

Design Conditions

Code:	ASME VIII-1
Year:	2010
Addenda:	2011
MAWP Shell Side:	1400 psi
Shell Side Temp:	650 °F
MAWP Tube Side:	150 psi
Tube Side Temp:	650 °F
MDMT:	-20 °F
Corrosion All. Shell:	0.0625 in
Corrosion All. Tube:	0.0625 in
Corrosion All. Tubes:	0 in
Hydrotest Shell:	1820 psi
Hydrotest Tube:	195 psi
Impact Exemption:	Exempt per UG-20(f)
Radiography:	Full

UG-22 Loadings Considered

Internal Press.:	Yes
External Press.:	Yes
Vessel Weight:	Yes
Weight of Attachments:	Yes
Attachment of Internals:	No
Attachment of Externals:	No
Cyclic or Dynamic Reactions:	No
Wind Loading:	No
Seismic Loading:	No
Fluid Impact Shock Reactions:	No
Temperature Gradients:	No
Differential Thermal Expansion:	No
Abnormal Pressures:	No
Hydrotest Loads:	No

ASME Section VIII-1 Calculations

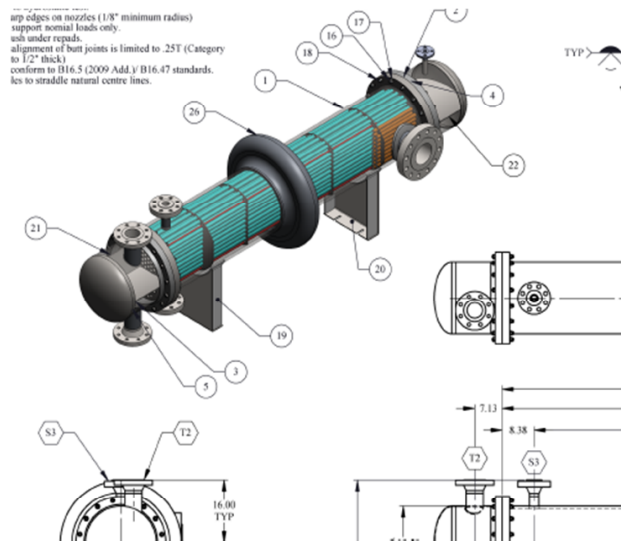
Cust: PVEng, H&C Heat Transfer

File: PVE-4293

Desc: Heat Exchanger Sample

Dwg: PVEdwg-4293-0.0 Heat Exchanger

Date: September 20, 2012



Author: Jakub Luszczycki CET
Reviewer: Laurence Brundrett, P. Eng.

Conclusion: The sample heat exchanger has been analyzed according to ASME Sec VIII-1 and found to be acceptable.

H&C Heat Transfer Sample
PVE-4293

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Cover Page

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DESIGN CALCULATION

In Accordance with ASME Section VIII Division 1

ASME Code Version : 2010 Edition, 2011a Addenda

Analysis Performed by : PRESSURE VESSEL ENGINEERING

Job File : \\PVESERVER\JOBS\4000-4999\4200-4299\4293 H&C HE

Date of Analysis : Sep 18,2012

PV Elite 2012, January 2012

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Warnings and Errors : Step: 0 5:58p Sep 18,2012

Class From To : Basic Element Checks.

=====

Class From To: Check of Additional Element Data

=====

There were no geometry errors or warnings.

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PV Elite Vessel Analysis Program: Input Data

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Exchanger Design Pressures and Temperatures

Shell Side Design Pressure	1400.0	psig
Channel Side Design Pressure	150.00	psig
Shell Side Design Temperature	650	F
Channel Side Design Temperature	650	F

Type of Hydrotest	UG99-b Note [34]	
Hydrotest Position	Horizontal	
Projection of Nozzle from Vessel Top	0.0000	in.
Projection of Nozzle from Vessel Bottom	0.0000	in.
Minimum Design Metal Temperature	-20	F
Type of Construction	Welded	
Special Service	None	
Degree of Radiography	RT 1	
Miscellaneous Weight Percent	0.0	
Use Higher Longitudinal Stresses (Flag)	Y	
Select t for Internal Pressure (Flag)	N	
Select t for External Pressure (Flag)	N	
Select t for Axial Stress (Flag)	N	
Select Location for Stiff. Rings (Flag)	N	
Consider Vortex Shedding	N	
Perform a Corroded Hydrotest	N	
Is this a Heat Exchanger	Yes	
User Defined Hydro. Press. (Used if > 0)	0.0000	psig
User defined MAWP	0.0000	psig
User defined MAPnc	0.0000	psig

Load Case 1	NP+EW+WI+FW+BW
Load Case 2	NP+EW+EE+FS+BS
Load Case 3	NP+OW+WI+FW+BW
Load Case 4	NP+OW+EQ+FS+BS
Load Case 5	NP+HW+HI
Load Case 6	NP+HW+HE
Load Case 7	IP+OW+WI+FW+BW
Load Case 8	IP+OW+EQ+FS+BS
Load Case 9	EP+OW+WI+FW+BW
Load Case 10	EP+OW+EQ+FS+BS
Load Case 11	HP+HW+HI
Load Case 12	HP+HW+HE
Load Case 13	IP+WE+EW
Load Case 14	IP+WF+CW
Load Case 15	IP+VO+OW
Load Case 16	IP+VE+EW
Load Case 17	NP+VO+OW
Load Case 18	FS+BS+IP+OW
Load Case 19	FS+BS+EP+OW

Wind Design Code No Wind Loads

Seismic Design Code No Seismic

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Input Echo : Step: 1 5:58p Sep 18,2012

Design Nozzle for Des. Press. + St. Head	Y
Consider MAP New and Cold in Noz. Design	N
Consider External Loads for Nozzle Des.	Y
Use ASME VIII-1 Appendix 1-9	N
Material Database Year	2009

Configuration Directives:

Do not use Nozzle MDMT Interpretation VIII-1 01-37	No
Use Table G instead of exact equation for "A"	Yes
Shell Head Joints are Tapered	Yes
Compute "K" in corroded condition	Yes
Use Code Case 2286	No
Use the MAWP to compute the MDMT	Yes
Using Metric Material Databases, ASME II D	No

Complete Listing of Vessel Elements and Details:

Element From Node	10
Element To Node	20
Element Type	Elliptical
Description	Left Head
Distance "FROM" to "TO"	0.08330 ft.
Element Outside Diameter	18.500 in.
Element Thickness	0.1680 in.
Internal Corrosion Allowance	0.06250 in.
Nominal Thickness	0.1875 in.
External Corrosion Allowance	0.0000 in.
Design Internal Pressure	150.00 psig
Design Temperature Internal Pressure	650 F
Design External Pressure	0.0000 psig
Design Temperature External Pressure	650 F
Effective Diameter Multiplier	1.2
Material Name	SA-516 70
Allowable Stress, Ambient	20000. psi
Allowable Stress, Operating	18800. psi
Allowable Stress, Hydrotest	26000. psi
Material Density	0.2830 lb./in ³
P Number Thickness	1.2500 in.
Yield Stress, Operating	28200. psi
UCS-66 Chart Curve Designation	B
External Pressure Chart Name	CS-2
UNS Number	K02700
Product Form	Plate
Efficiency, Longitudinal Seam	1.0
Efficiency, Circumferential Seam	1.0
Elliptical Head Factor	2.0
Element From Node	10
Detail Type	Liquid
Detail ID	HEAD
Dist. from "FROM" Node / Offset dist	0.1600 ft.
Height/Length of Liquid	1.5137 ft.
Liquid Density	62.400 lb./ft ³

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Input Echo :

Step: 1 5:58p Sep 18,2012

```

-----
Element From Node                20
Element To Node                  30
Element Type                      Cylinder
Description                      Left Channel
Distance "FROM" to "TO"         0.7290 ft.
Element Outside Diameter         18.500 in.
Element Thickness                 0.1875 in.
Internal Corrosion Allowance     0.06250 in.
Nominal Thickness                0.1875 in.
External Corrosion Allowance     0.0000 in.
Design Internal Pressure         150.00 psig
Design Temperature Internal Pressure 650 F
Design External Pressure         0.0000 psig
Design Temperature External Pressure 650 F
Effective Diameter Multiplier    1.2
Material Name                    SA-516 70
Efficiency, Longitudinal Seam    1.0
Efficiency, Circumferential Seam 1.0

```

```

Element From Node                20
Detail Type                      Liquid
Detail ID                        CHANNEL
Dist. from "FROM" Node / Offset dist 0.8300 ft.
Height/Length of Liquid         1.5104 ft.
Liquid Density                   62.400 lb./ft³

```

```

Element From Node                20
Detail Type                      Nozzle
Detail ID                        T1/T2
Dist. from "FROM" Node / Offset dist 0.3600 ft.
Nozzle Diameter                  4.0 in.
Nozzle Schedule                  40
Nozzle Class                     300
Layout Angle                     110.57
Blind Flange (Y/N)              N
Weight of Nozzle ( Used if > 0 ) 0.0000 lb.
Grade of Attached Flange        GR 1.1
Nozzle Matl                      SA-106 B

```

```

-----
Element From Node                30
Element To Node                  40
Element Type                      Flange
Description                      Channel Flange
Distance "FROM" to "TO"         0.2396 ft.
Flange Inside Diameter          18.125 in.
Element Thickness                1.7500 in.
Internal Corrosion Allowance     0.06250 in.
Nominal Thickness                1.7500 in.
External Corrosion Allowance     0.0000 in.
Design Internal Pressure         150.00 psig
Design Temperature Internal Pressure 650 F
Design External Pressure         0.0000 psig
Design Temperature External Pressure 650 F

```

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Input Echo :

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Effective Diameter Multiplier	1.2	
Material Name	SA-105	
Allowable Stress, Ambient	20000.	psi
Allowable Stress, Operating	17800.	psi
Allowable Stress, Hydrotest	26000.	psi
Material Density	0.2830	lb./in ³
P Number Thickness	1.2500	in.
Yield Stress, Operating	26700.	psi
UCS-66 Chart Curve Designation	B	
External Pressure Chart Name	CS-2	
UNS Number	K03504	
Product Form	Forgings	
Perform Flange Stress Calculation (Y/N)	Y	
Weight of ANSI B16.5/B16.47 Flange	0.0000	lb.
Class of ANSI B16.5/B16.47 Flange		
Grade of ANSI B16.5/B16.47 Flange		

Element From Node	40	
Element To Node	50	
Element Type	Cylinder	
Description	Shell	
Distance "FROM" to "TO"	7.6670	ft.
Element Outside Diameter	18.500	in.
Element Thickness	0.7500	in.
Internal Corrosion Allowance	0.06250	in.
Nominal Thickness	0.7500	in.
External Corrosion Allowance	0.0000	in.
Design Internal Pressure	1400.0	psig
Design Temperature Internal Pressure	650	F
Design External Pressure	15.000	psig
Design Temperature External Pressure	650	F
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	
Allowable Stress, Ambient	20000.	psi
Allowable Stress, Operating	18800.	psi
Allowable Stress, Hydrotest	26000.	psi
Material Density	0.2830	lb./in ³
P Number Thickness	1.2500	in.
Yield Stress, Operating	28200.	psi
UCS-66 Chart Curve Designation	B	
External Pressure Chart Name	CS-2	
UNS Number	K02700	
Product Form	Plate	
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	

Element From Node	40	
Detail Type	Saddle	
Detail ID	Lft Sd1	
Dist. from "FROM" Node / Offset dist	1.5989	ft.
Width of Saddle	4.0000	in.
Height of Saddle at Bottom	25.000	in.
Saddle Contact Angle	120.0	
Height of Composite Ring Stiffener	0.0000	in.
Width of Wear Plate	5.0000	in.

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Input Echo :

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Thickness of Wear Plate 0.2500 in.
Contact Angle, Wear Plate (degrees) 132.0

Element From Node 40
Detail Type Saddle
Detail ID Rgt Sdl
Dist. from "FROM" Node / Offset dist 6.4000 ft.
Width of Saddle 4.0000 in.
Height of Saddle at Bottom 25.000 in.
Saddle Contact Angle 120.0
Height of Composite Ring Stiffener 0.0000 in.
Width of Wear Plate 5.0000 in.
Thickness of Wear Plate 0.2500 in.
Contact Angle, Wear Plate (degrees) 132.0

Element From Node 40
Detail Type Liquid
Detail ID SHELL
Dist. from "FROM" Node / Offset dist 0.0000 ft.
Height/Length of Liquid 1.4427 ft.
Liquid Density 62.400 lb./ft³

Element From Node 40
Detail Type Nozzle
Detail ID S2
Dist. from "FROM" Node / Offset dist 0.5000 ft.
Nozzle Diameter 3.0 in.
Nozzle Schedule XXS
Nozzle Class 900
Layout Angle 270.0
Blind Flange (Y/N) N
Weight of Nozzle (Used if > 0) 0.0000 lb.
Grade of Attached Flange GR 1.1
Nozzle Matl SA-106 B

Element From Node 40
Detail Type Nozzle
Detail ID S1
Dist. from "FROM" Node / Offset dist 7.2900 ft.
Nozzle Diameter 6.0 in.
Nozzle Schedule XXS
Nozzle Class 900
Layout Angle 270.0
Blind Flange (Y/N) N
Weight of Nozzle (Used if > 0) 0.0000 lb.
Grade of Attached Flange GR 1.1
Nozzle Matl SA-106 B

Element From Node 40
Detail Type Nozzle
Detail ID S3
Dist. from "FROM" Node / Offset dist 0.5000 ft.
Nozzle Diameter 2.0 in.
Nozzle Schedule XXS
Nozzle Class 900
Layout Angle 90.0
Blind Flange (Y/N) N

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Input Echo :

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Weight of Nozzle (Used if > 0)	0.0000	lb.
Grade of Attached Flange	GR 1.1	
Nozzle Matl	SA-106 B	

Element From Node	50	
Element To Node	60	
Element Type	Flange	
Description	Channel Flange	
Distance "FROM" to "TO"	0.2396	ft.
Flange Inside Diameter	18.125	in.
Element Thickness	1.7500	in.
Internal Corrosion Allowance	0.06250	in.
Nominal Thickness	1.7500	in.
External Corrosion Allowance	0.0000	in.
Design Internal Pressure	150.00	psig
Design Temperature Internal Pressure	650	F
Design External Pressure	0.0000	psig
Design Temperature External Pressure	650	F
Effective Diameter Multiplier	1.2	
Material Name	SA-105	
Allowable Stress, Ambient	20000.	psi
Allowable Stress, Operating	17800.	psi
Allowable Stress, Hydrotest	26000.	psi
Material Density	0.2830	lb./in ³
P Number Thickness	1.2500	in.
Yield Stress, Operating	26700.	psi
UCS-66 Chart Curve Designation	B	
External Pressure Chart Name	CS-2	
UNS Number	K03504	
Product Form	Forgings	
Perform Flange Stress Calculation (Y/N)	Y	
Weight of ANSI B16.5/B16.47 Flange	0.0000	lb.
Class of ANSI B16.5/B16.47 Flange		
Grade of ANSI B16.5/B16.47 Flange		

Element From Node	60	
Element To Node	70	
Element Type	Cylinder	
Description	Channel	
Distance "FROM" to "TO"	0.7290	ft.
Element Outside Diameter	18.500	in.
Element Thickness	0.1875	in.
Internal Corrosion Allowance	0.06250	in.
Nominal Thickness	0.1875	in.
External Corrosion Allowance	0.0000	in.
Design Internal Pressure	150.00	psig
Design Temperature Internal Pressure	650	F
Design External Pressure	0.0000	psig
Design Temperature External Pressure	650	F
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	
Allowable Stress, Ambient	20000.	psi
Allowable Stress, Operating	18800.	psi

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Input Echo :

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Allowable Stress, Hydrotest	26000.	psi
Material Density	0.2830	lb./in ³
P Number Thickness	1.2500	in.
Yield Stress, Operating	28200.	psi
UCS-66 Chart Curve Designation	B	
External Pressure Chart Name	CS-2	
UNS Number	K02700	
Product Form	Plate	
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	

Element From Node	60	
Detail Type	Liquid	
Detail ID	CHANNEL	
Dist. from "FROM" Node / Offset dist	0.0000	ft.
Height/Length of Liquid	1.5104	ft.
Liquid Density	62.400	lb./ft ³

Element From Node	60	
Detail Type	Nozzle	
Detail ID	T3 & T4	
Dist. from "FROM" Node / Offset dist	0.5000	ft.
Nozzle Diameter	1.0	in.
Nozzle Schedule	80	
Nozzle Class	300	
Layout Angle	90.0	
Blind Flange (Y/N)	N	
Weight of Nozzle (Used if > 0)	0.0000	lb.
Grade of Attached Flange	GR 1.1	
Nozzle Matl	SA-106 B	

Element From Node	70	
Element To Node	80	
Element Type	Elliptical	
Description	Right Head	
Distance "FROM" to "TO"	0.08330	ft.
Element Outside Diameter	18.500	in.
Element Thickness	0.1680	in.
Internal Corrosion Allowance	0.06250	in.
Nominal Thickness	0.1875	in.
External Corrosion Allowance	0.0000	in.
Design Internal Pressure	150.00	psig
Design Temperature Internal Pressure	650	F
Design External Pressure	0.0000	psig
Design Temperature External Pressure	650	F
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	
Elliptical Head Factor	2.0	

Element From Node	70	
Detail Type	Liquid	
Detail ID	HEAD	
Dist. from "FROM" Node / Offset dist	0.0000	ft.

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Input Echo : Step: 1 5:58p Sep 18,2012

Height/Length of Liquid

1.5137 ft.

Liquid Density

62.400 lb./ft³

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Flg Calc [Int P] : Ch. Flange Flng: 3 5:58p Sep 18,2012

Flange Input Data Values

Description: Ch. Flange :

Channel Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	150.00	psig
Design Temperature		650	F
Internal Corrosion Allowance	ci	0.0625	in.
External Corrosion Allowance	ce	0.0000	in.
Use Corrosion Allowance in Thickness Calcs.		No	
Flange Inside Diameter	B	18.125	in.
Flange Outside Diameter	A	23.500	in.
Flange Thickness	t	1.7500	in.
Thickness of Hub at Small End	go	0.1875	in.
Thickness of Hub at Large End	gl	0.5625	in.
Length of Hub	h	1.1250	in.
Flange Material		SA-105	
Flange Material UNS number		K03504	
Flange Allowable Stress At Temperature	Sfo	17800.00	psi
Flange Allowable Stress At Ambient	Sfa	20000.00	psi
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Diameter of Bolt Circle	C	21.750	in.
Nominal Bolt Diameter	dB	0.6250	in.
Type of Threads		UNC Thread Series	
Number of Bolts		16	
Flange Face Outside Diameter	Fod	19.750	in.
Flange Face Inside Diameter	Fid	18.125	in.
Flange Facing Sketch		1, Code Sketch 1a	
Gasket Outside Diameter	Go	19.125	in.
Gasket Inside Diameter	Gi	18.125	in.
Gasket Factor	m	2.5000	
Gasket Design Seating Stress	y	2900.00	psi
Column for Gasket Seating		2, Code Column II	
Gasket Thickness	tg	0.1250	in.
Length of Partition Gasket	lp	18.1250	in.
Width of Partition Gasket	tp	0.3130	in.
Partition Gasket Factor	mPart	2.5000	
Partition Gasket Design Seating Stress	yPart	2900.00	psi

ASME Code, Section VIII, Division 1, 2010, 2011a

Hub Small End Required Thickness due to Internal Pressure:

$$= (P*(D/2+Ca))/(S*E-0.6*P) \text{ per UG-27 (c)(1)}$$

$$= (150.00*(18.1250/2+0.0625))/(17800.00*1.00-0.6*150.00)+Ca$$

$$= 0.1398 \text{ in.}$$

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Flg Calc [Int P] : Ch. Flange Flng: 3 5:58p Sep 18,2012

Hub Small End Hub MAWP:

$$\begin{aligned} &= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c)(1)} \\ &= (17800.00 \cdot 1.00 \cdot 0.1250) / (9.1250 + 0.6 \cdot 0.1250) \\ &= 241.848 \text{ psig} \end{aligned}$$

Corroded Flange ID,	Bcor = B+2*Fcor	18.250	in.
Corroded Large Hub,	g1Cor = g1-ci	0.500	in.
Corroded Small Hub,	g0Cor = go-ci	0.125	in.
Code R Dimension,	R = ((C-Bcor)/2)-g1cor	1.250	in.

Gasket Contact Width,	N = (Go - Gi) / 2	0.500	in.
Basic Gasket Width,	bo = N / 2	0.250	in.
Effective Gasket Width,	b = bo	0.250	in.
Gasket Reaction Diameter,	G = (Go + Gi) / 2	18.625	in.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$\begin{aligned} &= 0.785 \cdot G^2 \cdot Peq \\ &= 0.785 \cdot 18.6250^2 \cdot 150.000 \\ &= 40867.090 \text{ lb.} \end{aligned}$$

Contact Load on Gasket Surfaces [Hp]:

$$\begin{aligned} &= 2 \cdot b \cdot Pi \cdot G \cdot m \cdot P + 2 \cdot lp \cdot bPart \cdot mPart \cdot P \\ &= 2 \cdot 0.2500 \cdot 3.1416 \cdot 18.6250 \cdot 2.5000 \cdot 150.00 \\ &\quad + 2.0 \cdot 18.1250 \cdot 0.1565 \cdot 2.5000 \cdot 150.0000 \\ &= 13098.453 \text{ lb.} \end{aligned}$$

Hydrostatic End Load at Flange ID [Hd]:

$$\begin{aligned} &= Pi \cdot Bcor^2 \cdot P / 4 \\ &= 3.1416 \cdot 18.2500^2 \cdot 150.0000 / 4 \\ &= 39238.004 \text{ lb.} \end{aligned}$$

Pressure Force on Flange Face [Ht]:

$$\begin{aligned} &= H - Hd \\ &= 40867 - 39238 \\ &= 1629.086 \text{ lb.} \end{aligned}$$

Operating Bolt Load [Wm1]:

$$\begin{aligned} &= \max(H + Hp + H'p, 0) \\ &= \max(40867 + 13098 + 0, 0) \\ &= 53965.543 \text{ lb.} \end{aligned}$$

Gasket Seating Bolt Load [Wm2]:

$$\begin{aligned} &= y \cdot b \cdot Pi \cdot G + yPart \cdot bPart \cdot lp \\ &= 2900.00 \cdot 0.2500 \cdot 3.141 \cdot 18.625 + 2900.00 \cdot 0.1565 \cdot 18.12 \\ &= 50647.352 \text{ lb.} \end{aligned}$$

Required Bolt Area [Am]:

$$\begin{aligned} &= \text{Maximum of } Wm1/Sb, Wm2/Sa \\ &= \text{Maximum of } 53965/25000, 50647/25000 \\ &= 2.159 \text{ in}^2 \end{aligned}$$

ASME Maximum Circumferential Spacing between Bolts per App. 2 eq. (3) [Bsmax]:

$$\begin{aligned} &= 2a + 6t / (m + 0.5) \\ &= 2 \cdot 0.625 + 6 \cdot 1.750 / (2.50 + 0.5) \\ &= 4.750 \text{ in.} \end{aligned}$$

Actual Circumferential Bolt Spacing [Bs]:

$$\begin{aligned} &= C \cdot \sin(pi / n) \\ &= 21.750 \cdot \sin(3.142 / 16) \\ &= 4.243 \text{ in.} \end{aligned}$$

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Flg Calc [Int P] : Ch. Flange Flng: 3 5:58p Sep 18,2012

ASME Moment Multiplier for Bolt Spacing per App. 2 eq. (7) [Bsc]:

$$= \max(\text{sqrt}(Bs/(2a + t)), 1)$$

$$= \max(\text{sqrt}(4.243/(2 * 0.625 + 1.750)), 1)$$

$$= 1.1893$$

Bolting Information for UNC Thread Series (Non Mandatory):

	Minimum	Actual	Maximum
Bolt Area, in ²	2.159	3.232	
Radial distance bet. hub and bolts	0.938	1.250	
Radial distance bet. bolts and the edge	0.750	0.875	
Circumferential spacing between bolts	1.500	4.243	4.750

Min. Gasket Contact Width (Brownell Young) [Not an ASME Calc] [Nmin]:

$$= Ab * Sa / (y * Pi * (Go + Gi))$$

$$= 3.232 * 25000.00 / (2900.00 * 3.14 * (19.125 + 18.12))$$

$$= 0.238 \text{ in.}$$

Flange Design Bolt Load, Gasket Seating [W]:

$$= Sa * (Am + Ab) / 2$$

$$= 25000.00 * (2.1586 + 3.2320) / 2$$

$$= 67382.77 \text{ lb.}$$

Gasket Load for the Operating Condition [HG]:

$$= Wm1 - H$$

$$= 53965 - 40867$$

$$= 13098.45 \text{ lb.}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

$$= (C - G) / 2$$

$$= (21.7500 - 18.6250) / 2$$

$$= 1.5625 \text{ in.}$$

Distance to Face Pressure Reaction [ht]:

$$= (R + g1 + hg) / 2$$

$$= (1.2500 + 0.5000 + 1.5625) / 2$$

$$= 1.6562 \text{ in.}$$

Distance to End Pressure Reaction [hd]:

$$= R + (g1 / 2)$$

$$= 1.2500 + (0.5000 / 2.0)$$

$$= 1.5000 \text{ in.}$$

Summary of Moments for Internal Pressure:

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	39238.	1.5000	1.1893	5833. ft.lb.
Face Pressure, Mt	1629.	1.6562	1.1893	267. ft.lb.
Gasket Load, Mg	13098.	1.5625	1.1893	2028. ft.lb.
Gasket Seating, Matm	67383.	1.5625	1.1893	10435. ft.lb.

Total Moment for Operation, Mop 8129. ft.lb.
 Total Moment for Gasket seating, Matm 10435. ft.lb.

Effective Hub Length, ho = sqrt(Bcor*goCor) 1.510 in.
 Hub Ratio, h/h0 = HL / H0 0.745
 Thickness Ratio, g1/g0 = (g1Cor/goCor) 4.000

H&C Heat Transfer Sample

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Flg Calc [Int P] : Ch. Flange Flng: 3 5:58p Sep 18,2012

Flange Factors for Integral Flange:

Factor F per 2-7.2			0.736
Factor V per 2-7.3			0.081
Factor f per 2-7.6			2.930
Factors from Figure 2-7.1		K =	1.288
	T =	1.803	U = 8.595
	Y =	7.822	Z = 4.039
	d =	2.513 in. ³	e = 0.4875 in. ⁻¹
Stress Factors		ALPHA =	1.853
	BETA =	2.138	GAMMA = 1.028
	DELTA =	2.133	Lamda = 3.161

Longitudinal Hub Stress, Operating [SHo]:

$$\begin{aligned} &= (f * Mop / Bcor) / (L * g1^2) \\ &= (2.9303*97547/18.2500) / (3.1608*0.5000^2) \\ &= 19821.01 \text{ psi} \end{aligned}$$

Longitudinal Hub Stress, Seating [SHa]:

$$\begin{aligned} &= (f * Matm / Bcor) / (L * g1^2) \\ &= (2.9303*125214/18.2500) / (3.1608*0.5000^2) \\ &= 25442.91 \text{ psi} \end{aligned}$$

Radial Flange Stress, Operating [SRo]:

$$\begin{aligned} &= (Beta * Mop / Bcor) / (L * t^2) \\ &= (2.1376*97547/18.2500) / (3.1608*1.7500^2) \\ &= 1180.33 \text{ psi} \end{aligned}$$

Radial Flange Stress, Seating [SRa]:

$$\begin{aligned} &= (Beta * Matm / Bcor) / (L * t^2) \\ &= (2.1376*125214/18.2500) / (3.1608*1.7500^2) \\ &= 1515.11 \text{ psi} \end{aligned}$$

Tangential Flange Stress, Operating [STo]:

$$\begin{aligned} &= (Y * Mo / (t^2 * Bcor)) - Z * SRo \\ &= (7.8217*97547 / (1.7500^2 * 18.2500)) - 4.0391 * 1180 \\ &= 8883.92 \text{ psi} \end{aligned}$$

Tangential Flange Stress, Seating [STa]:

$$\begin{aligned} &= (y * Matm / (t^2 * Bcor)) - Z * SRa \\ &= (7.8217*125214 / (1.7500^2 * 18.2500)) - 4.0391 * 1515 \\ &= 11403.70 \text{ psi} \end{aligned}$$

Average Flange Stress, Operating [SAo]:

$$\begin{aligned} &= (SHo + \max(SRo, STo)) / 2 \\ &= (19821 + \max(1180, 8883)) / 2 \\ &= 14352.46 \text{ psi} \end{aligned}$$

Average Flange Stress, Seating [SAa]:

$$\begin{aligned} &= (SHa + \max(SRa, STa)) / 2 \\ &= (25442 + \max(1515, 11403)) / 2 \\ &= 18423.30 \text{ psi} \end{aligned}$$

Bolt Stress, Operating [BSo]:

$$\begin{aligned} &= (Wm1 / Ab) \\ &= (53965 / 3.2320) \\ &= 16697.26 \text{ psi} \end{aligned}$$

Bolt Stress, Seating [BSa]:

$$\begin{aligned} &= (Wm2 / Ab) \\ &= (50647 / 3.2320) \\ &= 15670.59 \text{ psi} \end{aligned}$$

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Flg Calc [Int P] : Ch. Flange Flng: 3 5:58p Sep 18,2012

Stress Computation Results:	Operating		Gasket Seating	
	Actual	Allowed	Actual	Allowed
Longitudinal Hub	19821.	26700.	25443.	30000. psi
Radial Flange	1180.	17800.	1515.	20000. psi
Tangential Flange	8884.	17800.	11404.	20000. psi
Maximum Average	14352.	17800.	18423.	20000. psi
Bolting	16697.	25000.	15671.	25000. psi

Minimum Required Flange Thickness	1.684 in.
Estimated M.A.W.P. (Operating)	186.0 psig
Estimated Finished Weight of Flange at given Thk.	94.0 lbm
Estimated Unfinished Weight of Forging at given Thk	143.0 lbm

Flange Rigidity Based on Required Thickness [ASME]:

Flange Rigidity Index, Seating (rotation check) per APP. 2 [Js]:

$$\begin{aligned} &= 52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^{(2)} * ho * Ki) \\ &= 52.14 * 10434.6/1.2026 * 12.000 * 0.081/(2.908 * 29400000 * \\ &\quad 0.125^{(2)} * 1.510 * 0.300) \\ &= 0.724 \quad (\text{should be } \leq 1) \end{aligned}$$

Flange Rigidity Index Operating (rotation check) per APP. 2 [J]:

$$\begin{aligned} &= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^{(2)} * ho * Ki) \\ &= 52.14 * 8128.9/1.2026 * 12.000 * 0.081/(2.908 * 26000000 \\ &\quad * 0.125^{(2)} * 1.510 * 0.300) \\ &= 0.638 \quad (\text{should be } \leq 1) \end{aligned}$$

Flange Rigidity Based on Given Thickness [ASME]:

Flange Rigidity Index, Seating (rotation check) per APP. 2 [Js]:

$$\begin{aligned} &= 52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^{(2)} * ho * Ki) \\ &= 52.14 * 10434.6/1.1893 * 12.000 * 0.081/(3.161 * 29400000 * \\ &\quad 0.125^{(2)} * 1.510 * 0.300) \\ &= 0.674 \quad (\text{should be } \leq 1) \end{aligned}$$

Flange Rigidity Index Operating (rotation check) per APP. 2 [J]:

$$\begin{aligned} &= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^{(2)} * ho * Ki) \\ &= 52.14 * 8128.9/1.1893 * 12.000 * 0.081/(3.161 * 26000000 \\ &\quad * 0.125^{(2)} * 1.510 * 0.300) \\ &= 0.593 \quad (\text{should be } \leq 1) \end{aligned}$$

Minimum Design Metal Temperature Results:

Stress Ratio = 0.775 , Temperature Reduction per Fig. UCS 66.1 = 23 F

Min Metal Temp. w/o impact per UCS-66 -20 F

Min Metal Temp. at Required thickness (UCS 66.1) -43 F

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Flg Calc [Int P] : Ch. Flange Flng: 4 5:58p Sep 18,2012

Flange Input Data Values Description: Ch. Flange :

Channel Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	150.00	psig
Design Temperature		650	F
Internal Corrosion Allowance	ci	0.0625	in.
External Corrosion Allowance	ce	0.0000	in.
Use Corrosion Allowance in Thickness Calcs.		No	
Flange Inside Diameter	B	18.125	in.
Flange Outside Diameter	A	23.500	in.
Flange Thickness	t	1.7500	in.
Thickness of Hub at Small End	go	0.1875	in.
Thickness of Hub at Large End	gl	0.5625	in.
Length of Hub	h	1.1250	in.
Flange Material		SA-105	
Flange Material UNS number		K03504	
Flange Allowable Stress At Temperature	Sfo	17800.00	psi
Flange Allowable Stress At Ambient	Sfa	20000.00	psi
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Diameter of Bolt Circle	C	21.750	in.
Nominal Bolt Diameter	dB	0.6250	in.
Type of Threads	UNC Thread Series		
Number of Bolts		16	
Flange Face Outside Diameter	Fod	19.750	in.
Flange Face Inside Diameter	Fid	18.125	in.
Flange Facing Sketch	1, Code Sketch 1a		
Gasket Outside Diameter	Go	19.125	in.
Gasket Inside Diameter	Gi	18.125	in.
Gasket Factor	m	2.5000	
Gasket Design Seating Stress	y	2900.00	psi
Column for Gasket Seating	2, Code Column II		
Gasket Thickness	tg	0.1250	in.
Length of Partition Gasket	lp	18.1250	in.
Width of Partition Gasket	tp	0.3130	in.
Partition Gasket Factor	mPart	2.5000	
Partition Gasket Design Seating Stress	yPart	2900.00	psi

ASME Code, Section VIII, Division 1, 2010, 2011a

Hub Small End Required Thickness due to Internal Pressure:

$$\begin{aligned}
 &= (P*(D/2+Ca))/(S*E-0.6*P) \text{ per UG-27 (c)(1)} \\
 &= (150.00*(18.1250/2+0.0625))/(17800.00*1.00-0.6*150.00)+Ca \\
 &= 0.1398 \text{ in.}
 \end{aligned}$$

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Flg Calc [Int P] : Ch. Flange Flng: 4 5:58p Sep 18,2012

Hub Small End Hub MAWP:

$$\begin{aligned} &= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c)(1)} \\ &= (17800.00 \cdot 1.00 \cdot 0.1250) / (9.1250 + 0.6 \cdot 0.1250) \\ &= 241.848 \text{ psig} \end{aligned}$$

Corroded Flange ID,	Bcor = B+2*Fcor	18.250	in.
Corroded Large Hub,	g1Cor = g1-ci	0.500	in.
Corroded Small Hub,	g0Cor = go-ci	0.125	in.
Code R Dimension,	R = ((C-Bcor)/2)-g1cor	1.250	in.

Gasket Contact Width,	N = (Go - Gi) / 2	0.500	in.
Basic Gasket Width,	bo = N / 2	0.250	in.
Effective Gasket Width,	b = bo	0.250	in.
Gasket Reaction Diameter,	G = (Go + Gi) / 2	18.625	in.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$\begin{aligned} &= 0.785 \cdot G^2 \cdot Peq \\ &= 0.785 \cdot 18.6250^2 \cdot 150.000 \\ &= 40867.090 \text{ lb.} \end{aligned}$$

Contact Load on Gasket Surfaces [Hp]:

$$\begin{aligned} &= 2 \cdot b \cdot Pi \cdot G \cdot m \cdot P + 2 \cdot lp \cdot bPart \cdot mPart \cdot P \\ &= 2 \cdot 0.2500 \cdot 3.1416 \cdot 18.6250 \cdot 2.5000 \cdot 150.00 \\ &\quad + 2.0 \cdot 18.1250 \cdot 0.1565 \cdot 2.5000 \cdot 150.0000 \\ &= 13098.453 \text{ lb.} \end{aligned}$$

Hydrostatic End Load at Flange ID [Hd]:

$$\begin{aligned} &= Pi \cdot Bcor^2 \cdot P / 4 \\ &= 3.1416 \cdot 18.2500^2 \cdot 150.0000 / 4 \\ &= 39238.004 \text{ lb.} \end{aligned}$$

Pressure Force on Flange Face [Ht]:

$$\begin{aligned} &= H - Hd \\ &= 40867 - 39238 \\ &= 1629.086 \text{ lb.} \end{aligned}$$

Operating Bolt Load [Wm1]:

$$\begin{aligned} &= \max(H + Hp + H'p, 0) \\ &= \max(40867 + 13098 + 0, 0) \\ &= 53965.543 \text{ lb.} \end{aligned}$$

Gasket Seating Bolt Load [Wm2]:

$$\begin{aligned} &= y \cdot b \cdot Pi \cdot G + yPart \cdot bPart \cdot lp \\ &= 2900.00 \cdot 0.2500 \cdot 3.141 \cdot 18.625 + 2900.00 \cdot 0.1565 \cdot 18.12 \\ &= 50647.352 \text{ lb.} \end{aligned}$$

Required Bolt Area [Am]:

$$\begin{aligned} &= \text{Maximum of } Wm1/Sb, Wm2/Sa \\ &= \text{Maximum of } 53965/25000, 50647/25000 \\ &= 2.159 \text{ in}^2 \end{aligned}$$

ASME Maximum Circumferential Spacing between Bolts per App. 2 eq. (3) [Bsmax]:

$$\begin{aligned} &= 2a + 6t / (m + 0.5) \\ &= 2 \cdot 0.625 + 6 \cdot 1.750 / (2.50 + 0.5) \\ &= 4.750 \text{ in.} \end{aligned}$$

Actual Circumferential Bolt Spacing [Bs]:

$$\begin{aligned} &= C \cdot \sin(pi / n) \\ &= 21.750 \cdot \sin(3.142 / 16) \\ &= 4.243 \text{ in.} \end{aligned}$$

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Flg Calc [Int P] : Ch. Flange

Flng: 4

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ASME Moment Multiplier for Bolt Spacing per App. 2 eq. (7) [Bsc]:

$$= \max(\text{sqrt}(Bs/(2a + t)), 1)$$

$$= \max(\text{sqrt}(4.243/(2 * 0.625 + 1.750)), 1)$$

$$= 1.1893$$

Bolting Information for UNC Thread Series (Non Mandatory):

	Minimum	Actual	Maximum
Bolt Area, in ²	2.159	3.232	
Radial distance bet. hub and bolts	0.938	1.250	
Radial distance bet. bolts and the edge	0.750	0.875	
Circumferential spacing between bolts	1.500	4.243	4.750

Min. Gasket Contact Width (Brownell Young) [Not an ASME Calc] [Nmin]:

$$= Ab * Sa / (y * Pi * (Go + Gi))$$

$$= 3.232 * 25000.00 / (2900.00 * 3.14 * (19.125 + 18.12))$$

$$= 0.238 \text{ in.}$$

Flange Design Bolt Load, Gasket Seating [W]:

$$= Sa * (Am + Ab) / 2$$

$$= 25000.00 * (2.1586 + 3.2320) / 2$$

$$= 67382.77 \text{ lb.}$$

Gasket Load for the Operating Condition [HG]:

$$= Wm1 - H$$

$$= 53965 - 40867$$

$$= 13098.45 \text{ lb.}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

$$= (C - G) / 2$$

$$= (21.7500 - 18.6250) / 2$$

$$= 1.5625 \text{ in.}$$

Distance to Face Pressure Reaction [ht]:

$$= (R + g1 + hg) / 2$$

$$= (1.2500 + 0.5000 + 1.5625) / 2$$

$$= 1.6562 \text{ in.}$$

Distance to End Pressure Reaction [hd]:

$$= R + (g1 / 2)$$

$$= 1.2500 + (0.5000 / 2.0)$$

$$= 1.5000 \text{ in.}$$

Summary of Moments for Internal Pressure:

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	39238.	1.5000	1.1893	5833. ft.lb.
Face Pressure, Mt	1629.	1.6562	1.1893	267. ft.lb.
Gasket Load, Mg	13098.	1.5625	1.1893	2028. ft.lb.
Gasket Seating, Matm	67383.	1.5625	1.1893	10435. ft.lb.

Total Moment for Operation, Mop 8129. ft.lb.
 Total Moment for Gasket seating, Matm 10435. ft.lb.

Effective Hub Length, ho = sqrt(Bcor*goCor) 1.510 in.
 Hub Ratio, h/h0 = HL / H0 0.745
 Thickness Ratio, g1/g0 = (g1Cor/goCor) 4.000

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Flg Calc [Int P] : Ch. Flange

Flng:

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Flange Factors for Integral Flange:

Factor F per 2-7.2 0.736

Factor V per 2-7.3 0.081

Factor f per 2-7.6 2.930

Factors from Figure 2-7.1 K = 1.288

T = 1.803 U = 8.595

Y = 7.822 Z = 4.039

d = 2.513 in.³ e = 0.4875 in.⁻¹

Stress Factors ALPHA = 1.853

BETA = 2.138 GAMMA = 1.028

DELTA = 2.133 Lamda = 3.161

Longitudinal Hub Stress, Operating [SHo]:

$$\begin{aligned} &= (f * Mop / Bcor) / (L * g1^2) \\ &= (2.9303*97547/18.2500) / (3.1608*0.5000^2) \\ &= 19821.01 \text{ psi} \end{aligned}$$

Longitudinal Hub Stress, Seating [SHa]:

$$\begin{aligned} &= (f * Matm / Bcor) / (L * g1^2) \\ &= (2.9303*125214/18.2500) / (3.1608*0.5000^2) \\ &= 25442.91 \text{ psi} \end{aligned}$$

Radial Flange Stress, Operating [SRo]:

$$\begin{aligned} &= (Beta * Mop / Bcor) / (L * t^2) \\ &= (2.1376*97547/18.2500) / (3.1608*1.7500^2) \\ &= 1180.33 \text{ psi} \end{aligned}$$

Radial Flange Stress, Seating [SRa]:

$$\begin{aligned} &= (Beta * Matm / Bcor) / (L * t^2) \\ &= (2.1376*125214/18.2500) / (3.1608*1.7500^2) \\ &= 1515.11 \text{ psi} \end{aligned}$$

Tangential Flange Stress, Operating [STo]:

$$\begin{aligned} &= (Y * Mo / (t^2 * Bcor)) - Z * SRo \\ &= (7.8217*97547 / (1.7500^2 * 18.2500)) - 4.0391 * 1180 \\ &= 8883.92 \text{ psi} \end{aligned}$$

Tangential Flange Stress, Seating [STa]:

$$\begin{aligned} &= (y * Matm / (t^2 * Bcor)) - Z * SRa \\ &= (7.8217*125214 / (1.7500^2 * 18.2500)) - 4.0391 * 1515 \\ &= 11403.70 \text{ psi} \end{aligned}$$

Average Flange Stress, Operating [SAo]:

$$\begin{aligned} &= (SHo + \max(SRo, STo)) / 2 \\ &= (19821 + \max(1180, 8883)) / 2 \\ &= 14352.46 \text{ psi} \end{aligned}$$

Average Flange Stress, Seating [SAa]:

$$\begin{aligned} &= (SHa + \max(SRa, STa)) / 2 \\ &= (25442 + \max(1515, 11403)) / 2 \\ &= 18423.30 \text{ psi} \end{aligned}$$

Bolt Stress, Operating [BSo]:

$$\begin{aligned} &= (Wm1 / Ab) \\ &= (53965 / 3.2320) \\ &= 16697.26 \text{ psi} \end{aligned}$$

Bolt Stress, Seating [BSa]:

$$\begin{aligned} &= (Wm2 / Ab) \\ &= (50647 / 3.2320) \\ &= 15670.59 \text{ psi} \end{aligned}$$

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Flg Calc [Int P] : Ch. Flange Flng: 4 5:58p Sep 18,2012

Stress Computation Results:	Operating		Gasket Seating	
	Actual	Allowed	Actual	Allowed
Longitudinal Hub	19821.	26700.	25443.	30000. psi
Radial Flange	1180.	17800.	1515.	20000. psi
Tangential Flange	8884.	17800.	11404.	20000. psi
Maximum Average	14352.	17800.	18423.	20000. psi
Bolting	16697.	25000.	15671.	25000. psi

Minimum Required Flange Thickness 1.684 in.
Estimated M.A.W.P. (Operating) 186.0 psig
Estimated Finished Weight of Flange at given Thk. 94.0 lbm
Estimated Unfinished Weight of Forging at given Thk 143.0 lbm

Flange Rigidity Based on Required Thickness [ASME]:

Flange Rigidity Index, Seating (rotation check) per APP. 2 [Js]:
= $52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^{(2)} * ho * Ki)$
= $52.14 * 10434.6/1.2026 * 12.000 * 0.081/(2.908 * 29400000 * 0.125^{(2)} * 1.510 * 0.300)$
= 0.724 (should be <= 1)

Flange Rigidity Index Operating (rotation check) per APP. 2 [J]:
= $52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^{(2)} * ho * Ki)$
= $52.14 * 8128.9/1.2026 * 12.000 * 0.081/(2.908 * 26000000 * 0.125^{(2)} * 1.510 * 0.300)$
= 0.638 (should be <= 1)

Flange Rigidity Based on Given Thickness [ASME]:

Flange Rigidity Index, Seating (rotation check) per APP. 2 [Js]:
= $52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^{(2)} * ho * Ki)$
= $52.14 * 10434.6/1.1893 * 12.000 * 0.081/(3.161 * 29400000 * 0.125^{(2)} * 1.510 * 0.300)$
= 0.674 (should be <= 1)

Flange Rigidity Index Operating (rotation check) per APP. 2 [J]:
= $52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^{(2)} * ho * Ki)$
= $52.14 * 8128.9/1.1893 * 12.000 * 0.081/(3.161 * 26000000 * 0.125^{(2)} * 1.510 * 0.300)$
= 0.593 (should be <= 1)

Minimum Design Metal Temperature Results:

Stress Ratio = 0.775 , Temperature Reduction per Fig. UCS 66.1 = 23 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -43 F

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Internal Pressure Calculations : Step: 5 5:58p Sep 18,2012

Element Thickness, Pressure, Diameter and Allowable Stress :

From	To	Int. Press + Liq. Hd psig	Nominal Thickness in.	Total Corr Allowance in.	Element Diameter in.	Allowable Stress(SE) psi
Left Head		150.658	0.18750	0.062500	18.5000	18800.0
Left Chann		150.657	0.18750	0.062500	18.5000	18800.0
Channel Fl		150.000	1.75000	0.062500	18.1250	17800.0
Shell		1400.63	0.75000	0.062500	18.5000	18800.0
Channel Fl		150.000	1.75000	0.062500	18.1250	17800.0
Channel		150.657	0.18750	0.062500	18.5000	18800.0
Right Head		150.658	0.18750	0.062500	18.5000	18800.0

Element Required Thickness and MAWP :

From	To	Design Pressure psig	M.A.W.P. Corroded psig	M.A.P. New & Cold psig	Minimum Thickness in.	Required Thickness in.
Left Head		150.000	217.957	369.279	0.16800	0.13544
Left Chann		150.000	254.778	408.719	0.18750	0.13639
Channel Fl		150.000	161.970	194.220	1.75000	1.68400
Shell		1400.00	1439.48	1675.98	0.75000	0.73170
Channel Fl		150.000	161.970	194.220	1.75000	1.68400
Channel		150.000	254.778	408.719	0.18750	0.13639
Right Head		150.000	217.957	369.279	0.16800	0.13544

Summary of Heat Exchanger Maximum Allowable Working Pressures :

Note: For ASME UHX designs, the following values include MAWPs that consider the tubesheet, tubes, tube/tubesheet joint etc. These values were determined by iteration. Review the tubesheet analysis report for more information.

Shell Side MAWP = 1439.484 psig
Shell Side MAPnc = 1675.978 psig
Channel Side MAWP = 161.970 psig
Channel Side MAPnc = 194.220 psig

Internal Pressure Calculation Results :

ASME Code, Section VIII, Division 1, 2010, 2011a

Elliptical Head From 10 To 20 SA-516 70 , UCS-66 Crv. B at 650 F

Left Head

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$= (P \cdot D_o \cdot K_{cor}) / (2 \cdot S \cdot E + 2 \cdot P \cdot (K_{cor} - 0.1)) \text{ per Appendix 1-4 (c)}$$

$$= (150.658 \cdot 18.5000 \cdot 0.991) / (2 \cdot 18800.00 \cdot 1.00 + 2 \cdot 150.658 \cdot (0.99 - 0.1))$$

$$= 0.0729 + 0.0625 = 0.1354 \text{ in.}$$

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

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Internal Pressure Calculations : Step: 5 5:58p Sep 18,2012

Less Operating Hydrostatic Head Pressure of 0.658 psig

$$\begin{aligned} &= (2 * S * E * t) / (K_{cor} * D_o - 2 * t * (K_{cor} - 0.1)) \text{ per Appendix 1-4 (c)} \\ &= (2 * 18800.00 * 1.00 * 0.1055) / (0.991 * 18.5000 - 2 * 0.1055 * (0.99 - 0.1)) \\ &= 218.615 - 0.658 = 217.957 \text{ psig} \end{aligned}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (2 * S * E * t) / (K * D_o - 2 * t * (K - 0.1)) \text{ per Appendix 1-4 (c)} \\ &= (2 * 20000.00 * 1.00 * 0.1680) / (1.000 * 18.5000 - 2 * 0.1680 * (1.000 - 0.1)) \\ &= 369.279 \text{ psig} \end{aligned}$$

Actual stress at given pressure and thickness, corroded [Sact]:

$$\begin{aligned} &= (P * (K_{cor} * D_o - 2 * t * (K_{cor} - 0.1))) / (2 * E * t) \\ &= (150.658 * (0.991 * 18.5000 - 2 * 0.1055 * (0.991 - 0.1))) / (2 * 1.00 * 0.1055) \\ &= 12955.980 \text{ psi} \end{aligned}$$

Straight Flange Required Thickness:

$$\begin{aligned} &= (P * R_o) / (S * E + 0.4 * P) + c \text{ per Appendix 1-1 (a)(1)} \\ &= (150.658 * 9.2500) / (18800.00 * 1.00 + 0.4 * 150.658) + 0.063 \\ &= 0.136 \text{ in.} \end{aligned}$$

Straight Flange Maximum Allowable Working Pressure:

Less Operating Hydrostatic Head Pressure of 0.658 psig

$$\begin{aligned} &= (S * E * t) / (R_o - 0.4 * t) \text{ per Appendix 1-1 (a)(1)} \\ &= (18800.00 * 1.00 * 0.1250) / (9.2500 - 0.4 * 0.1250) \\ &= 255.435 - 0.658 = 254.777 \text{ psig} \end{aligned}$$

Factor K, corroded condition [Kcor]:

$$\begin{aligned} &= (2 + (\text{Inside Diameter} / (2 * \text{Inside Head Depth}))^2) / 6 \\ &= (2 + (18.289 / (2 * 4.603))^2) / 6 \\ &= 0.990980 \end{aligned}$$

Percent Elong. per UCS-79, VIII-1-01-57 $(75 * t_{nom} / R_f) * (1 - R_f / R_o)$ 4.429 %

MDMT Calculations in the Knuckle Portion:

Govrn. thk, $t_g = 0.168$, $t_r = 0.074$, $c = 0.0625$ in., $E^* = 1.00$

Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.699$, Temp. Reduction = 30 F

Min Metal Temp. w/o impact per UCS-66 -20 F

Min Metal Temp. at Required thickness (UCS 66.1) -50 F

MDMT Calculations in the Head Straight Flange:

Govrn. thk, $t_g = 0.188$, $t_r = 0.075$, $c = 0.0625$ in., $E^* = 1.00$

Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.597$, Temp. Reduction = 41 F

Min Metal Temp. w/o impact per UCS-66 -20 F

Min Metal Temp. at Required thickness (UCS 66.1) -55 F

Cylindrical Shell From 20 To 30 SA-516 70, UCS-66 Crv. B at 650 F

Left Channel

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$= (P * R_o) / (S * E + 0.4 * P) \text{ per Appendix 1-1 (a)(1)}$$

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Internal Pressure Calculations : Step: 5 5:58p Sep 18,2012

$$= (150.657*9.2500)/(18800.00*1.00+0.4*150.657)$$
$$= 0.0739 + 0.0625 = 0.1364 \text{ in.}$$

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

Less Operating Hydrostatic Head Pressure of 0.657 psig

$$= (S*E*t)/(Ro-0.4*t) \text{ per Appendix 1-1 (a)(1)}$$
$$= (18800.00*1.00*0.1250)/(9.2500-0.4*0.1250)$$
$$= 255.435 - 0.657 = 254.778 \text{ psig}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$= (S*E*t)/(Ro-0.4*t) \text{ per Appendix 1-1 (a)(1)}$$
$$= (20000.00*1.00*0.1875)/(9.2500-0.4*0.1875)$$
$$= 408.719 \text{ psig}$$

Actual stress at given pressure and thickness, corroded [Sact]:

$$= (P*(Ro-0.4*t))/(E*t)$$
$$= (150.657*((9.2500-0.4*0.1250)))/(1.00*0.1250)$$
$$= 11088.338 \text{ psi}$$

Percent Elongation per UCS-79 $(50*t_{nom}/R_f)*(1-R_f/R_o)$ 1.024 %

Minimum Design Metal Temperature Results:

Govrn. thk, $t_g = 0.188$, $t_r = 0.075$, $c = 0.0625$ in., $E^* = 1.00$

Stress Ratio = $t_r * (E^*)/(t_g - c) = 0.597$, Temp. Reduction = 41 F

Min Metal Temp. w/o impact per UCS-66

-20 F

Min Metal Temp. at Required thickness (UCS 66.1)

-55 F

Cylindrical Shell From 40 To 50 SA-516 70, UCS-66 Crv. B at 650 F

Shell

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$= (P*Ro) / (S*E+0.4*P) \text{ per Appendix 1-1 (a)(1)}$$
$$= (1400.627*9.2500)/(18800.00*1.00+0.4*1400.627)$$
$$= 0.6692 + 0.0625 = 0.7317 \text{ in.}$$

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

Less Operating Hydrostatic Head Pressure of 0.627 psig

$$= (S*E*t)/(Ro-0.4*t) \text{ per Appendix 1-1 (a)(1)}$$
$$= (18800.00*1.00*0.6875)/(9.2500-0.4*0.6875)$$
$$= 1440.111 - 0.627 = 1439.484 \text{ psig}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$= (S*E*t)/(Ro-0.4*t) \text{ per Appendix 1-1 (a)(1)}$$
$$= (20000.00*1.00*0.7500)/(9.2500-0.4*0.7500)$$
$$= 1675.978 \text{ psig}$$

Actual stress at given pressure and thickness, corroded [Sact]:

$$= (P*(Ro-0.4*t))/(E*t)$$
$$= (1400.627*((9.2500-0.4*0.6875)))/(1.00*0.6875)$$
$$= 18284.555 \text{ psi}$$

Percent Elongation per UCS-79 $(50*t_{nom}/R_f)*(1-R_f/R_o)$ 4.225 %

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Internal Pressure Calculations : Step: 5 5:58p Sep 18,2012

Minimum Design Metal Temperature Results:

Govrn. thk, tg = 0.750 , tr = 0.647 , c = 0.0625 in. , E* = 1.00
Stress Ratio = tr * (E*)/(tg - c) = 0.941 , Temp. Reduction = 6 F

Min Metal Temp. w/o impact per UCS-66	16 F
Min Metal Temp. at Required thickness (UCS 66.1)	10 F
Min Metal Temp. w/o impact per UG-20(f)	-20 F

Cylindrical Shell From 60 To 70 SA-516 70 , UCS-66 Crv. B at 650 F

Channel

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:
= (P*Ro) / (S*E+0.4*P) per Appendix 1-1 (a)(1)
= (150.657*9.2500)/(18800.00*1.00+0.4*150.657)
= 0.0739 + 0.0625 = 0.1364 in.

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

Less Operating Hydrostatic Head Pressure of 0.657 psig

= (S*E*t)/(Ro-0.4*t) per Appendix 1-1 (a)(1)
= (18800.00*1.00*0.1250)/(9.2500-0.4*0.1250)
= 255.435 - 0.657 = 254.778 psig

Maximum Allowable Pressure, New and Cold [MAPNC]:

= (S*E*t)/(Ro-0.4*t) per Appendix 1-1 (a)(1)
= (20000.00*1.00*0.1875)/(9.2500-0.4*0.1875)
= 408.719 psig

Actual stress at given pressure and thickness, corroded [Sact]:

= (P*(Ro-0.4*t))/(E*t)
= (150.657*((9.2500-0.4*0.1250)))/(1.00*0.1250)
= 11088.338 psi

Percent Elongation per UCS-79 (50*tnom/Rf)*(1-Rf/Ro) 1.024 %

Minimum Design Metal Temperature Results:

Govrn. thk, tg = 0.188 , tr = 0.075 , c = 0.0625 in. , E* = 1.00
Stress Ratio = tr * (E*)/(tg - c) = 0.597 , Temp. Reduction = 41 F

Min Metal Temp. w/o impact per UCS-66	-20 F
Min Metal Temp. at Required thickness (UCS 66.1)	-55 F

Elliptical Head From 70 To 80 SA-516 70 , UCS-66 Crv. B at 650 F

Right Head

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:
= (P*Do*Kcor)/(2*S*E+2*P*(Kcor-0.1)) per Appendix 1-4 (c)
= (150.658*18.5000*0.991)/(2*18800.00*1.00+2*150.658*(0.99-0.1))
= 0.0729 + 0.0625 = 0.1354 in.

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Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

Less Operating Hydrostatic Head Pressure of 0.658 psig

$$\begin{aligned} &= (2 * S * E * t) / (K_{cor} * D_o - 2 * t * (K_{cor} - 0.1)) \text{ per Appendix 1-4 (c)} \\ &= (2 * 18800.00 * 1.00 * 0.1055) / (0.991 * 18.5000 - 2 * 0.1055 * (0.99 - 0.1)) \\ &= 218.615 - 0.658 = 217.957 \text{ psig} \end{aligned}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (2 * S * E * t) / (K * D_o - 2 * t * (K - 0.1)) \text{ per Appendix 1-4 (c)} \\ &= (2 * 20000.00 * 1.00 * 0.1680) / (1.000 * 18.5000 - 2 * 0.1680 * (1.000 - 0.1)) \\ &= 369.279 \text{ psig} \end{aligned}$$

Actual stress at given pressure and thickness, corroded [Sact]:

$$\begin{aligned} &= (P * (K_{cor} * D_o - 2 * t * (K_{cor} - 0.1))) / (2 * E * t) \\ &= (150.658 * (0.991 * 18.5000 - 2 * 0.1055 * (0.991 - 0.1))) / (2 * 1.00 * 0.1055) \\ &= 12955.980 \text{ psi} \end{aligned}$$

Straight Flange Required Thickness:

$$\begin{aligned} &= (P * R_o) / (S * E + 0.4 * P) + c \text{ per Appendix 1-1 (a)(1)} \\ &= (150.658 * 9.2500) / (18800.00 * 1.00 + 0.4 * 150.658) + 0.062 \\ &= 0.136 \text{ in.} \end{aligned}$$

Straight Flange Maximum Allowable Working Pressure:

Less Operating Hydrostatic Head Pressure of 0.658 psig

$$\begin{aligned} &= (S * E * t) / (R_o - 0.4 * t) \text{ per Appendix 1-1 (a)(1)} \\ &= (18800.00 * 1.00 * 0.1250) / (9.2500 - 0.4 * 0.1250) \\ &= 255.435 - 0.658 = 254.777 \text{ psig} \end{aligned}$$

Factor K, corroded condition [Kcor]:

$$\begin{aligned} &= (2 + (\text{Inside Diameter} / (2 * \text{Inside Head Depth}))^2) / 6 \\ &= (2 + (18.289 / (2 * 4.603))^2) / 6 \\ &= 0.990980 \end{aligned}$$

Percent Elong. per UCS-79, VIII-1-01-57 $(75 * t_{nom} / R_f) * (1 - R_f / R_o)$ 4.429 %

MDMT Calculations in the Knuckle Portion:

Govrn. thk, $t_g = 0.168$, $t_r = 0.074$, $c = 0.0625$ in., $E^* = 1.00$

Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.699$, Temp. Reduction = 30 F

Min Metal Temp. w/o impact per UCS-66 -20 F

Min Metal Temp. at Required thickness (UCS 66.1) -50 F

MDMT Calculations in the Head Straight Flange:

Govrn. thk, $t_g = 0.188$, $t_r = 0.075$, $c = 0.0625$ in., $E^* = 1.00$

Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.597$, Temp. Reduction = 41 F

Min Metal Temp. w/o impact per UCS-66 -20 F

Min Metal Temp. at Required thickness (UCS 66.1) -55 F

Note: Heads and Shells Exempted to -20F (-29C) by paragraph UG-20F

Hydrostatic Test Pressure Results:

Exchanger Shell Side Hydrostatic Test Pressures:

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Pressure per UG99b = 1.3 * M.A.W.P. * Sa/S 1871.329 psig
Pressure per UG99b[34] = 1.3 * Design Pres * Sa/S 1820.000 psig
Pressure per UG99c = 1.3 * M.A.P. - Head(Hyd) 2178.771 psig
Pressure per UG100 = 1.1 * M.A.W.P. * Sa/S 1583.432 psig
Pressure per PED = 1.43 * MAWP 2058.462 psig

Exchanger Channel Side Hydrostatic Test Pressures:

Pressure per UG99b = 1.3 * M.A.W.P. * Sa/S 210.561 psig
Pressure per UG99b[34] = 1.3 * Design Pres * Sa/S 195.000 psig
Pressure per UG99c = 1.3 * M.A.P. - Head(Hyd) 252.486 psig
Pressure per UG100 = 1.1 * M.A.W.P. * Sa/S 178.167 psig
Pressure per PED = 1.43 * MAWP 231.617 psig

UG-99(b) Note 34, Test Pressure Calculation [Shell Side]:

= Test Factor * Design Pressure * Stress Ratio
= 1.3 * 1400.000 * 1.000
= 1820.000 psig

UG-99(b) Note 34, Test Pressure Calculation [Channel Side]:

= Test Factor * Design Pressure * Stress Ratio
= 1.3 * 150.000 * 1.000
= 195.000 psig

Horizontal Test performed per: UG-99b (Note 34)

Please note that Nozzle, Shell, Head, Flange, etc MAWPs are all considered when determining the hydrotest pressure for those test types that are based on the MAWP of the vessel.

Stresses on Elements due to Hydrostatic Test Pressure:

From To	Stress	Allowable	Ratio	Pressure
Left Head	10597.3	26000.0	0.408	195.67
Left Channel	9574.7	26000.0	0.368	195.67
Shell	21726.6	26000.0	0.836	1820.67
Channel	9574.7	26000.0	0.368	195.67
Right Head	10597.3	26000.0	0.408	195.67

Elements Suitable for Internal Pressure.

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External Pressure Calculations : Step: 6 5:58p Sep 18,2012

External Pressure Calculation Results :

ASME Code, Section VIII, Division 1, 2010, 2011a

Elliptical Head From 10 to 20 Ext. Chart: CS-2 at 650 F

Left Head

Elastic Modulus from Chart: CS-2 at 650 F : 0.251E+08 psi

Results for Maximum Allowable External Pressure (MAEP):

Tca	OD	D/t	Factor A	B
0.105	18.50	175.36	0.0007920	8040.84

EMAP = $B / (K_0 * D / t) = 8040.8364 / (0.9000 * 175.3555) = 50.9494$ psig

*Check the requirements of UG-33(a)(1) using $P = 1.67 * \text{External Design pressure for this head.}$*

Material UNS Number: K02700

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= $((2 * S * E * t) / (K_{cor} * D_o - 2 * t * (K_{cor} - 0.1))) / 1.67$ per Appendix 1-4 (c)
= $((2 * 18800.00 * 1.00 * 0.1055) / (0.991 * 18.5000 - 2 * 0.1055 * (0.99 - 0.1))) / 1.67$
= 130.907 psig

Maximum Allowable External Pressure [MAEP]:

= min(MAEP, MAWP)
= min(50.95 , 130.9073)
= 50.949 psig

Cylindrical Shell From 20 to 30 Ext. Chart: CS-2 at 650 F

Left Channel

Elastic Modulus from Chart: CS-2 at 650 F : 0.251E+08 psi

Results for Maximum Allowable External Pressure (MAEP):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.125	18.50	11.26	148.00	0.6087	0.0012594	8959.65

EMAP = $(4 * B) / (3 * (D / t)) = (4 * 8959.6514) / (3 * 148.0000) = 80.7176$ psig

Results for Maximum Stiffened Length (Slen):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.125	18.50	11.26	148.00	0.6087	0.0012594	8959.65

EMAP = $(4 * B) / (3 * (D / t)) = (4 * 8959.6514) / (3 * 148.0000) = 80.7176$ psig

Cylindrical Shell From 40 to 50 Ext. Chart: CS-2 at 650 F

Shell

Elastic Modulus from Chart: CS-2 at 650 F : 0.251E+08 psi

Results for Maximum Allowable External Pressure (MAEP):

Tca	OD	SLEN	D/t	L/D	Factor A	B
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H&C Heat Transfer Sample

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External Pressure Calculations : Step: 6 5:58p Sep 18,2012

0.688 18.50 92.00 26.91 4.9732 0.0018108 9670.06
EMAP = (4*B)/(3*(D/t)) = (4*9670.0586)/(3*26.9091) = 479.1471 psig

Results for Required Thickness (Tca):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.124	18.50	92.00	149.23	4.9732	0.0001336	1678.88

EMAP = (4*B)/(3*(D/t)) = (4*1678.8820)/(3*149.2293) = 15.0005 psig

Results for Maximum Stiffened Length (Slen):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.688	18.50	2890.10	26.91	50.0000	0.0015359	9348.96

EMAP = (4*B)/(3*(D/t)) = (4*9348.9580)/(3*26.9091) = 463.2367 psig

Cylindrical Shell From 60 to 70 Ext. Chart: CS-2 at 650 F

Channel

Elastic Modulus from Chart: CS-2 at 650 F : 0.251E+08 psi

Results for Maximum Allowable External Pressure (MAEP):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.125	18.50	11.26	148.00	0.6087	0.0012594	8959.65

EMAP = (4*B)/(3*(D/t)) = (4*8959.6514)/(3*148.0000) = 80.7176 psig

Results for Maximum Stiffened Length (Slen):

Tca	OD	SLEN	D/t	L/D	Factor A	B
0.125	18.50	11.26	148.00	0.6087	0.0012594	8959.65

EMAP = (4*B)/(3*(D/t)) = (4*8959.6514)/(3*148.0000) = 80.7176 psig

Elliptical Head From 70 to 80 Ext. Chart: CS-2 at 650 F

Right Head

Elastic Modulus from Chart: CS-2 at 650 F : 0.251E+08 psi

Results for Maximum Allowable External Pressure (MAEP):

Tca	OD	D/t	Factor A	B
0.105	18.50	175.36	0.0007920	8040.84

EMAP = B/(K0*D/t) = 8040.8364/(0.9000 *175.3555) = 50.9494 psig

*Check the requirements of UG-33(a)(1) using P = 1.67 * External Design pressure for this head.*

Material UNS Number: K02700

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= ((2*S*E*t)/(Kcor*Do-2*t*(Kcor-0.1)))/1.67 per Appendix 1-4 (c)
= ((2*18800.00*1.00*0.1055)/(0.991*18.5000-2*0.1055*(0.99-0.1)))/1.67
= 130.907 psig

Maximum Allowable External Pressure [MAEP]:

= min(MAEP, MAWP)
= min(50.95 , 130.9073)
= 50.949 psig

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External Pressure Calculations : Step: 6 5:58p Sep 18,2012

External Pressure Calculations

From	To	Section Length ft.	Outside Diameter in.	Corroded Thickness in.	Factor A	Factor B psi
10	20	No Calc	18.5000	0.10550	0.00079204	8040.84
20	30	0.93844	18.5000	0.12500	0.0012594	8959.65
30	40	No Calc	...	1.68750	No Calc	No Calc
40	50	7.66700	18.5000	0.68750	0.0018108	9670.06
50	60	No Calc	...	1.68750	No Calc	No Calc
60	70	0.93844	18.5000	0.12500	0.0012594	8959.65
70	80	No Calc	18.5000	0.10550	0.00079204	8040.84

External Pressure Calculations

From	To	External Actual T. in.	External Required T. in.	External Des. Press. psig	External M.A.W.P. psig
10	20	0.16800	0.12500	...	50.9494
20	30	0.18750	No Calc	...	80.7176
30	40	1.75000	1.66400	...	No Calc
40	50	0.75000	0.18647	15.0000	479.147
50	60	1.75000	1.66400	...	No Calc
60	70	0.18750	No Calc	...	80.7176
70	80	0.16800	0.12500	...	50.9494
Minimum					50.949

External Pressure Calculations

From	To	Actual Len. Bet. Stiff. ft.	Allow. Len. Bet. Stiff. ft.	Ring Inertia Required in**4	Ring Inertia Available in**4
10	20	No Calc	No Calc	No Calc	No Calc
20	30	0.93844	No Calc	No Calc	No Calc
30	40	No Calc	No Calc	No Calc	No Calc
40	50	7.66700	240.842	No Calc	No Calc
50	60	No Calc	No Calc	No Calc	No Calc
60	70	0.93844	No Calc	No Calc	No Calc
70	80	No Calc	No Calc	No Calc	No Calc

Elements Suitable for External Pressure.

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Element and Detail Weights : Step: 7 5:58p Sep 18,2012

Element and Detail Weights

From	To	Element Metal Wgt. lb.	Element ID Volume in ³	Corroded Metal Wgt. lb.	Corroded ID Volume in ³	Extra due Misc % lb.
10	20	24.4461	1043.49	16.2974	1063.37	...
20	30	26.7051	2257.12	17.8642	2288.36	...
30	40	93.9645	748.044	91.0590	752.062	...
40	50	1088.94	13368.1	1001.71	13676.3	...
50	60	93.9645	748.044	91.0590	752.062	...
60	70	26.7051	2257.12	17.8642	2288.36	...
70	80	24.4461	1043.49	16.2974	1063.37	...
Total		1379	21465	1252	21883	0

For elements specified as shell side elements, the volume(s) shown above for those elements, reflects the displacement of the tubes.

Weight of Details

From	Type	Weight of Detail lb.	X Offset, Dtl. Cent. ft.	Y Offset, Dtl. Cent. ft.	Description
10	Liqd	37.6815	-0.12614	...	HEAD
20	Liqd	81.5071	0.77950	...	CHANNEL
20	Nozl	29.8733	0.36000	0.94271	T1/T2
40	Sadl	56.7760	1.59890	1.35417	Lft Sdl
40	Sadl	56.7760	6.40000	1.35417	Rgt Sdl
40	Liqd	482.735	3.83350	-0.013002	SHELL
40	Nozl	47.5039	0.50000	0.85417	S2
40	Nozl	140.290	7.29000	0.98438	S1
40	Nozl	29.1602	0.50000	0.80729	S3
60	Liqd	81.5071	0.36450	...	CHANNEL
60	Nozl	4.59655	0.50000	0.81000	T3 & T4
70	Liqd	37.6815	0.20944	...	HEAD
30	FTsh	199.263	0.32292	...	Tubesheet
30	Tube	862.075	4.23958	...	
30	RTsh	199.263	8.15625	...	

Total Weight of Each Detail Type

Total Weight of Saddles	113.6
Total Weight of Liquid	721.1
Total Weight of Nozzles	251.4
Total Weight of Exchanger Components	1260.6
Total Weight of Liquid in Tubes	173.2
Sum of the Detail Weights	2519.8 lb.

Weight Summation

Fabricated	Shop Test	Shipping	Erected	Empty	Operating
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Element and Detail Weights : Step: 7 5:58p Sep 18,2012

1379.2	3004.7	1379.2	3004.7	1379.2	3004.7
113.6	775.1	113.6	...	113.6	721.1
251.4	...	251.4
...	173.2
...
...
...	251.4	173.2
1260.6	...	1260.6
...	1260.6	...

3004.7	3953.0	3004.7	3004.7	3004.7	3899.0 lb.

Note: The shipping total has been modified because some items have been specified as being installed in the shop.

Weight Summary

Fabricated Wt.	- Bare Weight W/O Removable Internals	3004.7 lb.
Shop Test Wt.	- Fabricated Weight + Water (Full)	3953.0 lb.
Shipping Wt.	- Fab. Wt + Rem. Intls.+ Shipping App.	3004.7 lb.
Erected Wt.	- Fab. Wt + Rem. Intls.+ Insul. (etc)	3004.7 lb.
Ope. Wt. no Liq	- Fab. Wt + Intls. + Details + Wghts.	3004.7 lb.
Operating Wt.	- Empty Wt + Operating Liq. Uncorroded	3899.0 lb.
Oper. Wt. + CA	- Corr Wt. + Operating Liquid	3772.0 lb.
Field Test Wt.	- Empty Weight + Water (Full)	3953.0 lb.

Exchanger Tube Data

Volume of Exchanger tubes :	4795.2 in ³
Weight of Ope Liq in tubes :	173.2 lb.
Weight of Water in tubes :	173.2 lb.

Note: The Corroded Weight and thickness are used in the Horizontal Vessel Analysis (Ope Case) and Earthquake Load Calculations.

Outside Surface Areas of Elements

From	To	Surface Area in ²
10	20	430.779
20	30	508.429
30	40	372.954
40	50	5347.22
50	60	372.954
60	70	508.429
70	80	430.779

Total		7971.545 in ² [55.4 Square Feet]

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Nozzle Flange MAWP :

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Nozzle Flange MAWP Results :

Nozzle Description	----- Flange Rating		Temperature F	Class	Grade Group
	Operating psig	Ambient psig			
T1/T2	550.0	740.0	650	300	GR 1.1
S2	1650.0	2220.0	650	900	GR 1.1
S1	1650.0	2220.0	650	900	GR 1.1
S3	1650.0	2220.0	650	900	GR 1.1
T3 & T4	550.0	740.0	650	300	GR 1.1

Shellside Flange Rating

Lowest Flange Pressure Rating was (Ope)[ShellSide]: 1650.000 psig

Lowest Flange Pressure Rating was (Amb)[ShellSide]: 2220.000 psig

Channelside Flange Rating

Lowest Flange Pressure Rating was (Ope)[TubeSide]: 550.000 psig

Lowest Flange Pressure Rating was (Amb)[TubeSide]: 740.000 psig

Note: ANSI Ratings are per ANSI/ASME B16.5 2009 Edition

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Center of Gravity Calculation : Step: 9 5:58p Sep 18,2012

Shop/Field Installation Options :

Note : The CG is computed from the first Element From Node

Center of Gravity of Saddles	5.218 ft.
Center of Gravity of Liquid	5.099 ft.
Center of Gravity of Nozzles	5.503 ft.
Center of Gravity of Tubesheet(s)	5.052 ft.
Center of Gravity of Tubes	5.052 ft.
Center of Gravity of Bare Shell New and Cold	5.052 ft.
Center of Gravity of Bare Shell Corroded	5.052 ft.
Vessel CG in the Operating Condition	5.098 ft.
Vessel CG in the Fabricated (Shop/Empty) Condition	5.096 ft.

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ASME Horizontal Vessel Analysis: Stresses for the Left Saddle
(per ASME Sec. VIII Div. 2 based on the Zick method.)

Horizontal Vessel Stress Calculations : Operating Case

Note: Wear Pad Width (5.00) is less than $1.56\sqrt{rm*t}$
and less than 2a. The wear plate will be ignored.

Minimum Wear Plate Width to be considered in analysis [b1]:

$$= \min(b + 1.56\sqrt{ Rm * t }, 2a)$$

$$= \min(4.000 + 1.56\sqrt{ 8.9062 * 0.6875 }, 2 * 6.625)$$

$$= 7.8602 \text{ in.}$$

Input and Calculated Values:

Vessel Mean Radius	Rm	8.91	in.
Stiffened Vessel Length per 4.15.6	L	7.67	ft.
Distance from Saddle to Vessel tangent	a	6.62	in.
Saddle Width	b	4.00	in.
Saddle Bearing Angle	theta	120.00	degrees
Shell Allowable Stress used in Calculation		18800.00	psi
Head Allowable Stress used in Calculation		17800.00	psi
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Saddle Force Q, Operating Case		1696.94	lb.

Horizontal Vessel Analysis Results:	Actual	Allowable	

Long. Stress at Top of Midspan	8903.77	18800.00	psi
Long. Stress at Bottom of Midspan	9236.72	18800.00	psi
Long. Stress at Top of Saddles	9074.51	18800.00	psi
Long. Stress at Bottom of Saddles	9067.88	18800.00	psi
Tangential Shear in Shell	277.72	15040.00	psi
Circ. Stress at Horn of Saddle	253.77	23500.00	psi
Circ. Compressive Stress in Shell	23.87	18800.00	psi

Load Combination Results for Q + Wind or Seismic [Q]:

$$= \text{Saddle Load} + \text{Max}(Fwl, Fwt, Fsl, Fst)$$

$$= 1696 + \text{Max}(0, 0, 0, 0)$$

$$= 1696.9 \text{ lb.}$$

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	1753.71	lb.
Transverse Shear Load Saddle	0.00	lb.
Longitudinal Shear Load Saddle	0.00	lb.

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, $k = 0.1$

The Computed K values from Table 4.15.1:

$$K1 = 0.1066 \quad K2 = 1.1707 \quad K3 = 0.8799 \quad K4 = 0.4011$$

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

K5 = 0.7603 K6 = 0.0529 K7 = 0.0325 K8 = 0.3405
K9 = 0.2711 K10 = 0.0581 K1* = 0.1923

Note: Dimension a is greater than or equal to Rm / 2.

Moment per Equation 4.15.3 [M1]:

$$\begin{aligned} &= -Q*a [1 - (1 - a/L + (R^2 - h^2)/(2a*L))/(1 + (4h^2)/3L)] \\ &= -1696*0.55[1 - (1 - 0.55/7.67 + (0.742^2 - 0.000^2)/ \\ &\quad (2*0.55*7.67))/(1 + (4*0.00)/(3*7.67))] \\ &= -6.5 \text{ ft.lb.} \end{aligned}$$

Moment per Equation 4.15.4 [M2]:

$$\begin{aligned} &= Q*L/4(1 + 2(R^2 - h^2)/(L^2))/(1 + (4h^2)/(3L)) - 4a/L \\ &= 1696*7.7/4(1 + 2(0.742^2 - 0.000^2)/(7.67^2))/(1 + (4*0.000)/ \\ &\quad (3*7.667)) - 4*0.55/7.67 \\ &= 2376.7 \text{ ft.lb.} \end{aligned}$$

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$$\begin{aligned} &= P * Rm/(2t) - M2/(pi*Rm^2*t) \\ &= 1400.32 * 8.906/(2*0.688) - 28520.5/(pi*8.9^2*0.688) \\ &= 8903.77 \text{ psi} \end{aligned}$$

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$$\begin{aligned} &= P * Rm/(2t) + M2/(pi * Rm^2 * t) \\ &= 1400.32 * 8.906/(2 * 0.688) + 28520.5/(pi * 8.9^2 * 0.688) \\ &= 9236.72 \text{ psi} \end{aligned}$$

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$$\begin{aligned} &= P * Rm/(2t) - M1/(K1*pi*Rm^2*t) \\ &= 1400.32*8.906/(2*0.688) - 78.0/(0.1066*pi*8.9^2*0.688) \\ &= 9074.51 \text{ psi} \end{aligned}$$

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:

$$\begin{aligned} &= P * Rm/(2t) + M1/(K1* * pi * Rm^2 * t) \\ &= 1400.32*8.906/(2*0.688) + 78.0/(0.1923*pi*8.9^2*0.688) \\ &= 9067.88 \text{ psi} \end{aligned}$$

Maximum Shear Force in the Saddle (4.15.5) [T]:

$$\begin{aligned} &= Q(L - 2a)/(L + (4*h^2/3)) \\ &= 1696 (7.67 - 2 * 0.55)/(7.67 + (4 * 0.00/3)) \\ &= 1452.6 \text{ lb.} \end{aligned}$$

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$$\begin{aligned} &= K2 * T / (Rm * t) \\ &= 1.1707 * 1452.55/(8.9062 * 0.6875) \\ &= 277.72 \text{ psi} \end{aligned}$$

Decay Length (4.15.22) [x1,x2]:

$$\begin{aligned} &= 0.78 * \text{sqrt}(Rm * t) \\ &= 0.78 * \text{sqrt}(8.906 * 0.688) \\ &= 1.930 \text{ in.} \end{aligned}$$

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

$$\begin{aligned} &= -K5 * Q * k / (t * (b + X1 + X2)) \\ &= -0.7603 * 1696 * 0.1/(0.688 * (4.00 + 1.93 + 1.93)) \\ &= -23.87 \text{ psi} \end{aligned}$$

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

Circ. Comp. Stress at Horn of Saddle, L>=8Rm (4.15.24) [sigma7]:

$$= -Q/(4*t*(b+X1+X2)) - 3*K7*Q/(2*t^2)$$

$$= -1696/(4*0.688*(4.000+1.930+1.930)) - 3*0.0325*1696/(2*0.688^2)$$

$$= -253.77 \text{ psi}$$

Effective reinforcing plate width (4.15.1) [B1]:

$$= \min(b + 1.56 * \text{sqrt}(Rm * t), 2a)$$

$$= \min(4.00 + 1.56 * \text{sqrt}(8.906 * 0.688), 2 * 6.625)$$

$$= 7.86 \text{ in.}$$

Free Un-Restrained Thermal Expansion between the Saddles [Exp]:

$$= \text{Alpha} * Ls * (\text{Design Temperature} - \text{Ambient Temperature})$$

$$= 0.750E-05 * 57.613 * (650.0 - 70.0)$$

$$= 0.251 \text{ in.}$$

Results for Vessel Ribs, Web and Base:

Baseplate Length	Bplen	16.4000	in.
Baseplate Thickness	Bpthk	0.2500	in.
Baseplate Width	Bpwid	4.0000	in.
Number of Ribs (inc. outside ribs)	Nribs	2	
Rib Thickness	Ribtk	0.2500	in.
Web Thickness	Webtk	0.2500	in.
Web Location	Webloc	Side	

Moment of Inertia of Saddle - Lateral Direction

	Y	A	AY	Io
Shell	0.3438	6.0397	2.0761	0.9516
Wearplate	0.8125	1.2500	1.0156	0.8317
Web	12.3438	5.7031	70.3979	1116.3051
BasePlate	23.8750	1.0000	23.8750	570.0208
Totals	37.3750	13.9928	97.3647	1688.1091

$$\text{Value } C1 = \text{Sumof}(Ay) / \text{Sumof}(A) = 6.9582 \text{ in.}$$

$$\text{Value } I = \text{Sumof}(Io) - C1 * \text{Sumof}(Ay) = 1010.6252 \text{ in}^{*4}$$

$$\text{Value } As = \text{Sumof}(A) - A_{\text{shell}} = 7.9531 \text{ in}^2$$

$$K1 = (1 + \text{Cos}(\text{beta}) - .5 * \text{Sin}(\text{beta})^2) / (\text{pi} - \text{beta} + \text{Sin}(\text{beta}) * \text{Cos}(\text{beta})) = 0.2035$$

$$Fh = K1 * Q = 0.2035 * 1696.937 = 345.3636 \text{ lb.}$$

$$\text{Tension Stress, } St = (Fh / As) = 43.4249 \text{ psi}$$

$$\text{Allowed Stress, } Sa = 0.6 * \text{Yield Str} = 21600.0000 \text{ psi}$$

$$d = B - R * \text{Sin}(\text{theta}) / \text{theta} = 25.4814 \text{ in.}$$

$$\text{Bending Moment, } M = Fh * d = 733.3613 \text{ ft.lb.}$$

$$\text{Bending Stress, } Sb = (M * C1 / I) = 60.5908 \text{ psi}$$

$$\text{Allowed Stress, } Sa = 2/3 * \text{Yield Str} = 24000.0000 \text{ psi}$$

Minimum Thickness of Baseplate per Moss :

$$= (3 * (Q + \text{Saddle_Wt}) * \text{BasePlateWidth} / (2 * \text{BasePlateLength} * \text{AllStress}))^{1/2}$$

$$= (3 * (1696 + 56) * 4.00 / (2 * 16.400 * 24000.000))^{1/2}$$

$$= 0.164 \text{ in.}$$

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

Calculation of Axial Load, Intermediate Values and Compressive Stress

Effective Baseplate Length [e]:

$$= (Bpln - Clearance) / (Nrbs - 1)$$
$$= (16.4000 - 1.0) / (2 - 1) = 15.4000 \text{ in.}$$

Baseplate Pressure Area [Ap]:

$$= e * Bpwid / 2$$
$$= 15.4000 * 4.0000 / 2 = 30.8000 \text{ in}^2$$

Axial Load [P]:

$$= Ap * Bp$$
$$= 30.8 * 25.87 = 796.7 \text{ lb.}$$

Area of the Rib and Web [Ar]:

$$= (Bpwid - Clearance - Webtk) * Ribtk + e/2 * Webtk$$
$$= (4.000 - 1.0 - 0.250) * 0.250 + 15.4000/2 * 0.250$$
$$= 2.612 \text{ in}^2$$

Compressive Stress [Sc]:

$$= P/Ar$$
$$= 796.7/2.6125 = 304.9695 \text{ psi}$$

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	Y	A	AY	Ay ²	Io
Rib	1.6250	0.8125	1.3203	0.9040	0.8932
Web	0.1250	1.9250	0.2406	0.3816	0.0201
Values	0.5702	2.7375	1.5609	1.2855	0.9133

Bending Moment [Rm]:

$$= Fl / (2 * Bplen) * e * r1 / 2$$
$$= 0.0 / (2 * 16.40) * 15.400 * 27.94 / 2$$
$$= 0.000 \text{ ft.lb.}$$

KL/R < Cc (30.4524 < 126.0993) per AISC E2-1

$$Sca = (1 - (Klr)^2 / (2 * Cc^2)) * Fy / (5/3 + 3 * (Klr) / (8 * Cc) - (Klr^3) / (8 * Cc^3))$$

$$Sca = (1 - (30.45)^2 / (2 * 126.10^2)) * 36000 /$$
$$(5/3 + 3 * (30.45) / (8 * 126.10) - (30.45^3) / (8 * 126.10^3))$$

$$Sca = 19909.37 \text{ psi}$$

AISC Unity Check on Outside Ribs (must be <= 1.0)

$$\text{Check} = Sc/Sca + (Rm/Z)/Sba$$

$$\text{Check} = 304.97/19909.37 + (0.00/0.905) / 24000.00$$

$$\text{Check} = 0.02$$

ASME Horizontal Vessel Analysis: Stresses for the Right Saddle
(per ASME Sec. VIII Div. 2 based on the Zick method.)

Note: Wear Pad Width (5.00) is less than 1.56*sqrt(rm*t)
and less than 2a. The wear plate will be ignored.

Minimum Wear Plate Width to be considered in analysis [b1]:

$$= \min(b + 1.56 * \sqrt{ Rm * t }, 2a)$$
$$= \min(4.000 + 1.56 * \sqrt{ 8.9062 * 0.6875 }, 2 * 6.625)$$
$$= 7.8602 \text{ in.}$$

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

Input and Calculated Values:

Vessel Mean Radius	Rm	8.91	in.
Stiffened Vessel Length per 4.15.6	L	7.67	ft.
Distance from Saddle to Vessel tangent	a	6.62	in.
Saddle Width	b	4.00	in.
Saddle Bearing Angle	theta	120.00	degrees
Shell Allowable Stress used in Calculation		18800.00	psi
Head Allowable Stress used in Calculation		17800.00	psi
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Saddle Force Q, Operating Case		1961.51	lb.

Horizontal Vessel Analysis Results: Actual Allowable

	Actual	Allowable	
Long. Stress at Top of Midspan	8877.81	18800.00	psi
Long. Stress at Bottom of Midspan	9262.67	18800.00	psi
Long. Stress at Top of Saddles	9075.18	18800.00	psi
Long. Stress at Bottom of Saddles	9067.51	18800.00	psi
Tangential Shear in Shell	321.02	15040.00	psi
Circ. Stress at Horn of Saddle	293.34	23500.00	psi
Circ. Compressive Stress in Shell	27.60	18800.00	psi

Load Combination Results for Q + Wind or Seismic [Q]:

= Saddle Load + Max(Fwl, Fwt, Fsl, Fst)
= 1961 + Max(0 , 0 , 0 , 0)
= 1961.5 lb.

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	2018.29	lb.
Transverse Shear Load Saddle	0.00	lb.
Longitudinal Shear Load Saddle	0.00	lb.

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, k = 0.1

The Computed K values from Table 4.15.1:

K1 = 0.1066	K2 = 1.1707	K3 = 0.8799	K4 = 0.4011
K5 = 0.7603	K6 = 0.0529	K7 = 0.0325	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	

Note: Dimension a is greater than or equal to Rm / 2.

Moment per Equation 4.15.3 [M1]:

= -Q*a [1 - (1- a/L + (R²-h²)/(2a*L))/(1+(4h²)/3L)]
= -1961*0.55[1-(1-0.55/7.67+(0.742²-0.000²)/
(2*0.55*7.67))/(1+(4*0.00)/(3*7.67))]
= -7.5 ft.lb.

Moment per Equation 4.15.4 [M2]:

= Q*L/4(1+2(R²-h²)/(L²))/(1+(4h²)/(3L))-4a/L

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

$$= 1961 * 7.7 / 4 (1 + 2(0.742^2 - 0.000^2) / (7.67^2)) / (1 + (4 * 0.000) / (3 * 7.667)) - 4 * 0.55 / 7.67$$
$$= 2747.3 \text{ ft.lb.}$$

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$$= P * Rm / (2t) - M2 / (\pi * Rm^2 * t)$$
$$= 1400.32 * 8.906 / (2 * 0.688) - 32967.2 / (\pi * 8.9^2 * 0.688)$$
$$= 8877.81 \text{ psi}$$

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$$= P * Rm / (2t) + M2 / (\pi * Rm^2 * t)$$
$$= 1400.32 * 8.906 / (2 * 0.688) + 32967.2 / (\pi * 8.9^2 * 0.688)$$
$$= 9262.67 \text{ psi}$$

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$$= P * Rm / (2t) - M1 / (K1 * \pi * Rm^2 * t)$$
$$= 1400.32 * 8.906 / (2 * 0.688) - 90.2 / (0.1066 * \pi * 8.9^2 * 0.688)$$
$$= 9075.18 \text{ psi}$$

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:

$$= P * Rm / (2t) + M1 / (K1 * \pi * Rm^2 * t)$$
$$= 1400.32 * 8.906 / (2 * 0.688) + 90.2 / (0.1923 * \pi * 8.9^2 * 0.688)$$
$$= 9067.51 \text{ psi}$$

Maximum Shear Force in the Saddle (4.15.5) [T]:

$$= Q(L - 2a) / (L + (4 * h^2 / 3))$$
$$= 1961 (7.67 - 2 * 0.55) / (7.67 + (4 * 0.00 / 3))$$
$$= 1679.0 \text{ lb.}$$

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$$= K2 * T / (Rm * t)$$
$$= 1.1707 * 1679.02 / (8.9062 * 0.6875)$$
$$= 321.02 \text{ psi}$$

Decay Length (4.15.22) [x1,x2]:

$$= 0.78 * \text{sqrt}(Rm * t)$$
$$= 0.78 * \text{sqrt}(8.906 * 0.688)$$
$$= 1.930 \text{ in.}$$

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

$$= -K5 * Q * k / (t * (b + X1 + X2))$$
$$= -0.7603 * 1961 * 0.1 / (0.688 * (4.00 + 1.93 + 1.93))$$
$$= -27.60 \text{ psi}$$

Circ. Comp. Stress at Horn of Saddle, L>=8Rm (4.15.24) [sigma7]:

$$= -Q / (4 * t * (b + X1 + X2)) - 3 * K7 * Q / (2 * t^2)$$
$$= -1961 / (4 * 0.688 * (4.000 + 1.930 + 1.930)) -$$
$$3 * 0.0325 * 1961 / (2 * 0.688^2)$$
$$= -293.34 \text{ psi}$$

Effective reinforcing plate width (4.15.1) [B1]:

$$= \text{min}(b + 1.56 * \text{sqrt}(Rm * t), 2a)$$
$$= \text{min}(4.00 + 1.56 * \text{sqrt}(8.906 * 0.688), 2 * 6.625)$$
$$= 7.86 \text{ in.}$$

Results for Vessel Ribs, Web and Base

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Horizontal Vessel Analysis (Ope.) : Step: 10 5:58p Sep 18,2012

Baseplate Length	Bplen	16.4000	in.
Baseplate Thickness	Bpthk	0.2500	in.
Baseplate Width	Bpwid	4.0000	in.
Number of Ribs (inc. outside ribs)	Nribs	2	
Rib Thickness	Ribtk	0.2500	in.
Web Thickness	Webtk	0.2500	in.
Web Location	Webloc	Side	

Moment of Inertia of Saddle - Lateral Direction

	Y	A	AY	Io
Shell	0.3438	6.0397	2.0761	0.9516
Wearplate	0.8125	1.2500	1.0156	0.8317
Web	12.3438	5.7031	70.3979	1116.3051
BasePlate	23.8750	1.0000	23.8750	570.0208
Totals	37.3750	13.9928	97.3647	1688.1091

Value C1 = Sumof(Ay)/Sumof(A) = 6.9582 in.
Value I = Sumof(Io) - C1*Sumof(Ay) = 1010.6252 in**4
Value As = Sumof(A) - Ashell = 7.9531 in²

$K1 = (1 + \cos(\beta) - 0.5 \sin(\beta)^2) / (\pi - \beta + \sin(\beta) \cos(\beta)) = 0.2035$

$Fh = K1 * Q = 0.2035 * 1961.509 = 399.2097 \text{ lb.}$

Tension Stress, St = (Fh/As) = 50.1953 psi
Allowed Stress, Sa = 0.6 * Yield Str = 21600.0000 psi

d = B - R * Sin(theta) / theta = 25.4814 in.
Bending Moment, M = Fh * d = 847.7009 ft.lb.

Bending Stress, Sb = (M * C1 / I) = 70.0376 psi
Allowed Stress, Sa = 2/3 * Yield Str = 24000.0000 psi

Minimum Thickness of Baseplate per Moss :

= (3 * (Q + Saddle_Wt) * BasePlateWidth / (2 * BasePlateLength * AllStress))^{1/2}
= (3 * (1961 + 56) * 4.00 / (2 * 16.400 * 24000.000))^{1/2}
= 0.175 in.

Calculation of Axial Load, Intermediate Values and Compressive Stress

Effective Baseplate Length [e]:

= (Bplen - Clearance) / (Nribs - 1)
= (16.4000 - 1.0) / (2 - 1) = 15.4000 in.

Baseplate Pressure Area [Ap]:

= e * Bpwid / 2
= 15.4000 * 4.0000 / 2 = 30.8000 in²

Axial Load [P]:

= Ap * Bp
= 30.8 * 29.90 = 921.0 lb.

Area of the Rib and Web [Ar]:

= (Bpwid - Clearance - Webtk) * Ribtk + e/2 * Webtk
= (4.000 - 1.0 - 0.250) * 0.250 + 15.4000/2 * 0.250

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= 2.612 in²

Compressive Stress [Sc]:

= P/Ar

= 921.0/2.6125 = 352.5177 psi

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	Y	A	AY	Ay ²	Io
Rib	1.6250	0.8125	1.3203	0.9040	0.8932
Web	0.1250	1.9250	0.2406	0.3816	0.0201
Values	0.5702	2.7375	1.5609	1.2855	0.9133

Bending Moment [Rm]:

= Fl / (2 * Bplen) * e * rl / 2

= 0.0 / (2 * 16.40) * 15.400 * 27.94 / 2

= 0.000 ft.lb.

KL/R < Cc (30.4524 < 126.0993) per AISC E2-1

Sca = (1 - (Klr)² / (2 * Cc²)) * Fy / (5/3 + 3 * (Klr) / (8 * Cc) - (Klr³) / (8 * Cc³))

Sca = (1 - (30.45)² / (2 * 126.10²)) * 36000 / (5/3 + 3 * (30.45) / (8 * 126.10) - (30.45³) / (8 * 126.10³))

Sca = 19909.37 psi

AISC Unity Check on Outside Ribs (must be <= 1.0)

Check = Sc/Sca + (Rm/Z)/Sba

Check = 352.52/19909.37 + (0.00/0.905)/24000.00

Check = 0.02

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Horizontal Vessel Analysis (Test) : Step: 11 5:58p Sep 18,2012

ASME Horizontal Vessel Analysis: Stresses for the Left Saddle
(per ASME Sec. VIII Div. 2 based on the Zick method.)

Horizontal Vessel Stress Calculations : Test Case

Note: Wear Pad Width (5.00) is less than $1.56 \cdot \sqrt{r_m \cdot t}$
and less than $2a$. The wear plate will be ignored.

Minimum Wear Plate Width to be considered in analysis [b1]:

$$\begin{aligned} &= \min(b + 1.56 \cdot \sqrt{ R_m \cdot t }, 2a) \\ &= \min(4.000 + 1.56 \cdot \sqrt{ 8.8750 \cdot 0.7500 }, 2 \cdot 6.625) \\ &= 8.0248 \text{ in.} \end{aligned}$$

Input and Calculated Values:

Vessel Mean Radius	Rm	8.88	in.
Stiffened Vessel Length per 4.15.6	L	7.67	ft.
Distance from Saddle to Vessel tangent	a	6.62	in.
Saddle Width	b	4.00	in.
Saddle Bearing Angle	theta	120.00	degrees
Shell Allowable Stress used in Calculation		20000.00	psi
Head Allowable Stress used in Calculation		20000.00	psi
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Saddle Force Q, Test Case, no Ext. Forces		1800.76	lb.

Horizontal Vessel Analysis Results:	Actual	Allowable	
Long. Stress at Top of Midspan	10607.18	20000.00	psi
Long. Stress at Bottom of Midspan	10933.28	20000.00	psi
Long. Stress at Top of Saddles	10774.69	20000.00	psi
Long. Stress at Bottom of Saddles	10767.76	20000.00	psi
Tangential Shear in Shell	271.10	16000.00	psi
Circ. Stress at Horn of Saddle	232.08	25000.00	psi
Circ. Compressive Stress in Shell	22.75	20000.00	psi

Load Combination Results for Q + Wind or Seismic [Q]:

$$\begin{aligned} &= \text{Saddle Load} + \text{Max}(F_{wl}, F_{wt}, F_{sl}, F_{st}) \\ &= 1800 + \text{Max}(0, 0, 0, 0) \\ &= 1800.8 \text{ lb.} \end{aligned}$$

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	1857.54	lb.
Transverse Shear Load Saddle	0.00	lb.
Longitudinal Shear Load Saddle	0.00	lb.

Hydrostatic Test Pressure at center of Vessel: 1820.320 psig

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, $k = 0.1$

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Horizontal Vessel Analysis (Test) : Step: 11 5:58p Sep 18,2012

The Computed K values from Table 4.15.1:

K1 = 0.1066 K2 = 1.1707 K3 = 0.8799 K4 = 0.4011
K5 = 0.7603 K6 = 0.0529 K7 = 0.0328 K8 = 0.3405
K9 = 0.2711 K10 = 0.0581 K1* = 0.1923

Note: Dimension a is greater than or equal to Rm / 2.

Moment per Equation 4.15.3 [M1]:

$$\begin{aligned} &= -Q*a [1 - (1 - a/L + (R^2 - h^2)/(2a*L))/(1 + (4h^2)/3L)] \\ &= -1800*0.55[1 - (1 - 0.55/7.67 + (0.740^2 - 0.000^2)/ \\ &\quad (2*0.55*7.67))/(1 + (4*0.00)/(3*7.67))] \\ &= -7.4 \text{ ft.lb.} \end{aligned}$$

Moment per Equation 4.15.4 [M2]:

$$\begin{aligned} &= Q*L/4(1 + 2(R^2 - h^2)/(L^2))/(1 + (4h^2)/(3L)) - 4a/L \\ &= 1800*7.7/4(1 + 2(0.740^2 - 0.000^2)/(7.67^2))/(1 + (4*0.000)/ \\ &\quad (3*7.667)) - 4*0.55/7.67 \\ &= 2521.7 \text{ ft.lb.} \end{aligned}$$

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$$\begin{aligned} &= P * Rm/(2t) - M2/(pi*Rm^2*t) \\ &= 1820.32 * 8.875/(2*0.750) - 30260.1/(pi*8.9^2*0.750) \\ &= 10607.18 \text{ psi} \end{aligned}$$

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$$\begin{aligned} &= P * Rm/(2t) + M2/(pi * Rm^2 * t) \\ &= 1820.32 * 8.875/(2 * 0.750) + 30260.1/(pi * 8.9^2 * 0.750) \\ &= 10933.28 \text{ psi} \end{aligned}$$

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$$\begin{aligned} &= P * Rm/(2t) - M1/(K1*pi*Rm^2*t) \\ &= 1820.32*8.875/(2*0.750) - 88.2/(0.1066*pi*8.9^2*0.750) \\ &= 10774.69 \text{ psi} \end{aligned}$$

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:

$$\begin{aligned} &= P * Rm/(2t) + M1/(K1* * pi * Rm^2 * t) \\ &= 1820.32*8.875/(2*0.750) + 88.2/(0.1923*pi*8.9^2*0.750) \\ &= 10767.76 \text{ psi} \end{aligned}$$

Maximum Shear Force in the Saddle (4.15.5) [T]:

$$\begin{aligned} &= Q(L - 2a)/(L + (4*h^2/3)) \\ &= 1800 (7.67 - 2 * 0.55)/(7.67 + (4 * 0.00/3)) \\ &= 1541.4 \text{ lb.} \end{aligned}$$

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$$\begin{aligned} &= K2 * T / (Rm * t) \\ &= 1.1707 * 1541.43/(8.8750 * 0.7500) \\ &= 271.10 \text{ psi} \end{aligned}$$

Decay Length (4.15.22) [x1,x2]:

$$\begin{aligned} &= 0.78 * \text{sqrt}(Rm * t) \\ &= 0.78 * \text{sqrt}(8.875 * 0.750) \\ &= 2.012 \text{ in.} \end{aligned}$$

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

$$\begin{aligned} &= -K5 * Q * k / (t * (b + X1 + X2)) \\ &= -0.7603 * 1800 * 0.1/(0.750 * (4.00 + 2.01 + 2.01)) \end{aligned}$$

= -22.75 psi

Circ. Comp. Stress at Horn of Saddle, L>=8Rm (4.15.24) [sigma7]:

= -Q/(4*t*(b+X1+X2)) - 3*K7*Q/(2*t^2)
= -1800/(4*0.750 *(4.000 +2.012 +2.012)) -
3*0.0328 *1800/(2*0.750^2)
= -232.08 psi

Effective reinforcing plate width (4.15.1) [B1]:

= min(b + 1.56 * sqrt(Rm * t), 2a)
= min(4.00 + 1.56 * sqrt(8.875 * 0.750), 2 * 6.625)
= 8.02 in.

Results for Vessel Ribs, Web and Base:

Table with 4 columns: Parameter, Bplen, Bpthk, Bpwid, Nr ribs, Ribtk, Webtk, Webloc. Rows include Baseplate Length, Baseplate Thickness, Baseplate Width, Number of Ribs, Rib Thickness, Web Thickness, and Web Location.

Moment of Inertia of Saddle - Lateral Direction

Table with 5 columns: Component, Y, A, AY, Io. Rows include Shell, Wearplate, Web, BasePlate, and Totals.

Value C1 = Sumof(Ay)/Sumof(A) = 6.6841 in.
Value I = Sumof(Io) - C1*Sumof(Ay) = 1034.3906 in**4
Value As = Sumof(A) - Ashell = 7.9375 in^2

K1 = (1+Cos(beta)-.5*Sin(beta)^2)/(pi-beta+Sin(beta)*Cos(beta)) = 0.2035

Fh = K1 * Q = 0.2035 * 1800.763 = 366.4943 lb.

Tension Stress, St = (Fh/As) = 46.1725 psi
Allowed Stress, Sa = 0.6 * Yield Str = 21600.0000 psi

d = B - R*Sin(theta) / theta = 25.4706 in.
Bending Moment, M = Fh * d = 777.9012 ft.lb.

Bending Stress, Sb = (M * C1 / I) = 60.3202 psi
Allowed Stress, Sa = 2/3 * Yield Str = 24000.0000 psi

Minimum Thickness of Baseplate per Moss :

= (3 * (Q + Saddle_Wt) * BasePlateWidth / (2 * BasePlateLength *
AllStress))^1/2
= (3 * (1800 + 56) * 4.00/(2 * 16.400 * 24000.000))^1/2
= 0.168 in.

Calculation of Axial Load, Intermediate Values and Compressive Stress

Effective Baseplate Length [e]:

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$$= (B_{plen} - Clearance) / (N_{ribs} - 1)$$
$$= (16.4000 - 1.0) / (2 - 1) = 15.4000 \text{ in.}$$

Baseplate Pressure Area [Ap]:

$$= e * B_{pwid} / 2$$
$$= 15.4000 * 4.0000 / 2 = 30.8000 \text{ in}^2$$

Axial Load [P]:

$$= A_p * B_p$$
$$= 30.8 * 27.45 = 845.5 \text{ lb.}$$

Area of the Rib and Web [Ar]:

$$= (B_{pwid} - Clearance - Web_{tk}) * Rib_{tk} + e / 2 * Web_{tk}$$
$$= (4.000 - 1.0 - 0.250) * 0.250 + 15.4000 / 2 * 0.250$$
$$= 2.612 \text{ in}^2$$

Compressive Stress [Sc]:

$$= P / A_r$$
$$= 845.5 / 2.6125 = 323.6288 \text{ psi}$$

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	Y	A	AY	Ay ²	Io
Rib	1.6250	0.8125	1.3203	0.9040	0.8932
Web	0.1250	1.9250	0.2406	0.3816	0.0201
Values	0.5702	2.7375	1.5609	1.2855	0.9133

Bending Moment [Rm]:

$$= F_l / (2 * B_{plen}) * e * r_l / 2$$
$$= 0.0 / (2 * 16.40) * 15.400 * 27.88 / 2$$
$$= 0.000 \text{ ft.lb.}$$

KL/R < Cc (30.3843 < 126.0993) per AISC E2-1

$$S_{ca} = (1 - (K_{lr})^2 / (2 * C_c^2)) * F_y / (5/3 + 3 * (K_{lr}) / (8 * C_c) - (K_{lr}^3) / (8 * C_c^3))$$

$$S_{ca} = (1 - (30.38)^2 / (2 * 126.10^2)) * 36000 / (5/3 + 3 * (30.38) / (8 * 126.10) - (30.38^3) / (8 * 126.10^3))$$

$$S_{ca} = 19914.21 \text{ psi}$$

AISC Unity Check on Outside Ribs (must be <= 1.0)

$$\text{Check} = S_c / S_{ca} + (R_m / Z) / S_{ba}$$

$$\text{Check} = 323.63 / 19914.21 + (0.00 / 0.905) / 24000.00$$

$$\text{Check} = 0.02$$

ASME Horizontal Vessel Analysis: Stresses for the Right Saddle
(per ASME Sec. VIII Div. 2 based on the Zick method.)

Note: Wear Pad Width (5.00) is less than $1.56 * \sqrt{r_m * t}$
and less than 2a. The wear plate will be ignored.

Minimum Wear Plate Width to be considered in analysis [b1]:

$$= \min(b + 1.56 * \sqrt{ R_m * t }, 2a)$$
$$= \min(4.000 + 1.56 * \sqrt{ 8.8750 * 0.7500 }, 2 * 6.625)$$
$$= 8.0248 \text{ in.}$$

Input and Calculated Values:

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Horizontal Vessel Analysis (Test) : Step: 11 5:58p Sep 18,2012

Vessel Mean Radius	Rm	8.88	in.
Stiffened Vessel Length per 4.15.6	L	7.67	ft.
Distance from Saddle to Vessel tangent	a	6.62	in.
Saddle Width	b	4.00	in.
Saddle Bearing Angle	theta	120.00	degrees
Shell Allowable Stress used in Calculation		20000.00	psi
Head Allowable Stress used in Calculation		20000.00	psi
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Saddle Force Q, Test Case, no Ext. Forces		2038.73	lb.

Horizontal Vessel Analysis Results:	Actual	Allowable	
Long. Stress at Top of Midspan	10585.63	20000.00	psi
Long. Stress at Bottom of Midspan	10954.83	20000.00	psi
Long. Stress at Top of Saddles	10775.28	20000.00	psi
Long. Stress at Bottom of Saddles	10767.43	20000.00	psi
Tangential Shear in Shell	306.93	16000.00	psi
Circ. Stress at Horn of Saddle	262.75	25000.00	psi
Circ. Compressive Stress in Shell	25.75	20000.00	psi

Load Combination Results for Q + Wind or Seismic [Q]:
= Saddle Load + Max(Fwl, Fwt, Fsl, Fst)
= 2038 + Max(0 , 0 , 0 , 0)
= 2038.7 lb.

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	2095.50	lb.
Transverse Shear Load Saddle	0.00	lb.
Longitudinal Shear Load Saddle	0.00	lb.

Hydrostatic Test Pressure at center of Vessel: 1820.320 psig

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, k = 0.1

The Computed K values from Table 4.15.1:

K1 = 0.1066	K2 = 1.1707	K3 = 0.8799	K4 = 0.4011
K5 = 0.7603	K6 = 0.0529	K7 = 0.0328	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	

Note: Dimension a is greater than or equal to Rm / 2.

Moment per Equation 4.15.3 [M1]:

$$\begin{aligned} &= -Q*a [1 - (1 - a/L + (R^2 - h^2)/(2a*L))/(1 + (4h^2)/3L)] \\ &= -2038*0.55[1 - (1 - 0.55/7.67 + (0.740^2 - 0.000^2)/ \\ &\quad (2*0.55*7.67))/(1 + (4*0.00)/(3*7.67))] \\ &= -8.3 \text{ ft.lb.} \end{aligned}$$

Moment per Equation 4.15.4 [M2]:

$$\begin{aligned} &= Q*L/4(1 + 2(R^2 - h^2)/(L^2))/(1 + (4h^2)/(3L)) - 4a/L \\ &= 2038*7.7/4(1 + 2(0.740^2 - 0.000^2)/(7.67^2))/(1 + (4*0.000)/ \end{aligned}$$

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$$(3*7.667)) - 4*0.55/7.67 \\ = 2854.9 \text{ ft.lb.}$$

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$$= P * Rm / (2t) - M2 / (\pi * Rm^2 * t) \\ = 1820.32 * 8.875 / (2 * 0.750) - 34258.9 / (\pi * 8.9^2 * 0.750) \\ = 10585.63 \text{ psi}$$

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$$= P * Rm / (2t) + M2 / (\pi * Rm^2 * t) \\ = 1820.32 * 8.875 / (2 * 0.750) + 34258.9 / (\pi * 8.9^2 * 0.750) \\ = 10954.83 \text{ psi}$$

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$$= P * Rm / (2t) - M1 / (K1 * \pi * Rm^2 * t) \\ = 1820.32 * 8.875 / (2 * 0.750) - 99.9 / (0.1066 * \pi * 8.9^2 * 0.750) \\ = 10775.28 \text{ psi}$$

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:

$$= P * Rm / (2t) + M1 / (K1 * \pi * Rm^2 * t) \\ = 1820.32 * 8.875 / (2 * 0.750) + 99.9 / (0.1923 * \pi * 8.9^2 * 0.750) \\ = 10767.43 \text{ psi}$$

Maximum Shear Force in the Saddle (4.15.5) [T]:

$$= Q(L - 2a) / (L + (4 * h^2 / 3)) \\ = 2038 (7.67 - 2 * 0.55) / (7.67 + (4 * 0.00 / 3)) \\ = 1745.1 \text{ lb.}$$

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$$= K2 * T / (Rm * t) \\ = 1.1707 * 1745.12 / (8.8750 * 0.7500) \\ = 306.93 \text{ psi}$$

Decay Length (4.15.22) [x1,x2]:

$$= 0.78 * \text{sqrt}(Rm * t) \\ = 0.78 * \text{sqrt}(8.875 * 0.750) \\ = 2.012 \text{ in.}$$

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

$$= -K5 * Q * k / (t * (b + X1 + X2)) \\ = -0.7603 * 2038 * 0.1 / (0.750 * (4.00 + 2.01 + 2.01)) \\ = -25.75 \text{ psi}$$

Circ. Comp. Stress at Horn of Saddle, $L \geq 8Rm$ (4.15.24) [sigma7]:

$$= -Q / (4 * t * (b + X1 + X2)) - 3 * K7 * Q / (2 * t^2) \\ = -2038 / (4 * 0.750 * (4.000 + 2.012 + 2.012)) - \\ 3 * 0.0328 * 2038 / (2 * 0.750^2) \\ = -262.75 \text{ psi}$$

Effective reinforcing plate width (4.15.1) [B1]:

$$= \text{min}(b + 1.56 * \text{sqrt}(Rm * t), 2a) \\ = \text{min}(4.00 + 1.56 * \text{sqrt}(8.875 * 0.750), 2 * 6.625) \\ = 8.02 \text{ in.}$$

Results for Vessel Ribs, Web and Base

Baseplate Length

Bplen

16.4000 in.

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Horizontal Vessel Analysis (Test) : Step: 11 5:58p Sep 18,2012

Baseplate Thickness	Bpthk	0.2500	in.
Baseplate Width	Bpwid	4.0000	in.
Number of Ribs (inc. outside ribs)	Nribs	2	
Rib Thickness	Ribtk	0.2500	in.
Web Thickness	Webtk	0.2500	in.
Web Location	Webloc	Side	

Moment of Inertia of Saddle - Lateral Direction

	Y	A	AY	Io
Shell	0.3750	6.7041	2.5140	1.2570
Wearplate	0.8750	1.2500	1.0938	0.9635
Web	12.3750	5.6875	70.3828	1116.2904
BasePlate	23.8750	1.0000	23.8750	570.0208
Totals	37.5000	14.6416	97.8656	1688.5317

Value C1 = Sumof(Ay)/Sumof(A) = 6.6841 in.
 Value I = Sumof(Io) - C1*Sumof(Ay) = 1034.3906 in**4
 Value As = Sumof(A) - Ashell = 7.9375 in²

K1 = (1+Cos(beta)-.5*Sin(beta)²)/(pi-beta+Sin(beta)*Cos(beta)) = 0.2035

Fh = K1 * Q = 0.2035 * 2038.728 = 414.9255 lb.

Tension Stress, St = (Fh/As) = 52.2741 psi
 Allowed Stress, Sa = 0.6 * Yield Str = 21600.0000 psi

d = B - R*Sin(theta) / theta = 25.4706 in.
 Bending Moment, M = Fh * d = 880.6987 ft.lb.

Bending Stress, Sb = (M * C1 / I) = 68.2913 psi
 Allowed Stress, Sa = 2/3 * Yield Str = 24000.0000 psi

Minimum Thickness of Baseplate per Moss :

= (3 * (Q + Saddle_Wt) * BasePlateWidth / (2 * BasePlateLength *
 AllStress))^{1/2}
 = (3 * (2038 + 56) * 4.00 / (2 * 16.400 * 24000.000))^{1/2}
 = 0.179 in.

Calculation of Axial Load, Intermediate Values and Compressive Stress

Effective Baseplate Length [e]:

= (Bplen - Clearance) / (Nribs - 1)
 = (16.4000 - 1.0) / (2 - 1) = 15.4000 in.

Baseplate Pressure Area [Ap]:

= e * Bpwid / 2
 = 15.4000 * 4.0000 / 2 = 30.8000 in²

Axial Load [P]:

= Ap * Bp
 = 30.8 * 31.08 = 957.2 lb.

Area of the Rib and Web [Ar]:

= (Bpwid - Clearance - Webtk) * Ribtk + e/2 * Webtk
 = (4.000 - 1.0 - 0.250) * 0.250 + 15.4000/2 * 0.250
 = 2.612 in²

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Horizontal Vessel Analysis (Test) : Step: 11 5:58p Sep 18,2012

Compressive Stress [Sc]:

= P/Ar

= 957.2/2.6125 = 366.3954 psi

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	Y	A	AY	Ay ²	Io
Rib	1.6250	0.8125	1.3203	0.9040	0.8932
Web	0.1250	1.9250	0.2406	0.3816	0.0201
Values	0.5702	2.7375	1.5609	1.2855	0.9133

Bending Moment [Rm]:

= Fl / (2 * Bplen) * e * rl / 2

= 0.0 / (2 * 16.40) * 15.400 * 27.88 / 2

= 0.000 ft.lb.

KL/R < Cc (30.3843 < 126.0993) per AISC E2-1

Sca = (1 - (Klr)² / (2 * Cc²)) * Fy / (5/3 + 3 * (Klr) / (8 * Cc) - (Klr³) / (8 * Cc³))

Sca = (1 - (30.38)² / (2 * 126.10²)) * 36000 / (5/3 + 3 * (30.38) / (8 * 126.10) - (30.38³) / (8 * 126.10³))

Sca = 19914.21 psi

AISC Unity Check on Outside Ribs (must be <= 1.0)

Check = Sc/Sca + (Rm/Z)/Sba

Check = 366.40/19914.21 + (0.00/0.905)/24000.00

Check = 0.02

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Nozzle Calcs. : T1/T2

Nozl: 6 5:58p Sep 18,2012

INPUT VALUES, Nozzle Description: T1/T2 From : 20

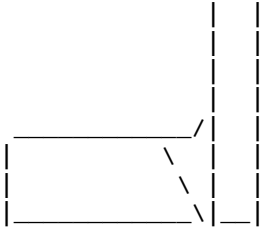
Pressure for Reinforcement Calculations	P	150.021	psig
Temperature for Internal Pressure	Temp	650	F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	S	18800.00	psi
Shell Allowable Stress At Ambient	Sa	20000.00	psi
Inside Diameter of Cylindrical Shell	D	18.1250	in.
Shell Finished (Minimum) Thickness	t	0.1875	in.
Shell Internal Corrosion Allowance	c	0.0625	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Cylinder/Cone Centerline	L1	3.2500	in.
Distance from Bottom/Left Tangent		0.4433	ft.
User Entered Minimum Design Metal Temperature		-20.00	F

Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		OD	
Layout Angle		110.57	deg
Diameter		4.0000	in.
Size and Thickness Basis		Nominal	
Nominal Thickness	tn	40	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0625	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	7.3400	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.2500	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.1875	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		300	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure + static head.

Nozzle Sketch (may not represent actual weld type/configuration)



Insert Nozzle No Pad, no Inside projection

Note : Checking Nozzle 90 degrees to the Longitudinal axis.

Reinforcement CALCULATION, Description: T1/T2

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 4.500 in.
Actual Thickness Used in Calculation 0.237 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= (P*R)/(S*E-0.6*P) per UG-27 (c)(1)
= (150.02*9.1250)/(18800*1.00-0.6*150.02)
= 0.0732 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= (P*Ro)/(S*E+0.4*P) per Appendix 1-1 (a)(1)
= (150.02*2.2500)/(17100*1.00+0.4*150.02)
= 0.0197 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit)	D1	9.0105	in.
Parallel to Vessel Wall, opening length	d	4.5053	in.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	0.3125	in.

Results of Nozzle Reinforcement Area Calculations:

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required Ar	0.166	NA	NA in ²
Area in Shell A1	0.396	NA	NA in ²
Area in Nozzle Wall A2	0.089	NA	NA in ²
Area in Inward Nozzle A3	0.000	NA	NA in ²
Area in Welds A41+A42+A43	0.057	NA	NA in ²
Area in Element A5	0.000	NA	NA in ²
TOTAL AREA AVAILABLE Atot	0.541	NA	NA in ²

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 67.13 Degs.

The area available without a pad is Sufficient.

Area Required [A]:
= (d * tr*F + 2 * tn * tr*F * (1-fr1)) UG-37(c)
= (4.5053*0.0732*0.5+2*0.1745*0.0732*0.5*(1-0.91))
= 0.166 in²

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

$$\begin{aligned} &= d(E1*tr - F*tr) - 2 * tn(E1*tr - F*tr) * (1 - fr1) \\ &= 4.505 (1.00 * 0.1250 - 0.5 * 0.073) - 2 * 0.175 \\ &\quad (1.00 * 0.1250 - 0.5 * 0.0732) * (1 - 0.910) \\ &= 0.396 \text{ in}^2 \end{aligned}$$

Area Available in Nozzle Projecting Outward [A2]:

$$\begin{aligned} &= (2 * tlnp) * (tn - trn) * fr2/\sin(alpha3) \\ &= (2 * 0.312) * (0.1745 - 0.0197) * 0.9096/\sin(83.4) \\ &= 0.089 \text{ in}^2 \end{aligned}$$

Note: See Appendix L, L-7.7.7(b) for more information.

Area Available in Inward Weld + Outward Weld [A41 + A43]:

$$\begin{aligned} &= Wo^2 * fr2 + (Wi-can/0.707)^2 * fr2 \\ &= 0.2500^2 * 0.9096 + (0.0000)^2 * 0.9096 \\ &= 0.057 \text{ in}^2 \end{aligned}$$

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, tg = 0.207 , tr = 0.020 , c = 0.0625 in. , E* = 1.00
Stress Ratio = tr * (E*)/(tg - c) = 0.136 , Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -155 F

MDMT of Nozzle-Shell/Head Weld for the Nozzle (UCS-66(a)1(b)), Curve: B

Govrn. thk, tg = 0.188 , tr = 0.069 , c = 0.0625 in. , E* = 1.00
Stress Ratio = tr * (E*)/(tg - c) = 0.550 , Temp. Reduction = 50 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -55 F

Governing MDMT of all the sub-joints of this Junction : -55 F

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ANSI B16.5/47 flanges per UCS-66(c) -20 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(b) -155 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(c) -155 F

Where the Stress Reduction Ratio per UCS-66(b)(1)(b) is :

$$\text{Design Pressure/Ambient Rating} = 150.02/740.00 = 0.203$$

Note: Using the minimum value from (b)(1)(b) and (b)(1)(c) above
as the calculated nozzle flange MDMT.

Weld Size Calculations. Description: T1/T2

Intermediate Calc. for nozzle/shell Welds Tmin 0.1250 in.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.0875 = 0.7 * tmin.	0.1768 = 0.7 * Wo in.

Weld Strength and Weld Loads per UG-41.1, Sketch (a) or (b)

Weld Load [W]:

$$\begin{aligned} &= (A-A1+2*tn*fr1*(E1*t-tr))*Sv \\ &= (0.1660 - 0.3956 + 2 * 0.1745 * 0.9096 * \\ &\quad (1.00 * 0.1250 - 0.0366)) * 18800 \\ &= 0.00 \text{ lb.} \end{aligned}$$

Note: F is always set to 1.0 throughout the calculation.

Weld Load [W1]:

$$\begin{aligned} &= (A2+A5+A4-(Wi-Can/.707)^2*fr2)*Sv \\ &= (0.0886 + 0.0000 + 0.0568 - 0.0000 * 0.91) * 18800 \\ &= 2734.52 \text{ lb.} \end{aligned}$$

Weld Load [W2]:

$$\begin{aligned} &= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv \\ &= (0.0886 + 0.0000 + 0.0568 + (0.0397)) * 18800 \\ &= 3480.50 \text{ lb.} \end{aligned}$$

Weld Load [W3]:

$$\begin{aligned} &= (A2+A3+A4+A5+(2*tn*t*fr1))*S \\ &= (0.0886 + 0.0000 + 0.0568 + 0.0000 + (0.0397)) * 18800 \\ &= 3480.50 \text{ lb.} \end{aligned}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$\begin{aligned} &= (\pi/2) * Dlo * Wo * 0.49 * Snw \\ &= (3.1416/2.0) * 4.8840 * 0.2500 * 0.49 * 17100 \\ &= 16071. \text{ lb.} \end{aligned}$$

Shear, Nozzle Wall [Snw]:

$$\begin{aligned} &= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn \\ &= (3.1416 * 2.3473) * (0.2370 - 0.0625) * 0.7 * 17100 \\ &= 15403. \text{ lb.} \end{aligned}$$

Tension, Shell Groove Weld [Tngw]:

$$\begin{aligned} &= (\pi/2) * Dlo * (Wgnvi-Cas) * 0.74 * Sng \\ &= (3.1416/2.0) * 4.8840 * (0.1875 - 0.0625) * 0.74 * 18800 \\ &= 13341. \text{ lb.} \end{aligned}$$

Strength of Failure Paths:

$$\begin{aligned} \text{PATH11} &= (\text{SONW} + \text{SNW}) = (16070 + 15403) = 31473 \text{ lb.} \\ \text{PATH22} &= (\text{Sonw} + \text{Tpgw} + \text{Tngw} + \text{Sinw}) \\ &= (16070 + 0 + 13341 + 0) = 29411 \text{ lb.} \\ \text{PATH33} &= (\text{Sonw} + \text{Tngw} + \text{Sinw}) \\ &= (16070 + 13341 + 0) = 29411 \text{ lb.} \end{aligned}$$

Summary of Failure Path Calculations:

Path 1-1 = 31473 lb., must exceed W = 0 lb. or W1 = 2734 lb.
Path 2-2 = 29411 lb., must exceed W = 0 lb. or W2 = 3480 lb.

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Nozzle Calcs. : T1/T2 Nozl: 6 5:58p Sep 18,2012

Path 3-3 = 29411 lb., must exceed W = 0 lb. or W3 = 3480 lb.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 254.799 psig

Note: The MAWP of this junction was limited by the parent Shell/Head.

Note : Checking Nozzle in plane parallel to the vessel axis.

Reinforcement CALCULATION, Description: T1/T2

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 4.500 in.
Actual Thickness Used in Calculation 0.237 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= (P*R)/(S*E-0.6*P) per UG-27 (c)(1)
= (150.02*9.1250)/(18800*1.00-0.6*150.02)
= 0.0732 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= (P*Ro)/(S*E+0.4*P) per Appendix 1-1 (a)(1)
= (150.02*2.2500)/(17100*1.00+0.4*150.02)
= 0.0197 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit) D1 8.3020 in.
Parallel to Vessel Wall, opening length d 4.1510 in.
Normal to Vessel Wall (Thickness Limit), no pad Tlnp 0.3125 in.

Results of Nozzle Reinforcement Area Calculations:

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required Ar	0.306	NA	NA in ²
Area in Shell A1	0.214	NA	NA in ²
Area in Nozzle Wall A2	0.088	NA	NA in ²
Area in Inward Nozzle A3	0.000	NA	NA in ²
Area in Welds A41+A42+A43	0.057	NA	NA in ²
Area in Element A5	0.000	NA	NA in ²
TOTAL AREA AVAILABLE Atot	0.358	NA	NA in ²

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 90.00 Degs.

The area available without a pad is Sufficient.

Area Required [A]:
= (d * tr*F + 2 * tn * tr*F * (1-fr1)) UG-37(c)
= (4.1510*0.0732*1.0+2*0.1745*0.0732*1.0*(1-0.91))
= 0.306 in²

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

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Nozzle Calcs. : T1/T2 Nozl: 6 5:58p Sep 18,2012

$$\begin{aligned}
&= d(E1*t - F*tr) - 2 * tn(E1*t - F*tr) * (1 - fr1) \\
&= 4.151 (1.00 * 0.1250 - 1.0 * 0.073) - 2 * 0.175 \\
&\quad (1.00 * 0.1250 - 1.0 * 0.0732) * (1 - 0.910) \\
&= 0.214 \text{ in}^2
\end{aligned}$$

Area Available in Nozzle Projecting Outward [A2]:

$$\begin{aligned}
&= (2 * tlnp) * (tn - trn) * fr2 \\
&= (2 * 0.312) * (0.1745 - 0.0197) * 0.9096 \\
&= 0.088 \text{ in}^2
\end{aligned}$$

Area Available in Inward Weld + Outward Weld [A41 + A43]:

$$\begin{aligned}
&= Wo^2 * fr2 + (Wi-can/0.707)^2 * fr2 \\
&= 0.2500^2 * 0.9096 + (0.0000)^2 * 0.9096 \\
&= 0.057 \text{ in}^2
\end{aligned}$$

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures	ta = 0.0822 in.
Wall Thickness per UG16(b),	tr16b = 0.1250 in.
Wall Thickness, shell/head, internal pressure	trb1 = 0.1357 in.
Wall Thickness	tb1 = max(trb1, tr16b) = 0.1357 in.
Wall Thickness	tb2 = max(trb2, tr16b) = 0.1250 in.
Wall Thickness per table UG-45	tb3 = 0.2695 in.

Determine Nozzle Thickness candidate [tb]:

$$\begin{aligned}
&= \min[tb3, \max(tb1, tb2)] \\
&= \min[0.270 , \max(0.136 , 0.125)] \\
&= 0.1357 \text{ in.}
\end{aligned}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

$$\begin{aligned}
&= \max(ta, tb) \\
&= \max(0.0822 , 0.1357) \\
&= 0.1357 \text{ in.}
\end{aligned}$$

Available Nozzle Neck Thickness = 0.875 * 0.237 = 0.207 in. --> OK

Weld Size Calculations. Description: T1/T2

Intermediate Calc. for nozzle/shell welds Tmin 0.1250 in.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.0875 = 0.7 * tmin.	0.1768 = 0.7 * Wo in.

Weld Strength and Weld Loads per UG-41.1, Sketch (a) or (b)

Weld Load [W]:

$$\begin{aligned}
&= (A-A1+2*tn*fr1*(E1*t-tr)) * Sv \\
&= (0.3060 - 0.2135 + 2 * 0.1745 * 0.9096 * \\
&\quad (1.00 * 0.1250 - 0.0732)) * 18800 \\
&= 2048.27 \text{ lb.}
\end{aligned}$$

Note: F is always set to 1.0 throughout the calculation.

Weld Load [W1]:

$$\begin{aligned}
&= (A2+A5+A4-(Wi-Can/.707)^2*fr2) * Sv \\
&= (0.0880 + 0.0000 + 0.0568 - 0.0000 * 0.91) * 18800 \\
&= 2723.49 \text{ lb.}
\end{aligned}$$

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Nozzle Calcs. : T1/T2

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Weld Load [W2]:

$$\begin{aligned} &= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv \\ &= (0.0880 + 0.0000 + 0.0568 + (0.0397)) * 18800 \\ &= 3469.48 \text{ lb.} \end{aligned}$$

Weld Load [W3]:

$$\begin{aligned} &= (A2+A3+A4+A5+(2*tn*t*fr1))*S \\ &= (0.0880 + 0.0000 + 0.0568 + 0.0000 + (0.0397)) * 18800 \\ &= 3469.48 \text{ lb.} \end{aligned}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$\begin{aligned} &= (\pi/2) * Dlo * Wo * 0.49 * Snw \\ &= (3.1416/2.0) * 4.5000 * 0.2500 * 0.49 * 17100 \\ &= 14807. \text{ lb.} \end{aligned}$$

Shear, Nozzle Wall [Snw]:

$$\begin{aligned} &= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn \\ &= (3.1416 * 2.1628) * (0.2370 - 0.0625) * 0.7 * 17100 \\ &= 14192. \text{ lb.} \end{aligned}$$

Tension, Shell Groove Weld [Tngw]:

$$\begin{aligned} &= (\pi/2) * Dlo * (Wgnvi-Cas) * 0.74 * Sng \\ &= (3.1416/2.0) * 4.5000 * (0.1875 - 0.0625) * 0.74 * 18800 \\ &= 12292. \text{ lb.} \end{aligned}$$

Strength of Failure Paths:

$$\begin{aligned} \text{PATH11} &= (\text{SONW} + \text{SNW}) = (14806 + 14192) = 28998 \text{ lb.} \\ \text{PATH22} &= (\text{Sonw} + \text{Tpgw} + \text{Tngw} + \text{Sinw}) \\ &= (14806 + 0 + 12292 + 0) = 27099 \text{ lb.} \\ \text{PATH33} &= (\text{Sonw} + \text{Tngw} + \text{Sinw}) \\ &= (14806 + 12292 + 0) = 27099 \text{ lb.} \end{aligned}$$

Summary of Failure Path Calculations:

Path 1-1 = 28998 lb., must exceed W = 2048 lb. or W1 = 2723 lb.
Path 2-2 = 27099 lb., must exceed W = 2048 lb. or W2 = 3469 lb.
Path 3-3 = 27099 lb., must exceed W = 2048 lb. or W3 = 3469 lb.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 162.656 psig

The Drop for this Nozzle is : 1.2570 in.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 8.7976 in.

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Nozzle Calcs. : S2

Noz1: 7 5:58p Sep 18,2012

INPUT VALUES, Nozzle Description: S2 From: 40

Pressure for Reinforcement Calculations	P	1400.625	psig
Temperature for Internal Pressure	Temp	650	F
Design External Pressure	Pext	15.00	psig
Temperature for External Pressure	Tempex	650	F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	S	18800.00	psi
Shell Allowable Stress At Ambient	Sa	20000.00	psi
Inside Diameter of Cylindrical Shell	D	17.0000	in.
Design Length of Section	L	92.0040	in.
Shell Finished (Minimum) Thickness	t	0.7500	in.
Shell Internal Corrosion Allowance	c	0.0625	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		1.5519	ft.
User Entered Minimum Design Metal Temperature		-20.00	F

Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		OD	
Layout Angle		270.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Nominal	
Nominal Thickness	tn	XXS	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0625	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	7.7500	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3750	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7500	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
Pad Material		SA-516 70	
Pad Allowable Stress at Temperature	Sp	18800.00	psi
Pad Allowable Stress At Ambient	Spa	20000.00	psi
Diameter of Pad along vessel surface	Dp	8.0000	in.
Thickness of Pad	te	0.6250	in.
Weld leg size between Pad and Shell	Wp	0.5000	in.
Groove weld depth between Pad and Nozzle	Wgpn	0.6250	in.

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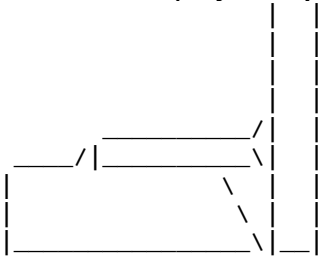
Nozzle Calcs. : S2 Nozl: 7 5:58p Sep 18,2012

Reinforcing Pad Width 2.2500 in.
ASME Code Weld Type per UW-16 None

Class of attached Flange 900
Grade of attached Flange GR 1.1

The Pressure Design option was Design Pressure + static head.

Nozzle Sketch (may not represent actual weld type/configuration)



Insert Nozzle With Pad, no Inside projection

Reinforcement CALCULATION, Description: S2

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 3.500 in.
Actual Thickness Used in Calculation 0.600 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= $(P \cdot R) / (S \cdot E - 0.6 \cdot P)$ per UG-27 (c)(1)
= $(1400.63 \cdot 8.5625) / (18800 \cdot 1.00 - 0.6 \cdot 1400.63)$
= 0.6678 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= $(P \cdot R_o) / (S \cdot E + 0.4 \cdot P)$ per Appendix 1-1 (a)(1)
= $(1400.63 \cdot 1.7500) / (17100 \cdot 1.00 + 0.4 \cdot 1400.63)$
= 0.1388 in.

Required Nozzle thickness under External Pressure per UG-28 : 0.0166 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit)	D1	4.8750 in.
Parallel to Vessel Wall	Rn+tn+t	2.4375 in.
Normal to Vessel Wall (Thickness Limit), pad side Tlwp		1.7188 in.

Note : The Pad diameter is greater than the Diameter Limit, the excess will not be considered .

Weld Strength Reduction Factor [fr1]:
= $\min(1, S_n/S)$
= $\min(1, 17100.0/18800.0)$
= 0.910

Weld Strength Reduction Factor [fr2]:
= $\min(1, S_n/S)$

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Nozzle Calcs. : S2

Noz1: 7 5:58p Sep 18,2012

$$= \min(1, 17100.0/18800.0)$$
$$= 0.910$$

Weld Strength Reduction Factor [fr4]:

$$= \min(1, Sp/S)$$
$$= \min(1, 18800.0/18800.0)$$
$$= 1.000$$

Weld Strength Reduction Factor [fr3]:

$$= \min(fr2, fr4)$$
$$= \min(0.9, 1.0)$$
$$= 0.910$$

Results of Nozzle Reinforcement Area Calculations:

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required Ar	1.684	0.156	NA in ²
Area in Shell A1	0.046	1.326	NA in ²
Area in Nozzle Wall A2	1.247	1.629	NA in ²
Area in Inward Nozzle A3	0.000	0.000	NA in ²
Area in Welds A41+A42+A43	0.128	0.128	NA in ²
Area in Element A5	0.859	0.859	NA in ²
TOTAL AREA AVAILABLE Atot	2.280	3.942	NA in ²

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 90.00 Degr.

The area available without a pad is Insufficient.

The area available with the given pad is Sufficient.

SELECTION OF POSSIBLE REINFORCING PADS:	Diameter	Thickness
Based on given Pad Thickness:	3.9375	0.6250 in.
Based on given Pad Diameter:	8.0000	0.2500 in.
Based on Shell or Nozzle Thickness:	3.9375	0.6250 in.

Area Required [A]:

$$= (d * tr * F + 2 * tn * tr * F * (1 - fr1)) UG-37(c)$$
$$= (2.4250 * 0.6678 * 1.0 + 2 * 0.5375 * 0.6678 * 1.0 * (1 - 0.91))$$
$$= 1.684 \text{ in}^2$$

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

$$= d (E1 * t - F * tr) - 2 * tn (E1 * t - F * tr) * (1 - fr1)$$
$$= 2.450 (1.00 * 0.6875 - 1.0 * 0.668) - 2 * 0.538$$
$$(1.00 * 0.6875 - 1.0 * 0.6678) * (1 - 0.910)$$
$$= 0.046 \text{ in}^2$$

Area Available in Nozzle Wall Projecting Outward [A2]:

$$= (2 * Tlwp) * (tn - trn) * fr2$$
$$= (2 * 1.719) * (0.5375 - 0.1388) * 0.9096$$
$$= 1.247 \text{ in}^2$$

Area Available in Welds [A41 + A42 + A43]:

$$= Wo^2 * fr3 + (Wi - can / 0.707)^2 * fr2 + Wp^2 * fr4$$
$$= 0.3750^2 * 0.91 + (0.0000)^2 * 0.91 + 0.0000^2 * 1.00$$
$$= 0.128 \text{ in}^2$$

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Nozzle Calcs. : S2

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Area Available in Element [A5]:

$$\begin{aligned} &= (\min(D_p, DL) - (\text{Nozzle OD})) * (\min(t_p, T_{lwp}, t_e)) * fr_4 \\ &= (4.8750 - 3.5000) * 0.6250 * 1.0000 \\ &= 0.859 \text{ in}^2 \end{aligned}$$

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures $t_a = 0.2013 \text{ in.}$
Wall Thickness per UG16(b), $tr_{16b} = 0.1250 \text{ in.}$
Wall Thickness, shell/head, internal pressure $tr_{b1} = 0.7303 \text{ in.}$
Wall Thickness $tb_1 = \max(tr_{b1}, tr_{16b}) = 0.7303 \text{ in.}$
Wall Thickness $tb_2 = \max(tr_{b2}, tr_{16b}) = 0.1250 \text{ in.}$
Wall Thickness per table UG-45 $tb_3 = 0.2515 \text{ in.}$

Determine Nozzle Thickness candidate [tb]:

$$\begin{aligned} &= \min[tb_3, \max(tb_1, tb_2)] \\ &= \min[0.251, \max(0.730, 0.125)] \\ &= 0.2515 \text{ in.} \end{aligned}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

$$\begin{aligned} &= \max(t_a, t_b) \\ &= \max(0.2013, 0.2515) \\ &= 0.2515 \text{ in.} \end{aligned}$$

Available Nozzle Neck Thickness = $0.875 * 0.600 = 0.525 \text{ in.}$ --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, $t_g = 0.525$, $t_r = 0.139$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.300$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -4 F
Min Metal Temp. at Required thickness (UCS 66.1) -144 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

MDMT of Nozzle Neck to Pad Weld for the Nozzle, Curve: B

Govrn. thk, $t_g = 0.525$, $t_r = 0.139$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.300$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -4 F
Min Metal Temp. at Required thickness (UCS 66.1) -144 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

MDMT of Nozzle Neck to Pad Weld for Reinforcement pad, Curve: B

Govrn. thk, $t_g = 0.525$, $t_r = 0.139$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.300$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -4 F
Min Metal Temp. at Required thickness (UCS 66.1) -144 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

MDMT of Shell to Pad Weld at Pad OD for pad, Curve: B

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Nozzle Calcs. : S2

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Govrn. thk, tg = 0.625 , c = 0.0625 in. , E* = 1.00

Stress Ratio = tr * (E*)/(tg - c) = 0.910 , Temp. Reduction = 9 F

Pad governing, Conservatively assuming Pad stress = Shell stress(Div. 1 L-9.3)

Min Metal Temp. w/o impact per UCS-66	6 F
Min Metal Temp. at Required thickness (UCS 66.1)	-3 F
Min Metal Temp. w/o impact per UG-20(f)	-20 F

MDMT of Nozzle-Shell/Head Weld for the Nozzle (UCS-66(a)1(b)), Curve: B

Govrn. thk, tg = 0.525 , tr = 0.139 , c = 0.0625 in. , E* = 1.00

Stress Ratio = tr * (E*)/(tg - c) = 0.300 , Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66	-4 F
Min Metal Temp. at Required thickness (UCS 66.1)	-144 F
Min Metal Temp. w/o impact per UG-20(f)	-20 F

Governing MDMT of the Nozzle	: -144 F
Governing MDMT of the Reinforcement Pad	: -20 F
Governing MDMT of all the sub-joints of this Junction	: -20 F

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ANSI B16.5/47 flanges per UCS-66(c)	-20 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(b)	-55 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(c)	-155 F

Where the Stress Reduction Ratio per UCS-66(b)(1)(b) is :

Design Pressure/Ambient Rating = 1400.63/2220.00 = 0.631

Note: Using the minimum value from (b)(1)(b) and (b)(1)(c) above as the calculated nozzle flange MDMT.

Weld Size Calculations, Description: S2

Intermediate Calc. for nozzle/shell Welds	Tmin	0.5375	in.
Intermediate Calc. for pad/shell Welds	TminPad	0.6250	in.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2500 = Min per Code	0.2651 = 0.7 * Wo in.
Pad Weld	0.3125 = 0.5*TminPad	0.3535 = 0.7 * Wp in.

Weld Strength and Weld Loads per UG-41.1, Sketch (a) or (b)

Weld Load [W]:

$$= (A-A1+2*tn*fr1*(E1*t-tr))*Sv$$

$$= (1.6842 - 0.0464 + 2 * 0.5375 * 0.9096 * (1.00 * 0.6875 - 0.6678)) * 18800$$

$$= 31153.78 \text{ lb.}$$

Note: F is always set to 1.0 throughout the calculation.

Weld Load [W1]:

$$= (A2+A5+A4-(Wi-Can/.707)^2*fr2)*Sv$$

$$= (1.2466 + 0.8594 + 0.1279 - 0.0000 * 0.91) * 18800$$

$$= 41997.52 \text{ lb.}$$

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Nozzle Calcs. : S2

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Weld Load [W2]:

$$\begin{aligned} &= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv \\ &= (1.2466 + 0.0000 + 0.1279 + (0.6722)) * 18800 \\ &= 38479.24 \text{ lb.} \end{aligned}$$

Weld Load [W3]:

$$\begin{aligned} &= (A2+A3+A4+A5+(2*tn*t*fr1))*S \\ &= (1.2466 + 0.0000 + 0.1279 + 0.8594 + (0.6722)) * 18800 \\ &= 54635.49 \text{ lb.} \end{aligned}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$\begin{aligned} &= (\pi/2) * Dlo * Wo * 0.49 * Snw \\ &= (3.1416/2.0) * 3.5000 * 0.3750 * 0.49 * 17100 \\ &= 17275. \text{ lb.} \end{aligned}$$

Shear, Pad Element Weld [Spew]:

$$\begin{aligned} &= (\pi/2) * DP * WP * 0.49 * SEW \\ &= (3.1416/2.0) * 8.0000 * 0.5000 * 0.49 * 18800 \\ &= 57881. \text{ lb.} \end{aligned}$$

Shear, Nozzle Wall [Snw]:

$$\begin{aligned} &= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn \\ &= (3.1416 * 1.4813) * (0.6000 - 0.0625) * 0.7 * 17100 \\ &= 29940. \text{ lb.} \end{aligned}$$

Tension, Pad Groove Weld [Tpgw]:

$$\begin{aligned} &= (\pi/2) * Dlo * Wgpn * 0.74 * Seg \\ &= (3.1416/2) * 3.5000 * 0.6250 * 0.74 * 18800 \\ &= 47803. \text{ lb.} \end{aligned}$$

Tension, Shell Groove Weld [Tngw]:

$$\begin{aligned} &= (\pi/2) * Dlo * (Wgnvi-Cas) * 0.74 * Sng \\ &= (3.1416/2.0) * 3.5000 * (0.7500 - 0.0625) * 0.74 * 18800 \\ &= 52584. \text{ lb.} \end{aligned}$$

Strength of Failure Paths:

$$\begin{aligned} \text{PATH11} &= (\text{SPEW} + \text{SNW}) = (57880 + 29939) = 87820 \text{ lb.} \\ \text{PATH22} &= (\text{Sonw} + \text{Tpgw} + \text{Tngw} + \text{Sinw}) \\ &= (17274 + 47803 + 52583 + 0) = 117661 \text{ lb.} \\ \text{PATH33} &= (\text{Spew} + \text{Tngw} + \text{Sinw}) \\ &= (57880 + 52583 + 0) = 110464 \text{ lb.} \end{aligned}$$

Summary of Failure Path Calculations:

Path 1-1 = 87820 lb., must exceed W = 31153 lb. or W1 = 41997 lb.
Path 2-2 = 117661 lb., must exceed W = 31153 lb. or W2 = 38479 lb.
Path 3-3 = 110464 lb., must exceed W = 31153 lb. or W3 = 54635 lb.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 1440.109 psig

Note: The MAWP of this junction was limited by the parent Shell/Head.

Nozzle is O.K. for the External Pressure 15.000 psig

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Nozzle Calcs. : S2

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The Drop for this Nozzle is : 0.1821 in.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 8.6821 in.

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Nozzle Calcs. : S1

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INPUT VALUES, Nozzle Description: S1 From: 40

Pressure for Reinforcement Calculations	P	1400.625	psig
Temperature for Internal Pressure	Temp	650	F
Design External Pressure	Pext	15.00	psig
Temperature for External Pressure	Tempex	650	F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	S	18800.00	psi
Shell Allowable Stress At Ambient	Sa	20000.00	psi
Inside Diameter of Cylindrical Shell	D	17.0000	in.
Design Length of Section	L	92.0040	in.
Shell Finished (Minimum) Thickness	t	0.7500	in.
Shell Internal Corrosion Allowance	c	0.0625	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		8.3419	ft.
User Entered Minimum Design Metal Temperature		-20.00	F

Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		OD	
Layout Angle		270.00	deg
Diameter		6.0000	in.
Size and Thickness Basis		Nominal	
Nominal Thickness	tn	XXS	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0625	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	7.7500	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3750	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7500	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
Pad Material		SA-516 70	
Pad Allowable Stress at Temperature	Sp	18800.00	psi
Pad Allowable Stress At Ambient	Spa	20000.00	psi
Diameter of Pad along vessel surface	Dp	10.1250	in.
Thickness of Pad	te	0.6250	in.
Weld leg size between Pad and Shell	Wp	0.5000	in.
Groove weld depth between Pad and Nozzle	Wgpn	0.6250	in.

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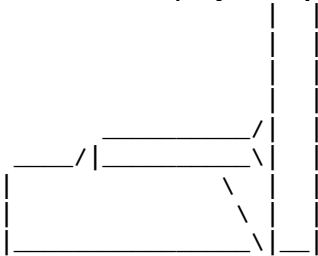
Nozzle Calcs. : S1 Nozl: 8 5:58p Sep 18,2012

Reinforcing Pad Width 1.7500 in.
ASME Code Weld Type per UW-16 None

Class of attached Flange 900
Grade of attached Flange GR 1.1

The Pressure Design option was Design Pressure + static head.

Nozzle Sketch (may not represent actual weld type/configuration)



Insert Nozzle With Pad, no Inside projection

Reinforcement CALCULATION, Description: S1

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 6.625 in.
Actual Thickness Used in Calculation 0.864 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= $(P \cdot R) / (S \cdot E - 0.6 \cdot P)$ per UG-27 (c)(1)
= $(1400.63 \cdot 8.5625) / (18800 \cdot 1.00 - 0.6 \cdot 1400.63)$
= 0.6678 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= $(P \cdot R_o) / (S \cdot E + 0.4 \cdot P)$ per Appendix 1-1 (a)(1)
= $(1400.63 \cdot 3.3125) / (17100 \cdot 1.00 + 0.4 \cdot 1400.63)$
= 0.2627 in.

Required Nozzle thickness under External Pressure per UG-28 : 0.0240 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit) D1 10.0440 in.
Parallel to Vessel Wall, opening length d 5.0220 in.
Normal to Vessel Wall (Thickness Limit), pad side Tlwp 1.7188 in.

Note : The Pad diameter is greater than the Diameter Limit, the excess will not be considered .

Weld Strength Reduction Factor [fr1]:
= $\min(1, S_n / S)$
= $\min(1, 17100.0 / 18800.0)$
= 0.910

Weld Strength Reduction Factor [fr2]:
= $\min(1, S_n / S)$

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Nozzle Calcs. : S1

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$$= \min(1, 17100.0/18800.0)$$
$$= 0.910$$

Weld Strength Reduction Factor [fr4]:

$$= \min(1, Sp/S)$$
$$= \min(1, 18800.0/18800.0)$$
$$= 1.000$$

Weld Strength Reduction Factor [fr3]:

$$= \min(fr2, fr4)$$
$$= \min(0.9, 1.0)$$
$$= 0.910$$

Results of Nozzle Reinforcement Area Calculations:

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required Ar	3.450	0.320	NA in ²
Area in Shell A1	0.096	2.748	NA in ²
Area in Nozzle Wall A2	1.685	2.431	NA in ²
Area in Inward Nozzle A3	0.000	0.000	NA in ²
Area in Welds A41+A42+A43	0.128	0.128	NA in ²
Area in Element A5	2.137	2.137	NA in ²
TOTAL AREA AVAILABLE Atot	4.046	7.444	NA in ²

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 90.00 Degr.

The area available without a pad is Insufficient.

The area available with the given pad is Sufficient.

SELECTION OF POSSIBLE REINFORCING PADS:	Diameter	Thickness
Based on given Pad Thickness:	9.1250	0.6250 in.
Based on given Pad Diameter:	10.1250	0.5000 in.
Based on Shell or Nozzle Thickness:	8.6875	0.7500 in.

Area Required [A]:

$$= (d * tr * F + 2 * tn * tr * F * (1 - fr1)) UG-37(c)$$
$$= (5.0220 * 0.6678 * 1.0 + 2 * 0.8015 * 0.6678 * 1.0 * (1 - 0.91))$$
$$= 3.450 \text{ in}^2$$

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

$$= d(E1 * t - F * tr) - 2 * tn(E1 * t - F * tr) * (1 - fr1)$$
$$= 5.022 (1.00 * 0.6875 - 1.0 * 0.668) - 2 * 0.802$$
$$(1.00 * 0.6875 - 1.0 * 0.6678) * (1 - 0.910)$$
$$= 0.096 \text{ in}^2$$

Area Available in Nozzle Wall Projecting Outward [A2]:

$$= (2 * Tlwp) * (tn - trn) * fr2$$
$$= (2 * 1.719) * (0.8015 - 0.2627) * 0.9096$$
$$= 1.685 \text{ in}^2$$

Area Available in Welds [A41 + A42 + A43]:

$$= Wo^2 * fr3 + (Wi - can / 0.707)^2 * fr2 + Wp^2 * fr4$$
$$= 0.3750^2 * 0.91 + (0.0000)^2 * 0.91 + 0.0000^2 * 1.00$$
$$= 0.128 \text{ in}^2$$

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Nozzle Calcs. : S1

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Area Available in Element [A5]:

$$\begin{aligned} &= (\min(D_p, D_L) - (\text{Nozzle OD})) * (\min(t_p, T_{lwp}, t_e)) * f_{r4} \\ &= (10.0440 - 6.6250) * 0.6250 * 1.0000 \\ &= 2.137 \text{ in}^2 \end{aligned}$$

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures $t_a = 0.3252 \text{ in.}$
Wall Thickness per UG16(b), $tr_{16b} = 0.1250 \text{ in.}$
Wall Thickness, shell/head, internal pressure $tr_{b1} = 0.7303 \text{ in.}$
Wall Thickness $tb_1 = \max(tr_{b1}, tr_{16b}) = 0.7303 \text{ in.}$
Wall Thickness $tb_2 = \max(tr_{b2}, tr_{16b}) = 0.1250 \text{ in.}$
Wall Thickness per table UG-45 $tb_3 = 0.3074 \text{ in.}$

Determine Nozzle Thickness candidate [tb]:

$$\begin{aligned} &= \min[tb_3, \max(tb_1, tb_2)] \\ &= \min[0.307, \max(0.730, 0.125)] \\ &= 0.3074 \text{ in.} \end{aligned}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

$$\begin{aligned} &= \max(t_a, t_b) \\ &= \max(0.3252, 0.3074) \\ &= 0.3252 \text{ in.} \end{aligned}$$

Available Nozzle Neck Thickness = $0.875 * 0.864 = 0.756 \text{ in.}$ --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, $t_g = 0.756$, $t_r = 0.263$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.379$, Temp. Reduction = 118 F

Min Metal Temp. w/o impact per UCS-66 17 F
Min Metal Temp. at Required thickness (UCS 66.1) -55 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

MDMT of Nozzle Neck to Pad Weld for the Nozzle, Curve: B

Govrn. thk, $t_g = 0.625$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.910$, Temp. Reduction = 9 F
Pad governing, Conservatively assuming Pad stress = Shell stress(Div. 1 L-9.3)

Min Metal Temp. w/o impact per UCS-66 6 F
Min Metal Temp. at Required thickness (UCS 66.1) -3 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

MDMT of Nozzle Neck to Pad Weld for Reinforcement pad, Curve: B

Govrn. thk, $t_g = 0.625$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.910$, Temp. Reduction = 9 F
Pad governing, Conservatively assuming Pad stress = Shell stress(Div. 1 L-9.3)

Min Metal Temp. w/o impact per UCS-66 6 F
Min Metal Temp. at Required thickness (UCS 66.1) -3 F
Min Metal Temp. w/o impact per UG-20(f) -20 F

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Nozzle Calcs. : S1

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MDMT of Shell to Pad Weld at Pad OD for pad, Curve: B

Govrn. thk, tg = 0.625 , c = 0.0625 in. , E* = 1.00
Stress Ratio = $tr * (E^*) / (tg - c) = 0.910$, Temp. Reduction = 9 F
Pad governing, Conservatively assuming Pad stress = Shell stress(Div. 1 L-9.3)

Min Metal Temp. w/o impact per UCS-66	6 F
Min Metal Temp. at Required thickness (UCS 66.1)	-3 F
Min Metal Temp. w/o impact per UG-20(f)	-20 F

MDMT of Nozzle-Shell/Head Weld for the Nozzle (UCS-66(a)1(b)), Curve: B

Govrn. thk, tg = 0.750 , tr = 0.626 , c = 0.0625 in. , E* = 1.00
Stress Ratio = $tr * (E^*) / (tg - c) = 0.910$, Temp. Reduction = 9 F

Min Metal Temp. w/o impact per UCS-66	16 F
Min Metal Temp. at Required thickness (UCS 66.1)	7 F
Min Metal Temp. w/o impact per UG-20(f)	-20 F

Governing MDMT of the Nozzle	: -20 F
Governing MDMT of the Reinforcement Pad	: -20 F
Governing MDMT of all the sub-joints of this Junction	: -20 F

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ANSI B16.5/47 flanges per UCS-66(c)	-20 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(b)	-55 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(c)	-55 F

Where the Stress Reduction Ratio per UCS-66(b)(1)(b) is :
Design Pressure/Ambient Rating = $1400.63 / 2220.00 = 0.631$

Note: Using the minimum value from (b)(1)(b) and (b)(1)(c) above
as the calculated nozzle flange MDMT.

Weld Size Calculations, Description: S1

Intermediate Calc. for nozzle/shell Welds	Tmin	0.6250	in.
Intermediate Calc. for pad/shell Welds	TminPad	0.6250	in.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	$0.2500 = \text{Min per Code}$	$0.2651 = 0.7 * W_o \text{ in.}$
Pad Weld	$0.3125 = 0.5 * T_{minPad}$	$0.3535 = 0.7 * W_p \text{ in.}$

Weld Strength and Weld Loads per UG-41.1, Sketch (a) or (b)

Weld Load [W]:
 $= (A - A1 + 2 * t_n * f_{r1} * (E1 * t - tr)) * S_v$
 $= (3.4503 - 0.0962 + 2 * 0.8015 * 0.9096 * (1.00 * 0.6875 - 0.6678)) * 18800$
 $= 63597.71 \text{ lb.}$

Note: F is always set to 1.0 throughout the calculation.

Weld Load [W1]:
 $= (A2 + A5 + A4 - (W_i - Can / .707)^2 * f_{r2}) * S_v$

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Nozzle Calcs. : S1 Nozl: 8 5:58p Sep 18,2012

$$= (1.6846 + 2.1369 + 0.1279 - 0.0000 * 0.91) * 18800$$
$$= 74248.52 \text{ lb.}$$

Weld Load [W2]:

$$= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv$$
$$= (1.6846 + 0.0000 + 0.1279 + (1.0024)) * 18800$$
$$= 52920.55 \text{ lb.}$$

Weld Load [W3]:

$$= (A2+A3+A4+A5+(2*tn*t*fr1))*S$$
$$= (1.6846 + 0.0000 + 0.1279 + 2.1369 + (1.0024)) * 18800$$
$$= 93093.80 \text{ lb.}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$= (\pi/2) * Dlo * Wo * 0.49 * Snw$$
$$= (3.1416/2.0) * 6.6250 * 0.3750 * 0.49 * 17100$$
$$= 32699. \text{ lb.}$$

Shear, Pad Element Weld [Spew]:

$$= (\pi/2) * DP * WP * 0.49 * SEW$$
$$= (3.1416/2.0) * 10.1250 * 0.5000 * 0.49 * 18800$$
$$= 73255. \text{ lb.}$$

Shear, Nozzle Wall [Snw]:

$$= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn$$
$$= (3.1416 * 2.9117) * (0.8640 - 0.0625) * 0.7 * 17100$$
$$= 87761. \text{ lb.}$$

Tension, Pad Groove Weld [Tpgw]:

$$= (\pi/2) * Dlo * Wgpn * 0.74 * Seg$$
$$= (3.1416/2) * 6.6250 * 0.6250 * 0.74 * 18800$$
$$= 90485. \text{ lb.}$$

Tension, Shell Groove Weld [Tngw]:

$$= (\pi/2) * Dlo * (Wgnvi-Cas) * 0.74 * Sng$$
$$= (3.1416/2.0) * 6.6250 * (0.7500 - 0.0625) * 0.74 * 18800$$
$$= 99533. \text{ lb.}$$

Strength of Failure Paths:

$$\text{PATH11} = (\text{SPEW} + \text{SNW}) = (73255 + 87761) = 161016 \text{ lb.}$$

$$\text{PATH22} = (\text{Sonw} + \text{Tpgw} + \text{Tngw} + \text{Sinw})$$
$$= (32698 + 90484 + 99533 + 0) = 222716 \text{ lb.}$$

$$\text{PATH33} = (\text{Spew} + \text{Tngw} + \text{Sinw})$$
$$= (73255 + 99533 + 0) = 172788 \text{ lb.}$$

Summary of Failure Path Calculations:

Path 1-1 = 161016 lb., must exceed W = 63597 lb. or W1 = 74248 lb.

Path 2-2 = 222716 lb., must exceed W = 63597 lb. or W2 = 52920 lb.

Path 3-3 = 172788 lb., must exceed W = 63597 lb. or W3 = 93093 lb.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 1440.109 psig

Note: The MAWP of this junction was limited by the parent Shell/Head.

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Nozzle Calcs. : S1 Nozl: 8 5:58p Sep 18,2012

Nozzle is O.K. for the External Pressure 15.000 psig

The Drop for this Nozzle is : 0.6720 in.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 9.1720 in.

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Nozzle Calcs. : S3

Noz1: 9 5:58p Sep 18,2012

INPUT VALUES, Nozzle Description: S3 From: 40

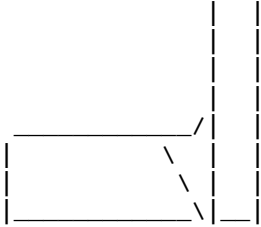
Pressure for Reinforcement Calculations	P	1400.011	psig
Temperature for Internal Pressure	Temp	650	F
Design External Pressure	Pext	15.00	psig
Temperature for External Pressure	Tempex	650	F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	S	18800.00	psi
Shell Allowable Stress At Ambient	Sa	20000.00	psi
Inside Diameter of Cylindrical Shell	D	17.0000	in.
Design Length of Section	L	92.0040	in.
Shell Finished (Minimum) Thickness	t	0.7500	in.
Shell Internal Corrosion Allowance	c	0.0625	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		1.5519	ft.
User Entered Minimum Design Metal Temperature		-20.00	F

Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		OD	
Layout Angle		90.00	deg
Diameter		2.0000	in.
Size and Thickness Basis		Nominal	
Nominal Thickness	tn	XXS	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0625	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	7.7500	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3750	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7500	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		900	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure + static head.

Nozzle Sketch (may not represent actual weld type/configuration)



Insert Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: S3

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 2.375 in.
Actual Thickness Used in Calculation 0.436 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= $(P \cdot R) / (S \cdot E - 0.6 \cdot P)$ per UG-27 (c)(1)
= $(1400.01 \cdot 8.5625) / (18800 \cdot 1.00 - 0.6 \cdot 1400.01)$
= 0.6675 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= $(P \cdot R_o) / (S \cdot E + 0.4 \cdot P)$ per Appendix 1-1 (a)(1)
= $(1400.01 \cdot 1.1875) / (17100 \cdot 1.00 + 0.4 \cdot 1400.01)$
= 0.0941 in.

Required Nozzle thickness under External Pressure per UG-28 : 0.0132 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit)	Dl	3.7500	in.
Parallel to Vessel Wall	Rn+tn+t	1.8750	in.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	0.9337	in.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: S3.
This calculation is valid for nozzles that meet all the requirements of paragraph UG-36. Please check the Code carefully, especially for nozzles that are not isolated or do not meet Code spacing requirements. To force the computation of areas for small nozzles go to Tools->Configuration and check the box to force the UG-37 area or App 1-10 computation.*

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures	ta	= 0.1566	in.
Wall Thickness per UG16(b),	tr16b	= 0.1250	in.
Wall Thickness, shell/head, internal pressure	trb1	= 0.7300	in.
Wall Thickness	tb1 = max(trb1, tr16b)	= 0.7300	in.
Wall Thickness	tb2 = max(trb2, tr16b)	= 0.1250	in.
Wall Thickness per table UG-45	tb3	= 0.1971	in.

Determine Nozzle Thickness candidate [tb]:

= min[tb3, max(tb1,tb2)]

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Nozzle Calcs. : S3 Nozl: 9 5:58p Sep 18,2012

$$= \min[0.197 , \max(0.730 , 0.125)]$$
$$= 0.1971 \text{ in.}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

$$= \max(t_a , t_b)$$
$$= \max(0.1566 , 0.1971)$$
$$= 0.1971 \text{ in.}$$

Available Nozzle Neck Thickness = $0.875 * 0.436 = 0.382 \text{ in.}$ --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, $t_g = 0.382$, $t_r = 0.094$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.295$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -155 F

MDMT of Nozzle-Shell/Head Weld for the Nozzle (UCS-66(a)1(b)), Curve: B

Govrn. thk, $t_g = 0.382$, $t_r = 0.094$, $c = 0.0625 \text{ in.}$, $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.295$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -155 F

Governing MDMT of all the sub-joints of this Junction : -155 F

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ANSI B16.5/47 flanges per UCS-66(c) -20 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(b) -55 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(c) -155 F

Where the Stress Reduction Ratio per UCS-66(b)(1)(b) is :

$$\text{Design Pressure/Ambient Rating} = 1400.01/2220.00 = 0.631$$

Note: Using the minimum value from (b)(1)(b) and (b)(1)(c) above
as the calculated nozzle flange MDMT.

Weld Size Calculations. Description: S3

Intermediate Calc. for nozzle/shell Welds T_{min} 0.3735 in.

Results Per UW-16.1:

Required Thickness Actual Thickness
Nozzle Weld 0.2500 = Min per Code 0.2651 = 0.7 * W_o in.

NOTE : Skipping the nozzle attachment weld strength calculations.
Per UW-15(b)(2) the nozzles exempted by UG-36(c)(3)(a)
(small nozzles) do not require a weld strength check.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 1439.495 psig

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Nozzle Calcs. : T3 & T4 Nozl: 10 5:58p Sep 18,2012

INPUT VALUES, Nozzle Description: T3 & T4 From : 60

Pressure for Reinforcement Calculations	P	150.000	psig
Temperature for Internal Pressure	Temp	650	F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	S	18800.00	psi
Shell Allowable Stress At Ambient	Sa	20000.00	psi
Inside Diameter of Cylindrical Shell	D	18.1250	in.
Shell Finished (Minimum) Thickness	t	0.1875	in.
Shell Internal Corrosion Allowance	c	0.0625	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		9.7918	ft.
User Entered Minimum Design Metal Temperature		-20.00	F

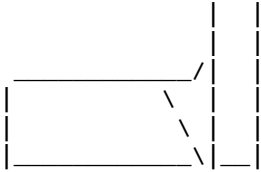
Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		OD	
Layout Angle		90.00	deg
Diameter		1.0000	in.
Size and Thickness Basis		Nominal	
Nominal Thickness	tn	80	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0625	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	6.0000	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.1875	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.1875	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		300	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure + static head.

Nozzle Sketch (may not represent actual weld type/configuration)





Insert Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: T3 & T4

ASME Code, Section VIII, Division 1, 2010, 2011a, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 1.315 in.
Actual Thickness Used in Calculation 0.179 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]
= $(P \cdot R) / (S \cdot E - 0.6 \cdot P)$ per UG-27 (c)(1)
= $(150.00 \cdot 9.1250) / (18800 \cdot 1.00 - 0.6 \cdot 150.00)$
= 0.0732 in.

Reqd thk per UG-37(a)of Nozzle Wall, Trn [Int. Press]
= $(P \cdot R_o) / (S \cdot E + 0.4 \cdot P)$ per Appendix 1-1 (a)(1)
= $(150.00 \cdot 0.6575) / (17100 \cdot 1.00 + 0.4 \cdot 150.00)$
= 0.0057 in.

UG-40, Limits of Reinforcement : [Internal Pressure]

Parallel to Vessel Wall (Diameter Limit)	D1	2.1640	in.
Parallel to Vessel Wall, opening length	d	1.0820	in.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	0.2913	in.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: T3 & T4.
This calculation is valid for nozzles that meet all the requirements of paragraph UG-36. Please check the Code carefully, especially for nozzles that are not isolated or do not meet Code spacing requirements. To force the computation of areas for small nozzles go to Tools->Configuration and check the box to force the UG-37 area or App 1-10 computation.*

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures	ta	= 0.0682	in.
Wall Thickness per UG16(b),	tr16b	= 0.1250	in.
Wall Thickness, shell/head, internal pressure	trb1	= 0.1357	in.
Wall Thickness	tb1 = max(trb1, tr16b)	= 0.1357	in.
Wall Thickness	tb2 = max(trb2, tr16b)	= 0.1250	in.
Wall Thickness per table UG-45	tb3	= 0.1785	in.

Determine Nozzle Thickness candidate [tb]:
= min[tb3, max(tb1, tb2)]
= min[0.178 , max(0.136 , 0.125)]
= 0.1357 in.

Minimum Wall Thickness of Nozzle Necks [tUG-45]:
= max(ta, tb)

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Nozzle Calcs. : T3 & T4

Noz1: 10 5:58p Sep 18,2012

$$= \max(0.0682 , 0.1357)$$
$$= 0.1357 \text{ in.}$$

Available Nozzle Neck Thickness = $0.875 * 0.179 = 0.157$ in. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, $t_g = 0.157$, $t_r = 0.006$, $c = 0.0625$ in. , $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.061$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -155 F

MDMT of Nozzle-Shell/Head Weld for the Nozzle (UCS-66(a)1(b)), Curve: B

Govrn. thk, $t_g = 0.157$, $t_r = 0.006$, $c = 0.0625$ in. , $E^* = 1.00$
Stress Ratio = $t_r * (E^*) / (t_g - c) = 0.061$, Temp. Reduction = 140 F

Min Metal Temp. w/o impact per UCS-66 -20 F
Min Metal Temp. at Required thickness (UCS 66.1) -155 F

Governing MDMT of all the sub-joints of this Junction : -155 F

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ANSI B16.5/47 flanges per UCS-66(c) -20 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(b) -155 F
Flange MDMT with Temp reduction per UCS-66(b)(1)(c) -155 F

Where the Stress Reduction Ratio per UCS-66(b)(1)(b) is :

$$\text{Design Pressure/Ambient Rating} = 150.00/740.00 = 0.203$$

*Note: Using the minimum value from (b)(1)(b) and (b)(1)(c) above
as the calculated nozzle flange MDMT.*

Weld Size Calculations. Description: T3 & T4

Intermediate Calc. for nozzle/shell Welds $T_{min} = 0.1165$ in.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	$0.0816 = 0.7 * t_{min}$	$0.1326 = 0.7 * W_o$ in.

NOTE : Skipping the nozzle attachment weld strength calculations.
Per UW-15(b)(2) the nozzles exempted by UG-36(c)(3)(a)
(small nozzles) do not require a weld strength check.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 192.729 psig

The Drop for this Nozzle is : 0.0239 in.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 6.2114 in.

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Nozzle Schedule:

Description	Nominal Flange			Noz. O/Dia in.	Wall Thk in.	Re-Pad		Cut Length in.
	Size in.	Sch/Type Cls				ODia in.	Thick in.	
T3 & T4	1.000	80	WNF	1.315	0.179	-	-	6.21
S3	2.000	XXS	WNF	2.375	0.436	-	-	8.58
S2	3.000	XXS	WNF	3.500	0.600	8.00	0.625	8.68
T1/T2	4.000	40	WNF	4.500	0.237	-	-	8.80
S1	6.000	XXS	WNF	6.625	0.864	10.12	0.625	9.17

General Notes for the above table:

The Cut Length is the Outside Projection + Inside Projection + Drop + In Plane Shell Thickness. This value does not include weld gaps, nor does it account for shrinkage.

In the case of Oblique Nozzles, the Outside Diameter must be increased. The Re-Pad WIDTH around the nozzle is calculated as follows:
Width of Pad = (Pad Outside Dia. (per above) - Nozzle Outside Dia.)/2

For hub nozzles, the thickness and diameter shown are those of the smaller and thinner section.

Nozzle Material and Weld Fillet Leg Size Details:

Nozzle	Material	Shl Grve Weld in.	Noz Shl/Pad Weld in.	Pad OD Weld in.	Pad Grve Weld in.	Inside Weld in.
T3 & T4	SA-106 B	0.188	0.188	-	-	-
S3	SA-106 B	0.750	0.375	-	-	-
S2	SA-106 B	0.750	0.375	0.500	0.625	-
T1/T2	SA-106 B	0.188	0.250	-	-	-
S1	SA-106 B	0.750	0.375	0.500	0.625	-

Note: The Outside projections below do not include the flange thickness.

Nozzle Miscellaneous Data:

Nozzle	Elevation/Distance From Datum ft.	Layout Angle deg.	Projection		Installed In Component
			Outside in.	Inside in.	
T3 & T4	9.792	90.00	6.00	0.00	Channel
S3	1.719	90.00	7.75	0.00	Shell
S2	1.719	270.00	7.75	0.00	Shell
T1/T2	0.443	110.57	7.34	0.00	Left Channel
S1	8.509	270.00	7.75	0.00	Shell

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Nozzle Summary :

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Nozzle Calculation Summary:

Description	MAWP psig	Ext	MAPNC psig	UG45 [tr]	Weld Path	Areas or Stresses
T1/T2	254.78	OK 0.136	OK	Passed
T1/T2	162.64	OK 0.136	OK	Passed
S2	1439.48	OK	...	OK 0.251	OK	Passed
S1	1439.48	OK	...	OK 0.325	OK	Passed
S3	1439.48	OK 0.197	OK	NoCalc[*]
T3 & T4	192.73	OK 0.136	OK	NoCalc[*]
Min. - Nozzles	162.64	T1/T2				

[*] - This was a small opening and the areas were not computed or the MAWP of this connection could not be computed because the longitudinal bending stress was greater than the hoop stress.

Note: MAWPs (Internal Case) shown above are at the High Point.

Check the Spatial Relationship between the Nozzles

From Node	Nozzle Description	X Coordinate,	Layout Angle,	Dia. Limit
20	T1/T2	5.320	110.570	8.302
40	S2	18.623	270.000	4.875
40	S1	100.103	270.000	10.044
40	S3	18.623	90.000	3.750
60	T3 & T4	117.502	90.000	2.164

The nozzle spacing is computed by the following:

= Sqrt(ll² + lc²) where

ll - Arc length along the inside vessel surface in the long. direction.

lc - Arc length along the inside vessel surface in the circ. direction

If any interferences/violations are found, they will be noted below.

No interference violations have been detected !

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Minimum Design Metal Temperature Results Summary :

Description	Notes	Curve	Basic MDMT F	Reduced MDMT F	UG-20(f) MDMT F	Thickness ratio	Gov Thk in.	E*
Channel Flang[11]		B	-20	-43	-20	0.775	0.188	1.000
Channel Flang[11]		B	-20	-43	-20	0.775	0.188	1.000
Left Head [10]		B	-20	-50	-20	0.699	0.168	1.000
Left Head [7]		B	-20	-55	-20	0.597	0.188	1.000
Left Channel [8]		B	-20	-55	-20	0.597	0.188	1.000
Shell [8]		B	16	10	-20	0.941	0.750	1.000
Channel [8]		B	-20	-55	-20	0.597	0.188	1.000
Right Head [10]		B	-20	-50	-20	0.699	0.168	1.000
Right Head [7]		B	-20	-55	-20	0.597	0.188	1.000
T1/T2 [1]		B	-20	-55		0.550	0.188	1.000
Nozzle Flg [4]			-20	-155		0.136		
S2 [1]		B	6	-20	-20	0.910	0.625	1.000
Nozzle Flg [4]			-20	-155		0.300		
S1 [1]		B	6	-20	-20	0.910	0.625	1.000
Nozzle Flg [4]			-20	-55		0.379		
S3 [1]		B	-20	-155		0.295	0.382	1.000
Nozzle Flg [4]			-20	-155		0.295		
T3 & T4 [1]		B	-20	-155		0.061	0.157	1.000
Nozzle Flg [4]			-20	-155		0.061		

Required Minimum Design Metal Temperature -20 F
Warmest Computed Minimum Design Metal Temperature -20 F

Notes:

- [!] - This was an impact tested material.
- [1] - Governing Nozzle Weld.
- [4] - ANSI Flange MDMT Calcs; Thickness ratio per UCS-66(b)(1)(c).
- [5] - ANSI Flange MDMT Calcs; Thickness ratio per UCS-66(b)(1)(b).
- [6] - MDMT Calculations at the Shell/Head Joint.
- [7] - MDMT Calculations for the Straight Flange.
- [8] - Cylinder/Cone/Flange Junction MDMT.
- [9] - Calculations in the Spherical Portion of the Head.
- [10] - Calculations in the Knuckle Portion of the Head.
- [11] - Calculated (Body Flange) Flange MDMT.
- [12] - Calculated Flat Head MDMT per UCS-66(3)

UG-84(b)(2) was not considered.
UCS-66(g) was not considered.
UCS-66(i) was not considered.

Notes:

Impact test temps were not entered in and not considered in the analysis.
UCS-66(i) applies to impact tested materials not by specification and
UCS-66(g) applies to materials impact tested per UG-84.1 General Note (c).
The Basic MDMT includes the (30F) PWHT credit if applicable.

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ASME TS Calc :

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Input Echo, Tubesheet Number 1, Description: Tubesheet

Shell Data:

Main Shell Description: Shell

Shell Design Pressure	Ps	1400.00	psig
Shell Thickness	ts	0.7500	in.
Shell Corrosion Allowance	cas	0.0625	in.
Inside Diameter of Shell	Ds	17.000	in.
Shell Circumferential Joint Efficiency	Esw	1.000	
Shell Temperature for Internal Pressure	Ts	650.00	F
Shell Material		SA-516	70

*Note: Using 2 * Yield for Discontinuity Stress Allowable (UG-23(e)), Sps.
Make sure that material properties at this temperature are not
time-dependent for Material: SA-516 70*

Shell Material UNS Number		K02700	
Shell Allowable Stress at Temperature	Ss	18800.00	psi
Shell Allowable Stress at Ambient		20000.00	psi

Channel Description: Left Channel

Channel Type:		Cylinder	
Channel Design Pressure	Pt	150.00	psig
Channel Thickness	tc	0.1875	in.
Channel Corrosion Allowance	cac	0.0625	in.
Inside Diameter of Channel	Dc	18.125	in.
Channel Design Temperature	TEMPC	650.00	F
Channel Material		SA-516	70

*Note: Using 2 * Yield for Discontinuity Stress Allowable (UG-23(e)), Sps.
Make sure that material properties at this temperature are not
time-dependent for Material: SA-516 70*

Channel Material UNS Number		K02700	
Channel Allowable Stress at Temperature	Sc	18800.00	psi
Channel Allowable Stress at Ambient		20000.00	psi

Tube Data:

Number of Tube Holes	Nt	104	
Tube Wall Thickness	et	0.1090	in.
Tube Outside Diameter	D	1.0000	in.
Total Straight Tube Length	Lt	96.00	in.
Straight Tube Length (bet. inner tubsht faces) L		92.00	in.
Design Temperature of the Tubes		650.00	F
Tube Material		SA-178	C
Tube Material UNS Number		K03503	
Is this a Welded Tube		No	
Tube Material Specification used		Wld. tube	
Tube Allowable Stress at Temperature		14600.00	psi
Tube Allowable Stress At Ambient		14600.00	psi
Tube Yield Stress At design Temperature	Syt	27400.00	psi
Tube Pitch (Center to Center Spacing)	P	1.2500	in.
Tube Layout Pattern		Triangular	
Fillet Weld Leg	af	0.0000	in.
Groove Weld Leg	ag	0.1090	in.

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ASME TS Calc :

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Tube-Tubesheet Joint Weld Type Full Strength
Method for Tube-Tubesheet Jt. Allow. UW-20
Tube-Tubesheet Joint Classification b-1

Radius to Outermost Tube Hole Center ro 8.005 in.
Largest Center-to-Center Tube Distance Ul 1.7500 in.
Length of Expanded Portion of Tube ltx 0.0000 in.
Tube-side pass partition groove depth hg 0.0000 in.

Tubesheet Data:

Tubesheet TYPE: Fixed Tubesheet Exchanger, Conf B

Tubesheet Design Metal Temperature T 650.00 F
Tubesheet Material Specification SA-516 70

*Note: Using 2 * Yield for Discontinuity Stress Allowable (UG-23(e)), Sps.
Make sure that material properties at this temperature are not
time-dependent for Material: SA-516 70*

Tubesheet Material UNS Number K02700
Tubesheet Allowable Stress at Temperature S 18800.00 psi
Tubesheet Allowable Stress at Ambient Tt 20000.00 psi
Thickness of Tubesheet h 2.0000 in.
Tubesheet Corr. Allowance (Shell side) Cats 0.0625 in.
Tubesheet Corr. Allowance (Channel side) Catc 0.0625 in.
Tubesheet Outside Diameter A 23.500 in.
Area of the Untubed Lanes AL 31.718 in²

Additional Data for Fixed/Floating Tubesheet Exchangers:

Unsupported Tube Span under consideration l 15.500 in.
Tube End condition corresponding to Span (l) k 1.00
Tubesheet Metal Temp. at Rim T' 500.00 F
Shell Metal Temp. at Tubesheet T'S 571.00 F
Channel Metal Temp. at Tubesheet T'C 429.00 F
Perform Differential Pressure Design N
Run Multiple Load Cases YES
Shell Side Vacuum Pressure Pexts 15.0000 psig

Mean Shell Metal Temp. along Shell len. Tsm 572.59 F
Mean Tube Metal Temp. along Tube length Ttm 475.00 F

Expansion Joint Type Thick Flanged/Flued type
User Exp. Joint Spring Rate (Corroded) Kjc 2149840.00 lb./in.
User Exp. Joint Spring Rate (Uncorroded) Kjnc 2149840.00 lb./in.
Expansion Joint Inside Diameter Dj 17.000 in.
Outside Diameter of Expansion Joint RODE 31.000 in.
Expansion Joint Corrosion Allowance Caj 0.0625 in.
Junction Stress Reduction option Perform Plastic Calculation

Additional Data for Gasketed Tubesheets:

Tubesheet Gasket on which Side Channel
Flange Outside Diameter A 23.500 in.
Flange Inside Diameter B 18.125 in.
Flange Face Outside Diameter Fod 19.750 in.
Flange Face Inside Diameter Fid 18.125 in.
Gasket Outside Diameter Go 19.125 in.

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Gasket Inside Diameter	Gi	18.125	in.
Small end Hub thk.	g0	0.1875	in.
Large end Hub thk.	g1	0.5625	in.
Gasket Factor,	m	2.50	
Gasket Design Seating Stress	y	2900.00	psi
Flange Facing Sketch	Code Sketch	1a	
Column for Gasket Seating	Code Column	II	
Gasket Thickness	tg	0.1250	in.
Full face Gasket Flange Option	Program Selects		
Length of Partition Gasket	lp	18.125	in.
Width of Partition Gasket	wp	0.3130	in.
Partition Gasket Factor,	mPart	2.5000	
Partition Gasket Design Seating Stress	yPart	2900.00	psi
Partition Gasket Facing Sketch	Code Sketch	1a	
Partition Gasket Column for Gasket Seating	Code Column	II	

Bolting Information:

Diameter of Bolt Circle	C	21.750	in.
Nominal Bolt Diameter	dB	0.6250	in.
Type of Thread Series	UNC Thread Series		
Number of Bolts	n	16	
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Weld between Flange and Shell/Channel		0.0000	in.

Tubesheet Integral with		Shell	
Tubesheet Extended as Flange		Yes	
Thickness of Extended Portion of Tubesheet	Tf	2.0000	in.
Is Bolt Load Transferred to the Tubesheet		Yes	

Is Exchanger in Creep range (skip EP, Use 3S for Sps) NO

ASME TubeSheet Results per Part UHX, 2010, 2011a

Elasticity/Expansion Material Properties:

Shell - TE-1 Carbon & Low Alloy Steels, Group 1
Shell - TM-1 Carbon Steels with C<= 0.3%

Th. Exp. Coeff. Metal Temp. along Len	572.6	F	0.0000073452	/F
Elastic Mod. at Design Temperature	650.0	F	0.26000E+08	psi
Th. Exp. Coeff. Metal Temp. at Tubsht	571.0	F	0.0000073420	/F
Elastic Mod. at Metal Temp. along Len	572.6	F	0.26719E+08	psi
Elastic Mod. at Ambient Temperature	70.0	F	0.29400E+08	psi

Channel - TE-1 Carbon & Low Alloy Steels, Group 1
Channel - TM-1 Carbon Steels with C<= 0.3%

Th. Exp. Coeff. Metal Temp. at Tubsht	429.0	F	0.0000071580	/F
Elastic Mod. at Design Temperature	650.0	F	0.26000E+08	psi
Elastic Mod. at Ambient Temperature	70.0	F	0.29400E+08	psi

Tubes - TE-1 Carbon & Low Alloy Steels, Group 1
Tubes - TM-1 Carbon Steels with C<= 0.3%

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Th. Exp. Coeff. Metal Temp. along Len	475.0 F	0.0000072500 /F
Elastic Mod. at Design Temperature	650.0 F	0.26000E+08 psi
Elastic Mod. at Metal Temp. along Len	475.0 F	0.27450E+08 psi
Elastic Mod. at Tubsht. Design Temp.	650.0 F	0.26000E+08 psi
Elastic Mod. at Ambient Temperature	70.0 F	0.29400E+08 psi

TubeSheet - TE-1 Carbon & Low Alloy Steels, Group 1
TubeSheet - TM-1 Carbon Steels with C<= 0.3%

Th. Exp. Coeff. Metal Temp. at Rim	500.0 F	0.0000073000 /F
Elastic Mod. at Design Temperature	650.0 F	0.26000E+08 psi
Elastic Mod. at Metal Temp. at Rim	500.0 F	0.27300E+08 psi
Elastic Mod. at Ambient Temperature	70.0 F	0.29400E+08 psi

Note:

The Elasticity and Alpha values are taken from Tables in ASME II D.
Please insure these properties are consistent with the
type of Material for the tubes, shell, channel etc.

Tube Required Thickness under Internal Pressure (Tubeside pressure) :

Thickness Due to Internal Pressure:

$$= (P*(D/2-CAE)) / (S*E+0.4*P) \text{ per Appendix 1-1 (a)(1)}$$
$$= (165.00*(1.0000/2-0.000))/(14600.00*1.00+0.4*165.00)$$
$$= 0.0056 + 0.0000 = 0.0056 \text{ in.}$$

Tube Required Thickness under External Pressure (Shellside pressure) :

External Pressure Chart CS-2 at 650.00 F
Elastic Modulus for Material 25125000.00 psi

Results for Max. Allowable External Pressure (Emawp):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
0.1090	1.00	92.00	9.17	50.0000	0.0130691	13399.55

$$\text{EMAWP} = (2.167/(D/T)-0.0833)*B = 2048.8315 \text{ psig}$$

Results for Reqd Thickness for Ext. Pressure (Tca):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
0.0843	1.00	92.00	11.86	50.0000	0.0078218	12453.27

$$\text{EMAWP} = (4*B)/(3*(D/T)) = (4 * 12453.2715)/(3 * 11.8589) = 1400.1616 \text{ psig}$$

Summary of Tube Required Thickness Results:

Total Required Thickness including Corrosion all.	0.0843 in.
Allowable Internal Pressure at Corroded thickness	3486.85 psig
Required Internal Design Pressure	165.00 psig
Allowable External Pressure at Corroded thickness	2048.83 psig
Required External Design Pressure	1400.00 psig
Required Thickness due to Shell Side pressure	0.0843 in.

Detailed Results for load Case 3 un-corr. (Ps + Pt - Th)

Intermediate Calculations For Tubesheets Extended As Flanges:

ASME Code, Section VIII, Div. 1, 2010, 2011a

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ASME TS Calc :

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Gasket Contact Width, $N = (Goc - Gic) / 2$ 0.500 in.
Basic Gasket Width, $b_0 = N / 2.0$ 0.250 in.
Effective Gasket Width, $b = b_0$ 0.250 in.
Gasket Reaction Diameter, $G = (Go + Gi) / 2.0$ 18.625 in.

ASME Maximum Circumferential Spacing between Bolts per App. 2 eq. (3) [Bsmax]:

$$\begin{aligned} &= 2a + 6t / (m + 0.5) \\ &= 2 * 0.625 + 6 * 2.000 / (2.50 + 0.5) \\ &= 5.250 \text{ in.} \end{aligned}$$

Actual Circumferential Bolt Spacing [Bs]:

$$\begin{aligned} &= C * \sin(\pi / n) \\ &= 21.750 * \sin(3.142 / 16) \\ &= 4.243 \text{ in.} \end{aligned}$$

ASME Moment Multiplier for Bolt Spacing per App. 2 eq. (7) [Bsc]:

$$\begin{aligned} &= \max(\sqrt{Bs / (2a + t)}, 1) \\ &= \max(\sqrt{4.243 / (2 * 0.625 + 2.000)}, 1) \\ &= 1.1426 \end{aligned}$$

Bolting Information for UNC Thread Series (Non Mandatory):

Distance Across Corners for Nuts 1.175 in.
Circular Wrench End Diameter a 1.750 in.

	Minimum	Actual	Maximum
Bolt Area, in ²	2.159	3.232	
Radial distance bet. hub and bolts	0.750	1.625	
Radial distance bet. bolts and the edge	0.750	0.875	
Circumferential spacing between bolts	1.500	4.243	5.250

Flange Design Bolt Load, Seating Condition W : 67382.77 lb.
Flange Design Bolt Load, Operating Condition Wm1: 53965.54 lb.

Results for ASME Fixed Tubesheet Calculations for Configuration b.

Results for Tubesheet Calculations Original Thickness :

UHX-13.5.1 Step 1:

Compute the Tube Expansion Depth Ratio [rho]:

$$\begin{aligned} &= ltx / h \text{ (modified for corrosion if present)} \\ &= 0.0000 / 2.0000 = 0.0000 \text{ (must be } 0 \leq \rho \leq 1 \text{)} \end{aligned}$$

Compute the Effective Tube Hole Diameter [d*]:

$$\begin{aligned} &= \text{Max}(dt - 2tt * (Et / E) (St / S) (rho), dt - 2tt) \\ &= \text{Max}(1.0000 - 2 * 0.1090 * (26000000 / 26000000) * \\ &\quad (14600 / 18800) * (0.000), 1.0000 - 2 * 0.1090) \\ &= 1.0000 \text{ in.} \end{aligned}$$

Compute the Equivalent Outer Tube Limit Circle Diameter [Do]:

$$= 2 * ro + dt = 2 * 8.005 + 1.000 = 17.010 \text{ in.}$$

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ASME TS Calc :

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Determine the Basic Ligament Efficiency for Shear [μ]:

$$= (p - dt)/p = (1.2500 - 1.0000) / 1.2500 = 0.2000$$

Compute the Equivalent Outer Tube Limit Radius [a_o]:

$$= D_o / 2 = 17.0100 / 2 = 8.5050 \text{ in.}$$

Compute the Effective Tube Pitch [p^*]:

$$\begin{aligned} &= p / \sqrt{1 - 4 * \min(AL * CNV_factor, 4 * D_o * p) / (\pi * D_o^2) } \\ &= 1.2500 / \sqrt{1 - 4 * \min(31.72 * 1.000, 4 * 17.010 * 1.250)} \\ &\quad / (3.141 * 17.010^2) \\ &= 1.3476 \text{ in.} \end{aligned}$$

Compute the Effective Ligament Efficiency for Bending [μ^*]:

$$= (p^* - d^*) / p^* = (1.3476 - 1.0000) / 1.3476 = 0.2579$$

Compute the Ratio [Rhos]:

$$= a_s / a_o = 8.5000 / 8.5050 = 0.999412$$

Compute the Ratio [Rhoc]:

$$= a_c / a_o = 9.3125 / 8.5050 = 1.094944$$

Compute Parameter [x_t]:

$$\begin{aligned} &= 1 - N_t * ((dt - 2 * t_t) / (2 * a_o))^2 \\ &= 1 - 104 * ((1.0000 - 2 * 0.1090) / (2 * 8.5050))^2 = 0.7802 \end{aligned}$$

Determine Parameter [x_s]:

$$\begin{aligned} &= 1 - N_t * (dt / (2 * a_o))^2 \\ &= 1 - 104 * (1.0000 / (2 * 8.5050))^2 = 0.6406 \end{aligned}$$

Determine the Value [$h'g$]:

$$\begin{aligned} &= \text{Max}((h_g - \text{CATC}), 0) \quad (\text{For pressure only cases}) \\ &= \text{Max}((0.000 - 0.000), 0) = 0.000 \text{ in.} \end{aligned}$$

UHX-13.5.2 Step 2:

Determine the Axial Shell Stiffness [K_s]:

$$\begin{aligned} &= \pi * t_s (D_s + t_s) E_s / L \\ &= 3.1416 * 0.7500 (17.0000 + 0.7500) 26000000 / 92.000 \\ &= 11819389.0000 \text{ psi} * \text{in.} \end{aligned}$$

Determine the Axial Tube Stiffness [K_t]:

$$\begin{aligned} &= \pi * t_t (D_t - t_t) E_t / L \\ &= 3.1416 * 0.1090 (1.0000 - 0.1090) 26000000 / 92.000 \\ &= 86226.2734 \text{ psi} * \text{in.} \end{aligned}$$

Compute the Stiffness Factor [K_{st}]:

$$= K_s / (N_t * K_t) = 11819389 / (104 * 86226.273) = 1.31802$$

Compute Factor [J]:

$$\begin{aligned} &= 1 / (1 + K_s / K_j) \\ &= 1 / (1 + 11819389 / 2149840) = 0.1538983 \quad (= 1 \text{ if No Exp. } J_t.) \end{aligned}$$

Compute Shell Coefficient [β_s]:

$$\begin{aligned} &= ((12 * (1 - \nu_s^2))^{0.25}) / ((D_s + t_s) * t_s)^{0.5} \\ &= ((12 * (1 - 0.30^2))^{0.25}) / ((17.0000 + 0.7500) * 0.7500)^{0.5} \\ &= 0.4982 \text{ 1/in.} \end{aligned}$$

Determine Shell Coefficient [ks]:

$$\begin{aligned}
 &= \text{betas} * \text{Es} * \text{ts}^3 / (6 * (1 - \text{nus}^2)) \\
 &= 0.498 * 26000000 * 0.750 ^{(3)} / (6 * (1 - 0.300^2)) \\
 &= 1000899.1875 \text{ psig}\cdot\text{in.}^2
 \end{aligned}$$

Determine Shell Coefficient [Lambdas]:

$$\begin{aligned}
 &= (6 * \text{Ds} * \text{ks}) / (\text{h}^3) * (1 + \text{h} * \text{betas} + 0.5 * (\text{h} * \text{betas})^2) \\
 &= 6 * 17.000 * 1000899 / (2.000^3) * (1 + 2.000 * 0.498 + 0.496) \\
 &= 31813154.0000 \text{ psig}
 \end{aligned}$$

Determine Shell Coefficient [deltaS]:

$$\begin{aligned}
 &= \text{Ds}^2 / (4 * \text{Es} * \text{Ts}) * (1 - \text{nus} / 2) \\
 &= 17.000 ^{(2)} / (4 * 26000000 * 0.750) * (1 - 0.3 / 2) \\
 &= 0.0000031494 \text{ in./psi}
 \end{aligned}$$

Intermediate parameters for Tubesheet Gasketed on the Channel Side:

betac, kc, deltaC, Lambdac = 0

UHX-13.5.3 Step 3:**E*/E and nu* for Triangular pattern from Fig. UHX-11.3.**

$$\begin{aligned}
 \text{h/p} &= 1.600000 ; \mu^* = 0.257927 \\
 \text{E}^*/\text{E} &= 0.240335 ; \nu^* = 0.371831 ; \text{E}^* = 6248720. \text{ psi}
 \end{aligned}$$

Compute the Tube Bundle Stiffness Factor [Xa]:

$$\begin{aligned}
 &= ((24 * (1 - \nu^{*2}) * \text{Nt} * \text{Et} * \text{tt} * (\text{dt} - \text{tt}) * \text{ao}^2) / \\
 & \quad (\text{E}^* * \text{L} * \text{H}^3))^{(0.25)} \\
 &= ((24 * (1 - 0.372^2) * 104 * 26000000 * 0.1090 * \\
 & \quad (1.0000 - 0.1090) * 8.5050^2) / (6248720 * \\
 & \quad 92.00 * 2.000^3))^{(0.25)} \\
 &= 3.0401
 \end{aligned}$$

Values from Table UHX-13.1

$$\begin{aligned}
 \text{Zd} &= 0.052796 ; \text{Zv} = 0.103117 ; \text{Zm} = 0.491601 \\
 \text{Za} &= 0.226483\text{E}+01 ; \text{Zw} = 0.103117
 \end{aligned}$$

UHX-13.5.4 Step 4:**Compute the Diameter Ratio [K]:**

$$= \text{A} / \text{Do} = 23.5000 / 17.0100 = 1.3815$$

Compute Coefficient [F]:

$$\begin{aligned}
 &= (1 - \nu^*) / (\text{E}^*) * (\text{Lambdas} + \text{Lambdac} + \text{E} * \ln(\text{K})) \\
 &= (1 - 0.37) / (6248720) * (31813154 + 0.00 + \\
 & \quad 26000000 * \ln(1.38)) \\
 &= 4.0429
 \end{aligned}$$

Compute Parameter [Phi]:

$$= (1 + \nu^*) * \text{F} = (1 + 0.3718) * 4.0429 = 5.5461$$

Compute Parameter [Q1]:

$$\begin{aligned}
 &= (\text{Rhos} - 1 - \text{Phi} * \text{Zv}) / (1 + \text{Phi} * \text{Zm}) \\
 &= (0.9994 - 1 - 5.5461 * 0.1031) / (1 + 5.5461 * 0.4916) \\
 &= -0.153626248
 \end{aligned}$$

Compute Parameter [Qz1]:

$$= (\text{Zd} + \text{Q1} * \text{Zw}) / 2 * \text{Xa}^4$$

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$$= (0.05280 + -0.15363 * 0.10312) / 2 * 3.04015 ^{(4)} = 1.5784$$

Compute Parameter [Qz2]:

$$= (Zv + Q1 * Zm) / 2 * Xa^{(4)}$$

$$= (0.10312 + -0.15363 * 0.49160) / 2 * 3.04015 ^{(4)} = 1.1786$$

Compute Parameter [U]:

$$= (Zw + (Rhos - 1) * Zm) * Xa^{(4)} / (1 + Phi * Zm)$$

$$= (0.1031 + (0.9994 - 1) * 0.4916) * 3.04015 ^{(4)} / (1 + 5.5461 * 0.4916)$$

$$= 2.3572$$

UHX-13.5.5 Step 5:

Determine factor [gamab]:

$$= (Gc - C) / Do \text{ (config b)}$$

$$= (18.6250 - 21.7500) / 17.0100 = -0.18372$$

Compute Parameter [gamma]:

$$= 0.000 \text{ in. (For Pressure only cases)}$$

Calculate Parameter [OmegaS]:

$$= rhos * ks * Betas * deltaS(1 + h * Betas)$$

$$= 0.9994 * 1000899 * 0.4982 * 0.000003 (1 + 2.0000 * 0.4982)$$

$$= 3.1336 \text{ in.}^2$$

Calculate Parameter [Omega*S]:

$$= Ao^2 * (Rhos^2 - 1) * (Rhos - 1) / 4 - OmegaS$$

$$= 8.505^2 * (0.999^2 - 1) * (0.999 - 1) / 4 - 3.134$$

$$= -3.1336 \text{ in.}^2$$

Calculate Parameter [OmegaC]:

$$= rhoc * kc * Betac * deltaC(1 + h * Betac)$$

$$= 1.0949 * 0.00 * 0.0000 * 0.000000 (1 + 2.0000 * 0.0000)$$

$$= 0.0000 \text{ in.}^2$$

Calculate Parameter [Omega*C]:

$$= ao^2 [(Rhoc^2 + 1) * (Rhoc - 1) / 4 - (Rhos - 1) / 2] - OmegaC$$

$$= 8.50500^2 [(1.09494^2 + 1) * (1.09494 - 1) / 4 - (0.99941 - 1) / 2] - 0.00000$$

$$= 3.7967 \text{ in.}^2$$

Compute the Pressure [P*S]:

$$= 0 \text{ For Pressure only cases or Configurations d,e,f,A,B,C,D}$$

Compute the Pressure [P*C]:

$$= 0 \text{ For Pressure only cases or Configurations b,c,d,B,C,D}$$

UHX-13.5.6 Step 6:

Compute the Pressure [P's]:

$$= Ps * \{xs + 2(1 - xs)nut + [2/Kst(Ds/Do)^2]nus - [(rhos^2 - 1)/(J * Kst)] - [(1 - J)/(2J * Kst)] [(Dj^2 - (Ds)^2) / Do^2]\}$$

$$= 1400.000 * \{0.641 + 2(1 - 0.641) 0.300 +$$

$$[2/1.318 (17.000 / 17.010)^2] 0.300 -$$

$$[(0.999^2 - 1) / (0.154 * 1.318)] -$$

$$[(1 - 0.154) / (2 * 0.154 * 1.318)] [(17.000^2 - (17.000)^2) / 17.010^2]\}$$

$$= 1843.3975 \text{ psig}$$

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Compute the Pressure [P't]:

$$\begin{aligned} &= [xt + 2(1 - xt)nut + 1/(J * Kst)] * Pt \\ &= [0.780 + 2(1 - 0.780)0.300 + \\ &\quad 1/(0.15390 * 1.318)] * 150.000 \\ &= 876.3071 \text{ psig} \end{aligned}$$

Compute the Pressure [Pgama]:

$$\begin{aligned} &= Nt * Kt * gama / (pi * ao^2) \\ &= 104 * 86226.273 * 0.000 / (3.142 * 8.505^2) = 0.000 \text{ psig} \end{aligned}$$

Compute the Pressure [Pw]:

$$\begin{aligned} &= -gamab * U * W* / (2 * pi * ao^2) \\ &= --0.184 * 2.357 * 53965.54 / (2 * 3.142 * 8.505^2) \\ &= 51.4191 \text{ psig} \end{aligned}$$

Calculate the Pressure [Prim]:

$$\begin{aligned} &= -(U/ao^2)(Omega*S * Ps - Omega*C * Pt) \\ &= -(2.357 / 8.505^2)(-3.134 * 1400.000 - 3.797 * 150.000) \\ &= 161.5167 \text{ psig} \end{aligned}$$

Calculate the Pressure [POmega]:

$$\begin{aligned} &= U/ao^2(OmegaS * P*s - OmegaC * P*c) \\ &= 2.357 /8.5050^2(3.1336 * 0.0000 - 0.0000 * 0.0000) \\ &= 0.0000 \text{ psig} \end{aligned}$$

Determine the Effective Pressure [Pe]:

$$\begin{aligned} &= J * Kst / (1 + J * Kst *(Qz1 + (Rhos - 1) * Qz2)) * \\ &\quad (P's - P't + Pgama + Pw + Prim) \\ &= 0.1539E+00 * 1.318 / (1 + 0.154 * 1.318 *(1.578 + (0.999 - \\ &\quad 1) * 1.179)) * (1843.397 - 876.307 + 0.000 + 51.419 + 161.517) \\ &= 181.3287 \text{ psig} \end{aligned}$$

UHX-13.5.7 Step 7:

Determine Factor [Q2]:

$$\begin{aligned} &= [((Omega*S*Ps - Omega*C*Pt) - (Omegas*P*s - Omegac P*c))CNV_FAC + \\ &\quad W* * gamab/(2*pi)]/(1 + Phi*Zm) \\ &= [((-3.134 * 1400.000 - 3.797 * 150.000) - \\ &\quad (3.134 * 0.000 - 0.000 * 0.000)) * 1.000 + \\ &\quad 53965.5 * -0.184 / (2*3.141)]/(1 + 5.54611 * 0.49160) \\ &= -1753.511718750 \text{ lb.} \end{aligned}$$

Calculate Factor [Q3]:

$$\begin{aligned} &= Q1 + 2 * Q2 / (Pe * ao^(2)) \\ &= -0.154 + 2 * -1753.512 / (181.329 * 8.505 ^ (2)) = -0.421003 \end{aligned}$$

Fm Value from Table UHX-13.1 = 0.210501

The Tubesheet Bending Stress - Original Thickness [Sigma]:

$$\begin{aligned} &= (1.5 * Fm / mu*) * (2 * ao/(H - h'g))^ (2) * Pe \\ &= (1.5 * 0.2105 /0.2579) * (2 * 8.5050 / (2.000 - 0.000))^ (2) * 181.33 \\ &= 16057.0020 \text{ psi} \end{aligned}$$

The Allowable Tubesheet Bending Stress [Sigma allowed]:

$$= 1.5 * S = 1.5 * 18800.00 = 28200.00 \text{ psi}$$

The Tubesheet Bending Stress - Final Thickness [Sigmaaf]:

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ASME TS Calc :

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$$\begin{aligned} &= (1.5 * Fm/mu*) * (2 * ao/(h - h'g)^2 * Pe \\ &= (1.5 * 0.1440 / 0.2579) * (2 * 8.5050 / (1.238 - 0.000))^2 * 178.50 \\ &= 28199.3555 \text{ psi} \end{aligned}$$

Reqd Tubesheet Thickness, for Bending Stress (Including CA) [HReqB]:

$$= h + Cats + Catc = 1.2383 + 0.0000 + 0.0000 = 1.2383 \text{ in.}$$

UHX-13.5.8 Step 8:

Shear Stress check [Tau_limit]:

$$\begin{aligned} &= 3.2 * S * MU * h / Do \\ &= 3.2 * 18800.00 * 0.200 * 2.000 / 17.01 = 1414.70 \text{ psig} \end{aligned}$$

The Tubesheet Average Shear Stress - Original Thickness [Tau]:

$$\begin{aligned} &= (1 / (2*mu)) * (ao/h) * Pe \\ &= (1 / (2*0.200)) * (8.5050 / 2.000) * 181.329 \\ &= 1927.7510 \text{ psi} \end{aligned}$$

The Allowable Tubesheet Shear Stress [Tau allowed]:

$$= 0.8 * S = 0.8 * 18800.00 = 15040.00 \text{ psi}$$

Note: Tubesheet Shear Stress is probably low, use the following req. thk:

Tubesheet thickness (Incl. Corr.)= 0.1500 in.

Tubesheet Shear Stress = 12768.5088 psi

Reqd Tubesheet Thickness for Given Loadings (Including CA) [Hreqd] :

$$= \text{Max}(HreqB, HreqS) = \text{Max}(1.2383 , 0.1500) = 1.2383 \text{ in.}$$

UHX-13.5.9 Step 9:

The Ftmin and Ftmax Coefficients from Table UHX-13.2:

$$Ftmin = 0.4008 , Ftmax = 1.2041$$

First Extreme Tube Axial Stress from among all the tubes [Sigmat1]:

$$\begin{aligned} &= ((Ps * xs - Pt * xt) - Pe * Ftmin) / (Xt - Xs) \\ &= ((1400.00 * 0.6406 - 150.00 * 0.7802) - (181.329) * 0.401) / \\ &\quad (0.7802 - 0.6406)) \\ &= 5063.8682 \text{ psi} \end{aligned}$$

Second Extreme value of Tube Axial Stress from among all the tubes [Sigmat2]:

$$\begin{aligned} &= ((Ps * xs - Pt * xt) - Pe * Ftmax) / (Xt - Xs) \\ &= ((1400.00 * 0.6406 - 150.00 * 0.7802) - (181.329) * 1.204) / \\ &\quad (0.7802 - 0.6406)) \\ &= 4020.6965 \text{ psi} \end{aligned}$$

Maximum Tube Axial Stress [Sigmat,max]:

$$= \text{MAX}(|Sigmat1|, |Sigmat2|) = 5063.868 \text{ psi}$$

The Allowable Tube Stress, [SigmatA]

$$= Sot = 14600.0000 \text{ psi}$$

The Largest tube-to-tubesheet Joint Load [Wt]:

$$= \text{Sigmat,max} * \text{Tube Area} = 5063.87 * 0.3051 = 1545.03 \text{ lb.}$$

Tube Weld Size Results per UW-20:

Tube Strength [Ft]:

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$$= 3.1415 * t * (do - t) * Sa$$
$$= 3.1415 * 0.109 * (1.000 - 0.109) * 14600.00 = 4454.582 \text{ lb.}$$

Fillet Weld Strength, $F_f = 0.0$

Groove Weld Strength [Fg]:

$$= .85 * 3.1415 * a_g * (do + 0.67 * a_g) * S_w \text{ (but not } > F_t)$$
$$= .85 * 3.1415 * 0.109 * (1.000 + 0.67 * 0.109) * 14600.00$$
$$= 4454.5815 \text{ lb.}$$

Max. Allow. Tube-Tubesheet Joint load, L_{max}

$$= F_t = 4454.5815 \text{ lb.}$$

Design Strength Ratio [fd]:

$$= 1.0000$$

Weld Strength Factor [fw]:

$$= S_{ot} / (\text{Min}(S_{ot}, S)) = 1.0000$$

Min Weld Length [ar]:

$$= ((0.75 * do)^2 + 1.76 * t * (do - t) * f_w * f_d)^{1/2} - .75 * do$$
$$= 0.1064 \text{ in.}$$

Minimum Required Groove Weld Leg $a_{gr} = 0.1090 \text{ in.}$

Tube-Tubesheet Jt allowable, 4454.58 is \geq tube strength 4454.58 lb.

Note: This tube-tubesheet joint is a Full Strength joint

UHX-13.5.10 Step 10:

Shell Axial Membrane Allowable Stress:

$$= S_s * E_{sw} = 18800.00 * 1.00 = 18800.00 \text{ psi}$$

Axial Membrane Stress in Shell [Sigmasm_ax]:

$$= a_o^2 / ((D_s + t_s) * t_s) * [P_e + (R_{hos}^2 - 1) (P_s - P_t)] + a_s^2 * P_t / ((D_s + t_s) * t_s)$$
$$= 8.505^2 / ((17.000 + 0.7500) * 0.7500) * [181.33 + (0.999^2 - 1) (1400.00 - 150.00)] + 8.500^2 * 150.00 / ((17.00 + 0.7500) * 0.75)$$
$$= 1791.3718 \text{ psi}$$

UHX-13.5.11 Step 11:

Note: For a given Shell thickness of: 0.750 in.

Min. Shell len. adjacent to the tubesheet should be: 6.427 in.

The Shell Membrane Stress due to Joint Interaction [Sigmasm]:

$$= a_o^2 / ((D_s + t_s) * t_s) [P_e + (R_{hos}^2 - 1) (P_s - P_t)] + a_s^2 * P_t / ((D_s + t_s) * t_s)$$
$$= 8.505^2 / ((17.000 + 0.7500) * 0.7500) [181.33 + (0.999^2 - 1) (1400.00 - 150.00)] + 8.500^2 * 150.00 / ((17.00 + 0.7500) * 0.75)$$
$$= 1791.3718 \text{ psi}$$

The Shell Bending Stress due to Joint Interaction [Sigmasb]

$$= 6 * k_s / t_s^2 \{ \text{betas} [\delta S * P_s + a_s^2 * P_{starS} / (E_s * t_s)] + 6(1 - \nu^{*2}) / (E^{*}) (a_o / h)^3 (1 + h * \text{betas} / 2) [P_e (Z_v + Z_m * Q_1) + 2 / a_o^2 * Z_m * Q_2] \}$$

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ASME TS Calc :

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$$\begin{aligned}
&= 6 * 1000899 / 0.7500^2 \{ 0.498 [0.000 * 1400.000 + 8.5000^2 \\
&\quad * 0.0000 / (19500000)] + \\
&\quad 6(1 - 0.37^2) / (6248720) (8.51 / 2.00)^3 (1 + 2.00 * \\
&\quad 0.50 / 2) [181.3 (0.103 + 0.492 * -0.154) + \\
&\quad 2 / 8.51^2 * 0.492 * -1753.512] \} \\
&= 4286.6968 \text{ psi}
\end{aligned}$$

Shell Stress Summation vs. Allowable

$$\begin{aligned}
&| \text{Sigmasm} | + | \text{Sigmasb} | \leq 1.5 * S_s \\
&| 1791.4 | + | 4286.7 | \leq 28200.00 \text{ psi} \\
&6078.07 \text{ must be } < \text{ or } = 28200.00 \text{ psi}
\end{aligned}$$

Computations Completed for ASME Tubesheet Configuration b

Stress/Force summary for loadcase 3 un-corr. (Ps + Pt - Th):

Stress Description	Actual		Allowable	Pass/Fail
Tubesheet Bend. Stress	16057.0	<=	28200.0 psi	Ok
Tubesheet Shear Stress	1927.8	<=	15040.0 psi	Ok
Maximum Tube Stress	5063.9	<=	14600.0 psi	Ok
Maximum Force on any one Tube	1545.0	<=	4454.6 lb.	Ok
Axial Membrane Stress in Shell	1791.4	<=	18800.0 psi	Ok
Shell Stress (jt. inter.)	6078.1	<=	28200.0 psi	Ok

Thickness results for loadcase 3 un-corr. (Ps + Pt - Th):

Thickness (in.)	Required	Actual	P/F
Tubesheet Thickness :	1.2383	2.0000	Ok
Tube-Tubesheet Groove Weld Leg :	0.1090	0.1090	Ok

Fixed Tubesheet results per ASME UHX-13 2010, 2011a

Results for 16 Load Cases:

Case#	--Reqd. Thk. + CA Tbsht Extnsn	----- Tubesheet Stresses		Case	Pass/ Fail
		Bend	Allwd	Type	
1uc	0.376 0.733	4332	28200	Fvs+Pt-Th	Ok
2uc	1.332 ...	16709	28200	Ps+Fvt-Th	Ok
3uc	1.238 ...	16057	28200	Ps+Pt-Th	Ok
4uc	0.644 ...	11814	56400	Fvs+Fvt+Th	Ok
5uc	0.789 ...	15457	56400	Fvs+Pt+Th	Ok
6uc	0.174 ...	10320	56400	Ps+Fvt+Th	Ok
7uc	0.340 ...	11915	56400	Ps+Pt+Th	Ok
8uc	0.299 ...	2539	28200	Fvs+Fvt-Th	Ok
1c	0.473 0.733	4661	28200	Fvs+Pt-Th-Ca	Ok
2c	1.433 ...	17802	28200	Ps+Fvt-Th-Ca	Ok
3c	1.393 ...	17637	28200	Ps+Pt-Th-Ca	Ok
4c	0.732 ...	11831	56400	Fvs+Fvt+Th-Ca	Ok
5c	0.868 ...	15463	56400	Fvs+Pt+Th-Ca	Ok
6c	0.288 ...	11659	56400	Ps+Fvt+Th-Ca	Ok
7c	0.443 ...	13189	56400	Ps+Pt+Th-Ca	Ok

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ASME TS Calc :

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8c	0.333	...	3111	28200	72	15040	Fvs+Fvt-Th-Ca	Ok

Max:	1.4330	0.733	in.	0.631		0.397	(Str. Ratio)	

Load Case Definitions:

Fvs,Fvt - User-defined Shell-side and Tube-side vacuum pressures or 0.0.

Ps, Pt - Shell-side and Tube-side Design Pressures.

(+)-Th - With or Without Thermal Expansion.

Ca - With or Without Corrosion Allowance.

Shell Axial Membrane Stress Summary:

Case#	Shell Stresses				Shell Band Stress				Pass Fail
	Ten	Allwd	Cmp	Allwd	Ten	Allwd	Cmp	Allwd	
1uc	124	18800	Ok
2uc	1650	18800	Ok
3uc	1791	18800	Ok
4uc	2224	56400	-2224	-12932	Ok
5uc	2126	56400	-2126	-12932	Ok
6uc	557	56400	-557	-12932	Ok
7uc	458	56400	-458	-12932	Ok
8uc	25	18800	Ok
1c	157	18800	Ok
2c	1854	18800	Ok
3c	2031	18800	Ok
4c	2345	56400	-2345	-12784	Ok
5c	2223	56400	-2223	-12784	Ok
6c	471	56400	-471	-12784	Ok
7c	349	56400	-349	-12784	Ok
8c	36	18800	Ok
Max RATIO	0.108		0.183			

Tube, Shell and Channel Stress Summary:

Case#	Tube Stresses				Tube Loads Ld	Tube Loads Allwd	Shell Stress		Channel Stress		Pass Fail
	Ten	Allwd	Cmp	Allwd			Stress	Allwd	Stress	Allwd	
1uc	1096	14600	-1096	-11255	334	4454	11745	28200	Ok
2uc	5213	14600	1590	4454	17388	28200	Ok
3uc	5064	14600	1545	4454	6078	28200	Ok
4uc	5261	29200	-14	-11255	1605	4454	18338	56400	Ok
5uc	6046	29200	-764	-11255	1845	4454	24055	56400	Ok
6uc	9303	29200	2838	4454	763	56400	Ok
7uc	10088	29200	3078	4454	6480	56400	Ok
8uc	345	14600	-345	-11255	105	4454	5830	28200	Ok
1c	1232	14600	-1232	-11255	376	4454	13347	28200	Ok
2c	5416	14600	1652	4454	19998	28200	Ok
3c	5197	14600	1586	4454	7180	28200	Ok
4c	5476	29200	-358	-11255	1671	4454	21008	56400	Ok
5c	6338	29200	-1212	-11255	1934	4454	27410	56400	Ok
6c	9529	29200	2907	4454	795	56400	Ok
7c	10391	29200	3170	4454	7197	56400	Ok
8c	378	14600	-378	-11255	115	4454	6703	28200	Ok
Max RATIO	0.371		0.109		0.712		0.709		...		

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ASME TS Calc :

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Summary of Thickness Comparisons for 16 Load Cases:

Thickness (in.)	Required	Actual	P/F
Tubesheet Thickness :	1.4330	2.0000	Ok
Tubesheet Thickness Flanged Extension :	0.7328	2.0000	Ok
Tube Thickness :	0.0843	0.1090	Ok
Tube-Tubesheet Groove Weld Leg :	0.1090	0.1090	Ok

Min Shell length of thk, (0.750) adj. to tubesheet: 6.427 in.

Note: This is a full strength Tube to Tubesheet Joint.

Summary of Axial Differential Expansion bet. Shell and Tubes :

Due to Thermal Expansion Exp. Jt. Compresses by : -0.070 in.
Due to Pressure Exp. Jt. Extends by : +0.0302 in.
Due to Pressure + Thermal Exp. Jt. Compresses by : -0.070 in.

Tubesheet MAWP used to Compute Hydrotest Pressure:

Stress / Force Condition	Tubeside (0 shellside)		Shellside (0 tubeside)	
	MAWP	Stress Rat.	MAWP	Stress Rat.
Tubesheet Bending Stress	910.98	1.000	2215.76	1.000
Tubesheet Shear Stress	1115.96	1.000	6296.51	1.000
Tube Tensile Stress	1846.01	1.000	3773.98	1.000
Tube Compressive Stress	1115.96	0.683	----	----
Tube-Tubesheet Joint load	1354.01	1.000	3173.88	1.000
Shell Stress (Axial, Junc)	605.83	1.000	2048.82	0.519
Tube Pressure Stress	3486.84	1.000	2048.82	1.000
Tubesheet Extension Stress	224.58	0.000	----	----
Minimum MAWP	224.58		2048.82	

Tubesheet MAPnc used to Compute Hydrotest Pressure:

Stress / Force Condition	Tubeside (0 shellside)		Shellside (0 tubeside)	
	MAPnc	Stress Rat.	MAPnc	Stress Rat.
Tubesheet Bending Stress	1036.92	1.000	2633.09	1.000
Tubesheet Shear Stress	1792.75	1.000	7478.11	1.000
Tube Tensile Stress	2013.14	1.000	3896.03	1.000
Tube Compressive Stress	2111.44	1.000	----	----
Tube-Tubesheet Joint load	2013.14	1.000	3896.03	1.000
Shell Stress (Axial, Junc)	916.20	0.999	2633.09	0.411
Tube Pressure Stress	3486.84	1.000	2721.66	1.000
Tubesheet Extension Stress	224.58	0.000	----	----
Minimum MAPnc	224.58		2633.09	

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Vessel Design Summary : Step: 20 5:58p Sep 18,2012

Design Code: ASME Code Section VIII Division 1, 2010, 2011a

Diameter Spec : 18.500 in. OD
Vessel Design Length, Tangent to Tangent 10.10 ft.
Specified Datum Line Distance 0.00 ft.
Shell Material Specification SA-516 70
Nozzle Material Specification SA-106 B
Re-Pad Material Specification SA-516 70
Shell Side Design Temperature 650 F
Channel Side Design Temperature 650 F
Shell Side Design Pressure 1400.000 psig
Channel Side Design Pressure 150.000 psig
Shell Side Hydrostatic Test Pressure 1820.000 psig
Channel Side Hydrostatic Test Pressure 195.000 psig
Required Minimum Design Metal Temperature -20 F
Warmest Computed Minimum Design Metal Temperature -20 F
Wind Design Code No Wind Loads
Earthquake Design Code No Seismic

Element Pressures and MAWP: psig

Element Desc	Design Pres. + Stat. head	External Pressure	M.A.W.P	Corrosion Allowance
Left Head	150.658	0.000	217.957	0.0625
Left Channel	150.657	0.000	254.778	0.0625
Channel Flange	150.000	0.000	161.970	0.0625
Shell	1400.627	15.000	1439.484	0.0625
Channel Flange	150.000	0.000	161.970	0.0625
Channel	150.657	0.000	254.778	0.0625
Right Head	150.658	0.000	217.957	0.0625

Liquid Level: 1.51 ft. Dens.: 62.400 lb./ft³ Sp. Gr.: 1.000

Element Type	"To" Elev ft.	Length ft.	Element Thk in.	R e q d Int.	T h k Ext.	Joint Eff Long	Eff Circ
Ellipse	0.08	0.083	0.188	0.135	0.125	1.00	1.00
Cylinder	0.81	0.729	0.188	0.136	No Calc	1.00	1.00
Body Flg	1.05	0.240	1.750	1.684	1.664	1.00	1.00
Cylinder	8.89	7.667	0.750	0.732	0.186	1.00	1.00
Body Flg	9.29	0.240	1.750	1.684	1.664	1.00	1.00
Cylinder	10.02	0.729	0.188	0.136	No Calc	1.00	1.00
Ellipse	10.10	0.083	0.188	0.135	0.125	1.00	1.00

Element thicknesses are shown as Nominal if specified, otherwise are Minimum

Saddle Parameters:

Saddle Width 4.000 in.
Saddle Bearing Angle 120.000 deg.

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Vessel Design Summary : Step: 20 5:58p Sep 18,2012

Centerline Dimension	25.000	in.
Wear Pad Width	5.000	in.
Wear Pad Thickness	0.250	in.
Wear Pad Bearing Angle	132.000	deg.
Distance from Saddle to Tangent	6.625	in.

Baseplate Length	16.400	in.
Baseplate Thickness	0.250	in.
Baseplate Width	4.000	in.
Number of Ribs (including outside ribs)	2	
Rib Thickness	0.250	in.
Web Thickness	0.250	in.
Height of Center Web	24.000	in.

Summary of Maximum Saddle Loads, Operating Case :

Maximum Vertical Saddle Load	2018.29	lb.
Maximum Transverse Saddle Shear Load	0.00	lb.
Maximum Longitudinal Saddle Shear Load	0.00	lb.

Summary of Maximum Saddle Loads, Hydrotest Case :

Maximum Vertical Saddle Load	2095.50	lb.
Maximum Transverse Saddle Shear Load	0.00	lb.
Maximum Longitudinal Saddle Shear Load	0.00	lb.

Weights:

Fabricated - Bare W/O Removable Internals	3004.7	lbm
Shop Test - Fabricated + Water (Full)	3953.0	lbm
Shipping - Fab. + Rem. Intls.+ Shipping App.	3004.7	lbm
Erected - Fab. + Rem. Intls.+ Insul. (etc)	3004.7	lbm
Empty - Fab. + Intls. + Details + Wghts.	3004.7	lbm
Operating - Empty + Operating Liquid (No CA)	3899.0	lbm
Field Test - Empty Weight + Water (Full)	3953.0	lbm

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EJMA 1998, A2000 and ASME VIII-1 App 5 and App 26

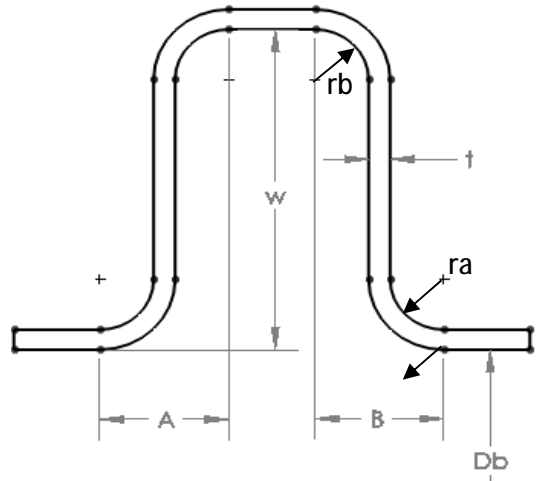
Shell Expansion Joint (Corroded State) Description

Materials and Conditions:

SA-299 B	Bellows material
26,000,000	E_b [psi] - bellows modulus
22,900	S_{ab} [psi] - allowed stress - bellows
3.00	C_m - material factor (1.5 Ann, 3 - cf)
1400.0	P [psi] - operating pressure

Dimensions:

17.000	D_b - inside diameter of bellows
6.500	w - convolution height
10.500	q - Convolution Pitch (A+B)
0.750	t - bellows thickness
2.250	ra - outside bellows radius
2.250	rb - inside bellows radius



Motion:

0.070	xc - axial compression - inch
0.000	xe - axial extension - inch

Constants: (per EJMA A-2, Cp, Cd, and Cf come from Fig C-24)

e = xc+xe	0.07+0 = 0.070
Dm = Db+w+t	17+6.5+0.75 = 24.250
Kr = 2*(q+xe)/(2*q)	2*(10.5+0)/(2*10.5) = 1.000
fv = q/(2.2*sqrt(Dm*t))	10.5/(2.2*SQRT(24.25*0.75)) = 1.119
fh = q/(2*w)	10.5/(2*6.5) = 0.808
Cp = PVELookup("CpTable", "Int2DLin", fh, fv)	0.505
Cf = PVELookup("CfTable", "Int2DLin", fh, fv)	1.205
Cd = PVELookup("CdTable", "Int2DLin", fh, fv)	2.424
Checkr = Min(ra,rb) >= (3*t)	MIN(2.25,2.25) >= (3*0.75) = Acceptable

Pressure and Deflection Stresses: (per EJMA A-2 Same as ASME VIII-1 App26-6 except Nc and Fi)

S2 [psi] = (P*Dm/(2*t))*(Kr/(0.571+2*w/q)) Circ. Stress from pressure (C-22)	(1400*24.25/(2*0.75))*(1/(0.571+2*6.5/10.5)) = 12,511
S3 [psi] = (P*w)/(2*t) Meridional membrane from pressure (C-23)	(1400*6.5)/(2*0.75) = 6,067
S4 [psi] = (P/2)*(w/t)^2*Cp Meridional bending from pressure (C-24)	(1400/2)*(6.5/0.75)^2*0.505 = 26,541
S5 [psi] = (Eb*t^2*e)/(2*w^3*Cf) Meridional membrane from deflection (C-25)	(26000000*0.75^2*0.07)/(2*6.5^3*1.205) = 1,547
S6 [psi] = (5*Eb*t*e)/(3*w^2*Cd) Meridional bending from deflection (C-26)	(5*26000000*0.75*0.07)/(3*6.5^2*2.424) = 22,215
St [psi] = 0.7*(S3+S4)+(S5+S6) Combined total stress	0.7*(6067+26541)+(1547+22215) = 46,587
Nc = IF(St>54000,(1860000/(St-54000))^3.4,100000000) EJMA predicted cycle life (C-27)	IF(46587>54000,(1860000/(46587-54000))^3.4,100000000) = 100,000,000
Fi [lb/in] = 1.7*(Dm*Eb*t^3)/(w^3*Cf) Bellows axial spring rate (lb/in) (C-29)	1.7*(24.25*26000000*0.75^3)/(6.5^3*1.205) = 1,366,173

Stress Evaluation:

CheckS2a = S2 <= Sab EJMA C-24	12511 <= 22900 = Acceptable
CheckS3S4 = S3 + S4 <= Cm*Sab EJMA C-24	6067 + 26541 <= 3*22900 = Acceptable
CheckS2b = S2 <= 1.5*Sab ASME Appendix 5-3(b)	12511 <= 1.5*22900 = Acceptable
CheckS3 = S3 <= 1.5*Sab ASME Appendix 5-3(b)	6067 <= 1.5*22900 = Acceptable
CheckS4 = S4 <= 1.5*Sab ASME Appendix 5-3(b)	26541 <= 1.5*22900 = Acceptable
CheckS4S5 = S4 + S5 <= 3*Sab ASME Appendix 5-3(b)	26541 + 1547 <= 3*22900 = Acceptable
CheckNc = Nc >= 100 ASME Appendix 5-3(b)	100000000 >= 100 = Acceptable