Pressure Vessel Engineering, Ltd.
120 Randall Dr.   Waterloo, Ontario N2V 1C6

Date Printed: 11/20/2008

CUSTOMER

Pressure Vessel Engineering
120 Randall Drive, Suite 'B'
Waterloo, Ontario N2V 1C6

VESSEL LOCATION

Pressure Vessel Engineering
Vessel With Large Opening
Sample Vessel 5
,

Vessel designed per the ASME Boiler & Pressure Vessel Code,
Section VIII, Division 1,2007 Edition
with Advanced Pressure Vessel, Version: 10.0.2
Vessel is ASME Code Stamped

Job No: PVE-Sample5
Vessel Number: Sample Vessel 5

NAMEPLATE INFORMATION

Vessel MAWP:	200.00 PSI at 350 °F
MDMT:	-20 °F at 200.00 PSI
Serial Number(s):	________________________________
National Board Number(s):	________________________________
Year Built:	2007
Radiography:	NONE
Postweld Heat Treated:	NONE
Construction Type:	W

Notes

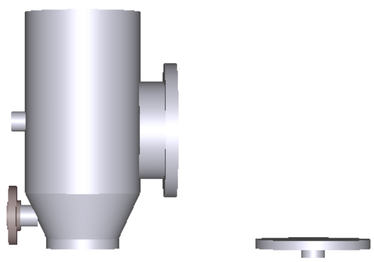
2" FERRULE CRN ASLOA 7213.51246089
8" FERRULE PROOF TESTED TO UG-101(m)

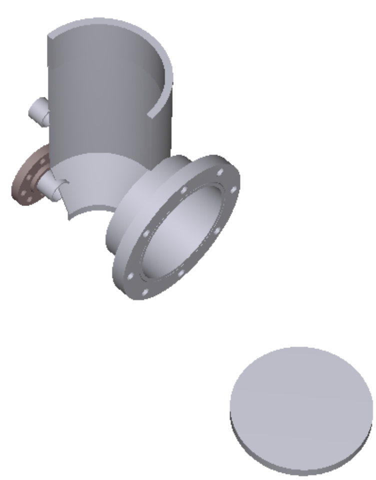
Signatures

P.Eng:____ Date: ____/____/____
Laurence Brundrett

Mechanical Technologist:____ Date: ____/____/____
Ben Vanderloo

Table of Contents





Cylindrical Shell Design Information

Design Pressure:	200.00	PSI	Design Temperature:	350	°F
Static Head:	1.00	PSI	Long. Joint Efficiency:	85	%
Shell Material:	SA-479 304, High		Factor B Chart:	HA-1	
			Material Stress (hot):	18600	PSI
Shell Length:	1.1020	in.	Material Stress (cold):	20000	PSI
			Compressive Stress:	9170	PSI
Corrosion Allowance:	0.0000	in.	Actual Circumferential Stress:	12078	PSI
External Corrosion Allowance:	0.0000	in.	Actual Longitudinal Stress:	7189	PSI
Outside Diameter (new):	8.0300	in.			
Outside Diameter (corroded):	8.0300	in.	Specific Gravity:	1.00	
Shell Surface Area:	0.19	Sq. Ft.	Weight of Fluid:	1.94	lb.
Shell Estimated Volume:	0.23	Gal.	Total Flooded Shell Weight:	2.56	lb.
Circ. Joint Efficiency:	70	%	Shell Weight:	0.62	lb.

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Design Thickness Calculations

Longitudinal Stress Calculations per Paragraph UG-27(c)(2)

$$\mathbf{t} = \frac{PR}{2SE + 0.4P} = \frac{201.00 * 3.9370}{2 * 18600 * 0.70 + 0.4 * 201.00}$$

= Greater Of (0.0303$_{\text{(Calculated)}}$, 0.0625$_{\text{(Minimum Allowed)}}$) + 0.0000 $_{\text{(corrosion)}}$ + 0.0000 $_{\text{(ext. corrosion)}}$ = minimum of **0.0625** in.

Circumferential Stress Calculations per Appendix 1-1(a)(1)

$$\mathbf{t} = \frac{PR_o}{SE + 0.4P} = \frac{201.00 * 4.0150}{18600 * 0.85 + 0.4 * 201.00}$$

= Greater of (0.0508$_{\text{(Calculated)}}$, 0.0625$_{\text{(Minimum Allowed)}}$) + 0.0000 $_{\text{(corrosion)}}$ + 0.0000 $_{\text{(ext. corrosion)}}$ = minimum of **0.0625** in.

Extreme Fiber Elongation Calculation per Paragraph UHA-44

$$\textbf{Elongation} = \frac{50t}{Rf} = \frac{50 * 0.0780}{3.9760}$$

= elongation of **0.98** %

Nominal Shell Thickness Selected = **0.0780** in.

Cylindrical Shell Design Information

Design Pressure:	200.00 PSI		Design Temperature:	350	°F
Static Head:	1.00 PSI		Long. Joint Efficiency:	70	%
Shell Material:	SA-240 304, High		Factor B Chart:	HA-1	
			Material Stress (hot):	18600	PSI
Shell Length:	15.5000 in.		Material Stress (cold):	20000	PSI
			Compressive Stress:	10102	PSI
Corrosion Allowance:	0.0000 in.		Actual Circumferential Stress:	9622	PSI
External Corrosion Allowance:	0.0000 in.		Actual Longitudinal Stress:	4667	PSI
Outside Diameter (new):	12.7500 in.				
Outside Diameter (corroded):	12.7500 in.		Specific Gravity:	1.00	
Shell Surface Area:	4.31 Sq. Ft.		Weight of Fluid:	67.42	lb.
Shell Estimated Volume:	8.07 Gal.		Total Flooded Shell Weight:	100.77	lb.
Circ. Joint Efficiency:	70 %		Shell Weight:	33.35	lb.

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Design Thickness Calculations

Longitudinal Stress Calculations per Paragraph UG-27(c)(2)

$$\mathbf{t} = \frac{PR}{2SE + 0.4P} = \frac{201.00 * 6.1870}{2 * 18600 * 0.70 + 0.4 * 201.00}$$

$= \text{Greater Of } (0.0476_{(\text{Calculated})}, 0.0625_{(\text{Minimum Allowed})}) + 0.0000_{\ (\text{corrosion})} + 0.0000_{\ (\text{ext. corrosion})}$ = minimum of **0.0625** in.

Circumferential Stress Calculations per Appendix 1-1(a)(1)

$$\mathbf{t} = \frac{PR_o}{SE + 0.4P} = \frac{201.00 * 6.3750}{18600 * 0.70 + 0.4 * 201.00} = 0.0979 + 0.0000_{\ (\text{corrosion})} + 0.0000_{\ (\text{ext. corrosion})}$$

= minimum of **0.0979** in.

Extreme Fiber Elongation Calculation per Paragraph UHA-44

$$\mathbf{Elongation} = \frac{50t}{Rf} = \frac{50 * 0.1880}{6.2810}$$

= elongation of **1.50** %

Nominal Shell Thickness Selected = **0.1880** in.

Pressure Vessel Engineering, Ltd.

thick shell

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 3

Vessel Number: Sample Vessel 5
Mark Number: S3

Date Printed: 11/20/2008

Cylindrical Shell Design Information

Design Pressure:	200.00 PSI		Design Temperature:	350	°F
Static Head:	1.00 PSI		Long. Joint Efficiency:	85	%
Shell Material:	SA-312 TP304 SMLS, High		Factor B Chart:	HA-1	
			Material Stress (hot):	18600	PSI
Shell Length:	4.5000 in.		Material Stress (cold):	20000	PSI
			Compressive Stress:	11376	PSI
Corrosion Allowance:	0.0000 in.		Actual Circumferential Stress:	2413	PSI
External Corrosion Allowance:	0.0000 in.		Actual Longitudinal Stress:	1301	PSI
Outside Diameter (new):	12.7500 in.				
Outside Diameter (corroded):	12.7500 in.		Specific Gravity:	1.00	
Shell Surface Area:	1.25 Sq. Ft.		Weight of Fluid:	16.54	lb.
Shell Estimated Volume:	1.98 Gal.		Total Flooded Shell Weight:	50.52	lb.
Circ. Joint Efficiency:	70 %		Shell Weight:	33.98	lb.

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -20 °F
Material is exempt from impact testing per UHA-51(d)

Design Thickness Calculations

Longitudinal Stress Calculations per Paragraph UG-27(c)(2)

$$\mathbf{t} = \frac{PR}{2SE + 0.4P} = \frac{201.00 * 5.6880}{2 * 18600 * 0.70 + 0.4 * 201.00}$$

= Greater Of (0.0438$_{\text{(Calculated)}}$, 0.0625$_{\text{(Minimum Allowed)}}$) + 0.0000 $_{\text{(corrosion)}}$ + 0.0000 $_{\text{(ext. corrosion)}}$ + 0.0859 $_{\text{(12 1/2\% for pipe)}}$

= minimum of **0.1484** in.

Circumferential Stress Calculations per Appendix 1-1(a)(1)

$$\mathbf{t} = \frac{PR_o}{SE + 0.4P} = \frac{201.00 * 6.3750}{18600 * 0.85 + 0.4 * 201.00}$$ = 0.0807 + 0.0000 $_{\text{(corrosion)}}$ + 0.0000 $_{\text{(ext. corrosion)}}$ + 0.0859 $_{\text{(12 1/2\% for pipe)}}$

= minimum of **0.1666** in.

Pipe Selected: Size = **12 in.**, Schedule = **80**, Diameter = **12.7500** in., Wall = **0.6870** in.

Pressure Vessel Engineering, Ltd.

Conical Reducer 1

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 1

Vessel Number: Sample Vessel 5
Mark Number: R1

Date Printed: 11/20/2008

Conical Reducer Design Information

Design Pressure:	200.00 PSI		Design Temperature:	350	°F
Static Head:	1.00 PSI		Joint Efficiency:	70	%
Conical Reducer Material:	SA-240 304, High		Factor B Chart:	HA-1	
			Material Stress (hot):	18600	PSI
Corrosion Allowance:	0.0000 in.		Material Stress (cold):	20000	PSI
External Corrosion Allowance:	0.0000 in.		Actual Conical Reducer Stress:	10681	PSI
Small End of Cone Located at:	Bottom		Outside Ds :	8.0300	in.
Outside Diameter :	12.7500 in.		Cone Height (h) :	4.9994	in.
angle a (α):	25.27 °				
Conical Reducer Surface Area:	1.25 Sq. Ft.		Specific Gravity:	1.00	
Conical Reducer Estimated Volume:	1.72 Gal.		Weight of Fluid:	14.37	lb.
Conical Reducer Weight:	9.62 lb.		Total Flooded Conical Reducer Weight:	23.99	lb.

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -20 °F
Other Exemption

Design Thickness Calculations

Design Thickness Calculations per Appendix 1-4(e)

$$\mathbf{t} = \frac{PD_o}{2\cos\alpha(SE + 0.4P)} = \frac{201.00 * 12.7500}{2 * 0.9043 * (18600 * 0.70 + 0.4 * 201.00)} = 0.1082 + 0.0000_{\text{(corrosion)}} + 0.0000_{\text{(ext. corrosion)}}$$

= minimum of **0.1082** in.

Extreme Fiber Elongation Calculation per Paragraph UHA-44

$$\textbf{elongation} = \frac{50t}{R_f} = \frac{50 * 0.1875}{3.9213}$$

= elongation of **2.39** %

Nominal Conical Reducer Thickness Selected = **0.1875** in.

Pressure Vessel Engineering, Ltd.
10" Inlet Nozzle C

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 1
ID Number: N-C

Vessel Number: Sample Vessel 5
Mark Number: N-C

Date Printed: 11/20/2008

Nozzle Design Information

Design Pressure:	200.00	PSI	Design Temperature:	350	°F
Static Head:	1.00	PSI	Nozzle Efficiency (E):	85	%
Nozzle Material:	SA-312 TP304 SMLS, High		Joint Efficiency (E_1):	1.00	
			Factor B Chart:	HA-1	
External Projection:	4.0000	in.	Allowable Stress at Design Temperature (S_n):	18600	PSI
Internal Projection:	0.0000	in.	Allowable Stress at Ambient Temperature:	20000	PSI
Inside Corrosion Allowance:	0.0000	in.	Correction Factor (F):	1.00	
External Corrosion Allowance:	0.0000	in.	Nozzle Path:	None	
Nozzle Pipe Size:	10		Nozzle Pipe Schedule:	40	
Nozzle ID (new):	10.0200	in.	Nozzle Wall Thickness(new):	0.3650	in.
Nozzle ID (corroded):	10.0200	in.	Nozzle Wall Thickness(corroded):	0.3650	in.
Outer "h" Limit:	0.4700	in.	Upper Weld Leg Size(Weld 41):	0.2500	in.
Internal "h" Limit:	0.4700	in.	Internal Weld Leg Size(Weld 43):	0.0000	in.
OD, Limit of Reinforcement:	20.0400	in.	Appendix 1-7 OD, Limit of Reinforcement(d_{LDR}):	15.0300	in.
Outside Groove Weld Depth:	0.1880	in.			

Minimum Design Metal Temperature

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Host Component: Shell 2 - Shell 2 - thin shell

Material:	SA-240 304, High		Shell wall thickness(new):	0.1880	in.
Material Stress(S_v):	18600	PSI	Shell wall thickness(corroded):	0.1880	in.

Nozzle Detail Information

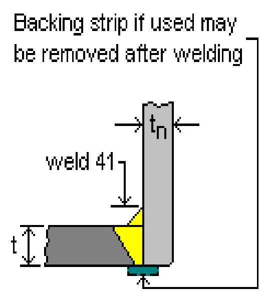


Fig. UW-16.1 (c)

Upper Weld Leg Size(Weld 41): 0.2500 in.

Nozzle Wall Thickness(t_n): 0.3650 in.

Outside Groove Weld Depth: 0.1880 in.

Nozzle passes through the vessel, attached by a groove weld.
Pipe Size: 10 Schedule: 40
Nozzle is adequate for UG-45 requirements.
Opening is adequately reinforced for Internal Pressure.
Large opening meets Appendix 1-7 stress calculation requirements
Weld Strength Paths are adequate.

Required Shell Thickness per Paragraph UG-37(a)

$$\mathbf{tr} = \frac{PRo}{SE + 0.4P} = \frac{201.00 * 6.3750}{18600 * 1 + 0.4 * 201.00} = \mathbf{0.0686} \text{ in.}$$

Nozzle Required Thickness Calculations

Required Nozzle Thickness for Internal Pressure per Paragraph UG-37(a)

$$\mathbf{trn} = \frac{PRn}{SE - 0.6P} = \frac{201.00 * 5.0100}{18600 * 1 - 0.6 * 201.00} = \mathbf{0.0545} \text{ in.}$$

Strength Reduction Factors

$$\mathbf{fr1} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr2} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr3} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

UG-45 Thickness Calculations

Nozzle Thickness for Pressure Loading (plus corrosion) per Paragraph UG-45(a)

$$\mathbf{t} = \frac{PRn}{SE - 0.6P} + Ca + ext.\ Ca = \frac{201.00 * 5.0100}{18600 * 0.85 - 0.6 * 201.00} + 0.0000 + 0.0000 = \mathbf{0.0642} \text{ in.}$$

Nozzle Thickness for Internal Pressure (plus corrosion) per Paragraph UG-45(b)(1)

$$\mathbf{t} = \frac{PRo}{SE + 0.4P} + Ca + ext.\ Ca = \frac{201.00 * 6.3750}{18600 * 1 + 0.4 * 201.00} + 0.0000 + 0.0000 = \mathbf{0.0686} \text{ in.}$$

Minimum Thickness of Standard Wall Pipe (plus corrosion) per Paragraph UG-45(b)(4)

t = minimum thickness of standard wall pipe + Ca + ext. Ca = **0.3194** in.

Nozzle Minimum Thickness per Paragraph UG-45(b)

t = Smallest of UG-45(b)(1) or UG-45(b)(4) = **0.0686** in.

Wall thickness = tn * 0.875(pipe) = **0.3194** is greater than or equal to UG-45 value of **0.0686**

Nozzle Reinforcement Calculations

Area Required for Internal Pressure

A = d tr F + 2 tn tr F (1 - fr1) = (10.0200 * 0.0686 * 1.00) + (2 * 0.3650 * 0.0686 * 1.00 * (1 - 1.0000)) = **0.6874** sq. in.

Area Available - Internal Pressure

A1 Formula 1 = d(E1 t - F tr) - 2tn(E1 t - F tr)(1 - fr1) =
10.0200 * (1.00 * 0.1880 - 1.00 * 0.0686) - 2 * 0.3650 * (1.00 * 0.1880 - 1.00 * 0.0686) * (1 - 1.0000) = *1.1964* sq. in.

A1 Formula 2 = 2(t + tn)(E1 t - F tr) - 2tn(E1 t - F tr)(1 - fr1) =
2 * (0.1880 + 0.3650)(1.00 * 0.1880 - 1.00 * 0.0686) - 2 * 0.3650 * (1.00 * 0.1880 - 1.00 * 0.0686) * (1 - 1.0000)
= *0.1321* sq. in.

A1 = Larger value of *A1 Formula 1* and *A1 Formula 2* = **1.1964** sq. in.

A2 Formula 1 = 5(tn - trn) fr2 t = 5(0.3650 - 0.0545) * 1.0000 * 0.1880 = *0.2919* sq. in.
A2 Formula 2 = 5(tn - trn) fr2 tn = 5(0.3650 - 0.0545) * 1.0000 * 0.3650 = *0.5667* sq. in.
A2 = Smaller value of *A2 Formula 1* and *A2 Formula 2* = **0.2919** sq. in.

A3 = Smaller value of the following :
$5 * t * t_i * f_{r2}$ = 5 * 0.1880 * 0.3650 * 1.0000 = 0.3431 sq. in.
$5 * t_i * t_i * f_{r2}$ = 5 * 0.3650 * 0.3650 * 1.0000 = 0.6661 sq. in.
$2 * h * t_i * f_{r2}$ = 2 * 0.0000 * 0.3650 * 1.0000= 0.0000 sq. in.
= **0.0000** sq. in.

A41 = (leg)² * fr2 = (0.2500)² * 1.0000 = **0.0625** sq. in.

A43 = (leg)² * fr2 = 0 * 1.0000 = **0.0000** sq. in.

Area Available (Internal Pressure) = A1 + A2 + A3 + A41 + A43 = 1.5508 sq. in., which is **greater** than A (0.6874)

Appendix 1-7(a) (Large Opening in Shell) Calculations

Area Required for Internal Pressure

A = 2/3 * (d tr F + 2 tn tr F (1 - fr1)) = 2/3 * ((10.0200 * 0.0686 * 1.00) + (2 * 0.3650 * 0.0686 * 1.00 * (1 - 1.0000))) = **0.4583** sq. in.

Area Available - Internal Pressure

A1 = (d_{LDR} - d)(E1 t - F tr) - 2tn(E1 t - F tr)(1 - fr1) =
(15.0300 - 10.0200) * (1.00 * 0.1880 - 1.00 * 0.0686) - 2 * 0.3650 * (1.00 * 0.1880 - 1.00 * 0.0686) * (1 - 1.0000) = **0.5982** sq. in.

A2 Formula 1 = 5(tn - trn) fr2 t = 5(0.3650 - 0.0545) * 1.0000 * 0.1880 = *0.2919* sq. in.
A2 Formula 2 = 5(tn - trn) fr2 tn = 5(0.3650 - 0.0545) * 1.0000 * 0.3650 = *0.5667* sq. in.
A2 = Smaller value of *A2 Formula 1* and *A2 Formula 2* = **0.2919** sq. in.

A3 = Smaller value of the following :
5 * t * t_i * f_{r2} = 5 * 0.1880 * 0.3650 * 1.0000 = 0.3431 sq. in.
5 * t_i * t_i * f_{r2} = 5 * 0.3650 * 0.3650 * 1.0000 = 0.6661 sq. in.
2 * h * t_i * f_{r2} = 2 * 0.0000 * 0.3650 * 1.0000= 0.0000 sq. in.
= **0.0000** sq. in.

A41 = (leg)² * fr2 = (0.2500)² * 1.0000 = **0.0625** sq. in.

A43 = (leg)² * fr2 = 0 * 1.0000 = **0.0000** sq. in.

Area Available (Internal Pressure) = A1 + A2 + A3 + A41 + A43 = 0.9526 sq. in., which is **greater** than A (0.4583)

Nozzle Weld Strength Calculations

Attachment Weld Strength per Paragraph UW-16

Weld 41 tmin = smaller of 0.75, t, or tn = smaller of 0.75, 0.1880, or 0.3650 = **0.1880** in.

Weld 41 Leg min. = $\frac{\text{(smaller of 0.25 or (tmin * 0.7)) + ext. CA}}{0.7} = \frac{0.1316}{0.7}$ = **0.1880** in.

Weld 41, actual weld leg = **0.2500** in.

Unit Stresses per Paragraphs UG-45(c) and UW-15

Nozzle wall in shear = 0.70 * Sn = 0.70 * 18600 = **13020** PSI
Upper fillet, Weld 41, in shear = 0.49 * Material Stress = 0.49 * 18600 = **9114** PSI
Vessel groove weld, in tension = 0.74 * Material Stress = 0.74 * 18600 = **13764** PSI

Strength of Connection Elements

Nozzle wall in shear = ½ * π * mean nozzle diameter * tn * Nozzle wall in shear unit stress =
½ * π * 10.3850 * 0.3650 * 13020 = **77500** lb.
Upper fillet in shear = ½ * π * Nozzle OD * weld leg * upper fillet in shear unit stress = ½ * π * 10.7500 * 0.2500 * 9114 = **38500** lb.
Groove Weld in Tension = ½ * π * Nozzle OD * groove depth * groove weld tension unit stress =
½ * π * 10.7500 * 0.1880 * 13764 = **43700** lb.

Load to be carried by welds, per UG-41(b)(1) and Fig. UG-41.1 sketch (a)

W = [A - A1 + 2 tn fr1(E1t - Ftr)] Sv = [0.6874 - 1.1964 + 2 * 0.3650 * 1.0000 * (1.00 * 0.1880 - 1.0000 * 0.0686)] * 18600 = **-7846** lb.
W1-1 = (A2 + A5 + A41 + A42) * Sv = (0.2919 + 0.0000 + 0.0625 + 0.0000) * 18600 = **6590** lb.
W2-2 = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv = (0.2919 + 0.0000 + 0.0625 + 0.0000 + 2 * 0.3650 * 0.1880 * 1.0000) * 18600 = **9140** lb.
W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) * Sv =
(0.2919 + 0.0000 + 0.0000 + 0.0625 + 0.0000 + 0.0000 + 2 * 0.3650 * 0.1880 * 1.0000) * 18600 = **9140** lb.

Check Strength Paths

Path 1-1 = Upper fillet in shear + Nozzle wall in shear = 38500 + 77500 = **116000** lb.
Path 2-2 = Upper fillet in shear + Groove weld in tension + Inner fillet in shear =
38500 + 43700 + 0 = **82200** lb.
Path 3-3 = Upper fillet in shear + Inner fillet in shear + Groove weld in tension = 38500 + 0 + 43700 = **82200** lb.

Pressure Vessel Engineering, Ltd.
2" Inlet Nozzle A

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 2
ID Number: N-A

Vessel Number: Sample Vessel 5
Mark Number: N-A

Date Printed: 11/20/2008

Nozzle Design Information

Design Pressure:	200.00	PSI	Design Temperature:	350	°F
Static Head:	1.00	PSI	Nozzle Efficiency (E):	85	%
Nozzle Material:	SA-312 TP304 SMLS, High		Joint Efficiency (E_1):	1.00	
			Factor B Chart:	HA-1	
External Projection:	3.0000	in.	Allowable Stress at Design Temperature (S_n):	18600	PSI
Internal Projection:	0.0000	in.	Allowable Stress at Ambient Temperature:	20000	PSI
Inside Corrosion Allowance:	0.0000	in.	Correction Factor (F):	1.00	
External Corrosion Allowance:	0.0000	in.	Nozzle Path:	None	
Nozzle Pipe Size:	2		Nozzle Pipe Schedule:	40	
Nozzle ID (new):	2.0670	in.	Nozzle Wall Thickness(new):	0.1540	in.
Nozzle ID (corroded):	2.0670	in.	Nozzle Wall Thickness(corroded):	0.1540	in.
Developed Opening:	2.1920	in.	Tangential Dimension L:	7.0000	in.
Outer "h" Limit:	0.3850	in.	Upper Weld Leg Size(Weld 41):	0.1880	in.
Internal "h" Limit:	0.3850	in.	Internal Weld Leg Size(Weld 43):	0.0000	in.
OD, Limit of Reinforcement:	4.3840	in.	Outside Groove Weld Depth:	0.1875	in.

Minimum Design Metal Temperature

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Host Component: Cone 1 - Conical Reducer 1

Material:	SA-240 304, High		Conical Reducer wall thickness(new):	0.1875	in.
Material Stress(S_v):	18600	PS	Ionical Reducer wall thickness - thin out (corroded):	0.1875	in.
Distance from small end of cone:	2.5000	in.	Diameter of Cone at Nozzle:	10.3903	in.

Nozzle Detail Information

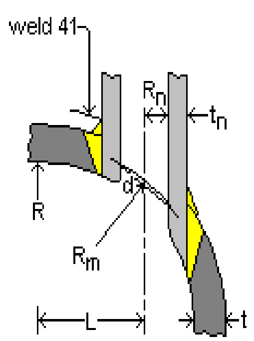


Fig. UW-16.1 (c)

Upper Weld Leg Size(Weld 41): 0.1880 in.

Nozzle Wall Thickness(t_n): 0.1540 in.

Outside Groove Weld Depth: 0.1875 in.

tangential to the vessel wall, attached by a groove weld.
Pipe Size: 2 Schedule: 40
Nozzle is adequate for UG-45 requirements.
Opening is adequately reinforced for Internal Pressure.
Reinforcement calculations are not required per UG-36(c)(3)(a)See Uw-14 for exceptions.
Weld Strength Paths are adequate.

Required Conical Reducer Thickness per Paragraph UG-37(a)

$$\mathbf{tr} = \frac{PDo}{(2 * \cos\alpha(SE + 0.4P))} = \frac{201.00 * 10.3903}{(2 * 0.9043 * (18600 * 1 + 0.4 * 201.00))} = \mathbf{0.0618} \text{ in.}$$

Nozzle Required Thickness Calculations

Required Nozzle Thickness for Internal Pressure per Paragraph UG-37(a)

$$\mathbf{trn} = \frac{PRn}{SE - 0.6P} = \frac{201.00 * 1.0335}{18600 * 1 - 0.6 * 201.00} = \mathbf{0.0112} \text{ in.}$$

Strength Reduction Factors

$$\mathbf{fr1} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr2} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr3} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

UG-45 Thickness Calculations

Nozzle Thickness for Pressure Loading (plus corrosion) per Paragraph UG-45(a)

$$\mathbf{t} = \frac{PRn}{SE - 0.6P} + Ca + ext.\ Ca = \frac{201.00 * 1.0335}{18600 * 0.85 - 0.6 * 201.00} + 0.0000 + 0.0000 = \mathbf{0.0132} \text{ in.}$$

Nozzle Thickness for Internal Pressure (plus corrosion) per Paragraph UG-45(b)(1)

$$\mathbf{t} = \frac{PDo}{(2 * \cos\alpha(SE + 0.4P))} + Ca + ext.\ Ca = \frac{201.00 * 10.3903}{(2 * 0.9043 * (18600 * 1 + 0.4 * 201.00))} + 0.0000 + 0.0000$$

$$= \text{Greater Of } (0.0618_{(Calculated)}, 0.0625_{(Minimum\ Allowed)}) + 0.0000_{(corrosion)} + 0.0000_{(ext.\ corrosion)} = \mathbf{0.0625} \text{ in.}$$

Minimum Thickness of Standard Wall Pipe (plus corrosion) per Paragraph UG-45(b)(4)

t = minimum thickness of standard wall pipe + Ca + ext. Ca = **0.1347** in.

Nozzle Minimum Thickness per Paragraph UG-45(b)

t = Smallest of UG-45(b)(1) or UG-45(b)(4) = **0.0625** in.

Wall thickness = tn * 0.875(pipe) = **0.1347** is greater than or equal to UG-45 value of **0.0625**

Nozzle Weld Strength Calculations

Attachment Weld Strength per Paragraph UW-16

Weld 41 tmin = smaller of 0.75, t, or tn = smaller of 0.75, 0.1875, or 0.1540 = **0.1540** in.

Weld 41 Leg min. = $\frac{\text{(smaller of 0.25 or (tmin * 0.7)) + ext. CA}}{0.7} = \frac{0.1078}{0.7}$ = **0.1540** in.

Weld 41, actual weld leg = **0.1880** in.

Unit Stresses per Paragraphs UG-45(c) and UW-15

Nozzle wall in shear = 0.70 * Sn = 0.70 * 18600 = **13020** PSI
Upper fillet, Weld 41, in shear = 0.49 * Material Stress = 0.49 * 18600 = **9114** PSI
Vessel groove weld, in tension = 0.74 * Material Stress = 0.74 * 18600 = **13764** PSI

Strength of Connection Elements

Nozzle wall in shear = ½ * π * mean nozzle diameter * tn * Nozzle wall in shear unit stress =
½ * π * 2.2210 * 0.1540 * 13020 = **6990** lb.
Upper fillet in shear = ½ * π * Nozzle OD * weld leg * upper fillet in shear unit stress = ½ * π * 2.3750 * 0.1880 * 9114 = **6390** lb.
Groove Weld in Tension = ½ * π * Nozzle OD * groove depth * groove weld tension unit stress =
½ * π * 2.3750 * 0.1875 * 13764 = **9620** lb.

Load to be carried by welds, per UG-41(b)(1) and Fig. UG-41.1 sketch (a)

W = [A - A1 + 2 tn fr1(E1t - Ftr)] Sv = [0.1355 - 0.2755 + 2 * 0.1540 * 1.0000 * (1.00 * 0.1875 - 1.0000 * 0.0618)] * 18600 = **-1883** lb.
W1-1 = (A2 + A5 + A41 + A42) * Sv = (0.1100 + 0.0000 + 0.0353 + 0.0000) * 18600 = **2700** lb.
W2-2 = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv = (0.1100 + 0.0000 + 0.0353 + 0.0000 + 2 * 0.1540 * 0.1875 * 1.0000) * 18600 = **3780** lb.
W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) * Sv =
(0.1100 + 0.0000 + 0.0000 + 0.0353 + 0.0000 + 0.0000 + 2 * 0.1540 * 0.1875 * 1.0000) * 18600 = **3780** lb.

Check Strength Paths

Path 1-1 = Upper fillet in shear + Nozzle wall in shear = 6390 + 6990 = **13380** lb.
Path 2-2 = Upper fillet in shear + Groove weld in tension + Inner fillet in shear =
6390 + 9620 + 0 = **16010** lb.
Path 3-3 = Upper fillet in shear + Inner fillet in shear + Groove weld in tension = 6390 + 0 + 9620 = **16010** lb.

Pressure Vessel Engineering, Ltd.
2"3000# Cplg N-E

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 3
ID Number: N-E

Vessel Number: Sample Vessel 5
Mark Number: N-E

Date Printed: 11/20/2008

Nozzle Design Information

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	0.00 PSI	Nozzle Efficiency (E):	70 %
Nozzle Material:	SA-182 F304 <=5", High	Joint Efficiency (E_1):	1.00
		Factor B Chart:	HA-1
External Projection:	1.0000 in.	Allowable Stress at Design Temperature (S_n):	18600 PSI
		Allowable Stress at Ambient Temperature:	20000 PSI
Inside Corrosion Allowance:	0.0000 in.	Correction Factor (F):	1.00
External Corrosion Allowance:	0.0000 in.	Nozzle Path:	None
Nozzle ID (new):	2.0970 in.	Nozzle Wall Thickness(new):	0.2730 in.
Nozzle ID (corroded):	2.0970 in.	Nozzle Wall Thickness(corroded):	0.2730 in.
Outer "h" Limit:	0.6825 in.	Upper Weld Leg Size(Weld 41):	0.3130 in.
OD, Limit of Reinforcement:	4.6430 in.	Outside Groove Weld Depth:	0.2730 in.

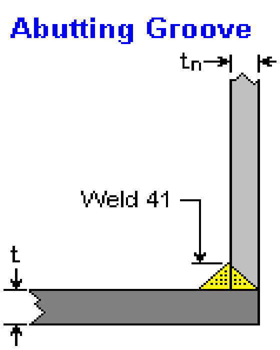
Minimum Design Metal Temperature

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Host Component: Flange 2 - Cover Flange

Material:	SA-240 304, Low	Host Flange wall thickness(new):	1.0000 in.
Material Stress(S_v):	14400 PSI	Host Flange wall thickness(corroded):	1.0000 in.

Nozzle Detail Information



Upper Weld Leg Size(Weld 41): 0.3130 in.

Nozzle Wall Thickness(t_n): 0.2730 in.

Outside Groove Weld Depth: 0.2730 in.

Nozzle abuts the vessel, attached by a groove weld.
Nozzle is adequate for UG-45 requirements.
Opening is adequately reinforced for Internal Pressure.
Reinforcement calculations are not required per UG-36(c)(3)(a)See Uw-14 for exceptions.
Weld Strength Paths are adequate.

Required Host Flange Thickness per Paragraph UG-39(b)(1)

$$\textbf{tr} = G * \sqrt{\frac{CP}{SE} + \frac{1.9\,W_{m1}\,h_G}{SE\,G^3}} = 12.4800 * \sqrt{\frac{0.3000 * 200.00}{14400 * 1} + \frac{1.9 * 25923 * 1.1350}{14400 * 1 * 12.4800^3}} = \textbf{0.9798} \text{ in.}$$

Nozzle Required Thickness Calculations

Required Nozzle Thickness for Internal Pressure per Paragraph UG-37(a)

$$\textbf{trn} = \frac{PRn}{SE - 0.6P} = \frac{200.00 * 1.0485}{18600 * 1 - 0.6 * 200.00} = \textbf{0.0113} \text{ in.}$$

Strength Reduction Factors

$$\textbf{fr2} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{14400}, 1.0000\right) = 1.0000 \qquad \textbf{fr3} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{14400}, 1.0000\right) = 1.0000$$

UG-45 Thickness Calculations

Nozzle Thickness for Pressure Loading (plus corrosion) per Paragraph UG-45(a)

$$\textbf{t} = \frac{PRn}{SE - 0.6P} + Ca + \text{ext. } Ca = \frac{200.00 * 1.0485}{18600 * 0.70 - 0.6 * 200.00} + 0.0000 + 0.0000 = \textbf{0.0163} \text{ in.}$$

Nozzle Thickness for Internal Pressure (plus corrosion) per Paragraph UG-45(b)(1)

$$\textbf{t} = G * \sqrt{\frac{CP}{SE} + \frac{1.9\,W_{m1}\,h_G}{SE\,G^3}} + Ca + \text{ext. } Ca = 12.4800 * \sqrt{\frac{0.3000 * 200.00}{14400 * 1} + \frac{1.9 * 25923 * 1.1350}{14400 * 1 * 12.4800^3}} + 0.0000 + 0.0000$$

= = **0.9798** in.

Minimum Thickness of Standard Wall Pipe (plus corrosion) per Paragraph UG-45(b)(4)

t = minimum thickness of standard wall pipe + Ca + ext. Ca = **0.1776** in.

Nozzle Minimum Thickness per Paragraph UG-45(b)

t = Smallest of UG-45(b)(1) or UG-45(b)(4) = **0.1776** in.

Wall thickness = tn = **0.2730** is greater than or equal to UG-45 value of **0.1776**

Nozzle Weld Strength Calculations

Attachment Weld Strength per Paragraph UW-16

Weld 41 tmin = smaller of 0.75, t, or tn = smaller of 0.75, 1.0000, or 0.2730 = **0.2730** in.

$$\text{Weld 41 Leg min.} = \frac{(\text{smaller of } 0.25 \text{ or } (\text{tmin} * 0.7)) + \text{ext. CA}}{0.7} = \frac{0.1911}{0.7} = \mathbf{0.2730} \text{ in.}$$

Weld 41, actual weld leg = **0.3130** in.

Unit Stresses per Paragraphs UG-45(c) and UW-15

Groove weld in shear = 0.60 * Groove Material Stress = 0.60 * 14400 = **8640** PSI
Upper fillet, Weld 41, in shear = 0.49 * Material Stress = 0.49 * 14400 = **7056** PSI

Strength of Connection Elements

Groove weld in shear = ½ * π * mean nozzle diameter * tn * Groove weld in shear unit stress =
½ * π * 2.3700 * 0.2730 * 8640 = **8780** lb.
Upper fillet in shear = ½ * π * Nozzle OD * weld leg * upper fillet in shear unit stress = ½ * π * 2.6430 * 0.3130 * 7056 = **9160** lb.

Load to be carried by welds, per UG-41(b)(1) and Fig. UG-41.1 sketch (b)

W = (A - A1) Sv = (1.0273 - 0.0514) * 14400 = **14100** lb.
W1-1 = (A2 + A5 + A41 + A42) * Sv = (0.3572 + 0.0000 + 0.0980 + 0.0000) * 14400 = **6550** lb.
W2-2 = (A2 + A41) Sv = (0.3572 + 0.0980) * 14400 = **6550** lb.

Check Strength Paths

Path 1-1 = Upper fillet in shear + Groove weld in shear = 9160 + 8780 = **17940** lb.
Path 2-2 = Upper fillet in shear + Groove weld in shear =
9160 + 8780 = **17940** lb.

Pressure Vessel Engineering, Ltd.

2" Heavy Ferrule Nozzle B

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 4
ID Number: N-B

Vessel Number: Sample Vessel 5
Mark Number: N-B

Date Printed: 11/20/2008

Nozzle Design Information

Design Pressure:	200.00	PSI	Design Temperature:	350	°F
Static Head:	0.00	PSI	Nozzle Efficiency (E):	70	%
Nozzle Material:	SA-479 304, High		Joint Efficiency (E_1):	1.00	
			Factor B Chart:	HA-1	
External Projection:	1.5000	in.	Allowable Stress at Design Temperature (S_n):	18600	PSI
Internal Projection:	0.0000	in.	Allowable Stress at Ambient Temperature:	20000	PSI
Inside Corrosion Allowance:	0.0000	in.	Correction Factor (F):	1.00	
External Corrosion Allowance:	0.0000	in.	Nozzle Path:	None	
Nozzle ID (new):	1.8700	in.	Nozzle Wall Thickness(new):	0.1610	in.
Nozzle ID (corroded):	1.8700	in.	Nozzle Wall Thickness(corroded):	0.1610	in.
Outer "h" Limit:	0.4025	in.	Upper Weld Leg Size(Weld 41):	0.1880	in.
Internal "h" Limit:	0.4025	in.	Internal Weld Leg Size(Weld 43):	0.0000	in.
OD, Limit of Reinforcement:	3.7400	in.	Outside Groove Weld Depth:	0.1880	in.

Minimum Design Metal Temperature

Minimum Design Metal Temperature: -20 °F
Material exempt from impact testing per UCS-66(b), stress ratio <= 0.35

Host Component: Shell 2 - Shell 2 - thin shell

Material:	SA-240 304, High		Shell wall thickness(new):	0.1880	in.
Material Stress(S_v):	18600	PSI	Shell wall thickness(corroded):	0.1880	in.

Nozzle Detail Information

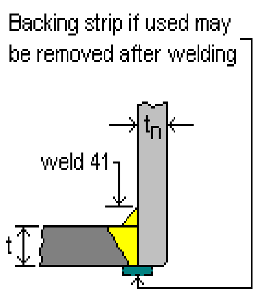


Fig. UW-16.1 (c)

Upper Weld Leg Size(Weld 41): 0.1880 in.

Nozzle Wall Thickness(t_n): 0.1610 in.

Outside Groove Weld Depth: 0.1880 in.

Nozzle passes through the vessel, attached by a groove weld.
Nozzle is adequate for UG-45 requirements.
Opening is adequately reinforced for Internal Pressure.
Reinforcement calculations are not required per UG-36(c)(3)(a)See Uw-14 for exceptions.
Weld Strength Paths are adequate.

Required Shell Thickness per Paragraph UG-37(a)

$$\mathbf{tr} = \frac{PRo}{SE + 0.4P} = \frac{200.00 * 6.3750}{18600 * 1 + 0.4 * 200.00} = \mathbf{0.0683} \text{ in.}$$

Nozzle Required Thickness Calculations

Required Nozzle Thickness for Internal Pressure per Paragraph UG-37(a)

$$\mathbf{trn} = \frac{PRn}{SE - 0.6P} = \frac{200.00 * 0.9350}{18600 * 1 - 0.6 * 200.00} = \mathbf{0.0101} \text{ in.}$$

Strength Reduction Factors

$$\mathbf{fr1} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr2} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

$$\mathbf{fr3} = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{18600}{18600}, 1.0000\right) = 1.0000$$

UG-45 Thickness Calculations

Nozzle Thickness for Pressure Loading (plus corrosion) per Paragraph UG-45(a)

$$\mathbf{t} = \frac{PRn}{SE - 0.6P} + Ca + ext.\ Ca = \frac{200.00 * 0.9350}{18600 * 0.70 - 0.6 * 200.00} + 0.0000 + 0.0000 = \mathbf{0.0145} \text{ in.}$$

Nozzle Thickness for Internal Pressure (plus corrosion) per Paragraph UG-45(b)(1)

$$\mathbf{t} = \frac{PRo}{SE + 0.4P} + Ca + ext.\ Ca = \frac{200.00 * 6.3750}{18600 * 1 + 0.4 * 200.00} + 0.0000 + 0.0000 = \mathbf{0.0683} \text{ in.}$$

Minimum Thickness of Standard Wall Pipe (plus corrosion) per Paragraph UG-45(b)(4)

t = minimum thickness of standard wall pipe + Ca + ext. Ca = **0.1347** in.

Nozzle Minimum Thickness per Paragraph UG-45(b)

t = Smallest of UG-45(b)(1) or UG-45(b)(4) = **0.0683** in.

Wall thickness = tn = **0.1610** is greater than or equal to UG-45 value of **0.0683**

Nozzle Weld Strength Calculations

Attachment Weld Strength per Paragraph UW-16

Weld 41 tmin = smaller of 0.75, t, or tn = smaller of 0.75, 0.1880, or 0.1610 = **0.1610** in.

Weld 41 Leg min. = $\dfrac{\text{(smaller of 0.25 or (tmin * 0.7)) + ext. CA}}{0.7} = \dfrac{0.1127}{0.7}$ = **0.1610** in.

Weld 41, actual weld leg = **0.1880** in.

Unit Stresses per Paragraphs UG-45(c) and UW-15

Nozzle wall in shear = 0.70 * Sn = 0.70 * 18600 = **13020** PSI
Upper fillet, Weld 41, in shear = 0.49 * Material Stress = 0.49 * 18600 = **9114** PSI
Vessel groove weld, in tension = 0.74 * Material Stress = 0.74 * 18600 = **13764** PSI

Strength of Connection Elements

Nozzle wall in shear = ½ * π * mean nozzle diameter * tn * Nozzle wall in shear unit stress =
½ * π * 2.0310 * 0.1610 * 13020 = **6680** lb.
Upper fillet in shear = ½ * π * Nozzle OD * weld leg * upper fillet in shear unit stress = ½ * π * 2.1920 * 0.1880 * 9114 = **5900** lb.
Groove Weld in Tension = ½ * π * Nozzle OD * groove depth * groove weld tension unit stress =
½ * π * 2.1920 * 0.1880 * 13764 = **8910** lb.

Load to be carried by welds, per UG-41(b)(1) and Fig. UG-41.1 sketch (a)

W = [A - A1 + 2 tn fr1(E1t - Ftr)] Sv = [0.1277 - 0.2238 + 2 * 0.1610 * 1.0000 * (1.00 * 0.1880 - 1.0000 * 0.0683)] * 18600 = **-1070** lb.
W1-1 = (A2 + A5 + A41 + A42) * Sv = (0.1215 + 0.0000 + 0.0353 + 0.0000) * 18600 = **2920** lb.
W2-2 = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv = (0.1215 + 0.0000 + 0.0353 + 0.0000 + 2 * 0.1610 * 0.1880 * 1.0000) * 18600 = **4040** lb.
W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) * Sv =
(0.1215 + 0.0000 + 0.0000 + 0.0353 + 0.0000 + 0.0000 + 2 * 0.1610 * 0.1880 * 1.0000) * 18600 = **4040** lb.

Check Strength Paths

Path 1-1 = Upper fillet in shear + Nozzle wall in shear = 5900 + 6680 = **12580** lb.
Path 2-2 = Upper fillet in shear + Groove weld in tension + Inner fillet in shear =
5900 + 8910 + 0 = **14810** lb.
Path 3-3 = Upper fillet in shear + Inner fillet in shear + Groove weld in tension = 5900 + 0 + 8910 = **14810** lb.

Pressure Vessel Engineering, Ltd.
10" Inlet Flange

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 1

Vessel Number: Sample Vessel 5
Mark Number: F1

Date Printed: 11/20/2008

Loose Flange Design Information

Design Pressure:	200.00 PSI		Design Temperature:	350	°F
Static Head:	1.00 PSI		Corrosion Allowance:	0.0000	in.
Material:	SA-240 304H, High		Factor B Chart:	HA-1	
			Material Stress Hot(S_{fo}):	18600	PSI
Outside Diameter (A):	14.7500 in.		Material Stress Cold(S_{fa}):	20000	PSI
Bolt Circle (C):	13.0000 in.		Flange I.D. (B):	10.8750	in.
Flange Weight:	31.10 lb.		Flange MAWP (at design):	476.81	PSI

Minimum Design Metal Temperature

Material is exempt from impact testing per UHA-51(d)

Bolting Information

Material:	SA-193 Gr B8 <=3/4"	Material Stress Hot (S_b):	25000	PSI
Material Condition:	Class 2	Material Stress Cold (S_a):	25000	PSI
Bolt Size:	3/4"	Threads Per Inch:	10	
Nominal Bolt Diameter (a):	0.7500 in.	Number of Bolts:	8	
Bolt Hole Diameter:	0.8750 in.	Bolt Root Area:	0.3020	sq. in.

Gasket & Facing Information

Material:	Self energizing	Configuration:	Ring	
Type:		Seating Stress (y): 200	PSI	
O.D. Contact Face:	12.1250 in.	Gasket Width (N):	0.5000	in.
Factor m:	1.00			

Host Component: Nozzle 1 - 10" Inlet Nozzle C

Material:	SA-312 TP304 SMLS, High	Material Stress Hot (S_{no}):	18600	PSI
		Material Stress Cold (S_{na}):	20000	PSI
Inside Diameter:	10.0200 in.	Wall Thickness (t_n):	0.3650	in.

ASME Flange Calculations per Appendix 2

Gasket Seating Calculations(Table 2-5.2)

$\mathbf{b}_0 = \frac{N}{2} = \frac{0.5000}{2}$ = **0.2500** in.

Since b_0 <= 1/4 in., $\mathbf{b} = b_0$ = **0.2500** in.

G = O.D. contact face - N = 12.1250 - 0.5000 = **11.6250** in.

Bolting is Adequate for Flange Design

Nominal Thickness is Adequate for Seating Conditions

Nominal Thickness is Adequate for Operating Conditions

Flange Thickness is Adequate for Flange Design

Flange Rigidity is Adequate.

Load and Bolting Calculations - Internal Pressure

The absolute value of effective pressure "P" is used for calculations.

$\mathbf{P} = \frac{16\ M}{\pi\ G^3} + \frac{4\ F_A}{\pi\ G^2} + P_{int} + \text{Static Head} = \frac{16 * 0}{3.14159 * 11.6250^3} + \frac{4 * 0}{3.14159 * 11.6250^2} + 200.00 + 1.00$ = **201.00** PSI

Minimum $\mathbf{W}_{m2} = \pi bGy = 3.14159 * 0.2500 * 11.6250 * 200$ = **1826** lb.

$\mathbf{H} = \frac{\pi}{4} G^2 P = \frac{3.14159}{4} * 11.6250^2 * 201.00$ = **21334** lb.

$\mathbf{H}_p = 2b\pi GmP = 2 * 0.2500 * 3.14159 * 11.6250 * 1.00 * 201.00$ = **3670** lb.

Minimum $\mathbf{W}_{m1} = H + H_p = 21334 + 3670$ = **25004** lb.

$\mathbf{A}_{m1} = \frac{W_{m1}}{S_b} = \frac{41173}{25000}$ = **1.6469** sq. in.

$\mathbf{A}_{m2} = \frac{W_{m2}}{S_a} = \frac{1826}{25000}$ = **0.0730** sq. in.

$\mathbf{A}_m$ = Greater of A_{m1} or A_{m2} = greater of 1.6469 or 0.0730 = **1.6469** sq. in.

$\mathbf{A}_b$ = Number of Bolts * Bolt Root Area = 8 * 0.3020 = **2.4160** sq. in.

$\mathbf{W} = \frac{(A_m + A_b)S_a}{2} = \frac{(1.6469 + 2.4160) * 25000}{2}$ = **50786** lb.

Ab >= Am, Bolting is Adequate for Flange Design

Internal Pressure Moment Calculations - Operating Conditions

$\mathbf{H}_D = \frac{\pi}{4} B^2 P = \frac{3.1416}{4} * 10.8750^2 * 201.00$ = **18670** lb.

$\mathbf{H}_G = W_{m1} - H = 41173 - 21334$ = **19839** lb.

$\mathbf{H}_T = H - H_D = 21334 - 18670$ = **2664** lb.

$\mathbf{h}_D = \frac{C - B}{2} = \frac{13.0000 - 10.8750}{2}$ = **1.0625** in.

$\mathbf{h}_G = \frac{C - G}{2} = \frac{13.0000 - 11.6250}{2}$ = **0.6875** in.

$\mathbf{h}_T = \frac{h_D + h_G}{2} = \frac{1.0625 + 0.6875}{2}$ = **0.8750** in.

$\mathbf{M}_D = H_D h_D = 18670 * 1.0625$ = **19837** in.-lb.

$\mathbf{M}_G = H_G h''_G = 19839 * 0.6875$ = **13639** in.-lb.

$\mathbf{M}_T = H_T h_T = 2664 * 0.8750$ = **2331** in.-lb.

$\mathbf{M}_o = M_D + M_G + M_T = 19837 + 13639 + 2331$ = **35807** in.-lb.

Internal Pressure Moment Calculations - Gasket Seating

$\mathbf{M_S} = Wh_G = 50786 * 0.6875$ = **34915** in.-lb.

Shape Constants
Calculated from Figure 2-7.1

$$\mathbf{K} = \frac{A}{B} = \frac{14.7500}{10.8750}$$ = **1.3563**

$$\mathbf{Y} = \frac{1}{K-1}\left(0.66845 + 5.71690\frac{K^2 \log_{10}K}{K^2 - 1}\right) = \frac{1}{1.3563 - 1}\left(0.66845 + \left[5.7169 * \frac{1.3563^2 * \log_{10}1.3563}{1.3563^2 - 1}\right]\right)$$ = **6.5293**

Bolt Spacing Calculations

$\mathbf{C_f} = 1$, Correction factor not applied.

Internal Pressure Stress Calculations - Operating Conditions

$$\mathbf{S_T} = \frac{YC_fM_o}{Bt^2} = \frac{6.5293 * 1.0000 * 35807}{10.8750 * 1.3750^2}$$ = **11371** PSI

Since $S_T \le S_{fo}$, nominal thickness is **ADEQUATE** for operating conditions.

Internal Pressure Stress Calculations - Gasket Seating

$$\mathbf{S_T} = \frac{YC_fM_S}{Bt^2} = \frac{6.5293 * 1.0000 * 34915}{10.8750 * 1.3750^2}$$ = **11088** PSI

Since $S_T \le S_{fa}$, nominal thickness is **ADEQUATE** for seating conditions.

Internal Pressure Rigidity Index per Appendix 2-14 - Operating Conditions

$$\mathbf{J} = \frac{109.4\ M_o}{Et^3\ Ln(K)\ K_L} = \frac{109.4 * 35807}{26.7x10^6 * 1.3750^3 * Ln(1.3563) * 0.2}$$ = **0.93**

J <= 1, design meets Flange Rigidity requirements for Operating Conditions

Internal Pressure Rigidity Index per Appendix 2-14 - Seating Conditions

$$\mathbf{J} = \frac{109.4\ M_S}{Et^3\ Ln(K)\ K_L} = \frac{109.4 * 34915}{28.3x10^6 * 1.3750^3 * Ln(1.3563) * 0.2}$$ = **0.85**

J <= 1, design meets Flange Rigidity requirements for Seating Conditions

Internal Pressure Minimum Thickness = **1.3380** in.

Nominal Thickness Selected = **1.3750** in.

Pressure Vessel Engineering, Ltd.

Cover Flange

Customer: **Pressure Vessel Engineering**

Job No: PVE-Sample5 | Vessel Number: Sample Vessel 5

Number: 2 | Mark Number: F2

Date Printed: 11/20/2008

Blind Flange Design Information

Design Pressure:	200.00 PSI		Design Temperature:	350	°F
Static Head:	0.00 PSI		Corrosion Allowance:	0.0000	in.
Material:	SA-240 304, Low		Factor B Chart:	HA-1	
			Material Stress Hot(S_{fo}):	14400	PSI
Outside Diameter (A):	12.7500 in.		Material Stress Cold(S_{fa}):	20000	PSI
Bolt Circle (C):	14.7500 in.		End Diameter (ID):	11.5000	in.
Flange Weight:	37.03 lb.		Flange MAWP (at design):	201.31	PSI
Head Factor C:	0.3000				
Weld Efficiency:	100 %				

Minimum Design Metal Temperature

Material is exempt from impact testing per UHA-51(d)

Bolting Information

Material:	SA-193 Gr B8	Material Stress Hot (S_b):	14400	PSI
Material Condition:	Class 1	Material Stress Cold (S_a):	18800	PSI
Bolt Size:	3/4"	Threads Per Inch:	10	
Nominal Bolt Diameter (a):	0.7500 in.	Number of Bolts:	6	
		Bolt Root Area:	0.3020	sq. in.

Gasket & Facing Information

Material:	Asbestos with suitable binder	Configuration:	Ring	
Type:	1/16 in. thick	Seating Stress (y):	3700	PSI
O.D. Contact Face:	12.7500 in.	Gasket Width (N):	0.2700	in.
Factor m:	2.75			
Facing Sketch:	6	Seating Column:	Column II	
Facing Width (w):	0.2700 in.			

ASME Flange Calculations per Appendix 2

Gasket Seating Calculations(Table 2-5.2)

Since b_0 <= 1/4 in., **b** = b_0 = **0.0338** in.

G = O.D. contact face - w = 12.7500 - 0.2700 = **12.4800** in.

Bolting is Adequate for Flange Design

Nominal Thickness is Adequate for Seating Conditions

Nominal Thickness is Adequate for Operating Conditions

Flange Thickness is Adequate for Flange Design

Nominal Thickness Selected = **1.0000** in.

Load and Bolting Calculations - Internal Pressure

The absolute value of effective pressure “P” is used for calculations.

$$\mathbf{P} = \frac{16\,M}{\pi\,G^3} + \frac{4\,F_A}{\pi\,G^2} + P_{int} + \text{Static Head} = \frac{16 * 0}{3.14159 * 12.4800^3} + \frac{4 * 0}{3.14159 * 12.4800^2} + 200.00 + 0.00 = \mathbf{200.00}\ \text{PSI}$$

Minimum $\mathbf{W_{m2}} = \pi bGy = 3.14159 * 0.0338 * 12.4800 * 3700$ = **4903** lb.

$$\mathbf{H} = \frac{\pi}{4} G^2 P = \frac{3.14159}{4} * 12.4800^2 * 200.00 = \mathbf{24465}\ \text{lb.}$$

$\mathbf{H_p} = 2b\pi GmP = 2 * 0.0338 * 3.14159 * 12.4800 * 2.75 * 200.00$ = **1458** lb.

Minimum $\mathbf{W_{m1}} = H + H_p = 24465 + 1458$ = **25923** lb.

$$\mathbf{A_{m1}} = \frac{W_{m1}}{S_b} = \frac{25923}{14400} = \mathbf{1.8002}\ \text{sq. in.}$$

$$\mathbf{A_{m2}} = \frac{W_{m2}}{S_a} = \frac{4903}{18800} = \mathbf{0.2608}\ \text{sq. in.}$$

$\mathbf{A_m}$ = Greater of A_{m1} or A_{m2} = greater of 1.8002 or 0.2608 = **1.8002** sq. in.

$\mathbf{A_b}$ = Number of Bolts * Bolt Root Area = 6 * 0.3020 = **1.8120** sq. in.

$$\mathbf{W} = \frac{(A_m + A_b)S_a}{2} = \frac{(1.8002 + 1.8120) * 18800}{2} = \mathbf{33955}\ \text{lb.}$$

$$\mathbf{h_G} = \frac{(C - G)}{2} = \frac{(14.7500 - 12.4800)}{2} = \mathbf{1.1350}\ \text{in.}$$

Ab >= Am, Bolting is Adequate for Flange Design

Thickness Calculations

$$\textbf{Operating Minimum t} = G\sqrt{\frac{CP}{SE} + \frac{1.9W_{m1}h_G}{SEG^3}} = 12.4800 * \sqrt{\frac{0.3000 * 200.00}{14400 * 1.00} + \frac{1.9 * 25923 * 1.1350}{14400 * 1.00 * 12.4800^3}} = \mathbf{0.9798}\ \text{in.}$$

$$\textbf{Seating Minimum t} = G\sqrt{\frac{1.9Wh_G}{SEG^3}} = 12.4800 * \sqrt{\frac{1.9 * 33955 * 1.1350}{20000 * 1.00 * 12.4800^3}} = \mathbf{0.5416}\ \text{in.}$$

Mininum t = maximum(0.9798, 0.5416) + CA = 0.9798 + 0.0000 = **0.9798** in.

ASME Flange Design Information

Host	Description	Type	Size (in.)	Material	ASME Class	Material Group	MAP (PSI)
2" Inlet Nozzle A	ASME Flange 1	Slip-On	2	SA-182 F304	300	2.1	517.50

Pressure Vessel Engineering, Ltd.
Cone to Cylinder 1

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5
Number: 1

Vessel Number: Sample Vessel 5

Date Printed: 11/20/2008

Cone-to-Cylinder Reinforcement
Design Calculations for Small End Juncture
Juncture is not a line of support

Description:	Cone to Cylinder 1	Design Temperature:	350 °F
Design Pressure:	200.00 PSI	Static Head:	1.00 PSI
Axial Load in Compression (f_2):	0 lb./in.		

Shell Design Information
8.03 OD Ferrule Advanced Coupling

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Shell Material:	SA-479 304, Low	Shell B Factor Table:	HA-1
		Material Stress (hot) (S_S):	14400 PSI
Modulus of Elasticity (E_S):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Shell Length (L_S):	1024.0000 in.	Longitudinal Efficiency (E_1):	70.00 %
Inside Radius:	3.9370 in.	Nominal Thickness (t_S):	0.0780 in.
Minimum Thickness (t):	0.0655 in.		

Conical Reducer Design Information
Conical Reducer 1

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Cone Material:	SA-240 304, High	Cone B Factor Table:	HA-1
		Material Stress (hot) (S_c):	18600 PSI
Modulus of Elasticity (E_c):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Cone Surface Length (L_c):	5.5284 in.		
Cone Small End Diameter:	8.0300 in.	Cone Large End Diameter:	12.7500 in.
Nominal Thickness (t_c):	0.1875 in.	Minimum Thickness (t_r):	0.0682 in.
Cone Angle (α):	25 °		

Internal Pressure per Appendix 1-5

For $\frac{\mathbf{P}}{\mathbf{S}_S E_1} = \frac{201.00}{14400 * 0.70}$ = 0.019940, maximum cone angle (Δ) = **12.5**°

Actual Cone Angle of 25° > maximum cone angle of 12.5°, reinforcement area requirements must be checked.

$\mathbf{Q}_S = f_2 + \left(\frac{PR_S}{2}\right) = 0.00 + \left(\frac{201.00 * 3.9370}{2}\right)$ = **395.67** lb./in.

$\mathbf{Y} = S_S E_S = 14400 * 26.7\text{x}10^6$ = **384480** 10^6 PSI²

k = 1, No ring stiffener.

$\mathbf{A}_{rS} = \left(\frac{kQ_S R_S}{S_S E_1}\right)\left(1 - \frac{\Delta}{\alpha}\right)\tan(\alpha) = \frac{1.0000 * 395.67 * 3.9370}{18600 * 0.70}\left(1 - \frac{12.5}{25.3}\right) 0.4721$ = **0.0369** sq. in.

$\mathbf{A}_{eS} = \frac{Pi}{4}\sqrt{R_S\, t_S}\left[(t_S - t) + \left(\frac{t_c - t_r}{\cos\alpha}\right)\right] = \frac{3.14159}{4}\sqrt{(3.9370 * 0.0780)}\left[(0.0780 - 0.0655) + \left(\frac{0.1875 - 0.0682}{0.9043}\right)\right]$ = **0.0629** sq. in.

$\mathbf{A}_{eS} + A_S = 0.0629 + 0.0000$ = **0.0629** sq. in.

Cone area + shell area >= required area for internal reinforcement
JUNCTURE PASSES

Cone-to-Cylinder Reinforcement
Design Calculations for Small End Juncture
Juncture is not a line of support

Description:	Cone to Cylinder 2	Design Temperature:	350 °F
Design Pressure:	200.00 PSI	Static Head:	1.00 PSI
Axial Load in Compression (f_2):	0 lb./in.		

Shell Design Information
8.03 OD Ferrule Advanced Coupling

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Shell Material:	SA-479 304, Low	Shell B Factor Table:	HA-1
		Material Stress (hot) (S_S):	14400 PSI
Modulus of Elasticity (E_S):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Shell Length (L_S):	2.0000 in.	Longitudinal Efficiency (E_1):	70.00 %
Inside Radius:	3.9370 in.	Nominal Thickness (t_S):	0.0780 in.
Minimum Thickness (t):	0.0655 in.		

Conical Reducer Design Information
Conical Reducer 1

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Cone Material:	SA-240 304, High	Cone B Factor Table:	HA-1
		Material Stress (hot) (S_C):	18600 PSI
Modulus of Elasticity (E_C):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Cone Surface Length (L_C):	5.5284 in.		
Cone Small End Diameter:	8.0300 in.	Cone Large End Diameter:	12.7500 in.
Nominal Thickness (t_C):	0.1875 in.	Minimum Thickness (t_r):	0.0682 in.
Cone Angle (α):	25 °		

Internal Pressure per Appendix 1-5

For $\frac{\mathbf{P}}{\mathbf{S}_S E_1} = \frac{201.00}{14400 * 0.70} = 0.019940$, maximum cone angle ($\Delta$) $= \mathbf{12.5}°$

Actual Cone Angle of 25° > maximum cone angle of 12.5°, reinforcement area requirements must be checked.

$$\mathbf{Q}_S = f_2 + \left(\frac{PR_S}{2}\right) = 0.00 + \left(\frac{201.00 * 3.9370}{2}\right) = \mathbf{395.67} \text{ lb./in.}$$

$\mathbf{Y} = S_S E_S = 14400 * 26.7 \times 10^6$ $= \mathbf{384480}\ 10^6$ PSI²

k = 1, No ring stiffener.

$$\mathbf{A}_{rS} = \left(\frac{kQ_S R_S}{S_S E_1}\right)\left(1 - \frac{\Delta}{\alpha}\right)\tan(\alpha) = \frac{1.0000 * 395.67 * 3.9370}{18600 * 0.70}\left(1 - \frac{12.5}{25.3}\right) 0.4721 = \mathbf{0.0369} \text{ sq. in.}$$

$$\mathbf{A}_{eS} = \frac{Pi}{4}\sqrt{R_S t_S}\left[(t_S - t) + \left(\frac{t_c - t_r}{\cos\alpha}\right)\right] = \frac{3.14159}{4}\sqrt{(3.9370 * 0.0780)}\left[(0.0780 - 0.0655) + \left(\frac{0.1875 - 0.0682}{0.9043}\right)\right] = \mathbf{0.0629} \text{ sq. in.}$$

$\mathbf{A}_{eS} + A_S = 0.0629 + 0.0000$ $= \mathbf{0.0629}$ sq. in.

Cone area + shell area >= required area for internal reinforcement
JUNCTURE PASSES

Pressure Vessel Engineering, Ltd.
Cone to Cylinder 3

Customer: **Pressure Vessel Engineering**
Job No: PVE-Sample5 Vessel Number: Sample Vessel 5
Number: 3

Date Printed: 11/20/2008

Cone-to-Cylinder Reinforcement
Design Calculations for Large End Juncture
Juncture is not a line of support

Description:	Cone to Cylinder 3	Design Temperature:	350 °F
Design Pressure:	200.00 PSI	Static Head:	0.00 PSI
Axial Load in Compression (f_1):	0 lb./in.		

Shell Design Information
Shell 2

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Shell Material:	SA-240 304H, High	Shell B Factor Table:	HA-1
		Material Stress (hot) (S_S):	18600 PSI
Modulus of Elasticity (E_S):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Shell Length (L_L):	15.5000 in.	Longitudinal Efficiency (E_1):	70.00 %
Inside Radius:	6.1870 in.	Nominal Thickness (t_S):	0.1880 in.
Minimum Thickness (t):	0.0974 in.		

Conical Reducer Design Information
Conical Reducer 1

Design Pressure:	200.00 PSI	Design Temperature:	350 °F
Static Head:	1.00 PSI	Corrosion Allowance:	0.0000 in.
Cone Material:	SA-240 304, High	Cone B Factor Table:	HA-1
		Material Stress (hot) (S_C):	18600 PSI
Modulus of Elasticity (E_C):	26.7 10^6 PSI	Material Stress (cold):	20000 PSI
Cone Surface Length (L_C):	5.5284 in.		
Cone Small End Diameter:	8.0300 in.	Cone Large End Diameter:	12.7500 in.
Nominal Thickness (t_C):	0.1875 in.	Minimum Thickness (t_r):	0.1077 in.
Cone Angle (α):	25 °		

Internal Pressure per Appendix 1-5

For $\frac{\mathbf{P}}{\mathbf{S}_S E_1} = \frac{200.00}{18600 * 0.70}$ = 0.015361, maximum cone angle (Δ) = **30.0**°

Actual Cone Angle of 25° <= maximum cone angle of 30.0° , reinforcement not required for internal pressure.

Maximum cone angle > Cone angle, reinforcement not needed for internal pressure
JUNCTURE PASSES

MDMT Report by Components

Design MDMT is -20 °F

Component	Material	Curve	Pressure	MDMT
Outlet 8" Ferrule Nozzle D	SA-479 304, High			Exempt per UCS-66(b)
Shell 2 - thin shell	SA-240 304, High			Exempt per UCS-66(b)
10" Inlet Nozzle C	SA-312 TP304 SMLS,			Exempt per UCS-66(b)
10" Inlet Flange	SA-240 304H, High			Exempt per UHA-51(d)
2" Heavy Ferrule Nozzle B	SA-479 304, High			Exempt per UCS-66(b)
thick shell	SA-312 TP304 SMLS,			Exempt per UHA-51(d)
Conical Reducer 1	SA-240 304, High			Other Exemption
2" Inlet Nozzle A	SA-312 TP304 SMLS,			Exempt per UCS-66(b)
Cover Flange	SA-240 304, Low			Exempt per UHA-51(d)
2"3000# Cplg N-E	SA-182 F304 <=5", Hig			Exempt per UCS-66(b)

The required design MDMT of -20 °F has been met or exceeded for the calculated MDMT values.

ASME Flanges Are Not Included in MDMT Calculations.

MAWP Report by Components

Component	Design Pressure	Static Head	Vessel MAWP New & Cold UG-98(a)	Component MAWP Hot & Corroded UG-98(b)	Vessel MAWP Hot & Corroded UG-98(a)
Outlet 8" Ferrule Nozzle D	200.00 PSI	1.00 PSI	331.85 PSI	309.55 PSI	308.55 PSI
Shell 2 - thin shell	200.00 PSI	1.00 PSI	341.24 PSI	318.29 PSI	317.29 PSI
10" Inlet Nozzle C	200.00 PSI	1.00 PSI	341.24 PSI	318.29 PSI	317.29 PSI
10" Inlet Flange	200.00 PSI	1.00 PSI	NC	476.81 PSI	475.81 PSI
2" Heavy Ferrule Nozzle B	200.00 PSI	0.00 PSI	597.00 PSI	555.21 PSI	555.21 PSI
thick shell	200.00 PSI	1.00 PSI	1664.83 PSI	1549.16 PSI	1548.16 PSI
Conical Reducer 1	200.00 PSI	1.00 PSI	375.37 PSI	350.02 PSI	349.02 PSI
2" Inlet Nozzle A	200.00 PSI	1.00 PSI	660.56 PSI	615.25 PSI	614.25 PSI
ASME Flange Class: 300 Gr:2.1		1.00 PSI	719.00 PSI	517.50 PSI	516.50 PSI
Cover Flange	200.00 PSI	0.00 PSI	262.82 PSI	201.31 PSI	201.31 PSI
2"3000# Cplg N-E	200.00 PSI	0.00 PSI	289.37 PSI	208.34 PSI	208.34 PSI

NC = Not Calculated Inc = Incomplete

Summary

Component with the lowest vessel MAWP(New & Cold) : **Cover Flange**
The lowest vessel MAWP(New & Cold) : **262.82** PSI

Component with the lowest vessel MAWP(Hot & Corroded) : **Cover Flange**
The lowest vessel MAWP(Hot & Corroded) : **201.31** PSI

Pressures are exclusive of any external loads.

Flange pressures listed here do not consider external loadings

Summary Information

	Dry Weight	Flooded Weight
Shell	67.95 lb.	153.85 lb.
Conical Reducer	9.62 lb.	23.99 lb.
Nozzle	17.14 lb.	17.14 lb.
Flange	68.13 lb.	68.13 lb.
ASME Flange	7.00 lb.	7.00 lb.
Totals	169.83 lb.	270.11 lb.

	Volume
Shell	10.28 Gal.
Conical Reducer	1.72 Gal.
Nozzle	1.68 Gal.
Totals	13.68 Gal.

	Area
Shell	5.76 Sq. Ft.
Conical Reducer	1.25 Sq. Ft.
Nozzle	2.67 Sq. Ft.
Totals	9.68 Sq. Ft.

Hydrostatic Test Information Par. UG-99(b)

Gauge at Top

Calculated Test Pressure: **279.57 PSI**

Specified Test Pressure: **260.00 PSI**

This calculation assumes one chamber.

This calculation is limited by the lowest component pressure per chamber.