Calculation report

EN 13445 Ed. 2014 Issue 4

Project:	Víko čtvrtprovozního reaktoru
Item:	Víko
Customer:	VUOS a.s.
Drawing:	400-101
Revision:	-
Date:	04.08.2018

Internal design pressure	P =	4.00 MPa
Internal design temperature	T =	200.00 °C
Internal corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Joint efficiency	z =	0.85
Jacket internal design pressure	=	1.00 MPa
Jacket internal design temperature	=	200.00 °C
Jacket corrosion allowance	=	0 mm
Vacuum in jacket?	=	No
Minimum design temperature	=	0 °C

Test pressure (MPa)

Component	*P*	*Static head (design)*	*Static head (test)*	*Stress ratio*	*1.25·P·f0/f*	*1.43·P*
Cylindrical shell #1	4.00	0	0.007	1,25	6.25	5.72
Torispherical head #1	4.00	0	0.008	1,25	6.25	5.72
Welding neck flange #1	4.00	0	0.003	1,2	6.00	5.72
Welding neck flange #2	4.00	0	0.002	1,2	6.00	5.72
Torispherical head #2	4.00	0	0.001	1,25	6.25	5.72

All pressures in MPa.

Item design pressure P = 4.00 MPa
Item maximum allowable design pressure (Pmax) = 4.13 MPa (limited by Welding neck flange #1)
Item lowest stress ratio = 1.200 (limited by Welding neck flange #1)
Item test pressure = Pt=max[1.25·Pd·(Item f0/f);1.43·Pd] = 6.00 MPa

Maximum Pressures (MPa)

Component	*Internal, test*	*Internal*
Cylindrical shell #1	14.07	6.48
Torispherical head #1	12.06	6.32
Welding neck flange #1	4.95	4.13
Welding neck flange #2	4.95	4.13
Torispherical head #2	12.07	6.32

All pressures in MPa.

Weights

Component	*Dead*	*Live*	*Liquid*	*Full of water*	*Operating*
Cylindrical shell #1	71 kg	0 kg	0 kg	147 kg	71 kg
Torispherical head #1	34 kg	0 kg	0 kg	51 kg	34 kg
Welding neck flange #1	24 kg	0 kg	0 kg	37 kg	24 kg
Welding neck flange #2	24 kg	0 kg	0 kg	37 kg	24 kg
Torispherical head #2	34 kg	0 kg	0 kg	51 kg	34 kg
Totals:	187 kg	0 kg	0 kg	324 kg	187 kg

Total volume: 0.13725 m³
Center of gravity (erection): Cx=0 mm, Cy=0 mm, Cz=438.39 mm, W = 187 kg
Center of gravity (operating): Cx=0 mm, Cy=0 mm, Cz=438.39 mm, W = 187 kg
Center of gravity (test): Cx=0 mm, Cy=0 mm, Cz=427.74 mm, W = 324 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: sum of component's proper weight and contained water in test conditions

Operating: sum of component's proper weight and contained liquid in operating conditions

Bill of materials

Component	*Dimensions*	*Material*
Cylindrical shell #1	Id = 488.00 mm, Od = 516.00 mm, Tk = 14.00 mm, L = 406.00 mm	P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm) - No.: 1.0425
Torispherical head #1	Id = 488.00 mm, Od = 516.00 mm, Crown = 508.00 mm, Knuckle = 58.00 mm, Tk = 14.00 mm	P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm) - No.: 1.0425
Welding neck flange #1 - Flange	Id = 488.00 mm, Od = 568.00 mm, Tk = 46.00 mm	P245GH (N,NT,QT) (EN 10222-2:1999) - Forging (35 ≤ t ≤ 50) - No.: 1.0352
Welding neck flange #1 - Gasket	Flat metal, jacketed asbestos or mineral fibre filled - Stainless steels	
Welding neck flange #1 - Bolts	20 x ISO M27 x 3.00	25CrMo4 (EN 10269:2009) - Bolting (t ≤ 100.00 mm) - No.: 1.7218
Welding neck flange #2 - Flange	Id = 488.00 mm, Od = 568.00 mm, Tk = 46.00 mm	P245GH (N,NT,QT) (EN 10222-2:1999) - Forging (35 ≤ t ≤ 50) - No.: 1.0352
Torispherical head #2	Id = 488.00 mm, Od = 516.00 mm, Crown = 508.00 mm, Knuckle = 58.00 mm, Tk = 14.00 mm	P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm) - No.: 1.0425

Material properties summary

25CrMo4 (EN 10269:2009) - Bolting (t ≤ 100.00 mm) - No.: 1.7218

Temp.	*Allowable*	*Yield strength*	*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	412.00 MPa	0 MPa	196 000.00 MPa	0.000012100 1/°C
Test	419.05 MPa				

P245GH (N,NT,QT) (EN 10222-2:1999) - Forging (35 ≤ t ≤ 50) - No.: 1.0352

Temp.	*Allowable*	*Yield strength*	*Tensile strength*	*Elasticity*	*Thermal expansion*
Room	140.00 MPa	210.00 MPa	410.00 MPa	200 021.00 MPa	0.000011500 1/°C
Design	116.67 MPa	175.00 MPa	0 MPa	190 496.00 MPa	0.000012754 1/°C
Test	200.00 MPa				

P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm) - No.: 1.0425

Temp.	*Allowable*	*Yield strength*	*Tensile strength*	*Elasticity*	*Thermal expansion*
Room	170.83 MPa	265.00 MPa	410.00 MPa	200 021.00 MPa	0.000011500 1/°C
Design	136.67 MPa	205.00 MPa	410.00 MPa	190 496.00 MPa	0.000012754 1/°C
Test	252.38 MPa				

Notes
Allowable stress calculation may vary upon component type or characteristics

Yield strength shown refers to 0.2% plastic strain

Cylindrical shell - Cylindrical shell #1

According to: EN 13445 Ed. 2014 Issue 4, Part 3, Clause 7 and 8

Design data

Internal design temperature	Ti =	200.00 °C
Internal design pressure	Pi =	4.00 MPa
Joint efficiency	z =	0.85

Material: P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	136.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	170.83 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	252.38 MPa

Geometry

Inside diameter	Di =	488.00 mm
Outside diameter	De =	516.00 mm
Length	L =	406.00 mm
Nominal thickness	en =	14.00 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 =	4.00 MPa
Inside diameter	Di'=Di+2δ+2c =	488.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	8.55 mm

e/De ≤ 0,16 (0.01700 ≤ 0.16000): Ok
en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	14.07 MPa
Maximum allowable design pressure	=	6.48 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	2.789 %

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.20000
Coincident design pressure for the maximum pressure load case	Pd =	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	6.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	6.00 MPa
Overpressure due to static head in test condition	Pht =	0.007 MPa
Calculation pressure	Pc=Pt+Pht =	6.01 MPa
Inside diameter	Di'=Di+2δ =	488.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	5.88 mm

e/De ≤ 0,16 (0.01100 ≤ 0.16000): Ok
en ≥ e: Ok

Torispherical head - Torispherical head #1

According to: EN 13445 Ed. 2014 Issue 4, Part 3, Clause 7 and 8

Design data

Internal design temperature	Ti	=	200.00 °C
Internal design pressure	Pi	=	4.00 MPa
Joint efficiency	z	=	0.85

Material: P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$	=	136.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$	=	170.83 MPa
Nominal design stress in test condition	$f_{\text{test}} = (\frac{R_{\text{p0.2/Ttest}}}{1.05})$	=	252.38 MPa

Geometry

Inside diameter	Di	=	488.00 mm
Outside diameter	De	=	516.00 mm
Head outside height	H	=	139.24 mm
Nominal thickness	en	=	14.00 mm
Minimum head thickness after forming	t-c'	=	14.00 mm
Corrosion allowance	c	=	0 mm
External corrosion allowance	ce	=	0 mm
Undertolerance	δ	=	0 mm
Straight flange length	l(sf)	=	27.00 mm
Straight flange undertolerance	δ(sf)	=	0 mm
Straight flange thickness	en(sf)	=	14.00 mm
Straight flange joint efficiency	z(sf)	=	1.00000
Knuckle thickness	en(k)	=	14.00 mm
Inside spherical radius of central part of torispherical head	R	=	508.00 mm
Inside knuckle radius	r	=	58.00 mm

Internal pressure

Overpressure due to static head	Ph	=	0 MPa
Calculation pressure	P=Pi+Ph	=	4.00 MPa
Parameter Y	Y=min(ec/R;0.04)	=	0.01846
Parameter Z	Z=log10(1/Y)	=	1.73375
Ratio X	X=r/Di	=	0.11885
Parameter N	$N = 1.006 - \frac{1}{[6.2 + (90Y)^4]}$	=	0.93364
Parameter β(0.1)	$\beta_{0.1} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837)$	=	0.70841
Parameter β(0.2)	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2); 0.5]$	=	0.50000
Parameter β	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$	=	0.66912
Joint efficiency	z	=	0.85000
Inside spherical radius of central part of torispherical head	R'=R+c	=	508.00 mm
Inside diameter	Di'=Di+2·c+2·δ	=	488.00 mm
Inside knuckle radius	r'=r+c	=	58.00 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{\text{PR'}}{2\text{fz} - 0.5P} + c + \text{ce} + \delta$	=	8.82 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R + 0.2D_i')}{f} + c + \text{ce} + \delta$	=	9.37 mm
Minimum required thickness	e=max(ey;es)	=	9.37 mm
Straight flange minimum required thickness	e(sf)	=	7.25 mm

en(sf) ≥ e(sf): Ok
en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure		=	12.06 MPa

Maximum allowable design pressure		=	6.32 MPa

Deformation according to EN13445-4 Clause 9

Manufactured in one piece (9.2.1)	F(1)=100·ln[(1.11·De)/(De-2·en)]	=	16.015 %
Spherical part (9.2.1)	F(2)=100·ln{2·R·asin[(0.4·De/R)/(0.8·De-2·en)]}	=	9.959 %
Segments deformation of multi-sectional torispherical heads or spheres (9.2.5)	F(3)=(100·en)/(r+en/2)	=	21.538 %

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f	=	1.20000
Coincident design pressure for the maximum pressure load case	Pd	=	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f)	=	6.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd	=	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2)	=	6.00 MPa
Overpressure due to static head in test condition	Pht	=	0.008 MPa
Calculation pressure	Pc=Pt+Pht	=	6.01 MPa
Parameter Y	Y=min(ec/R;0.04)	=	0.01546
Parameter Z	Z=log10(1/Y)	=	1.81072
Ratio X	X=r/Di	=	0.11885
Parameter N	$N = 1.006 - \frac{1}{[6.2 + (90Y)^4]}$	=	0.90550
Parameter β(0.1)	$\beta_{0.1} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837)$	=	0.73296
Parameter β(0.2)	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2), 0.5]$	=	0.50000
Parameter β	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$	=	0.68904
Joint efficiency	z	=	1.00000
Inside spherical radius of central part of torispherical head	R'=R	=	508.00 mm
Inside diameter	Di'=Di+2·δ	=	488.00 mm
Inside knuckle radius	r'=r	=	58.00 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fz - 0.5P} + \delta$	=	6.08 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R + 0.2D_i')}{f} + \delta$	=	7.85 mm
Minimum required thickness	e=max(ey;es)	=	7.85 mm
			en ≥ e: Ok

Full-face gasket Welding neck flange - Welding neck flange #1

According to: EN 13445 Ed. 2014 Issue 4, Part 3, Clause 11

Flange material	**P245GH (N,NT,QT) (EN 10222-2:1999) - Forging (35 ≤ t ≤ 50)**
Shell material	**P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm)**
Bolting material	**25CrMo4 (EN 10269:2009) - Bolting (t ≤ 100.00 mm)**
Full face gasket	**Flat metal, jacketed asbestos or mineral fibre filled - Stainless steels**

Allowable stresses	**Flange - f**	**Hub - fH**	**Bolting - fB**
Design condition	116.67 MPa / 16 921.1 psi	116.67 MPa / 16 921.1 psi	137.33 MPa / 19 918.5 psi
Seating condition	140.00 MPa / 20 305.3 psi	140.00 MPa / 20 305.3 psi	146.67 MPa / 21 272.2 psi
Test condition	200.00 MPa / 29 007.5 psi	200.00 MPa / 29 007.5 psi	220.00 MPa / 31 908.3 psi

Internal pressure	Pd	=	4.00 MPa
Overpressure due to static head	Ph	=	0 MPa
Calculation pressure	P	=	4.00 MPa
Calculation temperature	T	=	200.00 °C

Geometry

Corrosion allowance	c	=	0 mm
Flange external diameter	A	=	568.00 mm
Inside diameter	B	=	488.00 mm
Inside diameter (corroded)	B* = B + 2c	=	488.00 mm
Bolt circle	C	=	608.00 mm
Flange thickness	en	=	46.00 mm
Hub length	h	=	25.00 mm
Thickness of hub at back of flange	g1	=	18.00 mm
Thickness of hub at back of flange (corroded)	g1*	=	18.00 mm
Thickness of hub at small end	g0	=	14.00 mm
Thickness of hub at small end (corroded)	g0*	=	14.00 mm

Gasket parameters

Gasket factor	m	=	3.75
Minimum gasket seating pressure	y	=	62.00 MPa
Gasket inside diameter	A1	=	508.00 mm
Gasket outside diameter	GO=min(A,Gasket OD)	=	568.00 mm
Effective gasket pressure width	2b"	=	5.00 mm
Basic assembly width effective under initial tightening up	$b'_O = \min(G_O - C; C - A_1)$	=	-40.00 mm
Effective assembly width	$b' = 4\sqrt{b'_O}$	=	NaN mm
Diameter of gasket load reaction	G = C - 2hG	=	573.00 mm

Bolt loads

Number of bolts	n	=	20
Bolt type		=	ISO M27 x 3.00
Root area of one bolt		=	427.0 mm²
Distance between centre lines of adjacent bolts	δb	=	95.50 mm
Bolt outside diameter	db	=	24.19 mm
Bolt hole diameter	dh	=	30.00 mm
Total hydrostatic end force	$H = \frac{\pi}{4} \cdot (C - d_h)^2 \cdot P$	=	1 049 556 N
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^{*2} P\right)$	=	748 151 N
Hydrostatic end force due to pressure on flange face	HT = H - HD	=	301 404 N
Compression load on gasket to ensure tight joint	HG = 2b"·π·G·m·P	=	135 010 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2	=	51.00 mm
Radial distance from bolt circle to circle on which HT acts	hT = (C + dh + 2b" - B*) / 4	=	38.75 mm
Radial distance from gasket load reaction to bolt circle	hG = (dh + 2b") / 2	=	17.50 mm
Radial distance from bolt circle to circle on which HR acts;	hR = (GO - C + dh) / 4	=	-2.50 mm
Balancing radial moment in flange along line of bolt holes	$M_R = H_D h_D + H_T h_T + H_G h_G$	=	52 197.8 N·m

Balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle	HR = MR / hR	=	-20 879 127 N
Minimum required bolt load for operating condition	Wop = H + HG + HR	=	-19 694 561 N
	$H_t = \dfrac{G^2 \pi P_t}{4}$	=	1 575 022 N
	HGt = 2b"·π·G·m·Pt	=	202 603 N
	HRt = MRt / hR	=	-28 993 429 N
Minimum required bolt load for the test condition (Welding neck flange #2)	Wt	=	-27 213 188 N
Minimum required bolt load for assembly condition	WA = π·C·b'·y	=	NaN N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\dfrac{W_A}{f_{B,A}}; \dfrac{W_{op}}{f_B}; \dfrac{W_t}{f_{Bt}}\right]$	=	NaN mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB	=	8 540.0 mm²
Design bolt load for assembly condition	$W = 0{,}5(A_{B,\min} + A_B) f_{B,A}$	=	NaN N

AB ≥ AB,min: Ok

Length of screws in threaded hole to 11.4.3.3

Length of screws in threaded hole	ls	=	52.00 mm
	Rp,component	=	175.00 MPa
	Rp,screw	=	412.00 MPa
	$l_{s,\min} = \max\left[0{,}8 \cdot d_n \cdot \dfrac{R_{p,\text{screw}}}{R_{p,\text{component}}};\ 0{,}8 \cdot d_n\right]$	=	50.85 mm

ls ≥ ls,min (52.00 mm ≥ 50.85 mm): Ok

Flange design

Elastic modulus	E	=	200 021.00 MPa
Minimum required flange thickness	$e_{\min} = \max\left[\sqrt{\dfrac{6 \cdot \max(M_F; M_2)}{f(\pi C - n d_h)}}; \dfrac{m+0.5}{(E/200000)^{0.25}} \cdot \dfrac{\delta_b - 2d_b}{6}; \dfrac{(A_1 + 2g_1^*)P}{2f}\right]$	=	45.27 mm

e ≥ e_min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	4.96 MPa
Hot & corroded (flange)	=	4.13 MPa
New & cold (bolts)	=	250.75 MPa
Hot & corroded (bolts)	=	250.75 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	6.00 MPa
Overpressure due to static head	Ph	=	0.003 MPa
Calculation pressure	P = Pt + Ph	=	6.00 MPa

Flange design

Elastic modulus	E	=	200 021.00 MPa
Minimum required flange thickness	$e_{\min} = \max\left[\sqrt{\dfrac{6 \cdot \max(M_F; M_2)}{f(\pi C - n d_h)}}; \dfrac{m+0.5}{(E/200000)^{0.25}} \cdot \dfrac{\delta_b - 2d_b}{6}; \dfrac{(A_1 + 2g_1^*)P}{2f}\right]$	=	34.57 mm

e ≥ e_min: Ok

Full-face gasket Welding neck flange - Welding neck flange #2

According to: EN 13445 Ed. 2014 Issue 4, Part 3, Clause 11

Flange material	**P245GH (N,NT,QT) (EN 10222-2:1999) - Forging (35 ≤ t ≤ 50)**
Shell material	**P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm)**
Bolting material	**25CrMo4 (EN 10269:2009) - Bolting (t ≤ 100.00 mm)**
Full face gasket	**Flat metal, jacketed asbestos or mineral fibre filled - Stainless steels**

Allowable stresses	**Flange - f**	**Hub - fH**	**Bolting - fB**
Design condition	116.67 MPa / 16 921.1 psi	116.67 MPa / 16 921.1 psi	137.33 MPa / 19 918.5 psi
Seating condition	140.00 MPa / 20 305.3 psi	140.00 MPa / 20 305.3 psi	146.67 MPa / 21 272.2 psi
Test condition	200.00 MPa / 29 007.5 psi	200.00 MPa / 29 007.5 psi	220.00 MPa / 31 908.3 psi

Internal pressure	Pd =	4.00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	4.00 MPa
Calculation temperature	T =	200.00 °C

Geometry

Corrosion allowance	c =	0 mm
Flange external diameter	A =	568.00 mm
Inside diameter	B =	488.00 mm
Inside diameter (corroded)	B* = B + 2c =	488.00 mm
Bolt circle	C =	608.00 mm
Flange thickness	en =	46.00 mm
Hub length	h =	25.00 mm
Thickness of hub at back of flange	g1 =	18.00 mm
Thickness of hub at back of flange (corroded)	g1* =	18.00 mm
Thickness of hub at small end	g0 =	14.00 mm
Thickness of hub at small end (corroded)	g0* =	14.00 mm

Gasket parameters

Gasket factor	m =	3.75
Minimum gasket seating pressure	y =	62.00 MPa
Gasket inside diameter	A1 =	508.00 mm
Gasket outside diameter	GO=min(A,Gasket OD) =	568.00 mm
Effective gasket pressure width	2b" =	5.00 mm
Basic assembly width effective under initial tightening up	$b'_O = \min(G_O - C; C - A_1)$ =	-40.00 mm
Effective assembly width	$b' = 4\sqrt{b'_O}$ =	NaN mm
Diameter of gasket load reaction	G = C - 2hG =	573.00 mm

Bolt loads

Number of bolts	n =	20
Bolt type	=	ISO M27 x 3.00
Root area of one bolt	=	427.0 mm²
Distance between centre lines of adjacent bolts	δb =	95.50 mm
Bolt outside diameter	db =	24.19 mm
Bolt hole diameter	dh =	30.00 mm
Total hydrostatic end force	$H = \frac{\pi}{4} \cdot (C - d_h)^2 \cdot P$ =	1 049 556 N
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}\left(B^{*2} P\right)$ =	748 151 N
Hydrostatic end force due to pressure on flange face	HT = H - HD =	301 404 N
Compression load on gasket to ensure tight joint	HG = 2b"·π·G·m·P =	135 010 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2 =	51.00 mm
Radial distance from bolt circle to circle on which HT acts	hT = (C + dh + 2b" - B*) / 4 =	38.75 mm
Radial distance from gasket load reaction to bolt circle	hG = (dh + 2b") / 2 =	17.50 mm
Radial distance from bolt circle to circle on which HR acts;	hR = (GO - C + dh) / 4 =	-2.50 mm
Balancing radial moment in flange along line of bolt holes	$M_R = H_D h_D + H_T h_T + H_G h_G$ =	52 197.8 N·m

Balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle	HR = MR / hR =	-20 879 127 N
Minimum required bolt load for operating condition	Wop = H + HG + HR =	-19 694 561 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	1 574 839 N
	HGt = 2b"·π·G·m·Pt =	202 580 N
	HRt = MRt / hR =	-28 990 607 N
Minimum required bolt load for the test condition	Wt = Ht + HGt + HRt =	-27 213 188 N
Minimum required bolt load for assembly condition	WA = π·C·b'·y =	NaN N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}}; \frac{W_{op}}{f_B}; \frac{W_t}{f_{Bt}}\right]$ =	NaN mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB =	8 540.0 mm²
Design bolt load for assembly condition	$W = 0{,}5(A_{B,\min} + A_B) f_{B,A}$ =	NaN N
		AB ≥ AB,min: Ok

Flange design

Elastic modulus	E =	200 021.00 MPa
Minimum required flange thickness	$e_{\min} = \max\left[\sqrt{\frac{6 \cdot \max(M_F; M_2)}{f(\pi C - n d_h)}}; \frac{m + 0.5}{(E/200000)^{0.25}} \cdot \frac{\delta_b - 2d_b}{6}; \frac{(A_1 + 2g_1^*)P}{2f}\right]$ =	45.27 mm
		e ≥ e_min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	4.96 MPa
Hot & corroded (flange)	=	4.13 MPa
New & cold (bolts)	=	250.75 MPa
Hot & corroded (bolts)	=	250.75 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt =	6.00 MPa
Overpressure due to static head	Ph =	0.002 MPa
Calculation pressure	P = Pt + Ph =	6.00 MPa

Flange design

Elastic modulus	E =	200 021.00 MPa
Minimum required flange thickness	$e_{\min} = \max\left[\sqrt{\frac{6 \cdot \max(M_F; M_2)}{f(\pi C - n d_h)}}; \frac{m + 0.5}{(E/200000)^{0.25}} \cdot \frac{\delta_b - 2d_b}{6}; \frac{(A_1 + 2g_1^*)P}{2f}\right]$ =	34.57 mm
		e ≥ e_min: Ok

Torispherical head - Torispherical head #2

According to: EN 13445 Ed. 2014 Issue 4, Part 3, Clause 7 and 8

Design data

Internal design temperature	Ti =	200.00 °C
Internal design pressure	Pi =	4.00 MPa
Joint efficiency	z =	0.85

Material: P265GH (EN 10028-2:2009) - Plate (t ≤ 16.00 mm)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	136.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	170.83 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	252.38 MPa

Geometry

Inside diameter	Di =	488.00 mm
Outside diameter	De =	516.00 mm
Head outside height	H =	139.24 mm
Nominal thickness	en =	14.00 mm
Minimum head thickness after forming	t-c' =	14.00 mm
Corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Straight flange length	l(sf) =	27.00 mm
Straight flange undertolerance	δ(sf) =	0 mm
Straight flange thickness	en(sf) =	14.00 mm
Straight flange joint efficiency	z(sf) =	1.00000
Knuckle thickness	en(k) =	14.00 mm
Inside spherical radius of central part of torispherical head	R =	508.00 mm
Inside knuckle radius	r =	58.00 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	4.00 MPa
Parameter Y	Y=min(ec/R;0.04) =	0.01846
Parameter Z	Z=log10(1/Y) =	1.73375
Ratio X	X=r/Di =	0.11885
Parameter N	$N = 1.006 - \frac{1}{[6.2 + (90Y)^4]}$ =	0.93364
Parameter β(0.1)	$\beta_{0.1} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837)$ =	0.70841
Parameter β(0.2)	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2); 0.5]$ =	0.50000
Parameter β	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$ =	0.66912
Joint efficiency	z =	0.85000
Inside spherical radius of central part of torispherical head	R'=R+c =	508.00 mm
Inside diameter	Di'=Di+2·c+2·δ =	488.00 mm
Inside knuckle radius	r'=r+c =	58.00 mm

Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fz - 0.5P} + c + ce + \delta$	=	8.82 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R + 0.2D_i')}{f} + c + ce + \delta$	=	9.37 mm
Minimum required thickness	e=max(ey;es)	=	9.37 mm
Straight flange minimum required thickness	e(sf)	=	7.25 mm
			en(sf) ≥ e(sf): Ok
			en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure		=	12.07 MPa
Maximum allowable design pressure		=	6.32 MPa

Deformation according to EN13445-4 Clause 9

Manufactured in one piece (9.2.1)	F(1)=100·ln[(1.11·De)/(De-2·en)]	=	16.015 %
Spherical part (9.2.1)	F(2)=100·ln{2·R·asin[(0.4·De/R)/(0.8·De-2·en)]}	=	9.959 %
Segments deformation of multi-sectional torispherical heads or spheres (9.2.5)	F(3)=(100·en)/(r+en/2)	=	21.538 %

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f	=	1.20000
Coincident design pressure for the maximum pressure load case	Pd	=	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f)	=	6.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd	=	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2)	=	6.00 MPa
Overpressure due to static head in test condition	Pht	=	0.001 MPa
Calculation pressure	Pc=Pt+Pht	=	6.00 MPa
Parameter Y	Y=min(ec/R;0.04)	=	0.01545
Parameter Z	Z=log10(1/Y)	=	1.81113
Ratio X	X=r/Di	=	0.11885
Parameter N	$N = 1.006 - \frac{1}{[6.2 + (90Y)^4]}$	=	0.90536
Parameter β(0.1)	$\beta_{0.1} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837)$	=	0.73309
Parameter β(0.2)	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2), 0.5]$	=	0.50000
Parameter β	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$	=	0.68915
Joint efficiency	z	=	1.00000
Inside spherical radius of central part of torispherical head	R'=R	=	508.00 mm
Inside diameter	Di'=Di+2·δ	=	488.00 mm
Inside knuckle radius	r'=r	=	58.00 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fz - 0.5P} + \delta$	=	6.08 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R + 0.2D_i')}{f} + \delta$	=	7.84 mm
Minimum required thickness	e=max(ey;es)	=	7.84 mm
			en ≥ e: Ok