Pressure Vessel Engineering Note
For the
1.3-GHz Helium Vessel #1 in Cryomodule 1

Authors: B. Wands, M. Wong	Date: 1 April 2010

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PRESSURE VESSEL ENGINEERING NOTE

PER FESHM CHAPTER 5031

Prepared by: Bob Wands, Mayling Wong
Preparation date: 1 April 2010

1. Description and Identification
Fill in the label information below:

This vessel conforms to Fermilab ES&H Manual Chapter 5031

Vessel Title CM1 Helium Vessel Numbers 1

Vessel Number IND-116 (Vessel #1)

Vessel Drawing Number DESY 1-98-8427/0.000

Maximum Allowable Working Pressures (MAWP):
Warm Internal Pressure 2.0-bar(15.0-psig)@300K
Cold Internal Pressure 4.0-bar (43.3-psig)@ 2K
External Pressure 1.0-bar (0-psig)

Working Temperature Range -457°F - 100°F

Contents Superfluid helium

Designer/Manufacturer DESY / Accel, Zanon

Test Pressure (if tested at Fermi) Acceptance Date: ______ ←Document per Chapter 5034 of the Fermilab ES&H Manual

______PSIG, Hydraulic______ Pneumatic______

Accepted as conforming to standard by
/Giorgio APOLLINARI

of Division/Section Technical Div Date: 7/23/10 ←Actual signature required

NOTE: Any subsequent changes in contents, pressures, temperatures, valving, etc., which affect the safety of this vessel shall require another review.

Reviewed by: ______ Date: 7/1/10

Director's signature (or designee) if the vessel is for manned areas but doesn't conform to the requirements of the chapter.

S.H. ______ Date: 7/16/10

______ Date: 7/23/10
ES&H Director Concurrence

Amendment No.:	Reviewed by:	Date:
1	[signature]	11/15/10

Lab Property Number(s):

Lab Location Code: FIMS #700 (Muon Beam Enclosure at NML)

Purpose of Vessel(s): Liquid helium containment for nine-cell 1.3-GHz Superconducting radio frequency cavity

Vessel Capacity/Size: 23-L Diameter: 9.3 in (237mm) Length: 50.5 in (1.3-m)

Normal Operating Pressure (OP) 0.02-bar (0.25-psia)

MAWP-OP = 28.75 PSID

List the numbers of all pertinent drawings and the location of the originals.

Drawing #	Location of Original
1-98-8427/0.000	DESY
1-98-8427/8.000	DESY

2. Design Verification

Is this vessel designed and built to meet the Code or 'In-House Built' requirements?
Yes_____ No X .

If "No" state the standard that was used It is not known what standard the vessels followed. Note that the vessels were built and manufactured at DESY.

Demonstrate that design calculations of that standard have been made and that other requirements of that standard have been satisfied.
Skip to part 3 'system venting verification.'

Does the vessel(s) have a U stamp? Yes_____ No X . If "Yes", complete section 2A; if "No", complete section 2B.

A. Staple photo of U stamp plate below.
Copy "U" label details to the side

Copy data here:

Provide ASME design calculations in an appendix. On the sketch below, circle all applicable sections of the ASME code per Section VIII, Division I. (Only for non-coded vessels)

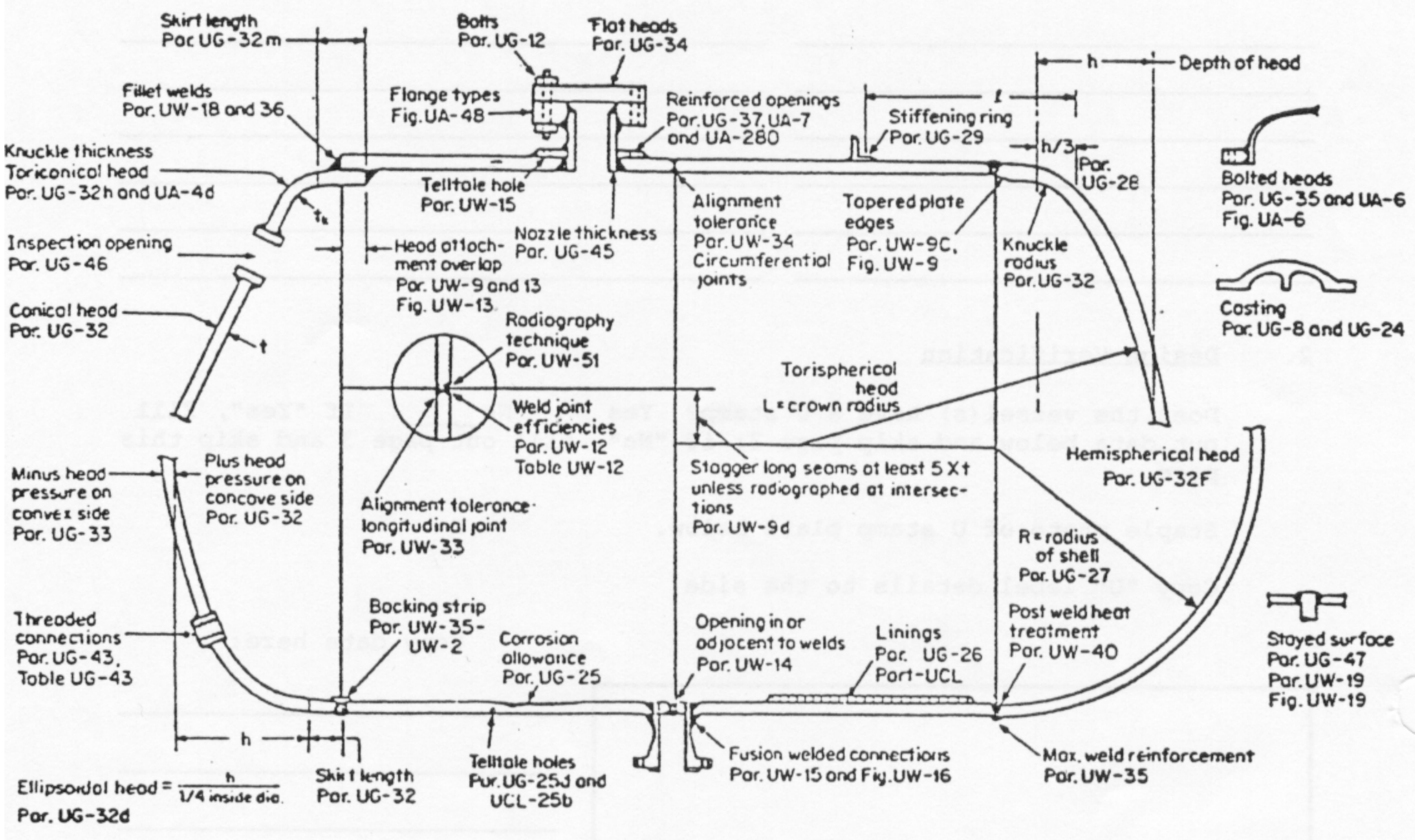


Figure 1. ASME Code: Applicable Sections

2B.

<u>Summary of ASME Code</u>

Item	Reference ASME Code Section	CALCULATION RESULT (Required thickness or stress level vs. actual thickness calculated stress level)
______	______	______ vs ______
______	______	______ vs ______
______	______	______ vs ______
______	______	______ vs ______
______	______	______ vs ______

3. System Venting Verification Provide the vent system schematic.

Does the venting system follow the Code UG-125 through UG-137?
Yes___ No_X__

Does the venting system also follow the Compressed Gas Association Standards S-1.1 and S-1.3?
Yes __X__   No_____

A 'no' response to both of the two proceeding questions requires a justification and statement regarding what standards were applied to verify system venting is adequate.

List of reliefs and settings:

Manufacturer	Model #	Set Pressure	Flow Rate	Size
Leser	4414.4722	43-psig	8053-SCFM air	6in x 8in
Leser	4414.7942	15-psig	951-SCFM air	2in x 3in

**Flow rates updated on 3 November 2010 as part of Amendment 1 to the note. See Amendment 1 for detailed explanation.

4. Operating Procedure

Is an operating procedure necessary for the safe operation of this vessel?
Yes_____ No__X___ (If "Yes", it must be appended)

5. Welding Information

Has the vessel been fabricated in a non-code shop? Yes__X__ No_____
If "Yes", append a copy of the welding shop statement of welder qualification (Procedure Qualification Record, PQR) which references the Welding Procedure Specification (WPS) used to weld this vessel.
The helium vessel was manufactured at DESY. At the time of fabrication, there was no plan to use it at Fermilab. So, no welding information is available.

6. Existing, Used and Unmanned Area Vessels

Is this vessel or any part thereof in the above categories?
Yes_____ No__X__

If "Yes", follow the requirements for an Extended Engineering Note for Existing, Used and Unmanned Area Vessels.

7. Exceptional Vessels

Is this vessel or any part thereof in the above category?
Yes__X__ No_____

If "Yes", follow the requirements for an Extended Engineering Note for Exceptional Vessels.

Appendix A
Extended Engineering Note for Exceptional Vessel

Introduction

Cryomodule #1 (CM1) contains a string of eight 1.3-GHz dressed cavities. A dressed cavity is a niobium superconducting RF cavity surrounded by a titanium shell. The shell acts as a vessel that contains superfluid helium so that the helium surrounds the RF cavity at temperatures as low as 1.8-K. The vessel also mechanically supports the cavity and takes part in tuning it. The maximum pressure that the helium can reach is 15.0-psig (2-bar), so the vessel is defined as a Pressure Vessel, according to FESM 5031. [(1)] This engineering note follows the guidelines as presented in FESHM 5031. This note, along with related documents, is stored online on the ILC Document Management System. The website is:

http://ilc-dms.fnal.gov/Workgroups/CryomoduleDocumentation/CM1folder/he-vessel-folder/

Figure 2 shows the cross-section of CM1 (DESY drawing 0-06-8205-0-000). Note how the dressed cavities are numbered. This engineering note pertains to Vessel Number IND-116 (Cavity #1). Subsequent engineering notes for the remaining dressed cavities will refer to this engineering note since all vessels have the same design and purpose. Figure 3 shows the assembly drawing of the 1.3-GHz dressed cavity for CM1 (DESY drawing 1-98-8427-0.000).

History of CM1

As part of an agreement between FNAL and DESY, a cryomodule "kit" was put together jointly by DESY and INFN-Milano and shipped to FNAL in July, 2007. The kit included the vacuum vessel and cold mass assemblies, as well as eight individual 1.3-GHz dressed cavities. The cryomodule was assembled at Fermilab by FNAL personnel assisted by DESY personnel. Now completed, the cryomodule resides at its final location at the Beam Enclosure in the New Muon Lab in preparation for the commissioning of the cryogenic system at that facility.

Exceptional Vessel Discussion

Reasons for Exception

Pressure vessels, as defined in FESHM Chapter 5031, are designed and fabricated following the ASME Boiler and Pressure Vessel Code (the Code) [(2)]. The 1.3-GHz dressed cavity as a helium pressure vessel has materials and complex geometry that are not conducive to complete design and fabrication following the Code.

The 1.3-GHz dressed cavities in CM1 were designed at DESY. The fabrication of the cavities took place at various vendors with oversight by DESY personnel. The fabrication took place over a span of many years (1998-2006). The dressed cavities were selected by DESY to be included in the CM1 kit sent to FNAL.

Since the dressed cavities, also known as helium vessels, were designed and built outside of FNAL oversight, detailed information about the vessels are not available. The information that is usually included in a pressure vessel engineering note is not available for the CM1 dressed cavities. The missing information includes detailed engineering drawings, material and fabrication certification by the manufacturer, and pressure test results. However, we show that the vessel is safe in accordance with FESHM 5031. Since the vessel design and fabrication methods cannot exactly follow the guidelines given by the Code, the vessel requires a Director's Exception. Table 1 lists the specific areas of exception to the Code.

Table 1 – Areas of Exception to the Code

Item or Procedure	Reference	Explanation for Exception	How the Vessel is Safe
Niobium material	Pg 28	Used for its superconducting properties; Not an established material listed by the Code	There has been extensive testing done on the niobium used in the cavity. The Code procedure for determining Div.1 allowable stresses (see Section II, Part D, Mandatory Appendix 1) are conservatively applied to the measured yield and ultimate stresses to establish allowable stresses which are consistent with Code philosophy.
Niobium-Titanium material	Pg 28	Used for as a transition material between niobium and titanium materials for welding purposes; Not an established material listed by the Code	Material properties were provided by the vendor of the material.
No information about the vessel's weld design is available.	Pg 22	Category B joints in titanium must be either Type 1 butt welds (welded from both sides) or Type 2 butt welds (welded from one side with backing strip) only (see the Code, Div. 1, UNF-19(a)).	The evaluation of these welds is based on a weld efficiency of 0.5. This weld efficiency is lower than the lowest efficiency specified by the Code for any weld.
No information about liquid penetrant testing on the titanium sub-assembly is available.	Pg 22	All joints in titanium vessels must be examined by the liquid penetrant method (see the Code, Div. 1, UNF-58(b)).	The evaluation of these welds is based on a weld efficiency of 0.5. This weld efficiency is lower than the lowest efficiency specified by the Code for any weld.
No information about ultrasonically testing the electron beam welds in the niobium and niobium-titanium assemblies is available.	Pg 22	All electron beam welds in any material are required to be ultrasonically examined along their entire length (see the Code, UW-11(e)).	The evaluation of these welds is based on a weld efficiency of 0.5. This weld efficiency is lower than the lowest efficiency specified by the Code for any weld.
No information about radiography inspection on the titanium welds is available.	Pg 22	All titanium welds require radiography inspection (see the Code, UNF-57(b))	The evaluation of these welds is based on a weld efficiency of 0.5. This weld efficiency is lower than the lowest efficiency specified by the Code for any weld.

Table 1 (continued) – Areas of Exception to the Code

Item or Procedure	Reference	Explanation for Exception	How the Vessel is Safe
Calculated stresses for longitudinal weld in titanium bellows exceed allowable stresses.	Pg 37	Calculated stresses must be at or less than allowable stresses. The allowable stresses include a 0.7 weld joint efficiency due to lack of examination results.	The calculated stress does not exceed the allowable stress with a joint efficiency of 1.0. This design of the bellows has been used extensively at DESY for over the past decade.
Calculated stress in the bellows using FEA shows a higher membrane plus bending stress than allowed.	Pg 48	Calculated stresses must be at or less than allowable stresses.	The design of the bellows is addressed by the Code in Div 1, Appendix 26. The sum S_3+S_4 is less than allowed K_fS (see pg 37).
Use of enhanced material properties at cryogenic temperatures in stress analysis	Pg 28	Titanium is not a material with established material properties at temperatures less than 38°C by the Code (see the Code, ULT-5(b))	Published material properties for titanium (outside the Code) at cryogenic temperatures were used.
Weld documents, including the WPS, PQR, or WPQ, are not available.	Pg 28	All welds must follow the rules of specifying the weld procedure, qualifying the weld procedure, and qualifying the welder according to Part UW, which refers to the Code, Sec. IX.	• The evaluation of these welds is based on a weld efficiency of 0.5. This weld efficiency is lower than the lowest efficiency specified by the Code for any weld. • The RF performance of the niobium cavity is acceptable, showing indirectly that all welds in the cavity are full penetration
Pressure test results are not available.	Pg 44, Table 9	All Exceptional Vessels require a pressure test, according to FESHM 5034.	The analysis shows that the stresses in the vessel, when pressurized at room temperature (Load Case No. 1), are within the allowable stress.

Analysis and use of the ASME Code

The extended engineering note presents the results of the analysis that was performed on the entire vessel.

Analytical Tools

Analysis was done using ANSYS Workbench 11.

Fabrication

The cavity processing data for each cavity is available online at the DESY database:

http://tesla-new.desy.de/cavity_database/summaries/

Included in the processing data for each cavity are material properties of the niobium. However, no material certifications for the niobium are available by the cavity manufacturer. No material data exists for the niobium-titanium parts.

Regarding the niobium-titanium parts, the titanium parts and the entire vessel assembly, weld specifications, welder qualifications, or weld samples from the manufacturer are not available. Material certifications for these items are not available. Inspection results are not available.

Hazard Analysis

When in operation as part of CM1, the 1.3-GHz dressed cavity is completely contained within a multilayered vessel that protects personnel. The 5K aluminum thermal shield completely surrounds the dressed cavity. The 70K aluminum thermal shield, in turn, completely surrounds the 5K shield. The shields sit within the carbon steel vacuum vessel. From a personnel safety standpoint, the dressed cavity is well contained within the CM1 vacuum vessel.

Two relief valves vent any helium spill from the dressed cavity. The section titled "System Venting Verification" details the venting analysis in this engineering note.

Pressure Test

No pressure tests for the individual dressed cavities were performed, so no pressure test results exist. However, every dressed cavity successfully performed at operating pressures and temperatures during horizontal testing at DESY.

Additional Information

The design for the dressed cavities utilized in CM1 has been proven at DESY and used in a number of facilities. More than 100 dressed cavities of this design have been built and tested. Many of them have been used in the cryomodules presently installed in the TESLA Test Facility (TTF).

The design will be used in the cryomodules for the European XFEL facility. As part of the preparations for building the XFEL cryomodules, DESY performed several pressure tests on a cryomodule to demonstrate compliance with European safety standards and to understand the design safety factor. [(3)] The Module Crash Test was a series of pressure tests. The first test took place while the cavities were at 2K. The dressed cavities began the test at operating conditions (2K and 30-mbar). The pressure in the 2K helium circuit was then increased to 6.1-bar without an increase in the cavity pressure or leak rate. The cavity did not experience plastic deformation during the cold test, only elastic deformations. The second and third tests took place with the cavities at room temperature and at 1-bar. The pressure in the helium circuit was increased to 5.3-bar without plastic deformations in both tests. A fourth test took place with the cavities at

2K. Again, the helium pressure was increased to over 6-bar without problems. The results show that the cavities are safe within the warm MAWP of 2-bar and cold MAWP of 4-bar. The next step for the XFEL cryomodule will be to create a simple test procedure as part of certifying the dressed cavity according to the European pressure vessel safety standard (TÜV).

In another set of tests in the Module Crash Test, air was leaked into different parts of the cryomodule in order to mimic fault conditions such as the sudden loss of insulating vacuum or venting of the cavity vacuum to air. [(4)] For this series of tests, the cavities began the test cold and with a vacuum pulled in the cavity. Either the insulating vacuum or the cavity vacuum was let up to air. The maximum pressures in the 2K helium circuit, the 5K helium circuit, and the nitrogen circuit were then measured. The temperature of the helium vessel during the test was also monitored. The air heat flow and average heat transfer densities estimated based on measurements of the test. Despite the catastrophic nature of the venting scenarios, the maximum pressures that were measured were less than the design pressure for each of the cryogen circuits, proving that the design of the system is safe.

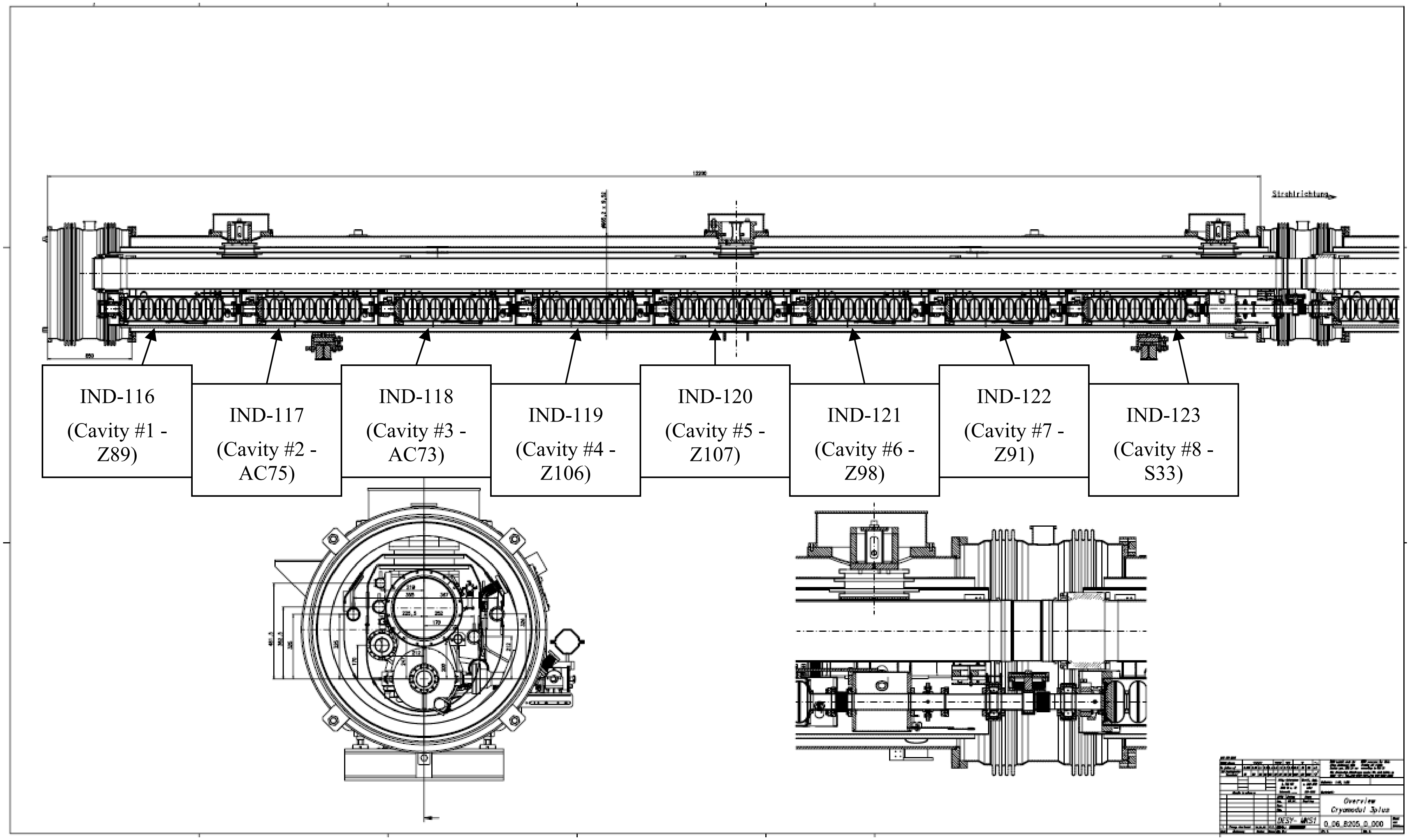


Figure 2 – Cross Section of CM1 (DESY drawing 0-06-8205-0-000)

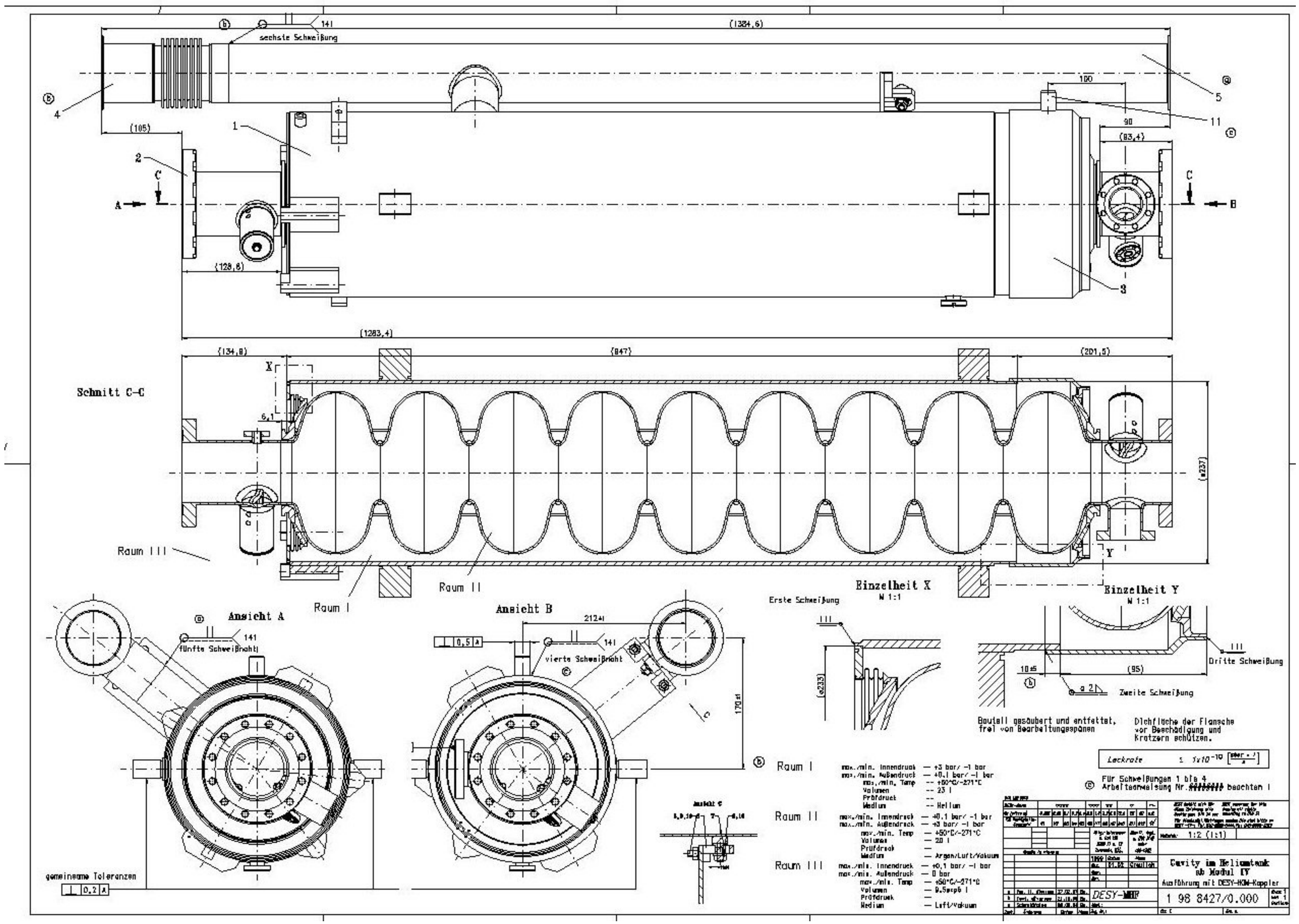


Figure 3 – 1.3-GHz Dressed Cavity for CM1 (DESY drawing 1-98-8427-0.000)

Description and Identification

CM1 contains a string of eight dressed cavities. The first dressed cavity in the string ("Cavity #1") is arbitrarily named pressure vessel number IND-116. Table 2 identifies the cavity number, with the corresponding Fermilab pressure vessel number and the DESY label. Table 2 also lists the year that the bare cavity was manufactured.

Table 2 – Cavity Identification Numbers

Cavity No.	Fermilab Pressure Vessel No.	DESY Label	Manufacturing Year
1	IND-116	Z89	2005
2	IND-117	AC75	2001
3	IND-118	AC73	2001
4	IND-119	Z106	2006
5	IND-120	Z107	2005
6	IND-121	Z98	2005
7	IND-122	Z91	2005
8	IND-123	S33	1998

The top assembly drawing of the assembly, DESY drawing 1-98-8427-0.000, is shown in Figure 3. The dressed cavity consists essentially of two sub-assemblies: the niobium SRF cavity and the titanium helium vessel.

Since the dressed cavities of CM1 were designed and fabricated at DESY, detailed drawings of the assembly are not available. However, a 3D model is available, so nominal dimensions of the parts are taken from it.

The niobium SRF cavity is an elliptical nine-cell assembly. A drawing of the nine-cell cavity is shown in Figure 4 (DESY drawing 1-98-8427/8.000). The cavity assembly consists of the niobium RF cavity and the end units. A single cell, or a dumbbell, consists of two half-cells that are welded together at the equator of the cell. Rings between the cells stiffen the assembly to a point. Some flexibility in the length of the nine-cell cavity is required to tune the cavity and optimize its resonance frequency. The end units each consist of a half cell, an end disk flange, and a transition flange. The transition flange is made of a titanium-niobium alloy. A titanium bellows assembly is attached to the longer end unit. The iris' minimum inner diameter is 35-mm (1.4-in), and the maximum diameter of a dumbbell is 211.1-mm (8.3-in). The length of the cavity, flange-to-flange, is 1247.4-mm (49.1-in.).

The titanium helium vessel encases the niobium SRF cavity. The inner diameter of the cylindrical part of the vessel is roughly 237-mm (9.3-in.). The shell is welded to the bellows on one end and to the cavity's niobium-titanium end cap disk at the other end. The vessel has a helium fill port at the bottom. Close to the top of the vessel is the two-phase helium return line. At the sides of the vessel are tabs which support the vessel within the CM1 vacuum vessel. The vessel is flexible in length due to a bellows at the middle of its length. This flexibility in the vessel allows for accommodating the change in the nine-cell cavity length due to thermal contraction at cryogenic temperature and to turning the cavity during operation. A titanium bellows allows for adjusting the cavity length. A slow-control tuner system that consists of a stepper motor that changes the vessel length to accommodate thermal shrinkage.

The vessel contains liquid helium at 2K during operating. The vessel's operating pressure is 30-mbar internal. The vessel's internal maximum allowable working pressure (MAWP) is 2.0-bar (15.0-psig) at room temperature. At the operating temperature of 2K, the vessel's internal MAWP is 4.0-bar (43.3-psig). The increased strength of the materials at the cryogenic temperature allows for a higher MAWP.

The external MAWP of the vessel is 1.0-bar (0-psig).

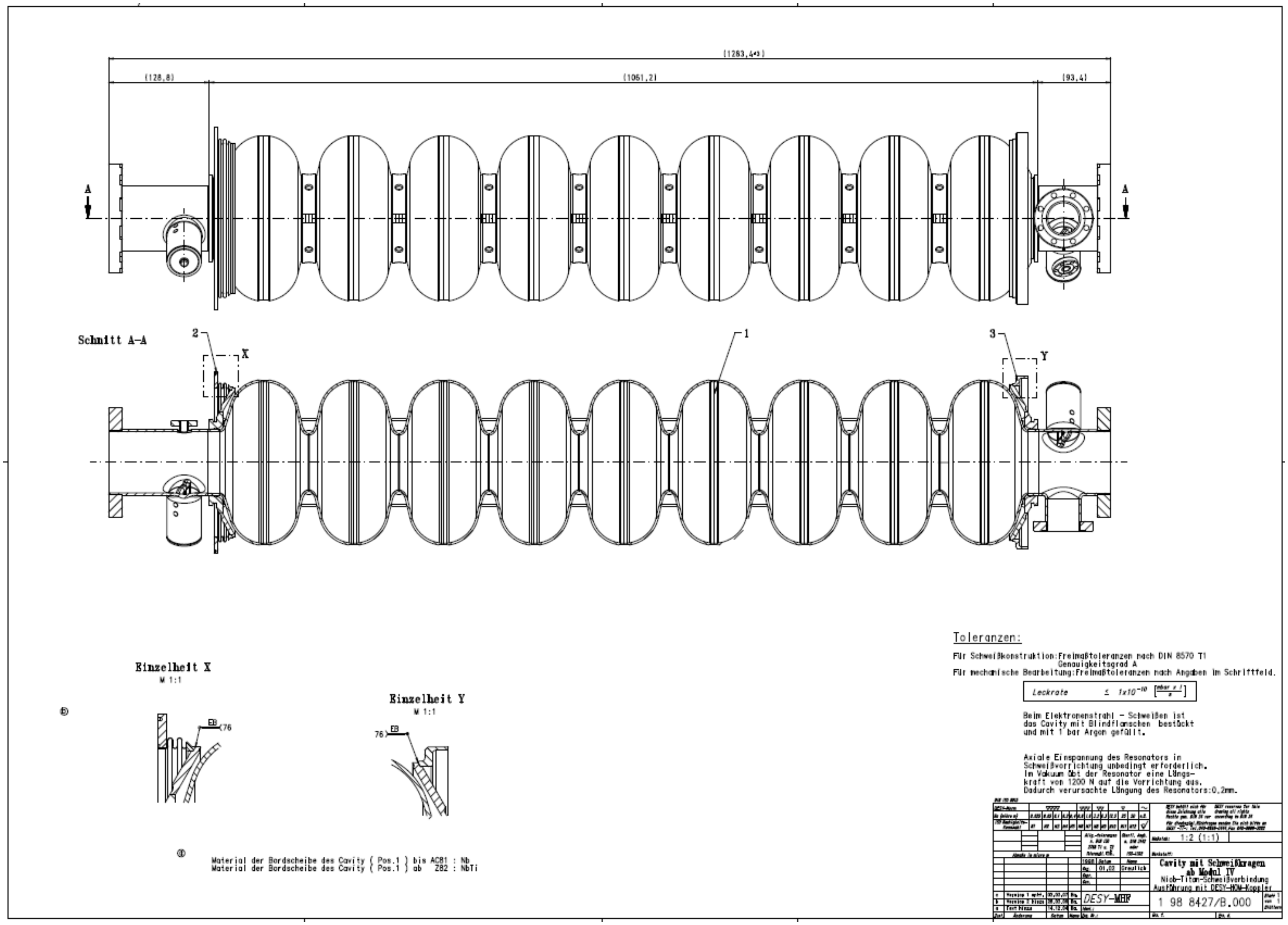


Figure 4 – RF Cavity Assembly (DESY drawing 1-98-8427/8.000)

Design Verification

Introduction and Summary

This analysis is intended to demonstrate that the CM1 1.3 GHz SRF cavity conforms to the ASME Boiler and Pressure Vessel Code (the "Code"), Section VIII, Div. 1, to the greatest extent possible.

Where Div. 1 formulas or procedures are prescribed, they are applied to this analysis. For those cases where no rules are available, the provisions of Div. 1, U-2(g) are invoked. This paragraph of the Code allows alternative analyses to be used in the absence of Code guidance.

This cavity contains several features which are not supported by the Code. These are related primarily to materials, weld types, and non-destructive examination, and are addressed in detail in the next section of this report, titled "Non-Code Elements." These are accepted as unavoidable in the context of SRF cavities, and every effort is made to demonstrate thorough consideration of their implications in the analysis.

The CM-1 cavity construction details are poorly understood. It is known to contain welds of questionable quality. Therefore, weld fusion zone dimensions for this analysis were taken from those measured on sectioned weld samples. For additional conservatism, all welds in the CM-1 were given a weld efficiency of 0.5, which is lower than any weld efficiency specified by the Code.

Advantage is taken of the increase in yield and ultimate strength which occurs in the Nb and Ti components at the operating temperature of 1.88 K.

The design pressures specified for this analysis are 30 psi (2.0-bar) at 293 K, and 60 psi (4.0-bar) at 1.88 K. This analysis confirms that the MAWPs of the vessel can be safely set at these pressures. Negligible margin for increase is available at 293 K, but the cold MAWP could be increased somewhat above 60 psi (4.0-bar).

Of all the stress limits checked in this analysis, the only violation was a slight (1%) overstress in the Ti bellows for secondary stresses due to thermal contraction, tuner extension, and pressure effects. Primary stresses in the bellows were well below the stress limits in both warm and cold operation. The bellows design was analyzed following the Code for expansion joints. The analysis shows that the bellows at cold temperatures has a design safety factor of 2.5.

In addition to these fundamental operating limits, the cavity was also shown to be stable at external pressures on the Ti shell of 15 psid (1.0-bar), and internal pressures on the Nb cavity of 15 psid (1.0-bar); these loadings could occur under fault conditions, when the beam and insulating vacuums have been compromised, and the helium volume has been evacuated.

Non-Code Elements

With regards to the Design Verification, the CM-1 1.3 GHz cavity does not comply with Div. 1 of the Code in the following ways (these are the first six items from Table 1):

1. Pure niobium, and Ti-45Nb titanium alloy are not “Code” materials, i.e., they have not been approved for use in Div. 1 or Div. 2 vessels, and there are no mechanical properties available from Code sources.
2. Category A and B joints in titanium must be either Type 1 butt welds (welded from both sides) or Type 2 butt welds (welded from one side with backing strip) only (see Div. 1, UNF-19(a)). The welds in the CM-1 are undocumented, and cannot at this point be verified to comply with any Code requirements.
3. All category A and B welds in titanium must be fully radiographed (see Div. 1, UNF-57(b)). No radiography results are available.
4. All joints in titanium vessels must be examined by the liquid penetrant method. (see Div. 1, UNF-58(b)). No liquid penetrant testing results are available.
5. All electron beam welds in any material are required to be ultrasonically examined along their entire length. (see UW-11(e)). No ultrasonic examination results are available.
6. The use of enhanced material properties for cold operation is permitted by Part ULT of Div. 1 for five materials: 5%,8%, and 9% nickel steels; 5083-Al; and Type 304 SS. The use of enhanced material properties for the cavity materials is not permitted. For this design analysis, published material properties for titanium (outside the Code) at cryogenic temperatures are used.

Although material properties are not available for Nb or Ti-45Nb from Code sources, there has been extensive testing done on the Nb used in the cavity. The Code procedures for determining Div. 1 allowable stresses (see Section II, Part D, Mandatory Appendix 1) are conservatively applied to the measured yield and ultimate stresses to establish allowable stresses which are consistent with Code philosophy.

To compensate for the poor understanding of welds, a uniform de-rating of 0.5 was applied to every weld in the structure.

Geometry

General

This analysis is based on geometry obtained from a solid model created by engineers at DESY. There are no released FNAL engineering drawings available for this cavity.

Fig. 5 shows the dressed cavity, complete with shielding, piping and blade tuner.

For the analysis, only the Nb cavity, conical Ti-45Nb heads, and titanium shells and bellows are modeled, as well as the flanges to which the blade tuner attaches to the Ti cylindrical shell. These components are shown in Fig. 6.

The geometric limits of the analysis are further clarified in Fig. 7.

The individual cavity component names used in this report are shown in Fig. 8.

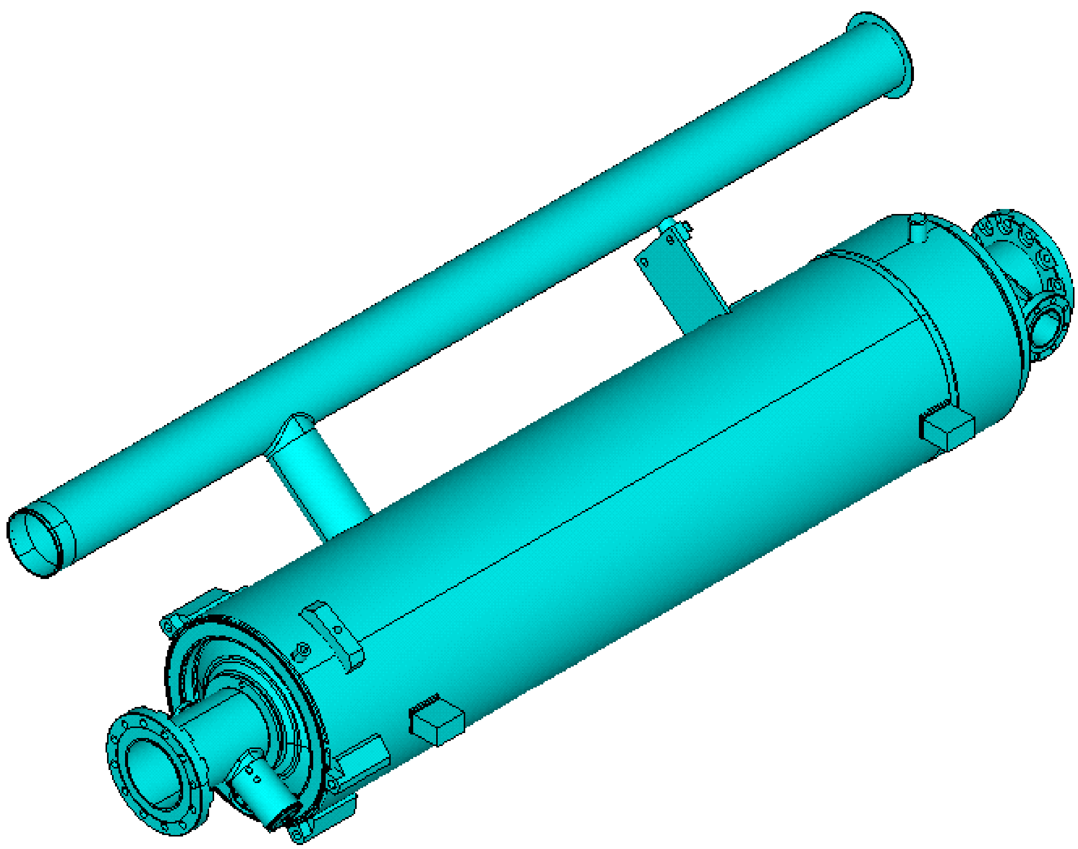


Figure 5 - CM-1 SRF cavity

Cavity assembly as analyzed (Nb cavity not fully visible)

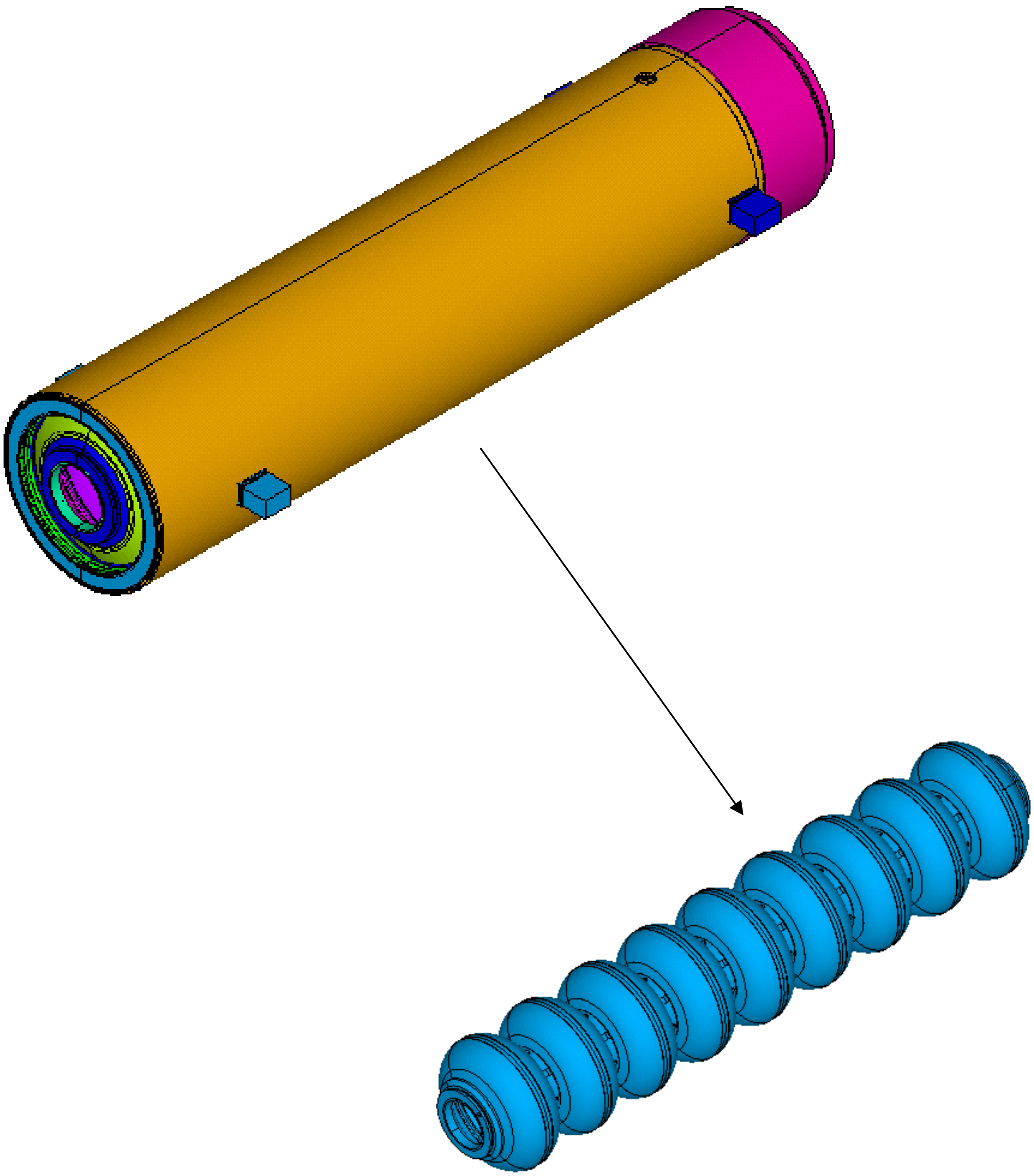


Figure 6 - Cavity components considered in the analysis

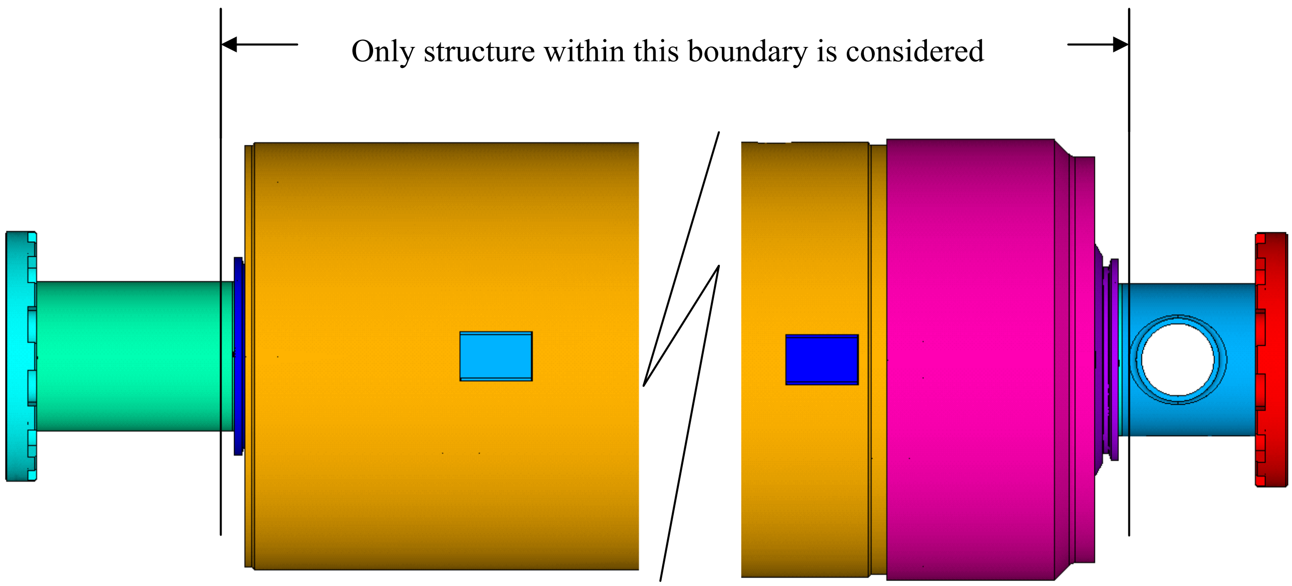


Figure 7 - Geometric limits of analysis

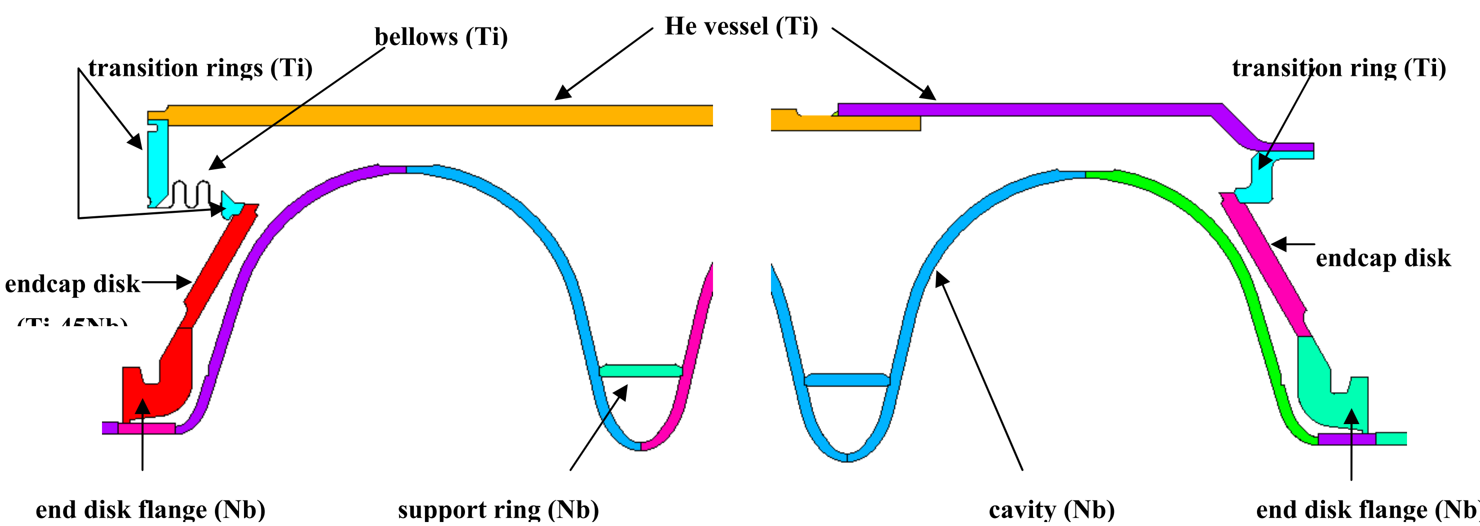


Figure 8 - Parts and Materials

Welds

Welds are produced by the EB process (in the Nb, and Nb-to-Ti transitions), and the TIG (GTAW) process (Ti-Ti welds).

Locations of the welds are shown in Figure 9. Detailed weld configurations and assumed zones of fusion are illustrated in Figs. 10-13.

Most construction details of the CM-1 are not a matter of record. Therefore, in an attempt to understand weld geometries, sectioning of DESY test welds was performed. [5] This sectioning yielded the minimum fusion zone dimensions for several welds in this analysis, specifically welds 8,9, and 10 (see Figs. 10-13)

Faced with undocumented welds of non-Code dimension and unproven quality, the decision was made to give all welds, regardless of material joined, a weld efficiency of 0.5. This is lower than the lowest efficiency specified by the Code for any weld, corresponding to less than that typically applied to uninspected fillet welds. It is felt that between using minimum fusion zone dimensions, and this large derating factor, confidence in the analysis can be asserted.

Table 3 – Summary of Welds

Weld	Weld Description	Materials Joined	Weld Process	Joint Efficiency
1	End tube spool piece to end cap flange	Nb-Nb	EB	0.50
2	End tube spool piece to RF Half Cell	Nb-Nb	EB	0.50
3	End cap flange to RF half cell	Nb-Nb	EB	0.50
4	End cap flange to end cap disk	Nb-Ti45Nb	EB	0.50
5	End cap disk to flanged ring	Ti45Nb-Ti	EB	0.50
6	Cavity bellows to flanged ring	Ti-Ti	TIG	0.50
7	Flanged wheel to cavity bellows	Ti-Ti	TIG	0.50
8	Vessel tube to flanged wheel	Ti-Ti	TIG	0.50
9	Vessel end cap to reducing collar	Ti-Ti	TIG	0.50
10	Vessel tube to reducing collar	Ti-Ti	TIG	0.50
11	Dumbbell to dumbbell	Nb-Nb	EB	0.50
12	Support ring to half cell	Nb-Nb	EB	0.50
13	Half cell to half cell	Nb-Nb	EB	0.50
--	Longitudinal weld in bellows	Ti-Ti	TIG	0.50

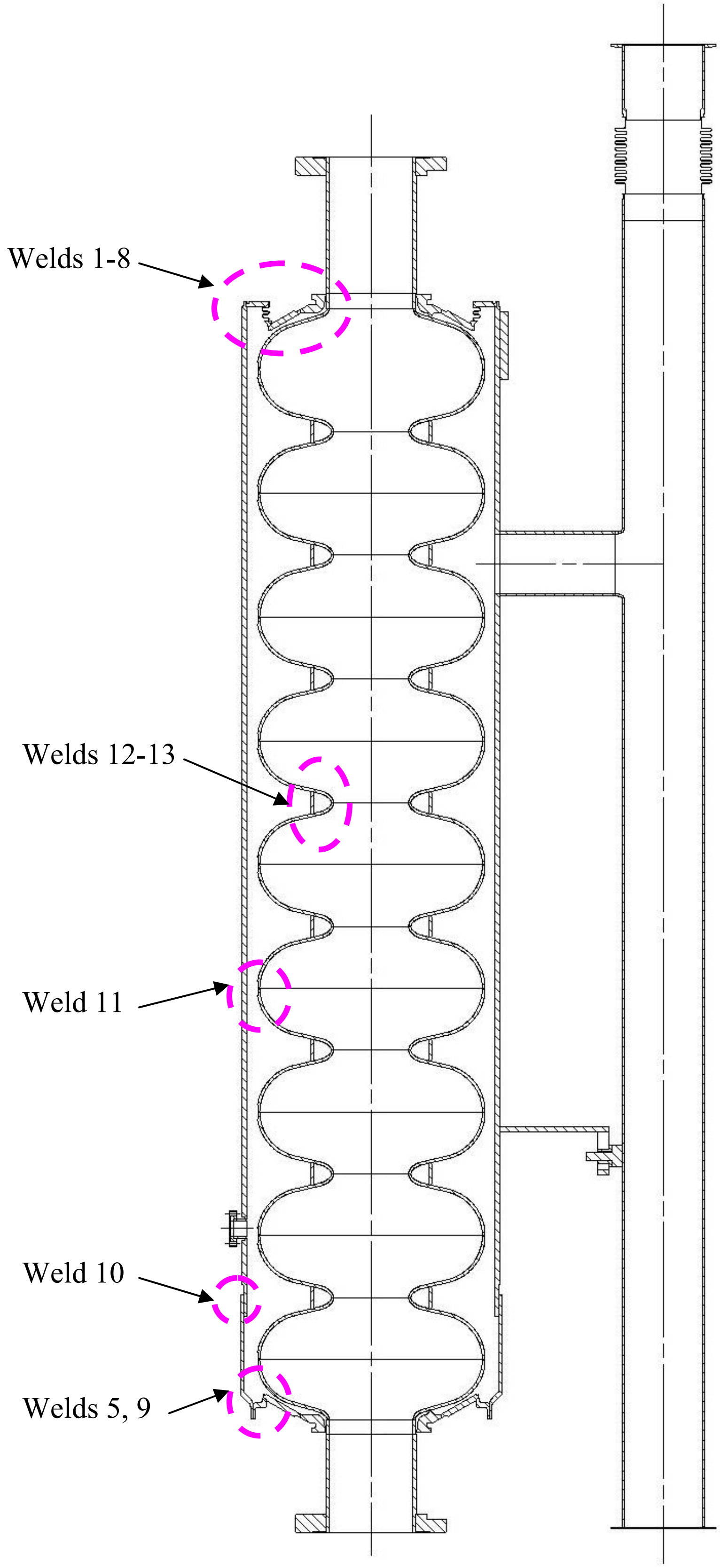


Figure 9 – Welds Numbered as in Table 3

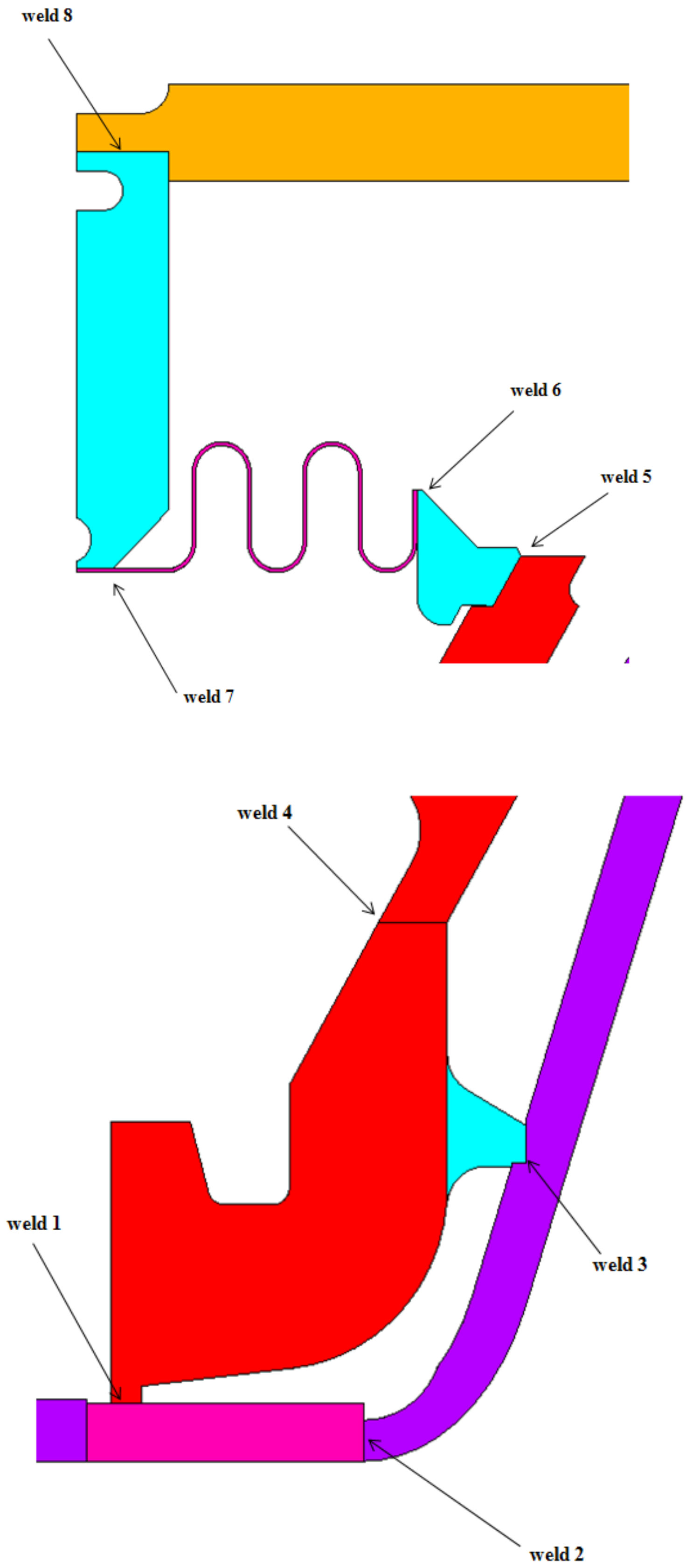


Figure 10 - Location of Welds 1-8

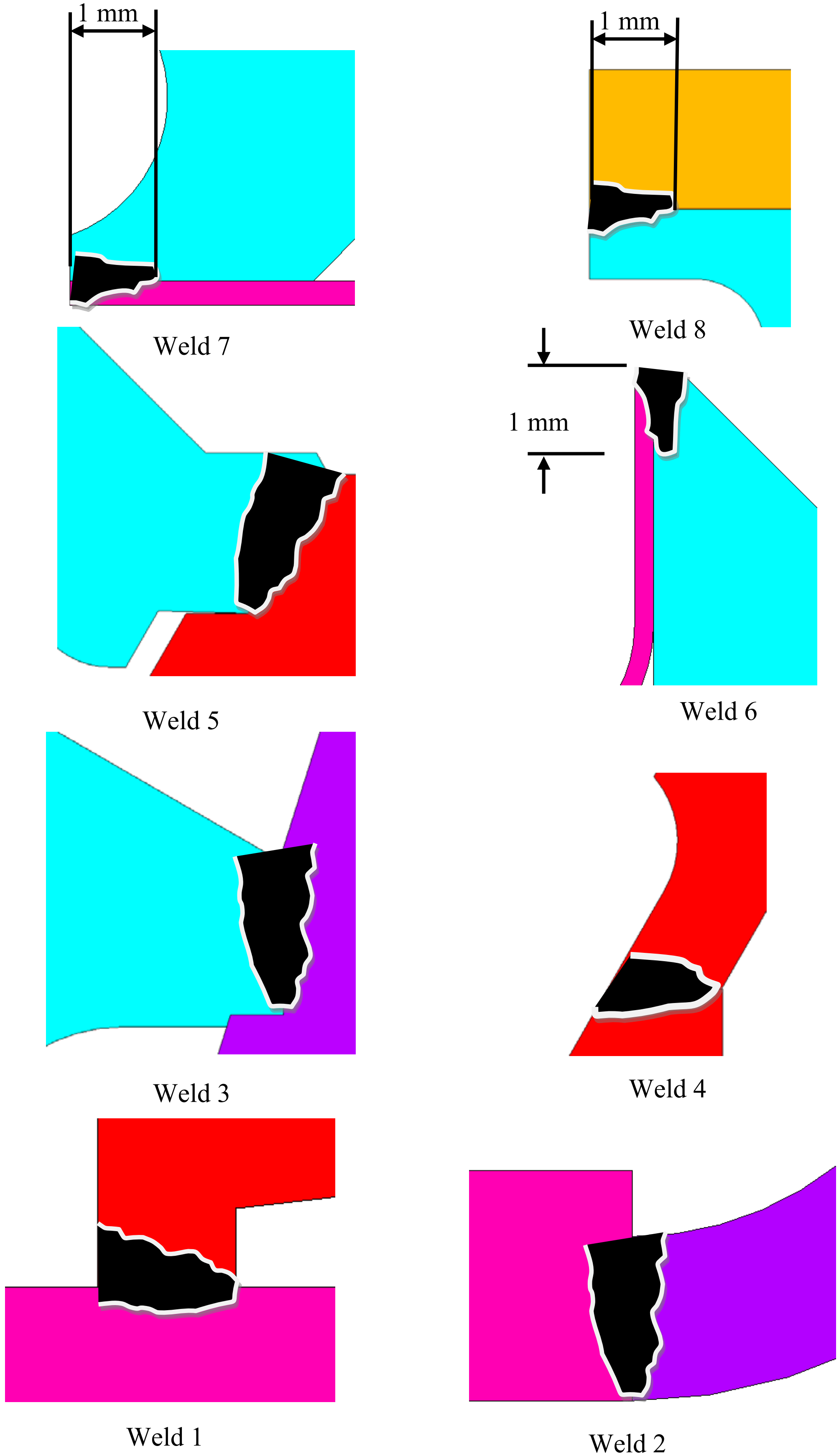


Figure 11 - Assumed fusion zones – welds 1 - 8

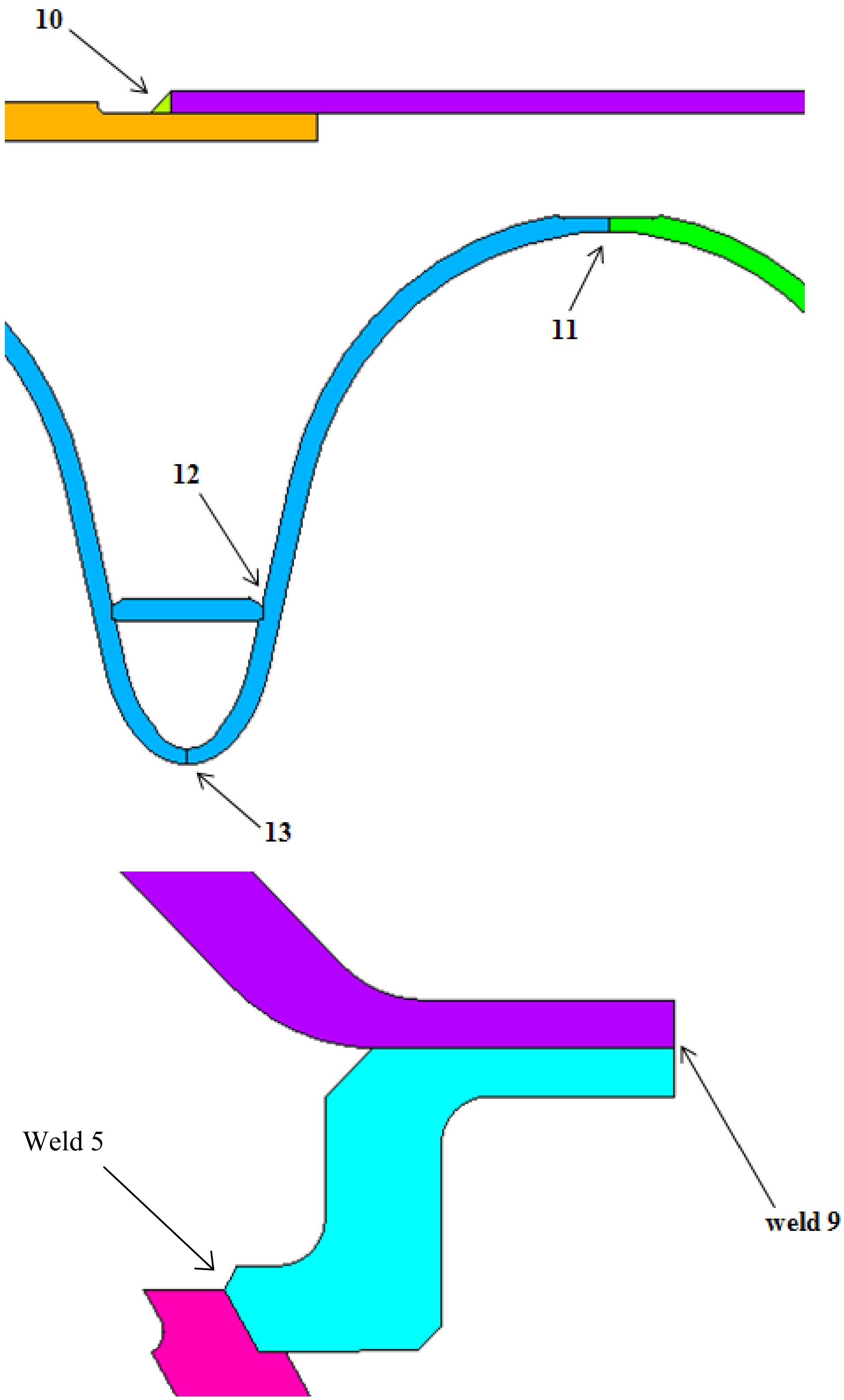
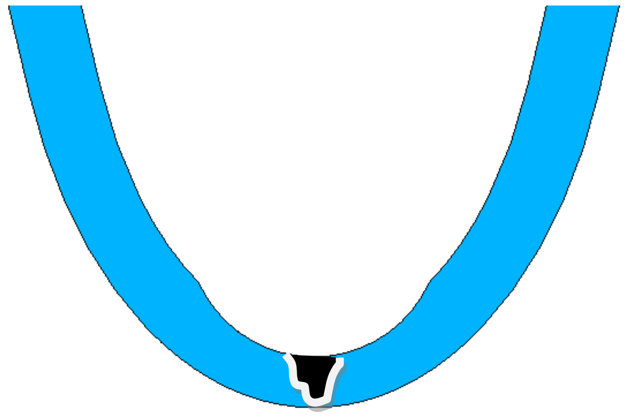
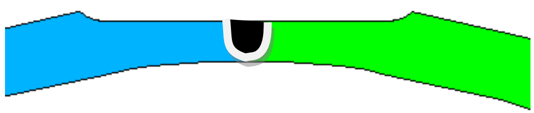


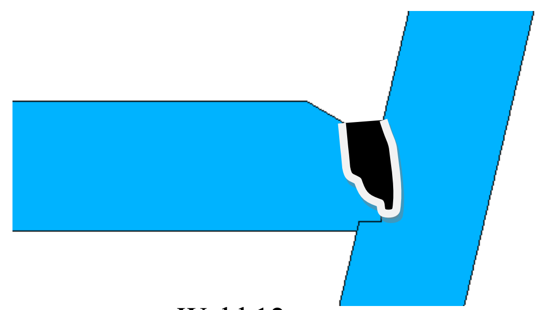
Figure 12 - Location of welds 9 - 13



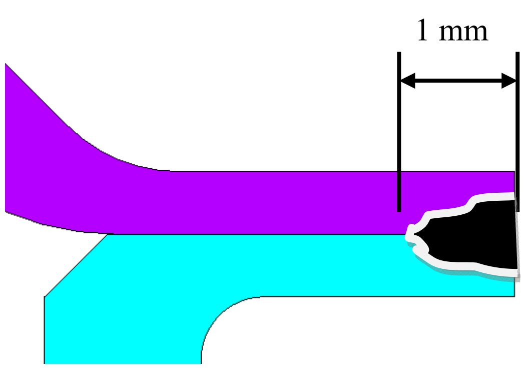
Weld 13



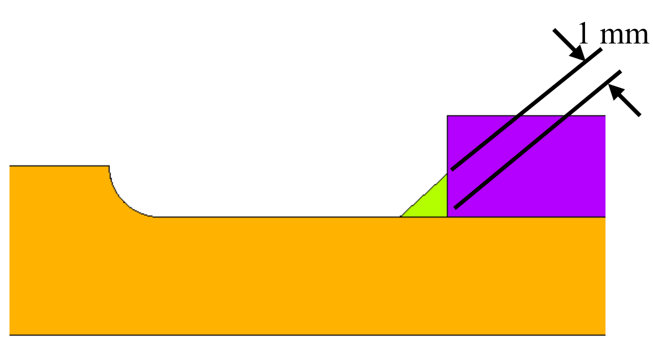
Weld 11



Weld 12



Weld 9



Weld 10

Figure 13 - Assumed fusion zones – welds 9 - 13

Material Properties

General

The dressed cavity is constructed of three materials: Pure niobium, Ti-45Nb alloy, and Grade 2 titanium. Of these materials, only Grade 2 Ti is approved by Div. 1 of the Code, and hence has properties and allowable stresses available from Section II, Part D.

The room temperature material properties and allowable stresses for this analysis are identical to those established in the analysis of the 3.9 GHz elliptical cavity(6). The determination of the allowable stresses was based on Code procedures, and employed a multiplier of 0.8 for additional conservatism.

For the cryogenic temperature load cases, advantage was taken of the increase in yield and ultimate stress for the Nb and Ti. As with the room temperature properties, the properties for these materials at cryogenic temperature were also established by previous work related to the 3.9 GHz cavity(7).

Room temperature properties were used for the Ti-45Nb alloy for all temperatures, as no low temperature data on that alloy were available. However, it is highly likely that, like the elemental Nb and Ti, substantial increases in strength occur.

Material Properties

The elastic modulus, yield strength, ultimate strength, and integrated thermal contraction from 293 K to 1.88 K are given in Table 4 for each material used in the construction of the cavity.

Table 4 – Material Properties

Material	Property					
	Elastic Modulus (psi)	Yield Strength (psi)		Ultimate Strength (psi)		Integrated Thermal Contraction 293K to 1.88K (in/in)
		293K	1.88 K	293K	1.88 K	
Niobium	**1.52E+07**	**5500**	**46000**	**16600**	**87000**	**0.0014**
55Ti-45Nb	**9.00E+06**	**69000**	**N/A**	**79000**	**N/A**	**0.0019**
Titanium, Gr. 2	**1.55E+07**	**40000**	**121000**	**50000**	**162000**	**0.0015**

Allowable Stresses

The Code-allowable stresses for unwelded materials for the various categories of stress (see "Stress Analysis Approach" of this report) are given in Table 5. The allowed stresses for each stress category are defined in the Code, Division 2, Paragraphs 5.2.2.4(e) and 5.5.6.1(d).

The Code-allowable stresses for welded materials are calculated by multiplying the values of Table 5 by the joint efficiency given in Table 3.

Table 5 – Allowable Stresses for Each Stress Category (Units in PSI)

	Stress Category							
	P_m		P_l		$P_l + P_b$		$P_l + P_b + Q$	
Material	**1.88K**	**293K**	**1.88K**	**293K**	**1.88K**	**293K**	**1.88K**	**293K**
Nb	19800	2900	29700	4350	29700	4350	59400	8700
Ti-45Nb	15300	15300	22950	22950	22950	22950	45900	45900
Gr. 2Ti	24500	9630	14520	36750	14520	36750	73500	29040

Note:
P_m **= primary membrane stress**
P_l **= primary local membrane stress**
P_b **= primary bending stress**
Q = secondary stress

The allowable stresses for each stress category in Table 5 are based on the value S, which is the allowable stress of the material at the design temperature. Paragraphs 5.2.2.4(e) and 5.5.6.1(d) define the stress categories:

$P_m \leq S$
$P_l \leq 1.5*S$
$(P_l + P_b) \leq 1.5*S$
$(P_l + P_b + Q) \leq 3*S$

Table 6 shows the values of S for each material at 1.88K and 293K. Note that S includes the de-rating factor of 0.8 of the established allowable stress for a material for an experimental vessel. The de-rating follows the guidelines in FESHM Chapter 5031.

Table 6 – Allowable Stress "S" (Units in MPa [PSI])

	Allowable Stress (S)		Established Values	
Material	**1.88°K**	**293°K**	**1.88°K**	**293°K**
Nb	137 [19870]	20 [2900]	171 [24801]	25 [3626]
Ti-45Nb	106 [15374]	106 [15374]	133 [19290]	133 [19290]
Gr. 2Ti	169 [24511]	66.4 [9630]	213 [30893]	83 [12038]

Loadings

General

The CM-1 cavity is shown in cross section in Fig. 14.

There are three volumes which may be pressurized or evacuated:

1. The LHe volume of the helium vessel
2. The volume outside the cavity typically evacuated for insulation
3. The volume through which the beam passes on the inside of the Nb cavity itself.

The pressures in these volumes are denoted as P_1, P_2, and P_3, respectively.

With regards to pressure, typical operation involves insulating vacuum, beam vacuum, and a pressurized LHe volume. Atypical operation may occur if the insulating or beam vacuums are spoiled, and the LHe space simultaneously evacuated. This reverses the normal operational stress state of the device, producing an external pressure on the Ti shell, and an internal pressure on the Nb cavity; however, this pressure is limited to a maximum differential of 15 psid.

In addition to the pressure loads, the cavity also sees dead weight forces due to gravity, as well thermal contractions when cooled to the operating temperature of 1.88 K, and a strain-controlled extension by the tuner after cool down.

All of these loadings are considered in this analysis. Specific load cases are defined in the next section.

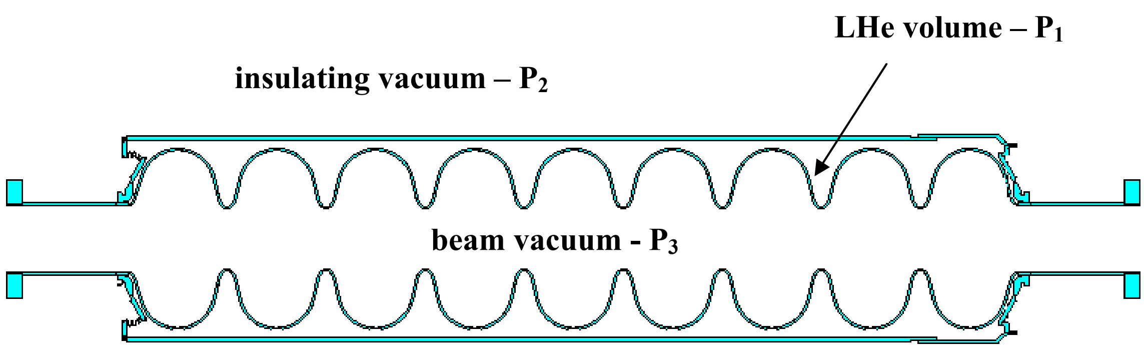


Figure 14 - Volumes for Pressure/Vacuum

Load Cases

The cavity is subjected to five basic loads:

1. Gravity
2. LHe liquid head
3. Thermal contraction
4. Tuner extension
5. Pressure (internal and external)

Three of these loads – gravity, liquid head, and pressure – produce both primary and secondary stresses. The remaining loads – thermal contraction and tuner extension – are displacement-controlled loads which produce secondary stresses only. This results in five load cases. These load cases are shown in Table 7, along with the temperatures at which the resulting stresses were assessed, and the stress categories that were applied.

Table 7 – Load Case Descriptions

Load Case	Loads	Condition Simulated	Temperature for Stress Assessment	Applicable Stress Categories
1	1. Gravity 2. P_1=30 psi 3. P_2=P_3 = 0	Warm Pressurization	293 K	P_m, P_l , P_l + Q
2	1. Gravity 2. LHe liquid head 3. P_1=60 psi 4. P_2=P_3 = 0	Cold operation, full, maximum pressure – no thermal contraction	1.88 K	P_m, P_l , P_l + Q
3	1. Cool down to 1.88 K 2. Tuner extension of 0.083 in	Cool down and tuner extension, no primary loads	1.88 K	Q
4	1. Gravity 2. LHe liquid head 3. Cool down to 1.88 K 4. Tuner extension 5. P_1=60 psi 6. P_2=P_3 = 0	Cold operation, full LHe inventory, maximum pressure – primary and secondary loads	1.88 K	Q
5	1. Gravity 2. P_1 = 0 3. P_2 = P_3 = 15 psi	Insulating and beam vacuum upset, helium volume evacuated	293 K	P_m, P_l , P_l + Q

Stress Analysis Approach

The goal of the analysis is to qualify the vessel to the greatest extent possible in accordance with the rules of the Code, Section VIII, Div. 1. This Division of the Code provides rules covering many cases; however, there are features of this cavity and its loadings for which the Division has no rules. This does not mean that the vessel cannot be qualified by Div. 1, since Div. 1 explicitly acknowledges the fact that it does not prevent formulaic procedures ("rules") covering all design possibilities. From U-2(g)

"This Division of Section VIII does not contain rules to cover all details of design and construction. Where complete details are not given, it is intended that the Manufacturer, subject to the acceptance of the Inspector, shall provide details of design and construction which will be as safe as those provided by the rules of this Division."

Applying Division I Rules to the Cavity

Division 1 rules relate to both geometries and loads. For either, there are few rules applicable to the features of the cavity.

The only components of the cavity which can be designed for internal and external pressure by the rules of Div. 1 are the Ti shells and the Ti bellows. In the Ti shell, there are two penetrations for connection of externals for which the required reinforcement can also be determined by Code rules.

The conical heads have half-apex angles exceeding 30 degrees, and no knuckles; Div. 1, Appendix 1, 1-5(g) states that their geometry falls under U-2(g).

The Nb cavity itself resembles an expansion joint, but does not conform to the geometries covered in Div. 1, Appendix 26. Therefore, U-2(g) is again applied.

UG-22(h) states that "temperature gradients and differential thermal contractions" are to be considered in vessel design, but provides no rules to cover the cavity. In this analysis, all thermal contraction effects are addressed under U-2(g).

The cavity is also subjected to a controlled displacement loading from blade tuner. There are no rules in Div. 1 covering such a loading, so U-2(g) is applied.

The applicable Code rules for each component are summarized in Table 8.

Table 8 - Applicable Code, Div. 1 Rules for 1.3 GHz Cavity

Component	Loading		
	Internal/External Pressure	Thermal Contraction	Tuner Extension
Nb cavity	U-2(g)	U-2(g)	U-2(g)
Conical heads	U-2(g)	U-2(g)	U-2(g)
Ti shells	UG-27/UG-28	U-2(g)	U-2(g)
Ti bellows	Appendix 26	U-2(g)	U-2(g)

Applying U-2(g)

U-2(g) is satisfied in this analysis by the application of the design-by-analysis rules of the Code, Section VIII, Div. 2, Part 5.

These rules provide protection against plastic collapse, local failure, buckling, fatigue, and ratcheting. The specific sections of Part 5 applied here are:

1. Plastic collapse – satisfied by an elastic stress analysis performed according to 5.2.2.
2. Ratcheting - satisfied by an elastic stress analysis performed according to 5.5.6.1
3. Local failure – satisfied by an elastic stress analysis performed according to 5.3.2
4. Buckling – satisfied by a linear buckling analysis performed according to 5.4.1.2(a).
5. Fatigue assessment – the need for a fatigue analysis is assessed according to 5.5.2.3

In general, an elastic stress analysis begins by establishing stress classification lines (SCLs) through critical sections in the structures according to the procedures of Part 5, Annex 5A. The stresses along these lines are then calculated (in this case, by an FEA), and "linearized" to produce statically equivalent membrane stress and bending stress components. The allowable stress for each component depends on the category of the stress. This category (or classification) depends on the location of the SCL in the structure, and the origin of the load. Stresses near discontinuities have higher allowables to reflect their ability to redistribute small amounts of plasticity into surrounding elastic material. Stresses produced solely by strain-controlled loads (e.g., thermal contractions and blade tuner extension) are given higher allowables regardless of their location in the structure.

Allowable stresses are expressed in terms of multiples of S, which is the allowable general primary membrane stress. The values of S used in this analysis are given in Table 6.

Division 1 Calculations by Rule

Ti Cylindrical Shells

Thickness for Internal Pressure

The minimum thickness required for the Ti cylindrical shells under internal pressure can be calculated from UG-27(c)(1):

$$t = \frac{PR}{SE - 0.6P}$$

where: t = required thickness
P = pressure = 30 psi (warm), 60 psi (cold)
R = inside radius of shell = 4.53 in
E = efficiency of seam weld = 0.5
S = maximum allowable membrane stress = 9630 psi (warm), 24500 psi (cold)

Substituting, the minimum required thickness when warm and pressurized to 30 psi is 0.028 in. The minimum required thickness when cold and pressurized to 60 psi is 0.022 in. The actual minimum thickness of the shells is 0.13 in (3.3 mm). Therefore, the Ti cylindrical shells meet the minimum thickness requirements of UG-27 for internal pressure.

Thickness for External Pressure (Buckling)

The minimum thickness required for the Ti cylindrical shells under external pressure can be calculated from UG-28(c). This procedure uses charts found in the Code, Section II, Part D. These charts are based on the geometric and material characteristics of the vessel.

Let: L = 20 in
D_o = 9.32 in
t = 0.07 in

Then: L/D = 4
D_o/t = 133

From the Code, Section II, Part D, Subpart 3, Fig. G, the factor A is 0.0002. From Fig. NFT-2 (the material chart for Grade 2 Ti), the factor B is 1600.

The allowable pressure is then

$$P = \frac{4B}{3(D_o / t)}$$

Substituting give P = 16 psi. This is approximately equal to the 15 psi maximum external vessel for which the vessel must be qualified.

The actual minimum thickness of the Ti shell is 0.13 inches. This occurs near one end, and it is unlikely that the collapse is well predicted by this thickness, due to its short length, and proximity to the conical head, which will tend to stiffen the region. If we assume, however, that the entire shell is this thickness, and repeat the calculations above, the allowable external pressure is 71 psi.

If we assume the collapse is better predicted by the predominate thickness of 0.2 inches, then the factor A = 0.00085, the factor B = 6400, and the allowable external pressure is 181 psi.

In any case, the required minimum thickness of 0.07 inches is less than the actual minimum thickness anywhere on the Ti cylindrical shell. Therefore, the Ti shell satisfies the Code requirement for external pressure.

Penetrations

The Ti cylindrical shell contains two penetrations. The penetration to the 2-phase helium return pipe is 2.16 inches (55 mm) in diameter. The penetration for the bottom-fill line has a through diameter of 0.71 inches (18-mm).

From UG-36(c)(3):

"Openings in vessels not subject to rapid fluctuations in pressure do not require reinforcement other that inherent in the construction under the following conditions: welded, brazed, and flued connections meeting applicable rules and with a finished opening not larger than 3.5 in diameter – in vessel shells or heads with a required minimum thickness of 3/8 inch or less."

The minimum required thickness of the shell is largest for the case of 30 psi pressurization, warm. This thickness (calculated a previous section titled "Thickness for Internal Pressure") is 0.022 in. This is less than 3/8 in. Therefore, since the penetrations are smaller than 3.5 in. in diameter, no additional reinforcement is required for either penetration in the Ti shells.

Titanium Bellows

The design of metallic expansion joints (bellows) is addressed by Appendix 26 of the Code. The formulas permit calculation of internal and external pressure limits. In a bellows, the pressure may be limited not only by stress, but by squirm (internal pressure), and collapse (external pressure.)

The geometry of the Ti bellows is not precisely covered by Appendix 26; there is only one end which has a horizontal tangent; the other end is comprised of a vertically terminated convolution. The bellows is nonetheless modeled as though it consisted of two convolutions with horizontal tangents on each end. The FEA is assumed to qualify the flange region of the bellows.

The Appendix 26 parameters are:

D_m = mean diameter of bellows convolutions = 7.75 in
D_b = inside diameter of bellows convolutions end tangent = 7.458 in
C_p = coefficient from Fig. 26.4 = 0.73
n = number of plies = 1
t = thickness of one ply = 0.00787 in
t_c = thickness of collar = 0 in
t_p = thickness of ply corrected for thinning during forming = 0.00757 in
L_t = length of bellows tangent = 0.2 in
L_c = length of bellows collar = 0 in
L = total bellows length = 0.4684 in
w = convolution height = 0.2848 in
A = cross sectional metal area of one convolution = 0.005535 in^2
q = convolution pitch = 0.2342 in
υ_b = Poisson's ratio = 0.35
I_{xx} = moment of inertia of one convolution = 4.79e-5 in^4
S = allowable stress (warm) = 9630 psi

Internal Pressure

For end convolutions, the circumferential membrane stress due to internal pressure shall comply with

$$S_{2,E} = \frac{1}{2}\frac{(qD_m + L_t(D_b + nt))}{(A + nt_p + t_cL_c)}P \le S$$

For intermediate convolutions, the circumferential membrane stress due to internal pressure shall comply with

$$S_{2,I} = \frac{1}{2}\frac{qD_m}{A}P \le S$$

Assuming P = 30 psi, substituting gives

$$S_{2,E} = 7040 \text{ psi} \le 9680 \text{ psi}$$

and

$$S_{2,I} = 4900 \text{ psi} \leq 9680 \text{ psi}$$

The bellows is formed from a rolled tube with a longitudinal weld. Based on Fermilab's experience in fabricating bellows from thin walled rolled tubes, it is assumed the weld is a Type 1 butt weld, where "the same quality of deposited weld metal on the inside and outside weld surfaces" have complete penetration and full fusion (Table UW-12 in the Code). () All welds in a bellows assembly must be examined using liquid penetrant. Since no information is available regarding examination of the weld and the liquid penetrant examination can be equivalent to radiography of a Type 1 weld, a weld joint efficiency of 0.7 is factored into the allowable stress for the circumferential stresses in the convolutions (Table UW-12). Therefore:

$$0.7 * S = 6776\, psi$$

$$S_{2,I} = 7040\, psi > 0.7 * S$$

$$S_{2,E} = 4900\, psi < 0.7 * S$$

The circumferential membrane stress in the intermediate convolutions is greater than the de-rated allowable stress. However, it is noted that the membrane stress is still less than the allowable stress with a weld joint efficiency of 1.0. Also, it is noted that the bellows design has been used extensively at DESY for over a decade. So the bellows is considered safe.

For meridional membrane stress, S_3, where

$$S_3 = \frac{w}{2nt_p} P = 564\, psi$$

and meridional bending stress, S_4, where

$$S_4 = \frac{1}{2n}\left(\frac{w}{t_p}\right)^2 C_p P = 15499\, psi$$

the sum $S_3 + S_4 = 16063$ psi must be less than $K_f S$, where K_f = bellows forming factor = 3 (for as-formed bellows). Substituting, $K_f S = 3S = 29040$ psi, and the criterion is satisfied.

The external pressure requirements as relate to stress are therefore satisfied at 293 K. At 1.88 K, the pressure increases by a factor of 2, but the allowable stress increases by a factor of 24500/9680, or 2.5. Therefore, the pressure requirements are also satisfied at 1.88 K.

External Pressure – Stability

For external pressure, the procedures for cylindrical shells, given in UG-28 of Div. 1 are used with an equivalent diameter defined as

$$D_{eq} = D_b + w + 2e_{eq}$$

and an equivalent thickness defined as

$$e_{eq} = (12(1 - \upsilon_b{}^2)\frac{I_{xx}}{q})^{\frac{1}{3}}$$

Substituting gives D_{eq} = 8.00 in, and e_{eq} = 0.129 in.

For use with the Section II, Part D procedures required by UG-28:

D/t = 61.9
L/D = 0.05854

The required factor A is found from Fig. A of Section II, Part D. The combination of D/t and L/D used here does not lie on this chart, so the maximum factor A = 0.1 is used. This is conservative, as the actual A is somewhat higher.

Using this factor with the relevant Section II material chart for Grade 2 titanium (Fig. NFT-2) gives the factor B = 20,000. From UG-27(c)(1),

$$P_a = \frac{4B}{3D/t} = 430\, psi$$

The required maximum allowable external working pressure is 60 psi. Therefore, the bellows meets the necessary external pressure requirements.

Internal Pressure – Stability

The bellows will experience a maximum internal pressure of 15 psid, occurring in Load Case 5. There are two failure modes which must be assessed: Column stability, and in-plane stability.

For column stability: $P \le P_{sc}$, where

$$P_{sc} = 0.34\frac{\pi K_b}{Nq}$$

where K_b = bellows axial stiffness = 1740 lbs/in (from FEA of isolated bellows)

Substituting give P_{sc} = 3968 psi. This is much larger than the maximum internal pressure of 15 psid.

For in-plane stability: $P \le P_{si}$, where

$$P_{si} = (\pi - 2)\frac{AS^*}{D_m q\sqrt{\alpha}}$$

where $= 1 + 2\delta^2 + \sqrt{1 - 2\delta^2 + 4\delta^2}$, and $\delta = \frac{1}{3}\frac{S_4}{S_{2,I}}$, and S^* = yield stress of bellows material at temperature = 40000 psi. Substituting gives P_{si} = 80.4 psi. This is much greater than the maximum internal pressure of 15 psid.

Fatigue Analysis for Titanium Bellows

The equations in the Code for fatigue analysis of a bellows are not valid for titanium. The manufacturer of the titanium bellows for the helium vessel provided design calculations following the Standards of the Expansion Joint Manufacturers Association [(7)]. The allowable fatigue life is calculated with the equation

$$N_C = \left(\frac{c}{S_T - b} \right)^a$$

The parameters a, b, and c are material and manufacturing constants. While the actual data from the manufacturer is not available, data is available for a titanium bellows that is made of the same material.[(14)] The manufacturer uses the same material and manufacturing constants as what EJMA uses for austenitic stainless steel. The calculated an allowable number of cycles to be N_C = 30521.

The slow tuner system is expected to extend the bellows a maximum length of 1.6-mm after each cooldown of the vessel. [(8)] The extension compensates for the thermal contraction and brings the SRF cavity back to its desired resonance frequency. The bellows extension will occur 200 times over the lifetime of the vessel. This is far less than the allowable number of cycles, so the bellows is designed well within the limits of fatigue failure.

The detailed calculations using EJMA guidelines are shown here:

Bellows Description:	Bellows for titanium helium vessel for 1.3-GHz dressed cavity CM1		
Prepared By:	M. Wong		
Date:	3/24/2010		
Design Basis:	Expansion Joint Manufacturers Association Standard, 7th Edition, ERRATA 2002		
Allowable Stress Basis:	ASME Section II, Part D, 2007 Edition		
Bellows Geometry		***Design Parameters***	
Bellows Inside Diameter, Db, (in.)	**7.46**	Design Pressure, P, (psi)	**43.3**
Number of Plies, n	**1**	Axial Extension, (in.)	**0.062**
Ply Thickness, t, (in.)	**0.00787**	Axial Precompression, (in.)	**0.000**
Free length, Lb, (in.)	**0.5**	Lateral Deflection, y, (in.)	**0.000**
Number of Convolutions, N	**2**	Minimum Fatigue Cycles	**200**
Depth of Convolution, w, (in.)	**0.146**	***Collar Geometry***	
Bellows Tangent Length, Lt, (in.)	**0.200**	Collar Thickness, tc, (in.)	**0.000**
Bellows Material	**Ti Gr2**	Collar Length, Lc, (in.)	**0.000**
Allowable Stress, Sab, (psi)	**9,630**	Collar Modulus of Elasticity, Ec, (psi)	**1.55E+07**
Modulus of Elasticity, Eb, (psi)	**1.55E+07**	Allowable Stress, Sac, (psi)	**9,630**
Intermediate Calculations			
Convolution Pitch, q, (in.)	0.234	Stiffening Factor, k	0.6
Bellows Mean Diameter, Dm, (in.)	7.612	Material Constant, c	**1.89E+06**
Bellows Outside Diameter, (in.)	7.766	Material Constant, b	**5.40E+04**
Collar Mean Diameter, Dc, (in.)	0.000	Manufacturing Constant, a	**3.4**
Total Axial Movement, (in.)	0.062	Factor from Figure C24, Cp	0.5745
Axial Movement Per Convolution, ex	0.0310	Factor from Figure C25, Cf	2.0949
Lateral Movement Per Convolution, ey	0.0000	Factor from Figure C26, Cd	2.1071
Bellows Mat. Thickness Factor, tp	0.0078	Material Strength Factor, Cm	**3.0**
Circumferential Stress Factor, Kr	1.133	Transition Point Factor, Cz	0.0000
X-Sect. Area for 1 Conv., Ac, (in.^2)	0.0033	Inplane Instability Stress Ratio, delta	0.1251
Yield Strength at Design Temp., Sy	40,000	Inplane Interaction Factor, alpha	2.0160

Fatigue Characteristics		*Minimum*	
Total Stress Range for All Movements, St, (psi)	**144,670**	*N/A*	
Fatigue Life (cycles to failure), Nc	**30521**	200	Pass
			Rev. 2, 8/12/02

Finite Element Model

A 3-d finite element half model was created in ANSYS. Elements were 10-node 41etrahedral, and 20-node hexahedra. Material behavior was linear elastic.

The cavity is supported against gravity by four support blocks. Two of these blocks at one end are mounted on low-friction surfaces which allow them to expand and contract with temperature changes without inducing stresses on the Ti shell.

The CM-1 is tuned by applying force to the end disk flange on the bellows end of the device.

The finite element model is shown in Fig. 15. Fig. 16 shows mesh details at various locations within the model.

The complete model was used to demonstrate satisfaction of the plastic collapse, ratcheting, and local failure criteria. Subsets of the model were also used to address the linear buckling of the Nb cavity and conical head.

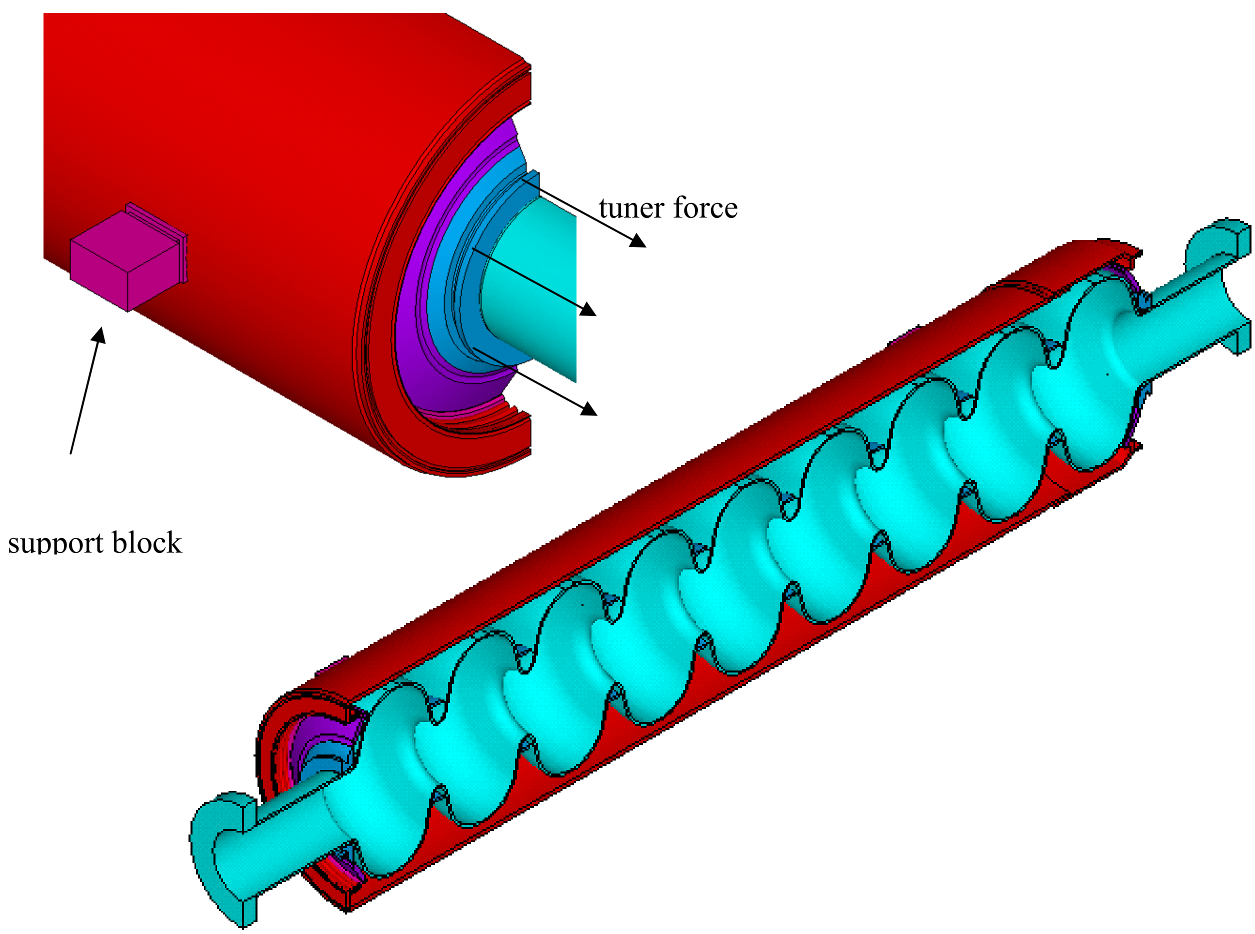


Figure 15 - The Finite Element Model

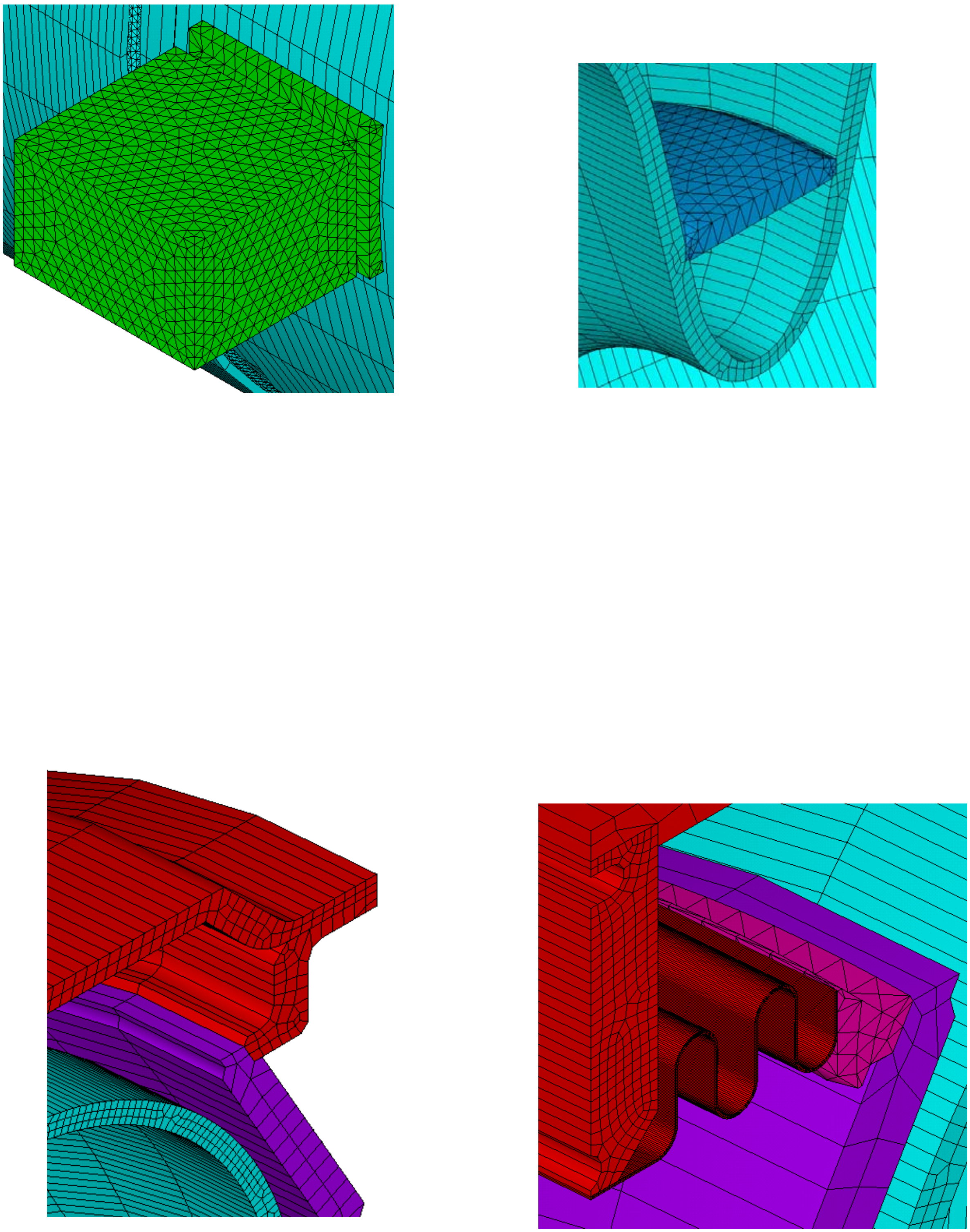


Figure 16 - Mesh Details

Stress Analysis Results

General

The complete finite element model was run for the five load cases. Stress classification lines (SCLs), shown in Fig. 17, were established through the critical sections of the structure. The stresses along these lines were linearized with ANSYS, and separated into membrane and bending components. The linearized stresses (expressed in terms of von Mises equivalent stress, as required by 5.2.2.1(b)) are categorized according to the Code, Div. 2, Part 5, 5.2.2.2 into primary and secondary stresses.

The primary and secondary stresses along each SCL for each of the five load cases are given in Tables 9-13.

The stresses from Tables 9-13 are used to demonstrate satisfaction of two of the criteria listed in 5.2 of this report: Protection against plastic collapse, and protection against ratcheting. Demonstrating protection against local failure employs the complete model, but requires the extraction of different quantities.

Note: The required minimum thicknesses of the Ti shells for internal and external pressure and the stresses in the titanium bellows are calculated by Div. 1 rules in an earlier section of this report. Therefore, no SCLs addressing the Ti shell thickness or the bellows design far from welds or other discontinuities are established here.

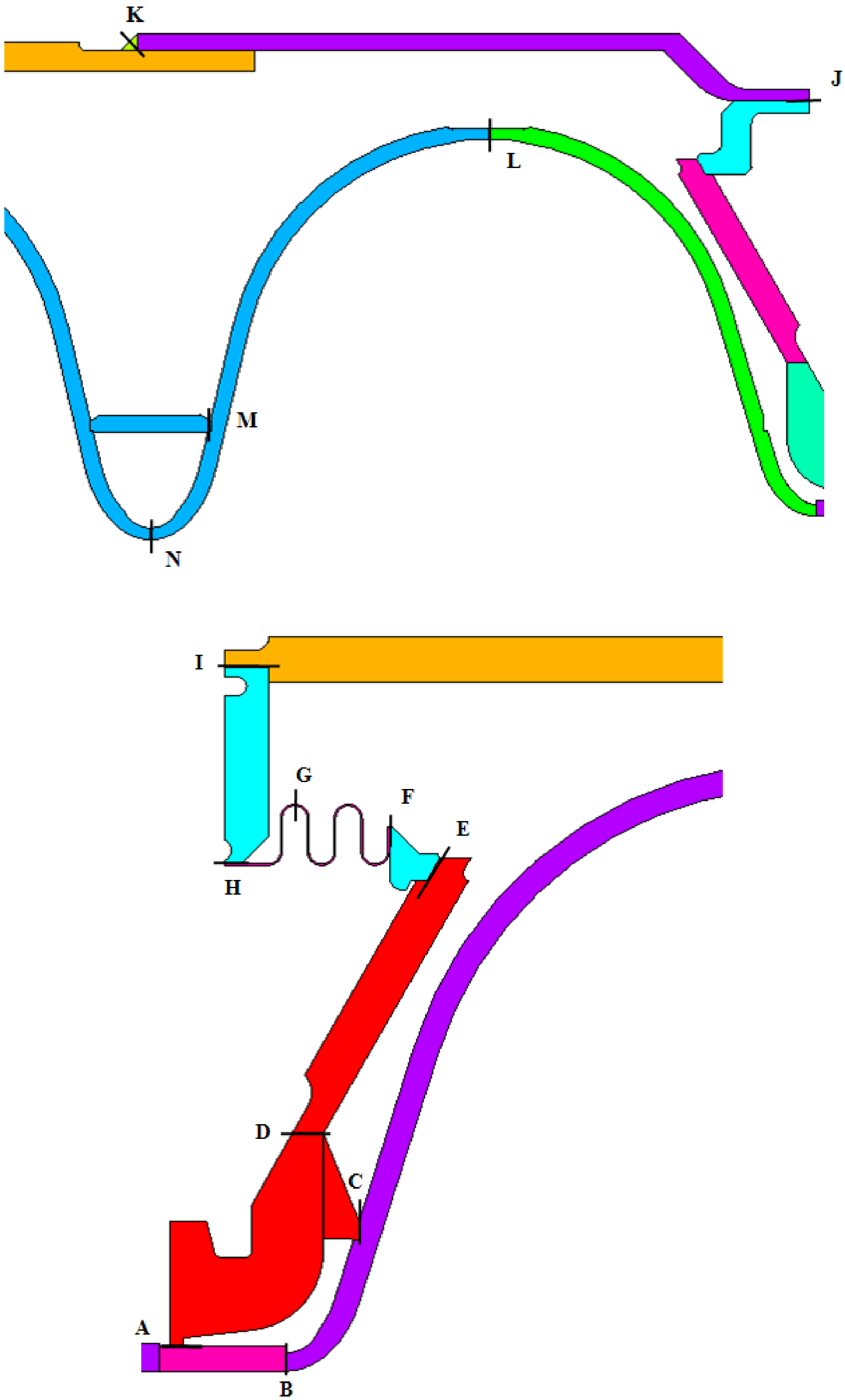


Figure 17 - Stress classification lines (SCLs)

Table 9 – Load Case 1 – Stress Results

Material	SCL	Weld #	Membrane Stress (psi)	Classification	Allowable (psi)	Ratio
Nb weld	**A**	**1**	302	Pm	1450	0.21
Nb weld	**B**	**2**	413	Pm	2175	0.19
Nb weld	**C**	**3**	1344	Pm	1450	0.93
Nb weld to Ti-Nb	**D**	**4**	700	Pm	1450	0.48
Ti-Nb weld to Ti	**E**	**5**	1726	Pm	4840	0.36
Ti weld	**F**	**6**	719	Pl	7260	0.10
Ti	**G**		5674	Pm	9680	0.59
Ti weld	**H**	**7**	2134	Pl	4840	0.44
Ti weld	**I**	**8**	6683	Pl	7260	0.92
Ti weld	**J**	**9**	3445	Pl	7260	0.47
Ti weld	**K**	**10**	1595	Pm	4840	0.33
Nb weld	**L**	**11**	510	Pm	1450	0.35
Nb weld	**M**	**12**	1313	Pm	1450	0.91
Nb weld	**N**	**13**	676	Pm	1450	0.47
Material	**SCL**	**Weld #**	**Membrane + Bending Stress (psi)**	**Classification**	**Allowable (psi)**	**Ratio**
Nb weld	**A**	**1**	336	Pm + Pb	2175	0.15
Nb weld	**B**	**2**	495	Pm + Pb	2175	0.15
Nb weld	**C**	**3**	3892	Q	4350	0.89
Nb weld to Ti-Nb	**D**	**4**	2073	Pm + Pb	2175	0.95
Ti-Nb weld to Ti	**E**	**5**	1901	Pm + Pb	7260	0.26
Ti weld	**F**	**6**	1358	Pl + Q	14520	0.09
Ti	**G**		10113	Pm + Pb	14520	0.70
Ti weld	**H**	**7**	2894	Pl + Q	14520	0.20
Ti weld	**I**	**8**	7715	Pl + Q	14520	0.53
Ti weld	**J**	**9**	3810	Pl + Q	14520	0.26
Ti weld	**K**	**10**	1829	Pm + Pb	7260	0.25
Nb weld	**L**	**11**	634	Pm + Pb	2175	0.29
Nb weld	**M**	**12**	2871	Q	4350	0.66
Nb weld	**N**	**13**	705	Pm + Pb	2175	0.32

Table 10 – Load Case 2 – Stress Results

Material	SCL	Weld #	Membrane Stress (psi)	Classification	Allowable (psi)	Ratio
Nb weld	**A**	**1**	564	Pm	9900	0.06
Nb weld	**B**	**2**	878	Pm	9900	0.09
Nb weld	**C**	**3**	2764	Pm	9900	0.28
Nb weld to Ti-Nb	**D**	**4**	1408	Pm	7650	0.18
Ti-Nb weld to Ti	**E**	**5**	3487	Pm	7650	0.46
Ti weld	**F**	**6**	1440	Pl	18375	0.08
Ti	**G**		11125	Pm	24500	0.45
Ti weld	**H**	**7**	4302	Pl	18375	0.23
Ti weld	**I**	**8**	13458	Pl	18375	0.73
Ti weld	**J**	**9**	7065	Pl	18375	0.38
Ti weld	**K**	**10**	3229	Pm	12250	0.26
Nb weld	**L**	**11**	1015	Pm	9900	0.10
Nb weld	**M**	**12**	2382	Pm	9900	0.24
Nb weld	**N**	**13**	1339	Pm	9900	0.14
Material	**SCL**	**Weld #**	**Membrane + Bending Stress (psi)**	**Classification**	**Allowable (psi)**	**Ratio**
Nb weld	**A**	**1**	686	Pm + Pb	14850	0.05
Nb weld	**B**	**2**	1004	Pm + Pb	14850	0.07
Nb weld	**C**	**3**	7193	Q	29700	0.24
Nb weld to Ti-Nb	**D**	**4**	4192	Pm + Pb	14850	0.28
Ti-Nb weld to Ti	**E**	**5**	3839	Pm + Pb	11475	0.33
Ti weld	**F**	**6**	2984	Pl + Q	36750	0.08
Ti	**G**		20811	Pm + Pb	36750	0.57
Ti weld	**H**	**7**	5845	Pl + Q	36750	0.16
Ti weld	**I**	**8**	15535	Pl + Q	36750	0.42
Ti weld	**J**	**9**	7816	Pl + Q	36750	0.21
Ti weld	**K**	**10**	3689	Pm + Pb	18375	0.20
Nb weld	**L**	**11**	1301	Pm + Pb	14850	0.09
Nb weld	**M**	**12**	5256	Q	29700	0.18
Nb weld	**N**	**13**	1396	Pm + Pb	14850	0.09

Table 11. Load Case 3 – Stress Results

Material	SCL	Weld #	Membrane Stress (psi)	Classification	Allowable (psi)	Ratio
Nb weld	**A**	**1**	397	N/A	N/A	N/A
Nb weld	**B**	**2**	634	N/A	N/A	N/A
Nb weld	**C**	**3**	4602	N/A	N/A	N/A
Nb weld to Ti-Nb	**D**	**4**	2357	N/A	N/A	N/A
Ti-Nb weld to Ti	**E**	**5**	5660	N/A	N/A	N/A
Ti weld	**F**	**6**	14404	N/A	N/A	N/A
Ti	**G**		10580	N/A	N/A	N/A
Ti weld	**H**	**7**	1758	N/A	N/A	N/A
Ti weld	**I**	**8**	3823	N/A	N/A	N/A
Ti weld	**J**	**9**	5333	N/A	N/A	N/A
Ti weld	**K**	**10**	1452	N/A	N/A	N/A
Nb weld	**L**	**11**	1831	N/A	N/A	N/A
Nb weld	**M**	**12**	12840	N/A	N/A	N/A
Nb weld	**N**	**13**	2768	N/A	N/A	N/A
Material	**SCL**	**Weld #**	**Membrane + Bending Stress (psi)**	**Classification**	**Allowable (psi)**	**Ratio**
Nb weld	**A**	**1**	1362	Q	29700	0.05
Nb weld	**B**	**2**	723	Q	29700	0.02
Nb weld	**C**	**3**	8552	Q	29700	0.29
Nb weld to Ti-Nb	**D**	**4**	6766	Q	22950	0.29
Ti-Nb weld to Ti	**E**	**5**	6940	Q	22950	0.30
Ti weld	**F**	**6**	28618	Q	36750	0.78
Ti	**G**		59354	Q	73500	0.81
Ti weld	**H**	**7**	3702	Q	36750	0.10
Ti weld	**I**	**8**	4460	Q	36750	0.12
Ti weld	**J**	**9**	5744	Q	36750	0.16
Ti weld	**K**	**10**	1587	Q	36750	0.04
Nb weld	**L**	**11**	2281	Q	29700	0.08
Nb weld	**M**	**12**	21415	Q	29700	0.72
Nb weld	**N**	**13**	4125	Q	29700	0.14

Table 12 – Load Case 4 – Stress Results

Material	SCL	Weld #	Membrane Stress (psi)	Classification	Allowable (psi)	Ratio
Nb weld	**A**	**1**	831	N/A	N/A	N/A
Nb weld	**B**	**2**	362	N/A	N/A	N/A
Nb weld	**C**	**3**	3414	N/A	N/A	N/A
Nb weld to Ti-Nb	**D**	**4**	2336	N/A	N/A	N/A
Ti-Nb weld to Ti	**E**	**5**	3574	N/A	N/A	N/A
Ti weld	**F**	**6**	13935	N/A	N/A	N/A
Ti	**G**		557	N/A	N/A	N/A
Ti weld	**H**	**7**	5826	N/A	N/A	N/A
Ti weld	**I**	**8**	17290	N/A	N/A	N/A
Ti weld	**J**	**9**	1838	N/A	N/A	N/A
Ti weld	**K**	**10**	2179	N/A	N/A	N/A
Nb weld	**L**	**11**	2123	N/A	N/A	N/A
Nb weld	**M**	**12**	14256	N/A	N/A	N/A
Nb weld	**N**	**13**	1455	N/A	N/A	N/A
Material	**SCL**	**Weld #**	**Membrane + Bending Stress (psi)**	**Classification**	**Allowable (psi)**	**Ratio**
Nb weld	**A**	**1**	1567	Q	29700	0.05
Nb weld	**B**	**2**	837	Q	29700	0.03
Nb weld	**C**	**3**	13008	Q	29700	0.44
Nb weld to Ti-Nb	**D**	**4**	3129	Q	22950	0.14
Ti-Nb weld to Ti	**E**	**5**	5247	Q	22950	0.23
Ti weld	**F**	**6**	27398	Q	36750	0.75
Ti	**G**		74419	Q	73500	1.01
Ti weld	**H**	**7**	7999	Q	36750	0.22
Ti weld	**I**	**8**	20005	Q	36750	0.54
Ti weld	**J**	**9**	2156	Q	36750	0.06
Ti weld	**K**	**10**	2459	Q	36750	0.07
Nb weld	**L**	**11**	2172	Q	29700	0.07
Nb weld	**M**	**12**	25713	Q	29700	0.87
Nb weld	**N**	**13**	3238	Q	29700	0.11

Table 13. Load Case 5 – Stress Results

Material	SCL	Weld #	Membrane Stress (psi)	Classification	Allowable (psi)	Ratio
Nb weld	**A**	**1**	134	Pm	1450	0.09
Nb weld	**B**	**2**	43	Pm	2175	0.02
Nb weld	**C**	**3**	266	Pm	1450	0.18
Nb weld to Ti-Nb	**D**	**4**	38	Pm	1450	0.03
Ti-Nb weld to Ti	**E**	**5**	157	Pm	4840	0.03
Ti weld	**F**	**6**	558	Pl	7260	0.08
Ti	**G**		186	Pm	9680	0.02
Ti weld	**H**	**7**	60	Pl	4840	0.01
Ti weld	**I**	**8**	257	Pl	7260	0.04
Ti weld	**J**	**9**	266	Pl	7260	0.04
Ti weld	**K**	**10**	44	Pm	4840	0.01
Nb weld	**L**	**11**	74	Pm	1450	0.05
Nb weld	**M**	**12**	505	Pm	1450	0.35
Nb weld	**N**	**13**	15	Pm	1450	0.01
Material	**SCL**	**Weld #**	**Membrane + Bending Stress (psi)**	**Classification**	**Allowable (psi)**	**Ratio**
Nb weld	**A**	**1**	144	Pm + Pb	2175	0.07
Nb weld	**B**	**2**	52	Pm + Pb	3262.5	0.02
Nb weld	**C**	**3**	815	Q	4350	0.19
Nb weld to Ti-Nb	**D**	**4**	142	Pm + Pb	2175	0.07
Ti-Nb weld to Ti	**E**	**5**	161	Pm + Pb	7260	0.02
Ti weld	**F**	**6**	1063	Pl + Q	14520	0.07
Ti	**G**		721	Pm + Pb	14520	0.05
Ti weld	**H**	**7**	96	Pl + Q	14520	0.01
Ti weld	**I**	**8**	319	Pl + Q	14520	0.02
Ti weld	**J**	**9**	319	Pl + Q	14520	0.02
Ti weld	**K**	**10**	55	Pm + Pb	7260	0.01
Nb weld	**L**	**11**	100	Pm + Pb	2175	0.05
Nb weld	**M**	**12**	807	Q	4350	0.19
Nb weld	**N**	**13**	16	Pm + Pb	2175	0.01

Plastic Collapse

The criterion for protection against plastic collapse is given in Div. 2, 5.2.2. The criterion is applied to load cases in which primary (load-controlled) stresses are produced. For this analysis, this is Load Case 1, Load Case 2, and Load Case 5.

The following stress limits must be met (per 5.2.2.4(e)):

1. P_m = primary membrane stress $\leq S$
2. P_l = primary local membrane stress $\leq 1.5\ S$
3. $P_l + P_b$ = primary local membrane + primary bending $\leq 1.5\ S$

where S = maximum allowable primary membrane stress.

In this work, the P_l classification is applied to SCLs F,H,I, and J (welds 6-9). This is justified by the discontinuities at which these welds are used. All other membrane stresses extracted on the SCLs are classified as the more conservative P_m, which is then used in place of P_l in 3) above.

The Nb end disk flange is intended to stiffen the Nb iris against axial motion only through the membrane stress in the weld, which means that any bending stresses are self-limiting and small rotations will satisfy the conditions that produce them. For this reason, the membrane stresses in this weld are classified as primary, while the bending stresses are secondary.

Examining Tables 9, 10, and 13, it is found that the closest approach to the limiting stress for any load case occurs at SCL C (weld #3, the weld between the end disk flange and the end cell of the Nb cavity) in Load Case 1, where the primary membrane stress of 1344 psi compares to an allowable of 1450 psi. This weld was also the limiting weld in the AES-004 analysis. (13) In this model, a better simulation of the end disk flange was used, resulting in somewhat lower calculated stresses, which permitted the calculated stress to not exceed the lower allowable assigned to weld materials in the CM-1.

For Load Case 2, the closest approach to the limiting stress occurs at SCL I (weld #8, the weld between the Ti transition piece and the Ti outer shell). The primary local membrane stress in this weld is 13458 psi, comparing to an allowable stress of 18375 psi.

For Load Case 5, all stresses are well below the allowable stresses at all locations.

Ratcheting

Protection against ratcheting, the progressive distortion of a component under repeated loadings, is provided by meeting the requirements of Div. 2, 5.5.6. Specifically, the following limit must be satisfied:

$$\Delta S_{n,k} \leq S_{PS}$$

where: $\Delta S_{n,k}$ = primary plus secondary equivalent stress range
S_{PS} = allowable limit on primary plus secondary stress range

The stress range $\Delta S_{n,k}$ must take into account stress reversals; however, there are no stress reversals in normal operation of the cavity, so for this analysis $\Delta S_{n,k}$ is equal to the primary plus secondary stresses given in Tables 9-13.

Examination of the tables shows that the cavity violates the ratcheting criterion; for Load Case 4 (gravity + liquid head + 60 psi + tuner extension + cool down), the calculated primary plus secondary stress range in the Ti bellows reaches 1.01 of the allowable. This is a very slight violation, and highlights one of the understood shortcomings of the CM-1 cavity design.

Local Failure

The criterion for protection against local failure is given in Div. 2, 5.3.2:

$$\sigma_1 + \sigma_2 + \sigma_3 \leq 4S$$

where σ_1, σ_2, σ_3 are the principal stresses at any point in the structure, and S is the maximum allowable primary membrane stress (see Table 5), multiplied by a joint efficiency factor if applicable.

This criterion is assumed to be satisfied if the sum of the principal stresses calculated at every element centroid in the model meets the stress limit for the material.

Table 14 lists the maximum allowable sum of principal stresses for each material at each load case. For Nb and Ti-Nb, the maximum sums occur at welds. Therefore, these values are four times the full values given for maximum primary membrane stress times a joint efficiency of 0.5. For Ti, the maximum sums occur in the bellows, away from the weld. Therefore, these values are four times the full value for the maximum primary membrane stress.

The results for each material and each load case are given in Tables 15-17. The closest approach to the allowable limit occurs in the iris support ring welds for Load Case 1 (warm, 30 psi internal pressure), which reaches 0.92 of the allowable. For all other materials/load cases, the principal stress sum lies well below the allowable.

Table 14 – Maximum Allowable Sum of Principal Stresses

Load Case (Temp)	**Maximum Allowable Sum of Principal Stresses (psi)**		
	Nb(weld)	**TiNb(weld)**	**Ti**
1 (293 K)	**5800**	**30600**	**38720**
2 (1.88 K)	**39600**	**30600**	**98000**
3 (1.88 K)	**39600**	**30600**	**98000**
4 (1.88 K)	**39600**	**30600**	**98000**
5 (293 K)	**5800**	**30600**	**38720**

Table 15 – Local Failure Criterion - Nb

Load Case	Maximum Principal Stress Sum (psi)	Allowable Stress (psi)	Location	Ratio Sfe/Sa
1	5347	5800	Weld #3	0.92
2	9354	39600	Weld #3	0.23
3	22163	39600	Weld #12	0.55
4	25457	39600	Weld #12	0.64
5	1466	5800	Weld #3	0.25

Table 16 – Local Failure Criterion – Ti-45Nb

Load Case	Maximum Principal Stress Sum (psi)	Allowable Stress (psi)	Location	Ratio Sfe/Sa
1	2589	30600	Weld #4	0.08
2	4871	30600	Weld #4	0.15
3	8179	30600	Weld #4	0.26
4	4767	30600	Weld #5	0.15
5	239	30600	Weld #4	0.01

Table 17 – Local Failure Criterion – Ti Grade 2

Load Case	Maximum Principal Stress Sum (psi)	Allowable Stress (psi)	Location	Ratio Sfe/Sa
1	14000	38720	SCL G	0.36
2	26490	98000	SCL G	0.27
3	73098	98000	SCL G	0.74
4	78204	98000	SCL G	0.79
5	1397	38720	SCL G	0.04

Buckling

Ti Shells and Bellows

The buckling of the Ti shells and bellows is addressed by Div. 1 rules in an section of this report.

The Nb Cavity

The Code, Div. 1, does not contain the necessary geometric and material information to perform a Div. 1 calculation of Nb cavity collapse. Therefore, the procedures of Div. 2, Part 5, 5.4 "Protection Against Collapse from Buckling" are applied.

A linear elastic buckling analysis was performed with ANSYS. A design factor was applied to the predicted collapse pressure to give the maximum allowable external working pressure. This design factor, taken from 5.4.1.3(c) for spherical shells, is 16.

Only the cavity was modeled. The ends are constrained in all degrees of freedom to simulate the effect of attachment to the conical heads and Ti shells of the helium vessel.

The predicted buckled shape is shown in Fig. 18. The critical pressure is 12450 psi. Applying the design factor gives this component a maximum allowable external working pressure of 778 psi, which is far greater than the required MAWP of 60 psi external.

Conical Heads

The buckling pressure of the conical heads was calculated by the linear buckling approach used for the Nb cavity. The results presented here are for the AES-004 cavity. [13] The relevant geometry is very similar between the two vessels, and the calculated buckling pressures are well above the required pressure of 15 psi; therefore, it was felt unnecessary to perform a separate analysis specific to the CM-1.

A model of the head only was made. It was constrained against axial motion where it connects to the Ti shell, but allowed to rotate freely, and translate radially.

The predicted buckling shape is shown in Fig. 19. The critical buckling pressure is 3880 psi. Applying the design factor of 2.5 (from 5.4.1.3(b) for conical shells under external pressure) gives an MAWP for external pressure of 1550 psi, which is well below the actual maximum pressure of 15 psi.

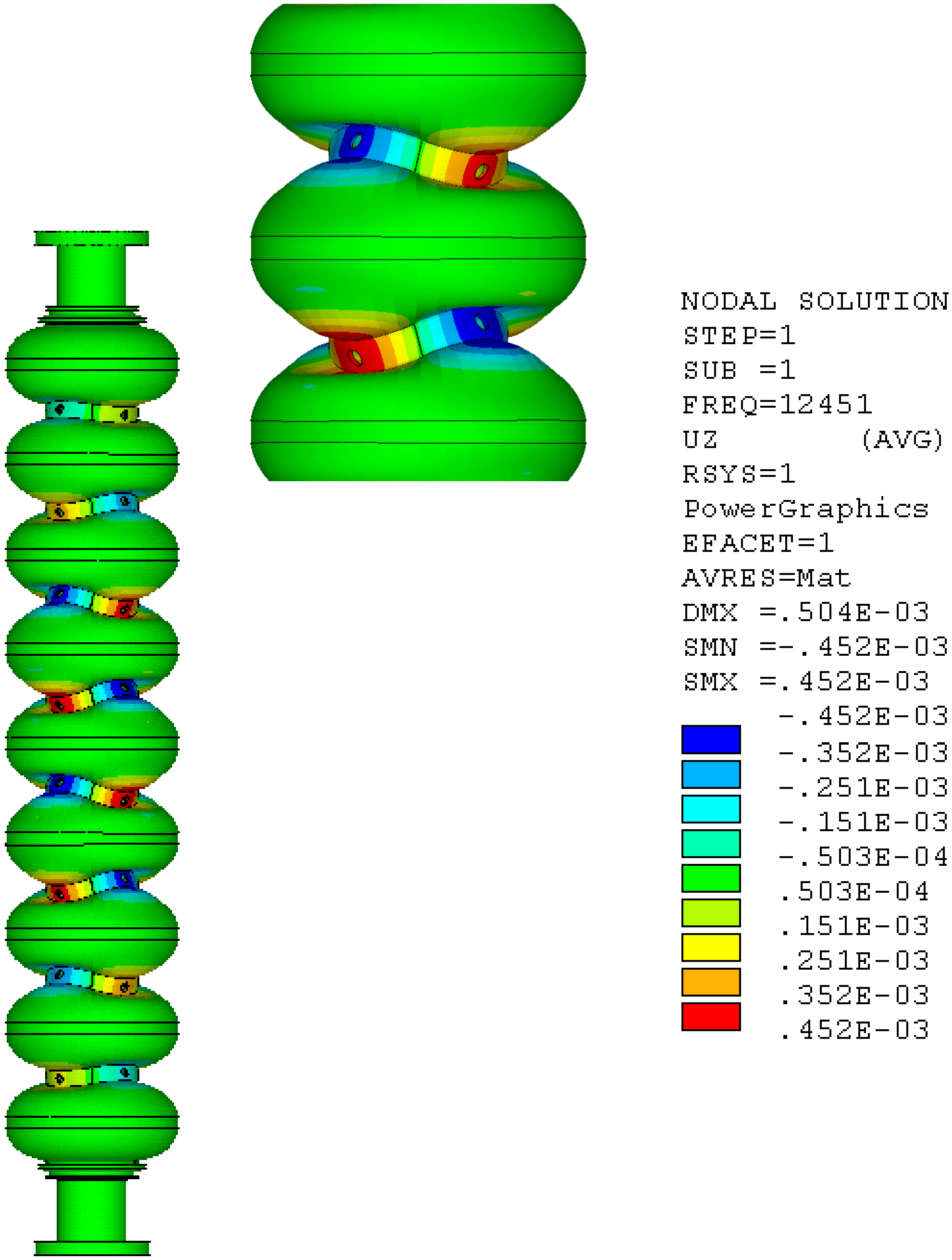


Figure 18 - Lowest buckling mode of Nb Cavity (P_{cr} = 12450 psi)

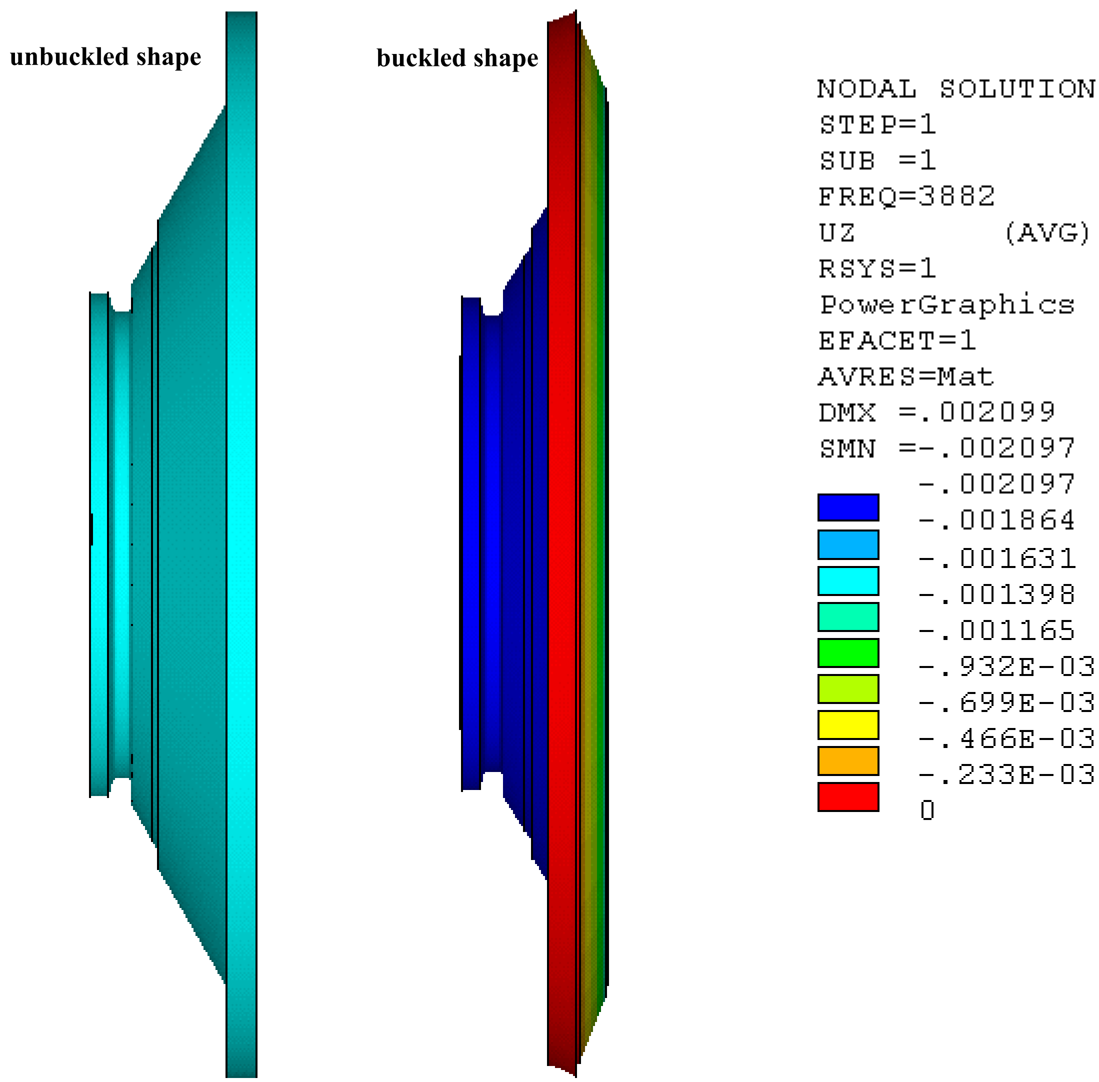


Figure 19 – Buckling of conical head

Fatigue Assessment

The need for a fatigue analysis can be determined by applying the fatigue assessment procedures of Div. 2, Part 5, 5.5.2.3, "Fatigue Analysis Screening, Method A."

In this procedure, a load history is established which determines the number of cycles of each loading experienced by the dressed cavity. These numbers are compared against criteria which determine whether a detailed fatigue analysis is necessary.

The load history consists of multiple cool down, pressurization, and tuning cycles. Estimates for the number of cycles of each load a cavity might experience are given in Table 18.

Table 18 – Estimated Load History of Dressed Cavity

Loading	Designation	Number of Cycles
Cool down	$N_{\Delta TE}$	**100**
Pressurization	$N_{\Delta FP}$	**200**
Tuning	$N_{\Delta tuner}$	**200**

The information of Table 18 is used with the criterion of Table 19 (a reproduction of Table 5.9 of Part 5) to determine whether a fatigue analysis is necessary.

The tuning load has no direct analog to the cycle definitions of Table 19. Therefore, it will be assigned its own definition as a cyclic load, $N_{\Delta tuner}$, and treated additively.

For the Nb cavity, construction is integral, and there are no attachments or nozzles in the knuckle regions of the heads. Therefore, the applicable criterion is

$$N_{\Delta TE} + N_{\Delta FP} + N_{\Delta tuner} \leq 1000$$

$$100 + 200 + 200 = 500 \leq 1000$$

The criterion is satisfied, and no fatigue assessment is necessary for the Nb cavity.

Table 19 – Reproduction of Table 5.9 of Part 5, "Fatigue Screening Criteria for Method A"

Description		
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq = 350$
	All other components that do not contain a flaw	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq = 1000$
Non-integral Construction	Attachments and nozzles in the knuckle region of formed head	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq = 60$
	All other components that do not contain a flaw	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq = 400$
$N_{\Delta FP}$ = expected number of full-range pressure cycles, including startup and shutdown $N_{\Delta PO}$ = expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction $N_{\Delta TE}$ = effective number of changes in metal temperature difference between any two adjacent points $N_{\Delta T\alpha}$ = number of temperature cycles for components involving welds between materials having different coefficients of thermal expansion that cause the value of $(\alpha_1 - \alpha_2)\Delta T$ to exceed 0.00034		

Welds Between Ti Support Blocks and Ti cylindrical Shells

The welds between the Ti support blocks and the Ti cylindrical are structural support welds. The Code, Div. 1, Nonmandatory Appendix G, "Suggested Good Practice Regarding Piping Reactions and Design of Supports and Attachments" was applied to their analysis. This appendix states that supports should conform to good structural practice. As a guide to this practice, the Manual of Steel Construction is suggested[9].

Unlike the AES-004, which supported its cavity from the blade tuner flange and thus saw loads associated with thermal contraction and tuner displacement, the supports in the dressed cavities of CM-1 see only the dead weight of the cavity and appurtenances

Fig. 20 shows the block and its welds to the shell. The welds are assumed to be a fillet weld with a 1 mm (0.03937 in) throat. Each weld (top and bottom) extends the length of the block parallel to the cavity axis a distance of 1.5 inches. The total shear stress area of one weld is therefore (0.03937)(1.5) = 0.059 in^2. For this analysis, the supporting force F is applied at its maximum distance from the weld.

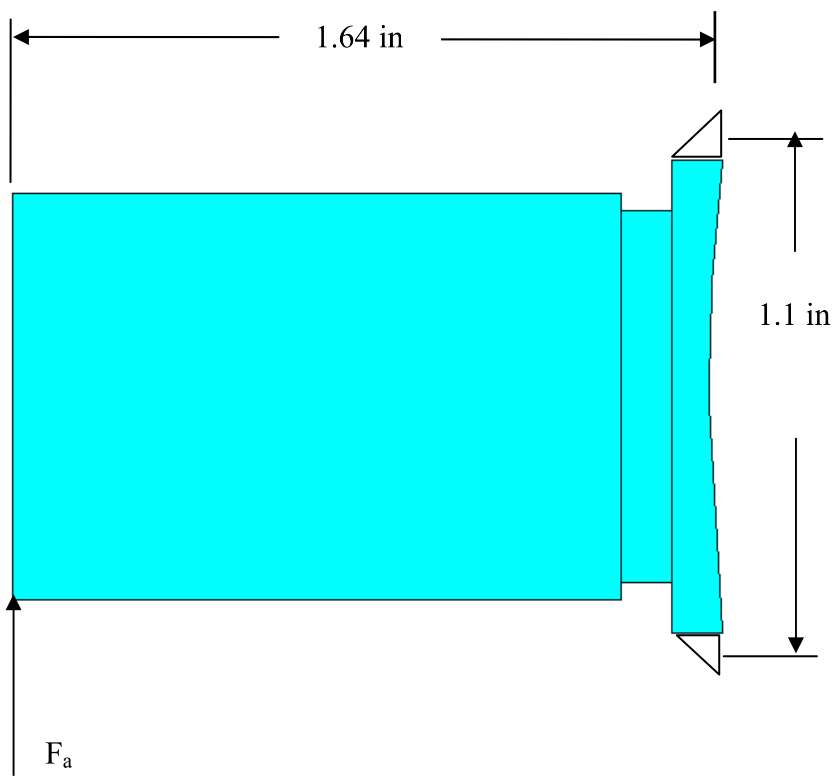


Figure 20 - Support block welds and applied force

The force on a single weld due to the moment produced by F_a is $F_{wm} = (1.64/1.1)F_a$. The force on a single weld due to pure shear from F_a is $F_{ws} = F_a/2$. Adding these forces algebraically gives a conservative estimate of total weld force, $F_{wt} = 2F_a$. If this force is applied to the weld throat, the resulting weld shear stress is then $S_w = 2F_a/0.059 = 34F_a$.

From Ref. 5, the shear stress on the throat of a fillet weld is limited to 0.3 of the ultimate strength of the weld. The shear stress on the base metal is limited to 0.4 of the yield stress of the base material. To be conservative in this analysis, the shear stress on the throat of the fillet weld will be limited to 1/2 of the primary membrane stress allowable for welded Ti. The primary membrane stress allowable for welded Ti is found by applying a weld efficiency of 0.5 to the values given in Table 5 of this report.

The maximum allowable shear stress on the throat of the fillet weld is then 9860(0.5)(0.5) = 2420 psi. Therefore, the maximum force that the support block can sustain is $F_a = 2420/34 = 71$ lbs.

From the FEA, the total weight of the cavity as modeled is 96 lbs. The most heavily loaded support sees a load of 26 lbs. Assuming that additional weight will be supported when the various shields and piping are attached, the present support system is capable of supporting a total cavity + appurtenances weight of 260 lbs. This is well in excess of what will actually be supported.

The stresses in the Ti shell from the support block reactions is shown in Fig. 21, for the case where the cavity weight has been artificially increased to 260 lbs. These stresses, a maximum of approximately 1900 psi, are well below the allowable stress of 4840 psi for welded Ti at room temperature.

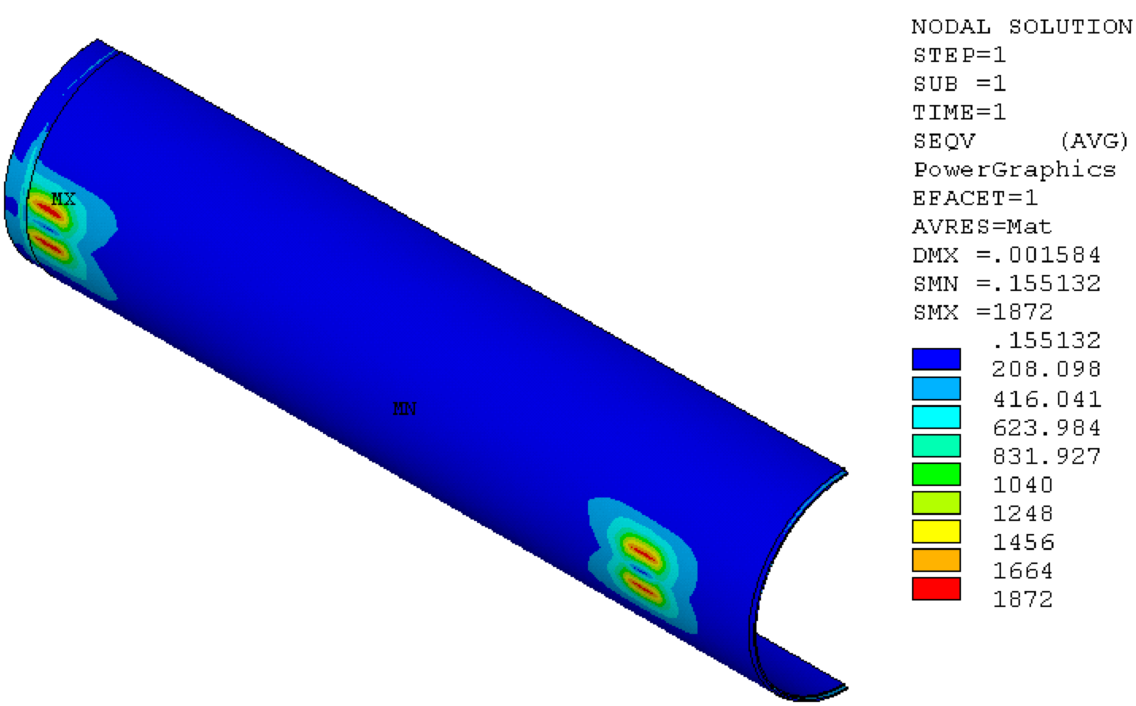


Figure 21 - Stresses in Ti shell from support block reactions

System Venting Verification

The venting system must protect the vessel against various sources of pressure. Figure 22 shows the schematic of the venting system at NML for CM1 (drawing 5520.320-ME-458097). There are two safety relief valves for venting helium from CM1 and are considered in the system venting calculations. Both are rupture disks, as detailed below:

- SV-803-H: Set pt. = 43 psig (4-bar), Leser Model 4414.4722, nominal size = 6"x8", 15,000-SCFM air
- SV-806-H: Set pt. = 15 psig (2-bar), Leser Model 4414.7942, nominal size = 2"x3", 1074-SCFM air

For this note, the required relief capacities are calculated for helium sources related only to the dressed cavity helium vessels. Table 20 summarizes the possible sources of helium pressure, the calculated maximum flow rate, and the capacity of the available relief valve. The design of the helium venting system follows the guidelines in CGA S-1.3-1995 (10), so the relief type is also shown. The available relief capacity is adequate for venting the possible sources of helium pressure.

Table 20 – Summary of Required and Available Relief Capacities for 8 Helium Vessels

Source of Helium Pressure	Required relief capacity	Available relief capacity	Relief Type (CGA-defined)
Room temperature helium supply from cryoplant	198 SCFM air (2-bar)	1074 SCFM air (set point 2-bar)	Primary Relief
2K helium supply from cryoplant	549 SCFM air (2-bar)	1074 SCFM air (set point 2-bar)	Primary Relief
Fire condition	7520 SCFM air	15,000-SCFM air (set point 4-bar)	Fire Relief
Loss of cavity vacuum		15,000-SCFM air (set point 4-bar)	Secondary Relief
- measured in DESY crash test	2273 SCFM air (4-bar)		
- calculated using 4 W/cm^2	6146 SCFM air		
Loss of insulating vacuum (measured in DESY crash test)	12,804 SCFM air (4-bar)	15,000-SCFM air (set point 4-bar)	Secondary Relief

Detailed calculations and test results show the required relief capacity for each of the sources of helium pressure.

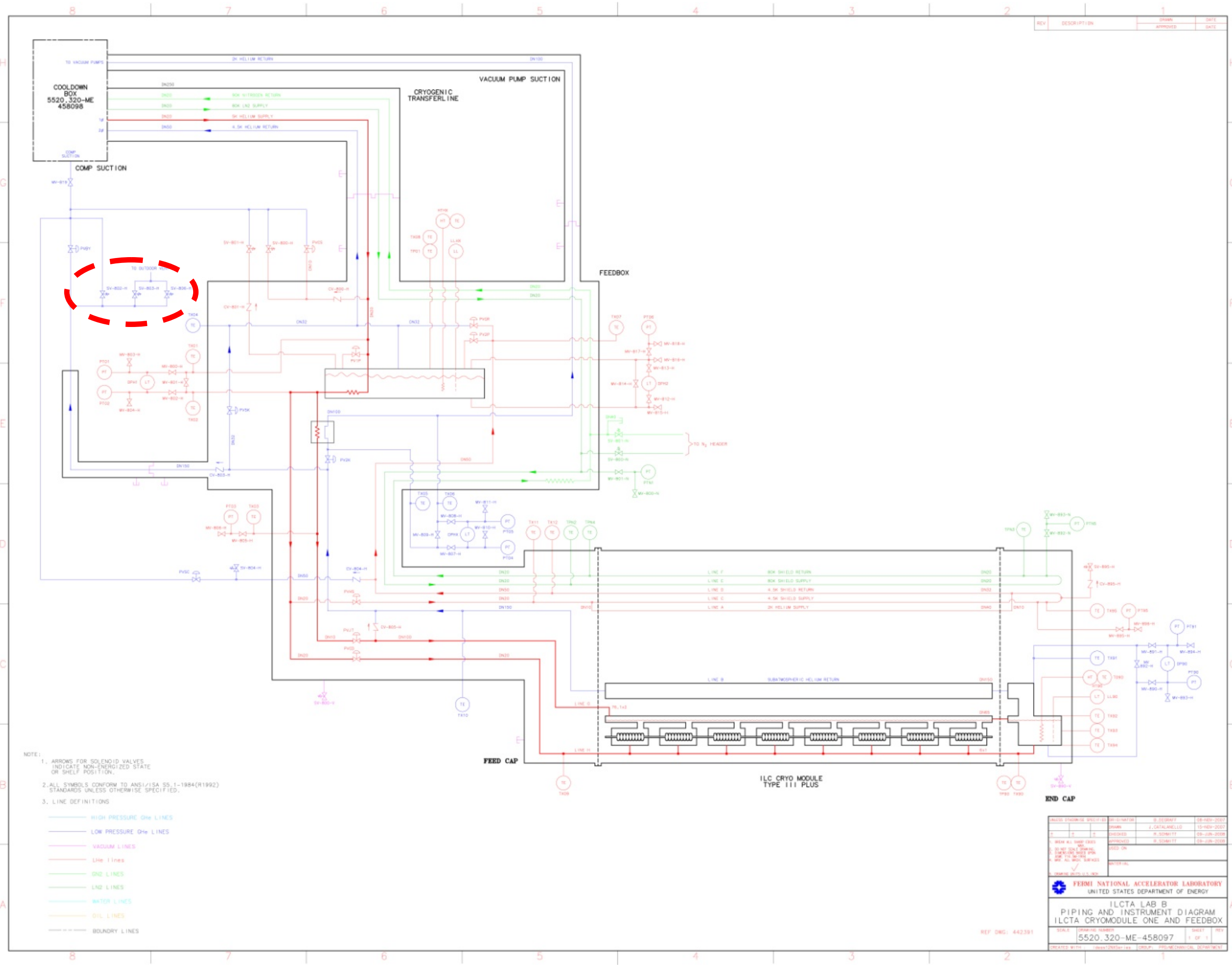


Figure 22 – P&ID of NML System where CM1 is Installed (Drawing 5520.000-ME-458097) (LHe relief valves circled in dashed line)

Room temperature helium supply from cryoplant

The maximum possible mass flow rate of helium from the cryoplant is 80-g/sec. At a pressure of 2-bar, the equivalent volumetric flow rate is calculated [11]:

$$Q_a = \frac{13.1 * W * C_a}{60 * C} \sqrt{\frac{Z * T * M_a}{M * Z_a * T_a}}$$

Where:

P	helium pressure	2	bar
T	helium temperature	300	K
m_dot	mass flow rate of helium	80	g/sec
m_dot_ASME	Correction factor to mass flow rate for ASME relief valve	88.9	g/sec
W	Corrected mass flow rate of helium	704	lbm/hr
Ca	gas constant of air	356	
Za	compressibility factor of air	1	
Ta	air temperature at standard conditions	520	R
Ma	air molecular weight	28.97	
C	helium gas constant	378	
M	molecular weight of helium	4	kg/kmol
ρ	helium density at T, P	0.32	kg/m^3
Z	compressibility factor of helium at T, P	0.25	
Qa	volumetric flow rate	198.5	SCFM air

2K helium supply from cryoplant

With the maximum helium mass flow rate at 80-g/sec, the equivalent volumetric flow rate at 2K is calculated using the following equation from the CGA S-1.3-2005, Paragraph 6.2.2 for primary relief: [10]

$$Q_a = \frac{(590 - T)}{4 * (1660 - T)} * F * G_i * U * A$$

The volumetric flow rate is calculated for pressure at 2-bar and at 4-bar. The flow rate at 2-bar is higher at 549-SCFM air.

		Warm system	Cold System	
P	relief pressure	2	4	bar
T	temperature at which the square root of specific volume divided by specific heat input @ relief pressure	5.1	6.5	K
		9.18	11.7	R
F	correction factor for cryogenic systems	1	1	
Gi	gas factor for insulated containers for LHe	52.5	52.5	
k_shield	thermal conductivity of helium gas at 80K	0.037	0.037	Btu/hr-ft-F
t_total	assume helium gas thickness of 1-inch	1	1	in
		0.083	0.083	ft
U	overall heat transfer coefficient of the insulating material	0.444	0.444	Btu/hr-ft^2-F
A_inner	inner surface of thermal shields	24860607	24860607	mm^2
		267.6	267.6	ft^2
Qa_primary	total minimum required flow capacity for primary PRD	548.7	547.1	SCFM air

Fire Condition

The required volumetric flow rate for fire condition in vessel is calculated following the CGA S-1.3-2005, Paragraph 6.3.3: (10)

$$Q_{a-fire} = F * G_i * U * A^{0.82}$$

Where:

F	correction factor for cryogenic systems	1	
Gi	gas factor for insulated containers for LHe	52.5	
k_shield	mean thermal conductivity of helium gas at between saturation temp and 1200 deg F at 1-bar (Table 3 of S-1.3)	0.122	Btu/hr-ft-F
t_total	assume helium gas thickness of 1-inch	1	in
		0.083	ft
U	overall heat transfer coefficient of the insulating material	1.464	Btu/hr-ft^2-F
A		267.6	ft^2
Qa_fire	flow capacity of relief device for fire conditions	7520.3	SCFM air

Loss of cavity vacuum and insulating vacuum

A large rate of helium vaporization can occur due to two scenarios:

- the loss of RF cavity vacuum
- the loss of insulating vacuum

DESY recently measured the air heat flows in each of these scenarios in a "crash test" of a cryomodule that was similar in design as CM1. [4] The air heat flow from the loss of RF cavity vacuum was measured at 99-kW. For the loss of insulating vacuum, the air heat flow was measured at 560-kW.

For this large mass flow of helium through CM1 at NML, the larger relief device, with the set point at 4-bar, will have adequate relief capacity.

		Loss of RF cavity vacuum	Loss of insulating vacuum	
q	air heat flow in DESY crash test	99400	560000	W
P_{relief}	relief set pressure	4	4	bar
		400	400	kPa
T	temperature when specific heat input	6	6	K
	is at a minimum for relief pressure	10.8	10.8	R
LH	specific heat input for helium at T, P_relief	19.4	19.4	J/g
m_dot	mass flow rate of helium during vaporization	5123.7	28866.0	g/sec
m_dot_{ASME}	Correction factor to mass flow rate	5693.0	32073.3	g/sec
	for ASME relief valve	45088.7	254020.6	lbm/hr
C	helium gas constant	378	378	
M	molecular weight of helium	4	4	kg/kmol
ρ	helium density at T, P_relief	80.29	80.29	kg/m^3
Z	compressibility factor for helium at flow condition	0.40	0.40	
C_a	air gas constant	356	356	
Z_a	air at Ta	1	1	
T_a	air at room temperature	520	520	R
M_a	air molecular weight	28.97	28.97	
Q_a	mass flow rate of helium during vaporization	2272.7	12804.1	SCFM air

For comparison, the helium boil-off during the loss of RF cavity vacuum is calculated based on the total surface area of the RF cavity, which is 1302-in^2 (0.84-m^2). For a loss of cavity vacuum due to an air leak, the heat flux of 4.0-W/cm^2 is used [15]. The specific heat input at the relief pressure of 4-bar and temperature 6°K is 19.4-J/g. The maximum mass flow rate can be calculated:

$$\dot{m}_{RF\_cavity} = \frac{A_{cavity} * Q}{LH}$$

And the equivalent volumetric flow rate

$$Q_a = \frac{13.1 * W * C_a}{60 * C} \sqrt{\frac{Z * T * M_a}{M * Z_a * T_a}}$$

As seen in the table below, the equivalent volumetric flow rate is 6146 SCFM-air.

q	based on surf area and heat flux	268800	W
P_relief	relief set pressure	4	bar
		400	kPa
T	temperature when specific heat input	6	K
	is at a minimum for relief pressure	10.8	R
LH	specific heat input for helium at T, P_relief	19.4	J/g
m_dot	mass flow rate of helium during vaporization	13855.7	g/sec
m_dot_ASME	Correction factor to mass flow rate	15395.2	g/sec
	for ASME relief valve	121929.9	lbm/hr
C	helium gas constant	378	
M	molecular weight of helium	4	kg/kmol
ρ	helium density at T, P_relief	80.29	kg/m^3
Z	compressibility factor for helium at flow condition	0.40	
Ca	air gas constant	356	
Za	air at Ta	1	
Ta	air at room temperature	520	R
Ma	air molecular weight	28.97	
Qa	mass flow rate of helium during vaporization	6146.0	SCFM air

Appendix B – Technical Specifications of Relief Valves

The technical specifications for the relief valves are shown in Appendix B. Both valves are from Leser. [12] The larger rupture disk (model 4414.4722 – size 6”x8”, set pressure 43-psig) is not listed in the catalog. However, its capacity is shown on a page from the company’s sizing software for the model 4414.4722.

LESER The-Safety-Valve.com	Sizing acc. to ASME VIII for Gas VALVESTAR® - v.7.1.4_07_26.0	Page:	1 of 6
		Date:	2008-12-08 12:54:46
		Project:	New project
		Tag No:	
		LESER Job №	

Sizing - Medium				
1000	Designation		Air	
1004	Formula			
1001	Molar mass	M	29	kg/kmol
1002	Ratio of specific heats	k	1.400	
1003	Compressibility factor	Z	1.000	
Sizing - Service condition				
1100	Maximum allowable working pressure			
1101	Set pressure	p	43	psi-g
1102	Superimposed back pressure	paf	0	psi-g
1103	Built up back pressure	pae		
1104	Backpressure		0	psi-g
1105	Overpressure	dp	10.00	%
1106	Environmental pressure	pu	14.696	psi
1107	Temperature	T	68	°F
1108	Required massflow	qm,ab	68,805.695	lb/h
1109	Volume flow to be discharged (working condition)	qvb,ab	216,710.786	ft³/h
1110	Volume flow to be discharged (std condition) [T=60 °F P=14.7 psi]	qvn,ab	14,999.72	SCFM
1120	Rupture disc correction factor	Kc	1.000	
Sizing - Calculation				
1200	Certified massflow	qm,zu	68,805.695	lb/h
1201	Certified volumeflow (operating condition)	qvb,zu	216,710.786	ft³/h
1203	Certified volumeflow (standard condition)	qvn,zu	25,488.055	m³/h
1204	Maximum mass flow	qm,max	76,450.772	lb/h
1205	Maximum volume flow (working condition)	qvb,max	240,789.762	ft³/h
1206	Maximum volume flow (standard condition)	qvn,max	28,320.061	m³/h
1207	Capacity exceed		0.00	%
Valve - General				
1500	Article number		4414.4722	
1512	Reseller article number		-	
1513	Quantity of safety valve		1	
1501	Certified coefficient of discharge for steam and gases	K,DG	0.699	
1502	Certified coefficient of discharge for liquid	K,F	0.521	
1453	Orifice		Q	
1505	Bonnet / Lifting device		Cap H2	
1506	Body-/ Inlet base material		1.4408 / SA 351 CF8M	
1511	Bonnet		Closed Bonnet	
1514	Order code	4414.4722-43 psi-g-H47H51-3.1		

Name	ASME VIII				
Date	2008-12-08 12:54:46				
Rev.No	1				

The capacity of the Leser rupture disk (4414.7942) is shown in the catalog dated June, 2004:

LESER | **Series 441 ANSI** | **Flanged Safety Relief Valves Spring Loaded**
How to order · Article Numbers · Dimensions and Weights

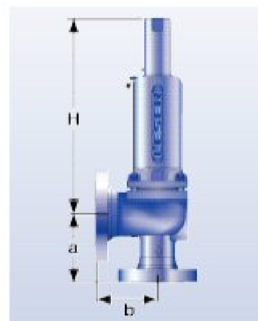
How to Order

Article Numbers Type 441 / 442

Valve Size [inch]	Orifice Area [inch²]	Orifice Diameter [mm]	Body Material SA 216 WCB				Body Material SA 351 CF8M	
			Type 4412 Closed bonnet			Type 4422 Open bonnet	Type 4414 Closed bonnet	
			Cap H2	Packed lever H4	Plain lever H3	Plain lever H3	Cap H2	Packed lever H4
1" x 2"	0.644	23	4412.4812	4412.4814	4412.4813	4422.4815	4414.7912	4414.7914
1 1/2" x 2"	1.024	29	4412.4822	4412.4824	4412.4823	4422.4825	–	–
1 1/2" x 2 1/2"	1.667	37	4412.4832	4412.4834	4412.4833	4422.4835	4414.7932	4414.7934
2" x 3"	2.576	46	4412.4842	4412.4844	4412.4843	4422.4845	4414.7942	4414.7944
3" x 4"	4.383	60	4412.4862	4412.4864	4412.4863	4422.4865	4414.7962	4414.7964
4" x 6"	10.304	92	4412.4872	4412.4874	4412.4873	4422.4875	4414.7972	4414.7974

Dimensions and Weights US Units

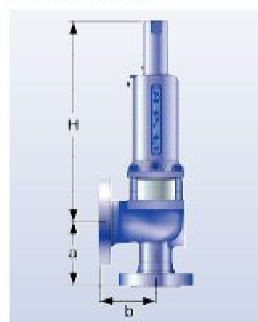
Valve Size [inch]	Center to Face [inch]		Height [inch]		Weight [lbs]
	Inlet	Outlet	Standard	Bellows	
	a	b	H	H	m
1" x 2"	4 1/8	4 1/2	9 3/16	10 11/16	22
1 1/2" x 2"	4 7/8	4 3/4	13	14 5/8	29
1 1/2" x 2 1/2"	4 7/8	4 3/4	14 5/8	16 1/8	35
2" x 3"	5 3/8	4 7/8	16 1/2	18 3/16	49
3" x 4"	6 1/8	6 1/2	20 13/16	23	73
4" x 6"	7 1/8	9	26 1/16	28 1/4	165



Conventional Design

Dimensions and Weights Metric Units

Valve Size [inch]	Center to Face [mm]		Height [mm]		Weight [kg]
	Inlet	Outlet	Standard	Bellows	
	a	b	H	H	m
1" x 2"	105	114	234	273	10
1 1/2" x 2"	124	121	331	373	13
1 1/2" x 2 1/2"	124	121	372	410	16
2" x 3"	137	124	419	465	22
3" x 4"	156	165	529	585	33
4" x 6"	181	229	663	721	75



Balanced Bellows Design

14 LWN 472.12

AIR

ASME Section VIII
[S.C.F.M.]

Capacities US Units

Valve Size	1" x 2"	1½" x 2"	1½"x2½"	2" x 3"	3" x 4"	4" x 6"
	Orifice [inch²]					
Set Pressure [psig]	0.644	1.024	1.667	2.576	4.383	10.304
15	269	427	695	1074	1828	4296
20	310	493	802	1239	2109	4957
30	392	624	1016	1570	2671	6278
40	483	768	1251	1933	3289	7732
50	574	913	1486	2296	3907	9186
60	665	1057	1721	2660	4526	10639
70	756	1202	1956	3023	5144	12093
80	847	1346	2192	3387	5762	13546
90	938	1491	2427	3750	6381	15000
100	1028	1635	2662	4113	6999	16454
120	1210	1924	3132	4840	8235	19361
140	1392	2213	3603	5567	9472	22268
160	1573	2502	4073	6294	10709	25175
180	1755	2791	4543	7021	11945	28082
200	1937	3080	5014	7747	13182	30989
220	2119	3369	5484	8474	14419	33897
240	2300	3658	5954	9201	15655	36804
260	2482	3946	6425	9928	16892	39711
280	2664	4235	6895	10655	18128	42618
300	2845	4524	7365	11381	19365	45525
320	3027	4813	7836	12108	20602	48432
340	3209	5102	8306	12835	21838	51340
360	3390	5391	8776	13562	23075	54247
380	3572	5680	9247	14289	24312	57154
400	3754	5969	9717	15015	25548	60061
420	3936	6258	10187	15742	26785	62968
440	4117	6547	10657	16469	28021	65875
460	4299	6836	11128	17196	29258	68783
480	4481	7124	11598	17922	30495	71690
500	4662	7413	12068	18649	31731	
550	5117	8136	13244	20466	34823	
600	5571	8858	14420	22283		
650	6025	9580	15596	24100		
700	6479			25917		
740	6843			27371		

AIR

ASME Section VIII
[m_n^3/h]

Capacities Metric Units

Valve Size	1" x 2"	1½" x 2"	1½"x2½"	2" x 3"	3" x 4"	4" x 6"
	Orifice [mm²]					
Set Pressure [bar]	416	661	1075	1662	2827	6648
1	437	695	1132	1749	2975	6994
2	635	1010	1644	2541	4323	10163
3	852	1354	2205	3407	5797	13628
4	1070	1701	2769	4279	7280	17115
5	1288	2047	3333	5150	8763	20601
6	1505	2394	3897	6022	10246	24087
7	1723	2740	4461	6893	11729	27573
8	1941	3087	5025	7765	13212	31060
9	2159	3433	5589	8637	14695	34546
10	2377	3780	6153	9508	16178	38032
12	2813	4473	7281	11251	19144	45005
14	3249	5165	8409	12994	22110	51977
16	3684	5858	9537	14737	25075	58950
18	4120	6551	10665	16481	28041	65922
20	4556	7244	11793	18224	31007	72895
22	4992	7937	12921	19967	33973	79867
24	5428	8630	14049	21710	36939	86840
26	5863	9323	15177	23453	39905	93813
28	6299	10016	16305	25196	42871	100785
30	6735	10709	17433	26939	45837	107758
32	7171	11402	18561	28683	48803	114730
34	7606	12095	19689	30426	51769	121703
36	8042	12788	20817	32169	54735	
38	8478	13481	21945	33912	57700	
40	8914	14174	23073	35655	60666	
42	9350	14866	24201	37398		
44	9785	15559	25330	39141		
46	10221	16252	26458	40885		
48	10657	16945		42628		
50	11093			44371		
51	11311			45242		

Capacities for air according to ASME Section VIII (UV), based on set pressure plus 10 % overpressure at 60 °F (16 °C).
Capacities at 30 psig (2.07 bar) and below are based on 3 psig (0.207 bar) overpressure.

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Amendment 1

4 November 2010

Updates on the System Venting Verification

The AD/Cryo document titled "New Muon Lab Cryomodule, Feed Cap, and End Cap Relief Valve System Analysis" (6 Sept 2010) (located online http://ilctanmlcryo.fnal.gov/) lists the most up-to-date calculations on the design relief capacity for at the New Muon Lab (NML). As mentioned in the original pressure vessel engineering note, there are two safety relief valves for venting helium from CM1 and are considered in the system venting calculations. Both are rupture disks, as detailed below (see Tables 1, 2, and 3 in the AD/Cryo document):

- SV-803-H: Set pt. = 43 psig (4-bar), Leser Model 4414.4722, nominal size = 6"x8", 8053-SCFM air (16,175-g/sec)
- SV-806-H: Set pt. = 15 psig (2-bar), Leser Model 4414.7942, nominal size = 2"x3", 951-SCFM air (217-g/sec)

Table AM-1 summarizes the possible sources of helium pressure, the calculated maximum flow rate, and the capacity of the available relief valve.

Table AM-1 – Summary of Required and Available Relief Capacities at NML

Source of Helium Pressure	Required relief capacity (SCFM air)	Available relief capacity (SCFM air)	Relief type (CGA-defined)	Relief Device Name
Room temperature helium supply from cryoplant	351	951	Primary relief	SV-806-H
2K helium supply from cryoplant	452	951	Primary relief	SV-806-H
Fire condition	6213	8053	Secondary relief	SV-803-H
Loss of cavity vacuum	5176	8053	Secondary relief	SV-803-H
Loss of insulating vacuum	1849	8053	Secondary relief	SV-803-H

Room temperature helium supply from cryoplant

(The calculation for the required relief capacity for room temperature helium from the cryoplant supply was updated to reflect the helium compressibility factor Z = 1 for room temperature.)

The maximum possible mass flow rate of helium from the cryoplant is 80-g/sec. At a pressure of 2-bar, the equivalent volumetric flow rate is calculated [11]:

$$Q_a = \frac{13.1 * W * C_a}{60 * C} \sqrt{\frac{Z * T * M_a}{M * Z_a * T_a}}$$

Where:

P	helium pressure	2	bar
T	helium temperature	520	R
m_dot	mass flow rate of helium from cryoplant	80	g/sec
W	mass flow rate of helium	633.6	lbm/hr
C_a	gas constant of air	356	
Z_a	compressibility factor of air	1	
T_a	air temperature at standard conditions	520	R
M_a	air molecular weight	28.97	
C	helium gas constant	378	
M	molecular weight of helium	4	kg/kmol
Z	compressibility factor of helium at T, P	1	
Q_a	volumetric flow rate of room temperature helium from cryoplant	351	SCFM air

2K helium supply from cryoplant

(This calculations is updated to reflect the corrected value of G_i = 43.4 for supercritical helium and to take into account a relief pressure 110% of the set pressure.)

The required capacity for venting 2K helium from the cryoplant supply is calculated using the following equation from the CGA S-1.3-2005, Paragraph 6.2.2 for primary relief: [10]

$$Q_a = \frac{(590 - T)}{4 * (1660 - T)} * F * G_i * U * A$$

P	relief pressure	4.4	bar
T	temperature at which the square root of specific volume divided by specific heat input @ relief pressure	6.5	K
		11.7	R
t		-457.97	F
F	correction factor for cryogenic systems	1	
L'	Heat absorbed per pound of helium vapor leaving the helium vessel	23	J/g
		9.91	Btu/lb
M	Helium molecular weight	4	kg/kmol
ρ	helium density at P_cold, T	53.39	kg/m^3
Z	Compressibility factor for helium at flow conditions	0.583	
G_i	gas factor for insulated containers for LHe	43.4	
k_{shield}	thermal conductivity of helium gas at 80K	0.037	Btu/hr-ft-F
t_{total}	assume helium gas thickness of 1-inch	1	in
		0.083	ft
U	overall heat transfer coefficient of the insulating material	0.444	$Btu/hr\text{-}ft^2\text{-}F$
A_{inner}	inner surface of thermal shields	24860607	mm^2
		267.6	ft^2
$Q_{a\ primary}$	total minimum required flow capacity for 2K helium from cryoplant	451.7	SCFM air

Fire Condition

(This calculation is updated to reflect the corrected value of G_i = 43.4 for supercritical helium, which is the same value as used in calculating the flow capacity for 2K helium from the cryoplant.)

The required volumetric flow rate for fire condition in vessel is calculated following the CGA S-1.3-2005, Paragraph 6.3.3: [10]

$$Q_{a-fire} = F * G_i * U * A^{0.82}$$

Where:

F	correction factor for cryogenic systems	1	
G_i	gas factor for insulated containers for LHe (same as for the primary relief)	43.4	
k_{shield}	mean thermal conductivity of helium gas at between saturation temp and 1200 deg F at 1-bar (Table 3 of S-1.3)	0.122	Btu/hr-ft-F
t_{total}	assume helium gas thickness of 1-inch	1	in
		0.083	ft
U	overall heat transfer coefficient of the insulating material	1.464	Btu/hr-ft^2-F
A		267.6	ft^2
$Q_{a\ fire}$	flow capacity of relief device for fire conditions	6213.1	SCFM air

Loss of cavity vacuum and insulating vacuum

(In the original note, the values of the heat transfer used to calculate the relief capacity included all cryogenic circuits in the cryomodule of the DESY crash test. These calculations show updated relief capacities to reflect two issues. One is to use the measured heat transfer densities from the DESY crash test that is specific to the 2K circuit, which is listed in Table 3 of the test results. [4] Also, the relief pressure is 110% of the set pressure.)

A large rate of helium vaporization can occur due to two scenarios: the loss of RF cavity vacuum, and the loss of insulating vacuum

DESY recently measured the air heat flows in each of these scenarios in a "crash test" of a cryomodule that was similar in design as CM1. [4] Table 3 of the crash test results shows the measured heat transfer densities each of the cryogenic circuits. For the 2K circuit, the largest heat transfer density from the loss of RF cavity vacuum was measured at 23-kW/m^2. For the loss of insulating vacuum, the largest air heat flow was measured as 6.5-kW/m^2. The authors estimate that the measurements have an accuracy of ±50%. For the purposes of calculating the required relief capacity, the DESY results are used, including a 50% increase to account for inaccuracies in the measurements. So, to calculate the mass flow rate of helium during vaporization, the values of 34.5- kW/m^2 and 9.75- kW/m^2 are used for the loss of RF cavity vacuum and loss of insulating vacuum, respectively. Note that for the heat flux of 34.5-kW/m^2 (3.45-W/cm^2) is less than the 4-W/cm^2 heat flux that has been used in system venting analysis to date. The equation to calculate the mass flow rate is

$$\dot{m} = \frac{A * Q}{\theta}$$

And the equivalent volumetric flow rate is

$$Q_a = \frac{13.1 * W * C_a}{60 * C} \sqrt{\frac{Z * T * M_a}{M * Z_a * T_a}}$$

Where:

		Cavity Vacuum Loss (8 RF cavities)	Loss of Insulating Vacuum	
Q	measured heat density in DESY crash test, including 50% error	34.5	9.75	kW/cm^2
P_relief	110% of set pressure of relief device	4.4	4.4	bar
		440	440	kPa
T	temperature when specific heat input	6.8	6.8	K
	is at a minimum for relief pressure	12.24	12.24	R
θ	specific heat input for helium at T, P_relief	23	23	J/g
A	Surface area of helium-to-vacuum boundary	6.72	8.50	m^2
m_dot	mass flow rate of helium during vaporization	10080.0	3602.0	g/sec
W	mass flow rate of helium during vaporization	79833.6	28527.8	lbm/hr
C	helium gas constant	378	378	
M	molecular weight of helium	4	4	kg/kmol
ρ	helium density at T, P_relief	53.39	53.39	kg/m^3
Z	compressibility factor for helium at flow condition	0.58	0.58	
C_a	air gas constant	356	356	
Z_a	air at Ta	1	1	
T_a	air at room temperature	520	520	R
M_a	air molecular weight	28.97	28.97	kg/kmol
Q_a	mass flow rate of helium during vaporization	5175.6	1849.5	SCFM air