{ mm}$$

$$_e1 = 0^\circ$$

$$_e2 = 0^\circ$$

$$r_{os1} = \frac{r_{is1}}{\sin^2(\phi)} + 0.5e_{as} = 554.80 \text{ mm}$$

$$r_{os2} = \frac{r_{is2}}{\sin^2(\phi)} + 0.5e_{as} = 554.80 \text{ mm}$$

$$\delta_1 = \frac{d_{eb1}}{2r_{os1}} = 0.02866$$

$$\delta_2 = \frac{d_{eb2}}{2r_{os2}} = 0.02866$$

$$a_1 = r_{os1} [\arcsin(\delta_1 + \sin(\phi_{e1})) - \phi_{e1}] = 15.90 \text{ mm}$$

$$a_2 = r_{os2} [\arcsin(\delta_2 + \sin(\phi_{e2})) - \phi_{e2}] = 15.90 \text{ mm}$$

Af of the shell contained along the length Lb

$$Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{cs} = 5\,883.0 \text{ mm}^2$$

$$Ap\phi1 = 54.1 \text{ mm}^2$$

$$Ap\phi2 = 54.1 \text{ mm}^2$$

$$Apb1 = 431.4 \text{ mm}^2$$

$$Apb2 = 431.4 \text{ mm}^2$$

$$Afb1 = 197.9 \text{ mm}^2$$

$$Afb2 = 197.9 \text{ mm}^2$$

$$Afp1 = 0 \text{ mm}^2$$

$$Afp2 = 0 \text{ mm}^2$$

$$fob1 = 266.67 \text{ MPa}$$

$$fob2 = 266.67 \text{ MPa}$$

$$fop1 = 0 \text{ MPa}$$

$$fop2 = 0 \text{ MPa}$$

Reactive force

$$F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{b1}(f_{ob1} - 0.5P) + Af_{p1}(f_{op1} - 0.5P) +$$

$$+ Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op2} - 0.5P) = 1\,682\,269 \text{ N}$$

Pressure load

$$F_{req} = P(Ap_{Ls} + Ap_{b1} + 0.5Ap_{\phi1} + Ap_{b2} + 0.5Ap_{\phi2}) = 990\,930 \text{ N}$$

F ≥ Freq: Ok

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Reinforcement of opening - Trubka-výpust' komory 2*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	187.50 MPa
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Nozzle material: 10CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	182.07 MPa
	fob = min(fs, fb) =	182.07 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Hillside / Axial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset k between nozzle and shell axis	=	120.00 mm
Angular offset	=	180.00 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	31.80 mm
Internal diameter	d_ib = Id + 2(c + δ) =	28.20 mm
External diameter of the nozzle	d_eb = Od - 2ce =	31.80 mm
Nominal thickness of the nozzle	e_ab =	5.00 mm
Length of nozzle extending outside the shell	l_b =	92.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	1.80 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	7.35 mm
	$l_{bo} = \min(l_b, l_{bo(max)})$ =	7.35 mm
	$l'_b = \min(l_{bo}, l_b)$ =	7.35 mm
	$l'_{bi} = \min(l_{bi}, 0.5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement	Afw =	0 mm²
Effective length of nozzle inside the shell for reinforcement	$\delta^* = \frac{d_{eb}}{2r_{ms}}$ =	0.02858
Stress loaded cross-sectional area effective as reinforcement - welds	=	12.74 °
Obliquity angle in the longitudinal or transversal cross-section	=	≤ arcsin(1-δ*): Ok

Pad geometry

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

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Head inside radius	R =	540.80 mm
Inside radius of curvature of the shell at the opening centre	r_is = R + cs + cs' =	544.00 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

Head's analysis thickness

$$e_k = 24.80 \text{ mm}$$

Head's knuckle radius

$$r_k = 107.30 \text{ mm}$$

Distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w(7.7.2) = 281.07 \text{ mm}$$

Limit distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w_{\min}(7.7.2) = 2.5 \cdot \sqrt{(e_k \cdot r_k)} = 128.97 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 145.08 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 145.08 \text{ mm}$$

Minimum value for w which has no influence on ls from shell discontinuities

$$w_p = l_{so} = 166.12 \text{ mm}$$

Required minimum value for w

$$w_{\min} = 0 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Transverse section

Distance taken along the mid-thickness of the shell between the centre of the opening and the external edge of the nozzle

$$\delta = \frac{d_{eb}}{2r_{ms}} = 0.02858$$

$$a = 0.5r_{ms} [\arcsin(\delta + \sin(\varphi)) + \arcsin(\delta - \sin(\varphi))] = 16.30 \text{ mm}$$

Mean radius of curvature

$$r_{ms} = r_{is} + 0.5e_{as} = 556.40 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l'_s = \min[l_s, l_{so}] = 145.08 \text{ mm}$$

Pressure loaded area - shell

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 42\,917.2 \text{ mm}^2$$

Length of penetration into shell wall

$$e's = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{f_b} = e_b \cdot (l'_b + l'_{bi} + e'_s) = 57.9 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 3\,597.9 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{p_b} = 0.5d_{ib} \cdot (l'_b + e_{as}) = 453.3 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p\varphi} = 89.9 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{fp} = 0 \text{ mm}^2$$

Reactive force

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{f_p}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 665\,600 \text{ N}$$

Pressure load

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) = 464\,111 \text{ N}$$

Maximum allowable pressure

$$P_{\max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{f_p} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{f_p})} = 15.14 \text{ MPa}$$

Fr ≥ Fp: Ok**Adjacent openings**

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Trubka-výpust' komory 2 - Trubka-výpust' komory 1

Centre-to-centre distance taken on the mean surface of the shell

$$L_b = 241.91 \text{ mm}$$

Ligament length

$$L = 209.30 \text{ mm}$$

Mean shell radius at the centres of adjacent nozzles

$$r_{is} = 544.00 \text{ mm}$$

Minimum required ligament length

$$L_{min} = \max \left[3e_{as}, 0.2\sqrt{(2r_{is} + e_{cs})e_{cs}} \right] = 74.40 \text{ mm}$$

$$d1 = 31.80 \text{ mm}$$

$$d2 = 31.80 \text{ mm}$$

L ≥ Lmin: Ok**d1+d2 ≤ 0.2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required****Ligament check**

Inside radius of curvature of the shell at the opening centre

$$r_{is} = \frac{D_e}{2} - e_{as} = 544.00 \text{ mm}$$

Angle between the centre-to-centre line of openings and the generatrix of the shell

$$\phi = 90.00^\circ$$

Ap of the shell for the length Lb

$$A_{p_{Ls}} = \frac{0.5r_{is}^2 \cdot L_b (1 + \cos(\phi))}{r_{is} + 0.5e_{as} \sin(\phi)} = 64\,333.6 \text{ mm}^2$$

$$deb1 = 31.80 \text{ mm}$$

$$deb2 = 31.80 \text{ mm}$$

$$_e1 = 0^\circ$$

$$_e2 = 0^\circ$$

$$r_{os1} = \frac{r_{is1}}{\sin^2(\phi)} + 0.5e_{as} = 556.40 \text{ mm}$$

$$r_{os2} = \frac{r_{is2}}{\sin^2(\phi)} + 0.5e_{as} = 556.40 \text{ mm}$$

$$\delta_1 = \frac{d_{eb1}}{2r_{os1}} = 0.02858$$

$$\delta_2 = \frac{d_{eb2}}{2r_{os2}} = 0.02858$$

$$a_1 = r_{os1} [\arcsin(\delta_1 + \sin(\phi_{e1})) - \phi_{e1}] = 15.90 \text{ mm}$$

$$a_2 = r_{os2} [\arcsin(\delta_2 + \sin(\phi_{e2})) - \phi_{e2}] = 15.90 \text{ mm}$$

Af of the shell contained along the length Lb

$$A_{f_{Ls}} = (L_b - a_1 - a_2) \cdot e_{cs} = 5\,210.7 \text{ mm}^2$$

$$A_{p\phi1} = 89.9 \text{ mm}^2$$

$$A_{p\phi2} = 89.9 \text{ mm}^2$$

$$A_{pb1} = 453.3 \text{ mm}^2$$

$$A_{pb2} = 453.3 \text{ mm}^2$$

$$A_{fb1} = 57.9 \text{ mm}^2$$

$$A_{fb2} = 57.9 \text{ mm}^2$$

$$A_{fp1} = 0 \text{ mm}^2$$

$$A_{fp2} = 0 \text{ mm}^2$$

$$f_{ob1} = 182.07 \text{ MPa}$$

$$f_{ob2} = 182.07 \text{ MPa}$$

$$f_{op1} = 0 \text{ MPa}$$

$$f_{op2} = 0 \text{ MPa}$$

Reactive force

$$F = (A_{f_{Ls}} + A_{fw}) (f_s - 0.5P) + A_{fb1} (f_{ob1} - 0.5P) + A_{fp1} (f_{op1} - 0.5P) + A_{fb2} (f_{ob2} - 0.5P) + A_{fp2} (f_{op2} - 0.5P) = 969\,603 \text{ N}$$

Pressure load

$$F_{req} = P (A_{p_{Ls}} + A_{pb1} + 0.5A_{p\phi1} + A_{pb2} + 0.5A_{p\phi2}) = 698\,379 \text{ N}$$

F ≥ Freq: Ok

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Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Calculation pressure	P = Pt + Ph =	15.32 MPa

Transverse section

	$\delta = \frac{d_{eb}}{2r_{ms}}$	=	0.02866
Distance taken along the mid-thickness of the shell between the centre of the opening and the external edge of the nozzle	$a = 0,5r_{ms}[\arcsin(\delta + \sin(\varphi)) + \arcsin(\delta - \sin(\varphi))]$	=	16.31 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	554.80 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	176.26 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	145.15 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	42 555.8 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l'_b + l'_{bi} + e'_s)$	=	197.9 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	4 064.1 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib} \cdot (l'_b + e_{as})$	=	431.4 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	54.1 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 142 582 N
Pressure load	$Fp = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	659 008 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	26.03 MPa

Fr ≥ Fp: Ok**Adjacent openings****Trubka-výpust' komory 2 - Trubka-výpust' komory 1**

Centre-to-centre distance taken on the mean surface of the shell	Lb	=	241.91 mm
Ligament length	L	=	209.30 mm
Mean shell radius at the centres of adjacent nozzles	r_is	=	540.80 mm
Minimum required ligament length	$L_{\min} = \max \left[3e_{as}; 0.2\sqrt{(2r_{is} + e_{cs})e_{cs}} \right]$	=	84.00 mm
	d1	=	31.80 mm
	d2	=	31.80 mm

L ≥ Lmin: Ok**d1+d2 ≤ 0.2√[(2r_is+e_c,s)·e_c,s]: Not satisfied, ligament check required**

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Ligament check

Inside radius of curvature of the shell at the opening centre

$$r_{is} = \frac{D_e}{2} - e_{as} = 540.80 \text{ mm}$$

Angle between the centre-to-centre line of openings and the generatrix of the shell

$$\phi = 90.00^\circ$$

Ap of the shell for the length Lb

$$Ap_{Ls} = \frac{0.5r_{is}^2 \cdot L_b (1 + \cos(\phi))}{r_{is} + 0.5e_{as} \cdot \sin(\phi)} = 63\,762.3 \text{ mm}^2$$

$$deb1 = 31.80 \text{ mm}$$

$$deb2 = 31.80 \text{ mm}$$

$$_e1 = 0^\circ$$

$$_e2 = 0^\circ$$

$$r_{os1} = \frac{r_{is1}}{\sin^2(\phi)} + 0.5e_{as} = 554.80 \text{ mm}$$

$$r_{os2} = \frac{r_{is2}}{\sin^2(\phi)} + 0.5e_{as} = 554.80 \text{ mm}$$

$$\delta_1 = \frac{d_{eb1}}{2r_{os1}} = 0.02866$$

$$\delta_2 = \frac{d_{eb2}}{2r_{os2}} = 0.02866$$

$$a_1 = r_{os1} [\arcsin(\delta_1 + \sin(\phi_{e1})) - \phi_{e1}] = 15.90 \text{ mm}$$

$$a_2 = r_{os2} [\arcsin(\delta_2 + \sin(\phi_{e2})) - \phi_{e2}] = 15.90 \text{ mm}$$

Af of the shell contained along the length Lb

$$Af_{Ls} = (L_b - a_1 - a_2) \cdot e_{cs} = 5\,883.0 \text{ mm}^2$$

$$Ap\phi1 = 54.1 \text{ mm}^2$$

$$Ap\phi2 = 54.1 \text{ mm}^2$$

$$Apb1 = 431.4 \text{ mm}^2$$

$$Apb2 = 431.4 \text{ mm}^2$$

$$Afb1 = 197.9 \text{ mm}^2$$

$$Afb2 = 197.9 \text{ mm}^2$$

$$Afp1 = 0 \text{ mm}^2$$

$$Afp2 = 0 \text{ mm}^2$$

$$fob1 = 266.67 \text{ MPa}$$

$$fob2 = 266.67 \text{ MPa}$$

$$fop1 = 0 \text{ MPa}$$

$$fop2 = 0 \text{ MPa}$$

Reactive force

$$F = (Af_{Ls} + Af_w)(f_s - 0.5P) + Af_{b1}(f_{ob1} - 0.5P) + Af_{p1}(f_{op1} - 0.5P) + \\ + Af_{b2}(f_{ob2} - 0.5P) + Af_{p2}(f_{op2} - 0.5P) = 1\,682\,269 \text{ N}$$

Pressure load

$$F_{req} = P(Ap_{Ls} + Ap_{b1} + 0.5Ap_{\phi1} + Ap_{b2} + 0.5Ap_{\phi2}) = 990\,930 \text{ N}$$

F ≥ Freq: Ok

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Standard Welding neck flange - Příruba-výpust' komory 1*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 13CrMo4-5 (NT) (EN 10222-2) - Forging ($35,001 \leq t \leq 60$) - No.: 1.7335**Shell material** 10CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 40.00$ mm) - No.: 1.7380**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	25
Number of bolts	=	4
Bolt type	=	ISO M16 x 2.00
Material group	=	5E0

Calculation temperature	T =	37.78 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa

Bolt loads

Number of bolts	n =	4
Bolt type	=	ISO M16 x 2.00
Root area of one bolt	=	144.0 mm ²
Distance between centre lines of adjacent bolts	δb =	78.54 mm
Bolt effective diameter	db =	14.12 mm

Notes

Up to and including 50°C all flange types are suitable for the given PN

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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Nozzle - Trubka-výpust' komory 2*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Access or inspection opening	=	No

Material: 10CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7380

Nominal design stress at internal design temperature	$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_m/20}{2.4}\right) =$	182.07 MPa
Nominal design stress at room temperature	$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_m/20}{2.4}\right) =$	186.67 MPa
Nominal design stress in test condition	$f_{test} = \left(\frac{R_{p0.2/T_{test}}}{1.05}\right) =$	266.67 MPa

Geometry

Inside diameter	Di =	21.80 mm
Outside diameter	De =	31.80 mm
Length	L =	120.00 mm
Nominal thickness	en =	5.00 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	28.20 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	4.05 mm
e/De ≤ 0,16 (0.12700 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	26.00 MPa
Maximum allowable design pressure	=	15.14 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	$F = 50 \cdot en / (Di/2 + en/2) =$	18.657 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.32 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	21.80 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	0.64 mm
e/De ≤ 0,16 (0.02000 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-výpust' komory 2*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 13CrMo4-5 (NT) (EN 10222-2) - Forging ($35,001 \leq t \leq 60$) - No.: 1.7335**Shell material** 10CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 40.00$ mm) - No.: 1.7380**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	25
Number of bolts	=	4
Bolt type	=	ISO M16 x 2.00
Material group	=	5E0

Calculation temperature	T =	37.78 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa

Bolt loads

Number of bolts	n =	4
Bolt type	=	ISO M16 x 2.00
Root area of one bolt	=	144.0 mm ²
Distance between centre lines of adjacent bolts	δb =	78.54 mm
Bolt effective diameter	db =	14.12 mm

Notes

Up to and including 50°C all flange types are suitable for the given PN

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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Revision

Cylindrical shell - Komora*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature

Ti = 37.78 °C

Internal design pressure

Pi = 10.69 MPa

Joint efficiency

z = 1.00

Material: 13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335

Nominal design stress at internal design temperature

$$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_m/20}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress at room temperature

$$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_m/20}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress in test condition

$$f_{\text{test}} = \left(\frac{R_{p0.2/T_{\text{test}}}}{1.05}\right) = 276.19 \text{ MPa}$$

Geometry

Inside diameter

Di = 620.00 mm

Outside diameter

De = 676.00 mm

Length

L = 540.00 mm

Nominal thickness

en = 28.00 mm

Corrosion allowance

c = 3.20 mm

External corrosion allowance

ce = 0 mm

Undertolerance

δ = 0 mm

Internal pressure

Overpressure due to static head

Ph = 0 MPa

Calculation pressure

P=Pi+Ph = 10.69 MPa

Inside diameter

Di'=Di+2δ+2c = 626.40 mm

Minimum required thickness

$$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta = 21.58 \text{ mm}$$

e/De ≤ 0,16 (0.03200 ≤ 0.16000): Ok**en ≥ e: Ok****Maximum allowable pressures (at the top of the vessel)**

Maximum allowable test pressure

= 23.84 MPa

Maximum allowable design pressure

= 14.28 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation

F=50·en/(Di/2+en/2) = 4.321 %

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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.32 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	620.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	17.68 mm
e/De ≤ 0,16 (0.02600 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Reinforcement of opening - Trubka- vstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	187.50 MPa
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Nozzle material: 11CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	192.67 MPa
	fob = min(fs, fb) =	187.50 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Radial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset from shell border	=	270.00 mm
Angular offset	=	0 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	355.60 mm
Internal diameter	d_ib = Id + 2(c + δ) =	298.00 mm
External diameter of the nozzle	d_eb = Od - 2ce =	355.60 mm
Nominal thickness of the nozzle	e_ab =	32.00 mm
Length of nozzle extending outside the shell	l_b =	272.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	28.80 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	101.76 mm
	$l_{bo} = \min(l_b, l_{bo(max)})$ =	101.76 mm
	$l_b' = \min(l_{bo}, l_b)$ =	101.76 mm
	$l_{bi}' = \min(l_{bi}, 0, 5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement		
Effective length of nozzle inside the shell for reinforcement		
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm ²

ea,b / ea,s ≤ 2,297: Ok**Pad geometry**

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	Af_p = e_p · l_p' =	0 mm ²

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Shell external diameter	De =	676.00 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$ =	313.20 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 92.20 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 92.20 \text{ mm}$$

Minimum value for w which has no influence on l_s from shell discontinuities

$$w_p = l_{so} = 127.08 \text{ mm}$$

Required minimum value for w

$$w_{min} = \max[0.2\sqrt{(2r_{is} + e_{cs})e_{cs}}; 3e_{as}] = 74.40 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Longitudinal section

Inside radius of curvature of the shell at the opening centre

$$a = \frac{d_{cb}}{2} = 177.80 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$r_{is} = \frac{D_e}{2} - e_{as} = 313.20 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Pressure loaded area - shell

$$l'_s = \min[l_s, l_{so}] = 92.20 \text{ mm}$$

Length of penetration into shell wall

$$A_{p_s} = r_{is}(l'_s + a) = 84\,564.0 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area - shell

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 4\,049.9 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,286.6 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 18\,857.5 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p\phi} = 0 \text{ mm}^2$$

Reactive force

$$A_{fp} = 0 \text{ mm}^2$$

Pressure load

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{fp}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 1\,154\,224 \text{ N}$$

Maximum allowable pressure

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) = 1\,105\,576 \text{ N}$$

$$P_{max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{fp} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{fp})} = 11.15 \text{ MPa}$$

Fr ≥ Fp: Ok**Transverse section**

Mean radius of curvature

$$\delta = \frac{d_{cb}}{2r_{ms}} = 0.54607$$

Maximum length of shell contributing to opening reinforcement

$$a = r_{ms} \arcsin(\delta) = 188.09 \text{ mm}$$

Effective length of shell for opening reinforcement

$$r_{ms} = r_{is} + 0.5e_{as} = 325.60 \text{ mm}$$

Pressure loaded area - shell

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 127.08 \text{ mm}$$

Length of penetration into shell wall

$$l'_s = \min[l_s, l_{so}] = 92.20 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 42\,221.4 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Pressure loaded area - nozzle

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 4\,049.9 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,286.6 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 18\,857.5 \text{ mm}^2$$

$$A_{p\phi} = 0 \text{ mm}^2$$

$$A_{fp} = 0 \text{ mm}^2$$

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Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 154 224 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	652 934 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	18.49 MPa
			Fr ≥ Fp: Ok

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	15.29 MPa
Overpressure due to static head	Ph	=	0.03 MPa
Calculation pressure	P = Pt + Ph	=	15.32 MPa

Longitudinal section

	$a = \frac{d_{cb}}{2}$	=	177.80 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$	=	310.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	92.20 mm
Pressure loaded area - shell	$Ap_s = r_{is}(l'_s + a)$	=	83 700.0 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	4 152.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 581.6 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	18 919.1 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 595 720 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	1 571 723 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	15.54 MPa
			Fr ≥ Fp: Ok

Transverse section

	$\delta = \frac{d_{cb}}{2r_{ms}}$	=	0.54877
	$a = r_{ms} \arcsin(\delta)$	=	188.21 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	324.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	92.20 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	41 585.1 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	4 152.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 581.6 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	18 919.1 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²

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Reactive force

$$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P) =$$

1 595 720 N

Pressure load

$$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi) =$$

926 687 N

Maximum allowable pressure

$$P_{\max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)} =$$

25.79 MPa

Fr ≥ Fp: Ok

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Reinforcement of opening - Trubka- výstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	187.50 MPa
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Nozzle material: 11CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	192.67 MPa
	fob = min(fs, fb) =	187.50 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Radial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset from shell border	=	270.00 mm
Angular offset	=	180.00 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	355.60 mm
Internal diameter	d_ib = Id + 2(c + δ) =	298.00 mm
External diameter of the nozzle	d_eb = Od - 2ce =	355.60 mm
Nominal thickness of the nozzle	e_ab =	32.00 mm
Length of nozzle extending outside the shell	l_b =	272.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	28.80 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	101.76 mm
	$l_{bo} = \min(l_b, l_{bo(max)})$ =	101.76 mm
	$l'_b = \min(l_{bo}, l_b)$ =	101.76 mm
	$l'_{bi} = \min(l_{bi}, 0, 5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement		
Effective length of nozzle inside the shell for reinforcement		
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm ²

ea,b / ea,s ≤ 2,297: Ok**Pad geometry**

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	Af_p = e_p · l_p' =	0 mm ²

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Shell external diameter	De =	676.00 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$ =	313.20 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 92.20 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 92.20 \text{ mm}$$

Minimum value for w which has no influence on l_s from shell discontinuities

$$w_p = l_{so} = 127.08 \text{ mm}$$

Required minimum value for w

$$w_{min} = \max[0.2\sqrt{(2r_{is} + e_{cs})e_{cs}}; 3e_{as}] = 74.40 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Longitudinal section

Inside radius of curvature of the shell at the opening centre

$$a = \frac{d_{cb}}{2} = 177.80 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$r_{is} = \frac{D_e}{2} - e_{as} = 313.20 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Pressure loaded area - shell

$$l'_s = \min[l_s, l_{so}] = 92.20 \text{ mm}$$

Length of penetration into shell wall

$$A_{p_s} = r_{is}(l'_s + a) = 84\,564.0 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area - shell

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 4\,049.9 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,286.6 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 18\,857.5 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p\phi} = 0 \text{ mm}^2$$

Reactive force

$$A_{fp} = 0 \text{ mm}^2$$

Pressure load

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{fp}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 1\,154\,224 \text{ N}$$

Maximum allowable pressure

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) = 1\,105\,576 \text{ N}$$

$$P_{max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{fp} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{fp})} = 11.15 \text{ MPa}$$

Fr ≥ Fp: Ok**Transverse section**

Mean radius of curvature

$$\delta = \frac{d_{cb}}{2r_{ms}} = 0.54607$$

Maximum length of shell contributing to opening reinforcement

$$a = r_{ms} \arcsin(\delta) = 188.09 \text{ mm}$$

Effective length of shell for opening reinforcement

$$r_{ms} = r_{is} + 0.5e_{as} = 325.60 \text{ mm}$$

Pressure loaded area - shell

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 127.08 \text{ mm}$$

Length of penetration into shell wall

$$l'_s = \min[l_s, l_{so}] = 92.20 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 42\,221.4 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Pressure loaded area - nozzle

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 4\,049.9 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,286.6 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 18\,857.5 \text{ mm}^2$$

$$A_{p\phi} = 0 \text{ mm}^2$$

$$A_{fp} = 0 \text{ mm}^2$$

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Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 154 224 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	652 934 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	18.49 MPa
			Fr ≥ Fp: Ok

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	15.29 MPa
Overpressure due to static head	Ph	=	0.03 MPa
Calculation pressure	P = Pt + Ph	=	15.32 MPa

Longitudinal section

	$a = \frac{d_{cb}}{2}$	=	177.80 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$	=	310.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	92.20 mm
Pressure loaded area - shell	$Ap_s = r_{is}(l'_s + a)$	=	83 700.0 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l'_b + l'_{bi} + e'_s)$	=	4 152.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 581.6 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib} \cdot (l'_b + e_{as})$	=	18 919.1 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 595 720 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	1 571 723 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	15.54 MPa
			Fr ≥ Fp: Ok

Transverse section

	$\delta = \frac{d_{cb}}{2r_{ms}}$	=	0.54877
	$a = r_{ms} \arcsin(\delta)$	=	188.21 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	324.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	92.20 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	41 585.1 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l'_b + l'_{bi} + e'_s)$	=	4 152.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 581.6 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib} \cdot (l'_b + e_{as})$	=	18 919.1 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²

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Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P) =$	1 595 720 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi) =$	926 687 N
Maximum allowable pressure	$P_{\max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)} =$	25.79 MPa
		Fr ≥ Fp: Ok

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Nozzle - Trubka- vstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Material: 11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383

Nominal design stress at internal design temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress at room temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4}) =$	225.00 MPa

Geometry

Inside diameter	Di =	291.60 mm
Outside diameter	De =	355.60 mm
Length	L =	300.00 mm
Nominal thickness	en =	32.00 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	298.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	11.70 mm
e/De ≤ 0,16 (0.03300 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	15.51 MPa
Maximum allowable design pressure	=	11.15 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	9.889 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.32 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	291.60 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	10.27 mm
e/De ≤ 0,16 (0.02900 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-vstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($100,001 \leq t \leq 150$) - No.: 1.7335**Shell material** 11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	300
Number of bolts	=	16
Bolt type	=	ISO M39 x 4.00
Material group	=	5E0

Calculation temperature	T =	37.78 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa

Bolt loads

Number of bolts	n =	16
Bolt type	=	ISO M39 x 4.00
Root area of one bolt	=	913.0 mm ²
Distance between centre lines of adjacent bolts	δb =	98.17 mm
Bolt effective diameter	db =	35.25 mm

Notes

Up to and including 50°C all flange types are suitable for the given PN

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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Nozzle - Trubka- výstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	37.78 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Material: 11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383

Nominal design stress at internal design temperature	$f = \min(f_{nc}; \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress at room temperature	$f = \min(f_{nc}; \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4}) =$	225.00 MPa

Geometry

Inside diameter	Di =	291.60 mm
Outside diameter	De =	355.60 mm
Length	L =	300.00 mm
Nominal thickness	en =	32.00 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	298.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	11.70 mm
e/De ≤ 0,16 (0.03300 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	15.51 MPa
Maximum allowable design pressure	=	11.15 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	9.889 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.32 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	291.60 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	10.27 mm
e/De ≤ 0,16 (0.02900 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-výstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($100,001 \leq t \leq 150$) - No.: 1.7335**Shell material** 11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	300
Number of bolts	=	16
Bolt type	=	ISO M39 x 4.00
Material group	=	5E0

Calculation temperature	T =	37.78 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa

Bolt loads

Number of bolts	n =	16
Bolt type	=	ISO M39 x 4.00
Root area of one bolt	=	913.0 mm ²
Distance between centre lines of adjacent bolts	δb =	98.17 mm
Bolt effective diameter	db =	35.25 mm

Notes

Up to and including 50°C all flange types are suitable for the given PN

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Maximum pressure at temperature allowed by the specifications	Pmax =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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Welding neck flange - Komorová příruba*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 11CrMo9-10 (NT) (EN 10222-2) - Forging ($t \leq 200.00$ mm) - No.: 1.7383**Shell material** 13CrMo4-5 (EN 10028-2) - Plate ($16,001 \leq t \leq 60$) - No.: 1.7335**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Allowable stresses	Flange - f	Hub - fH	Bolting - fB
Design condition	194.81 MPa / 28 255.4 psi	194.81 MPa / 28 255.4 psi	145.68 MPa / 21 128.9 psi
Seating condition	200.00 MPa / 29 007.5 psi	200.00 MPa / 29 007.5 psi	146.67 MPa / 21 272.2 psi
Test condition	285.71 MPa / 41 439.4 psi	285.71 MPa / 41 439.4 psi	220.00 MPa / 31 908.3 psi

Internal pressure Pd = 10.69 MPa

Overpressure due to static head Ph = 0 MPa

Calculation pressure P = 10.69 MPa

Calculation temperature T = 37.78 °C

Geometry

Corrosion allowance c = 3.20 mm

Flange external diameter A = 1 018.55 mm

Inside diameter B = 620.00 mm

Inside diameter (corroded) B* = B + 2c = 626.40 mm

Bolt circle C = 920.55 mm

Flange thickness en = 102.00 mm

Mean gasket diameter Gmean = 637.00 mm

Hub length h = 67.00 mm

Thickness of hub at back of flange g1 = 56.50 mm

Thickness of hub at back of flange (corroded) g1* = 53.30 mm

Thickness of hub at small end g0 = 28.00 mm

Thickness of hub at small end (corroded) g0* = 24.80 mm

Gasket parameters

Gasket factor m = 1.25

Minimum gasket seating pressure y = 2.80 MPa

Gasket contact width w = 13.00 mm

Basic gasket seating width b0 = w / 2 = 6.50 mm

Effective gasket seating width b = 2.52·√(b0) = 6.42 mm

Diameter of gasket load reaction G = Gmean + w - 2b = 637.15 mm

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Bolt loads

Number of bolts	n =	28
Bolt type	=	ISO_TEMA M42 x 4.50
Root area of one bolt	=	1 018.2 mm ²
Distance between centre lines of adjacent bolts	δb =	103.29 mm
Bolt effective diameter	db =	36.01 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	3 408 408 N
Compression load on gasket to ensure tight joint	HG = 2π·G·b·m·P =	343 690 N
Minimum required bolt load for operating condition	Wop = H + HG =	3 752 098 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	4 882 092 N
Minimum required bolt load for the test condition (Plášťová příruba)	Wt =	5 376 003 N
Minimum required bolt load for assembly condition	WA = πb·G·y =	36 009 N
Total required cross-sectional area of bolts	$A_{Bmin} = \max \left[\frac{W_A}{f_{B,A}}, \frac{W_{op}}{f_B}, \frac{W_t}{f_{B,t}} \right]$ =	25 755.9 mm ²
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	28 510.1 mm ²
Maximum bolts area for gasket crush	$A_{bmax} = \frac{2\pi \cdot y \cdot G \cdot w}{f_{B,A}}$ =	993.6 mm ²
Design bolt load for assembly condition	$W = 0,5(A_{Bmin} + A_B)f_{B,A}$ =	3 979 511 N

AB ≥ AB,min: Ok**Flange constants**

Bolt pitch correction factor	$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}}; 1 \right]$ =	1.00000
Ratio of the flange diameters	K = A / B* =	1.62604
Length parameter	$l_0 = \sqrt{B^* \cdot g_0^*}$ =	124.64 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4} (B^2 P)$ =	3 294 360 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	114 048 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2 =	120.42 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2 =	141.70 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B* - G) / 4 =	144.39 mm
Flange stress factor	$\beta_T = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{(10472 + 19448 K^2) (K - 1)}$ =	1.65645
Flange stress factor	$\beta_U = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)}$ =	4.58080
Flange stress factor	$\beta_Y = \frac{1}{K-1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right]$ =	4.16854
	βF =	0.81332
	βV =	0.20191
Hub stress correction factor	φ =	1.29001
	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3 \beta_V}{\beta_U l_0 g_0^2} \right)$ =	1.61571

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Flange moments

Total moment acting upon flange for assembly condition

$$MA = W \cdot h_G = 563\,893.7 \text{ N}\cdot\text{m}$$

Total moment acting upon flange for operating condition

$$M_{op} = |H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G| = 461\,889.3 \text{ N}\cdot\text{m}$$

Moment factor used to design split rings

$$F_s = 1.00$$

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

$$M = F_s \cdot M_{op} \frac{C_F}{B^*} = 737.4 \text{ N}\cdot\text{m}$$

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

$$M = F_s \cdot M_A \frac{C_F}{B^*} = 900.2 \text{ N}\cdot\text{m}$$

Flange stresses - operating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 207.24 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \cdot \beta_F + I_0) M}{\lambda \cdot e^2 \cdot I_0} = 82.78 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 111.94 \text{ MPa}$$

Stress factor

$$k = 1.00000$$

$$k \cdot \sigma_H \leq 1.5 \min(f; f_H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

Flange stresses - seating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 253.00 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \cdot \beta_F + I_0) M}{\lambda \cdot e^2 \cdot I_0} = 101.07 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 136.66 \text{ MPa}$$

$$k \cdot \sigma_H \leq 1.5 \min(f; f_H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)

$$= 13.69 \text{ MPa}$$

Hot & corroded (flange)

$$= 11.29 \text{ MPa}$$

New & cold (bolts)

$$= 11.91 \text{ MPa}$$

Hot & corroded (bolts)

$$= 11.83 \text{ MPa}$$

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Hydrostatic test

Item or side hydrostatic test pressure

Pt = 15.29 MPa

Overpressure due to static head

Ph = 0.03 MPa

Calculation pressure

P = Pt + Ph = 15.31 MPa

Flange constants

Bolt pitch correction factor

$$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0.5}}}; 1 \right] = 1.00000$$

Ratio of the flange diameters

$$K = A / B = 1.64282$$

Length parameter

$$l_0 = \sqrt{B \cdot g_0^*} = 131.76 \text{ mm}$$

Hydrostatic end force applied via shell to flange

$$H_D = \frac{\pi}{4} (B^2 P) = 4622803 \text{ N}$$

Hydrostatic end force due to pressure on flange face

$$HT = H - HD = 259290 \text{ N}$$

Radial distance from bolt circle to circle on which HD acts

$$hD = (C - B - g_1^*) / 2 = 122.02 \text{ mm}$$

Radial distance from gasket load reaction to bolt circle

$$hG = (C - G) / 2 = 141.70 \text{ mm}$$

Radial distance from bolt circle to circle on which HT acts

$$hT = (2C - B - G) / 4 = 145.99 \text{ mm}$$

Flange stress factor

$$\beta_T = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{(10472 + 19448 K^2) (K - 1)} = 1.64933$$

Flange stress factor

$$\beta_U = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)} = 4.48990$$

Flange stress factor

$$\beta_Y = \frac{1}{K - 1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right] = 4.08582$$

$$\beta F = 0.82329$$

$$\beta V = 0.22295$$

Hub stress correction factor

$$\varphi = 1.22033$$

$$\lambda = \left(\frac{e \beta_F + l_0}{\beta_T l_0} + \frac{e^3 \beta_V}{\beta_U l_0 g_0^2} \right) = 1.50287$$

Flange moments

Total moment acting upon flange for operating condition

$$M_{op} = |H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G| = 650648.7 \text{ N}\cdot\text{m}$$

Moment factor used to design split rings

$$F_s = 1.00$$

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

$$M = F_s \cdot M_{op} \frac{C_F}{B} = 1049.4 \text{ N}\cdot\text{m}$$

Flange stresses - operating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 266.94 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \beta_F + l_0) M}{\lambda e^2 l_0} = 124.14 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 141.85 \text{ MPa}$$

Stress factor

$$k = 1.00000$$

$$k \cdot \sigma_H \leq 1.5 \min(f; f_H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

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Welding neck flange - Plášťová příruba*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 11CrMo9-10 (NT) (EN 10222-2) - Forging ($t \leq 200.00$ mm) - No.: 1.7383**Shell material** 13CrMo4-5 (EN 10028-2) - Plate ($16,001 \leq t \leq 60$) - No.: 1.7335**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Allowable stresses	Flange - f	Hub - fH	Bolting - fB
Design condition	172.44 MPa / 25 011.0 psi	172.44 MPa / 25 011.0 psi	141.54 MPa / 20 528.8 psi
Seating condition	200.00 MPa / 29 007.5 psi	200.00 MPa / 29 007.5 psi	146.67 MPa / 21 272.2 psi
Test condition	285.71 MPa / 41 439.4 psi	285.71 MPa / 41 439.4 psi	220.00 MPa / 31 908.3 psi

Internal pressure

Pd = 10.69 MPa

Overpressure due to static head

Ph = 0 MPa

Calculation pressure

P = 10.69 MPa

Calculation temperature

T = 121.11 °C

Geometry

Corrosion allowance

c = 3.20 mm

Flange external diameter

A = 1 018.55 mm

Inside diameter

B = 620.00 mm

Inside diameter (corroded)

B* = B + 2c = 626.40 mm

Bolt circle

C = 920.55 mm

Flange thickness

en = 102.00 mm

Mean gasket diameter

Gmean = 637.00 mm

Hub length

h = 67.00 mm

Thickness of hub at back of flange

g1 = 56.50 mm

Thickness of hub at back of flange (corroded)

g1* = 53.30 mm

Thickness of hub at small end

g0 = 28.00 mm

Thickness of hub at small end (corroded)

g0* = 24.80 mm

Gasket parameters

Gasket factor

m = 1.25

Minimum gasket seating pressure

y = 2.80 MPa

Gasket contact width

w = 13.00 mm

Basic gasket seating width

b0 = w / 2 = 6.50 mm

Effective gasket seating width

b = 2.52·√(b0) = 6.42 mm

Diameter of gasket load reaction

G = Gmean + w - 2b = 637.15 mm

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Bolt loads

Number of bolts	n =	28
Bolt type	=	ISO_TEMA M42 x 4.50
Root area of one bolt	=	1 018.2 mm ²
Distance between centre lines of adjacent bolts	δb =	103.29 mm
Bolt effective diameter	db =	36.01 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	3 408 408 N
Compression load on gasket to ensure tight joint	HG = 2π·G·b·m·P =	343 690 N
Minimum required bolt load for operating condition	Wop = H + HG =	3 752 098 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	4 883 564 N
Minimum required bolt load for the test condition	Wt = Ht + 2b·π·G·m·Pt =	5 376 003 N
Minimum required bolt load for assembly condition	WA = πb·G·y =	36 009 N
Total required cross-sectional area of bolts	$A_{Bmin} = \max \left[\frac{W_A}{f_{B,A}}, \frac{W_{op}}{f_B}, \frac{W_t}{f_{B,t}} \right]$ =	26 509.0 mm ²
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	28 510.1 mm ²
Maximum bolts area for gasket crush	$A_{bmax} = \frac{2\pi \cdot y \cdot G \cdot w}{f_{B,A}}$ =	993.6 mm ²
Design bolt load for assembly condition	$W = 0,5(A_{Bmin} + A_B)f_{B,A}$ =	4 034 731 N

AB ≥ AB,min: Ok**Flange constants**

Bolt pitch correction factor	$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}}; 1 \right]$ =	1.00000
Ratio of the flange diameters	K = A / B* =	1.62604
Length parameter	$l_0 = \sqrt{B^* \cdot g_0^*}$ =	124.64 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4} (B^2 P)$ =	3 294 360 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	114 048 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2 =	120.42 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2 =	141.70 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B* - G) / 4 =	144.39 mm
Flange stress factor	$\beta_T = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{(10472 + 19448 K^2) (K - 1)}$ =	1.65645
Flange stress factor	$\beta_U = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)}$ =	4.58080
Flange stress factor	$\beta_Y = \frac{1}{K-1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right]$ =	4.16854
	βF =	0.81332
	βV =	0.20191
Hub stress correction factor	φ =	1.29001
	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3 \beta_V}{\beta_U l_0 g_0^2} \right)$ =	1.61571

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Flange moments

Total moment acting upon flange for assembly condition

$$MA = W \cdot h_G = 571\,718.4 \text{ N}\cdot\text{m}$$

Total moment acting upon flange for operating condition

$$M_{op} = |H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G| = 461\,889.3 \text{ N}\cdot\text{m}$$

Moment factor used to design split rings

$$F_s = 1.00$$

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

$$M = F_s \cdot M_{op} \frac{C_F}{B^*} = 737.4 \text{ N}\cdot\text{m}$$

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

$$M = F_s \cdot M_A \frac{C_F}{B^*} = 912.7 \text{ N}\cdot\text{m}$$

Flange stresses - operating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 207.24 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \cdot \beta_F + I_0) M}{\lambda \cdot e^2 \cdot I_0} = 82.78 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 111.94 \text{ MPa}$$

Stress factor

$$k = 1.00000$$

$$k \cdot \sigma_H \leq 1.5 \min(f; H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

Flange stresses - seating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 256.51 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \cdot \beta_F + I_0) M}{\lambda \cdot e^2 \cdot I_0} = 102.47 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 138.56 \text{ MPa}$$

$$k \cdot \sigma_H \leq 1.5 \min(f; H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)

$$= 13.69 \text{ MPa}$$

Hot & corroded (flange)

$$= 10.97 \text{ MPa}$$

New & cold (bolts)

$$= 11.91 \text{ MPa}$$

Hot & corroded (bolts)

$$= 11.50 \text{ MPa}$$

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Hydrostatic test

Item or side hydrostatic test pressure

Pt = 15.29 MPa

Overpressure due to static head

Ph = 0.03 MPa

Calculation pressure

P = Pt + Ph = 15.32 MPa

Flange constants

Bolt pitch correction factor

$$C_F = \max \left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0.5}}}; 1 \right] = 1.00000$$

Ratio of the flange diameters

$$K = A / B = 1.64282$$

Length parameter

$$l_0 = \sqrt{B \cdot g_0^*} = 131.76 \text{ mm}$$

Hydrostatic end force applied via shell to flange

$$H_D = \frac{\pi}{4} (B^2 P) = 4624197 \text{ N}$$

Hydrostatic end force due to pressure on flange face

$$HT = H - HD = 259368 \text{ N}$$

Radial distance from bolt circle to circle on which HD acts

$$hD = (C - B - g_1^*) / 2 = 122.02 \text{ mm}$$

Radial distance from gasket load reaction to bolt circle

$$hG = (C - G) / 2 = 141.70 \text{ mm}$$

Radial distance from bolt circle to circle on which HT acts

$$hT = (2C - B - G) / 4 = 145.99 \text{ mm}$$

Flange stress factor

$$\beta_T = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{(10472 + 19448 K^2) (K - 1)} = 1.64933$$

Flange stress factor

$$\beta_U = \frac{K^2 (1 + 8.55246 \log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)} = 4.48990$$

Flange stress factor

$$\beta_Y = \frac{1}{K - 1} \left[0.66845 + 5.7169 \left(\frac{K^2 \log_{10} K}{K^2 - 1} \right) \right] = 4.08582$$

$$\beta F = 0.82329$$

$$\beta V = 0.22295$$

Hub stress correction factor

$$\varphi = 1.22033$$

$$\lambda = \left(\frac{e \beta_F + l_0}{\beta_T l_0} + \frac{e^3 \beta_V}{\beta_U l_0 g_0^2} \right) = 1.50287$$

Flange moments

Total moment acting upon flange for operating condition

$$M_{op} = |H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G| = 650830.2 \text{ N}\cdot\text{m}$$

Moment factor used to design split rings

$$F_s = 1.00$$

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

$$M = F_s \cdot M_{op} \frac{C_F}{B} = 1049.7 \text{ N}\cdot\text{m}$$

Flange stresses - operating condition

Longitudinal stress in hub

$$\sigma_H = \frac{\phi M}{\lambda g_s^2} = 267.01 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{(1333e \beta_F + l_0) M}{\lambda e^2 l_0} = 124.17 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2 + 1}{K^2 - 1} = 141.89 \text{ MPa}$$

Stress factor

$$k = 1.00000$$

$$k \cdot \sigma_H \leq 1.5 \min(f; f_H): \text{ Ok}$$

$$k \cdot \sigma_r \leq f: \text{ Ok}$$

$$k \cdot \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

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Cylindrical shell - Plášť*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature

Ti = 121.11 °C

Internal design pressure

Pi = 10.69 MPa

Joint efficiency

z = 1.00

Material: 13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335

Nominal design stress at internal design temperature

$$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_m/20}{2.4}\right) = 179.11 \text{ MPa}$$

Nominal design stress at room temperature

$$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_m/20}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress in test condition

$$f_{\text{test}} = \left(\frac{R_{p0.2/T_{\text{test}}}}{1.05}\right) = 276.19 \text{ MPa}$$

Geometry

Inside diameter

Di = 620.00 mm

Outside diameter

De = 676.00 mm

Length

L = 2 400.00 mm

Nominal thickness

en = 28.00 mm

Corrosion allowance

c = 3.20 mm

External corrosion allowance

ce = 0 mm

Undertolerance

δ = 0 mm

Internal pressure

Overpressure due to static head

Ph = 0 MPa

Calculation pressure

P=Pi+Ph = 10.69 MPa

Inside diameter

Di'=Di+2δ+2c = 626.40 mm

Minimum required thickness

$$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta = 22.47 \text{ mm}$$

e/De ≤ 0,16 (0.03300 ≤ 0.16000): Ok**en ≥ e: Ok****Maximum allowable pressures (at the top of the vessel)**

Maximum allowable test pressure

= 23.84 MPa

Maximum allowable design pressure

= 13.64 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation

F=50·en/(Di/2+en/2) = 4.321 %

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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.04651
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.98 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.31 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	620.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	17.68 mm
e/De ≤ 0,16 (0.02600 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Reinforcement of opening - Trubka-vstup vody*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	179.11 MPa
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Nozzle material: 11CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	192.67 MPa
	fob = min(fs, fb) =	179.11 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Radial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset from shell border	=	2 255.00 mm
Angular offset	=	180.00 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	101.66 mm
Internal diameter	d_ib = Id + 2(c + δ) =	83.06 mm
External diameter of the nozzle	d_eb = Od - 2ce =	101.66 mm
Nominal thickness of the nozzle	e_ab =	12.50 mm
Length of nozzle extending outside the shell	l_b =	272.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	9.30 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	33.38 mm
	$l_{bo} = \min(l_b; l_{bo(max)})$ =	33.38 mm
	$l'_b = \min(l_{bo}; l_b)$ =	33.38 mm
	$l'_{bi} = \min(l_{bi}; 0,5 l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement		
Effective length of nozzle inside the shell for reinforcement		
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm ²

ea,b / ea,s ≤ 3: Ok**Pad geometry**

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	Af_p = e_p · l_p' =	0 mm ²

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Shell external diameter	De =	676.00 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$ =	313.20 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 94.17 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 94.17 \text{ mm}$$

Minimum value for w which has no influence on l_s from shell discontinuities

$$w_p = l_{so} = 127.08 \text{ mm}$$

Required minimum value for w

$$w_{min} = \max[0.2\sqrt{(2r_{is} + e_{cs})e_{cs}}; 3e_{as}] = 74.40 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Longitudinal section

$$a = \frac{d_{cb}}{2} = 50.83 \text{ mm}$$

Inside radius of curvature of the shell at the opening centre

$$r_{is} = \frac{D_e}{2} - e_{as} = 313.20 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l'_s = \min[l_s, l_{so}] = 94.17 \text{ mm}$$

Pressure loaded area - shell

$$A_{p_s} = r_{is}(l'_s + a) = 45\,414.0 \text{ mm}^2$$

Length of penetration into shell wall

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 727.3 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,335.4 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 2\,416.4 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p\phi} = 0 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{fp} = 0 \text{ mm}^2$$

Reactive force

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{f_p}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 532\,197 \text{ N}$$

Pressure load

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) = 511\,307 \text{ N}$$

Maximum allowable pressure

$$P_{max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{f_p} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{f_p})} = 11.11 \text{ MPa}$$

Fr ≥ Fp: Ok**Transverse section**

$$\delta = \frac{d_{cb}}{2r_{ms}} = 0.15611$$

Mean radius of curvature

$$a = r_{ms} \arcsin(\delta) = 51.04 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$r_{ms} = r_{is} + 0.5e_{as} = 325.60 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 127.08 \text{ mm}$$

Pressure loaded area - shell

$$l'_s = \min[l_s, l_{so}] = 94.17 \text{ mm}$$

Length of penetration into shell wall

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 21\,873.7 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area - shell

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 727.3 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,335.4 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 2\,416.4 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p\phi} = 0 \text{ mm}^2$$

$$A_{fp} = 0 \text{ mm}^2$$

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Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	532 197 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	259 661 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	21.24 MPa
			Fr ≥ Fp: Ok

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	15.29 MPa
Overpressure due to static head	Ph	=	0.006 MPa
Calculation pressure	P = Pt + Ph	=	15.29 MPa

Longitudinal section

	$a = \frac{d_{cb}}{2}$	=	50.83 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$	=	310.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	94.17 mm
Pressure loaded area - shell	$Ap_s = r_{is}(l'_s + a)$	=	44 950.0 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	767.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 636.8 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	2 352.9 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	874 861 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	723 413 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	18.38 MPa
			Fr ≥ Fp: Ok

Transverse section

	$\delta = \frac{d_{cb}}{2r_{ms}}$	=	0.15688
	$a = r_{ms} \arcsin(\delta)$	=	51.04 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	324.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	94.17 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	21 535.1 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	767.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 636.8 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	2 352.9 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²

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Reactive force

$$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P) = 874\,861 \text{ N}$$

Pressure load

$$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi) = 365\,324 \text{ N}$$

Maximum allowable pressure

$$P_{\max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)} = 35.20 \text{ MPa}$$

Fr ≥ Fp: Ok

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Reinforcement of opening - Trubka-výstup vody

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9

Design data

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	179.11 MPa
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Nozzle material: 11CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	192.67 MPa
	fob = min(fs, fb) =	179.11 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Radial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset from shell border	=	145.00 mm
Angular offset	=	0 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	101.66 mm
Internal diameter	d_ib = Id + 2(c + δ) =	83.06 mm
External diameter of the nozzle	d_eb = Od - 2ce =	101.66 mm
Nominal thickness of the nozzle	e_ab =	12.50 mm
Length of nozzle extending outside the shell	l_b =	272.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	9.30 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	33.38 mm
	$l_{bo} = \min(l_b, l_{bo(max)})$ =	33.38 mm
	$l'_b = \min(l_{bo}, l_b)$ =	33.38 mm
	$l'_{bi} = \min(l_{bi}, 0, 5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement		
Effective length of nozzle inside the shell for reinforcement		
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm ²

ea,b / ea,s ≤ 3: Ok**Pad geometry**

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	Af_p = e_p · l_p' =	0 mm ²

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Shell external diameter	De =	676.00 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$ =	313.20 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 94.17 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 94.17 \text{ mm}$$

Minimum value for w which has no influence on l_s from shell discontinuities

$$w_p = l_{so} = 127.08 \text{ mm}$$

Required minimum value for w

$$w_{min} = \max[0.2\sqrt{(2r_{is} + e_{cs})e_{cs}}; 3e_{as}] = 74.40 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Longitudinal section

Inside radius of curvature of the shell at the opening centre

$$a = \frac{d_{cb}}{2} = 50.83 \text{ mm}$$

Maximum length of shell contributing to opening reinforcement

$$r_{is} = \frac{D_e}{2} - e_{as} = 313.20 \text{ mm}$$

Effective length of shell for opening reinforcement

$$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}} = 127.08 \text{ mm}$$

Pressure loaded area - shell

$$l'_s = \min[l_s, l_{so}] = 94.17 \text{ mm}$$

Length of penetration into shell wall

$$A_{p_s} = r_{is}(l'_s + a) = 45\,414.0 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area - shell

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 727.3 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,335.4 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 2\,416.4 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p\phi} = 0 \text{ mm}^2$$

Reactive force

$$A_{fp} = 0 \text{ mm}^2$$

Pressure load

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{fp}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 532\,197 \text{ N}$$

Maximum allowable pressure

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) = 511\,307 \text{ N}$$

$$P_{max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{fp} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\phi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{fp})} = 11.11 \text{ MPa}$$

Fr ≥ Fp: Ok**Transverse section**

Mean radius of curvature

$$\delta = \frac{d_{cb}}{2r_{ms}} = 0.15611$$

Maximum length of shell contributing to opening reinforcement

$$a = r_{ms} \arcsin(\delta) = 51.04 \text{ mm}$$

Effective length of shell for opening reinforcement

$$r_{ms} = r_{is} + 0.5e_{as} = 325.60 \text{ mm}$$

Pressure loaded area - shell

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 127.08 \text{ mm}$$

Length of penetration into shell wall

$$l'_s = \min[l_s, l_{so}] = 94.17 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 21\,873.7 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$e'_s = e_{as} = 24.80 \text{ mm}$$

Pressure loaded area - nozzle

$$A_{f_b} = e_b(l'_b + l'_{bi} + e'_s) = 727.3 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 2\,335.4 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{p_b} = 0.5d_{ib}(l'_b + e_{as}) = 2\,416.4 \text{ mm}^2$$

$$A_{p\phi} = 0 \text{ mm}^2$$

$$A_{fp} = 0 \text{ mm}^2$$

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Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	532 197 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	259 661 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	21.24 MPa
			Fr ≥ Fp: Ok

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	15.29 MPa
Overpressure due to static head	Ph	=	0.03 MPa
Calculation pressure	P = Pt + Ph	=	15.31 MPa

Longitudinal section

	$a = \frac{d_{cb}}{2}$	=	50.83 mm
Inside radius of curvature of the shell at the opening centre	$r_{is} = \frac{D_e}{2} - e_{as}$	=	310.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{[(D_e - 2e_{as}) + e_{cs}]e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	94.17 mm
Pressure loaded area - shell	$Ap_s = r_{is}(l'_s + a)$	=	44 950.0 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	767.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 636.8 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	2 352.9 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	874 826 N
Pressure load	$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi)$	=	724 392 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	18.38 MPa
			Fr ≥ Fp: Ok

Transverse section

	$\delta = \frac{d_{cb}}{2r_{ms}}$	=	0.15688
	$a = r_{ms} \arcsin(\delta)$	=	51.04 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	324.00 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	134.70 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_s]$	=	94.17 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	21 535.1 mm ²
Length of penetration into shell wall	$e's = e_{a,s}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b(l'_b + l'_{bi} + e'_s)$	=	767.3 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	2 636.8 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib}(l'_b + e_{as})$	=	2 352.9 mm ²
Additional area due to obliquity of the nozzle	Ap_φ	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²

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Reactive force

$$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P) =$$

874 826 N

Pressure load

$$F_p = P(Ap_s + Ap_b + 0.5Ap_\varphi) =$$

365 819 N

Maximum allowable pressure

$$P_{\max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap_\varphi) + 0.5(Af_s + Af_w + Af_b + Af_p)} =$$

35.20 MPa

Fr ≥ Fp: Ok

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Nozzle - Trubka-vstup vody*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Material: 11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383

Nominal design stress at internal design temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress at room temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4}) =$	225.00 MPa

Geometry

Inside diameter	Di =	76.66 mm
Outside diameter	De =	101.66 mm
Length	L =	300.00 mm
Nominal thickness	en =	12.50 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	83.06 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	5.57 mm
e/De ≤ 0,16 (0.05500 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	18.38 MPa
Maximum allowable design pressure	=	11.11 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	14.020 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.04651
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.98 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.006 MPa
Calculation pressure	Pc=Pt+Pht =	15.29 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	76.66 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	2.70 mm
e/De ≤ 0,16 (0.02700 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-vstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11*

Flange material 13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($70 \leq t \leq 90$) - No.: 1.7335
Shell material 11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383
Bolting material 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218
Gasket Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	80
Number of bolts	=	8
Bolt type	=	ISO M24 x 3.00
Material group	=	5E0

Calculation temperature	T =	121.11 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa

Bolt loads

Number of bolts	n =	8
Bolt type	=	ISO M24 x 3.00
Root area of one bolt	=	324.0 mm ²
Distance between centre lines of adjacent bolts	δb =	70.69 mm
Bolt effective diameter	db =	21.19 mm

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.006 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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Nozzle - Trubka-výstup vody*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Material: 11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383

Nominal design stress at internal design temperature	$f = \min(f_{nc}; \frac{R_m/T_t}{1.5}; R_{p1/T_t}) =$	192.67 MPa
Nominal design stress at room temperature	$f = \min(f_{nc}; \frac{R_m/T_t}{1.5}; R_{p1/T_t}) =$	192.67 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_m/20}{2.4}) =$	225.00 MPa

Geometry

Inside diameter	Di =	76.66 mm
Outside diameter	De =	101.66 mm
Length	L =	300.00 mm
Nominal thickness	en =	12.50 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	83.06 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	5.57 mm
e/De ≤ 0,16 (0.05500 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	18.36 MPa
Maximum allowable design pressure	=	11.11 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	14.020 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.04651
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.98 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.03 MPa
Calculation pressure	Pc=Pt+Pht =	15.31 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	76.66 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	2.70 mm
e/De ≤ 0,16 (0.02700 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-výstup plynu*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11*

Flange material 13CrMo4-5 (NT,QT) (EN 10222-2) - Forging ($70 \leq t \leq 90$) - No.: 1.7335
Shell material 11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383
Bolting material 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218
Gasket Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	80
Number of bolts	=	8
Bolt type	=	ISO M24 x 3.00
Material group	=	5E0

Calculation temperature	T =	121.11 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa

Bolt loads

Number of bolts	n =	8
Bolt type	=	ISO M24 x 3.00
Root area of one bolt	=	324.0 mm ²
Distance between centre lines of adjacent bolts	δb =	70.69 mm
Bolt effective diameter	db =	21.19 mm

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.03 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa

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U-Tube tubesheet - Trubkovnice*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 13***Design conditions**

	Design temperature	Design pressure
Shell	T _s = 121.11 °C / 250.00 °F	P _s = 10.69 MPa / 1 550.5 psi
Channel	T _c = 37.78 °C / 100.00 °F	P _t = 10.69 MPa / 1 550.5 psi
Tubesheet	T = 121.11 °C / 250.00 °F	
Tubes	T _t = 121.11 °C / 250.00 °F	

Tubesheet material 13CrMo4-5 (NT,QT) (EN 10222-2)

Tubesheet design temperature

T = 121.11 °C

Modulus of elasticity for tubesheet material at T

E = 198 193.42 MPa

Nominal design stress of tubesheet material at T

f = 162.44 MPa

Modulus of elasticity for tubesheet material at test temperature

E_t = 204 842.00 MPa

Allowable stress for tubesheet material for hydraulic test

f_h = 242.86 MPa**Tubes material 13CrMo4-5 (EN 10216-2)**

Tube design temperature

T_t = 121.11 °CModulus of elasticity for tube material at T_tE_t = 198 193.42 MPaNominal design stress of tube material at T_tf_t = 172.90 MPa

Modulus of elasticity for tube material at test temperature

E_{t,t} = 204 842.00 MPa**Channel material 13CrMo4-5 (EN 10028-2)**

Channel design temperature

T_c = 37.78 °CModulus of elasticity for channel material at T_cE_c = 203 548.34 MPa

Poisson's ratio of channel material

ν_c = 0.30Nominal design stress of channel material at T_cf_c = 187.50 MPa

Modulus of elasticity for channel material at test temperature

E_{c,t} = 204 842.00 MPa

Allowable stress for channel material for hydraulic test

f_{ch} = 276.19 MPa**Shell material 13CrMo4-5 (EN 10028-2)**

Shell design temperature

T_s = 121.11 °CModulus of elasticity for shell material at T_sE_s = 198 193.42 MPa

Poisson's ratio of shell material

ν_s = 0.30Nominal design stress of shell material at T_sf_s = 179.11 MPa

Modulus of elasticity for shell material at test temperature

E_{s,t} = 204 842.00 MPa

Allowable stress for shell material for hydraulic test

f_{sh} = 276.19 MPa

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Geometric data

Tubesheet configuration	=	d) Gasketed with shell and channel
Outside diameter of tubesheet	A =	860.00 mm
Shell corrosion allowance	cs =	3.20 mm
Shell undertolerance	δs =	0 mm
Shell thickness	$e_s = e_{\text{shell}} - c_s - \delta_s =$	24.80 mm
Channel corrosion allowance	cc =	3.20 mm
Channel undertolerance	δc =	0 mm
Channel thickness	$e_c = e_{\text{channel}} - c_c - \delta_c =$	24.80 mm
Tubeside corrosion allowance	c_ts =	3.20 mm
Shellside corrosion allowance	c_ss =	3.20 mm
Tubesheet undertolerance	δ =	0 mm
Tubesheet thickness	e_tubesheet =	133.35 mm
Tubesheet thickness for calculation	$e = e_{\text{tubesheet}} - c_{ts} - c_{ss} - \delta =$	126.95 mm
Nominal outside diameter of tubes	dt =	40.00 mm
Radius to outermost tube hole center	ro =	167.49 mm
Equivalent diameter of outer tube limit circle	$D_0 = 2r_0 + d_t =$	374.98 mm
Triangular tube pitch	p =	50.00 mm
Nominal tube wall thickness	et =	2.60 mm
Tube side pass partition groove depth	hg =	0 mm
Effective tube side pass partition groove depth	$h'_g = \max[(h_g - c_t), 0.0] =$	0 mm
Expanded length of tube in tubesheet	ltx =	0 mm
Total area of untubed lanes	S =	0 mm ²
Largest centre-to-centre distance between adjacent tube rows	UL =	0 mm
Basic ligament efficiency of perforated tubesheet for shear	$\mu = \frac{p - d_t}{p} =$	0.20000
Effective tube pitch	p* = p =	50.00 mm
Tube expansion depth ratio	$\rho = \frac{l_{tx}}{h} =$	0
Effective tube hole diameter	$d^* = \max\left[\left\{d_t - 2e_t\left(\frac{E_t}{E}\right)\left(\frac{f_t}{f}\right)\rho\right\}, (d_t - 2e_t)\right] =$	40.00 mm
Effective ligament efficiency of perforated tubesheet for bending	$\mu^* = \frac{p - d^*}{p^*} =$	0.20000
	α0 =	-0.00290
	α1 =	0.21260
	α2 =	3.99060
	α3 =	-6.17300
	α4 =	3.43070
	β0 =	0.99660
	β1 =	-4.19780
	β2 =	9.04780
	β3 =	-7.99550
	β4 =	2.23980
	$E^*/E = \alpha_0 + \alpha_1\mu^* + \alpha_2(\mu^*)^2 + \alpha_3(\mu^*)^3 + \alpha_4(\mu^*)^4 =$	0.15535
Effective Poisson ratio of perforated tubesheet	$\nu^* = \beta_0 + \beta_1\mu^* + \beta_2(\mu^*)^2 + \beta_3(\mu^*)^3 + \beta_4(\mu^*)^4 =$	0.45857
Effective elastic modulus of perforated tubesheet at design temperature	E* =	30 789.17 MPa
		UL ≤ 4p: Ok

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Tube to tubesheet joints

Weld throat thickness	at	=	2.00 mm
Nominal tube wall thickness	et	=	2.60 mm
Minimum nominal design stress of tubesheet or tubes material	fmin = min(f; ft)	=	162.44 MPa
Maximum permissible stress of the tube-to-tubesheet joint	$f_{t,j} = \min \left[f_{\min} \cdot \frac{a_t}{e_t}, f_t \right]$	=	124.96 MPa

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Loading case 1: Pt = 10.69 MPa, Ps = 0 MPa, thermal exp.: N, corr.: Y, vacuum: N

Diameter of shell gasket load reaction	Gs =	637.15 mm
Diameter of channel gasket load reaction	Gc =	637.15 mm
Shell diameter ratio	ps = Gs / Do =	1.69917
Channel diameter ratio	pc = Gc / Do =	1.69917
Tubesheet diameter ratio	K = A / Do =	2.29347
	$F = \frac{(1 - \nu^*) (E \cdot \ln[K])}{E^*}$ =	2.89297
Channel flange design bolt load for the gasket seating condition	Wc =	3 979 511 N
Shell flange design bolt load for the gasket seating condition	Ws =	4 034 731 N
Maximum flange design bolt load for the assembly condition	Wmax = Max[Ws; Wc] =	4 034 731 N

Tubesheet bending stress

Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$ =	-255 320 N
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \frac{W_{max}(G_c - G_s)}{2\pi D_o}$ =	-255 320 N
Moment acting at periphery of tubesheet	$M_p = \frac{M^* - \frac{D_o^2}{32} F (P_s - P_t)}{1 + F}$ =	-30 679 N
Moment acting at centre of tubesheet	$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t)$ =	-111 907 N
Maximum bending moment acting on the tubesheet	$M = \max[M_p , M_o]$ =	111 907 N
Maximum radial bending stress in the tubesheet	$\sigma = \frac{6M}{\mu^* (e - h'_g)^2}$ =	208.31 MPa
	$\sigma \leq 2f$: Ok	

Tubesheet shear stress

Maximum shear stress in the tubesheet	$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{D_o}{e} \right) \cdot P_s - P_t $ =	39.47 MPa
	$\tau \leq 0.8f$: Ok	

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Loading case 2: Pt = 0 MPa, Ps = 10.69 MPa, thermal exp.: N, corr.: Y, vacuum: N

Diameter of shell gasket load reaction	Gs =	637.15 mm
Diameter of channel gasket load reaction	Gc =	637.15 mm
Shell diameter ratio	ps = Gs / Do =	1.69917
Channel diameter ratio	pc = Gc / Do =	1.69917
Tubesheet diameter ratio	K = A / Do =	2.29347
	$F = \frac{(1 - \nu^*) (E \cdot \ln[K])}{E^*}$ =	2.89297
Channel flange design bolt load for the gasket seating condition	Wc =	3 979 511 N
Shell flange design bolt load for the gasket seating condition	Ws =	4 034 731 N
Maximum flange design bolt load for the assembly condition	Wmax = Max[Ws; Wc] =	4 034 731 N

Tubesheet bending stress

Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$ =	255 320 N
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \frac{W_{max}(G_c - G_s)}{2\pi D_o}$ =	255 320 N
Moment acting at periphery of tubesheet	$M_p = \frac{M^* - \frac{D_o^2}{32} F (P_s - P_t)}{1 + F}$ =	30 679 N
Moment acting at centre of tubesheet	$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t)$ =	111 907 N
Maximum bending moment acting on the tubesheet	$M = \max[M_p , M_o]$ =	111 907 N
Maximum radial bending stress in the tubesheet	$\sigma = \frac{6M}{\mu^* (e - h'_g)^2}$ =	208.31 MPa
	$\sigma \leq 2f$: Ok	

Tubesheet shear stress

Maximum shear stress in the tubesheet	$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{D_o}{e} \right) \cdot P_s - P_t $ =	39.47 MPa
	$\tau \leq 0.8f$: Ok	

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Loading case 3: Pt = 10.69 MPa, Ps = 10.69 MPa, thermal exp.: N, corr.: Y, vacuum: N

Diameter of shell gasket load reaction	Gs =	637.15 mm
Diameter of channel gasket load reaction	Gc =	637.15 mm
Shell diameter ratio	$\rho_s = G_s / D_o =$	1.69917
Channel diameter ratio	$\rho_c = G_c / D_o =$	1.69917
Tubesheet diameter ratio	$K = A / D_o =$	2.29347
	$F = \frac{(1 - \nu^*) (E \cdot \ln[K])}{E^*} =$	2.89297
Channel flange design bolt load for the gasket seating condition	Wc =	3 979 511 N
Shell flange design bolt load for the gasket seating condition	Ws =	4 034 731 N
Maximum flange design bolt load for the assembly condition	Wmax = Max[Ws; Wc] =	4 034 731 N

Tubesheet bending stress

Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16} =$	0 N
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \frac{W_{\max}(G_c - G_s)}{2\pi D_o} =$	0 N
Moment acting at periphery of tubesheet	$M_p = \frac{M^* - \frac{D_o^2}{32} F (P_s - P_t)}{1 + F} =$	0 N
Moment acting at centre of tubesheet	$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t) =$	0 N
Maximum bending moment acting on the tubesheet	$M = \max[M_p , M_o] =$	0 N
Maximum radial bending stress in the tubesheet	$\sigma = \frac{6M}{\mu^* (e - h'_g)^2} =$	0 MPa
		$\sigma \leq 2f$: Ok

Tubesheet shear stress

Maximum shear stress in the tubesheet	$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{D_o}{e} \right) \cdot P_s - P_t =$	0 MPa
		$\tau \leq 0.8f$: Ok

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Loading case Hydrostatic test tubes side: $P_t = 15.32$ MPa, $P_s = 0$ MPa, thermal exp.: N, corr.: N, vacuum: N

Tubesheet bending stress

Moment acting on the unperforated tubesheet rim

$$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16} = -365\,853 \text{ N}$$

Moment acting on the unperforated tubesheet rim

$$M^* = M_{TS} + \frac{W_{\max}(G_c - G_s)}{2\pi D_o} = -365\,853 \text{ N}$$

Moment acting at periphery of tubesheet

$$M_p = \frac{M^* - \frac{D_o^2}{32} F(P_s - P_t)}{1 + F} = -43\,960 \text{ N}$$

Moment acting at centre of tubesheet

$$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t) = -160\,353 \text{ N}$$

Maximum bending moment acting on the tubesheet

$$M = \max[|M_p|, |M_o|] = 160\,353 \text{ N}$$

Maximum radial bending stress in the tubesheet

$$\sigma = \frac{6M}{\mu^* (e - h_g')^2} = 270.53 \text{ MPa}$$

$\sigma \leq 2f$: Ok

Tubesheet shear stress

Maximum shear stress in the tubesheet

$$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{D_o}{e} \right) \cdot |P_s - P_t| = 53.84 \text{ MPa}$$

$|\tau| \leq 0.8f$: Ok

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Loading case Hydrostatic test shell side: Pt = 0 MPa, Ps = 15.32 MPa, thermal exp.: N, corr.: N, vacuum: N

Tubesheet bending stress

Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	365 853 N
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \frac{W_{\max}(G_c - G_s)}{2\pi D_o}$	=	365 853 N
Moment acting at periphery of tubesheet	$M_p = \frac{M^* - \frac{D_o^2}{32} F(P_s - P_t)}{1 + F}$	=	43 960 N
Moment acting at centre of tubesheet	$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t)$	=	160 353 N
Maximum bending moment acting on the tubesheet	$M = \max[M_p , M_o]$	=	160 353 N
Maximum radial bending stress in the tubesheet	$\sigma = \frac{6M}{\mu^* (e - h_g')^2}$	=	270.53 MPa
			$\sigma \leq 2f$: Ok

Tubesheet shear stress

Maximum shear stress in the tubesheet	$\tau = \left(\frac{1}{4\mu} \right) \left(\frac{D_o}{e} \right) \cdot P_s - P_t $	=	53.84 MPa
			$\tau \leq 0.8f$: Ok

Maximum allowable pressures (at the top of the vessel)

Shell side New & Cold	=	19.25 MPa
Shell side Hot & Corroded	=	19.25 MPa
Tube side New & Cold	=	16.67 MPa
Tube side Hot & Corroded	=	16.67 MPa

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Tube bundle - Svazek trubek*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
External design temperature	Te =	121.11 °C
External design pressure	Pe =	10.69 MPa
Joint efficiency	z =	1.00

Material: 13CrMo4-5 (EN 10216-2) - Seamless tube (t ≤ 40.00 mm) - No.: 1.7335

Nominal design stress at design temperature	$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_{m/20}}{2.4}\right) =$	172.90 MPa
Nominal design stress at room temperature	$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_{m/20}}{2.4}\right) =$	183.33 MPa
Nominal design stress in test condition	$f_{test} = \left(\frac{R_{p0.2/T_{test}}}{1.05}\right) =$	276.19 MPa

Geometry

Inside diameter	Di =	34.80 mm
Outside diameter	De =	40.00 mm
Length	L =	2 380.00 mm
Nominal thickness	en =	2.60 mm
Corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	34.80 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	1.11 mm
e/De ≤ 0,16 (0.02800 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	38.38 MPa
Maximum allowable design pressure	=	24.04 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	6.952 %
Radius of curvature	R =	60.00 mm
Ratio of deformation of tube bends	F=100·De/2R =	33.333 %

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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.00000
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.02 MPa
Calculation pressure	Pc=Pt+Pht =	15.31 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	34.80 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	0.99 mm
e/De ≤ 0,16 (0.02500 ≤ 0.16000): Ok		
en ≥ e: Ok		

External pressure

Overpressure due to static head	Phe =	0 MPa
Calculation pressure	P=Pe+Phe =	10.69 MPa
Unsupported length of shell	L =	2 246.65 mm
Elastic modulus	E =	198 193.42 MPa
Proof strength	σe = Rp0.2/t =	259.36 MPa
Poisson ratio	ν =	0.30000
Mean radius	R=(Di+en+c+δ)/2 =	18.70 mm
Z factor	Z=πR/L =	0.02615
Number of circumferential waves	ncyl =	2.00000
	$\varepsilon = \frac{1}{n_{cyl}^2 - 1 + \frac{Z^2}{2}} \left\{ \frac{1}{\left(\frac{n_{cyl}^2}{Z^2} + 1\right)^2} + \frac{e_a^2}{12R^2(1-\nu^2)}(n_{cyl}^2 - 1 + Z^2)^2 \right\} =$	0.00531
Yield point pressure	Py=[σe*(en-c-ce-δ)]/R =	36.06 MPa
Theoretical elastic instability pressure	Pm=[E*(en-c-ce-δ)*ε]/R =	146.40 MPa
Pm/Py ratio	Pm/Py =	4.06 MPa
Pr/Py ratio	Pr/Py =	0.88 MPa
Lower bound collapse pressure	Pr=(Pr/Py)*Py =	31.77 MPa
Safety factor	S =	1.50000
	Pr/S =	21.18 MPa

Minimum required thickness for external pressure - use greater of:

Minimum required thickness according to external pressure procedure	e =	1.65 mm
Minimum required thickness for internal pressure with z=1	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	1.11 mm
P < Pr/S: Ok		

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TEMA RCB Requirements

Outside diameter	do =	40.00 mm
Mean radius of bend	R =	60.00 mm
Required tube wall thickness prior to bending	$t_0 = t_r \left[1 + \frac{d_o}{4R} \right]$ =	1.93 mm
Outside diameter of the tube	od =	40.00 mm
Tube pitch	Pitch =	50.00 mm
Minimum tube pitch	Pitch(TEMA) = 1.25 * od =	50.00 mm
	t ≥ t0: Ok	
	Pitch ≥ Pitch(TEMA): Ok	
	en ≥ e: Ok	

Maximum allowable external pressures

Maximum allowable test pressure	=	31.89 MPa
Maximum allowable design pressure	=	21.18 MPa

Hydrostatic test (external pressure)

Item or side minimum allowables ratio	Item f0/f =	1.04651
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.98 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.005 MPa
Calculation pressure	Pc=Pt+Pht =	15.29 MPa
Unsupported length of shell	L =	2 246.65 mm
Elastic modulus	E =	204 842.00 MPa
Proof strength	σe = Rp0.2/t =	290.00 MPa
Poisson ratio	ν =	0.30000
Mean radius	R=(Di+en+δ)/2 =	18.70 mm
Z factor	Z=πR/L =	0.02615
Number of circumferential waves	ncyl =	2.00000

$$\varepsilon = \frac{1}{n_{cyl}^2 - 1 + \frac{Z^2}{2}} \left\{ \frac{1}{\left(\frac{n_{cyl}^2}{Z^2} + 1 \right)^2} + \frac{e_a^2}{12R^2(1-\nu^2)} (n_{cyl}^2 - 1 + Z^2)^2 \right\} = 0.00531$$

Yield point pressure	Py=[σe*(en-c-ce-δ)]/R =	40.32 MPa
Theoretical elastic instability pressure	Pm=[E*(en-c-ce-δ)*ε]/R =	151.31 MPa
Pm/Py ratio	Pm/Py =	3.75 MPa
Pr/Py ratio	Pr/Py =	0.87 MPa
Lower bound collapse pressure	Pr=(Pr/Py)*Py =	35.08 MPa
Safety factor	S =	1.10000
	Pr/S =	31.89 MPa
Minimum required thickness according to external pressure procedure	e =	1.62 mm
Minimum required thickness for internal pressure with z=1	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	0.99 mm

P < Pr/S: Ok**en ≥ e: Ok**

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Torispherical head - Torosferické dno pláště*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature

Ti = 121.11 °C

Internal design pressure

Pi = 10.69 MPa

Joint efficiency

z = 1.00

Material: 13CrMo4-5 (EN 10028-2) - Plate (16,001 ≤ t ≤ 60) - No.: 1.7335

Nominal design stress at internal design temperature

$$f = \min\left(\frac{R_{p0.2/T}}{1.5}, \frac{R_{m/20}}{2.4}\right) = 179.11 \text{ MPa}$$

Nominal design stress at room temperature

$$f = \min\left(\frac{R_{p0.2/20}}{1.5}, \frac{R_{m/20}}{2.4}\right) = 187.50 \text{ MPa}$$

Nominal design stress in test condition

$$f_{\text{test}} = \left(\frac{R_{p0.2/T_{\text{test}}}}{1.05}\right) = 276.19 \text{ MPa}$$

Geometry

Inside diameter

Di = 620.00 mm

Outside diameter

De = 676.00 mm

Head outside height

H = 267.69 mm

Nominal thickness

en = 28.00 mm

Minimum head thickness after forming

t-c' = 28.00 mm

Corrosion allowance

c = 3.20 mm

External corrosion allowance

ce = 0 mm

Undertolerance

δ = 0 mm

Straight flange length

l(sf) = 84.00 mm

Straight flange undertolerance

δ(sf) = 0 mm

Straight flange thickness

en(sf) = 28.00 mm

Straight flange joint efficiency

z(sf) = 1.00000

Knuckle thickness

en(k) = 28.00 mm

Inside spherical radius of central part of torispherical head

R = 540.80 mm

Inside knuckle radius

r = 104.10 mm

Internal pressure

Overpressure due to static head

Ph = 0 MPa

Calculation pressure

P=Pi+Ph = 10.69 MPa

Parameter Y

Y=min(ec/R;0.04) = 0.03112

Parameter Z

Z=log10(1/Y) = 1.50702

Ratio X

X=r/Di = 0.17130

Parameter N

$$N = 1006 - \frac{1}{[62 + (90Y)^4]} = 0.99123$$

Parameter β(0.1)

$$\beta_{0.1} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837) = 0.61178$$

Parameter β(0.2)

$$\beta_{0.2} = \max[0.95(0.56 - 194Y - 82.5Y^2), 0.5] = 0.50000$$

Parameter β

$$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}] = 0.53208$$

Joint efficiency

z = 1.00000

Inside spherical radius of central part of torispherical head

R'=R+c = 544.00 mm

Inside diameter

Di'=Di+2·c = 626.40 mm

Inside knuckle radius

r'=r+c = 107.30 mm

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Required thickness of end to limit membrane stress in central part

$$e_s = \frac{PR'}{2fZ - 0.5P} + c + ce + \delta = 19.68 \text{ mm}$$

Required thickness of knuckle to avoid axisymmetric yielding

$$e_y = \frac{\beta P(0.75R + 0.2D_i)}{f} + c + ce + \delta = 20.13 \text{ mm}$$

Minimum required thickness

$$e = \max(e_y, e_s) = 20.13 \text{ mm}$$

Straight flange minimum required thickness

$$e(sf) = 22.47 \text{ mm}$$

en(sf) ≥ e(sf): Ok**en ≥ e: Ok****Maximum allowable pressures (at the top of the vessel)**

Maximum allowable test pressure

$$= 27.87 \text{ MPa}$$

Maximum allowable design pressure

$$= 15.97 \text{ MPa}$$

Deformation according to EN13445-4 Clause 9

Manufactured in one piece (9.2.1)

$$F(1) = 100 \cdot \ln[(1.11 \cdot De)/(De - 2 \cdot en)] = 19.083 \%$$

Spherical part (9.2.1)

$$F(2) = 100 \cdot \ln\{2 \cdot R \cdot \sin[(0.4 \cdot De/R)/(0.8 \cdot De - 2 \cdot en)]\} = 15.543 \%$$

Segments deformation of multi-sectional torispherical heads or spheres (9.2.5)

$$F(3) = (100 \cdot en)/(r + en/2) = 23.708 \%$$

Hydrostatic test

Item or side minimum allowables ratio

$$\text{Item } f_0/f = 1.04651$$

Coincident design pressure for the maximum pressure load case

$$P_d = 10.69 \text{ MPa}$$

Test pressure as per EN13445-5 formula 10.2.3.3.1-1

$$P_{t1} = 1.25 \cdot P_d \cdot (\text{Item } f_0/f) = 13.98 \text{ MPa}$$

Test pressure as per EN13445-5 formula 10.2.3.3.1-2

$$P_{t2} = 1.43 \cdot P_d = 15.29 \text{ MPa}$$

Item or side hydrostatic test pressure

$$P_t = \max(P_{t1}, P_{t2}) = 15.29 \text{ MPa}$$

Overpressure due to static head in test condition

$$P_{ht} = 0.004 \text{ MPa}$$

Calculation pressure

$$P_c = P_t + P_{ht} = 15.29 \text{ MPa}$$

Joint efficiency

$$Z = 1.00000$$

Parameter Y

$$Y = \min(ec/R; 0.04) = 0.02929$$

Parameter Z

$$Z = \log_{10}(1/Y) = 1.53334$$

Ratio X

$$X = r/D_i = 0.16791$$

Parameter N

$$N = 1006 - \frac{1}{[6.2 + (90Y)^4]} = 0.98764$$

Parameter β_{01}

$$\beta_{01} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837) = 0.62494$$

Parameter β_{02}

$$\beta_{02} = \max[0.95(0.56 - 1.94Y - 82.5Y^2), 0.5] = 0.50000$$

Parameter β

$$\beta = 10[(0.2 - X)\beta_{01} + (X - 0.1)\beta_{02}] = 0.54009$$

Inside spherical radius of central part of torispherical head

$$R' = R = 540.80 \text{ mm}$$

Inside diameter

$$D_i' = D_i = 620.00 \text{ mm}$$

Inside knuckle radius

$$r' = r = 104.10 \text{ mm}$$

Required thickness of end to limit membrane stress in central part

$$e_s = \frac{PR'}{2fZ - 0.5P} + \delta = 15.18 \text{ mm}$$

Required thickness of knuckle to avoid axisymmetric yielding

$$e_y = \frac{\beta P(0.75R + 0.2D_i)}{f} + \delta = 15.84 \text{ mm}$$

Minimum required thickness

$$e = \max(e_y, e_s) = 15.84 \text{ mm}$$

en ≥ e: Ok

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Reinforcement of opening - Trubka-odvzdušnění*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 9***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Shell material: 13CrMo4-5 (EN 10028-2) - Plate

Nominal design stress of shell material	fs =	179.11 MPa
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Nozzle material: 11CrMo9-10 (EN 10216-2) - Seamless tube

Nominal design stress of the nozzle material	fb =	192.67 MPa
	fob = min(fs, fb) =	179.11 MPa

Nozzle geometry

Nozzle connection	=	Set in
Nozzle position	=	Radial
Fatigue assessed using Clause 17 and opening is a critical area	=	No
Offset k between nozzle and shell axis	=	0 mm
Angular offset	=	0 °
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum width of the opening on shell without nozzle	d =	51.00 mm
Internal diameter	d_ib = Id + 2(c + δ) =	47.40 mm
External diameter of the nozzle	d_eb = Od - 2ce =	51.00 mm
Nominal thickness of the nozzle	e_ab =	5.00 mm
Length of nozzle extending outside the shell	l_b =	122.00 mm
Analysis thickness of the nozzle	e_b = e_ab - c - ce - δ =	1.80 mm
	$l_{bo(max)} = \sqrt{(d_{eb} - e_b) e_b}$ =	15.17 mm
	$l_{bo} = \min(l_b, l_{bo(max)})$ =	15.17 mm
	$l'_b = \min(l_{bo}, l_b)$ =	15.17 mm
	$l'_{bi} = \min(l_{bi}, 0, 5l_{bo})$ =	0 mm
Effective length of nozzle outside the shell for reinforcement		
Effective length of nozzle inside the shell for reinforcement		
Stress loaded cross-sectional area effective as reinforcement - welds	Afw =	0 mm ²

ea,b / ea,s ≤ 3: Ok**Pad geometry**

Effective width of reinforcing plate for reinforcement	lp' = min(l_so, l_p) =	0 mm
Stress loaded cross-sectional area effective as reinforcement - pad	Af _p = e _p · l _p ' =	0 mm ²

Shell geometry

Analysis thickness of shell wall	$e_{as} = t_{shell} - c_{shell} - \delta_{shell}$ =	24.80 mm
Analysis thickness of shell wall	e_cs =	24.80 mm
Head inside radius	R =	540.80 mm
Inside radius of curvature of the shell at the opening centre	r_is = R + cs + cs' =	544.00 mm

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Check of distance from shell discontinuity

Maximum length of shell contributing to opening reinforcement

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

Head's analysis thickness

$$e_k = 24.80 \text{ mm}$$

Head's knuckle radius

$$r_k = 107.30 \text{ mm}$$

Distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w(7.7.2) = 392.79 \text{ mm}$$

Limit distance between nozzle edge and knuckle-shell tangent line as per 7.7.2

$$w_{\min}(7.7.2) = 2.5 \cdot \sqrt{(e_k \cdot r_k)} = 128.97 \text{ mm}$$

Distance between an opening and a shell discontinuity

$$w = 256.86 \text{ mm}$$

Length of shell from the edge of the opening to a shell discontinuity

$$l_s = w = 256.86 \text{ mm}$$

Minimum value for w which has no influence on ls from shell discontinuities

$$w_p = l_{so} = 166.12 \text{ mm}$$

Required minimum value for w

$$w_{\min} = 0 \text{ mm}$$

w ≥ w_min: Ok**Internal pressure**

Overpressure due to static head

$$P_h = 0 \text{ MPa}$$

Calculation pressure

$$P = P_i + P_h = 10.69 \text{ MPa}$$

Transverse section

Mean radius of curvature

$$\delta = \frac{d_{eb}}{2r_{ms}} = 0.04583$$

Maximum length of shell contributing to opening reinforcement

$$a = r_{ms} \arcsin(\delta) = 25.51 \text{ mm}$$

Effective length of shell for opening reinforcement

$$r_{ms} = r_{is} + 0.5e_{as} = 556.40 \text{ mm}$$

$$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}} = 166.12 \text{ mm}$$

$$l'_s = \min[l_{so}, l_s] = 166.12 \text{ mm}$$

Pressure loaded area - shell

$$A_{p_s} = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}} = 50962.7 \text{ mm}^2$$

Length of penetration into shell wall

$$e'_s = e_{a,s} = 24.80 \text{ mm}$$

Stress loaded cross-sectional area effective as reinforcement - nozzle

$$A_{f_b} = e_b \cdot (l'_b + l'_{bi} + e'_s) = 199.8 \text{ mm}^2$$

Stress loaded cross-sectional area - shell

$$A_{f_s} = (l'_s + e_{ab} - c_{shell})e_{cs} = 4119.9 \text{ mm}^2$$

Pressure loaded area - nozzle

$$A_{p_b} = 0.5d_{ib} \cdot (l'_b + e_{as}) = 947.2 \text{ mm}^2$$

Additional area due to obliquity of the nozzle

$$A_{p\varphi} = 0 \text{ mm}^2$$

Stress loaded cross-sectional area effective as reinforcement - pad

$$A_{f_p} = 0 \text{ mm}^2$$

Reactive force

$$F_r = (A_{f_s} + A_{f_w})(f_s - 0.5P) + A_{f_p}(f_{op} - 0.5P) + A_{f_b}(f_{ob} - 0.5P) = 750622 \text{ N}$$

Pressure load

$$F_p = P(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) = 554917 \text{ N}$$

Maximum allowable pressure

$$P_{\max} = \frac{(A_{f_s} + A_{f_w}) \cdot f_s + A_{f_b} \cdot f_{ob} + A_{f_p} \cdot f_{op}}{(A_{p_s} + A_{p_b} + 0.5A_{p\varphi}) + 0.5(A_{f_s} + A_{f_w} + A_{f_b} + A_{f_p})} = 14.31 \text{ MPa}$$

Fr ≥ Fp: Ok**Hydrostatic test**

Item or side hydrostatic test pressure

$$P_t = 15.29 \text{ MPa}$$

Overpressure due to static head

$$P_h = 0.002 \text{ MPa}$$

Calculation pressure

$$P = P_t + P_h = 15.29 \text{ MPa}$$

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Transverse section

	$\delta = \frac{d_{eb}}{2r_{ms}}$	=	0.04596
	$a = r_{ms} \arcsin(\delta)$	=	25.51 mm
Mean radius of curvature	$r_{ms} = r_{is} + 0.5e_{as}$	=	554.80 mm
Maximum length of shell contributing to opening reinforcement	$l_{so} = \sqrt{(2r_{is} + e_{cs})e_{cs}}$	=	176.26 mm
Effective length of shell for opening reinforcement	$l'_s = \min[l_{so}, l_{so}]$	=	176.26 mm
Pressure loaded area - shell	$Ap_s = 0.5r_{is}^2 \frac{l'_s + a}{0.5e_{as} + r_{is}}$	=	53 182.5 mm ²
Length of penetration into shell wall	$e's = e_{as}$	=	28.00 mm
Stress loaded cross-sectional area effective as reinforcement - nozzle	$Af_b = e_b \cdot (l'_b + l'_{bi} + e'_s)$	=	215.8 mm ²
Stress loaded cross-sectional area - shell	$Af_s = (l'_s + e_{ab} - c_{shell})e_{cs}$	=	4 935.4 mm ²
Pressure loaded area - nozzle	$Ap_b = 0.5d_{ib} \cdot (l'_b + e_{as})$	=	884.9 mm ²
Additional area due to obliquity of the nozzle	$Ap\phi$	=	0 mm ²
Stress loaded cross-sectional area effective as reinforcement - pad	Afp	=	0 mm ²
Reactive force	$F_r = (Af_s + Af_w)(f_s - 0.5P) + Af_p(f_{op} - 0.5P) + Af_b(f_{ob} - 0.5P)$	=	1 372 287 N
Pressure load	$Fp = P(Ap_s + Ap_b + 0.5Ap\phi)$	=	826 641 N
Maximum allowable pressure	$P_{max} = \frac{(Af_s + Af_w) \cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0.5Ap\phi) + 0.5(Af_s + Af_w + Af_b + Af_p)}$	=	24.92 MPa
			Fr ≥ Fp: Ok

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Nozzle - Trubka-odvzdušnění*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8***Design data**

Internal design temperature	Ti =	121.11 °C
Internal design pressure	Pi =	10.69 MPa
Joint efficiency	z =	1.00
Specified lifetime (in creep range)	=	100'000 Hours
Weld creep strength reduction factor	zc =	1.00
Access or inspection opening	=	No

Material: 11CrMo9-10 (EN 10216-2) - Seamless tube (t ≤ 60.00 mm) - No.: 1.7383

Nominal design stress at internal design temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress at room temperature	$f = \min(f_{nc} \cdot \frac{R_{m/Tt}}{1.5}; R_{p1/Tt}) =$	192.67 MPa
Nominal design stress in test condition	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4}) =$	225.00 MPa

Geometry

Inside diameter	Di =	41.00 mm
Outside diameter	De =	51.00 mm
Length	L =	150.00 mm
Nominal thickness	en =	5.00 mm
Corrosion allowance	c =	3.20 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	10.69 MPa
Inside diameter	Di'=Di+2δ+2c =	47.40 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta =$	4.55 mm
e/De ≤ 0,16 (0.08900 ≤ 0.16000): Ok		
en ≥ e: Ok		

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	24.92 MPa
Maximum allowable design pressure	=	14.10 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	10.870 %
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Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.04651
Coincident design pressure for the maximum pressure load case	Pd =	10.69 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	13.98 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	15.29 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	15.29 MPa
Overpressure due to static head in test condition	Pht =	0.002 MPa
Calculation pressure	Pc=Pt+Pht =	15.29 MPa
Joint efficiency	z =	1.00000
Inside diameter	Di'=Di+2δ =	41.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta =$	1.44 mm
e/De ≤ 0,16 (0.02800 ≤ 0.16000): Ok		
en ≥ e: Ok		

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Standard Welding neck flange - Příruba-odvzdušnění*According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11***Flange material** 13CrMo4-5 (NT) (EN 10222-2) - Forging ($35,001 \leq t \leq 60$) - No.: 1.7335**Shell material** 11CrMo9-10 (EN 10216-2) - Seamless tube ($t \leq 60.00$ mm) - No.: 1.7383**Bolting material** 25CrMo4 (EN 10269) - Bolting ($t \leq 100.00$ mm) - No.: 1.7218**Gasket** Rubber with cotton fabric insertion

Calculation performed as a standard flange	=	Yes
Flange standard / specification	=	EN 1092-1:2007
Flange rating	PN =	160
Nominal size	DN =	40
Number of bolts	=	4
Bolt type	=	ISO M20 x 2.50
Material group	=	5E0

Calculation temperature	T =	121.11 °C
Internal pressure	Pd =	10.69 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	10.69 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa

Bolt loads

Number of bolts	n =	4
Bolt type	=	ISO M20 x 2.50
Root area of one bolt	=	225.0 mm ²
Distance between centre lines of adjacent bolts	δb =	98.17 mm
Bolt effective diameter	db =	17.65 mm

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.00 MPa
Hot & corroded (flange)	=	16.00 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	15.29 MPa
Overpressure due to static head	Ph =	0.0006 MPa
Maximum pressure at temperature allowed by the specifications	PS (Annex G) =	16.00 MPa
	PMax(test) = 1.5 · PS =	24.00 MPa