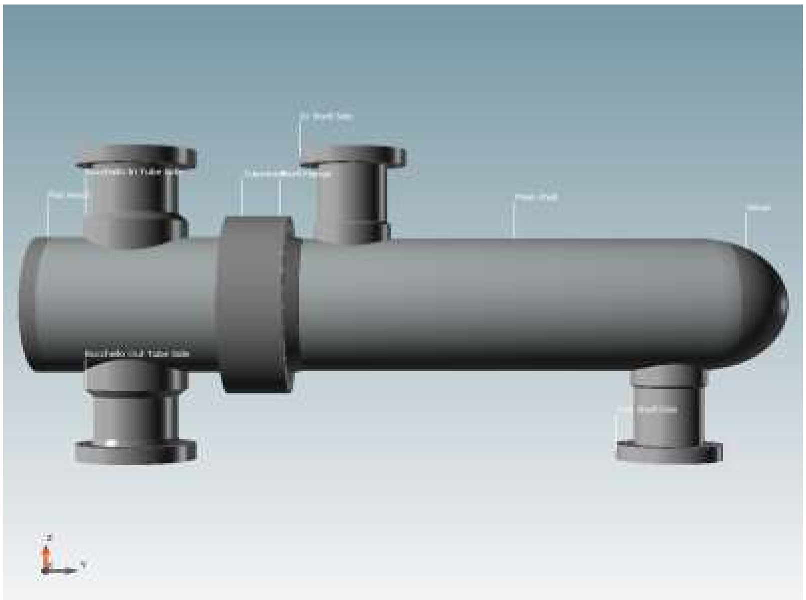
Calculation report

Asme VIII Div. 2 Ed. 2015 - Metric Units

Project: U_150_Divisione_1_Start
Item: U_150_Div_2
Customer: Riccardo Petrelli
Drawing: U_150
Revision:
Date: 01/07/2016

	Tube side		Shell side	
Internal design pressure	18.44 MPa	2 674.0 psi	14.22 MPa	2 062.4 psi
Internal design temperature	454.00 °C	849.20 °F	420.00 °C	788.00 °F
Corrosion allowance	0 mm	0 in	3.00 mm	0.118 in
Vacuum?	Yes		Yes	

	Both sides	
Minimum design temperature	-4.00 °C	24.80 °F



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Test Pressure - Tube side (MPa)

Component	*P*	*Static head (design)*	*Static head (test)*	*MAP N&C*	*MAWP H&C*	*Stress ratio*
Flat Head	18.44	0	0.02	25.91	18.75	1,381
Left channel	18.44	0	0.02	25.91	18.75	1,381
Bocchello In Tube Side	18.44	0	0.01	25.61	18.54	1,381
Flange In Tube Side	18.44	0	0.002	20.77	19.00	1,381
Bocchello Out Tube Side	18.44	0	0.03	25.61	18.54	1,381
Flange Out Tube Side	18.44	0	0.03	20.77	19.00	1,381
Tubesheet	18.44	0	0.02	31.26	22.60	1,381
Tubes bundle	18.54	0	0.02	41.99	41.99	1

All pressures in MPa

Tubes side design pressure P = 18.54 MPa

Design pressure used for test pressure calculation increased due to vacuum

Tubes side MAWP (Hot & Corroded conditions) = 18.54 MPa (limited by Bocchello In Tube Side)

Tubes side MAP (New & Cold conditions) = 20.77 MPa (limited by Flange In Tube Side)

Tubes side Lowest Stress Ratio = 1.000

Tubes side test pressure = Pt = 1.43 * MAWP = 26.51 MPa

Maximum Pressures - Tube side (MPa)

Component	*MAP N&C*	*MAWP H&C*	*MAEP N&C*	*MAEWP H&C*
Flat Head	25.91	18.75	18.75	18.75
Left channel	25.91	18.75	21.88	21.88
Bocchello In Tube Side	25.61	18.54	23.09	23.09
Flange In Tube Side	20.77	19.00	19.00	19.00
Bocchello Out Tube Side	25.61	18.54	23.09	23.09
Flange Out Tube Side	20.77	19.00	19.00	19.00
Tubesheet	31.26	22.60		
Tubes bundle	41.99	41.99	23.69	19.90

All pressures in MPa

Test Pressure - Shell side (MPa)

Component	*P*	*Static head (design)*	*Static head (test)*	*MAP N&C*	*MAWP H&C*	*Stress ratio*
Shell Flange	14.22	0	0.02	22.47	21.58	1,305
Main shell	14.22	0	0.02	20.10	14.65	1,305
Out Shell Side	14.22	0	0.03	25.86	17.67	1,305
In Shell Side	14.22	0	0.01	25.86	17.67	1,305
Tubesheet	14.22	0	0.02	19.04	14.22	1,381
Tubes bundle	14.32	0	0.01	23.69	19.90	1
Head	14.22	0	0.02	21.01	14.65	1,305

All pressures in MPa

Shell side design pressure P = 14.32 MPa

Design pressure used for test pressure calculation increased due to vacuum

Shell side MAWP (Hot & Corroded conditions) = 14.22 MPa (limited by Tubesheet)

Shell side MAP (New & Cold conditions) = 19.04 MPa (limited by Tubesheet)

Shell side Lowest Stress Ratio = 1.000

Shell side test pressure = Pt = 1.43 * MAWP = 20.34 MPa

Maximum Pressures - Shell side (MPa)

Component	*MAP N&C*	*MAWP H&C*	*MAEP N&C*	*MAEWP H&C*
Shell Flange	22.47	21.58	21.58	21.58
Main shell	20.10	14.65	17.21	16.41
Out Shell Side	25.86	17.67	17.67	17.67
In Shell Side	25.86	17.67	17.67	17.67
Tubesheet	19.04	14.22		
Tubes bundle	23.69	19.90		
Head	21.01	14.65	14.52	12.88

All pressures in MPa

Weights

Component	***Dead***	***Live***	***Liquid***	***Full of water***	***Operating***
Flat Head	3 559 kg	0 kg	0 kg	3 559 kg	3 559 kg
Left channel	3 530 kg	0 kg	0 kg	5 445 kg	3 530 kg
Bocchello In Tube Side	937 kg	0 kg	0 kg	1 183 kg	937 kg
Flange In Tube Side	635 kg	0 kg	0 kg	710 kg	635 kg
Bocchello Out Tube Side	937 kg	0 kg	0 kg	1 183 kg	937 kg
Flange Out Tube Side	635 kg	0 kg	0 kg	710 kg	635 kg
Shell Flange	3 231 kg	0 kg	0 kg	3 769 kg	3 231 kg
Main shell	7 696 kg	0 kg	0 kg	10 519 kg	7 696 kg
Out Shell Side	1 437 kg	0 kg	0 kg	1 464 kg	1 437 kg
In Shell Side	1 437 kg	0 kg	0 kg	1 464 kg	1 437 kg
Tubesheet	3 674 kg	0 kg	0 kg	3 674 kg	3 674 kg
Tubes bundle	15 229 kg	0 kg	0 kg	17 027 kg	15 229 kg
Head	744 kg	0 kg	0 kg	1 327 kg	744 kg
Totals:	43 681 kg	0 kg	0 kg	52 035 kg	43 681 kg

Total shell side volume: 3.99688 m³

Total tube side volume: 4.35666 m³

Total volume: 8.35355 m³

Bill of materials

Component	*Dimensions*	*Material*
Flat Head	Id = 1 275.00 mm, Od = 1 445.00 mm, Tk = 280.00 mm	SA-387 22 2 - Plate
Left channel	Id = 1 275.00 mm, Od = 1 445.00 mm, Tk = 85.00 mm, L = 1 500.00 mm	SA-387 22 2 - Plate
Bocchello In Tube Side	Id = 635.00 mm, Od = 725.00 mm, Tk = 45.00 mm, L = 778.00 mm	SA-182 F22 3 - Forging
Flange In Tube Side - Flange	Id = 635.00 mm, Od = 1 026.00 mm, Tk = 170.00 mm	SA-182 F22 3 - Forging
Flange In Tube Side - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Flange In Tube Side - Bolts	20 x ANSI_TEMA 2-1/2"	SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting
Bocchello Out Tube Side	Id = 635.00 mm, Od = 725.00 mm, Tk = 45.00 mm, L = 778.00 mm	SA-182 F22 3 - Forging
Flange Out Tube Side - Flange	Id = 635.00 mm, Od = 1 026.00 mm, Tk = 170.00 mm	SA-182 F22 3 - Forging
Flange Out Tube Side - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Flange Out Tube Side - Bolts	20 x ANSI_TEMA 2-1/2"	SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting
Shell Flange - Flange	Id = 1 277.00 mm, Od = 1 900.00 mm, Tk = 300.00 mm	SA-387 22 2 - Plate
Shell Flange - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Shell Flange - Bolts	28 x ANSI_TEMA 4-1/2"	SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting
Main shell	Id = 1 275.00 mm, Od = 1 405.00 mm, Tk = 65.00 mm, L = 3 629.00 mm	SA-387 22 2 - Plate
Out Shell Side - Flange	Id = 457.20 mm, Od = 915.00 mm, Tk = 69.90 mm	SA-182 F22 3 - Forging
Out Shell Side - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Out Shell Side - Bolts	16 x ANSI_TEMA 2-1/2"	SA-193 B16 - Bolting
In Shell Side - Flange	Id = 457.20 mm, Od = 915.00 mm, Tk = 69.90 mm	SA-182 F22 3 - Forging
In Shell Side - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
In Shell Side - Bolts	16 x ANSI_TEMA 2-1/2"	SA-193 B16 - Bolting
Tubesheet - Flange	Od = 1 900.00 mm, Tk = 230.00 mm	SA-182 F22 3 - Forging
Tubes bundle	Id = 23.37 mm, Od = 31.75 mm, Tk = 4.19 mm, L = 3 658.00 mm	SB-517 Cold drawn/ann. N06600 - Pipe / tube
Head	Id = 1 306.00 mm, Od = 1 374.00 mm, Tk = 34.00 mm	SA-387 22 2 - Plate

Nozzle connections

Name	*Flange*	*Material*	*OD*	*Tk*
Bocchello In Tube Side	WN NotStandard	SA-182 F22 3	725.00 mm	45.00 mm
Bocchello Out Tube Side	WN NotStandard	SA-182 F22 3	725.00 mm	45.00 mm
Out Shell Side	18" LWN 1500 ANSI	SA-182 F22 3	597.00 mm	69.90 mm
In Shell Side	18" LWN 1500 ANSI	SA-182 F22 3	597.00 mm	69.90 mm

Nozzle positions

Name	*Placed on*	*Type*	*Distance from reference*	*Orientation*	*Notes*
Bocchello In Tube Side	Left channel	Radial/ Reinforced	700.00 mm	0 °	
Bocchello Out Tube Side	Left channel	Radial/ Reinforced	700.00 mm	180.00 °	
Out Shell Side	Main shell	Radial/ Reinforced	3 150.00 mm	180.00 °	
In Shell Side	Main shell	Radial/ Reinforced	420.00 mm	0 °	

Nozzle welds

Name	*Nozzle to wall*	*Pad to wall*	*Shell groove*	*Pad groove*	*Inside*
Bocchello In Tube Side	15.00 mm				
Bocchello Out Tube Side	15.00 mm				
Out Shell Side	15.00 mm				
In Shell Side	15.00 mm				

Minimum Design Metal Temperature (MDMT)

Component		***MDMT***	***Tmin > MDMT***
Flat Head		Impact tests required	
Left channel		Impact tests required	
Bocchello In Tube Side		Impact tests required	
Flange In Tube Side		Impact tests required	
Flange In Tube Side (bolting)		-29.00 °C / -20.20 °F	Yes
Bocchello Out Tube Side		Impact tests required	
Flange Out Tube Side		Impact tests required	
Flange Out Tube Side (bolting)		-29.00 °C / -20.20 °F	Yes
Shell Flange		Impact tests required	
Shell Flange (bolting)		-29.00 °C / -20.20 °F	Yes
Main shell		Impact tests required	
Out Shell Side		Impact tests required	
Out Shell Side (bolting)		-29.00 °C / -20.20 °F	Yes
In Shell Side		Impact tests required	
In Shell Side (bolting)		-29.00 °C / -20.20 °F	Yes
Tubesheet		Impact tests required	
Tubesheet (bolting)		-29.00 °C / -20.20 °F	Yes
Tubes bundle		-104.00 °C / -155.20 °F	Yes
Head		39.50 °C / 103.10 °F	No

Item minimum design temperature Tmin: -4.00 °C / 24.80 °F

Impact tests required by Code

One or more components have a MDMT higher then item minimum design temperature.

Welded flat cover - Flat Head

According to: Asme VIII Div. 2 Ed. 2015, 4.6 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Material:	**SA-387 22 2 - Plate**			
Allowable stress	S	=	149.84 MPa	21 732.5 psi
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Inside diameter	D	=	1 275.00 mm	50.197 in
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Factor C	C	=		0.30000
Outside diameter	d = Do	=	1 445.00 mm	56.890 in
Adopted thickness	t	=	280.00 mm	11.024 in
Head type		=		Type9

Internal pressure

Allowable stress	S	=	149.84 MPa	21 732.5 psi
Internal pressure	Pi	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	18.44 MPa	2 674.0 psi
Required thickness	$t_r = d\sqrt{\frac{\mathrm{CP}}{\mathrm{SE}}} + c + c_e + c'$	=	277.63 mm	10.930 in
				t ≥ tr: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold	=	25.91 MPa	3 757.6 psi
Hot & corroded	=	18.75 MPa	2 720.0 psi

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
Required thickness	$t_r = d\sqrt{\frac{\mathrm{CP}}{\mathrm{SE}}} + c + c_e + c'$	=	17.66 mm	0.695 in
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	18.75 MPa	2 720.0 psi
Hot & corroded	=	18.75 MPa	2 720.0 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Cylindrical shell - Left channel

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Material: SA-387 22 2 - Plate				
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	1 275.00 mm	50.197 in
Length	L	=	1 500.00 mm	59.055 in
Adopted thickness	t	=	85.00 mm	3.346 in

Ligament Efficiency

Reference figure		=		None
Diameter of tube holes	d	=	0 mm	0 in

Internal pressure

Allowable stress	S	=	149.84 MPa	21 732.5 psi
Internal pressure	Pi	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	18.44 MPa	2 674.0 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	83.47 mm	3.286 in
				t ≥ tr: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold		=	25.91 MPa	3 757.8 psi
Hot & corroded		=	18.75 MPa	2 720.1 psi

External pressure

External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	1 445.00 mm	56.890 in
Max unsupported length	L	=	1 500.00 mm	59.055 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		6.05289
	$C_h = \frac{0.92}{M_x - 0.579}$	=		0.16807
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	3 327.40 MPa	482 598.8 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	21.88 MPa	3 173.1 psi
Minimum required thickness	tr	=	4.31 mm	0.170 in
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	21.88 MPa	3 173.1 psi
Hot & corroded	=	21.88 MPa	3 173.1 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Cylindrical shell - Left channel

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Cylindrical shell - Left channel

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Validation warnings:

- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]

Reinforcement of opening - Bocchello In Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Nozzle material	**SA-182 F22 3 - Forgings**			
Shell material	**SA-387 22 2 - Plate**			
Pad material	**-**			
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	149.84 MPa	21 732.5 psi
Shell allowable stress at room temperature	S0	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	149.84 MPa	21 732.5 psi
Nozzle allowable stress at room temperature	Sn0	=	207.00 MPa	30 022.8 psi
Shell thickness	t	=	85.00 mm	3.346 in
Nozzle thickness	tn	=	130.00 mm	5.118 in
Nominal wall thickness of the nozzle thinner portion	tn2	=	45.00 mm	1.772 in
Tapering angle	ta	=		45.00 °
Nozzle inside diameter	d	=	635.00 mm	25.000 in
Nozzle outside diameter	Od	=	725.00 mm	28.543 in
Joint efficiency	E	=		1.00000
Nozzle internal corrosion allowance	cni	=	0 mm	0 in
Nozzle external corrosion allowance	cne	=	0 mm	0 in
Nozzle total corrosion allowance	cn	=	0 mm	0 in
Nozzle undertolerance	cn'	=	0 mm	0 in
Nozzle position		=		Radial
Nozzle connection		=		Integrally reinforced
Weld joint type		=		7 - Full penetration welds
Offset from shell border		=	700.00 mm	27.559 in
Angular offset		=		0 °
Width of the reinforcing pad	W	=	0 mm	0 in
Thickness of the reinforcing pad	te	=	0 mm	0 in
Nozzle inside radius	Rn = d/2 + cni + cn'	=	317.50 mm	12.500 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{\text{eff}} = 0.5 \cdot D_i + c'$	=	637.50 mm	25.098 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	693.00 mm	27.283 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	170.00 mm	6.693 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	255.00 mm	10.039 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Corner radius	r	=	10.00 mm	0.394 in
Effective length along the vessel wall	$L_R = \min\left[\sqrt{R_{\text{eff}} \cdot (t - c - c')}, 2R_n\right]$	=	232.78 mm	9.165 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	203.16 mm	7.999 in
	$L_{H2} = L_{\text{pr1}}$	=	693.00 mm	27.283 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	680.00 mm	26.772 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	288.16 mm	11.345 in
Effective thickness	$t_{\text{eff}} = t - c - c' + \left(\frac{A_5 f_{\text{rp}}}{L_R}\right)$	=	85.00 mm	3.346 in
	LI1	=	0 mm	0 in
	LI2	=	0 mm	0 in

	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{\text{eff}})t_{\text{eff}}}}\right\}, 12.0\right]$	=		2.25000
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R)\cdot\max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	19 786.5 mm²	30.669 in²
Wall thickness at the variable thickness portion of the nozzle	$t_{\text{nx}} = \left[1 + \frac{t_n - t_{\text{n2}}}{t_{\text{n2}} - c_n - c_n'}\cdot\frac{L_{\text{x4}} - L_H}{L_{\text{pr4}} - L_{\text{pr3}}}\right](t_{\text{n2}} - c_n - c_n')$	=	96.84 mm	3.813 in
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3	$A_{2a} = (t_n - c_n - c_n')L_{x3}$	=	33 150.0 mm²	51.383 in²
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3	$A_{2b} = \frac{t_n - c_n - c_n' + t_{\text{nx}}}{2}\cdot\min\left[0.78\sqrt{R_n\left(\frac{t_n - c_n - c_n' + t_{\text{nx}}}{2}\right)}, L_H - L_{x3}\right]$	=	3 761.2 mm²	5.830 in²
Nozzle outs. vessel wall area	$A_2 = A_{2a} + A_{2b} - r^2\left(1 - \frac{\pi}{4}\right)$	=	36 889.8 mm²	57.179 in²
Nozzle material factor	$f_{\text{rn}} = \frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{\text{rp}} = \frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a} = W t_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{\text{rp}}\ A_5$	=	56 788.8 mm²	88.023 in²
Radius of the nozzle opening	Rnc = Rn	=	317.50 mm	12.500 in
Nozzle radius for force calculation	$R_{\text{xn}} = \frac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$	=	378.79 mm	14.913 in
Shell radius for force calculation	$R_{\text{xs}} = \frac{t_{\text{eff}}}{\ln\left[\frac{R_{\text{eff}} + t_{\text{eff}}}{R_{\text{eff}}}\right]}$	=	679.11 mm	26.737 in
Force from internal pressure in the nozzle	$f_N = P R_{\text{xn}} L_H$	=	2 012 397 N	452 404.82 lbf
Force from internal pressure in the shell	$f_S = P R_{\text{xs}}(L_R + t_n - c_n - c_n')$	=	4 542 206 N	1 021 128.44 lbf
Discontinuity force from internal pressure	$f_Y = P R_{\text{xs}} R_{\text{nc}}$	=	3 975 253 N	893 672.24 lbf
Average primary membrane stress	$\sigma_{\text{avg}} = \frac{(f_N + f_S + f_Y)}{A_T}$	=	185.42 MPa	26 893.1 psi
General primary membrane stress	$\sigma_{\text{circ}} = \frac{P R_{\text{xs}}}{t_{\text{eff}}}$	=	147.30 MPa	21 364.0 psi
Allowable stress	$S_{\text{allow}} = 1.5\text{SE}$	=	224.76 MPa	32 598.7 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{\text{avg}} - \sigma_{\text{circ}}), \sigma_{\text{circ}}]$	=	223.54 MPa	32 422.2 psi
				r ≥ min[0.25t,3mm(0.125in)]: Ok
				PL ≤ Sallow: Ok
	$P_{\text{max1}} = \frac{S_{\text{allow}}}{\frac{2A_p}{A_T} - \frac{R_{\text{xs}}}{t_{\text{eff}}}}$	=	18.54 MPa	2 688.5 psi
	$P_{\text{max2}} = S\left(\frac{t - c - c'}{R_{\text{xs}}}\right)$	=	18.75 MPa	2 720.1 psi
Area resisting pressure	$A_p = \frac{f_N + f_S + f_Y}{P} + r^2\left(1 - \frac{\pi}{4}\right)$	=	571 163.2 mm²	885.305 in²
Nozzle maximum allowable pressure (bottom)	$P_{\text{max}} = \min[P_{\text{max1}}, P_{\text{max2}}]$	=	18.54 MPa	2 688.5 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2	=	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \frac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$	=		1.40945
Nozzle to shell weld length	$L_\tau = \frac{\pi}{2}(R_n + t_n - c_n - c_n')$	=	702.93 mm	27.674 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$	=	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$	=	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$	=	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$	=	2 899 709 N	651 880.56 lbf
Nozzle to shell groove weld depth	tw1 = twall	=	85.00 mm	3.346 in
Average effective shear stress	$\tau = \frac{f_{welds}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$	=	73.41 MPa	10 646.5 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	207.00 MPa	30 022.8 psi
Nozzle inside radius	Rn = d/2 + cni + cn'	=	317.50 mm	12.500 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{eff} = 0.5 \cdot D_i + c'$	=	637.50 mm	25.098 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	693.00 mm	27.283 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	170.00 mm	6.693 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	255.00 mm	10.039 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Effective length along the vessel wall	$L_R = \min\left[\sqrt{R_{eff} \cdot (t - c - c')},\ 2R_n\right]$	=	232.78 mm	9.165 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	203.16 mm	7.999 in
	$L_{H2} = L_{pr1}$	=	693.00 mm	27.283 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	680.00 mm	26.772 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	288.16 mm	11.345 in
Effective thickness	$t_{eff} = t - c - c' + \left(\frac{A_5 f_{rp}}{L_R}\right)$	=	85.00 mm	3.346 in
	LI1	=	0 mm	0 in
	LI2	=	0 mm	0 in
	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{eff})t_{eff}}}\right\},\ 12.0\right]$	=		2.25000
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85},\ 1.0\right]$	=	19 786.5 mm²	30.669 in²
Wall thickness at the variable thickness portion of the nozzle	$t_{nx} = \left[1 + \frac{t_n - t_{n2}}{t_{n2} - c_n - c_n'} \cdot \frac{L_{x4} - L_H}{L_{pr4} - L_{pr3}}\right](t_{n2} - c_n - c_n')$	=	96.84 mm	3.813 in
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3	$A_{2a} = (t_n - c_n - c_n')L_{x3}$	=	33 150.0 mm²	51.383 in²

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3	$A_{2b}=\frac{t_n-c_n-c_n'+t_{nx}}{2}\cdot\min\left[0.78\sqrt{R_n\left(\frac{t_n-c_n-c_n'+t_{nx}}{2}\right)},L_H-L_{x3}\right]$	=	3 761.2 mm²	5.830 in²
Nozzle outs. vessel wall area	$A_2=A_{2a}+A_{2b}-r^2\left(1-\frac{\pi}{4}\right)$	=	36 889.8 mm²	57.179 in²
Nozzle material factor	$f_{rn}=\frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{rp}=\frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3=(t_n-2c_n-c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41}=0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42}=0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43}=0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a}=Wt_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5=\min[A_{5a},A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T=A_1+f_m(A_2+A_3)+A_{41}+A_{42}+A_{43}+f_{rp}\ A_5$	=	56 788.8 mm²	88.023 in²
Radius of the nozzle opening	Rnc = Rn	=	317.50 mm	12.500 in
Nozzle radius for force calculation	$R_{xn}=\frac{t_n-c_n-c_n'}{\ln\left[\frac{R_n+t_n-c_n-c_n'}{R_n}\right]}$	=	378.79 mm	14.913 in
Shell radius for force calculation	$R_{xs}=\frac{t_{eff}}{\ln\left[\frac{R_{eff}+t_{eff}}{R_{eff}}\right]}$	=	679.11 mm	26.737 in
Force from internal pressure in the nozzle	$f_N=PR_{xn}L_H$	=	11 243 N	2 527.47 lbf
Force from internal pressure in the shell	$f_S=PR_{xs}(L_R+t_n-c_n-c_n')$	=	25 376 N	5 704.78 lbf
Discontinuity force from internal pressure	$f_Y=PR_{xs}R_{nc}$	=	22 209 N	4 992.72 lbf
Average primary membrane stress	$\sigma_{avg}=\frac{(f_N+f_S+f_Y)}{A_T}$	=	1.04 MPa	150.2 psi
General primary membrane stress	$\sigma_{circ}=\frac{PR_{xs}}{t_{eff}}$	=	0.82 MPa	119.4 psi
Allowable stress	$S_{allow}=F_{ha\ shell}$	=	185.96 MPa	26 971.6 psi
Maximum local primary membrane stress	$P_L=\max[(2\sigma_{avg}-\sigma_{circ}),\sigma_{circ}]$	=	1.25 MPa	181.1 psi
				PL ≤ Sallow: Ok
	$P_{max1}=\frac{S_{allow}}{\frac{2A_p}{A_T}-\frac{R_{xs}}{t_{eff}}}$	=	15.34 MPa	2 224.5 psi
	$P_{max2}=S\left(\frac{t-c-c'}{R_{xs}}\right)$	=	25.91 MPa	3 757.7 psi
Area resisting pressure	$A_p=\frac{f_N+f_S+f_Y}{P}+r^2\left(1-\frac{\pi}{4}\right)$	=	571 163.2 mm²	885.305 in²
Nozzle maximum allowable pressure (bottom)	$P_{max}=\min[P_{max1},P_{max2}]$	=	15.34 MPa	2 224.5 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2	=	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \dfrac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$	=		1.40945
Nozzle to shell weld length	$L_\tau = \dfrac{\pi}{2}(R_n + t_n - c_n - c_n')$	=	702.93 mm	27.674 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$	=	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$	=	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$	=	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \dfrac{\pi}{4} P R_n^2 k_y^2 \right]$	=	16 200 N	3 641.89 lbf
Nozzle to shell groove weld depth	tw1 = twall	=	85.00 mm	3.346 in
Average effective shear stress	$\tau = \dfrac{f_{welds}}{L_\tau (0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$	=	0.41 MPa	59.5 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

Reinforcement of opening - Bocchello Out Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Nozzle material	**SA-182 F22 3 - Forgings**			
Shell material	**SA-387 22 2 - Plate**			
Pad material	**-**			
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	149.84 MPa	21 732.5 psi
Shell allowable stress at room temperature	S0	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	149.84 MPa	21 732.5 psi
Nozzle allowable stress at room temperature	Sn0	=	207.00 MPa	30 022.8 psi
Shell thickness	t	=	85.00 mm	3.346 in
Nozzle thickness	tn	=	130.00 mm	5.118 in
Nominal wall thickness of the nozzle thinner portion	tn2	=	45.00 mm	1.772 in
Tapering angle	ta	=		45.00 °
Nozzle inside diameter	d	=	635.00 mm	25.000 in
Nozzle outside diameter	Od	=	725.00 mm	28.543 in
Joint efficiency	E	=		1.00000
Nozzle internal corrosion allowance	cni	=	0 mm	0 in
Nozzle external corrosion allowance	cne	=	0 mm	0 in
Nozzle total corrosion allowance	cn	=	0 mm	0 in
Nozzle undertolerance	cn'	=	0 mm	0 in
Nozzle position		=		Radial
Nozzle connection		=		Integrally reinforced
Weld joint type		=		7 - Full penetration welds
Offset from shell border		=	700.00 mm	27.559 in
Angular offset		=		180.00 °
Width of the reinforcing pad	W	=	0 mm	0 in
Thickness of the reinforcing pad	te	=	0 mm	0 in
Nozzle inside radius	Rn = d/2 + cni + cn'	=	317.50 mm	12.500 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{\text{eff}} = 0.5 \cdot D_i + c$	=	637.50 mm	25.098 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	693.00 mm	27.283 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	170.00 mm	6.693 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	255.00 mm	10.039 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Corner radius	r	=	10.00 mm	0.394 in
Effective length along the vessel wall	$L_R = \min\left[\sqrt{R_{\text{eff}} \cdot (t - c - c')}, 2R_n\right]$	=	232.78 mm	9.165 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	203.16 mm	7.999 in
	$L_{H2} = L_{\text{pr1}}$	=	693.00 mm	27.283 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	680.00 mm	26.772 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	288.16 mm	11.345 in
Effective thickness	$t_{\text{eff}} = t - c - c' + \left(\frac{A_5 f_{\text{rp}}}{L_R}\right)$	=	85.00 mm	3.346 in
	Ll1	=	0 mm	0 in
	Ll2	=	0 mm	0 in

	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{\text{eff}})t_{\text{eff}}}}\right\}, 12.0\right]$	=		2.25000
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R)\cdot\max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	19 786.5 mm²	30.669 in²
Wall thickness at the variable thickness portion of the nozzle	$t_{\text{nx}} = \left[1 + \frac{t_n - t_{n2}}{t_{n2} - c_n - c_n'}\cdot\frac{L_{x4} - L_H}{L_{\text{pr4}} - L_{\text{pr3}}}\right](t_{n2} - c_n - c_n')$	=	96.84 mm	3.813 in
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3	$A_{2a} = (t_n - c_n - c_n')L_{x3}$	=	33 150.0 mm²	51.383 in²
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3	$A_{2b} = \frac{t_n - c_n - c_n' + t_{\text{nx}}}{2}\cdot\min\left[0.78\sqrt{R_n\left(\frac{t_n - c_n - c_n' + t_{\text{nx}}}{2}\right)}, L_H - L_{x3}\right]$	=	3 761.2 mm²	5.830 in²
Nozzle outs. vessel wall area	$A_2 = A_{2a} + A_{2b} - r^2\left(1 - \frac{\pi}{4}\right)$	=	36 889.8 mm²	57.179 in²
Nozzle material factor	$f_{\text{rn}} = \frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{\text{rp}} = \frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a} = W t_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T = A_1 + f_{\text{rn}}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{\text{rp}} A_5$	=	56 788.8 mm²	88.023 in²
Radius of the nozzle opening	Rnc = Rn	=	317.50 mm	12.500 in
Nozzle radius for force calculation	$R_{\text{xn}} = \frac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$	=	378.79 mm	14.913 in
Shell radius for force calculation	$R_{\text{xs}} = \frac{t_{\text{eff}}}{\ln\left[\frac{R_{\text{eff}} + t_{\text{eff}}}{R_{\text{eff}}}\right]}$	=	679.11 mm	26.737 in
Force from internal pressure in the nozzle	$f_N = P R_{\text{xn}} L_H$	=	2 012 397 N	452 404.82 lbf
Force from internal pressure in the shell	$f_S = P R_{\text{xs}}(L_R + t_n - c_n - c_n')$	=	4 542 206 N	1 021 128.44 lbf
Discontinuity force from internal pressure	$f_Y = P R_{\text{xs}} R_{\text{nc}}$	=	3 975 253 N	893 672.24 lbf
Average primary membrane stress	$\sigma_{\text{avg}} = \frac{(f_N + f_S + f_Y)}{A_T}$	=	185.42 MPa	26 893.1 psi
General primary membrane stress	$\sigma_{\text{circ}} = \frac{P R_{\text{xs}}}{t_{\text{eff}}}$	=	147.30 MPa	21 364.0 psi
Allowable stress	$S_{\text{allow}} = 1.5SE$	=	224.76 MPa	32 598.7 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{\text{avg}} - \sigma_{\text{circ}}), \sigma_{\text{circ}}]$	=	223.54 MPa	32 422.2 psi

r ≥ min[0.25t,3mm(0.125in)]: Ok

PL ≤ Sallow: Ok

	$P_{\text{max1}} = \frac{S_{\text{allow}}}{\frac{2A_p}{A_T} - \frac{R_{\text{xs}}}{t_{\text{eff}}}}$	=	18.54 MPa	2 688.5 psi
	$P_{\text{max2}} = S\left(\frac{t - c - c'}{R_{\text{xs}}}\right)$	=	18.75 MPa	2 720.1 psi
Area resisting pressure	$A_p = \frac{f_N + f_S + f_Y}{P} + r^2\left(1 - \frac{\pi}{4}\right)$	=	571 163.2 mm²	885.305 in²
Nozzle maximum allowable pressure (bottom)	$P_{\max} = \min[P_{\text{max1}}, P_{\text{max2}}]$	=	18.54 MPa	2 688.5 psi

P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2	=	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \frac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$	=		1.40945
Nozzle to shell weld length	$L_\tau = \frac{\pi}{2}(R_n + t_n - c_n - c_n')$	=	702.93 mm	27.674 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$	=	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$	=	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$	=	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$	=	2 899 709 N	651 880.56 lbf
Nozzle to shell groove weld depth	tw1 = twall	=	85.00 mm	3.346 in
Average effective shear stress	$\tau = \frac{f_{welds}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$	=	73.41 MPa	10 646.5 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	207.00 MPa	30 022.8 psi
Nozzle inside radius	Rn = d/2 + cni + cn'	=	317.50 mm	12.500 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{eff} = 0.5 \cdot D_i + c'$	=	637.50 mm	25.098 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	693.00 mm	27.283 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	170.00 mm	6.693 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	255.00 mm	10.039 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Effective length along the vessel wall	$L_R = \min\left[\sqrt{R_{eff} \cdot (t - c - c')},\ 2R_n\right]$	=	232.78 mm	9.165 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	203.16 mm	7.999 in
	$L_{H2} = L_{pr1}$	=	693.00 mm	27.283 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	680.00 mm	26.772 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	288.16 mm	11.345 in
Effective thickness	$t_{eff} = t - c - c' + \left(\frac{A_5 f_{rp}}{L_R}\right)$	=	85.00 mm	3.346 in
	LI1	=	0 mm	0 in
	LI2	=	0 mm	0 in
	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{eff})t_{eff}}}\right\}, 12.0\right]$	=		2.25000
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	19 786.5 mm²	30.669 in²
Wall thickness at the variable thickness portion of the nozzle	$t_{nx} = \left[1 + \frac{t_n - t_{n2}}{t_{n2} - c_n - c_n'} \cdot \frac{L_{x4} - L_H}{L_{pr4} - L_{pr3}}\right](t_{n2} - c_n - c_n')$	=	96.84 mm	3.813 in
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3	$A_{2a} = (t_n - c_n - c_n')L_{x3}$	=	33 150.0 mm²	51.383 in²

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3	$A_{2b}=\frac{t_n-c_n-c_n'+t_{nx}}{2}\cdot\min\left[0.78\sqrt{R_n\left(\frac{t_n-c_n-c_n'+t_{nx}}{2}\right)},L_H-L_{x3}\right]$	=	3 761.2 mm²	5.830 in²
Nozzle outs. vessel wall area	$A_2=A_{2a}+A_{2b}-r^2\left(1-\frac{\pi}{4}\right)$	=	36 889.8 mm²	57.179 in²
Nozzle material factor	$f_{rn}=\frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{rp}=\frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3=(t_n-2c_n-c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41}=0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42}=0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43}=0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a}=Wt_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5=\min[A_{5a},A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T=A_1+f_m(A_2+A_3)+A_{41}+A_{42}+A_{43}+f_{rp}\ A_5$	=	56 788.8 mm²	88.023 in²
Radius of the nozzle opening	Rnc = Rn	=	317.50 mm	12.500 in
Nozzle radius for force calculation	$R_{xn}=\frac{t_n-c_n-c_n'}{\ln\left[\frac{R_n+t_n-c_n-c_n'}{R_n}\right]}$	=	378.79 mm	14.913 in
Shell radius for force calculation	$R_{xs}=\frac{t_{eff}}{\ln\left[\frac{R_{eff}+t_{eff}}{R_{eff}}\right]}$	=	679.11 mm	26.737 in
Force from internal pressure in the nozzle	$f_N=PR_{xn}L_H$	=	11 243 N	2 527.47 lbf
Force from internal pressure in the shell	$f_S=PR_{xs}(L_R+t_n-c_n-c_n')$	=	25 376 N	5 704.78 lbf
Discontinuity force from internal pressure	$f_Y=PR_{xs}R_{nc}$	=	22 209 N	4 992.72 lbf
Average primary membrane stress	$\sigma_{avg}=\frac{(f_N+f_S+f_Y)}{A_T}$	=	1.04 MPa	150.2 psi
General primary membrane stress	$\sigma_{circ}=\frac{PR_{xs}}{t_{eff}}$	=	0.82 MPa	119.4 psi
Allowable stress	$S_{allow}=F_{ha\ shell}$	=	185.96 MPa	26 971.6 psi
Maximum local primary membrane stress	$P_L=\max[(2\sigma_{avg}-\sigma_{circ}),\sigma_{circ}]$	=	1.25 MPa	181.1 psi
				PL ≤ Sallow: Ok
	$P_{max1}=\frac{S_{allow}}{\frac{2A_p}{A_T}-\frac{R_{xs}}{t_{eff}}}$	=	15.34 MPa	2 224.5 psi
	$P_{max2}=S\left(\frac{t-c-c'}{R_{xs}}\right)$	=	25.91 MPa	3 757.7 psi
Area resisting pressure	$A_p=\frac{f_N+f_S+f_Y}{P}+r^2\left(1-\frac{\pi}{4}\right)$	=	571 163.2 mm²	885.305 in²
Nozzle maximum allowable pressure (bottom)	$P_{max}=\min[P_{max1},P_{max2}]$	=	15.34 MPa	2 224.5 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2 =		10.61 mm	0.418 in
Discontinuity force factor	$k_y = \frac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$ =			1.40945
Nozzle to shell weld length	$L_\tau = \frac{\pi}{2}(R_n + t_n - c_n - c_n')$ =		702.93 mm	27.674 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$ =		10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$ =		0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$ =		0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2 \right]$ =		16 200 N	3 641.89 lbf
Nozzle to shell groove weld depth	tw1 = twall =		85.00 mm	3.346 in
Average effective shear stress	$\tau = \frac{f_{welds}}{L_\tau (0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$ =		0.41 MPa	59.5 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

Nozzle - Bocchello In Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Material: **SA-182 F22 3 - Forgings**				
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	635.00 mm	25.000 in
Length	L	=	778.00 mm	30.630 in
Adopted thickness	t	=	45.00 mm	1.772 in
Internal pressure				
Allowable stress	S	=	149.84 MPa	21 732.5 psi
Internal pressure	Pi	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	18.44 MPa	2 674.0 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	41.57 mm	1.637 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	27.44 MPa	3 979.4 psi
Hot & corroded		=	19.86 MPa	2 880.6 psi
External pressure				
External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	725.00 mm	28.543 in
Max unsupported length	L	=	778.00 mm	30.630 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		6.09143
	$C_h = \frac{0.92}{M_x - 0.579}$	=		0.16690
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	3 486.44 MPa	505 664.7 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	23.09 MPa	3 348.2 psi
Minimum required thickness	tr	=	2.20 mm	0.087 in
				t ≥ tr: Ok
Maximum allowable external pressures				
New & cold		=	23.09 MPa	3 348.2 psi
Hot & corroded		=	23.09 MPa	3 348.2 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Nozzle - Bocchello In Tube Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Nozzle - Bocchello In Tube Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Welding neck flange - Flange In Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

Design data

Internal design temperature	T	=	454.00 °C	849.20 °F
Internal design pressure	P	=	18.44 MPa	2 674.0 psi
External design temperature	Te	=	20.00 °C	68.00 °F
External design pressure	Pe	=	0.10 MPa	14.9 psi
Joint efficiency	E	=		1.00

Flange material **SA-182 F22 3 - Forgings**

Shell material **SA-182 F22 3 - Forgings**

Bolting material **SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting**

Gasket **Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=149.84 MPa / 21 732.5 psi	Sno=149.84 MPa / 21 732.5 psi	Sbo=161.44 MPa / 23 414.9 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=172.00 MPa / 24 946.5 psi
Test condition	Sft= /	Snt= /	Sbt= /

Internal pressure	Pd	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P	=	18.44 MPa	2 674.0 psi
Modulus of Elasticity at the operating load case temperature	Eyo	=	179 600.00 MPa	26 048 777.7 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi
Corrosion allowance	c	=	0 mm	0 in
Flange external diameter	A	=	1 026.00 mm	40.394 in
Inside diameter	B	=	635.00 mm	25.000 in
Bolt circle	C	=	905.00 mm	35.630 in
Flange thickness	t	=	170.00 mm	6.693 in
Mean gasket diameter	Gmean	=	651.00 mm	25.630 in
Thickness of the hub at the small end	g0	=	45.00 mm	1.772 in
Thickness of the hub at the large end	g1	=	57.00 mm	2.244 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	45.00 mm	1.772 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	57.00 mm	2.244 in
Hub length	h	=	68.00 mm	2.677 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	70.00 MPa	10 152.6 psi
Gasket contact width	N	=	16.00 mm	0.630 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	8.00 mm	0.315 in
Conversion factor for length	Cul	=		25.40000
Effective gasket contact width	$b = 0.5C_{ul}\sqrt{\frac{b_0}{C_{ul}}}$	=	7.13 mm	0.281 in
Outside diameter of the gasket contact area	$G_c = G_{mean} + N$	=	667.00 mm	26.260 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	652.75 mm	25.699 in

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 2-1/2"
Bolt spacing	Bs	=	142.16 mm	5.597 in
Nominal bolt diameter	a	=	63.50 mm	2.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \text{GmP}$	=	8 456 915 N	1 901 189.99 lbf
Design bolt load for the test condition	$W_t = 0.785G^2P_t + 2b\pi \text{Gm}P_t$	=	0 N	0 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \text{bG}y$	=	1 023 113 N	230 004.85 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right), \left(\frac{W_t + F_{At} + \frac{4M_{Et}}{G}}{S_{bt}}\right)\right]$	=	52 384.3 mm²	81.196 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	55 380.5 mm²	85.840 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	26 706.3 mm²	41.395 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	9 267 772 N	2 083 477.94 lbf
				Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.61575
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		4.22139
Stress factor T	$T = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{(1.04720 + 1.9448K^2)(K-1)}$	=		1.66082
Stress factor U	$U = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{1.36136(K^2-1)(K-1)}$	=		4.63888
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		2.24174
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	169.04 mm	6.655 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.26667
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.40227
Flange stress factor for integral type flanges	$F = \left(\begin{array}{l} 0.897697 - 0.297012X_g + 9.5257(10^{-3})X_g + \\ 0.123586(X_g)^2 + 0.0358580(X_h)^2 - 0.194422(X_g)(X_h) - \\ 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - \\ 0.0377693(X_g)(X_h)^2 + 0.0273791(X_g)^2(X_h) \end{array}\right)$	=		0.87850
Flange stress factor for integral type flanges	$V = \left(\begin{array}{l} 0.500244 + \frac{0.227914}{X_g} - 1.87071X_h - \frac{0.344410}{X_g^2} + 2.49189X_h^2 + \\ 0.873446\left(\frac{X_h}{X_g}\right) + \frac{0.189953}{X_g^3} - 1.06082X_h^3 - 1.49970\left(\frac{X_h^2}{X_g}\right) + \\ 0.719413\left(\frac{X_h}{X_g^2}\right) \end{array}\right)$	=		0.40681
Hub stress correction factor for integral flanges	$f = \max\left[1.0, \left(\frac{0.0927779 - 0.0336633X_g + 0.964176X_g^2 + 0.0566286X_h + 0.347076X_h^2 - 4.18699X_h^3}{1 - 5.96093(10^{-3})X_g + 1.62904X_h + 3.49329X_h^2 + 1.39052X_h^3}\right)\right]$	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00520

Stress factor d	$d = \frac{U g_0^2 h_o}{V}$	=		3 903 371.60229
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		2.39272
Gasket load for the operating condition	$H_G = W_o - H$	=	2 290 461 N	514 916.01 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785 B^2 P$	=	5 835 736 N	1 311 925.51 lbf
Total hydrostatic end force	$H = 0.785 G^2 P$	=	6 166 454 N	1 386 273.97 lbf
Difference	$H_T = H - H_D$	=	330 718 N	74 348.46 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	106.50 mm	4.193 in
Moment arm for load HG	$h_G = \frac{C - G}{2}$	=	126.13 mm	4.966 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C - B'}{2} + h_G\right]$	=	130.56 mm	5.140 in
	$A_R = 0.5(A - B')$	=	195.50 mm	7.697 in
Average of the hub thicknesses	$G_{avg} = 0.5(g_0' + g_1')$	=	51.00 mm	2.008 in
	AA = AR	=	195.50 mm	7.697 in
	BB = t	=	170.00 mm	6.693 in
	CC = h	=	68.00 mm	2.677 in
	DDG = Gavg	=	51.00 mm	2.008 in
Moment of inertia KAB	$K_{AB} = (A_A B_B^3)\left[\frac{1}{3} - 0.21\left(\frac{B_B}{A_A}\right)\left(1 - \frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	153 126 579 mm^4	367.888 in^4
Moment of inertia KCD	$K_{CD} = (C_C D_{DG}^3)\left[\frac{1}{3} - 0.105\left(\frac{D_{DG}}{C_C}\right)\left(1 - \frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	2 297 581 mm^4	5.520 in^4
Bending moment of inertia of the flange cross-section	$I = \frac{0.0874 L g_0'^2 h_0 B'}{V}$	=	111 738 738 mm^4	268.453 in^4
Cross-section polar moment of inertia	$I_P = K_{AB} + K_{CD}$	=	155 424 159 mm^4	373.408 in^4

Flange moments

Nominal bolt diameter	dB	=	63.50 mm	2.500 in
TEMA Load concentration factor	$\mathrm{cF} = \mathrm{MAX}\left[\sqrt{\frac{\frac{\pi \cdot C}{n}}{2\mathrm{dB} + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = \mathrm{cF} \cdot \mathrm{abs}[(H_D h_D + H_T h_T + H_G h_G + M_{oe}) F_s]$	=	953 575.6 N·m	8 439 854.3 lbf·in
Flange design moment for the gasket seating condition	$M_g = \mathrm{cF} \cdot \frac{W_g (C - G) F_s}{2}$	=	1 168 920.1 N·m	10 345 814.1 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	692.00 mm	27.244 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	177.26 MPa	25 709.2 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1) M_o}{L t^2 B'}$	=	47.23 MPa	6 850.7 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	113.46 MPa	16 456.5 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \frac{fM_g}{Lg_1^2B_1'}$ =	217.29 MPa	31 515.1 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2B'}$ =	57.90 MPa	8 397.8 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2B'} - ZS_R$ =	139.09 MPa	20 172.9 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR =	0.30
Flange rigidity index	$J = \frac{52.14VM_o}{LE_{yo}g_0^2K_Rh_o}$ =	0.45833

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2K_Rh_o}$ =	0.47970

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min =	41.57 mm	1.637 in

g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	26.18 MPa	3 796.5 psi
Hot & corroded (flange)	=	19.00 MPa	2 756.4 psi
New & cold (bolts)	=	20.77 MPa	3 011.8 psi
Hot & corroded (bolts)	=	19.49 MPa	2 826.9 psi

External pressure

Allowable stresses	Flange	Hub	Bolting
Design condition	Sfo=207.00 MPa / 30 022.8 psi	Sno=207.00 MPa / 30 022.8 psi	Sbo=172.00 MPa / 24 946.5 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=172.00 MPa / 24 946.5 psi

External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Modulus of Elasticity at the operating load case temperature	Eyo	=	210 350.00 MPa	30 508 688.2 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 2-1/2"
Bolt spacing	Bs	=	142.16 mm	5.597 in
Nominal bolt diameter	a	=	63.50 mm	2.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi\mathrm{GmP}$	=	47 247 N	10 621.46 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi\mathrm{bG}y$	=	1 023 113 N	230 004.85 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	5 948.3 mm²	9.220 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	55 380.5 mm²	85.840 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	26 706.3 mm²	41.395 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	5 274 282 N	1 185 705.72 lbf

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.61575
Stress factor Y	$Y=\frac{1}{K-1}\left[0.66845+5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		4.22139
Stress factor T	$T=\frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.66082
Stress factor U	$U=\frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		4.63888
Stress factor Z	$Z=\frac{(K^2+1)}{(K^2-1)}$	=		2.24174
Hub length parameter	$h_o=\sqrt{Bg_0'}$	=	169.04 mm	6.655 in
Hub thickness ratio	$X_g=\frac{g_1'}{g_0'}$	=		1.26667
Hub length ratio	$X_h=\frac{h}{h_0}$	=		0.40227
Flange stress factor for integral type flanges	$F=\begin{pmatrix}0.897697-0.297012X_g+9.5257(10^{-3})X_g+\\0.123586(X_g)^2+0.0358580(X_h)^2-0.194422(X_g)(X_h)-\\0.0181259(X_g)^3+0.0129360(X_h)^3-\\0.0377693(X_g)(X_h)^2+0.0273791(X_g)^2(X_h)\end{pmatrix}$	=		0.87850
Flange stress factor for integral type flanges	$V=\begin{pmatrix}0.500244+\frac{0.227914}{X_g}-1.87071X_h-\frac{0.344410}{X_g^2}+2.49189X_h^2+\\0.873446\left(\frac{X_h}{X_g}\right)+\frac{0.189953}{X_g^3}-1.06082X_h^3-1.49970\left(\frac{X_h^2}{X_g}\right)+\\0.719413\left(\frac{X_h}{X_g^2}\right)\end{pmatrix}$	=		0.40681
Hub stress correction factor for integral flanges	$f=\max\left[1.0,\left(\frac{0.0927779-0.0336633X_g+0.964176X_g^2+0.0566286X_h+0.347076X_h^2-4.18699X_h^3}{1-5.96093(10^{-3})X_g+1.62904X_h+3.49329X_h^2+1.39052X_h^3}\right)\right]$	=		1.00000
Stress factor e	$e=\frac{F}{h_o}$	=		0.00520
Stress factor d	$d=\frac{Ug_0^2h_o}{V}$	=		3 903 371.60229
Stress factor L	$L=\frac{\mathrm{te}+1}{T}+\frac{t^3}{d}$	=		2.39272
Gasket load for the operating condition	$H_G=W_o-H$	=	12 796 N	2 876.70 lbf
Total hydrostatic end force on the area inside of the flange	$H_D=0.785B^2P$	=	32 603 N	7 329.39 lbf
Total hydrostatic end force	$H=0.785G^2P$	=	34 450 N	7 744.76 lbf
Difference	$H_T=H-H_D$	=	1 848 N	415.37 lbf
Moment arm for load HD	$\mathrm{hD}=\frac{C-B'-g_1'}{2}$	=	106.50 mm	4.193 in
Moment arm for load HG	$h_G=\frac{C-G}{2}$	=	126.13 mm	4.966 in
Moment arm for load HT	$h_T=\frac{1}{2}\left[\frac{C-B'}{2}+h_G\right]$	=	130.56 mm	5.140 in
	$A_R=0.5(A-B')$	=	195.50 mm	7.697 in
Average of the hub thicknesses	$G_{avg}=0.5(g_0'+g_1')$	=	51.00 mm	2.008 in
	AA = AR	=	195.50 mm	7.697 in
	BB = t	=	170.00 mm	6.693 in
	CC = h	=	68.00 mm	2.677 in
	DDG = Gavg	=	51.00 mm	2.008 in
Moment of inertia KAB	$K_{AB}=(A_AB_B^3)\left[\frac{1}{3}-0.21\left(\frac{B_B}{A_A}\right)\left(1-\frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	153 126 579 mm^4	367.888 in^4
Moment of inertia KCD	$K_{CD}=(C_CD_{DG}^3)\left[\frac{1}{3}-0.105\left(\frac{D_{DG}}{C_C}\right)\left(1-\frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	2 297 581 mm^4	5.520 in^4

Bending moment of inertia of the flange cross-section	$I = \frac{0.0874 L g_0^2 h_0 B'}{V}$	=	111 738 738 mm^4	268.453 in^4
Cross-section polar moment of inertia	$I_p = K_{AB} + K_{CD}$	=	155 424 159 mm^4	373.408 in^4

Flange moments

Nominal bolt diameter	dB	=	63.50 mm	2.500 in
TEMA Load concentration factor	$cF = MAX\left[\sqrt{\frac{\frac{\pi \cdot C}{n}}{2dB + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = abs[\ (H_D(h_D - h_G) + H_T(h_T - h_G) + M_{oe})F_s\ ]$	=	631.7 N·m	5 591.1 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_g F_s$	=	665 231.6 N·m	5 887 795.0 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	692.00 mm	27.244 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	0.12 MPa	17.0 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1)M_o}{L t^2 B'}$	=	0.03 MPa	4.5 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	0.08 MPa	10.9 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \frac{f M_g}{L g_1^2 B_1'}$	=	123.66 MPa	17 935.2 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te + 1)M_g}{L t^2 B'}$	=	32.95 MPa	4 779.2 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{Y M_g}{t^2 B'} - Z S_R$	=	79.15 MPa	11 480.4 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR	=	0.30
Flange rigidity index	$J = \frac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=	0.00026

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=	0.27300

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Flange In Tube Side (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Flange In Tube Side

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	45.00 mm	1.772 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Welding neck flange - Flange In Tube Side (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Flange In Tube Side

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	45.00 mm	1.772 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Validation warnings:

- Gasket overloaded: Ab > AbMax

Nozzle - Bocchello Out Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Material:	**SA-182 F22 3 - Forgings**			
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	635.00 mm	25.000 in
Length	L	=	778.00 mm	30.630 in
Adopted thickness	t	=	45.00 mm	1.772 in
Internal pressure				
Allowable stress	S	=	149.84 MPa	21 732.5 psi
Internal pressure	Pi	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	18.44 MPa	2 674.0 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	41.57 mm	1.637 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	27.44 MPa	3 979.4 psi
Hot & corroded		=	19.86 MPa	2 880.6 psi
External pressure				
External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	725.00 mm	28.543 in
Max unsupported length	L	=	778.00 mm	30.630 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		6.09143
	$C_h = \frac{0.92}{M_x - 0.579}$	=		0.16690
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	3 486.44 MPa	505 664.7 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	23.09 MPa	3 348.2 psi
Minimum required thickness	tr	=	2.20 mm	0.087 in
				t ≥ tr: Ok
Maximum allowable external pressures				
New & cold		=	23.09 MPa	3 348.2 psi
Hot & corroded		=	23.09 MPa	3 348.2 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Nozzle - Bocchello Out Tube Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Nozzle - Bocchello Out Tube Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Welding neck flange - Flange Out Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

Design data

Internal design temperature	T	=	454.00 °C	849.20 °F
Internal design pressure	P	=	18.44 MPa	2 674.0 psi
External design temperature	Te	=	20.00 °C	68.00 °F
External design pressure	Pe	=	0.10 MPa	14.9 psi
Joint efficiency	E	=		1.00

Flange material **SA-182 F22 3 - Forgings**

Shell material **SA-182 F22 3 - Forgings**

Bolting material **SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting**

Gasket **Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=149.84 MPa / 21 732.5 psi	Sno=149.84 MPa / 21 732.5 psi	Sbo=161.44 MPa / 23 414.9 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=172.00 MPa / 24 946.5 psi
Test condition	Sft= /	Snt= /	Sbt= /

Internal pressure	Pd	=	18.44 MPa	2 674.0 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P	=	18.44 MPa	2 674.0 psi
Modulus of Elasticity at the operating load case temperature	Eyo	=	179 600.00 MPa	26 048 777.7 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi
Corrosion allowance	c	=	0 mm	0 in
Flange external diameter	A	=	1 026.00 mm	40.394 in
Inside diameter	B	=	635.00 mm	25.000 in
Bolt circle	C	=	905.00 mm	35.630 in
Flange thickness	t	=	170.00 mm	6.693 in
Mean gasket diameter	Gmean	=	651.00 mm	25.630 in
Thickness of the hub at the small end	g0	=	45.00 mm	1.772 in
Thickness of the hub at the large end	g1	=	57.00 mm	2.244 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	45.00 mm	1.772 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	57.00 mm	2.244 in
Hub length	h	=	68.00 mm	2.677 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	70.00 MPa	10 152.6 psi
Gasket contact width	N	=	16.00 mm	0.630 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	8.00 mm	0.315 in
Conversion factor for length	Cul	=		25.40000
Effective gasket contact width	$b = 0.5C_{ul}\sqrt{\frac{b_0}{C_{ul}}}$	=	7.13 mm	0.281 in
Outside diameter of the gasket contact area	$G_c = G_{mean} + N$	=	667.00 mm	26.260 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	652.75 mm	25.699 in

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 2-1/2"
Bolt spacing	Bs	=	142.16 mm	5.597 in
Nominal bolt diameter	a	=	63.50 mm	2.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	8 456 915 N	1 901 189.99 lbf
Design bolt load for the test condition	$W_t = 0.785G^2P_t + 2b\pi \mathrm{Gm}P_t$	=	0 N	0 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	1 023 113 N	230 004.85 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right), \left(\frac{W_t + F_{At} + \frac{4M_{Et}}{G}}{S_{bt}}\right)\right]$	=	52 384.3 mm²	81.196 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	55 380.5 mm²	85.840 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	26 706.3 mm²	41.395 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	9 267 772 N	2 083 477.94 lbf
				Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.61575
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2 \log_{10} K}{K^2 - 1}\right)\right]$	=		4.22139
Stress factor T	$T = \frac{K^2(1 + 8.55246 \log_{10} K) - 1}{(1.04720 + 1.9448K^2)(K-1)}$	=		1.66082
Stress factor U	$U = \frac{K^2(1 + 8.55246 \log_{10} K) - 1}{1.36136(K^2 - 1)(K-1)}$	=		4.63888
Stress factor Z	$Z = \frac{(K^2 + 1)}{(K^2 - 1)}$	=		2.24174
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	169.04 mm	6.655 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.26667
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.40227
Flange stress factor for integral type flanges	$F = \left(\begin{array}{l} 0.897697 - 0.297012X_g + 9.5257(10^{-3})X_g + \\ 0.123586(X_g)^2 + 0.0358580(X_h)^2 - 0.194422(X_g)(X_h) - \\ 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - \\ 0.0377693(X_g)(X_h)^2 + 0.0273791(X_g)^2(X_h) \end{array} \right)$	=		0.87850
Flange stress factor for integral type flanges	$V = \left(\begin{array}{l} 0.500244 + \frac{0.227914}{X_g} - 1.87071X_h - \frac{0.344410}{X_g^2} + 2.49189X_h^2 + \\ 0.873446\left(\frac{X_h}{X_g}\right) + \frac{0.189953}{X_g^3} - 1.06082X_h^3 - 1.49970\left(\frac{X_h^2}{X_g}\right) + \\ 0.719413\left(\frac{X_h}{X_g^2}\right) \end{array} \right)$	=		0.40681
Hub stress correction factor for integral flanges	$f = \max\left[1.0, \left(\frac{\begin{array}{l} 0.0927779 - 0.0336633X_g + 0.964176X_g^2 + \\ 0.0566286X_h + 0.347076X_h^2 - 4.18699X_h^3 \end{array}}{\begin{array}{l} 1 - 5.96093(10^{-3})X_g + 1.62904X_h + \\ 3.49329X_h^2 + 1.39052X_h^3 \end{array}} \right)\right]$	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00520

Stress factor d	$d = \frac{U g_0^2 h_o}{V}$	=		3 903 371.60229
Stress factor L	$L = \frac{te + 1}{T} + \frac{t^3}{d}$	=		2.39272
Gasket load for the operating condition	$H_G = W_o - H$	=	2 290 461 N	514 916.01 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785 B'^2 P$	=	5 835 736 N	1 311 925.51 lbf
Total hydrostatic end force	$H = 0.785 G^2 P$	=	6 166 454 N	1 386 273.97 lbf
Difference	$H_T = H - H_D$	=	330 718 N	74 348.46 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	106.50 mm	4.193 in
Moment arm for load HG	$h_G = \frac{C - G}{2}$	=	126.13 mm	4.966 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C - B'}{2} + h_G\right]$	=	130.56 mm	5.140 in
	$A_R = 0.5(A - B')$	=	195.50 mm	7.697 in
Average of the hub thicknesses	$G_{avg} = 0.5(g_0' + g_1')$	=	51.00 mm	2.008 in
	AA = AR	=	195.50 mm	7.697 in
	BB = t	=	170.00 mm	6.693 in
	CC = h	=	68.00 mm	2.677 in
	DDG = Gavg	=	51.00 mm	2.008 in
Moment of inertia KAB	$K_{AB} = (A_A B_B^3)\left[\frac{1}{3} - 0.21\left(\frac{B_B}{A_A}\right)\left(1 - \frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	153 126 579 mm^4	367.888 in^4
Moment of inertia KCD	$K_{CD} = (C_C D_{DG}^3)\left[\frac{1}{3} - 0.105\left(\frac{D_{DG}}{C_C}\right)\left(1 - \frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	2 297 581 mm^4	5.520 in^4
Bending moment of inertia of the flange cross-section	$I = \frac{0.0874 L g_0'^2 h_0 B'}{V}$	=	111 738 738 mm^4	268.453 in^4
Cross-section polar moment of inertia	$I_P = K_{AB} + K_{CD}$	=	155 424 159 mm^4	373.408 in^4

Flange moments

Nominal bolt diameter	dB	=	63.50 mm	2.500 in
TEMA Load concentration factor	$\mathrm{cF} = \mathrm{MAX}\left[\sqrt{\frac{\frac{\pi \cdot C}{n}}{2\mathrm{dB} + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4 M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = \mathrm{cF} \cdot \mathrm{abs}[(H_D h_D + H_T h_T + H_G h_G + M_{oe}) F_s]$	=	953 575.6 N·m	8 439 854.3 lbf·in
Flange design moment for the gasket seating condition	$M_g = \mathrm{cF} \cdot \frac{W_g (C - G) F_s}{2}$	=	1 168 920.1 N·m	10 345 814.1 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	692.00 mm	27.244 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	177.26 MPa	25 709.2 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33 te + 1) M_o}{L t^2 B'}$	=	47.23 MPa	6 850.7 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	113.46 MPa	16 456.5 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \frac{fM_g}{Lg_1^2B_1'}$	=	217.29 MPa	31 515.1 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2B'}$	=	57.90 MPa	8 397.8 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2B'} - ZS_R$	=	139.09 MPa	20 172.9 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR	=	0.30
Flange rigidity index	$J = \frac{52.14VM_o}{LE_{yo}g_0^2K_Rh_o}$	=	0.45833

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2K_Rh_o}$	=	0.47970

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min	=	41.57 mm	1.637 in

g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	26.18 MPa	3 796.5 psi
Hot & corroded (flange)	=	19.00 MPa	2 756.4 psi
New & cold (bolts)	=	20.77 MPa	3 011.8 psi
Hot & corroded (bolts)	=	19.49 MPa	2 826.9 psi

External pressure

Allowable stresses	Flange	Hub	Bolting
Design condition	Sfo=207.00 MPa / 30 022.8 psi	Sno=207.00 MPa / 30 022.8 psi	Sbo=172.00 MPa / 24 946.5 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=172.00 MPa / 24 946.5 psi

External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Modulus of Elasticity at the operating load case temperature	Eyo	=	210 350.00 MPa	30 508 688.2 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 2-1/2"
Bolt spacing	Bs	=	142.16 mm	5.597 in
Nominal bolt diameter	a	=	63.50 mm	2.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	47 247 N	10 621.46 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	1 023 113 N	230 004.85 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	5 948.3 mm²	9.220 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	55 380.5 mm²	85.840 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	26 706.3 mm²	41.395 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	5 274 282 N	1 185 705.72 lbf

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.61575
Stress factor Y	$Y=\frac{1}{K-1}\left[0.66845+5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		4.22139
Stress factor T	$T=\frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.66082
Stress factor U	$U=\frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		4.63888
Stress factor Z	$Z=\frac{(K^2+1)}{(K^2-1)}$	=		2.24174
Hub length parameter	$h_o=\sqrt{Bg_0'}$	=	169.04 mm	6.655 in
Hub thickness ratio	$X_g=\frac{g_1'}{g_0'}$	=		1.26667
Hub length ratio	$X_h=\frac{h}{h_0}$	=		0.40227
Flange stress factor for integral type flanges	$F=\begin{vmatrix}0.897697-0.297012X_g+9.5257(10^{-3})X_g+\\ 0.123586(X_g)^2+0.0358580(X_h)^2-0.194422(X_g)(X_h)-\\ 0.0181259(X_g)^3+0.0129360(X_h)^3-\\ 0.0377693(X_g)(X_h)^2+0.0273791(X_g)^2(X_h)\end{vmatrix}$	=		0.87850
Flange stress factor for integral type flanges	$V=\begin{vmatrix}0.500244+\frac{0.227914}{X_g}-1.87071X_h-\frac{0.344410}{X_g^2}+2.49189X_h^2+\\ 0.873446\left(\frac{X_h}{X_g}\right)+\frac{0.189953}{X_g^3}-1.06082X_h^3-1.49970\left(\frac{X_h^2}{X_g}\right)+\\ 0.719413\left(\frac{X_h}{X_g^2}\right)\end{vmatrix}$	=		0.40681
Hub stress correction factor for integral flanges	$f=\max\left[1.0,\left(\frac{0.0927779-0.0336633X_g+0.964176X_g^2+0.0566286X_h+0.347076X_h^2-4.18699X_h^3}{1-5.96093(10^{-3})X_g+1.62904X_h+3.49329X_h^2+1.39052X_h^3}\right)\right]$	=		1.00000
Stress factor e	$e=\frac{F}{h_o}$	=		0.00520
Stress factor d	$d=\frac{Ug_0^2h_o}{V}$	=		3 903 371.60229
Stress factor L	$L=\frac{te+1}{T}+\frac{t^3}{d}$	=		2.39272
Gasket load for the operating condition	$H_G=W_o-H$	=	12 796 N	2 876.70 lbf
Total hydrostatic end force on the area inside of the flange	$H_D=0.785B^2P$	=	32 603 N	7 329.39 lbf
Total hydrostatic end force	$H=0.785G^2P$	=	34 450 N	7 744.76 lbf
Difference	$H_T=H-H_D$	=	1 848 N	415.37 lbf
Moment arm for load HD	$\mathrm{hD}=\frac{C-B'-g_1'}{2}$	=	106.50 mm	4.193 in
Moment arm for load HG	$h_G=\frac{C-G}{2}$	=	126.13 mm	4.966 in
Moment arm for load HT	$h_T=\frac{1}{2}\left[\frac{C-B'}{2}+h_G\right]$	=	130.56 mm	5.140 in
	$A_R=0.5(A-B')$	=	195.50 mm	7.697 in
Average of the hub thicknesses	$G_{avg}=0.5(g_0'+g_1')$	=	51.00 mm	2.008 in
	AA = AR	=	195.50 mm	7.697 in
	BB = t	=	170.00 mm	6.693 in
	CC = h	=	68.00 mm	2.677 in
	DDG = Gavg	=	51.00 mm	2.008 in
Moment of inertia KAB	$K_{AB}=(A_AB_B^3)\left[\frac{1}{3}-0.21\left(\frac{B_B}{A_A}\right)\left(1-\frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	153 126 579 mm^4	367.888 in^4
Moment of inertia KCD	$K_{CD}=(C_CD_{DG}^3)\left[\frac{1}{3}-0.105\left(\frac{D_{DG}}{C_C}\right)\left(1-\frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	2 297 581 mm^4	5.520 in^4

Bending moment of inertia of the flange cross-section	$I = \frac{0.0874 L g_0{}^2 h_0 B'}{V}$	=	111 738 738 mm^4	268.453 in^4
Cross-section polar moment of inertia	$I_p = K_{AB} + K_{CD}$	=	155 424 159 mm^4	373.408 in^4

Flange moments

Nominal bolt diameter	dB	=	63.50 mm	2.500 in
TEMA Load concentration factor	$cF = MAX\left[\sqrt{\frac{\frac{\pi \cdot C}{n}}{2dB + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = abs[\,(H_D(h_D - h_G) + H_T(h_T - h_G) + M_{oe})F_s\,]$	=	631.7 N·m	5 591.1 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_g F_s$	=	665 231.6 N·m	5 887 795.0 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	692.00 mm	27.244 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	0.12 MPa	17.0 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1)M_o}{L t^2 B'}$	=	0.03 MPa	4.5 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	0.08 MPa	10.9 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \frac{f M_g}{L g_1^2 B_1'}$	=	123.66 MPa	17 935.2 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te + 1)M_g}{L t^2 B'}$	=	32.95 MPa	4 779.2 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{Y M_g}{t^2 B'} - Z S_R$	=	79.15 MPa	11 480.4 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR	=		0.30
Flange rigidity index	$J = \frac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=		0.00026

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=		0.27300

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Flange Out Tube Side (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Flange Out Tube Side

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	45.00 mm	1.772 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Welding neck flange - Flange Out Tube Side (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Flange Out Tube Side

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	45.00 mm	1.772 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Validation warnings:

- Gasket overloaded: Ab > AbMax

Welding neck flange - Shell Flange

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

Design data

Internal design temperature	T	=	420.00 °C	788.00 °F
Internal design pressure	P	=	14.22 MPa	2 062.4 psi
External design temperature	Te	=	20.00 °C	68.00 °F
External design pressure	Pe	=	0.10 MPa	14.9 psi
Joint efficiency	E	=		1.00

Flange material **SA-387 22 2 - Plate**

Shell material **SA-387 22 2 - Plate**

Bolting material **SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting**

Gasket **Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=158.60 MPa / 23 003.0 psi	Sno=158.60 MPa / 23 003.0 psi	Sbo=138.00 MPa / 20 015.2 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=138.00 MPa / 20 015.2 psi
Test condition	Sft= /	Snt= /	Sbt= /

Internal pressure	Pd	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P	=	14.22 MPa	2 062.4 psi
Modulus of Elasticity at the operating load case temperature	Eyo	=	182 400.00 MPa	26 454 883.4 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi
Corrosion allowance	c	=	3.00 mm	0.118 in
Flange external diameter	A	=	1 900.00 mm	74.803 in
Inside diameter	B	=	1 277.00 mm	50.276 in
Bolt circle	C	=	1 700.00 mm	66.929 in
Flange thickness	t	=	300.00 mm	11.811 in
Mean gasket diameter	Gmean	=	1 297.00 mm	51.063 in
Thickness of the hub at the small end	g0	=	65.00 mm	2.559 in
Thickness of the hub at the large end	g1	=	100.00 mm	3.937 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	62.00 mm	2.441 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	97.00 mm	3.819 in
Hub length	h	=	120.00 mm	4.724 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	1 000.00 MPa	145 037.7 psi
Gasket contact width	N	=	20.00 mm	0.787 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	10.00 mm	0.394 in
Conversion factor for length	Cul	=		25.40000
Effective gasket contact width	$b = 0.5C_{\mathrm{ul}}\sqrt{\frac{b_0}{C_{\mathrm{ul}}}}$	=	7.97 mm	0.314 in
Outside diameter of the gasket contact area	$G_c = G_{\mathrm{mean}} + N$	=	1 317.00 mm	51.850 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	1 301.06 mm	51.223 in

Bolt loads

Number of bolts	n	=		28
Bolt type		=		ANSI_TEMA 4-1/2"
Bolt spacing	Bs	=	190.74 mm	7.509 in
Nominal bolt diameter	a	=	114.30 mm	4.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	22 832 134 N	5 132 867.40 lbf
Design bolt load for the test condition	$W_t = 0.785G^2P_t + 2b\pi \mathrm{Gm}P_t$	=	0 N	0 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	32 571 288 N	7 322 316.25 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_g}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right), \left(\frac{W_t + F_{At} + \frac{4M_{Et}}{G}}{S_{bt}}\right)\right]$	=	236 023.8 mm²	365.838 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	261 393.0 mm²	405.160 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	1 184 756.2 mm²	1 836.376 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	34 321 763 N	7 715 838.57 lbf
				Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.48090
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2 \log_{10} K}{K^2 - 1}\right)\right]$	=		5.11631
Stress factor T	$T = \frac{K^2(1 + 8.55246 \log_{10} K) - 1}{(1.04720 + 1.9448K^2)(K-1)}$	=		1.71899
Stress factor U	$U = \frac{K^2(1 + 8.55246 \log_{10} K) - 1}{1.36136(K^2 - 1)(K-1)}$	=		5.62231
Stress factor Z	$Z = \frac{(K^2 + 1)}{(K^2 - 1)}$	=		2.67634
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	282.04 mm	11.104 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.56452
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.42547
Flange stress factor for integral type flanges	$F = \left(\begin{array}{l} 0.897697 - 0.297012X_g + 9.5257(10^{-3})X_g + \\ 0.123586(X_g)^2 + 0.0358580(X_h)^2 - 0.194422(X_g)(X_h) - \\ 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - \\ 0.0377693(X_g)(X_h)^2 + 0.0273791(X_g)^2(X_h) \end{array}\right)$	=		0.85520
Flange stress factor for integral type flanges	$V = \left(\begin{array}{l} 0.500244 + \frac{0.227914}{X_g} - 1.87071X_h - \frac{0.344410}{X_g^2} + 2.49189X_h^2 + \\ 0.873446\left(\frac{X_h}{X_g}\right) + \frac{0.189953}{X_g^3} - 1.06082X_h^3 - 1.49970\left(\frac{X_h^2}{X_g}\right) + \\ 0.719413\left(\frac{X_h}{X_g^2}\right) \end{array}\right)$	=		0.31733
Hub stress correction factor for integral flanges	$f = \max\left[1.0, \left(\frac{\begin{array}{l} 0.0927779 - 0.0336633X_g + 0.964176X_g^2 + \\ 0.0566286X_h + 0.347076X_h^2 - 4.18699X_h^3 \end{array}}{\begin{array}{l} 1 - 5.96093(10^{-3})X_g + 1.62904X_h + \\ 3.49329X_h^2 + 1.39052X_h^3 \end{array}}\right)\right]$	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00303

Stress factor d	$d = \frac{U g_0^2 h_o}{V}$	=		19 208 417.78589
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		2.51655
Gasket load for the operating condition	$H_G = W_o - H$	=	3 936 793 N	885 026.11 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785 B'^2 P$	=	18 374 336 N	4 130 714.64 lbf
Total hydrostatic end force	$H = 0.785 G^2 P$	=	18 895 341 N	4 247 841.29 lbf
Difference	$H_T = H - H_D$	=	521 005 N	117 126.65 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	160.00 mm	6.299 in
Moment arm for load HG	$h_G = \frac{C - G}{2}$	=	199.47 mm	7.853 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C - B'}{2} + h_G\right]$	=	203.98 mm	8.031 in
	$A_R = 0.5(A - B')$	=	308.50 mm	12.146 in
Average of the hub thicknesses	$G_{avg} = 0.5(g_0' + g_1')$	=	79.50 mm	3.130 in
	AA = AR	=	308.50 mm	12.146 in
	BB = t	=	300.00 mm	11.811 in
	CC = h	=	120.00 mm	4.724 in
	DDG = Gavg	=	79.50 mm	3.130 in
Moment of inertia KAB	$K_{AB} = (A_A B_B^3)\left[\frac{1}{3} - 0.21\left(\frac{B_B}{A_A}\right)\left(1 - \frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	1 202 261 513 mm^4	2 888.445 in^4
Moment of inertia KCD	$K_{CD} = (C_C D_{DG}^3)\left[\frac{1}{3} - 0.105\left(\frac{D_{DG}}{C_C}\right)\left(1 - \frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	15 908 319 mm^4	38.220 in^4
Bending moment of inertia of the flange cross-section	$I = \frac{0.0874 L g_0'^2 h_0 B'}{V}$	=	964 097 900 mm^4	2 316.254 in^4
Cross-section polar moment of inertia	$I_P = K_{AB} + K_{CD}$	=	1 218 169 832 mm^4	2 926.665 in^4

Flange moments

Nominal bolt diameter	dB	=	114.30 mm	4.500 in
TEMA Load concentration factor	$\mathrm{cF} = \mathrm{MAX}\left[\sqrt{\frac{\frac{\pi \cdot C}{n}}{2\mathrm{dB} + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_P + I}\right]\left[\frac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = \mathrm{cF} \cdot \mathrm{abs}[(H_D h_D + H_T h_T + H_G h_G + M_{oe}) F_s]$	=	3 831 437.5 N·m	33 911 076.2 lbf·in
Flange design moment for the gasket seating condition	$M_g = \mathrm{cF} \cdot \frac{W_g (C - G) F_s}{2}$	=	6 846 117.0 N·m	60 593 235.7 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	1 380.00 mm	54.331 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1}$	=	117.26 MPa	17 006.5 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1) M_o}{L t^2 B'}$	=	29.14 MPa	4 226.0 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	91.78 MPa	13 312.2 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \frac{fM_g}{Lg_1^2 B_1'}$	=	209.52 MPa	30 387.6 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2 B'}$	=	52.06 MPa	7 551.1 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2 B'} - ZS_R$	=	164.00 MPa	23 786.6 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR	=	0.30
Flange rigidity index	$J = \frac{52.14VM_o}{LE_{yo}g_0^2 K_R h_o}$	=	0.42462

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2 K_R h_o}$	=	0.65791

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min	=	63.17 mm	2.487 in

g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	26.08 MPa	3 782.4 psi
Hot & corroded (flange)	=	21.58 MPa	3 129.5 psi
New & cold (bolts)	=	22.47 MPa	3 258.3 psi
Hot & corroded (bolts)	=	22.47 MPa	3 258.3 psi

External pressure

Allowable stresses	Flange	Hub	Bolting
Design condition	Sfo=207.00 MPa / 30 022.8 psi	Sno=207.00 MPa / 30 022.8 psi	Sbo=138.00 MPa / 20 015.2 psi
Seating condition	Sfg=207.00 MPa / 30 022.8 psi	Sng=207.00 MPa / 30 022.8 psi	Sbg=138.00 MPa / 20 015.2 psi

External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Modulus of Elasticity at the operating load case temperature	Eyo	=	210 350.00 MPa	30 508 688.2 psi
Modulus of Elasticity at the gasket seating load case temperature	Eyg	=	210 350.00 MPa	30 508 688.2 psi

Bolt loads

Number of bolts	n	=		28
Bolt type		=		ANSI_TEMA 4-1/2"
Bolt spacing	Bs	=	190.74 mm	7.509 in
Nominal bolt diameter	a	=	114.30 mm	4.500 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	165 385 N	37 179.93 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	32 571 288 N	7 322 316.25 lbf
External tensile net-section axial force	FA	=	0 N	0 lbf
Absolute value of the external net-section bending moment	ME	=	0 N·m	0 lbf·in
External tensile net-section axial force (test condition)	FAt	=	0 N	0 lbf
External tensile net-section bending moment (test condition)	MEt	=	0 N·m	0 lbf·in
Total minimum required cross-sectional area of the bolts	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	236 023.8 mm²	365.838 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	261 393.0 mm²	405.160 in²
Maximum bolts area for gasket crush	$A_b \max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	1 184 756.2 mm²	1 836.376 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	34 321 763 N	7 715 838.57 lbf

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.48090
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		5.11631
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.71899
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		5.62231
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		2.67634
Hub length parameter	$h_o = \sqrt{Bg_0{}'}$	=	282.04 mm	11.104 in
Hub thickness ratio	$X_g = \frac{g_1{}'}{g_0{}'}$	=		1.56452
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.42547
Flange stress factor for integral type flanges	$F = \begin{pmatrix} 0.897697 - 0.297012X_g + 9.5257(10^{-3})X_g + \\ 0.123586(X_g)^2 + 0.0358580(X_h)^2 - 0.194422(X_g)(X_h) - \\ 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - \\ 0.0377693(X_g)(X_h)^2 + 0.0273791(X_g)^2(X_h) \end{pmatrix}$	=		0.85520
Flange stress factor for integral type flanges	$V = \begin{pmatrix} 0.500244 + \frac{0.227914}{X_g} - 1.87071X_h - \frac{0.344410}{X_g^2} + 2.49189X_h^2 + \\ 0.873446\left(\frac{X_h}{X_g}\right) + \frac{0.189953}{X_g^3} - 1.06082X_h^3 - 1.49970\left(\frac{X_h^2}{X_g}\right) + \\ 0.719413\left(\frac{X_h}{X_g^2}\right) \end{pmatrix}$	=		0.31733
Hub stress correction factor for integral flanges	$f = \max\left[1.0, \left(\frac{0.0927779 - 0.0336633X_g + 0.964176X_g^2 + 0.0566286X_h + 0.347076X_h^2 - 4.18699X_h^3}{1 - 5.96093(10^{-3})X_g + 1.62904X_h + 3.49329X_h^2 + 1.39052X_h^3}\right)\right]$	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00303
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		19 208 417.78589
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		2.51655
Gasket load for the operating condition	$H_G = W_o - H$	=	28 516 N	6 410.69 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785B^2P$	=	133 095 N	29 920.84 lbf
Total hydrostatic end force	$H = 0.785G^2P$	=	136 868 N	30 769.24 lbf
Difference	$H_T = H - H_D$	=	3 774 N	848.41 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1{}'}{2}$	=	160.00 mm	6.299 in
Moment arm for load HG	$h_G = \frac{C - G}{2}$	=	199.47 mm	7.853 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C - B'}{2} + h_G\right]$	=	203.98 mm	8.031 in
	$A_R = 0.5(A - B')$	=	308.50 mm	12.146 in
Average of the hub thicknesses	$G_{avg} = 0.5(g_0{}' + g_1{}')$	=	79.50 mm	3.130 in
	AA = AR	=	308.50 mm	12.146 in
	BB = t	=	300.00 mm	11.811 in
	CC = h	=	120.00 mm	4.724 in
	DDG = Gavg	=	79.50 mm	3.130 in
Moment of inertia KAB	$K_{AB} = (A_AB_B^3)\left[\frac{1}{3} - 0.21\left(\frac{B_B}{A_A}\right)\left(1 - \frac{1}{12}\left\{\frac{B_B}{A_A}\right\}^4\right)\right]$	=	1 202 261 513 mm^4	2 888.445 in^4
Moment of inertia KCD	$K_{CD} = (C_CD_{DG}^3)\left[\frac{1}{3} - 0.105\left(\frac{D_{DG}}{C_C}\right)\left(1 - \frac{1}{192}\left\{\frac{D_{DG}}{C_C}\right\}^4\right)\right]$	=	15 908 319 mm^4	38.220 in^4

Bending moment of inertia of the flange cross-section	$I = \dfrac{0.0874 L g_0{}^2 h_0 B'}{V}$	=	964 097 900 mm^4	2 316.254 in^4
Cross-section polar moment of inertia	$I_P = K_{AB} + K_{CD}$	=	1 218 169 832 mm^4	2 926.665 in^4

Flange moments

Nominal bolt diameter	dB	=	114.30 mm	4.500 in
TEMA Load concentration factor	$\mathrm{cF} = \mathrm{MAX}\left[\sqrt{\dfrac{\frac{\pi \cdot C}{n}}{2\mathrm{dB} + \frac{6t}{m+0.5}}}, 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Component of the flange design moment resulting from a net section bending moment and/or axial force	$M_{oe} = 4M_E\left[\dfrac{I}{0.3846 I_P + I}\right]\left[\dfrac{h_D}{(C - 2h_D)}\right] + F_A h_D$	=	0 N·m	0 lbf·in
Flange design moment for the operating condition	$M_o = \mathrm{abs}[\,(H_D(h_D - h_G) + H_T(h_T - h_G) + M_{oe})F_s\,]$	=	5 236.0 N·m	46 342.7 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_g F_s$	=	6 846 117.0 N·m	60 593 235.7 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	1 380.00 mm	54.331 in
Flange hub stress - operating condition	$S_H = \dfrac{f M_o}{L g_1^2 B_1'}$	=	0.16 MPa	23.2 psi
Flange radial stress - operating condition	$S_R = \dfrac{(1.33te + 1)M_o}{L t^2 B'}$	=	0.04 MPa	5.8 psi
Flange tangential stress - operating condition	$S_T = \dfrac{Y M_o}{t^2 B'} - Z S_R$	=	0.13 MPa	18.2 psi

SHo ≤ min[1.5Sfo, 2.5Sno]: Ok
SRo ≤ Sfo: Ok
STo ≤ Sfo: Ok
(SHo + SRo) / 2 ≤ Sfo: Ok
(SHo + STo) / 2 ≤ Sfo: Ok

Flange stresses - seating condition

Flange hub stress - gasket seating condition	$S_H = \dfrac{f M_g}{L g_1^2 B_1'}$	=	209.52 MPa	30 387.6 psi
Flange radial stress - gasket seating condition	$S_R = \dfrac{(1.33te + 1)M_g}{L t^2 B'}$	=	52.06 MPa	7 551.1 psi
Flange tangential stress - gasket seating condition	$S_T = \dfrac{Y M_g}{t^2 B'} - Z S_R$	=	164.00 MPa	23 786.6 psi

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
(SHg + SRg) / 2 ≤ Sfg: Ok
(SHg + STg) / 2 ≤ Sfg: Ok

Flange rigidity - operating condition

Rigidity index factor	KR	=		0.30
Flange rigidity index	$J = \dfrac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=		0.00050

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \dfrac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=		0.65791

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Shell Flange (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	114.30 mm	4.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Shell Flange

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	65.00 mm	2.559 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Welding neck flange - Shell Flange (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	114.30 mm	4.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Welding neck flange - Shell Flange

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	65.00 mm	2.559 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Cylindrical shell - Main shell

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

Calculation temperature	T	=	420.00 °C	788.00 °F
Material: SA-387 22 2 - Plate				
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	1 275.00 mm	50.197 in
Length	L	=	3 629.00 mm	142.874 in
Adopted thickness	t	=	65.00 mm	2.559 in
Ligament Efficiency				
Reference figure		=		None
Diameter of tube holes	d	=	0 mm	0 in
Internal pressure				
Allowable stress	S	=	158.60 MPa	23 003.0 psi
Internal pressure	Pi	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	14.22 MPa	2 062.4 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	63.08 mm	2.483 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	20.10 MPa	2 915.0 psi
Hot & corroded		=	14.65 MPa	2 125.4 psi
External pressure				
External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	1 405.00 mm	55.315 in
Max unsupported length	L	=	3 808.40 mm	149.937 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		18.24835
	$C_h = 1.12 \cdot M_x^{-1.058}$	=		0.05186
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	770.23 MPa	111 712.7 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	16.41 MPa	2 380.4 psi
Minimum required thickness	tr	=	9.26 mm	0.365 in
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	17.21 MPa	2 495.6 psi
Hot & corroded	=	16.41 MPa	2 380.4 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Cylindrical shell - Main shell

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	65.00 mm	2.559 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Cylindrical shell - Main shell

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	65.00 mm	2.559 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Validation warnings:

- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]

Reinforcement of opening - Out Shell Side

According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units

Calculation temperature	T	=	420.00 °C	788.00 °F
Nozzle material	**SA-182 F22 3 - Forgings**			
Shell material	**SA-387 22 2 - Plate**			
Pad material	**-**			
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	158.60 MPa	23 003.0 psi
Shell allowable stress at room temperature	S0	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	158.60 MPa	23 003.0 psi
Nozzle allowable stress at room temperature	Sn0	=	207.00 MPa	30 022.8 psi
Shell thickness	t	=	65.00 mm	2.559 in
Nozzle thickness	tn	=	100.00 mm	3.937 in
Nominal wall thickness of the nozzle thinner portion	tn2	=	69.90 mm	2.752 in
Tapering angle	ta	=		45.00 °
Nozzle inside diameter	d	=	457.20 mm	18.000 in
Nozzle outside diameter	Od	=	597.00 mm	23.504 in
Joint efficiency	E	=		1.00000
Nozzle internal corrosion allowance	cni	=	3.00 mm	0.118 in
Nozzle external corrosion allowance	cne	=	0 mm	0 in
Nozzle total corrosion allowance	cn	=	3.00 mm	0.118 in
Nozzle undertolerance	cn'	=	0 mm	0 in
Nozzle position		=		Radial
Nozzle connection		=		Integrally reinforced
Weld joint type		=		7 - Full penetration welds
Offset from shell border		=	3 150.00 mm	124.016 in
Angular offset		=		180.00 °
Width of the reinforcing pad	W	=	0 mm	0 in
Thickness of the reinforcing pad	te	=	0 mm	0 in
Nozzle inside radius	Rn = d/2 + cni + cn'	=	231.60 mm	9.118 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{\text{eff}} = 0.5 \cdot D_i + c'$	=	640.50 mm	25.217 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	881.10 mm	34.689 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	150.00 mm	5.906 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	88.10 mm	3.469 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Corner radius	r	=	10.00 mm	0.394 in
Effective length along the vessel wall (limited by shell offset)	$L_R = \min\left[\sqrt{R_{\text{eff}} \cdot (t - c - c')}, 2R_n\right]$	=	1.60 mm	0.063 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	149.88 mm	5.901 in
	$L_{H2} = L_{pr1}$	=	881.10 mm	34.689 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	496.00 mm	19.528 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	211.88 mm	8.342 in
Effective thickness	$t_{\text{eff}} = t - c - c' + \left(\dfrac{A_5 f_{rp}}{L_R}\right)$	=	62.00 mm	2.441 in
	LI1	=	0 mm	0 in

	LI2	=	0 mm	0 in
	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{\text{eff}})t_{\text{eff}}}}\right\}, 12.0\right]$	=		1.94137
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	99.2 mm²	0.154 in²
Nozzle outs. vessel wall area	$A_2 = (t_n - c_n - c_n')L_h$	=	20 552.7 mm²	31.857 in²
Nozzle material factor	$f_{\text{rn}} = \frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{\text{rp}} = \frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a} = W t_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T = A_1 + f_m(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{\text{rp}} A_5$	=	20 764.4 mm²	32.185 in²
Radius of the nozzle opening	Rnc = Rn	=	231.60 mm	9.118 in
Nozzle radius for force calculation	$R_{\text{xn}} = \frac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$	=	277.28 mm	10.916 in
Shell radius for force calculation	$R_{\text{xs}} = \frac{t_{\text{eff}}}{\ln\left[\frac{R_{\text{eff}} + t_{\text{eff}}}{R_{\text{eff}}}\right]}$	=	671.02 mm	26.418 in
Force from internal pressure in the nozzle	$f_N = P R_{\text{xn}} L_H$	=	835 415 N	187 808.68 lbf
Force from internal pressure in the shell	$f_S = P R_{\text{xs}}(L_R + t_n - c_n - c_n')$	=	940 812 N	211 502.91 lbf
Discontinuity force from internal pressure	$f_Y = P R_{\text{xs}} R_{\text{nc}}$	=	2 209 858 N	496 795.88 lbf
Average primary membrane stress	$\sigma_{\text{avg}} = \frac{(f_N + f_S + f_Y)}{A_T}$	=	191.97 MPa	27 842.4 psi
General primary membrane stress	$\sigma_{\text{circ}} = \frac{P R_{\text{xs}}}{t_{\text{eff}}}$	=	153.90 MPa	22 321.1 psi
Allowable stress	$S_{\text{allow}} = 1.5SE$	=	237.90 MPa	34 504.5 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{\text{avg}} - \sigma_{\text{circ}}), \sigma_{\text{circ}}]$	=	230.04 MPa	33 363.8 psi
				r ≥ min[0.25t,3mm(0.125in)]: Ok
				PL ≤ Sallow: Ok
	$P_{\max 1} = \frac{S_{\text{allow}}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{\text{eff}}}}$	=	14.71 MPa	2 132.9 psi
	$P_{\max 2} = S\left(\frac{t - c - c'}{R_{\text{xs}}}\right)$	=	14.65 MPa	2 125.4 psi
Area resisting pressure	$A_p = \frac{f_N + f_S + f_Y}{P}$	=	280 322.4 mm²	434.501 in²
Nozzle maximum allowable pressure (bottom)	$P_{\max} = \min[P_{\max 1}, P_{\max 2}]$	=	14.65 MPa	2 125.4 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2 =	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \frac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$ =		1.41883
Nozzle to shell weld length	$L_\tau = \frac{\pi}{2}(R_n + t_n - c_n - c_n')$ =	516.16 mm	20.321 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$ =	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$ =	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$ =	0 mm	0 in
Welds force	$f_{\text{welds}} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$ =	1 205 907 N	271 098.57 lbf
Nozzle to shell groove weld depth	tw1 = twall =	62.00 mm	2.441 in
Average effective shear stress	$\tau = \frac{f_{\text{welds}}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$ =	55.10 MPa	7 992.3 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

External pressure

External design temperature	Te =	20.00 °C	68.00 °F
Allowable stress from Annex 3.A for the vessel at the design temperature	S =	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn =	207.00 MPa	30 022.8 psi
Nozzle inside radius	Rn = d/2 + cni + cn' =	231.60 mm	9.118 in
Shell inside diameter	Di =	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{\text{eff}} = 0.5 \cdot D_i + c'$ =	640.50 mm	25.217 in
Nozzle projection from the outside of the vessel wall	Lpr1 =	881.10 mm	34.689 in
Nozzle projection from the inside of the vessel wall	Lpr2 =	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3 =	150.00 mm	5.906 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4 =	88.10 mm	3.469 in
Weld leg length of the outside nozzle fillet weld	L41 =	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42 =	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43 =	0 mm	0 in
Effective length along the vessel wall (limited by shell offset)	$L_R = \min\left[\sqrt{R_{\text{eff}} \cdot (t - c - c')},\ 2R_n\right]$ =	1.60 mm	0.063 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$ =	149.88 mm	5.901 in
	$L_{H2} = L_{pr1}$ =	881.10 mm	34.689 in
	$L_{H3} = 8(t - c - c' + t_e)$ =	496.00 mm	19.528 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$ =	211.88 mm	8.342 in
Effective thickness	$t_{\text{eff}} = t - c - c' + \left(\frac{A_5 f_{rp}}{L_R}\right)$ =	62.00 mm	2.441 in
	LI1 =	0 mm	0 in
	LI2 =	0 mm	0 in
	LI3 =	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$ =	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{\text{eff}}) t_{\text{eff}}}}\right\},\ 12.0\right]$ =		1.94137
Area contributed by the vessel wall	$A_1 = ((t - c - c') L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85},\ 1.0\right]$ =	99.2 mm²	0.154 in²
Nozzle outs. vessel wall area	$A_2 = (t_n - c_n - c_n') L_h$ =	20 552.7 mm²	31.857 in²
Nozzle material factor	$f_{rn} = \frac{S_n}{S}$ =		1.00000
Pad material factor	$f_{rp} = \frac{S_p}{S}$ =		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n') L_I$ =	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5 L_{41}^2$ =	112.5 mm²	0.174 in²

Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$ =	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$ =	0 mm²	0 in²
	$A_{5a} = W t_e$ =	0 mm²	0 in²
	A5b =	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$ =	0 mm²	0 in²
Total area	$A_T = A_1 + f_m(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}\ A_5$ =	20 764.4 mm²	32.185 in²
Radius of the nozzle opening	Rnc = Rn =	231.60 mm	9.118 in
Nozzle radius for force calculation	$R_{xn} = \dfrac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$ =	277.28 mm	10.916 in
Shell radius for force calculation	$R_{xs} = \dfrac{t_{eff}}{\ln\left[\frac{R_{eff} + t_{eff}}{R_{eff}}\right]}$ =	671.02 mm	26.418 in
Force from internal pressure in the nozzle	$f_N = P R_{xn} L_H$ =	6 051 N	1 360.39 lbf
Force from internal pressure in the shell	$f_S = P R_{xs}(L_R + t_n - c_n - c_n')$ =	6 815 N	1 532.02 lbf
Discontinuity force from internal pressure	$f_Y = P R_{xs} R_{nc}$ =	16 007 N	3 598.54 lbf
Average primary membrane stress	$\sigma_{avg} = \dfrac{(f_N + f_S + f_Y)}{A_T}$ =	1.39 MPa	201.7 psi
General primary membrane stress	$\sigma_{circ} = \dfrac{P R_{xs}}{t_{eff}}$ =	1.11 MPa	161.7 psi
Allowable stress	$S_{allow} = F_{ha\ shell}$ =	185.96 MPa	26 971.6 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{avg} - \sigma_{circ}), \sigma_{circ}]$ =	1.67 MPa	241.7 psi
			PL ≤ Sallow: Ok
	$P_{max1} = \dfrac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{eff}}}$ =	11.50 MPa	1 667.3 psi
	$P_{max2} = S\left(\dfrac{t - c - c'}{R_{xs}}\right)$ =	19.13 MPa	2 774.0 psi
Area resisting pressure	$A_p = \dfrac{f_N + f_S + f_Y}{P}$ =	280 322.4 mm²	434.501 in²
Nozzle maximum allowable pressure (bottom)	$P_{max} = \min[P_{max1}, P_{max2}]$ =	11.50 MPa	1 667.3 psi
			P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2 =	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \dfrac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$ =		1.41883
Nozzle to shell weld length	$L_\tau = \dfrac{\pi}{2}(R_n + t_n - c_n - c_n')$ =	516.16 mm	20.321 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$ =	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$ =	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$ =	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n(A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$ =	8 735 N	1 963.70 lbf
Nozzle to shell groove weld depth	tw1 = twall =	62.00 mm	2.441 in
Average effective shear stress	$\tau = \dfrac{f_{welds}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$ =	0.40 MPa	57.9 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok
τ ≤ S: Ok

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Date ________ Calc. ________ Contr.________ Appr.________

Reinforcement of opening - In Shell Side

According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units

Calculation temperature	T	=	420.00 °C	788.00 °F
Nozzle material **SA-182 F22 3 - Forgings**				
Shell material **SA-387 22 2 - Plate**				
Pad material **-**				
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	158.60 MPa	23 003.0 psi
Shell allowable stress at room temperature	S0	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	158.60 MPa	23 003.0 psi
Nozzle allowable stress at room temperature	Sn0	=	207.00 MPa	30 022.8 psi
Shell thickness	t	=	65.00 mm	2.559 in
Nozzle thickness	tn	=	100.00 mm	3.937 in
Nominal wall thickness of the nozzle thinner portion	tn2	=	69.90 mm	2.752 in
Tapering angle	ta	=		45.00 °
Nozzle inside diameter	d	=	457.20 mm	18.000 in
Nozzle outside diameter	Od	=	597.00 mm	23.504 in
Joint efficiency	E	=		1.00000
Nozzle internal corrosion allowance	cni	=	3.00 mm	0.118 in
Nozzle external corrosion allowance	cne	=	0 mm	0 in
Nozzle total corrosion allowance	cn	=	3.00 mm	0.118 in
Nozzle undertolerance	cn'	=	0 mm	0 in
Nozzle position		=		Radial
Nozzle connection		=		Integrally reinforced
Weld joint type		=		7 - Full penetration welds
Offset from shell border		=	420.00 mm	16.535 in
Angular offset		=		0 °
Width of the reinforcing pad	W	=	0 mm	0 in
Thickness of the reinforcing pad	te	=	0 mm	0 in
Nozzle inside radius	Rn = d/2 + cni + cn'	=	231.60 mm	9.118 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{\text{eff}} = 0.5 \cdot D_i + c'$	=	640.50 mm	25.217 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	881.10 mm	34.689 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	150.00 mm	5.906 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	88.10 mm	3.469 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Corner radius	r	=	10.00 mm	0.394 in
Effective length along the vessel wall (limited by shell offset)	$L_R = \min\left[\sqrt{R_{\text{eff}} \cdot (t - c - c')}, 2R_n\right]$	=	91.40 mm	3.598 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	149.88 mm	5.901 in
	$L_{H2} = L_{pr1}$	=	881.10 mm	34.689 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	496.00 mm	19.528 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	211.88 mm	8.342 in
Effective thickness	$t_{\text{eff}} = t - c - c' + \left(\frac{A_5 f_{rp}}{L_R}\right)$	=	62.00 mm	2.441 in
	LI1	=	0 mm	0 in

	LI2	=	0 mm	0 in
	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{\text{eff}})t_{\text{eff}}}}\right\}, 12.0\right]$	=		1.94137
Area contributed by the vessel wall	$A_1 = ((t - c - c')L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	5 666.8 mm²	8.784 in²
Nozzle outs. vessel wall area	$A_2 = (t_n - c_n - c_n')L_h$	=	20 552.7 mm²	31.857 in²
Nozzle material factor	$f_{rn} = \frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{rp} = \frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n')L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5L_{41}^2$	=	112.5 mm²	0.174 in²
Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a} = W t_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp} A_5$	=	26 332.0 mm²	40.815 in²
Radius of the nozzle opening	Rnc = Rn	=	231.60 mm	9.118 in
Nozzle radius for force calculation	$R_{xn} = \frac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$	=	277.28 mm	10.916 in
Shell radius for force calculation	$R_{xs} = \frac{t_{\text{eff}}}{\ln\left[\frac{R_{\text{eff}} + t_{\text{eff}}}{R_{\text{eff}}}\right]}$	=	671.02 mm	26.418 in
Force from internal pressure in the nozzle	$f_N = P R_{xn} L_H$	=	835 415 N	187 808.68 lbf
Force from internal pressure in the shell	$f_S = P R_{xs}(L_R + t_n - c_n - c_n')$	=	1 797 657 N	404 129.29 lbf
Discontinuity force from internal pressure	$f_Y = P R_{xs} R_{nc}$	=	2 209 858 N	496 795.88 lbf
Average primary membrane stress	$\sigma_{\text{avg}} = \frac{(f_N + f_S + f_Y)}{A_T}$	=	183.92 MPa	26 675.0 psi
General primary membrane stress	$\sigma_{\text{circ}} = \frac{P R_{xs}}{t_{\text{eff}}}$	=	153.90 MPa	22 321.1 psi
Allowable stress	$S_{\text{allow}} = 1.5SE$	=	237.90 MPa	34 504.5 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{\text{avg}} - \sigma_{\text{circ}}), \sigma_{\text{circ}}]$	=	213.94 MPa	31 028.9 psi
				r ≥ min[0.25t,3mm(0.125in)]: Ok
				PL ≤ Sallow: Ok
	$P_{\max 1} = \frac{S_{\text{allow}}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{\text{eff}}}}$	=	15.81 MPa	2 293.4 psi
	$P_{\max 2} = S\left(\frac{t - c - c'}{R_{xs}}\right)$	=	14.65 MPa	2 125.4 psi
Area resisting pressure	$A_p = \frac{f_N + f_S + f_Y}{P}$	=	340 580.3 mm²	527.900 in²
Nozzle maximum allowable pressure (bottom)	$P_{\max} = \min[P_{\max 1}, P_{\max 2}]$	=	14.65 MPa	2 125.4 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2	=	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \frac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$	=		1.41883
Nozzle to shell weld length	$L_\tau = \frac{\pi}{2}(R_n + t_n - c_n - c_n')$	=	516.16 mm	20.321 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$	=	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$	=	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$	=	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$	=	1 205 907 N	271 098.57 lbf
Nozzle to shell groove weld depth	tw1 = twall	=	62.00 mm	2.441 in
Average effective shear stress	$\tau = \frac{f_{welds}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$	=	55.10 MPa	7 992.3 psi

tc ≥ min[0.7tn, 6mm (0.25in)]: Ok

τ ≤ S: Ok

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
Allowable stress from Annex 3.A for the vessel at the design temperature	S	=	207.00 MPa	30 022.8 psi
Allowable stress from Annex 3.A for the nozzle at the design temperature	Sn	=	207.00 MPa	30 022.8 psi
Nozzle inside radius	Rn = d/2 + cni + cn'	=	231.60 mm	9.118 in
Shell inside diameter	Di	=	1 275.00 mm	50.197 in
Effective radius of the shell	$R_{eff} = 0.5 \cdot D_i + c'$	=	640.50 mm	25.217 in
Nozzle projection from the outside of the vessel wall	Lpr1	=	881.10 mm	34.689 in
Nozzle projection from the inside of the vessel wall	Lpr2	=	0 mm	0 in
Length of variable thickness from the outside of the vessel wall	Lpr3	=	150.00 mm	5.906 in
Nozzle projection from the outside of the vessel wall to tn2	Lpr4	=	88.10 mm	3.469 in
Weld leg length of the outside nozzle fillet weld	L41	=	15.00 mm	0.591 in
Weld leg length of the pad to vessel fillet weld	L42	=	0 mm	0 in
Weld leg length of the inside nozzle fillet weld	L43	=	0 mm	0 in
Effective length along the vessel wall (limited by shell offset)	$L_R = \min\left[\sqrt{R_{eff} \cdot (t - c - c')},\ 2R_n\right]$	=	91.40 mm	3.598 in
	$L_{H1} = \min[1.5(t - c - c'),\ t_e] + \sqrt{R_n(t_n - c_n - c_n')}$	=	149.88 mm	5.901 in
	$L_{H2} = L_{pr1}$	=	881.10 mm	34.689 in
	$L_{H3} = 8(t - c - c' + t_e)$	=	496.00 mm	19.528 in
Effective length along the nozzle wall outside the vessel	$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t - c - c'$	=	211.88 mm	8.342 in
Effective thickness	$t_{eff} = t - c - c' + \left(\frac{A_5 f_{rp}}{L_R}\right)$	=	62.00 mm	2.441 in
	LI1	=	0 mm	0 in
	LI2	=	0 mm	0 in
	LI3	=	0 mm	0 in
Effective length along the nozzle wall inside the vessel	$L_I = \min[L_{I1}, L_{I2}, L_{I3}]$	=	0 mm	0 in
	$\lambda = \min\left[\left\{\frac{(2R_n + t_n - c_n - c_n')}{\sqrt{(D_i + t_{eff}) t_{eff}}}\right\}, 12.0\right]$	=		1.94137
Area contributed by the vessel wall	$A_1 = ((t - c - c') L_R) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right]$	=	5 666.8 mm²	8.784 in²
Nozzle outs. vessel wall area	$A_2 = (t_n - c_n - c_n') L_h$	=	20 552.7 mm²	31.857 in²
Nozzle material factor	$f_{rn} = \frac{S_n}{S}$	=		1.00000
Pad material factor	$f_{rp} = \frac{S_p}{S}$	=		0
Nozzle ins. vessel wall area	$A_3 = (t_n - 2c_n - c_n') L_I$	=	0 mm²	0 in²
Area contributed by the outside nozzle fillet weld	$A_{41} = 0.5 L_{41}^2$	=	112.5 mm²	0.174 in²

Area contributed by the pad to vessel fillet weld	$A_{42} = 0.5L_{42}^2$	=	0 mm²	0 in²
Area contributed by the inside nozzle fillet weld	$A_{43} = 0.5L_{43}^2$	=	0 mm²	0 in²
	$A_{5a} = W t_e$	=	0 mm²	0 in²
	A5b	=	0 mm²	0 in²
Area contributed by the reinforcing pad	$A_5 = \min[A_{5a}, A_{5b}]$	=	0 mm²	0 in²
Total area	$A_T = A_1 + f_m(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}\ A_5$	=	26 332.0 mm²	40.815 in²
Radius of the nozzle opening	Rnc = Rn	=	231.60 mm	9.118 in
Nozzle radius for force calculation	$R_{xn} = \dfrac{t_n - c_n - c_n'}{\ln\left[\frac{R_n + t_n - c_n - c_n'}{R_n}\right]}$	=	277.28 mm	10.916 in
Shell radius for force calculation	$R_{xs} = \dfrac{t_{eff}}{\ln\left[\frac{R_{eff} + t_{eff}}{R_{eff}}\right]}$	=	671.02 mm	26.418 in
Force from internal pressure in the nozzle	$f_N = P R_{xn} L_H$	=	6 051 N	1 360.39 lbf
Force from internal pressure in the shell	$f_S = P R_{xs}(L_R + t_n - c_n - c_n')$	=	13 021 N	2 927.31 lbf
Discontinuity force from internal pressure	$f_Y = P R_{xs} R_{nc}$	=	16 007 N	3 598.54 lbf
Average primary membrane stress	$\sigma_{avg} = \dfrac{(f_N + f_S + f_Y)}{A_T}$	=	1.33 MPa	193.2 psi
General primary membrane stress	$\sigma_{circ} = \dfrac{P R_{xs}}{t_{eff}}$	=	1.11 MPa	161.7 psi
Allowable stress	$S_{allow} = F_{ha\ shell}$	=	185.96 MPa	26 971.6 psi
Maximum local primary membrane stress	$P_L = \max[(2\sigma_{avg} - \sigma_{circ}), \sigma_{circ}]$	=	1.55 MPa	224.8 psi
				PL ≤ Sallow: Ok
	$P_{max1} = \dfrac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{eff}}}$	=	12.36 MPa	1 792.7 psi
	$P_{max2} = S\left(\dfrac{t - c - c'}{R_{xs}}\right)$	=	19.13 MPa	2 774.0 psi
Area resisting pressure	$A_p = \dfrac{f_N + f_S + f_Y}{P}$	=	340 580.3 mm²	527.900 in²
Nozzle maximum allowable pressure (bottom)	$P_{max} = \min[P_{max1}, P_{max2}]$	=	12.36 MPa	1 792.7 psi
				P ≤ Pmax: Ok

Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld	tc = L41 / √2	=	10.61 mm	0.418 in
Discontinuity force factor	$k_y = \dfrac{R_{nc} + t_n - c_n - c_n'}{R_{nc}}$	=		1.41883
Nozzle to shell weld length	$L_\tau = \dfrac{\pi}{2}(R_n + t_n - c_n - c_n')$	=	516.16 mm	20.321 in
Throat dimension of the outside nozzle fillet weld	$L_{41T} = 0.7071 L_{41}$	=	10.61 mm	0.418 in
Throat dimension for the pad to vessel fillet weld	$L_{42T} = 0.7071 L_{42}$	=	0 mm	0 in
Throat dimension for inside nozzle fillet weld	$L_{43T} = 0.7071 L_{43}$	=	0 mm	0 in
Welds force	$f_{welds} = \min\left[f_Y k_y,\ 1.5 S_n (A_2 + A_3),\ \frac{\pi}{4} P R_n^2 k_y^2\right]$	=	8 735 N	1 963.70 lbf
Nozzle to shell groove weld depth	tw1 = twall	=	62.00 mm	2.441 in
Average effective shear stress	$\tau = \dfrac{f_{welds}}{L_\tau(0.49 L_{41T} + 0.6 t_{w1} + 0.49 L_{43T})}$	=	0.40 MPa	57.9 psi
				tc ≥ min[0.7tn, 6mm (0.25in)]: Ok
				τ ≤ S: Ok

Validation warnings:

- Gasket overloaded: Ab > AbMax

Standard Long Welding Neck flange - Out Shell Side

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

Flange material	**SA-182 F22 3 - Forgings**
Nozzle material	**SA-182 F22 3 - Forgings**
Bolting material	**SA-193 B16 - Bolting**
Gasket	**Grooved Metal - Stainless steels and nickel-base alloys**

Flange standard / specification		=		ASME B16.5 2013
Flange rating		=		1 500
Nominal size		=		18"
Number of bolts		=		16
Bolt type		=		ANSI_TEMA 2-1/2"
Material group		=		1.10
Calculation temperature	T	=	420.00 °C	788.00 °F
Internal pressure	Pd	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P	=	14.22 MPa	2 062.4 psi
Maximum pressure at temperature allowed by the specifications	Pmax	=	17.67 MPa	2 562.8 psi

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	25.86 MPa	3 750.7 psi
Hot & corroded (flange)	=	17.67 MPa	2 562.8 psi
New & cold (bolts)	=	25.86 MPa	3 750.7 psi
Hot & corroded (bolts)	=	17.67 MPa	2 562.8 psi
New & cold (cylinder)	=	55.23 MPa	8 010.0 psi
Hot & corroded (cylinder)	=	40.25 MPa	5 837.2 psi

External pressure

External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
Maximum pressure at temperature allowed by the specifications	Pmax	=	17.67 MPa	2 562.8 psi

Cylindrical shell

Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	457.20 mm	18.000 in
Length	L	=	1 016.00 mm	40.000 in
Adopted thickness	t	=	69.90 mm	2.752 in
Allowable stress	S	=	158.60 MPa	23 003.0 psi
Internal pressure	Pi	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	14.22 MPa	2 062.4 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	24.72 mm	0.973 in
				t ≥ tr: Ok

External pressure

External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	915.00 mm	36.024 in
Max unsupported length	L	=	1 016.00 mm	40.000 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		5.80744
	$C_h = \frac{0.92}{M_x - 0.579}$	=		0.17596
Elastic buckling stress	$F_{\mathrm{he}} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	4 329.95 MPa	628 006.1 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{\mathrm{ha}}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	27.19 MPa	3 944.0 psi
Minimum required thickness	tr	=	5.81 mm	0.229 in
				t ≥ tr: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Long Welding Neck flange - Out Shell Side (Bolting)

Material		=		SA-193 B16
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Long Welding Neck flange - Out Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	69.90 mm	2.752 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Long Welding Neck flange - Out Shell Side (Bolting)

Material		=		SA-193 B16
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Long Welding Neck flange - Out Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	69.90 mm	2.752 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Standard Long Welding Neck flange - In Shell Side

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

Flange material	**SA-182 F22 3 - Forgings**
Nozzle material	**SA-182 F22 3 - Forgings**
Bolting material	**SA-193 B16 - Bolting**
Gasket	**Grooved Metal - Stainless steels and nickel-base alloys**

Flange standard / specification		=		ASME B16.5 2013
Flange rating		=		1 500
Nominal size		=		18"
Number of bolts		=		16
Bolt type		=		ANSI_TEMA 2-1/2"
Material group		=		1.10
Calculation temperature	T	=	420.00 °C	788.00 °F
Internal pressure	Pd	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P	=	14.22 MPa	2 062.4 psi
Maximum pressure at temperature allowed by the specifications	Pmax	=	17.67 MPa	2 562.8 psi

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	25.86 MPa	3 750.7 psi
Hot & corroded (flange)	=	17.67 MPa	2 562.8 psi
New & cold (bolts)	=	25.86 MPa	3 750.7 psi
Hot & corroded (bolts)	=	17.67 MPa	2 562.8 psi
New & cold (cylinder)	=	55.23 MPa	8 010.0 psi
Hot & corroded (cylinder)	=	40.25 MPa	5 837.2 psi

External pressure

External design pressure	Pe	=	0.10 MPa	14.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
Maximum pressure at temperature allowed by the specifications	Pmax	=	17.67 MPa	2 562.8 psi

Cylindrical shell

Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	457.20 mm	18.000 in
Length	L	=	1 016.00 mm	40.000 in
Adopted thickness	t	=	69.90 mm	2.752 in
Allowable stress	S	=	158.60 MPa	23 003.0 psi
Internal pressure	Pi	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	14.22 MPa	2 062.4 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	24.72 mm	0.973 in
				t ≥ tr: Ok

External pressure

External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside diameter	Do	=	915.00 mm	36.024 in
Max unsupported length	L	=	1 016.00 mm	40.000 in
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		5.80744
	$C_h = \frac{0.92}{M_x - 0.579}$	=		0.17596
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	4 329.95 MPa	628 006.1 psi
Buckling stress	Fic = Sy	=	310.00 MPa	44 961.7 psi
Yield strength at design temperature	Sy	=	310.00 MPa	44 961.7 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	Fha = Fic / FS	=	185.96 MPa	26 971.6 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	27.19 MPa	3 944.0 psi
Minimum required thickness	tr	=	5.81 mm	0.229 in
				t ≥ tr: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Long Welding Neck flange - In Shell Side (Bolting)

Material		=		SA-193 B16
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Long Welding Neck flange - In Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	69.90 mm	2.752 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

External pressure

Long Welding Neck flange - In Shell Side (Bolting)

Material		=		SA-193 B16
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	63.50 mm	2.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

Long Welding Neck flange - In Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	69.90 mm	2.752 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

U-Tube tubesheet - Tubesheet

According to: Asme VIII Div. 2 Ed. 2015, 4.18 - Metric Units

Operating conditions

	Design temperature		Design pressure	
Shell	Ts =	420.00 °C / 788.00 °F	Ps =	14.22 MPa / 2 062.4 psi
Channel	Tc =	454.00 °C / 849.20 °F	Pt =	18.44 MPa / 2 674.0 psi
Tubesheet	T =	454.00 °C / 849.20 °F		
Tubes	Tt =	454.00 °C / 849.20 °F		

Tubesheet material SA-182 F22 3 - Forgings

Tubesheet design temperature	T	=	454.00 °C	849.20 °F
Modulus of elasticity for tubesheet material at T	E	=	179 600.00 MPa	26 048 777.7 psi
Allowable stress for tubesheet material at T	S	=	149.84 MPa	21 732.5 psi
Allowable primary plus secondary stress for tubesheet material	SPS = 2·Sy	=	461.76 MPa	66 972.6 psi

Tubes material SB-517 Cold drawn/ann. N06600 - Wld. pipe

Tube design temperature	Tt	=	454.00 °C	849.20 °F
Modulus of elasticity for tube material at Tt	Et	=	188 760.00 MPa	27 377 323.4 psi
Allowable stress for tube material at tubesheet design temperature	StT	=	161.18 MPa	23 376.7 psi

Allowable stress of the welded product divided by 0.85

Channel material SA-387 22 2 - Plate

Channel design temperature	Tc	=	454.00 °C	849.20 °F
Modulus of elasticity for channel material at Tc	Ec	=	179 600.00 MPa	26 048 777.7 psi
Poisson's ratio of channel material	vc	=		0.30
Allowable stress for channel material at Tc	Sc	=	149.84 MPa	21 732.5 psi
Allowable primary plus secondary stress for channel material at Tc	SPSc = 2·Sy	=	461.76 MPa	66 972.6 psi

Shell material SA-387 22 2 - Plate

Shell design temperature	Ts	=	420.00 °C	788.00 °F
Modulus of elasticity for shell material at Ts	Es	=	182 400.00 MPa	26 454 883.4 psi
Poisson's ratio of shell material	vs	=		0.30
Allowable stress for shell material at Ts	Ss	=	158.60 MPa	23 003.0 psi
Allowable primary plus secondary stress for shell material at Ts	SPSs = 3·S	=	475.80 MPa	69 009.0 psi

Bolting material SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting

Allowable stress for the bolt evaluated at the design temperature	Sbo	=	129.92 MPa	18 843.3 psi
Allowable stress for the bolt evaluated at the gasket seating temperature	Sbg	=	138.00 MPa	20 015.2 psi

Geometric data

Outside diameter of tubesheet	A	=	1 900.00 mm	74.803 in
Bolt circle diameter	C	=	1 700.00 mm	66.929 in
Shell corrosion allowance	cs	=	3.00 mm	0.118 in
Shell undertolerance	c's	=	0 mm	0 in
Shell thickness	$t_s = t_{\text{shell}} - c_s - c'_s$	=	62.00 mm	2.441 in
Channel corrosion allowance	cc	=	0 mm	0 in
Channel undertolerance	c'c	=	0 mm	0 in
Channel thickness	$t_c = t_{\text{channel}} - c_c - c'_c$	=	85.00 mm	3.346 in
Perimeter of the tube layout	Cp	=	3 764.42 mm	148.206 in
Area enclosed by perimeter Cp	Ap	=	1 127 677.1 mm²	1 747.903 in²
Tubeside corrosion allowance	c_ts	=	0 mm	0 in
Shellside corrosion allowance	c_ss	=	3.00 mm	0.118 in
Tubesheet undertolerance	c'	=	0 mm	0 in
Tubesheet thickness	t_tubesheet	=	230.00 mm	9.055 in
Tubesheet thickness for calculation	$h = t_{\text{tubesheet}} - c_{\text{ts}} - c_{\text{ss}} - c'$	=	227.00 mm	8.937 in
Nominal outside diameter of tubes	dt	=	31.75 mm	1.250 in
Radius to outermost tube hole center	ro	=	599.13 mm	23.588 in
Equivalent diameter of outer tube limit circle	$D_0 = 2r_0 + d_t$	=	1 230.00 mm	48.425 in
Triangular tube pitch	p	=	42.33 mm	1.667 in
Nominal tube wall thickness	tt	=	4.19 mm	0.165 in
Total area of untubed lanes	AL	=	0 mm²	0 in²
Expanded length of tube in tubesheet	ltx	=	227.00 mm	8.937 in
Tube side pass partition groove depth	hg	=	0 mm	0 in
Effective tube side pass partition groove depth	$h'_g = \max[(h_g - c_t), 0.0]$	=	0 mm	0 in
Diameter of shell gasket load reaction	Gs	=	1 301.06 mm	51.223 in
Diameter ratio	$\rho_s = \frac{G_s}{D_o}$	=		1.05777
Inside channel diameter	Dc	=	1 275.00 mm	50.197 in
Diameter ratio	$\rho_c = \frac{D_c}{D_o}$	=		1.03659
	$\beta_c = \frac{[12(1-\nu_c^2)]^{0.25}}{[(D_c+t_c)t_c]^{0.5}}$	=	0.0053 mm^-1	0.136 in^-1
	$k_c = \frac{\beta_c E_c t_c^3}{6(1-\nu_c^2)}$	=	108 005 833 N	24 280 674.98 lbf
	$\lambda_c = \frac{6D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right)$	=	208 391.28 MPa	30 224 599.9 psi
	$\delta_c = \frac{D_c^2}{4E_c t_c}\left(1 - \frac{\nu_c}{2}\right)$	=		0.02263
	$\omega_c = \rho_c k_c \beta_c \delta_c (1 + h\beta_c)$	=	29 984.5 mm²	46.476 in²
Diameter ratio	$K = \frac{A}{D_o}$	=		1.54472
Coefficient	$F = \frac{(1-\nu^*)(\lambda_c + E \cdot \ln[K])}{E^*}$	=		2.33305

Minimum RCB 7.11 thickness

TEMA Class	=	R

t - ct - cs ≥ do: Ok

t ≥ 19.10 mm: Ok

Flanged extension

Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi GmP$	=	22 832 134 N	5 132 867.40 lbf
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	30 162 175 N	6 780 726.10 lbf
Moment arm for load HG	hG = (C-G) / 2	=	199.47 mm	7.853 in
Allowable stress for the tubesheet extension at design temperature	So	=	149.84 MPa	21 732.5 psi
Allowable stress for the tubesheet extension at gasket seating temperature	Sg	=	207.00 MPa	30 022.8 psi
Flanged extension thickness	$t_{fe} = t_{flanged\ extension} - c$	=	773.75 mm	30.463 in
Minimum required thickness of the tubesheet flanged extension (operating)	$hr_o = \sqrt{\frac{1.9W_o h_G}{S_o G}}$	=	210.68 mm	8.295 in
Minimum required thickness of the tubesheet flanged extension (gasket seating)	$hr_g = \sqrt{\frac{1.9W_g h_G}{S_g G}}$	=	206.02 mm	8.111 in
Minimum required thickness of the tubesheet flanged extension.	$h_r = \left(\frac{1.9W h_g}{S \cdot G}\right)^{0.5}$	=	210.68 mm	8.295 in
				tfe ≥ hr: Ok

Tube to tubesheet joints

Fillet weld leg	af	=	10.00 mm	0.394 in
Min. required length of the weld leg(s)	$a_r = \sqrt{(0.75d_o)^2 + 2.73t(d_o - t)f_w f_d} - 0.75d_o$	=	5.89 mm	0.232 in
Allowable stress of the tube	Sa	=	137.00 MPa	19 870.2 psi
Allowable stress of the material to which the tube is welded	St	=	149.84 MPa	21 732.5 psi
Allowable stress in weld	$S_w = \min[S_a, S_t]$	=	137.00 MPa	19 870.2 psi
Fillet weld strength	$F_f = 0.55\pi a_f(d_o + 0.67a_f)S_w$	=	91 018 N	20 461.76 lbf
Groove weld strength	$F_g = 0.85\pi a_g(d_o + 0.67a_g)S_w$	=	0 N	0 lbf
Axial tube strength	$F_t = \pi t(d_o - t)S_a$	=	49 711 N	11 175.45 lbf
Ratio of the design strength to the tube strength	fd	=		1.00000
Ratio of the fillet weld strength to the design strength	$f_f = 1 - \frac{F_g}{f_d F_t}$	=		1.00000
Weld strength factor	$f_w = \frac{S_a}{S_w}$	=		1.00000
Max axial load (pressure only)	Lmax = Ft	=	49 711 N	11 175.45 lbf
Max axial load (pressure or thermally induced)	Lmax = 2Ft	=	99 422 N	22 350.90 lbf
				af ≥ max[ar, t]: Ok

Loading case Design 1: Pt = 18.44 MPa, Ps = -0.10 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear	$\mu = \frac{p - d_t}{p}$	=		0.24998
Effective tube pitch	$p^* = p\left(1 - \frac{4 \cdot \min[A_L, (4D_o p)]}{\pi D_o^2}\right)^{-0.5}$	=	42.33 mm	1.667 in
Tube expansion depth ratio	$\rho = \frac{l_{tx}}{h}$	=		1.00000
Effective tube hole diameter	$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\}, (d_t - 2t_t)\right]$	=	23.37 mm	0.920 in
Effective ligament efficiency for bending	$\mu^* = \frac{p^* - d^*}{p^*}$	=		0.44798
	A0	=		-0.00290
	A1	=		0.21260
	A2	=		3.99060
	A3	=		-6.17300
	A4	=		3.43070
	B0	=		0.99660
	B1	=		-4.19780
	B2	=		9.04780
	B3	=		-7.99550
	B4	=		2.23980
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.47640
Effective Poisson ratio in perforated region	$\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.30322

Tubesheet bending stress

Tubesheet design bold load	W*	=	0 N	0 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	-133 502 N	-30 012.42 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	419 308 N	94 264.11 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	739 340 N	166 210.12 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + \nu^*)(P_s - P_t)}{64}$	=	-708 319 N	-159 236.38 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	739 340 N	166 210.12 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h_g')^2}$	=	192.17 MPa	27 871.7 psi

σ ≤ 2S: Ok

Tubesheet shear stress

|Ps - Pt| ≤ 3.2·S·µ·h/Do: Shear stress is not required to be calculated

Shell and channel stresses

Axial membr. stress in the channel at tubesheet junction	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	64.82 MPa	9 400.7 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1 - \nu^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	301.64 MPa	43 749.5 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	366.46 MPa	53 150.2 psi

Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation	$E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$	=	140 654.57 MPa	20 400 220.3 psi
	$k_c = \dfrac{\beta_c E_c^* t_c^3}{6(1-\nu_c^2)}$	=	84 585 265 N	19 015 522.45 lbf
	$\lambda_c = \dfrac{6D_c k_c}{h^3}\left(1+h\beta_c+\dfrac{h^2\beta_c^2}{2}\right)$	=	163 202.59 MPa	23 670 534.7 psi
	$\omega_c = \rho_c k_c \beta_c \delta_c (1+h\beta_c)$	=	23 482.5 mm²	36.398 in²
Diameter ratio	$K = \dfrac{A}{D_o}$	=		1.54472
Coefficient	$F = \dfrac{(1-\nu^*)(\lambda_c + E\cdot\ln[K])}{E^*}$	=		1.96505
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \dfrac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	722 315 N	162 382.78 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \dfrac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	-725 344 N	-163 063.73 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	725 344 N	163 063.73 lbf
Bending stress (simplified Elastic-plastic calculation performed)	$\sigma = \dfrac{6M}{\mu^*(h-h'_g)^2}$	=	188.53 MPa	27 344.1 psi
				σ ≤ 2S: Ok

Walter Tosto S.p.A. via Erasmo Piaggio Chieti Telephone, Fax Website, Email address Date ________ Calc. ________ Contr.________ Appr.________	Customer Riccardo Petrelli Drawing U_150 Revision

Loading case Design 2: Pt = -0.10 MPa, Ps = 14.22 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear	$\mu = \frac{p-d_t}{p}$	=		0.24998
Effective tube pitch	$p^* = p\left(1-\frac{4\cdot\min[A_L,\ (4D_o p)]}{\pi D_o^2}\right)^{-0.5}$	=	42.33 mm	1.667 in
Tube expansion depth ratio	$\rho = \frac{l_{tx}}{h}$	=		1.00000
Effective tube hole diameter	$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\},\ (d_t - 2t_t)\right]$	=	23.37 mm	0.920 in
Effective ligament efficiency for bending	$\mu^* = \frac{p^*-d^*}{p^*}$	=		0.44798
	A0	=		-0.00290
	A1	=		0.21260
	A2	=		3.99060
	A3	=		-6.17300
	A4	=		3.43070
	B0	=		0.99660
	B1	=		-4.19780
	B2	=		9.04780
	B3	=		-7.99550
	B4	=		2.23980
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.47640
Effective Poisson ratio in perforated region	$\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.30322

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	22 832 134 N	5 132 867.40 lbf
Tubesheet design bold load	W* = Wds	=	22 832 134 N	5 132 867.40 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2+1)P_s - (\rho_c - 1)(\rho_c^2+1)P_t]}{16}$	=	165 337 N	37 169.12 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C-G_s)W^*}{2\pi D_o}$	=	1 340 848 N	301 434.51 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	-71 697 N	-16 118.20 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	1 046 687 N	235 304.64 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	1 046 687 N	235 304.64 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h-h'_g)^2}$	=	272.05 MPa	39 458.1 psi

σ ≤ 2S: Ok

Tubesheet shear stress

|Ps - Pt| ≤ 3.2·S·μ·h/Do: Shear stress is not required to be calculated

Shell and channel stresses

Axial membr. stress in the channel at tubesheet junction	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c+t_c)}$	=	-0.36 MPa	-52.5 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1-\nu^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1+\frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s-P_t)\right)\right]$	=	-449.46 MPa	-65 188.2 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	449.82 MPa	65 240.7 psi

Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation	$E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$	=	126 954.19 MPa	18 413 148.0 psi
	$k_c = \frac{\beta_c E_c^* t_c^3}{6(1-\nu_c^2)}$	=	76 346 284 N	17 163 325.87 lbf
	$\lambda_c = \frac{6D_c k_c}{h^3}\left(1+h\beta_c+\frac{h^2\beta_c^2}{2}\right)$	=	147 305.93 MPa	21 364 919.2 psi
	$\omega_c = \rho_c k_c \beta_c \delta_c(1+h\beta_c)$	=	21 195.2 mm²	32.853 in²
Diameter ratio	$K = \frac{A}{D_o}$	=		1.54472
Coefficient	$F = \frac{(1-\nu^*)(\lambda_c+E\cdot\ln[K])}{E^*}$	=		1.83559
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	34 518 N	7 760.01 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	1 152 903 N	259 182.85 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	1 152 903 N	259 182.85 lbf
Bending stress (simplified Elastic-plastic calculation performed)	$\sigma = \frac{6M}{\mu^*(h-h_g')^2}$	=	299.66 MPa	43 462.2 psi
				σ ≤ 2S: Ok

Loading case Design 3: Pt = 18.44 MPa, Ps = 14.22 MPa, thermal exp.: N, corr.: Y, vacuum: N

Basic ligament efficiency for shear	$\mu = \frac{p - d_t}{p}$	=		0.24998
Effective tube pitch	$p^* = p\left(1 - \frac{4 \cdot \min[A_L, (4D_o p)]}{\pi D_o^2}\right)^{-0.5}$	=	42.33 mm	1.667 in
Tube expansion depth ratio	$\rho = \frac{l_{tx}}{h}$	=		1.00000
Effective tube hole diameter	$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\}, (d_t - 2t_t)\right]$	=	23.37 mm	0.920 in
Effective ligament efficiency for bending	$\mu^* = \frac{p^* - d^*}{p^*}$	=		0.44798
	A0	=		-0.00290
	A1	=		0.21260
	A2	=		3.99060
	A3	=		-6.17300
	A4	=		3.43070
	B0	=		0.99660
	B1	=		-4.19780
	B2	=		9.04780
	B3	=		-7.99550
	B4	=		2.23980
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.47640
Effective Poisson ratio in perforated region	$v^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.30322

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	22 832 134 N	5 132 867.40 lbf
Tubesheet design bold load	W* = Wds	=	22 832 134 N	5 132 867.40 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	32 288 N	7 258.56 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	1 763 697 N	396 494.77 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	668 705 N	150 330.81 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + v^*)(P_s - P_t)}{64}$	=	339 431 N	76 307.12 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	668 705 N	150 330.81 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h_g')^2}$	=	173.81 MPa	25 208.9 psi

σ ≤ 2S: Ok

Tubesheet shear stress

|Ps - Pt| ≤ 3.2·S·μ·h/Do: Shear stress is not required to be calculated

Shell and channel stresses

Axial membr. stress in the channel at tubesheet junction	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	64.82 MPa	9 400.7 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1 - v^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	-147.48 MPa	-21 390.7 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	212.30 MPa	30 791.5 psi

Lc >= 1.8√(Dc·tc): Ok

σc ≤ 1.5 Sc: Ok

Loading case Design 4: Pt = -0.10 MPa, Ps = -0.10 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear	$\mu = \frac{p - d_t}{p}$	=		0.24998
Effective tube pitch	$p^* = p\left(1 - \frac{4 \cdot \min[A_L, (4D_o p)]}{\pi D_o^2}\right)^{-0.5}$	=	42.33 mm	1.667 in
Tube expansion depth ratio	$\rho = \frac{l_{tx}}{h}$	=		1.00000
Effective tube hole diameter	$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\}, (d_t - 2t_t)\right]$	=	23.37 mm	0.920 in
Effective ligament efficiency for bending	$\mu^* = \frac{p^* - d^*}{p^*}$	=		0.44798
	A0	=		-0.00290
	A1	=		0.21260
	A2	=		3.99060
	A3	=		-6.17300
	A4	=		3.43070
	B0	=		0.99660
	B1	=		-4.19780
	B2	=		9.04780
	B3	=		-7.99550
	B4	=		2.23980
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.47640
Effective Poisson ratio in perforated region	$v^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.30322

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	-165 385 N	-37 179.93 lbf
Tubesheet design bold load	W* = Wds	=	-165 385 N	-37 179.93 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	-453 N	-101.86 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	-12 079 N	-2 715.39 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	-3 624 N	-814.69 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + v^*)(P_s - P_t)}{64}$	=	-3 624 N	-814.69 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	3 624 N	814.69 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h_g')^2}$	=	0.94 MPa	136.6 psi

σ ≤ 2S: Ok

Tubesheet shear stress

|Ps - Pt| ≤ 3.2·S·μ·h/Do: Shear stress is not required to be calculated

Shell and channel stresses

Axial membr. stress in the channel at tubesheet junction	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	-0.36 MPa	-52.5 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1 - v^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	1.57 MPa	227.1 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	1.93 MPa	279.6 psi

Lc >= 1.8√(Dc·tc): Ok

σc ≤ 1.5 Sc: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

U-Tube tubesheet - Tubesheet (Bolting)

Material		=		SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	114.30 mm	4.500 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-29.00 °C	-20.20 °F

U-Tube tubesheet - Tubesheet

Material		=		SA-182 F22 3
Curve of fig. 3.7 / 3.8		=		B
Governing Thickness	tg	=	85.00 mm	3.346 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=		
Impact tests required by Code		=		Yes

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

Validation warnings:

- Gasket overloaded: Ab > AbMax

Tube bundle - Tubes bundle

According to: Asme VIII Div. 2 Ed. 2015, 4.3 & 4.18 - Metric Units

Calculation temperature	T	=	454.00 °C	849.20 °F
Material:	**SB-517 Cold drawn/ann. N06600 - Wld. pipe**			
Allowable stress at room temperature	ST	=	137.00 MPa	19 870.2 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	23.37 mm	0.920 in
Length	L	=	3 658.00 mm	144.016 in
Adopted thickness	t	=	4.19 mm	0.165 in
Internal pressure				
Allowable stress	S	=	137.00 MPa	19 870.2 psi
Internal pressure	Pi	=	18.54 MPa	2 688.9 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	18.54 MPa	2 688.9 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}(e^{\frac{P}{SE}}-1)+c+c_e+c'$	=	1.69 mm	0.067 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	41.99 MPa	6 090.7 psi
Hot & corroded		=	41.99 MPa	6 090.7 psi
External pressure				
External pressure	Pe	=	14.32 MPa	2 077.3 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	14.32 MPa	2 077.3 psi
External design temperature	Te	=	420.00 °C	788.00 °F
Outside diameter	Do	=	31.75 mm	1.250 in
Max unsupported length	L	=	3 428.00 mm	134.961 in
Modulus of elasticity	Ey	=	190 800.00 MPa	27 673 200.4 psi
Shell parameter	$M_x = \frac{L}{\sqrt{R_o(t-c-c')}}$	=		420.26683
	$C_h = 0.55\left(\frac{t}{D_o}\right)$	=		0.07260
Elastic buckling stress	$F_{he} = \frac{16 \cdot C_h \cdot E_y \cdot (t-c-c_e-c')}{D_O}$	=	2 925.50 MPa	424 307.3 psi
Buckling stress	Fic = Sy	=	198.60 MPa	28 804.5 psi
Factor A	A	=		0.01947
	B	=	79.78 MPa	11 570.7 psi
Tangent modulus of elasticity	Et = 2B / A	=	8 196.75 MPa	1 188 837.7 psi
Yield strength at design temperature	Sy	=	198.60 MPa	28 804.5 psi
Design factor	FS	=		1.66700
Hoop compressive membrane stress	$F_{ha} = \frac{F_{he} \cdot E_t}{FS \cdot E_y}$	=	75.39 MPa	10 934.7 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{D_O}\right)$	=	19.90 MPa	2 886.8 psi
Minimum required thickness	tr	=	3.13 mm	0.123 in
				t ≥ tr: Ok

TEMA RCB Requirements

Outside diameter	do	=	31.75 mm	1.250 in
Mean radius of bend	R	=	47.63 mm	1.875 in
Required tube wall thickness prior to bending	$t_0 = t_r\left[1+\frac{d_o}{4R}\right]$	=	3.65 mm	0.144 in
Outside diameter of the tube	od	=	31.75 mm	1.250 in
Tube pitch	Pitch	=	42.33 mm	1.667 in
Minimum tube pitch	Pitch(TEMA) = 1.25 * od	=	39.69 mm	1.563 in
				t ≥ t0: Ok
				Pitch ≥ Pitch(TEMA): Ok

Maximum allowable external pressures

New & cold	=	23.69 MPa	3 436.3 psi
Hot & corroded	=	19.90 MPa	2 886.8 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Tube bundle - Tubes bundle

Material		=	SB-517 Cold drawn/ann. N06600	
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	4.19 mm	0.165 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-104.00 °C	-155.20 °F

Note: Component made from DN 100 (NPS 4) pipe or smaller or equivalent size of tubes of P-No. 1 materials

External pressure

Tube bundle - Tubes bundle

Material		=	SB-517 Cold drawn/ann. N06600	
Curve of fig. 3.7 / 3.8		=		None
Governing Thickness	tg	=	4.19 mm	0.165 in
PostWeld Heat Treatment		=		No
Minimum Design Metal Temperature (MDMT)		=	-104.00 °C	-155.20 °F

Note: Component made from DN 100 (NPS 4) pipe or smaller or equivalent size of tubes of P-No. 1 materials

Hemispherical head - Head

According to: Asme VIII Div. 2 Ed. 2015, 4.3.5 - Metric Units

Calculation temperature	T	=	420.00 °C	788.00 °F
Material: **SA-387 22 2 - Plate**				
Allowable stress at room temperature	ST	=	207.00 MPa	30 022.8 psi
Joint efficiency	E	=		1.00
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Inside diameter	D	=	1 306.00 mm	51.417 in
Length	L	=	538.20 mm	21.189 in
Adopted thickness	t	=	34.00 mm	1.339 in
Internal pressure				
Allowable stress	S	=	158.60 MPa	23 003.0 psi
Internal pressure	Pi	=	14.22 MPa	2 062.4 psi
Overpressure due to static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pi + Ph	=	14.22 MPa	2 062.4 psi
Required thickness	$t_r = \frac{D+2(c+c')}{2}\left(e^{\frac{P}{2SE}}-1\right)+c+c_e+c'$	=	33.08 mm	1.302 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	21.01 MPa	3 047.8 psi
Hot & corroded		=	14.65 MPa	2 124.3 psi
External pressure				
External pressure	Pe	=	0.10 MPa	14.9 psi
External static head	Ph	=	0 MPa	0 psi
Calculation pressure	P = Pe + Ph	=	0.10 MPa	14.9 psi
External design temperature	Te	=	20.00 °C	68.00 °F
Outside radius	Ro = D/2 + t	=	687.00 mm	27.047 in
Yield stress	Sy	=	310.00 MPa	44 961.7 psi
Modulus of elasticity	Ey	=	210 350.00 MPa	30 508 688.2 psi
Elastic buckling stress	$F_{he} = 0.075E_y\left(\frac{t-c-c_e-c'}{R_o}\right)$	=	711.88 MPa	103 249.9 psi
Buckling stress	$F_{ic} = \frac{1.31S_y}{\left(1.15+\frac{S_y}{F_{he}}\right)}$	=	256.14 MPa	37 149.9 psi
Design factor	$FS = 2.407 - 0.741\left(\frac{F_{ic}}{S_y}\right)$	=		1.79474
Hoop compressive membrane stress	$F_{ha} = \frac{F_{ic}}{FS}$	=	142.72 MPa	20 699.3 psi
Allowable external pressure in the absence of other loads	$P_a = 2F_{ha}\left(\frac{t-c-c_e-c'}{R_o}\right)$	=	12.88 MPa	1 868.1 psi
Minimum required thickness	tr	=	4.69 mm	0.185 in
				t ≥ tr: Ok
Maximum allowable external pressures				
New & cold		=	14.52 MPa	2 105.3 psi
Hot & corroded		=	12.88 MPa	1 868.1 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Hemispherical head - Head

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	34.00 mm	1.339 in
PostWeld Heat Treatment		=		No
Unadjusted MDMT from figure 3.7		=	40.90 °C	105.62 °F
Coincident ratio	Rts	=		0.97000
Reduction in MDMT based on available excess thickness	TR	=	1.40 °C	34.52 °F
Adjusted MDMT from fig. 3.12		=	39.50 °C	103.10 °F

External pressure

Hemispherical head - Head

Material		=		SA-387 22 2
Curve of fig. 3.7 / 3.8		=		A
Governing Thickness	tg	=	34.00 mm	1.339 in
PostWeld Heat Treatment		=		No
Unadjusted MDMT from figure 3.7		=	40.90 °C	105.62 °F
Coincident ratio	Rts	=		0.05500
Adjusted MDMT from fig. 3.12		=	-104.00 °C	-155.20 °F

Note: Stress ratio Rts ≤ 0.24

Validation warnings:

- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]