

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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 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 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 =	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)	$\beta_{0.2} = \max[0.95(0.56 - 1.94Y - 82.5Y^2), 0.5]$ =	0.50000
Parameter β	$\beta = 10[(0.2 - X)\beta_{0.1} + (X - 0.1)\beta_{0.2}]$ =	0.53464
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

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____		Customer Drawing Revision
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fZ - 0.5P} + c + ce + \delta$	= 18.93 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 mm
Minimum required thickness	$e = \max(e_y; e_s)$	= 19.46 mm
Straight flange minimum required thickness	$e(sf)$	= 21.58 mm
		en(sf) ≥ e(sf): Ok en ≥ e: Ok
Maximum allowable pressures (at the top of the vessel)		
Maximum allowable test pressure		= 27.85 MPa
Maximum allowable design pressure		= 16.71 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)	$F(2) = 100 \cdot \ln\{2 \cdot R \cdot \arcsin[(0.4 \cdot De/R)/(0.8 \cdot De - 2 \cdot en)]\}$	= 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	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	$Pt1 = 1.25 \cdot Pd \cdot (\text{Item f0/f})$	= 13.36 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	$Pt2 = 1.43 \cdot Pd$	= 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
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	$N = 1006 - \frac{1}{[62 + (90Y)^4]}$	= 0.98775
Parameter β(0.1)	$\beta_{01} = N(-0.1833Z^3 + 10383Z^2 - 12943Z + 0.837)$	= 0.62459
Parameter β(0.2)	$\beta_{02} = \max[0.95(0.56 - 194Y - 82.5Y^2), 0.5]$	= 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 mm
Inside diameter	$Di' = Di$	= 620.00 mm
Inside knuckle radius	$r' = r$	= 104.10 mm
Required thickness of end to limit membrane stress in central part	$e_s = \frac{PR'}{2fZ - 0.5P} + \delta$	= 15.21 mm
Required thickness of knuckle to avoid axisymmetric yielding	$e_y = \frac{\beta P(0.75R + 0.2D_i)}{f} + \delta$	= 15.86 mm
Minimum required thickness	$e = \max(e_y; e_s)$	= 15.86 mm
		en ≥ e: Ok

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Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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 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 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 =	620.00 mm
Outside diameter	De =	676.00 mm
Length	L =	2400.00 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	$e = \frac{P \cdot D_i'}{2f \cdot z \cdot P} + c + ce + \delta =$	22.47 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	$F = 50 \cdot en / (Di/2 + en/2) =$	4.321 %
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Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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		

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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	Ts = 121.11 °C / 250.00 °F	Ps = 10.69 MPa / 1 550.5 psi
Channel	Tc = 37.78 °C / 100.00 °F	Pt = 10.69 MPa / 1 550.5 psi
Tubesheet	T = 121.11 °C / 250.00 °F	
Tubes	Tt = 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	fh = 242.86 MPa

Tubes material 13CrMo4-5 (EN 10216-2)

Tube design temperature	Tt = 121.11 °C
Modulus of elasticity for tube material at Tt	Et = 198 193.42 MPa
Nominal design stress of tube material at Tt	ft = 172.90 MPa
Modulus of elasticity for tube material at test temperature	Et_t = 204 842.00 MPa

Channel material 13CrMo4-5 (EN 10028-2)

Channel design temperature	Tc = 37.78 °C
Modulus of elasticity for channel material at Tc	Ec = 203 548.34 MPa
Poisson's ratio of channel material	vc = 0.30
Nominal design stress of channel material at Tc	fc = 187.50 MPa
Modulus of elasticity for channel material at test temperature	Ec_t = 204 842.00 MPa
Allowable stress for channel material for hydraulic test	fch = 276.19 MPa

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

Shell design temperature	Ts = 121.11 °C
Modulus of elasticity for shell material at Ts	Es = 198 193.42 MPa
Poisson's ratio of shell material	vs = 0.30
Nominal design stress of shell material at Ts	fs = 179.11 MPa
Modulus of elasticity for shell material at test temperature	Es_t = 204 842.00 MPa
Allowable stress for shell material for hydraulic test	fsh = 276.19 MPa

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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

<div>Company name</div> <div>Address</div> <div>City</div> <div>Telephone, Fax</div> <div>Website, Email address</div> <div>Date _____ Calc. _____ Contr. _____ Appr. _____</div>	<div>Customer</div> <div>Drawing</div> <div>Revision</div>
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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

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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	$\rho_s = G_s / D_o =$	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	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	-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
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$|\tau| \leq 0.8f$: Ok

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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	$\rho_s = G_s / D_o =$	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	$M_{TS} = \frac{D_o^2 [(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	255 320 N
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Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \frac{W_{\max}(G_c - G_s)}{2\pi D_o}$	=	255 320 N
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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
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Moment acting at centre of tubesheet	$M_o = M_p + \frac{D_o^2}{64} (3 + \nu^*) (P_s - P_t)$	=	111 907 N
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Maximum bending moment acting on the tubesheet	$M = \max[M_p , M_o]$	=	111 907 N
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Maximum radial bending stress in the tubesheet	$\sigma = \frac{6M}{\mu^* (e - h_g')^2}$	=	208.31 MPa
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$\sigma \leq 2f$: 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
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$|\tau| \leq 0.8f$: Ok

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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
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$|\tau| \leq 0.8f$: Ok

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
---	---

Loading case Hydrostatic test tubes side: Pt = 15.32 MPa, Ps = 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 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

Company name Address City Telephone, Fax Website, Email address Date _____ Calc. _____ Contr. _____ Appr. _____	Customer Drawing Revision
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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