



Toase-e Park Sanati Gohar Ofogh
Petrochemical Co.
**CONCEPTUAL, BASIC and DETAIL DESIGN
ENGINEERING OF STYRENE PARK OFFSITE**



Document Title: Chiller (Evaporator) Mechanical calculation

Document No.: EI027-HSE-VD –ME–CAL–009- R1

Rev. R1

Page 1 of 193

STYRENE PARK OFFSITE

Document Title:
Chiller (Evaporator) Mechanical calculation

Rev.	Issued Date	DESCRIPTION	PREPARED	CHECKED	APPROVED
R1	25-05-2024	IFA	F.sh	M.O	A.M
R0	20-04-2024	IFA	F.sh	M.O	A.M



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REVISION RECORD SHEET

Page Page	Revisions							Page	Revisions						
	R0	R1	R2	R3	R4	R5	R6		R0	R1	R2	R3	R4	R5	R6
1	X	X						41	X	X					
2	X	X						42	X	X					
3	X	X						43	X	X					
4	X	X						44	X	X					
5	X	X						45	X	X					
6	X	X						46	X	X					
7	X	X						47	X	X					
8	X	X						48	X	X					
9	X	X						49	X	X					
10	X	X						50	X	X					
11	X	X						51	X	X					
12	X	X						52	X	X					
13	X	X						53	X	X					
14	X	X						54	X	X					
15	X	X						55	X	X					
16	X	X						56	X	X					
17	X	X						57	X	X					
18	X	X						58	X	X					
19	X	X						59	X	X					
20	X	X						60	X	X					
21	X	X						61	X	X					
22	X	X						62	X	X					
23	X	X						63	X	X					
24	X	X						64	X	X					
25	X	X						65	X	X					
26	X	X						66	X	X					
27	X	X						67	X	X					
28	X	X						68	X	X					
29	X	X						69	X	X					
30	X	X						70	X	X					
31	X	X						71	X	X					
32	X	X						72	X	X					
33	X	X						73	X	X					
34	X	X						74	X	X					
35	X	X						75	X	X					
36	X	X						76	X	X					
37	X	X						77	X	X					
38	X	X						78	X	X					
39	X	X						79	X	X					
40	X	X						80-193	X	X					

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DESIGN CALCULATION

In Accordance with ASME Section VIII Division 1

ASME Code Version : 2019

Analysis Performed by : #Replace this text with your company name and th

Job File : D:\ JAM-PETROELECTRIC\CHILLER-Rev.01

Date of Analysis : May 21,2024 12:29pm

PV Elite 23, January 2021

FileName : Chiller-Rev.01 -----

Vessel Design Summary: Step: 28 12:29pm May 21,2024

Vessel Design Summary:

ASME Code, Section VIII Division 1, 2019

Diameter Spec : 600.000 x 600.000 x 925.000 mm. ID
 Vessel Design Length, Tangent to Tangent 3563.53 mm.
 Specified Datum Line Distance 0.00 mm.
 Shell Side Design Temperature 120 °C
 Channel Side Design Temperature 85 °C
 Shell Side Design Pressure 22.000 bars
 Channel Side Design Pressure 6.800 bars
 Shop Shell Side Test Pressure 28.600 bars
 Shop Channel Side Test Pressure 8.840 bars
 Wind Design Code UBC
 Earthquake Design Code UBC-97

Materials of Construction:

Component Type	Material	Class	Thickness	UNS #	Normal ized	Impact Tested
Shell	SA-516 70	K02700	Yes	No
Head	SA-516 70	K02700	Yes	No
Cone	SA-516 70	K02700	Yes	No
Flange	SA-266 2	K03506	No	No
Flange	SA-350 LF2	1	...	K03011	No	Yes
Nozzle	SA-106 B	K03006	No	No
Nozzle	SA-333 6	K03006	No	Yes
Nozzle	SA-350 LF2	1	...	K03011	No	Yes
Re-Pad	SA-516 70	K02700	Yes	No
Nozzle Flg	SA-105	K03504	No	No
Nozzle Flg	SA-350 LF2	1	...	K03011	No	Yes
Tubes	SA-334 6	K03006	No	Yes
Tubesheet	SA-350 LF2	1	...	K03011	No	Yes
Flg Bolting	SA-193 B7	...	<= 2 1/2	G41400	No	No
Hrz Bolting	SA-193 B7	...	<= 2 1/2	G41400	No	No

Normalized is determined based on the UCS-66 material curve selection and Figure UCS-66.
 Impact Tested is based on material selection and material data properties.

Element Pressures and MAWP (bars & mm.):

Element Description or Type	Design Pressure + Stat. head	Ext. Press.	Element M.A.W.P	Corrosion Allowance	Str. Flg. Gov.	In Creep Range
CH. Head	6.854	1.00	32.100	3.0000	No	No
CH. Barrel	6.854	1.03	31.400	3.0000	N/A	No
CH. Flange	6.854	1.03	14.600	3.0000	N/A	No
SH. Flange	22.031	1.03	23.400	3.0000	N/A	No
Port Barrel	22.032	1.03	40.200	3.0000	N/A	No
SH. Cone	22.048	1.03	22.800	3.0000	N/A	No
SH. Barrel	22.048	1.03	26.300	3.0000	N/A	No
SH. Head	22.048	1.03	26.800	3.0000	No	No

Liquid Level: 925.00 mm. Dens.: 0.001 kg./cm³ Sp. Gr.: 0.533

Element Types and Properties:

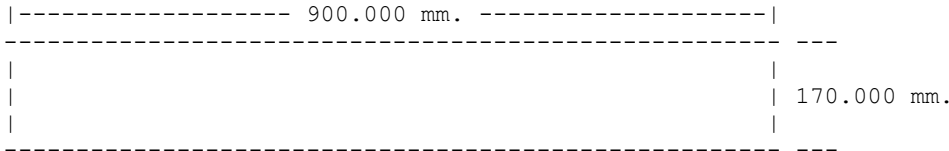
Vessel Design Summary: Step: 28 12:29pm May 21,2024

Element Type	"To" Elev mm.	Element Length mm.	Nominal Thickness mm.	Finished Thickness mm.	Reqd Thk Internal mm.	Reqd Thk External mm.	Long Eff	Circ Eff
Ellipse	50.0	50.0	12.0	10.0	4.5	4.6	1.00	1.00
Cylinder	350.0	300.0	10.0	10.0	4.5	4.7	1.00	1.00
Body Flg	435.0	85.0	90.0	60.0	58.2	57.3	1.00	1.00
Body Flg	598.5	85.0	90.0	60.0	57.9	53.5	1.00	1.00
Cylinder	748.5	150.0	12.0	12.0	7.9	4.1	1.00	1.00
Conical	1313.5	565.0	12.0	12.0	11.7	8.6	1.00	1.00
Cylinder	3513.5	2200.0	12.0	12.0	10.5	7.8	1.00	1.00
Ellipse	3563.5	50.0	14.0	12.0	10.4	5.5	1.00	1.00

Saddle Parameters:

Saddle Width	140.000	mm.
Saddle Bearing Angle	120.000	deg.
Centerline Dimension	750.000	mm.
Wear Pad Width	300.000	mm.
Wear Pad Thickness	12.000	mm.
Wear Pad Bearing Angle	132.000	deg.
Distance from Saddle to Tangent	800.000	mm.
Baseplate Length	900.000	mm.
Baseplate Thickness	15.000	mm.
Baseplate Width	170.000	mm.
Number of Ribs (including outside ribs)	3	
Rib Thickness	12.000	mm.
Web Thickness	12.000	mm.
Height of Center Web	367.500	mm.
Number of Bolts in Baseplate	4	

Baseplate Sketch



Baseplate Plan View



Baseplate Side View

Maximum Tensile Bolt Load 114. Kgf

Summary of Maximum Saddle Loads, Operating Case:

Maximum Vertical Saddle Load	4630.43	Kgf
Maximum Transverse Saddle Shear Load	481.01	Kgf
Maximum Longitudinal Saddle Shear Load	962.03	Kgf

Summary of Maximum Saddle Loads, Operating Case, Un-Factored:

Maximum Vertical Saddle Load	5180.88	Kgf
Maximum Transverse Saddle Shear Load	1284.39	Kgf
Maximum Longitudinal Saddle Shear Load	1374.33	Kgf

Summary of Maximum Saddle Loads, Hydrotest Case :

Maximum Vertical Saddle Load	4025.37	Kgf
Maximum Transverse Saddle Shear Load	78.72	Kgf
Maximum Longitudinal Saddle Shear Load	37.89	Kgf

Vessel Design Summary: Step: 28 12:29pm May 21,2024

Local Stress Analysis Results:

Description	Analysis Type	Max Stress Ratio	High Stress Location	Pass Fail
S2 (6in.)	WRC-107/537	0.833	n/a	Passed

Weights:

Fabricated - Bare W/O Removable Internals	2800.6 kg.
Shop Test - Fabricated + Water (Full)	4834.0 kg.
Shipping - Fab. + Rem. Intls.+ Shipping App.	2800.6 kg.
Erected - Fab. + Rem. Intls.+ Insul. (etc)	2973.9 kg.
Empty - Fab. + Intls. + Details + Wghts.	2973.9 kg.
Operating - Empty + Operating Liquid (No CA)	4198.2 kg.
Field Test - Empty Weight + Water (Full)	4766.1 kg.

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FileName : Chiller-Rev.01 -----

Nozzle Summary:

Step: 26 12:29pm May 21,2024

Nozzle Calculation Summary:

Description	MAWP bars	Ext	MAPNC bars	UG-45	[tr] mm.	Weld Path	Areas or Stresses
T1 (3in.)	16.6	OK	4.50	OK	No Calc[*]
T1 (3in.)	16.6	OK	4.50	OK	No Calc[*]
T2 (3in.)	16.6	OK	4.51	OK	No Calc[*]
T2 (3in.)	16.6	OK	4.51	OK	No Calc[*]
S1 (4in.)	22.8	OK	...	OK	8.26	OK	Passed
S2 (6in.)	26.3	OK	...	OK	9.22	OK	Passed
V (2in.)	26.3	OK	7.80	OK	No Calc[*]
D (2in.)	26.3	OK	7.80	OK	No Calc[*]
PSV (3in.)	26.3	OK	7.80	OK	No Calc[*]
LG1 (2in.)	26.3	OK	6.42	OK	No Calc[*]
LG2 (2in.)	26.3	OK	6.42	OK	No Calc[*]

Nozzle MAWP Summary:

Minimum MAWP Nozzles : 22.8 Nozzle : S1 (4in.) [Shellside]
 Minimum MAWP Nozzles : 16.6 Nozzle : T2 (3in.) [Tubeside]

[*] - This was a small opening and the areas were not computed.

Note: MAWPs (Internal Case) shown above are at the High Point.

Check the Spatial Relationship between the Nozzles:

From Node	Nozzle Description	X Coordinate mm.	Layout Angle deg	Dia. Limit mm.
20	T1 (3in.)	200.000	108.819	163.130
20	T2 (3in.)	200.000	251.181	163.130
60	S1 (4in.)	898.525	270.000	201.662
70	S2 (6in.)	2413.525	90.000	310.145
70	V (2in.)	3363.526	90.000	113.600
70	D (2in.)	3363.526	270.000	113.600
70	PSV (3in.)	1483.525	90.000	150.862
70	LG1 (2in.)	1913.525	90.000	102.068
70	LG2 (2in.)	1913.525	270.000	102.068

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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FileName : Chiller-Rev.01 -----

Nozzle Schedule:

Step: 25 12:29pm May 21,2024

Nozzle Schedule:

Description	Nominal or Actual Size	Schd or FVC Type	Flg Type	Nozzle O/Dia in	Wall Thk mm	Reinforcing Diameter	Pad Thk mm	Cut Length mm	Flg Class
V (2in.)	2.000 in	Actual	LW	3.307	16.600	213.9	300
D (2in.)	2.000 in	Actual	LW	3.307	16.600	213.9	300
LG1 (2in.)	2.000 in	160	WN	2.375	8.738	162.9	300
LG2 (2in.)	2.000 in	160	WN	2.375	8.738	162.9	300
T1 (3in.)	3.000 in	80	WN	3.500	7.620	190.00	10.00	230.4	150
T2 (3in.)	3.000 in	80	WN	3.500	7.620	190.00	10.00	230.4	150
PSV (3in.)	3.000 in	160	WN	3.500	11.125	190.00	12.00	214.1	300
S1 (4in.)	4.000 in	120	WN	4.500	11.125	220.00	12.00	196.8	300
S2 (6in.)	6.000 in	80	WN	6.625	10.973	300.00	12.00	199.7	300

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 (mm.):

Description	Material	Shl Grve Weld	Noz Shl/Pad Weld	Pad OD Weld	Pad Grve Weld	Inside Weld
V (2in.)	SA-350 LF2	12.000	10.000
D (2in.)	SA-350 LF2	12.000	10.000
LG1 (2in.)	SA-333 6	12.000	10.000
LG2 (2in.)	SA-333 6	12.000	10.000
T1 (3in.)	SA-106 B	10.000	10.000	8.000	10.000	...
T2 (3in.)	SA-106 B	10.000	10.000	8.000	10.000	...
PSV (3in.)	SA-333 6	12.000	10.000	10.000	12.000	...
S1 (4in.)	SA-333 6	12.000	10.000	10.000	12.000	...
S2 (6in.)	SA-333 6	12.000	10.000	10.000	12.000	...

Note: The Outside projections below do not include the flange thickness.

Nozzle Miscellaneous Data:

Description	Elev/Distance From Datum mm.	Layout Angle deg	Proj Outside mm.	Proj Inside mm.	Installed in Component
V (2in.)	3363.525	90.0	200.00	0.00	SH. Barrel
D (2in.)	3363.525	270.0	200.00	0.00	SH. Barrel
LG1 (2in.)	1913.525	90.0	150.00	0.00	SH. Barrel
LG2 (2in.)	1913.525	270.0	150.00	0.00	SH. Barrel
T1 (3in.)	200.000	108.8	200.00	0.00	CH. Barrel
T2 (3in.)	200.000	251.2	200.00	0.00	CH. Barrel
PSV (3in.)	1483.525	90.0	200.00	0.00	SH. Barrel
S1 (4in.)	898.525	270.0	180.00	0.00	SH. Cone
S2 (6in.)	2413.525	90.0	180.00	0.00	SH. Barrel

FileName : Chiller-Rev.01

Bill of Material:

Step: 24 12:29pm May 21,2024

Bill of Materials:

QTY	DESCRIPTION	MATERIAL
1	ELLIPTICAL HEAD: 2.0 X 1, 12.0mm. THK X 600.0mm. ID X 50.0mm.	SA-516 70
1	CYLINDER: 10.0mm. THK X 600.0mm. ID X 300.0mm.	SA-516 70
1	BODY FLANGE: 90.0mm. THK X 760.0mm. OD	SA-266 2
1	BODY FLANGE: 90.0mm. THK X 760.0mm. OD	SA-350 LF2
1	CYLINDER: 12.0mm. THK X 600.0mm. ID X 150.0mm.	SA-516 70
1	CONE: 12.0mm. THK X 925.0mm. ID X 565.0mm.	SA-516 70
1	CYLINDER: 12.0mm. THK X 925.0mm. ID X 2200.0mm.	SA-516 70
1	ELLIPTICAL HEAD: 2.0 X 1, 14.0mm. THK X 925.0mm. ID X 50.0mm.	SA-516 70
1	INSULATION: 200mm X 60mm THK	
1	INSULATION: 300mm X 60mm THK	
1	INSULATION: 85mm X 60mm THK	
1	TUBESHEET: 650mm X 60mm THK	SA-350 LF2
376	TUBES: 2300mm X 19mm DIA X 1mm THK	SA-334 6
1	INSULATION: 84mm X 60mm THK	
1	INSULATION: 150mm X 60mm THK	
1	INSULATION: 565mm X 60mm THK	
2	SADDLE: 140mm X 120 DEG	SA-516 70
1	INSULATION: 2200mm X 60mm THK	
1	INSULATION: 281mm X 60mm THK	
1	GASKET: 650mm. OD X 624mm. ID	...
20	BODY FLANGE BOLTS: 22mm. DIA	SA-193 B7
40	NUTS FOR BODY FLANGE BOLTS: 22mm. DIA	...
1	NAMEPLATE	...

MDMT Summary: Step: 27 12:29pm May 21,2024

Minimum Design Metal Temperature Results Summary :

Description	Notes	Curve	Basic MDMT °C	Reduced MDMT °C	UG-20 (f) MDMT °C	Thickness ratio	Gov Thk mm.	E*	PWHT reqd
SH. Flange	[11]	!	-46	-46		0.566	12.000	1.00	No
Port Barrel	[8]	D	-48	-48	-29	0.562	12.000	1.00	No
SH. Cone	[8]	D	-48	-48	-29	0.998	12.000	1.00	No
SH. Barrel	[8]	D	-48	-48	-29	0.863	12.000	1.00	No
SH. Head	[10]	D	-48	-48	-29	0.849	12.000	1.00	No
SH. Head	[7]	D	-48	-48	-29	0.706	14.000	1.00	No
S1 (4in.)	[1]	D	-48	-48	-29	0.716	12.000	1.00	No
Nozzle Flg	[5]	!	-46	-89					
S2 (6in.)	[1]	D	-48	-48	-29	0.833	12.000	1.00	No
Nozzle Flg	[5]	!	-46	-89					
V (2in.)	[1]	D	-48	-48	-29	0.833	12.000	1.00	No
Nozzle Flg	[5]	!	-46	-46					
D (2in.)	[1]	D	-48	-48	-29	0.835	12.000	1.00	No
Nozzle Flg	[5]	!	-46	-46					
PSV (3in.)	[1]	D	-48	-48	-29	0.833	12.000	1.00	No
Nozzle Flg	[5]	!	-46	-89					
LG1 (2in.)	[1]	D	-46	-104		0.104	7.645	1.00	No
Nozzle Flg	[5]	!	-46	-89					
LG2 (2in.)	[1]	D	-46	-104		0.104	7.645	1.00	No
Nozzle Flg	[5]	!	-46	-89					
Tubesheet: SS	[13]	!	-46	-46		0.838	15.000	1.00	No

Warmest MDMT: -46 -46

CH. Flange	[11]	C	-44	-48	-29	0.466	10.000	1.00	No
CH. Head	[10]	D	-48	-48	-29	0.454	10.000	1.00	No
CH. Head	[7]	D	-48	-48	-29	0.359	12.000	1.00	No
CH. Barrel	[8]	D	-48	-48	-29	0.462	10.000	1.00	No
T1 (3in.)	[1]	B	-8	-104	-29	0.070	6.668	1.00	No
Nozzle Flg	[5]	A	-18	-104					
T2 (3in.)	[1]	B	-8	-104	-29	0.070	6.668	1.00	No
Nozzle Flg	[5]	A	-18	-104					
Tubesheet: CS	[14]	!	-46	-46		0.838	15.000	1.00	No
Bolting	[21]		-48						

Warmest MDMT: -8 -46

Exchanger Side	Computed MDMT °C	Required MDMT °C	Pass/Fail
Shell	-46.0	-45.0	Pass
Channel/Tube	-46.0	-29.0	Pass

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
- [13] - Tubesheet MDMT, shell side, if applicable
- [14] - Tubesheet MDMT, tube side, if applicable

MDMT Summary: Step: 27 12:29pm May 21,2024

- [15] - Nozzle Material
- [16] - Shell or Head Material
- [17] - Impact Testing required
- [18] - Impact Testing not required, see UCS-66(b)(3)
- [20] - Cylinder/Cone Junction MDMT based on Longitudinal Stress considerations
- [21] - Body Flange Bolting Material
- [22] - Nozzle Flange Bolting Material

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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FileName : Chiller-Rev.01 -----

Warnings and Errors: Step: 0 12:29pm May 21,2024

Class From To : Basic Element Checks.

Class From To: Check of Additional Element Data

Warning: CH. Flange and SH. Flange

There are mating flanges in this model that have different design pressures. This will generate different bolt loads. If a tubesheet is in between these flanges, PV Elite will compute the mating flange loads for you. If there is not a tubesheet physically separating these flanges, you must enter in the mating flange loads into the flange dialog.

Warning:

There is a flange in this model connected to an element whose temperature is different. The bolt allowable stress is based on it's parent design temperature. You may need to manually adjust the operating bolt allowable stress to a value based upon a representative maximum temperature the bolts will be subject to.

Note:

PV Elite performs all calculations internally in Imperial Units to remain compliant with the ASME Code and any built in assumptions in the ASME Code formulas. The finalized results are reflected to show the set of selected units for this analysis.

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FileName : Chiller-Rev.01 -----

Input Echo:

Step: 1 12:29pm May 21,2024

PV Elite Vessel Analysis Program: Input Data

Exchanger Design Pressures and Temperatures

Shell Side Design Pressure	22	bars
Channel Side Design Pressure	6.8	bars
Shell Side Design Temperature	120.0	°C
Channel Side Design Temperature	85.0	°C
Radiography, Shell Side	RT-1	
Radiography, Channel Side	RT-1	
Service Type, Shell Side	None	
Service Type, Channel Side	None	
MDMT (CET), Shell Side	-45.0	°C
MDMT (CET), Tube Side	-29.0	°C
User defined MAWP, Shell Side	0	bars
User defined MAWP, Channel Side	0	bars
User defined MAPnc, Shell Side	0	bars
User defined MAPnc, Channel Side	0	bars
User defined Test Pres., Shell Side	0	bars
User defined Test Pres., Channel Side	0	bars

Projection of Nozzle from Vessel Top	0	mm.
Projection of Nozzle from Vessel Bottom	0	mm.
Type of Construction	Welded	
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	Y	

Shop Pressure Test:

Type of Pressure Test	UG-99(b) Note [35]
Pressure Test Position	Horizontal

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	UBC-94/97	
UBC Design Wind Speed	125	Km/hr
UBC Exposure Constant	D: Flat, unobstructed	
UBC Importance Factor	1.0	
UBC Base Elevation	20000	mm.
UBC Percent Wind for Hydrotest	33.0	

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Using User defined Wind Press. Vs Elev.	N
Damping Factor (Beta) for Wind (Ope)	0.0100
Damping Factor (Beta) for Wind (Empty)	0.0000
Damping Factor (Beta) for Wind (Filled)	0.0000
Seismic Design Code	UBC 1997
UBC Seismic Zone (1=1,2=2a,3=2b,4=3,5=4)	5
UBC Importance Factor	1.250
UBC Seismic Coefficient Ca	0.400
UBC Seismic Coefficient Cv	0.560
UBC Seismic Coefficient Nv	1.000
UBC Horizontal Force Factor	3.000
Apply Allowables per paragraph 1612.3.2	No
Design Pressure + Static 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 Current w/Addenda or Code Year

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
For thickness ratios <= 0.35, MDMT will be -155F (-104C)	Yes
For PWHT & P1 Materials the MDMT can be < -55F (-48C)	No
Using Metric Material Databases, ASME II D	No
Calculate B31.3 type stress for Nozzles with Loads	Yes
Reduce the MDMT due to lower membrane stress	Yes
Consider Longitudinal Stress in MDMT calcs. (Div. 1)	Yes

Complete Listing of Vessel Elements and Details:

Element From Node	10
Element To Node	20
Element Type	Elliptical
Description	CH. Head
Distance "FROM" to "TO"	50 mm.
Inside Diameter	600 mm.
Element Thickness	10 mm.
Internal Corrosion Allowance	3 mm.
Nominal Thickness	12 mm.
External Corrosion Allowance	0 mm.
Design Internal Pressure	6.8 bars
Design Temperature Internal Pressure	85 °C
Design External Pressure	1 bars
Design Temperature External Pressure	85 °C
Effective Diameter Multiplier	1.2
Material Name	SA-516 70 [Normalized]
Allowable Stress, Ambient	137.9 N./mm ²
Allowable Stress, Operating	137.9 N./mm ²
Allowable Stress, Hydrotest	235.8 N./mm ²
Material Density	0.00775 kg./cm ³
P Number Thickness	30.988 mm.
Yield Stress, Operating	241.8 N./mm ²
UCS-66 Chart Curve Designation	D
External Pressure Chart Name	CS-2
UNS Number	K02700
Product Form	Plate
Efficiency, Longitudinal Seam	1.0

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Efficiency, Circumferential Seam 1.0
 Elliptical Head Factor 2.0
 Weld is pre-Heated No

Element From Node 10
 Detail Type Liquid
 Detail ID Liquid: 10
 Dist. from "FROM" Node / Offset dist 0 mm.
 Height/Length of Liquid 600 mm.
 Liquid Density 0.0009181 kg./cm³

Element From Node 10
 Detail Type Insulation
 Detail ID Ins: 10
 Dist. from "FROM" Node / Offset dist -150 mm.
 Height/Length of Insulation 200 mm.
 Thickness of Insulation 60 mm.
 Density 0.00023 kg./cm³

 Element From Node 20
 Element To Node 30
 Element Type Cylinder
 Description CH. Barrel
 Distance "FROM" to "TO" 300 mm.
 Inside Diameter 600 mm.
 Element Thickness 10 mm.
 Internal Corrosion Allowance 3 mm.
 Nominal Thickness 10 mm.
 External Corrosion Allowance 0 mm.
 Design Internal Pressure 6.8 bars
 Design Temperature Internal Pressure 85 °C
 Design External Pressure 1.0342 bars
 Design Temperature External Pressure 85 °C
 Effective Diameter Multiplier 1.2
 Material Name SA-516 70 [Normalized]
 Efficiency, Longitudinal Seam 1.0
 Efficiency, Circumferential Seam 1.0
 Weld is pre-Heated No

Element From Node 20
 Detail Type Liquid
 Detail ID Liquid: 20
 Dist. from "FROM" Node / Offset dist 0 mm.
 Height/Length of Liquid 600 mm.
 Liquid Density 0.0009181 kg./cm³

Element From Node 20
 Detail Type Insulation
 Detail ID Ins: 20
 Dist. from "FROM" Node / Offset dist 0 mm.
 Height/Length of Insulation 300 mm.
 Thickness of Insulation 60 mm.
 Density 0.00023 kg./cm³

Element From Node 20
 Detail Type Nozzle
 Detail ID T1 (3in.)
 Dist. from "FROM" Node / Offset dist 150 mm.
 Nozzle Diameter 3 in.
 Nozzle Schedule 80
 Nozzle Class 150
 Layout Angle 108.819
 Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 11.862 Kgf

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Grade of Attached Flange	GR 1.1
Nozzle Matl	SA-106 B
Element From Node	20
Detail Type	Nozzle
Detail ID	T2 (3in.)
Dist. from "FROM" Node / Offset dist	150 mm.
Nozzle Diameter	3 in.
Nozzle Schedule	80
Nozzle Class	150
Layout Angle	251.181
Blind Flange (Y/N)	N
Weight of Nozzle (Used if > 0)	11.862 Kgf
Grade of Attached Flange	GR 1.1
Nozzle Matl	SA-106 B
Element From Node	20
Detail Type	Weight
Detail ID	PASS PAR.0
Dist. from "FROM" Node / Offset dist	150 mm.
Miscellaneous Weight	70 Kgf
Offset from Element Centerline	0 mm.

Element From Node	30
Element To Node	40
Element Type	Flange
Description	CH. Flange
Distance "FROM" to "TO"	85 mm.
Flange Inside Diameter	600 mm.
Element Thickness	60 mm.
Internal Corrosion Allowance	3 mm.
Nominal Thickness	90 mm.
External Corrosion Allowance	0 mm.
Design Internal Pressure	6.8 bars
Design Temperature Internal Pressure	85 °C
Design External Pressure	1.0342 bars
Design Temperature External Pressure	85 °C
Effective Diameter Multiplier	1.2
Material Name	SA-266 2
Allowable Stress, Ambient	137.9 N./mm ²
Allowable Stress, Operating	137.9 N./mm ²
Allowable Stress, Hydrotest	223.4 N./mm ²
Material Density	0.00775 kg./cm ³
P Number Thickness	30.988 mm.
Yield Stress, Operating	229.2 N./mm ²
UCS-66 Chart Curve Designation	C
External Pressure Chart Name	CS-2
UNS Number	K03506
Product Form	Forgings
Perform Flange Stress Calculation (Y/N)	Y
Weight of Standard Flange	0 Kgf
Class of Standard Flange	
Grade of Standard Flange	
Weld is pre-Heated	No
Element From Node	30
Detail Type	Liquid
Detail ID	Liquid: 30
Dist. from "FROM" Node / Offset dist	0 mm.
Height/Length of Liquid	600 mm.
Liquid Density	0.0009181 kg./cm ³
Element From Node	30
Detail Type	Insulation

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Detail ID	Ins: 30
Dist. from "FROM" Node / Offset dist	0 mm.
Height/Length of Insulation	85 mm.
Thickness of Insulation	60 mm.
Density	0.00023 kg./cm ³

Element From Node	40
Element To Node	50
Element Type	Flange
Description	SH. Flange
Distance "FROM" to "TO"	85 mm.
Flange Inside Diameter	600 mm.
Element Thickness	60 mm.
Internal Corrosion Allowance	3 mm.
Nominal Thickness	90 mm.
External Corrosion Allowance	0 mm.
Design Internal Pressure	22 bars
Design Temperature Internal Pressure	120 °C
Design External Pressure	1.0342 bars
Design Temperature External Pressure	120 °C
Effective Diameter Multiplier	1.2
Material Name	SA-350 LF2 [Impact Tested]
Allowable Stress, Ambient	137.9 N./mm ²
Allowable Stress, Operating	137.9 N./mm ²
Allowable Stress, Hydrotest	223.4 N./mm ²
Material Density	0.00775 kg./cm ³
P Number Thickness	30.988 mm.
Yield Stress, Operating	223.6 N./mm ²
UCS-66 Chart Curve Designation	Impact Tested
External Pressure Chart Name	CS-2
UNS Number	K03011
Class / Thickness / Grade	1::
Product Form	Forgings
Perform Flange Stress Calculation (Y/N)	Y
Weight of Standard Flange	0 Kgf
Class of Standard Flange	
Grade of Standard Flange	
Weld is pre-Heated	No

Element From Node	40
Detail Type	Liquid
Detail ID	Liquid: 40
Dist. from "FROM" Node / Offset dist	0 mm.
Height/Length of Liquid	600 mm.
Liquid Density	0.0005329 kg./cm ³

Element From Node	40
Detail Type	Insulation
Detail ID	Ins: 40
Dist. from "FROM" Node / Offset dist	0 mm.
Height/Length of Insulation	85 mm.
Thickness of Insulation	60 mm.
Density	0.00023 kg./cm ³

Element From Node	50
Element To Node	60
Element Type	Cylinder
Description	Port Barrel
Distance "FROM" to "TO"	150 mm.
Inside Diameter	600 mm.
Element Thickness	12 mm.
Internal Corrosion Allowance	3 mm.

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Nominal Thickness	12	mm.
External Corrosion Allowance	0	mm.
Design Internal Pressure	22	bars
Design Temperature Internal Pressure	120	°C
Design External Pressure	1.0342	bars
Design Temperature External Pressure	120	°C
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	[Normalized]
Allowable Stress, Ambient	137.9	N./mm ²
Allowable Stress, Operating	137.9	N./mm ²
Allowable Stress, Hydrotest	235.8	N./mm ²
Material Density	0.00775	kg./cm ³
P Number Thickness	30.988	mm.
Yield Stress, Operating	236	N./mm ²
UCS-66 Chart Curve Designation	D	
External Pressure Chart Name	CS-2	
UNS Number	K02700	
Product Form	Plate	
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	
Weld is pre-Heated	No	

Element From Node	50	
Detail Type	Liquid	
Detail ID	Liquid: 50	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Liquid	600	mm.
Liquid Density	0.0005329	kg./cm ³

Element From Node	50	
Detail Type	Insulation	
Detail ID	Ins: 50	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Insulation	150	mm.
Thickness of Insulation	60	mm.
Density	0.00023	kg./cm ³

Element From Node	60	
Element To Node	70	
Element Type	Conical	
Description	SH. Cone	
Distance "FROM" to "TO"	565	mm.
Inside Diameter	600	mm.
Element Thickness	12	mm.
Internal Corrosion Allowance	3	mm.
Nominal Thickness	12	mm.
External Corrosion Allowance	0	mm.
Design Internal Pressure	22	bars
Design Temperature Internal Pressure	120	°C
Design External Pressure	1.0342	bars
Design Temperature External Pressure	120	°C
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	[Normalized]
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	
Cone Diameter at "To" End	925	mm.
Design Length of Cone	565	mm.
Half Apex Angle of Cone	30.0	degrees
Toriconical (Y/N)	N	
Weld is pre-Heated	No	

Element From Node	60	
Detail Type	Liquid	
Detail ID	Liquid: 60	

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Dist. from "FROM" Node / Offset dist 0 mm.
 Height/Length of Liquid 925 mm.
 Liquid Density 0.0005329 kg./cm³

Element From Node 60
 Detail Type Insulation
 Detail ID Ins: 60
 Dist. from "FROM" Node / Offset dist 0 mm.
 Height/Length of Insulation 565 mm.
 Thickness of Insulation 60 mm.
 Density 0.00023 kg./cm³

Element From Node 60
 Detail Type Nozzle
 Detail ID S1 (4in.)
 Dist. from "FROM" Node / Offset dist 150 mm.
 Nozzle Diameter 4 in.
 Nozzle Schedule 120
 Nozzle Class 300
 Layout Angle 270.0
 Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 25.056 Kgf
 Grade of Attached Flange GR 1.1
 Nozzle Matl SA-333 6 [Impact Tested]

Element From Node 70
 Element To Node 80
 Element Type Cylinder
 Description SH. Barrel
 Distance "FROM" to "TO" 2200 mm.
 Inside Diameter 925 mm.
 Element Thickness 12 mm.
 Internal Corrosion Allowance 3 mm.
 Nominal Thickness 12 mm.
 External Corrosion Allowance 0 mm.
 Design Internal Pressure 22 bars
 Design Temperature Internal Pressure 120 °C
 Design External Pressure 1.0342 bars
 Design Temperature External Pressure 120 °C
 Effective Diameter Multiplier 1.2
 Material Name SA-516 70 [Normalized]
 Efficiency, Longitudinal Seam 1.0
 Efficiency, Circumferential Seam 1.0
 Weld is pre-Heated No

Element From Node 70
 Detail Type Saddle
 Detail ID Left Saddle
 Dist. from "FROM" Node / Offset dist 230 mm.
 Width of Saddle 140 mm.
 Height of Saddle at Bottom 750 mm.
 Saddle Contact Angle 120.0
 Height of Composite Ring Stiffener 0 mm.
 Width of Wear Plate 300 mm.
 Thickness of Wear Plate 12 mm.
 Contact Angle, Wear Plate (degrees) 132.0
 Friction coefficient 0.3000001
 Moment Factor 3.0
 Dimension E at base (optional) 0 mm.
 Circumferential Eff. over Saddle 1.0
 Circumferential Eff. at Midspan 1.0
 Tangent to Tangent dist. (optional) 0 mm.

Element From Node 70

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Detail Type	Saddle	
Detail ID	Right Saddle	
Dist. from "FROM" Node / Offset dist	1650	mm.
Width of Saddle	140	mm.
Height of Saddle at Bottom	750	mm.
Saddle Contact Angle	120.0	
Height of Composite Ring Stiffener	0	mm.
Width of Wear Plate	300	mm.
Thickness of Wear Plate	12	mm.
Contact Angle, Wear Plate (degrees)	132.0	
Friction coefficient	0.30000001	
Moment Factor	3.0	
Dimension E at base (optional)	0	mm.
Circumferential Eff. over Saddle	1.0	
Circumferential Eff. at Midspan	1.0	
Tangent to Tangent dist. (optional)	0	mm.
Element From Node	70	
Detail Type	Liquid	
Detail ID	Liquid: 70	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Liquid	925	mm.
Liquid Density	0.0005329	kg./cm ³
Element From Node	70	
Detail Type	Insulation	
Detail ID	Ins: 70	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Insulation	2200	mm.
Thickness of Insulation	60	mm.
Density	0.00023	kg./cm ³
Element From Node	70	
Detail Type	Nozzle	
Detail ID	S2 (6in.)	
Dist. from "FROM" Node / Offset dist	1100	mm.
Nozzle Diameter	6	in.
Nozzle Schedule	80	
Nozzle Class	300	
Layout Angle	90.0	
Blind Flange (Y/N)	N	
Weight of Nozzle (Used if > 0)	47.688	Kgf
Grade of Attached Flange	GR 1.1	
Nozzle Matl	SA-333 6	[Impact Tested]
Element From Node	70	
Detail Type	Nozzle	
Detail ID	V (2in.)	
Dist. from "FROM" Node / Offset dist	2050	mm.
Nozzle Diameter	2	in.
Nozzle Schedule	None	
Nozzle Class	300	
Layout Angle	90.0	
Blind Flange (Y/N)	N	
Weight of Nozzle (Used if > 0)	9.9881	Kgf
Grade of Attached Flange	GR 1.1	
Nozzle Matl	SA-350 LF2	[Impact Tested]
Element From Node	70	
Detail Type	Nozzle	
Detail ID	D (2in.)	
Dist. from "FROM" Node / Offset dist	2050	mm.
Nozzle Diameter	2	in.
Nozzle Schedule	None	
Nozzle Class	300	
Layout Angle	270.0	

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Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 9.9881 Kgf
 Grade of Attached Flange GR 1.1
 Nozzle Matl SA-350 LF2 [Impact Tested]

Element From Node 70
 Detail Type Nozzle
 Detail ID PSV (3in.)
 Dist. from "FROM" Node / Offset dist 170 mm.
 Nozzle Diameter 3 in.
 Nozzle Schedule 160
 Nozzle Class 300
 Layout Angle 90.0
 Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 16.474 Kgf
 Grade of Attached Flange GR 1.1
 Nozzle Matl SA-333 6 [Impact Tested]

Element From Node 70
 Detail Type Nozzle
 Detail ID LG1 (2in.)
 Dist. from "FROM" Node / Offset dist 600 mm.
 Nozzle Diameter 2 in.
 Nozzle Schedule 160
 Nozzle Class 300
 Layout Angle 90.0
 Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 6.0973 Kgf
 Grade of Attached Flange GR 1.1
 Nozzle Matl SA-333 6 [Impact Tested]

Element From Node 70
 Detail Type Nozzle
 Detail ID LG2 (2in.)
 Dist. from "FROM" Node / Offset dist 600 mm.
 Nozzle Diameter 2 in.
 Nozzle Schedule 160
 Nozzle Class 300
 Layout Angle 270.0
 Blind Flange (Y/N) N
 Weight of Nozzle (Used if > 0) 8.7691 Kgf
 Grade of Attached Flange GR 1.1
 Nozzle Matl SA-333 6 [Impact Tested]

Element From Node 70
 Detail Type Weight
 Detail ID BUNDLE MISS.0
 Dist. from "FROM" Node / Offset dist 100 mm.
 Miscellaneous Weight 100 Kgf
 Offset from Element Centerline 160 mm.

Element From Node 70
 Detail Type Weight
 Detail ID BUNDLE MISS.0
 Dist. from "FROM" Node / Offset dist 1250 mm.
 Miscellaneous Weight 100 Kgf
 Offset from Element Centerline 160 mm.

Element From Node 80
 Element To Node 90
 Element Type Elliptical
 Description SH. Head
 Distance "FROM" to "TO" 50 mm.
 Inside Diameter 925 mm.

Input Echo: Step: 1 12:29pm May 21,2024

Element Thickness	12	mm.
Internal Corrosion Allowance	3	mm.
Nominal Thickness	14	mm.
External Corrosion Allowance	0	mm.
Design Internal Pressure	22	bars
Design Temperature Internal Pressure	120	°C
Design External Pressure	1.0342	bars
Design Temperature External Pressure	120	°C
Effective Diameter Multiplier	1.2	
Material Name	SA-516 70	[Normalized]
Efficiency, Longitudinal Seam	1.0	
Efficiency, Circumferential Seam	1.0	
Elliptical Head Factor	2.0	
Weld is pre-Heated	No	

Element From Node	80	
Detail Type	Liquid	
Detail ID	Liquid: 80	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Liquid	925	mm.
Liquid Density	0.0005329	kg./cm ³

Element From Node	80	
Detail Type	Insulation	
Detail ID	Ins: 80	
Dist. from "FROM" Node / Offset dist	0	mm.
Height/Length of Insulation	281.25	mm.
Thickness of Insulation	60	mm.
Density	0.00023	kg./cm ³

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FileName : Chiller-Rev.01 -----

XY Coordinate Calculations: Step: 2 12:29pm May 21,2024

XY Coordinate Calculations:

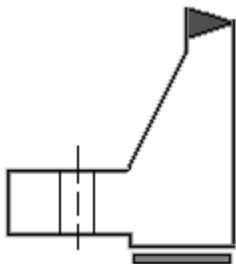
From	To	X (Horiz.) mm.	Y (Vert.) mm.	DX (Horiz.) mm.	DY (Vert.) mm.
CH. Head		50	...	50	...
CH. Barrel		350	...	300	...
CH. Flange		435	...	85	...
SH. Flange		598.525	...	85	...
Port Barrel		748.525	...	150	...
SH. Cone		1313.53	...	565	...
SH. Barrel		3513.53	...	2200	...
SH. Head		3563.53	...	50	...

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Flange Input Data Values Description: CH. FLANGE :

CH. Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	6.85	bars
Design Temperature		85	°C
Internal Corrosion Allowance	ci	3.0000	mm.
External Corrosion Allowance	ce	0.0000	mm.
Use Corrosion Allowance in Thickness Calcs.		Yes	
Flange Inside Diameter	B	600.000	mm.
Flange Outside Diameter	A	760.000	mm.
Flange Thickness	t	60.0000	mm.
Thickness of Hub at Small End	go	10.0000	mm.
Thickness of Hub at Large End	gl	15.0000	mm.
Length of Hub	h	25.0000	mm.
Flange Material		SA-266 2	
Flange Material UNS number		K03506	
Flange Allowable Stress At Temperature	Sfo	137.90	N./mm ²
Flange Allowable Stress At Ambient	Sfa	137.90	N./mm ²
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	172.38	N./mm ²
Bolt Allowable Stress At Ambient	Sa	172.38	N./mm ²
Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	a	22.2250	mm.
Type of Threads	TEMA Thread Series		
Number of Bolts		20	
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Flange Facing Sketch	2, Code Sketch 1b		
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Gasket Factor	m	3.0000	
Gasket Design Seating Stress	y	69.00	N./mm ²
Column for Gasket Seating	2, Code Column II		
Gasket Thickness	tg	4.5000	mm.



ASME Code, Section VIII Division 1, 2019

Hub Small End Required Thickness due to Internal Pressure:

$$\begin{aligned}
 &= (P*(D/2+Ca))/(S*E-0.6*P) \text{ per UG-27 (c) (1)} \\
 &= (6.85*(600/2+3))/(138*1-0.6*6.85)+Ca \\
 &= 4.5106 \text{ mm.}
 \end{aligned}$$

Hub Small End Hub MAWP:

$$= (S*E*t)/(R+0.6*t) \text{ per UG-27 (c) (1)}$$

Flg Calc [Int P]: CH. FLANGE Flng: 11 12:29pm May 21,2024

$$= (138 * 1 * 7) / (303 + 0.6 * 7)$$

$$= 31.421 \text{ bars}$$

Corroded Flange Thickness, tc = T-ci	57.000	mm.
Corroded Flange ID, Bcor = B+2*Fcor	606.000	mm.
Corroded Large Hub, glCor = gl-ci	12.000	mm.
Corroded Small Hub, goCor = go-ci	7.000	mm.
Code R Dimension, R = ((C-Bcor)/2)-glcor	37.500	mm.
Gasket Contact Width, N = (Go - Gi) / 2	13.000	mm.
Basic Gasket Width, bo = N / 2	6.500	mm.
Effective Gasket Width, b = Cb sqrt(bo)	6.425	mm.
Gasket Reaction Diameter, G = Go - 2 * b	637.151	mm.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$= 0.785 * G^2 * Peq$$

$$= 0.79 * 637^2 * 6.85$$

$$= 22285.168 \text{ Kgf}$$

Contact Load on Gasket Surfaces [Hp]:

$$= 2 * b * Pi * G * m * P$$

$$= 2 * 6.42 * 3.14 * 637 * 3 * 6.85$$

$$= 5392.977 \text{ Kgf}$$

Hydrostatic End Load at Flange ID [Hd]:

$$= Pi * Bcor^2 * P / 4$$

$$= 3.14 * 606^2 * 6.85 / 4$$

$$= 20159.355 \text{ Kgf}$$

Pressure Force on Flange Face [Ht]:

$$= H - Hd$$

$$= 22285 - 20159$$

$$= 2125.812 \text{ Kgf}$$

Operating Bolt Load [Wm1]:

$$= \max(H + Hp + H'p, 0)$$

$$= \max(22285 + 5393 + 0, 0)$$

$$= 27678.143 \text{ Kgf}$$

$$= 88967.883 \text{ Kgf, Mating Flange Load Governs}$$

Gasket Seating Bolt Load [Wm2]:

$$= y * b * Pi * G + yPart * bPart * lp$$

$$= 69 * 6.42 * 3.141 * 637 + 0 * 0 * 0$$

$$= 90480.758 \text{ Kgf}$$

Required Bolt Area [Am]:

$$= \text{Maximum of } Wm1/Sb, Wm2/Sa$$

$$= \text{Maximum of } 88968/172, 90481/172$$

$$= 51.477 \text{ cm}^2$$

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

$$= 2a + 6t / (m + 0.5)$$

$$= 2 * 22.2 + 6 * 57 / (3 + 0.5)$$

$$= 142.164 \text{ mm.}$$

Actual Circumferential Bolt Spacing [Bs]:

$$= C * \sin(pi / n)$$

$$= 705 * \sin(3.14 / 20)$$

$$= 110.286 \text{ mm.}$$

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

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

$$= \max(\sqrt{ 110 / (2 * 22.2 + 57) }, 1)$$

$$= 1.0426$$

Bolting Information for TEMA Imperial Thread Series (Non Mandatory):

-----	Minimum	Actual	Maximum

Flg Calc [Int P]: CH. FLANGE Flng: 11 12:29pm May 21,2024

Bolt Area, cm ²	51.477	54.064	
Radial Distance between Hub and Bolts:	31.750	37.500	
Radial Distance between Bolts and Edge:	23.812	27.500	
Circ. Spacing between the Bolts:	52.400	110.286	142.164

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

$$= A_b * S_a / (y * \pi * (G_o + G_i))$$

$$= 54.1 * 172 / (69 * 3.14 * (650 + 624))$$

$$= 3.375 \text{ mm.}$$

Flange Design Bolt Load, Gasket Seating [W]:

$$= S_a * (A_m + A_b) / 2$$

$$= 172 * (51.5 + 54.1) / 2$$

$$= 92754.98 \text{ Kg}$$

Gasket Load for the Operating Condition [HG]:

$$= W_{m1} - H$$

$$= 88968 - 22285$$

$$= 66682.71 \text{ Kg}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

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

$$= (705 - 637) / 2$$

$$= 33.9246 \text{ mm.}$$

Distance to Face Pressure Reaction [ht]:

$$= (R + g_1 + h_g) / 2$$

$$= (37.5 + 12 + 33.9) / 2$$

$$= 41.7123 \text{ mm.}$$

Distance to End Pressure Reaction [hd]:

$$= R + (g_1 / 2)$$

$$= 37.5 + (12 / 2.0)$$

$$= 43.5000 \text{ mm.}$$

Summary of Moments for Internal Pressure: (Kg-m.)

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	20159.	43.5000	1.0426	914.
Face Pressure, Mt	2126.	41.7123	1.0426	92.
Gasket Load, Mg	66683.	33.9246	1.0426	2359.
Gasket Seating, Matm	92755.	33.9246	1.0426	3281.

Total Moment for Operation, Mop 3365. Kg-m.
 Total Moment for Gasket seating, Matm 3281. Kg-m.

Effective Hub Length, ho = sqrt(Bcor*goCor) 65.131 mm.
 Hub Ratio, h/h0 = HL / H0 0.384
 Thickness Ratio, g1/g0 = (g1Cor/goCor) 1.714

Flange Factors for Integral Flange:

Factor F 0.858
 Factor V 0.302
 Factor f 1.217
 Factors from Figure 2-7.1 K = 1.254
 T = 1.817 U = 9.566
 Y = 8.705 Z = 4.491
 d = 0.10120E+06 mm.³ e = 0.0132 mm.⁻¹
 Stress Factors ALPHA = 1.751
 BETA = 2.001 GAMMA = 0.964
 DELTA = 1.830 Lamda = 2.794

Longitudinal Hub Stress, Operating [SHo]:

$$= (f * Mop / Bcor) / (L * g_1^2)$$

$$= (1.22 * 3365 / 606) / (2.79 * 12^2)$$

FileName : Chiller-Rev.01 -----

Flg Calc [Int P]: CH. FLANGE

Flng: 11 12:29pm May 21,2024

= 164.79 N./mm²

Longitudinal Hub Stress, Seating [SHa]:

= (f * Matm / Bcor) / (L * g1²)
 = (1.22*3281/606) / (2.79*12²)
 = 160.65 N./mm²

Radial Flange Stress, Operating [SRo]:

= (Beta * Mop / Bcor) / (L * t²)
 = (2*3365/606) / (2.79*57²)
 = 12.01 N./mm²

Radial Flange Stress, Seating [SRa]:

= (Beta * Matm/Bcor) / (L * t²)
 = (2*3281/606) / (2.79*57²)
 = 11.71 N./mm²

Tangential Flange Stress, Operating [STo]:

= (Y * Mo / (t² * Bcor)) - Z * SRO
 = (8.7*3365 / (57²*606)) - 4.49*12
 = 91.99 N./mm²

Tangential Flange Stress, Seating [STa]:

= (y * Matm / (t² * Bcor)) - Z * SRA
 = (8.7*3281 / (57²*606)) - 4.49*11.7
 = 89.68 N./mm²

Average Flange Stress, Operating [SAo]:

= (SHo + max(SRO, STo)) / 2
 = (165+max(12, 92)) / 2
 = 128.39 N./mm²

Average Flange Stress, Seating [SAa]:

= (SHa + max(SRA, STa)) / 2
 = (161+max(11.7, 89.7)) / 2
 = 125.16 N./mm²

Bolt Stress, Operating [BSo]:

= Wm1 / Ab
 = 88968/54.1
 = 161.38 N./mm²

Bolt Stress, Seating [BSa]:

= (Wm2 / Ab)
 = (90481/54.1)
 = 164.12 N./mm²

Flange Stress Analysis Results: N./mm²

	Actual	Operating Allowed	Gasket Seating Actual	Gasket Seating Allowed
Longitudinal Hub	164.79	206.85	160.65	206.85
Radial Flange	12.01	137.90	11.71	137.90
Tangential Flange	91.99	137.90	89.68	137.90
Maximum Average	128.39	137.90	125.16	137.90
Bolting	161.38	172.38	164.12	172.38

Minimum Required Flange Thickness [Rigidity] 58.242 mm.
 Estimated M.A.W.P. (Operating) 14.7 bars
 Estimated Finished Weight of Flange at given Thk. 84.1 kg.
 Estimated Unfinished Weight of Forging at given Thk 112.6 kg.

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 * go² * ho * Ki)
 = 52.14*3310/1.05*9807*0.3/(2.61*202713*7²*65.1*0.3)
 = 0.956 (should be <= 1)

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Flg Calc [Int P]: CH. FLANGE Flng: 11 12:29pm May 21,2024

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

$$= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^2 * ho * Ki)$$

$$= 52.14*3395/1.05*9807*0.3/(2.61*199054*7^2*65.1*0.3)$$

$$= 0.999 \quad (\text{should be } \leq 1)$$

Flange Rigidity Based on Given Thickness [ASME]:

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

$$= 52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^2 * ho * Ki)$$

$$= 52.14*3281/1.04*9807*0.3/(2.79*202713*7^2*65.1*0.3)$$

$$= 0.895 \quad (\text{should be } \leq 1)$$

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

$$= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^2 * ho * Ki)$$

$$= 52.14*3365/1.04*9807*0.3/(2.79*199054*7^2*65.1*0.3)$$

$$= 0.935 \quad (\text{should be } \leq 1)$$

Minimum Design Metal Temperature Results:

Thickness Ratio = 0.47, Temperature Reduction per Fig. UCS 66.1 = 37 °C

Min Metal Temp. w/o impact per UCS-66, Curve C	-44 °C
Min Metal Temp. at Required thickness (UCS 66.1)	-48 °C

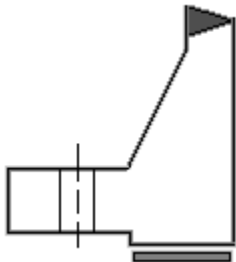
Note: UCS-66(b)(-c) was considered in the flange MDMT calculation.

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Flange Input Data Values Description: SH. FLANGE :

SH. Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	22.03	bars
Design Temperature		120	°C
Internal Corrosion Allowance	ci	3.0000	mm.
External Corrosion Allowance	ce	0.0000	mm.
Use Corrosion Allowance in Thickness Calcs.		Yes	
Flange Inside Diameter	B	600.000	mm.
Flange Outside Diameter	A	760.000	mm.
Flange Thickness	t	60.0000	mm.
Thickness of Hub at Small End	go	12.0000	mm.
Thickness of Hub at Large End	gl	18.0000	mm.
Length of Hub	h	25.0000	mm.
Flange Material		SA-350 LF2	
Flange Material UNS number		K03011	
Flange Allowable Stress At Temperature	Sfo	137.90	N./mm ²
Flange Allowable Stress At Ambient	Sfa	137.90	N./mm ²
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	172.38	N./mm ²
Bolt Allowable Stress At Ambient	Sa	172.38	N./mm ²
Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	a	22.2250	mm.
Type of Threads	TEMA Thread Series		
Number of Bolts		20	
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Flange Facing Sketch	2, Code Sketch 1b		
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Gasket Factor	m	3.0000	
Gasket Design Seating Stress	y	69.00	N./mm ²
Column for Gasket Seating	2, Code Column II		
Gasket Thickness	tg	4.5000	mm.



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Hub Small End Required Thickness due to Internal Pressure:

$$\begin{aligned}
 &= (P*(D/2+Ca))/(S*E-0.6*P) \text{ per UG-27 (c) (1)} \\
 &= (22*(600/2+3))/(138*1-0.6*22)+Ca \\
 &= 7.8880 \text{ mm.}
 \end{aligned}$$

Hub Small End Hub MAWP:

$$= (S*E*t)/(R+0.6*t) \text{ per UG-27 (c) (1)}$$

Flg Calc [Int P]: SH. FLANGE Flng: 12 12:29pm May 21,2024

$$= (138 * 1 * 9) / (303 + 0.6 * 9)$$

$$= 40.241 \text{ bars}$$

Corroded Flange Thickness, tc = T-ci	57.000	mm.
Corroded Flange ID, Bcor = B+2*Fcor	606.000	mm.
Corroded Large Hub, glCor = gl-ci	15.000	mm.
Corroded Small Hub, g0Cor = go-ci	9.000	mm.
Code R Dimension, R = ((C-Bcor)/2)-glcor	34.500	mm.
Gasket Contact Width, N = (Go - Gi) / 2	13.000	mm.
Basic Gasket Width, bo = N / 2	6.500	mm.
Effective Gasket Width, b = Cb sqrt(bo)	6.425	mm.
Gasket Reaction Diameter, G = Go - 2 * b	637.151	mm.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$= 0.785 * G^2 * Peq$$

$$= 0.79 * 637^2 * 22$$

$$= 71632.844 \text{ Kgf}$$

Contact Load on Gasket Surfaces [Hp]:

$$= 2 * b * Pi * G * m * P$$

$$= 2 * 6.42 * 3.14 * 637 * 3 * 22$$

$$= 17335.039 \text{ Kgf}$$

Hydrostatic End Load at Flange ID [Hd]:

$$= Pi * Bcor^2 * P / 4$$

$$= 3.14 * 606^2 * 22 / 4$$

$$= 64799.688 \text{ Kgf}$$

Pressure Force on Flange Face [Ht]:

$$= H - Hd$$

$$= 71633 - 64800$$

$$= 6833.158 \text{ Kgf}$$

Operating Bolt Load [Wm1]:

$$= \max(H + Hp + H'p, 0)$$

$$= \max(71633 + 17335 + 0, 0)$$

$$= 88967.883 \text{ Kgf}$$

Gasket Seating Bolt Load [Wm2]:

$$= y * b * Pi * G + yPart * bPart * lp$$

$$= 69 * 6.42 * 3.141 * 637 + 0 * 0 * 0$$

$$= 90480.758 \text{ Kgf}$$

Required Bolt Area [Am]:

$$= \text{Maximum of } Wm1/Sb, Wm2/Sa$$

$$= \text{Maximum of } 88968/172, 90481/172$$

$$= 51.477 \text{ cm}^2$$

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

$$= 2a + 6t / (m + 0.5)$$

$$= 2 * 22.2 + 6 * 57 / (3 + 0.5)$$

$$= 142.164 \text{ mm.}$$

Actual Circumferential Bolt Spacing [Bs]:

$$= C * \sin(pi / n)$$

$$= 705 * \sin(3.14 / 20)$$

$$= 110.286 \text{ mm.}$$

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

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

$$= \max(\sqrt{ 110 / (2 * 22.2 + 57) }, 1)$$

$$= 1.0426$$

Bolting Information for TEMA Imperial Thread Series (Non Mandatory):

	Minimum	Actual	Maximum
Bolt Area, cm ²	51.477	54.064	

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FileName : Chiller-Rev.01 -----

Flg Calc [Int P]: SH. FLANGE Flng: 12 12:29pm May 21,2024

Radial Distance between Hub and Bolts: 31.750 34.500
 Radial Distance between Bolts and Edge: 23.812 27.500
 Circ. Spacing between the Bolts: 52.400 110.286 142.164

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

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

$$= 54.1 * 172 / (69 * 3.14 * (650 + 624))$$

$$= 3.375 \text{ mm.}$$

Flange Design Bolt Load, Gasket Seating [W]:

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

$$= 172 * (51.5 + 54.1) / 2$$

$$= 92754.98 \text{ KgF}$$

Gasket Load for the Operating Condition [HG]:

$$= Wm1 - H$$

$$= 88968 - 71633$$

$$= 17335.04 \text{ KgF}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

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

$$= (705 - 637) / 2$$

$$= 33.9246 \text{ mm.}$$

Distance to Face Pressure Reaction [ht]:

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

$$= (34.5 + 15 + 33.9) / 2$$

$$= 41.7123 \text{ mm.}$$

Distance to End Pressure Reaction [hd]:

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

$$= 34.5 + (15 / 2.0)$$

$$= 42.0000 \text{ mm.}$$

Summary of Moments for Internal Pressure: (Kg-m.)

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	64800.	42.0000	1.0426	2838.
Face Pressure, Mt	6833.	41.7123	1.0426	297.
Gasket Load, Mg	17335.	33.9246	1.0426	613.
Gasket Seating, Matm	92755.	33.9246	1.0426	3281.
Total Moment for Operation, Mop				3748. Kg-m.
Total Moment for Gasket seating, Matm				3281. Kg-m.

Effective Hub Length, $h_o = \text{sqrt}(Bcor * goCor)$ 73.851 mm.
 Hub Ratio, $h/h_o = HL / H0$ 0.339
 Thickness Ratio, $g1/g_o = (g1Cor/goCor)$ 1.667

Flange Factors for Integral Flange:

Factor F 0.868
 Factor V 0.326
 Factor f 1.295
 Factors from Figure 2-7.1 K = 1.254
 T = 1.817 U = 9.566
 Y = 8.705 Z = 4.491
 d = 0.17564E+06 mm.³ e = 0.0118 mm.⁻¹
 Stress Factors ALPHA = 1.670
 BETA = 1.893 GAMMA = 0.919
 DELTA = 1.054 Lamda = 1.973

Longitudinal Hub Stress, Operating [SHo]:

$$= (f * Mop / Bcor) / (L * g1^2)$$

$$= (1.3 * 3748 / 606) / (1.97 * 15^2)$$

$$= 176.95 \text{ N./mm}^2$$

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Flg Calc [Int P]: SH. FLANGE Flng: 12 12:29pm May 21,2024

Longitudinal Hub Stress, Seating [SHa]:
 = (f * Matm / Bcor) / (L * g1²)
 = (1.3*3281/606) / (1.97*15²)
 = 154.89 N./mm²

Radial Flange Stress, Operating [SRo]:
 = (Beta * Mop / Bcor) / (L * t²)
 = (1.89*3748/606) / (1.97*57²)
 = 17.91 N./mm²

Radial Flange Stress, Seating [SRa]:
 = (Beta * Matm/Bcor) / (L * t²)
 = (1.89*3281/606) / (1.97*57²)
 = 15.68 N./mm²

Tangential Flange Stress, Operating [STo]:
 = (Y * Mo / (t² * Bcor)) - Z * SRO
 = (8.7*3748 / (57²*606)) - 4.49*17.9
 = 82.07 N./mm²

Tangential Flange Stress, Seating [STa]:
 = (y * Matm / (t² * Bcor)) - Z * SRA
 = (8.7*3281 / (57²*606)) - 4.49*15.7
 = 71.84 N./mm²

Average Flange Stress, Operating [SAo]:
 = (SHo + max(SRo, STo)) / 2
 = (177+max(17.9, 82.1)) / 2
 = 129.51 N./mm²

Average Flange Stress, Seating [SAa]:
 = (SHa + max(SRa, STa)) / 2
 = (155+max(15.7, 71.8)) / 2
 = 113.37 N./mm²

Bolt Stress, Operating [BSo]:
 = Wm1 / Ab
 = 88968/54.1
 = 161.38 N./mm²

Bolt Stress, Seating [BSa]:
 = (Wm2 / Ab)
 = (90481/54.1)
 = 164.12 N./mm²

Flange Stress Analysis Results: N./mm²

	Actual	Operating Allowed	Gasket Actual	Seating Allowed
Longitudinal Hub	176.95	206.85	154.89	206.85
Radial Flange	17.91	137.90	15.68	137.90
Tangential Flange	82.07	137.90	71.84	137.90
Maximum Average	129.51	137.90	113.37	137.90
Bolting	161.38	172.38	164.12	172.38

Minimum Required Flange Thickness 57.937 mm.
 Estimated M.A.W.P. (Operating) 23.5 bars
 Estimated Finished Weight of Flange at given Thk. 85.1 kg.
 Estimated Unfinished Weight of Forging at given Thk 112.6 kg.

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 * go² * ho * Ki)
 = 52.14*3315/1.05*9807*0.33/(1.85*202713*9²*73.9*0.3)
 = 0.779 (should be <= 1)

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

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Flg Calc [Int P]: SH. FLANGE Flng: 12 12:29pm May 21,2024

$$= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^2 * ho * Ki)$$

$$= 52.14*3787/1.05*9807*0.33/(1.85*196922*9^2*73.9*0.3)$$

$$= 0.916 \quad (\text{should be } \leq 1)$$

Flange Rigidity Based on Given Thickness [ASME]:

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

$$= 52.14 * Ma / Bsc * Cnv_fac * V / (Lambda * Eamb * go^2 * ho * Ki)$$

$$= 52.14*3281/1.04*9807*0.33/(1.97*202713*9^2*73.9*0.3)$$

$$= 0.730 \quad (\text{should be } \leq 1)$$

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

$$= 52.14 * Mo / Bsc * Cnv_fac * V / (Lambda * Eop * goc^2 * ho * Ki)$$

$$= 52.14*3748/1.04*9807*0.33/(1.97*196922*9^2*73.9*0.3)$$

$$= 0.858 \quad (\text{should be } \leq 1)$$

Minimum Design Metal Temperature Results:

Note:

This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C

Note: UCS-66(b)(-c) was considered in the flange MDMT calculation.

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Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

Internal Pressure Results Summary:

Element Thickness, Pressure, Diameter and Allowable Stress :

From	To	Int. Press + Liq. Hd bars	Nominal Thickness mm.	Total Corr Allowance mm.	Element Diameter mm.	Allowable Stress (SE) N./mm ²
CH. Head		6.8543	12	3	600	137.9
CH. Barrel		6.8543	10	3	600	137.9
CH. Flange		6.854	90	3	600	137.9
SH. Flange		22.031	90	3	600	137.9
Port Barrel		22.032	12	3	600	137.9
SH. Cone		22.048	12	3	925	137.9
SH. Barrel		22.048	12	3	925	137.9
SH. Head		22.048	14	3	925	137.9

Element Required Thickness and MAWP :

From	To	Design Pressure bars	M.A.W.P. Corroded bars	M.A.P. New & Cold bars	Minimum Thickness mm.	Required Thickness mm.
CH. Head		6.8	32.1	45.8	10	4.5
CH. Barrel		6.8	31.4	45.1	10	4.51065
CH. Flange		6.8	14.6	23.5	60	58.2422
SH. Flange		22	23.4	23.5	60	57.9374
Port Barrel		22	40.2	53.9	12	7.888
SH. Cone		22	22.8	30.6	12	11.6866
SH. Barrel		22	26.3	35.2	12	10.5153
SH. Head		22	26.8	35.7	12	10.3917

Summary of Heat Exchanger Maximum Allowable Working Pressures :

Note:

For Exchanger 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 = 22.8 bars
 Shell Side MAPnc = 23.5 bars
 Channel Side MAWP = 14.6 bars
 Channel Side MAPnc = 19.6 bars

Elements Suitable for Design Internal Pressure.

Internal Pressure Calculation Results:

ASME Code, Section VIII Division 1, 2019

Elliptical Head From 10 To 20 SA-516 70 , UCS-66 Crv. D at 85 °C

CH. Head

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$= (P \cdot D \cdot K_{cor}) / (2 \cdot S \cdot E - 0.2 \cdot P) \text{ Appendix 1-4(c)}$$

$$= (6.85 \cdot 606 \cdot 0.99) / (2 \cdot 138 \cdot 1 - 0.2 \cdot 6.85)$$

$$= 1.4873 + 3.0000 = 4.4873 \text{ mm.}$$

Note: The thickness required was less than the Code Minimum, therefore

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Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

the Code Minimum value of 1.5000 mm. per UG-16 will be used.

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

Less Operating Hydrostatic Head Pressure of 0.054 bars

$$= (2 * S * E * t) / (K_{cor} * D + 0.2 * t) \text{ per Appendix 1-4 (c)}$$

$$= (2 * 138 * 1 * 7) / (0.99 * 606 + 0.2 * 7)$$

$$= 32.2 - 0.054 = 32.1 \text{ bars}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$= (2 * S * E * t) / (K * D + 0.2 * t) \text{ per Appendix 1-4 (c)}$$

$$= (2 * 138 * 1 * 10) / (1 * 600 + 0.2 * 10)$$

$$= 45.8 \text{ bars}$$

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

$$= (P * (K_{cor} * D + 0.2 * t)) / (2 * E * t)$$

$$= (6.85 * (0.99 * 606 + 0.2 * 7)) / (2 * 1 * 7)$$

$$= 29.354 \text{ N./mm}^2$$

Straight Flange Required Thickness:

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

$$= (6.85 * 303) / (138 * 1 - 0.6 * 6.85) + 3$$

$$= 4.511 \text{ mm.}$$

Straight Flange Maximum Allowable Working Pressure:

Less Operating Hydrostatic Head Pressure of 0.054 bars

$$= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)}$$

$$= (138 * 1 * 9) / (303 + 0.6 * 9)$$

$$= 40.2 - 0.054 = 40.2 \text{ bars}$$

Factor K, corroded condition [Kcor]:

$$= (2 + (\text{Inside Diameter} / (2 * \text{Inside Head Depth}))^2) / 6$$

$$= (2 + (606 / (2 * 153))^2) / 6$$

$$= 0.986992$$

Percent Elong. per UCS-79, VIII-1-01-57 $(75 * t_{nom} / R_f) * (1 - R_f / R_o)$ 8.333 %

Note: Please Check Requirements of UCS-79 as Elongation is > 5%.

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 10, tr = 3.18, c = 3 mm., E* = 1

Thickness Ratio = $tr * (E^*) / (tg - c) = 0.45$, Temp. Reduction = 39 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

MDMT Calculations in the Head Straight Flange:

Govrn. thk, tg = 12, tr = 3.24, c = 3 mm., E* = 1

Thickness Ratio = $tr * (E^*) / (tg - c) = 0.36$, Temp. Reduction = 71 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Cylindrical Shell From 20 To 30 SA-516 70 , UCS-66 Crv. D at 85 °C

CH. Barrel

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

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

$$= (6.85 * 303) / (138 * 1 - 0.6 * 6.85)$$

$$= 1.5106 + 3.0000 = 4.5106 \text{ mm.}$$

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

Less Operating Hydrostatic Head Pressure of 0.054 bars

$$= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)}$$

$$= (138 * 1 * 7) / (303 + 0.6 * 7)$$

Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

$$= 31.4 - 0.054 = 31.4 \text{ bars}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c) (1)} \\ &= (138 \cdot 1 \cdot 10) / (300 + 0.6 \cdot 10) \\ &= 45.1 \text{ bars} \end{aligned}$$

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

$$\begin{aligned} &= (P \cdot (R + 0.6 \cdot t)) / (E \cdot t) \\ &= (6.85 \cdot (303 + 0.6 \cdot 7)) / (1 \cdot 7) \\ &= 30.082 \text{ N./mm}^2 \end{aligned}$$

% Elongation per Table UG-79-1 ($50 \cdot t_{nom} / R_f \cdot (1 - R_f / R_o)$) 1.639 %

Minimum Design Metal Temperature Results:

Govrn. thk, $t_g = 10$, $t_r = 3.24$, $c = 3 \text{ mm.}$, $E^* = 1$
Thickness Ratio = $t_r \cdot (E^*) / (t_g - c) = 0.46$, Temp. Reduction = $38 \text{ }^\circ\text{C}$

Min Metal Temp. w/o impact per UCS-66, Curve D -48 $^\circ\text{C}$

Cylindrical Shell From 50 To 60 SA-516 70 , UCS-66 Crv. D at 120 $^\circ\text{C}$

Port Barrel

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$\begin{aligned} &= (P \cdot R) / (S \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)} \\ &= (22 \cdot 303) / (138 \cdot 1 - 0.6 \cdot 22) \\ &= 4.8880 + 3.0000 = 7.8880 \text{ mm.} \end{aligned}$$

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

Less Operating Hydrostatic Head Pressure of 0.032 bars

$$\begin{aligned} &= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c) (1)} \\ &= (138 \cdot 1 \cdot 9) / (303 + 0.6 \cdot 9) \\ &= 40.2 - 0.032 = 40.2 \text{ bars} \end{aligned}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c) (1)} \\ &= (138 \cdot 1 \cdot 12) / (300 + 0.6 \cdot 12) \\ &= 53.9 \text{ bars} \end{aligned}$$

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

$$\begin{aligned} &= (P \cdot (R + 0.6 \cdot t)) / (E \cdot t) \\ &= (22 \cdot (303 + 0.6 \cdot 9)) / (1 \cdot 9) \\ &= 75.499 \text{ N./mm}^2 \end{aligned}$$

% Elongation per Table UG-79-1 ($50 \cdot t_{nom} / R_f \cdot (1 - R_f / R_o)$) 1.961 %

Minimum Design Metal Temperature Results:

Govrn. thk, $t_g = 12$, $t_r = 5.06$, $c = 3 \text{ mm.}$, $E^* = 1$
Thickness Ratio = $t_r \cdot (E^*) / (t_g - c) = 0.56$, Temp. Reduction = $26 \text{ }^\circ\text{C}$

Min Metal Temp. w/o impact per UCS-66, Curve D -48 $^\circ\text{C}$

Conical Section From 60 To 70 SA-516 70 , UCS-66 Crv. D at 120 $^\circ\text{C}$

SH. Cone

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$\begin{aligned} &= (P \cdot D) / (2 \cdot \cos(a) \cdot (S \cdot E - 0.6 \cdot P)) \text{ per Appendix 1-4 (e)} \\ &= (22 \cdot 932) / (2 \cdot 0.87 \cdot (138 \cdot 1 - 0.6 \cdot 22)) \end{aligned}$$

Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

$$= 8.6866 + 3.0000 = 11.6866 \text{ mm.}$$

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

Less Operating Hydrostatic Head Pressure of 0.048 bars

$$\begin{aligned} &= (2 * S * E * t * \cos(a)) / (D + 1.2 * t * \cos(a)) \text{ per App 1-4 (e)} \\ &= (2 * 138 * 1 * 9 * 0.87) / (932 + 1.2 * 9 * 0.87) \\ &= 22.8 - 0.048 = 22.8 \text{ bars} \end{aligned}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (2 * S * E * t * \cos(a)) / (D + 1.2 * t * \cos(a)) \text{ per App 1-4 (e)} \\ &= (2 * 138 * 1 * 12 * 0.87) / (925 + 1.2 * 12 * 0.87) \\ &= 30.6 \text{ bars} \end{aligned}$$

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

$$\begin{aligned} &= (P * (D + 1.2 * t * \cos(a))) / (2 * E * t * \cos(a)) \\ &= (22 * (932 + 1.2 * 9 * 0.87)) / (2 * 1 * 9 * 0.87) \\ &= 133.144 \text{ N./mm}^2 \end{aligned}$$

% Elongation per Table UG-79-1 (50*tnom/Rf*(1-Rf/Ro)) 1.955 %

Minimum Design Metal Temperature Results:

Govrn. thk, tg = 12, tr = 8.98, c = 3 mm., E* = 1
Thickness Ratio = tr *(E*)/(tg - c) = 1, Temp. Reduction = 0 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Cylindrical Shell From 70 To 80 SA-516 70 , UCS-66 Crv. D at 120 °C

SH. Barrel

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$\begin{aligned} &= (P * R) / (S * E - 0.6 * P) \text{ per UG-27 (c) (1)} \\ &= (22 * 466) / (138 * 1 - 0.6 * 22) \\ &= 7.5153 + 3.0000 = 10.5153 \text{ mm.} \end{aligned}$$

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

Less Operating Hydrostatic Head Pressure of 0.048 bars

$$\begin{aligned} &= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)} \\ &= (138 * 1 * 9) / (466 + 0.6 * 9) \\ &= 26.4 - 0.048 = 26.3 \text{ bars} \end{aligned}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$\begin{aligned} &= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)} \\ &= (138 * 1 * 12) / (462 + 0.6 * 12) \\ &= 35.2 \text{ bars} \end{aligned}$$

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

$$\begin{aligned} &= (P * (R + 0.6 * t)) / (E * t) \\ &= (22 * (466 + 0.6 * 9)) / (1 * 9) \\ &= 115.369 \text{ N./mm}^2 \end{aligned}$$

% Elongation per Table UG-79-1 (50*tnom/Rf*(1-Rf/Ro)) 1.281 %

Minimum Design Metal Temperature Results:

Govrn. thk, tg = 12, tr = 7.77, c = 3 mm., E* = 1
Thickness Ratio = tr *(E*)/(tg - c) = 0.86, Temp. Reduction = 8 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Elliptical Head From 80 To 90 SA-516 70 , UCS-66 Crv. D at 120 °C

FileName : Chiller-Rev.01 -----

Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

SH. Head

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

$$= (P \cdot D \cdot K_{cor}) / (2 \cdot S \cdot E - 0.2 \cdot P) \text{ Appendix 1-4 (c)}$$

$$= (22 \cdot 931 \cdot 0.99) / (2 \cdot 138 \cdot 1 - 0.2 \cdot 22)$$

$$= 7.3917 + 3.0000 = 10.3917 \text{ mm.}$$

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

Less Operating Hydrostatic Head Pressure of 0.048 bars

$$= (2 \cdot S \cdot E \cdot t) / (K_{cor} \cdot D + 0.2 \cdot t) \text{ per Appendix 1-4 (c)}$$

$$= (2 \cdot 138 \cdot 1 \cdot 9) / (0.99 \cdot 931 + 0.2 \cdot 9)$$

$$= 26.8 - 0.048 = 26.8 \text{ bars}$$

Maximum Allowable Pressure, New and Cold [MAPNC]:

$$= (2 \cdot S \cdot E \cdot t) / (K \cdot D + 0.2 \cdot t) \text{ per Appendix 1-4 (c)}$$

$$= (2 \cdot 138 \cdot 1 \cdot 12) / (1 \cdot 925 + 0.2 \cdot 12)$$

$$= 35.7 \text{ bars}$$

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

$$= (P \cdot (K_{cor} \cdot D + 0.2 \cdot t)) / (2 \cdot E \cdot t)$$

$$= (22 \cdot (0.99 \cdot 931 + 0.2 \cdot 9)) / (2 \cdot 1 \cdot 9)$$

$$= 113.296 \text{ N./mm}^2$$

Straight Flange Required Thickness:

$$= (P \cdot R) / (S \cdot E - 0.6 \cdot P) + c \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22) + 3$$

$$= 10.515 \text{ mm.}$$

Straight Flange Maximum Allowable Working Pressure:

Less Operating Hydrostatic Head Pressure of 0.048 bars

$$= (S \cdot E \cdot t) / (R + 0.6 \cdot t) \text{ per UG-27 (c) (1)}$$

$$= (138 \cdot 1 \cdot 11) / (466 + 0.6 \cdot 11)$$

$$= 32.1 - 0.048 = 32.1 \text{ bars}$$

Factor K, corroded condition [Kcor]:

$$= (2 + (\text{Inside Diameter} / (2 \cdot \text{Inside Head Depth}))^2) / 6$$

$$= (2 + (931 / (2 \cdot 234))^2) / 6$$

$$= 0.991489$$

Percent Elong. per UCS-79, VIII-1-01-57 (75*tnom/Rf)*(1-Rf/Ro) 6.393 %

Note: Please Check Requirements of UCS-79 as Elongation is > 5%.

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 12, tr = 7.64, c = 3 mm., E* = 1

Thickness Ratio = tr*(E*)/(tg - c) = 0.85, Temp. Reduction = 8 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

MDMT Calculations in the Head Straight Flange:

Govrn. thk, tg = 14, tr = 7.77, c = 3 mm., E* = 1

Thickness Ratio = tr*(E*)/(tg - c) = 0.71, Temp. Reduction = 16 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

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

Hydrostatic Test Pressure Results:

Exchanger Shell Side Hydrostatic Test Pressures:

Pressure per UG99b = 1.30 * M.A.W.P. * Sa/S 29.624 bars

Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

Pressure per UG99b[35] = 1.30 * Design Pres * Sa/S 28.600 bars
 Pressure per UG99c = 1.30 * M.A.P. - Head(Hyd) 30.501 bars
 Pressure per UG100 = 1.10 * M.A.W.P. * Sa/S 25.066 bars
 Pressure per PED = max(1.43*DP, 1.25*DP*ratio) 31.460 bars
 Pressure per App 27-4 = M.A.W.P. 22.788 bars

Exchanger Channel Side Hydrostatic Test Pressures:

Pressure per UG99b = 1.30 * M.A.W.P. * Sa/S 19.018 bars
 Pressure per UG99b[35] = 1.30 * Design Pres * Sa/S 8.840 bars
 Pressure per UG99c = 1.30 * M.A.P. - Head(Hyd) 25.480 bars
 Pressure per UG100 = 1.10 * M.A.W.P. * Sa/S 16.092 bars
 Pressure per PED = max(1.43*DP, 1.25*DP*ratio) 9.724 bars
 Pressure per App 27-4 = M.A.W.P. 14.629 bars

UG-99(b) Note 35, Test Pressure Calculation [Shell Side]:

= Test Factor * Design Pressure * Stress Ratio
 = 1.3 * 22 * 1
 = 28.600 bars

UG-99(b) Note 35, Test Pressure Calculation [Channel Side]:

= Test Factor * Design Pressure * Stress Ratio
 = 1.3 * 6.8 * 1
 = 8.840 bars

Horizontal Test performed per: UG-99b (Note 35)

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 Test Pressure (N./mm² & bars):

From To	Stress	Allowable	Ratio	Pressure
CH. Head	38.1	235.8	0.162	8.90
CH. Barrel	39.1	235.8	0.166	8.90
Port Barrel	98.2	235.8	0.416	28.66
SH. Cone	173.3	235.8	0.735	28.69
SH. Barrel	150.1	235.8	0.637	28.69
SH. Head	147.4	235.8	0.625	28.69

Stress ratios for Nozzle and Pad Materials (N./mm²):

Description	Pad/Nozzle	Ambient	Operating	Ratio
T1 (3in.)	Nozzle	117.90	117.90	1.000
T1 (3in.)	Pad	137.90	137.90	1.000
T2 (3in.)	Nozzle	117.90	117.90	1.000
T2 (3in.)	Pad	137.90	137.90	1.000
S1 (4in.)	Nozzle	117.90	117.90	1.000
S1 (4in.)	Pad	137.90	137.90	1.000
S2 (6in.)	Nozzle	117.90	117.90	1.000
S2 (6in.)	Pad	137.90	137.90	1.000
V (2in.)	Nozzle	137.90	137.90	1.000
D (2in.)	Nozzle	137.90	137.90	1.000
PSV (3in.)	Nozzle	117.90	117.90	1.000
PSV (3in.)	Pad	137.90	137.90	1.000
LG1 (2in.)	Nozzle	117.90	117.90	1.000
LG2 (2in.)	Nozzle	117.90	117.90	1.000
Minimum				1.000

Stress ratios for Pressurized Vessel Elements (N./mm²):

Internal Pressure Calculations: Step: 5 12:29pm May 21,2024

Description	Ambient	Operating	Ratio
CH. Head	137.90	137.90	1.000
CH. Barrel	137.90	137.90	1.000
CH. Flange	137.90	137.90	1.000
SH. Flange	137.90	137.90	1.000
Port Barrel	137.90	137.90	1.000
SH. Cone	137.90	137.90	1.000
SH. Barrel	137.90	137.90	1.000
SH. Head	137.90	137.90	1.000
Minimum			1.000

Stress ratios for Exchanger Materials (N./mm²):

Description	Ambient	Operating	Ratio
Tube Material	117.90	117.90	1.000
Tubesheet Material	137.90	137.90	1.000
Minimum			1.000

Hoop Stress in Nozzle Wall during Pressure Test (N./mm²):

Description	Ambient	Operating	Ratio
T1 (3in.)	10.43	217.19	0.048
T2 (3in.)	10.43	217.19	0.048
S1 (4in.)	23.20	217.20	0.107
S2 (6in.)	35.42	217.19	0.163
V (2in.)	7.71	223.40	0.035
D (2in.)	7.71	223.40	0.035
PSV (3in.)	17.79	217.19	0.082
LG1 (2in.)	17.48	217.19	0.080
LG2 (2in.)	17.48	217.19	0.080

Elements Suitable for Test Pressure.

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External Pressure Calculations: Step: 6 12:29pm May 21,2024

External Pressure Calculation Results :

External Pressure Calculations:

From	To	Section Length mm.	Outside Diameter mm.	Corroded Thickness mm.	Factor A	Factor B N./mm²
10	20	No Calc	620	7	0.0015681	97.9287
20	30	400	620	7	0.0026194	109.396
30	40	No Calc	...	57	No Calc	No Calc
40	50	No Calc	...	57	No Calc	No Calc
50	60	150	624	9	0.012325	122.731
60	70	2892.08	952.713	9	0.00030613	30.6064
70	80	2892.08	949	9	0.00038034	38.0256
80	90	No Calc	949	9	0.0013172	93.319

External Pressure Calculations:

From	To	External Actual T. mm.	External Required T. mm.	External Design Pressure bars	External M.A.W.P. bars
10	20	10	4.57853	1	12.3
20	30	10	4.65419	1.03419	16.5
30	40	60	57.277	1.03419	No Calc
40	50	60	53.5432	1.03419	No Calc
50	60	12	4.10182	1.03419	23.6
60	70	12	8.61227	1.03419	3.34
70	80	12	7.84861	1.03419	4.81
80	90	12	5.45716	1.03419	9.83
Minimum					3.34

External Pressure Calculations:

From	To	Actual Length Bet. Stiffeners mm.	Allowable Length Bet. Stiffeners mm.	Ring Inertia Required cm**4	Ring Inertia Available cm**4
10	20	No Calc	No Calc	No Calc	No Calc
20	30	400	11389	No Calc	No Calc
30	40	No Calc	No Calc	No Calc	No Calc
40	50	No Calc	No Calc	No Calc	No Calc
50	60	150	3156.37	No Calc	No Calc
60	70	2892.08	2892.08	No Calc	No Calc
70	80	2892.08	61083.1	No Calc	No Calc
80	90	No Calc	No Calc	No Calc	No Calc

Elements Suitable for External Pressure.

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Element and Detail Weights: Step: 7 12:29pm May 21,2024

Element and Detail Weights:

From	To	Element Metal Wgt. kg.	Element ID Volume Cm.	Corroded Metal Wgt. kg.	Corroded ID Volume Cm.	Extra due Misc %
10	20	50.6331	42419.1	37.9748	43560.2	3.03799
20	30	44.5585	84838.2	31.3443	86543.5	2.67351
30	40	123.873	33032.9	118.807	33175	7.43237
40	50	124.828	33032.9	119.762	33175	7.48967
50	60	26.8228	26340.9	20.2156	27193.6	1.60936
60	70	147.993	201391	111.49	206101	8.87957
70	80	602.315	1272183	453.181	1291428	36.1389
80	90	130.511	137226	102.544	139693	7.83066
Total		1251	1830464.25	995	1860870.12	75

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 kg.	X Offset, Dtl. Cent. mm.	Y Offset, Dtl. Cent. mm.	Z Offset, Dtl. Cent. mm.	Description
10	Liqd	38.9339	-50	0.18167E-04	...	Liquid: 10
10	Insl	10.0491	-50	Ins: 10
20	Liqd	77.8679	150	Liquid: 20
20	Insl	8.84351	150	Ins: 20
20	Nozl	12.5737	150	326.037	-111.113	T1 (3in.)
20	Nozl	12.5737	150	-326.037	-111.113	T2 (3in.)
20	Wght	70	150	PASS PAR.
30	Liqd	30.3189	42.5	Liquid: 30
30	Insl	4.60465	42.5	Ins: 30
40	Liqd	17.5991	42.5	Liquid: 40
40	Insl	4.64015	42.5	Ins: 40
50	Liqd	14.0338	75	Liquid: 50
50	Insl	4.44777	75	Ins: 50
60	Liqd	107.296	217.188	-81.25	...	Liquid: 60
60	Insl	23.6443	282.5	Ins: 60
60	Nozl	26.5597	150	-519.65	...	S1 (4in.)
70	Sadl	105.632	230	594.25	...	Left Saddle
70	Sadl	105.632	1650	594.25	...	Right Saddle
70	Liqd	677.787	1100	Liquid: 70
70	Insl	96.2295	1100	Ins: 70
70	Nozl	50.5487	1100	546.637	...	S2 (6in.)
70	Nozl	10.5874	2050	487.9	...	V (2in.)
70	Nozl	10.5874	2050	-487.9	...	D (2in.)
70	Nozl	17.4622	170	506.95	...	PSV (3in.)
70	Nozl	6.46311	600	485.017	...	LG1 (2in.)
70	Nozl	9.29522	600	-485.017	...	LG2 (2in.)
70	Wght	100	100	...	160	BUNDLE MISS.
70	Wght	100	1250	...	160	BUNDLE MISS.
80	Liqd	73.1104	127.083	Liquid: 80
80	Insl	20.8513	140.625	Ins: 80
30	FTsh	110.743	124	Tubesheet
30	Tube	725.306	1244	Tubes

Total Weight of Each Detail Type:

Saddles 211.3
 Liquid 1036.9

Element and Detail Weights: Step: 7 12:29pm May 21,2024

Insulation	173.3
Nozzles	156.7
Weights	270.0
Exchanger Components	836.0
Liquid in Tubes	187.4

Sum of the Detail Weights	2871.6 kg.

Weight Summation Results: (kg.)

	Fabricated	Shop Test	Shipping	Erected	Empty	Operating
Main Elements	1326.6	1326.6	1326.6	1326.6	1326.6	1326.6
Saddles	211.3	211.3	211.3	211.3	211.3	211.3
Nozzles	156.7	156.7	156.7	156.7	156.7	156.7
Wld Weights	270.0	270.0	270.0	270.0	270.0	270.0
Exchanger	836.0	836.0	836.0	836.0	836.0	836.0
Insulation	173.3	173.3	173.3
Ope. Liquid	1036.9
Tube Ope Lqd	187.4
Test Liquid	...	1829.3
Tube Tst Lqd	...	204.0

Totals	2800.6	4834.0	2800.6	2973.9	2973.9	4198.2

Field Installation Options:

Insulation installed before lifting.

Miscellaneous Weight Percent: 6.0 %

Note that the above value for the miscellaneous weight percent has been applied to the shells/heads/flange/tubesheets/tubes etc. in the weight calculations for metallic components.

Weight Summary:

Fabricated Wt.	- Bare Weight without Removable Internals	2800.6 kg.
Shop Test Wt.	- Fabricated Weight + Water (Full)	4834.0 kg.
Shipping Wt.	- Fab. Weight + removable Intls.+ Shipping App.	2800.6 kg.
Erected Wt.	- Fab. Wt + or - loose items (trays,platforms etc.)	2973.9 kg.
Ope. Wt. no Liq	- Fab. Weight + Internals. + Details + Weights	2973.9 kg.
Operating Wt.	- Empty Weight + Operating Liq. Uncorroded	4198.2 kg.
Oper. Wt. + CA	- Corr Wt. + Operating Liquid	3926.6 kg.
Field Test Wt.	- Empty Weight + Water (Full)	4766.1 kg.

Exchanger Tube Data

Volume of Exchanger tubes :	204161.0 Cm.
Weight of Ope Liq in tubes :	187.4 kg.
Weight of Water in tubes :	204.0 kg.

Note:

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

Note: The Field Test weight as computed in the corroded condition.

Outside Surface Areas of Elements:

From	To	Surface Area
		cm ²
10	20	5200.96
20	30	5843.36
30	40	4352.68

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Element and Detail Weights: Step: 7 12:29pm May 21,2024

40	50	4357.39
50	60	2940.53
60	70	14594.9
70	80	65590.2
80	90	11341.8

Total 114221.789 cm²

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FileName : Chiller-Rev.01 -----

Nozzle Flange MAWP: Step: 8 12:29pm May 21,2024

Nozzle Flange MAWP Results: (bars & °C)

Nozzle Description	Flange Rating Ope.	Flange Rating Ambient	Design Temp	Class	Grade/Group	Equiv. Press	UG-44 (b)	Max Pressure 50%	DNV
T1 (3in.)	18.15	19.60	85	150	GR 1.1
T2 (3in.)	18.15	19.60	85	150	GR 1.1
S1 (4in.)	46.00	51.10	120	300	GR 1.1
S2 (6in.)	46.00	51.10	120	300	GR 1.1
V (2in.)	46.00	51.10	120	300	GR 1.1
D (2in.)	46.00	51.10	120	300	GR 1.1
PSV (3in.)	46.00	51.10	120	300	GR 1.1
LG1 (2in.)	46.00	51.10	120	300	GR 1.1
LG2 (2in.)	46.00	51.10	120	300	GR 1.1

Shellside Flange Rating

Lowest Flange Pressure Rating was (Ope) [ShellSide]: 46.000 bars
 Lowest Flange Pressure Rating was (Amb) [ShellSide]: 51.100 bars

Channelside Flange Rating

Lowest Flange Pressure Rating was (Ope) [TubeSide]: 18.150 bars
 Lowest Flange Pressure Rating was (Amb) [TubeSide]: 19.600 bars

Pressure Ratings are per ASME B16.5 2013 Metric Edition

Warning:

There are nozzles in this model, but no flange MAWPs were de-rated. Be sure this is what you intended. There is a check box in the nozzle dialog that instructs PV Elite to perform the de-rating for each nozzle flange. See ASME VIII-1, UG-44(b) for more information.

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Wind Load Calculation: Step: 9 12:29pm May 21,2024

Wind Load Calculation Results:

Wind Analysis Results per UBC 1994 or UBC 1997

Importance Factor as Entered by the User is 1.000
 Wind Stagnation Pressure (qs) from Table 16-F 75.6 Kgs/m²
 Pressure Coefficient from Table 16-H Cq 0.800
 User Entered Basic Wind Speed 125.0 Km/hr

$P(\text{height}) = C_e(\text{height,Exp}) * C_q * q_s * \text{Imp Fact. [18-1](1994) or [20-1](1997)}$

The values of Ce are shown as the in the table below:

Element	Ce
CH. Head	1.7635
CH. Barrel	1.7635
CH. Flange	1.7635
SH. Flange	1.7635
Port Barrel	1.7635
SH. Cone	1.7635
SH. Barrel	1.7635
SH. Head	1.7635

Wind Loads on Masses/Equipment/Piping

ID	Wind Area cm ²	Elevation mm.	Pressure Kgs/m ²	Force Kgf
PASS PAR.	0.00	20750.00	106.62	0.00
BUNDLE MISS.	0.00	20750.00	106.62	0.00
BUNDLE MISS.	0.00	20750.00	106.62	0.00

Wind Load Calculation:

From	To	Wind Height mm.	Wind Diameter mm.	Wind Area cm ²	Wind Pressure Kgs/m ²	Element Wind Load Kgf
10	20	20750	888	1722.63	106.618	18.3664
20	30	20750	888	2664	106.618	28.4032
30	40	20750	864	734.4	106.618	7.83008
40	50	20750	864	734.4	106.618	7.83008
50	60	20750	892.8	1339.2	106.618	14.2784
60	70	20750	1087.8	6146.07	106.618	65.5286
70	80	20750	1282.8	28221.6	106.618	300.895
80	90	20750	1282.8	3187.46	106.618	33.9843

Earthquake Load Calculation Results:

Earthquake Analysis Results per UBC 1997

The UBC Zone Factor for the Vessel is	0.4000	
The Importance Factor as Specified by the User is	1.250	
The UBC Force Factor as Specified by the User is ..	3.000	
The UBC Total Weight (W) for the Vessel is	3926.6	Kgf
The UBC Total Shear (V) for the Vessel is	962.0	Kgf
The UBC Seismic Coefficient Value Ca is	0.400	
The UBC Seismic Coefficient Value Cv is	0.560	
The UBC Near Source Factor Nv (used with zone 4).	1.000	

Note: The base shear printed above has been modified
 by the user defined Earthquake scalar.

Calculation Steps for Computing the design Base Shear (V) per UBC 1997

Computation of V per equation (34-1):

$$V = 0.7 * Ca * I * W$$

$$V = 0.7 * 0.4 * 1.25 * 3927$$

$$V = 1374.3 \text{ Kgf}$$

Computation of V per equation (30-5):

$$V = 2.5 * Ca * I * W / R$$

$$V = 2.5 * 0.4 * 1.25 * 3927/3$$

$$V = 1636.1 \text{ Kgf}$$

The computed base shear is the minimum of V from 34-1 and 30-5.

Computation of V per equation (34-2), minimum V. V cannot be less than this value !

$$V = 0.56 * Ca * I * W$$

$$V = 0.56 * 0.4 * 1.25 * 3927$$

$$V = 1099.5 \text{ Kgf}$$

In Zone 4, the Base Shear V may also not be less than the following(34-3):

$$V = 1.6 * Z * Nv * I * W / R$$

$$V = 1.6 * 0.4 * 1 * 1.25 * 3927/3$$

$$V = 1047.1 \text{ Kgf}$$

Total Adjusted Base Shear V:

$$= V * \text{Scalar Multiplier} = 1374.3 * 0.7000 = 962.0 \text{ Kgf}$$

Next Sum the earthquake weights times their heights (wi*hi):

Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 118.
Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 236.
Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 353.
Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 471.
Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 589.
Current Sum = Prev. Sum + Wght 393. * Hght 300.000 = 707.
Current Sum = Prev. Sum + Wght 393. * Hght 462.500 = 888.
Current Sum = Prev. Sum + Wght 393. * Hght 462.500 = 1070.
Current Sum = Prev. Sum + Wght 393. * Hght 462.500 = 1252.
Current Sum = Prev. Sum + Wght 393. * Hght 462.500 = 1433.

Compute the load at each level based on equation 30-15 and multiply by the load case scalar. The sum will be the total adjusted shear.

$$F_x = (V * w_x * h_x / (\text{sum of } (w_i * h_i))) * \text{EqFact}$$

$$F_x = [(1374.) * 393. * 300.000 / 1433.] * .7000 = 79.$$

$$F_x = [(1374.) * 393. * 300.000 / 1433.] * .7000 = 79.$$

$$F_x = [(1374.) * 393. * 300.000 / 1433.] * .7000 = 79.$$

Earthquake Load Calculation: Step: 10 12:29pm May 21,2024

Fx = [(1374.) * 393. * 300.000 / 1433.]*.7000 = 79.
 Fx = [(1374.) * 393. * 300.000 / 1433.]*.7000 = 79.
 Fx = [(1374.) * 393. * 300.000 / 1433.]*.7000 = 79.
 Fx = [(1374.) * 393. * 462.500 / 1433.]*.7000 = 122.
 Fx = [(1374.) * 393. * 462.500 / 1433.]*.7000 = 122.
 Fx = [(1374.) * 393. * 462.500 / 1433.]*.7000 = 122.
 Fx = [(1374.) * 393. * 462.500 / 1433.]*.7000 = 122.

Earthquake Load Calculation:

From	To	Earthquake Height mm.	Earthquake Weight Kgf	Element Ope Load Kgf
10	20	300	392.665	79.0708
20	30	300	392.665	79.0708
30	40	300	392.665	79.0708
40	50	300	392.665	79.0708
50	60	300	392.665	79.0708
60	70	300	392.665	79.0708
70	Sad1	462.5	392.665	121.901
Sad1	80	462.5	392.665	121.901
70	80	462.5	392.665	121.901
80	90	462.5	392.665	121.901

Note:
 The Earthquake Loads calculated and printed in the Earthquake
 Load calculation report have been factored by the input
 scalar/load reduction factor of: 0.700.

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Center of Gravity Calculation: Step: 11 12:29pm May 21,2024

Shop/Field Installation Options :

Insulation is installed in the Field before being lifted.

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

Center of Gravity of the Saddles	2253.525 mm.
Center of Gravity of the Liquid	1977.348 mm.
Center of Gravity of the Insulation	1970.300 mm.
Center of Gravity of the Nozzles	1775.768 mm.
Center of Gravity of the Added Weights (Operating)	1524.833 mm.
Center of Gravity of the Added Weights (Empty)	1524.833 mm.
Center of Gravity of the Tubesheet(s)	474.000 mm.
Center of Gravity of the Tubes	1594.000 mm.
Center of Gravity of Bare Shell New and Cold	1781.005 mm.
Center of Gravity of Bare Shell Corroded	1725.221 mm.
Vessel CG in the Operating Condition	1753.492 mm.
Vessel CG in the Fabricated (Shop/Empty) Condition	1691.546 mm.
Vessel CG in the Test Condition	1842.804 mm.

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Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

ASME VIII Division 2 Horizontal Vessel Analysis, Left Saddle:

Horizontal Vessel Stress Calculations : Operating Case

*Warning - Distance to Saddle (a) > 0.25 * Tangent Distance (L) - 4.15.3.2*

Input and Calculated Values:

Vessel Mean Radius	Rm	470.00	mm.
Shell Thickness used in this Case	t	12.000	mm.
Stiffened Vessel Length per 4.15.6	L	3050.00	mm.
Distance from Saddle to Vessel tangent	a1 or a	800.00	mm.
Saddle Width	b1 or b	140.00	mm.
Saddle Bearing Angle	delta or theta	120.00	degrees
Wear Plate Width	b2 or b1	300.00	mm.
Wear Plate Bearing Angle	delta2 or theta1	132.00	degrees
Wear Plate Thickness	e2 or tr	12.0	mm.
Wear Plate Allowable Stress	fw or Sr	137.90	N./mm ²
Shell Allowable Stress used in Calculation		137.90	N./mm ²
Head Allowable Stress used in Calculation		137.90	N./mm ²
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Distance from Saddle Base to Centerline	B	750.00	mm.
Coefficient of Friction	mu	0.30	
Saddle Force Q, Operating Case		4524.79	Kgf
Pressure used in Analysis	P	22.024	bars
Horizontal Vessel Analysis Results:	Actual Allowable		
	N./mm ² N./mm ²		

Long. Stress at Top of Midspan	57.52	137.90	
Long. Stress at Bottom of Midspan	57.50	137.90	
Long. Stress at Top of Saddles	69.08	137.90	
Long. Stress at Bottom of Saddles	51.10	137.90	

Tangential Shear in Shell	5.84	110.32	
Circ. Stress at Horn of Saddle	12.02	172.37	
Circ. Compressive Stress in Shell	0.67	137.90	

Intermediate Results: Saddle Reaction Q due to Wind or Seismic:

Saddle Reaction Force due to Wind Ft [Fwt]:

$$\begin{aligned}
 &= F_{tr} (F_t / \text{Num of Saddles} + Z \text{ Force Load}) * B / E \\
 &= 3 (477/2 + 0) * 750/843 \\
 &= 637.0 \text{ Kgf}
 \end{aligned}$$

Saddle Reaction Force due to Wind Fl [Fwl]:

$$\begin{aligned}
 &= \max (F_l, \text{Sum of X Forces}) * B / L_s \\
 &= \max (115, 0) * 750/1420 \\
 &= 60.7 \text{ Kgf}
 \end{aligned}$$

Saddle Reaction Force due to Earthquake Fl [Fsl]:

$$\begin{aligned}
 &= \max (F_l, \text{Sum of X Forces}) * B / L_s \\
 &= \max (962, 0) * 750/1420 \\
 &= 508.1 \text{ Kgf}
 \end{aligned}$$

Saddle Reaction Force due to Earthquake Ft [Fst]:

Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

$$= \text{Ftr}(\text{Ft}/\text{Num of Saddles} + \text{Z Force Load}) * \text{B} / \text{E}$$

$$= 3(962/2 + 0) * 750/843$$

$$= 1284.4 \text{ Kgf}$$

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

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

$$= 3240 + \max(60.7, 637, 508, 1284)$$

$$= 4524.8 \text{ Kgf}$$

Longitudinal Wind Force [F]:

$$= \text{WindScalar} * \text{WindPress}(\text{Platform Area} + (\text{End Area} * \text{WindDiaMult}))$$

$$= 1 * 1046(0 + (0.9 * 1.2))$$

$$= 1126.093 \text{ N}$$

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)		4630.43	Kgf
Transverse Shear Load Saddle	Ft	481.01	Kgf
Longitudinal Shear Load Saddle		962.03	Kgf

Formulas and Substitutions for Horizontal Vessel Analysis:

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

Saddle Dimension [E]:

$$= \min(2(\text{ShellID}/2 + t + \text{WearPadThickness})\sin(\text{theta}/2), 2 * \text{Rm})$$

$$= \min(2(925/2 + 12 + 12)\sin(120/2), 2 * 470)$$

$$= 842.643 \text{ mm.}$$

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 = 0.0529	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434
K7p = 0.0434			

The suffix 'p' denotes the values for a wear plate if it exists.

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

Moment per Equation 4.15.3 [M1]:

$$= -Q * a [1 - (1 - a/L + (\text{Rm}^2 - h^2) / (2a * L)) / (1 + (4h^2) / (3L))]$$

$$= -4525 * 800 [1 - (1 - 800/3050 + (470^2 - 0^2) / (2 * 800 * 3050)) / (1 + (4 * 0) / (3 * 3050))]$$

$$= -785.6 \text{ Kg-m.}$$

Moment per Equation 4.15.4 [M2]:

$$= Q * L / 4 (1 + 2(\text{Rm}^2 - h^2) / (L^2)) / (1 + (4h^2) / (3L)) - 4a / L$$

$$= 4525 * 3050 / 4 (1 + 2(470^2 - 0^2) / (3050^2)) / (1 + (4 * 0) / (3 * 3050)) - 4 * 800 / 3050$$

$$= -5.8 \text{ Kg-m.}$$

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

$$= P * \text{Rm} / (2t) - M2 / (\text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) - -5.82 / (\text{pi} * 470^2 * 9)$$

$$= 57.52 \text{ N./mm}^2$$

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

$$= P * \text{Rm} / (2t) + M2 / (\text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) + -5.82 / (\text{pi} * 470^2 * 9)$$

$$= 57.50 \text{ N./mm}^2$$

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

$$= P * \text{Rm} / (2t) - M1 / (K1 * \text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) - -786 / (0.11 * \text{pi} * 470^2 * 9)$$

$$= 69.08 \text{ N./mm}^2$$

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Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

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

$$= P * Rm / (2t) + M1 / (K1 * pi * Rm^2 * t)$$

$$= 22 * 470 / (2 * 9) + 786 / (0.19 * pi * 470^2 * 9)$$

$$= 51.10 \text{ N./mm}^2$$

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

$$= Q(L-2a) / (L + (4 * h^2 / 3))$$

$$= 4525(3050 - 2 * 800) / (3050 + (4 * 0 / 3))$$

$$= 2151.1 \text{ Kgf}$$

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

$$= K2 * T / (Rm * t)$$

$$= 1.17 * 2151 / (470 * 9)$$

$$= 5.84 \text{ N./mm}^2$$

Decay Length (4.15.22) [x1,x2]:

$$= 0.78 * \text{sqrt}(Rm * t)$$

$$= 0.78 * \text{sqrt}(470 * 9)$$

$$= 50.730 \text{ mm.}$$

Effective reinforcing plate width (4.15.1) [B1]:

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

$$= \text{min}(140 + 1.56 * \text{sqrt}(470 * 9), 2 * 800)$$

$$= 241.46 \text{ mm.}$$

Wear Plate/Shell Stress ratio (4.15.29) [eta]:

$$= \text{min}(Sr/S, 1)$$

$$= \text{min}(138/138, 1)$$

$$= 1.0000$$

Circumferential Stress at Saddle Base with Wear Plate (4.15.26) [sigma6,r]:

$$= -K5 * Q * k / (B1(t + eta * tr))$$

$$= -0.76 * 4525 * 0.1 / (241(9 + 1 * 12))$$

$$= -0.67 \text{ N./mm}^2$$

Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:

$$= -Q / (4(t + eta * tr)b1) - 12 * K7 * Q * Rm / (L(t + eta * tr)^2)$$

$$= -4525 / (4(9 + 1 * 12)241) - 12 * 0.053 * 4525 * 470 / (3050(9 + 1 * 12)^2)$$

$$= -12.02 \text{ N./mm}^2$$

Distance between Saddle Supports [Ls]:

$$= 1420.0 \text{ mm.}$$

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

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

$$= 0.12232E-04 * 1420(120 - 21.1)$$

$$= 1.718 \text{ mm.}$$

Results for Vessel Ribs, Web and Base:

Baseplate Length	Bplen	900.0000	mm.
Baseplate Thickness	Bpthk	15.0000	mm.
Baseplate Width	Bpwid	170.0000	mm.
Number of Ribs (inc. outside ribs)	Nribs	3	
Rib Thickness	Ribtk	12.0000	mm.
Web Thickness	Webtk	12.0000	mm.
Web Location	Webloc	Center	
Saddle Yield Stress	Sy	206.9	N./
Height of Web at Center	Hw,c	367.5	mm.
Friction Coefficient	mu	0.300	

Note: In the tables below lo is I for the rectangle + Area * Centroid Distance^2

Moment of Inertia of Saddle - Transverse Direction (90 degrees to long axis)

Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

	B	D	Y	A	AY	Io
Shell	401.0	9.0	4.5	36.1	16239.4	0.295E+04
Wearplate	300.0	12.0	15.0	36.0	54000.0	0.231E+04
Web	12.0	248.5	145.2	29.8	433135.5	0.229E+04
BasePlate	170.0	15.0	277.0	25.5	706350.0	0.846E+04
Totals	127.4	1209724.9	0.160E+05

Distance to Centroid [C1]:

$$= AY / A$$

$$= 476/127$$

$$= 94.949 \text{ mm.}$$

Angle [beta]:

$$= 180 - \text{Saddle Angle}/2$$

$$= 180 - 120/2$$

$$= 120.0$$

Saddle Splitting Coefficient [K1]:

$$= (1 + \cos(\text{beta}) - 0.5 * \sin(\text{beta})^2) / (\pi - \text{beta} + \sin(\text{beta}) \cos(\text{beta}))$$

$$= (1 + \cos(120) - 0.5 * \sin(120)^2) / (\pi - 2.09 + \sin(120) \cos(120))$$

$$= 0.2035$$

Saddle Splitting Force [Fh]:

$$= K1 * Q$$

$$= 0.2 * 4525$$

$$= 920.8936 \text{ KgF}$$

$$\text{Tension Stress, } St = (Fh/As) = 0.9889 \text{ N./mm}^2$$

$$\text{Allowed Stress, } Sa = 0.6 * \text{Yield Str} = 124.1100 \text{ N./mm}^2$$

Saddle Splitting Dimension [d]:

$$= B - R * \sin(\text{theta}/2) / (\text{theta}/2 \text{ in radians})$$

$$= 750 - 466 * \sin(120/2) / 1.05$$

$$= 365.035 \text{ mm.}$$

$$\text{Bending Moment, } M = Fh * d = 336.1647 \text{ Kg-m.}$$

$$\text{Bending Stress, } Sb = (M * C1 / I) = 1.9557 \text{ N./mm}^2$$

$$\text{Allowed Stress, } Sa = 2/3 * \text{Yield Str} = 137.9000 \text{ N./mm}^2$$

Minimum Thickness of Baseplate per Moss:

$$= (3 (Q + \text{Saddle_Wt}) \text{BasePlateWidth} / (4 * \text{BasePlateLength} * \text{AllStress}))^{1/2}$$

$$= (3(4525 + 106)170 / (4 * 900 * 138))^{1/2}$$

$$= 6.830 \text{ mm.}$$

Calculation of Axial Load, Intermediate Values and Compressive Stress:

Web Length Dimension [Web Length]:

$$= 2 * \cos(90 - \text{Saddle Angle}/2) (\text{Inside Radius} + \text{Shell Thk} + \text{Wear Plate Thk})$$

$$= 2 * \cos(90 - 120/2) (462 + 12 + 12)$$

$$= 842.643 \text{ mm.}$$

Distance between Ribs [e]:

$$= \text{Web Length} / (\text{Nr ribs} - 1)$$

$$= 843 / (3 - 1)$$

$$= 421.321 \text{ mm.}$$

Baseplate Pressure Area [Ap]:

$$= e * \text{Bpwid} / 2$$

$$= 421 * 170 / 2$$

$$= 358.123 \text{ cm}^2$$

Bearing Pressure [Bp]:

$$= Q / (\text{BasePlateLength} * \text{BasePlateWidth})$$

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$$= 4525 / (900 * 170)$$
$$= 2.957 \text{ Kgf/cm}^2$$

Axial Load [P]:

$$= A_p * B_p$$
$$= 358 * 2.96$$
$$= 1059.107 \text{ Kgf}$$

Area of the Rib and Web [Ar]:

$$= \text{Rib Area} + \text{Web Area}$$
$$= 15.4 + 25.3$$
$$= 40.639 \text{ cm}^2$$

Compressive Stress [Sc]:

$$= P / A_r$$
$$= 1059 / 40.6$$
$$= 2.556 \text{ N./mm}^2$$

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	B	D	Y	A	AY	Io
-----	-----	-----	-----	-----	-----	-----
Rib+Web	12.0	140.0	...	16.8	...	274.

Rib dimension [D]:

$$= \text{Saddle Width} - \text{Web Thickness}$$
$$= 140 - 12$$
$$= 128.000 \text{ mm.}$$

Distance to Centroid from Datum [ytot]:

$$= AY / A$$
$$= 0 / 40.6$$
$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= \text{Saddle Width} / 2$$
$$= 140 / 2$$
$$= 70.000 \text{ mm.}$$

Radius of Gyration [r]:

$$= \sqrt{ \text{Total Inertia} / \text{Total Area} }$$
$$= \sqrt{ 274 / 40.6 }$$
$$= 25.985 \text{ mm.}$$

Length of Outer Rib [L]:

$$= \text{Saddle Height} - \cos(\text{theta} / 2) (\text{radius} + \text{shlthk} + \text{wpdthk}) - \text{bpthk}$$
$$= 750 - \cos(120 / 2) (462 + 12 + 12) - 15$$
$$= 491.750 \text{ mm.}$$

Intermediate Term [Cc]:

$$= \sqrt{ 2 * \pi^2 * \text{Elastic Modulus} / \text{Yield Stress} }$$
$$= \sqrt{ 2 * \pi^2 * 199943008 / 207 }$$
$$= 138.135$$

Slenderness ratio [KL/r]:

$$= KL / r$$
$$= 1 * 492 / 26$$
$$= 18.925$$

Bending Moment [Rm]:

$$= Fl / (2 * B_{plen}) * e * L / 2$$
$$= 962 / (2 * 900) * 421 * 492 / 2$$
$$= 55.367 \text{ Kg-m.}$$

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Compressive Allowable, $KL/r < Cc(18.9 < 138)$ per AISC E2-1 [Sca]:

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

$$= (1 - (18.9)^2 / (2 * 138^2)) 207 / (5/3 + 3 * (18.9) / (8 * 138) - (18.9^3) / (8 * 138^3))$$

$$= 119.3 \text{ N./mm}^2$$

AISC Unity Check of Outside Ribs (must be ≤ 1)

$$= Sc / Sca + (Rm * C1 / I) / Sba$$

$$= 2.56 / 119 + (55.4 * 70 / 2744000) / 138$$

$$= 0.122$$

Check of Inside Ribs:

Inertia of Saddle, Inner Ribs - Axial Direction

	B	D	Y	A	AY	Io
Rib	12.0	128.0	0.0	15.4	0.0	274.
Web	421.3	12.0	0.0	50.6	0.0	6.07
Totals	65.9	...	280.

Distance to Centroid from Datum [ytot]:

$$= AY / A$$

$$= 0 / 65.9$$

$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= \text{Saddle Width} / 2$$

$$= 140 / 2$$

$$= 70.000 \text{ mm.}$$

Length of Inner Rib [L]:

$$= \text{Saddle Height} - \text{Outside Radius} - \text{Bpthk}$$

$$= 750 - \cos(486 / 2) (15 + 0 + 0) - 0$$

$$= 248.500 \text{ mm.}$$

Radius of Gyration [r]:

$$= \text{sqrt}(\text{Total Inertia} / \text{Total Area})$$

$$= \text{sqrt}(280 / 65.9)$$

$$= 20.621 \text{ mm.}$$

Slenderness ratio [KL/r]:

$$= KL / r$$

$$= 1 * 248 / 20.6$$

$$= 12.051$$

Unit Force [Force,u]:

$$= F1 / (2 * \text{Baseplate Length})$$

$$= 962 / (2 * 900)$$

$$= 0.534 \text{ Kgf/mm.}$$

Moment at base of inner Rib [Mbase,c]:

$$= \text{Unit Force} * e * L$$

$$= 0.53 * 421 * 248$$

$$= 55.958 \text{ Kg-m.}$$

Bending Stress due to Transverse Force and Weight Load [SigmaB,base,c]:

$$= \text{Bending Moment} / \text{Section Modulus}$$

$$= 56 / 40042$$

$$= 13.705 \text{ N./mm}^2$$

Compressive Allowable, $KL/r < Cc(12.1 < 138)$ per AISC E2-1 [Sca]:

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

$$= (1 - (12.1)^2 / (2 * 138^2)) 207 / (5/3 + 3 * (12.1) / (8 * 138) - (12.1^3) / (8 * 138^3))$$

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$$= 121.3 \text{ N./mm}^2$$

AISC Unity Check of Inside Ribs (must be <= 1)

$$= Sc/Sca + (Mbase,c * C1/I)/Sba$$

$$= 3.13/121 + (56 * 70/280)/138$$

$$= 0.125$$

Input Data for Base Plate Bolting Calculations:

Total Number of Bolts per BasePlate	Nbolts	4	
Total Number of Bolts in Tension/Baseplate	Nbt	2	
Bolt Material Specification		SA-193 B7	
Bolt Allowable Stress	Stba	283.00	N./mm ²
Bolt Corrosion Allowance	Bca	0.0	mm.
Distance from Bolts to Edge	Edgedis	120.0	mm.
Nominal Bolt Diameter	Bnd	20.0000	mm.
Thread Series	Series	TEMA Metric	
BasePlate Allowable Stress	S	108.25	N./mm ²
Area Available in a Single Bolt	BltArea	2.1705	cm ²
Saddle Load QO (Weight)	QO	3346.0	Kgf
Saddle Load QL (Wind/Seismic contribution)	QL	508.1	Kgf
Maximum Transverse Force	Ft	481.0	Kgf
Maximum Longitudinal Force	F1	962.0	Kgf
Saddle Bolted to Steel Foundation	No		

Shear Stress in a Single Bolt, Longitudinal Direction [taub,l]:

$$= F1 / (Bolt Area * Number of Bolts)$$

$$= 962 / (2.17 * 4)$$

$$= 10.9 \text{ N./mm}^2. \text{ Must be less than } 146.2 \text{ N./mm}^2.$$

Shear Stress in a Single Bolt, Transverse Direction [taub,t]:

$$= Ft / (Bolt Area * Number of Bolts)$$

$$= 481 / (2.17 * 4)$$

$$= 5.4 \text{ N./mm}^2. \text{ Must be less than } 146.2 \text{ N./mm}^2.$$

Bolt Area Calculation per Dennis R. Moss

Bolt Area Requirement Due to Longitudinal Load [Bltarearl]:

$$= 0.0 \text{ (QO > QL --> No Uplift in Longitudinal direction)}$$

Bolt Area due to Shear Load [Bltarears]:

$$= F1 / (BoltShearAllowable * Nbolts)$$

$$= 962 / (146 * 4)$$

$$= 0.1613 \text{ cm}^2$$

Bolt Area due to Transverse Load:

Moment on Baseplate Due to Transverse Load [Rmom]:

$$= B * Ft + \text{Sum of X Moments}$$

$$= 750 * 481 + 0$$

$$= 360.77 \text{ Kg-m.}$$

Eccentricity (e):

$$= Rmom / QO$$

$$= 361/3346$$

$$= 107.82 \text{ mm. < Bplen/6 --> No Uplift in Transverse direction}$$

Bolt Area due to Qtransverse Load [Bltareart]:

$$= 0 \text{ (No Uplift)}$$

Required Area of a Single Bolt [Bltarear]:

$$= \text{max}[Bltarearl, Bltarears, Bltareart]$$

$$= \text{max}[0, 0.16, 0]$$

$$= 0.1613 \text{ cm}^2$$

FileName : Chiller-Rev.01 -----

Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

ASME VIII Division 2 Horizontal Vessel Analysis, Right Saddle:

Input and Calculated Values:

Vessel Mean Radius	Rm	470.00	mm.
Shell Thickness used in this Case	t	12.000	mm.
Stiffened Vessel Length per 4.15.6	L	3050.00	mm.
Distance from Saddle to Vessel tangent	a1 or a	600.00	mm.
Saddle Width	b1 or b	140.00	mm.
Saddle Bearing Angle	delta or theta	120.00	degrees
Wear Plate Width	b2 or b1	300.00	mm.
Wear Plate Bearing Angle	delta2 or theta1	132.00	degrees
Wear Plate Thickness	e2 or tr	12.0	mm.
Wear Plate Allowable Stress	fw or Sr	137.90	N./mm ²
Inside Depth of Head	Hi or h2	234.25	mm.
Shell Allowable Stress used in Calculation		137.90	N./mm ²
Head Allowable Stress used in Calculation		137.90	N./mm ²
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Distance from Saddle Base to Centerline	B	750.00	mm.
Coefficient of Friction	mu	0.30	
Saddle Force Q, Operating Case		1759.37	Kgf
Pressure used in Analysis	P	22.024	bars
Horizontal Vessel Analysis Results:	Actual Allowable		
	N./mm ² N./mm ²		

Long. Stress at Top of Midspan	57.19	137.90	
Long. Stress at Bottom of Midspan	57.83	137.90	
Long. Stress at Top of Saddles	61.09	137.90	
Long. Stress at Bottom of Saddles	55.53	137.90	

Tangential Shear in Shell	2.63	110.32	
Circ. Stress at Horn of Saddle	4.67	172.37	
Circ. Compressive Stress in Shell	0.26	137.90	

Intermediate Results: Saddle Reaction Q due to Wind or Seismic:

Saddle Reaction Force due to Wind Ft [Fwt]:

$$\begin{aligned}
 &= Ftr(Ft/Num of Saddles + Z Force Load) * B / E \\
 &= 3(477/2 + 0) * 750/843 \\
 &= 637.0 Kgf
 \end{aligned}$$

Saddle Reaction Force due to Wind Fl [Fwl]:

$$\begin{aligned}
 &= \max(Fl, Sum of X Forces) * B / Ls \\
 &= \max(115, 0) * 750/1420 \\
 &= 60.7 Kgf
 \end{aligned}$$

Saddle Reaction Force due to Earthquake Fl [Fsl]:

$$\begin{aligned}
 &= \max(Fl, Sum of X Forces) * B / Ls \\
 &= \max(962, 0) * 750/1420 \\
 &= 508.1 Kgf
 \end{aligned}$$

Saddle Reaction Force due to Earthquake Ft [Fst]:

$$\begin{aligned}
 &= Ftr(Ft/Num of Saddles + Z Force Load) * B / E \\
 &= 3(962/2 + 0) * 750/843
 \end{aligned}$$

FileName : Chiller-Rev.01 -----

Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

$$= 1284.4 \text{ Kgf}$$

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

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

$$= 475 + \max(60.7, 637, 508, 1284)$$

$$= 1759.4 \text{ Kgf}$$

Longitudinal Wind Force [FI]:

$$= \text{WindScalar} * \text{WindPress}(\text{Platform Area} + (\text{End Area} * \text{WindDiaMult}))$$

$$= 1 * 1046(0 + (0.9 * 1.2))$$

$$= 1126.093 \text{ N}$$

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)		1865.00	Kgf
Transverse Shear Load Saddle	Ft	481.01	Kgf
Longitudinal Shear Load Saddle		962.03	Kgf

Formulas and Substitutions for Horizontal Vessel Analysis:

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

Saddle Dimension [E]:

$$= \min(2(\text{ShellID}/2 + t + \text{WearPadThickness})\sin(\text{theta}/2), 2 * \text{Rm})$$

$$= \min(2(925/2 + 12 + 12)\sin(120/2), 2 * 470)$$

$$= 842.643 \text{ mm.}$$

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 = 0.0529	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434
K7p = 0.0434			

The suffix 'p' denotes the values for a wear plate if it exists.

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

Moment per Equation 4.15.3 [M1]:

$$= -Q * a [1 - (1 - a/L + (\text{Rm}^2 - h^2) / (2a * L)) / (1 + (4h^2) / (3L))]$$

$$= -1759 * 600 [1 - (1 - 600/3050 + (470^2 - 234^2) / (2 * 600 * 3050)) / (1 + (4 * 234) / (3 * 3050))]$$

$$= -243.0 \text{ Kg-m.}$$

Moment per Equation 4.15.4 [M2]:

$$= Q * L / 4 (1 + 2(\text{Rm}^2 - h^2) / (L^2)) / (1 + (4h^2) / (3L)) - 4a / L$$

$$= 1759 * 3050 / 4 (1 + 2(470^2 - 234^2) / (3050^2)) / (1 + (4 * 234) / (3 * 3050)) - 4 * 600 / 3050$$

$$= 204.7 \text{ Kg-m.}$$

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

$$= P * \text{Rm} / (2t) - M2 / (\text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) - 205 / (\text{pi} * 470^2 * 9)$$

$$= 57.19 \text{ N./mm}^2$$

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

$$= P * \text{Rm} / (2t) + M2 / (\text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) + 205 / (\text{pi} * 470^2 * 9)$$

$$= 57.83 \text{ N./mm}^2$$

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

$$= P * \text{Rm} / (2t) - M1 / (K1 * \text{pi} * \text{Rm}^2 * t)$$

$$= 22 * 470 / (2 * 9) - 243 / (0.11 * \text{pi} * 470^2 * 9)$$

$$= 61.09 \text{ N./mm}^2$$

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

$$= P * \text{Rm} / (2t) + M1 / (K1 * \text{pi} * \text{Rm}^2 * t)$$

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$$= 22*470/(2*9)+-243/(0.19*pi*470^2*9)$$

$$= 55.53 \text{ N./mm}^2$$

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

$$= Q(L-2a)/(L+(4*h^2/3))$$

$$= 1759(3050 - 2 * 600)/(3050 + (4 * 234/3))$$

$$= 968.0 \text{ Kgf}$$

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

$$= K2 * T / (Rm * t)$$

$$= 1.17 * 968 / (470 * 9)$$

$$= 2.63 \text{ N./mm}^2$$

Decay Length (4.15.22) [x1,x2]:

$$= 0.78 * \text{sqrt}(Rm * t)$$

$$= 0.78 * \text{sqrt}(470 * 9)$$

$$= 50.730 \text{ mm.}$$

Effective reinforcing plate width (4.15.1) [B1]:

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

$$= \min(140 + 1.56 * \text{sqrt}(470 * 9), 2 * 600)$$

$$= 241.46 \text{ mm.}$$

Wear Plate/Shell Stress ratio (4.15.29) [eta]:

$$= \min(Sr/S, 1)$$

$$= \min(138/138, 1)$$

$$= 1.0000$$

Circumferential Stress at Saddle Base with Wear Plate (4.15.26) [sigma6,r]:

$$= -K5 * Q * k / (B1(t + eta * tr))$$

$$= -0.76 * 1759 * 0.1 / (241(9 + 1 * 12))$$

$$= -0.26 \text{ N./mm}^2$$

Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:

$$= -Q/(4(t+eta*tr)b1) - 12*K7*Q*Rm/(L(t+eta*tr)^2)$$

$$= -1759/(4(9 + 1 * 12)241) -$$

$$12*0.053*1759*470/(3050(9+1*12)^2)$$

$$= -4.67 \text{ N./mm}^2$$

Results for Vessel Ribs, Web and Base:

Baseplate Length	Bplen	900.0000	mm.
Baseplate Thickness	Bpthk	15.0000	mm.
Baseplate Width	Bpwid	170.0000	mm.
Number of Ribs (inc. outside ribs)	Nribs	3	
Rib Thickness	Ribtk	12.0000	mm.
Web Thickness	Webtk	12.0000	mm.
Web Location	Webloc	Center	
Saddle Yield Stress	Sy	206.9	N./
Height of Web at Center	Hw,c	367.5	mm.
Friction Coefficient	mu	0.300	

Note: In the tables below lo is I for the rectangle + Area * Centroid Distance^2

Moment of Inertia of Saddle - Transverse Direction (90 degrees to long axis)

	B	D	Y	A	AY	Io
Shell	401.0	9.0	4.5	36.1	16239.4	0.295E+04
Wearplate	300.0	12.0	15.0	36.0	54000.0	0.231E+04
Web	12.0	248.5	145.2	29.8	433135.5	0.229E+04
BasePlate	170.0	15.0	277.0	25.5	706350.0	0.846E+04
Totals	127.4	1209724.9	0.160E+05

Distance to Centroid [C1]:

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$$\begin{aligned} &= AY / A \\ &= 476/127 \\ &= 94.949 \text{ mm.} \end{aligned}$$

Angle [beta]:

$$\begin{aligned} &= 180 - \text{Saddle Angle}/2 \\ &= 180 - 120/2 \\ &= 120.0 \end{aligned}$$

Saddle Splitting Coefficient [K1]:

$$\begin{aligned} &= (1 + \cos(\text{beta}) - 0.5*\sin(\text{beta})^2) / (\pi - \text{beta} + \sin(\text{beta})\cos(\text{beta})) \\ &= (1 + \cos(120) - 0.5*\sin(120)^2) / (\pi - 2.09 + \sin(120)\cos(120)) \\ &= 0.2035 \end{aligned}$$

Saddle Splitting Force [Fh]:

$$\begin{aligned} &= K1 * Q \\ &= 0.2 * 1759 \\ &= 358.0696 \text{ Kgf} \end{aligned}$$

$$\begin{aligned} \text{Tension Stress, St} &= (Fh/As) = 0.3845 \text{ N./mm}^2 \\ \text{Allowed Stress, Sa} &= 0.6 * \text{Yield Str} = 124.1100 \text{ N./mm}^2 \end{aligned}$$

Saddle Splitting Dimension [d]:

$$\begin{aligned} &= B - R * \sin(\text{theta}/2) / (\text{theta}/2 \text{ in radians}) \\ &= 750 - 466 * \sin(120/2) / 1.05 \\ &= 365.035 \text{ mm.} \end{aligned}$$

$$\text{Bending Moment, M} = Fh * d = 130.7104 \text{ Kg-m.}$$

$$\begin{aligned} \text{Bending Stress, Sb} &= (M * C1 / I) = 0.7604 \text{ N./mm}^2 \\ \text{Allowed Stress, Sa} &= 2/3 * \text{Yield Str} = 137.9000 \text{ N./mm}^2 \end{aligned}$$

Minimum Thickness of Baseplate per Moss:

$$\begin{aligned} &= (3(Q + \text{Saddle_Wt}) \text{BasePlateWidth} / (4 * \text{BasePlateLength} * \text{AllStress}))^{1/2} \\ &= (3(1759 + 106)170 / (4 * 900 * 138))^{1/2} \\ &= 4.335 \text{ mm.} \end{aligned}$$

Calculation of Axial Load, Intermediate Values and Compressive Stress:

Web Length Dimension [Web Length]:

$$\begin{aligned} &= 2 * \cos(90 - \text{Saddle Angle}/2) (\text{Inside Radius} + \text{Shell Thk} + \text{Wear Plate Thk}) \\ &= 2 * \cos(90 - 120/2) (462 + 12 + 12) \\ &= 842.643 \text{ mm.} \end{aligned}$$

Distance between Ribs [e]:

$$\begin{aligned} &= \text{Web Length} / (\text{Nr ribs} - 1) \\ &= 843 / (3 - 1) \\ &= 421.321 \text{ mm.} \end{aligned}$$

Baseplate Pressure Area [Ap]:

$$\begin{aligned} &= e * \text{Bpwid} / 2 \\ &= 421 * 170 / 2 \\ &= 358.123 \text{ cm}^2 \end{aligned}$$

Bearing Pressure [Bp]:

$$\begin{aligned} &= Q / (\text{BasePlateLength} * \text{BasePlateWidth}) \\ &= 1759 / (900 * 170) \\ &= 1.150 \text{ Kgf/cm}^2 \end{aligned}$$

Axial Load [P]:

$$\begin{aligned} &= Ap * Bp \\ &= 358 * 1.15 \\ &= 411.811 \text{ Kgf} \end{aligned}$$

Area of the Rib and Web [Ar]:

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$$\begin{aligned}
 &= \text{Rib Area} + \text{Web Area} \\
 &= 15.4 + 25.3 \\
 &= 40.639 \text{ cm}^2
 \end{aligned}$$

Compressive Stress [Sc]:

$$\begin{aligned}
 &= P/A_r \\
 &= 412/40.6 \\
 &= 0.994 \text{ N./mm}^2
 \end{aligned}$$

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	B	D	Y	A	AY	Io
-----	-----	-----	-----	-----	-----	-----
Rib+Web	12.0	140.0	...	16.8	...	274.

Rib dimension [D]:

$$\begin{aligned}
 &= \text{Saddle Width} - \text{Web Thickness} \\
 &= 140 - 12 \\
 &= 128.000 \text{ mm.}
 \end{aligned}$$

Distance to Centroid from Datum [ytot]:

$$\begin{aligned}
 &= AY / A \\
 &= 0/40.6 \\
 &= 0.000 \text{ mm.}
 \end{aligned}$$

Distance to Centroid [C1]:

$$\begin{aligned}
 &= \text{Saddle Width} / 2 \\
 &= 140/2 \\
 &= 70.000 \text{ mm.}
 \end{aligned}$$

Radius of Gyration [r]:

$$\begin{aligned}
 &= \text{sqrt}(\text{Total Inertia} / \text{Total Area}) \\
 &= \text{sqrt}(274/40.6) \\
 &= 25.985 \text{ mm.}
 \end{aligned}$$

Length of Outer Rib [L]:

$$\begin{aligned}
 &= \text{Saddle Height} - \cos(\text{theta}/2) (\text{radius} + \text{shlthk} + \text{wpdthk}) - \text{bpthk} \\
 &= 750 - \cos(120/2) (462 + 12 + 12) - 15 \\
 &= 491.750 \text{ mm.}
 \end{aligned}$$

Intermediate Term [Cc]:

$$\begin{aligned}
 &= \text{sqrt}(2 * \text{pi}^2 * \text{Elastic Modulus} / \text{Yield Stress}) \\
 &= \text{sqrt}(2 * \text{pi}^2 * 199943008/207) \\
 &= 138.135
 \end{aligned}$$

Slenderness ratio [KL/r]:

$$\begin{aligned}
 &= KL/r \\
 &= 1 * 492/26 \\
 &= 18.925
 \end{aligned}$$

Bending Moment [Rm]:

$$\begin{aligned}
 &= F1 / (2 * Bplen) * e * L / 2 \\
 &= 962 / (2 * 900) * 421 * 492/2 \\
 &= 55.367 \text{ Kg-m.}
 \end{aligned}$$

Compressive Allowable, KL/r < Cc(18.9 < 138) per AISC E2-1 [Sca]:

$$\begin{aligned}
 &= (1 - (KL/r)^2 / (2 * Cc^2)) Fy / (5/3 + 3 * (KL/r) / (8 * Cc) - (KL/r^3) / (8 * Cc^3)) \\
 &= (1 - (18.9)^2 / (2 * 138^2)) 207 / \\
 &\quad (5/3 + 3 * (18.9) / (8 * 138) - (18.9^3) / (8 * 138^3)) \\
 &= 119.3 \text{ N./mm}^2
 \end{aligned}$$

AISC Unity Check of Outside Ribs (must be <= 1)

$$= Sc/Sca + (Rm * C1 / I) / Sba$$

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$$= 0.99/119 + (55.4 * 70/2744000)/138$$

$$= 0.109$$

Check of Inside Ribs:

Inertia of Saddle, Inner Ribs - Axial Direction

	B	D	Y	A	AY	Io
Rib	12.0	128.0	0.0	15.4	0.0	274.
Web	421.3	12.0	0.0	50.6	0.0	6.07
Totals	65.9	...	280.

Distance to Centroid from Datum [ytot]:

$$= AY / A$$

$$= 0/65.9$$

$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= \text{Saddle Width} / 2$$

$$= 140/2$$

$$= 70.000 \text{ mm.}$$

Length of Inner Rib [L]:

$$= \text{Saddle Height} - \text{Outside Radius} - \text{Bpthk}$$

$$= 750 - \cos(486/2) (15 + 0 + 0) - 0$$

$$= 248.500 \text{ mm.}$$

Radius of Gyration [r]:

$$= \sqrt{ \text{Total Inertia} / \text{Total Area} }$$

$$= \sqrt{ 280/65.9 }$$

$$= 20.621 \text{ mm.}$$

Slenderness ratio [KL/r]:

$$= KL/r$$

$$= 1 * 248/20.6$$

$$= 12.051$$

Unit Force [Force,u]:

$$= F1 / (2 * \text{Baseplate Length})$$

$$= 962 / (2 * 900)$$

$$= 0.534 \text{ Kg/mm.}$$

Moment at base of inner Rib [Mbase,c]:

$$= \text{Unit Force} * e * L$$

$$= 0.53 * 421 * 248$$

$$= 55.958 \text{ Kg-m.}$$

Bending Stress due to Transverse Force and Weight Load [SigmaB,base,c]:

$$= \text{Bending Moment} / \text{Section Modulus}$$

$$= 56/40042$$

$$= 13.705 \text{ N./mm}^2$$

Compressive Allowable, KL/r < Cc(12.1 < 138) per AISC E2-1 [Sca]:

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

$$= (1 - (12.1)^2 / (2 * 138^2)) 207 /$$

$$(5/3 + 3*(12.1) / (8 * 138) - (12.1^3) / (8 * 138^3))$$

$$= 121.3 \text{ N./mm}^2$$

AISC Unity Check of Inside Ribs (must be <= 1)

$$= Sc/Sca + (Mbase,c * C1/I)/Sba$$

$$= 1.22/121 + (56 * 70/280)/138$$

$$= 0.109$$

Input Data for Base Plate Bolting Calculations:

Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

Total Number of Bolts per BasePlate	Nbolts	4	
Total Number of Bolts in Tension/Baseplate	Nbt	2	
Bolt Material Specification		SA-193 B7	
Bolt Allowable Stress	Stba	283.00	N./mm ²
Bolt Corrosion Allowance	Bca	0.0	mm.
Distance from Bolts to Edge	Edgedis	120.0	mm.
Nominal Bolt Diameter	Bnd	20.0000	mm.
Thread Series	Series	TEMA Metric	
BasePlate Allowable Stress	S	108.25	N./mm ²
Area Available in a Single Bolt	BltArea	2.1705	cm ²
Saddle Load QO (Weight)	QO	580.6	Kgf
Saddle Load QL (Wind/Seismic contribution)	QL	508.1	Kgf
Maximum Transverse Force	Ft	481.0	Kgf
Maximum Longitudinal Force	F1	962.0	Kgf
Saddle Bolted to Steel Foundation		No	

Shear Stress in a Single Bolt, Longitudinal Direction [taub,l]:
 = F1 / (Bolt Area * Number of Bolts)
 = 962 / (2.17 * 4)
 = 10.9 N./mm². Must be less than 146.2 N./mm².

Shear Stress in a Single Bolt, Transverse Direction [taub,t]:
 = Ft / (Bolt Area * Number of Bolts)
 = 481 / (2.17 * 4)
 = 5.4 N./mm². Must be less than 146.2 N./mm².

Bolt Area Calculation per Dennis R. Moss

Bolt Area Requirement Due to Longitudinal Load [Bltarearl]:
 = 0.0 (QO > QL --> No Uplift in Longitudinal direction)

Bolt Area due to Shear Load [Bltarears]:
 = F1 / (BoltShearAllowable * Nbolts)
 = 962 / (146 * 4)
 = 0.1613 cm²

Bolt Area due to Transverse Load:

Moment on Baseplate Due to Transverse Load [Rmom]:
 = B * Ft + Sum of X Moments
 = 750 * 481 + 0
 = 360.77 Kg-m.

Eccentricity (e):
 = Rmom / QO
 = 361/581
 = 621.35 mm. > Bplen/6 --> Uplift in Transverse direction

f = Bplen / 2 - Edgedis
 = 900/2 - 120
 = 330.00 mm.

Modular Ratio Of Steel/Concrete (n1):
 = ES / EC
 = 203402/21526
 = 9.45

K1 = 3 (e - 0.5 * Bplen)
 = 3(621 - 0.5*900)
 = 514.04 mm.

K2 = 6 * n1 * At / Bpwid * (f + e)
 = 6 * 9.45 * 4.34/170 * (330 + 621)
 = 137727.18 mm.²

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$$\begin{aligned} K3 &= -K2 * (0.5 * Bplen + f) \\ &= -137727 * (0.5 * 900 + 330) \\ &= -107427192.90 \text{ mm.}^3 \end{aligned}$$

Iteratively Solving for the Effective Bearing Length:

$$\begin{aligned} Y^3 + K1 * Y^2 + K2 * Y + K3 &= 0 \\ Y^3 + 514 * Y^2 + 137727 * Y + -107427200 &= 0 \\ Y &= 289.81 \text{ mm.} \end{aligned}$$

$$\begin{aligned} \text{Num} &= (Bplen / 2 - Y / 3 - e) \\ &= (900/2 - 290/3 - 621) \\ &= -267.95 \end{aligned}$$

$$\begin{aligned} \text{Denom} &= (Bplen / 2 - Y / 3 + f) \\ &= (900/2 - 290/3 + 330) \\ &= 683.40 \end{aligned}$$

Total Bolt Tension Force [Tforce]:

$$\begin{aligned} &= - QO * \text{Num} / \text{Denom} \\ &= - 581 * -268/683 \\ &= 227.65 \text{ Kgf} \end{aligned}$$

Bolt Area Required due to Transverse Load [Bltareart]:

$$\begin{aligned} &= Tforce / (\text{Stba} * \text{Nbt}) \\ &= 228 / (283 * 2) \\ &= 0.0394 \text{ cm}^2 \end{aligned}$$

Required Area of a Single Bolt [Bltarear]:

$$\begin{aligned} &= \max[\text{Bltarearl}, \text{Bltarears}, \text{Bltareart}] \\ &= \max[0, 0.16, 0.039] \\ &= 0.1613 \text{ cm}^2 \end{aligned}$$

Baseplate Thickness Calculation per D. Moss:

Bearing Pressure (fc)

$$\begin{aligned} &= 2(QO + Tforce) / (Y * Bpwid) \\ &= 2(581 + 228) / (290 * 170) \\ &= 3.22 \text{ bars} \end{aligned}$$

Distance from Baseplate Edge to the Web [ADIST]:

$$\begin{aligned} &= (Bplen - \text{Weblength}) / 2 \\ &= (900 - 849) / 2 \\ &= 25.4000 \text{ mm.} \end{aligned}$$

Overturning Moment due To Bolt Tension [Mt]:

$$\begin{aligned} &= Tforce * \text{Adist} \\ &= 228 * 25.4 \\ &= 5.78 \text{ Kg-m.} \end{aligned}$$

Equivalent Bearing Pressure (f1):

$$\begin{aligned} &= fc * (Y - \text{Adist}) / Y \\ &= 3.22 * (290 - 25.4) / 290 \\ &= 2.94 \text{ bars} \end{aligned}$$

Overturning Moment due to Bearing Pressure [Mc]:

$$\begin{aligned} &= (\text{Adist}^2 * Bpwid / 6) * (f1 + 2 * fc) \\ &= (25.4^2 * 170/6) * (2.94 + 2 * 3.22) \\ &= 1.75 \text{ Kg-m.} \end{aligned}$$

Baseplate Required Thickness [Treq]:

$$\begin{aligned} &= (6 * \max(\text{Mt}, \text{Mc}) / (\text{Bpwid} * \text{Sba}))^{1/2} \\ &= (6 * \max(5.78, 1.75) / (170 * 162))^{1/2} \\ &= 3.5108 \text{ mm.} \end{aligned}$$

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Saddle Calcs: Operating Case: Step: 12 12:29pm May 21,2024

Warning:

Please note that nozzle loadings, if included, are assumed to be local in nature and will not contribute to or create a net section bending moment. Therefore, the addition of nozzle loads will not affect the support load calculation. If you wish to create a load on the legs/lugs etc. from forces or moments, use the Force/Moment dialog and add the appropriate forces and moments at the correct locations in the model. Ensure that all supports are designed to take into account all possible loading conditions.

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ASME VIII Division 2 Horizontal Vessel Analysis, Left Saddle:

Horizontal Vessel Stress Calculations : Test Case

*Warning - Distance to Saddle (a) > 0.25 * Tangent Distance (L) - 4.15.3.2*

Input and Calculated Values:

Vessel Mean Radius	Rm	470.00	mm.
Shell Thickness used in this Case	t	12.000	mm.
Stiffened Vessel Length per 4.15.6	L	3050.00	mm.
Distance from Saddle to Vessel tangent	a1 or a	800.00	mm.
Saddle Width	b1 or b	140.00	mm.
Saddle Bearing Angle	delta or theta	120.00	degrees
Wear Plate Width	b2 or b1	300.00	mm.
Wear Plate Bearing Angle	delta2 or theta1	132.00	degrees
Wear Plate Thickness	e2 or tr	12.0	mm.
Wear Plate Allowable Stress	fw or Sr	137.90	N./mm ²
Shell Allowable Stress used in Calculation		235.81	N./mm ²
Head Allowable Stress used in Calculation		235.81	N./mm ²
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Distance from Saddle Base to Centerline	B	750.00	mm.
Coefficient of Friction	mu	0.30	
Saddle Force Q, Test Case, no Ext. Forces		3919.74	Kgf
Pressure used in Analysis	P	28.646	bars
Horizontal Vessel Analysis Results:	Actual Allowable		
	N./mm ² N./mm ²		

Long. Stress at Top of Midspan	74.81	235.81	
Long. Stress at Bottom of Midspan	74.79	235.81	
Long. Stress at Top of Saddles	84.83	235.81	
Long. Stress at Bottom of Saddles	69.25	235.81	

Tangential Shear in Shell	5.06	139.65	
Circ. Stress at Horn of Saddle	10.41	353.71	
Circ. Compressive Stress in Shell	0.58	235.81	

Intermediate Results: Saddle Reaction Q due to Wind or Seismic:

Saddle Reaction Force due to Wind Ft [Fwt]:

$$\begin{aligned}
 &= F_{tr} (F_t / \text{Num of Saddles} + Z \text{ Force Load}) * B / E \\
 &= 3 (157/2 + 0) * 750/843 \\
 &= 210.2 \text{ Kgf}
 \end{aligned}$$

Saddle Reaction Force due to Wind Fl [Fwl]:

$$\begin{aligned}
 &= \max (F_l, \text{Sum of X Forces}) * B / L_s \\
 &= \max (37.9, 0) * 750/1420 \\
 &= 20.0 \text{ Kgf}
 \end{aligned}$$

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

$$\begin{aligned}
 &= \text{Saddle Load} + \max (F_{wl}, F_{wt}, F_{sl}, F_{st}) \\
 &= 3710 + \max (20, 210, 0, 0) \\
 &= 3919.7 \text{ Kgf}
 \end{aligned}$$

Longitudinal Wind Force [Fl]:

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$$= \text{WindScalar} * \text{WindPress}(\text{Platform Area} + (\text{End Area} * \text{WindDiaMult}))$$

$$= 1 * 1046(0 + (0.9 * 1.2))$$

$$= 1126.093 \text{ N}$$

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)		4025.37	Kgf
Transverse Shear Load Saddle	Ft	78.72	Kgf
Longitudinal Shear Load Saddle		37.89	Kgf

Formulas and Substitutions for Horizontal Vessel Analysis:

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

Saddle Dimension [E]:

$$= \min(2(\text{ShellID}/2 + t + \text{WearPadThickness})\sin(\text{theta}/2), 2 * R_m)$$

$$= \min(2(925/2 + 12 + 12)\sin(120/2), 2 * 470)$$

$$= 842.643 \text{ mm.}$$

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 = 0.0529	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434
K7p = 0.0434			

The suffix 'p' denotes the values for a wear plate if it exists.

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

Moment per Equation 4.15.3 [M1]:

$$= -Q * a [1 - (1 - a/L + (R_m^2 - h^2) / (2a * L)) / (1 + (4h^2) / (3L))]$$

$$= -3920 * 800 [1 - (1 - 800 / 3050 + (470^2 - 0^2) / (2 * 800 * 3050)) / (1 + (4 * 0) / (3 * 3050))]$$

$$= -680.6 \text{ Kg-m.}$$

Moment per Equation 4.15.4 [M2]:

$$= Q * L / 4 (1 + 2(R_m^2 - h^2) / (L^2)) / (1 + (4h^2) / (3L)) - 4a / L$$

$$= 3920 * 3050 / 4 (1 + 2(470^2 - 0^2) / (3050^2)) / (1 + (4 * 0) / (3 * 3050)) - 4 * 800 / 3050$$

$$= -5.0 \text{ Kg-m.}$$

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

$$= P * R_m / (2t) - M2 / (\pi * R_m^2 * t)$$

$$= 28.6 * 470 / (2 * 9) - -5.04 / (\pi * 470^2 * 9)$$

$$= 74.81 \text{ N./mm}^2$$

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

$$= P * R_m / (2t) + M2 / (\pi * R_m^2 * t)$$

$$= 28.6 * 470 / (2 * 9) + -5.04 / (\pi * 470^2 * 9)$$

$$= 74.79 \text{ N./mm}^2$$

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

$$= P * R_m / (2t) - M1 / (K1 * \pi * R_m^2 * t)$$

$$= 28.6 * 470 / (2 * 9) - -681 / (0.11 * \pi * 470^2 * 9)$$

$$= 84.83 \text{ N./mm}^2$$

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

$$= P * R_m / (2t) + M1 / (K1 * \pi * R_m^2 * t)$$

$$= 28.6 * 470 / (2 * 9) + -681 / (0.19 * \pi * 470^2 * 9)$$

$$= 69.25 \text{ N./mm}^2$$

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

$$= Q(L - 2a) / (L + (4 * h^2) / 3)$$

$$= 3920(3050 - 2 * 800) / (3050 + (4 * 0) / 3)$$

$$= 1863.5 \text{ Kgf}$$

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Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:
 = $K2 * T / (Rm * t)$
 = $1.17 * 1863 / (470 * 9)$
 = 5.06 N./mm²

Decay Length (4.15.22) [x1,x2]:
 = $0.78 * \text{sqrt}(Rm * t)$
 = $0.78 * \text{sqrt}(470 * 9)$
 = 50.730 mm.

Effective reinforcing plate width (4.15.1) [B1]:
 = $\text{min}(b + 1.56 * \text{sqrt}(Rm * t), 2a)$
 = $\text{min}(140 + 1.56 * \text{sqrt}(470 * 9), 2 * 800)$
 = 241.46 mm.

Wear Plate/Shell Stress ratio (4.15.29) [eta]:
 = 1.0000 Materials are the same, test case

Circumferential Stress at Saddle Base with Wear Plate (4.15.26) [sigma6,r]:
 = $-K5 * Q * k / (B1(t + \text{eta} * \text{tr}))$
 = $-0.76 * 3920 * 0.1 / (241(9 + 1 * 12))$
 = -0.58 N./mm²

Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:
 = $-Q / (4(t + \text{eta} * \text{tr}) b1) - 12 * K7 * Q * Rm / (L(t + \text{eta} * \text{tr})^2)$
 = $-3920 / (4(9 + 1 * 12) 241) - 12 * 0.053 * 3920 * 470 / (3050(9 + 1 * 12)^2)$
 = -10.41 N./mm²

Results for Vessel Ribs, Web and Base:

Baseplate Length	Bplen	900.0000	mm.
Baseplate Thickness	Bpthk	15.0000	mm.
Baseplate Width	Bpwid	170.0000	mm.
Number of Ribs (inc. outside ribs)	Nribs	3	
Rib Thickness	Ribtck	12.0000	mm.
Web Thickness	Webtk	12.0000	mm.
Web Location	Webloc	Center	
Saddle Yield Stress	Sy	206.9	N./
Height of Web at Center	Hw,c	367.5	mm.
Friction Coefficient	mu	0.300	

Note: In the tables below lo is I for the rectangle + Area * Centroid Distance^2

Moment of Inertia of Saddle - Transverse Direction (90 degrees to long axis)

	B	D	Y	A	AY	Io
Shell	401.0	9.0	4.5	36.1	16239.4	0.295E+04
Wearplate	300.0	12.0	15.0	36.0	54000.0	0.231E+04
Web	12.0	248.5	145.2	29.8	433135.5	0.229E+04
BasePlate	170.0	15.0	277.0	25.5	706350.0	0.846E+04
Totals	127.4	1209724.9	0.160E+05

Distance to Centroid [C1]:

= AY / A
 = 476/127
 = 94.949 mm.

Angle [beta]:

= 180 - Saddle Angle/2
 = 180 - 120/2
 = 120.0

Saddle Splitting Coefficient [K1]:

= $(1 + \cos(\text{beta}) - 0.5 * \sin(\text{beta})^2) / (\pi - \text{beta} + \sin(\text{beta}) \cos(\text{beta}))$

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$$= (1 + \cos(120) - 0.5 * \sin(120)^2) / (\pi - 2.09 + \sin(120) \cos(120))$$
$$= 0.2035$$

Saddle Splitting Force [Fh]:

$$= K1 * Q$$
$$= 0.2 * 3920$$
$$= 797.7523 \text{ Kgf}$$

$$\text{Tension Stress, } St = (Fh / As) = 0.8567 \text{ N./mm}^2$$
$$\text{Allowed Stress, } Sa = 0.6 * \text{Yield Str} = 124.1100 \text{ N./mm}^2$$

Saddle Splitting Dimension [d]:

$$= B - R * \sin(\text{theta}/2) / (\text{theta}/2 \text{ in radians})$$
$$= 750 - 466 * \sin(120/2) / 1.05$$
$$= 365.035 \text{ mm.}$$

$$\text{Bending Moment, } M = Fh * d = 291.2129 \text{ Kg-m.}$$

$$\text{Bending Stress, } Sb = (M * C1 / I) = 1.6942 \text{ N./mm}^2$$
$$\text{Allowed Stress, } Sa = 2/3 * \text{Yield Str} = 137.9000 \text{ N./mm}^2$$

Minimum Thickness of Baseplate per Moss:

$$= (3 (Q + \text{Saddle_Wt}) \text{BasePlateWidth} / (4 * \text{BasePlateLength} * \text{AllStress}))^{1/2}$$
$$= (3(3920 + 106)170 / (4 * 900 * 138))^{1/2}$$
$$= 6.368 \text{ mm.}$$

Calculation of Axial Load, Intermediate Values and Compressive Stress:

Web Length Dimension [Web Length]:

$$= 2 * \cos(90 - \text{Saddle Angle}/2) (\text{Inside Radius} + \text{Shell Thk} + \text{Wear Plate Thk})$$
$$= 2 * \cos(90 - 120/2) (462 + 12 + 12)$$
$$= 842.643 \text{ mm.}$$

Distance between Ribs [e]:

$$= \text{Web Length} / (\text{Nrings} - 1)$$
$$= 843 / (3 - 1)$$
$$= 421.321 \text{ mm.}$$

Baseplate Pressure Area [Ap]:

$$= e * \text{Bpwid} / 2$$
$$= 421 * 170 / 2$$
$$= 358.123 \text{ cm}^2$$

Bearing Pressure [Bp]:

$$= Q / (\text{BasePlateLength} * \text{BasePlateWidth})$$
$$= 3920 / (900 * 170)$$
$$= 2.562 \text{ Kgf/cm}^2$$

Axial Load [P]:

$$= Ap * Bp$$
$$= 358 * 2.56$$
$$= 917.484 \text{ Kgf}$$

Area of the Rib and Web [Ar]:

$$= \text{Rib Area} + \text{Web Area}$$
$$= 15.4 + 25.3$$
$$= 40.639 \text{ cm}^2$$

Compressive Stress [Sc]:

$$= P / Ar$$
$$= 917 / 40.6$$
$$= 2.214 \text{ N./mm}^2$$

Check of Outside Ribs:

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Inertia of Saddle, Outer Ribs - Longitudinal Direction

	B	D	Y	A	AY	Io
-----	-----	-----	-----	-----	-----	-----
Rib+Web	12.0	140.0	...	16.8	...	274.

Rib dimension [D]:

= Saddle Width - Web Thickness
 = 140 - 12
 = 128.000 mm.

Distance to Centroid from Datum [ytot]:

= AY / A
 = 0/40.6
 = 0.000 mm.

Distance to Centroid [C1]:

= Saddle Width / 2
 = 140/2
 = 70.000 mm.

Radius of Gyration [r]:

= sqrt(Total Inertia / Total Area)
 = sqrt(274/40.6)
 = 25.985 mm.

Length of Outer Rib [L]:

= Saddle Height - cos(theta/2) (radius + shlthk + wpdthk) - bpthk
 = 750 - cos(120/2) (462 + 12 + 12) - 15
 = 491.750 mm.

Intermediate Term [Cc]:

= sqrt(2 * pi² * Elastic Modulus / Yield Stress)
 = sqrt(2 * pi² * 199943008/207)
 = 138.135

Slenderness ratio [KL/r]:

= KL/r
 = 1 * 492/26
 = 18.925

Bending Moment [Rm]:

= Fl / (2 * Bplen) * e * L / 2
 = 37.9 / (2 * 900) * 421 * 492/2
 = 2.181 Kg-m.

Compressive Allowable, KL/r < Cc(18.9 < 138) per AISC E2-1 [Sca]:

= (1 - (KLr)² / (2 * Cc²)) Fy / (5/3 + 3 * (KLr) / (8 * Cc) - (KLr³) / (8 * Cc³))
 = (1 - (18.9)² / (2 * 138²)) 207 /
 (5/3 + 3 * (18.9) / (8 * 138) - (18.9³) / (8 * 138³))
 = 119.3 N./mm²

AISC Unity Check of Outside Ribs (must be <= 1)

= Sc/Sca + (Rm * C1 / I) / Sba
 = 2.21/119 + (2.18 * 70/2744000) / 138
 = 0.023

Check of Inside Ribs:

Inertia of Saddle, Inner Ribs - Axial Direction

	B	D	Y	A	AY	Io
-----	-----	-----	-----	-----	-----	-----
Rib	12.0	128.0	0.0	15.4	0.0	274.
Web	421.3	12.0	0.0	50.6	0.0	6.07
Totals	65.9	...	280.

Distance to Centroid from Datum [ytot]:

$$= AY / A$$

$$= 0/65.9$$

$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= \text{Saddle Width} / 2$$

$$= 140/2$$

$$= 70.000 \text{ mm.}$$

Length of Inner Rib [L]:

$$= \text{Saddle Height} - \text{Outside Radius} - \text{Bpthk}$$

$$= 750 - \cos(486/2)(15 + 0 + 0) - 0$$

$$= 248.500 \text{ mm.}$$

Radius of Gyration [r]:

$$= \sqrt{\text{Total Inertia} / \text{Total Area}}$$

$$= \sqrt{280/65.9}$$

$$= 20.621 \text{ mm.}$$

Slenderness ratio [KL/r]:

$$= KL/r$$

$$= 1 * 248/20.6$$

$$= 12.051$$

Unit Force [Force,u]:

$$= F1 / (2 * \text{Baseplate Length})$$

$$= 37.9 / (2 * 900)$$

$$= 0.021 \text{ Kgf/mm.}$$

Moment at base of inner Rib [Mbase,c]:

$$= \text{Unit Force} * e * L$$

$$= 0.021 * 421 * 248$$

$$= 2.204 \text{ Kg-m.}$$

Bending Stress due to Transverse Force and Weight Load [SigmaB,base,c]:

$$= \text{Bending Moment} / \text{Section Modulus}$$

$$= 2.2/40042$$

$$= 0.540 \text{ N./mm}^2$$

Compressive Allowable, $KL/r < Cc$ (12.1 < 138) per AISC E2-1 [Sca]:

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

$$= (1 - (12.1)^2 / (2 * 138^2)) 207 / (5/3 + 3 * (12.1) / (8 * 138) - (12.1^3) / (8 * 138^3))$$

$$= 121.3 \text{ N./mm}^2$$

AISC Unity Check of Inside Ribs (must be <= 1)

$$= Sc/Sca + (Mbase,c * C1/I) / Sba$$

$$= 2.71/121 + (2.2 * 70/280) / 138$$

$$= 0.026$$

Input Data for Base Plate Bolting Calculations:

Total Number of Bolts per BasePlate	Nbolts	4	
Total Number of Bolts in Tension/Baseplate	Nbt	2	
Bolt Material Specification		SA-193 B7	
Bolt Allowable Stress	Stba	283.00	N./mm ²
Bolt Corrosion Allowance	Bca	0.0	mm.
Distance from Bolts to Edge	Edgedis	120.0	mm.
Nominal Bolt Diameter	Bnd	20.0000	mm.
Thread Series	Series	TEMA Metric	
BasePlate Allowable Stress	S	108.25	N./mm ²
Area Available in a Single Bolt	BltArea	2.1705	cm ²
Saddle Load QO (Weight)	QO	3815.2	Kgf

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Saddle Load QL (Wind/Seismic contribution)	QL	20.0	Kgf
Maximum Transverse Force	Ft	78.7	Kgf
Maximum Longitudinal Force	F1	37.9	Kgf
Saddle Bolted to Steel Foundation			No

Shear Stress in a Single Bolt, Longitudinal Direction [taub,l]:
 = F1 / (Bolt Area * Number of Bolts)
 = 37.9/(2.17 * 4)
 = 0.4 N./mm². Must be less than 146.2 N./mm².

Shear Stress in a Single Bolt, Transverse Direction [taub,t]:
 = Ft / (Bolt Area * Number of Bolts)
 = 78.7/(2.17 * 4)
 = 0.9 N./mm². Must be less than 146.2 N./mm².

Bolt Area Calculation per Dennis R. Moss

Bolt Area Requirement Due to Longitudinal Load [Bltarearl]:
 = 0.0 (Q0 > QL --> No Uplift in Longitudinal direction)

Bolt Area due to Shear Load [Bltarears]:
 = F1 / (BoltShearAllowable * Nbolts)
 = 37.9/(146 * 4)
 = 0.0064 cm²

Bolt Area due to Transverse Load:

Moment on Baseplate Due to Transverse Load [Rmom]:
 = B * Ft + Sum of X Moments
 = 750 * 78.7 + 0
 = 59.04 Kg-m.

Eccentricity (e):
 = Rmom / Q0
 = 59/3815
 = 15.48 mm. < Bplen/6 --> No Uplift in Transverse direction

Bolt Area due to Transverse Load [Bltareart]:
 = 0 (No Uplift)

Required Area of a Single Bolt [Bltarear]:
 = max[Bltarearl, Bltarears, Bltareart]
 = max[0, 0.0064, 0]
 = 0.0064 cm²

ASME VIII Division 2 Horizontal Vessel Analysis, Right Saddle:

Input and Calculated Values:

Vessel Mean Radius	Rm	470.00	mm.
Shell Thickness used in this Case	t	12.000	mm.
Stiffened Vessel Length per 4.15.6	L	3050.00	mm.
Distance from Saddle to Vessel tangent	a1 or a	600.00	mm.
Saddle Width	b1 or b	140.00	mm.
Saddle Bearing Angle	delta or theta	120.00	degrees
Wear Plate Width	b2 or b1	300.00	mm.
Wear Plate Bearing Angle	delta2 or theta1	132.00	degrees
Wear Plate Thickness	e2 or tr	12.0	mm.
Wear Plate Allowable Stress	fw or Sr	137.90	N./mm ²
Inside Depth of Head	Hi or h2	234.25	mm.

Saddle Calcs: Test Case: Step: 13 12:29pm May 21,2024

Shell Allowable Stress used in Calculation		235.81	N./mm ²
Head Allowable Stress used in Calculation		235.81	N./mm ²
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Distance from Saddle Base to Centerline	B	750.00	mm.
Coefficient of Friction	mu	0.30	
Saddle Force Q, Test Case, no Ext. Forces		1123.38	Kgf
Pressure used in Analysis	P	28.646	bars
Horizontal Vessel Analysis Results:			
	Actual	Allowable	
	N./mm ²	N./mm ²	

Long. Stress at Top of Midspan	74.60	235.81	
Long. Stress at Bottom of Midspan	75.01	235.81	
Long. Stress at Top of Saddles	77.09	235.81	
Long. Stress at Bottom of Saddles	73.54	235.81	

Tangential Shear in Shell	1.68	139.65	
Circ. Stress at Horn of Saddle	2.98	353.71	
Circ. Compressive Stress in Shell	0.17	235.81	

Intermediate Results: Saddle Reaction Q due to Wind or Seismic:

Saddle Reaction Force due to Wind Ft [Fwt]:
 = Ftr(Ft/Num of Saddles + Z Force Load) * B / E
 = 3(157/2 + 0) * 750/843
 = 210.2 Kgf

Saddle Reaction Force due to Wind Fl [Fwl]:
 = max(Fl, Sum of X Forces) * B / Ls
 = max(37.9, 0) * 750/1420
 = 20.0 Kgf

Load Combination Results for Q + Wind or Seismic [Q]:
 = Saddle Load + max(Fwl, Fwt, Fsl, Fst)
 = 913 + max(20, 210, 0, 0)
 = 1123.4 Kgf

Longitudinal Wind Force [Fl]:
 = WindScalar * WindPress(Platform Area + (End Area * WindDiaMult))
 = 1 * 1046(0 + (0.9 * 1.2))
 = 1126.093 N

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)		1229.01	Kgf
Transverse Shear Load Saddle	Ft	78.72	Kgf
Longitudinal Shear Load Saddle		37.89	Kgf

Formulas and Substitutions for Horizontal Vessel Analysis:

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

Saddle Dimension [E]:
 = min(2(ShellID/2 + t + WearPadThickness)sin(theta/2), 2*Rm)
 = min(2(925/2 + 12 + 12)sin(120/2), 2*470)
 = 842.643 mm.

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 = 0.0529	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434

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Saddle Calcs: Test Case: Step: 13 12:29pm May 21,2024

$$K7p = 0.0434$$

The suffix 'p' denotes the values for a wear plate if it exists.

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

Moment per Equation 4.15.3 [M1]:

$$\begin{aligned} &= -Q*a [1 - (1 - a/L + (Rm^2-h^2)/(2a*L)) / (1+(4h^2)/3L)] \\ &= -1123*600 [1 - (1 - 600/3050 + (470^2-234^2) / (2*600*3050)) / (1+(4*234)/(3*3050))] \\ &= -155.2 \text{ Kg-m.} \end{aligned}$$

Moment per Equation 4.15.4 [M2]:

$$\begin{aligned} &= Q*L/4 (1+2(Rm^2-h^2)/(L^2)) / (1+(4h^2)/(3L)) - 4a/L \\ &= 1123*3050/4 (1+2(470^2-234^2)/(3050^2)) / (1+(4*234)/(3*3050)) - 4*600/3050 \\ &= 130.7 \text{ Kg-m.} \end{aligned}$$

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

$$\begin{aligned} &= P * Rm/(2t) - M2/(pi*Rm^2*t) \\ &= 28.6 * 470/(2*9) - 131/(pi*470^2*9) \\ &= 74.60 \text{ N./mm}^2 \end{aligned}$$

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

$$\begin{aligned} &= P * Rm/(2t) + M2/(pi * Rm^2 * t) \\ &= 28.6 * 470/(2 * 9) + 131/(pi * 470^2 * 9) \\ &= 75.01 \text{ N./mm}^2 \end{aligned}$$

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

$$\begin{aligned} &= P * Rm/(2t) - M1/(K1*pi*Rm^2*t) \\ &= 28.6*470/(2*9) - 155/(0.11*pi*470^2*9) \\ &= 77.09 \text{ N./mm}^2 \end{aligned}$$

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

$$\begin{aligned} &= P * Rm/(2t) + M1/(K1* * pi * Rm^2 * t) \\ &= 28.6*470/(2*9) + 155/(0.19*pi*470^2*9) \\ &= 73.54 \text{ N./mm}^2 \end{aligned}$$

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

$$\begin{aligned} &= Q(L-2a)/(L+(4*h^2/3)) \\ &= 1123(3050 - 2 * 600)/(3050 + (4 * 234/3)) \\ &= 618.1 \text{ Kg} \end{aligned}$$

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

$$\begin{aligned} &= K2 * T / (Rm * t) \\ &= 1.17 * 618 / (470 * 9) \\ &= 1.68 \text{ N./mm}^2 \end{aligned}$$

Decay Length (4.15.22) [x1,x2]:

$$\begin{aligned} &= 0.78 * \text{sqrt}(Rm * t) \\ &= 0.78 * \text{sqrt}(470 * 9) \\ &= 50.730 \text{ mm.} \end{aligned}$$

Effective reinforcing plate width (4.15.1) [B1]:

$$\begin{aligned} &= \text{min}(b + 1.56 * \text{sqrt}(Rm * t), 2a) \\ &= \text{min}(140 + 1.56 * \text{sqrt}(470 * 9), 2 * 600) \\ &= 241.46 \text{ mm.} \end{aligned}$$

Wear Plate/Shell Stress ratio (4.15.29) [eta]:

$$= 1.0000 \text{ Materials are the same, test case}$$

Circumferential Stress at Saddle Base with Wear Plate (4.15.26) [sigma6,r]:

$$\begin{aligned} &= -K5 * Q * k / (B1(t + eta * tr)) \\ &= -0.76 * 1123 * 0.1 / (241(9 + 1 * 12)) \\ &= -0.17 \text{ N./mm}^2 \end{aligned}$$

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Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:

$$= -Q / (4(t + \eta * tr) b l) - 12 * K7 * Q * Rm / (L(t + \eta * tr)^2)$$

$$= -1123 / (4(9 + 1 * 12) 241) - 12 * 0.053 * 1123 * 470 / (3050(9 + 1 * 12)^2)$$

$$= -2.98 \text{ N./mm}^2$$

Results for Vessel Ribs, Web and Base:

Baseplate Length	Bplen	900.0000	mm.
Baseplate Thickness	Bpthk	15.0000	mm.
Baseplate Width	Bpwid	170.0000	mm.
Number of Ribs (inc. outside ribs)	Nribs	3	
Rib Thickness	Ribtk	12.0000	mm.
Web Thickness	Webtk	12.0000	mm.
Web Location	Webloc	Center	
Saddle Yield Stress	Sy	206.9	N./
Height of Web at Center	Hw,c	367.5	mm.
Friction Coefficient	mu	0.300	

Note: In the tables below lo is I for the rectangle + Area * Centroid Distance^2

Moment of Inertia of Saddle - Transverse Direction (90 degrees to long axis)

	B	D	Y	A	AY	Io
Shell	401.0	9.0	4.5	36.1	16239.4	0.295E+04
Wearplate	300.0	12.0	15.0	36.0	54000.0	0.231E+04
Web	12.0	248.5	145.2	29.8	433135.5	0.229E+04
BasePlate	170.0	15.0	277.0	25.5	706350.0	0.846E+04
Totals	127.4	1209724.9	0.160E+05

Distance to Centroid [C1]:

$$= AY / A$$

$$= 476 / 127$$

$$= 94.949 \text{ mm.}$$

Angle [beta]:

$$= 180 - \text{Saddle Angle} / 2$$

$$= 180 - 120 / 2$$

$$= 120.0$$

Saddle Splitting Coefficient [K1]:

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

$$= (1 + \cos(120) - 0.5 * \sin(120)^2) / (\pi - 2.09 + \sin(120) \cos(120))$$

$$= 0.2035$$

Saddle Splitting Force [Fh]:

$$= K1 * Q$$

$$= 0.2 * 1123$$

$$= 228.6325 \text{ Kgf}$$

Tension Stress, St = (Fh/As)	=	0.2455	N./mm ²
Allowed Stress, Sa = 0.6 * Yield Str	=	124.1100	N./mm ²

Saddle Splitting Dimension [d]:

$$= B - R * \sin(\theta/2) / (\theta/2 \text{ in radians})$$

$$= 750 - 466 * \sin(120/2) / 1.05$$

$$= 365.035 \text{ mm.}$$

Bending Moment, M = Fh * d	=	83.4604	Kg-m.
----------------------------	---	---------	-------

Bending Stress, Sb = (M * C1 / I)	=	0.4855	N./mm ²
Allowed Stress, Sa = 2/3 * Yield Str	=	137.9000	N./mm ²

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Minimum Thickness of Baseplate per Moss:

$$= (3 (Q + Saddle_Wt) BasePlateWidth / (4 * BasePlateLength * AllStress))^{1/2}$$

$$= (3 (1123 + 106) 170 / (4 * 900 * 138))^{1/2}$$

$$= 3.519 \text{ mm.}$$

Calculation of Axial Load, Intermediate Values and Compressive Stress:

Web Length Dimension [Web Length]:

$$= 2 * \cos(90 - Saddle \text{ Angle} / 2) (Inside \text{ Radius} + Shell \text{ Thk} + Wear \text{ Plate} \text{ Thk})$$

$$= 2 * \cos(90 - 120 / 2) (462 + 12 + 12)$$

$$= 842.643 \text{ mm.}$$

Distance between Ribs [e]:

$$= Web \text{ Length} / (Nribs - 1)$$

$$= 843 / (3 - 1)$$

$$= 421.321 \text{ mm.}$$

Baseplate Pressure Area [Ap]:

$$= e * Bpwid / 2$$

$$= 421 * 170 / 2$$

$$= 358.123 \text{ cm}^2$$

Bearing Pressure [Bp]:

$$= Q / (BasePlateLength * BasePlateWidth)$$

$$= 1123 / (900 * 170)$$

$$= 0.734 \text{ Kgf/cm}^2$$

Axial Load [P]:

$$= Ap * Bp$$

$$= 358 * 0.73$$

$$= 262.947 \text{ Kgf}$$

Area of the Rib and Web [Ar]:

$$= Rib \text{ Area} + Web \text{ Area}$$

$$= 15.4 + 25.3$$

$$= 40.639 \text{ cm}^2$$

Compressive Stress [Sc]:

$$= P / Ar$$

$$= 263 / 40.6$$

$$= 0.635 \text{ N./mm}^2$$

Check of Outside Ribs:

Inertia of Saddle, Outer Ribs - Longitudinal Direction

	B	D	Y	A	AY	Io
-----	-----	-----	-----	-----	-----	-----
Rib+Web	12.0	140.0	...	16.8	...	274.

Rib dimension [D]:

$$= Saddle \text{ Width} - Web \text{ Thickness}$$

$$= 140 - 12$$

$$= 128.000 \text{ mm.}$$

Distance to Centroid from Datum [ytot]:

$$= AY / A$$

$$= 0 / 40.6$$

$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= Saddle \text{ Width} / 2$$

$$= 140 / 2$$

$$= 70.000 \text{ mm.}$$

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Radius of Gyration [r]:

$$= \sqrt{\text{Total Inertia} / \text{Total Area}}$$

$$= \sqrt{274/40.6}$$

$$= 25.985 \text{ mm.}$$

Length of Outer Rib [L]:

$$= \text{Saddle Height} - \cos(\text{theta}/2) (\text{radius} + \text{shlthk} + \text{wpdthk}) - \text{bpthk}$$

$$= 750 - \cos(120/2) (462 + 12 + 12) - 15$$

$$= 491.750 \text{ mm.}$$

Intermediate Term [Cc]:

$$= \sqrt{2 * \pi^2 * \text{Elastic Modulus} / \text{Yield Stress}}$$

$$= \sqrt{2 * \pi^2 * 199943008/207}$$

$$= 138.135$$

Slenderness ratio [KL/r]:

$$= KL/r$$

$$= 1 * 492/26$$

$$= 18.925$$

Bending Moment [Rm]:

$$= Fl / (2 * Bplen) * e * L / 2$$

$$= 37.9 / (2 * 900) * 421 * 492/2$$

$$= 2.181 \text{ Kg-m.}$$

Compressive Allowable, $KL/r < Cc$ (18.9 < 138) per AISC E2-1 [Sca]:

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

$$= (1 - (18.9)^2 / (2 * 138^2)) 207 / (5/3 + 3 * (18.9) / (8 * 138) - (18.9^3) / (8 * 138^3))$$

$$= 119.3 \text{ N./mm}^2$$

AISC Unity Check of Outside Ribs (must be <= 1)

$$= Sc/Sca + (Rm * C1 / I) / Sba$$

$$= 0.63/119 + (2.18 * 70/2744000) / 138$$

$$= 0.009$$

Check of Inside Ribs:

Inertia of Saddle, Inner Ribs - Axial Direction

	B	D	Y	A	AY	Io
Rib	12.0	128.0	0.0	15.4	0.0	274.
Web	421.3	12.0	0.0	50.6	0.0	6.07
Totals	65.9	...	280.

Distance to Centroid from Datum [ytot]:

$$= AY / A$$

$$= 0/65.9$$

$$= 0.000 \text{ mm.}$$

Distance to Centroid [C1]:

$$= \text{Saddle Width} / 2$$

$$= 140/2$$

$$= 70.000 \text{ mm.}$$

Length of Inner Rib [L]:

$$= \text{Saddle Height} - \text{Outside Radius} - \text{Bpthk}$$

$$= 750 - \cos(486/2) (15 + 0 + 0) - 0$$

$$= 248.500 \text{ mm.}$$

Radius of Gyration [r]:

$$= \sqrt{\text{Total Inertia} / \text{Total Area}}$$

$$= \sqrt{280/65.9}$$

$$= 20.621 \text{ mm.}$$

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Slenderness ratio [KL/r]:

$$= KL/r$$

$$= 1 * 248/20.6$$

$$= 12.051$$

Unit Force [Force,u]:

$$= F1 / (2 * Baseplate Length)$$

$$= 37.9 / (2 * 900)$$

$$= 0.021 Kgf/mm.$$

Moment at base of inner Rib [Mbase,c]:

$$= Unit Force * e * L$$

$$= 0.021 * 421 * 248$$

$$= 2.204 Kg-m.$$

Bending Stress due to Transverse Force and Weight Load [SigmaB,base,c]:

$$= Bending Moment / Section Modulus$$

$$= 2.2/40042$$

$$= 0.540 N./mm²$$

Compressive Allowable, KL/r < Cc(12.1 < 138) per AISC E2-1 [Sca]:

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

$$= (1 - (12.1)^2 / (2 * 138^2)) 207 / (5/3 + 3 * (12.1) / (8 * 138) - (12.1^3) / (8 * 138^3))$$

$$= 121.3 N./mm²$$

AISC Unity Check of Inside Ribs (must be <= 1)

$$= Sc/Sca + (Mbase,c * C1/I) / Sba$$

$$= 0.78/121 + (2.2 * 70/280) / 138$$

$$= 0.010$$

Input Data for Base Plate Bolting Calculations:

Total Number of Bolts per BasePlate	Nbolts	4	
Total Number of Bolts in Tension/Baseplate	Nbt	2	
Bolt Material Specification		SA-193 B7	
Bolt Allowable Stress	Stba	283.00	N./mm ²
Bolt Corrosion Allowance	Bca	0.0	mm.
Distance from Bolts to Edge	Edgedis	120.0	mm.
Nominal Bolt Diameter	Bnd	20.0000	mm.
Thread Series	Series	TEMA Metric	
BasePlate Allowable Stress	S	108.25	N./mm ²
Area Available in a Single Bolt	BltArea	2.1705	cm ²
Saddle Load QO (Weight)	QO	1018.8	Kgf
Saddle Load QL (Wind/Seismic contribution)	QL	20.0	Kgf
Maximum Transverse Force	Ft	78.7	Kgf
Maximum Longitudinal Force	F1	37.9	Kgf
Saddle Bolted to Steel Foundation		No	

Shear Stress in a Single Bolt, Longitudinal Direction [taub,l]:

$$= F1 / (Bolt Area * Number of Bolts)$$

$$= 37.9 / (2.17 * 4)$$

$$= 0.4 N./mm². Must be less than 146.2 N./mm².$$

Shear Stress in a Single Bolt, Transverse Direction [taub,t]:

$$= Ft / (Bolt Area * Number of Bolts)$$

$$= 78.7 / (2.17 * 4)$$

$$= 0.9 N./mm². Must be less than 146.2 N./mm².$$

Bolt Area Calculation per Dennis R. Moss

Bolt Area Requirement Due to Longitudinal Load [Bltarearl]:

$$= 0.0 (QO > QL --> No Uplift in Longitudinal direction)$$

Bolt Area due to Shear Load [Bltarears]:

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$$= F1 / (BoltShearAllowable * Nbolts)$$

$$= 37.9 / (146 * 4)$$

$$= 0.0064 \text{ cm}^2$$

Bolt Area due to Transverse Load:

Moment on Baseplate Due to Transverse Load [Rmom]:

$$= B * Ft + \text{Sum of X Moments}$$

$$= 750 * 78.7 + 0$$

$$= 59.04 \text{ Kg-m.}$$

Eccentricity (e):

$$= Rmom / QO$$

$$= 59 / 1019$$

$$= 57.95 \text{ mm.} < Bplen / 6 \text{ --> No Uplift in Transverse direction}$$

Bolt Area due to Transverse Load [Bltareart]:

$$= 0 \text{ (No Uplift)}$$

Required Area of a Single Bolt [Bltarear]:

$$= \max[Bltarearl, Bltarears, Bltareart]$$

$$= \max[0, 0.0064, 0]$$

$$= 0.0064 \text{ cm}^2$$

Warning:

Please note that nozzle loadings, if included, are assumed to be local in nature and will not contribute to or create a net section bending moment. Therefore, the addition of nozzle loads will not affect the support load calculation. If you wish to create a load on the legs/lugs etc. from forces or moments, use the Force/Moment dialog and add the appropriate forces and moments at the correct locations in the model. Ensure that all supports are designed to take into account all possible loading conditions.

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Conical Reinforcement Calculations, ASME VIII Div. 1, App. 1

Input Values:

SH. Cone

Design Internal Pressure, Small End		22.048	bars
Design Internal Pressure, Large End		22.048	bars
Temperature for Internal Pressure		120.0	°C
Design External Pressure		1.034	bars
Temperature for External Pressure		120.0	°C
Cone Material		SA-516 70	
Cone Allowable Stress at Temperature	Ss	137.90	N./mm ²
Cone Allowable Stress At Ambient		137.90	N./mm ²
Longitudinal Joint Efficiency of Cone		1.00	
Circumferential Joint Efficiency of Cone		1.00	
Actual Thickness of Cone	Tc	12.0000	mm.
Corrosion Allowance for Cone	Cac	3.0000	mm.
Diameter Basis for Cone and Cylinders		ID	
Diameter of Small End of Cone	Ds	600.000	mm.
Diameter of Large End of Cone	Dl	925.000	mm.
Half Apex Angle for Cone		30.00	degrees
Axial Length of Cone	Lc	565.000	mm.
Small End Cylinder Material		SA-516 70	
Small Cylinder Allowable Stress at Operating	Ss	137.90	N./mm ²
Small Cylinder Allowable Stress At Ambient		137.90	N./mm ²
Joint Efficiency of Small Cylinder	Es	1.0000	
Actual Thickness of Small Cylinder	Ts	12.0000	mm.
Corrosion Allowance for Small Cylinder	Cas	3.0000	mm.
Axial Length of Small Cylinder	Ls	150.000	mm.
Large End Cylinder Material		SA-516 70	
Large Cylinder Allowable Stress at Operating	Sl	137.90	N./mm ²
Large Cylinder Allowable Stress At Ambient		137.90	N./mm ²
Joint Efficiency of Large Cylinder	El	1.0000	
Actual Thickness of Large Cylinder	Tl	12.0000	mm.
Corrosion Allowance for Large Cylinder	Cal	3.0000	mm.
Axial Length of Large Cylinder	Ll	2892.083	mm.
Type of Reinforcement at Large End of Cone:		None	
Large End Reinforcing/Knuckle Material			
Large Reinforcing/Knuckle Allowable, Operating		137.90	N./mm ²
Large Reinforcing/Knuckle Allowable, Ambient		137.90	N./mm ²
Type of Reinforcement at Small End of Cone:		None	
Small End Reinforcing/Knuckle Material			
Small Reinforcing/Knuckle Allowable, Operating		137.90	N./mm ²
Small Reinforcing/Knuckle Allowable, Ambient		137.90	N./mm ²

Calculated Values:

Elastic Modulus Data from ASME Section II Part D at 120 °C

Elastic Modulus of Cone Material	0.197E+09	KPa. at 119 °C
Elastic Modulus of Small Cylinder Material	0.197E+09	KPa. at 119 °C
Elastic Modulus of Large Cylinder Material	0.197E+09	KPa. at 119 °C
Elastic Modulus of Large End Reinforcement	0.197E+09	KPa. at 119 °C
Elastic Modulus of Small End Reinforcement	0.197E+09	KPa. at 119 °C

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Elastic Modulus Data from ASME Section II Part D at 120 °C

Elastic Modulus of Cone Material	0.197E+09 KPa. at 119 °C
Elastic Modulus of Small Cylinder Material	0.197E+09 KPa. at 119 °C
Elastic Modulus of Large Cylinder Material	0.197E+09 KPa. at 119 °C
Elastic Modulus of Large End Reinforcement	0.197E+09 KPa. at 119 °C
Elastic Modulus of Small End Reinforcement	0.197E+09 KPa. at 119 °C

Axial forces and moments are not computed for Horizontal geometries.
 Compute them manually and enter the loads using the Force/Moment dialog.
 Place the loads at the ends of the cone as needed.

Note: Small end of the Cone is taken as a Line of Support

Maximum Centroid Reinforcement Distance Large End	16.3373 mm.
Maximum Centroid Reinforcement Distance Small End	13.2476 mm.

Note: No ring was found close enough to the large end to be considered.

Note: No ring was found close enough to the small end to be considered.

Reinforcement Calculations for Cone / Large Cylinder:

Required Area of Reinforcement for Large End Under Internal Pressure

Large end ratio of pressure to allowable stress	P/Ss E1	0.01599	
Large end max. half apex angle w/o reinforcement	delta	30.000	degrees
Large end actual half apex angle	alpha	30.000	degrees

Intermediate Value [k]:

= max(Y / (Sr * Erl), 1)
 = max(27.15E+09/(138 * 196910448), 1)
 = 1.0000

where [Y] is:

= Large End All. Stress * Large End Elastic Modulus (Int. temp.)
 = 138 * 196910448
 = 27153948672.0 N./mm²

Decay Length, Cone Large End:

= 2 * sqrt(RL(ts - ca))
 = 2 * sqrt(466(12 - 3))
 = 129.453 mm.

Required Area of Reinforcement, Large End, Internal [Arl]:

= k*QL*RL/(Ss*E1) * (1-delta/alpha) * tan(alpha)
 = 1 * 52.3 * 466/(138 * 1) * (1 - 30/30) * 0.58
 = 0 cm²

Force per Length, Cone Large End [QL]:

= P(RL/2) + f1
 = 22(466/2) + 0
 = 52.332 Kgf/mm.

Axial Load per Unit Circumference excluding Pressure [f1]:

= Faxial/(pi(Ds + Ts)) + Moment/(pi(Rs + Ts/2)(Rs + Ts/2))
 = -0/(pi(931+12))+0/(pi(466+12/2)(466+12/2))
 = 0.0 Kgf/mm.

Area of Reinforcement Available in Large end Shell [Ael]:

= (Ts-t) * sqrt(RL*Ts) + (Tc-Tr) * sqrt(RL*Tc/cos(alpha))
 = (9 - 7.52) * SQRT(466 * 9) +
 (9 - 8.68) * SQRT(466 * 9/0.87)
 = 1.1850 cm²

Summary of Reinforcement area, Large end, Internal Pressure:

Area of reinforcement required per App. 1-5(1) 0.0000 cm²
 Area of reinforcement in shell per App. 1-5(2) 1.1850 cm²
 Area of reinforcement in stiffening ring 0.0000 cm²

Required Area of Reinforcement for Large End Under External Pressure

Large end ratio of pressure to allowable stress P/Ss E1 0.00075
 Large end max. half apex angle w/o reinforcement delta 1.875 degrees
 Large end actual half apex angle alpha 30.000 degrees

Intermediate Value [k]:

= max(Y / (Srl * Erl), 1)
 = max(27.15E+09/(138 * 196910448), 1)
 = 1.0000

where [Y] is:

= Large End All. Stress * Large End Elastic Modulus (Ext. temp.)
 = 138 * 196910448
 = 27153948672.0 N./mm²

Allowable Stress of Large End Material (Ext. Temp) 137.9 N./mm²
 Allowable Stress of Cone Material (Ext. Temp) 137.9 N./mm²

Required Area of Reinforcement, Large End, External [Arl]:

= (k*QL*Rl*tan(angle)/(Ss*E1))*(1-1/4((P*Rl-QL)/QL))*(delta/alpha)
 = (1*2.5*474*0.58/(138*1))*
 (1-1/4((1.03*474-2.5)/2.5))*(1.88/30)
 = 0.48 cm²

Force per Length, Cone Large End External Pressure [QL]:

= Pext(Rl/2) + f1
 = 1.03(474/2) + 0
 = 2.502 Kgf/mm.

Axial Load per Unit Circumference excluding Pressure [f1]:

= Faxial/(pi(Ds + Ts)) + Moment/(pi(Rs + Ts/2)(Rs + Ts/2))
 = +0/(pi(949-12))+0/(pi(474-12/2)(474-12/2))
 = 0.0 Kgf/mm.

Available Area of Reinforcement, Large End, External [Ael]:

= 0.55*(D1*ts)^½ * (ts + tc/cos(alpha))
 = 0.55 * (949 * 9)^½ * (9 + 9/0.87)
 = 9.8570 cm²

Summary of Reinforcement Area, Large End, External Pressure:

Area of reinforcement required per App. 1-8(1) 0.4798 cm²
 Area of reinforcement in shell per App. 1-8(2) 9.8570 cm²
 Area of reinforcement in stiffening ring 0.0000 cm²

Reinforcement Calculations for Cone / Small Cylinder:

Required Area of Reinforcement for Small End under Internal Pressure

Small end ratio of pressure to allowable stress P/Ss E1 0.01599
 Small end max. half apex angle w/o reinforcement delta 11.096 degrees
 Small end actual half apex angle alpha 30.000 degrees

Intermediate Value [k]:

= max(Y / (Sr * Ers), 1)
 = max(27.15E+09/(138 * 196910448), 1)
 = 1.0000

where [Y] is:

= Small End All. Stress * Small End Elastic Modulus (Int. temp.)
 = 138 * 196910448
 = 27153948672.0 N./mm²

Decay Length, Cone Small End:
 $= 1.4 * \text{sqrt}(Rs(ts - ca))$
 $= 1.4 * \text{sqrt}(303(12 - 3))$
 $= 73.109 \text{ mm.}$

Required Area of Reinforcement, Small End, Internal [Ars]:
 $= k*QS*Rs/(Ss*E1)*(1-\text{delta}/\alpha)*\tan(\alpha)$
 $= 1 * 34.1 * 303/(138 * 1) * (1 - 11.1/30) 0.58$
 $= 2.67 \text{ cm}^2$

Force per Length, Cone Small End [QS]:
 $= P(Rs/2) + f2$
 $= 22(303/2) + 0$
 $= 34.063 \text{ Kgf/mm.}$

Axial Load per Unit Circumference excluding Pressure [f2]:
 $= \text{Faxial}/(\text{pi}(Ds + Ts)) + \text{Moment}/(\text{pi}(Rs + Ts/2)(Rs + Ts/2))$
 $= -0/(\text{pi}(606+12))+0/(\text{pi}(303+12/2)(303+12/2))$
 $= 0.0 \text{ Kgf/mm.}$

Area of Reinforcement Available in Small End Shell [Aes]:
 $= 0.78(Rs*Ts)^{1/2} * ((Ts-t)+(Tc-Tr)/\cos(\alpha)))$
 $= 0.78(303 * 9)^{1/2} * ((9 -4.89)+(9 -5.65)/0.87)$
 $= 3.2496 \text{ cm}^2$

Summary of Reinforcement Area, Small End, Internal Pressure:

Area of reinforcement required per App. 1-5(3)	2.6703	cm ²
Area of reinforcement in shell per App. 1-5(4)	3.2496	cm ²
Area of reinforcement in stiffening ring	0.0000	cm ²

Required Area of Reinforcement for Small End Under External Pressure

Allowable Stress of Small End Material (Ext. Temp)	137.9	N./mm ²
Allowable Stress of Cone Material (Ext. Temp)	137.9	N./mm ²

Intermediate Value [k]:
 $= \max(Y / (Srs * Ers), 1)$
 $= \max(27.15\text{E}+09/(138 * 196910448), 1)$
 $= 1.0000$

where [Y] is:
 $= \text{Small End All. Stress} * \text{Small End Elastic Modulus (Ext. temp.)}$
 $= 138 * 196910448$
 $= 27153948672.0 \text{ N./mm}^2$

Area of Reinforcement Required in Small End Shell [Ars]:
 $= k * QS * Rs * \tan(\alpha) / (Ss * E1)$
 $= (1*1.65*312*0.58/(138*1))$
 $= 0.211 \text{ cm}^2$

Force per Length, Cone Small End [QS]:
 $= \text{Pext}(Rs/2) + f2$
 $= 1.03(312/2) + 0$
 $= 1.645 \text{ Kgf/mm.}$

Axial Load per Unit Circumference excluding Pressure [f2]:
 $= \text{Faxial}/(\text{pi}(Ds + Ts)) + \text{Moment}/(\text{pi}(Rs + Ts/2)(Rs + Ts/2))$
 $= +0/(\text{pi}(624-12))+0/(\text{pi}(312-12/2)(312-12/2))$
 $= 0.0 \text{ Kgf/mm.}$

Area of Reinforcement Available in Small End Shell [Aes]:
 $= 0.55*(Ds*ts)^{1/2} * [(ts-t)+(tc-tr)/\cos(\alpha)])$
 $= 0.55*(624*9)^{1/2} * [(9-1.1)+(9-2.05)/0.87]$
 $= 6.5630 \text{ cm}^2$

Summary of Reinforcement Area, Small End, External Pressure:

Area of reinforcement required per App. 1-8(3)	0.2108	cm ²
Area of reinforcement in shell per App. 1-8(4)	6.5630	cm ²
Area of reinforcement in stiffening ring	0.0000	cm ²

Intermediate Results, Small End, External Pressure:

Area Available in Cone, Shell, and Reinforcement	33.21	cm ²
Force per Unit Length on Shell / Cone Junction	2.99	Kgf/mm.
Actual Buckling Stress associated with this Force	4.13	N./mm ²
Material Strain associated with this stress	0.000042	

Required Moment of Inertia, Small End, External Pressure [I's]:

= A * Ds² * Ats / 10.9
 = 0.41944E-04 * 624 * 624 * 33.2/10.9
 = 4975.36 mm.⁴

Available Moment of Inertia, Small End, External Pressure:

	Area	Centroid	Ar*Ce	Dist	I	Ar*Di ²
Shl	3.710	0.0000	0.000	6.3763	0.250	150.819
Con	4.283	11.8983	50.965	-5.5220	2.407	130.613
Sec	0.000	4.5000	0.000	1.8763	0.000	0.000
TOT	7.993		50.965		2.657	281.431

Centroid of Section			6.3763	Moment of Inertia		5.47

Summary of Small End Inertia Calculations

Available Moment of Inertia (Small End)	0.5472E+01	cm**4
Required Moment of Inertia (Small End)	0.4975E+00	cm**4

Note: The following calculations are only required per 1-5(g)(1) and do include external loads due to wind or seismic. These discontinuity stresses are computed at the shell/cone junction and do not include effects of local stiffening from a junction ring.

Results for Discontinuity Stresses per Bednar p. 236 2nd Edition

Stress Type	Stress	Allowable	Location
Tensile Stress	180.72	413.70	Small Cyl. Long.
Compres. Stress	-105.38	-413.70	Small Cyl. Long.
Membrane Stress	154.08	-206.85	Small End Tang.
Tensile Stress	186.54	413.70	Cone Longitudinal
Compres. Stress	-99.55	-413.70	Cone Longitudinal
Compres Stress	165.73	-206.85	Cone Tangential
Tensile Stress	327.88	413.70	Large Cyl. Long.
Compres. Stress	-212.74	-413.70	Large Cyl. Long.
Membrane Stress	-33.64	-206.85	Large End Tang.
Tensile Stress	336.79	413.70	Cone Longitudinal
Compres. Stress	-203.83	-413.70	Cone Longitudinal
Compres Stress	-15.83	-206.85	Cone Tangential

Note: An asterisk (*) denotes that this stress was not applicable for this combination of loads.

Maximum Allowable Pressure Calculations for Cone to Shell Junction:

Pressure Case	Pressure bars	Reason for Failure at this Pressure
MAWP	22.861	Thickness due to internal pressure, Cone Large End
MAPnc	30.576	Thickness due to internal pressure, Cone Large End

These pressures were determined by iteration.

Input, Nozzle Desc: T1 (3in.) From: 20

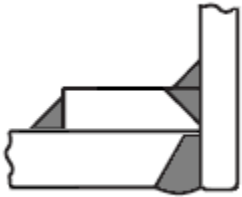
Pressure for Reinforcement Calculations	P	6.801	bars
Temperature for Internal Pressure	Temp	85	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	85	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	600.00	mm.
Design Length of Section	L	400.0000	mm.
Shell Finished (Minimum) Thickness	t	10.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Cylinder/Cone Centerline	L1	100.0000	mm.
Distance from Bottom/Left Tangent		200.00	mm.
User Entered Minimum Design Metal Temperature		-29.00	°C

Type of Element Connected to the Parent : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Outside	
Layout Angle		108.82	deg
Diameter		3.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		80	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	200.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	10.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Pad Material [Normalized]		SA-516 70	
Pad Allowable Stress at Temperature	Sp	137.90	N./mm ²
Pad Allowable Stress At Ambient	Spa	137.90	N./mm ²
Diameter of Pad along vessel surface	Dp	190.0000	mm.
Thickness of Pad	te	10.0000	mm.
Weld leg size between Pad and Shell	Wp	8.0000	mm.
Groove weld depth between Pad and Nozzle	Wgpn	10.0000	mm.
Reinforcing Pad Width		50.5500	mm.
Flange Class		150	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle With Pad, no Inside projection

Note : Checking Nozzle 90 degrees to the Longitudinal axis.

Reinforcement CALCULATION, Description: T1 (3in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation	3.500 in.
Actual Thickness Used in Calculation	0.263 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (6.8 \cdot 303) / (138 \cdot 1 - 0.6 \cdot 6.8)$$

$$= 1.4990 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (6.8 \cdot 44.5) / (118 \cdot 1 + 0.4 \cdot 6.8)$$

$$= 0.2558 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3989 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	173.5622 mm.
Parallel to Vessel Wall, opening length	d	86.7811 mm.
Normal to Vessel Wall (Thickness Limit), pad side Tlwp		17.5000 mm.

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

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: T1 (3in.).
 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 small nozzle area calculation or force the Appendix 1-10 computation in Nozzle Design Options.*

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld, min(Curve:B, Curve:A)

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = tr * (E*) / (tg - c) = 0.07, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve A	-8 °C
Min Metal Temp. at Required thickness (UCS 66.1)	-104 °C

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: T1 (3in.) Nozl: 19 12:29pm May 21,2024

Min Metal Temp. w/o impact per UG-20(f) -29 °C

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

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Nozzle Neck to Pad Weld for Reinforcement pad, Curve: D

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Shell to Pad Weld Junction at Pad OD, Curve: D

Govrn. thk, tg = 10, tr = 1.5, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.21$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), min(Curve:B, Curve:D)

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the Nozzle : -104 °C
 Gov. MDMT of the Reinforcement Pad : -104 °C
 Gov. MDMT of the nozzle to shell joint welded assembly : -104 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ASME B16.5/47 flanges per UCS-66(c) -18 °C
 Flange MDMT with Temp reduction per UCS-66(b) (1) (-b) -104 °C

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

Design Pressure/Ambient Rating = 6.80/19.60 = 0.347

Weld Size Calculations, Description: T1 (3in.)

Intermediate Calc. for nozzle/shell Welds Tmin 3.6675 mm.
 Intermediate Calc. for pad/shell Welds TminPad 7.0000 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	2.5673 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.
Pad Weld	3.5000 = 0.5*TminPad	5.6560 = 0.7 * Wp mm.

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 16.6 bars

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

Reinforcement CALCULATION, Description: T1 (3in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 3.500 in.
 Actual Thickness Used in Calculation 0.263 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (6.8 \cdot 303) / (138 \cdot 1 - 0.6 \cdot 6.8)$$

$$= 1.4990 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (6.8 \cdot 44.5) / (118 \cdot 1 + 0.4 \cdot 6.8)$$

$$= 0.2558 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3989 mm.

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

Parallel to Vessel Wall (Diameter Limit) D1 163.1300 mm.
 Parallel to Vessel Wall, opening length d 81.5650 mm.
 Normal to Vessel Wall (Thickness Limit), pad side Tlwp 17.5000 mm.

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

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: T1 (3in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures ta = 3.3989 mm.
 Wall Thickness per UG16(b), tr16b = 4.5000 mm.
 Wall Thickness, shell/head, internal pressure trb1 = 4.4990 mm.
 Wall Thickness tb1 = max(trb1, tr16b) = 4.5000 mm.
 Wall Thickness tb2 = max(trb2, tr16b) = 4.5000 mm.
 Wall Thickness per table UG-45 tb3 = 7.8000 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[7.8, \max(4.5, 4.5)]$$

$$= 4.5000 \text{ mm.}$$

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

$$= \max(ta, tb)$$

$$= \max(3.4, 4.5)$$

$$= 4.5000 \text{ mm.}$$

Available Nozzle Neck Thickness = 6.6675 mm. --> OK

Weld Size Calculations, Description: T1 (3in.)

Intermediate Calc. for nozzle/shell Welds Tmin 3.6675 mm.
 Intermediate Calc. for pad/shell Welds TminPad 7.0000 mm.

Results Per UW-16.1:

Required Thickness Actual Thickness

PV Elite 23 Licensee: #Replace this text with your company name and th
FileName : Chiller-Rev.01 -----

Nozzle Calcs.: T1 (3in.) Nozl: 19 12:29pm May 21,2024

Nozzle Weld 2.5673 = 0.7 * tmin. 7.0700 = 0.7 * Wo mm.
Pad Weld 3.5000 = 0.5*TminPad 5.6560 = 0.7 * Wp mm.

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 16.6 bars

The Drop for this Nozzle is : 19.9089 mm.
The Cut Length for this Nozzle is, Drop + Ho + H + T : 230.4942 mm.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: T2 (3in.)

Nozl: 20 12:29pm May 21,2024

Input, Nozzle Desc: T2 (3in.)

From: 20

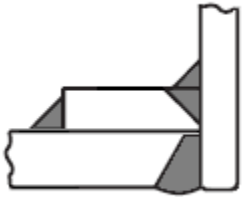
Pressure for Reinforcement Calculations	P	6.853	bars
Temperature for Internal Pressure	Temp	85	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	85	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	600.00	mm.
Design Length of Section	L	400.0000	mm.
Shell Finished (Minimum) Thickness	t	10.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Cylinder/Cone Centerline	L1	100.0000	mm.
Distance from Bottom/Left Tangent		200.00	mm.
User Entered Minimum Design Metal Temperature		-29.00	°C

Type of Element Connected to the Parent : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Outside	
Layout Angle		251.18	deg
Diameter		3.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		80	
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	200.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	10.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Pad Material [Normalized]		SA-516 70	
Pad Allowable Stress at Temperature	Sp	137.90	N./mm ²
Pad Allowable Stress At Ambient	Spa	137.90	N./mm ²
Diameter of Pad along vessel surface	Dp	190.0000	mm.
Thickness of Pad	te	10.0000	mm.
Weld leg size between Pad and Shell	Wp	8.0000	mm.
Groove weld depth between Pad and Nozzle	Wgpn	10.0000	mm.
Reinforcing Pad Width		50.5500	mm.
Flange Class		150	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle With Pad, no Inside projection

Note : Checking Nozzle 90 degrees to the Longitudinal axis.

Reinforcement CALCULATION, Description: T2 (3in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation	3.500 in.
Actual Thickness Used in Calculation	0.263 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (6.85 \cdot 303) / (138 \cdot 1 - 0.6 \cdot 6.85)$$

$$= 1.5103 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (6.85 \cdot 44.5) / (118 \cdot 1 + 0.4 \cdot 6.85)$$

$$= 0.2578 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3989 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	173.5622 mm.
Parallel to Vessel Wall, opening length	d	86.7811 mm.
Normal to Vessel Wall (Thickness Limit), pad side Tlwp		17.5000 mm.

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

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: T2 (3in.).
 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 small nozzle area calculation or force the Appendix 1-10 computation in Nozzle Design Options.*

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld, min(Curve:B, Curve:A)

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = tr * (E*) / (tg - c) = 0.07, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve A	-8 °C
Min Metal Temp. at Required thickness (UCS 66.1)	-104 °C

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: T2 (3in.) Nozl: 20 12:29pm May 21,2024

Min Metal Temp. w/o impact per UG-20(f) -29 °C

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

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Nozzle Neck to Pad Weld for Reinforcement pad, Curve: D

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Shell to Pad Weld Junction at Pad OD, Curve: D

Govrn. thk, tg = 10, tr = 1.51, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.22$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), min(Curve:B, Curve:D)

Govrn. thk, tg = 6.67, tr = 0.26, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.07$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the Nozzle : -104 °C
 Gov. MDMT of the Reinforcement Pad : -104 °C
 Gov. MDMT of the nozzle to shell joint welded assembly : -104 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

Unadjusted MDMT of ASME B16.5/47 flanges per UCS-66(c) -18 °C
 Flange MDMT with Temp reduction per UCS-66(b) (1) (-b) -104 °C

Where the Stress Reduction Ratio per UCS-66(b)(1)(-b) is :
 Design Pressure/Ambient Rating = 6.85/19.60 = 0.350

Weld Size Calculations, Description: T2 (3in.)

Intermediate Calc. for nozzle/shell Welds Tmin 3.6675 mm.
 Intermediate Calc. for pad/shell Welds TminPad 7.0000 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	2.5673 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.
Pad Weld	3.5000 = 0.5 * TminPad	5.6560 = 0.7 * Wp mm.

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 16.6 bars

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

Reinforcement CALCULATION, Description: T2 (3in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 3.500 in.
 Actual Thickness Used in Calculation 0.263 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (6.85 \cdot 303) / (138 \cdot 1 - 0.6 \cdot 6.85)$$

$$= 1.5103 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (6.85 \cdot 44.5) / (118 \cdot 1 + 0.4 \cdot 6.85)$$

$$= 0.2578 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3989 mm.

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

Parallel to Vessel Wall (Diameter Limit) D1 163.1300 mm.
 Parallel to Vessel Wall, opening length d 81.5650 mm.
 Normal to Vessel Wall (Thickness Limit), pad side Tlwp 17.5000 mm.

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

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: T2 (3in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures ta = 3.3989 mm.
 Wall Thickness per UG16(b), tr16b = 4.5000 mm.
 Wall Thickness, shell/head, internal pressure trb1 = 4.5103 mm.
 Wall Thickness tb1 = max(trb1, tr16b) = 4.5103 mm.
 Wall Thickness tb2 = max(trb2, tr16b) = 4.5000 mm.
 Wall Thickness per table UG-45 tb3 = 7.8000 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[7.8, \max(4.51, 4.5)]$$

$$= 4.5103 \text{ mm.}$$

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

$$= \max(ta, tb)$$

$$= \max(3.4, 4.51)$$

$$= 4.5103 \text{ mm.}$$

Available Nozzle Neck Thickness = 6.6675 mm. --> OK

Weld Size Calculations, Description: T2 (3in.)

Intermediate Calc. for nozzle/shell Welds Tmin 3.6675 mm.
 Intermediate Calc. for pad/shell Welds TminPad 7.0000 mm.

Results Per UW-16.1:

Required Thickness Actual Thickness

Nozzle Calcs.: T2 (3in.) Nozl: 20 12:29pm May 21,2024

Nozzle Weld 2.5673 = 0.7 * tmin. 7.0700 = 0.7 * Wo mm.
Pad Weld 3.5000 = 0.5*TminPad 5.6560 = 0.7 * Wp mm.

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 16.6 bars

The Drop for this Nozzle is : 19.9089 mm.
The Cut Length for this Nozzle is, Drop + Ho + H + T : 230.4942 mm.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: S1 (4in.)

Nozl: 21 12:29pm May 21,2024

Input, Nozzle Desc: S1 (4in.)

From: 60

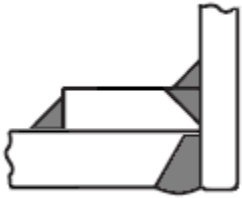
Pressure for Reinforcement Calculations	P	22.031	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cone at Nozzle Location	D	685.40	mm.
Equivalent Length of Conical Section	Le	2892.0833	mm.
Cone Half Apex Angle	Alpha	30.00	Degrees
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		898.52	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-333 6	
Material UNS Number		K03006	
Material Specification/Type	Smls. & wld. pipe		
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Outside	
Layout Angle		270.00	deg
Diameter		4.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		120	
Flange Material		SA-350 LF2	
Flange Type	Weld Neck Flange		
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	180.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Pad Material [Normalized]		SA-516 70	
Pad Allowable Stress at Temperature	Sp	137.90	N./mm ²
Pad Allowable Stress At Ambient	Spa	137.90	N./mm ²
Diameter of Pad along vessel surface	Dp	220.0000	mm.
Thickness of Pad	te	12.0000	mm.
Weld leg size between Pad and Shell	Wp	10.0000	mm.
Groove weld depth between Pad and Nozzle	Wgpn	12.0000	mm.
Reinforcing Pad Width		52.8500	mm.
Flange Class		300	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle With Pad, no Inside projection

Reinforcement CALCULATION, Description: S1 (4in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 4.500 in.
 Actual Thickness Used in Calculation 0.383 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Conical Transition, Tr [Int. Press]
 = $(P \cdot D) / (2 \cdot \cos(a) \cdot (S_v \cdot E - 0.6 \cdot P))$ Appendix 1-4 (e)
 = $(22 \cdot (692 + 2 \cdot 0)) / (2 \cdot 0.87 \cdot (138 \cdot 1 - 0.6 \cdot 22))$
 = 6.4481 mm.

Reqd Cone thickness at Nozzle Location under External Pres. : 8.6123 mm.

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]
 = $(P \cdot R_o) / (S_n \cdot E + 0.4 \cdot P)$ per Appendix 1-1 (a) (1)
 = $(22 \cdot 57.1) / (118 \cdot 1 + 0.4 \cdot 22)$
 = 1.0601 mm.

Required Nozzle thickness under External Pressure per UG-28 : 0.4431 mm.

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

Parallel to Vessel Wall (Diameter Limit) D1 201.6618 mm.
 Parallel to Vessel Wall, opening length d 100.8309 mm.
 Normal to Vessel Wall (Thickness Limit), pad side Tlwp 22.5000 mm.

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

Weld Strength Reduction Factor [fr1]:
 = $\min(1, S_n / S_v)$
 = $\min(1, 118 / 138)$
 = 0.855

Weld Strength Reduction Factor [fr2]:
 = $\min(1, S_n / S_v)$
 = $\min(1, 118 / 138)$
 = 0.855

Weld Strength Reduction Factor [fr4]:
 = $\min(1, S_p / S_v)$
 = $\min(1, 138 / 138)$
 = 1.000

Weld Strength Reduction Factor [fr3]:
 = $\min(fr2, fr4)$
 = $\min(0.85, 1)$
 = 0.855

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: S1 (4in.) Nozl: 21 12:29pm May 21,2024

Results of Nozzle Reinforcement Area Calculations: (cm²)

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required Ar	6.628	2.884	NA
Area in Shell A1	2.523	3.350	NA
Area in Nozzle Wall A2	2.183	2.421	NA
Area in Inward Nozzle A3	0.000	0.000	NA
Area in Welds A41+A42+A43	0.855	0.855	NA
Area in Element A5	10.483	10.483	NA
TOTAL AREA AVAILABLE Atot	16.045	17.109	NA

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 90.00 Degs.

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:	123.1852	12.0000 mm.
Based on given Pad Diameter:	220.0000	1.2205 mm.
Based on Shell or Nozzle Thickness:	125.2530	9.7345 mm.

Area Required [A]:

$$= (d * tr * F + 2 * tn * tr * F * (1 - fr1)) \text{ UG-37(c)}$$

$$= (101 * 6.45 * 1 + 2 * 6.73 * 6.45 * 1 * (1 - 0.85))$$

$$= 6.628 \text{ cm}^2$$

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

$$= d (E1 * t - F * tr) - 2 * tn (E1 * t - F * tr) * (1 - fr1)$$

$$= 101 (1 * 9 - 1 * 6.45) - 2 * 6.73$$

$$(1 * 9 - 1 * 6.45) * (1 - 0.85)$$

$$= 2.523 \text{ cm}^2$$

Area Available in Nozzle Wall Projecting Outward [A2]:

$$= (2 * Tlwp) * (tn - trn) * fr2$$

$$= (2 * 22.5) * (6.73 - 1.06) * 0.85$$

$$= 2.183 \text{ cm}^2$$

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

$$= Wo^2 * fr3 + (Wi - can / 0.707)^2 * fr2 + Wp^2 * fr4$$

$$= 10^2 * 0.85 + (0)^2 * 0.85 + 0^2 * 1$$

$$= 0.855 \text{ cm}^2$$

Area Available in Element [A5]:

$$= (\min(Dp, DL) - (\text{Nozzle OD})) * (\min(tp, Tlwp, te)) * fr4$$

$$= (202 - 114) * 12 * 1$$

$$= 10.483 \text{ cm}^2$$

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

Wall Thickness for Internal/External pressures	ta = 4.0601 mm.
Wall Thickness per UG16(b),	tr16b = 4.5000 mm.
Wall Thickness, shell/head, internal pressure	trb1 = 9.4481 mm.
Wall Thickness	tb1 = max(trb1, tr16b) = 9.4481 mm.
Wall Thickness, shell/head, external pressure	trb2 = 3.2999 mm.
Wall Thickness	tb2 = max(trb2, tr16b) = 4.5000 mm.
Wall Thickness per table UG-45	tb3 = 8.2578 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[8.26, \max(9.45, 4.5)]$$

$$= 8.2578 \text{ mm.}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:
 = max(ta, tb)
 = max(4.06, 8.26)
 = 8.2578 mm.

Available Nozzle Neck Thickness = 9.7345 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld (Impact tested) :

Note:
This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle Neck to Pad Weld for the Nozzle (Impact tested) :

Note:
This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle Neck to Pad Weld for Reinforcement pad, Curve: D

Govrn. thk, tg = 9.73, tr = 1.06, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.16$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Shell to Pad Weld Junction at Pad OD, Curve: D

Govrn. thk, tg = 12, tr = 6.45, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.72$, Temp. Reduction = 16 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 9.73, tr = 1.06, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.16$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the Nozzle : -104 °C
 Gov. MDMT of the Reinforcement Pad : -48 °C
 Gov. MDMT of the nozzle to shell joint welded assembly : -48 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

MDMT of ASME B16.5/47 flange per Matl. Specification -46 °C
 Flange MDMT with Temp reduction per UCS-66(i) (2) -89 °C

Where the Stress Reduction Ratio per UCS-66(i)(2) is :
 Design Pressure/Ambient Rating = 22.03/51.10 = 0.431

Weld Size Calculations, Description: S1 (4in.)

Nozzle Calcs.: S1 (4in.) Nozl: 21 12:29pm May 21,2024

Intermediate Calc. for nozzle/shell Welds Tmin 6.7345 mm.
 Intermediate Calc. for pad/shell Welds TminPad 9.0000 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	4.7142 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.
Pad Weld	4.5000 = 0.5*TminPad	7.0700 = 0.7 * Wp mm.

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

Weld Load [W]:

$$= \max(0, (A-A1+2*tn*fr1*(E1*t-tr))Sv)$$

$$= \max(0, (6.63 - 2.52 + 2 * 6.73 * 0.85 * (1 * 9 - 6.45)) / 138)$$

$$= 6184.64 \text{ Kgf}$$

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

Weld Load [W1]:

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

$$= (2.18 + 10.5 + 0.85 - 0 * 0.85) * 138$$

$$= 19013.51 \text{ Kgf}$$

Weld Load [W2]:

$$= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv$$

$$= (2.18 + 0 + 0.85 + (1.04)) * 138$$

$$= 5729.49 \text{ Kgf}$$

Weld Load [W3]:

$$= (A2+A3+A4+A5+(2*tn*t*fr1))*S$$

$$= (2.18 + 0 + 0.85 + 10.5 + (1.04)) * 138$$

$$= 20470.87 \text{ Kgf}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$= (\pi/2) * Dlo * Wo * 0.49 * Snw$$

$$= (3.14/2.0) * 114 * 10 * 0.49 * 118$$

$$= 10577. \text{ Kgf}$$

Shear, Pad Element Weld [Spew]:

$$= (\pi/2) * DP * WP * 0.49 * SEW$$

$$= (3.14/2.0) * 220 * 10 * 0.49 * 138$$

$$= 23811. \text{ Kgf}$$

Shear, Nozzle Wall [Snw]:

$$= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn$$

$$= (3.14 * 53.8) * (9.73 - 3) * 0.7 * 118$$

$$= 9576. \text{ Kgf}$$

Tension, Pad Groove Weld [Tpgw]:

$$= (\pi/2) * Dlo * Wgpn * 0.74 * Seg$$

$$= (3.14/2) * 114 * 12 * 0.74 * 138$$

$$= 22419. \text{ Kgf}$$

Tension, Shell Groove Weld [Tngw]:

$$= (\pi/2) * Dlo * (T - cas) * 0.74 * Sng$$

$$= (3.14/2.0) * 114 * (12 - 3) * 0.74 * 138$$

$$= 16814. \text{ Kgf}$$

Strength of Failure Paths:

$$PATH11 = (SPEW + SNW) = (23811 + 9576) = 33387 \text{ Kgf}$$

$$PATH22 = (Sonw + Tpgw + Tngw + Sinw)$$

$$= (10577 + 22419 + 16814 + 0) = 49810 \text{ Kgf}$$

Nozzle Calcs.: S1 (4in.) Nozl: 21 12:29pm May 21,2024

$$\begin{aligned} \text{PATH33} &= (\text{Spew} + \text{Tngw} + \text{Sinw}) \\ &= (23811 + 16814 + 0) = 40625 \text{ Kgf} \end{aligned}$$

Summary of Failure Path Calculations:

Path 1-1 = 33386 Kgf, must exceed W = 6184 Kgf or W1 = 19013 Kgf
Path 2-2 = 49809 Kgf, must exceed W = 6184 Kgf or W2 = 5729 Kgf
Path 3-3 = 40625 Kgf, must exceed W = 6184 Kgf or W3 = 20470 Kgf

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 22.8 bars

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

Nozzle is O.K. for the External Pressure 1.03 bars

The Drop for this Nozzle is : 4.7989 mm.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 196.7989 mm.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: S2 (6in.)

Nozl: 22 12:29pm May 21,2024

Input, Nozzle Desc: S2 (6in.)

From: 70

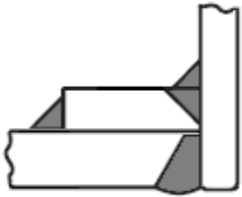
Pressure for Reinforcement Calculations	P	22.000	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		2413.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-333 6	
Material UNS Number		K03006	
Material Specification/Type	Smls. & wld. pipe		
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Outside	
Layout Angle		90.00	deg
Diameter		6.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		80	
Flange Material		SA-350 LF2	
Flange Type	Weld Neck Flange		
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	180.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Pad Material [Normalized]		SA-516 70	
Pad Allowable Stress at Temperature	Sp	137.90	N./mm ²
Pad Allowable Stress At Ambient	Spa	137.90	N./mm ²
Diameter of Pad along vessel surface	Dp	300.0000	mm.
Thickness of Pad	te	12.0000	mm.
Weld leg size between Pad and Shell	Wp	10.0000	mm.
Groove weld depth between Pad and Nozzle	Wgpn	12.0000	mm.
Reinforcing Pad Width		65.8625	mm.
Flange Class		300	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle With Pad, no Inside projection

Reinforcement CALCULATION, Description: S2 (6in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 6.625 in.
 Actual Thickness Used in Calculation 0.378 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.4986 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (22 \cdot 84.1) / (118 \cdot 1 + 0.4 \cdot 22)$$

$$= 1.5584 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.5566 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	310.1452	mm.
Parallel to Vessel Wall, opening length	d	155.0726	mm.
Normal to Vessel Wall (Thickness Limit), pad side Tlwp		22.5000	mm.

Weld Strength Reduction Factor [fr1]:

$$= \min(1, S_n / S_v)$$

$$= \min(1, 118 / 138)$$

$$= 0.855$$

Weld Strength Reduction Factor [fr2]:

$$= \min(1, S_n / S_v)$$

$$= \min(1, 118 / 138)$$

$$= 0.855$$

Weld Strength Reduction Factor [fr4]:

$$= \min(1, S_p / S_v)$$

$$= \min(1, 138 / 138)$$

$$= 1.000$$

Weld Strength Reduction Factor [fr3]:

$$= \min(fr2, fr4)$$

$$= \min(0.86, 1)$$

$$= 0.855$$

Results of Nozzle Reinforcement Area Calculations: (cm²)

AREA AVAILABLE, A1 to A5	Design	External	Mapnc
Area Required	Ar	11.772	3.806
			NA

Nozzle Calcs.: S2 (6in.) Nozl: 22 12:29pm May 21,2024

Area in Shell	A1	2.300	6.358	NA
Area in Nozzle Wall	A2	1.940	2.326	NA
Area in Inward Nozzle	A3	0.000	0.000	NA
Area in Welds	A41+A42+A43	1.612	1.612	NA
Area in Element	A5	15.807	15.807	NA
TOTAL AREA AVAILABLE	Atot	21.659	26.103	NA

The Internal Pressure Case Governs the Analysis.

Nozzle Angle Used in Area Calculations 90.00 Degs.

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:	217.6076	12.0000 mm.
Based on given Pad Diameter:	300.0000	4.4941 mm.
Based on Shell or Nozzle Thickness:	229.9330	9.6012 mm.

Area Required [A]:

$$= (d * tr * F + 2 * tn * tr * F * (1 - fr1)) \text{ UG-37(c)}$$

$$= (155 * 7.5 * 1 + 2 * 6.6 * 7.5 * 1 * (1 - 0.86))$$

$$= 11.772 \text{ cm}^2$$

Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:

$$= d (E1 * t - F * tr) - 2 * tn (E1 * t - F * tr) * (1 - fr1)$$

$$= 155 (1 * 9 - 1 * 7.5) - 2 * 6.6$$

$$(1 * 9 - 1 * 7.5) * (1 - 0.86)$$

$$= 2.300 \text{ cm}^2$$

Area Available in Nozzle Wall Projecting Outward [A2]:

$$= (2 * Tlwp) * (tn - trn) * fr2$$

$$= (2 * 22.5) * (6.6 - 1.56) * 0.86$$

$$= 1.940 \text{ cm}^2$$

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

$$= (Wo^2 - Ar \text{ Lost}) * Fr3 + ((Wi - can / 0.707)^2 - Ar \text{ Lost}) * fr2 + Trapfr4$$

$$= (1) * 0.86 + (0) * 0.86 + 75.7^2 * 1$$

$$= 1.612 \text{ cm}^2$$

Area Available in Element [A5]:

$$= (\min(Dp, DL) - (\text{Nozzle OD})) * (\min(tp, Tlwp, te)) * fr4$$

$$= (300 - 168) * 12 * 1$$

$$= 15.807 \text{ cm}^2$$

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

Wall Thickness for Internal/External pressures	ta = 4.5584 mm.
Wall Thickness per UG16(b),	tr16b = 4.5000 mm.
Wall Thickness, shell/head, internal pressure	trb1 = 10.4986 mm.
Wall Thickness	tb1 = max(trb1, tr16b) = 10.4986 mm.
Wall Thickness, shell/head, external pressure	trb2 = 3.3493 mm.
Wall Thickness	tb2 = max(trb2, tr16b) = 4.5000 mm.
Wall Thickness per table UG-45	tb3 = 9.2200 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[9.22, \max(10.5, 4.5)]$$

$$= 9.2200 \text{ mm.}$$

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

$$= \max(ta, tb)$$

$$= \max(4.56, 9.22)$$

$$= 9.2200 \text{ mm.}$$

Available Nozzle Neck Thickness = 9.6012 mm. --> OK

Stresses on Nozzle due to External and Pressure Loads per the ASME

B31.3 Piping Code (see 319.4.4 and 302.3.5):

Sustained	:	14.9,	Allowable	:	117.9 N./mm ²	Passed
Expansion	:	0.0,	Allowable	:	279.8 N./mm ²	Passed
Occasional	:	12.4,	Allowable	:	156.8 N./mm ²	Passed
Shear	:	6.5,	Allowable	:	82.5 N./mm ²	Passed

Note : The number of cycles on this nozzle was assumed to be 7000 or less for the determination of the expansion stress allowable.

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld (Impact tested) :

Note:

This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification	-46 °C
Calculated Minimum Design Metal Temperature	-104 °C

Nozzle Neck to Pad Weld for the Nozzle (Impact tested) :

Note:

This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification	-46 °C
Calculated Minimum Design Metal Temperature	-104 °C

Nozzle Neck to Pad Weld for Reinforcement pad, Curve: D

Govrn. thk, tg = 9.6, tr = 1.56, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.24$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D	-48 °C
Min Metal Temp. at Required thickness (UCS 66.1)	-104 °C

Shell to Pad Weld Junction at Pad OD, Curve: D

Govrn. thk, tg = 12, tr = 7.5, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.83$, Temp. Reduction = 9 °C

Min Metal Temp. w/o impact per UCS-66, Curve D	-48 °C
--	--------

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 9.6, tr = 1.56, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.24$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D	-48 °C
Min Metal Temp. at Required thickness (UCS 66.1)	-104 °C

Gov. MDMT of the Nozzle	:	-104 °C
Gov. MDMT of the Reinforcement Pad	:	-48 °C
Gov. MDMT of the nozzle to shell joint welded assembly	:	-48 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

MDMT of ASME B16.5/47 flange per Matl. Specification	-46 °C
Flange MDMT with Temp reduction per UCS-66(i) (2)	-89 °C

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: S2 (6in.)

Nozl: 22 12:29pm May 21,2024

Where the Stress Reduction Ratio per UCS-66(i)(2) is :

$$\text{Design Pressure/Ambient Rating} = 22.00/51.10 = 0.431$$

Weld Size Calculations, Description: S2 (6in.)

Intermediate Calc. for nozzle/shell Welds Tmin 6.6012 mm.
 Intermediate Calc. for pad/shell Welds TminPad 9.0000 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	4.6208 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.
Pad Weld	4.5000 = 0.5*TminPad	7.0700 = 0.7 * Wp mm.

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

Weld Load [W]:

$$= \max(0, (A-A1+2*tn*fr1*(E1*t-tr))Sv)$$

$$= \max(0, (11.8 - 2.3 + 2 * 6.6 * 0.86 * (1 * 9 - 7.5)) 138)$$

$$= 13557.96 \text{ Kgf}$$

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

Weld Load [W1]:

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

$$= (1.94 + 15.8 + 1.61 - 0 * 0.86) * 138$$

$$= 27222.50 \text{ Kgf}$$

Weld Load [W2]:

$$= (A2 + A3 + A4 + (2 * tn * t * fr1)) * Sv$$

$$= (1.94 + 0 + 0.86 + (1.02)) * 138$$

$$= 5359.09 \text{ Kgf}$$

Weld Load [W3]:

$$= (A2+A3+A4+A5+(2*tn*t*fr1))*S$$

$$= (1.94 + 0 + 1.61 + 15.8 + (1.02)) * 138$$

$$= 28651.06 \text{ Kgf}$$

Strength of Connection Elements for Failure Path Analysis

Shear, Outward Nozzle Weld [Sonw]:

$$= (\pi/2) * Dlo * Wo * 0.49 * Snw$$

$$= (3.14/2.0) * 168 * 10 * 0.49 * 118$$

$$= 15572. \text{ Kgf}$$

Shear, Pad Element Weld [Spew]:

$$= (\pi/2) * DP * WP * 0.49 * SEW$$

$$= (3.14/2.0) * 300 * 10 * 0.49 * 138$$

$$= 32469. \text{ Kgf}$$

Shear, Nozzle Wall [Snw]:

$$= (\pi * (Dlr + Dlo) / 4) * (Thk - Can) * 0.7 * Sn$$

$$= (3.14 * 80.8) * (9.6 - 3) * 0.7 * 118$$

$$= 14109. \text{ Kgf}$$

Tension, Pad Groove Weld [Tpgw]:

$$= (\pi/2) * Dlo * Wgpn * 0.74 * Seg$$

$$= (3.14/2) * 168 * 12 * 0.74 * 138$$

$$= 33006. \text{ Kgf}$$

Tension, Shell Groove Weld [Tngw]:

$$= (\pi/2) * Dlo * (T - cas) * 0.74 * Sng$$

$$= (3.14/2.0) * 168 * (12 - 3) * 0.74 * 138$$

$$= 24754. \text{ Kgf}$$

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: S2 (6in.) Nozl: 22 12:29pm May 21,2024

Strength of Failure Paths:

PATH11 = (SPEW + SNW) = (32469 + 14109) = 46578 Kgf
 PATH22 = (Sonw + Tpgw + Tngw + Sinw)
 = (15572 + 33006 + 24754 + 0) = 73332 Kgf
 PATH33 = (Spew + Tngw + Sinw)
 = (32469 + 24754 + 0) = 57224 Kgf

Summary of Failure Path Calculations:

Path 1-1 = 46577 Kgf, must exceed W = 13557 Kgf or W1 = 27222 Kgf
 Path 2-2 = 73331 Kgf, must exceed W = 13557 Kgf or W2 = 5359 Kgf
 Path 3-3 = 57223 Kgf, must exceed W = 13557 Kgf or W3 = 28651 Kgf

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 26.3 bars

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

Nozzle is O.K. for the External Pressure 1.03 bars

The Drop for this Nozzle is : 7.7175 mm.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 199.7175 mm.

Input Echo, WRC107/537 Item 1, Description: S2 (6in.) :

Diameter Basis for Vessel	Vbasis	ID	
Cylindrical or Spherical Vessel	Cylsph	Cylindrical	
Internal Corrosion Allowance	Cas	3.0000	mm.
Vessel Diameter	Dv	925.000	mm.
Vessel Thickness	Tv	12.000	mm.
Design Temperature	T1	120.0	°C
Vessel Material		SA-516 70	
Vessel UNS Number		K02700	
Vessel Cold S.I. Allowable	Smc	137.90	N./mm ²
Vessel Hot S.I. Allowable	Smh	137.90	N./mm ²

Note:

Using 2 * Yield for Discontinuity Stress Allowable (Div 2, 4.1.6.3), Sps.
 Make sure that material properties at this temperature are not time-dependent for Material: SA-516 70

Attachment Type	Type	Round	
Diameter Basis for Nozzle	Nbasis	OD	
Corrosion Allowance for Nozzle	Can	3.0000	mm.
Nozzle Diameter	Dn	168.275	mm.
Nozzle Thickness	Tn	9.601	mm.
Nozzle Material		SA-333 6	
Nozzle UNS Number		K03006	
Nozzle Cold S.I. Allowable	SNmc	117.90	N./mm ²
Nozzle Hot S.I. Allowable	SNmh	117.90	N./mm ²
Thickness of Reinforcing Pad	Tpad	12.000	mm.
Diameter of Reinforcing Pad	Dpad	300.000	mm.
Design Internal Pressure	Dp	22.000	bars
Include Pressure Thrust		No	

External Forces and Moments in WRC 107/537 Convention:

Radial Load (SUS)	P	856.6	Kgf
Longitudinal Shear (SUS)	V1	856.6	Kgf
Circumferential Shear (SUS)	Vc	642.5	Kgf
Circumferential Moment (SUS)	Mc	0.3	Kg-m.

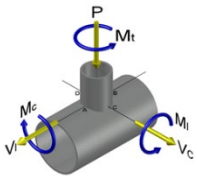
Nozzle Calcs.: S2 (6in.) Nozl: 22 12:29pm May 21,2024

Longitudinal Moment (SUS) Ml 0.3 Kg-m.
 Torsional Moment (SUS) Mt 0.4 Kg-m.

Use Interactive Control No
 WRC107 Version Version March 1979

Include Pressure Stress Indices per Div. 2 No
 Compute Pressure Stress per WRC-368 No
 Local Loads applied at end of Nozzle/Attachment No

Note:
 WRC Bulletin 537 provides equations for the dimensionless curves found in bulletin 107. As noted in the foreword to bulletin 537, "537 is equivalent to WRC 107". Where 107 is printed in the results below, "537" can be interchanged with "107".



Stress Attenuation Diameter (for Insert Plates) per WRC 297:
 = NozzleOD + 2 * 1.65 * sqrt(Rmean(t - ca))
 = 168.275 + 2 * 1.65 * sqrt(470.0 (12.0 - 3.0))
 = 382.902 mm.

WRC 107 Stress Calculation for SUSstained loads:

Radial Load P 856.6 Kgf
 Circumferential Shear VC 642.5 Kgf
 Longitudinal Shear VL 856.6 Kgf
 Circumferential Moment MC 0.3 Kg-m.
 Longitudinal Moment ML 0.3 Kg-m.
 Torsional Moment MT 0.4 Kg-m.

Dimensionless Parameters used : Gamma = 22.67

Dimensionless Loads for Cylindrical Shells at Attachment Junction:

Curves read for 1979	Beta	Figure	Value	Location
N(PHI) / (P/Rm)	0.155	4C	3.739	(A,B)
N(PHI) / (P/Rm)	0.155	3C	3.087	(C,D)
M(PHI) / (P)	0.155	2C1	0.073	(A,B)
M(PHI) / (P)	0.155	1C	0.106	(C,D)
N(PHI) / (MC/ (Rm**2 * Beta))	0.155	3A	0.768	(A,B,C,D)
M(PHI) / (MC/ (Rm * Beta))	0.155	1A	0.093	(A,B,C,D)
N(PHI) / (ML/ (Rm**2 * Beta))	0.155	3B	2.456	(A,B,C,D)
M(PHI) / (ML/ (Rm * Beta))	0.155	1B	0.042	(A,B,C,D)
N(x) / (P/Rm)	0.155	3C	3.087	(A,B)
N(x) / (P/Rm)	0.155	4C	3.739	(C,D)
M(x) / (P)	0.155	1C1	0.111	(A,B)
M(x) / (P)	0.155	2C	0.074	(C,D)
N(x) / (MC/ (Rm**2 * Beta))	0.155	4A	1.170	(A,B,C,D)
M(x) / (MC/ (Rm * Beta))	0.155	2A	0.050	(A,B,C,D)
N(x) / (ML/ (Rm**2 * Beta))	0.155	4B	0.742	(A,B,C,D)
M(x) / (ML/ (Rm * Beta))	0.155	2B	0.068	(A,B,C,D)

Stress Concentration Factors: Kn = 1.00, Kb = 1.00

Stresses in the Vessel at the Attachment Junction (N./mm²)

FileName : Chiller-Rev.01

Nozzle Calcs.: S2 (6in.)

Noz1: 22 12:29pm May 21,2024

		Stress Intensity Values at							
Type of Stress	Load	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
Circ. Memb.	P	-3.1	-3.1	-3.1	-3.1	-2.6	-2.6	-2.6	-2.6
Circ. Bend.	P	-8.4	8.4	-8.4	8.4	-12.1	12.1	-12.1	12.1
Circ. Memb.	MC	0.0	0.0	0.0	0.0	-0.0	-0.0	0.0	0.0
Circ. Memb.	ML	-0.0	-0.0	0.0	0.0	0.0	0.0	0.0	0.0
Circ. Bend.	ML	-0.0	0.0	0.0	-0.0	0.0	0.0	0.0	0.0
Tot. Circ. Str.		-11.6	5.3	-11.5	5.2	-14.7	9.5	-14.7	9.5
Long. Memb.	P	-2.6	-2.6	-2.6	-2.6	-3.1	-3.1	-3.1	-3.1
Long. Bend.	P	-12.6	12.6	-12.6	12.6	-8.4	8.4	-8.4	8.4
Long. Memb.	MC	0.0	0.0	0.0	0.0	-0.0	-0.0	0.0	0.0
Long. Bend.	MC	0.0	0.0	0.0	0.0	-0.0	0.0	0.0	-0.0
Long. Memb.	ML	-0.0	-0.0	0.0	0.0	0.0	0.0	0.0	0.0
Long. Bend.	ML	-0.0	0.0	0.0	-0.0	0.0	0.0	0.0	0.0
Tot. Long. Str.		-15.3	10.1	-15.2	10.0	-11.6	5.3	-11.5	5.2
Shear	VC	1.1	1.1	-1.1	-1.1	0.0	0.0	0.0	0.0
Shear	VL	0.0	0.0	0.0	0.0	-1.5	-1.5	1.5	1.5
Shear	MT	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Tot. Shear		1.1	1.1	-1.1	-1.1	-1.5	-1.5	1.5	1.5
Str. Int.		15.6	10.3	15.5	10.3	15.3	10.0	15.3	10.0

Dimensionless Parameters used : Gamma = 52.22

Dimensionless Loads for Cylindrical Shells at Pad edge:

Curves read for 1979	Beta	Figure	Value	Location
N(PHI) / (P/Rm)	0.279	4C	5.662	(A,B)
N(PHI) / (P/Rm)	0.279	3C	2.676	(C,D)
M(PHI) / (P)	0.279	2C1	0.015	(A,B)
M(PHI) / (P)	0.279	1C !	0.065	(C,D)
N(PHI) / (MC/ (Rm**2 * Beta))	0.279	3A	1.741	(A,B,C,D)
M(PHI) / (MC/ (Rm * Beta))	0.279	1A	0.061	(A,B,C,D)
N(PHI) / (ML/ (Rm**2 * Beta))	0.279	3B	3.690	(A,B,C,D)
M(PHI) / (ML/ (Rm * Beta))	0.279	1B	0.011	(A,B,C,D)
N(x) / (P/Rm)	0.279	3C	2.676	(A,B)
N(x) / (P/Rm)	0.279	4C	5.662	(C,D)
M(x) / (P)	0.279	1C1	0.037	(A,B)
M(x) / (P)	0.279	2C !	0.034	(C,D)
N(x) / (MC/ (Rm**2 * Beta))	0.279	4A	4.862	(A,B,C,D)
M(x) / (MC/ (Rm * Beta))	0.279	2A	0.025	(A,B,C,D)
N(x) / (ML/ (Rm**2 * Beta))	0.279	4B	1.882	(A,B,C,D)
M(x) / (ML/ (Rm * Beta))	0.279	2B	0.017	(A,B,C,D)

Note - The ! mark next to the figure name denotes curve value exceeded.

Stress Concentration Factors: Kn = 1.00, Kb = 1.00

Stresses in the Vessel at the Edge of Reinforcing Pad (N./mm²)

		Stress Intensity Values at							
Type of Stress	Load	Au	Al	Bu	Bl	Cu	Cl	Du	Dl

FileName : Chiller-Rev.01

Nozzle Calcs.: S2 (6in.)

Nozl: 22 12:29pm May 21,2024

Circ. Memb. P	-11.2	-11.2	-11.2	-11.2	-5.3	-5.3	-5.3	-5.3
Circ. Bend. P	-9.2	9.2	-9.2	9.2	-40.7	40.7	-40.7	40.7
Circ. Memb. MC	0.0	0.0	0.0	0.0	-0.0	-0.0	0.0	0.0
Circ. Memb. ML	0.0	0.0	0.0	0.0	-0.1	0.1	0.1	-0.1
Circ. Memb. ML	-0.0	-0.0	0.0	0.0	0.0	0.0	0.0	0.0
Circ. Bend. ML	-0.0	0.0	0.0	-0.0	0.0	0.0	0.0	0.0
Tot. Circ. Str.	-20.5	-2.0	-20.4	-2.0	-46.1	35.5	-45.9	35.3
Long. Memb. P	-5.3	-5.3	-5.3	-5.3	-11.2	-11.2	-11.2	-11.2
Long. Bend. P	-22.7	22.7	-22.7	22.7	-20.9	20.9	-20.9	20.9
Long. Memb. MC	0.0	0.0	0.0	0.0	-0.0	-0.0	0.0	0.0
Long. Bend. MC	0.0	0.0	0.0	0.0	-0.0	0.0	0.0	-0.0
Long. Memb. ML	-0.0	-0.0	0.0	0.0	0.0	0.0	0.0	0.0
Long. Bend. ML	-0.0	0.0	0.0	-0.0	0.0	0.0	0.0	0.0
Tot. Long. Str.	-28.1	17.4	-28.0	17.4	-32.2	9.7	-32.1	9.7
Shear VC	1.5	1.5	-1.5	-1.5	0.0	0.0	0.0	0.0
Shear VL	0.0	0.0	0.0	0.0	-2.0	-2.0	2.0	2.0
Shear MT	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Tot. Shear	1.5	1.5	-1.5	-1.5	-2.0	-2.0	2.0	2.0
Str. Int.	28.4	19.7	28.3	19.6	46.4	35.6	46.2	35.5

WRC 107/537 Stress Summations:

Vessel Stress Summation at Attachment Junction (N./mm²)

Type of Stress	Load	Stress Intensity Values at							
		Au	Al	Bu	Bl	Cu	Cl	Du	Dl
Circ. Pm (SUS)		47.7	49.9	47.7	49.9	47.7	49.9	47.7	49.9
Circ. Pl (SUS)		-3.2	-3.2	-3.1	-3.1	-2.6	-2.6	-2.6	-2.6
Circ. Q (SUS)		-8.4	8.4	-8.4	8.4	-12.1	12.1	-12.1	12.1
Long. Pm (SUS)		23.8	23.8	23.8	23.8	23.8	23.8	23.8	23.8
Long. Pl (SUS)		-2.6	-2.6	-2.6	-2.6	-3.1	-3.1	-3.1	-3.1
Long. Q (SUS)		-12.7	12.7	-12.6	12.6	-8.4	8.4	-8.4	8.4
Shear Pm (SUS)		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Shear Pl (SUS)		1.1	1.1	-1.1	-1.1	-1.5	-1.5	1.5	1.5
Shear Q (SUS)		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Pm (SUS)		47.7	49.9	47.7	49.9	47.7	49.9	47.7	49.9
Pm+Pl (SUS)		44.6	46.8	44.6	46.8	45.2	47.4	45.2	47.4
Pm+Pl+Q (Total)		36.2	55.2	36.2	55.2	33.1	59.5	33.2	59.4

Vessel Stress Summation Comparison (N./mm²):

Type of Stress Int.	Max. S.I.	S.I. Allowable	Result
Pm (SUS)	49.89	137.90	Passed
Pm+Pl (SUS)	47.39	206.85	Passed
Pm+Pl+Q (TOTAL)	59.52	413.70	Passed

Because only sustained loads were specified, the Pm+Pl+Q allowable was 3 * Smh.

WRC 107/537 Stress Summations:

Vessel Stress Summation at Reinforcing Pad Edge (N./mm²)

Type of Stress	Load	Stress Intensity Values at							
		Au	Al	Bu	Bl	Cu	Cl	Du	Dl
Circ. Pm (SUS)		112.7	114.9	112.7	114.9	112.7	114.9	112.7	114.9
Circ. Pl (SUS)		-11.3	-11.3	-11.2	-11.2	-5.3	-5.3	-5.3	-5.3
Circ. Q (SUS)		-9.2	9.2	-9.2	9.2	-40.8	40.8	-40.6	40.6
Long. Pm (SUS)		56.4	56.4	56.4	56.4	56.4	56.4	56.4	56.4
Long. Pl (SUS)		-5.3	-5.3	-5.3	-5.3	-11.3	-11.3	-11.2	-11.2
Long. Q (SUS)		-22.8	22.8	-22.7	22.7	-21.0	21.0	-20.9	20.9
Shear Pm (SUS)		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Shear Pl (SUS)		1.5	1.5	-1.5	-1.5	-2.0	-2.0	2.0	2.0
Shear Q (SUS)		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Pm (SUS)		112.7	114.9	112.7	114.9	112.7	114.9	112.7	114.9
Pm+Pl (SUS)		101.5	103.7	101.5	103.7	107.4	109.6	107.5	109.7
Pm+Pl+Q (Total)		92.2	112.9	92.3	112.9	66.7	150.4	66.9	150.3

Vessel Stress Summation Comparison (N./mm²):

Type of Stress Int.	Max. S.I.	S.I. Allowable	Result
Pm (SUS)	114.91	137.90	Passed
Pm+Pl (SUS)	109.66	206.85	Passed
Pm+Pl+Q (TOTAL)	150.43	413.70	Passed

Because only sustained loads were specified, the Pm+Pl+Q allowable was 3 * Smh.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: V (2in.)

Nozl: 23 12:29pm May 21,2024

Input, Nozzle Desc: V (2in.)

From: 70

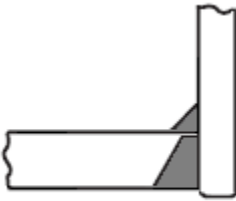
Pressure for Reinforcement Calculations	P	22.000	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		3363.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-350 LF2	
Material UNS Number		K03011	
Material Specification/Type		Forgings	
Allowable Stress at Temperature	Sn	137.90	N./mm ²
Allowable Stress At Ambient	Sna	137.90	N./mm ²
Diameter Basis (for tr calc only)		Inside	
Layout Angle		90.00	deg
Diameter		2.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	16.6000	mm.
Flange Material [Normalized]		SA-350 LF2	
Flange Type		Long Weld Neck	
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	200.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Flange Class		300	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: V (2in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 2.000 in.
 Actual Thickness Used in Calculation 0.654 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.4986 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

$$= (P \cdot R) / (S_n \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 28.4) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 0.4575 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3862 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	113.6000	mm.
Parallel to Vessel Wall, opening length	d	56.8000	mm.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	22.5000	mm.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: V (2in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures	ta	= 3.4575	mm.
Wall Thickness per UG16(b),	tr16b	= 4.5000	mm.
Wall Thickness, shell/head, internal pressure	trb1	= 10.4986	mm.
Wall Thickness	tb1 = max(trb1, tr16b)	= 10.4986	mm.
Wall Thickness, shell/head, external pressure	trb2	= 3.3493	mm.
Wall Thickness	tb2 = max(trb2, tr16b)	= 4.5000	mm.
Wall Thickness per table UG-45	tb3	= 7.8000	mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[7.8, \max(10.5, 4.5)]$$

$$= 7.8000 \text{ mm.}$$

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

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: V (2in.)

Nozl: 23 12:29pm May 21,2024

= max(ta, tb)
= max(3.46, 7.8)
= 7.8000 mm.

Available Nozzle Neck Thickness = 16.6000 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 12, tr = 7.5, c = 3 mm., E* = 1
Thickness Ratio = tr *(E*)/(tg - c) = 0.83, Temp. Reduction = 9 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Gov. MDMT of the nozzle to shell joint welded assembly : -48 °C

Weld Size Calculations, Description: V (2in.)

Intermediate Calc. for nozzle/shell Welds Tmin 9.0000 mm.

Results Per UW-16.1:

Required Thickness Actual Thickness
Nozzle Weld 6.0000 = Min per Code 7.0700 = 0.7 * Wo mm.

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 26.3 bars

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

The Drop for this Nozzle is : 1.9110 mm.
The Cut Length for this Nozzle is, Drop + Ho + H + T : 213.9110 mm.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: D (2in.)

Nozl: 24 12:29pm May 21,2024

Input, Nozzle Desc: D (2in.)

From: 70

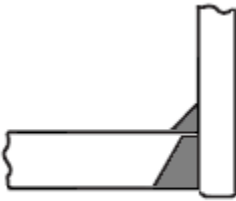
Pressure for Reinforcement Calculations	P	22.048	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		3363.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-350 LF2	
Material UNS Number		K03011	
Material Specification/Type		Forgings	
Allowable Stress at Temperature	Sn	137.90	N./mm ²
Allowable Stress At Ambient	Sna	137.90	N./mm ²
Diameter Basis (for tr calc only)		Inside	
Layout Angle		270.00	deg
Diameter		2.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	16.6000	mm.
Flange Material [Normalized]		SA-350 LF2	
Flange Type		Long Weld Neck	
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	200.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Flange Class		300	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: D (2in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 2.000 in.
 Actual Thickness Used in Calculation 0.654 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.5152 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

$$= (P \cdot R) / (S_n \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 28.4) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 0.4585 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3862 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	113.6000	mm.
Parallel to Vessel Wall, opening length	d	56.8000	mm.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	22.5000	mm.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: D (2in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures	ta	= 3.4585	mm.
Wall Thickness per UG16(b),	tr16b	= 4.5000	mm.
Wall Thickness, shell/head, internal pressure	trb1	= 10.5152	mm.
Wall Thickness	tb1 = max(trb1, tr16b)	= 10.5152	mm.
Wall Thickness, shell/head, external pressure	trb2	= 3.3493	mm.
Wall Thickness	tb2 = max(trb2, tr16b)	= 4.5000	mm.
Wall Thickness per table UG-45	tb3	= 7.8000	mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[7.8, \max(10.5, 4.5)]$$

$$= 7.8000 \text{ mm.}$$

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

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: D (2in.)

Nozl: 24 12:29pm May 21,2024

= max(ta, tb)
= max(3.46, 7.8)
= 7.8000 mm.

Available Nozzle Neck Thickness = 16.6000 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 12, tr = 7.52, c = 3 mm., E* = 1
Thickness Ratio = tr *(E*)/(tg - c) = 0.84, Temp. Reduction = 9 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Gov. MDMT of the nozzle to shell joint welded assembly : -48 °C

Weld Size Calculations, Description: D (2in.)

Intermediate Calc. for nozzle/shell Welds Tmin 9.0000 mm.

Results Per UW-16.1:

Required Thickness Actual Thickness
Nozzle Weld 6.0000 = Min per Code 7.0700 = 0.7 * Wo mm.

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 26.4 bars

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

The Drop for this Nozzle is : 1.9110 mm.
The Cut Length for this Nozzle is, Drop + Ho + H + T : 213.9110 mm.

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FileName : Chiller-Rev.01 -----

Nozzle Calcs.: PSV (3in.)

Nozl: 25 12:29pm May 21,2024

Input, Nozzle Desc: PSV (3in.)

From: 70

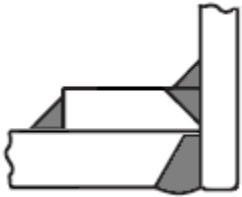
Pressure for Reinforcement Calculations	P	22.000	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		1483.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-333 6	
Material UNS Number		K03006	
Material Specification/Type	Smls. & wld. pipe		
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Outside	
Layout Angle		90.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		160	
Flange Material		SA-350 LF2	
Flange Type	Weld Neck Flange		
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	200.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Pad Material [Normalized]		SA-516 70	
Pad Allowable Stress at Temperature	Sp	137.90	N./mm ²
Pad Allowable Stress At Ambient	Spa	137.90	N./mm ²
Diameter of Pad along vessel surface	Dp	190.0000	mm.
Thickness of Pad	te	12.0000	mm.
Weld leg size between Pad and Shell	Wp	10.0000	mm.
Groove weld depth between Pad and Nozzle	Wgpn	12.0000	mm.
Reinforcing Pad Width		50.5500	mm.
Flange Class		300	
Flange Grade		GR 1.1	

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle With Pad, no Inside projection

Reinforcement CALCULATION, Description: PSV (3in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Outside Diameter Used in Calculation 3.500 in.
 Actual Thickness Used in Calculation 0.383 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.4986 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

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

$$= (22 \cdot 44.5) / (118 \cdot 1 + 0.4 \cdot 22)$$

$$= 0.8233 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.3989 mm.

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

Parallel to Vessel Wall (Diameter Limit) D1 150.8618 mm.
 Parallel to Vessel Wall, opening length d 75.4309 mm.
 Normal to Vessel Wall (Thickness Limit), pad side Tlwp 22.5000 mm.

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

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: PSV (3in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures ta = 3.8233 mm.
 Wall Thickness per UG16(b), tr16b = 4.5000 mm.
 Wall Thickness, shell/head, internal pressure trb1 = 10.4986 mm.
 Wall Thickness tb1 = max(trb1, tr16b) = 10.4986 mm.
 Wall Thickness, shell/head, external pressure trb2 = 3.3493 mm.
 Wall Thickness tb2 = max(trb2, tr16b) = 4.5000 mm.
 Wall Thickness per table UG-45 tb3 = 7.8000 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[7.8, \max(10.5, 4.5)]$$

$$= 7.8000 \text{ mm.}$$

Minimum Wall Thickness of Nozzle Necks [tUG-45]:
 = max(ta, tb)
 = max(3.82, 7.8)
 = 7.8000 mm.

Available Nozzle Neck Thickness = 9.7345 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld (Impact tested) :

Note:
This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle Neck to Pad Weld for the Nozzle (Impact tested) :

Note:
This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle Neck to Pad Weld for Reinforcement pad, Curve: D

Govrn. thk, tg = 9.73, tr = 0.82, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.12$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Shell to Pad Weld Junction at Pad OD, Curve: D

Govrn. thk, tg = 12, tr = 7.5, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.83$, Temp. Reduction = 9 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 9.73, tr = 0.82, c = 3 mm., E* = 1
 Thickness Ratio = $tr \cdot (E^*) / (tg - c) = 0.12$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the Nozzle : -104 °C
 Gov. MDMT of the Reinforcement Pad : -48 °C
 Gov. MDMT of the nozzle to shell joint welded assembly : -48 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

MDMT of ASME B16.5/47 flange per Matl. Specification -46 °C
 Flange MDMT with Temp reduction per UCS-66(i) (2) -89 °C

Where the Stress Reduction Ratio per UCS-66(i)(2) is :
 Design Pressure/Ambient Rating = 22.00/51.10 = 0.431

Weld Size Calculations, Description: PSV (3in.)

Nozzle Calcs.: PSV (3in.) Nozl: 25 12:29pm May 21,2024

Intermediate Calc. for nozzle/shell Welds Tmin 6.7345 mm.
Intermediate Calc. for pad/shell Welds TminPad 9.0000 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	$4.7142 = 0.7 * t_{min}$	$7.0700 = 0.7 * W_o$ mm.
Pad Weld	$4.5000 = 0.5 * T_{minPad}$	$7.0700 = 0.7 * W_p$ mm.

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 26.3 bars

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

The Drop for this Nozzle is : 2.1410 mm.

The Cut Length for this Nozzle is, Drop + Ho + H + T : 214.1409 mm.

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FileName : Chiller-Rev.01

Nozzle Calcs.: LG1 (2in.)

Nozl: 26 12:29pm May 21,2024

Input, Nozzle Desc: LG1 (2in.)

From: 70

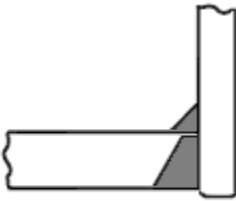
Pressure for Reinforcement Calculations	P	22.000	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		1913.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-333 6	
Material UNS Number		K03006	
Material Specification/Type	Smls. & wld. pipe		
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Inside	
Layout Angle		90.00	deg
Diameter		2.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		160	
Flange Material		SA-350 LF2	
Flange Type	Weld Neck Flange		
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	150.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Flange Class		300	
Flange Grade		GR 1.1	
Piping Exit Angle [North Clockwise reference]		270.0	deg
Bend Radius Multiplier		1.50	
Horizontal Run Length		550.0	mm.
Centerline Distance to Tangent		150.0	mm.

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: LG1 (2in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Inside Diameter Used in Calculation	1.773 in.
Actual Thickness Used in Calculation	0.301 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.4986 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

$$= (P \cdot R) / (S_n \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 25.5) / (118 \cdot 1 - 0.6 \cdot 22)$$

$$= 0.4815 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.2826 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	102.0684 mm.
Parallel to Vessel Wall, opening length	d	51.0342 mm.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	11.6135 mm.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: LG1 (2in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures	ta = 3.4815 mm.
Wall Thickness per UG16(b),	tr16b = 4.5000 mm.
Wall Thickness, shell/head, internal pressure	trb1 = 10.4986 mm.
Wall Thickness	tb1 = max(trb1, tr16b) = 10.4986 mm.
Wall Thickness, shell/head, external pressure	trb2 = 3.3493 mm.
Wall Thickness	tb2 = max(trb2, tr16b) = 4.5000 mm.
Wall Thickness per table UG-45	tb3 = 6.4200 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[6.42, \max(10.5, 4.5)]$$

$$= 6.4200 \text{ mm.}$$

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

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: LG1 (2in.) Nozl: 26 12:29pm May 21,2024

= max(ta, tb)
 = max(3.48, 6.42)
 = 6.4200 mm.

Available Nozzle Neck Thickness = 7.6454 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld (Impact tested) :

Note:
 This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 7.65, tr = 0.48, c = 3 mm., E* = 1
 Thickness Ratio = tr *(E*)/(tg - c) = 0.1, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the nozzle to shell joint welded assembly : -104 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

MDMT of ASME B16.5/47 flange per Matl. Specification -46 °C
 Flange MDMT with Temp reduction per UCS-66(i) (2) -89 °C

Where the Stress Reduction Ratio per UCS-66(i)(2) is :
 Design Pressure/Ambient Rating = 22.00/51.10 = 0.431

Weld Size Calculations, Description: LG1 (2in.)

Intermediate Calc. for nozzle/shell Welds Tmin 4.6454 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	3.2518 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.

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 26.3 bars

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

The Drop for this Nozzle is : 0.9846 mm.
 The Cut Length for this Nozzle is, Drop + Ho + H + T : 162.9846 mm.

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FileName : Chiller-Rev.01

Nozzle Calcs.: LG2 (2in.)

Nozl: 27 12:29pm May 21,2024

Input, Nozzle Desc: LG2 (2in.)

From: 70

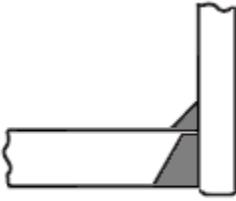
Pressure for Reinforcement Calculations	P	22.048	bars
Temperature for Internal Pressure	Temp	120	°C
Design External Pressure	Pext	1.03	bars
Temperature for External Pressure	Tempex	120	°C
Parent Material [Normalized]		SA-516 70	
Parent Allowable Stress at Temperature	Sv	137.90	N./mm ²
Parent Allowable Stress At Ambient	Sva	137.90	N./mm ²
Inside Diameter of Cylindrical Shell	D	925.00	mm.
Design Length of Section	L	2892.0833	mm.
Shell Finished (Minimum) Thickness	t	12.0000	mm.
Shell Internal Corrosion Allowance	c	3.0000	mm.
Shell External Corrosion Allowance	co	0.0000	mm.
Distance from Bottom/Left Tangent		1913.53	mm.
User Entered Minimum Design Metal Temperature		-45.00	°C

Type of Element Connected to the Parent : Nozzle

Material [Impact Tested]		SA-333 6	
Material UNS Number		K03006	
Material Specification/Type	Smls. & wld. pipe		
Allowable Stress at Temperature	Sn	117.90	N./mm ²
Allowable Stress At Ambient	Sna	117.90	N./mm ²
Diameter Basis (for tr calc only)		Inside	
Layout Angle		270.00	deg
Diameter		2.0000	in.
Size and Thickness Basis		Minimum	
Nominal Thickness		160	
Flange Material		SA-350 LF2	
Flange Type	Weld Neck Flange		
Corrosion Allowance	can	3.0000	mm.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	150.0000	mm.
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000	mm.
Groove weld depth between Nozzle and Vessel	Wgnv	12.0000	mm.
Inside Projection	h	0.0000	mm.
Weld leg size, Inside Element to Shell	Wi	0.0000	mm.
Flange Class		300	
Flange Grade		GR 1.1	
Piping Exit Angle [North Clockwise reference]		90.0	deg
Bend Radius Multiplier		1.50	
Horizontal Run Length		550.0	mm.
Centerline Distance to Tangent		150.0	mm.

The Pressure Design option was Design Pressure + static head.

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



Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: LG2 (2in.)

ASME Code, Section VIII, Div. 1, 2019, UG-37 to UG-45

Actual Inside Diameter Used in Calculation	1.773 in.
Actual Thickness Used in Calculation	0.301 in.

Nozzle input data check completed without errors.

Reqd thk per UG-37(a) of Cylindrical Shell, Tr [Int. Press]

$$= (P \cdot R) / (S_v \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 466) / (138 \cdot 1 - 0.6 \cdot 22)$$

$$= 7.5152 \text{ mm.}$$

Reqd thk per UG-37(a) of Nozzle Wall, Trn [Int. Press]

$$= (P \cdot R) / (S_n \cdot E - 0.6 \cdot P) \text{ per UG-27 (c) (1)}$$

$$= (22 \cdot 25.5) / (118 \cdot 1 - 0.6 \cdot 22)$$

$$= 0.4826 \text{ mm.}$$

Required Nozzle thickness under External Pressure per UG-28 : 0.2826 mm.

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

Parallel to Vessel Wall (Diameter Limit)	D1	102.0684 mm.
Parallel to Vessel Wall, opening length	d	51.0342 mm.
Normal to Vessel Wall (Thickness Limit), no pad	Tlnp	11.6135 mm.

Note:

*Taking a UG-36(c)(3)(a) exemption for nozzle: LG2 (2in.).
 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 small nozzle area calculation or
 force the Appendix 1-10 computation in Nozzle Design Options.*

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

Wall Thickness for Internal/External pressures	ta = 3.4826 mm.
Wall Thickness per UG16(b),	tr16b = 4.5000 mm.
Wall Thickness, shell/head, internal pressure	trb1 = 10.5152 mm.
Wall Thickness	tb1 = max(trb1, tr16b) = 10.5152 mm.
Wall Thickness, shell/head, external pressure	trb2 = 3.3493 mm.
Wall Thickness	tb2 = max(trb2, tr16b) = 4.5000 mm.
Wall Thickness per table UG-45	tb3 = 6.4200 mm.

Determine Nozzle Thickness candidate [tb]:

$$= \min[tb3, \max(tb1, tb2)]$$

$$= \min[6.42, \max(10.5, 4.5)]$$

$$= 6.4200 \text{ mm.}$$

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

FileName : Chiller-Rev.01 -----

Nozzle Calcs.: LG2 (2in.) Nozl: 27 12:29pm May 21,2024

= max(ta, tb)
 = max(3.48, 6.42)
 = 6.4200 mm.

Available Nozzle Neck Thickness = 7.6454 mm. --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

Nozzle Neck to Flange Weld (Impact tested) :

Note:
This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C
 Calculated Minimum Design Metal Temperature -104 °C

Nozzle-Shell/Head Weld (UCS-66(a)1(b)), Curve: D

Govrn. thk, tg = 7.65, tr = 0.48, c = 3 mm., E* = 1
 Thickness Ratio = $tr * (E*) / (tg - c) = 0.1$, Temp. Reduction = 78 °C

Min Metal Temp. w/o impact per UCS-66, Curve D -48 °C
 Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Gov. MDMT of the nozzle to shell joint welded assembly : -104 °C

ANSI Flange MDMT including Temperature reduction per UCS-66.1:

MDMT of ASME B16.5/47 flange per Matl. Specification -46 °C
 Flange MDMT with Temp reduction per UCS-66(i) (2) -89 °C

Where the Stress Reduction Ratio per UCS-66(i)(2) is :
 Design Pressure/Ambient Rating = 22.05/51.10 = 0.431

Weld Size Calculations, Description: LG2 (2in.)

Intermediate Calc. for nozzle/shell Welds Tmin 4.6454 mm.

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	3.2518 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm.

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 26.4 bars

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

The Drop for this Nozzle is : 0.9846 mm.
 The Cut Length for this Nozzle is, Drop + Ho + H + T : 162.9846 mm.

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Input Echo, Tubesheet Number 1, Description: Tubesheet

Shell Data:

Main Shell Description: Port Barrel

Shell Maximum Design Pressure	Psd,max	22.00	bars
Shell Maximum Operating Pressure	Psox,max	22.00	bars
Shell Minimum Operating Pressure	Psox,min	0.00	bars
Shell Thickness	ts	12.0000	mm.
Shell Internal Corrosion Allowance	cas	3.0000	mm.
Shell External Corrosion Allowance	caext	0.0000	mm.
Inside Diameter of Shell	Ds	600.000	mm.
Shell Temperature for Internal Pressure	Ts	120.00	°C
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	137.90	N./mm ²
Shell Allowable Stress at Ambient		137.90	N./mm ²

Channel Description: CH. Barrel

Channel Type:		Cylinder	
Channel Maximum Design Pressure	Ptd,max	6.80	bars
Channel Maximum Operating Pressure	Ptox,max	6.80	bars
Channel Minimum Operating Pressure	Ptox,min	0.00	bars
Channel Thickness	tc	10.0000	mm.
Channel Corrosion Allowance	cac	3.0000	mm.
Inside Diameter of Channel	Dc	600.000	mm.
Channel Design Temperature	TEMPC	85.00	°C
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	137.90	N./mm ²
Channel Allowable Stress at Ambient		137.90	N./mm ²

Tube Data:

Number of Tube Holes	Nt	376	
Tube Wall Thickness	et	1.6500	mm.
Tube Outside Diameter	D	19.0500	mm.
Total Straight Tube Length	Lt	2300.00	mm.
Straight Tube Length (bet. inner tubsht faces) L		2240.00	mm.
Design Temperature of the Tubes		120.00	°C
Tube Material		SA-334 6	
Tube Material UNS Number		K03006	
Is this a Welded Tube		No	
Tube Material Specification used	Smls. & wld. tube		
Tube Allowable Stress at Temperature		117.90	N./mm ²
Tube Allowable Stress At Ambient		117.90	N./mm ²
Tube Yield Stress At design Temperature	Syt	217.36	N./mm ²
Tube Pitch (Center to Center Spacing)	P	23.8130	mm.
Tube Layout Pattern		Triangular	
Radius to Outermost Tube Hole Center	ro	283.510	mm.
Largest Center-to-Center Tube Distance	Ul	52.3870	mm.
Length of Expanded Portion of Tube	ltx	57.0000	mm.
Tube-side pass partition groove depth	hg	5.0000	mm.

Tubesheet Data:

Tubesheet TYPE: U-tube, Gasketed both Sides, Conf. d

Tubesheet Design Metal Temperature T 120.00 °C
 Tubesheet Material SA-350 LF2

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-350 LF2

Tubesheet Material UNS Number K03011
 Tubesheet Allowable Stress at Temperature S 137.90 N./mm²
 Tubesheet Allowable Stress at Ambient Tt 137.90 N./mm²
 Thickness of Tubesheet h 60.0000 mm.
 Tubesheet Corr. Allowance (Shell side) Cats 3.0000 mm.
 Tubesheet Corr. Allowance (Channel side) Catc 3.0000 mm.
 Tubesheet Outside Diameter A 650.000 mm.

Dimension G for the Channel Side Gc 637.151 mm.
 Area of the Untubed Lanes AL 425.5 cm²

Junction Stress Reduction option Increase Tubesheet thickness
 Perform Differential Pressure Design NO
 Run Multiple Load Cases YES

Additional Data for Gasketed Tubesheets:

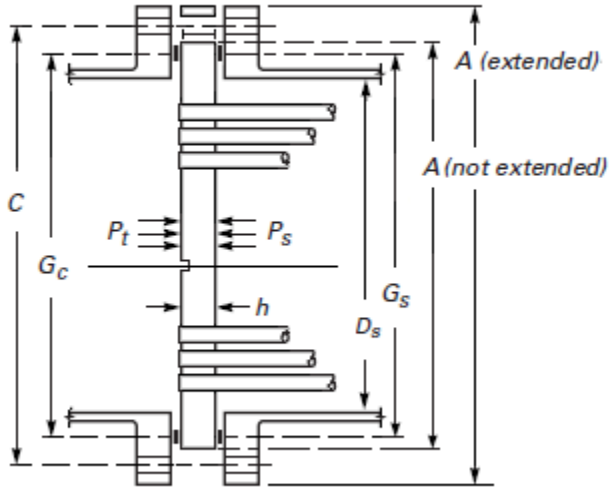
Tubesheet Gasket on which Side Both
 Flange Outside Diameter A 760.000 mm.
 Flange Inside Diameter B 600.000 mm.
 Flange Face Outside Diameter Fod 653.000 mm.
 Flange Face Inside Diameter Fid 600.000 mm.
 Gasket Outside Diameter Go 650.000 mm.
 Gasket Inside Diameter Gi 624.000 mm.
 Small end Hub thk. g0 12.0000 mm.
 Large end Hub thk. g1 18.0000 mm.
 Gasket Factor, m 3.00
 Gasket Design Seating Stress y 69.00 N./mm²
 Flange Facing Sketch Code Sketch 1b
 Column for Gasket Seating Code Column II
 Gasket Thickness tg 4.5000 mm.
 Full face Gasket Flange Option Program Selects

Bolting Information:

Diameter of Bolt Circle C 705.000 mm.
 Nominal Bolt Diameter dB 22.2250 mm.
 Type of Thread Series TEMA Thread Series
 Number of Bolts n 20
 Bolt Material SA-193 B7
 Bolt Allowable Stress At Temperature Sb 172.38 N./mm²
 Bolt Allowable Stress At Ambient Sa 172.38 N./mm²
 Weld between Flange and Shell/Channel 0.0000 mm.
 Alternate Flange Operating Bolt Load, Wm1 88967.88 Kgf
 Alternate Flange Seating Bolt Load, Wm2 90480.76 Kgf
 Alternate Flange Design Bolt Load, W 92754.98 Kgf

Tubesheet Integral with None
 Tubesheet Extended as Flange No

Tubesheet Gasketed With Shell and Channel



Configuration d:

ASME TubeSheet Results per Part UHX, 2019

Elasticity/Expansion Material Properties :

Shell - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	120.0 °C	0.19691E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Channel - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	85.0 °C	0.19904E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tubes - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Tubsht. Design Temp.	120.0 °C	0.19691E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

TubeSheet - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	120.0 °C	0.19691E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tube Required Thickness under Internal Pressure (Tubeside pressure):

Thickness Due to Internal Pressure:

$$= (P*(D/2-CAE)) / (S*E+0.4*P) \text{ per Appendix 1-1 (a) (1)}$$

$$= (6.8*(19/2-0) / (118*1+0.4*6.8))$$

$$= 0.0548 + 0.0000 = 0.0548 \text{ mm.}$$

Tube Required Thickness under External Pressure (Shellside pressure) :

External Pressure Chart	CS-2	at	120.00 °C
Elastic Modulus for Material			199943392.00 KPa.

Results for Max. Allowable External Pressure (Emawp):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
1.6500	19.05	2523.51	11.55	50.0000	0.0082522	122.73
EMAWP = (4*B) / (3*(D/T)) = (4 *123) / (3 *11.5) = 142 bars						

Results for Req'd Thickness for Ext. Pressure (Tca):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
0.4699	19.05	2523.51	40.54	50.0000	0.0006692	66.90

EMAWP = (4*B)/(3*(D/T)) = (4 *66.9)/(3 *40.5) = 22 bars

Summary of Tube Required Thickness Results:

Total Required Thickness including Corrosion all.	0.4699	mm.
Allowable Internal Pressure at Corroded thickness	219.44	bars
Required Internal Design Pressure	6.80	bars
Allowable External Pressure at Corroded thickness	141.73	bars
Required External Design Pressure	22.00	bars
Required Thickness due to Shell Side pressure	0.4699	mm.

Detailed Results for load Case D3 un-corr. (Psd,max + Ptd,max)

Intermediate Calculations For Gasketed Tubesheets:

ASME Code, Section VIII Division 1, 2019

Gasket Contact Width,	N = (Goc-Gic) / 2	13.000	mm.
Basic Gasket Width,	b0 = N / 2.0	6.500	mm.
Effective Gasket Width,	b = SQRT(b0) * 2.5	6.422	mm.
Gasket Reaction Diameter,	G = Go-2.0*b	637.156	mm.

Flange Design Bolt Load, Seating Condition	W :	92754.98	Kgf
Flange Design Bolt Load, Operating Condition	Wm1:	88836.14	Kgf

Results for ASME U-tube Tubesheet Calculations for Configuration d,
 Per Edition 2019, Original Thickness :

Minimum Required Thickness for Shear [HreqS]:

$$= 1/(4 * \mu) * (Do/(\min(0.8*S, 0.533*Sy))) [Ps - Pt] + Cats + Catc$$

$$= 1/(4 * 0.2) * (586/110) [22 - 6.8] + 0$$

$$= 10.0934 \text{ mm.}$$

UHX-12.5.1 Step 1:

Compute the Equivalent Outer Tube Limit Circle Diameter [Do]:

$$= 2 * ro + dt$$

$$= 2 * 284 + 19 = 586 \text{ mm.}$$

Determine the Basic Ligament Efficiency for Shear [mu]:

$$= (p - dt) / p$$

$$= (23.8 - 19) / 23.8 = 0.2$$

UHX-12.5.2 Step 2:

Compute the Ratio [Rhos]:

$$= Gs / Do \text{ (Configurations d, e, f)}$$

$$= 637/586 = 1.09$$

Compute the Ratio [Rhoc]:

$$= Gc / Do \text{ (Configurations d)}$$

$$= 637/586 = 1.09$$

Moment on Tubesheet due to Pressures (Ps, Pt) [Mts]:

$$= Do^2/16 * [(Rhos-1)*(Rhos^2+1)* Ps - (Rhoc-1) * (Rhoc^2+1) * Pt]$$

$$= 586^2/16 * [(1.09 - 1) * (1.09^2 + 1) * 22 -$$

$$(1.09 - 1) * (1.09^2 + 1) * 6.8]$$

$$= 62062.8672 \text{ bars*mm.}^2$$

UHX-12.5.3 Step 3, Determination of Effective Elastic Properties :

Compute the Ratio [rho]:

$$= l_{tx}/h = 57/60 = 0.95 \text{ (must be } 0 \leq \rho \leq 1 \text{)}$$

Compute the Effective Tube Hole Diameter [d*]:

$$= \max(dt - 2tt*(Et/E)(St/S)(rho), dt - 2tt)$$

$$= \max(19 - 2*1.65 *(196910448/196910448)*$$

$$(118/138)*(0.95), 19 - 2*1.65)$$

$$= 16.3696 \text{ mm.}$$

Compute the Effective Tube Pitch [p*]:

$$= p / \sqrt{ 1 - 4 * \min(AL * CNV_factor, 4*Do*p) / (Pi * Do^2) }$$

$$= 23.8 / \sqrt{ 1 - 4 * \min(425 * 100, 4*586 * 23.8)$$

$$(3.141* 586^2) }$$

$$= 25.9469 \text{ mm.}$$

Compute the Effective Ligament Efficiency for Bending [mu*]:

$$= (p^* - d^*)/p^* = (25.9 - 16.4)/25.9 = 0.37$$

E*/E and nu* for Triangular pattern from Fig. UHX-11.3.

$$h/p = 2.519632 ; \mu^* = 0.369112$$

$$E^*/E = 0.372514 ; \nu^* = 0.319336 ; E^* = 73351928. \text{ KPa.}$$

Note: As h/p (2.520) is > 2, data values for h/p = 2 were used.

Skip Step 4 for Configuration d :

UHX-12.5.5 Step 5:

Diameter ratio [K]:

$$= A/Do = 650/586 = 1.11$$

Determine Coefficient [F]:

$$= (1 - \nu^*) / E^* * (E * \ln(K))$$

$$= (1 - 0.32) / 73351928 * (196910448 * \ln(1.11))$$

$$= 0.1892$$

UHX-12.5.6 Step 6:

Moment Acting on Unperforated Tubesheet Rim [M*]

$$= M_{ts} + W^* * (G_c - G_s) / (2 * \pi * Do)$$

$$= 62063 + 88968 * (637 - 637) / (2 * \pi * 586)$$

$$= 62062.8672 \text{ bars*mm.}^2$$

Note: W* is the maximum of the bolt loads between the shell and channel sides.

UHX-12.5.7 Step 7:

Maximum Bending Moment acting on Periphery of Tubesheet [Mp]:

$$= ((M^*) - Do^2/32 * F * (Ps - Pt)) / (1 + F)$$

$$= ((62063) - 586^2/32 * 0.19 * (22 - 6.8)) / (1 + 0.19)$$

$$= 26235.0996 \text{ bars*mm.}^2$$

Maximum Bending Moment acting on Center of Tubesheet [Mo]:

$$= M_p + Do^2/64 * (3 + r\nu^*) (Ps - Pt)$$

$$= 26235 + 586^2/64 * (3 + 0.32) (22 - 6.8)$$

$$= 297013.3750 \text{ bars*mm.}^2$$

Maximum Bending Moment acting on Tubesheet [M]:

$$= \text{Max}(\text{abs}(M_p), \text{abs}(M_o))$$

$$= \text{Max}(\text{abs}(26235), \text{abs}(297013))$$

$$= 297013.3750 \text{ bars*mm.}^2$$

UHX-12.5.8 Step 8:

FileName : Chiller-Rev.01 -----

ASME TS Calc: DESIGN

Case: 1 12:29p May 21,2024

Tubesheet Bending Stress at Original Thickness:

$$= 6 * M / ((\mu*) * (h - hg')^2)$$

$$= 6 * 297013 / ((0.37) * (60 - 5)^2)$$

$$= 159.6132 \text{ N./mm}^2$$

The Allowable Tubesheet Bending Stress [SigmaAll]:

$$= 2 * S = 2 * 138 = 276 \text{ N./mm}^2$$

Tubesheet Bending Stress at Final Thickness [Sigma]:

$$= 6 * M / ((\mu*) * (h - hg')^2)$$

$$= 6 * 297361 / ((0.37) * (46.6 - 5)^2)$$

$$= 275.7786 \text{ N./mm}^2$$

Required Tubesheet Thickness, for Bending Stress [HreqB]:

$$= H + CATS + CATC = 46.6 + 0 + 0 = 46.6 \text{ mm.}$$

Required Tubesheet Thickness for Given Loadings (includes CA) [Hreq]:

$$= \text{Max}(HreqB, HreqS) = \text{Max}(46.6, 10.1) = 46.6 \text{ mm.}$$

UHX-12.5.9 Step 9:

$$\text{abs}(Ps - Pt) = \text{abs}(22 - 6.8) = 15.2 \text{ bars}$$

Shear Stress check [Tau_limit]:

$$= 3.2 * S * \mu * h/Do$$

$$= 3.2 * 138 * 0.2 * 60/586 = 90.4 \text{ bars}$$

Average Shear Stress at the Outer Edge of Perforated Region [Tau]:

$$= 1 / (4 * \mu) * (Do/h) * [Ps - Pt]$$

$$= 1 / (4 * 0.2) * (586/60) * [22 - 6.8] \text{ N./mm}^2$$

$$= 18.56 \text{ N./mm}^2$$

Note: Analysis Completed for Tubesheet Configuration d.

Stress/Force summary for loadcase D3 un-corr. (Psd,max + Ptd,max):

Stress Description	Actual	Allowable	Pass/Fail
Tubesheet bend. stress	159.6 <=	275.8 N./mm ²	Ok
Tubesheet shear stress	18.6 <=	110.3 N./mm ²	Ok

Thickness results for loadcase D3 un-corr. (Psd,max + Ptd,max):

Thickness (mm.)	Required	Actual	P/F
Tubesheet Thickness :	46.562	60.000	Ok

U-Tube Tubesheet results per ASME UHX-12 2019

Results for 6 Load Cases:

Case#	--Reqd. Thk. + CA		---- Tubesheet Stresses				Case Type	Pass/Fail
	Tbsht	Extnsn	Bend	Allwd	Shear	Allwd		
D1uc	32.806	0.962	71	276	8	110	Ps+Pt D1	Ok
D2uc	55.009	3.111	231	276	27	110	Ps+Pt D2	Ok
D3uc	46.562	2.150	160	276	19	110	Ps+Pt D3	Ok
D1c	35.858	0.962	79	276	9	110	Ps+Pt-c D1	Ok
D2c	58.003	3.111	255	276	30	110	Ps+Pt-c D2	Ok
D3c	49.579	2.150	176	276	21	110	Ps+Pt-c D3	Ok
Max:	58.0031	3.111	mm.	0.925		0.271	(Str. Ratio)	

Load Case Definitions:

[Ps & Pt]:
 Shell-side and Tube-side Design or Operating Pressures
 derived from Psd,min Ptd,max, Psox,min, Ptox,max etc. per the
 Load Case Tables

[c]:
 With or Without Corrosion Allowance

[D1, D2, D3]:
 Design Load Cases using the Maximum and Minimum Design Pressures

[D4]:
 Design Load Case using the Minimum (Vacuum) Pressures (if specified)

Note:
 Because there was no net pressure differential across the tubesheet due to
 the specified pressures psd,min and ptd,min; load case D4 was not processed as
 the resulting tubesheet bending stress would be zero.

Summary of Thickness Comparisons for 6 Load Cases:

Thickness (mm.)	Required	Actual	P/F	
Tubesheet Thickness :	58.003	60.000	Ok	
Tubesheet Thickness Flanged Extension :	3.111	50.000	Ok	
Tube Thickness :	0.470	1.650	Ok	

Tubesheet MAWP used to Compute Hydrotest Pressure:

Stress / Force Condition	Tube-side MAWP	Shell-side Stress Rat.	Shell-side MAWP	Tube-side Stress Rat.
Tubesheet Bending Stress	23.795	1.000	23.794	1.000
Tubesheet Shear Stress	81.319	1.000	81.319	1.000
Tube Pressure Stress	219.436	1.000	141.727	1.000
Minimum MAWP	23.795		23.794	

Tubesheet MAPnc used to Compute Hydrotest Pressure:

Stress / Force Condition	Tube-side MAPnc	Shell-side Stress Rat.	Shell-side MAPnc	Tube-side Stress Rat.
Tubesheet Bending Stress	26.264	1.000	26.264	1.000
Tubesheet Shear Stress	90.355	1.000	90.355	1.000
Tube Pressure Stress	219.436	1.000	141.727	1.000
Minimum MAPnc	26.264		26.264	

(* All load cases were analyzed to compute the MAWP for determining the test pressure.

Tubesheet MDMT Calculations:

Note: The loading conditions from this case will be used to determine the tubesheet MDMT.

Determine the governing MDMT considering the governing condition:

Governing thickness on the shell side per figure UCS-66.3 (c):
 = tubesheet thickness/4

FileName : Chiller-Rev.01 -----

ASME TS Calc: DESIGN

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= 60/4
= 15.000 mm.

Note:

This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C

where the MDMT reduction ratio per UCS 66 (b)(1)(b) is:

= max(pt/Tubeside MAPnc, ps/Shellside MAPnc), must be <= 1
= max(6.8/26.3, 22/26.3)
= 0.838

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Input Echo, Tubesheet Number 2, Description: Tubesheet

Shell Data:

Main Shell Description: Port Barrel

Shell Maximum Design Pressure	Psd,max	22.00	bars
Shell Maximum Operating Pressure	Psox,max	22.00	bars
Shell Minimum Operating Pressure	Psox,min	0.00	bars
Shell Thickness	ts	12.0000	mm.
Shell Internal Corrosion Allowance	cas	3.0000	mm.
Shell External Corrosion Allowance	caext	0.0000	mm.
Inside Diameter of Shell	Ds	600.000	mm.
Shell Temperature for Internal Pressure	Ts	-45.00	°C
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	137.90	N./mm ²
Shell Allowable Stress at Ambient		137.90	N./mm ²

Channel Description: CH. Barrel

Channel Type:		Cylinder	
Channel Maximum Design Pressure	Ptd,max	6.80	bars
Channel Maximum Operating Pressure	Ptox,max	6.80	bars
Channel Minimum Operating Pressure	Ptox,min	0.00	bars
Channel Thickness	tc	10.0000	mm.
Channel Corrosion Allowance	cac	3.0000	mm.
Inside Diameter of Channel	Dc	600.000	mm.
Channel Design Temperature	TEMPC	-29.00	°C
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	137.90	N./mm ²
Channel Allowable Stress at Ambient		137.90	N./mm ²

Tube Data:

Number of Tube Holes	Nt	376	
Tube Wall Thickness	et	1.6500	mm.
Tube Outside Diameter	D	19.0500	mm.
Total Straight Tube Length	Lt	2300.00	mm.
Straight Tube Length (bet. inner tubsht faces) L		2240.00	mm.
Design Temperature of the Tubes		-45.00	°C
Tube Material		SA-334 6	
Tube Material UNS Number		K03006	
Is this a Welded Tube		No	
Tube Material Specification used	Smls. & wld. tube		
Tube Allowable Stress at Temperature		117.90	N./mm ²
Tube Allowable Stress At Ambient		117.90	N./mm ²
Tube Yield Stress At design Temperature	Syt	241.32	N./mm ²
Tube Pitch (Center to Center Spacing)	P	23.8130	mm.
Tube Layout Pattern		Triangular	
Radius to Outermost Tube Hole Center	ro	283.510	mm.
Largest Center-to-Center Tube Distance	Ul	52.3870	mm.
Length of Expanded Portion of Tube	ltx	57.0000	mm.
Tube-side pass partition groove depth	hg	5.0000	mm.

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ASME TS Calc: MDMT

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Tubesheet Data:

Tubesheet TYPE: U-tube, Gasketed both Sides, Conf. d

Tubesheet Design Metal Temperature	T	-45.00	°C
Tubesheet Material		SA-350 LF2	
Tubesheet Material UNS Number		K03011	
Tubesheet Allowable Stress at Temperature	S	223.40	N./mm ²
Tubesheet Allowable Stress at Ambient	Tt	223.40	N./mm ²
Thickness of Tubesheet	h	60.0000	mm.
Tubesheet Corr. Allowance (Shell side)	Cats	3.0000	mm.
Tubesheet Corr. Allowance (Channel side)	Catc	3.0000	mm.
Tubesheet Outside Diameter	A	650.000	mm.
Dimension G for the Channel Side	Gc	637.151	mm.
Area of the Untubed Lanes	AL	425.5	cm ²
Junction Stress Reduction option		Increase Tubesheet thickness	
Perform Differential Pressure Design		NO	
Run Multiple Load Cases		YES	

Additional Data for Gasketed Tubesheets:

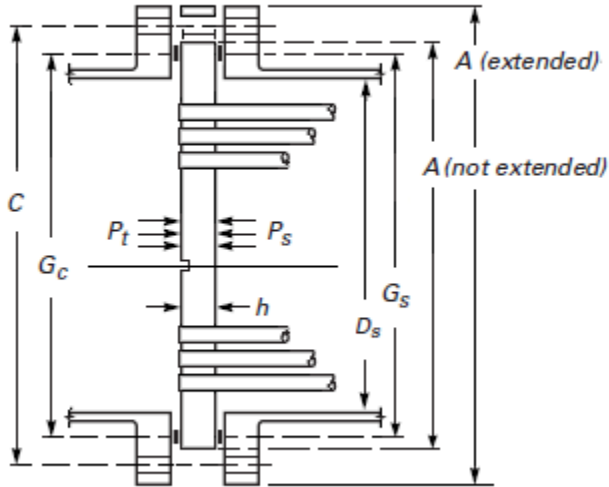
Tubesheet Gasket on which Side		Both	
Flange Outside Diameter	A	760.000	mm.
Flange Inside Diameter	B	600.000	mm.
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Small end Hub thk.	g0	12.0000	mm.
Large end Hub thk.	g1	18.0000	mm.
Gasket Factor,	m	3.00	
Gasket Design Seating Stress	y	69.00	N./mm ²
Flange Facing Sketch		Code Sketch 1b	
Column for Gasket Seating		Code Column II	
Gasket Thickness	tg	4.5000	mm.
Full face Gasket Flange Option		Program Selects	

Bolting Information:

Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	dB	22.2250	mm.
Type of Thread Series		TEMA Thread Series	
Number of Bolts	n	20	
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	172.38	N./mm ²
Bolt Allowable Stress At Ambient	Sa	172.38	N./mm ²
Weld between Flange and Shell/Channel		0.0000	mm.
Alternate Flange Operating Bolt Load, Wm1		88967.88	Kgf
Alternate Flange Seating Bolt Load, Wm2		90480.76	Kgf
Alternate Flange Design Bolt Load, W		92754.98	Kgf

Tubesheet Integral with	None
Tubesheet Extended as Flange	No

Tubesheet Gasketed With Shell and Channel



Configuration d:

ASME TubeSheet Results per Part UHX, 2019

Elasticity/Expansion Material Properties :

Shell - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	-45.0 °C	0.20704E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Channel - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	-29.0 °C	0.20599E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tubes - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Tubsht. Design Temp.	-45.0 °C	0.20704E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

TubeSheet - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	-45.0 °C	0.20704E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tube Required Thickness under Internal Pressure (Tubeside pressure):

Thickness Due to Internal Pressure:

$$= (P*(D/2-CAE)) / (S*E+0.4*P) \text{ per Appendix 1-1 (a) (1)}$$

$$= (6.8*(19/2-0) / (118*1+0.4*6.8))$$

$$= 0.0548 + 0.0000 = 0.0548 \text{ mm.}$$

Tube Required Thickness under External Pressure (Shellside pressure) :

External Pressure Chart	CS-2	at	-45.00 °C
Elastic Modulus for Material			199943392.00 KPa.

Results for Max. Allowable External Pressure (Emawp):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
1.6500	19.05	2523.51	11.55	50.0000	0.0082522	122.73
EMAWP = (4*B) / (3*(D/T)) = (4 *123) / (3 *11.5) = 142 bars						

FileName : Chiller-Rev.01 -----

ASME TS Calc: MDMT

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Results for Req'd Thickness for Ext. Pressure (Tca):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
0.4699	19.05	2523.51	40.54	50.0000	0.0006692	66.90

EMAWP = (4*B)/(3*(D/T)) = (4 *66.9)/(3 *40.5) = 22 bars

Summary of Tube Required Thickness Results:

Total Required Thickness including Corrosion all.	0.4699 mm.
Allowable Internal Pressure at Corroded thickness	219.44 bars
Required Internal Design Pressure	6.80 bars
Allowable External Pressure at Corroded thickness	141.73 bars
Required External Design Pressure	22.00 bars
Required Thickness due to Shell Side pressure	0.4699 mm.

U-Tube Tubesheet results per ASME UHX-12 2019

Results for 6 Load Cases:

Case#	--Reqd. Thk. + CA		---- Tubesheet Stresses				Case Type	Pass/Fail
	Tbsht	Extnsn	Bend	Allwd	Shear	Allwd		
D1uc	28.074	0.594	79	447	8	132	Ps+Pt D1	Ok
D2uc	46.475	1.920	256	447	27	132	Ps+Pt D2	Ok
D3uc	39.472	1.327	177	447	19	132	Ps+Pt D3	Ok
D1c	31.041	0.594	88	447	9	132	Ps+Pt-c D1	Ok
D2c	49.489	1.920	284	447	30	132	Ps+Pt-c D2	Ok
D3c	42.502	1.327	196	447	21	132	Ps+Pt-c D3	Ok
Max:	49.4892	1.920 mm.		0.636		0.226	(Str. Ratio)	

Load Case Definitions:

[Ps & Pt]:

Shell-side and Tube-side Design or Operating Pressures derived from Psd,min Ptd,max, Psox,min, Ptox,max etc. per the Load Case Tables

[c]:

With or Without Corrosion Allowance

[D1, D2, D3]:

Design Load Cases using the Maximum and Minimum Design Pressures

[D4]:

Design Load Case using the Minimum (Vacuum) Pressures (if specified)

Note:

Because there was no net pressure differential across the tubesheet due to the specified pressures psd,min and ptd,min; load case D4 was not processed as the resulting tubesheet bending stress would be zero.

Summary of Thickness Comparisons for 6 Load Cases:

Thickness (mm.)	Required	Actual	P/F	
Tubesheet Thickness :	49.489	60.000	Ok	
Tubesheet Thickness Flanged Extension :	1.920	50.000	Ok	
Tube Thickness :	0.470	1.650	Ok	

Tubesheet MAWP used to Compute Hydrotest Pressure:

Stress / Force Condition	Tube side MAWP	Shell side Stress Rat.	Shell side MAWP	Tube side Stress Rat.

ASME TS Calc: MDMT Case: 2 12:29p May 21,2024

Tubesheet Bending Stress		34.584		1.000
Tubesheet Shear Stress		97.515		1.000
Tube Pressure Stress		219.436		1.000

Minimum MAWP		34.584		34.583

Tubesheet MAPnc used to Compute Hydrotest Pressure:

Stress / Force Condition		Tubeside MAPnc		0 shellside Stress Rat.		Shellside MAPnc		0 tubeside Stress Rat.

Tubesheet Bending Stress		38.329		1.000		38.329		1.000
Tubesheet Shear Stress		108.358		1.000		108.358		1.000
Tube Pressure Stress		219.436		1.000		141.727		1.000

Minimum MAPnc		38.329				38.329		

(*) All load cases were analyzed to compute the MAWP for determining the test pressure.

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Input Echo, Tubesheet Number 3, Description: Tubesheet

Shell Data:

Main Shell Description: Port Barrel

Shell Maximum Design Pressure	Psd,max	28.60	bars
Shell Maximum Operating Pressure	Psox,max	28.60	bars
Shell Minimum Operating Pressure	Psox,min	0.00	bars
Shell Thickness	ts	12.0000	mm.
Shell Internal Corrosion Allowance	cas	3.0000	mm.
Shell External Corrosion Allowance	caext	0.0000	mm.
Inside Diameter of Shell	Ds	600.000	mm.
Shell Temperature for Internal Pressure	Ts	20.00	°C
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	137.90	N./mm ²
Shell Allowable Stress at Ambient		137.90	N./mm ²

Channel Description: CH. Barrel

Channel Type:		Cylinder	
Channel Maximum Design Pressure	Ptd,max	8.84	bars
Channel Maximum Operating Pressure	Ptox,max	8.84	bars
Channel Minimum Operating Pressure	Ptox,min	0.00	bars
Channel Thickness	tc	10.0000	mm.
Channel Corrosion Allowance	cac	3.0000	mm.
Inside Diameter of Channel	Dc	600.000	mm.
Channel Design Temperature	TEMPC	20.00	°C
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	137.90	N./mm ²
Channel Allowable Stress at Ambient		137.90	N./mm ²

Tube Data:

Number of Tube Holes	Nt	376	
Tube Wall Thickness	et	1.6500	mm.
Tube Outside Diameter	D	19.0500	mm.
Total Straight Tube Length	Lt	2300.00	mm.
Straight Tube Length (bet. inner tubsht faces) L		2240.00	mm.
Design Temperature of the Tubes		20.00	°C
Tube Material		SA-334 6	
Tube Material UNS Number		K03006	
Is this a Welded Tube		No	
Tube Material Specification used	Smls. & wld. tube		
Tube Allowable Stress at Temperature		117.90	N./mm ²
Tube Allowable Stress At Ambient		117.90	N./mm ²
Tube Yield Stress At design Temperature	Syt	241.32	N./mm ²
Tube Pitch (Center to Center Spacing)	P	23.8130	mm.
Tube Layout Pattern		Triangular	
Radius to Outermost Tube Hole Center	ro	283.510	mm.
Largest Center-to-Center Tube Distance	U1	52.3870	mm.
Length of Expanded Portion of Tube	ltx	57.0000	mm.
Tube-side pass partition groove depth	hg	5.0000	mm.

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ASME TS Calc: HYDROTEST

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Tubesheet Data:

Tubesheet TYPE: U-tube, Gasketed both Sides, Conf. d

Tubesheet Design Metal Temperature	T	20.00	°C
Tubesheet Material		SA-350 LF2	
Tubesheet Material UNS Number		K03011	
Tubesheet Allowable Stress at Temperature	S	223.40	N./mm ²
Tubesheet Allowable Stress at Ambient	Tt	223.40	N./mm ²
Thickness of Tubesheet	h	60.0000	mm.
Tubesheet Corr. Allowance (Shell side)	Cats	3.0000	mm.
Tubesheet Corr. Allowance (Channel side)	Catc	3.0000	mm.
Tubesheet Outside Diameter	A	650.000	mm.
Dimension G for the Channel Side	Gc	637.151	mm.
Area of the Untubed Lanes	AL	425.5	cm ²
Junction Stress Reduction option		Increase Tubesheet thickness	
Perform Differential Pressure Design		NO	
Run Multiple Load Cases		YES	

Additional Data for Gasketed Tubesheets:

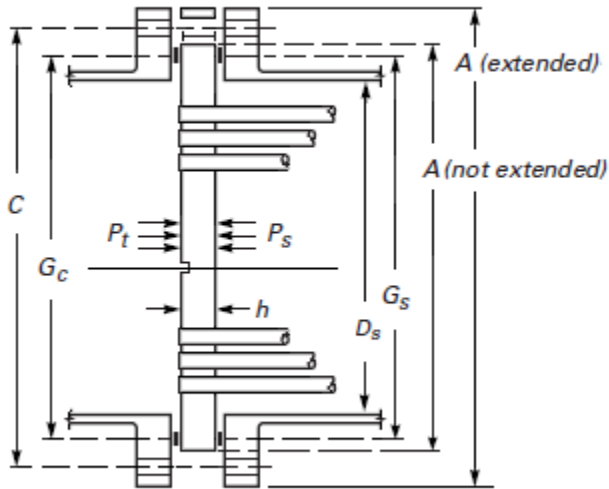
Tubesheet Gasket on which Side		Both	
Flange Outside Diameter	A	760.000	mm.
Flange Inside Diameter	B	600.000	mm.
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Small end Hub thk.	g0	12.0000	mm.
Large end Hub thk.	g1	18.0000	mm.
Gasket Factor,	m	3.00	
Gasket Design Seating Stress	y	69.00	N./mm ²
Flange Facing Sketch		Code Sketch 1b	
Column for Gasket Seating		Code Column II	
Gasket Thickness	tg	4.5000	mm.
Full face Gasket Flange Option		Program Selects	

Bolting Information:

Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	dB	22.2250	mm.
Type of Thread Series	TEMA	Thread Series	
Number of Bolts	n	20	
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	172.38	N./mm ²
Bolt Allowable Stress At Ambient	Sa	172.38	N./mm ²
Weld between Flange and Shell/Channel		0.0000	mm.
Alternate Flange Operating Bolt Load, Wm1		115486.97	Kgf
Alternate Flange Seating Bolt Load, Wm2		90480.76	Kgf
Alternate Flange Design Bolt Load, W		105258.09	Kgf

Tubesheet Integral with	None
Tubesheet Extended as Flange	No

Tubesheet Gasketed With Shell and Channel



Configuration d:

ASME TubeSheet Results per Part UHX, 2019

Elasticity/Expansion Material Properties :

Shell - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	20.0 °C	0.20277E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Channel - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	20.0 °C	0.20277E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tubes - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Tubsht. Design Temp.	20.0 °C	0.20277E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

TubeSheet - TM-1 Carbon Steels with C<= 0.3%

Elastic Mod. at Design Temperature	20.0 °C	0.20277E+09 KPa.
Elastic Mod. at Ambient Temperature	21.1 °C	0.20270E+09 KPa.

Tube Required Thickness under Internal Pressure (Tubeside pressure):

Thickness Due to Internal Pressure:

$$= (P*(D/2-CAE)) / (S*E+0.4*P) \text{ per Appendix 1-1 (a) (1)}$$

$$= (8.84*(19/2-0) / (118*1+0.4*8.84))$$

$$= 0.0712 + 0.0000 = 0.0712 \text{ mm.}$$

Tube Required Thickness under External Pressure (Shellside pressure) :

External Pressure Chart	CS-2	at	20.00 °C
Elastic Modulus for Material			199943392.00 KPa.

Results for Max. Allowable External Pressure (Emawp):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
1.6500	19.05	2523.51	11.55	50.0000	0.0082522	122.73
EMAWP = (4*B)/(3*(D/T)) = (4 *123)/(3 *11.5) = 142 bars						

FileName : Chiller-Rev.01 -----

ASME TS Calc: HYDROTEST

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Results for Req'd Thickness for Ext. Pressure (Tca):

TCA	ODCA	SLEN	D/T	L/D	Factor A	B
0.5187	19.05	2523.51	36.72	50.0000	0.0008156	78.79

EMAWP = (4*B)/(3*(D/T)) = (4 *78.8)/(3 *36.7) = 28.6 bars

Summary of Tube Required Thickness Results:

Total Required Thickness including Corrosion all.	0.5187	mm.
Allowable Internal Pressure at Corroded thickness	219.44	bars
Required Internal Design Pressure	8.84	bars
Allowable External Pressure at Corroded thickness	141.73	bars
Required External Design Pressure	28.60	bars
Required Thickness due to Shell Side pressure	0.5187	mm.

U-Tube Tubesheet results per ASME UHX-12 2019

Results for 6 Load Cases:

Case#	--Reqd. Thk. + CA		---- Tubesheet Stresses				Case Type	Pass/ Fail
	Tbsht	Extnsn	Bend	Allwd	Shear	Allwd		
D1uc	31.295	0.772	103	447	11	132	Ps+Pt D1	Ok
D2uc	52.337	2.497	333	447	35	132	Ps+Pt D2	Ok
D3uc	44.306	1.725	230	447	24	132	Ps+Pt D3	Ok
D1c	34.274	0.772	114	447	12	132	Ps+Pt-c D1	Ok
D2c	55.295	2.497	369	447	39	132	Ps+Pt-c D2	Ok
D3c	47.324	1.725	255	447	27	132	Ps+Pt-c D3	Ok
Max:	55.2952	2.497	mm.	0.827		0.293	(Str. Ratio)	

Load Case Definitions:

[Ps & Pt]:
Shell-side and Tube-side Design or Operating Pressures derived from Psd,min Ptd,max, Psox,min, Ptox,max etc. per the Load Case Tables

[c]:
With or Without Corrosion Allowance

[D1, D2, D3]:
Design Load Cases using the Maximum and Minimum Design Pressures

[D4]:
Design Load Case using the Minimum (Vacuum) Pressures (if specified)

Note:
Because there was no net pressure differential across the tubesheet due to the specified pressures psd,min and ptd,min; load case D4 was not processed as the resulting tubesheet bending stress would be zero.

Summary of Thickness Comparisons for 6 Load Cases:

Thickness (mm.)	Required	Actual	P/F	
Tubesheet Thickness :	55.295	60.000	Ok	
Tubesheet Thickness Flanged Extension :	2.497	50.000	Ok	
Tube Thickness :	0.519	1.650	Ok	

Tubesheet MAWP used to Compute Hydrotest Pressure:

Stress / Force Condition	Tube side MAWP	Shell side Stress Rat.	Shell side MAWP	Tube side Stress Rat.

ASME TS Calc: HYDROTEST Case: 3 12:29p May 21,2024

Tubesheet Bending Stress	34.584	1.000	34.583	1.000
Tubesheet Shear Stress	97.515	1.000	97.515	1.000
Tube Pressure Stress	219.436	1.000	141.727	1.000
Minimum MAWP	34.584		34.583	

Tubesheet MAPnc used to Compute Hydrotest Pressure:

Stress / Force Condition	Tubeside MAPnc	Shellside Stress Rat.	Shellside MAPnc	Tube Side Stress Rat.
Tubesheet Bending Stress	38.329	1.000	38.329	1.000
Tubesheet Shear Stress	108.358	1.000	108.358	1.000
Tube Pressure Stress	219.436	1.000	141.727	1.000
Minimum MAPnc	38.329		38.329	

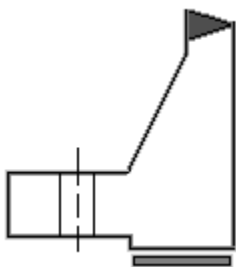
(*) All load cases were analyzed to compute the MAWP for determining the test pressure.

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Flange Input Data Values Description: CH. FLANGE :

CH. Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	8.89	bars
Design Temperature		20	°C
Internal Corrosion Allowance	ci	3.0000	mm.
External Corrosion Allowance	ce	0.0000	mm.
Use Corrosion Allowance in Thickness Calcs.		Yes	
Flange Inside Diameter	B	600.000	mm.
Flange Outside Diameter	A	760.000	mm.
Flange Thickness	t	60.0000	mm.
Thickness of Hub at Small End	go	10.0000	mm.
Thickness of Hub at Large End	gl	15.0000	mm.
Length of Hub	h	25.0000	mm.
Flange Material		SA-266 2	
Flange Material UNS number		K03506	
Flange Allowable Stress At Temperature	Sfo	223.40	N./mm ²
Flange Allowable Stress At Ambient	Sfa	223.40	N./mm ²
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	220.00	N./mm ²
Bolt Allowable Stress At Ambient	Sa	220.00	N./mm ²
Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	a	22.2250	mm.
Type of Threads		TEMA Thread Series	
Number of Bolts		20	
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Flange Facing Sketch		2, Code Sketch 1b	
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Gasket Factor	m	3.0000	
Gasket Design Seating Stress	y	69.00	N./mm ²
Column for Gasket Seating		2, Code Column II	
Gasket Thickness	tg	4.5000	mm.



Note:

The rigidity index calculation has been turned off. Please ensure the requirements of 2-14(a) are met.

ASME Code, Section VIII Division 1, 2019

Hub Small End Required Thickness due to Internal Pressure:
 $= (P*(D/2+Ca)) / (S*E-0.6*P)$ per UG-27 (c) (1)

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: CH. BODY FLANGE Flng: 11 1:02pm May 21,2024

$$= (8.89 * (600 / 2 + 3)) / (223 * 1 - 0.6 * 8.89) + Ca$$

$$= 4.2093 \text{ mm.}$$

Hub Small End Hub MAWP:

$$= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)}$$

$$= (223 * 1 * 7) / (303 + 0.6 * 7)$$

$$= 50.902 \text{ bars}$$

Corroded Flange Thickness, tc = T-ci	57.000	mm.
Corroded Flange ID, Bcor = B+2*Fcor	606.000	mm.
Corroded Large Hub, g1Cor = g1-ci	12.000	mm.
Corroded Small Hub, g0Cor = go-ci	7.000	mm.
Code R Dimension, R = ((C-Bcor)/2)-g1cor	37.500	mm.
Gasket Contact Width, N = (Go - Gi) / 2	13.000	mm.
Basic Gasket Width, bo = N / 2	6.500	mm.
Effective Gasket Width, b = Cb sqrt(bo)	6.425	mm.
Gasket Reaction Diameter, G = Go - 2 * b	637.151	mm.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$= 0.785 * G^2 * Peq$$

$$= 0.79 * 637^2 * 8.89$$

$$= 28918.035 \text{ Kgf}$$

Contact Load on Gasket Surfaces [Hp]:

$$= 2 * b * Pi * G * m * P$$

$$= 2 * 6.42 * 3.14 * 637 * 3 * 8.89$$

$$= 6998.121 \text{ Kgf}$$

Hydrostatic End Load at Flange ID [Hd]:

$$= Pi * Bcor^2 * P / 4$$

$$= 3.14 * 606^2 * 8.89 / 4$$

$$= 26159.504 \text{ Kgf}$$

Pressure Force on Flange Face [Ht]:

$$= H - Hd$$

$$= 28918 - 26160$$

$$= 2758.531 \text{ Kgf}$$

Operating Bolt Load [Wm1]:

$$= \max(H + Hp + H'p, 0)$$

$$= \max(28918 + 6998 + 0, 0)$$

$$= 35916.156 \text{ Kgf}$$

$$= 115620.258 \text{ Kgf, Mating Flange Load Governs}$$

Gasket Seating Bolt Load [Wm2]:

$$= y * b * Pi * G + yPart * bPart * lp$$

$$= 69 * 6.42 * 3.14 * 637 + 0 * 0 * 0$$

$$= 90480.758 \text{ Kgf}$$

Required Bolt Area [Am]:

$$= \text{Maximum of } Wm1/Sb, Wm2/Sa$$

$$= \text{Maximum of } 115620/220, 90481/220$$

$$= 51.539 \text{ cm}^2$$

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

$$= 2a + 6t / (m + 0.5)$$

$$= 2 * 22.2 + 6 * 57 / (3 + 0.5)$$

$$= 142.164 \text{ mm.}$$

Actual Circumferential Bolt Spacing [Bs]:

$$= C * \sin(pi / n)$$

$$= 705 * \sin(3.14 / 20)$$

$$= 110.286 \text{ mm.}$$

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

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

$$= \max(\sqrt{ 110 / (2 * 22.2 + 57) }, 1)$$

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: CH. BODY FLANGE Flng: 11 1:02pm May 21,2024

= 1.0426

Bolting Information for TEMA Imperial Thread Series (Non Mandatory):

	Minimum	Actual	Maximum
Bolt Area, cm ²	51.539	54.064	
Radial Distance between Hub and Bolts:	31.750	37.500	
Radial Distance between Bolts and Edge:	23.812	27.500	
Circ. Spacing between the Bolts:	52.400	110.286	142.164

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

$$= A_b * S_a / (y * \pi * (G_o + G_i))$$

$$= 54.1 * 220 / (69 * 3.14 * (650 + 624))$$

$$= 4.307 \text{ mm.}$$

Flange Design Bolt Load, Gasket Seating [W]:

$$= S_a * (A_m + A_b) / 2$$

$$= 220 * (51.5 + 54.1) / 2$$

$$= 118452.40 \text{ Kgf}$$

Gasket Load for the Operating Condition [HG]:

$$= W_{m1} - H$$

$$= 115620 - 28918$$

$$= 86702.22 \text{ Kgf}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

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

$$= (705 - 637) / 2$$

$$= 33.9246 \text{ mm.}$$

Distance to Face Pressure Reaction [ht]:

$$= (R + g_1 + h_g) / 2$$

$$= (37.5 + 12 + 33.9) / 2$$

$$= 41.7123 \text{ mm.}$$

Distance to End Pressure Reaction [hd]:

$$= R + (g_1 / 2)$$

$$= 37.5 + (12 / 2.0)$$

$$= 43.5000 \text{ mm.}$$

Summary of Moments for Internal Pressure: (Kg-m.)

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	26160.	43.5000	1.0426	1186.
Face Pressure, Mt	2759.	41.7123	1.0426	120.
Gasket Load, Mg	86702.	33.9246	1.0426	3067.
Gasket Seating, Matm	118452.	33.9246	1.0426	4190.
Total Moment for Operation, Mop				4373. Kg-m.
Total Moment for Gasket seating, Matm				4190. Kg-m.
Effective Hub Length, ho = sqrt(Bcor*goCor)			65.131	mm.
Hub Ratio, h/h0 = HL / H0			0.384	
Thickness Ratio, g1/g0 = (g1Cor/goCor)			1.714	

Flange Factors for Integral Flange:

Factor F		0.858
Factor V		0.302
Factor f		1.217
Factors from Figure 2-7.1		K = 1.254
T = 1.817		U = 9.566
Y = 8.705		Z = 4.491
d = 0.10120E+06 mm. ³		e = 0.0132 mm. ⁻¹

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: CH. BODY FLANGE

Flng: 11 1:02pm May 21,2024

Stress Factors ALPHA = 1.751
 BETA = 2.001 GAMMA = 0.964
 DELTA = 1.830 Lamda = 2.794

Longitudinal Hub Stress, Operating [SHo]:

$$= (f * Mop / Bcor) / (L * g1^2)$$

$$= (1.22*4373/606) / (2.79*12^2)$$

$$= 214.13 \text{ N./mm}^2$$

Longitudinal Hub Stress, Seating [SHa]:

$$= (f * Matm / Bcor) / (L * g1^2)$$

$$= (1.22*4190/606) / (2.79*12^2)$$

$$= 205.15 \text{ N./mm}^2$$

Radial Flange Stress, Operating [SRo]:

$$= (Beta * Mop / Bcor) / (L * t^2)$$

$$= (2*4373/606) / (2.79*57^2)$$

$$= 15.60 \text{ N./mm}^2$$

Radial Flange Stress, Seating [SRa]:

$$= (Beta * Matm / Bcor) / (L * t^2)$$

$$= (2*4190/606) / (2.79*57^2)$$

$$= 14.95 \text{ N./mm}^2$$

Tangential Flange Stress, Operating [STo]:

$$= (Y * Mo / (t^2 * Bcor)) - Z * SRO$$

$$= (8.7*4373 / (57^2 * 606)) - 4.49 * 15.6$$

$$= 119.53 \text{ N./mm}^2$$

Tangential Flange Stress, Seating [STa]:

$$= (y * Matm / (t^2 * Bcor)) - Z * SRA$$

$$= (8.7*4190 / (57^2 * 606)) - 4.49 * 14.9$$

$$= 114.52 \text{ N./mm}^2$$

Average Flange Stress, Operating [SAo]:

$$= (SHo + \max(SRO, STo)) / 2$$

$$= (214 + \max(15.6, 120)) / 2$$

$$= 166.83 \text{ N./mm}^2$$

Average Flange Stress, Seating [SAa]:

$$= (SHa + \max(SRA, STa)) / 2$$

$$= (205 + \max(14.9, 115)) / 2$$

$$= 159.84 \text{ N./mm}^2$$

Bolt Stress, Operating [BSo]:

$$= Wm1 / Ab$$

$$= 115620 / 54.1$$

$$= 209.73 \text{ N./mm}^2$$

Bolt Stress, Seating [BSa]:

$$= (Wm2 / Ab)$$

$$= (90481 / 54.1)$$

$$= 164.12 \text{ N./mm}^2$$

Flange Stress Analysis Results: N./mm²

	Actual	Operating Allowed	Gasket Seating Actual	Gasket Seating Allowed
Longitudinal Hub	214.13	335.10	205.15	335.10
Radial Flange	15.60	223.40	14.95	223.40
Tangential Flange	119.53	223.40	114.52	223.40
Maximum Average	166.83	223.40	159.84	223.40
Bolting	209.73	220.00	164.12	220.00

Minimum Required Flange Thickness 51.892 mm.
 Estimated M.A.W.P. (Operating) 30 bars

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: CH. BODY FLANGE Flng: 11 1:02pm May 21,2024

Estimated Finished Weight of Flange at given Thk. 84.1 kg.
Estimated Unfinished Weight of Forging at given Thk 112.6 kg.

Minimum Design Metal Temperature Results:

Thickness Ratio = 0.53, Temperature Reduction per Fig. UCS 66.1 = 29 °C

Min Metal Temp. w/o impact per UCS-66, Curve C -44 °C
Min Metal Temp. at Required thickness (UCS 66.1) -48 °C

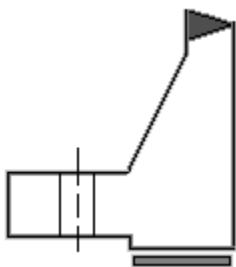
Note: UCS-66(b)(-c) was considered in the flange MDMT calculation.

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Flange Input Data Values Description: SH. FLANGE :

SH. Flange

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	28.63	bars
Design Temperature		20	°C
Internal Corrosion Allowance	ci	3.0000	mm.
External Corrosion Allowance	ce	0.0000	mm.
Use Corrosion Allowance in Thickness Calcs.		Yes	
Flange Inside Diameter	B	600.000	mm.
Flange Outside Diameter	A	760.000	mm.
Flange Thickness	t	60.0000	mm.
Thickness of Hub at Small End	go	12.0000	mm.
Thickness of Hub at Large End	gl	18.0000	mm.
Length of Hub	h	25.0000	mm.
Flange Material		SA-350 LF2	
Flange Material UNS number		K03011	
Flange Allowable Stress At Temperature	Sfo	223.40	N./mm ²
Flange Allowable Stress At Ambient	Sfa	223.40	N./mm ²
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	220.00	N./mm ²
Bolt Allowable Stress At Ambient	Sa	220.00	N./mm ²
Diameter of Bolt Circle	C	705.000	mm.
Nominal Bolt Diameter	a	22.2250	mm.
Type of Threads	TEMA Thread Series		
Number of Bolts		20	
Flange Face Outside Diameter	Fod	653.000	mm.
Flange Face Inside Diameter	Fid	600.000	mm.
Flange Facing Sketch	2, Code Sketch 1b		
Gasket Outside Diameter	Go	650.000	mm.
Gasket Inside Diameter	Gi	624.000	mm.
Gasket Factor	m	3.0000	
Gasket Design Seating Stress	y	69.00	N./mm ²
Column for Gasket Seating	2, Code Column II		
Gasket Thickness	tg	4.5000	mm.



Note:
 The rigidity index calculation has been turned off. Please ensure the requirements of 2-14(a) are met.

ASME Code, Section VIII Division 1, 2019

Hub Small End Required Thickness due to Internal Pressure:
 $= (P*(D/2+Ca)) / (S*E-0.6*P)$ per UG-27 (c) (1)

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: SH. BODY FLANGE Flng: 12 1:02pm May 21,2024

$$= (28.6 * (600/2+3)) / (223*1-0.6*28.6) + Ca$$

$$= 6.9136 \text{ mm.}$$

Hub Small End Hub MAWP:

$$= (S * E * t) / (R + 0.6 * t) \text{ per UG-27 (c) (1)}$$

$$= (223 * 1 * 9) / (303 + 0.6 * 9)$$

$$= 65.191 \text{ bars}$$

Corroded Flange Thickness, tc = T-ci	57.000	mm.
Corroded Flange ID, Bcor = B+2*Fcor	606.000	mm.
Corroded Large Hub, g1Cor = g1-ci	15.000	mm.
Corroded Small Hub, g0Cor = go-ci	9.000	mm.
Code R Dimension, R = ((C-Bcor)/2)-g1cor	34.500	mm.
Gasket Contact Width, N = (Go - Gi) / 2	13.000	mm.
Basic Gasket Width, bo = N / 2	6.500	mm.
Effective Gasket Width, b = Cb sqrt(bo)	6.425	mm.
Gasket Reaction Diameter, G = Go - 2 * b	637.151	mm.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure [H]:

$$= 0.785 * G^2 * Peq$$

$$= 0.79 * 637^2 * 28.6$$

$$= 93092.109 \text{ Kgf}$$

Contact Load on Gasket Surfaces [Hp]:

$$= 2 * b * Pi * G * m * P$$

$$= 2 * 6.42 * 3.14 * 637 * 3 * 28.6$$

$$= 22528.148 \text{ Kgf}$$

Hydrostatic End Load at Flange ID [Hd]:

$$= Pi * Bcor^2 * P / 4$$

$$= 3.14 * 606^2 * 28.6 / 4$$

$$= 84211.922 \text{ Kgf}$$

Pressure Force on Flange Face [Ht]:

$$= H - Hd$$

$$= 93092 - 84212$$

$$= 8880.184 \text{ Kgf}$$

Operating Bolt Load [Wm1]:

$$= \max(H + Hp + H'p, 0)$$

$$= \max(93092 + 22528 + 0, 0)$$

$$= 115620.258 \text{ Kgf}$$

Gasket Seating Bolt Load [Wm2]:

$$= y * b * Pi * G + yPart * bPart * lp$$

$$= 69 * 6.42 * 3.141 * 637 + 0 * 0 * 0$$

$$= 90480.758 \text{ Kgf}$$

Required Bolt Area [Am]:

$$= \text{Maximum of } Wm1/Sb, Wm2/Sa$$

$$= \text{Maximum of } 115620/220, 90481/220$$

$$= 51.539 \text{ cm}^2$$

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

$$= 2a + 6t / (m + 0.5)$$

$$= 2 * 22.2 + 6 * 57 / (3 + 0.5)$$

$$= 142.164 \text{ mm.}$$

Actual Circumferential Bolt Spacing [Bs]:

$$= C * \sin(pi / n)$$

$$= 705 * \sin(3.14/20)$$

$$= 110.286 \text{ mm.}$$

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

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

$$= \max(\sqrt{ 110 / (2 * 22.2 + 57) }, 1)$$

$$= 1.0426$$

Bolting Information for TEMA Imperial Thread Series (Non Mandatory):

	Minimum	Actual	Maximum
Bolt Area, cm ²	51.539	54.064	
Radial Distance between Hub and Bolts:	31.750	34.500	
Radial Distance between Bolts and Edge:	23.812	27.500	
Circ. Spacing between the Bolts:	52.400	110.286	142.164

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

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

$$= 54.1 * 220 / (69 * 3.14 * (650 + 624))$$

$$= 4.307 \text{ mm.}$$

Flange Design Bolt Load, Gasket Seating [W]:

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

$$= 220 * (51.5 + 54.1) / 2$$

$$= 118452.40 \text{ Kgf}$$

Gasket Load for the Operating Condition [HG]:

$$= Wm1 - H$$

$$= 115620 - 93092$$

$$= 22528.15 \text{ Kgf}$$

Moment Arm Calculations:

Distance to Gasket Load Reaction [hg]:

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

$$= (705 - 637) / 2$$

$$= 33.9246 \text{ mm.}$$

Distance to Face Pressure Reaction [ht]:

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

$$= (34.5 + 15 + 33.9) / 2$$

$$= 41.7123 \text{ mm.}$$

Distance to End Pressure Reaction [hd]:

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

$$= 34.5 + (15 / 2.0)$$

$$= 42.0000 \text{ mm.}$$

Summary of Moments for Internal Pressure: (Kg-m.)

Loading	Force	Distance	Bolt Corr	Moment
End Pressure, Md	84212.	42.0000	1.0426	3688.
Face Pressure, Mt	8880.	41.7123	1.0426	386.
Gasket Load, Mg	22528.	33.9246	1.0426	797.
Gasket Seating, Matm	118452.	33.9246	1.0426	4190.
Total Moment for Operation, Mop				4871. Kg-m.
Total Moment for Gasket seating, Matm				4190. Kg-m.

Effective Hub Length, ho = sqrt(Bcor*goCor) 73.851 mm.
 Hub Ratio, h/h0 = HL / H0 0.339
 Thickness Ratio, g1/g0 = (g1Cor/goCor) 1.667

Flange Factors for Integral Flange:

Factor F		0.868
Factor V		0.326
Factor f		1.295
Factors from Figure 2-7.1	K =	1.254
	T = 1.817	U = 9.566
	Y = 8.705	Z = 4.491
	d = 0.17564E+06 mm. ³	e = 0.0118 mm. ⁻¹
Stress Factors	ALPHA =	1.670

FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: SH. BODY FLANGE Flng: 12 1:02pm May 21,2024

BETA = 1.893 GAMMA = 0.919
 DELTA = 1.054 Lamda = 1.973

Longitudinal Hub Stress, Operating [SHo]:

$$= (f * Mop / Bcor) / (L * g1^2)$$

$$= (1.3*4871/606) / (1.97*15^2)$$

$$= 229.96 \text{ N./mm}^2$$

Longitudinal Hub Stress, Seating [SHa]:

$$= (f * Matm / Bcor) / (L * g1^2)$$

$$= (1.3*4190/606) / (1.97*15^2)$$

$$= 197.81 \text{ N./mm}^2$$

Radial Flange Stress, Operating [SRo]:

$$= (Beta * Mop / Bcor) / (L * t^2)$$

$$= (1.89*4871/606) / (1.97*57^2)$$

$$= 23.27 \text{ N./mm}^2$$

Radial Flange Stress, Seating [SRa]:

$$= (Beta * Matm / Bcor) / (L * t^2)$$

$$= (1.89*4190/606) / (1.97*57^2)$$

$$= 20.02 \text{ N./mm}^2$$

Tangential Flange Stress, Operating [STo]:

$$= (Y * Mo / (t^2 * Bcor)) - Z * SRO$$

$$= (8.7*4871 / (57^2 * 606)) - 4.49 * 23.3$$

$$= 106.66 \text{ N./mm}^2$$

Tangential Flange Stress, Seating [STa]:

$$= (y * Matm / (t^2 * Bcor)) - Z * SRA$$

$$= (8.7*4190 / (57^2 * 606)) - 4.49 * 20$$

$$= 91.75 \text{ N./mm}^2$$

Average Flange Stress, Operating [SAo]:

$$= (SHo + \max(SRo, STo)) / 2$$

$$= (230 + \max(23.3, 107)) / 2$$

$$= 168.31 \text{ N./mm}^2$$

Average Flange Stress, Seating [SAa]:

$$= (SHa + \max(SRa, STa)) / 2$$

$$= (198 + \max(20, 91.7)) / 2$$

$$= 144.78 \text{ N./mm}^2$$

Bolt Stress, Operating [BSo]:

$$= Wm1 / Ab$$

$$= 115620 / 54.1$$

$$= 209.73 \text{ N./mm}^2$$

Bolt Stress, Seating [BSa]:

$$= (Wm2 / Ab)$$

$$= (90481 / 54.1)$$

$$= 164.12 \text{ N./mm}^2$$

Flange Stress Analysis Results: N./mm²

	Actual	Operating Allowed	Gasket Seating Actual	Gasket Seating Allowed
Longitudinal Hub	229.96	335.10	197.81	335.10
Radial Flange	23.27	223.40	20.02	223.40
Tangential Flange	106.66	223.40	91.75	223.40
Maximum Average	168.31	223.40	144.78	223.40
Bolting	209.73	220.00	164.12	220.00

Minimum Required Flange Thickness 50.622 mm.
 Estimated M.A.W.P. (Operating) 30 bars
 Estimated Finished Weight of Flange at given Thk. 85.1 kg.

PV Elite 23 Licensee: #Replace this text with your company name and th

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FileName : Chiller_Hydrotest-Rev.01 -----

HYDROTEST For: SH. BODY FLANGE Flng: 12 1:02pm May 21,2024

Estimated Unfinished Weight of Forging at given Thk 112.6 kg.

Minimum Design Metal Temperature Results:

Note:

This Material was specified as being an Impact Tested (Low Temperature) Material.

Impact Test Temperature provided per Specification -46 °C

Note: UCS-66(b)(-c) was considered in the flange MDMT calculation.

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FEA for Nozzles T1 & T2 (3in.)

Tabular Results

Results were generated with the finite element program FE/Pipe®. Stress results are post-processed in accordance with the rules specified in ASME Section III and ASME Section VIII, Division 2.

Analysis Time Stamp: Wed Apr 17 11:30:56 2024.

- [Model Notes](#)
- [Load Case Report](#)
- [Solution Data](#)
- [ASME Code Stress Output Plots](#)
- [Stress Results - Notes](#)
- [ASME Overstressed Areas](#)
- [Highest Primary Stress Ratios](#)
- [Highest Secondary Stress Ratios](#)
- [Highest Fatigue Stress Ratios](#)
- [Highest Stress Ratios Per Region](#)
- [Stress Intensification Factors](#)
- [Allowable Loads](#)
- [Compressive Stress Summary](#)
- [Flexibilities](#)
- [Graphical Results](#)

Model Notes
Model Notes

Input Echo:

Model Type : Cylindrical Shell

Parent Geometry

Parent Outside Diam. : 620.000 mm.
Thickness : 7.000 mm.

Parent Properties:

Cold Allowable : 137.9 MPa
Hot Allowable : 137.9 MPa
Material DB # 1016422.
Ultimate Tensile (Amb) : 482.6 MPa
Yield Strength (Amb) : 262.0 MPa
Yield Strength (Hot) : 242.0 MPa
Elastic Modulus (Amb) : 202720.0 MPa
Poissons Ratio : 0.300
Expansion Coefficient : 0.1201E-04 mm./mm./deg.
Weight Density : 0.0000E+00 N /cu.mm. (NOT USED)

Hillside Offset Distance : 100.000 mm.

Nozzle Geometry

Nozzle Outside Diam. : 88.900 mm.
Thickness : 6.668 mm.

Length : 180.000 mm.
 RePad Width : 50.550 mm.
 RePad Thickness : 10.000 mm.
 Nozzle Tilt Angle : 0.000 deg.
 Distance from Top : 0.000 mm.
 Distance from Bottom : 0.000 mm.

Nozzle Properties

Cold Allowable : 117.9 MPa
 Hot Allowable : 117.9 MPa
 Material DB # 1009922.
 Ultimate Tensile (Amb) : 413.7 MPa
 Yield Strength (Amb) : 241.3 MPa
 Yield Strength (Hot) : 222.7 MPa
 Elastic Modulus (Amb) : 202720.0 MPa
 Poissons Ratio : 0.300
 Expansion Coefficient : 0.1201E-04 mm./mm./deg.
 Weight Density : 0.0000E+00 N /cu.mm. (NOT USED)

Design Operating Cycles : 0.
 Ambient Temperature (Deg.) : 21.10

Uniform thermal expansion produces no stress in this geometry.
 Any thermal loads will come through operating forces and moments applied through the nozzle.

Nozzle Inside Temperature : 85.00 deg.
 Nozzle Outside Temperature : 85.00 deg.
 Vessel Inside Temperature : 85.00 deg.
 Vessel Outside Temperature : 85.00 deg.

Nozzle Pressure : 0.680 MPa
 Vessel Pressure : 0.680 MPa

Operating Pressure : 0.7 MPa

The operating pressure is used for secondary and peak stress cases. The design pressure is used for primary cases. The ratio of the operating/design pressure = 1.000

User Defined Load Input Echo for the ATTACHMENT:
 Loads are given at the Nozzle/Header Junction
 Loads are defined in Global Coordinates

Forces(N) Moments (N-m)

Load Case	FX	FY	FZ	MX	MY	MZ
WEIGHT:	-3600.0	3600.0	2700.0	810.0	540.0	700.0

FEA Model Loads:

These are the actual Attachment loads applied to the FEA model.
 These are the User Defined Loads translated to the end of the nozzle and reported in global coordinates.

Forces(N) Moments (N-m)

Load Case	FX	FY	FZ	MX	MY	MZ
WEIGHT:	-3600.0	3600.0	2700.0	324.0	540.0	52.0

The "top" or "positive" end of this model is "free" in the axial and translational directions.

Stresses ARE nodally AVERAGED.

No weld dimensions have been given for the nozzle connection to the shell. This will produce conservative results for external loads and may tend to produce more realistic inside surface pressure stresses.

No pad weld dimensions have been given for the pad connection to the shell. Few correlations have been performed to investigate the sensitivity of peak stresses to this value. Reasonable lengths have been assumed.

The cylinder length or nozzle/branch location was adjusted

so that a better mesh could be generated at each end of the cylinder. The nozzle is now located 311.15 mm. down the length of the cylinder and the total cylinder length is 622.30 mm.

Vessel Centerline Vector : 1.000 0.000 0.000
Nozzle Orientation Vector : 0.000 1.000 0.000

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Load Case Report
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Load Case Report \$X

Inner and outer element temperatures are the same throughout the model. No thermal ratcheting calculations will be performed.

THE 9 LOAD CASES ANALYZED ARE:

1 WEIGHT ONLY (Wgt Only)

Weight ONLY case run to get the stress range between the installed and the operating states.

/----- Loads in Case 1
Loads due to Weight

2 SUSTAINED (Wgt+Pr)

Sustained case run to satisfy local primary membrane and bending stress limits.

/----- Loads in Case 2
Loads due to Weight
Pressure Case 1

3 OPERATING

Case run to compute the operating stresses used in secondary, peak and range calculations as needed.

/----- Loads in Case 3
Pressure Case 1
Loads from (Operating)

4 RANGE (Fatigue Calc Performed)

Case run to get the RANGE of stresses. as described in NB-3222.2, 5.5.3.2, 5.5.5.2 or 5.5.6.1.

/----- Combinations in Range Case 4
Plus Stress Results from CASE 3
Minus Stress Results from CASE 1

5 Program Generated -- Force Only

Case run to compute sif's and flexibilities.
/----- Loads in Case 5
Loads from (Axial)

6 Program Generated -- Force Only

Case run to compute sif's and flexibilities.
/----- Loads in Case 6
Loads from (Inplane)

7 Program Generated -- Force Only

Case run to compute sif's and flexibilities.
/----- Loads in Case 7

Loads from (Outplane)

8 Program Generated -- Force Only

Case run to compute sif's and flexibilities.
 /----- Loads in Case 8
 Loads from (Torsion)

9 Program Generated -- Force Only

Case run to compute sif's and flexibilities.
 /----- Loads in Case 9
 Pressure Case 1

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Solution Data
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Solution Data

Maximum Solution Row Size = 636
 Number of Nodes = 2652
 Number of Elements = 876
 Number of Solution Cases = 8

Summation of Loads per Case

Case #	FX	FY	FZ
1	-3600.	3600.	2700.
2	197092.	3038.	2700.
3	200692.	-562.	0.
4	0.	237524.	0.
5	3.	0.	1.
6	0.	0.	3.
7	0.	0.	0.
8	200692.	-562.	0.

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ASME Code Stress Output Plots
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ASME Code Stress Output Plots \$X

- 1) P1 < SPL (SUS,Membrane) Case 2
- 2) Qb < SPS (SUS,Bending) Case 2
- 3) P1+Pb+Q < SPS (SUS,Inside) Case 2
- 4) P1+Pb+Q < SPS (SUS,Outside) Case 2
- 5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
- 6) P1+Pb+Q < SPS (OPE,Inside) Case 3
- 7) P1+Pb+Q < SPS (OPE,Outside) Case 3
- 8) Membrane < User (OPE,Membrane) Case 3

- 9) Bending < User (OPE,Bending) Case 3
- 10) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 5
- 11) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 6
- 12) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 7
- 13) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 8
- 14) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 9
- 15) P1+Pb+Q < SPS (EXP,Inside) Case 4
- 16) P1+Pb+Q < SPS (EXP,Outside) Case 4
- 17) P1+Pb+Q+F < 2Sa (EXP,Inside) Case 4
- 18) P1+Pb+Q+F < 2Sa (EXP,Outside) Case 4

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Stress Results - Notes
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Stress Results - Notes

- Results in this analysis were generated using the finite element solution method.
- Using 2019 ASME Section VIII Division 2
- Use Polished Bar fatigue curve.
- Ratio between Operating and Design Pressure = 1.000000
 Range cases use operating pressure. Primary cases use design pressure.
- Assume free end displacements of attached pipe (e.g. thermal loads) are secondary loads.
- Primary bending stresses at discontinuities are treated like secondary stresses. (Pb=0)
- Use Equivalent Stress (Von Mises).
- TRIAXIAL Stress Guidelines:
 S1+S2+S3 evaluation omitted from operating stress.
 Include S1+S2+S3 evaluation in primary case evaluation.
 Bending stress NOT included for all S1+S2+S3 calculations.
- Use local tensor values for averaged and not averaged stresses.

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ASME Overstressed Areas
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ASME Overstressed Areas \$X

*** NO OVERSTRESSED NODES IN THIS MODEL ***

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Highest Primary Stress Ratios

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Highest Primary Stress Ratios \$X

Header/Pad at Junction

P1	SPL	Primary Membrane Load Case 2
42	242	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2
17%		

Branch at Junction

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
168	464	Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
36%		

Branch Transition

P1	SPL	Primary Membrane Load Case 2
30	223	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2
13%		

Pad Outer Edge Weld

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
116	504	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
23%		

Header/Pad removed from Junction

P1	SPL	Primary Membrane Load Case 2
33	242	Min Prin. Stress = -5. (25% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2
13%		

Branch removed from Junction

P1	SPL	Primary Membrane Load Case 2
32	223	Min Prin. Stress = -30. (94% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2
14%		

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Highest Secondary Stress Ratios

Highest Secondary Stress Ratios

\$X

In combination case 4 the max range stress divided by the max component stress is 1.98. The case tensor components are in some directions additive and so the combination case will have HIGHER stresses than the largest of any of the individual cases by more than 50%.

Load Case	Combined/Max (Inside)	Combined/Max (Outside)
4	1.968	1.983

Header/Pad at Junction

Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
85	504	Min Prin. Stress = -43. (81% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
16%		

Branch at Junction

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
201	464	Min Prin. Stress = -145. (85% Neg, 29% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
43%		

Branch Transition

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
36	464	Min Prin. Stress = -30. (88% Neg, 66% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
7%		

Pad Outer Edge Weld

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
116	504	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
23%		

Header/Pad removed from Junction

Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
46	504	Min Prin. Stress = -5. (25% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
9%		

Branch removed from Junction

Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
46	464	Min Prin. Stress = -42. (91% Neg, 58% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
10%		

Highest Fatigue Stress Ratios
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\$P

Highest Fatigue Stress Ratios

\$X

Header/Pad at Junction

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
115	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.005 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 2.2279E9
Allowable		"B31" Fatigue Stress Allowable = 689.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 7,592,540.
		WRC 474 99% Probability Cycles = 1,763,816.
0%		WRC 474 95% Probability Cycles = 2,448,836.
		BS5500 Allowed Cycles(Curve F) = 1,126,150.
		Membrane-to-Bending Ratio = 0.459
		Bending-to-PL+PB+Q Ratio = 0.685
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4

Branch at Junction

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
271	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.011 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 98,775.
Allowable		"B31" Fatigue Stress Allowable = 589.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 1,033,402.
		WRC 474 99% Probability Cycles = 240,069.
1%		WRC 474 95% Probability Cycles = 333,305.
		BS5500 Allowed Cycles(Curve F) = 85,077.
		Membrane-to-Bending Ratio = 0.325
		Bending-to-PL+PB+Q Ratio = 0.755
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Branch Transition

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
36	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.001 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 1.0000E11
Allowable		"B31" Fatigue Stress Allowable = 589.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 1.9594E8
		WRC 474 99% Probability Cycles = 45,519,084.
0%		WRC 474 95% Probability Cycles = 63,197,524.
		BS5500 Allowed Cycles(Curve F) = 20,613,216.
		Membrane-to-Bending Ratio = 9.953
		Bending-to-PL+PB+Q Ratio = 0.091
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Pad Outer Edge Weld

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
146	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.006 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 16,071,699.
Allowable		"B31" Fatigue Stress Allowable = 689.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 6,451,036.
		WRC 474 99% Probability Cycles = 1,498,634.
0%		WRC 474 95% Probability Cycles = 2,080,665.
		BS5500 Allowed Cycles(Curve F) = 540,636.
		Membrane-to-Bending Ratio = 0.519
		Bending-to-PL+PB+Q Ratio = 0.658
		Plot Reference:

18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Header/Pad removed from Junction

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
42	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.002 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 1.0000E11
24,846.2		"B31" Fatigue Stress Allowable = 689.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 1.1606E8
		WRC 474 99% Probability Cycles = 26,961,756.
0%		WRC 474 95% Probability Cycles = 37,433,004.
		BS5500 Allowed Cycles(Curve F) = 9,328,373.
		Membrane-to-Bending Ratio = 1.864
		Bending-to-PL+PB+Q Ratio = 0.349
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Branch removed from Junction

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
46	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.002 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 1.0000E11
24,846.2		"B31" Fatigue Stress Allowable = 589.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 86,172,024.
		WRC 474 99% Probability Cycles = 20,018,544.
0%		WRC 474 95% Probability Cycles = 27,793,228.
		BS5500 Allowed Cycles(Curve F) = 6,881,620.
		Membrane-to-Bending Ratio = 3.189
		Bending-to-PL+PB+Q Ratio = 0.239
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4

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Highest Stress Ratios Per Region

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Highest Stress Ratios Per Region

\$X

Header/Pad at Junction

Pl	SPL	Primary Membrane Load Case 2
42	242	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
17%		
Qb	SPS	Primary Bending Load Case 2
65	504	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
12%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
75	504	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
14%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
61	504	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
12%		

S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
42	552	Min Prin. Stress = -31. (67% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
		7%
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
41	504	Min Prin. Stress = -7. (29% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
		8%
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
13	504	Min Prin. Stress = -7. (29% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
		2%
Membrane	User	Component Evaluation Load Case 3
24	504	Min Prin. Stress = -7. (29% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		8) Membrane < User (OPE,Membrane) Case 3
		4%
Bending	User	Component Evaluation Load Case 3
19	504	Min Prin. Stress = -7. (29% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		9) Bending < User (OPE,Bending) Case 3
		3%
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
85	504	Min Prin. Stress = -43. (81% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
		16%
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
64	504	Min Prin. Stress = -43. (81% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
		12%
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
115	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.005 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 2.2279E9
Allowable		"B31" Fatigue Stress Allowable = 689.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 7,592,540.
		WRC 474 99% Probability Cycles = 1,763,816.
		WRC 474 95% Probability Cycles = 2,448,836.
		BS5500 Allowed Cycles(Curve F) = 1,126,150.
0%		Membrane-to-Bending Ratio = 0.459
		Bending-to-PL+PB+Q Ratio = 0.685
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
86	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.003 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 1.0000E11
Allowable		"B31" Fatigue Stress Allowable = 689.5
24,846.2		Markl Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 21,273,600.
		WRC 474 99% Probability Cycles = 4,942,045.
		WRC 474 95% Probability Cycles = 6,861,408.
		BS5500 Allowed Cycles(Curve F) = 2,635,012.
		Membrane-to-Bending Ratio = 0.330
		Bending-to-PL+PB+Q Ratio = 0.752
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4
Branch at Junction		
Pl	SPL	Primary Membrane Load Case 2
55	223	Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa	Plot Reference:

			1) Pl < SPL (SUS,Membrane) Case 2
24%			
Qb	SPS		Primary Bending Load Case 2
151	464		Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa		Plot Reference:
			2) Qb < SPS (SUS,Bending) Case 2
32%			
Pl+Pb+Q	SPS		Primary+Secondary (Inner) Load Case 2
148	464		Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa		Plot Reference:
			3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
31%			
Pl+Pb+Q	SPS		Primary+Secondary (Outer) Load Case 2
168	464		Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa		Plot Reference:
			4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
36%			
S1+S2+S3	4S		Part 5 (5.3.2) Load Case 2
83	472		Min Prin. Stress = -119. (82% Neg, 40% NegHi)
MPa	MPa		Plot Reference:
			5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
17%			
Pl+Pb+Q	SPS		Primary+Secondary (Inner) Load Case 3
33	464		Min Prin. Stress = -24. (63% Neg, 11% NegHi)
MPa	MPa		Plot Reference:
			6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
7%			
Pl+Pb+Q	SPS		Primary+Secondary (Outer) Load Case 3
39	464		Min Prin. Stress = -24. (63% Neg, 11% NegHi)
MPa	MPa		Plot Reference:
			7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
8%			
Membrane	User		Component Evaluation Load Case 3
14	464		Min Prin. Stress = -24. (63% Neg, 11% NegHi)
MPa	MPa		Plot Reference:
			8) Membrane < User (OPE,Membrane) Case 3
2%			
Bending	User		Component Evaluation Load Case 3
35	464		Min Prin. Stress = -24. (63% Neg, 11% NegHi)
MPa	MPa		Plot Reference:
			9) Bending < User (OPE,Bending) Case 3
7%			
Pl+Pb+Q	SPS		Primary+Secondary (Inner) Load Case 4
154	464		Min Prin. Stress = -145. (85% Neg, 29% NegHi)
MPa	MPa		Plot Reference:
			15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
33%			
Pl+Pb+Q	SPS		Primary+Secondary (Outer) Load Case 4
201	464		Min Prin. Stress = -145. (85% Neg, 29% NegHi)
MPa	MPa		Plot Reference:
			16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
43%			
Pl+Pb+Q+F	Damage Ratio		Primary+Secondary+Peak (Inner) Load Case 4
208	0.000	Life	Stress Concentration Factor = 1.350
MPa	0.008	Stress	Strain Concentration Factor = 1.000
			Cycles Allowed for this Stress = 276,618.
Allowable			"B31" Fatigue Stress Allowable = 589.5
24,846.2			Mark1 Fatigue Stress Allowable = 3378.5
MPa			WRC 474 Mean Cycles to Failure = 2,804,450.
			WRC 474 99% Probability Cycles = 651,499.
0%			WRC 474 95% Probability Cycles = 904,525.
			BS5500 Allowed Cycles(Curve F) = 188,899.
			Membrane-to-Bending Ratio = 0.307
			Bending-to-PL+PB+Q Ratio = 0.765
			Plot Reference:
			17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Pl+Pb+Q+F	Damage Ratio		Primary+Secondary+Peak (Outer) Load Case 4

271	0.000	Life	Stress Concentration Factor = 1.350
MPa	0.011	Stress	Strain Concentration Factor = 1.000
			Cycles Allowed for this Stress = 98,775.
Allowable			"B31" Fatigue Stress Allowable = 589.5
24,846.2			MarkI Fatigue Stress Allowable = 3378.5
MPa			WRC 474 Mean Cycles to Failure = 1,033,402.
			WRC 474 99% Probability Cycles = 240,069.
1%			WRC 474 95% Probability Cycles = 333,305.
			BS5500 Allowed Cycles(Curve F) = 85,077.
			Membrane-to-Bending Ratio = 0.325
			Bending-to-PL+PB+Q Ratio = 0.755
			Plot Reference:
			18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Branch Transition

Pl	SPL	Primary Membrane Load Case 2
30	223	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
	13%	
Qb	SPS	Primary Bending Load Case 2
9	464	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
	1%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
32	464	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
	6%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
32	464	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
	6%	
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
30	472	Min Prin. Stress = -26. (97% Neg, 78% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
	6%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
8	464	Min Prin. Stress = -3. (57% Neg, 37% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
	1%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
7	464	Min Prin. Stress = -3. (57% Neg, 37% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
	1%	
Membrane	User	Component Evaluation Load Case 3
7	464	Min Prin. Stress = -3. (57% Neg, 37% NegHi)
MPa	MPa	Plot Reference:
		8) Membrane < User (OPE,Membrane) Case 3
	1%	
Bending	User	Component Evaluation Load Case 3
6	464	Min Prin. Stress = -3. (57% Neg, 37% NegHi)
MPa	MPa	Plot Reference:
		9) Bending < User (OPE,Bending) Case 3
	1%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
31	464	Min Prin. Stress = -30. (88% Neg, 66% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
	6%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
36	464	Min Prin. Stress = -30. (88% Neg, 66% NegHi)

MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
	7%	
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
31	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.001 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 1.0000E11
24,846.2		"B31" Fatigue Stress Allowable = 589.5
MPa		Mark1 Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 3.0174E8
		WRC 474 99% Probability Cycles = 70,096,664.
0%		WRC 474 95% Probability Cycles = 97,320,392.
		BS5500 Allowed Cycles(Curve F) = 41,067,668.
		Membrane-to-Bending Ratio = 9.579
		Bending-to-PL+PB+Q Ratio = 0.095
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
36	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.001 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 1.0000E11
24,846.2		"B31" Fatigue Stress Allowable = 589.5
MPa		Mark1 Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 1.9594E8
		WRC 474 99% Probability Cycles = 45,519,084.
0%		WRC 474 95% Probability Cycles = 63,197,524.
		BS5500 Allowed Cycles(Curve F) = 20,613,216.
		Membrane-to-Bending Ratio = 9.953
		Bending-to-PL+PB+Q Ratio = 0.091
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4
Pad Outer Edge Weld		
Pl	SPL	Primary Membrane Load Case 2
41	242	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
	17%	
Qb	SPS	Primary Bending Load Case 2
81	504	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
	16%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
54	504	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
	10%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
116	504	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
	23%	
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
65	552	Min Prin. Stress = -10. (22% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
	11%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
32	504	Min Prin. Stress = -4. (6% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
	6%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
50	504	Min Prin. Stress = -4. (6% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
	10%	

Membrane 27 MPa	User 504 MPa	Component Evaluation Load Case 3 Min Prin. Stress = -4. (6% Neg, 0% NegHi) Plot Reference: 8) Membrane < User (OPE,Membrane) Case 3
5%		
Bending 26 MPa	User 504 MPa	Component Evaluation Load Case 3 Min Prin. Stress = -4. (6% Neg, 0% NegHi) Plot Reference: 9) Bending < User (OPE,Bending) Case 3
5%		
Pl+Pb+Q 56 MPa	SPS 504 MPa	Primary+Secondary (Inner) Load Case 4 Min Prin. Stress = -12. (41% Neg, 0% NegHi) Plot Reference: 15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
11%		
Pl+Pb+Q 109 MPa	SPS 504 MPa	Primary+Secondary (Outer) Load Case 4 Min Prin. Stress = -12. (41% Neg, 0% NegHi) Plot Reference: 16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
21%		
Pl+Pb+Q+F 75 MPa	Damage Ratio 0.000 Life 0.003 Stress	Primary+Secondary+Peak (Inner) Load Case 4 Stress Concentration Factor = 1.350 Strain Concentration Factor = 1.000 Cycles Allowed for this Stress = 1.0000E11 "B31" Fatigue Stress Allowable = 689.5 Markl Fatigue Stress Allowable = 3378.5 WRC 474 Mean Cycles to Failure = 48,120,244. WRC 474 99% Probability Cycles = 11,178,770. WRC 474 95% Probability Cycles = 15,520,316. BS5500 Allowed Cycles(Curve F) = 3,994,395. Membrane-to-Bending Ratio = 1.610 Bending-to-PL+PB+Q Ratio = 0.383 Plot Reference: 17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Allowable 24,846.2 MPa		
0%		
Pl+Pb+Q+F 146 MPa	Damage Ratio 0.000 Life 0.006 Stress	Primary+Secondary+Peak (Outer) Load Case 4 Stress Concentration Factor = 1.350 Strain Concentration Factor = 1.000 Cycles Allowed for this Stress = 16,071,699. "B31" Fatigue Stress Allowable = 689.5 Markl Fatigue Stress Allowable = 3378.5 WRC 474 Mean Cycles to Failure = 6,451,036. WRC 474 99% Probability Cycles = 1,498,634. WRC 474 95% Probability Cycles = 2,080,665. BS5500 Allowed Cycles(Curve F) = 540,636. Membrane-to-Bending Ratio = 0.519 Bending-to-PL+PB+Q Ratio = 0.658 Plot Reference: 18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4
Allowable 24,846.2 MPa		
0%		

Header/Pad removed from Junction

Pl 33 MPa	SPL 242 MPa	Primary Membrane Load Case 2 Min Prin. Stress = -5. (25% Neg, 0% NegHi) Plot Reference: 1) Pl < SPL (SUS,Membrane) Case 2
13%		
Qb 18 MPa	SPS 504 MPa	Primary Bending Load Case 2 Min Prin. Stress = -5. (25% Neg, 0% NegHi) Plot Reference: 2) Qb < SPS (SUS,Bending) Case 2
3%		
Pl+Pb+Q 46 MPa	SPS 504 MPa	Primary+Secondary (Inner) Load Case 2 Min Prin. Stress = -5. (25% Neg, 0% NegHi) Plot Reference: 3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
9%		
Pl+Pb+Q 39 MPa	SPS 504 MPa	Primary+Secondary (Outer) Load Case 2 Min Prin. Stress = -5. (25% Neg, 0% NegHi) Plot Reference: 4) Pl+Pb+Q < SPS (SUS,Outside) Case 2

		7%			
S1+S2+S3	4S		Part 5 (5.3.2) Load Case 2		
54	552		Min Prin. Stress = -5. (25% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2		
		9%			
Pl+Pb+Q	SPS		Primary+Secondary (Inner) Load Case 3		
31	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			6) Pl+Pb+Q < SPS (OPE,Inside) Case 3		
		6%			
Pl+Pb+Q	SPS		Primary+Secondary (Outer) Load Case 3		
34	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			7) Pl+Pb+Q < SPS (OPE,Outside) Case 3		
		6%			
Membrane	User		Component Evaluation Load Case 3		
28	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			8) Membrane < User (OPE,Membrane) Case 3		
		5%			
Bending	User		Component Evaluation Load Case 3		
8	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			9) Bending < User (OPE,Bending) Case 3		
		1%			
Pl+Pb+Q	SPS		Primary+Secondary (Inner) Load Case 4		
42	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			15) Pl+Pb+Q < SPS (EXP,Inside) Case 4		
		8%			
Pl+Pb+Q	SPS		Primary+Secondary (Outer) Load Case 4		
42	504		Min Prin. Stress = -2. (23% Neg, 0% NegHi)		
MPa	MPa		Plot Reference:		
			16) Pl+Pb+Q < SPS (EXP,Outside) Case 4		
		8%			
Pl+Pb+Q+F	Damage Ratio		Primary+Secondary+Peak (Inner) Load Case 4		
42	0.000 Life		Stress Concentration Factor = 1.000		
MPa	0.002 Stress		Strain Concentration Factor = 1.000		
			Cycles Allowed for this Stress = 1.0000E11		
Allowable			"B31" Fatigue Stress Allowable = 689.5		
24,846.2			Mark1 Fatigue Stress Allowable = 3378.5		
MPa			WRC 474 Mean Cycles to Failure = 1.1774E8		
			WRC 474 99% Probability Cycles = 27,351,872.		
0%			WRC 474 95% Probability Cycles = 37,974,628.		
			BS5500 Allowed Cycles(Curve F) = 9,493,398.		
			Membrane-to-Bending Ratio = 1.954		
			Bending-to-PL+PB+Q Ratio = 0.339		
			Plot Reference:		
			17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4		
Pl+Pb+Q+F	Damage Ratio		Primary+Secondary+Peak (Outer) Load Case 4		
42	0.000 Life		Stress Concentration Factor = 1.000		
MPa	0.002 Stress		Strain Concentration Factor = 1.000		
			Cycles Allowed for this Stress = 1.0000E11		
Allowable			"B31" Fatigue Stress Allowable = 689.5		
24,846.2			Mark1 Fatigue Stress Allowable = 3378.5		
MPa			WRC 474 Mean Cycles to Failure = 1.1606E8		
			WRC 474 99% Probability Cycles = 26,961,756.		
0%			WRC 474 95% Probability Cycles = 37,433,004.		
			BS5500 Allowed Cycles(Curve F) = 9,328,373.		
			Membrane-to-Bending Ratio = 1.864		
			Bending-to-PL+PB+Q Ratio = 0.349		
			Plot Reference:		
			18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4		
Branch removed from Junction					
Pl	SPL		Primary Membrane Load Case 2		
32	223		Min Prin. Stress = -30. (94% Neg, 40% NegHi)		

MPa	MPa	Plot Reference:
14%		1) Pl < SPL (SUS,Membrane) Case 2
Qb	SPS	Primary Bending Load Case 2
11	464	Min Prin. Stress = -30. (94% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
2%		2) Qb < SPS (SUS,Bending) Case 2
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
40	464	Min Prin. Stress = -30. (94% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
8%		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
32	464	Min Prin. Stress = -30. (94% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
6%		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
47	472	Min Prin. Stress = -30. (94% Neg, 40% NegHi)
MPa	MPa	Plot Reference:
9%		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
8	464	Min Prin. Stress = -6. (66% Neg, 8% NegHi)
MPa	MPa	Plot Reference:
1%		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
8	464	Min Prin. Stress = -6. (66% Neg, 8% NegHi)
MPa	MPa	Plot Reference:
1%		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
Membrane	User	Component Evaluation Load Case 3
7	464	Min Prin. Stress = -6. (66% Neg, 8% NegHi)
MPa	MPa	Plot Reference:
1%		8) Membrane < User (OPE,Membrane) Case 3
Bending	User	Component Evaluation Load Case 3
2	464	Min Prin. Stress = -6. (66% Neg, 8% NegHi)
MPa	MPa	Plot Reference:
0%		9) Bending < User (OPE,Bending) Case 3
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
46	464	Min Prin. Stress = -42. (91% Neg, 58% NegHi)
MPa	MPa	Plot Reference:
10%		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
37	464	Min Prin. Stress = -42. (91% Neg, 58% NegHi)
MPa	MPa	Plot Reference:
8%		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
46	0.000 Life	Stress Concentration Factor = 1.000
MPa	0.002 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 1.0000E11
24,846.2		"B31" Fatigue Stress Allowable = 589.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
0%		WRC 474 Mean Cycles to Failure = 86,172,024.
		WRC 474 99% Probability Cycles = 20,018,544.
		WRC 474 95% Probability Cycles = 27,793,228.
		BS5500 Allowed Cycles(Curve F) = 6,881,620.
		Membrane-to-Bending Ratio = 3.189
		Bending-to-PL+PB+Q Ratio = 0.239
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4

Pl+Pb+Q+F Damage Ratio Primary+Secondary+Peak (Outer) Load Case 4
 37 0.000 Life Stress Concentration Factor = 1.000
 MPa 0.001 Stress Strain Concentration Factor = 1.000
 Cycles Allowed for this Stress = 1.0000E11
 Allowable "B31" Fatigue Stress Allowable = 589.5
 24,846.2 Markl Fatigue Stress Allowable = 3378.5
 MPa WRC 474 Mean Cycles to Failure = 1.7048E8
 WRC 474 99% Probability Cycles = 39,603,136.
 0% WRC 474 95% Probability Cycles = 54,983,972.
 BS5500 Allowed Cycles(Curve F) = 16,564,323.
 Membrane-to-Bending Ratio = 25.559
 Bending-to-PL+PB+Q Ratio = 0.038
 Plot Reference:
 18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

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Stress Intensification Factors

FEPIPE Version 15.0 Jobname: NOZZLE \$P
 Released Jan. 2021 11:30am APR 17,2024

Stress Intensification Factors \$X

Branch/Nozzle Sif Summary

	Peak	Primary	Secondary	SSI
Axial :	6.168	4.569	9.137	1.520
Inplane :	2.465	1.826	3.652	1.350
Outplane:	3.291	2.438	4.875	1.315
Torsion :	1.064	1.060	1.576	1.032
Pressure:	1.130	0.920	1.674	1.029

The above stress intensification factors are to be used in a beam-type analysis of the piping system. Inplane, Outplane and Torsional sif's should be used with the matching branch pipe whose diameter and thickness is given below. The axial sif should be used to intensify the axial stress in the branch pipe calculated by F/A. The pressure sif should be used to intensify the nominal pressure stress in the PARENT or HEADER, calculated from PDo/2T. B31 calculations use mean diameters and Section VIII calculations use outside diameters. SSIs are based on peak stress factors and correlated test results.

Pipe OD : 88.900 mm.
 Pipe Thk: 6.668 mm.
 Z approx: 35411.109 cu.mm.
 Z exact : 32970.617 cu.mm.

(SSI = SIF^x)	Axial	Inpl	Outpl	Tors	Pres
SIF/SSI Exponents:	0.835	0.667	0.748	0.939	-0.679

SIF/SSI exponent based on relationship between primary and peak stress factors from the finite element analysis.

B31.3 Branch Pressure i-factor = 16.232
 Header Pressure i-factor = 2.286

The B31.3 pressure i-factors should be used with with F/A, where F is the axial force due to pressure, and A is the area of the pipe wall. This is equivalent to finding the pressure stress from (ip) (PD/4T).

B31.3 (Branch)		
Peak Stress Sif	0.000	Axial
	3.491	Inplane
	4.337	Outplane
	1.000	Torsional
B31.1 (Branch)		
Peak Stress Sif	0.000	Axial
	4.337	Inplane

Compressive Stress Summary (MPa)

\$X

Nomenclature:

Min Stress - Compressive Membrane and Bending Stress
Pts in Region - No. of nodes in the model region
>5% Compression - 5% or more of Compressive Stress Limit
>50% Compression - 50% or more of Compressive Stress Limit

Compressive Stress Limit = -0.55 Min(Sy, kEt/R), Section
slenderness ratio (elastic buckling) not considered.

#	Load Type	Case	Min Stress	Pts in Region	>5% Compression and Bending	>50% Compression and Bending	Region
1	SUSTAINED	2	-10.	1272	22%	0%	Pad Outer Edge Weld
2	OPERATING	3	-4.	1272	6%	0%	Pad Outer Edge Weld
3	EXPANSION	4	-12.	1272	41%	0%	Pad Outer Edge Weld
4	SUSTAINED	2	-5.	648	25%	0%	Header/Pad removed from Junction
5	OPERATING	3	-2.	648	23%	0%	Header/Pad removed from Junction
6	EXPANSION	4	-2.	648	23%	0%	Header/Pad removed from Junction
7	SUSTAINED	2	-119.	960	82%	40%	Branch at Junction
8	OPERATING	3	-24.	960	63%	11%	Branch at Junction
9	EXPANSION	4	-145.	960	85%	29%	Branch at Junction
10	SUSTAINED	2	-30.	432	94%	40%	Branch removed from Junction
11	OPERATING	3	-6.	432	66%	8%	Branch removed from Junction
12	EXPANSION	4	-42.	432	91%	58%	Branch removed from Junction
13	SUSTAINED	2	-26.	1104	97%	78%	Branch Transition
14	OPERATING	3	-3.	1104	57%	37%	Branch Transition
15	EXPANSION	4	-30.	1104	88%	66%	Branch Transition
16	SUSTAINED	2	-31.	576	67%	0%	Header/Pad at Junction
17	OPERATING	3	-7.	576	29%	0%	Header/Pad at Junction
18	EXPANSION	4	-43.	576	81%	0%	Header/Pad at Junction

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Flexibilities

Flexibilities

\$X

The following stiffnesses should be used in a piping, "beam-type" analysis of the intersection. The stiffnesses should be inserted at the surface of the branch/header or nozzle/vessel junction. The general characteristics used for the branch pipe should be:

Outside Diameter = 88.900 mm.
Wall Thickness = 6.668 mm.

Axial Translational Stiffness = 162521. N/mm.
Inplane Rotational Stiffness = 18359652. mm. N/deg
Outplane Rotational Stiffness = 8047242. mm. N/deg
Torsional Rotational Stiffness = 145807360. mm. N/deg

Estimated Radial Shell Displacement due to Allowable Loads

SECONDARY:

Axial (mm.) = 0.538 In-Plane (mm.) = 0.164 Out-Plane (mm.) = 0.280

PRIMARY

Axial (mm.) = 0.517 In-Plane (mm.) = 0.157 Out-Plane (mm.) = 0.269

Secondary Conservative Displacement = 0.260 mm.
Secondary Realistic Displacement = 0.450 mm.

Primary Conservative Displacement = 0.249 mm.
Primary Realistic Displacement = 0.430 mm.

Intersection Flexibility Factors for Branch/Nozzle

:

Find axial stiffness: $K = 3EI/(kd)^3$ N/mm.
Find bending and torsional stiffnesses: $K = EI/(kd)$ mm. N per radian.
The EI product is $0.29751E+12$ N mm.²
The value of (d) to use is: 82.232 mm..
The resulting bending stiffness is in units of force x length per radian.

Axial Flexibility Factor	(k) =	2.145
Inplane Flexibility Factor	(k) =	3.439
Outplane Flexibility Factor	(k) =	7.847
Torsional Flexibility Factor	(k) =	0.433

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FEA for Nozzle S1 (4in.)

Tabular Results

Results were generated with the finite element program FE/Pipe®. Stress results are post-processed in accordance with the rules specified in ASME Section III and ASME Section VIII, Division 2.

Analysis Time Stamp: Wed Apr 17 11:34:19 2024.

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Model Notes
Model Notes

Input Echo:

Model Type : Cylindrical Shell

Parent Geometry

Parent Outside Diam. : 712.000 mm.
Thickness : 9.000 mm.

Parent Properties:

Cold Allowable : 137.9 MPa
Hot Allowable : 137.9 MPa
Material DB # 1016422.
Ultimate Tensile (Amb) : 482.6 MPa
Yield Strength (Amb) : 262.0 MPa
Yield Strength (Hot) : 235.8 MPa
Elastic Modulus (Amb) : 202720.0 MPa
Poissons Ratio : 0.300
Expansion Coefficient : 0.1223E-04 mm./mm./deg.
Weight Density : 0.0000E+00 N /cu.mm. (NOT USED)

Nozzle Geometry

Nozzle Outside Diam. : 114.300 mm.
Thickness : 9.735 mm.
Length : 150.000 mm.
RePad Width : 52.850 mm.

RePad Thickness : 12.000 mm.
 Nozzle Tilt Angle : 0.000 deg.
 Distance from Top : 0.000 mm.
 Distance from Bottom : 0.000 mm.

Nozzle Properties

Cold Allowable : 117.9 MPa
 Hot Allowable : 117.9 MPa
 Material DB # 1010222.
 Ultimate Tensile (Amb) : 413.7 MPa
 Yield Strength (Amb) : 241.3 MPa
 Yield Strength (Hot) : 217.2 MPa
 Elastic Modulus (Amb) : 202720.0 MPa
 Poissons Ratio : 0.300
 Expansion Coefficient : 0.1223E-04 mm./mm./deg.
 Weight Density : 0.0000E+00 N /cu.mm. (NOT USED)

Design Operating Cycles : 0.
 Ambient Temperature (Deg.) : 21.10

Uniform thermal expansion produces no stress in this geometry.
 Any thermal loads will come through operating forces and moments applied through the nozzle.

Nozzle Inside Temperature : 120.00 deg.
 Nozzle Outside Temperature : 120.00 deg.
 Vessel Inside Temperature : 120.00 deg.
 Vessel Outside Temperature : 120.00 deg.

Nozzle Pressure : 2.200 MPa
 Vessel Pressure : 2.200 MPa

Operating Pressure : 2.2 MPa

The operating pressure is used for secondary and peak stress cases. The design pressure is used for primary cases. The ratio of the operating/design pressure = 1.000

User Defined Load Input Echo for the ATTACHMENT:
 Loads are given at the Nozzle/Header Junction
 Loads are defined in Global Coordinates

Forces(N) Moments (N-m)

Load Case	FX	FY	FZ	MX	MY	MZ
WEIGHT:	-5600.0	5600.0	4200.0	1680.0	1120.0	1460.0

FEA Model Loads:

These are the actual Attachment loads applied to the FEA model.
 These are the User Defined Loads translated to the end of the nozzle and reported in global coordinates.

Forces(N) Moments (N-m)

Load Case	FX	FY	FZ	MX	MY	MZ
WEIGHT:	-5600.0	5600.0	4200.0	1050.0	1120.0	620.0

The "top" or "positive" end of this model is "free" in the axial and translational directions.

Stresses ARE nodally AVERAGED.

No weld dimensions have been given for the nozzle connection to the shell. This will produce conservative results for external loads and may tend to produce more realistic inside surface pressure stresses.

No pad weld dimensions have been given for the pad connection to the shell. Few correlations have been performed to investigate the sensitivity of peak stresses to this value. Reasonable lengths have been assumed.

The cylinder length or nozzle/branch location was adjusted so that a better mesh could be generated at each end of the cylinder. The nozzle is now located 400.05 mm.

down the length of the cylinder and the total cylinder length is 800.10 mm.

Vessel Centerline Vector : 1.000 0.000 0.000
Nozzle Orientation Vector : 0.000 1.000 0.000

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Load Case Report
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Load Case Report \$X

Inner and outer element temperatures are the same throughout the model. No thermal ratcheting calculations will be performed.

THE 9 LOAD CASES ANALYZED ARE:

1 WEIGHT ONLY (Wgt Only)

Weight ONLY case run to get the stress range between the installed and the operating states.

/----- Loads in Case 1
Loads due to Weight

2 SUSTAINED (Wgt+Pr)

Sustained case run to satisfy local primary membrane and bending stress limits.

/----- Loads in Case 2
Loads due to Weight
Pressure Case 1

3 OPERATING

Case run to compute the operating stresses used in secondary, peak and range calculations as needed.

/----- Loads in Case 3
Pressure Case 1
Loads from (Operating)

4 RANGE (Fatigue Calc Performed)

Case run to get the RANGE of stresses. as described in NB-3222.2, 5.5.3.2, 5.5.5.2 or 5.5.6.1.

/----- Combinations in Range Case 4
Plus Stress Results from CASE 3
Minus Stress Results from CASE 1

5 Program Generated -- Force Only

Case run to compute sif's and flexibilities.

/----- Loads in Case 5
Loads from (Axial)

6 Program Generated -- Force Only

Case run to compute sif's and flexibilities.

/----- Loads in Case 6
Loads from (Inplane)

7 Program Generated -- Force Only

Case run to compute sif's and flexibilities.

/----- Loads in Case 7
Loads from (Outplane)

8 Program Generated -- Force Only
 Case run to compute sif's and flexibilities.
 /----- Loads in Case 8
 Loads from (Torsion)

9 Program Generated -- Force Only
 Case run to compute sif's and flexibilities.
 /----- Loads in Case 9
 Pressure Case 1

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Solution Data
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Solution Data

Maximum Solution Row Size = 690
 Number of Nodes = 2184
 Number of Elements = 720
 Number of Solution Cases = 8

Summation of Loads per Case

Case #	FX	FY	FZ
1	-5600.	5600.	4200.
2	848303.	2246.	4200.
3	853903.	-3354.	0.
4	0.	440969.	0.
5	0.	0.	0.
6	0.	0.	0.
7	0.	0.	0.
8	853903.	-3354.	0.

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ASME Code Stress Output Plots
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ASME Code Stress Output Plots \$X

- 1) P1 < SPL (SUS,Membrane) Case 2
- 2) Qb < SPS (SUS,Bending) Case 2
- 3) P1+Pb+Q < SPS (SUS,Inside) Case 2
- 4) P1+Pb+Q < SPS (SUS,Outside) Case 2
- 5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
- 6) P1+Pb+Q < SPS (OPE,Inside) Case 3
- 7) P1+Pb+Q < SPS (OPE,Outside) Case 3
- 8) Membrane < User (OPE,Membrane) Case 3
- 9) Bending < User (OPE,Bending) Case 3

- 10) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 5
- 11) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 6
- 12) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 7
- 13) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 8
- 14) P1+Pb+Q+F < 2Sa (SIF,Outside) Case 9
- 15) P1+Pb+Q < SPS (EXP,Inside) Case 4
- 16) P1+Pb+Q < SPS (EXP,Outside) Case 4
- 17) P1+Pb+Q+F < 2Sa (EXP,Inside) Case 4
- 18) P1+Pb+Q+F < 2Sa (EXP,Outside) Case 4

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Stress Results - Notes
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Stress Results - Notes

- Results in this analysis were generated using the finite element solution method.
- Using 2019 ASME Section VIII Division 2
- Use Polished Bar fatigue curve.
- Ratio between Operating and Design Pressure = 1.000000
 Range cases use operating pressure. Primary cases use design pressure.
- Assume free end displacements of attached pipe (e.g. thermal loads) are secondary loads.
- Primary bending stresses at discontinuities are treated like secondary stresses. (Pb=0)
- Use Equivalent Stress (Von Mises).
- TRIAXIAL Stress Guidelines:
 S1+S2+S3 evaluation omitted from operating stress.
 Include S1+S2+S3 evaluation in primary case evaluation.
 Bending stress NOT included for all S1+S2+S3 calculations.
- Use local tensor values for averaged and not averaged stresses.

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ASME Overstressed Areas
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ASME Overstressed Areas \$X

*** NO OVERSTRESSED NODES IN THIS MODEL ***

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Highest Primary Stress Ratios

FEPipe Version 15.0 Jobname: NOZZLE \$P
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Highest Primary Stress Ratios \$X

Pad/Header at Junction

P1	SPL	Primary Membrane Load Case 2
86	236	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2

36%

Branch at Junction

P1+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
218	458	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		4) P1+Pb+Q < SPS (SUS,Outside) Case 2

47%

Branch Transition

P1	SPL	Primary Membrane Load Case 2
38	217	Min Prin. Stress = -35. (87% Neg, 65% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2

17%

Pad Outer Edge Weld

P1+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
230	498	Min Prin. Stress = -2. (2% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) P1+Pb+Q < SPS (SUS,Outside) Case 2

46%

Header Outside Pad Area

P1	SPL	Primary Membrane Load Case 2
85	236	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) P1 < SPL (SUS,Membrane) Case 2

36%

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Highest Secondary Stress Ratios

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Highest Secondary Stress Ratios \$X

In combination case 4 the max range stress divided by the max component stress is 1.99. The case tensor components are in some directions additive and so the combination

case will have HIGHER stresses than the largest of any of the individual cases by more than 50%.

Load Case	Combined/Max (Inside)	Combined/Max (Outside)
4	1.990	1.984

Pad/Header at Junction

Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
164	498	Min Prin. Stress = -43. (73% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
32%		

Branch at Junction

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
249	458	Min Prin. Stress = -170. (82% Neg, 43% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
54%		

Branch Transition

Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
46	458	Min Prin. Stress = -39. (84% Neg, 64% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
10%		

Pad Outer Edge Weld

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
230	498	Min Prin. Stress = -2. (2% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
46%		

Header Outside Pad Area

Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
105	498	Min Prin. Stress = -3. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
21%		

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Highest Fatigue Stress Ratios

FEPipe Version 15.0 Jobname: NOZZLE \$P
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Highest Fatigue Stress Ratios

\$X

Pad/Header at Junction

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
221	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.009 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 208,642.
Allowable		"B31" Fatigue Stress Allowable = 689.5
24,846.2		MarkI Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 801,548.
		WRC 474 99% Probability Cycles = 186,207.

0% WRC 474 95% Probability Cycles = 258,525.
 BS5500 Allowed Cycles(Curve F) = 156,950.
 Membrane-to-Bending Ratio = 0.764
 Bending-to-PL+PB+Q Ratio = 0.567
 Plot Reference:
 17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4

Branch at Junction

Pl+Pb+Q+F Damage Ratio Primary+Secondary+Peak (Outer) Load Case 4
 336 0.000 Life Stress Concentration Factor = 1.350
 MPa 0.014 Stress Strain Concentration Factor = 1.000
 Cycles Allowed for this Stress = 39,497.
 Allowable "B31" Fatigue Stress Allowable = 589.5
 24,846.2 MPa Markl Fatigue Stress Allowable = 3378.5
 WRC 474 Mean Cycles to Failure = 417,204.
 WRC 474 99% Probability Cycles = 96,920.
 WRC 474 95% Probability Cycles = 134,562.
 BS5500 Allowed Cycles(Curve F) = 44,759.
 Membrane-to-Bending Ratio = 0.282
 Bending-to-PL+PB+Q Ratio = 0.780
 Plot Reference:
 18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Branch Transition

Pl+Pb+Q+F Damage Ratio Primary+Secondary+Peak (Inner) Load Case 4
 46 0.000 Life Stress Concentration Factor = 1.000
 MPa 0.002 Stress Strain Concentration Factor = 1.000
 Cycles Allowed for this Stress = 1.0000E11
 Allowable "B31" Fatigue Stress Allowable = 589.5
 24,846.2 MPa Markl Fatigue Stress Allowable = 3378.5
 WRC 474 Mean Cycles to Failure = 68,015,488.
 WRC 474 99% Probability Cycles = 15,800,615.
 WRC 474 95% Probability Cycles = 21,937,166.
 BS5500 Allowed Cycles(Curve F) = 7,122,108.
 Membrane-to-Bending Ratio = 7.575
 Bending-to-PL+PB+Q Ratio = 0.117
 Plot Reference:
 17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4

Pad Outer Edge Weld

Pl+Pb+Q+F Damage Ratio Primary+Secondary+Peak (Outer) Load Case 4
 280 0.000 Life Stress Concentration Factor = 1.350
 MPa 0.011 Stress Strain Concentration Factor = 1.000
 Cycles Allowed for this Stress = 85,213.
 Allowable "B31" Fatigue Stress Allowable = 689.5
 24,846.2 MPa Markl Fatigue Stress Allowable = 3378.5
 WRC 474 Mean Cycles to Failure = 707,346.
 WRC 474 99% Probability Cycles = 164,323.
 WRC 474 95% Probability Cycles = 228,142.
 BS5500 Allowed Cycles(Curve F) = 77,806.
 Membrane-to-Bending Ratio = 0.579
 Bending-to-PL+PB+Q Ratio = 0.634
 Plot Reference:
 18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Header Outside Pad Area

Pl+Pb+Q+F Damage Ratio Primary+Secondary+Peak (Outer) Load Case 4
 105 0.000 Life Stress Concentration Factor = 1.000
 MPa 0.004 Stress Strain Concentration Factor = 1.000
 Cycles Allowed for this Stress = 1.3294E10
 Allowable "B31" Fatigue Stress Allowable = 689.5
 24,846.2 MPa Markl Fatigue Stress Allowable = 3378.5
 WRC 474 Mean Cycles to Failure = 5,447,098.
 WRC 474 99% Probability Cycles = 1,265,410.
 WRC 474 95% Probability Cycles = 1,756,863.
 BS5500 Allowed Cycles(Curve F) = 591,216.
 Membrane-to-Bending Ratio = 2.602
 Bending-to-PL+PB+Q Ratio = 0.278
 Plot Reference:
 18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

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Highest Stress Ratios Per Region

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Highest Stress Ratios Per Region \$X

Pad/Header at Junction

Pl	SPL	Primary Membrane Load Case 2
86	236	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
36%		
Qb	SPS	Primary Bending Load Case 2
89	498	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
17%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
159	498	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
31%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
79	498	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
15%		
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
108	552	Min Prin. Stress = -39. (83% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
19%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
127	498	Min Prin. Stress = -19. (63% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
25%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
43	498	Min Prin. Stress = -19. (63% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
8%		
Membrane	User	Component Evaluation Load Case 3
72	498	Min Prin. Stress = -19. (63% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		8) Membrane < User (OPE,Membrane) Case 3
14%		
Bending	User	Component Evaluation Load Case 3
57	498	Min Prin. Stress = -19. (63% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		9) Bending < User (OPE,Bending) Case 3
11%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
164	498	Min Prin. Stress = -43. (73% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
32%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4

93	498	Min Prin. Stress = -43. (73% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
18%		
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
221	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.009 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 208,642.
24,846.2		"B31" Fatigue Stress Allowable = 689.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 801,548.
0%		WRC 474 99% Probability Cycles = 186,207.
		WRC 474 95% Probability Cycles = 258,525.
		BS5500 Allowed Cycles(Curve F) = 156,950.
		Membrane-to-Bending Ratio = 0.764
		Bending-to-PL+PB+Q Ratio = 0.567
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
125	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.005 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 3.6436E8
24,846.2		"B31" Fatigue Stress Allowable = 689.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 5,486,796.
0%		WRC 474 99% Probability Cycles = 1,274,632.
		WRC 474 95% Probability Cycles = 1,769,667.
		BS5500 Allowed Cycles(Curve F) = 861,803.
		Membrane-to-Bending Ratio = 0.437
		Bending-to-PL+PB+Q Ratio = 0.696
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4
Branch at Junction		
Pl	SPL	Primary Membrane Load Case 2
72	217	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
33%		
Qb	SPS	Primary Bending Load Case 2
194	458	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
42%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
182	458	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
39%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
218	458	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
47%		
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
111	472	Min Prin. Stress = -148. (81% Neg, 34% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
23%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
93	458	Min Prin. Stress = -69. (71% Neg, 33% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
20%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
112	458	Min Prin. Stress = -69. (71% Neg, 33% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
24%		

Membrane	User	Component Evaluation Load Case 3
42	458	Min Prin. Stress = -69. (71% Neg, 33% NegHi)
MPa	MPa	Plot Reference:
		8) Membrane < User (OPE,Membrane) Case 3
9%		
Bending	User	Component Evaluation Load Case 3
101	458	Min Prin. Stress = -69. (71% Neg, 33% NegHi)
MPa	MPa	Plot Reference:
		9) Bending < User (OPE,Bending) Case 3
22%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
209	458	Min Prin. Stress = -170. (82% Neg, 43% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
45%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4
249	458	Min Prin. Stress = -170. (82% Neg, 43% NegHi)
MPa	MPa	Plot Reference:
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
54%		
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4
282	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.011 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 81,702.
Allowable		"B31" Fatigue Stress Allowable = 589.5
24,846.2		MarkI Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 820,332.
		WRC 474 99% Probability Cycles = 190,570.
1%		WRC 474 95% Probability Cycles = 264,583.
		BS5500 Allowed Cycles(Curve F) = 75,851.
		Membrane-to-Bending Ratio = 0.282
		Bending-to-PL+PB+Q Ratio = 0.780
		Plot Reference:
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
336	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.014 Stress	Strain Concentration Factor = 1.000
		Cycles Allowed for this Stress = 39,497.
Allowable		"B31" Fatigue Stress Allowable = 589.5
24,846.2		MarkI Fatigue Stress Allowable = 3378.5
MPa		WRC 474 Mean Cycles to Failure = 417,204.
		WRC 474 99% Probability Cycles = 96,920.
1%		WRC 474 95% Probability Cycles = 134,562.
		BS5500 Allowed Cycles(Curve F) = 44,759.
		Membrane-to-Bending Ratio = 0.282
		Bending-to-PL+PB+Q Ratio = 0.780
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4
Branch Transition		
Pl	SPL	Primary Membrane Load Case 2
38	217	Min Prin. Stress = -35. (87% Neg, 65% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
17%		
Qb	SPS	Primary Bending Load Case 2
15	458	Min Prin. Stress = -35. (87% Neg, 65% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
3%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
39	458	Min Prin. Stress = -35. (87% Neg, 65% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
8%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
39	458	Min Prin. Stress = -35. (87% Neg, 65% NegHi)
MPa	MPa	Plot Reference:

			4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
	8%		
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2	
51	472	Min Prin. Stress = -35. (87% Neg, 65% NegHi)	
MPa	MPa	Plot Reference:	
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2	
	10%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3	
24	458	Min Prin. Stress = -14. (83% Neg, 60% NegHi)	
MPa	MPa	Plot Reference:	
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3	
	5%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3	
23	458	Min Prin. Stress = -14. (83% Neg, 60% NegHi)	
MPa	MPa	Plot Reference:	
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3	
	5%		
Membrane	User	Component Evaluation Load Case 3	
20	458	Min Prin. Stress = -14. (83% Neg, 60% NegHi)	
MPa	MPa	Plot Reference:	
		8) Membrane < User (OPE,Membrane) Case 3	
	4%		
Bending	User	Component Evaluation Load Case 3	
16	458	Min Prin. Stress = -14. (83% Neg, 60% NegHi)	
MPa	MPa	Plot Reference:	
		9) Bending < User (OPE,Bending) Case 3	
	3%		
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4	
46	458	Min Prin. Stress = -39. (84% Neg, 64% NegHi)	
MPa	MPa	Plot Reference:	
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4	
	10%		
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 4	
40	458	Min Prin. Stress = -39. (84% Neg, 64% NegHi)	
MPa	MPa	Plot Reference:	
		16) Pl+Pb+Q < SPS (EXP,Outside) Case 4	
	8%		
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Inner) Load Case 4	
46	0.000 Life	Stress Concentration Factor = 1.000	
MPa	0.002 Stress	Strain Concentration Factor = 1.000	
		Cycles Allowed for this Stress = 1.0000E11	
Allowable		"B31" Fatigue Stress Allowable = 589.5	
24,846.2		Markl Fatigue Stress Allowable = 3378.5	
MPa		WRC 474 Mean Cycles to Failure = 68,015,488.	
		WRC 474 99% Probability Cycles = 15,800,615.	
0%		WRC 474 95% Probability Cycles = 21,937,166.	
		BS5500 Allowed Cycles(Curve F) = 7,122,108.	
		Membrane-to-Bending Ratio = 7.575	
		Bending-to-PL+PB+Q Ratio = 0.117	
		Plot Reference:	
		17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4	
Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4	
40	0.000 Life	Stress Concentration Factor = 1.000	
MPa	0.002 Stress	Strain Concentration Factor = 1.000	
		Cycles Allowed for this Stress = 1.0000E11	
Allowable		"B31" Fatigue Stress Allowable = 589.5	
24,846.2		Markl Fatigue Stress Allowable = 3378.5	
MPa		WRC 474 Mean Cycles to Failure = 1.0071E8	
		WRC 474 99% Probability Cycles = 23,396,360.	
0%		WRC 474 95% Probability Cycles = 32,482,900.	
		BS5500 Allowed Cycles(Curve F) = 10,840,045.	
		Membrane-to-Bending Ratio = 14.690	
		Bending-to-PL+PB+Q Ratio = 0.064	
		Plot Reference:	
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4	
Pad Outer Edge Weld			
Pl	SPL	Primary Membrane Load Case 2	

97 MPa	236 MPa	Min Prin. Stress = -2. (2% Neg, 0% NegHi) Plot Reference: 1) Pl < SPL (SUS,Membrane) Case 2
41%		
Qb 152 MPa	SPS 498 MPa	Primary Bending Load Case 2 Min Prin. Stress = -2. (2% Neg, 0% NegHi) Plot Reference: 2) Qb < SPS (SUS,Bending) Case 2
30%		
Pl+Pb+Q 116 MPa	SPS 498 MPa	Primary+Secondary (Inner) Load Case 2 Min Prin. Stress = -2. (2% Neg, 0% NegHi) Plot Reference: 3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
23%		
Pl+Pb+Q 230 MPa	SPS 498 MPa	Primary+Secondary (Outer) Load Case 2 Min Prin. Stress = -2. (2% Neg, 0% NegHi) Plot Reference: 4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
46%		
S1+S2+S3 154 MPa	4S 552 MPa	Part 5 (5.3.2) Load Case 2 Min Prin. Stress = -2. (2% Neg, 0% NegHi) Plot Reference: 5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
27%		
Pl+Pb+Q 95 MPa	SPS 498 MPa	Primary+Secondary (Inner) Load Case 3 Min Prin. Stress = -1. (0% Neg, 0% NegHi) Plot Reference: 6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
19%		
Pl+Pb+Q 139 MPa	SPS 498 MPa	Primary+Secondary (Outer) Load Case 3 Min Prin. Stress = -1. (0% Neg, 0% NegHi) Plot Reference: 7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
27%		
Membrane 81 MPa	User 498 MPa	Component Evaluation Load Case 3 Min Prin. Stress = -1. (0% Neg, 0% NegHi) Plot Reference: 8) Membrane < User (OPE,Membrane) Case 3
16%		
Bending 73 MPa	User 498 MPa	Component Evaluation Load Case 3 Min Prin. Stress = -1. (0% Neg, 0% NegHi) Plot Reference: 9) Bending < User (OPE,Bending) Case 3
14%		
Pl+Pb+Q 126 MPa	SPS 498 MPa	Primary+Secondary (Inner) Load Case 4 Min Prin. Stress = -3. (3% Neg, 0% NegHi) Plot Reference: 15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
25%		
Pl+Pb+Q 207 MPa	SPS 498 MPa	Primary+Secondary (Outer) Load Case 4 Min Prin. Stress = -3. (3% Neg, 0% NegHi) Plot Reference: 16) Pl+Pb+Q < SPS (EXP,Outside) Case 4
41%		
Pl+Pb+Q+F 170 MPa	Damage Ratio 0.000 Life 0.007 Stress	Primary+Secondary+Peak (Inner) Load Case 4 Stress Concentration Factor = 1.350 Strain Concentration Factor = 1.000 Cycles Allowed for this Stress = 932,696. "B31" Fatigue Stress Allowable = 689.5 Markl Fatigue Stress Allowable = 3378.5 WRC 474 Mean Cycles to Failure = 3,152,230. WRC 474 99% Probability Cycles = 732,292. WRC 474 95% Probability Cycles = 1,016,695. BS5500 Allowed Cycles(Curve F) = 343,030. Membrane-to-Bending Ratio = 1.502 Bending-to-PL+PB+Q Ratio = 0.400 Plot Reference: 17) Pl+Pb+Q+F < 2Sa (EXP,Inside) Case 4
Allowable 24,846.2 MPa		
0%		

Pl+Pb+Q+F	Damage Ratio	Primary+Secondary+Peak (Outer) Load Case 4
280	0.000 Life	Stress Concentration Factor = 1.350
MPa	0.011 Stress	Strain Concentration Factor = 1.000
Allowable		Cycles Allowed for this Stress = 85,213.
24,846.2		"B31" Fatigue Stress Allowable = 689.5
MPa		MarkI Fatigue Stress Allowable = 3378.5
		WRC 474 Mean Cycles to Failure = 707,346.
		WRC 474 99% Probability Cycles = 164,323.
1%		WRC 474 95% Probability Cycles = 228,142.
		BS5500 Allowed Cycles(Curve F) = 77,806.
		Membrane-to-Bending Ratio = 0.579
		Bending-to-PL+PB+Q Ratio = 0.634
		Plot Reference:
		18) Pl+Pb+Q+F < 2Sa (EXP,Outside) Case 4

Header Outside Pad Area

Pl	SPL	Primary Membrane Load Case 2
85	236	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		1) Pl < SPL (SUS,Membrane) Case 2
	36%	
Qb	SPS	Primary Bending Load Case 2
26	498	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		2) Qb < SPS (SUS,Bending) Case 2
	5%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 2
101	498	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		3) Pl+Pb+Q < SPS (SUS,Inside) Case 2
	20%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 2
101	498	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		4) Pl+Pb+Q < SPS (SUS,Outside) Case 2
	20%	
S1+S2+S3	4S	Part 5 (5.3.2) Load Case 2
147	552	Min Prin. Stress = -4. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		5) S1+S2+S3 < 4S (SUS,S1+S2+S3) Case 2
	26%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 3
88	498	Min Prin. Stress = -2. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		6) Pl+Pb+Q < SPS (OPE,Inside) Case 3
	17%	
Pl+Pb+Q	SPS	Primary+Secondary (Outer) Load Case 3
96	498	Min Prin. Stress = -2. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		7) Pl+Pb+Q < SPS (OPE,Outside) Case 3
	19%	
Membrane	User	Component Evaluation Load Case 3
84	498	Min Prin. Stress = -2. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		8) Membrane < User (OPE,Membrane) Case 3
	16%	
Bending	User	Component Evaluation Load Case 3
21	498	Min Prin. Stress = -2. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		9) Bending < User (OPE,Bending) Case 3
	4%	
Pl+Pb+Q	SPS	Primary+Secondary (Inner) Load Case 4
100	498	Min Prin. Stress = -3. (12% Neg, 0% NegHi)
MPa	MPa	Plot Reference:
		15) Pl+Pb+Q < SPS (EXP,Inside) Case 4
	20%	

SIF/SSI Exponents: 0.842 0.660 0.754 0.338 -0.536

SIF/SSI exponent based on relationship between primary and peak stress factors from the finite element analysis.

B31.3 Branch Pressure i-factor = 15.841
Header Pressure i-factor = 2.178

The B31.3 pressure i-factors should be used with with F/A, where F is the axial force due to pressure, and A is the area of the pipe wall. This is equivalent to finding the pressure stress from (ip) (PD/4T).

B31.3 (Branch)		
Peak Stress Sif	0.000	Axial
	3.858	Inplane
	4.783	Outplane
	1.000	Torsional
B31.1 (Branch)		
Peak Stress Sif	0.000	Axial
	4.783	Inplane
	4.783	Outplane
	4.783	Torsional
WRC 330 (Branch)		
Peak Stress Sif	0.000	Axial
	4.422	Inplane
	4.783	Outplane
	4.422	Torsional

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Allowable Loads
 FEPipe Version 15.0 Jobname: NOZZLE \$P
 Released Jan. 2021 11:34am APR 17,2024

Allowable Loads \$X

SECONDARY	Maximum	Conservative	Realistic
Load Type (Range):	Individual	Simultaneous	Simultaneous
	Occuring	Occuring	Occuring
Axial Force (N)	148391.	37417.	56125.
Inplane Moment (mm. N)	9879366.	1761457.	3736615.
Outplane Moment (mm. N)	7055976.	1258056.	2668741.
Torsional Moment (mm. N)	37356572.	9419439.	14129159.
Pressure (MPa)	7.90	2.20	2.20

PRIMARY	Maximum	Conservative	Realistic
Load Type:	Individual	Simultaneous	Simultaneous
	Occuring	Occuring	Occuring
Axial Force (N)	140592.	34817.	52225.
Inplane Moment (mm. N)	9360079.	1639060.	3476972.
Outplane Moment (mm. N)	6685095.	1170639.	2483300.
Torsional Moment (mm. N)	19489534.	4828047.	7242070.
Pressure (MPa)	6.20	2.20	2.20

NOTES:

- 1) Maximum Individual Occuring Loads are the maximum allowed values of the respective loads if all other load components are zero, i.e. the listed axial force may be applied if the inplane, outplane and torsional moments, and the pressure are zero.
- 2) The Conservative Allowable Simultaneous loads are the maximum loads that can be applied simultaneously. A conservative stress combination equation is used that typically produces stresses within 50-70% of the allowable stress.
- 3) The Realistic Allowable Simultaneous loads are the

maximum loads that can be applied simultaneously. A more realistic stress combination equation is used based on experience at Paulin Research. Stresses are typically produced within 80-105% of the allowable.

- 4) Secondary allowable loads are limits for expansion and operating piping loads.
- 5) Primary allowable loads are limits for weight, primary and sustained type piping loads.
- 6) High D/T low pressure systems may be subject to instability and should be checked when compressive stresses are present.

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Compressive Stress Summary
 FEPipe Version 15.0 Jobname: NOZZLE \$P
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Compressive Stress Summary (MPa) \$X

Nomenclature:

- Min Stress - Compressive Membrane and Bending Stress
 Pts in Region - No. of nodes in the model region
 >5% Compression - 5% or more of Compressive Stress Limit
 >50% Compression - 50% or more of Compressive Stress Limit

Compressive Stress Limit = -0.55 Min(Sy, kEt/R), Section slenderness ratio (elastic buckling) not considered.

#	Load Type	Case	Min Stress	Pts in Region	Compressive Stress		Region
					>5%	>50%	
1	SUSTAINED	2	-39.	1344	83%	0%	Pad/Header at Junction
2	OPERATING	3	-19.	1344	63%	0%	Pad/Header at Junction
3	EXPANSION	4	-43.	1344	73%	0%	Pad/Header at Junction
4	SUSTAINED	2	-2.	456	2%	0%	Pad Outer Edge Weld
5	OPERATING	3	-1.	456	0%	0%	Pad Outer Edge Weld
6	EXPANSION	4	-3.	456	3%	0%	Pad Outer Edge Weld
7	SUSTAINED	2	-4.	312	12%	0%	Header Outside Pad Area
8	OPERATING	3	-2.	312	12%	0%	Header Outside Pad Area
9	EXPANSION	4	-3.	312	12%	0%	Header Outside Pad Area
10	SUSTAINED	2	-148.	1080	81%	34%	Branch at Junction
11	OPERATING	3	-69.	1080	71%	33%	Branch at Junction
12	EXPANSION	4	-170.	1080	82%	43%	Branch at Junction
13	SUSTAINED	2	-35.	648	87%	65%	Branch Transition
14	OPERATING	3	-14.	648	83%	60%	Branch Transition
15	EXPANSION	4	-39.	648	84%	64%	Branch Transition

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Flexibilities
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Flexibilities \$X

The following stiffnesses should be used in a piping, "beam-type" analysis of the intersection. The stiffnesses should be inserted at the surface of the

branch/header or nozzle/vessel junction. The general characteristics used for the branch pipe should be:

Outside Diameter = 114.300 mm.
Wall Thickness = 9.735 mm.

Axial Translational Stiffness = 190063. N /mm.
Inplane Rotational Stiffness = 35510780. mm. N /deg
Outplane Rotational Stiffness = 15159243. mm. N /deg
Torsional Rotational Stiffness = 663682560. mm. N /deg

Estimated Radial Shell Displacement due to Allowable Loads

SECONDARY:

Axial (mm.) = 0.781 In-Plane (mm.) = 0.254 Out-Plane (mm.) = 0.425

PRIMARY

Axial (mm.) = 0.740 In-Plane (mm.) = 0.241 Out-Plane (mm.) = 0.402

Secondary Conservative Displacement = 0.318 mm.

Secondary Realistic Displacement = 0.552 mm.

Primary Conservative Displacement = 0.296 mm.

Primary Realistic Displacement = 0.514 mm.

Intersection Flexibility Factors for Branch/Nozzle

:

Find axial stiffness: $K = 3EI/(kd)^3$ N /mm.

Find bending and torsional stiffnesses: $K = EI/(kd)$ mm. N per radian.

The EI product is $0.89492E+12$ N mm.²

The value of (d) to use is: 104.565 mm..

The resulting bending stiffness is in units of force x length per radian.

Axial Flexibility Factor (k) = 2.312

Inplane Flexibility Factor (k) = 4.206

Outplane Flexibility Factor (k) = 9.854

Torsional Flexibility Factor (k) = 0.225

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