



FERMILAB  
Technical  
Division

**Vacuum Vessel Engineering Note  
For Cryomodule 2 (CM2)**

**Vessel No. IND-184  
Rev. No. --**

**Vacuum Vessel Engineering Note  
For Cryomodule 2 (CM2)**

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Date: 18 July 2011

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

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 Reviewed by [Signature] Date 8/17/11 Div/Sec PPD/MD  
 Div/Sec Head Apollinari S. Jimeno Date 8/31/11 Div/Sec TD / HQ

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	<u>New Muon Lab - Type 3+ Cryomodule 2 (NML-CM2)</u>
Vessel Number	<u>IND-184</u>
Vessel Drawing Number	<u>D933265, 4904.280-ME-459303, 4904.280-ME-459304</u>
Internal MAWP	<u>14.7 psig</u>
External MAWP	<u>14.7 psid</u>
Working Temperature Range	<u>20 °F                      100 °F</u>
Design/Manufacturer	<u>Design: DESY/FNAL; Manufacturer: E.Zanon</u>
Date of Manufacture	<u>January, 2007</u>
Acceptance Date	<u>July, 2007</u>

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

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

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Laboratory location code	<u>FIMS #700 (Muon Beam Enclosure at NML)</u>
Laboratory property number	<u>---</u>
Purpose of vessel	<u>Provide insulating vacuum for the 1.3-GHz SCRF dressed cavities</u>

List all pertinent drawings

Drawing No.	Location of Original
D933265	DESY EDMS
4904.280-ME-459303	Fermilab I-DEAS TDM Library
4904.280-ME-459304	Fermilab I-DEAS TDM Library

2. Design Verification

See design calculations in the Note's Appendix A.

3. System Venting Verification

Can this vessel be pressurized either internally or externally?  Yes  No

*If "Yes", to what pressure? \_\_\_\_\_*

Manufacturer	Relief	Pressure Setting	Flow Rate	Size
Fermilab	Parallel plate relief	0.5-psig	See Table 9	4.3"

P&ID number 5520.000-ME-442391 (Figure 6 in Appendix A)

4. Operating Procedure Section

Is an operating procedure necessary for the safe operation of this vessel?

Yes  No *(If "Yes", it must be appended)*

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

Yes  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

Is the vessel code-stamped?  Yes  No

*If "No," append WPS, PQR, and WPQ.*

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

Is this vessel or any part thereof in the above categories?  Yes  No

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

## Appendix A – Design Note

### Description and Identification

The 1.3-GHz Cryomodule 2 has a vacuum vessel that houses the sub-assemblies and provides the insulating vacuum to the cold mass and the superconducting RF (SCRF) cavities within. The cavities and cold mass are cooled to less than 2°K with liquid helium. Thermal shields cooled to 80°K with liquid nitrogen also sit within the vacuum vessel. The vessel is a longitudinal welded pipe. There are openings along the vessel that provide feedthroughs for the RF power couplers, the magnet current leads, and instrumentation.

The vacuum vessel is a Type III+ assembly that was designed by DESY and built with their oversight at an Italian company called Zanon. The cryomodule is assembled by Fermilab’s Technical Division SRF Development Department. The completed assembly will be installed for use at New Muon Lab, which will be its final location.

This engineering note shows that the vacuum vessel for CM2 conforms to FESHM 5033 - “Vacuum Vessel Safety.”<sup>[1]</sup> Since the vacuum vessel houses the insulating vacuum for the SCRF cavities, its external differential MAWP is 14.7-psid [1.0-bar]. The vacuum vessel is designed to the pressures and temperatures and temperatures shown in Table 1:

Table 1 – Design and Operating Pressures and Temperatures

	Design	Operating
External pressure (psid) [bar]	14.7 [1.0]	14.7 [1.0]
Internal pressure (psig) [bar]	14.7 [2.0]	1.5E-10 [1E-9]
Minimum temperature (°F) [°C]	20 [-7]	77 [25]
Maximum temperature (°F) [°C]	100 [38]	77 [25]

Table 2 lists all drawings pertinent to the engineering note. The top assembly drawing (D933265) shows the entire CM2 assembly, as shown in Figures 1a and 1b. The DESY vacuum vessel shell assembly drawing, number 6850500040, was redrawn by Fermilab and made into two drawings (4904.280-ME-459303, 4904.280-ME-459304). A PDF file of the drawings is included online in the CM2 folder of the ILC Document (see Figures 2 and 3) Management System:

<http://ilc-dms.fnal.gov/Workgroups/CryomoduleDocumentation/CM2-folder/VacVessel/>

Table 2 – Engineering Drawings for the Vacuum Vessel of CM2

Drawing No.	Rev.	Title
D933265	A	CM2 Assembly
D919085	B	Valve Relief – 4-inch 0.5 psi
459303**	-	Vessel Weldment Assembly, Sheet 1
459304**	-	Vessel Weldment Assembly, Sheet 2
459305	-	Pick Point Weldment
459306	-	Vessel Support Weldment
459307	-	Flange D OD 350
459308	-	Flange DN 63 DF
459309	-	Flange - Instrumentation
459310	-	Flange - K1, K3
459311	-	Flange MC Port
459312	-	Flange Bellows Sliding
459313	-	Flange Fixed Upstream
459314	-	Flange Fixed Downstream
459315	-	Gusset Welded Outrigger
459316	-	Pad Weld Lower
459317	-	Pad Weld Upper
459318	-	Plate MC Mount
459319	-	Plate Mount Gusset
459320	-	Plate Mount Support
459321	-	Piping Support Plate
459322	-	Plate Welded Outrigger
459323	-	Port Middle Coldmass
459324	-	Port End Coldmass
459325	-	Rib - Flange Side
459326	-	Stiffening Rib
459327	-	Ring Flange Stop
459328	-	Vessel Shell
459329	-	MC Bracket Weldment - LH
459330	-	MC Bracket Weldment - RH
459331	-	Flange Weld Ring
459070	-	Flange G 150 OD
458097	-	ILCTA Cryomodule 1 and Feedbox

\*\* Same as drawing 00682050040 – Vacuum Vessel Shell Assembly (DESY)

As detailed in the drawings, the following materials are selected for the vacuum vessel:

- The shell wall and stiffening rings: ASTM A516 Grade 60
- The nozzles walls: AISI 304L (equivalent to ASTM A240 Type 304L)

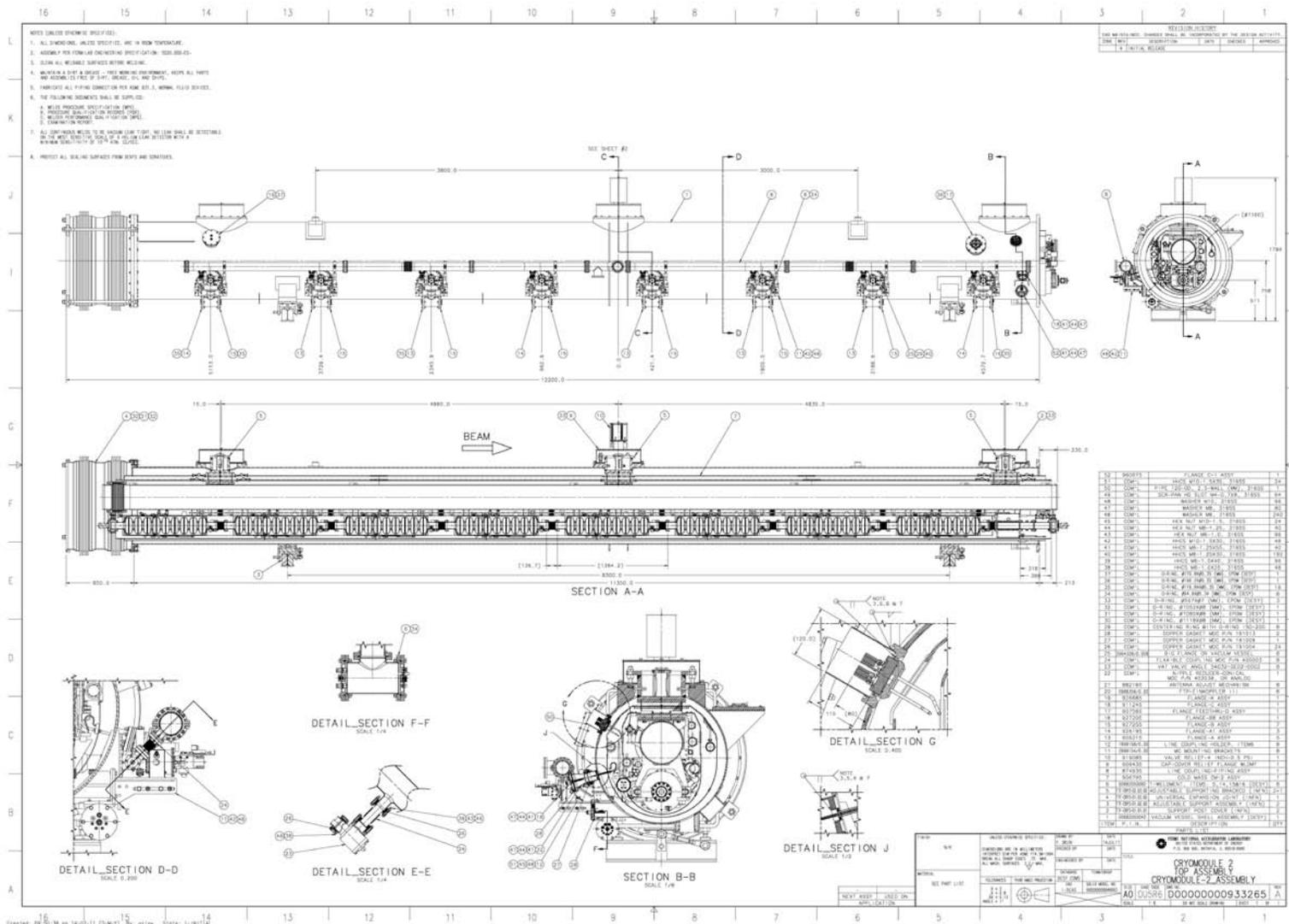


Figure 1a – CM2 Top Assembly Drawing, Sheet 1 (D933265)





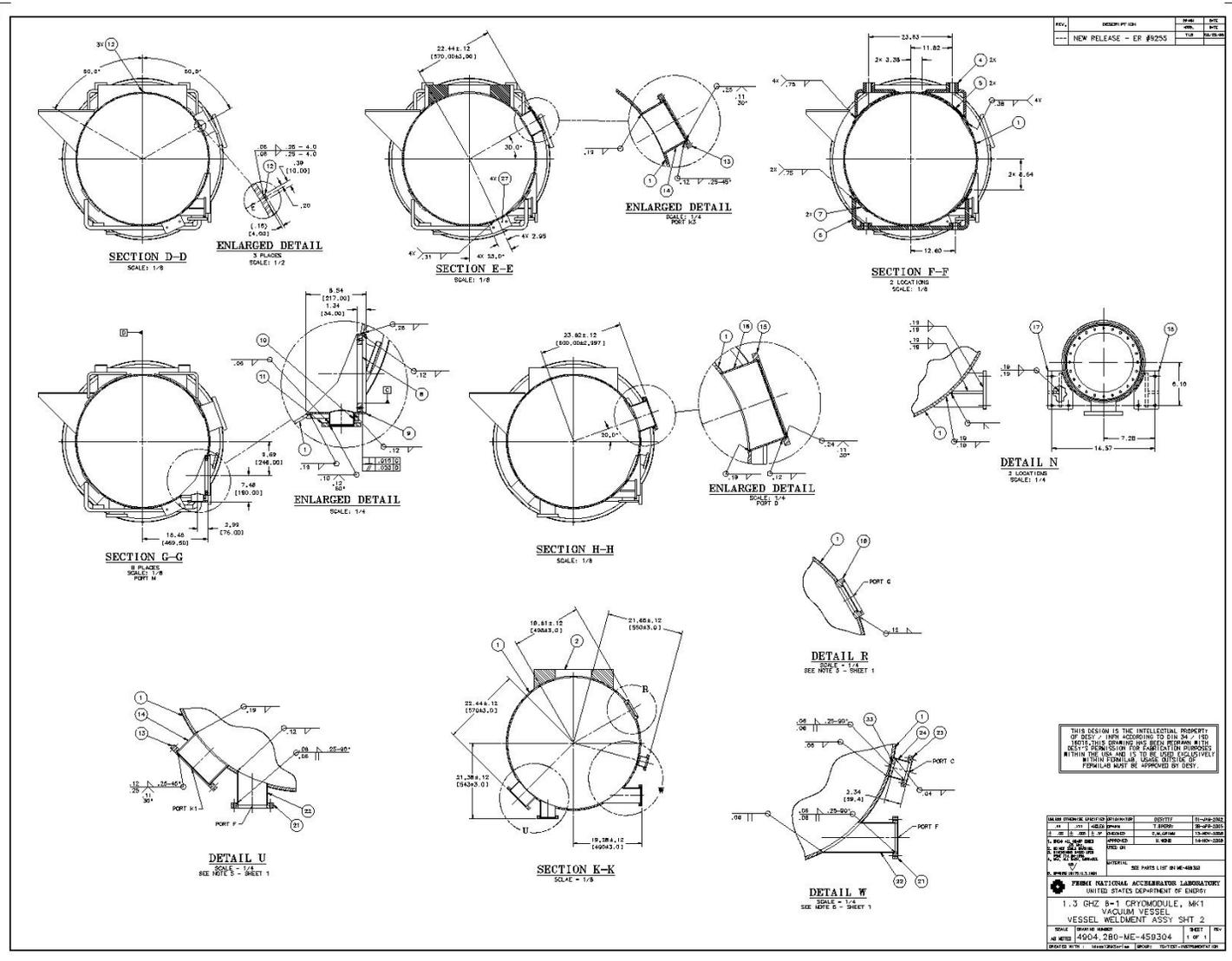


Figure 3 – Type III+ Vacuum Vessel, Sheet 2 - Drawing 4904.280-ME-459304

## Design Verification

The design of the vacuum vessel follows the applicable rules of the ASME Boiler and Pressure Vessel Code (the Code) Section VIII, Division 1 <sup>[2]</sup> as listed in FESHM 5033 – “Vacuum Vessel Safety.”

### Calculation for the Vacuum Vessel Cylindrical Shell Thickness (UG-28, UG-27)

#### *External Pressure:*

Calculate the maximum allowed external working pressure following UG-28. The minimum required thickness takes into account the support of the stiffening rings. Note that the actual vessel shell thickness  $t = 0.375$ -inch,

$D_o := 38.0$  inch (outer diameter of shell)

$t := 0.261$  inch (minimum required thickness of shell for external pressure of 14.5-psia)

$$\frac{D_o}{t} = 145.594 \quad \text{Since this ratio is greater than 10, follow UG-28(c)(1)}$$

$L := 235.7$  inch (length of stiffened shell)

$$\frac{L}{D_o} = 6.203$$

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

From Table G in Sec II, Subpart 3 to determine Factor A

$$A = 1.108 \cdot 10^{-4} \quad \text{(Factor A, from Table G of Sec. II, Subpart 3)}$$

$$P_{\text{ext}} := \frac{2 \cdot A \cdot E}{3 \cdot \left(\frac{D_o}{t}\right)}$$

$$P_{\text{ext}} = 14.709 \quad \text{psi (maximum allowable working external pressure for given } t \text{)}$$

A wall thickness of  $t = 0.261$ -inch results in a maximum allowable working external pressure of  $P_{\text{ext}} = 14.7$ -psia, which is greater than the external pressure that the vacuum vessel will see. *Since the actual wall thickness is 0.375-inch, the vessel design is adequate for the working external pressure.*

*Internal pressure*

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

$P_{int} := 29.0$  psia (internal design pressure)

$t_{actual} := 0.375$  inch (actual wall thickness)

$Do = 38$  inch (outer diameter)

$$R := \frac{Do}{2} - t_{actual}$$

$R = 18.625$  inch (inner radius of shell)

$S := 17100$  psi (maximum allowable stress value for SA-516, Gr 60, from Table 1A of Sec II, Part D)

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

$$t_{circum} := \frac{P_{int} \cdot R}{S \cdot E_{weld} - 0.6 \cdot P_{int}}$$

$t_{circum} = 0.045$  inch (minimum thickness that allows a circumferential stress to equal S)

---

$$t_{long} := \frac{P_{int} \cdot R}{2 \cdot S \cdot E_{weld} + 0.4 \cdot P_{int}}$$

$t_{long} = 0.023$  inch (minimum thickness that allows a longitudinal stress to equal S)

$$t_{internal} := \begin{cases} t_{circum} & \text{if } t_{circum} > t_{long} \\ t_{long} & \text{otherwise} \end{cases}$$

$t_{internal} = 0.045$  inch (minimum required wall thickness)

*Since the actual vessel thickness of 0.375-inch is greater than the minimum required thickness of 0.045-inch, the thickness is adequate for the vessel to withstand an internal pressure of 29.0-psia (2.0-bar).*

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

Following the Code, the moment of inertia of the stiffening ring is calculated. It is confirmed that the available moment of inertia is greater than the required moment of inertia. Figure 3 shows the area that is considered.

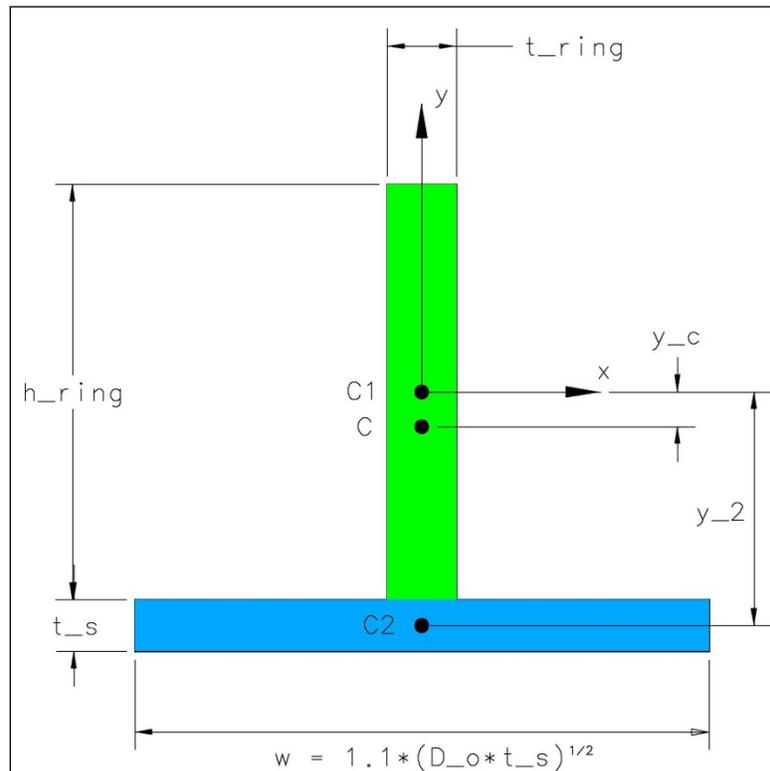


Figure 4 – Cross-Sectional Areas of Stiffening Ring and Shell Sector

The available moment of inertia in the stiffening ring is

$$I_1 := \frac{t_{ring} \cdot h_{ring}^3}{12}$$

Where  $I_1$  = available moment of inertia in the stiffening ring ( $\text{in}^4$ )

$t_{ring}$  = stiffening ring thickness (inch)

$h_{ring}$  = stiffening ring height (inch)

The required moment of inertia for the stiffening ring (from UG-29) alone is

$$I_s := \frac{\left[ D_o^2 \cdot L_s \cdot \left( t + \frac{A_s}{L_s} \right) \cdot A \right]}{14}$$

Where  $I_s$  = required moment of inertia for the stiffening ring (in<sup>4</sup>)

$D_o$  = outer diameter of the shell (inch)

$L_s$  = one-half of the distance from the centerline of the stiffening ring to the next line of support, plus one-half of the distance from the centerline of the stiffening ring to the next line of support on the other side of the stiffening ring (inch)

$t$  = minimum required shell thickness for shell at external pressure of 14.5-psia (1.0-bar) (inch)

$A_s = A_1$  = cross-sectional area of the stiffening ring (in<sup>2</sup>)

$A$  = factor determined from Table CS-2 of Subpart 3 of Section II, Part D of the Code for the stiffening ring that is made of SA-516, Gr60 (unitless)

Table 3 lists the values for the calculation of the required and available moments of inertia of the stiffening ring.

Table 3 – Values for Moment of Inertia Calculation

$t_{ring}$	0.5-inch
$h_{ring}$	3.0-inch
$A_s = A_1$	1.5-in <sup>2</sup>
$D_o$	38-inch
$L_s$	122.4-inch
$t$	0.26-inch
$A$	1.05*10 <sup>-4</sup>
$I_1$	1.125-in <sup>4</sup>
$I_s$	0.459-in <sup>4</sup>

As specified in UG-29 Step 8, *the required moment of inertia is less than the available moment of inertia for the ring ( $I_s < I_1$ ), so the ring design is adequate.*

### Attachment of Stiffening Rings (UG-30)

The stiffening ring is welded to the vacuum vessel shell with a continuous weld. The strength of the weld is calculated by comparing the actual load to the allowable load. The actual load ( $P_{\text{weld}}$ ) on the weld is a combination of the radial pressure load between the stiffeners, the weld shear flow due to the radial load through the stiffener, and the external design load carried by the stiffener. The weld shear flow is calculated by taking into account the load on the combined ring-shell cross section. The allowable load ( $P_{\text{allow}}$ ) for the weld is calculated based on the weld size and material strength.

The actual load on the weld is a combination of the radial pressure load ( $P \cdot L_s$ ) and the weld shear flow ( $VQ_x/I_s$ ):

$$P_{\text{actual}} := \sqrt{P \cdot L_s^2 + F_{\text{external}}^2 + \left( \frac{V \cdot |Q_x|}{I} \right)^2}$$

where

$$V := 0.01 \cdot P \cdot L_s \cdot D_o$$

And

$$Q_x := y_1 \cdot A_1 + y_2 \cdot A_2$$

Where  $P_{\text{actual}}$  = actual load on the attachment weld (lbf/in)

$P$  = external pressure on vessel (psi)

$F_{\text{external}}$  = external design load carried by the stiffener (lbf/in)

$V$  = radial shear load (lbf)

$Q_x$  = first moment of inertia for ring/shell shape (in<sup>3</sup>)

$L_s$ ,  $D_o$ ,  $y_1$ ,  $y_2$ ,  $A_1$ ,  $A_2$ , and  $I$  are defined in calculations for Stiffening Rings (UG-29)

The allowable load on a continuous attachment weld is calculated:

$$P_{\text{allow}} := 0.55 \cdot \text{leg} \cdot S_v$$

Where  $P_{\text{allow}}$  = allowable load on the attachment weld (lbf)

leg = weld size (inch)

$S_v$  = allowable stress for material (psi) – both the vessel and stiffening ring are made of ASTM A 516 Grade 60 (H II DIN 17155)

Table 4 lists the values that result in comparing the actual and allowable loads on the attachment weld.

Table 4 – Values for Attachment Weld Strength

P	14.5-psia
V	675-lbf
Q <sub>x</sub>	-2.63-in <sup>3</sup>
F <sub>external</sub>	0-lbf/in
Radial pressure load (P*L <sub>s</sub> )	1.8E3-lbf/in
Weld shear flow (VQ <sub>x</sub> /I <sub>s</sub> )	502-lbf/in
P <sub>actual</sub>	685-lbf/in
leg	0.19-inch
S <sub>v</sub>	17100-psi
P <sub>allow</sub>	1787-lbf/in

The actual design load P<sub>actual</sub> = 685-lbf/in is less than the allowable load P<sub>allow</sub> = 1787-lbf/in for the attachment weld. *So the continuous fillet weld with size of 0.19-inch is acceptable.*

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 a number of openings that are various sizes. Table 5 summarizes the port labels and their dimensions, which are taken from drawings 4904.280-ME-459303 and 4904.280-ME-459304 (Figures 1-2).

Table 5 – Dimensions of Openings in the Vacuum Vessel Shell

Geometric Parameters	Ports N, O, P	Coupler Port (M)	Port C	Port D	Port F-horiz	Port F-vert	Port G	Ports K1-K3
d - inner diameter of opening (inch)	15.5	10.9	2.6	7.2	4.1	4.1	3.9	6.2
t <sub>n</sub> - actual nozzle wall thickness (inch)	4.843	0.197	0.197	0.453	0.079	0.079	0.591	0.197
L <sub>nozzle</sub> - nozzle length (inch)	1.870	4.123	1.960	4.011	4.850	2.240	0.610	3.200

The size and reinforcement of each opening must follow the specifications according to the Code. For external pressure, the required area of reinforcement (A<sub>req\_ext</sub>) 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<sub>req\_ext</sub> = required area of reinforcement for external pressure (in<sup>2</sup>)

d = inner diameter of opening (in)

t<sub>r\_ext</sub> = required thickness of vacuum vessel shell for external pressure = 0.26-in.

t<sub>n</sub> = nozzle wall thickness (in.)

F = correction factor = 1.0

S<sub>v</sub> = maximum allowable stress for vessel material (SA-516, Gr 60) = 17.1-ksi

S<sub>n</sub> = maximum allowable stress for nozzle material (304L) = 16.7-ksi

f<sub>r1</sub> = strength reduction factor = S<sub>n</sub>/S<sub>v</sub> = 0.98

Other parameters that are used in the calculations for reinforcement calculations are:

t = 0.375-in. actual wall thickness of vessel shell

E1 = 1 weld joint efficiency for a nozzle that does not pass through a weld

Table 6 lists the required wall thickness of the nozzle. The wall thickness has to be thick enough to withstand the design external pressure of 14.5-psia.

Table 6 – Nozzle Wall Thickness for External Pressure of at Least 14.7-psi

Nozzle Wall Thick. External Pressure	Ports N, O, P	Coupler Port (M)	Port C	Port D	Port F-horiz	Port F-vert	Port G	Ports K1-K3
t <sub>rn</sub> - required nozzle wall thickness (inch)	0.022	0.023	0.008	0.019	0.014	0.011	0.006	0.015
A	0.0007	0.0004	0.0003	0.0003	0.0002	0.0003	0.0006	0.0003
B	7835	5345	4463	4896	3334	4509	7643	4737
Pa - external pressure allowed (psi)	14.8	15.0	18.3	17.1	15.2	16.2	15.5	15.3

For all nozzles, the actual nozzle wall thickness is greater than the required nozzle wall thickness ( $t_n > t_{rn}$ ).

Table 7 shows the available and required areas of reinforcement for each nozzle for external pressure.

**Table 7 – Available and Required Areas of Reinforcement for External Pressure**

Area of Reinforcement for External Pressure	Ports N, O, P	Coupler Port (M)	Port C	Port D	Port F-horiz	Port F-vert	Port G	Ports K1-K3
A1 (in2)	1.7	1.2	0.3	0.8	0.5	0.5	0.4	0.7
A2 (in2)	8.8	0.2	0.1	0.8	0.0	0.0	1.1	0.2
A3 (in2)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
A41, A42, A43 (in2)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
A <sub>avail</sub> - total available area of reinforcement (in2)	10.6	1.5	0.4	1.6	0.5	0.5	1.5	0.9
A <sub>req_ext</sub> - required area of reinforcement for external pressure (in2)	2.1	1.4	0.3	0.9	0.5	0.5	0.5	0.8

*For all nozzles at the external design pressure of 14.5-psia (1.0-bar), the total available area of reinforcement is at least equal to or greater than the required area of reinforcement ( $A_{avail} \geq A_{req\_ext}$ ), showing that the nozzle design follows the Code.*

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<sup>2</sup>)

$t_{r\_int}$  = required thickness of vacuum vessel shell for internal pressure = 0.045-in.

Table 8 lists the areas of reinforcement, both available and required, for all ports for an internal pressure of 29.0-psia (2.0-bar).

Table 8 – Available and Required Areas of Reinforcement for Internal Pressure

Area of Reinforcement for Internal Pressure	Ports N, O, P	Coupler Port (M)	Port C	Port D	Port F-horiz	Port F-vert	Port G	Ports K1-K3
trn - required nozzle wall thickness (inch)	2.E-07	2E-07	2E-08	1E-07	6E-08	6E-08	6E-08	1E-07
A1 (in2)	5.0	1.2	0.3	0.8	0.5	0.5	0.6	0.7
A2 (in2)	8.9	0.2	0.1	0.8	0.0	0.0	1.1	0.2
A3 (in2)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
A41, A42, A43 (in2)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
A_avail - total available area of reinforcement (in2)	13.9	1.5	0.4	1.7	0.5	0.5	1.7	0.9
A_req_ext - required area of reinforcement for external pressure (in2)	0.7	0.5	0.1	0.3	0.2	0.2	0.2	0.3

*For all nozzles at the internal design pressure of 29.0-psia (2.0-bar), the total available area of reinforcement is at least equal to or greater than the required area of reinforcement ( $A_{avail} \geq A_{req\_ext}$ ), showing that the nozzle design follows the Code.*

#### 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:

$$D_i = 37.25 \text{ in.} \quad \text{Inner diameter of the vessel shell}$$

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:

$$0.01 \times D_i = 0.373 \text{ in} \geq D_{i\_max} - D_{i\_min}$$

(per section UG-80 (a)(1) of ASME VIII, Div.1)

The exception occurs at a cross section through or near an opening. Here, the permissible difference shall not exceed 2% of the nominal inner diameter:

$$0.02 \times D_i = 0.746 \text{ in} \geq D_{i\_max} - D_{i\_min}$$

(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. The deviation from true circular form is calculated.

Given:

$$D_o = 38 \quad \text{inch (vessel outer diameter)}$$

$$t_{\text{actual}} = 0.375 \quad \text{inch (actual vessel wall thickness)}$$

$$L = 235.7 \quad \text{inch (unstiffened length of vessel)}$$

$$\frac{D_o}{t_{\text{actual}}} = 101.333$$

$$\frac{L}{D_o} = 6.203$$

$$e := 0.6 \cdot t_{\text{actual}}$$

$$e = 0.225 \quad \text{inch (maximum permissible deviation on radius, Fig. UG-80.1)}$$

## System Relief Verification

The CM2 assembly will be a component of the NML cryomodule testing facility. When installed at NML, the cryomodule's insulating vacuum becomes part of the insulating vacuum of the larger system, which includes the feed cap, the end cap, and the feedbox. CM2 has a 4-inch parallel plate relief (flow orifice 4.33-inch diameter, 14.75-in<sup>2</sup> area). The part number is D919085 (Figure 5). Figure 6 shows the P&ID of the system where CM2 sits.

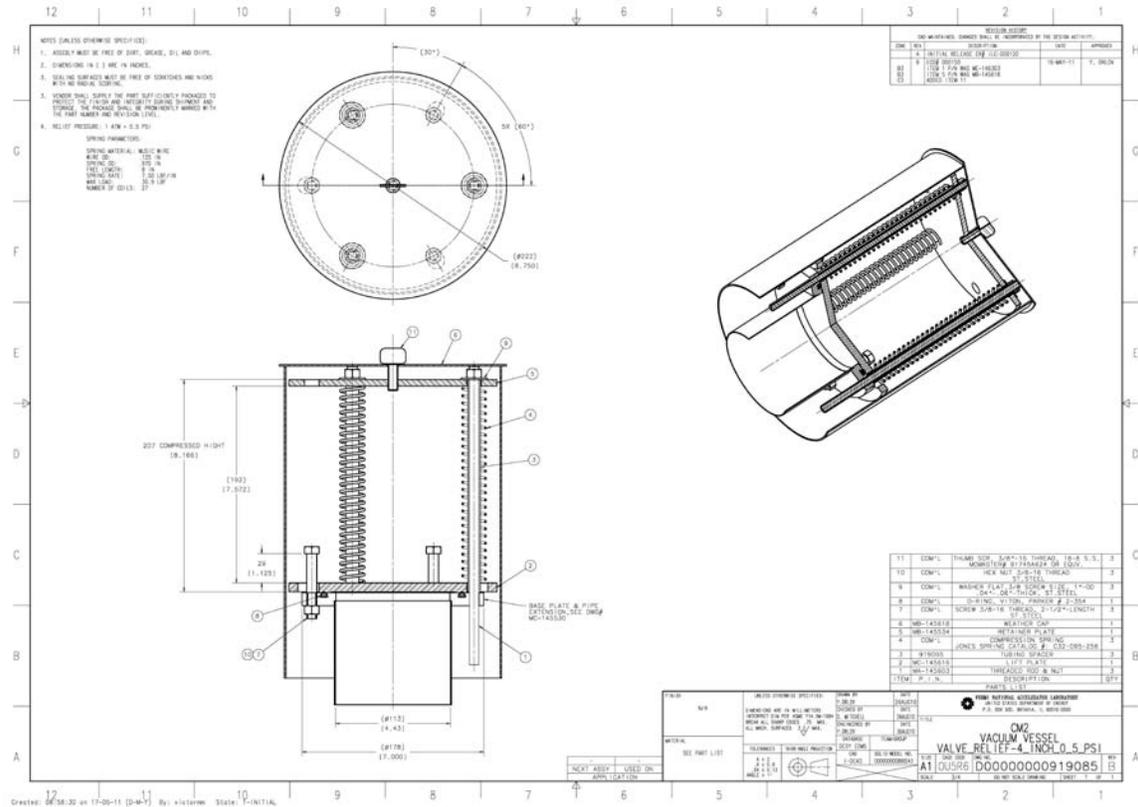


Figure 5 – 4-inch Parallel Plate Relief Device (drawing D919085)

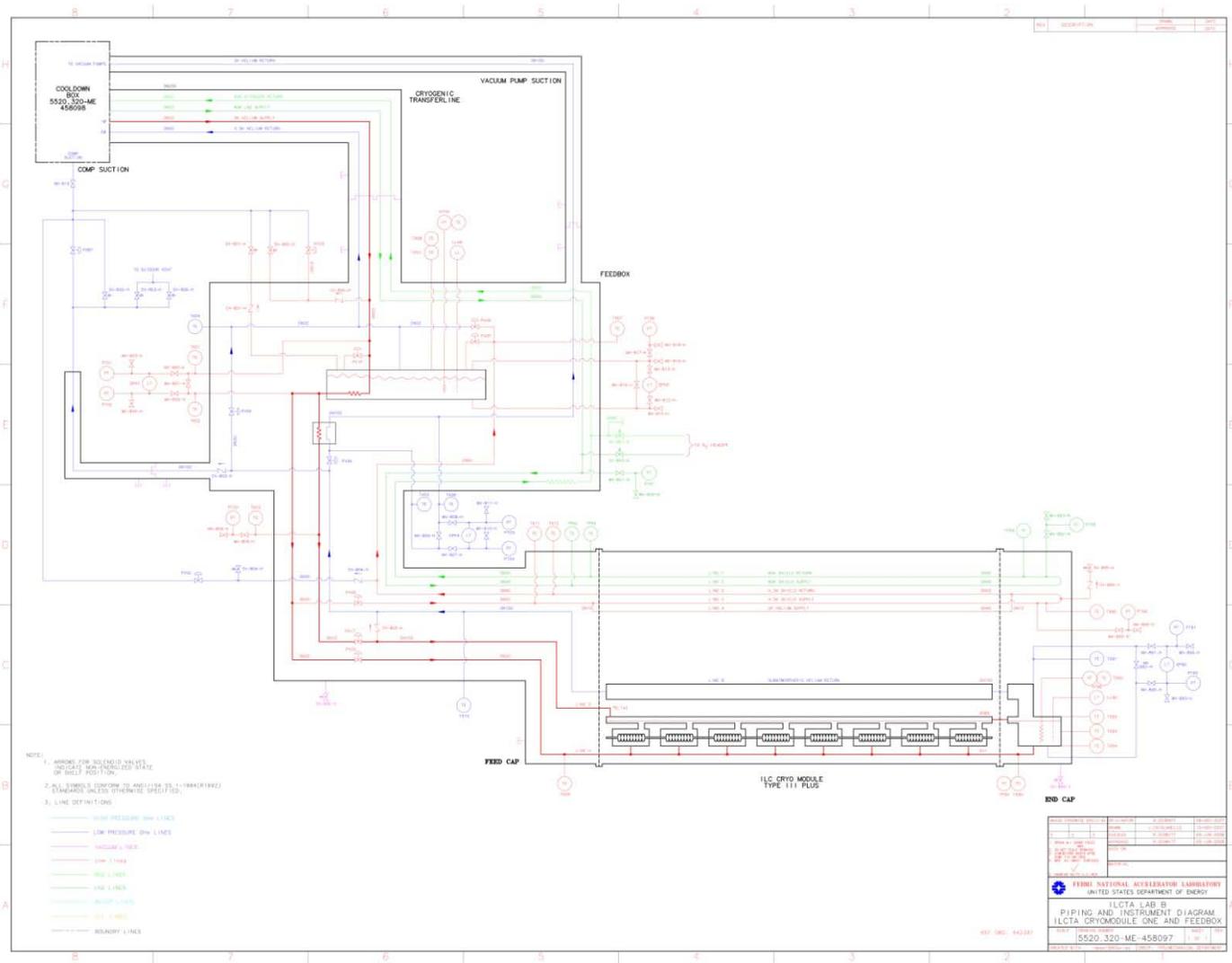


Figure 6 – P&ID of NML System where CM2 is Located (Drawing 5520.000-ME-458097)

Table 9 summarizes the mass flow rate through ruptures in the both lines and compares the flow capacity of the relief device for each cryogen. A single 4-inch relief device is adequately sized for cryogen flow in the event of a rupture in the line.

Table 9 – Comparing Maximum Mass Flow Rates of Cryogens through a Ruptured Line to the Flow Capacity of the Vacuum Relief Device

	Required	Available
Helium	80 g/sec	800 g/sec
Nitrogen	1318 g/sec	2018 g/sec
Discharge area - CGA	4.2 in <sup>2</sup>	14.7 in <sup>2</sup>

*Requirements for the Relief Device*

*Discharge Area Based on CGA S-1.3-2008, Section 5.4*

According to CGA S-1.3-2008, Section 5.4 – Jacket relief device: <sup>[6]</sup>

“The total discharge area of vacuum jacket relief devices on a container shall be at least 0.00024 in<sup>2</sup>/lb of water capacity of the container.”

The total volume of the vacuum vessel is calculated:

— —

Where

V = total volume of the vacuum vessel (ft<sup>3</sup>)

D<sub>i</sub> = inner diameter of vessel = 37.25 inch

L = vessel length = 448.23 inch

— —

The water capacity is then calculated:

—

The total required discharge area that is based on water capacity is:

—————

The available area is 14.75-in<sup>2</sup>, so the relief device is adequate.

### *Required Helium and Nitrogen Flow Rates*

Let us examine the scenario of a rupture in a cryogen line, which disrupts the insulating vacuum. Failure of the helium and the nitrogen lines are both examined. In the worst case scenario, a rupture can result from a crack in a weld or a stainless-to-aluminum transition. Some assumptions are made for the venting calculations:

- For a rupture in the line, assume that the hole diameter equals the diameter of the pipe. This leads to a maximum flow rate for a conservative analysis.
- At the rupture, assume that the cryogen is saturated liquid.
- By the time that the cryogen reaches a vacuum relief device, assume that the cryogen reaches a temperature of 150-K.
- Assume that the maximum cryogen pressure at the vacuum relief equals 0.5-psig.

The maximum helium mass flow rate is 80-g/sec<sup>[3]</sup>. This is the combined supply from the two refrigerators at NML.

The maximum nitrogen mass flow rate is 1318-g/sec (2.9-lbm/sec)<sup>[4]</sup>. The nitrogen is supplied from a dewar whose maximum allowed working pressure is 100-psig.

### *Flow Capacity of Relief Device*

The venting calculations show that the vacuum relief devices have the capacity to relieve the flow of a cryogen in the event of a rupture of an internal line in the CM2 vacuum vessel. The mass flow rate of a gas that is required to go through the parallel plate relief is calculated using the formulas from API 520<sup>[5]</sup> for sonic flow. For nitrogen, flow is sonic, so the equation used to calculate the mass flow rate through the relief is:

$$W = AK_D CP \sqrt{\frac{M}{TZ}}$$

where

—

Table 10 defines the variables and lists the mass flow rate for both helium and nitrogen at the parallel plate relief valve at 150°K.

Table 10 – Mass Flow Rate through Parallel Plate Relief Valve

		Helium	Nitrogen	
P	set pressure plus overpressure allowance	15.2	15.2	psia
K <sub>D</sub>	coefficient of discharge	0.62	0.62	
C	gas flow constant for sonic flow	377	356	
d	internal diameter of relief device	4.334	4.334	in
A	flow area	14.75	14.75	in <sup>2</sup>
M	molecular weight	4	28	
T	absolute temperature at inlet	270	270	R
Z	compressibility factor	1.014576	0.994669	
ρ	density at flow conditions	0.0207	0.1478	lbm/ft <sup>3</sup>
R	ideal gas constant	1544	1544	lbf/(lbm-R)
W	mass flow rate	6333.6	15981.3	lb/hr
		799.7	2017.8	g/sec

For both nitrogen and helium, the capacity of a vacuum relief device is adequate for either cryogen that leaks from a rupture line.

Operating Procedures

For safe operation of the CM2 vacuum vessel, operating procedures are not required.

Welding Information

E. Zanon, the manufacturer of the vacuum vessel, provided a data book that accompanied the vessel. This document includes material certifications, welding specifications, inspection results, and quality control documents on the vessel assembly. The welding certifications are on pages 150-223. An electronic copy of the document is located online in the CM2 folder of the ILC Document Management System (to view this document, log in to the ILC DMS):

<http://ilc-dms.fnal.gov/Workgroups/CryomoduleDocumentation/CM2folder/VacVessel/>

## References

1. ES&H Manual Chapter 5033, Fermilab.
2. ASME Boiler & Pressure Vessel Code 2007.
3. A. Klebaner, email 18 August 2008.
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6. Compressed Gas Association, "Pressure Relief Device Standards Part 3 – Stationary Storage containers for Compressed Gases," CGA S-1.3-2008, 8<sup>th</sup> Edition.