FERMILAB
Technical Division

Vacuum Vessel Engineering Note For the 3.9-GHz Cryomodule

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

Prepared by	Edward Chi Mayling Wong	Date	09-May-2008	Div/Sec	PPD/MD/ME TD/SRF Dev.
Reviewed by	[signature]	Date	5/23/08	Div/Sec	PPD/MD/ME
Div/Sec Head	[signature]	Date	5 Aug '08	Div/Sec	TD Head

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	*3.9 GHz, 3rd Harmonic Vacuum Vessel*
Vessel Number	IND-103
Vessel Drawing Number	*5520 – ME - 439317*
Internal MAWP	*15 psig*
External MAWP	*14.7 psi*
Working Temperature Range	*20* °F *100* °F
Design/Manufacturer	*Design: DESY/FNAL; Manufacturer: Precision Metal*
Date of Manufacture	*June, 2007*
Acceptance Date	*August, 2007*

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	(DESY)
Laboratory property number	
Purpose of vessel	Provide insulating vacuum for the 3.9-GHz SCRF dressed cavities

List all pertinent drawings

Drawing No.	Location of Original
5520 – ME – 439317 -1	
5520 – ME – 439317 -2	

2. Design Verification

Provide design calculations in the Note Appendix.
See attached.

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**

The vacuum vessel for the 3.9-GHz cryomodule will be used at the DESY TTF/FLASH system. Vacuum venting requirements are a system issue which DESY will resolve, and we expect the vacuum venting already in place for the 1.3 GHz cryomodules to cover the requirements for this, smaller cryomodule. The vacuum relief devices are a part of the DESY system, so there is no relief device on the individual vacuum vessel. As a reference, calculations are provided in the design note to show the required area of relief for the individual vacuum vessel.

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.

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.

References

1. ES&H Manual Chapter 5033, Fermilab.
2. ASME Boiler & Pressure Vessel Code
3. CGA S-1.3-1995: Compressed Gas Association Pressure Relief Device Standards – Part 3 – Stationary Containers for Compressed Gases

<u>Description and Identification</u>

The 3.9-GHz cryomodule is a component of the DESY TTF/FLASH accelerator complex in Hamburg, Germany. The cryomodule was designed by DESY personnel. Its sub-assemblies, including the dressed superconducting radio frequency (SCRF) cavities, the vacuum vessel, and the internal piping are fabricated with oversight by FNAL. The fabrication takes place at outside vendors and at FNAL. The final assembly will take place at FNAL and shipped to DESY. DESY is responsible for the design of the entire TTF/FLASH system. FNAL provides safety documentation to show that the design of the cryomodule and its sub-assemblies follow FESHM safety specifications, as agreed by DESY. This includes compliance with the ASME Boiler and Pressure Vessel Code, as stated in the appropriate FESHM chapters.

This engineering note shows that the vacuum vessel for the 3.9-GHz cryomodule conforms to FESHM 5033 - "Vacuum Vessel Safety." Since the vacuum vessel houses the insulating vacuum for the SCRF cavities, its external MAWP is 14.7-psig. Table 1 lists all drawings pertinent to the engineering note. Figures 1 and 2 show the two sheets of the engineering drawing of the vacuum vessel (5520.000-ME-439317). Figure 3 shows a 3D model of the vessel. Hard copies of the drawings are available, and electronic versions of the drawings are available online at the FERMI I-DEAS Team (RELEASED) Drawing Database. The web address for this database is:

http://www-admscad.fnal.gov/MSDMain/cgi-bin/TP_FERMIifind-web.pl

<u>Table 1 – Engineering Drawings for the Vacuum Vessel of the 3.9-GHz Cryomodule</u>

Drawing Title	Drawing Number
Vacuum Vessel Weldment Assy.	ME – 439317
Vessel Shell	ME – 439262
Port Fixed Coldmass	MD – 439251
Port Sliding Coldmass	MD – 439252
Pick Point Weldment	MC – 439257
Pad Weld Upper	MC – 439242
Vessel Support Weldment	MC – 439254
Pad Base Support	MC – 439241
Flange MC Support	MD – 439243
Flange Vessel Electronics	MD – 439246
Bracket, Weldment MC LH	MC - 439238
Bracket, Weldment MC LH	MC – 439239
Flange, Cryostat End	MD - 439240
Gusset, Outrigger	MC – 439247
Plate, Outrigger	MC – 439248

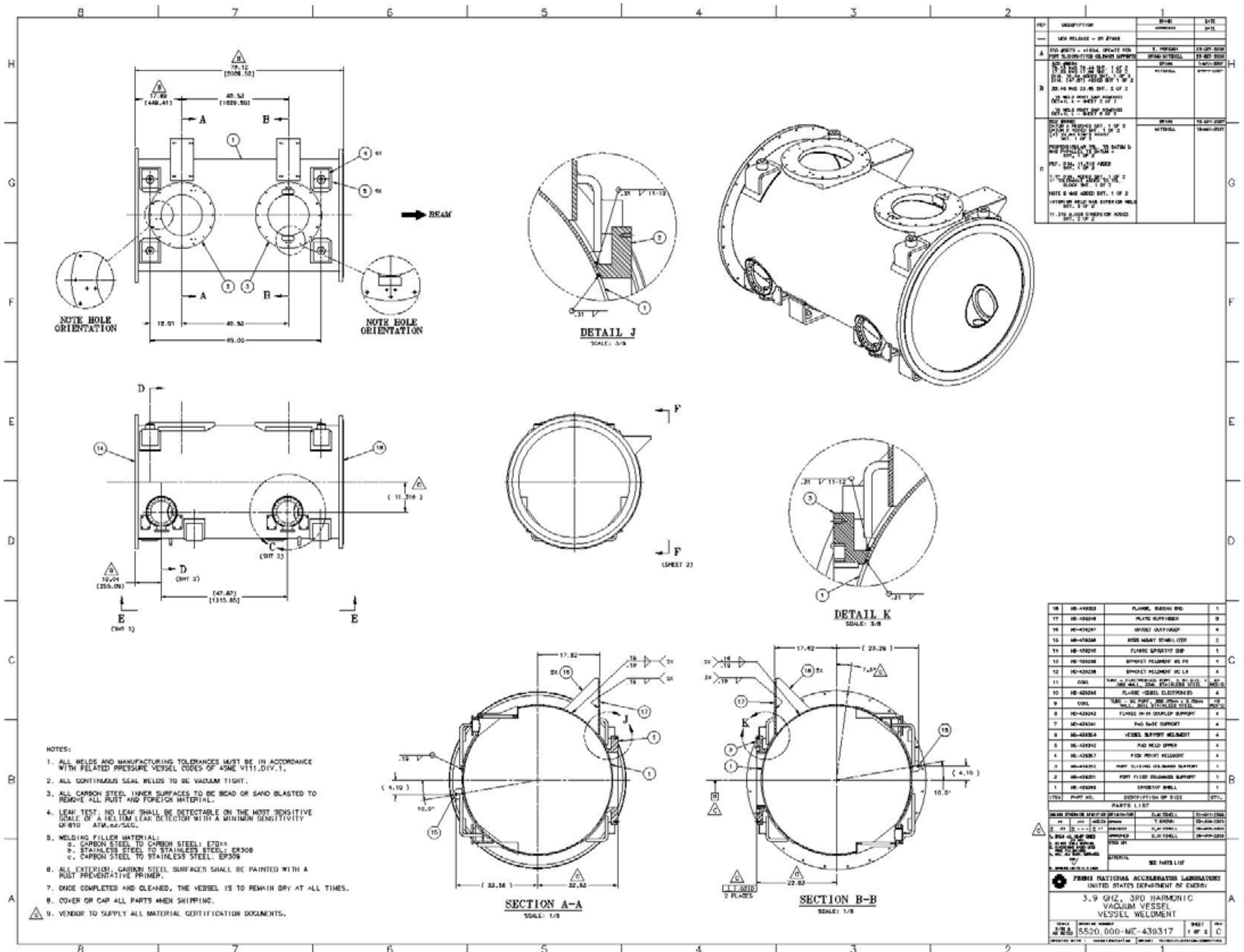

Figure 1 - 3.9-GHz, 3rd Harmonic Vacuum Vessel Drawing 5520.000-ME-439317, Sheet 1

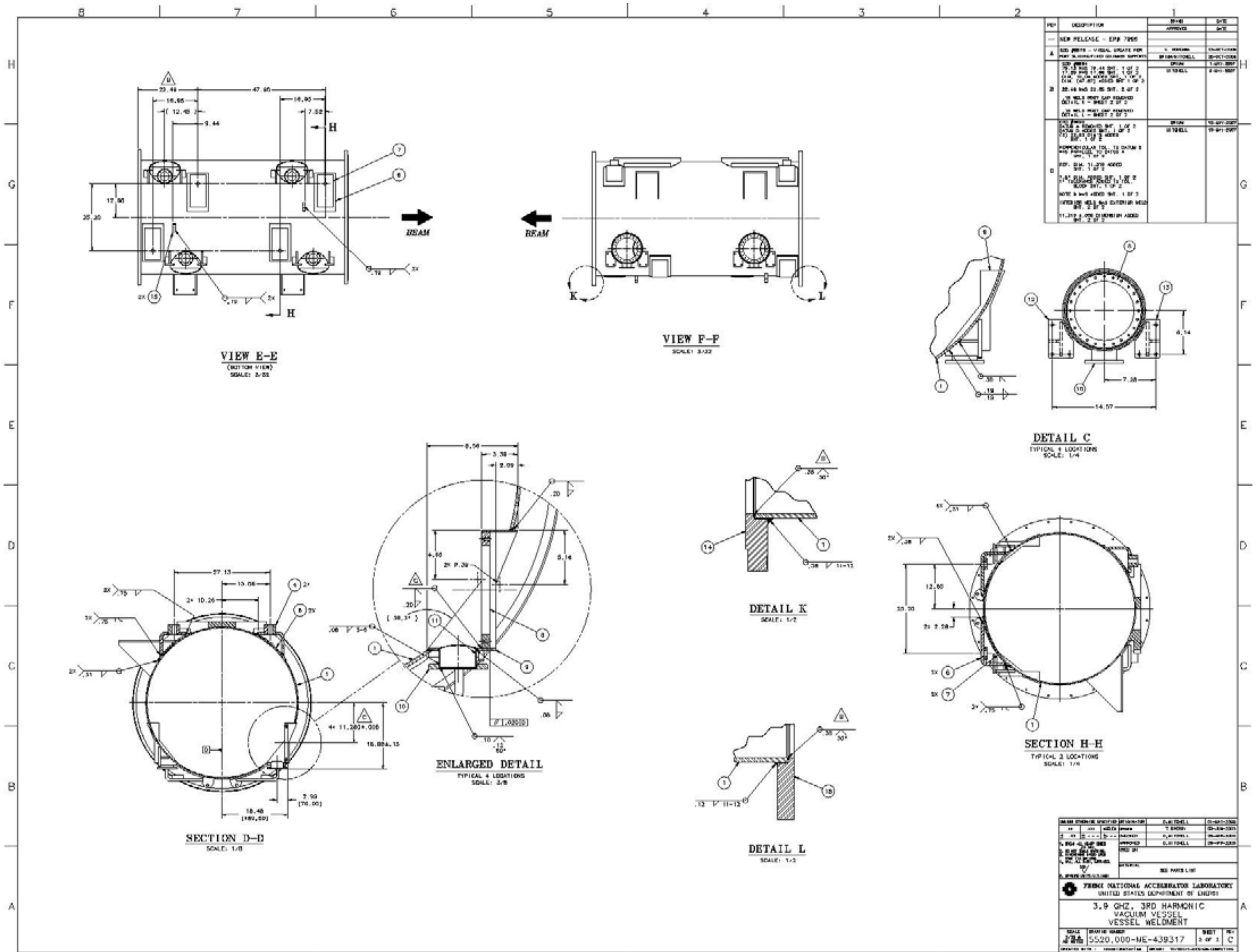

Figure 2 - 3.9-GHz, 3rd Harmonic Vacuum Vessel Drawing 5520.000-ME-439317, Sheet 2

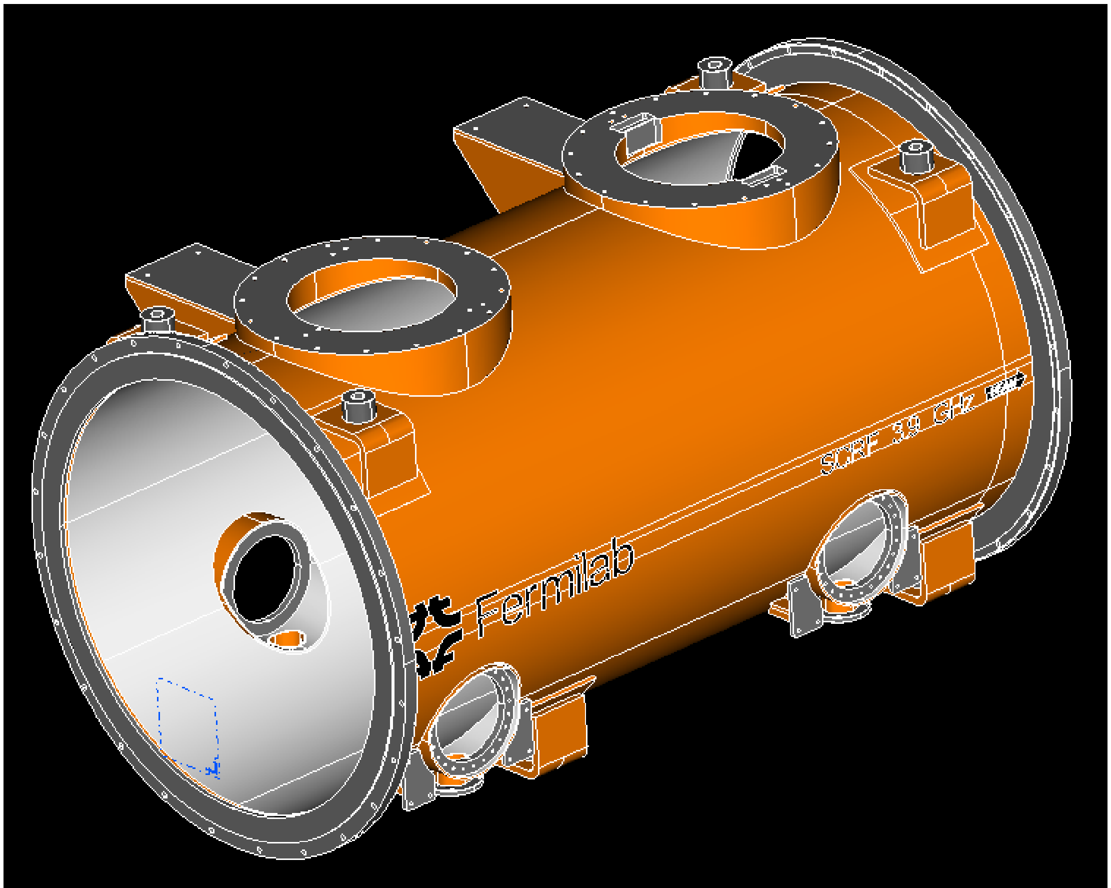

Figure 3 - An overall view of the vacuum vessel weldment of 3.9 GHz, 3rd Harmonic

Design Verification

The design of the vacuum vessel follows the applicable rules of the ASME BPVC (the Code) as listed in FESHM 5033 – “Vacuum Vessel Safety.” Unless specified, the paragraphs are from Section VIII, Division 1 of the Code.

Calculation for the Vacuum Vessel Cylindrical Shell Thickness (UG-28, UG-27)
External Pressure:

Calculate the maximum allowed external working pressure, given the vessel shell thickness t = 0.315-inch, following UG-28:

Given (dimensions from drawing ME-439262):

- P: 14.7 psi, the external design pressure,
- D_o: 42.87 in, outside diameter of the vessel shell
- D_i: 42.24 in, inside diameter of the vessel shell
- t: 0.315-in, The vessel shell thickness.
- L: 78.06 in, Design length of the vessel between lines of the support

$Do := 42.87$ inch (outer diameter of cylindrical shell of vacuum vessel)

$t := 0.17$ inch (minimum required thickness of the shell)

$\frac{Do}{t} = 252.176$ Since this ratio is greater than 10, follow UG-28(c)(1)

$L := 78.06$ inch (total length of tube)

$\frac{L}{Do} = 1.821$

$E := 29 \cdot 10^6$ psi (modulus of elasticity for SA-516 Gr 60 carbon steel)

$A := 0.00019$ (Factor A, obtained from Fig. G in Sec II, Subpart 3)

$B := 2800$ (Factor B, obtained from Fig CS-2 in Sec II, Subpart 3

$$Pa := \frac{4 \cdot B}{3 \cdot \left(\frac{Do}{t}\right)}$$

$Pa = 14.804$ psi (maximum allowable external working pressure)

Since the external design pressure requires a shell thickness of 0.17-inch, the specified thickness t=0.315-inch is acceptable. No stiffening rings are required, so UG-29 and UG-30 are not applicable.

<u>Internal pressure:</u>

Calculate the minimum required thickness of the vacuum vessel for an internal pressure of 15-psig, following UG-27.

Given:

- P: 15-psig, the internal design pressure,
- R: 21.12-inch, the inner radius of the vacuum vessel,
- S: 17.1-ksi, the maximum allowable stress value for SA-516, Gr 60 plate,
- E: 0.70, the weld joint efficiency for a double butt weld that is not radiographed.

$P := 15$ psig (internal design pressure)

$R := 21.12$ inch (inner radius of the vacuum vessel)

$S := 17100$ psi (maximum allowable stressvalue for SA-516, Gr 60)

$E := 0.70$ (weld joint efficiency for a double butt weld that is not radiographed)

$$t\_circum := \frac{P \cdot R}{S \cdot E - 0.6 \cdot P}$$

$t\_circum = 0.026$ inch (minimum thickness that allows a circumferential stress to equal S)

$$t\_long := \frac{P \cdot R}{2 \cdot S \cdot E + 0.4 \cdot P}$$

$t\_long = 0.013$ inch (minimum thickness that allows a longitudinal stress to equal S)

$$t := \begin{cases} t\_circum & \text{if } t\_circum > t\_long \\ t\_long & \text{otherwise} \end{cases}$$

$t = 0.026$ inch (minimum required thickness of vacuum vesse

Since the actual vessel thickness of 0.315-inch is greater than the minimum required thickness of 0.026-inch, the thickness is adequate for the vessel to withstand an internal pressure of 15-psig.

Stiffening Rings for Cylindrical Shells under External Pressure (UG-29)

Calculations for the cylindrical shell under external pressure show that the vessel does not require stiffening rings. This paragraph is thus not applicable.

Attachment of Stiffening Rings (UG-30)

Calculations for the cylindrical shell under external pressure show that the vessel does not require stiffening rings. This paragraph is thus not applicable.

Formed Heads with Pressure on Concave Side (UG-32)

The vacuum vessel does not include formed heads, so this paragraph is not applicable.

Formed Heads with Pressure on Convex Side (UG-33)

The vacuum vessel does not include formed heads, so this paragraph is not applicable.

Unstayed Flat Heads on Covers (UG-34)

The vacuum vessel does not include flat heads, so this paragraph is not applicable.

Calculations and Analysis for the Vessel Shell Openings (UG-37)

In the vacuum vessel are three distinct openings: the port for the sliding coldmass support (item 3 in drawing ME-439317), the port for the fixed coldmass support (item 2), and the port for the main coupler (items 8 and 9). The required areas (A_{req_int}) of reinforcement for all openings are calculated for internal pressure following UG-37(c):

$$A_{req\_int} = d*t_{r\_int}*F + 2*t_n*t_{r\_int}*F*(1-f_{r1})$$

Where A_{req_int} = required area of reinforcement for internal pressure (in^2)
d = diameter of opening (in)
t_{r_int} = required thickness of vacuum vessel shell for internal pressure = 0.026-in. (calculated in section 1)
t_n = nozzle wall thickness (in.)
F = correction factor = 1.0
f_{r1} = strength reduction factor

For external pressure, the required area of reinforcement (A_{req_ext}) for all openings follows UG-37(d)

$$A_{req\_ext} = 0.5*[d*t_{r\_ext}*F + 2*t_n*t_{r\_ext}*F*(1-f_{r1})]$$

Where A_{req_ext} = required area of reinforcement for external pressure (in^2)
t_{r_ext} = required thickness of vacuum vessel shell for external pressure = 0.17-in.

Calculate the available areas of reinforcement (A_{avail}) following UG-37. Table 2 lists the available and required areas of reinforcement for each opening under internal pressure. Table 3 lists the areas of reinforcement for openings under external pressure.

Table 2 – Available (A_{avail}) and Required (A_{req_int}) Areas of Reinforcement for Nozzles in Internal Pressure

Location	d (inch)	tn (inch)	Nozzle material	Sn (ksi)	fr2	A_req_int (in^2)	A_avail (in^2)
Fixed Coldmass port (#2)	15.5	1.25	304L	16.7	0.98	0.4	6.5
Sliding coldmass port (#3)	15.5	0.96	304L	16.7	0.98	0.4	6.0
Main coupler ports	8.4	0.20	304L	16.7	0.98	0.2	2.7

Table 3 – Available (A_{avail}) and Required (A_{req_ext}) Areas of Reinforcement for Nozzles in External Pressure

Location	d (inch)	tn (inch)	Nozzle material	Sn (ksi)	fr2	A_req_ext (in^2)	A_avail (in^2)
Fixed Coldmass port (#2)	15.5	1.25	304L	16.7	0.98	1.3	4.2
Sliding coldmass port (#3)	15.5	0.96	304L	16.7	0.98	1.3	3.8
Main coupler ports	8.4	0.20	304L	16.7	0.98	0.7	1.5

Detailed Reinforcement Calculations for the Sliding Coldmass Port

The dimensions of the two ports that allow access to the coldmass (fixed coldmass port #2 and sliding coldmass port #3) are similar. Detailed calculations for the port supporting the sliding coldmass (item 3 in drawing ME-439317) are presented. Results for both ports are summarized in Tables 2 and 3.

The port for the sliding coldmass:

Given (dimensions from drawings ME-439317, Detail K and MD-439252):

d = 15.512", inside diameter of the port.
t_n = 0.96", wall thickness of the port (taking into account the notches).
t_{r_int} = 0.026", required wall thickness of the vessel wall for internal pressure
t_{r_ext} = 0.170", required wall thickness of the vessel wall for external pressure
$E = 29 \times 10^6$ psi, modulus of elasticity of the material at the operating temp.
S_n = 16,700 psi for 304L SS
S_v = 17,100 psi for the vessel
L = 1.8", length of nozzle (the thickness portion of integral reinforcement of the port) beyond the outside surface of the vessel wall.
t_{rn}: required thickness of the port.

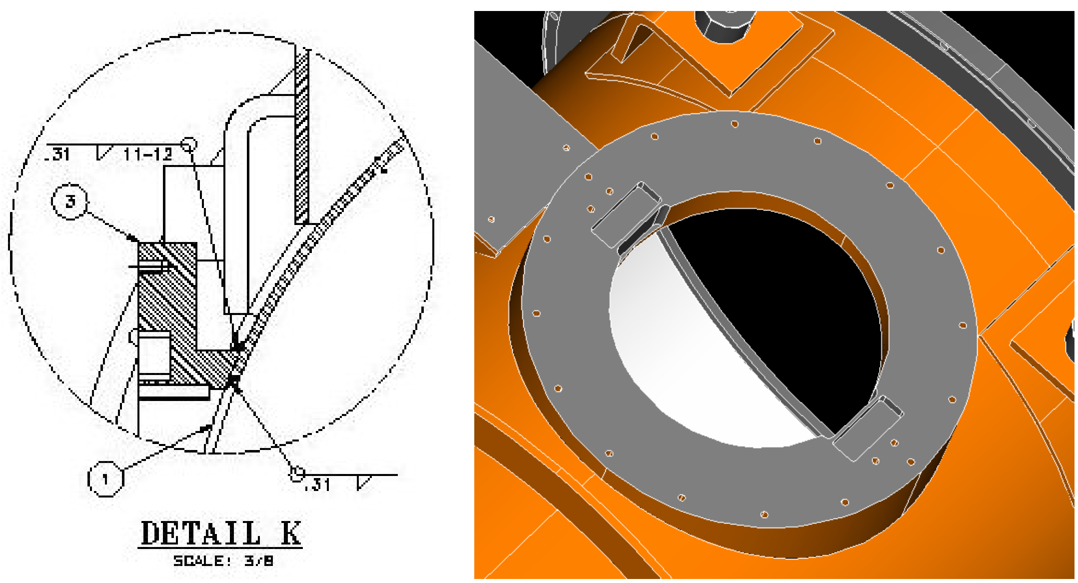

Figure 4 – Detailed drawing (Detail K of drawing ME-439317) and 3D view of the sliding coldmass support port #3 and the vessel shell

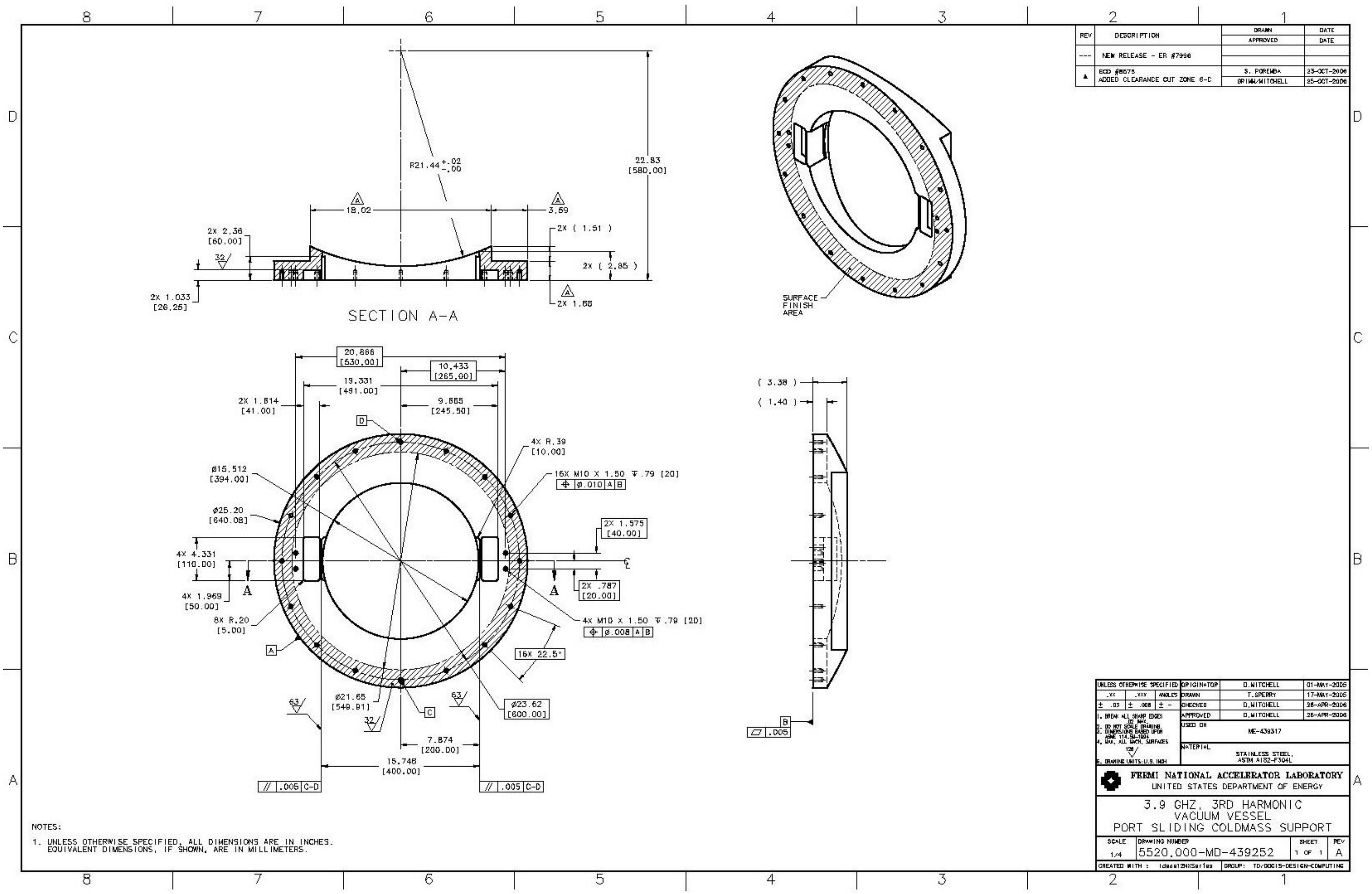

Figure 5 – Port for the Sliding Coldmass Support (Drawing 5520.000-MD-439252)

First, the minimum nozzle thickness t_{rn} is calculated for internal pressure

$E\_nozzle := 1.0$ (weld efficiency for seamless nozzle)

$P\_nozzle := 15.0$ psig (internal pressure for nozzle)

$$R\_nozzle := \frac{d}{2}$$

$R\_nozzle = 7.75$ inch (inner radius of nozzle)

$Sn = 16.7$ ksi (maximum allowable stress value for nozzle, made of 304L)

$$t\_nozzle\_circum := \frac{P\_nozzle \cdot R\_nozzle}{Sn \cdot 1000 \cdot E\_nozzle - 0.6 \cdot P\_nozzle}$$

$t\_nozzle\_circum = 6.965 \cdot 10^{-3}$ inch (minimum req. nozzle thickness for circumferential stress)

$$t\_nozzle\_longit := \frac{P\_nozzle \cdot R\_nozzle}{2 \cdot Sn \cdot 1000 \cdot E\_nozzle + 0.4 \cdot P\_nozzle}$$

$t\_nozzle\_longit = 3.48 \cdot 10^{-3}$ inch (minimum req. nozzle thickness for longitudinal stress)

$$trn\_int\_press := \begin{cases} t\_nozzle\_circum & \text{if } t\_nozzle\_circum > t\_nozzle\_longit \\ t\_nozzle\_longit & \text{otherwise} \end{cases}$$

$trn\_int\_press = 6.965 \cdot 10^{-3}$ inch (minimum req. nozzle wall thickness for internal pressure 15 psig)

Now, the required area of reinforcement is calculated.

$t := 0.315$ inch (specified vessel wall thickness)

$F := 1.0$ (correction factor)

$fr1 := 1.0$ (strength reduction factor for nozzle wal abutting the vessel wall)

$$A\_req := d \cdot tr \cdot F + 2 \cdot tn \cdot tr \cdot F \cdot (1 - fr1)$$

$A\_req = 0.403$ in^2 (required cross-sectional area of reinforcement - UG-37(c))

Next, the available reinforcement is calculated

$E1 := 1.0$ (joint weld efficiency for an opening that is not through a weld)

$A11 := d\cdot(E1\cdot t - F\cdot tr) - 2\cdot tn\cdot(E1\cdot t - F\cdot tr)\cdot(1 - fr1)$

$A11 = 4.472$

$A12 := (2\cdot(t + tn)\cdot(E1\cdot t - F\cdot tr)) - 0$

$A12 = 0.736$

$$A1 := \begin{cases} A11 & \text{if } A11 > A12 \\ A12 & \text{otherwise} \end{cases}$$

$A1 = 4.472$ in^2 (cross-sectional area available in head)

$$f\_r2 := \frac{Sn}{Sv}$$

$f\_r2 = 0.977$ (strength reducing factor)

$A21 := 5\cdot(tn - trn\_int\_press)\cdot f\_r2\cdot t$

$A21 = 1.466$

$A22 := 5\cdot(tn - trn\_int\_press)\cdot f\_r2\cdot tn$

$A22 = 4.468$

$$A2 := \begin{cases} A21 & \text{if } A21 < A22 \\ A22 & \text{otherwise} \end{cases}$$

$A2 = 1.466$ in^2 (area available in nozzle projecting outward)

$ti := 0$ inch (wall thickness of inward nozzle)

$h := 0$ inch (distance that nozzle projects inward)

$A31 := 5\cdot t\cdot ti\cdot f\_r2$

$A31 = 0$

$A32 := 5 \cdot ti^2 \cdot f\_r2$

$A32 = 0$

$A33 := 2 \cdot h \cdot ti \cdot f\_r2$

$A33 = 0$

$$A3 := \begin{array}{|lll} A31 & \text{if} & A31 < A32 < A33 \\ A31 & \text{if} & A31 < A33 < A33 \\ A32 & \text{if} & A32 < A31 < A33 \\ A32 & \text{if} & A32 < A33 < A31 \\ A33 & \text{otherwise} & \end{array}$$

$A3 = 0$ in^2 (area available in inward nozzle)

$leg\_out := 0.31$ inch (weld size)

$A41 := leg\_out^2 \cdot f\_r2$

$A41 = 0.094$ in^2 (area available in outward weld)

$leg\_in := 0$ inch (weld size)

$A43 := leg\_in^2 \cdot f\_r2$

$A43 = 0$ in^2 (area available in inward weld)

$A\_avail := A1 + A2 + A3 + A41 + A43$

$A\_avail = 6.032$ in^2 (total available area for reinforcement without reinforcing element for nozzle in internal pressure)

The available area of reinforcement of 6.0-in^2 far exceeds the required area of reinforcement of 0.4-in^2. So a reinforcing element is not needed for the nozzle when under internal pressure.

Now, calculate the required and available areas of reinforcement for the nozzle under external pressure. First, the required nozzle wall thickness is calculated for the external pressure.

$Do\_nozzle := d$

$Do\_nozzle = 15.5$ inch (inner diameter of the nozzle)

$trn\_ext\_press := 0.025$ inch (min. req'd nozzle thickness to obtain Pa_nozzle=15 psi)

$$\frac{Do\_nozzle}{trn\_ext\_press} = 620$$

Since this ratio is greater than 10, follow UG-28(c)(1)

$L\_nozzle := 1.8$ inch (length of nozzle)

$$\frac{L\_nozzle}{Do\_nozzle} = 0.116$$

$A\_nozzle := 0.0009$ (Factor A, obtained from Fig. G in Sec II, Subpart 3)

$B\_nozzle := 8750$ (Factor B, obtained from Fig. HA-3 - 304L SS)

$$Pa\_nozzle := \frac{4 \cdot B\_nozzle}{3 \cdot \left(\frac{Do\_nozzle}{trn\_ext\_press} \right)}$$

$Pa\_nozzle = 18.817$ psi (maximum allowable external working pressure)

So, the minimum required nozzle thickness is t_{rn} = 0.025-inch for a nozzle under an external pressure of at least 15-psi. Now, the required nozzle area of reinforcement is calculated for external pressure.

$tr\_ext := 0.17$ inch (required thickness of the vessel shell for external pressure)

$fr1 := 1.0$ (strength reduction factor for nozzle wal abutting the vessel wall)

$fr1 = 1$

$$A\_req := 0.5 \cdot (d \cdot tr\_ext \cdot F + 2 \cdot tn \cdot tr\_ext \cdot F \cdot (1 - fr1))$$

$A\_req = 1.318$ in^2 (required cross-sectional area of reinforcement for external pressure)

Next, the available reinforcement is calculated

$E1 := 1.0$ (joint weld efficiency for a port that does not pass through a weld

$A11 := d \cdot (E1 \cdot t - F \cdot tr\_ext) - 2 \cdot tn \cdot (E1 \cdot t - F \cdot tr\_ext) \cdot (1 - fr1)$

$A11 = 2.248$

$A12 := (2 \cdot (t + tn) \cdot (E1 \cdot t - F \cdot tr\_ext)) - 0$

$A12 = 0.37$

$$A1 := \begin{cases} A11 & \text{if } A11 > A12 \\ A12 & \text{otherwise} \end{cases}$$

$A1 = 2.248$ in^2 (cross-sectional area available in head)

$$f\_r2 := \frac{Sn}{Sv}$$

$f\_r2 = 0.977$ (strength reducing factor)

$A21 := 5 \cdot (tn - trn\_ext\_press) \cdot f\_r2 \cdot t$

$A21 = 1.438$

$A22 := 5 \cdot (tn - trn\_ext\_press) \cdot f\_r2 \cdot tn$

$A22 = 4.383$

$$A2 := \begin{cases} A21 & \text{if } A21 < A22 \\ A22 & \text{otherwise} \end{cases}$$

$A2 = 1.438$ in^2 (area available in nozzle projecting outward)

$ti := 0$ inch (wall thickness of inward nozzle)

$h := 0$ inch (distance that nozzle projects inward)

$A31 := 5 \cdot t \cdot ti \cdot f\_r2$

$A31 = 0$

$$A32 := 5 \cdot ti^2 \cdot f\_r2$$

$$A32 = 0$$

$$A33 := 2 \cdot h \cdot ti \cdot f\_r2$$

$$A33 = 0$$

$$A3 := \begin{array}{|lll} A31 & \text{if} & A31 < A32 < A33 \\ A31 & \text{if} & A31 < A33 < A33 \\ A32 & \text{if} & A32 < A31 < A33 \\ A32 & \text{if} & A32 < A33 < A31 \\ A33 & \text{otherwise} & \end{array}$$

$A3 = 0$ in^2 (area available in inward nozzle)

$leg\_out := 0.31$ inch (weld size)

$$A41 := leg\_out^2 \cdot f\_r2$$

$A41 = 0.094$ in^2 (area available in outward weld)

$leg\_in := 0$ inch (weld size)

$$A43 := leg\_in^2 \cdot f\_r2$$

$A43 = 0$ in^2 (area available in inward weld)

$$A\_avail := A1 + A2 + A3 + A41 + A43$$

$A\_avail = 3.78$ in^2 (total available area for reinforcement without reinforcing element for external pressure

For the nozzle under external pressure, the available area of reinforcement of 3.8in^2 far exceeds the required area of reinforcement of 1.3-in^2. So, the nozzle does not require a reinforcing element.

The required and available areas of reinforcement for both internal and external pressure are summarized in Tables 2 and 3.

The dimensions for the port for the fixed coldmass support are shown in Figures 6 (drawing MD-439251) and 7 (Detail J in drawing ME-439317).

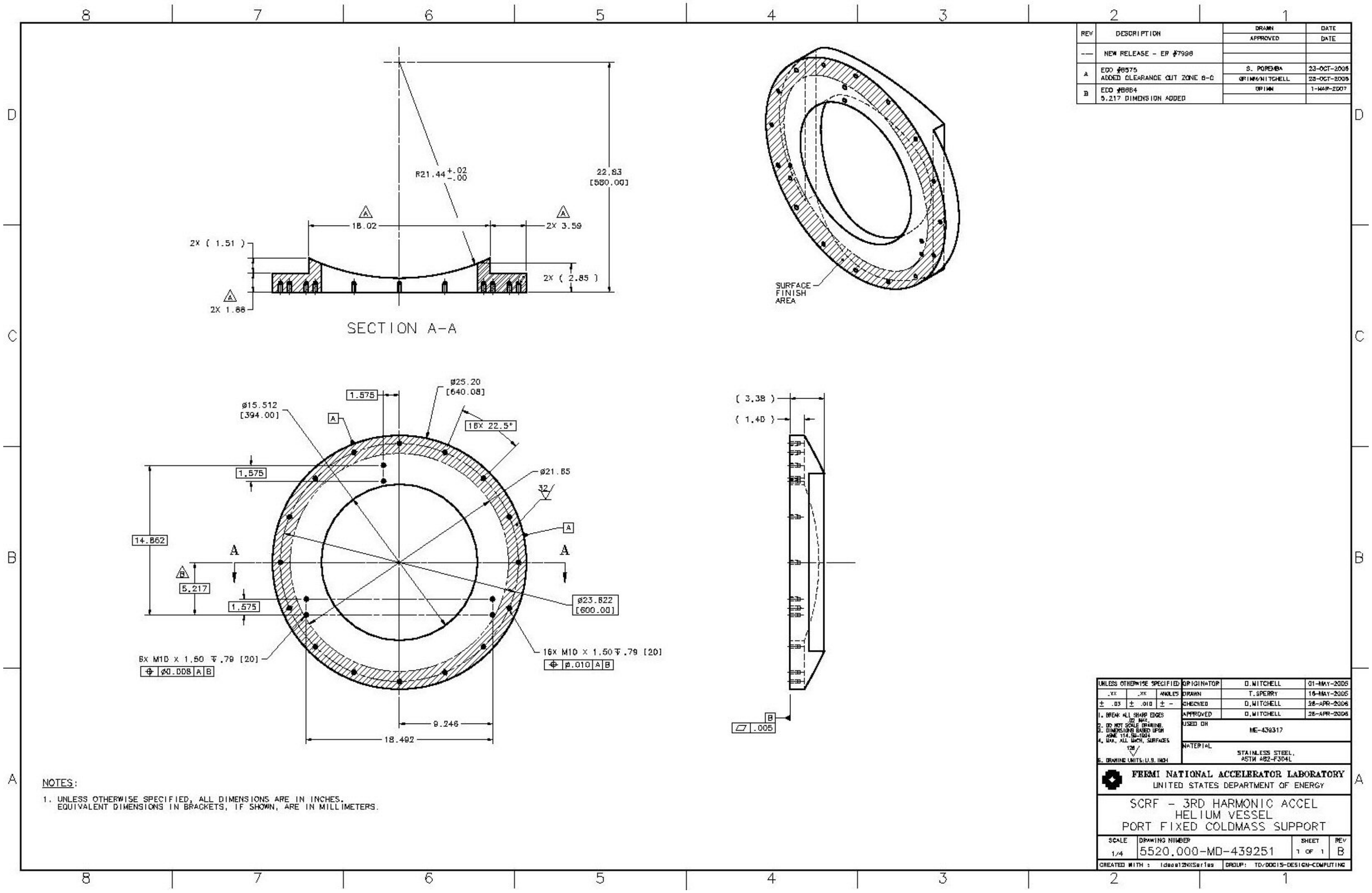

Figure 6 – Port for Fixed Coldmass Support (drawing MD-439251)

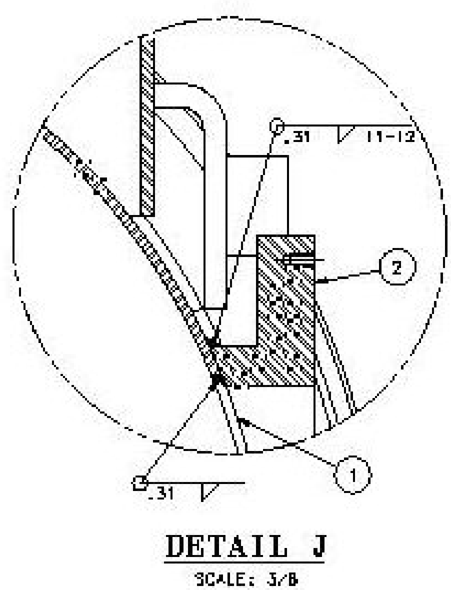

Figure 7 – Detail of Port for Fixed Coldmass Support (Detail J of drawing ME-439317)

Detailed Reinforcement Calculations for the Main Coupler Port

The dimensions for the nozzle opening for the main coupler are shown in the Enlarged Detail of Section D-D in drawing ME-439317 (Figure 8) and the drawing of the flange for the main coupler support (MC-439243) (Figure 9). According to the Parts List, Item 9 from the Enlarged Detail is a 304L stainless steel tube with a 5.00-mm (0.2-inch) wall thickness.

Given (dimensions from drawings ME-439317, Section D-D and MC-439243)

- d = 8.39”, inside diameter of the port.
- t_n = 0.79”, wall thickness of the port
- t_{r_int} = 0.026”, required wall thickness of the vessel wall for internal pressure
- t_{r_ext} = 0.170”, required wall thickness of the vessel wall for external pressure
- $E = 29 \times 10^6$ psi, modulus of elasticity of the material at the operating temp.
- S_n = 16,700 psi for 304L SS
- S_v = 17,100 psi for the vessel
- L = 0.45”, length of nozzle (the thickness portion of integral reinforcement of the port), shortest length, beyond the outside surface of the vessel wall.
- t_{rn}: required thickness of the port.

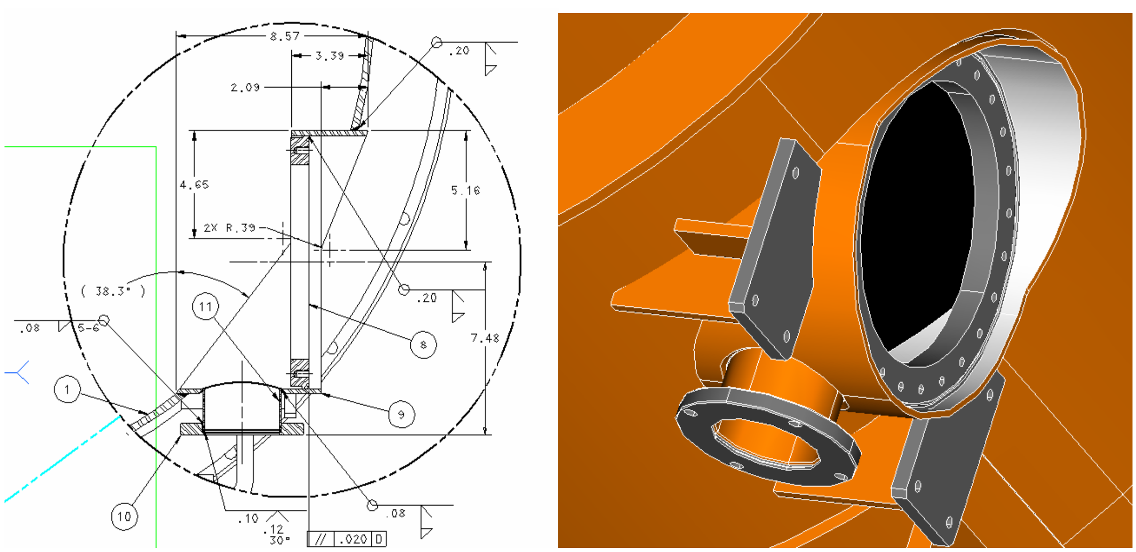

Figure 8 – Enlarged Detail of Section D-D from ME-439317 and 3D View of the Main Coupler Port

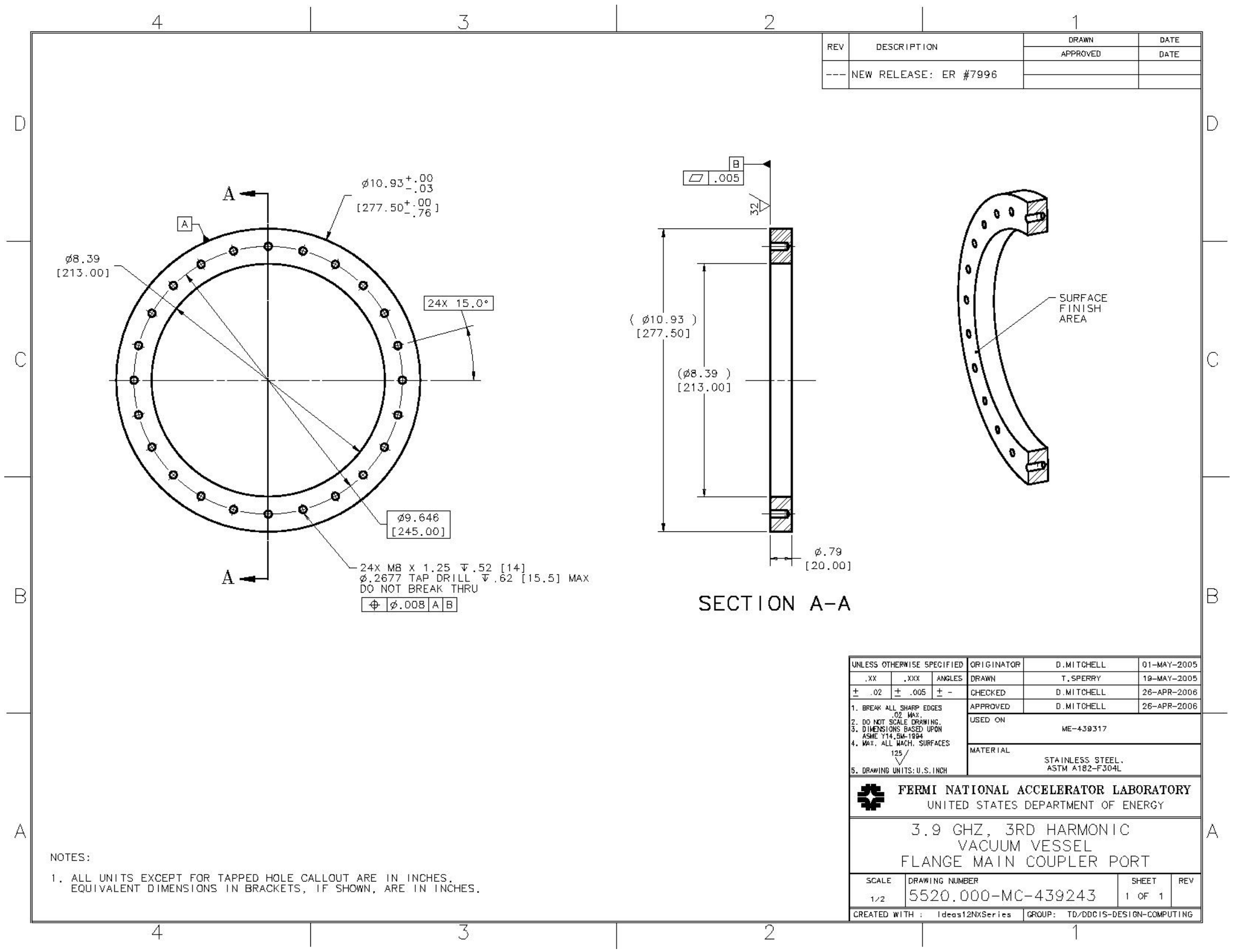

Figure 9 – Drawing of Flange for the Main Coupler Port (5520.000-MC-439243)

For the main coupler port, the minimum nozzle thickness t_{rn} is first calculated for internal pressure.

$E\_nozzle := 1.0$ (weld efficiency for a seamless nozzle)

$P\_nozzle := 15.0$ psig (internal pressure for nozzle)

$$R\_nozzle := \frac{d}{2}$$

$R\_nozzle = 4.2$ inch (inner radius of nozzle)

$Sn = 16.7$ ksi (maximum allowable stress value for nozzle, made of 304L)

$$t\_nozzle\_circum := \frac{P\_nozzle \cdot R\_nozzle}{Sn \cdot 1000 \cdot E\_nozzle - 0.6 \cdot P\_nozzle}$$

$t\_nozzle\_circum = 3.774 \cdot 10^{-3}$ inch (minimum req. nozzle thickness for circumferential stress)

$$t\_nozzle\_longit := \frac{P\_nozzle \cdot R\_nozzle}{2 \cdot Sn \cdot 1000 \cdot E\_nozzle + 0.4 \cdot P\_nozzle}$$

$t\_nozzle\_longit = 1.886 \cdot 10^{-3}$ inch (minimum req. nozzle thickness for longitudinal stress)

$$trn\_int\_press := \begin{cases} t\_nozzle\_circum & \text{if } t\_nozzle\_circum > t\_nozzle\_longit \\ t\_nozzle\_longit & \text{otherwise} \end{cases}$$

$trn\_int\_press = 3.774 \cdot 10^{-3}$ inch (minimum req. nozzle wall thickness for internal pressure 15 psig)

Now, the required area of reinforcement for the main coupler port is calculated.

$tr\_int := 0.026$ inch (required thickness of the vessel shell for internal pressure)

$F := 1.0$ (correction factor)

$fr1 := 1.0$ (strength reduction factor for nozzle wal abutting the vessel wall)

$$A\_req := d \cdot tr\_int \cdot F + 2 \cdot tn \cdot tr\_int \cdot F \cdot (1 - fr1)$$

$A\_req = 0.218$ in^2 (required cross-sectional area of reinforcement - UG-37(c))

Next, the available reinforcement for the main coupler port under internal pressure is calculated

$E1 := 1.0$ (joint weld efficiency for a port not passing through a weld)

$A11 := d\cdot(E1\cdot t - F\cdot tr\_int) - 2\cdot tn\cdot(E1\cdot t - F\cdot tr\_int)\cdot(1 - fr1)$

$A11 = 2.428$

$A12 := (2\cdot(t + tn)\cdot(E1\cdot t - F\cdot tr\_int)) - 0$

$A12 = 0.298$

$$A1 := \begin{array}{|l} A11 \ \text{ if } \ A11 > A12 \\ A12 \ \text{ otherwise} \end{array}$$

$A1 = 2.428$ in^2 (cross-sectional area available in vessel wall)

$f\_r2 := \dfrac{Sn}{Sv}$

$f\_r2 = 0.977$ (strength reducing factor)

$A21 := 5\cdot(tn - trn\_int\_press)\cdot f\_r2\cdot t$

$A21 = 0.302$

$A22 := 5\cdot(tn - trn\_int\_press)\cdot f\_r2\cdot tn$

$A22 = 0.192$

$$A2 := \begin{array}{|l} A21 \ \text{ if } \ A21 < A22 \\ A22 \ \text{ otherwise} \end{array}$$

$A2 = 0.192$ in^2 (area available in nozzle projecting outward)

$ti := 0$ inch (wall thickness of inward nozzle)

$h := 0$ inch (distance that nozzle projects inward)

$A31 := 5\cdot t\cdot ti\cdot f\_r2$

$A31 = 0$

Note that while the nozzle on the main coupler port does project inward from the vacuum vessel wall, it is not taken into account in the reinforcement calculations. This is why h = 0. This way the calculations are on the conservative side.

$A32 := 5 \cdot ti^2 \cdot f\_r2$

$A32 = 0$

$A33 := 2 \cdot h \cdot ti \cdot f\_r2$

$A33 = 0$

$$A3 := \begin{array}{|ll} A31 & \text{if } A31 < A32 < A33 \\ A31 & \text{if } A31 < A33 < A33 \\ A32 & \text{if } A32 < A31 < A33 \\ A32 & \text{if } A32 < A33 < A31 \\ A33 & \text{otherwise} \end{array}$$

$A3 = 0$ in^2 (area available in inward nozzle)

$leg\_out := 0.31$ inch (weld size)

$A41 := leg\_out^2 \cdot f\_r2$

$A41 = 0.094$ in^2 (area available in outward weld)

$leg\_in := 0$ inch (weld size)

$A43 := leg\_in^2 \cdot f\_r2$

$A43 = 0$ in^2 (area available in inward weld)

$A\_avail := A1 + A2 + A3 + A41 + A43$

$A\_avail = 2.713$ in^2 (total available area for reinforcement without reinforcing element)

The available are of reinforcement of 2.7-in^2 far exceeds required cross-sectional area of reinforcement of 0.2-in^2. So a reinforcing element is not needed for the nozzle when it is under internal pressure.

Now, for the main coupler port, calculate the required and available areas of reinforcement for the nozzle under external pressure. First the required nozzle wall thickness is calculated.

$Do\_nozzle := d$

$Do\_nozzle = 8.4$ inch (inner diameter of the nozzle)

$trn\_ext\_press := 0.01$ inch (minimum required nozzle thickness to obtain Pa_nozzle=15 ps

$\frac{Do\_nozzle}{trn\_ext\_press} = 840$ Since this ratio is greater than 10, follow UG-28(c)(1)

$L\_nozzle := 0.45$ inch (length of nozzle)

$\frac{L\_nozzle}{Do\_nozzle} = 0.054$

$A\_nozzle := 0.0015$ (Factor A, obtained from Fig. G in Sec II, Subpart 3)

$B\_nozzle := 9500$ (Factor B, obtained from Fig. HA-3 for 304L SS)

$$Pa\_nozzle := \frac{4 \cdot B\_nozzle}{3 \cdot \left(\frac{Do\_nozzle}{trn\_ext\_press}\right)}$$

$Pa\_nozzle = 15.079$ psi (maximum allowable external working pressure)

So, at the main coupler port, for a nozzle that has a maximum allowable external working pressure of about 15-psi, the minimum required nozzle thickness is 0.01-inch.

Now, the required nozzle area of reinforcement is calculated for the main coupler port in external pressure.

$tr\_ext := 0.17$ inch (required thickness of the vessel shell for external pressure)

$fr1 := 1.0$ (strength reduction factor for nozzle wal abutting the vessel wall)

$fr1 = 1$

$$A\_req := 0.5 \cdot (d \cdot tr\_ext \cdot F + 2 \cdot tn \cdot tr\_ext \cdot F \cdot (1 - fr1))$$

$A\_req = 0.714$ in^2 (required cross-sectional area of reinforcement for external pressure)

Next, for the main coupler port, the available reinforcement is calculated.

$E1 := 1.0$ (joint weld efficiency for port not passing through a weld)

$$A11 := d \cdot (E1 \cdot t - F \cdot tr\_ext) - 2 \cdot tn \cdot (E1 \cdot t - F \cdot tr\_ext) \cdot (1 - fr1)$$

$$A11 = 1.218$$

$$A12 := (2 \cdot (t + tn) \cdot (E1 \cdot t - F \cdot tr\_ext)) - 0$$

$$A12 = 0.149$$

$$A1 := \begin{cases} A11 & \text{if } A11 > A12 \\ A12 & \text{otherwise} \end{cases}$$

$A1 = 1.218$ in^2 (cross-sectional area available in vacuum vessel)

$$f\_r2 := \frac{Sn}{Sv}$$

$f\_r2 = 0.977$ (strength reducing factor)

$$A21 := 5 \cdot (tn - trn\_ext\_press) \cdot f\_r2 \cdot t$$

$$A21 = 0.292$$

$$A22 := 5 \cdot (tn - trn\_ext\_press) \cdot f\_r2 \cdot tn$$

$$A22 = 0.186$$

$$A2 := \begin{cases} A21 & \text{if } A21 < A22 \\ A22 & \text{otherwise} \end{cases}$$

$A2 = 0.186$ in^2 (area available in nozzle projecting outward)

$ti := 0$ inch (wall thickness of inward nozzle)

$h := 0$ inch (distance that nozzle projects inward)

$A31 := 5 \cdot t \cdot ti \cdot f\_r2$

$A31 = 0$

$A32 := 5 \cdot ti^2 \cdot f\_r2$

$A32 = 0$

$A33 := 2 \cdot h \cdot ti \cdot f\_r2$

$A33 = 0$

$$A3 := \left| \begin{array}{lll} A31 & \text{if} & A31 < A32 < A33 \\ A31 & \text{if} & A31 < A33 < A33 \\ A32 & \text{if} & A32 < A31 < A33 \\ A32 & \text{if} & A32 < A33 < A31 \\ A33 & \text{otherwise} & \end{array} \right.$$

$A3 = 0$ in^2 (area available in inward nozzle)

$leg\_out := 0.31$ inch (weld size)

$A41 := leg\_out^2 \cdot f\_r2$

$A41 = 0.094$ in^2 (area available in outward weld)

$leg\_in := 0$ inch (weld size)

$A43 := leg\_in^2 \cdot f\_r2$

$A43 = 0$ in^2 (area available in inward weld)

$A\_avail := A1 + A2 + A3 + A41 + A43$

$A\_avail = 1.497$ in^2 (total available area for reinforcement without reinforcing element for nozzle under external pressure

For the main coupler port's nozzle under external pressure, the available area of reinforcement of 1.5-in^2 exceeds the required area of 0.7-in^2. So, the nozzle does not require a reinforcing element when under external pressure.

For the main coupler port, the required and available areas of reinforcement for both internal and external pressure are summarized in Tables 2 and 3.

Flued Openings in Shells and Formed Heads (UG-38)

No flued openings exist in the shell of the vacuum vessel, so this paragraph does not apply.

Reinforcement Required for Openings in Flat Heads (UG-39)

This paragraph does not apply to the vacuum vessel.

Limits of Reinforcement (UG-40)

Calculations in this document show that the openings are adequately reinforced with additional reinforcement or attachment. So this paragraph does not apply.

Strength of Reinforcement (UG-41)

Calculations in this document show that the openings are adequately reinforced with additional reinforcement or attachment. So this paragraph does not apply.

Reinforcement of Multiple Openings (UG-42)

Calculations in this document show that the openings are adequately reinforced with additional reinforcement or attachment. So this paragraph does not apply.

Jacketed Vessels (Appendix 9)

The vacuum vessel is not a jacketed vessel as defined in Appendix 9.

Vessels of Noncircular Cross Section (Appendix 13)

This appendix is not applicable to the vacuum vessel.

The Permissible Out of Roundness of the Vessel Cylindrical Shell (UG-80)

Calculate the permissible out-of-roundness of the vessel's cylindrical shell, following UG-80.

Internal Pressure:

Given:
D_i: 42.24 in, inside diameter of the vessel shell

The difference between the maximum diameter and the minimum diameter of the vessel shell at any cross section shall not exceed 1% of the nominal inner diameter:

0.01 x 42.24 in = 0.422 in
(per section UG-80 (a)(1) of ASME VIII, Div.1)

For a cross section through or near an opening, the permissible difference shall not exceed 2% of the nominal inner diameter:

0.02 x 42.24-in = 0.844 in
(per section UG-80 (a)(2) of ASME VIII, Div.1)

External Pressure:

In addition to the out-of-roundness limitations prescribed for Internal Pressure, the shell shall meet the follow requirements at any cross section.

Given:
Do: 42.87 in, outer diameter of the vessel shell
D_i: 42.24 in, inside diameter of the vessel shell
L: 79.06 in, shell length

Per section UG-80(b)(2) and figure UG-80.1 of ASME VIII, Div.1, the maximum permissible radial deviation e is:

Calculated previously, using the outer diameter, the wall thickness, and the length:

$$\frac{Do}{t} = 134.095$$

$$\frac{L}{Do} = 1.848$$

$$e := 0.6 \cdot t$$

$e = 0.189$ inch (maximum permissible deviation on radius, Fig. UG-80.1)

System Relief Verification

The 3.9-GHz cryomodule assembly is a component of DESY's TTF/FLASH accelerator complex. The cryomodule's insulating vacuum is part of the insulating vacuum of a larger system. Thus, the vacuum relief devices are a part of the DESY system, and there is no relief device on the individual vacuum vessel. DESY is responsible for the design of the flow and venting of the insulating vacuum for the accelerator complex. The design is not known to FNAL at this time. The vacuum venting requirements are a system issue which DESY will resolve. We expect the vacuum venting already in place for the 1.3 GHz cryomodules (other components in FLASH) to cover the requirements for this, smaller cryomodule. Note, also, that the dressed cavities, which are viewed as helium pressure vessels and are housed within the vacuum vessel, have their own relief systems to vent helium.

As a reference, calculations are provided in this section to show the required area of relief for the individual vacuum vessel. These calculations are based on the size of the individual vacuum vessel. CGA S-1.3-1995, Section 4.4 defines the vacuum vessel's, or jacket, relief device's required discharge area as a function of the per pound water capacity of the vacuum vessel. The value of the water capacity is calculated using the entire capacity of an empty vacuum vessel. The required area of the relief valve is calculated:

$\rho\_water := 62.5$ lb/ft^3 (density of water)

$L\_shell := 79.12$ inch (length of vacuum vessel)

$D\_inner := 42.24$ inch (inner diameter of vacuum vessel)

$$V\_shell := \pi \cdot \left(\frac{D\_inner}{2}\right)^2 \cdot L\_shell$$

$V\_shell = 1.109 \cdot 10^5$ in^3 (inner volume of vacuum vessel)

$$V\_shell := \frac{V\_shell}{12^3}$$

$V\_shell = 64.162$ ft^3 (inner volume of vacuum vessel)

$$A\_req := 0.00024 \cdot \rho\_water \cdot V\_shell$$

$A\_req = 0.962$ in^2 (required discharge area of relief valve)

Assume that the relief valve is circular. Then the inner diameter of the required relief valve is calculated

$$d\_req := \sqrt{\frac{4 \cdot A\_req}{\pi}}$$

$$d\_req = 1.107$$

inch (required diameter of relief valve)