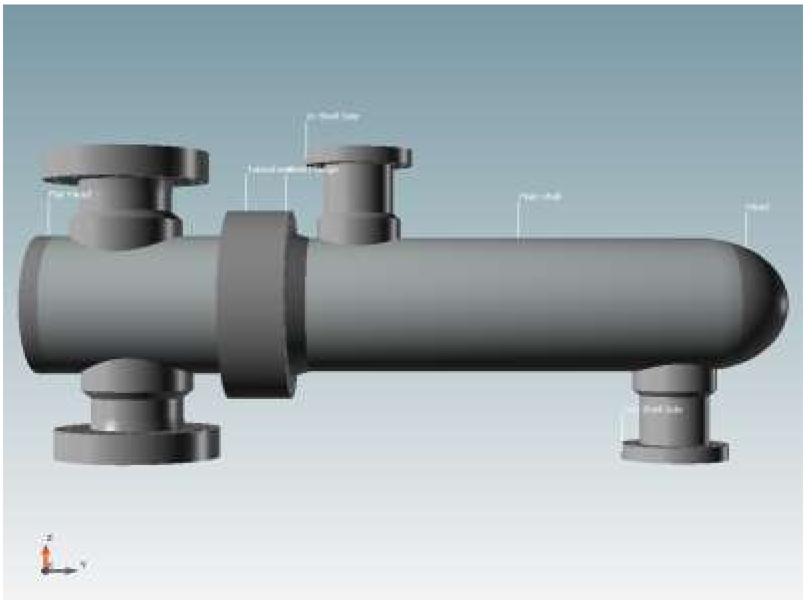
Calculation report

Asme VIII Div. 1 Ed. 2015 - Metric Units

Project: U_150_Divisione_1_Start
Item: U_150_Div_1
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*	*Max test pressure*
Flat Head	18.44	0	0.02	21.38	18.45	1,158	
Left channel	18.44	0	0.02	24.93	21.52	1,158	
Bocchello In Tube Side	18.44	0	0.01	21.61	18.66	1,158	
Flange In Tube Side 26"	18.44	0	0.004	21.72	18.94	1,158	
Bocchello Out Tube Side	18.44	0	0.03	21.76	18.79	1,158	
Flange Out Tube Side 26"	18.44	0	0.03	21.72	18.94	1,158	
Tubesheet	18.44	0	0.02	26.08	22.24	1,158	
Tubes bundle	18.54	0	0.02	54.31	54.31	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) = 18.45 MPa (limited by Flat Head)

Tubes side MAP (New & Cold conditions) = 21.38 MPa (limited by Flat Head)

Tubes side Lowest Stress Ratio = 1.000

Tubes side test pressure = Pt=1.3·MAWP H&C (Item)·St/S (Item) = 23.99 MPa

Tubes side maximum test pressure = (limited by Tubes bundle)

Maximum Pressures - Tube side (MPa)

Component	*MAP N&C*	*MAWP H&C*	*MAEP N&C*	*MAEWP H&C*
Flat Head	21.38	18.45	21.38	21.38
Left channel	24.93	21.52	15.37	15.37
Bocchello In Tube Side	21.61	18.66	19.91	19.91
Flange In Tube Side 26"	21.72	18.94	18.94	18.94
Bocchello Out Tube Side	21.76	18.79	19.91	19.91
Flange Out Tube Side 26"	21.72	18.94	18.94	18.94
Tubesheet	26.08	22.24		
Tubes bundle	54.31	54.31	27.98	20.93

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*	*Max test pressure*
Shell Flange	14.22	0	0.02	14.58	14.52	1,108	
Main shell	14.22	0	0.02	16.47	14.25	1,108	
In Shell Side	14.22	0	0.01	25.86	17.67	1,108	
Out Shell Side	14.22	0	0.03	25.86	17.67	1,108	
Tubesheet	14.22	0	0.02	16.90	14.36	1,158	
Tubes bundle	14.32	0	0.01	27.98	20.93	1	
Head	14.22	0	0.02	17.39	14.44	1,108	

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) = 14.25 MPa (limited by Main shell)

Shell side MAP (New & Cold conditions) = 14.58 MPa (limited by Shell Flange)

Shell side Lowest Stress Ratio = 1.000

Shell side test pressure = Pt=1.3·MAWP H&C (Item)·St/S (Item) = 18.53 MPa

Shell side maximum test pressure = (limited by Head)

Maximum Pressures - Shell side (MPa)

Component	*MAP N&C*	*MAWP H&C*	*MAEP N&C*	*MAEWP H&C*
Shell Flange	14.58	14.52	14.52	14.52
Main shell	16.47	14.25	9.51	9.11
In Shell Side	25.86	17.67	17.67	17.67
Out Shell Side	25.86	17.67	17.67	17.67
Tubesheet	16.90	14.36		
Tubes bundle	27.98	20.93		
Head	17.39	14.44	7.60	7.02

All pressures in MPa

Weights

Component	*Dead*	*Live*	*Liquid*	*Full of water*	*Operating*
Flat Head	4 025 kg	0 kg	0 kg	4 444 kg	4 025 kg
Left channel	4 622 kg	0 kg	0 kg	6 428 kg	4 622 kg
Bocchello In Tube Side	1 244 kg	0 kg	0 kg	1 428 kg	1 244 kg
Flange In Tube Side 26”	2 819 kg	0 kg	0 kg	2 957 kg	2 819 kg
Bocchello Out Tube Side	1 244 kg	0 kg	0 kg	1 428 kg	1 244 kg
Flange Out Tube Side 26”	2 819 kg	0 kg	0 kg	2 957 kg	2 819 kg
Shell Flange	4 027 kg	0 kg	0 kg	4 576 kg	4 027 kg
Main shell	8 985 kg	0 kg	0 kg	11 763 kg	8 985 kg
In Shell Side	1 437 kg	0 kg	0 kg	1 464 kg	1 437 kg
Out Shell Side	1 437 kg	0 kg	0 kg	1 464 kg	1 437 kg
Tubesheet	5 016 kg	0 kg	0 kg	5 016 kg	5 016 kg
Tubes bundle	15 284 kg	0 kg	0 kg	17 082 kg	15 284 kg
Head	867 kg	0 kg	0 kg	1 458 kg	867 kg
Totals:	53 826 kg	0 kg	0 kg	62 463 kg	53 826 kg

Total shell side volume: 3.97127 m³
Total tube side volume: 4.66602 m³

Total volume: 8.63729 m³

Bill of materials

Component	***Dimensions***	***Material***
Flat Head	Id = 1 238.00 mm, Od = 1 470.00 mm, Tk = 306.00 mm	SA-387 22 2 - Plate
Left channel	Id = 1 238.00 mm, Od = 1 470.00 mm, Tk = 116.00 mm, L = 1 500.00 mm	SA-387 22 2 - Plate
Bocchello In Tube Side	Id = 635.00 mm, Od = 787.40 mm, Tk = 76.20 mm, L = 581.00 mm	SA-182 F22 3 - Forging
Flange In Tube Side 26" - Flange	Id = 635.00 mm, Od = 1 420.00 mm, Tk = 310.00 mm	SA-182 F22 3 - Forging
Flange In Tube Side 26" - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Flange In Tube Side 26" - Bolts	20 x ANSI_TEMA 3-3/4"	SA-193 B16 - Bolting
Bocchello Out Tube Side	Id = 635.00 mm, Od = 787.40 mm, Tk = 76.20 mm, L = 581.00 mm	SA-182 F22 3 - Forging
Flange Out Tube Side 26" - Flange	Id = 635.00 mm, Od = 1 420.00 mm, Tk = 310.00 mm	SA-182 F22 3 - Forging
Flange Out Tube Side 26" - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Flange Out Tube Side 26" - Bolts	20 x ANSI_TEMA 3-3/4"	SA-193 B16 - Bolting
Shell Flange - Flange	Id = 1 275.00 mm, Od = 2 010.00 mm, Tk = 275.00 mm	SA-387 22 2 - Plate
Shell Flange - Gasket	Grooved Metal - Stainless steels and nickel-base alloys	
Shell Flange - Bolts	24 x ANSI_TEMA 4-1/2"	SA-193 B16 - Bolting
Main shell	Id = 1 275.00 mm, Od = 1 427.00 mm, Tk = 76.00 mm, L = 3 594.00 mm	SA-387 22 2 - Plate
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
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
Tubesheet - Flange	Od = 2 010.00 mm, Tk = 265.00 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-444 2 Solution ann. N06625 (high allowable stresses) - Pipe / tube
Head	Id = 1 312.00 mm, Od = 1 390.00 mm, Tk = 39.00 mm	SA-387 22 2 - Plate

Nozzle connections

Name	*Flange*	*Material*	*OD*	*Tk*
Bocchello In Tube Side	WN NotStandard	SA-182 F22 3	787.40 mm	76.20 mm
Bocchello Out Tube Side	WN NotStandard	SA-182 F22 3	787.40 mm	76.20 mm
In Shell Side	18" LWN 1500 ANSI	SA-182 F22 3	597.00 mm	69.90 mm
Out 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	600.00 mm	0 °	
Bocchello Out Tube Side	Left channel	Radial/ Reinforced	700.00 mm	180.00 °	
In Shell Side	Main shell	Radial/ Reinforced	420.00 mm	0 °	
Out Shell Side	Main shell	Radial/ Reinforced	3 150.00 mm	180.00 °	

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				
In Shell Side	15.00 mm				
Out Shell Side	15.00 mm				

Minimum Design Metal Temperature (MDMT)

Component		***MDMT***	***Tmin > MDMT***
Flat Head		48.95 °C / 120.11 °F	No
Left channel		40.28 °C / 104.50 °F	No
Bocchello In Tube Side		25.78 °C / 78.40 °F	No
Flange In Tube Side 26"		24.53 °C / 76.15 °F	No
Flange In Tube Side 26" (bolting)		-110.00 °C / -166.00 °F	Yes
Bocchello Out Tube Side		25.78 °C / 78.40 °F	No
Flange Out Tube Side 26"		24.53 °C / 76.15 °F	No
Flange Out Tube Side 26" (bolting)		-110.00 °C / -166.00 °F	Yes
Shell Flange		41.84 °C / 107.30 °F	No
Shell Flange (bolting)		-110.00 °C / -166.00 °F	Yes
Main shell		42.86 °C / 109.16 °F	No
In Shell Side		25.76 °C / 78.38 °F	No
In Shell Side (bolting)		-110.00 °C / -166.00 °F	Yes
Out Shell Side		25.76 °C / 78.38 °F	No
Out Shell Side (bolting)		-110.00 °C / -166.00 °F	Yes
Tubesheet		41.52 °C / 106.74 °F	No
Tubesheet (bolting)		-110.00 °C / -166.00 °F	Yes
Tubes bundle		-198.00 °C / -324.40 °F	Yes
Head		30.75 °C / 87.34 °F	No

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

Item MDMT: 48.95 °C / 120.11 °F

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

Welded flat cover - Flat Head

According to: Asme VIII Div. 1 Ed. 2015, UG-34 - 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

Material: SA-387 22 2 - Plate

Allowable stress	S	=	127.76 MPa	18 530.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi

Geometry

Inside diameter	D	=	1 238.00 mm	48.740 in
Outside diameter	Do	=	1 470.00 mm	57.874 in
Adopted thickness	t	=	306.00 mm	12.047 in
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Sketch		=		Sketch o
Factor C	C	=		0.30000
Outside diameter	d = Do	=	1 470.00 mm	57.874 in

Internal pressure

Allowable stress	S	=	127.76 MPa	18 530.0 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	$t = d\sqrt{\frac{CP}{SE}} + c + c_e + c'$	=	305.86 mm	12.042 in
Item service	Service	=		NotSpecified
Minimum required thickness as per UG-16(b), including corrosion	tr UG-16(b)	=	1.50 mm	0.059 in
				t ≥ tr UG-16(b): Ok
				t ≥ tr: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold		=	21.38 MPa	3 100.5 psi
Hot & corroded		=	18.45 MPa	2 676.5 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	$t = d\sqrt{\frac{CP}{SE}} + c + c_e + c'$	=	21.25 mm	0.837 in
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold		=	21.38 MPa	3 100.5 psi
Hot & corroded		=	21.38 MPa	3 100.5 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	0.05 °C	32.09 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Design pressure	P	=	18.44 MPa	2 674.0 psi
Maximum allowable working pressure	MAWP H&C	=	18.45 MPa	2 676.5 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.99907
Adjusted MDMT from fig. UCS-66.1		=	48.95 °C	120.11 °F

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	8.72 °C	47.70 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	97.79 mm	3.850 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.84304
Adjusted MDMT from fig. UCS-66.1		=	40.28 °C	104.50 °F

External pressure

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	18.45 MPa	2 676.5 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.00558
Adjusted MDMT from fig. UCS-66.1		=	-31.00 °C	-23.80 °F

Welded flat cover - Flat Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	4.77 mm	0.188 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.04112
Adjusted MDMT from fig. UCS-66.1		=	-31.00 °C	-23.80 °F

Cylindrical shell - Left channel

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - 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
Material: SA-387 22 2 - Plate				
Allowable stress	S	=	127.76 MPa	18 530.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi
Geometry				
Inside diameter	D	=	1 238.00 mm	48.740 in
Outside diameter	Do	=	1 470.00 mm	57.874 in
Length	L	=	1 500.00 mm	59.055 in
Adopted thickness	t	=	116.00 mm	4.567 in
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		8,57%
Ligament Efficiency				
Reference figure		=		None
Diameter of tube holes	d	=	0 mm	0 in
Internal pressure				
Allowable stress	S	=	127.76 MPa	18 530.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	619.00 mm	24.370 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE - 0.6P} + c + c_e + c'$	=	97.79 mm	3.850 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE + 0.4P} + c + c_e + c'$	=	43.41 mm	1.709 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	97.79 mm	3.850 in
Item service	Service	=		NotSpecified
Minimum required thickness as per UG-16(b), including corrosion	tr UG-16(b)	=	1.50 mm	0.059 in
				t ≥ tr: Ok
				t ≥ tr UG-16(b): Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	24.93 MPa	3 616.0 psi
Hot & corroded		=	21.52 MPa	3 121.5 psi

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
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
Outside diameter	Do	=	1 470.00 mm	57.874 in
Modulus of elasticity	E	=	200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L	=	1 500.00 mm	59.055 in
L / Do ratio		=		1.02041
Do / t ratio		=		12.67241
Factor A		=		0.03111
Factor B		=	146.07 MPa	21 185.3 psi
External pressure	$\mathrm{Pa} = \frac{4B}{3[D_o/(t-c-c_e-c')]}$	=	15.37 MPa	2 229.0 psi
Required thickness	tr	=	4.77 mm	0.188 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	15.37 MPa	2 229.0 psi
Hot & corroded	=	15.37 MPa	2 229.0 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Cylindrical shell - Left channel

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	8.72 °C	47.70 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	97.79 mm	3.850 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.84304
Adjusted MDMT from fig. UCS-66.1		=	40.28 °C	104.50 °F

External pressure

Cylindrical shell - Left channel

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	4.77 mm	0.188 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.04112
Adjusted MDMT from fig. UCS-66.1		=	-31.00 °C	-23.80 °F

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. 1 Ed. 2015, UG-36 - 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
Nozzle material: SA-182 F22 3 - Forgings				
Allowable stress in nozzle	Sn	=	127.76 MPa	18 530.0 psi
Shell material: SA-387 22 2 - Plate				
Allowable stress in vessel	Sv	=	127.76 MPa	18 530.0 psi
Nozzle geometry				
Outside diameter	do	=	787.40 mm	31.000 in
Inside diameter	di	=	635.00 mm	25.000 in
Nozzle thickness	tn	=	76.20 mm	3.000 in
Nozzle reinforcement thickness	tx	=	165.00 mm	6.496 in
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Tapering angle	ta	=		45.00 °
Corner radius	r	=	10.00 mm	0.394 in
Nozzle corrosion allowance	cn	=	0 mm	0 in
Nozzle undertolerance	c'n	=	0 mm	0 in
Nozzle connection		=		Integrally reinforced
Nozzle position		=		Radial
Offset from shell border		=	600.00 mm	23.622 in
Angular offset		=		0 °
Weld leg length of the outside nozzle fillet weld	two	=	15.00 mm	0.591 in
Minimum weld leg length of the outside nozzle fillet weld		=	8.49 mm	0.334 in
				r>=Min[1/4t,1/8 in. (3mm)]: Ok
Opening geometry				
Finished diameter of circular opening	d	=	635.00 mm	25.000 in
Finished radius of circular opening	Rn	=	317.50 mm	12.500 in
Nearest opening		=		Bocchello Out Tube Side
Distance to nearest opening		=	2 311.23 mm	90.994 in
Maximum distance before multiple openings calculation occurs		=	1 333.15 mm	52.486 in
Shell Geometry				
Inside shell diameter	D	=	1 238.00 mm	48.740 in
Inside radius of shell course under consideration	R	=	619.00 mm	24.370 in
Shell thickness	t	=	116.00 mm	4.567 in
Shell corrosion allowance	cs	=	0 mm	0 in
Shell undertolerance	c's	=	0 mm	0 in

Internal pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	116.00 mm	4.567 in
Net required thickness of a seamless shell or formed head	tr	=	97.79 mm	3.850 in
Net nozzle thickness	tn'=tn-cn-c'n	=	76.20 mm	3.000 in
Net required thickness of a seamless nozzle wall	trn	=	50.16 mm	1.975 in
Nozzle reinforcement thickness	tx'=tx-cn	=	165.00 mm	6.496 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	635.00 mm	25.000 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	117.50 mm	4.626 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	290.00 mm	11.417 in

Internal pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Correction factor	F	=		1.00000
Internal pressure	P	=	18.44 MPa	2 674.0 psi
Static head internal	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\left\{\begin{matrix} d(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \\ 2(t + t_x')(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \end{matrix}\right\}$	=	10 287.3 mm²	15.945 in²
Corner area to be subtracted from nozzle area	$A_7 = (r^2 - \pi\frac{r^2}{4})f_{r2}$	=	21.5 mm²	0.033 in²
Area available in outward nozzle	$A_2 = \left\{2[(t_n' - t_{rn})(h_o - L) + L(t_x' - t_{rn}) + (t_x' - t_n')^2\tan(t_a)/2] - 2A_7\right\}f_{r2}$	=	53 492.9 mm²	82.914 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=d·tr·F+2·tx'·tr·F(1-fr1)	=	62 098.2 mm²	96.252 in²
Total available area	At=A1+A2+A3+A41+A43	=	64 005.1 mm²	99.208 in²
				At ≥ A: Ok

Appendix 1-7(a)

Area available in shell, within Appendix 1-7 limits	A1 1-7(a)	=	6 999.0 mm²	10.848 in²
Area available in outward nozzle	A2	=	53 492.9 mm²	82.914 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41	=	225.0 mm²	0.349 in²
Area available in inward weld	A43	=	0 mm²	0 in²
Total area available within Appendix 1-7 limits	At (1-7)	=	60 716.8 mm²	94.111 in²
Area required as per Appendix 1-7(a)	2/3 A	=	41 398.8 mm²	64.168 in²
				At (1-7) ≥ 2/3 A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	18.66 MPa	2 705.9 psi
Total pressure	Pt	=	18.44 MPa	2 674.0 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	1.50 mm	0.059 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	50.16 mm	1.975 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	97.79 mm	3.850 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	97.79 mm	3.850 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	0.50 mm	0.020 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	1.50 mm	0.059 in
Minimum thickness of a standard wall pipe	tb3	=	8.34 mm	0.328 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	8.34 mm	0.328 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	50.16 mm	1.975 in
Nozzle thickness	tn	=	76.20 mm	3.000 in
				tn ≥ t(UG-45): Ok

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External pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	116.00 mm	4.567 in
Net required thickness of a seamless shell or formed head	tr	=	4.77 mm	0.188 in
Net nozzle thickness	tn'=tn-cn-c'n	=	76.20 mm	3.000 in
Net required thickness of a seamless nozzle wall	trn	=	1.77 mm	0.070 in
Nozzle reinforcement thickness	tx'=tx-cn	=	165.00 mm	6.496 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	635.00 mm	25.000 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	117.50 mm	4.626 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	290.00 mm	11.417 in

External pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Correction factor	F	=		1.00000
External pressure	P	=	0.10 MPa	14.9 psi
Static head external	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\left\{\begin{array}{c} d(E_1t' - Ft_r) - 2t_x'(E_1t' - Ft_r)(1-f_{r1}) \\ 2(t + t_x')(E_1t' - Ft_r) - 2t_x'(E_1t' - Ft_r)(1-f_{r1}) \end{array}\right\}$	=	62 845.0 mm²	97.410 in²
Area available in outward nozzle	$A_2 = \left\{2[(t_n' - t_{rn})(h_o - L) + L(t_x' - t_{rn}) + (t_x' - t_n')^2\tan(t_a)/2] - 2A_7\right\}f_{r2}$	=	81 559.1 mm²	126.417 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=[d·tr·F+2·tx'·tr·F(1-fr1)]/2	=	1 514.5 mm²	2.347 in²
Total available area	At=A1+A2+A3+A41+A43	=	144 629.1 mm²	224.175 in²
				At ≥ A: Ok

Appendix 1-7(a)

Area available in shell, within Appendix 1-7 limits	A1 1-7(a)	=	42 756.8 mm²	66.273 in²
Area available in outward nozzle	A2	=	81 559.1 mm²	126.417 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41	=	225.0 mm²	0.349 in²
Area available in inward weld	A43	=	0 mm²	0 in²
Total area available within Appendix 1-7 limits	At (1-7)	=	124 540.9 mm²	193.039 in²
Area required as per Appendix 1-7(a)	2/3 A	=	1 009.7 mm²	1.565 in²
				At (1-7) ≥ 2/3 A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	17.98 MPa	2 607.4 psi
Total pressure	Pt	=	0.10 MPa	14.9 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	1.50 mm	0.059 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	1.77 mm	0.070 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	97.79 mm	3.850 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	97.79 mm	3.850 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	0.50 mm	0.020 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	1.50 mm	0.059 in
Minimum thickness of a standard wall pipe	tb3	=	8.34 mm	0.328 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	8.34 mm	0.328 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	8.34 mm	0.328 in
Nozzle thickness	tn	=	76.20 mm	3.000 in
				tn ≥ t(UG-45): Ok

Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1 238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]

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Reinforcement of opening - Bocchello Out Tube Side

According to: Asme VIII Div. 1 Ed. 2015, UG-36 - 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

Nozzle material: SA-182 F22 3 - Forgings

Allowable stress in nozzle	Sn	=	127.76 MPa	18 530.0 psi

Shell material: SA-387 22 2 - Plate

Allowable stress in vessel	Sv	=	127.76 MPa	18 530.0 psi

Nozzle geometry

Outside diameter	do	=	787.40 mm	31.000 in
Inside diameter	di	=	635.00 mm	25.000 in
Nozzle thickness	tn	=	76.20 mm	3.000 in
Nozzle reinforcement thickness	tx	=	165.00 mm	6.496 in
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Tapering angle	ta	=		45.00 °
Corner radius	r	=	10.00 mm	0.394 in
Nozzle corrosion allowance	cn	=	0 mm	0 in
Nozzle undertolerance	c'n	=	0 mm	0 in
Nozzle connection		=		Integrally reinforced
Nozzle position		=		Radial
Offset from shell border		=	700.00 mm	27.559 in
Angular offset		=		180.00 °
Weld leg length of the outside nozzle fillet weld	two	=	15.00 mm	0.591 in
Minimum weld leg length of the outside nozzle fillet weld		=	8.49 mm	0.334 in
				r>=Min[1/4t,1/8 in. (3mm)]: Ok

Opening geometry

Finished diameter of circular opening	d	=	635.00 mm	25.000 in
Finished radius of circular opening	Rn	=	317.50 mm	12.500 in
Nearest opening		=		Bocchello In Tube Side
Distance to nearest opening		=	2 311.23 mm	90.994 in
Maximum distance before multiple openings calculation occurs		=	1 333.15 mm	52.486 in

Shell Geometry

Inside shell diameter	D	=	1 238.00 mm	48.740 in
Inside radius of shell course under consideration	R	=	619.00 mm	24.370 in
Shell thickness	t	=	116.00 mm	4.567 in
Shell corrosion allowance	cs	=	0 mm	0 in
Shell undertolerance	c's	=	0 mm	0 in

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Revision

Internal pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	116.00 mm	4.567 in
Net required thickness of a seamless shell or formed head	tr	=	97.79 mm	3.850 in
Net nozzle thickness	tn'=tn-cn-c'n	=	76.20 mm	3.000 in
Net required thickness of a seamless nozzle wall	trn	=	50.16 mm	1.975 in
Nozzle reinforcement thickness	tx'=tx-cn	=	165.00 mm	6.496 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	635.00 mm	25.000 in
Useful limit of reinforcement in vessel wall	Lo=LR-(d/2)-tx	=	152.50 mm	6.004 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	290.00 mm	11.417 in

Internal pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Correction factor	F	=		1.00000
Internal pressure	P	=	18.44 MPa	2 674.0 psi
Static head internal	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\begin{Bmatrix} d(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \\ 2(t + t_x')(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \end{Bmatrix}$	=	11 561.8 mm²	17.921 in²
Corner area to be subtracted from nozzle area	$A_7 = (r^2 - \pi\frac{r^2}{4})f_{r2}$	=	21.5 mm²	0.033 in²
Area available in outward nozzle	$A_2 = \{2[(t_n' - t_{rn})(h_o - L) + L(t_x' - t_{rn}) + (t_x' - t_n')^2\tan(t_a)/2] - 2A_7\}f_{r2}$	=	53 492.9 mm²	82.914 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=d·tr·F+2·tx'·tr·F(1-fr1)	=	62 098.2 mm²	96.252 in²
Total available area	At=A1+A2+A3+A41+A43	=	65 279.6 mm²	101.184 in²
				At ≥ A: Ok

Appendix 1-7(a)

Area available in shell, within Appendix 1-7 limits	A1 1-7(a)	=	6 999.0 mm²	10.848 in²
Area available in outward nozzle	A2	=	53 492.9 mm²	82.914 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41	=	225.0 mm²	0.349 in²
Area available in inward weld	A43	=	0 mm²	0 in²
Total area available within Appendix 1-7 limits	At (1-7)	=	60 716.8 mm²	94.111 in²
Area required as per Appendix 1-7(a)	2/3 A	=	41 398.8 mm²	64.168 in²
				At (1-7) ≥ 2/3 A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	18.79 MPa	2 724.7 psi
Total pressure	Pt	=	18.44 MPa	2 674.0 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	1.50 mm	0.059 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	50.16 mm	1.975 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	97.79 mm	3.850 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	97.79 mm	3.850 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	0.50 mm	0.020 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	1.50 mm	0.059 in
Minimum thickness of a standard wall pipe	tb3	=	8.34 mm	0.328 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	8.34 mm	0.328 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	50.16 mm	1.975 in
Nozzle thickness	tn	=	76.20 mm	3.000 in

tn ≥ t(UG-45): Ok

External pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	116.00 mm	4.567 in
Net required thickness of a seamless shell or formed head	tr	=	4.77 mm	0.188 in
Net nozzle thickness	tn'=tn-cn-c'n	=	76.20 mm	3.000 in
Net required thickness of a seamless nozzle wall	trn	=	1.77 mm	0.070 in
Nozzle reinforcement thickness	tx'=tx-cn	=	165.00 mm	6.496 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	635.00 mm	25.000 in
Useful limit of reinforcement in vessel wall	Lo=LR-(d/2)-tx	=	152.50 mm	6.004 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	290.00 mm	11.417 in

External pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	172.00 mm	6.772 in
Correction factor	F	=		1.00000
External pressure	P	=	0.10 MPa	14.9 psi
Static head external	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\begin{Bmatrix} d(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \\ 2(t + t_x')(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \end{Bmatrix}$	=	70 631.1 mm²	109.478 in²
Area available in outward nozzle	$A_2 = \{2[(t_n' - t_{rn})(h_o - L) + L(t_x' - t_{rn}) + (t_x' - t_n')^2 \tan(t_a)/2] - 2A_7\} f_{r2}$	=	81 559.1 mm²	126.417 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=[d·tr·F+2·tx'·tr·F(1-fr1)]/2	=	1 514.5 mm²	2.347 in²
Total available area	At=A1+A2+A3+A41+A43	=	152 415.2 mm²	236.244 in²
				At ≥ A: Ok

Appendix 1-7(a)

Area available in shell, within Appendix 1-7 limits	A1 1-7(a)	=	42 756.8 mm²	66.273 in²
Area available in outward nozzle	A2	=	81 559.1 mm²	126.417 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41	=	225.0 mm²	0.349 in²
Area available in inward weld	A43	=	0 mm²	0 in²
Total area available within Appendix 1-7 limits	At (1-7)	=	124 540.9 mm²	193.039 in²
Area required as per Appendix 1-7(a)	2/3 A	=	1 009.7 mm²	1.565 in²
				At (1-7) ≥ 2/3 A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	17.98 MPa	2 607.4 psi
Total pressure	Pt	=	0.10 MPa	14.9 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	1.50 mm	0.059 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	1.77 mm	0.070 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	97.79 mm	3.850 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	97.79 mm	3.850 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	0.50 mm	0.020 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	1.50 mm	0.059 in
Minimum thickness of a standard wall pipe	tb3	=	8.34 mm	0.328 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	8.34 mm	0.328 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	8.34 mm	0.328 in
Nozzle thickness	tn	=	76.20 mm	3.000 in

tn ≥ t(UG-45): Ok

Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1 238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]

Nozzle - Bocchello In Tube Side

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - 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
Material: SA-182 F22 3 - Forgings				
Allowable stress	S	=	127.76 MPa	18 530.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi
Geometry				
Inside diameter	D	=	635.00 mm	25.000 in
Outside diameter	Do	=	787.40 mm	31.000 in
Length	L	=	581.00 mm	22.874 in
Adopted thickness	t	=	76.20 mm	3.000 in
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		10,71%
Internal pressure				
Allowable stress	S	=	127.76 MPa	18 530.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	317.50 mm	12.500 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE - 0.6P} + c + c_e + c'$	=	50.16 mm	1.975 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE + 0.4P} + c + c_e + c'$	=	22.27 mm	0.877 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	50.16 mm	1.975 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold (opening)		=	21.61 MPa	3 134.6 psi
Hot & corroded (opening)		=	18.66 MPa	2 705.9 psi
New & cold (cylinder)		=	26.80 MPa	3 887.4 psi
Hot & corroded (cylinder)		=	26.80 MPa	3 887.4 psi

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
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
Outside diameter	Do	=	787.40 mm	31.000 in
Modulus of elasticity	E	=	200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L	=	331.00 mm	13.031 in
L / Do ratio		=		0.42037
Do / t ratio		=		10.33333
Factor A		=		0.09136
Factor B		=	154.29 MPa	22 377.8 psi
External pressure	$Pa = \frac{4B}{3[D_o/(t-c-c_e-c')]}$	=	19.91 MPa	2 887.5 psi
Required thickness	tr	=	1.77 mm	0.070 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	19.91 MPa	2 887.5 psi
Hot & corroded	=	19.91 MPa	2 887.5 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. UCS-66		=		B
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	8.72 °C	47.70 °F
Unadjusted MDMT from table UCS-66		=	34.50 °C	94.10 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	97.79 mm	3.850 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.84304
Adjusted MDMT from fig. UCS-66.1		=	25.78 °C	78.40 °F

External pressure

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

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	34.50 °C	94.10 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	4.77 mm	0.188 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.04112
Adjusted MDMT from fig. UCS-66.1		=	-45.50 °C	-49.90 °F

Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1 238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]

Welding neck flange - Flange In Tube Side 26"

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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 - Bolting**
Gasket	**Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=127.76 MPa / 18 530.0 psi	Sno=127.76 MPa / 18 530.0 psi	Sbo=144.76 MPa / 20 995.7 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 psi	Sbg=152.00 MPa / 22 045.7 psi

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	=	1 420.00 mm	55.906 in
Inside diameter	B	=	635.00 mm	25.000 in
Bolt circle	C	=	1 235.00 mm	48.622 in
Flange thickness	t	=	310.00 mm	12.205 in
Mean gasket diameter	Gmean	=	742.00 mm	29.213 in
Thickness of the hub at the small end	g0	=	76.20 mm	3.000 in
Thickness of the hub at the large end	g1	=	90.00 mm	3.543 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	76.20 mm	3.000 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	90.00 mm	3.543 in
Hub length	h	=	115.00 mm	4.528 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	70.00 MPa	10 152.6 psi
Gasket contact width	N	=	106.40 mm	4.189 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	53.20 mm	2.094 in
Conversion factor for length	Cb	=		2.50000
Effective gasket contact width	$b = C_b\sqrt{b_0}$	=	18.23 mm	0.718 in
Outside diameter of the gasket contact area	$G_c = G_{\mathrm{mean}} + N$	=	848.40 mm	33.402 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	811.93 mm	31.966 in

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 3-3/4"
Bolt spacing	Bs	=	193.99 mm	7.638 in
Nominal bolt diameter	a	=	95.25 mm	3.750 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	16 829 735 N	3 783 474.67 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	3 255 838 N	731 941.41 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_g}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	116 259.6 mm²	180.203 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	130 425.5 mm²	202.160 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	249 974.1 mm²	387.461 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	18 748 069 N	4 214 733.16 lbf
				Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		2.23622
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		2.56106
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.42288
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		2.81435
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		1.49991
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	219.97 mm	8.660 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.18110
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.52280
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.87835
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.42757
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00399
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		8 407 169.84802
Stress factor L	$L = \frac{\mathrm{te}+1}{T} + \frac{t^3}{d}$	=		5.11628
Gasket load for the operating condition	$H_G = W_o - H$	=	7 288 903 N	1 638 610.48 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785B^2P$	=	5 835 736 N	1 311 925.51 lbf
Total hydrostatic end force	$H = 0.785G^2P$	=	9 540 832 N	2 144 864.19 lbf
Difference	$H_T = H - H_D$	=	3 705 096 N	832 938.67 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	255.00 mm	10.039 in
Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	211.53 mm	8.328 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	255.77 mm	10.070 in

Flange moments

Nominal bolt diameter	dB	=	95.25 mm	3.750 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
Flange design moment for the operating condition	$M_o = cF \cdot abs[(H_D h_D + H_T h_T + H_G h_G) F_s]$	=	3 977 610.2 N·m	35 204 813.0 lbf·in
Flange design moment for the gasket seating condition	$M_g = cF \cdot \frac{W_g (C - G) F_s}{2}$	=	3 965 864.9 N·m	35 100 858.6 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	725.00 mm	28.543 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	132.39 MPa	19 201.1 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1) M_o}{L t^2 B'}$	=	33.71 MPa	4 889.9 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	116.37 MPa	16 877.4 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	$S_H = \frac{f M_g}{L g_1^2 B_1'}$	=	132.00 MPa	19 144.4 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te + 1) M_g}{L t^2 B'}$	=	33.61 MPa	4 875.4 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{Y M_g}{t^2 B'} - Z S_R$	=	116.02 MPa	16 827.6 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	$J = \frac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=	0.25185

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=	0.21440

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min	=	50.16 mm	1.975 in

g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	21.94 MPa	3 181.9 psi
Hot & corroded (flange)	=	18.94 MPa	2 746.7 psi
New & cold (bolts)	=	21.72 MPa	3 149.8 psi
Hot & corroded (bolts)	=	20.68 MPa	2 999.8 psi

External pressure

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=148.00 MPa / 21 465.6 psi	Sno=148.00 MPa / 21 465.6 psi	Sbo=152.00 MPa / 22 045.7 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 psi	Sbg=152.00 MPa / 22 045.7 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 3-3/4"
Bolt spacing	Bs	=	193.99 mm	7.638 in
Nominal bolt diameter	a	=	95.25 mm	3.750 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	94 023 N	21 137.30 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	3 255 838 N	731 941.41 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	21 420.0 mm²	33.201 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	130 425.5 mm²	202.160 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	249 974.1 mm²	387.461 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	11 540 260 N	2 594 353.52 lbf

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		2.23622
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		2.56106
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.42288
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		2.81435
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		1.49991
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	219.97 mm	8.660 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.18110
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.52280
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.87835
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.42757
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00399
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		8 407 169.84802
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		5.11628
Gasket load for the operating condition	$H_G = W_o - H$	=	40 721 N	9 154.50 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	$H = 0.785G^2P$	=	53 302 N	11 982.81 lbf
Difference	$H_T = H - H_D$	=	20 699 N	4 653.41 lbf
Moment arm for load HD	$hD = \frac{C - B' - g_1'}{2}$	=	255.00 mm	10.039 in
Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	211.53 mm	8.328 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	255.77 mm	10.070 in

Flange moments

Nominal bolt diameter	dB	=	95.25 mm	3.750 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
Flange design moment for the operating condition	$M_o = abs[(H_D(h_D - h_G) + H_T(h_T - h_G))F_s]$	=	2 332.7 N·m	20 646.0 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_G F_s$	=	2 441 164.2 N·m	21 606 121.4 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	725.00 mm	28.543 in
Flange hub stress - operating condition	$S_H = \frac{fM_o}{Lg_1^2 B_1'}$	=	0.08 MPa	11.3 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te+1)M_o}{Lt^2 B'}$	=	0.02 MPa	2.9 psi
Flange tangential stress - operating condition	$S_T = \frac{YM_o}{t^2 B'} - ZS_R$	=	0.07 MPa	9.9 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	$S_H = \frac{fM_g}{Lg_1^2 B_1'}$	=	81.25 MPa	11 784.2 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2 B'}$	=	20.69 MPa	3 001.0 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2 B'} - ZS_R$	=	71.42 MPa	10 358.1 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	$J = \frac{52.14VM_o}{LE_{yo}g_0^2 K_R h_o}$	=	0.00013

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2 K_R h_o}$	=	0.13197

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Flange In Tube Side 26" (Bolting)				
Material		=		SA-193 B16
Governing Thickness		=	95.25 mm	3.750 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F
Welding neck flange - Flange In Tube Side 26", Flange				
Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.20 mm	3.000 in
Reduction in MDMT based on available excess thickness	TR	=	1.47 °C	34.65 °F
Unadjusted MDMT from table UCS-66		=	26.00 °C	78.80 °F
Design pressure	P	=	18.44 MPa	2 674.0 psi
Maximum allowable working pressure	MAWP H&C	=	18.94 MPa	2 746.7 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\, H\ \&\ C}$	=		0.97351
Adjusted MDMT from fig. UCS-66.1		=	24.53 °C	76.15 °F

External pressure

Welding neck flange - Flange In Tube Side 26" (Bolting)				
Material		=		SA-193 B16
Governing Thickness		=	95.25 mm	3.750 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F
Welding neck flange - Flange In Tube Side 26", Flange				
Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.20 mm	3.000 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	26.00 °C	78.80 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	18.94 MPa	2 746.7 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\, H\ \&\ C}$	=		0.00544
Adjusted MDMT from fig. UCS-66.1		=	-54.00 °C	-65.20 °F

Nozzle - Bocchello Out Tube Side

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - 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
Material: SA-182 F22 3 - Forgings				
Allowable stress	S	=	127.76 MPa	18 530.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi
Geometry				
Inside diameter	D	=	635.00 mm	25.000 in
Outside diameter	Do	=	787.40 mm	31.000 in
Length	L	=	581.00 mm	22.874 in
Adopted thickness	t	=	76.20 mm	3.000 in
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		10,71%
Internal pressure				
Allowable stress	S	=	127.76 MPa	18 530.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	317.50 mm	12.500 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE-0.6P} + c + c_e + c'$	=	50.16 mm	1.975 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE+0.4P} + c + c_e + c'$	=	22.27 mm	0.877 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	50.16 mm	1.975 in
				t ≥ tr: Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold (opening)		=	21.76 MPa	3 156.3 psi
Hot & corroded (opening)		=	18.79 MPa	2 724.7 psi
New & cold (cylinder)		=	26.80 MPa	3 887.4 psi
Hot & corroded (cylinder)		=	26.80 MPa	3 887.4 psi

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
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
Outside diameter	Do	=	787.40 mm	31.000 in
Modulus of elasticity	E	=	200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L	=	331.00 mm	13.031 in
L / Do ratio		=		0.42037
Do / t ratio		=		10.33333
Factor A		=		0.09136
Factor B		=	154.29 MPa	22 377.8 psi
External pressure	$\text{Pa} = \frac{4B}{3[D_o/(t-c-c_e-c')]}$	=	19.91 MPa	2 887.5 psi
Required thickness	tr	=	1.77 mm	0.070 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	19.91 MPa	2 887.5 psi
Hot & corroded	=	19.91 MPa	2 887.5 psi

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Minimum Design Metal Temperature (MDMT)

Internal pressure

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

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	8.72 °C	47.70 °F
Unadjusted MDMT from table UCS-66		=	34.50 °C	94.10 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	97.79 mm	3.850 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.84304
Adjusted MDMT from fig. UCS-66.1		=	25.78 °C	78.40 °F

External pressure

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

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	34.50 °C	94.10 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	0 mm	0 in
Minimum required thickness in corroded condition	Tr	=	4.77 mm	0.188 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.04112
Adjusted MDMT from fig. UCS-66.1		=	-45.50 °C	-49.90 °F

Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1 238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]

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Welding neck flange - Flange Out Tube Side 26"

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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 - Bolting**
Gasket **Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	Flange	Hub	Bolting
Design condition	Sfo=127.76 MPa / 18 530.0 psi	Sno=127.76 MPa / 18 530.0 psi	Sbo=144.76 MPa / 20 995.7 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 psi	Sbg=152.00 MPa / 22 045.7 psi

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	=	1 420.00 mm	55.906 in
Inside diameter	B	=	635.00 mm	25.000 in
Bolt circle	C	=	1 235.00 mm	48.622 in
Flange thickness	t	=	310.00 mm	12.205 in
Mean gasket diameter	Gmean	=	742.00 mm	29.213 in
Thickness of the hub at the small end	g0	=	76.20 mm	3.000 in
Thickness of the hub at the large end	g1	=	90.00 mm	3.543 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	76.20 mm	3.000 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	90.00 mm	3.543 in
Hub length	h	=	115.00 mm	4.528 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	70.00 MPa	10 152.6 psi
Gasket contact width	N	=	106.40 mm	4.189 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	53.20 mm	2.094 in
Conversion factor for length	Cb	=		2.50000
Effective gasket contact width	$b = C_b\sqrt{b_0}$	=	18.23 mm	0.718 in
Outside diameter of the gasket contact area	$G_c = G_{\text{mean}} + N$	=	848.40 mm	33.402 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	811.93 mm	31.966 in

Bolt loads

Number of bolts	n	=		20
Bolt type		=		ANSI_TEMA 3-3/4"
Bolt spacing	Bs	=	193.99 mm	7.638 in
Nominal bolt diameter	a	=	95.25 mm	3.750 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	16 829 735 N	3 783 474.67 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	3 255 838 N	731 941.41 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	116 259.6 mm²	180.203 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	130 425.5 mm²	202.160 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	249 974.1 mm²	387.461 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	18 748 069 N	4 214 733.16 lbf
				Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		2.23622
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		2.56106
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K) - 1}{(1.04720 + 1.9448K^2)(K-1)}$	=		1.42288
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K) - 1}{1.36136(K^2-1)(K-1)}$	=		2.81435
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		1.49991
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	219.97 mm	8.660 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.18110
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.52280
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.87835
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.42757
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00399
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		8 407 169.84802
Stress factor L	$L = \frac{\mathrm{te}+1}{T} + \frac{t^3}{d}$	=		5.11628
Gasket load for the operating condition	$H_G = W_o - H$	=	7 288 903 N	1 638 610.48 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785B^2P$	=	5 835 736 N	1 311 925.51 lbf
Total hydrostatic end force	$H = 0.785G^2P$	=	9 540 832 N	2 144 864.19 lbf
Difference	$H_T = H - H_D$	=	3 705 096 N	832 938.67 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	255.00 mm	10.039 in
Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	211.53 mm	8.328 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	255.77 mm	10.070 in

Flange moments

Nominal bolt diameter	dB	=	95.25 mm	3.750 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
Flange design moment for the operating condition	$M_o = cF \cdot abs[(H_D h_D + H_T h_T + H_G h_G) F_s]$	=	3 977 610.2 N·m	35 204 813.0 lbf·in
Flange design moment for the gasket seating condition	$M_g = cF \cdot \frac{W_g (C - G) F_s}{2}$	=	3 965 864.9 N·m	35 100 858.6 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	725.00 mm	28.543 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	132.39 MPa	19 201.1 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te + 1) M_o}{L t^2 B'}$	=	33.71 MPa	4 889.9 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	116.37 MPa	16 877.4 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	$S_H = \frac{f M_g}{L g_1^2 B_1'}$	=	132.00 MPa	19 144.4 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te + 1) M_g}{L t^2 B'}$	=	33.61 MPa	4 875.4 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{Y M_g}{t^2 B'} - Z S_R$	=	116.02 MPa	16 827.6 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	$J = \frac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=	0.25185

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=	0.21440

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min	=	50.16 mm	1.975 in

g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	21.94 MPa	3 181.9 psi
Hot & corroded (flange)	=	18.94 MPa	2 746.7 psi
New & cold (bolts)	=	21.72 MPa	3 149.8 psi
Hot & corroded (bolts)	=	20.68 MPa	2 999.8 psi

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External pressure

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=148.00 MPa / 21 465.6 psi	Sno=148.00 MPa / 21 465.6 psi	Sbo=152.00 MPa / 22 045.7 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 psi	Sbg=152.00 MPa / 22 045.7 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 3-3/4"
Bolt spacing	Bs	=	193.99 mm	7.638 in
Nominal bolt diameter	a	=	95.25 mm	3.750 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	94 023 N	21 137.30 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	3 255 838 N	731 941.41 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	21 420.0 mm²	33.201 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	130 425.5 mm²	202.160 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	249 974.1 mm²	387.461 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	11 540 260 N	2 594 353.52 lbf

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		2.23622
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2 \log_{10} K}{K^2-1}\right)\right]$	=		2.56106
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.42288
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		2.81435
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		1.49991
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	219.97 mm	8.660 in
Hub thickness ratio	$X_g = \frac{g_1}{g_0'}$	=		1.18110
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.52280
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.87835
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.42757
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00399
Stress factor d	$d = \frac{Ug_0^2 h_o}{V}$	=		8 407 169.84802
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		5.11628
Gasket load for the operating condition	$H_G = W_o - H$	=	40 721 N	9 154.50 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	$H = 0.785G^2P$	=	53 302 N	11 982.81 lbf
Difference	$H_T = H - H_D$	=	20 699 N	4 653.41 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	255.00 mm	10.039 in
Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	211.53 mm	8.328 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	255.77 mm	10.070 in

Flange moments

Nominal bolt diameter	dB	=	95.25 mm	3.750 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
Flange design moment for the operating condition	$M_o = \mathrm{abs}[\,(H_D(h_D - h_G) + H_T(h_T - h_G))F_s\,]$	=	2 332.7 N·m	20 646.0 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_G F_s$	=	2 441 164.2 N·m	21 606 121.4 lbf·in

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Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1' =		725.00 mm	28.543 in
Flange hub stress - operating condition	$S_H = \frac{fM_o}{Lg_1^2 B_1'}$ =		0.08 MPa	11.3 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te+1)M_o}{Lt^2 B'}$ =		0.02 MPa	2.9 psi
Flange tangential stress - operating condition	$S_T = \frac{YM_o}{t^2 B'} - ZS_R$ =		0.07 MPa	9.9 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	$S_H = \frac{fM_g}{Lg_1^2 B_1'}$ =	81.25 MPa	11 784.2 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2 B'}$ =	20.69 MPa	3 001.0 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2 B'} - ZS_R$ =	71.42 MPa	10 358.1 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	$J = \frac{52.14VM_o}{LE_{yo}g_0^2 K_R h_o}$ =	0.00013

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2 K_R h_o}$ =	0.13197

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Flange Out Tube Side 26" (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	95.25 mm	3.750 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Welding neck flange - Flange Out Tube Side 26", Flange

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.20 mm	3.000 in
Reduction in MDMT based on available excess thickness	TR	=	1.47 °C	34.65 °F
Unadjusted MDMT from table UCS-66		=	26.00 °C	78.80 °F
Design pressure	P	=	18.44 MPa	2 674.0 psi
Maximum allowable working pressure	MAWP H&C	=	18.94 MPa	2 746.7 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.97351
Adjusted MDMT from fig. UCS-66.1		=	24.53 °C	76.15 °F

External pressure

Welding neck flange - Flange Out Tube Side 26" (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	95.25 mm	3.750 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Welding neck flange - Flange Out Tube Side 26", Flange

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.20 mm	3.000 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	26.00 °C	78.80 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	18.94 MPa	2 746.7 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.00544
Adjusted MDMT from fig. UCS-66.1		=	-54.00 °C	-65.20 °F

Welding neck flange - Shell Flange

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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 - Bolting**
Gasket	**Grooved Metal - Stainless steels and nickel-base alloys**

Allowable stresses	**Flange**	**Hub**	**Bolting**
Design condition	Sfo=133.60 MPa / 19 377.0 psi	Sno=133.60 MPa / 19 377.0 psi	Sbo=138.00 MPa / 20 015.2 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 psi	Sbg=138.00 MPa / 20 015.2 psi

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	=	2 010.00 mm	79.134 in
Inside diameter	B	=	1 275.00 mm	50.197 in
Bolt circle	C	=	1 810.00 mm	71.260 in
Flange thickness	t	=	275.00 mm	10.827 in
Mean gasket diameter	Gmean	=	1 352.85 mm	53.262 in
Thickness of the hub at the small end	g0	=	76.00 mm	2.992 in
Thickness of the hub at the large end	g1	=	133.00 mm	5.236 in
Thickness of the hub at the small end (Corroded)	g0' = g0 - c	=	73.00 mm	2.874 in
Thickness of the hub at the large end (Corroded)	g1' = g1 - c	=	130.00 mm	5.118 in
Hub length	h	=	155.00 mm	6.102 in

Gasket parameters

Gasket factor	m	=		4.25
Gasket factor	y	=	70.00 MPa	10 152.6 psi
Gasket contact width	N	=	77.85 mm	3.065 in
Basic gasket seating width	$b_0 = \frac{N}{2}$	=	38.93 mm	1.532 in
Conversion factor for length	Cb	=		2.50000
Effective gasket contact width	$b = C_b\sqrt{b_0}$	=	15.60 mm	0.614 in
Outside diameter of the gasket contact area	$G_c = G_{\mathrm{mean}} + N$	=	1 430.70 mm	56.327 in
Diameter at the location of the gasket load reaction	$G = G_c - 2b$	=	1 399.51 mm	55.099 in

Bolt loads

Number of bolts	n	=		24
Bolt type		=		ANSI_TEMA 4-1/2"
Bolt spacing	Bs	=	236.93 mm	9.328 in
Nominal bolt diameter	a	=	114.30 mm	4.500 in
Maximum bolt spacing	$B_{s\,\max} = 2a + \frac{6t}{m+0.5}$	=	575.97 mm	22.676 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	30 151 572 N	6 778 342.47 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi \mathrm{bG}y$	=	4 800 392 N	1 079 170.95 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	218 489.7 mm²	338.660 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	224 051.2 mm²	347.280 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	347 241.7 mm²	538.226 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	30 535 316 N	6 864 611.62 lbf

Bs <= BsMax: Ok

Ab ≥ Am: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.56909
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		4.48433
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K) - 1}{(1.04720+1.9448K^2)(K-1)}$	=		1.68080
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K) - 1}{1.36136(K^2-1)(K-1)}$	=		4.92782
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		2.36796
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	305.80 mm	12.039 in
Hub thickness ratio	$X_g = \frac{g_1'}{g_0'}$	=		1.78082
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.50687
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.83184
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.25473
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00272
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		31 524 790.28339
Stress factor L	$L = \frac{\mathrm{te}+1}{T} + \frac{t^3}{d}$	=		1.69971
Gasket load for the operating condition	$H_G = W_o - H$	=	8 288 697 N	1 863 373.06 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785B^2P$	=	18 317 095 N	4 117 846.38 lbf
Total hydrostatic end force	$H = 0.785G^2P$	=	21 862 875 N	4 914 969.42 lbf
Difference	$H_T = H - H_D$	=	3 545 780 N	797 123.04 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	199.50 mm	7.854 in

Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	205.25 mm	8.081 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	234.87 mm	9.247 in

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
Bolt spacing correction factor	$B_{sc} = \max\left[\sqrt{\frac{B_s}{2a+t}}; 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Flange design moment for the operating condition	$M_o = \mathrm{cF} \cdot \mathrm{abs}[((H_D h_D + H_T h_T + H_G h_G) B_{sc}) F_s]$	=	6 188 305.2 N·m	54 771 111.1 lbf·in
Flange design moment for the gasket seating condition	$M_g = \mathrm{cF} \cdot \frac{W_g (C-G) B_{sc} F_s}{2}$	=	6 267 296.6 N·m	55 470 243.8 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	1 411.00 mm	55.551 in
Flange hub stress - operating condition	$S_H = \frac{f M_o}{L g_1^2 B_1'}$	=	152.68 MPa	22 144.4 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te+1) M_o}{L t^2 B'}$	=	74.97 MPa	10 874.0 psi
Flange tangential stress - operating condition	$S_T = \frac{Y M_o}{t^2 B'} - Z S_R$	=	108.92 MPa	15 797.5 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	$S_H = \frac{f M_g}{L g_1^2 B_1'}$	=	154.63 MPa	22 427.0 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1) M_g}{L t^2 B'}$	=	75.93 MPa	11 012.8 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{Y M_g}{t^2 B'} - Z S_R$	=	110.31 MPa	15 999.1 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	$J = \frac{52.14 V M_o}{L E_{yo} g_0^2 K_R h_o}$	=		0.54228

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14 V M_g}{L E_{yg} g_0^2 K_R h_o}$	=		0.47623

Jg ≤ 1: Ok

Hub thickness

Minimum hub thickness as cylindrical shell	th,min	=	75.82 mm	2.985 in
				g0 ≥ th,min: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	16.41 MPa	2 380.4 psi
Hot & corroded (flange)	=	14.52 MPa	2 106.5 psi
New & cold (bolts)	=	14.58 MPa	2 114.9 psi
Hot & corroded (bolts)	=	14.58 MPa	2 114.9 psi

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Date ________ Calc. ________ Contr.________ Appr.________

External pressure

Allowable stresses	Flange	Hub	Bolting
Design condition	Sfo=148.00 MPa / 21 465.6 psi	Sno=148.00 MPa / 21 465.6 psi	Sbo=138.00 MPa / 20 015.2 psi
Seating condition	Sfg=148.00 MPa / 21 465.6 psi	Sng=148.00 MPa / 21 465.6 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	=		24
Bolt type		=		ANSI_TEMA 4-1/2"
Bolt spacing	Bs	=	236.93 mm	9.328 in
Nominal bolt diameter	a	=	114.30 mm	4.500 in
Maximum bolt spacing	$B_{s\,\max} = 2a + \frac{6t}{m+0.5}$	=	575.97 mm	22.676 in
Design bolt load for the operating condition	$W_o = 0.785G^2P + 2b\pi\mathrm{GmP}$	=	218 403 N	49 098.93 lbf
Design bolt load for the gasket seating (Bolt)	$W_{gs} = \pi\mathrm{bG}y$	=	4 800 392 N	1 079 170.95 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	$A_m = \max\left[\left(\frac{W_o + F_A + \frac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right]$	=	34 785.4 mm²	53.918 in²
Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion	Ab	=	224 051.2 mm²	347.280 in²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}$	=	347 241.7 mm²	538.226 in²
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right)S_{bg}$	=	17 859 726 N	4 015 025.86 lbf

Bs <= BsMax: Ok

Flange constants

Ratio of the flange outside diameter to the inside diameter	K = A/B'	=		1.56909
Stress factor Y	$Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=		4.48433
Stress factor T	$T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.04720+1.9448K^2)(K-1)}$	=		1.68080
Stress factor U	$U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=		4.92782
Stress factor Z	$Z = \frac{(K^2+1)}{(K^2-1)}$	=		2.36796
Hub length parameter	$h_o = \sqrt{Bg_0'}$	=	305.80 mm	12.039 in
Hub thickness ratio	$X_g = \frac{g_1}{g_0'}$	=		1.78082
Hub length ratio	$X_h = \frac{h}{h_0}$	=		0.50687
Flange stress factor for integral type flanges	F (fig. 2-7.2)	=		0.83184
Flange stress factor for integral type flanges	V (fig. 2-7.3)	=		0.25473
Hub stress correction factor for integral flanges	f (fig. 2-7.6)	=		1.00000
Stress factor e	$e = \frac{F}{h_o}$	=		0.00272
Stress factor d	$d = \frac{Ug_0^2h_o}{V}$	=		31 524 790.28339
Stress factor L	$L = \frac{te+1}{T} + \frac{t^3}{d}$	=		1.69971
Gasket load for the operating condition	$H_G = W_o - H$	=	60 039 N	13 497.35 lbf
Total hydrostatic end force on the area inside of the flange	$H_D = 0.785B^2P$	=	132 680 N	29 827.63 lbf
Total hydrostatic end force	$H = 0.785G^2P$	=	158 364 N	35 601.59 lbf
Difference	$H_T = H - H_D$	=	25 684 N	5 773.96 lbf
Moment arm for load HD	$\mathrm{hD} = \frac{C - B' - g_1'}{2}$	=	199.50 mm	7.854 in
Moment arm for load HG	$h_G = \frac{C-G}{2}$	=	205.25 mm	8.081 in
Moment arm for load HT	$h_T = \frac{1}{2}\left[\frac{C-B'}{2} + h_G\right]$	=	234.87 mm	9.247 in

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
Bolt spacing correction factor	$B_{sc} = \max\left[\sqrt{\frac{B_s}{2a+t}}; 1\right]$	=		1.00000
Moment factor used to design split rings	Fs	=		1.00
Flange design moment for the operating condition	$M_o = \mathrm{abs}[(H_D(h_D - h_G) + H_T(h_T - h_G))F_s]$	=	1.7 N·m	14.7 lbf·in
Flange design moment for the gasket seating condition	$M_g = W_g h_g F_s$	=	3 665 663.8 N·m	32 443 854.9 lbf·in

Flange stresses - operating condition

Corrected inside diameter of the flange	B1'=B'+g1'	=	1 411.00 mm	55.551 in
Flange hub stress - operating condition	$S_H = \frac{fM_o}{Lg_1^2B_1'}$	=	0.00004 MPa	0.006 psi
Flange radial stress - operating condition	$S_R = \frac{(1.33te+1)M_o}{Lt^2B'}$	=	0.00002 MPa	0.003 psi
Flange tangential stress - operating condition	$S_T = \frac{YM_o}{t^2B'} - ZS_R$	=	0.00003 MPa	0.004 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	$S_H = \frac{fM_g}{Lg_1^2B_1'}$	=	90.44 MPa	13 117.3 psi
Flange radial stress - gasket seating condition	$S_R = \frac{(1.33te+1)M_g}{Lt^2B'}$	=	44.41 MPa	6 441.2 psi
Flange tangential stress - gasket seating condition	$S_T = \frac{YM_g}{t^2B'} - ZS_R$	=	64.52 MPa	9 357.7 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	$J = \frac{52.14VM_o}{LE_{yo}g_0^2K_Rh_o}$	=	0.00000

Jo ≤ 1: Ok

Flange rigidity - seating condition

Flange rigidity index	$J = \frac{52.14VM_g}{LE_{yg}g_0^2K_Rh_o}$	=	0.27854

Jg ≤ 1: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Welding neck flange - Shell Flange (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	114.30 mm	4.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Welding neck flange - Shell Flange, Flange

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	1.16 °C	34.10 °F
Unadjusted MDMT from table UCS-66		=	43.00 °C	109.40 °F
Design pressure	P	=	14.22 MPa	2 062.4 psi
Maximum allowable working pressure	MAWP H&C	=	14.52 MPa	2 106.5 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.97904
Adjusted MDMT from fig. UCS-66.1		=	41.84 °C	107.30 °F

External pressure

Welding neck flange - Shell Flange (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	114.30 mm	4.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Welding neck flange - Shell Flange, Flange

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	43.00 °C	109.40 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	14.52 MPa	2 106.5 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.00709
Adjusted MDMT from fig. UCS-66.1		=	-37.00 °C	-34.60 °F

Cylindrical shell - Main shell

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - 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
Material: SA-387 22 2 - Plate				
Allowable stress	S	=	133.60 MPa	19 377.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi
Geometry				
Inside diameter	D	=	1 275.00 mm	50.197 in
Outside diameter	Do	=	1 427.00 mm	56.181 in
Length	L	=	3 594.00 mm	141.496 in
Adopted thickness	t	=	76.00 mm	2.992 in
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		5,63%
Ligament Efficiency				
Reference figure		=		None
Diameter of tube holes	d	=	0 mm	0 in
Internal pressure				
Allowable stress	S	=	133.60 MPa	19 377.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	637.50 mm	25.098 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE-0.6P}+c+c_e+c'$	=	75.82 mm	2.985 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE+0.4P}+c+c_e+c'$	=	36.38 mm	1.432 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	75.82 mm	2.985 in
Item service	Service	=		NotSpecified
Minimum required thickness as per UG-16(b), including corrosion	tr UG-16(b)	=	4.50 mm	0.177 in
				t ≥ tr: Ok
				t ≥ tr UG-16(b): Ok
Maximum allowable pressures (at the top of the vessel)				
New & cold		=	16.47 MPa	2 388.2 psi
Hot & corroded		=	14.25 MPa	2 067.1 psi

External pressure

External design temperature	Te =		20.00 °C	68.00 °F
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
Outside diameter	Do =		1 427.00 mm	56.181 in
Modulus of elasticity	E =		200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L =		3 771.04 mm	148.466 in
L / Do ratio	=			2.64263
Do / t ratio	=			19.54795
Factor A	=			0.00535
Factor B	=		133.56 MPa	19 371.3 psi
External pressure	$\mathrm{Pa} = \frac{4B}{3[D_o/(t-c-c_e-c')]}$ =		9.11 MPa	1 321.3 psi
Required thickness	tr =		9.88 mm	0.389 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	9.51 MPa	1 379.8 psi
Hot & corroded	=	9.11 MPa	1 321.3 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Cylindrical shell - Main shell

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	0.14 °C	32.24 °F
Unadjusted MDMT from table UCS-66		=	43.00 °C	109.40 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	72.82 mm	2.867 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.99756
Adjusted MDMT from fig. UCS-66.1		=	42.86 °C	109.16 °F

External pressure

Cylindrical shell - Main shell

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	43.00 °C	109.40 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	6.88 mm	0.271 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.09425
Adjusted MDMT from fig. UCS-66.1		=	-37.00 °C	-34.60 °F

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 - In Shell Side

According to: Asme VIII Div. 1 Ed. 2015, UG-36 - 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
Nozzle material: SA-182 F22 3 - Forgings				
Allowable stress in nozzle	Sn	=	133.60 MPa	19 377.0 psi
Shell material: SA-387 22 2 - Plate				
Allowable stress in vessel	Sv	=	133.60 MPa	19 377.0 psi
Nozzle geometry				
Outside diameter	do	=	597.00 mm	23.504 in
Inside diameter	di	=	457.20 mm	18.000 in
Nozzle thickness	tn	=	69.90 mm	2.752 in
Nozzle reinforcement thickness	tx	=	130.00 mm	5.118 in
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Tapering angle	ta	=		45.00 °
Corner radius	r	=	10.00 mm	0.394 in
Nozzle corrosion allowance	cn	=	3.00 mm	0.118 in
Nozzle undertolerance	c'n	=	0 mm	0 in
Nozzle connection		=		Integrally reinforced
Nozzle position		=		Radial
Offset from shell border		=	420.00 mm	16.535 in
Angular offset		=		0 °
Weld leg length of the outside nozzle fillet weld	two	=	15.00 mm	0.591 in
Minimum weld leg length of the outside nozzle fillet weld		=	8.49 mm	0.334 in
				r>=Min[1/4t,1/8 in. (3mm)]: Ok
Opening geometry				
Finished diameter of circular opening	d	=	463.20 mm	18.236 in
Finished radius of circular opening	Rn	=	231.60 mm	9.118 in
Nearest opening		=		Out Shell Side
Distance to nearest opening		=	3 532.33 mm	139.068 in
Maximum distance before multiple openings calculation occurs		=	928.91 mm	36.571 in
Shell Geometry				
Inside shell diameter	D	=	1 275.00 mm	50.197 in
Inside radius of shell course under consideration	R	=	637.50 mm	25.098 in
Shell thickness	t	=	76.00 mm	2.992 in
Shell corrosion allowance	cs	=	3.00 mm	0.118 in
Shell undertolerance	c's	=	0 mm	0 in

Internal pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	73.00 mm	2.874 in
Net required thickness of a seamless shell or formed head	tr	=	72.82 mm	2.867 in
Net nozzle thickness	tn'=tn-cn-c'n	=	66.90 mm	2.634 in
Net required thickness of a seamless nozzle wall	trn	=	26.33 mm	1.037 in
Nozzle reinforcement thickness	tx'=tx-cn	=	127.00 mm	5.000 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	463.20 mm	18.236 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	61.40 mm	2.417 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	182.50 mm	7.185 in

Internal pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Correction factor	F	=		1.00000
Internal pressure	P	=	14.22 MPa	2 062.4 psi
Static head internal	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\begin{Bmatrix} d(E_1 t - F t_r) - 2t_x'(E_1 t - F t_r)(1 - f_{r1}) \\ 2(t + t_x')(E_1 t - F t_r) - 2t_x'(E_1 t - F t_r)(1 - f_{r1}) \end{Bmatrix}$	=	67.2 mm²	0.104 in²
Corner area to be subtracted from nozzle area	$A_7 = (r^2 - \pi\frac{r^2}{4}) f_{r2}$	=	21.5 mm²	0.033 in²
Area available in outward nozzle	$A_2 = \min\begin{Bmatrix} 5(t_x' - t_{rn}) f_{r2} t \\ 5(t_x' - t_{rn}) f_{r2} t_x' \end{Bmatrix} - A_7$	=	36 701.0 mm²	56.887 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=d·tr·F+2·tx'·tr·F(1-fr1)	=	33 731.0 mm²	52.283 in²
Total available area	At=A1+A2+A3+A41+A43	=	36 993.2 mm²	57.340 in²
				At ≥ A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	15.20 MPa	2 204.2 psi
Total pressure	Pt	=	14.22 MPa	2 062.4 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	4.50 mm	0.177 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	29.33 mm	1.155 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	75.82 mm	2.985 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	75.82 mm	2.985 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	3.49 mm	0.138 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	4.50 mm	0.177 in
Minimum thickness of a standard wall pipe	tb3	=	11.34 mm	0.446 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	11.34 mm	0.446 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	29.33 mm	1.155 in
Nozzle thickness	tn	=	69.90 mm	2.752 in

tn ≥ t(UG-45): Ok

External pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	73.00 mm	2.874 in
Net required thickness of a seamless shell or formed head	tr	=	6.88 mm	0.271 in
Net nozzle thickness	tn'=tn-cn-c'n	=	66.90 mm	2.634 in
Net required thickness of a seamless nozzle wall	trn	=	3.08 mm	0.121 in
Nozzle reinforcement thickness	tx'=tx-cn	=	127.00 mm	5.000 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	463.20 mm	18.236 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	61.40 mm	2.417 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	182.50 mm	7.185 in

External pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Correction factor	F	=		1.00000
External pressure	P	=	0.10 MPa	14.9 psi
Static head external	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\begin{Bmatrix} d(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \\ 2(t + t_x')(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \end{Bmatrix}$	=	24 914.0 mm²	38.617 in²
Area available in outward nozzle	$A_2 = \min\begin{Bmatrix} 5(t_x' - t_{rn}) f_{r2} t' \\ 5(t_x' - t_{rn}) f_{r2} t_x' \end{Bmatrix} - A_7$	=	45 187.9 mm²	70.041 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=[d·tr·F+2·tx'·tr·F(1-fr1)]/2	=	1 593.4 mm²	2.470 in²
Total available area	At=A1+A2+A3+A41+A43	=	70 326.9 mm²	109.007 in²
				At ≥ A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	13.11 MPa	1 900.8 psi
Total pressure	Pt	=	0.10 MPa	14.9 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	4.50 mm	0.177 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	6.08 mm	0.239 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	75.82 mm	2.985 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	75.82 mm	2.985 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	3.49 mm	0.138 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	4.50 mm	0.177 in
Minimum thickness of a standard wall pipe	tb3	=	11.34 mm	0.446 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	11.34 mm	0.446 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	11.34 mm	0.446 in
Nozzle thickness	tn	=	69.90 mm	2.752 in

tn ≥ t(UG-45): Ok

Reinforcement of opening - Out Shell Side

According to: Asme VIII Div. 1 Ed. 2015, UG-36 - 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

Nozzle material: SA-182 F22 3 - Forgings

Allowable stress in nozzle	Sn	=	133.60 MPa	19 377.0 psi

Shell material: SA-387 22 2 - Plate

Allowable stress in vessel	Sv	=	133.60 MPa	19 377.0 psi

Nozzle geometry

Outside diameter	do	=	597.00 mm	23.504 in
Inside diameter	di	=	457.20 mm	18.000 in
Nozzle thickness	tn	=	69.90 mm	2.752 in
Nozzle reinforcement thickness	tx	=	130.00 mm	5.118 in
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Tapering angle	ta	=		45.00 °
Corner radius	r	=	10.00 mm	0.394 in
Nozzle corrosion allowance	cn	=	3.00 mm	0.118 in
Nozzle undertolerance	c'n	=	0 mm	0 in
Nozzle connection		=		Integrally reinforced
Nozzle position		=		Radial
Offset from shell border		=	3 150.00 mm	124.016 in
Angular offset		=		180.00 °
Weld leg length of the outside nozzle fillet weld	two	=	15.00 mm	0.591 in
Minimum weld leg length of the outside nozzle fillet weld		=	8.49 mm	0.334 in
				r>=Min[1/4t,1/8 in. (3mm)]: Ok

Opening geometry

Finished diameter of circular opening	d	=	463.20 mm	18.236 in
Finished radius of circular opening	Rn	=	231.60 mm	9.118 in
Nearest opening		=		In Shell Side
Distance to nearest opening		=	3 532.33 mm	139.068 in
Maximum distance before multiple openings calculation occurs		=	928.91 mm	36.571 in

Shell Geometry

Inside shell diameter	D	=	1 275.00 mm	50.197 in
Inside radius of shell course under consideration	R	=	637.50 mm	25.098 in
Shell thickness	t	=	76.00 mm	2.992 in
Shell corrosion allowance	cs	=	3.00 mm	0.118 in
Shell undertolerance	c's	=	0 mm	0 in

Internal pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	73.00 mm	2.874 in
Net required thickness of a seamless shell or formed head	tr	=	72.82 mm	2.867 in
Net nozzle thickness	tn'=tn-cn-c'n	=	66.90 mm	2.634 in
Net required thickness of a seamless nozzle wall	trn	=	26.33 mm	1.037 in
Nozzle reinforcement thickness	tx'=tx-cn	=	127.00 mm	5.000 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	463.20 mm	18.236 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	-78.49 mm	-3.090 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	182.50 mm	7.185 in

Internal pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Correction factor	F	=		1.00000
Internal pressure	P	=	14.22 MPa	2 062.4 psi
Static head internal	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\left\{\begin{matrix} d(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \\ 2(t' + t_x')(E_1 t' - F t_r) - 2t_x'(E_1 t' - F t_r)(1 - f_{r1}) \end{matrix}\right\}$	=	17.3 mm²	0.027 in²
Corner area to be subtracted from nozzle area	$A_7 = (r^2 - \pi\frac{r^2}{4}) f_{r2}$	=	21.5 mm²	0.033 in²
Area available in outward nozzle	$A_2 = \min\left\{\begin{matrix} 5(t_x' - t_{rn}) f_{r2} t' \\ 5(t_x' - t_{rn}) f_{r2} t_x' \end{matrix}\right\} - A_7$	=	36 701.0 mm²	56.887 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=d·tr·F+2·tx'·tr·F(1-fr1)	=	33 731.0 mm²	52.283 in²
Total available area	At=A1+A2+A3+A41+A43	=	36 943.3 mm²	57.262 in²
				At ≥ A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	15.20 MPa	2 204.2 psi
Total pressure	Pt	=	14.22 MPa	2 062.4 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	4.50 mm	0.177 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	29.33 mm	1.155 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	75.82 mm	2.985 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	75.82 mm	2.985 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	3.49 mm	0.138 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	4.50 mm	0.177 in
Minimum thickness of a standard wall pipe	tb3	=	11.34 mm	0.446 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	11.34 mm	0.446 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	29.33 mm	1.155 in
Nozzle thickness	tn	=	69.90 mm	2.752 in
				tn ≥ t(UG-45): Ok

External pressure

Net thicknesses

Net shell thickness	t'=t-cs-c's	=	73.00 mm	2.874 in
Net required thickness of a seamless shell or formed head	tr	=	6.88 mm	0.271 in
Net nozzle thickness	tn'=tn-cn-c'n	=	66.90 mm	2.634 in
Net required thickness of a seamless nozzle wall	trn	=	3.08 mm	0.121 in
Nozzle reinforcement thickness	tx'=tx-cn	=	127.00 mm	5.000 in
Limit of reinforcement parallel to the vessel wall	$L_R = \max(d, R_n + t_n + t)$	=	463.20 mm	18.236 in
Useful limit of reinforcement in vessel wall	Lo=(distance from border)	=	-78.49 mm	-3.090 in
Useful limit of reinforcement normal to the vessel wall	ho=min(2.5·t,2.5·tn)	=	182.50 mm	7.185 in

External pressure

Strength reduction factor	fr	=		1.00000
fr1 factor	fr1=max(Sn/Sv;1)	=		1.00000
fr2 factor	fr2=max(Sn/Sv;1)	=		1.00000
Lenght of projection defining the thickened portion of integral reinforcement	L	=	200.00 mm	7.874 in
Correction factor	F	=		1.00000
External pressure	P	=	0.10 MPa	14.9 psi
Static head external	Ph	=	0 MPa	0 psi
Area available in shell	$A_1 = \max\left\{\begin{matrix} d(E_1t' - Ft_r) - 2t_x'(E_1t' - Ft_r)(1-f_{r1}) \\ 2(t+t_x')(E_1t' - Ft_r) - 2t_x'(E_1t' - Ft_r)(1-f_{r1}) \end{matrix}\right\}$	=	6 415.0 mm²	9.943 in²
Area available in outward nozzle	$A_2 = \min\left\{\begin{matrix} 5(t_x' - t_{rn}) f_{r2} t' \\ 5(t_x' - t_{rn}) f_{r2} t_x' \end{matrix}\right\} - A_7$	=	45 187.9 mm²	70.041 in²
Area available in inward nozzle	A3	=	0 mm²	0 in²
Area available in outward weld	A41=(leg)²·fr2	=	225.0 mm²	0.349 in²
Area available in inward weld	A43=(leg)²·fr2	=	0 mm²	0 in²
Area required	A=[d·tr·F+2·tx'·tr·F(1-fr1)]/2	=	1 593.4 mm²	2.470 in²
Total available area	At=A1+A2+A3+A41+A43	=	51 827.8 mm²	80.333 in²
				At ≥ A: Ok

Opening maximum allowable pressure

Opening maximum allowable pressure	Pmax	=	13.11 MPa	1 900.8 psi
Total pressure	Pt	=	0.10 MPa	14.9 psi
				Pt ≤ Pmax: Ok

Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion	t(UG-16)	=	4.50 mm	0.177 in
Minimum neck thickness required for internal and external pressure using UG-27 and UG-28	ta	=	6.08 mm	0.239 in
Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance	tb1=tr(P.int)-c'	=	75.82 mm	2.985 in
tb1 no less than minimum thickness specified in UG-16(b)	tb1*=max[tb1,t(UG-16)]	=	75.82 mm	2.985 in
Min. thickness required using external pressure as internal (with E=1) for the shell or head	tb2	=	3.49 mm	0.138 in
tb2 no less than minimum thickness specified in UG-16(b)	tb2*=max[tb2,t(UG-16)]	=	4.50 mm	0.177 in
Minimum thickness of a standard wall pipe	tb3	=	11.34 mm	0.446 in
tb	tb=min[tb3,max(tb1*,tb2*]	=	11.34 mm	0.446 in
Minimum required nozzle neck thickness	t(UG-45) = max(ta,tb)	=	11.34 mm	0.446 in
Nozzle thickness	tn	=	69.90 mm	2.752 in

tn ≥ t(UG-45): Ok

Standard Long Welding Neck flange - In Shell Side

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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 (opening)	=	17.98 MPa	2 608.3 psi
Hot & corroded (opening)	=	15.20 MPa	2 204.2 psi
New & cold (cylinder)	=	34.52 MPa	5 006.5 psi
Hot & corroded (cylinder)	=	32.89 MPa	4 770.5 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

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

Cylindrical shell

Inside diameter	D	=	457.20 mm	18.000 in
Outside diameter	Do	=	597.00 mm	23.504 in
Length	L	=	1 016.00 mm	40.000 in
Adopted thickness	t	=	69.90 mm	2.752 in
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		7,10%
Allowable stress	S	=	133.60 MPa	19 377.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	228.60 mm	9.000 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE-0.6P}+c+c_e+c'$	=	29.33 mm	1.155 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE+0.4P}+c+c_e+c'$	=	15.07 mm	0.593 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	29.33 mm	1.155 in
				t ≥ tr: Ok

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
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
Outside diameter	Do	=	915.00 mm	36.024 in
Modulus of elasticity	E	=	200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L	=	1 016.00 mm	40.000 in
L / Do ratio		=		1.11038
Do / t ratio		=		13.67713
Factor A		=		0.02495
Factor B		=	144.44 MPa	20 949.3 psi
External pressure	$Pa = \frac{4B}{3[D_o/(t-c-c_e-c')]}$	=	14.08 MPa	2 042.3 psi
Required thickness	tr	=	6.08 mm	0.239 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Long Welding Neck flange - In Shell Side (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	63.50 mm	2.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Long Welding Neck flange - In Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	69.90 mm	2.752 in
Reduction in MDMT based on available excess thickness	TR	=	10.85 °C	51.53 °F
Unadjusted MDMT from table UCS-66		=	23.00 °C	73.40 °F
Design pressure	P	=	14.22 MPa	2 062.4 psi
Maximum allowable working pressure	MAWP H&C	=	17.67 MPa	2 562.8 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.80473
Impact test exempion temperature		=	12.15 °C	53.87 °F

Note: No impact testing is required for the ASME B16-5 and ASME B16-47 flanges when used at minimum design metal temperatures no colder than −20°F (−29°C)

Long Welding Neck flange - In Shell Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	0.14 °C	32.24 °F
Unadjusted MDMT from table UCS-66		=	25.90 °C	78.62 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	72.82 mm	2.867 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.99756
Adjusted MDMT from fig. UCS-66.1		=	25.76 °C	78.38 °F

External pressure

Long Welding Neck flange - In Shell Side (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	63.50 mm	2.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Long Welding Neck flange - In Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	69.90 mm	2.752 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	23.00 °C	73.40 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	17.67 MPa	2 562.8 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\,H\,\&\,C}$	=		0.00583
Impact test exempion temperature		=	-57.00 °C	-70.60 °F

Long Welding Neck flange - In Shell Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	25.90 °C	78.62 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	6.88 mm	0.271 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.09425
Adjusted MDMT from fig. UCS-66.1		=	-54.10 °C	-65.38 °F

Standard Long Welding Neck flange - Out Shell Side

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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 (opening)		=	17.98 MPa	2 608.3 psi
Hot & corroded (opening)		=	15.20 MPa	2 204.2 psi
New & cold (cylinder)		=	34.52 MPa	5 006.5 psi
Hot & corroded (cylinder)		=	32.89 MPa	4 770.5 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

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Date _______ Calc. _______ Contr._______ Appr._______

Customer
Riccardo Petrelli

Drawing
U_150

Revision

Cylindrical shell

Inside diameter	D	=	457.20 mm	18.000 in
Outside diameter	Do	=	597.00 mm	23.504 in
Length	L	=	1 016.00 mm	40.000 in
Adopted thickness	t	=	69.90 mm	2.752 in
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (cylinders formed from plate)	εf=50·t/(R+t/2)	=		7,10%
Allowable stress	S	=	133.60 MPa	19 377.0 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
Reference diameter		=		Inside
Calculation radius (inside)	R	=	228.60 mm	9.000 in
Required thickness for circumferential stress, UG-27(c)(1)	$t_r = \frac{P(R+c+c')}{SE-0.6P} + c + c_e + c'$	=	29.33 mm	1.155 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R+c+c')}{2SE+0.4P} + c + c_e + c'$	=	15.07 mm	0.593 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	29.33 mm	1.155 in
				t ≥ tr: Ok

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
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
Outside diameter	Do	=	915.00 mm	36.024 in
Modulus of elasticity	E	=	200 000.00 MPa	29 007 547.6 psi
Axial length between reinforcements	L	=	1 016.00 mm	40.000 in
L / Do ratio		=		1.11038
Do / t ratio		=		13.67713
Factor A		=		0.02495
Factor B		=	144.44 MPa	20 949.3 psi
External pressure	$\mathrm{Pa} = \frac{4B}{3[D_o/(t-c-c_e-c')]}$	=	14.08 MPa	2 042.3 psi
Required thickness	tr	=	6.08 mm	0.239 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Minimum Design Metal Temperature (MDMT)

Internal pressure

Long Welding Neck flange - Out Shell Side (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	63.50 mm	2.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Long Welding Neck flange - Out Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	69.90 mm	2.752 in
Reduction in MDMT based on available excess thickness	TR	=	10.85 °C	51.53 °F
Unadjusted MDMT from table UCS-66		=	23.00 °C	73.40 °F
Design pressure	P	=	14.22 MPa	2 062.4 psi
Maximum allowable working pressure	MAWP H&C	=	17.67 MPa	2 562.8 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\, H\,\&\,C}$	=		0.80473
Impact test exempion temperature		=	12.15 °C	53.87 °F

Note: No impact testing is required for the ASME B16-5 and ASME B16-47 flanges when used at minimum design metal temperatures no colder than −20°F (−29°C)

Long Welding Neck flange - Out Shell Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	0.14 °C	32.24 °F
Unadjusted MDMT from table UCS-66		=	25.90 °C	78.62 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	72.82 mm	2.867 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.99756
Adjusted MDMT from fig. UCS-66.1		=	25.76 °C	78.38 °F

External pressure

Long Welding Neck flange - Out Shell Side (Bolting)

Material		=		SA-193 B16
Governing Thickness		=	63.50 mm	2.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

Long Welding Neck flange - Out Shell Side (Flange)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	69.90 mm	2.752 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	23.00 °C	73.40 °F
Design pressure	P	=	0.10 MPa	14.9 psi
Maximum allowable working pressure	MAWP H&C	=	17.67 MPa	2 562.8 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\, H\,\&\,C}$	=		0.00583
Impact test exempion temperature		=	-57.00 °C	-70.60 °F

Long Welding Neck flange - Out Shell Side (Nozzle to wall - shell)

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	76.00 mm	2.992 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	25.90 °C	78.62 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	6.88 mm	0.271 in
Nominal noncorroded thickness	Tn	=	76.00 mm	2.992 in
Coincident ratio	$\frac{t_r E*}{t_n - c}$	=		0.09425
Adjusted MDMT from fig. UCS-66.1		=	-54.10 °C	-65.38 °F

U-Tube tubesheet - Tubesheet

According to: Asme VIII Div. 1 Ed. 2015, UHX-12 - 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	=	127.76 MPa	18 530.0 psi
Allowable primary plus secondary stress for tubesheet material	SPS = 2·Sy	=	461.76 MPa	66 972.6 psi

Tubes material **SB-444 2 Solution ann. N06625 (high allowable stresses) - Smls. pipe & tube**

Tube design temperature	Tt	=	454.00 °C	849.20 °F
Modulus of elasticity for tube material at Tt	Et	=	182 760.00 MPa	26 507 097.0 psi
Allowable stress for tube material at tubesheet design temperature	StT	=	184.00 MPa	26 686.9 psi

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	=	127.76 MPa	18 530.0 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	=	133.60 MPa	19 377.0 psi
Allowable primary plus secondary stress for shell material at Ts	SPSs = 2·Sy	=	475.60 MPa	68 979.9 psi

Bolting material **SA-193 B16 - 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	=	2 010.00 mm	79.134 in
Bolt circle diameter	C	=	1 810.00 mm	71.260 in
Shell corrosion allowance	cs	=	3.00 mm	0.118 in
Shell undertolerance	c's	=	0 mm	0 in
Shell thickness	$t_s = t_{\text{shell}} - c_s - c'_s$	=	73.00 mm	2.874 in
Channel corrosion allowance	cc	=	0 mm	0 in
Channel undertolerance	c'c	=	0 mm	0 in
Channel thickness	$t_c = t_{\text{channel}} - c_c - c'_c$	=	116.00 mm	4.567 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	=	265.00 mm	10.433 in
Tubesheet thickness for calculation	$h = t_{\text{tubesheet}} - c_{\text{ts}} - c_{\text{ss}} - c'$	=	262.00 mm	10.315 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
Square 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	=	262.00 mm	10.315 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	=	1 399.51 mm	55.099 in
Diameter ratio	$\rho_s = \frac{G_s}{D_o}$	=		1.13781
Inside channel diameter	Dc	=	1 238.00 mm	48.740 in
Diameter ratio	$\rho_c = \frac{D_c}{D_o}$	=		1.00650
	$\beta_c = \frac{[12(1-\nu_c^2)]^{0.25}}{[(D_c+t_c)t_c]^{0.5}}$	=	0.0046 mm^-1	0.117 in^-1
	$k_c = \frac{\beta_c E_c t_c^3}{6(1-\nu_c^2)}$	=	235 507 634 N	52 944 217.80 lbf
	$\lambda_c = \frac{6D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right)$	=	284 401.83 MPa	41 248 998.8 psi
	$\delta_c = \frac{D_c^2}{4E_c t_c}\left(1 - \frac{\nu_c}{2}\right)$	=		0.01563
	$\omega_c = \rho_c k_c \beta_c \delta_c (1 + h\beta_c)$	=	37 423.5 mm²	58.007 in²
Diameter ratio	$K = \frac{A}{D_o}$	=		1.63415
Coefficient	$F = \frac{(1-\nu^*)(\lambda_c + E \cdot \ln[K])}{E^*}$	=		2.75916

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	$W_o = 0.785G^2P + 2b\pi \mathrm{GmP}$	=	30 151 572 N	6 778 342.47 lbf
Design bolt load for the gasket seating (Flange)	$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg}$	=	31 472 911 N	7 075 391.36 lbf
Moment arm for load HG	hG = (C-G) / 2	=	205.25 mm	8.081 in
Allowable stress for the tubesheet extension at design temperature	So	=	127.76 MPa	18 530.0 psi
Allowable stress for the tubesheet extension at gasket seating temperature	Sg	=	148.00 MPa	21 465.6 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)	$hr_o = \sqrt{\frac{1.9 W_o h_G}{S_o G}}$	=	256.44 mm	10.096 in
Minimum required thickness of the tubesheet flanged extension (gasket seating)	$hr_g = \sqrt{\frac{1.9 W_g h_G}{S_g G}}$	=	243.43 mm	9.584 in
Minimum required thickness of the tubesheet flanged extension.	$h_r = \left(\frac{1.9 W h_g}{S \cdot G}\right)^{0.5}$	=	256.44 mm	10.096 in
				tfe ≥ hr: Ok

Tube to tubesheet joints

Fillet weld leg	af	=	9.00 mm	0.354 in
Min. required length of the weld leg(s)	$a_r = \sqrt{(0.75 d_o)^2 + 2.73 t (d_o - t) f_w f_d} - 0.75 d_o$	=	8.14 mm	0.321 in
Allowable stress of the tube	Sa	=	184.00 MPa	26 686.9 psi
Allowable stress of the material to which the tube is welded	St	=	127.76 MPa	18 530.0 psi
Allowable stress in weld	$S_w = \min[S_a, S_t]$	=	127.76 MPa	18 530.0 psi
Fillet weld strength	$F_f = 0.55 \pi a_f (d_o + 0.67 a_f) S_w$	=	75 061 N	16 874.29 lbf
Groove weld strength	$F_g = 0.85 \pi a_g (d_o + 0.67 a_g) S_w$	=	0 N	0 lbf
Axial tube strength	$F_t = \pi t (d_o - t) S_a$	=	66 765 N	15 009.37 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	$f_w = \frac{S_a}{S_w}$	=		1.44020
Max axial load (pressure only)	Lmax = Ft	=	66 765 N	15 009.37 lbf
Max axial load (pressure or thermally induced)	Lmax = 2Ft	=	133 530 N	30 018.73 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	=		0.03720
	A1	=		1.03140
	A2	=		-0.64020
	A3	=		2.62010
	A4	=		-2.19290
	B0	=		0.33410
	B1	=		0.12600
	B2	=		-0.69200
	B3	=		0.68770
	B4	=		-0.06000
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.51801
Effective Poisson ratio in perforated region	$\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.31108

Tubesheet bending stress

Tubesheet design bold load	W*	=	0 N	0 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	-25 905 N	-5 823.59 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	664 054 N	149 285.16 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	819 995 N	184 342.21 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + \nu^*)(P_s - P_t)}{64}$	=	-631 107 N	-141 878.52 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	819 995 N	184 342.21 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h'_g)^2}$	=	159.99 MPa	23 204.9 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	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	44.98 MPa	6 523.3 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6(1 - \nu^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	167.70 MPa	24 322.5 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	212.67 MPa	30 845.7 psi

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Chieti
Telephone, Fax
Website, Email address
Date ________ Calc. ________ Contr.________ Appr.________

Customer
Riccardo Petrelli

Drawing
U_150

Revision

Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation	$E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$	=	170 487.48 MPa	24 727 118.4 psi
	$k_c = \frac{\beta_c E_c^* t_c^3}{6(1-\nu_c^2)}$	=	223 558 480 N	50 257 941.34 lbf
	$\lambda_c = \frac{6D_c k_c}{h^3}\left(1+h\beta_c+\frac{h^2\beta_c^2}{2}\right)$	=	269 971.89 MPa	39 156 112.7 psi
	$\omega_c = \rho_c k_c \beta_c \delta_c (1+h\beta_c)$	=	35 524.7 mm²	55.063 in²
Diameter ratio	$K = \frac{A}{D_o}$	=		1.63415
Coefficient	$F = \frac{(1-\nu^*)(\lambda_c + E\cdot\ln[K])}{E^*}$	=		2.65231
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	818 342 N	183 970.49 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	-632 761 N	-142 250.24 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	818 342 N	183 970.49 lbf
Bending stress (simplified Elastic-plastic calculation performed)	$\sigma = \frac{6M}{\mu^*(h-h_g')^2}$	=	159.67 MPa	23 158.1 psi
				σ ≤ 2S: Ok

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	=		0.03720
	A1	=		1.03140
	A2	=		-0.64020
	A3	=		2.62010
	A4	=		-2.19290
	B0	=		0.33410
	B1	=		0.12600
	B2	=		-0.69200
	B3	=		0.68770
	B4	=		-0.06000
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.51801
Effective Poisson ratio in perforated region	$v^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.31108

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	30 151 572 N	6 778 342.47 lbf
Tubesheet design bold load	W* = Wds	=	30 151 572 N	6 778 342.47 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	425 300 N	95 611.24 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	2 022 967 N	454 781.04 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	41 128 N	9 245.95 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + v^*)(P_s - P_t)}{64}$	=	1 162 173 N	261 266.91 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o \rvert, \lvert M_p \rvert]$	=	1 162 173 N	261 266.91 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h'_g)^2}$	=	226.76 MPa	32 888.1 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	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	-0.25 MPa	-36.4 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6(1 - v^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	-367.69 MPa	-53 329.5 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	367.94 MPa	53 365.9 psi

Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation	$E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$	=	129 615.81 MPa	18 799 184.2 psi
	$k_c = \frac{\beta_c E_c^* t_c^3}{6(1-\nu_c^2)}$	=	169 963 883 N	38 209 397.55 lbf
	$\lambda_c = \frac{6D_c k_c}{h^3}\left(1+h\beta_c+\frac{h^2\beta_c^2}{2}\right)$	=	205 250.42 MPa	29 769 056.2 psi
	$\omega_c = \rho_c k_c \beta_c \delta_c(1+h\beta_c)$	=	27 008.2 mm²	41.863 in²
Diameter ratio	$K = \frac{A}{D_o}$	=		1.63415
Coefficient	$F = \frac{(1-\nu^*)(\lambda_c + E\cdot\ln[K])}{E^*}$	=		2.17304
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	173 806 N	39 073.21 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	1 294 852 N	291 094.18 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	1 294 852 N	291 094.18 lbf
Bending stress (simplified Elastic-plastic calculation performed)	$\sigma = \frac{6M}{\mu^*(h-h_g')^2}$	=	252.64 MPa	36 642.8 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	=		0.03720
	A1	=		1.03140
	A2	=		-0.64020
	A3	=		2.62010
	A4	=		-2.19290
	B0	=		0.33410
	B1	=		0.12600
	B2	=		-0.69200
	B3	=		0.68770
	B4	=		-0.06000
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.51801
Effective Poisson ratio in perforated region	$v^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.31108

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	30 151 572 N	6 778 342.47 lbf
Tubesheet design bold load	W* = Wds	=	30 151 572 N	6 778 342.47 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	402 348 N	90 451.34 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	2 693 827 N	605 596.44 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	862 934 N	193 995.24 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + v^*)(P_s - P_t)}{64}$	=	532 877 N	119 795.46 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	862 934 N	193 995.24 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h'_g)^2}$	=	168.37 MPa	24 420.0 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	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	44.98 MPa	6 523.3 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1 - v^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	-200.15 MPa	-29 028.7 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	245.12 MPa	35 551.9 psi

Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation	$E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$	=	158 802.89 MPa	23 032 411.5 psi
	$k_c = \dfrac{\beta_c E_c^* t_c^3}{6(1-\nu_c^2)}$	=	208 236 594 N	46 813 444.59 lbf
	$\lambda_c = \dfrac{6D_c k_c}{h^3}\left(1+h\beta_c+\dfrac{h^2\beta_c^2}{2}\right)$	=	251 469.00 MPa	36 472 495.0 psi
	$\omega_c = \rho_c k_c \beta_c \delta_c (1+h\beta_c)$	=	33 090.0 mm²	51.290 in²
Diameter ratio	$K = \dfrac{A}{D_o}$	=		1.63415
Coefficient	$F = \dfrac{(1-\nu^*)(\lambda_c + E\cdot\ln[K])}{E^*}$	=		2.51529
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \dfrac{M^* - \frac{D_o^2 F(P_s-P_t)}{32}}{1+F}$	=	908 968 N	204 344.12 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \dfrac{D_o^2(3+\nu^*)(P_s-P_t)}{64}$	=	578 911 N	130 144.33 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	908 968 N	204 344.12 lbf
Bending stress (simplified Elastic-plastic calculation performed)	$\sigma = \dfrac{6M}{\mu^*(h-h_g')^2}$	=	177.35 MPa	25 722.7 psi
				σ ≤ 2S: 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	=		0.03720
	A1	=		1.03140
	A2	=		-0.64020
	A3	=		2.62010
	A4	=		-2.19290
	B0	=		0.33410
	B1	=		0.12600
	B2	=		-0.69200
	B3	=		0.68770
	B4	=		-0.06000
	$E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$	=		0.51801
Effective Poisson ratio in perforated region	$\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$	=		0.31108

Tubesheet bending stress

Shell flange design bolt load for the operating condition	Wds	=	-218 403 N	-49 098.93 lbf
Tubesheet design bold load	W* = Wds	=	-218 403 N	-49 098.93 lbf
Moment acting on the unperforated tubesheet rim	$M_{TS} = \frac{D_o^2[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t]}{16}$	=	-2 952 N	-663.69 lbf
Moment acting on the unperforated tubesheet rim	$M^* = M_{TS} + \omega_c P_t + \frac{(C - G_s)W^*}{2\pi D_o}$	=	-18 407 N	-4 138.16 lbf
Maximum bending moments acting on the tubesheet at the periphery	$M_p = \frac{M^* - \frac{D_o^2 F(P_s - P_t)}{32}}{1 + F}$	=	-4 897 N	-1 100.82 lbf
Maximum bending moments acting on the tubesheet at the center	$M_o = M_p + \frac{D_o^2(3 + \nu^*)(P_s - P_t)}{64}$	=	-4 897 N	-1 100.82 lbf
Maximum bending moments acting on the tubesheet	$M = \max[\lvert M_o\rvert, \lvert M_p\rvert]$	=	4 897 N	1 100.82 lbf
Bending stress	$\sigma = \frac{6M}{\mu^*(h - h_g')^2}$	=	0.96 MPa	138.6 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	$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c(D_c + t_c)}$	=	-0.25 MPa	-36.4 psi
Bending stress in the channel at its junction to the tubesheet	$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6(1-\nu^*)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}(P_s - P_t)\right)\right]$	=	1.73 MPa	250.3 psi
Channel axial stress	$\sigma_c = \lvert\sigma_{c,m}\rvert + \lvert\sigma_{c,b}\rvert$	=	1.98 MPa	286.7 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
Governing Thickness		=	114.30 mm	4.500 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	-30.00 °C	-22.00 °F
Adjusted MDMT from fig. UCS-66.1		=	-110.00 °C	-166.00 °F

U-Tube tubesheet - Tubesheet, Flange

Material		=		SA-182 F22 3
Curve of fig. UCS-66		=		B
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	0.54 °C	32.98 °F
Unadjusted MDMT from table UCS-66		=	34.50 °C	94.10 °F
Design pressure	P	=	14.22 MPa	2 062.4 psi
Maximum allowable working pressure	MAWP H&C	=	14.36 MPa	2 082.7 psi
Coincident ratio	$\frac{P}{\mathrm{MAWP}\, H\, \&\, C}$	=		0.99025
Adjusted MDMT from fig. UCS-66.1		=	33.96 °C	93.12 °F

U-Tube tubesheet - Tubesheet, Shell

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	116.00 mm	4.567 in
Reduction in MDMT based on available excess thickness	TR	=	7.48 °C	45.46 °F
Unadjusted MDMT from table UCS-66		=	49.00 °C	120.20 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	97.79 mm	3.850 in
Nominal noncorroded thickness	Tn	=	116.00 mm	4.567 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.86542
Adjusted MDMT from fig. UCS-66.1		=	41.52 °C	106.74 °F

Tube bundle - Tubes bundle

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28, Appendix 1.1 - Metric Units

Design data

Internal design temperature	T	=	454.00 °C	849.20 °F
Internal design pressure	P	=	18.54 MPa	2 688.9 psi
External design temperature	Te	=	420.00 °C	788.00 °F
External design pressure	Pe	=	14.32 MPa	2 077.3 psi
Joint efficiency	E	=		1.00

Material: SB-444 2 Solution ann. N06625 (high allowable stresses) - Smls. pipe & tube

Allowable stress	S	=	184.00 MPa	26 686.9 psi
Allowable stress at room temperature	ST	=	184.00 MPa	26 686.9 psi

Geometry

Inside diameter	D	=	23.37 mm	0.920 in
Outside diameter	Do	=	31.75 mm	1.250 in
Length	L	=	3 658.00 mm	144.016 in
Adopted thickness	t	=	4.19 mm	0.165 in
Corrosion allowance	c	=	0 mm	0 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Forming strain (tubes and pipe bends)	εf=100·r/R	=		33,33%

Internal pressure

Allowable stress	S	=	184.00 MPa	26 686.9 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
Reference diameter		=		Outside
Calculation radius (outside)	Ro	=	15.88 mm	0.625 in
Required thickness for circumferential stress, Appendix 1.1(a)	$t_r = \frac{PR_o}{SE + 0.4P}$	=	1.54 mm	0.061 in
Required thickness for longitudinal stress, UG-27(c)(2)	$t_r = \frac{P(R + c + c')}{2SE + 0.4P} + c + c_e + c'$	=	0.58 mm	0.023 in
Minimum required thickness	tr=max[tr(circ),tr(long)]	=	1.54 mm	0.061 in
				t ≥ tr: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold		=	54.31 MPa	7 877.2 psi
Hot & corroded		=	54.31 MPa	7 877.2 psi

External pressure

External design temperature	Te	=	420.00 °C	788.00 °F
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
Outside diameter	Do	=	31.75 mm	1.250 in
Axial length between reinforcements	L	=	3 393.00 mm	133.583 in
L / Do ratio		=		106.86614
Do / t ratio		=		7.57576
Factor A		=		0.01943
Factor B		=	103.21 MPa	14 969.8 psi
Yield strength at design temperature	Sy	=		206.45450
Allowable stress	S	=	184.00 MPa	26 686.9 psi
	min(2S,0.9Sy)	=		185.80905
	$\mathrm{Pa1} = \left[\frac{2.167}{D_o/(t-c-c_e-c')} - 0.0833\right]\cdot B$	=	20.93 MPa	3 035.0 psi
	$\mathrm{Pa2} = \frac{2S_a}{D_o/(t-c-c_e-c')}\cdot\left[1-\frac{1}{D_o/(t-c-c_e-c')}\right]$	=	42.58 MPa	6 175.5 psi
External pressure	min(Pa1, Pa2)	=	20.93 MPa	3 035.0 psi
Required thickness	tr	=	3.26 mm	0.128 in
				Pa ≥ P: Ok
				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	$t_0 = t_r\left[1+\frac{d_o}{4R}\right]$	=	3.80 mm	0.150 in
				t ≥ t0: Ok

Maximum allowable external pressures

New & cold	=	27.98 MPa	4 058.0 psi
Hot & corroded	=	20.93 MPa	3 035.0 psi

Minimum Design Metal Temperature (MDMT)

Internal pressure

Tube bundle - Tubes bundle

Material	=	SB-444 2 Solution ann. N06625 (high allowable stresses)	
Governing Thickness	=	4.19 mm	0.165 in
Impact test exempion temperature	=	-198.00 °C	-324.40 °F

Note: MDMT according to UNF-65

External pressure

Tube bundle - Tubes bundle

Material	=	SB-444 2 Solution ann. N06625 (high allowable stresses)	
Governing Thickness	=	4.19 mm	0.165 in
Impact test exempion temperature	=	-198.00 °C	-324.40 °F

Note: MDMT according to UNF-65

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Customer
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Drawing
U_150

Revision

Hemispherical head - Head

According to: Asme VIII Div. 1 Ed. 2015, UG-32 / UG-33 - 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

Material: SA-387 22 2 - Plate

Allowable stress	S	=	133.60 MPa	19 377.0 psi
Allowable stress at room temperature	ST	=	148.00 MPa	21 465.6 psi

Geometry

Inside diameter	D	=	1 312.00 mm	51.654 in
Outside diameter	Do	=	1 390.00 mm	54.724 in
Inside crown radius (corroded)	L	=	659.00 mm	25.945 in
Adopted thickness	t	=	39.00 mm	1.535 in
Corrosion allowance	c	=	3.00 mm	0.118 in
External corrosion allowance	ce	=	0 mm	0 in
Wall undertolerance	c'	=	0 mm	0 in
Straight flange length	FL	=	0 mm	0 in
Forming strain (double curvature, e.g., heads)	εf=75·t/(L+t/2)	=		4,33%

Internal pressure

Allowable stress	S	=	133.60 MPa	19 377.0 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
Minimum required thickness	$t_r = \frac{PL}{2SE - 0.2P} + c + c_e + c'$	=	38.45 mm	1.514 in
Item service	Service	=		NotSpecified
Minimum required thickness as per UG-16(b), including corrosion	tr UG-16(b)	=	4.50 mm	0.177 in

t ≥ tr: Ok

t ≥ tr UG-16(b): Ok

Maximum allowable pressures (at the top of the vessel)

New & cold	=	17.39 MPa	2 522.3 psi
Hot & corroded	=	14.44 MPa	2 094.2 psi

External pressure

External design temperature	Te	=	20.00 °C	68.00 °F
Factor A	A=0.125/(R0/t)	=		0.00650
Factor B	B	=	134.90 MPa	19 565.2 psi
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
Outside radius	Ro	=	692.00 mm	27.244 in
Maximum allowable external working pressure	Pa	=	7.02 MPa	1 017.8 psi
Minimum required thickness	tr	=	4.89 mm	0.193 in
				Pa ≥ P: Ok
				t ≥ tr: Ok

Maximum allowable external pressures

New & cold	=	7.60 MPa	1 102.1 psi
Hot & corroded	=	7.02 MPa	1 017.8 psi

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Drawing
U_150

Revision

Minimum Design Metal Temperature (MDMT)

Internal pressure

Hemispherical head - Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	39.00 mm	1.535 in
Reduction in MDMT based on available excess thickness	TR	=	0.85 °C	33.54 °F
Unadjusted MDMT from table UCS-66		=	31.60 °C	88.88 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	35.45 mm	1.396 in
Nominal noncorroded thickness	Tn	=	39.00 mm	1.535 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.98465
Adjusted MDMT from fig. UCS-66.1		=	30.75 °C	87.34 °F

External pressure

Hemispherical head - Head

Material		=		SA-387 22 2
Curve of fig. UCS-66		=		A
Governing Thickness		=	39.00 mm	1.535 in
Reduction in MDMT based on available excess thickness	TR	=	80.00 °C	176.00 °F
Unadjusted MDMT from table UCS-66		=	31.60 °C	88.88 °F
Joint efficiency	E	=		1.00
Corrected joint efficiency	$E^* = \max(E, 0.8)$	=		1.00000
Corrosion allowance	c	=	3.00 mm	0.118 in
Minimum required thickness in corroded condition	Tr	=	1.89 mm	0.075 in
Nominal noncorroded thickness	Tn	=	39.00 mm	1.535 in
Coincident ratio	$\frac{t_r E *}{t_n - c}$	=		0.05263
Adjusted MDMT from fig. UCS-66.1		=	-48.40 °C	-55.12 °F

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]