Fermilab

Particle Physics Division
Mechanical Department Engineering Note

Number: MD-ENG- 079 Date: June 20, 2005

Project Internal Reference:

Project: **ILC**

Title: Vacuum Vessel Engineering Note for 1.3 GHz, β=1, Cryomodule

Author(s): Edward Chi

Reviewer(s):

Key Words:

Crymodule, Vacuum, Pressure, Operating Temperature, Vessel Shell, Stiffening Rib, Reinforcement, Flange, Weldment, Up & Downstream

Abstract Summary:

The vacuum vessel of 1.3 GHz, β=1, Cryomodule has 38" O. D, ~ 448" Length with multiple openings for different connections and instrumentation. This engineering note is presented extensive discussion, analysis and calculations for the vessel and nozzle shells, stiffening rings, multiple openings and reinforcements, different flanges, welds and others per Fermilab, ASME and other applicable codes.

Applicable Codes:

ES&H Manual Chapter 5033, Fermilab.

"Boiler & Pressure Vessel Code" ASME VIII, Div.1
Addenda July 1, 2003

"Allowable Stress Design", AISC, 9th edition

"Structural Welding Code-Steel", AWS D1.1-90

EXHIBIT A-1

Vacuum Vessel Engineering Note
(per Fermilab ES&H Manual Chapter 5033)

Prepared by	Edward Chi	Date	06-20-2005	Div/ Sec	PPD/ MD/ ME
Reviewed by		Date		Div/ Sec	
Div/ Sec Head		Date		Div/ Sec	

1. Identification and Verification of Compliance

Fill in the Fermilab Engineering Conformance Label information below:

This vessel conforms to Fermilab ES&H Manual Chapter 5033

Vessel Title	*1.3 GHz, β =1, Cryomodule Vacuum Vessel*
Vessel Number	
Vessel Drawing Number	*5520 – ME - 443027*
Internal MAWP	*1 psig*
External MAWP	*14.7 psi*
Working Temperature Range	*20* °F *100* °F
Design/ Manufacturer	*DESY*
Date of Manufacture	
Acceptance Date	

Director's signature (or designee) if vessel is for manned area and requires an exception to the provisions of this chapter.

____

Amendment No.	**Reviewed by:**	**Date:**

Laboratory location code	
Laboratory property number	
Purpose of vessel	

List all pertinent drawings

Drawing No.	Location of Original
5520 – ME – 443027 -1	
5520 – ME – 443027 -2	
5520 – ME - 443054	

2. Design Verification

Provide design calculations in the Note Appendix.
See the attached vacuum vessel engineering note #MD-Eng-079

3. System Venting Verification

Can this vessel be pressurized either internally or externally? [ ] Yes
[*X*] No

If "Yes", to what pressure? ____________

List all reliefs and settings. Provide a schematic of the relief system components and appropriate calculations or test results to prove that the vessel will not be subjected to pressures greater than 110% beyond the maximum allowable internal or external pressure.

Manufacturer	**Relief**	**Pressure Setting**	**Flow Rate**	**Size**

4. Operating Procedure Section

Is an operating procedure necessary for the safe operation of this vessel?
[ ] Yes [*X*] No *(If "Yes", it must be appended)*

Is a testing procedure necessary for the safe acceptance testing (acceptance testing) of this vessel?

[ ] Yes [*X*] No

If "Yes", the written procedure must be approved by the division head prior to testing and supplied with this Engineering Note.

5. Welding Information

Has the vessel been fabricated in a Fermilab shop? [ ] Yes [*X*] No

If "Yes," append a copy of the welding shop statement of welder qualification.

The fabrication will be performed by using qualified welder of an outside contractor.

6. Exceptional, Existing, Used and Non-Manned Area Vessels

Is this vessel or any part thereof in the above categories? [ ] Yes
[*X*] No

If "Yes" follow the Engineering Note requirements for documentation and append to note.

CONTENTS

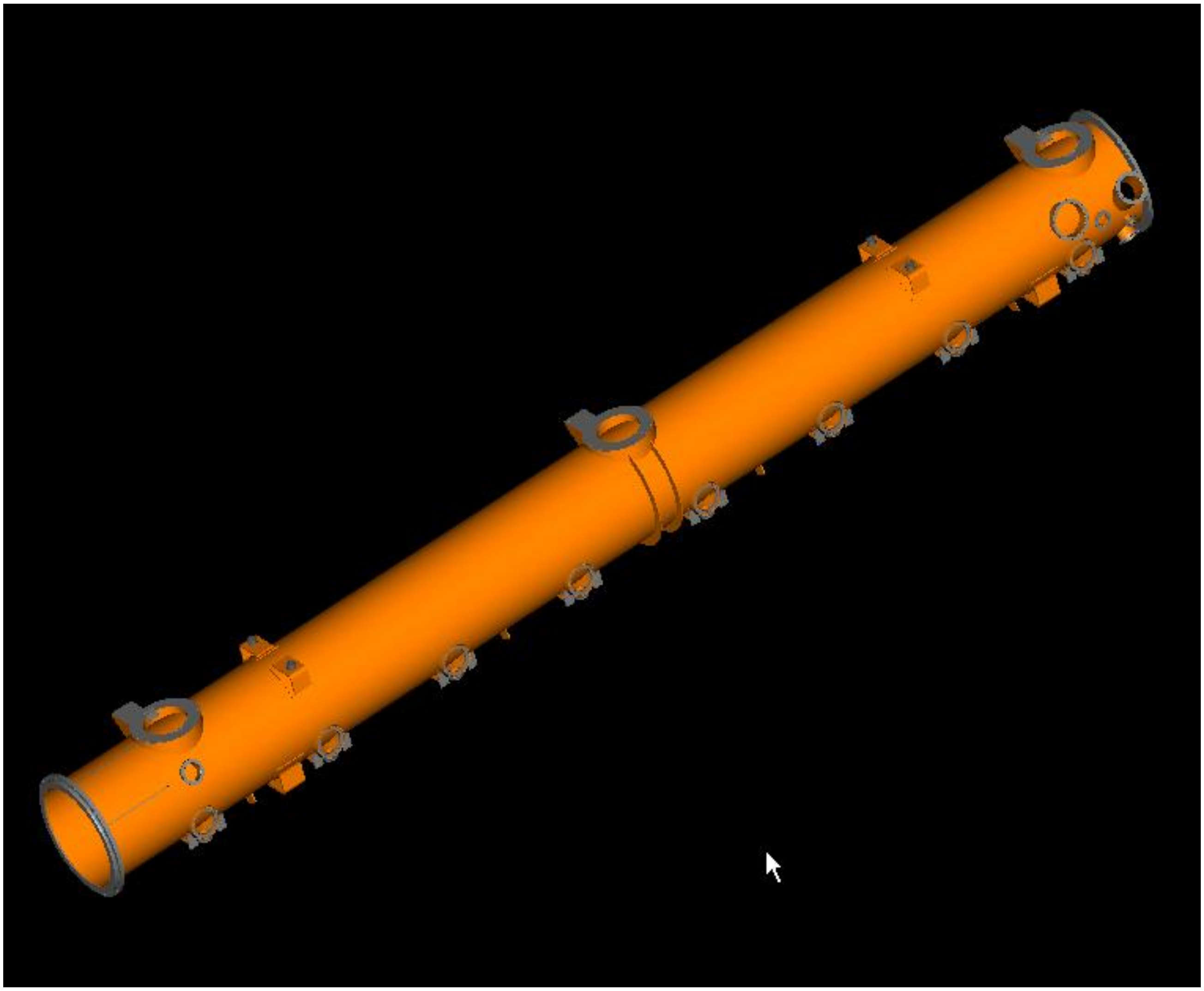

Figure i.1, An overall view of the vacuum vessel weldment of 1.3 MHz, β=1,Cryomodulue

Reference drawing: ME – 443027
Overall length = 448.23 in
W_{ws} = 7,745 lbs. the weight of the vessel shell weldment.

Assuming the vessel shell is operating under the room temperature.
Vessel shell material: SA 516-60
Materials for flanges, ports and nozzles: see respective notes and drawings.

1. Calculation for the Vacuum Vessel Cylindrical Shell Thickness

The Vacuum Vessel Cylindrical Shell:

Given:

P: 14.7 psi, the external design pressure,
D_o: 38.00 in, outside diameter of the vessel shell
D_i: 37.25 in, inside diameter of the vessel shell
t_n: The nominal vessel shell thickness.
F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_v. (See Page 1-1).
Vessel shell material: SA 516-60, carbon steel, see dwg. ME-443054
L_v: Design length of the vessel between lines of the support.
M.A: Material corrosion and other misc. allowance, ~ 1/16"

Assuming:

L_v = 228.30 in (per drawing ME-443027-01)

Find out the required vessel shell thickness t_n:

1. Try t_n = 0.375"
   L_v/D_o = 228.30" / 38.00" = 6.0079
   D_o/t_n = 38.00" / 0.375" = 101.3333

   Per section UG-28, also Fig. G and Fig. CS-2 in Subpart 3 of Section II, part D, ASME VIII, Div. 1:
   It is found out that:
   A = 0.0002
   B = 2850
   Then:
   $P_a = (4B) / (3D_o/t_n)$
   $= (4 \times 2850) \div (3 \times 101.3333)$
   $= 37.50$ psi
   where:
   P_a : The max. allowable external working pressure, psi
   A: Factor determined from Fig. G
   B: Factor determined from the applicable material chert in Fig. CS-2.

2. Try t_{r2} = 0.25
   L_v/D_o = 6.053
   D_o/t_{r2} = 38"/0.25" = 152

   Find:
   $A \approx 0.000115$
   Since the value of A falling to the left of the applicable material/temperature curve, so:

$P_{a2} = (2AE/3(Do/t_{r2})) = 15.13$ psi
So, $\mathbf{t_r = 0.250"}$

So the vessel shell thickness is required 0.25", using 3/8" as vessel shell thickness is above the shell thickness required and it is satisfactory.

2. The Permissible Out of Roundness of the Vessel Cylindrical Shell

Given:

P: 14.7 psi, the external design pressure,
D_o: 38.00 in, outside diameter of the vessel shell
D_i: 37.25 in, inside diameter of the vessel shell
t_n: 0.375", the nominal vessel shell thickness.
F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_v. (See Page 1-1).
Vessel shell material: SA 516-60, carbon steel, see dwg. ME-443054
L_v: ~230", design length of the vessel between lines of the support.

The difference between the max. dia and the mini. dia. of the vessel shell at any cross section shall not exceed:

0.01 x 38 in = 0.38 in
(per section UG-80 (b)(1) of ASME VIII, Div.1)

Per section UG-80(b)(2) and figure UG-80 of ASME VIII, Div.1, it is found out that the maximum plus or minus deviation from the true circular form, measured radially on the outside or inside of the cylindrical vessel shall not exceed the maximum permissible deviation e:

$$e \approx 0.85t$$
$$= 0.85 \times 0.375 \text{ in}$$
$$= 0.319 \text{ in}$$

where:

$D_o / t = 101$
$L_v / D_o = 6.052$

Per Figure UG-29.2, section UG-29 of ASME VIII, Div.1, it is found out that:

$$\text{Arc length} \approx 0.36\ D_o$$
$$= 0.36 \times 38 \text{ in}$$
$$= 13.68 \text{ in}$$
$$\text{Chord length} = 2 \times \text{Arc length}$$
$$= 2 \times 13.68 \text{ in}$$
$$= 27.36 \text{ in}$$

So, in a chord length of 27.36 in, the maximum plus or minus deviation from the true circle form shall not exceed 0.32 in.

3. Analysis and Calculation for the Stffening Rings

3.1 To Define the Stiffening Ring for the Vessel Shell Under the External Pressure and the Calculations of the Weld

Reference drawings: ME-443027

Given:

$E = 29 \times 10^6$ psi, Modules of elasticity of the stiffening materials

$l_e = 1.1\,(D_o t)^{0.5} = 1.1\,(38" \times 0.375")^{0.5}$

$= 4.152"$, the effective regional length of the vessel shell

F_y: 38 ksi, min. yield stress of the stiffening material SA516-70 under the operating temperature.

To find the required moment of inertia of the combined stiffening ring – shell cross section about the neutral axis parallel to the axial of the shell, in^4, I_s':

$I_s' = [D_o^2 L_s(t + A_s/L_s)A] / 10.9$ (eq. 3-1)

(per UG-29, ASME VIII, Div.1)

Where:

A_s: Cross-sectional area of the stiffening area, in^2

$= 1.50$ in^2, see figure 1.

L_s: Distance between the stiffening ring and support, ~230.0 in

(See drawing ME-443027)

B: Factor determined from the applicable chart,

$= 0.75\,[PD_o \div (t + As/Ls)]$

$= 0.75\,[14.7 \text{ psi} \times 38 \text{ in} \div (0.375 \text{ in} + 1.50 \text{ in}^2/230 \text{ in})]$

$= 1098$

Per figure CS-2 in Subpart 3 of section II, Part D, where B = 1098 falling below the left end of the material/temperature curve, so:

$A = 2B/E$

$= 2 \times 1098 / (30 \times 10^6)$

$= 0.0000732$

(per step 5 of UG-29, ASME VIII, Div. 1)

Then the Eq. 3-1 becomes:

$I_s' = \{[38^2 \times 230 \times (0.375 + 1.50/230) \times 0.0000732] / 10.9\}\text{in}^4$

$= 0.851$ in^4

To find the combined moment of inertia of the cross section area of the stiffening ring and shell

Figure 3.1 is the cross-section view of the stiffening ring with the vessel shell, also refer to view S-S of drawing ME-443027, MC-443052

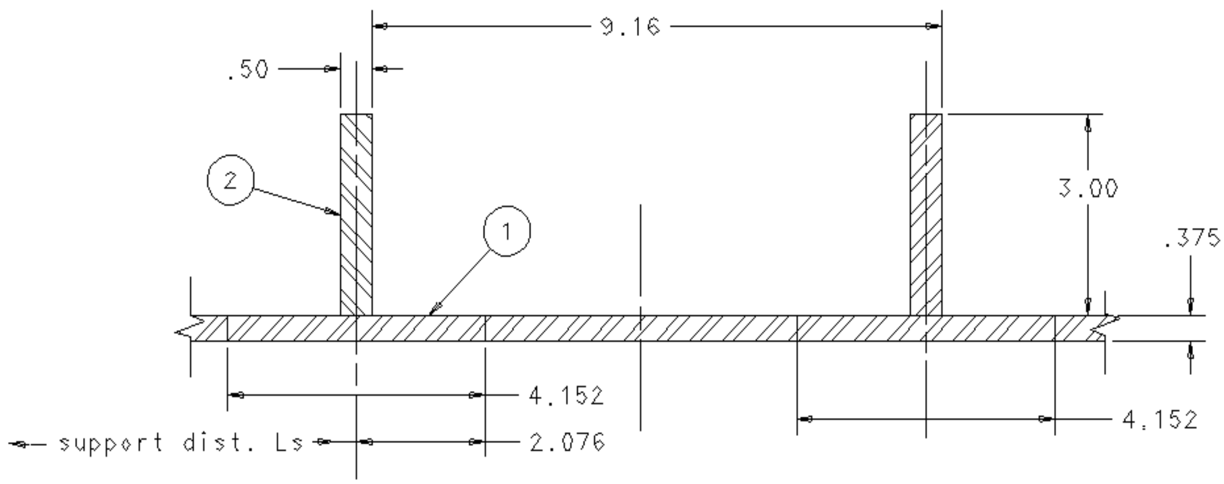

Figure 3.1, <u>Cross-section view of the vessel shell with the stiffening rings</u>

It is found that from the figure 3.1:

$A_1 = 4.152 \text{ in} \times 0.375 \text{ in} = 1.557 \text{ in}^2$
$A_2 = 0.5 \text{ in} \times 3.0 \text{ in} = 1.500 \text{ in}^2$
$y_1 = 0.1875 \text{ in}$
$y_2 = 1.8125 \text{ in}$
$y = (A_1 y_1 + A_2 y_2) \div (A_1 + A_2)$
$\quad = 0.9848 \text{ in}$
$dy_1^2 = (y - y_1)^2 = 0.6357 \text{ in}^2$
$dy_2^2 = (y - y_2)^2 = 0.6851 \text{ in}^2$
$I_1 = 0.0182 \text{ in}^4$
$I_2 = 1.125 \text{ in}^4$
$I = \sum I_i + \sum A_i dy_i^2$
$\quad = \underline{3.1507 \text{ in}^4} > I' = 0.851 \text{ in}^4$

Since the requirement moment of inertia I' is less than the value of I provided by the combined moment inertia of stiffening ring and vessel shell, so the stiffening ring design is satisfactory and it can be treated as a support of the vessel.

<u>Calculations of the attachment welds of the stiffening ring</u>:

Per UG-30(e), ASME VIII, Div. 1:

Radial pressure load PL_s = 14.7 psi x 230 in

$= 3381$ lb/in

Radial shear load $V = 0.01\ PL_sD_o = 0.01 \times 14.7$ psi $\times 230$ in $\times 38$ in

$= 1{,}285$ lb

Weld shear flow due to radial shear load: VQ/I

where Q is the first moment of area, per figure 3.1,

$Q = 4.152 \times 0.375\ (0.9848 – 0.375/2)$

$= 1.24\ in^3$

$VQ/I = 1{,}285\ lb \times 1.24\ in^3 / 3.1507\ in^4$

$= 505.73$ lb/in

The combined weld load (required)

$P_r = (3{,}381^2 + 506^2)^{1/2}$ lb/in

$= 3{,}419$ lb/in

The calculated welding load from the ring welds:

The most weaker material is the shell:

$S_v = 15$ ksi (SA 516-60)

The allowable fillet weld stress:

$F_{af} = 0.55S_v = 0.55 \times 15{,}000$ psi

$= 8{,}250$ psi (per UW-18(d), ASME VIII, Div.1)

The minimum fillet weld leg size is: 0.25" for the attachment.

(per UG-30(f), ASME VIII, Div.1)

Since there is weld on both sides of the ring (see view S-S, ME-443027)

The allowable load from the weld is:

$P_{aw} = 2 \times 0.25$ in $\times 8{,}250$ psi

$= 4{,}125$ lb/in $> P_r = 3{,}419$ lb/in

So the min. fillet size of 0.25" is acceptable for the current stiffening ring design.

3.2 To Define the Downstream End Flange as the Stiffening Ring for the Vessel Shell Under the External Pressure

Reference drawings: ME-443027, MD-443039

Givien:

$E = 30 \times 10^6$ psi, Modules of elasticity of the flange materials

$l_e = 1.1\ (D_o t)^{0.5} = 1.1\ (38\text{''} \times 0.375\text{''})^{0.5}$

$= 4.152$", the effective regional length of the vessel shell

F_y: 25 ksi, min. yield stress of the stiffening material A182-F304L under the operating temperature.

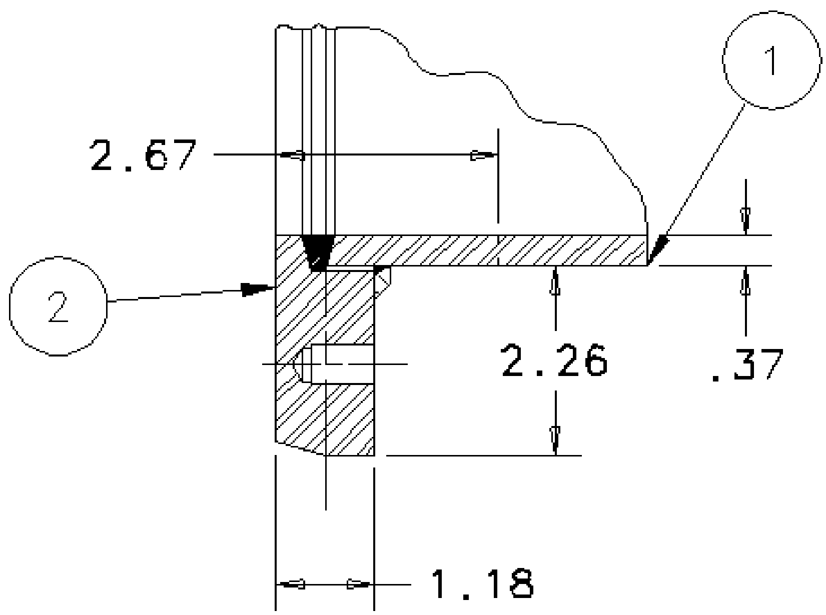

Figure 3.2, The cross-section view of the end flange with the vessel shell, also refer to detail L of drawing ME-443027, and drawing MD-443039..

To find the required moment of inertia of the combined end flange – shell cross section about the neutral axis parallel to the axia of the shell, in^4, I_s':

$I_s' = [D_o^2 L_s(t + A_s/L_s)A] / 10.9$ (eq. 3-2)

(per UG-29, ASME VIII, Div.1)

Where:

A_{sd}: Cross-sectional area of the downstream end flange stiffening area, in^2

$= (1.18 \times 2.26)\ \text{in}^2 = 2.6668$ in2, see figure 3.2

L_{sd}: Distance between the stiffening ring and downstream end flange, ~230.0 in
(See drawing ME-443027)

B: Factor determined from the applicable chart,

$= 0.75\ [PD_o \div (t + As/Ls)]$

$= 0.75\ [14.7\ \text{psi} \times 38\ \text{in} \div (0.375\ \text{in} + 2.6668\ \text{in}^2/230\ \text{in})]$

$= 1084$

Per figure CS-2 in Subpart 3 of section II, Part D, where B = 1084 falling below the left end of the material/temperature curve, so:

$A = 2B/E$

$= 2 \times 1084 / (30 \times 10^6)$

$= 0.0000723$

(per step 5 of UG-29, ASME VIII, Div. 1)

Then the eq.3-2 becomes:

$I_s' = \{[38^2 \times 230 \times (0.375 + 2.6668/230) \times 0.0000723] / 10.9\}\text{in}^4$

$= ((332{,}120 \times 0.3866 \times 0.0000723) / 10.9)\ \text{in}^4$

$= 0.852\ \text{in}^4$

<u>To find the combined moment of inertia of the cross section area of the stiffening ring and shell</u>

It is found that from the figure 3.2:

$A_1 = 2.67\ \text{in} \times 0.375\ \text{in} = 1.001\ \text{in}^2$

$A_2 = 1.18\ \text{in} \times 2.26\ \text{in} = 2.6668\ \text{in}^2$

$y_1 = 0.1875\ \text{in}$

$y_2 = 1.505\ \text{in}$

$y = (A_1y_1 + A_2y_2) \div (A_1 + A_2)$

$= 1.1454\ \text{in}$

$dy_1^2 = (y - y_1)^2 = 0.9176\ \text{in}^2$

$dy_2^2 = (y - y_2)^2 = 0.1293\ \text{in}^2$

$I_1 = 0.0117\ \text{in}^4$

$I_2 = 1.1351\ \text{in}^4$

$I = \sum I_i + \sum A_i dy_i^2$

$= (1.1468 + 1.2633)\ \text{in}^4$

$= \underline{2.4101\ \text{in}^4} > I' = 0.852\ \text{in}^4$

Since the requirement moment of inertia I' is less than the value of I provided by the combined moment inertia of the end flange(downstream) and vessel shell, so the downstream end flange is satisfactory and it can be treated as a support of the vessel.

3.3 To Define the Upstream End Flange as the Stiffening Ring for the Vessel Shell Under the External Pressure

Reference drawings: Detail M of drawing ME-443027-2,
MD- 443040

L_{su}: Distance between the stiffening ring and upstream end flange, ~ 45.0 in
(See figure 4-4.3 of Page 32)
A_{su}: ≈ 0.5 (45.67 - 38.12) x 1.57 in^2 = 5.927 in^2

Since $L_{su} < L_{sd}$; $A_{su} > A_{sd}$

From the calculation steps of section 3.2, we can conclude that the requirement moment of inertia I' is less than the value of I provided by the combined moment inertia of the end flange(upstream) and vessel shell, so the upstream end flange is satisfactory and it can be treated as a support of the vessel.

4. Calculations and Analysis for the Vessel Shell Openings

4.1 Calculations for the Opening of Coldmass Port for the Reinforcement and Weld Size

The main vessel shell:

P: 14.7 psi, external design pressure
D_o: 38.00 in, outside dia. Of the vessel shell.
D_i: 37.25, inside dia. of the vessel shell.
t: 0.375 in, the nominal pipe wall thickness.
F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_v. at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Vessel shell material: SA 516, grade 60, carbon steel (see ME-443054)
L_v: Design length of the vessel section between lines of support (see Page UG-28(b)), ≈ 230"

Find out the required vessel shell thickness t_r:

Try t_{r1} = 0.375"

L_v/D_o = 230"/38" = 6.053
D_o/t_r = 38/0.375 = 101.333

From section UG-28, also Fig. G & Fig. CS-2 in Subpart 3 of section II, part D, ASME VIII, Div I:
Find:
A = 0.0002
B = 2850
P_{a1}= (4B)/(3(D_o/t_r)) = 37.50 psi

where:
A: Factor determined from Fig. G.
B: Factor determined from applicable material chart in Fig. CS-2
P_a: Max. allowable external working pressure, psi

3. Try t_{r2} = 0.25
L_v/D_o = 6.053
D_o/t_{r2} = 38"/0.25" = 152
Find:

A ≈ 0.000115
Since the value of A falling to the left of the applicable material/temperature curve, so:

P_{a2} = (2AE/3(Do/t_{r2})) = 15.13 psi

So, $\mathbf{t_r = 0.250"}$

The coldmass port:

d_o: 25.197", outside dia. of the port.
D_i: 15.512", inside dia . of the port.
t_n: 4.842", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 29 x 10^6 psi, modulus of elasticity of the material at the operating temp.
Material: SA 182- 304L stainless steel pipe, $F_a = S_n$ = 16,700 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: MD-443049, MD-443050, ME-443027, and also as shown from figure 4.1.

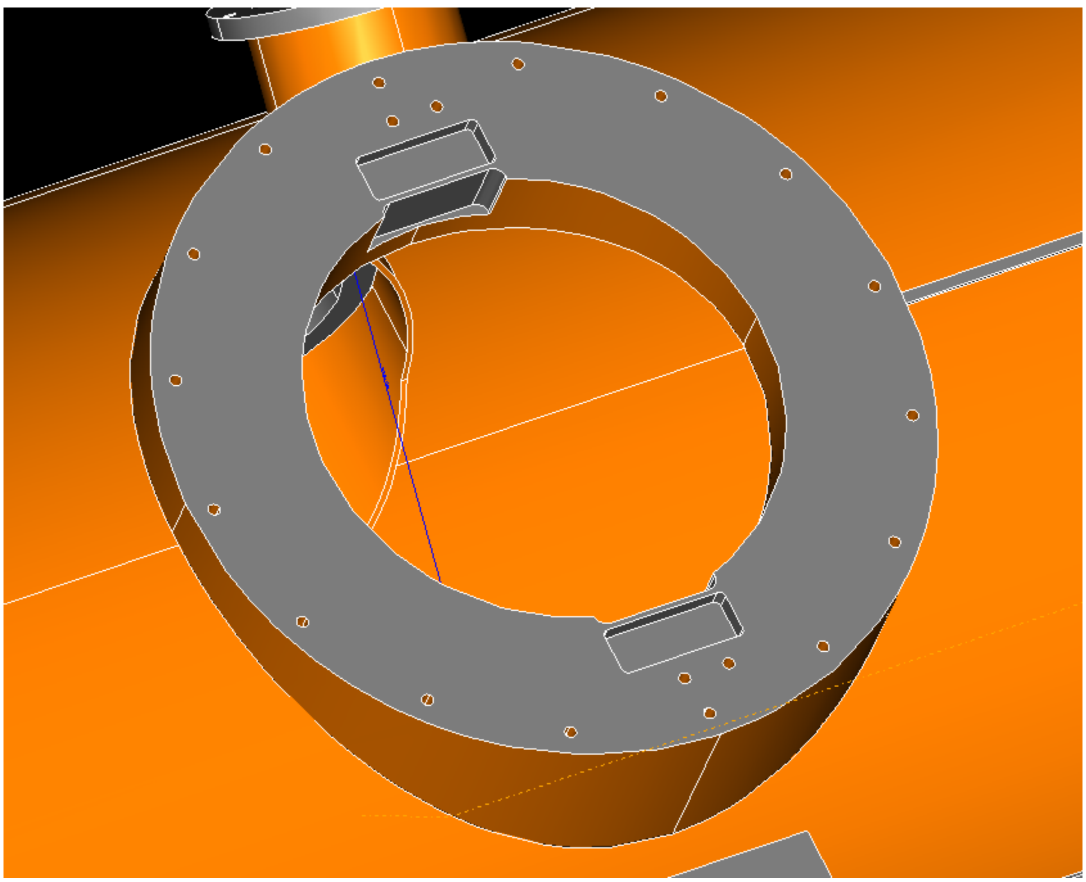

Figure 4.1, 3D view of the coldmass port and the vessel shell

Find out the thickness required:

1. Assuming $L_1 = 6.64$ in, $t_{r1} = 0.375$ in, then:

   $L_1/d_o = 6.64/25.197 = 0.2635$
   $d_o/t_{r1} = 25.197/0.375 = 67$

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   $A = 0.013$
   $B = 13300$
   $P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 13300)/(3 \times 67) = 264$ psi > P

2. Try: $L_2 = 6.64$ in, $t_{r2} = 0.0625$ in

   $L_2/d_o = 0.2635$
   $d_o/t_{r2} = 403$

   get:
   $A = 0.00073$
   $B = 8000$
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 8000)/(3 \times 403) = 26.47$ psi > P

3. Try: $L_3 = 6.64$ in, $t_{r3} = 0.046$ in

   $L_3/d_o = 0.2635$
   $d_o/t_{r3} = 25.197/0.046 = 548$

   get:
   $A = 0.000445$
   $B = 6100$
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 6100)/(3 \times 548) = 14.84$ psi ≈ P
   So: **$t_{rn} = t_{r3} = 0.046$ in**

Size of the weld required:

Per UG-37, UW-16 and Fig. UW-16.1(p) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $=0.5\ t_{min}$.
$t_{i:}$ the throat dimension of the inside of the shell cutout, $\geq 0.7 t_{min}$
$t_o = 0.5\ t_{min} = 0.5 \times 0.375” = 0.1875”$, **so the fillet leg size is 0.27”**
$t_i = 0.7\ t_{min} = 0.7 \times 0.375” = 0.26”$, **so the fillet leg size is 0.37”**

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1})$
$= 0.5 \times (16.299” \times 0.25” \times 1.0 +2 \times 4.842” \times 0.25” \times 1.0 \times 0)$

= **2.0374 in^2**

Where:

d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 16.299 in (per drawing ME-443054)

F: Correction factor

f_r : Strength reduction factor, not greater than 1.0
f_{r1} =1.0 for nozzle wall abutting the vessel wall as shown in figure UG-40(j), (k), (n) and (o).
$f_{r2} = S_n / S_v$

<u>*Area of reinforcement available A_a:*</u>

A_1: Larger of the following calculated value: (area available in shell):
= $d(E_1t – Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
= 16.299” x (1 x 0.375” – 1.0 x 0.25”) – 0
= 2.0374 in^2

or = $2(t + t_n)(E_1t –Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
= 2 (0.375” + 4.842”) x (1.0 x 0.375” – 1.0 x 0.25”) – 0
= 1.30425 in^2

So pick **A_1 = 2.0374 in^2**

A_2: Smaller of the following:
= $5 (t_n-t_{rn})tf_{r2}$
= 5 x (4.842 in – 0.046 in) x 0.375 in x 1.0
= 8.9925 in^2

or = $5 (t_n-t_{rn}) f_{r2}t_n$
= 5 x (4.842 in – 0.046 in) x 1.0 x 4.842 in
= 116.11 in^2

So pick **A_2 = 8.9925 in^2**

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick **A_3 = 0.0 in^2**

A_{41}: Outward nozzle weld
= $leg^2 f_{r2}$
= 0.27^2 in^2 x 1.0
=**0.0729 in^2**

A_{43}: Inward nozzle weld
= $leg^2 f_{r2}$
= 0.37^2 in^2 x 1.0
= **0.1369 in^2**

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

$= (2.0374 + 8.9925 + 0 + 0.0729 + 0.1369)\ in^2$

$= \mathbf{11.2397\ in^2} > A_r = 2.0374\ in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be also true for openings of #2 (have 2 openings @ both ends, see drawing ME-443027), this is because: the value of L_v for #2 opening is equal or smaller than the value for #3 opening, the smaller value of L_v will lead to smaller value of t_r, accordingly, it will lead to the smaller value of A_r.

4.2 <u>Calculations for the Opening of MC Port (#9) for the Reinforcement and Weld Size</u>

<u>The main vessel shell</u>:

From section 4.1, it is found that **t_r = 0.250"**

<u>The MC port:</u>

d_o: 11.339", outside dia. of the port.
D_i: 10.945", inside dia . of the port.
t_n: 0.197", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30 x 10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: ME-443027, and also as shown from figure 4.2.

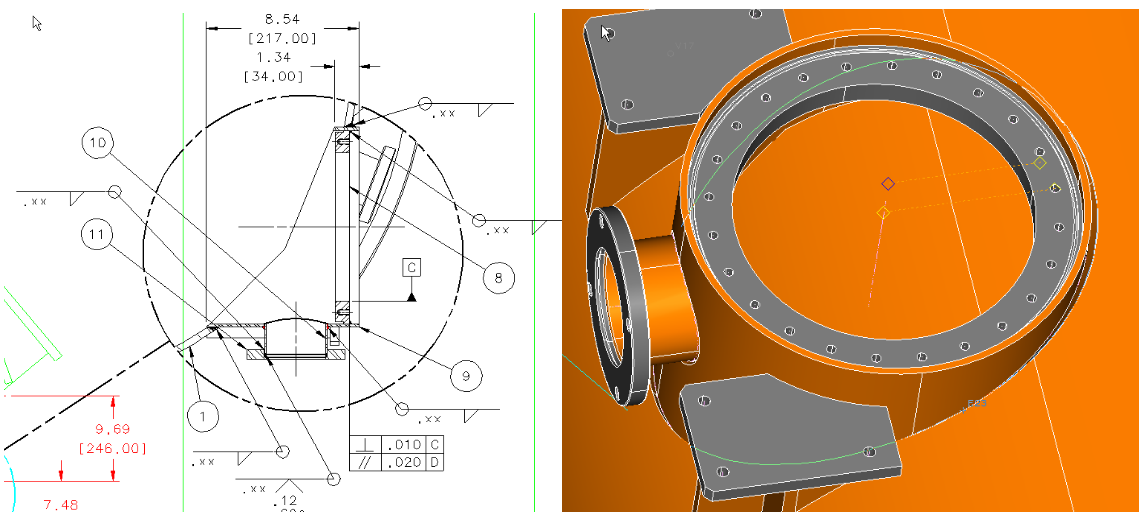

Figure 4.2, <u>The partial 2D and 3D views of the MC port from ME-443027</u>

Find out the wall thickness required:

1. Assuming L_1 = 8.54 in, t_{r1} = 0.125 in, then:

   L_1/d_o = 8.54/11.339 =0.7532
   d_o/t_{r1} = 11.339/0.125 = 91

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   A = 0.00225
   B = 10500
   $P_a = (4B)/(3(d_o/t_{r1}))$ = (4 x 10500)/(3 x 91) = 153 psi > P

2. Try: L_2 = 8.54 in, t_{r2} = 0.0625 in

   L_2/d_o = 0.7532
   d_o/t_{r2} = 181

   get:
   A = 0.00075
   B = 8050
   $P_a = (4B)/(3(d_o/t_{r2}))$ = (4 x 8050)/(3 x 181) = 59.3 psi > P

Try: L_3 = 8.54 in, t_{r3} = 0.033 in

   L_2/d_o = 0.7532
   d_o/t_{r3} = 344

   get:
   A = 0.000280
   B = 3950
   $P_a = (4B)/(3(d_o/t_{r2}))$ = (4 x 3950)/(3 x 344) = 15.31 psi ≈ P

So: **$t_{rn} = t_{r3}$ = 0.033 in**

Size of the weld required:

Per UG-37, UW-16 and Fig. UW-16.1© of ASME VIII, Div.1.
t_{min}: 0.197”
$t_{c:}$ ≥ $0.7t_{min}$ = 0.138”, **so the fillet leg size is 0.195”**
Full penetration weld plus the fillet weld $t_{c.}$

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1\text{-}f_{r1})$
$= 0.5 \times (11.339" \times 0.25" \times 1.0 + 2 \times 0.197" \times 0.25" \times 1.0 \times 0.053)$
$= \mathbf{1.4226\ in^2}$

Where:

d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 11.339 in (per drawing ME-443054)

F: Correction factor

f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

<u>Area of reinforcement available A_a</u>:

A_1: Larger of the following calculated value: (area available in shell):
$= d(E_1t – Ft_r)\text{-}2t_n(E_1t – Ft_r) \times (1\text{-}f_{r1})$
$= 11.339" \times (1 \times 0.375" – 1.0 \times 0.25") – 2 \times 0.197 (1.0 \times 0.375 – 1.0 \times 0.25) \times 0.053$
$= (1.4174 – 0.0026)\ in^2$
$= 1.4148\ in^2$

or $= 2(t + t_n)(E_1t –Ft_r)\text{-}2t_n(E_1t – Ft_r) \times (1\text{-}f_{r1})$
$= 2 (0.375" + 0.197") \times (1.0 \times 0.375" – 1.0 \times 0.25") – 2 \times 0.197 (1.0 \times 0.375 – 1.0 \times 0.25) \times 0.053$
$= (0.143 – 0.0026)\ in^2$
$= 0.1405\ in^2$

So pick $\mathbf{A_1 = 1.4148\ in^2}$

A_2: Smaller of the following:
$= 5 (t_n\text{-}t_{rn})tf_{r2}$
$= 5 \times (0.197\ in – 0.033\ in) \times 0.375\ in \times 0.947$
$= 0.2912\ in^2$

or $= 5 (t_n\text{-}t_{rn})\ f_{r2}t_n$
$= 5 \times (0.197\ in – 0.033\ in) \times 0.947 \times 0.197\ in$
$= 0.153\ in^2$

So pick $\mathbf{A_2 = 0.153\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld
$= leg^2\ f_{r2}$
$= 0.195^2\ in^2 \times 0.947$
$= \mathbf{0.036\ in^2}$

A_{43}: Inward nozzle weld

$= leg^2 \; f_{r2}$

$= 0.0^2 \; in^2 \times 1.0$

$= \mathbf{0.0 \; in^2}$

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

$= (1.4148 + 0.153 + 0 + 0.036 + 0.0) \; in^2$

$= \mathbf{1.6038 \; in^2} > A_r = 1.4226 \; in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be true for all 8 openings (#9 openings, see drawing ME-443027 & ME-443054). Any issue related the multiple openings will discuss in different section.

4.3 Calculations for the Opening of Instrumentation Port (#10 opening) for the Reinforcement and Weld Size

The main shell (Tube #9)

From section 4.2, it is found that: $\mathbf{t_r = 0.033}$ **in**

The instrumentation port:

d_o: 3.50", outside dia. of the port.
D_i: 3.334", inside dia . of the port.
t_n: 0.083", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30×10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n = 14,200$ psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: section G-G, ME-443027, and also as shown from figure 4.2.

Find out the wall thickness requires:

Assuming $L_1 = 2.0075$ in, $t_{r1} = 0.083$ in, then:
where: $L_1 = 7.48 - (11.339/2 - 0.197) = 2.0075"$

$L_1/d_o = 2.0075/3.50 = 0.5736$
$d_o/t_{r1} = 3.50/0.083 = 42$

From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

A = 0.009
B = 13200
$P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 13200)/(3 \times 42) = 419$ psi > P

Try: $L_2 = 2.0075$ in, $t_{r2} = 0.015625$ in

$L_2/d_o = 0.5736$
$d_o/t_{r2} = 224$

get:
A = 0.0007
B = 8000
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 8000)/(3 \times 224) = 47.62$ psi > P

Try: $L_3 = 2.0075$ in, $t_{r3} = 0.0094$ in

$L_3/d_o = 0.5732$
$d_o/t_{r3} = 372$

get:
$A = 0.0003$
$B = 4250$
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 4250)/(3 \times 372) = 15.23 \text{ psi} \approx P$

So: $\mathbf{t_{rn} = t_{r3} = 0.0094}$ **in**

<u>Size of the weld required</u>:

Per UG-37, UW-16 and Fig. UW-16.1© of ASME VIII, Div.1.
t_{min}: 0.083"
$t_{c:}$ $\geq 0.7t_{min} = 0.056$", **so the fillet leg size is 0.08"**
Full penetration weld plus the fillet weld $t_{c.}$

<u>Area of the reinforcement required A_r</u>:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1\text{-}f_{r1})$
$= 0.5 \times (3.5" \times 0.033" \times 1.0 + 2 \times 0.083" \times 0.033" \times 1.0 \times 0)$
$= \mathbf{0.0578\ in^2}$
Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 11.339 in (per drawing ME-443054)
F: Correction factor
f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 14.2 = 1.0$

<u>Area of reinforcement available A_a</u>:

A_1: Larger of the following calculated value: (area available in shell):
$= d(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r) \times (1\text{-}f_{r1})$
$= 3.50" \times (1 \times 0.197" - 1.0 \times 0.033") - 2 \times 0.083 (1.0 \times 0.197 - 1.0 \times 0.033) \times 0$
$= 0.574 \text{ in}^2$

or $= 2(t + t_n)(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r) \times (1\text{-}f_{r1})$
$= 2 (0.197" + 0.083") \times (1.0 \times 0.197" - 1.0 \times 0.033") - 2 \times 0.197 (1.0 \times 0.375 - 1.0 \times 0.25) \times 0.0$
$= (2 \times 0.28 \times 0.164) \text{ in}^2$
$= 0.0918 \text{ in}^2$
So pick $\mathbf{A_1 = 0.574\ in^2}$

Since $A_1 = 0.574$ in^2
Also: $A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

That leads to:
$A_a > A_r = 0.0578$ in^2

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be true for all 8 instrumentation ports (#10 openings, see section G-G, drawing ME-443027).

4.4. <u>Calculations for the Opening of Port #16 for the Reinforcement and Weld Size</u>

The main vessel shell:

From section 4.1, it is found that **t_r = 0.250"**

The port #16:

d_o: 11.811", outside dia. of the port.
D_i: 11.417", inside dia . of the port.
t_n: 0.197", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30 x 10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: Section H-H of ME-443027, and also as shown from figure 4-4.1 and figure 4-4.2..

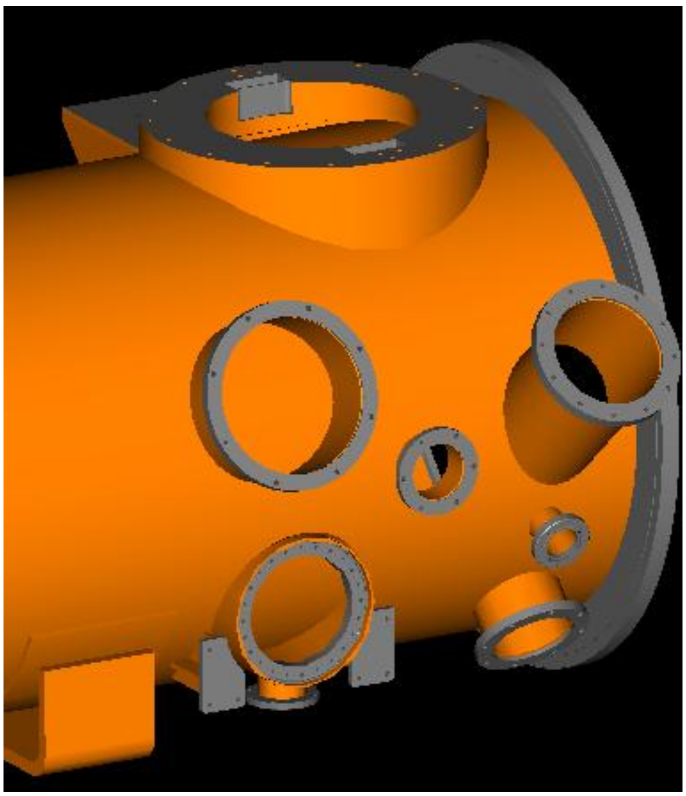

Figure 4-4.1 Multiple openings of the upstream end of the vessel

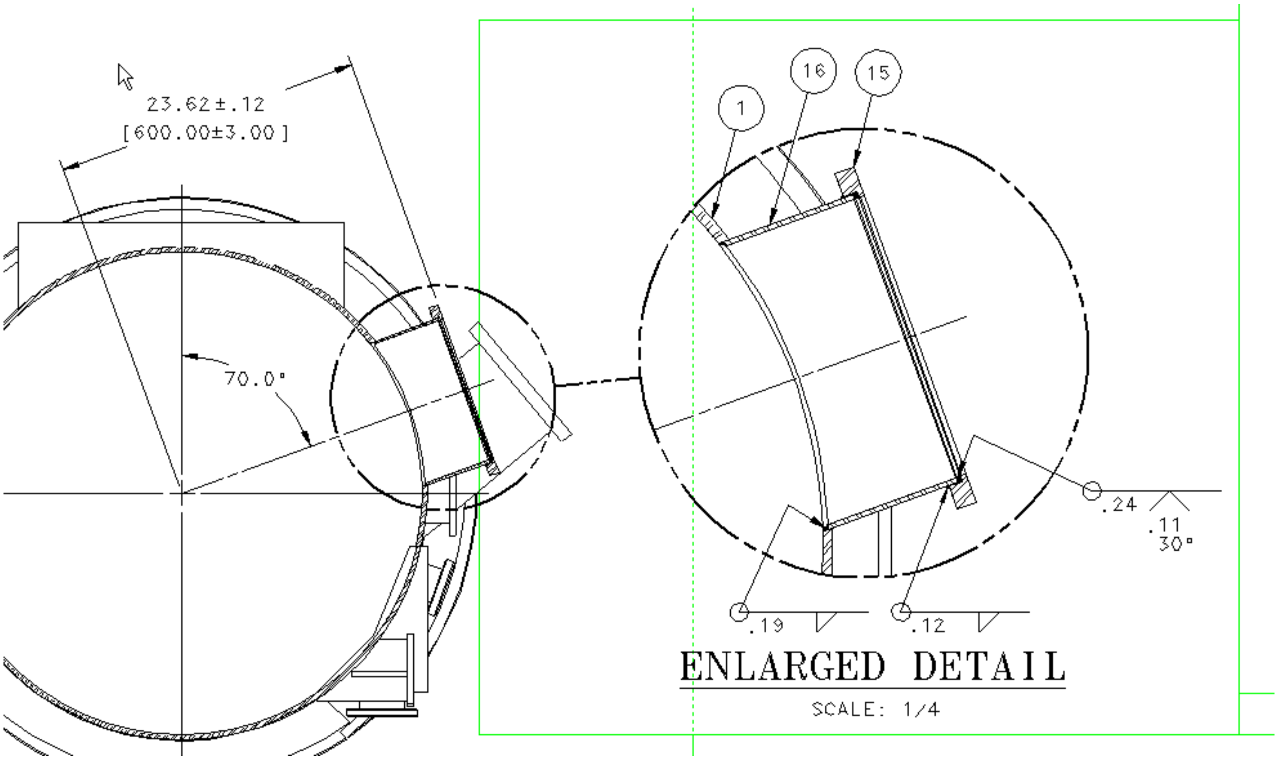

Figure 4-4.2, <u>The 2D view of the port #16 from drawing ME-443027</u>

Find out the wall thickness required:

Assuming L_1 = (23.62 – 18.625) in = 5.00 in, t_{r1} = 0.035 in, then:

L_1/d_o = 5.00/11.811 = 0.4233
d_o/t_{r1} = 11.811/0.035 = 337

From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

A = 0.00057
B = 7700
$P_a = (4B)/(3(d_o/t_{r1}))$ = (4 x 7700)/(3 x 337) = 30.46 psi > P

1. Try: L_2 = 5.00in, t_{r2} = 0.025 in

L_2/d_o = 0.4233
d_o/t_{r2} = 472

get:
A = 0.00033
B = 4650

$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 4650)/(3 \times 472) = 13.13$ psi < P

Try: $L_3 = 5.00$ in, $t_{r3} = 0.026$in

$L_2/d_o = 0.4233$
$d_o/t_{r3} = 454$

get:
A = 0.00035
B = 5000
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 5000)/(3 \times 454) = 14.68$ psi $> \approx$ P

So: $\mathbf{t_{rn} = t_{r3} = 0.026}$ **in**

<u>Size of the weld required</u>:

Per UG-37, UW-16 (d.1) and Fig. UW-16.1(k) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $\geq 0.7t_{min}$.
t_i: the throat dimension of the inside of the shell cutout, $\geq 0.7t_{min}$
$t_o = 0.7\ t_{min} = 0.7 \times 0.197" = 0.138"$, **so the fillet leg size is 0.19"**
$t_i = 0.7\ t_{min} = 0.7 \times 0.197" = 0.138"$, **so the fillet leg size is 0.19"**
$t_o + t_i = 0.138" + 0.138" = 0.276" \geq 1.25\ t_{min} = 0.246"$

<u>Area of the reinforcement required A_r</u>:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1}))$
$= 0.5 \times (11.417" \times 0.25" \times 1.0 + 2 \times 0.197" \times 0.25" \times 1.0 \times 0.053)$
$= (1.4271 + 0.0026)\ in^2$
$= \mathbf{1.4297\ in^2}$

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 11.417 in (per drawing ME-443054)
F: Correction factor
f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

<u>Area of reinforcement available A_a</u>:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{16}(E_1t - Ft_r) - 2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 4.90" \times (1 \times 0.375" - 1.0 \times 0.25") - 2 \times 0.197" \times 1.0 \times .125" \times 0.053$

$= (0.6125 – 0.0026)\ in^2$
$= 0.6099\ in^2$

or $= 2(t + t_n)(E_1t – Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
$= 2\ (0.375” + 0.197”) \times (1.0 \times 0.375” – 1.0 \times 0.25”) – 2 \times 0.197” \times 1.0 \times .125” \times 0.053$
$= (0.143 – 0.0026)\ in^2$
$= 0.1404\ in^2$

So pick $\mathbf{A_1 = 0.6099\ in^2}$

where:

$d_{16} \approx 2 \times 51\% \times [(\Pi\ d_o \times 13.3^o) \div 360^o + 2d_n] = 4.90”$, per UG-42 (a)(1) and section C-C of drawing ME-443054. 51% is the portion of the cross section can be considered between two openings (#16 & #9) by the ratio of the diameters.

A_2: Smaller of the following:
$= 5\ (t_n-t_{rn})tf_{r2}$
$= 5 \times (0.197\ in – 0.026\ in) \times 0.375\ in \times 0.947$
$= 0.3036\ in^2$

or $= 5\ (t_n-t_{rn})\ f_{r2}t_n$
$= 5 \times (0.197\ in – 0.026\ in) \times 0.947 \times 0.197\ in$
$= 0.1595\ in^2$

So pick $\mathbf{A_2 = 0.1595\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld
$= leg^2\ f_{r2}$
$= 0.19^2\ in^2 \times 1.0$
$= \mathbf{0.036\ in^2}$

A_{43}: Inward nozzle weld
$= leg^2\ f_{r2}$
$= 0.19^2\ in^2 \times 1.0$
$= \mathbf{0.036 in^2}$

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$
$= (0.6099 + 0.1595 + 0 + 0.036 + 0.036)\ in^2$
$= \mathbf{0.8414\ in^2} < A_r = 1.4297\ in^2$

The opening is not adequate, either to reduce the t_r value or adding reinforcing element.

Assuming to add a stiffening rib like #26 of drawing ME-443027 at the location as shown in figure 4-4.3

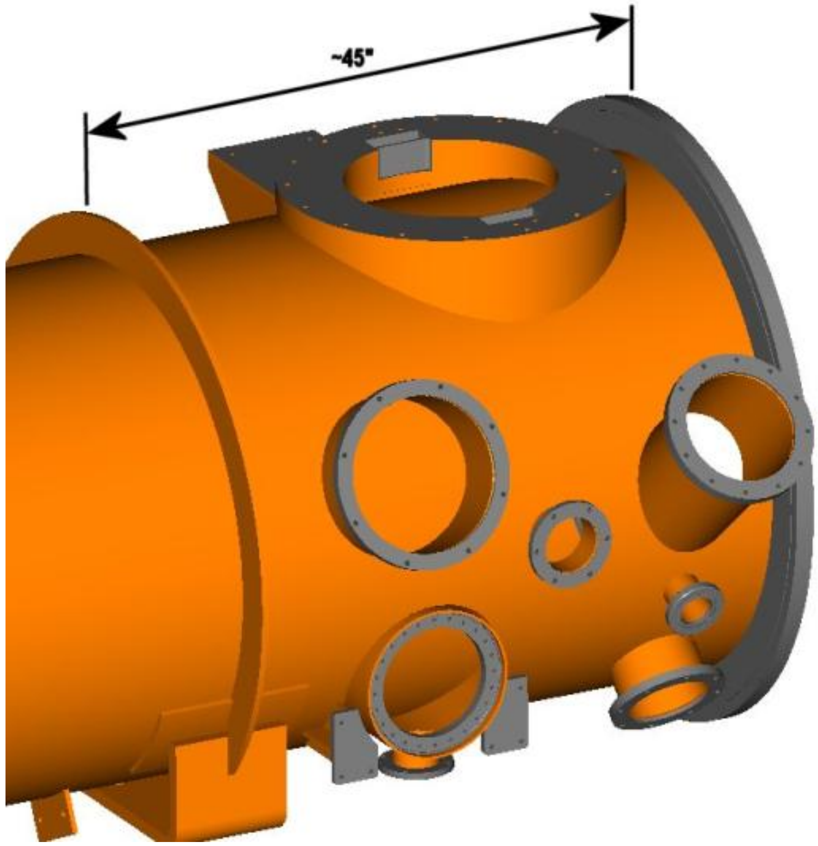

Figure 4-4.3, New version upstream end of the vessel with new stiffening rib.

If going through the similar procedures as shown on section 4.1, it was found that:

$t_r \approx 0.13$" when $L_v = 45$"

then:

Area of the reinforcement required A_r:
Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1\text{-}f_{r1}))$
$= 0.5 \times (11.417" \times 0.13" \times 1.0 + 2 \times 0.197" \times 0.13" \times 1.0 \times 0.053)$
$= (0.7421 + 0.0014)\ in^2$
$= \mathbf{0.7435\ in^2}$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{16}(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r) \times (1\text{-}f_{r1})$
$= 4.90" \times (1 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.197" \times 1.0 \times .245" \times 0.053$
$= (1.2005 - 0.0051)\ in^2$

$= 1.1954\ in^2$

or $= 2(t + t_n)(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r) \times (1 - f_{r1})$

$= 2\ (0.375" + 0.197") \times (1.0 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.197" \times 1.0 \times .245" \times 0.053$

$= (0.2803 - 0.0051)\ in^2$

$= 0.2752\ in^2$

So pick $\mathbf{A_1 = 1.1954\ in^2}$

where:

$d_{16} \approx 2 \times 51\% \times [(\Pi\ d_o \times 13.3^o) \div 360^o + 2d_n] = 4.90"$, per UG-42 (a)(1) and section C-C of drawing ME-443054. 51% is the portion of the cross section can be considered between two openings (#16 & #9) by the ratio of the diameters.

A_2: Smaller of the following:

$= 5\ (t_n - t_{rn}) t f_{r2}$

$= 5 \times (0.197\ in - 0.026\ in) \times 0.375\ in \times 0.947$

$= 0.3036\ in^2$

or $= 5\ (t_n - t_{rn})\ f_{r2} t_n$

$= 5 \times (0.197\ in - 0.026\ in) \times 0.947 \times 0.197\ in$

$= 0.1595\ in^2$

So pick $\mathbf{A_2 = 0.1595\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:

Since there is no inward nozzle

So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld

$= leg^2\ f_{r2}$

$= 0.19^2\ in^2 \times 1.0$

$= \mathbf{0.036\ in^2}$

A_{43}: Inward nozzle weld

$= leg^2\ f_{r2}$

$= 0.19^2\ in^2 \times 1.0$

$= \mathbf{0.036 in^2}$

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

$= (1.1954 + 0.1595 + 0 + 0.036 + 0.036)\ in^2$

$= \mathbf{1.4269\ in^2} > A_r = 0.7435\ in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the modified boundary conditions.

Applying the similar approach, it also will find out that the opening #9 at the upstream end is adequately reinforced under the modified boundary condition.

4.5 Calculations for the Opening of Port #22 for the Reinforcement and Weld Size

The main vessel shell:

From section 4.4, it is found that **t_r = 0.13"**

The port #22:

d_o: 8.780", outside dia. of the port.
D_i: 8.386", inside dia . of the port.
t_n: 0.197", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30×10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).

Reference drawings: Section K-K and detail L of ME-443027, and also as shown from figure 4-5.1

Find out the wall thickness required:

1. Assuming L_1 = 14.533 in, t_{r1} = 0.0625 in, then:

   L_1/d_o = 14.533/8.780 =1.6552
   d_o/t_{r1} = 8.780/0.0.0625 = 140

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:
   A = 0.00053
   B = 7400
   $P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 7400)/(3 \times 140) = 70.48$ psi > P

2. Try: L_2 = 14.533 in, t_{r2} = 0.03125 in

   L_2/d_o = 1.6552
   d_o/t_{r2} = 281

   get:
   A = 0.00017
   B = 2300
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 2300)/(3 \times 281)$
   = 10.91 psi < P

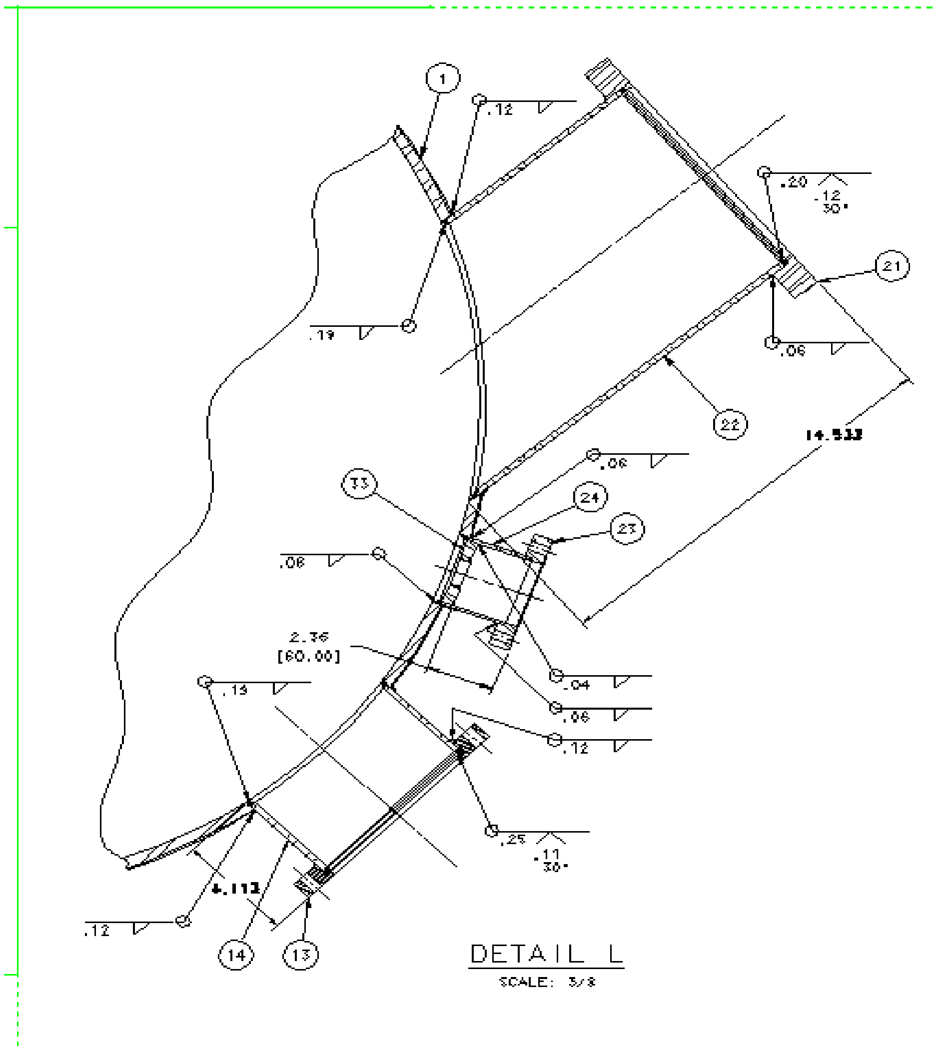

Figure 4-5.1, The 2D view of the ports of #22, #24 & #14

Try: L_3 = 14.533 in, t_{r3} = 0.035 in

L_2/d_o = 1.6552
d_o/t_{r3} = 251

get:
A = 0.00021
B = 2800
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 2800)/(3 \times 251) = 14.87$ psi ≈ P

So: $\mathbf{t_{rn} = t_{r3} = 0.035}$ **in**

Size of the weld required:

Per UG-37, UW-16 (d.1) and Fig. UW-16.1(k) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $=0.5\ t_{min}$.
t_i: the throat dimension of the inside of the shell cutout, $\geq 0.7t_{min}$
$t_o = 0.7\ t_{min} = 0.7 \times 0.197" = 0.138"$, **so the fillet leg size is 0.19”**
$t_i = 0.7\ t_{min} = 0.7 \times 0.197" = 0.138"$, **so the fillet leg size is 0.19”**
$t_o + t_i = 0.138" + 0.138" = 0.276" \geq 1.25\ t_{min} = 0.246"$

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1}))$
$= 0.5 \times (8.386" \times 0.13" \times 1.0 + 2 \times 0.197" \times 0.13" \times 1.0 \times 0.053)$
$= 0.5 \times (1.0902 + 0.0027)$
$=$ **$0.5465\ in^2$**

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 8.386 in (per drawing ME-443054)
F: Correction factor
f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{22}(E_1t - Ft_r) - 2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 3.243" \times (1 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.197 \times (1.0 \times 0.375" - 1.0 \times 0.13) \times 0.053$
$= (0.7945 - 0.0051)\ in^2$
$= 0.7894\ in^2$

or $= 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 2\ (0.375" + 0.197") \times (1.0 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.197 \times (1.0 \times 0.375" - 1.0 \times 0.13) \times 0.053$
$= 0.28 - 0.0051$
$= 0.2749\ in^2$

So pick **$A_1 = 0.7894\ in^2$**

where:
$d_{22} \approx 2 \times 77\% \times [(\Pi\ d_o \times 5.4^o) \div 360^o + d_{n22} + d_{n24}] = 3.243"$, per UG-42 (a)(1) and section C-C of drawing ME-443054. 51% is the portion of the cross section can be considered between two openings (#22 & #24) by the ratio of the diameters.

A_2: Smaller of the following:

$= 5\,(t_n - t_{rn}) t f_{r2}$

$= 5 \times (0.197 \text{ in} – 0.035 \text{ in}) \times 0.375 \text{ in} \times 0.947$

$= 0.2877 \text{ in}^2$

or $= 5\,(t_n - t_{rn})\, f_{r2} t_n$

$= 5 \times (0.197 \text{ in} – 0.046 \text{ in}) \times 0.947 \times 0.197 \text{ in}$

$= 0.1408 \text{ in}^2$

So pick $\mathbf{A_2 = 0.1408\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:

Since there is no inward nozzle

So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld

$= leg^2\, f_{r2}$

$= 0.19^2 \text{ in}^2 \times 1.0$

$= \mathbf{0.0361\ in^2}$

A_{43}: Inward nozzle weld

$= leg^2\, f_{r2}$

$= 0.19^2 \text{ in}^2 \times 1.0$

$= \mathbf{0.0361 in^2}$

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

$= (0.7894 + 0.1408 + 0 + 0.0361 + 0.0361) \text{ in}^2$

$= \mathbf{0.9979\ in^2} > A_r = 0.5465 \text{ in}^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be also true for openings of #2 (have 2 openings @ both ends, see drawing ME-443027), this is because: the value of L_v for #2 opening is equal or smaller than the value for #3 opening, the smaller value of L_v will lead to smaller value of t_r, accordingly, it will lead to the smaller value of A_r.

4.6 Calculations for the Opening of Port #24 for the Reinforcement and Weld Size

The main vessel shell:

From section 4.4, it is found that **t_r = 0.130"**

The port #24:

d_o: 2.756", outside dia. of the port.
d_i: 2.52", inside dia . of the port.
t_n: 0.118", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30×10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: Section K-K and detail L of ME-443027, and also as shown from figure 4-4.3 and figure 4-5.1.

Find out the wall thickness required:

1. Assuming L_1 = (22.44 – 18.625) in = 3.815 in, t_{r1} = 0.0625 in, then:

   L_1/d_o = 3.815/2.756 =1.3843
   d_o/t_{r1} = 2.756/0.0.0625 = 44

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   A = 0.0035
   B = 11500
   $P_a = (4B)/(3(d_o/t_{r1}))$ = (4 x 11500)/(3 x 44) = 348 psi > P

3. Try: L_2 = 3.815 in, t_{r2} = 0.03125 in

   L_2/d_o = 1.3843
   d_o/t_{r2} = 88

   get:
   A = 0.0012
   B = 8950
   $P_a = (4B)/(3(d_o/t_{r2}))$ = (4 x 8950)/(3 x 88) = 135.6 psi > P

3. Try: L_3 = 3.815 in, t_{r3} = 0.013 in

L_2/d_o = 1.3843
d_o/t_{r3} = 212

get:
A = 0.00033
B = 4550
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 2800)/(3 \times 251) = 28.62$ psi > P

4. Try: L_3 = 3.815 in, t_{r3} = 0.011 in

L_2/d_o = 1.3843
d_o/t_{r3} = 251

get:
A = 0.000215
B = 2900
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 2900)/(3 \times 251) = 15.41$ psi ≈ P

So: $\mathbf{t_{rn} = t_{r3} = 0.011}$ **in**

<u>Size of the weld required</u>:

Per UG-37, UW-16 (d.1) and Fig. UW-16.1(k) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, =0.5 t_{min}.
$t_{i:}$ the throat dimension of the inside of the shell cutout, ≥ 0.7t_{min}
t_o = 0.7 t_{min} = 0.7 x 0.118" =0.0826", **so the fillet leg size is 0.117"**
t_i = 0.7 t_{min} = 0.7 x 0.197" =0.0826", **so the fillet leg size is 0.117"**
$t_o + t_i$ = 0.0826" + 0.0826" = 0.165" ≥ 1.25 t_{min} =0.148"

<u>Area of the reinforcement required A_r</u>:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1}))$
= 0.5 x (2.52" x 0.13" x 1.0 + 2 x 0.118" x 0.13" x 1.0 x 0.053)
= 0.5 x (0.3276 + 0.0016) in^2
= **0.1646 $\mathbf{in^2}$**

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
d = 2.52 in (per drawing ME-443054)
F: Correction factor
f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall

$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{24}(E_1t – Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
$= 0.9686” \times (1 \times 0.375” – 1.0 \times 0.13”) – 2 \times 0.118” \times 1.0 \times .245” \times 0.053$
$= (0.237 - 0.0031)\ in^2$
$= 0.2339\ in^2$

or $= 2(t + t_n)(E_1t –Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
$= 2 (0.375” + 0.118”) \times (1.0 \times 0.375” – 1.0 \times 0.13”) – 2 \times 0.118” \times 1.0 \times .245” \times 0.053$
$= (0.1208 – 0.0031)\ in^2$
$= 0.1177\ in^2$

So pick **A_1 = 0.2339 in^2**

where:
$d_{24} \approx 2 \times 23\% \times [(\Pi\ d_o \times 5.4^o) \div 360^o + d_{n22} + d_{n24}] = 0.9686”$, per UG-42 (a)(1) and section C-C of drawing ME-443054. 51% is the portion of the cross section can be considered between two openings (#22 & #24) by the ratio of the diameters.

A_2: Smaller of the following:
$= 5 (t_n-t_{rn})tf_{r2}$
$= 5 \times (0.118\ in – 0.011\ in) \times 0.375\ in \times 0.947$
$= 0.190\ in^2$

or $= 5 (t_n-t_{rn}) f_{r2}t_n$
$= 5 \times (0.118\ in – 0.011\ in) \times 0.947 \times 0.118\ in$
$= 0.0598\ in^2$

So pick **A_2 = 0.0598 in^2**

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick **A_3 = 0.0 in^2**

A_{41}: Outward nozzle weld
$= leg^2 f_{r2}$
$= 0.11^2\ in^2 \times 1.0$
=**0.012 in^2**

A_{43}: Inward nozzle weld
$= leg^2 f_{r2}$
$= 0.11^2\ in^2 \times 1.0$
= **0.012in^2**

Then:
$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$
$= (0.2339 + 0.0598 + 0 + 0.0121 + 0.0121)\ \text{in}^2$
$= \mathbf{0.3179\ in^2} > A_r = 0.1646\ \text{in}^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

4.7 Analysis for the Opening of Port #14 for the Reinforcement and Weld Size

The main vessel shell:

From section 4.1, it is found that **t_r = 0.250”**

The port #14:

d_o: 6.625”, outside dia. of the port.
d_i: 6.187”, inside dia . of the port.
t_n: 0.219”, wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
L = 22.44”- 18.625 = 3.815”
E: 30 x 10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: Section K-K and detail L of ME-443027, and also as shown from figure 4-4.3 and figure 4-5.1.

Find out the wall thickness required:

1. Assuming L_1 = (22.44 – 18.625) in = 3.815 in, t_{r1} = 0.0625 in, then:

   L_1/d_o = 3.815/6.625 =0.5758
   d_o/t_{r1} = 6.625/0.0625 = 106

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   A = 0.0023
   B = 10500
   $P_a = (4B)/(3(d_o/t_{r1}))$ = (4 x 10500)/(3 x 106) = 132 psi > P

2. Try: L_2 = 3.815 in, t_{r2} = 0.02 in

   L_2/d_o = 0.5758
   d_o/t_{r2} = 331

   get:
   A = 0.0004
   B = 7000
   $P_a = (4B)/(3(d_o/t_{r2}))$ = (4 x 7000)/(3 x 331) = 28.2 psi > P

3. Try: L_3 = 3.815 in, t_{r3} = 0.0170 in

$L_3/d_o = 0.5758$
$d_o/t_{r3} = 390$

get:
$A = 0.00031$
$B = 4350$
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 4350)/(3 \times 390) = 14.87$ psi ≈ P

So: $\mathbf{t_{rn} = t_{r3} = 0.017}$ **in**

Size of the weld required:

Per UG-37, UW-16 (d.1) and Fig. UW-16.1(k) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $=0.5\ t_{min}$.
t_i: the throat dimension of the inside of the shell cutout, $\geq 0.7t_{min}$
$t_o = 0.7\ t_{min} = 0.7 \times 0.219” = 0.153”$, **so the fillet leg size is 0.216”**
$t_i = 0.7\ t_{min} = 0.7 \times 0.219” = 0.153”$, **so the fillet leg size is 0.216”**
$t_o + t_i = 0.153” + 0.153” = 0.306” \geq 1.25\ t_{min} = 0.274”$

$t_{i22} = t_{o22} = 0.19” < t_{i14} = t_{o14} = 0.21”$

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1}))$
$= 0.5 \times (6.187” \times 0.25” \times 1.0 + 2 \times 0.219” \times 0.25” \times 1.0 \times 0.053)$
$= (0.7734 + 0.0058)\ in^2$
$= \mathbf{0.7792\ in^2}$

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
d = 6.187 in (per drawing ME-443054)
F: Correction factor
f_r: Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{14}(E_1t – Ft_r)-2t_n(E_1t – Ft_r) \times (1-f_{r1})$
$= 5” \times (1 \times 0.375” – 1.0 \times 0.25”) – 2 \times 0.219” \times 1.0 \times .125” \times 0.053$
$= (0.625 - 0.0058)\ in^2$
$= 0.6192\ in^2$

or $= 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 2\ (0.375" + 0.219") \times (1.0 \times 0.375" - 1.0 \times 0.25") - 2 \times 0.219" \times 1.0 \times .125" \times 0.053$
$= (0.1485 - 0.0058)\ in^2$
$= 0.1427\ in^2$

So pick $\mathbf{A_1 = 0.6192\ in^2}$

where:

$d_{14} \approx 2 \times 71\% \times (\Pi\ d_o \times 10.9^o) \div 360^o = 5.08"$, per UG-42 (a)(1) and section C-C of drawing ME-443054. 71% is the portion of the cross section can be considered between two openings (#24 & #14) by the ratio of the diameters.

A_2: Smaller of the following:
$= 5\ (t_n-t_{rn})tf_{r2}$
$= 5 \times (0.219in - 0.017\ in) \times 0.375\ in \times 0.947$
$= 0.3587in^2$

or $= 5\ (t_n-t_{rn})\ f_{r2}t_n$
$= 5 \times (0.219\ in - 0.017\ in) \times 0.947 \times 0.219\ in$
$= 0.2095\ in^2$

So pick $\mathbf{A_2 = 0.2095\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld
$= leg^2\ f_{r2}$
$= 0.21^2\ in^2 \times 1.0$
$= \mathbf{0.044\ in^2}$

A_{43}: Inward nozzle weld
$= leg^2\ f_{r2}$
$= 0.21^2\ in^2 \times 1.0$
$= \mathbf{0.044in^2}$

Then:
$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$
$= (0.6192 + 0.2095 + 0 + 0.044 + 0.044)\ in^2$
$= \mathbf{0.9167\ in^2} > A_r = 0.7792\ in^2$

Per Fig. UG-37.1, section UG-37 and UG-42, the opening is adequately reinforced under the boundary conditions.

Notes:

1. The actual $t_r = 0.13"$ in the upstream end area after adding the stiffening rib as shown in figure x.1, that leads even smaller value of A_r and larger value of A_a,

so the additional reinforcing element is not necessary.

2. The analysis of the above is based on the opening in the upstream end of the Vessel. Per sections A-A and C-C of drawing ME-443054, sections A-A and K-K of drawing ME-443027, #14 opening in the downstream end of the vessel has larger space distance to its neighboring opening, to this end, we conclude that #14 opening at the downstream end is adequately reinforced under the current design configuration.

4.8 Calculations for the Opening of Port #20 for the Reinforcement and Weld Size

The main vessel shell:

From section x.x, it is found that **t_r = 0.130”**

The port #20:

d_o: 4.50”, outside dia. of the port.
d_i: 4.26”, inside dia . of the port.
t_n: 0.120”, wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 30 x 10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: Section J-J of ME-443027, and also as shown from figure 4-4.3 and figure 4-8.1..

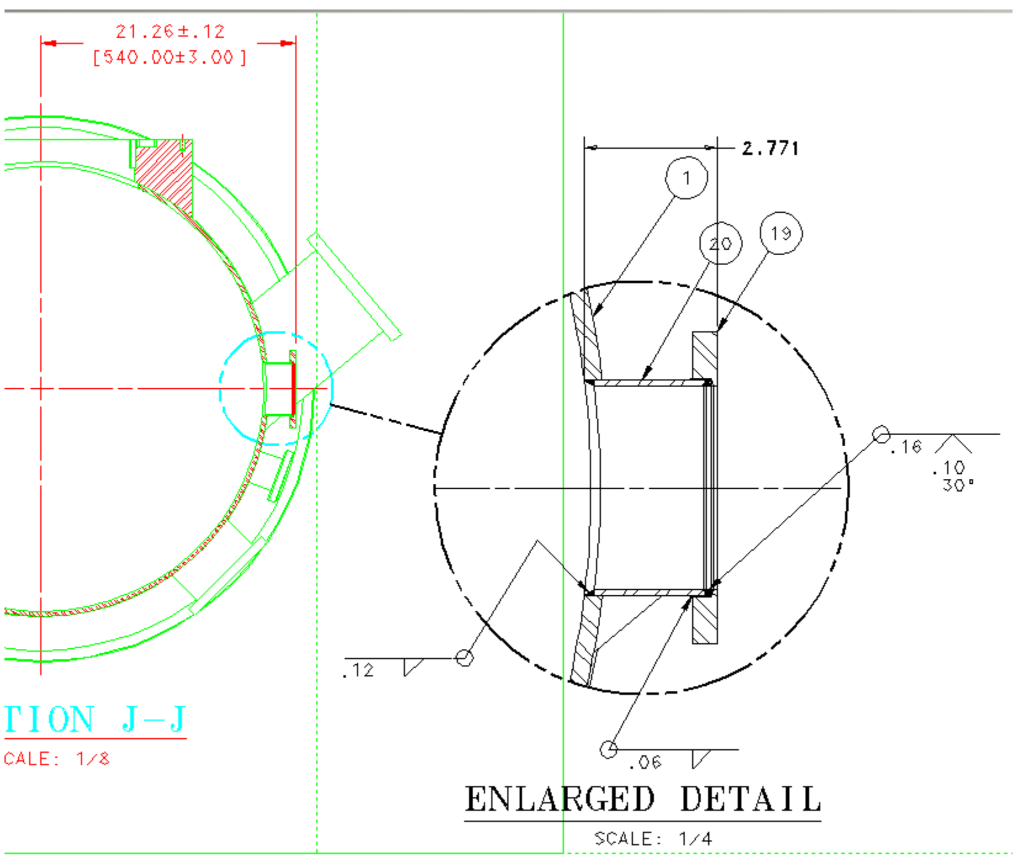

Figure 4-8.1, The 2D view of the port #20 from drawing ME-443027-2

Find out the wall thickness required:

1. Assuming L_1 = 2.771 in, t_{r1} = 0.035 in, then:

   L_1/d_o = 2.771/4.50 = 0.6158
   d_o/t_{r1} = 4.50/0.035 = 129

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   A = 0.0015
   B = 9500
   $P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 9500)/(3 \times 129) = 98$ psi > P

3. Try: L_2 = 2.771 in, t_{r2} = 0.025 in

   L_2/d_o = 0.6158
   d_o/t_{r2} = 180

   get:
   A = 0.00089
   B = 8400
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 8400)/(3 \times 180) = 62.2$ psi > P

3. Try: L_3 = 2.771 in, t_{r3} = 0.013 in

   L_2/d_o = 0.6158
   d_o/t_{r3} = 375

   get:
   A = 0.00032
   B = 4400
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 4400)/(3 \times 375) = 15.64$ psi > ≈ P

So: $\mathbf{t_{rn} = t_{r3} = 0.013}$ **in**

Size of the weld required:

Per UG-37, UW-16 (d.1) and Fig. UW-16.1(k) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $\geq 0.7t_{min}$.
$t_{i:}$ the throat dimension of the inside of the shell cutout, $\geq 0.7t_{min}$
$t_o = 0.7\ t_{min} = 0.7 \times 0.120” = 0.084”$, **so the fillet leg size is 0.119”**
$t_i = 0.7\ t_{min} = 0.7 \times 0.120” = 0.084”$, **so the fillet leg size is 0.119”**
$t_o + t_i = 0.084” + 0.084” = 0.168” \geq 1.25\ t_{min} = 0.15”$

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1}))$
$= 0.5 \times (4.26" \times 0.13" \times 1.0 + 2 \times 0.120" \times 0.13" \times 1.0 \times 0.053)$
$= 0.5 \times (0.5538 + 0.0017)\ in^2$
$=$ **$0.2778\ in^2$**

Where:

d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
$d = 4.26$ in (per drawing ME-443054)

F: Correction factor

f_r : Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

__Area of reinforcement available A_a__:

A_1: Larger of the following calculated value: (area available in shell):
$= d_{20}(E_1t - Ft_r)-2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 3.346" \times (1 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.120" \times 1.0 \times .245" \times 0.053$
$= (0.8198 - 0.0031)\ in^2$
$= 0.8167\ in^2$

or $= 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 2\ (0.375" + 0.120") \times (1.0 \times 0.375" - 1.0 \times 0.13") - 2 \times 0.120" \times 1.0 \times .245" \times 0.053$
$= (0.2426 - 0.0031)\ in^2$
$= 0.2395\ in^2$

So pick **$A_1 = 0.8167\ in^2$**

Where: $d_{20} \approx 2 \times 27\% \times 6.1975 = 3.346"$, per UG-42 (a)(1) and front view of drawing ME-443054. 27% is the portion of the cross section can be considered between two openings (#16 & #20) by the ratio of the diameters.
6.1975" = 14.036" (ctr. to ctr. distance of the openings #20 & #16) – 0.5 x 15.677"

A_2: Smaller of the following:
$= 5\ (t_n-t_{rn})tf_{r2}$
$= 5 \times (0.120\ in - 0.013\ in) \times 0.375\ in \times 0.947$
$= 0.190\ in^2$

or $= 5\ (t_n-t_{rn})\ f_{r2}t_n$
$= 5 \times (0.120\ in - 0.013\ in) \times 0.947 \times 0.120\ in$
$= 0.0608\ in^2$

So pick **$A_2 = 0.0608\ in^2$**

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld
$= leg^2\ f_{r2}$
$= 0.12^2\ in^2 \times 1.0$
$= \mathbf{0.014\ in^2}$

A_{43}: Inward nozzle weld
$= leg^2\ f_{r2}$
$= 0.12^2\ in^2 \times 1.0$
$= \mathbf{0.014 in^2}$

Then:
$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$
$= (0.8167 + 0.0608 + 0 + 0.014 + 0.014)\ in^2$
$= \mathbf{0.9055\ in^2} > A_r = 0.2778\ in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the designated boundary conditions.

5. Analysis and Calculations for the Flanges,Their Bolts and Welds

5.1 Upstream End Flange, Sliding Flange and Bellow Flange

References: ME-443027-1, MD-443039, MD-443038,
TTF-CRY3-01.01.00, TTF-CRY3-01.03.00,
and Figure 5.1

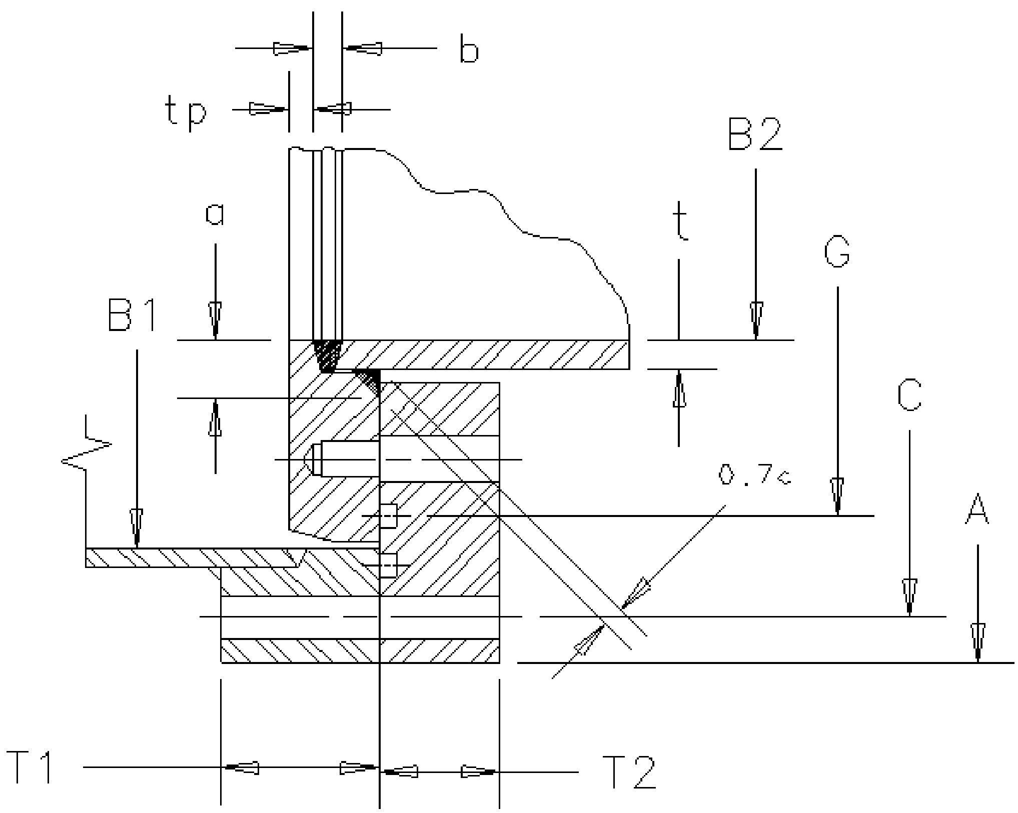

Figure 5.1, Upstream end flanges with the below flange

Figure 5.1 is the assembly configuration of the upstream flange connecting with the below flange, where the upstream flange is assumed as the combination of the flange MD-443038 and MD-443039.

It was found that:

A = 45.669" (dia.)
B_1 = 42.677" (dia.)
B_2 = 37.25" (dia.)

C = 44.488” (dia.)
G = 43.15” (41.85) (dia.)*
T_1 = 1.654”
T_2 = 1.181”
P = 14.7 psi
n = 24
$E_1 = E_2 = 29 \times 10^6$ psi
*: There are 2 o’ring grooves, conservatively pick the larger one.
Material: SA 182-F304L stainless steel, $F_a = S_f$= 16,700 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
assuming bolt spec: ½ - 13 UNC
$A_b = 24 \times 0.1257\ in^2 = 3.0168\ in^2$
The cross-sectional area of the bolts using the root dia. (0.4001”) of the thread in sq. in.

Per section Y-5.1, Y-5.2, figure Y-5.1.2 of Appendix Y, Section 2-4, figure 2-4 of Appendix 2 of ASME VIII:

B_1 > ½ C,
So it is defined as: Class 2, Category 3 flange

Flange Analysis:

The moment due to the pressure M'_p:

$M'_p = H_D'h_D' + H_T'h_T'$ (per Y-6.2(3), Appendix Y, ASME VIII)

$H_D' = 0.785\ B_2^2\ P$
$= 0.785 \times 37.25^2\ in^2 \times 14.7$ psi
= 16,012 lbs.

$H_T' = 0.785P\ (B_1^2 - B_2^2)$
$= 0.785 \times 14.7$ psi $(42.677^2 - 37.25^2)\ in^2$
= 5,006 lbs.

$h_D' = (B_1 - B_2) \div 2$
= (42.677 – 37.25) in ÷ 2
= 2.7135 in

$h_T' = (B_1 - B_2) \div 4$
= (42.677 – 37.25) in ÷ 4
= 1.3568 in

So: M'_p = (16,012 x 2.7135) lbs-in + (5,006 x 1.3568) lbs-in
= 50,241 lbs-in

$= C_5$

(per section Y-6.2(4)(a), Appendix Y, ASME VIII, Div.1)

Let: $C_6 = 0.829 / \log(B_1/B_2)$

$= 0.829 \div 0.059$

$= 14.051$

(per section Y-6.2(4)(b), Appendix Y, ASME VIII, Div.1)

$C_1 = [1 – 2.095 J_s \log(A/B_1)] \div (- C_6 – 1.738\ J_s)$

(per section Y-6.2(4)(c), Appendix Y, ASME VIII, Div.1)

Where:

$J_s = (1/B_1)\ [2h_D/\beta + h_C/\alpha] + \pi\ r_B$

$h_D = (C – B) / 2$

$= (44.488 – 42.677) / 2$ in

$= 0.9055$ in

$h_C = (A – C) / 2$

$= (45.669 – 44.488) / 2$ in

$= 0.5905$ in

$\beta = (C + B)/ 2B$

$= (44.488 + 42.677) / (2 \times 42.677)$

$= 1.021$

$\alpha = (A + C)/ 2B$

$= (45.669 + 44.488) / (2 \times 42.677)$

$= 1.056$

$r_B = (1/n)[4/(1 – (\overline{A}\overline{R})^2)^{1/2}\ \ \tan^{-1}((1+ \overline{A}\overline{R})/(1 - \overline{A}\overline{R}))^{1/2} – \pi -2 \overline{A}\overline{R}]$

where:

$\overline{A}\overline{R} = nD/ \pi C$

$= (24 \times 0.551”) / \pi\ 44.488”$

$= 0.0946$

D: the bolt hole diameter

$r_B = (1/24)[4/(1 – (0.0946)^2)^{1/2}\ \ \tan^{-1}((1+ 0.0946)/(1 – 0.0946))^{1/2} – \pi –$
$2 \times 0.0946]$

$= 0.041667 \times (4.018 \times 47.7141- \pi – 0.1892)$

$= 7.849$

$J_s = (1/42.677”)\ [2 \times 0.9055/1.021 + 0.5905/1.056] + \pi \times 7.849$

$= 0.0234 \times (1.774 + 0.559) + 24.658$

$= 24.713$

So:

$C_1 = [1 – 2.095 \times 24.713 \times \log(45.669/42.677)] \div (- 14.051 – 1.738 \times 24.713)$

$= (1 – 1.5236) \div (-57.0022)$

$= 0.0091$

$C2 = (1.738 J_p M_p – C_5 C_6) \div (- C_6 – 1.738\ J_s)$
(per section Y-6.2(4)(d), Appendix Y, ASME VIII, Div.1)

where:

$J_p = (1/B_1)\ [h_D/\beta + h_C/\alpha] + \pi\ r_B$
$= (1/42.677”)\ [0.9055/1.021 + 0.5905/1.056] + \pi \text{ x } 7.849$
$= 0.0234 \text{ x } (0.8869 + 0.559) + 24.658$
$= 24.692$

$M_p = H_D h_D + H_T h_T + H_G h_G$ (per sectionY-3, Appendix Y, ASME VIII)

$H_D = 0.785\ B_1^2\ P$
$= 0.785 \text{ x } 42.677^2 \text{ in}^2 \text{ x } 14.7 \text{ psi}$
$= 21{,}017$ lbs.

$H = 0.785 G^2 P$
$= 0.785 \text{ x } 43.15^2 \text{ in}^2 \text{ x } 14.7 \text{ psi}$
$= 21{,}486$ lbs.

$H_T = H - H_D$
$= 21{,}486 – 21{,}017$
$= 469$ lbs.

$h_G = (C – G)/2$
$= (44.488 – 43.15) \text{ in} / 2$
$= 0.669$ in

$h_D = 0.9055$ in

$h_T = (h_D + h_G) / 2$
$= 0.7873$ in

$H_G = W – H$, gasket load
(per section 2-3, appendix 2, ASME VIII)
$W = H + H_p$, flange design load for the operating condition
(per section 2-5(c)(1), appendix 2, ASME VIII)

which leads to:

$H_G = W – H$
$= H + H_p – H$
$= H_p$
$= 2b \text{ x } 3.14 GmP$
$= 2 \text{ x } 0.315 \text{ in x } 3.14 \text{ x } 43.15 \text{ in x } 6.00 \text{ x } 14.7 \text{ psi}$
$= 7{,}529$ lbs.

$M_p = H_D h_D + H_T h_T + H_G h_G$

$= (21{,}017 \times 0.9055 + 489 \times 0.7873 + 7{,}529 \times 0.669)$ lbs-in
$= (19{,}031 + 385 + 5{,}037)$ lbs-in
$= 24{,}453$ lbs-in

$C2 = (1.738 J_p M_p - C_5 C_6) \div (- C_6 - 1.738\ J_s)$
$= (1.738 \times 24.692 \times 24{,}453 - 14.051 \times 52{,}041)$ lbs-in $\div (-14.051 - 1.738 \times 24.713)$
$= 318{,}165$ lbs-in / $(- 57.002)$
$= - 5{,}582$ lbs-in

For the rigid body rotation of the flange times E*:

$E_1^* \theta_{rbi} = X(C_4 - C_2) \div (1.206 \log(A/B_1) - XC_3 - (1 - X)C_1)$

where:

$C_3 = - (0.575 - 1.206\ Js \log(A/B_1)) \div (Js + t_1^3/F_1')$
$\because\ F_1' = 0$ (per section Y-3(6c), Appendix Y, ASME VIII, Div.1)
$\therefore\ C_3 = - (0.575 - 1.206\ Js \log(A/B_1)) \div (Js)$
$= - (0.575 - 1.206 \times 24.713 \times 0.0294) / 24.713$
$= 0.0122$

$C4 = - (J_p M_p) \div (Js + t_1^3/F_1')$
$= - (24.692 \times 24{,}453$ lbs-in$) / 24.713$
$= - 24{,}432$ lbs-in

$X = E_1^* / (E_1^* + E_2^*)$
$E_1^* = E_1\ t_1^3$
$E_2^* = E_2\ t_2^3$
$\because\ E_1 = E_2$
$\therefore\ X = E_1\ t_1^3 / (E_1\ t_1^3 + E_2\ t_2^3)$
$= t_1^3 / (t_1^3 + t_2^3)$
$= 4.525 / (4.525 + 1.647)$
$= 0.733$

$E_1^* \theta_{rbi} = 0.733(-24{,}432 - (-5{,}582)) \div [(1.206 \times 0.0294) - (0.733 \times 0.0122) - (1 - 0.733) \times 0.00919)]$
$= 0.733 \times (-18{,}850)$ lbs-in $\div (0.0355 - 0.0089 - 0.00245)$
$= - 13{,}817$ lbs-in $\div 0.02415$
$= - 572{,}133$ lbs-in

$E_2^* \theta_{rbii} = - E_1^* \theta_{rbi} (E_2^* / (E_1^*)$
$= - (-572{,}133 \times 0.3635)$ lbs-in
$= 207{,}970$ lbs-in

Total Flange Moment at Diameter B_1:

$M_{s1} = C_3 (E_1 * \theta_{rbi}) + C_4$
$= 0.0122 \times (- 572{,}133 \text{ lbs-in}) + (- 24{,}432 \text{ lbs-in})$
$= - 31{,}412 \text{ lbs-in}$
$M_{s2} = C_1 (E_2 * \theta_{rbii}) + C_2$
$= 0.00919 \times 207{,}970 \text{ lbs-in} + (- 5{,}582 \text{ lbs-in})$
$= - 3{,}671 \text{ lbs-in}$

Unbalanced Flange Moment at Diameter B_1:

$M_{u1} = 1.206 (E_1 * \theta_{rbi}) \log(A/B_1)$
$= 1.206 \times (- 572{,}133 \text{ lbs-in}) \times 0.0294$
$= - 20{,}286 \text{ lbs-in}$
$M_{u2} = 1.206 (E_2 * \theta_{rbii}) \log(A/B_1)$
$= 1.206 \times (207{,}970 \text{ lbs-in}) \times 0.0294$
$= 7{,}374 \text{ lbs-in}$

Balanced Flange Moment at Diameter B_1:

$M_{b1} = M_{s1} - M_{u1}$
$= - 31{,}412 \text{ lbs-in} - (- 20{,}286 \text{ lbs-in})$
$= - 11{,}126 \text{ lbs-in}$
$M_{b2} = M_{s2} - M_{u2}$
$= - 3{,}671 \text{ lbs-in} – 7{,}374 \text{ lbs-in}$
$= - 11{,}045 \text{ lbs-in}$

Slope of Flange at Diameter B_1 times E:

$E_1\ \theta_{Bi} = (5.46/\pi\ t_1^3) (J_s M_{b1} + J_p M_p) + E_1 * \theta_{rbi} / t_1^3$
$= 0.384 \text{ in}^{-3} \times (24.713 \times (– 11{,}126 \text{ lbs-in}) + 24.692 \times 24{,}453 \text{ lbs-in}) + (-$
$572{,}133 \text{ lbs-in} / 4.525 \text{ in}^3)$
$= (126{,}273 + (-126{,}438)) \text{ psi}$
$= -165 \text{ psi}$
$E_2\ \theta_{Bii} = (5.46/\pi\ t_2^3) (J_s M_{b2} + J_p M_p) + E_2 * \theta_{rbii} / t_2^3$
(per section Y-6.2(e), Appendix Y, ASME VIII, Div.1)
$= [1.055 \times (24.713 \times (-11{,}045) + 24.692 \times 24{,}453) + 207{,}970 / 1.647] \text{ psi}$
$= (349{,}035 + 126{,}272) \text{ psi}$
$= 475{,}307 \text{ psi}$

Contact Force between Flange at h_c:

$H_c = (M_p + M_{b1}) / h_c$
$= (24{,}453 +(- 11{,}126)) \text{ lbs-in} / 0.5905 \text{ in}$
$= 22{,}569 \text{ lbs.}$

Bolt Load at Operating Conditions:

$W_{m1} = H + H_G + H_C$
$=(21{,}486 + 7{,}529 + 22{,}569)$ lbs
$= 51{,}584$ lbs.

Operating Bolt Stress f_b:

$f_b = W_{m1} / A_{b1}$
$= 51{,}584 \text{ lbs} / 3.0168 \text{ in}^2$ (See page 52, section 5.1 of the note)
$= \underline{17{,}099 \text{ psi}}$

Design Pre-stress in Bolts:

$S_i = f_b - 1.159\, h_c^2\, (M_p + M_{b1}) / (2(1 - X)a\, t_1^3\, l\, r_{e1} B_1)$
where:
$r_{e1:} \approx 1.0$, elasticity factor, the ratio between E_{flange} and E_{bolt}
$a = (A + C) / 2B_1$
$= (45.669 + 44.488) \text{ in} / 2 \times 42.677 \text{ in}$
$= 1.056$ (shape factor)
$l: = t_1 + t_2 + 0.5d_b$ (calculated strain length of the bolt)
$= (1.654 + 1.181 + 0.25)$ in
$= 3.085$ in

$S_i = 17{,}099 \text{ psi} - 1.159 \times 0.5905^2 \text{ in}^2 \times (24{,}453 - 11{,}126)\text{lbs-in} / (2(1 - 0.733) \times 1.056 \times 1.654^3 \text{ in}^3 \times 3.085 \text{ in} \times 1.0 \times 42.677 \text{ in})$
$= 17{,}099 \text{ psi} - 5{,}386 \text{ lbs-in}^3 / 336 \text{ in}^5$
$= 17{,}083$ psi

Redial Stress in Flange I @ Bolt Circle:

$S_{R1} = 6(M_p + M_{S1}) / t_1^2(\pi C - nD)$
$= 6 \times (24{,}453 - 31{,}412) \text{ lbs-in} / 1.654^2 \text{ in}^2 \times (\pi\, 44.488 \text{ in} - 24 \times 0.551 \text{ in})$
$= - 41{,}754 \text{ lbs-in} / 346.175 \text{ in}^3$
$= | -121 | \text{ psi} < S_n = 16{,}700$ psi

Tangential Stress in Flange I @ Inside Diameter:

$S_{t1} = t_1\, E_1\, \theta_{Bi} / B_1$
$= 1.654 \text{ in} \times -165 \text{ psi} / 42.677 \text{ in}$
$= |-6.39 | \text{ psi} < S_n = 16{,}700$ psi

Tangential Flange Stress Adjacent to Central Nozzle:

$S_{t2n} = Y(M'_p + M_{S2}) / t_2^2\, B_2$
where:
$Y = (1/(K-1))[0.66845 + 5.71690\, K^2 \log_{10}K/(K^2 -1)]$
(per figure 2-7.1, Appendix 2, ASME VIII, Div. 1)

$$K = A/B$$
$$= 45.669 / 37.25$$
$$= 1.226$$
$$Y = (1/(1.226-1))[0.66845 + 5.71690 \times 1.5031 \times \log_{10}K/(K^2 -1)]$$
$$= 4.4248 \times (0.6685 + 5.7169 \times 0.2643)$$
$$= 9.6437$$

$$\therefore\ S_{t2} = 9.6437 \times (50{,}241 - 3{,}671) \text{ lbs-in} / (1.181^2 \times 37.25) \text{ in}^3$$
$$= 8{,}644 \text{ psi} < S_n = 16{,}700 \text{ psi}$$

Longitudinal Hub Stress in Flange 1:

$$S_{h1} = 0$$
(per equation (34c), section Y-6.3, appendix Y, ASME VIII, Div.1)

Radial Stress in Flange 2 @ bolt circle:

$$S_{r2b} = 6(M_p + M_{S2}) / t_2^2(\pi C - nD)$$
$$= 6 \times (24{,}453 - 3{,}671) \text{ lbs-in} / 1{,}181^2 \text{ in}^2 \times (\pi\ 44.488 \text{ in} - 24 \times 0.551 \text{ in})$$
$$= 124{,}692 \text{ lbs-in} / 176.492 \text{ in}^3$$
$$= 707 \text{ psi} < S_n = 16{,}700 \text{ psi}$$

Radial Stress in Flange 2 @ Diameter B_1:

$$S_{r2B1} = 6\ M_{S2} / t_2^2\ \pi B_1$$
$$= 6 \times (-3{,}671) \text{ lbs-in} / 1.181^2 \text{ in}^2 \times \pi\ 42.677 \text{ in}$$
$$= |-118| \text{ psi} < S_n = 16{,}700 \text{ psi}$$

Tangential Stress in Flange 2 @ Diameter B_1:

$$S_{t2B1} = t_2\ E_2\ \theta_{Bii} / B_1 - 1.8\ M_{S2} / t_2^2\ \pi B_1$$
$$= (1.181 \text{ in} \times 475{,}307 \text{ psi} / 42.677 \text{ in}) - (1.8 \times (-3{,}671 \text{ lbs-in})) / 1.181^2 \text{ in}^2\ \pi\ 42.677 \text{ in}$$
$$= 13{,}153 \text{ psi} + 35 \text{ psi}$$
$$= 13{,}188 \text{ psi} < S_n = 16{,}700 \text{ psi}$$

Radial and Tangential Stress at Center of Flange 2:

$$S_{r2} = S_{t2} = (0.3094\ PB_1^2 / t_2^2) - (6\ M_{S2} / \pi B_1\ t_2^2)$$
$$= (0.3094 \times 14.7 \text{ psi} \times 42.677^2 \text{ in}^2 / 1.181^2 \text{ in}^2) - (6 \times (-3{,}671 \text{ lbs-in}) / \pi\ 42.677 \text{ in} \times 1{,}181^2 \text{ in}^2)$$
$$= 5{,}939 \text{ psi} + 118 \text{ psi}$$
$$= 6{,}057 \text{ psi} < S_n = 16{,}700 \text{ psi}$$

<u>*Welds Analysis and Calculations:*</u>

See UW-13 (e)(2), Fig. UW-13.2, section 2-3 and Fig. 2-4(4a), Appendix 2 of ASME VIII, Div.1. for the related codes and the physical representation of the nomenclatures.

Per above codes, it is required that the throat of the fillet $c = 0.7t_{min.} = 0.26$”,

So the fillet leg size is ~ 0.37”.
However, per the assembly configuration of the upstream flange connecting with the below flange (see drawing TTF-CRY3-01.01.00 & TTF-CRY3-01.03.00), it is found that there only has ~0.19 clearance between flange MD-443038 and MD-443039, so it is suggested to use 0.37” bevel groove instead of 0.37” fillet weld to meet the related code required and to avoid the parts interference.

It is found that:

t_p= 0.32” > 0.25”

where is the smallest of t_n, t_x and 0.25” is 0.25”

$c = t_n$ = 0.375” = leg of the fillet or bevel groove

b = 0.16” + 2 x tan 15^o x 0.43”

= 0.39”

a = 0.43” + 0.37”- 0.06”

= 0.74”

$\therefore$ a + b = 1.13” $\geq 3t_n$ = 1.125”

The weld design for the end flange is ok.

Also run a weld strength calculation for the noted location as the 2nd analysis approach:

Radial pressure load F_{rp}:

$F_{rp} = P\ L_{sd}$

= 14.7 psi x 230 in

= 3,381 lbs/in

Shear flow due to radial shear load F_{sr}:

F_{sr} = VQ/I

where:

Radial shear load V:

$V = 0.01\ P\ L_{sd}\ D_o$

= 0.01 x 14.7 psi x 230 in x 38,0 in

= 1,285 lbs.

The 1st moment of area Q:

Q = 2.67 in x 0.375 in x (1.1454 – 0.1875) in

= 0.959 in^3

I = 2.4101 in^4

$F_{sr} = VQ/I$
$= 1{,}285$ lbs. x 0.959 in^3 / 2.4101 in^4
$=$ 511 lbs/in

The required resultant weld load F_{rw}:
$F_{rw} = (3{,}381^2 + 511^2)^{1/2}$ lbs/in
$=$ 3,419 lbs/in

The welds for connecting the flange and the vessel shell is single groove weld inside and intermittent fillet outside, the lowest allowable weld stress F_{aw}:

$F_{aw} = 0.60\ S_n = 0.60 \times 16.7$ ksi
$= 10.02$ ksi

The requires calculated minimum weld size $C_{m:}$
$C_m = F_{rw} / F_{aw}$
$= 3{,}419 / 10{,}020$ (in)
$= 0.341$ in < 0.43 in (designated single groove weld)
If adding the value of the outside intermittent bevel groove weld (0.0.37”), the weld size even can go further smaller size.
So, the design of the weld for the upstream end flange is satisfactory.

Per the above calculations and analysis, the upstream end flange and the bellow flange have met the codes of ASME VIII, Div.1.

5.2. Downstream End Flange and Bellow Flange

References: ME-443027-1, MC - 443040,
TTF-CRY3-01.01.00, TTF-CRY3-01.03.00

Since there some information are missing at this moment, so it is to make the best assumption as followings:

1. The downstream end flange MC-443040 connect to the upstream end flange of the bellow (see drawing TTF-CRY3-01.03.00).
2. The above two flanges are connected by the clamps, the center line of the clamp C = 43.78" + (45.47 – 43.78") / 2 = 44.625".
3. A = 45.669", G = 44.488", T_1 = 1.57", T_2 = 1.57", B_1 = 42.677", B_2 = 37.37.25".
4. n = 12, D = 0.55"

Material: SA 182-F304L stainless steel, $F_a = S_f$= 16,700 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).

Per section Y-5.1, Y-5.2, figure Y-5.1.2 of Appendix Y,
Section 2-4, figure 2-4 of Appendix 2 of ASME VIII:

$B_1 > ½ C$,
So it is defined as: Class 2, Category 3 flange

In order to reduce the amount of work required as shown on section 5.1, per section Y-9, of Appendix Y, ASME VIII, Div.1, we apply the method of estimating flange thickness and bolting for the downstream flanges.

$t_a = 2.45\, [Mp / ((\pi C – nD)S_f)]^{1/2}$
(per eq. 39 of Y-9, appendix Y, ASME VIII, Div.1)

where:
$M_p = H_Dh_D + H_Th_T + H_Gh_G$ (per sectionY-3, Appendix Y, ASME VIII)

$H_D = 0.785\, B_1^2\, P$
$= 0.785 \times 42.677^2\ in^2 \times 14.7$ psi
= 21,017 lbs.

$H = 0.785G^2P$
$= 0.785 \times 44.488^2\ in^2 \times 14.7$ psi
= 22,839 lbs.

$H_T = H - H_D$
= 22,839 – 21,017
= 1,822 lbs.

$h_G = (C – G)/2$
$= (44.625 – 44.488)\text{in} / 2$
$= 0.0685$ in

$h_D = (C – B) / 2$
$= (44.625 – 42.677) / 2$
$= 0.974$ in

$h_T = (h_D + h_G) / 2$
$= 0.5213$ in

$H_G = W – H$, gasket load
(per section 2-3, appendix 2, ASME VIII)
$W = H + H_p$,
flange design load for the operating condition
(per section 2-5(c)(1), appendix 2, ASME VIII)

which leads to:

$H_G = W – H$
$= H + H_p – H$
$= H_p$
$= 2b \times 3.14GmP$
$= 2 \times 0.315$ in $\times 3.14 \times 44.488$ in $\times 6.00 \times 14.7$ psi
$= 7{,}766$ lbs.

$M_p = H_Dh_D + H_Th_T + H_Gh_G$
$= (21{,}017 \times 0.974 + 1{,}822 \times 0.5213 + 7{,}766 \times 0.0685)$ lbs-in
$= (20{,}471 + 950 + 532)$ lbs-in
$= 21{,}953$ lbs-in

$t_a = 2.45\,[Mp / ((\pi C – nD)S_f)]^{1/2}$
$= 2.45 \times [\ 21{,}953 \text{ lbs-in} / ((\pi\ 44.625 \text{ in} – 24 \times 0.551 \text{ in}) \times 16{,}700 \text{ psi})]^{1/2}$
$= 2.45 \times 0.102$ in
$= 0.250$ in

$t_b = 0.56\ B_1\ (P/S_f)^{1/2}$
$= 0.56 \times 42.677 \text{ in} \times (14.7 \text{ psi} / 16{,}700 \text{ psi})^{1/2}$
$= 0.56 \times 42.677 \text{ in} \times 0.0297$
$= 0.71$ in

Since t_c is the greater of t_a or t_b, so:
$t_c = t_b = 0.71$”

Per equation 42 of A-7, Appendix Y, ASME VIII, Div. 1:

$A'_b = (H + 2\ M_p / (A – C)) \div S_b$

$= (22,839 \text{ lbs} + 2 \times 21,953 \text{ lbs-in} / (45.669 – 44.625) \text{ in}) \div 20,000 \text{ psi}$
$= (22,839 \text{ lbs} + 42,056 \text{ lbs}) / 20,000 \text{ psi}$
$= 3.245 \text{ in}^2$

Per Table Y-9.1, Appendix Y, ASME VIII, Div.1, it is found:

$t_1 = 1.1t_a = 0.275” < T_1 = 1.57”$
$t_2 = 1.1t_c = 0.781” < T_2 = 1.57”$

So the thickness of the flanges are ok for the downstream end flanges.

Welds Analysis and Calculations:

The weld configurations are the same as the upstream flange, it designed per section UW-13 (e)(2), Fig. UW-13.2, section 2-3 and Fig. 2-4(4a), Appendix 2 of ASME VIII, Div.1

Also, since the valus of L_{sd} for downstream end flange is ~45” (~230” for upstream end flange), I value for downstream end flange is large than the upstream flange, all these lead to a smaller values of F_{rw} and C_m than the upstream end flange requires, however, the designated welds of the downstream end flange are the same as the upstream end flange, so the welds design are satisfactory.

Per the above calculations, the upstream end flange and the bellow flange have met the codes of ASME VIII, Div.1.

5.3. Flanges #8, #11, #13, #15, #19, #21 & #23

In order to reduce the amount of work required as shown on section 5.1, per section Y-9, of Appendix Y, ASME VIII, Div.1, we apply the method of estimating flange thickness and bolting for the above flanges.

Since the information of the most mating flanges are not available at this moment, I made some engineering judgments per the best of my logic knowledge.

Table 5.3, Estimated Calculated Flange thickness and bolting VS. the designated data

nomenclature	unit	Flanges (see drawing ME-443027)						
		#8	#11	#13	#15	#19	#21	#23
		MC-443037	MC-443034	MC-443036	MC-443031	MC-443033	MC-443035	MC-443032
P	psi	14.7	14.7	14.7	14.7	14.7	14.7	14.7
S_f	psi	18,800	18,800	18,800	18,800	18,800	18,800	18,800
A	in	10.945	5.512	8.67	13.78	6.535	11.22	4.472
B_1	in	8.386	3.334	6.194	11.418	4.26	8.386	2.522
B_2	in	3.75	3.334	0	0	0	0	0
C	in	9.646	4.882	7.874	12.99	5.748	10.236	3.628
G	in	8.793	4.012	7.034	12.204	5.004	9.311	3.024
Classification		3	1	3	3	3	3	3
Categorization		3	3	3	3	3	3	3
n		24	4	8	8	6	12	8
D	in	0.3125	0.394	0.394	0.394	0.394	0.335	0.335
m		6	6	6	6	6	6	6
b		0.21	0.21	0.21	0.21	0.21	0.21	0.21
H_D	lbs	812	128	443	1504	209	812	73
H	lbs	892	186	571	1719	289	1000	106
H_T	lbs	81	57	128	214	80	189	32
h_G	in	0.4265	0.435	0.42	0.393	0.372	0.4625	0.302
h_D	in	0.63	0.774	0.84	0.786	0.744	0.925	0.553
h_T	in	0.5283	0.6045	0.6300	0.5895	0.5580	0.6938	0.4275
H_G	lbs	1023	467	818	1420	582	1083	352
M_P	lbs-in	990	337	796	1867	417	1383	161
t_a	in	0.1178	0.0885	0.1086	0.1258	0.0921	0.1253	0.0767
t_b	in	0.1313	0.0522	0.0970	0.1788	0.0667	0.1313	0.0395
t_c	in	0.1313	0.0885	0.1086	0.1788	0.0921	0.1313	0.0767
A'_b	in^2	0.1285	0.0668	0.1368	0.3428	0.0717	0.2027	0.0258
t_1	in	0.1296	0.0885	0.1194	0.1384	0.1013	0.1378	0.0844
t_2	in	0.1444	0.0885	0.1194	0.1967	0.1013	0.1444	0.0844
A_b	in^2	0.1350	0.0668	0.1436	0.3599	0.0753	0.2128	0.0271
t_{1d}	in	0.787	0.512	0.709	0.79	0.51	0.984	0.68
t_{2d}	in	N/A	0.512	N/A	N/A	N/A	N/A	N/A
A_{bd}	in^2	1.088	0.271	0.542	0.542	0.4065	TBA	TBA

The most representation of nomenclature of the Table can be found from section 5.1 of this note, also from the appendix 2 and appendix Y of ASME VIII, Div.1, except:

t_1, Estimated calculated flange 1 thickness
t_2, Estimated calculated flange 2 thickness
A_b, Estimated calculated root diameter area of the bolts.
$t_{1d,}$ Designated flange 1 thickness
t_{2d}, Dersignated flange 2 thickness
A_{bd}, Designated root diameter area of the bolts.

Per table 5.3, it is found that all $t_1 > t_{1d}$; $t_2 > t_{2d}$ and $A_b > A_{bd}$ for all respective flanges, so the flange design are satisfactory.

Per figure UW-13.2, section UW-13, appendix 2 of ASME VIII, Div. 1 and drawing ME-443027, it is found that all flange welds are designed per the related codes.

6. Calculation and Analysis of the Saddle support

Reference drawing: ME-443027
Figure: 6.1 and Figure 6.2

Per drawing ME-443027 and figure x.x.1, it is found that:

L_c = 163.50 x 2 = 327 in (the ctr. distance between 2 saddle support.)

W_{ws} = 7,745 lbs. the weight of the vessel shell weldment.

W_{other} = 8,846 lbs.,
the weight of coldmass shields & pipe, 8 cavity string w/mech. turners, individual dressed cavity w/turner, main coupler.

$W_{tot} = 1.05\,(W_{ws} + W_{tot}) \approx 17{,}421$ lbs
(Total weight including 5% mice. parts weight applying to two saddle supports)

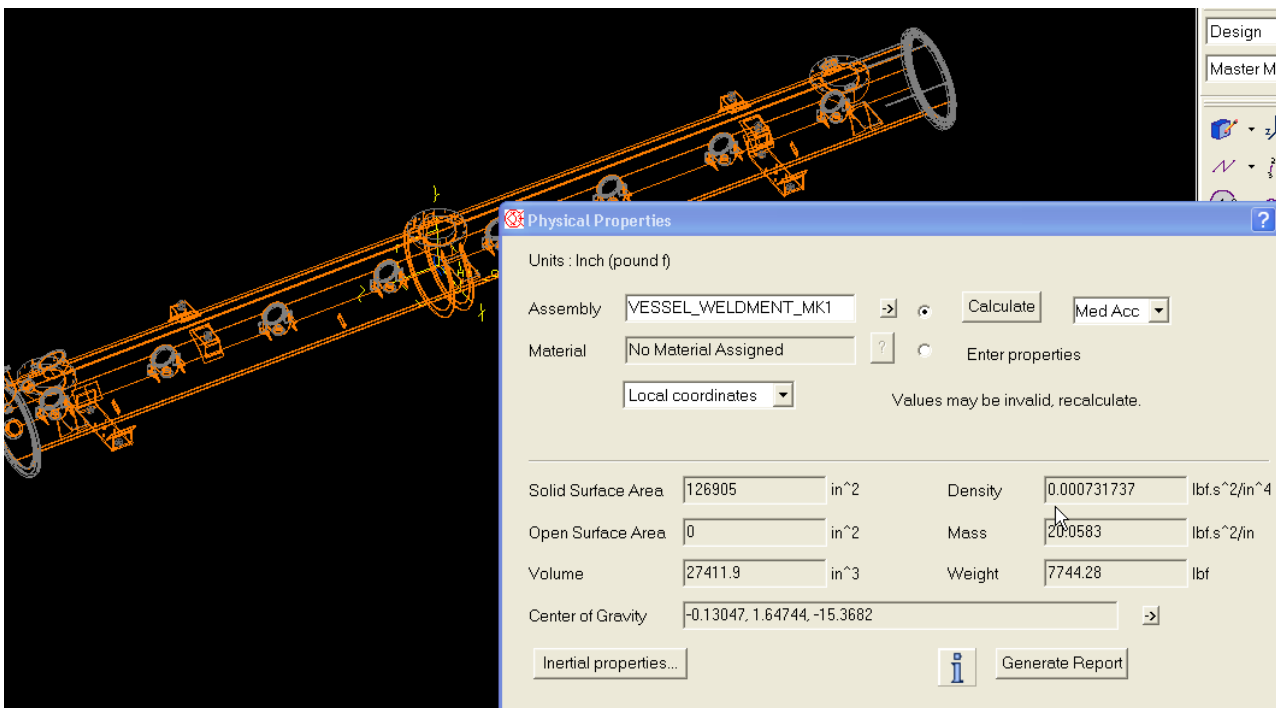

Figure 6.1, <u>The weight and the location of the ctr. of gravity of the vessel shell assembly</u>.

The force distribution diagram on figure x.x.2 is created conservatively based on the information from figure x.x.1 and drawing ME- 443027.

It is found that:

$$Q_1 + Q_2 = 17{,}421 \text{ lbs} = W_{tot}$$

$Q_2 = (W_{tot} \times 148.13) / 327$
the reaction force on the downstream saddle support.
= 7,892 lbs
Q_1 = 9,529 lbs.
the reaction force on the upstream saddle support.
A = 76.854 in
the distance from the end of the upstream end to the ctr, of the 1st saddle support.

$L_{tot} \approx 448$ in
$\Theta = 120^o$, the value of the assuming contact angle of the saddle support.

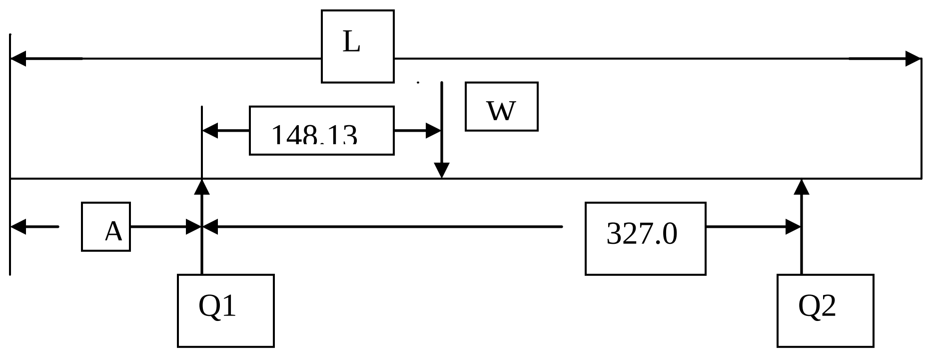

Figure 6.2, The force distribution diagram of the vacuum vessel.

To find the longitudinal stress on the saddle support 1:

$f_{1s} = QA [1 – (1 – A/L + (R^2 – H^2) / 2AL) / (1 + 4H/3L)] / KR^2t_s$

where H = 0,
R = 19.0"
K = constant, from page 88 of Pressure Vessel Handbook, 7th edition,
by E. Megyesy

f_{1s} = 9,529 lbs x 76.854" x [1 - (1 – 76.854 / 448 + (19^2 / 2 x 76.854 x 448))
÷ (1 + 0)] / (0.335 x 19^2 x 0.375 in^3)
= 732,342 lbs-in x [1- (1 – 0.1715 + 0.0052)] ÷ 45.351 in^3
= 2,686 psi

To find the longitudinal stress on the middle of the span:

$f_{1m} = (QL/4) [(1 + 2 (R^2 – H^2) /L^2)) / (1 + 4H/3L) – 4A/L] / KR^2t_s$
= 1,067,248 lbs-in x (1.0036 – 0.6862) ÷ 45.351 in^3
= 7,469 psi

Stress due to the pressure:

$f_p = PR / 2t_s$
$= 14.7$ psi x 19.0 in / 2 x 0.375 in
$= 373$ psi

So, the total longitudinal stress at the middle span of the vessel:

$f_{bm} = f_{1m} + f_p = 7{,}469$ psi + 373 psi
$= 7{,}842$ psi $< 0.80\ F_a = 12$ ksi
where:
$F_{a:}$ allowable stree, see page x.
0.80, joint efficiency, see Table UW-12, section UW-12, ASME VIII, Div.1.

To find the tangential stress on the vessel shell:

$f_{2t} = (K^2Q / Rt_s)(L - 2A) / (L + 1.333H)$
$= ((1.171^2 \times 9{,}529$ lbs./ 19 in x 0.375 in) x (448 – 2 x 76.854) in) ÷ 448 in
$= 1{,}834$ psi x 294.29 in / 448 in
$= 1{,}197$ psi $< 0.80\ F_a = 12$ ksi

To find the circumferential stress on the vessel shell:

$f_{4c} = (Q / (4t_s(b + 1.56(Rt_s)^{1/2}) - 3K_6Q / 2t_s^2$

where: b, the width of the saddle support, assuming 3/4 inch, so:
f_{4c} = [9,529 lbs / 4 x 0.375 in x (0.75 + 1.56 x $(19 \times 0.375)^{1/2}$ in] – (3 x 0.053 x 9,529 lbs / (2 x 0.375^2 in^2))
= -1,293 psi – 5,387 psi
= |- 6,680| psi $< 1.50\ F_a = 22.5$ ksi

So, all the stresses are less than the allowable stress respectively, so the saddle support design is satisfactory (Since the design hasn't finished yet, the calculation and analysis is based on the assumption of: the saddle support has al least 120^o degree contact angle with 3/8 in thick wear plate).

7. Appendix

7.1 The Lists of the Design Drawings of 1.3 MHz, β = 1, Cryomodule Vacuum Vessel (Fermilab)

1. Vacuum Vessel Weldment Assy. ME – 443027 – 1 (2)
2. Vessel Shell ME – 443054
3. Port End Coldmass MD – 443050
4. Port Middle Coldmass MD – 443049
5. Pick Point Weldment MC – 443028
6. Pad Weld Upper MC – 443043
7. Vessel Support Weldment MC – 443029
8. Pad Weld Lower MC – 443042
9. Flange MC Port MC – 443037
10. Instrumentation Flange MC – 443034
11. Flange Side Rib MB - 443051
12. Flange – K1, K3 MC – 443036
13. Flange D MC – 443031
14. MC Bracket Weldment – LH MC – 443058
15. MC Bracket Weldment – RH MC – 443059
16. Flange E.O.D. 166 MC – 443033
17. Flange G.O.D. 285 MC – 443035
18. Flange DN 63 DF MC – 443032
19. Ring Flange Stop MC – 443053
20. Stiffening Rib MC – 443052
21. Pipe Support Plate MC – 443047
22. Plate Welded Outrigger MC – 443048
23. Gusset Welded Outrigger MC – 443041
24. Flange, Fixed Downstream MC – 443040
25. Flange, Fixed Upstream MC – 443039
26. Flange, Bellow Sliding MD – 443038
27. Flange Weld Ring MC – 443060

7.2 The Lists of the Design Drawings of 1.3 MHz, β = 1, Cryomodule Vacuum Vessel (ARES - TTF)

1. Vacuum Vessel Shell Assy.	TTF – CRY3 – 01.01.00
2. Support Post Cover	TTF – CRY3 – 01.01.01
3. Vacuum Vessel Shell Details	TTF – CRY3 – 01.01.02
4. Adjustable Support Assy.	TTF – CRY3 - 01.02.00
5. Adjustable Support Details	TTF – CRY3 – 01.02.01
6. Universal Expansion Joint	TTF – CRY3 – 01.03.00