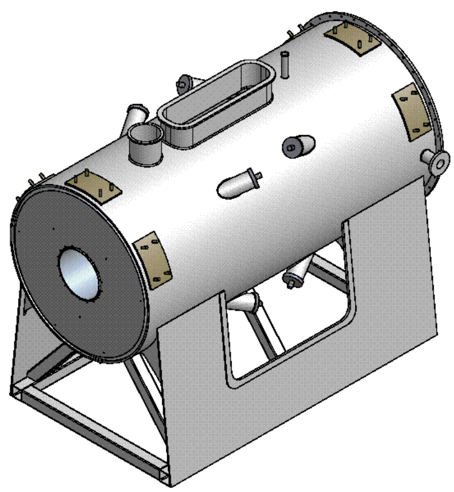
proj: **Muon Ionization Cooling Experiment (MICE)**
MICE - Spectrometer Solenoid Experiment

title: **Cryostat Pressure Safety**

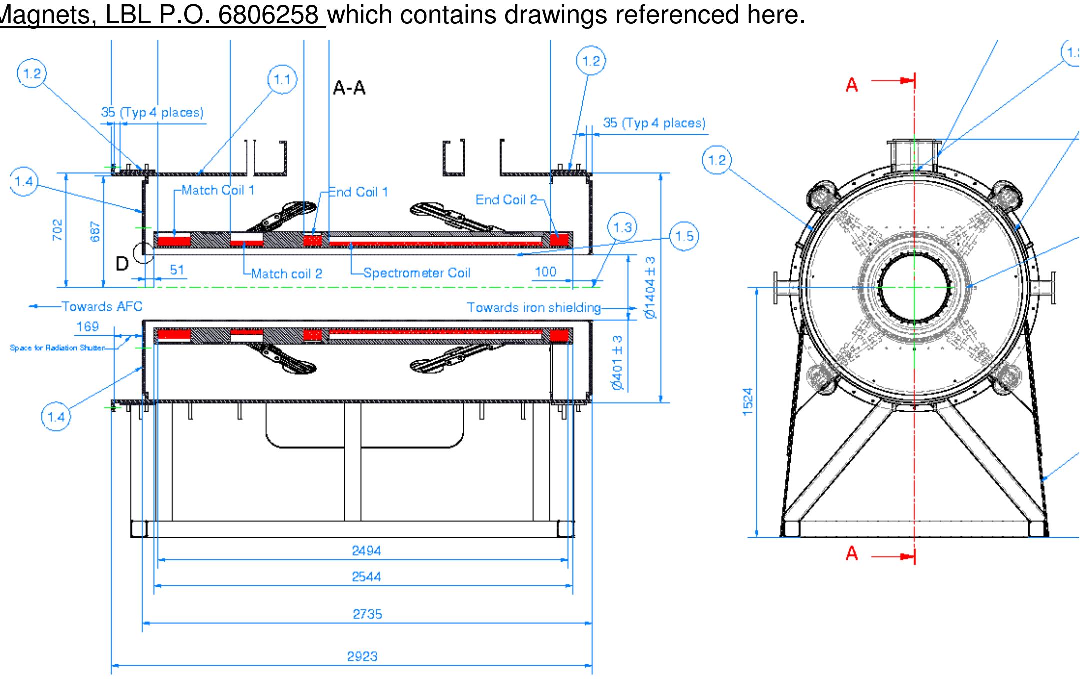


Assembly View

1. Introduction

This note addresses the design compliance of the MICE Tracker Module Solenoid Magnet (in fabrication as of Jan 15, 2008) with respect to the 1998 ASME Pressure Vessel code (with updates to 2000). Compliance with code sections dealing with material and welding qualification and certification is not addressed.

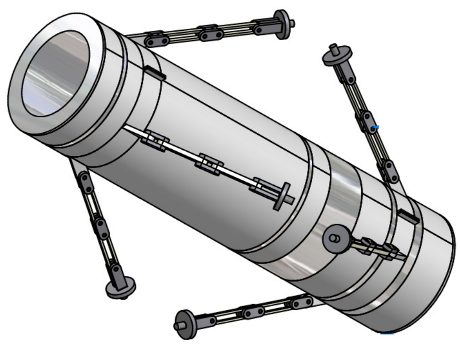
The MICE collaboration has developed the main magnetic and structural conceptual design which is presented in LBNL Engineering Specification 10154: Technical Specification, MICE Spectrometer Solenoid Magnets, Fabrication, Assembly, Test and Shipping. The subcontracting fabricator for the entire assembly, Wang NMR, Inc. has added the details and provided a design report: Final Engineering Design for MICE Spectrometer Solenoid Magnets, LBL P.O. 6806258 which contains drawings referenced here.



Assembly Cross Sections (from Specification dwg; final design differs slightly)

The MICE Tracker Module Solenoid Magnet consists of a cylindrical Vacuum Vessel, inside which is suspended a Helium Vessel containing a set of five superconducting solenoid magnet coils, all wound on a single, multiple flanged spool (bobbin). The coils are bathed in liquid helium during operation. The flanges between the individual coils are larger in diameter than the largest coil outer diameter, and a cylindrical cover is welded on to the end flanges over the coils thus forming a reservoir volume for LHe. The spool forms the Inner Helium Cylinder, the cover forms the Outer Helium Cylinder, and the spool end flanges form the endplates of the Helium Vessel. Together with the coils these form the cold mass.

Similarly, the Vacuum vessel consists of an Outer Vacuum Cylinder, an Inner Vacuum Cylinder, both welded to front and back Vacuum Endplates. The Inner Vacuum Cylinder bore provides the space for the spectrometer tracker. In the vacuum space between the Helium Vessel and the Vacuum Vessel, multi-layer insulation and a 77K nitrogen cooled shield are installed. The Helium Vessel is suspended inside the Vacuum Vessel using low heat leak fiberglass "endless loop" tension link supports. Drawing MICE-1000 shows overall assembly of the final design.



Cold Mass (Helium Vessel and coils) with Supports

Analyses here are based as much as possible on 1998 ASME Pressure Vessel Code (w/revisions to 2000), section VIII (UG, UW, and ULT), with subsections as noted. Other analyses use Roark's Formulas for Stress and Strain 6th ed. Both the Helium Vessel and the Vacuum Vessel were fabricated in a non-ASME code shop, so, per Fermilab rules, an additional factor for maximum allowable stress is included for the final value of maximum allowable stress for each component. In addition, this factor is included in (maximum) external pressure calculations, even though these values are determined by elastic stability, not material strength.

$K_{sr} := 0.8$ FNAL stress reduction factor for non-code certified shops

2. Helium Vessel

The Helium Vessel (dwg. MICE-C0000) consists of an aluminum spool (or bobbin), referred to here as the Inner Helium Cylinder, on which the coils are wound. There are multiple flanges to separate the coils. The coils are then covered by an aluminum cylinder, referred to here as Outer Helium Cylinder which is welded on each end to the end flanges of the Inner Helium Cylinder. All these parts are made from 6061-T6 aluminum. S.S. lead, vent (neck) and fill tubes (not shown) are attached using Al/S.S. bimetallic flanges. Machined channels in the inner flanges provide for leads and He flow from all coils to fill and vent lines.

Conditions

Maximum working internal pressure

$P_{max\_Hev} := 44psig$ (from Technical Specification 10154, section 3.3.3, quench pressure. Note that this assumes negligible pressure drop through vent system during quench conditions)

Minimum working internal (maximum external) pressure

$P_{min\_Hev} := -15psig$ (from Technical Specification 10154, section 3.3.3, external atmospheric pressure from vacuum during leak checking)

Burst disk pressures

$P_{bd\_vv} := 4.4psig$ (Vacuum Vessel)

$P_{bd\_Hev} := 44psig$ (Helium Vessel, specified to be between 32 and 44 psig)

Relief Valve Pressures

$P_{rv\_vv} := 2.3psig$ (Vacuum Vessel)

$P_{rv\_hev} := 29psig$ (Helium Vessel)

Note : all pressure calculations here are based on the burst disk pressures, not the relief valve pressures.

Outer Helium Cylinder

This cylinder (dwg. #MICE-C002, COIL FORM COVER) is rolled and welded 1/2" thick 6061-T6. It is formed from two semicircular halves which are placed over the coils and then welded together longitudinally (full penetration weld specified) and to the end flanges of the spool at each end (fillet welds). It has several small openings, and no intermediate reinforcing ribs. It is subject to internal pressure and longitudinal compression from both vacuum and from the cold mass support forces. It is subject to external pressure under leak checking, which could conceivably be necessary to perform in the assembled condition (with stabilized cryocooler sleeves, as described in section 2e).

Dimensions (from dwgs. MICE-C000, and MICE-C002):

wall thickness: $t_{oHec} := 12.7mm$

length: $L_{oHec} := 2.544m$

inner radius: $R_{i\_oHec} := 0.3495m$

outer radius: $R_{o\_oHec} := R_{i\_oHec} + t_{oHec}$; $R_{o\_oHec} = 0.3622\,m$

Material: 6061-T6 , welded. Yield strength: $S_{y\_6061T6\_w} := 10000psi$ (Section II Part D,Table 2A)

Young's modulus: $E_{Al} := 10.0\cdot 10^{6} psi$ (Section II Part D, Subpart 3)

ASME Maximum Allowable Stress : $S_{6061T6\_w} := 6000psi$ (Section II Part D,Table 2A)

Maximum Allowable Stress: $S_{oHec} := K_{sr}\cdot S_{6061T6\_w}$; $S_{oHec} = 4800\,psi$

Weld joint efficiency: $E_j := 0.6$ from table UW-12, assume single welded butt joint with no backing strip

UG-23b Maximum Allowable Longitudinal Compressive Stress

The eight cold mass supports exert a net axial compressive force on both the Outer Helium Cylinder and the Inner Helium Cylinder (parallel load paths). This force, transferred to the Helium Vessel near its ends means the length of each compressive element is approximately the same, thus deflection and strain are also approximately the same. Since the materials are also the same, the stress will be approximately the same :

$F_{support} := 200kN$ from M. A. Green, correspondence of 12/12/07 $\alpha_{support} := 45deg$ (approx)

Axial component of this force (total for all 8) :

$F_{axial\_oHec} := 4\cdot F_{support}\cdot \cos(\alpha_{support})$; $F_{axial\_oHec} = 1.2717 \times 10^{5}\,lbf$

Inner Helium Cylinder radius and thickness:

$R_{i\_iHec} := 0.245m$ $t_{iHec} := 12mm$

$$\sigma_{axial\_oHec} := \frac{-F_{axial\_oHec}}{2\cdot\pi\left(R_{i\_oHec}\cdot t_{oHec} + R_{i\_iHec}\cdot t_{iHec}\right)} \quad ; \quad \sigma_{axial\_oHec} = -1770\,psi$$

Maximum allowable longitudinal compressive stress is determined by the following procedure:
First, determine:

$$A_{long} := \frac{0.125}{\frac{R_{o\_oHec}}{t_{oHec}}} \qquad A_{long} = 0.004$$

from the appropriate chart in Sec II, Pt. D, Subpart 3 (fig NFA-13) find B:

$$B_{long} := 7200psi$$

check that actual stress is smaller than (smaller of B_{long} or max allowable stress): $S_{oHec} = 4800\,psi$

$$\left|\sigma_{axial\_ohec}\right| < S_{oHec} \quad \text{OK}$$

UG-27 Thickness of Shells under Internal Pressure

Internal pressure, maximum (worst case, under quench and relief valve failure conditions):

$$P_{i\_oHec} := P_{max\_Hev} + 15psi \quad ; \; P_{i\_oHec} = 59\,psid$$

Minimum required thickness based on circumferential stress:

$$t_{min\_oHec\_cs} := \frac{P_{i\_oHec} \cdot R_{i\_oHec}}{S_{oHec} \cdot E_j - 0.6P_{i\_oHec}} \quad ; \; t_{min\_oHec\_cs} = 0.285\,in$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_oHec\_ls} := \frac{P_{i\_oHec} \cdot R_{i\_oHec}}{2S_{oHec} \cdot E_j - 0.4P_{i\_oHec}} \quad ; \; t_{min\_oHec\_ls} = 0.142\,in$$

$$; \; t_{oHec} = 0.5\,in$$

$t_{oHec} > t_{min\_oHec\_cs}$ therefore the outer helium cylinder is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_oHec} := P_{i\_oHec} \cdot \frac{t_{oHec}}{t_{min\_oHec\_cs}} \quad ; \; P_{mawi\_oHec} = 103.4\,psi$$

UG-28 Thickness of Shells under External Pressure

The maximum external pressure condition would occur if leak checking the Helium Vessel in the assembled condition, possibly with some unanticipated overpressure in the insulating space. This is not an anticipated or planned condition, but could at some point become necessary. Currently, the cryocooler sleeves disallow this condition, but, if necessary, one could remove the cryocoolers and insert a mechanical stabilizing tool to keep the sleeves from collapsing.

External pressure, maximum: $P_{e\_oHec} := -P_{min\_Hev} + P_{bd\_vv} \quad ; \; P_{e\_oHec} = 19.4\,psi$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{oHec}}{2R_{o\_oHec}} = 3.5 \qquad \frac{2R_{o\_oHec}}{t_{oHec}} = 57$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$A_{oHec} := 0.0014$

Using the factor A in the applicable material (304 S.S.) chart (HA-1) in Subpart 3 of Section II, Part D, we find the factor B:

$B_{oHec} := 10000psi$

The maximum allowable working external pressure is then given by (including the FNAL factor K_{sr}):

$$P_{mawe\_oHec} := \frac{4B_{oHec} \cdot K_{sr}}{3\left(\frac{2R_{o\_oHec}}{t_{oHec}}\right)} \quad ; \; P_{mawe\_oHec} = 187\,psi$$

$P_{mawe\_oHec} > P_{e\_oHec}$ so the Outer Helium Cylinder is safe from buckling under vacuum load

UG-36 Openings in Pressure Vessels

Subsection (c)(3)(a) of UG-36 allows isolated unreinforced openings of 2 3/8" (60mm) or less, in finished diameter, for shell thicknesses t> 3/8", given no rapidly fluctuating loads. There are three holes in the Outer Helium Cylinder, one for the neck tube and two for the leads) which are 63.5mm (2.5") in diameter. However these holes are for the purpose of welding in bimetallic flanges, and the finished size of the holes is only 36mm, so no reinforcement appears to be necessary. However, the validity of this interpretation is not clear as the weld connecting the bimetal flange to the vessel wall is clearly important for load transfer, and furthermore, the weld itself is not fully specified and may not meet minimum requirements of section UW-16 Minimum requirements for Attachment Welds at Openings. Given this uncertainty, we check adequacy of reinforcement below:

UG-37 Reinforcement required for Openings in Shells and Formed Heads

The holes have bimetallic Al/S.S. junctions welded in them which can serve as reinforcement. In addition, there is excess shell thickness (over and above that required per section 26 and 27, which also counts as reinforcement. The weld design appears similar to that shown in figs. UG-40 (a-3) and UW-16.1 (v-2) except that a full penetration weld with a beveled weld prep is not called for on the drawing; it seems that a fillet weld without a weld groove is more likely to be what gets made. This would not be in compliance with ASME code for reinforcements.

For:

required minimum thickness for seamless vessel with no openings, using weld efficiency E_j=1:

$$t_{r\_oHec} := \frac{P_{i\_oHec} \cdot R_{i\_oHec}}{S_{oHec} - 0.6P_{i\_oHec}} \quad ; \quad t_{r\_oHec} = 0.1704\,in$$

stress variation factor (circumferential to longitudinal)

$F_{sv} := 1.0$

strength reduction factors for differing material strengths (assume all strengths equal)

$f_{r1} := 1.0 \quad f_{r3} := 1.0 \quad f_{r4} := 1.0$

opening weld joint efficiency (for opening passing through weld)

$E_1 := 1.0$

and the following dimensions as per fig UG-37.1:

finished hole (nozzle) diameter (inside); radius

$d_h := 36mm \qquad R_n := 0.5 \cdot d_h \quad ; \quad R_n = 18\,mm$

nozzle thickness (dwg. MICE-C008)

$t_n := 0mm$ (as shown in fig UG-40 (a-3))

reinforcement height (est. from dwgs MICE-C0008 (part) and MICE-C0000 (assembly))

$t_e := 21.6\text{mm} - t_{oHec}$ $\quad t_e = 8.9\,\text{mm}$

reinforcement outer diameter

$D_p := 63.5\text{mm}$ (scaled from dwg, note that the bimetal junction is the reinforcement)

weld leg size (est)

$d_w := 0.7t_e$ $\quad d_w = 6.2\,\text{mm}$

Area of reinforcement required:

$$A_{rnf\_oHec} := d_h \cdot t_{r\_oHec} \cdot F_{sv} + 2t_n \cdot t_{r\_oHec} \cdot F_{sv} \cdot (1 - f_{r1})$$

$$A_{rnf\_oHec} = 155.8\,\text{mm}^2$$

UG-40 Limits of Reinforcement

These define the allowable dimensional extents for any reinforcement (see fig UG-37.1)

(b) parallel to vessel wall. Use greater of the following:

$L_{1a} := d_h$; $L_{1a} = 36\,\text{mm}$ <<<

$L_{1b} := R_n + t_{oHec} + t_n$; $L_{1b} = 30.7\,\text{mm}$

(c) perpendicular to vessel wall. Use smaller of the following:

$L_{2a} := 2.5 \cdot t_{oHec}$; $L_{2a} = 31.75\,\text{mm}$

$L_{2b} := 2.5 \cdot t_n + t_e$; $L_{2b} = 8.9\,\text{mm}$ <<<

Available areas for reinforcement, per Fig UG-37.1 :

in excess wall thickness, use larger of (t_n=0):

$A_{1a\_oHec} := d_h \cdot (E_1 \cdot t_{oHec} - f_{r1} \cdot t_{r\_oHec}) - 0$; $A_{1a\_oHec} = 301.4\,\text{mm}^2$ <<<

$A_{1b\_oHec} := 2 \cdot (t_{oHec} + t_n) \cdot (E_1 \cdot t_{oHec} - f_{r1} \cdot t_{r\_oHec}) - 0$; $A_{1b\_oHec} = 212.7\,\text{mm}^2$

in outside weld:

$A_{41\_oHec} := d_w^2 \cdot f_{r3}$; $A_{41\_oHec} = 38.8\,\text{mm}^2$

in reinforcement thickness:

$A_{5\_oHec} := (D_p - d_h - 2t_n) \cdot t_e \cdot f_{r4}$; $A_{5\_oHec} = 244.8\,\text{mm}^2$

Total area of reinforcement

$A_{t\_oHec} := A_{1a\_oHec} + A_{41\_oHec} + A_{5\_oHec}$; $A_{t\_oHec} = 585\,\text{mm}^2$

compare with required area>> $A_{rnf\_oHec} = 155.8\,\text{mm}^2$

$A_{t\_oHec} > A_{rnf\_oHec}$ so these openings have sufficient reinforcement area.

UG-42 Reinforcement of Multiple Openings

For multiple holes whose centers are spaced less than twice the average of their diameters, this section describes additional requirements. However, it is the finished hole size of 36mm, that applies here, not the 63.5mm bored holes used for welding in the bimetal junctions. The center to center distance is 100mm, and the limits of reinforcement (L_{ia}=d_h=36mm) do not overlap, so this section does not apply here.

Inner Helium Cylinder (spool)

This cylinder is a machined cylinder of 6061-T6 aluminum with very thick flanges in between the coils. the thinnest section is for the main solenoid coil. It is subject to external pressure (internal positive pressure in He vessel) and a small amount of longitudinal compression from the support tension links transferred in from the end welds. It might be subject to internal pressure, under leak checking conditions with a vacuum in the He vessel and atmospheric pressure in the insulating space.

Dimensions (from dwgs. MICE-C000, and MICE-C001):

wall thickness: $t_{iHec} = 12\,mm$

length: $L_{iHec} := 1.3207m$ (longest thin section between flanges

total length $L_{t\_iHec} := L_{oHec}$; $L_{t\_iHec} = 2.544\,m$

inner radius: $R_{i\_iHec} = 0.245\,m$

outer radius: $R_{o\_iHec} := R_{i\_iHec} + t_{iHec}$; $R_{o\_iHec} = 0.257\,m$

inner diameter: $D_{i\_iHec} := 2R_{o\_iHec}$; $D_{i\_iHec} = 0.514\,m$

Material: 6061-T6. , welded. Yield strength: $S_{y\_6061T6} := 20000psi$ (Section II Part D,Table 2A)

Young's modulus: $E_{Al} = 1 \times 10^{7}\,psi$ (Section II Part D, Subpart 3)

Maximum Stress: $S_{6061T6} := 12000psi$ (Section II Part D,Table 2A)

Stress limit: $S_{iHec} := K_{sr} \cdot S_{6061T6}$; $S_{iHec} = 9.6 \times 10^{3}\,psi$

UG-23b Longitudinal Compressive Stress

From same section for Outer Helium Cylinder; above:

$\sigma_{axial\_oHec} = -1770\,psi$

$\sigma_{axial\_iHec} := \sigma_{axial\_oHec}$

Maximum allowable longitudinal compressive stress is determined by the following procedure:
First, determine:

$$A_{long\_iHec} := \frac{0.125}{\left(\frac{R_{o\_iHec}}{t_{iHec}}\right)} \qquad A_{long\_iHec} = 0.006$$

from the appropriate chart in Sec II, Pt. D, Subpart 3 (fig NFA-13) find B:

$B_{long\_iHec} := 7300psi$ $\quad S_{iHec} = 9600\,psi$

check that actual stress is smaller than (smaller of) B_{long} or max allowable stress

$\left|\sigma_{axial\_iHec}\right| < B_{long\_iHec}$; $\sigma_{axial\_iHec} < B_{long\_iHec}$ OK

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_iHec} := -P_{min\_Hev}$; $P_{i\_iHec} = 15\,psi$

Minimum required thickness based on circumferential stress:

$$t_{min\_iHec\_cs} := \frac{P_{i\_iHec} \cdot D_{i\_iHec}}{2S_{iHec} \cdot E_j - 0.6P_{i\_iHec}} \quad ; \; t_{min\_iHec\_cs} = 0.0264\,in$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_iHec\_ls} := \frac{P_{i\_iHec} \cdot D_{i\_iHec}}{4S_{iHec} \cdot E_j - 0.4P_{i\_iHec}} \quad ; \; t_{min\_iHec\_ls} = 0.0132\,in$$

$$; \; t_{iHec} = 0.4724\,in$$

$t_{ivc} > t_{min\_ivc\_cs}$ therefore the Inner Helium Cylinder is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_iHec} := P_{i\_iHec} \cdot \frac{t_{iHec}}{t_{min\_iHec\_cs}} \quad ; \; P_{mawi\_iHec} = 268.7\,psi$$

UG-28 Thickness of Shells under External Pressure

External pressure, maximum: $P_{e\_iHec} := P_{max\_Hev} + 15psi \quad ; \; P_{e\_iHec} = 59\,psid$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{iHec}}{D_{i\_iHec}} = 2.6 \qquad \frac{D_{i\_iHec}}{t_{iHec}} = 43$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{iHec} := 0.0017$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we (try to) find the factor B, but since factor A falls to the left of the appropriate material curve, we must compute the allowable external pressure as follows (including the FNAL factor K_{sr}):

Modulus of Elasticity (Young's Modulus):

$$E_{iHec} := E_{Al} \quad ; \; E_{iHec} = 1 \times 10^{7}\,psi$$

Maximum allowable working external pressure on inner helium cylinder (maximum internal gauge pressure of helium vessel):

$$P_{mawe\_iHec} := \frac{2A_{iHec} \cdot E_{iHec} \cdot K_{sr}}{3\left(\frac{D_{i\_iHec}}{t_{iHec}}\right)} \quad ; \; P_{mawe\_iHec} = 211.7\,psi$$ (internal positive gauge pressure in Helium vessel)

$$; \; P_{e\_iHec} = 59\,psi$$

$P_{mawe\_iHec} > P_{e\_iHec}$ so the vessel is safe from buckling under maximum internal pressure in helium vessel

Helium Neck Tube

This is a thin tube of 304 S.S for providing a vent for quench condition. Some instrument cables also are routed through this tube. It is shown on dwg MICE-C009. Note that the OD and thickness have been changed from the drawing. A separate fill tube is used, for which a drawing is not yet available. The pressure relief valve and burst disk are located at room temperature at the end of this tube. There is a reinforced section near one end for a 77K intercept. The tube is fabricated by turning down a seamless thick walled tube using a close fitting mandrel, (not clear if a follow rest is used to assure uniform thickness).

$d_{o\_nt} := 25.4\text{mm}$

$l_{nt} := 0.3746\text{m}$ (longest unreinforced length)

$t_{nt} := .015\text{in}$ per conversation, S. Virostek, 1/18/08

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} := 30000\text{psi}$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} := 28.3\cdot10^{6}\text{psi}$ (Section II Part D, Subpart 3)

Maximum allowable stress, ASME: $S_{304} := 18800\text{psi}$ (Table ULT-23)

Maximum Allowable Stress: $S_{nt} := K_{sr}\cdot S_{304}$; $S_{nt} = 1.504\times10^{4}\,\text{psi}$ $\quad \nu_{304} := .28$

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_nt} := P_{max\_Hev} + 15\text{psi}$; $P_{i\_nt} = 59\,\text{psid}$

Minimum required thickness based on circumferential stress:

$$t_{min\_nt\_cs} := \frac{P_{i\_nt}\cdot d_{o\_nt}}{2S_{nt} - 0.6P_{i\_nt}} \quad ; \; t_{min\_nt\_cs} = 0.0499\,\text{mm}$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_nt\_ls} := \frac{P_{i\_nt}\cdot d_{o\_nt}}{4S_{nt} - 0.4P_{i\_nt}} \quad ; \; t_{min\_nt\_ls} = 0.0249\,\text{mm}$$

$t_{nt} = 0.381\,\text{mm}$

$t_{nt} > t_{min\_nt\_cs}$ therefore the Neck Tube of the Helium Vessel is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_nt} := P_{i\_nt}\cdot\frac{t_{nt}}{t_{min\_nt\_cs}} \quad ; \; P_{mawi\_nt} = 450.7\,\text{psid}$$

UG-28 Thickness of Shells under External Pressure

External pressure condition occurs when the Helium vessel is pumped down to vacuum prior to installation into vacuum vessel. For unanticipated leak checking after full assembly (with stabilized cryocooler sleeves; see below) the possibility of overpressure in the insulating space (up to the Vacuum Vessel burst disk pressure) cannot be discounted. So maximum external pressure could be as much as:

External pressure, maximum: $P_{e\_nt} := -P_{min\_Hev} + P_{bd\_vv}$; $P_{e\_nt} = 19.4\,\text{psid}$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{l_{nt}}{d_{o\_nt}} = 14.7 \qquad \frac{d_{o\_nt}}{t_{nt}} = 67$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{nt} := 0.00026$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we find the factor B :

$$B_{nt} := 3500\text{psi}$$

Maximum allowable working external pressure on neck tube (includes K_{sr}):

$$P_{mawe\_nt} := 4 \cdot \frac{B_{nt} \cdot K_{sr}}{3 \cdot \left(\frac{d_{o\_nt}}{t_{nt}}\right)} \qquad ;\ P_{mawe\_nt} = 56\,\text{psid} \qquad ;\ P_{e\_nt} = 19.4\,\text{psid}$$

(internal positive differential pressure in Helium vessel)

$$P_{mawe\_nt} > P_{e\_nt}$$

so the neck tube is safe from buckling under maximum external pressure in helium vessel, according to ASME Design Code. This condition could occur if the helium vessel is pumped down to vacuum while the insulating space is at atmospheric pressure (intentional or accidental).

<u>Helium Fill Tube</u>

This is a thin tube of 304 S.S leading to the bottom of the He Vessel for LHe fill. It is not shown in the final design report. There are several 90 deg. bends; we analyze here for the longest straight section:

$d_{o\_ft} := 36\text{mm}$ per S. Virostek, conversation 1-18-08

$l_{ft} := 0.5\text{m}$ (longest unreinforced length, est. from Final design ASME calc spreadsheet)

$t_{ft} := .035\text{in}$ per conversation, S. Virostek, 1/18/08

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} = 3 \times 10^{4}\,\text{psi}$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} = 2.83 \times 10^{7}\,\text{psi}$ (Section II Part D, Subpart 3)

Maximum allowable stress, ASME: $S_{304} = 1.88 \times 10^{4}\,\text{psi}$ (Table ULT-23)

Maximum Allowable Stress: $S_{ft} := K_{sr} \cdot S_{304}$; $S_{nt} = 1.504 \times 10^{4}\,\text{psi}$ $\qquad \nu_{304} = 0.28$

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_ft} := P_{max\_Hev} + 15\text{psi}$; $P_{i\_ft} = 59\,\text{psid}$

Minimum required thickness based on circumferential stress:

$$t_{min\_ft\_cs} := \frac{P_{i\_ft} \cdot d_{o\_ft}}{2S_{ft} - 0.6P_{i\_ft}} \qquad ;\ t_{min\_ft\_cs} = 0.0707\,\text{mm}$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_ft\_ls} := \frac{P_{i\_ft} \cdot d_{o\_ft}}{4S_{ft} - 0.4P_{i\_ft}} \qquad ;\ t_{min\_ft\_ls} = 0.0353\,\text{mm}$$

$$t_{ft} = 0.889\,\text{mm}$$

$t_{ft} > t_{min\_ft\_cs}$ therefore the Neck Tube of the Helium Vessel is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_ft} := P_{i\_ft} \cdot \frac{t_{ft}}{t_{min\_ft\_cs}} \qquad ;\ P_{mawi\_ft} = 741.9\,\text{psid} \qquad \text{OK}$$

UG-28 Thickness of Shells under External Pressure

External pressure condition occurs when the Helium vessel is pumped down to vacuum prior to installation into vacuum vessel. For unanticipated leak checking after full assembly (with stabilized cryocooler sleeves; see below) the possibility of overpressure in the insulating space (up to the Vacuum Vessel burst disk pressure) cannot be discounted. So maximum external pressure could be as much as:

External pressure, maximum: $P_{e\_ft} := -P_{min\_Hev} + P_{bd\_vv}$; $P_{e\_ft} = 19.4\,\text{psid}$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{l_{ft}}{d_{o\_ft}} = 13.9 \qquad \frac{d_{o\_ft}}{t_{ft}} = 40$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{ft} := 0.0007$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we find the factor B :

$$B_{ft} := 7600\text{psi}$$

Maximum allowable working external pressure on neck tube (includes K_{sr}):

$$P_{mawe\_ft} := 4 \cdot \frac{B_{ft} \cdot K_{sr}}{3 \cdot \left(\frac{d_{o\_ft}}{t_{ft}}\right)} \qquad ;\ P_{mawe\_ft} = 200.2\,\text{psid}$$

(internal positive differential pressure in Helium vessel)

$$;\ P_{e\_ft} = 19.4\,\text{psid}$$

$$P_{mawe\_ft} > P_{e\_ft}$$

so the fill tube is safe from buckling under maximum external pressure in helium vessel, according to ASME Design Code. This condition will occur when the helium vessel is pumped down to vacuum for leak checking

Cryocooler Sleeve, 300K-80K

This is a large diameter thin wall tube of 304 S.S for providing a low heat leak helium enclosure for the first stage of the cryocooler. It is made from seamless tubing so the FNAL weld factor does not apply. One end is at 300K and the other end at 80K (shield). It is shown on dwg. MICE-C[]04 (not a typo). Thus is is part of the helium vessel system and sees the same internal pressures.

$$d_{o\_sl\_300\_80} := 5.28\text{in}$$

$$l_{sl\_300\_80} := 6.435\text{in}$$

$$t_{sl\_300\_80} := 0.5\text{mm}$$

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} = 3 \times 10^4\ \text{psi}$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} = 2.83 \times 10^7\ \text{psi}$ (Section II Part D, Subpart 3)

Maximum allowable stress, ASME: $S_{304} = 1.88 \times 10^4\ \text{psi}$ (Table ULT-23)

Maximum Allowable Stress: $S_{sl} := K_{sr} \cdot S_{304}$; $S_{nt} = 1.504 \times 10^4\ \text{psi}$

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_sl} := P_{max\_Hev} + 15\text{psi}$; $P_{i\_sl} = 59\ \text{psid}$

Minimum required thickness based on circumferential stress:

$$t_{min\_sl\_300\_80\_cs} := \frac{P_{i\_sl} \cdot d_{o\_sl\_300\_80}}{2S_{sl} - 0.6P_{i\_sl}} \quad ; \ t_{min\_sl\_300\_80\_cs} = 0.2634\ \text{mm}$$

Minimum required thickness based on longitudinal stress:
Note: there is substantial longitudinal tensile stress present from the conical mating surface of the first stage compressor. This is a secondary stress under ASME code, but we add it in here:

$$F_{l\_sl\_300\_80} := 7000\text{N}$$

$$t_{min\_sl\_300\_80\_ls} := \frac{P_{i\_sl} \cdot d_{o\_sl\_300\_80}}{4S_{sl} - 0.4P_{i\_sl}} + \frac{F_{l\_sl\_300\_80}}{2\pi \cdot d_{o\_sl\_300\_80} \cdot S_{sl}} \quad ; \ t_{min\_sl\_300\_80\_ls} = 0.2117\ \text{mm}$$

$$t_{sl\_300\_80} = 0.5\ \text{mm}$$

$t_{sl} - 300\_80 > t_{min\_sl300\_80\_cs}$ therefore the Cryocooler Sleeve from 300K to 80K of the Helium Vessel is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_sl\_300\_80} := P_{i\_sl} \cdot \frac{t_{sl\_300\_80}}{t_{min\_sl\_300\_80\_cs}} \quad ; \ P_{mawi\_sl\_300\_80} = 112\ \text{psid}$$

UG-28 Thickness of Shells under External Pressure

This sleeve, though part of the helium vessel system, will be leak checked separately, after full assembly from the rest of the system, as it is too thin to withstand full vacuum external pressure. This can be done by removing the cryocoolers and pulling a vacuum in the insulating space. We analyze here to find out the limits of stability for other unanticipated conditions, e.g. a loss of insulating vacuum during a full system pumpdown.

External pressure, maximum: $P_{e\_sl} := P_{bd\_vv}$; $P_{e\_sl} = 4.4\ \text{psid}$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{l_{sl\_300\_80}}{d_{o\_sl\_300\_80}} = 1.2 \qquad \frac{d_{o\_sl\_300\_80}}{t_{sl\_300\_80}} = 268$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{sl\_300\_80} := 0.00025$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we find the factor:

$$B_{sl\_300\_80} := 3500\text{psi}$$

Maximum allowable working external pressure on cryocooler sleeve, 300-80K

$$P_{mawe\_sl\_300\_80} := \frac{\left(4B_{sl\_300\_80}\right)}{3\left(\frac{d_{o\_sl\_300\_80}}{t_{sl\_300\_80}}\right)} \qquad P_{mawe\_sl\_300\_80} = 17.4\,\text{psid}$$

(internal positive differential pressure in Helium vessel)

$$P_{e\_sl} = 4.4\,\text{psid}$$

$$P_{mawe\_nt} > P_{e\_nt}$$

this sleeve meet ASME code requirements for maximum external pressure, under the condition that leak checking is not performed by pulling a vacuum in the Helium Vessel. Mechanical stabilization can possibly utilized should the need arise to pull vacuum on the helium vessel in the assembled condition (with the cryocoolers removed).

Cryocooler Sleeve, 80K-4K

This is a medium diameter thin wall tube of 304 S.S for providing a low heat leak helium enclosure for the first stage of the cryocooler. One end is at 80K (shield)and the other end at 4K (coldmass). It is shown on dwg. MICE-C[]06 (not a typo). Thus it is part of the helium vessel system and sees the same internal pressures.

$$d_{o\_sl\_80\_4} := 3.730\text{in}$$

$$l_{sl\_80\_4} := 11.709\text{in}$$

$$t_{sl\_80\_4} := .015\text{in}$$

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} = 3 \times 10^4\,\text{psi}$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} = 2.83 \times 10^7\,\text{psi}$ (Section II Part D, Subpart 3)

Maximum allowable stress, ASME: $S_{304} = 1.88 \times 10^4\,\text{psi}$ (Table ULT-23)

Maximum Allowable Stress: ; $S_{sl} = 1.504 \times 10^4\,\text{psi}$

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_sl} = 59\,\text{psid}$

Minimum required thickness based on circumferential stress:

$$t_{min\_sl\_80\_4\_cs} := \frac{P_{i\_sl} \cdot d_{o\_sl\_80\_4}}{2S_{sl} - 0.6P_{i\_sl}} \quad ; \; t_{min\_sl\_80\_4\_cs} = 0.007\,\text{in}$$

Minimum required thickness based on longitudinal stress:
Note: Unlike the 300-80K sleeve there is no substantial longitudinal tensile stress present.

$$t_{min\_sl\_80\_4\_ls} := \frac{P_{i\_sl} \cdot d_{o\_sl\_80\_4}}{4S_{sl} - 0.4P_{i\_sl}} \quad ; \; t_{min\_sl\_80\_4\_ls} = 0.004\,\text{in}$$

$$t_{sl\_80\_4} = 0.015\,\text{in}$$

$t_{sl\_80\_4} > t_{min\_sl\_80\_4\_cs}$ therefore the Cryocooler Sleeve from 80K to 4K of the Helium Vessel is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_sl\_80\_4} := P_{i\_sl} \cdot \frac{t_{sl\_80\_4}}{t_{min\_sl\_80\_4\_cs}} \quad ; \; P_{mawi\_sl\_80\_4} = 120.8\,\text{psid}$$

UG-28 Thickness of Shells under External Pressure

This sleeve, though part of the helium vessel system, will be leak checked separately, after full assembly from the rest of the system, as it is too thin to withstand full vacuum external pressure. This can be done by removing the cryocoolers and pulling a vacuum in the insulating space. We analyze here to find out the limits of stability for other unanticipated conditions, e.g. a loss of insulating vacuum during a full system pumpdown.

External pressure, maximum: $P_{e\_sl} = 4.4\,\text{psi}$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{l_{sl\_80\_4}}{d_{o\_sl\_80\_4}} = 3.1 \qquad \frac{d_{o\_sl\_80\_4}}{t_{sl\_80\_4}} = 249$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{sl\_80\_4} := 0.0001$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we(try to) find the factor B, but since factor A falls to the left of the appropriate material curve, we must compute the allowable external pressure as follows (including the FNAL factor K_{sr}):

Modulus of Elasticity (Young's Modulus):

$$E_{sl} := E_{304} \qquad E_{sl} = 2.83 \times 10^{7}\,\text{psi}$$

Maximum allowable working external pressure on inner vacuum cylinder (maximum internal gauge pressure of vacuum vessel):

$$P_{mawe\_sl\_80\_4} := \frac{2A_{sl\_80\_4} \cdot E_{sl} \cdot K_{sr}}{3\left(\frac{d_{o\_sl\_80\_4}}{t_{sl\_80\_4}}\right)} \quad ; \; P_{mawe\_sl\_80\_4} = 6.1\,\text{psid}$$

$$P_{e\_sl} = 4.4\,\text{psid}$$

$P_{mawe\_sl\_80\_4} > P_{e\_sl}$ this sleeve meets ASME code requirements for maximum external pressure, provided leak checking is not performed by pulling a vacuum on the helium vessel. Mechanical stabilization can possibly utilized should the need arise to pull vacuum on the helium vessel in the assembled condition (with the cryocoolers removed).

3. Vacuum Vessel

The Vacuum Vessel consists of a large diameter Outer Vacuum Cylinder, a small diameter Inner Vacuum Cylinder, connected to each other by flat endplates. All parts are made from 304 stainless steel and the cylinders are welded to the endplates with vacuum tight fillet welds. It contains the Helium Vessel and provides a vacuum insulated space for the shield and MLI.

Conditions

$P_{max\_vv} := 4.4psig$ (from Technical Specification 10154)

$P_{min\_vv} := -15psig$ (external pressure from vacuum)

Note: WANG report (chap III) simply states: "(vacuum) vessel will be tested to satisfy ASME pressure vessel code to sustain a pressure rating of 1 atmosphere pressure"

Drawings do not show a pressure relief valve and/or burst disk on the vacuum vessel (however a possible vent stub has been penned in on the overall assembly drawing in the vendor design report). Technical Specification 10154 calls for a 1" dia. relief valve set to +2 psid and a burst disk set to 4.4 psid.

Outer Vacuum Cylinder

This cylinder is rolled and welded 3/4" thick 304 S.S. and has penetrations and intermediate reinforcing ribs. It is welded on each end to the endplates. It is subject to both external pressure and longitudinal compression from vacuum. It might be subject to internal pressure, should a rupture of the He vessel system occur.

Dimensions (from dwgs. MICE-3100 and MICE-3110):

wall thickness: $t_{ovc} := 19mm$

length: $L_{ovc} := [2.923 - (.207 + .019 + .003 + .003)]m$; $L_{ovc} = 2.691\,m$ $L_{ovc} = 105.9449\,in$

inner radius: $R_{i\_ovc} := 0.683m$

outer radius: $R_{o\_ovc} := R_{i\_ovc} + t_{ovc}$; $R_{o\_ovc} = 0.702\,m$

inner diameter: $D_{i\_ovc} := 2R_{i\_ovc}$; $D_{i\_ovc} = 1.366\,m$

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} = 3 \times 10^4\,psi$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} = 2.83 \times 10^7\,psi$ (Section II Part D, Subpart 3)

ASME Maximum allowable Stress: $S_{304} = 1.88 \times 10^4\,psi$ (Section VIII, Table ULT-23)

Maximum Allowable Stress: $S_{ovc} := K_{sr} \cdot S_{304}$; $S_{ovc} = 1.504 \times 10^4\,psi$

Weld joint efficiency: $E_j = 0.6$ (Section VIII, Table UW-12)

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum (worst case, under quench and ruptured burst disk conditions):

$P_{i\_ovc} := P_{bd\_vv}$; $P_{i\_ovc} = 4.4\,psi$

Minimum required thickness based on circumferential stress:

$$t_{min\_ovc\_cs} := \frac{P_{i\_ovc} \cdot D_{i\_ovc}}{2S_{ovc} \cdot E_j - 0.6P_{i\_ovc}} ; \; t_{min\_ovc\_cs} = 0.3331\,mm$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_ovc\_ls} := \frac{P_{i\_ovc} \cdot D_{i\_ovc}}{4S_{ovc} \cdot E_j - 0.4P_{i\_ovc}} \;;\; t_{min\_ovc\_ls} = 0.1665\,mm$$

$$;\; t_{ovc} = 19\,mm$$

$t_{ovc} > t_{min\_ovc\_sc}$ therefore the outer vacuum cylinder is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_ovc} := P_{i\_ovc} \cdot \frac{t_{ovc}}{t_{min\_ovc\_cs}} \;;\; P_{mawi\_ovc} = 251\,psi$$

UG-28 Thickness of Shells under External Pressure

External pressure, maximum: $P_{e\_ovc} := -P_{min\_vv}$; $P_{e\_ovc} = 15\,psi$

The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{ovc}}{D_{i\_ovc}} = 1.97 \qquad \frac{D_{i\_ovc}}{t_{ovc}} = 71.8947$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{ovc} := 0.001$$

Using the factor A in the applicable material (304 S.S.) chart (HA-1) in Subpart 3 of Section II, Part D, we find the factor B:

$$B_{ovc} := 9000psi$$

The maximum allowable working external pressure is then given by (including the FNAL factor K_{sr}):

$$P_{mawe\_ovc} := \frac{4B_{ovc} \cdot K_{sr}}{3\left(\frac{D_{i\_ovc}}{t_{ovc}}\right)} \;;\; P_{mawe\_ovc} = 133.5\,psi$$

$P_{mawe\_ovc} > P_{e\_ovc}$ so the outer vacuum cylinder is safe from buckling under vacuum load

Note: forces from cold mass support struts probably do not have much effect on buckling strength as they would quickly relax for any radial inward motion. In addition, they might stabilize the cylinder against some mode shapes, while allowing others. Only a proper modal analysis can quantify this with certainty.

UG-36 Openings in Pressure Vessels

Subsection (c)(3)(a) of UG-36 allows isolated unreinforced openings of 2 3/8" (60mm) or less, in finished diameter, for shell thicknesses t> 3/8", given no rapidly fluctuating loads. There are five holes in the Outer Vacuum Cylinder, three rectangular openings for the cryocooleres, and two round holes for the fill and vent lines two for the leads) which are larger than 60mm in diameter (or width). Therefore reinforcement is necessary. This reinforcement may be in excess vessel wall thickness (no additional reinforcements are present in the design).

UG-37 Reinforcement required for Openings in Shells and Formed Heads

The holes have bimetallic Al/S.S. junctions welded in them which can serve as reinforcement. In addition, there is excess shell thickness (over and above that required per section 26 and 27, which also counts as reinforcement. The weld design appears similar to that shown in figs. UG-40 (a-3) and UW-16.1 (v-2) except that a full penetration weld with a beveled weld prep is not called for on the drawing; it seems that a fillet weld without a weld groove is more likely to be what gets made. This would not be in compliance with ASME code for reinforcements.

For:

required minimum thickness for seamless vessel with no openings, using weld efficiency E_j=1:

$$t_{r\_ovc} := \frac{P_{i\_ovc} \cdot R_{i\_ovc}}{S_{ovc} - 0.6P_{i\_ovc}} \quad ; \quad t_{r\_ovc} = 0.2\,\text{mm}$$

stress variation factor (circumferential to longitudinal)

$F_{sv} = 1$

strength reduction factors for differing material strengths (assume all strengths equal)

$f_{r1} = 1 \qquad f_{r3} = 1 \qquad f_{r4} = 1$

opening weld joint efficiency (for opening passing through weld)

$E_1 = 1$

and the following dimensions as per fig. UG-37.1 using the largest hole (for leads):

finished hole (nozzle) diameter (inside); radius

$d_{h\_l} := 402.5\text{mm} \qquad R_{n\_l} := 0.5 \cdot d_{h\_l} \; ; \quad R_{n\_l} = 201.25\,\text{mm}$

nozzle thickness (dwg. MICE-C008)

$t_{n\_l} := 0\text{mm}$

No reinforcements are added (tower does not count, as it lies outside the limits of reinforcement)

$t_{e\_l} := 0\text{mm}$

Area of reinforcement required for lead opening:

$$A_{rnf\_ovc\_l} := d_{h\_l} \cdot t_{r\_ovc} \cdot F_{sv} + 2t_n \cdot t_{r\_ovc} \cdot F_{sv} \cdot (1 - f_{r1})$$

$$A_{rnf\_ovc\_l} = 80.4\,\text{mm}^2$$

UG-40 Limits of Reinforcement

These define the allowable dimensional extents for any reinforcement (see fig UG-37.1)

(b) parallel to vessel wall. Use greater of the following:

$L_{1a\_l} := d_{h\_l} \quad ; \; L_{1a\_l} = 402.5\,\text{mm}$ <<<

$L_{1b\_l} := R_n + t_{ovc} + t_{n\_l} \quad ; \; L_{1b\_l} = 37\,\text{mm}$

(c) perpendicular to vessel wall. Use smaller of the following:

$L_{2a\_l} := 2.5 \cdot t_{ovc} \quad ; \; L_{2a\_l} = 47.5\,\text{mm}$

$L_{2b\_l} := 2.5 \cdot t_{n\_l} + t_{e\_l} \quad ; \; L_{2b\_l} = 0\,\text{mm}$ <<<

Available areas for reinforcement, per Fig UG-37.1 :

in excess wall thickness, use larger of (t_{n_l}=0):

$A_{1a\_ovc\_l} := d_{h\_l} \cdot (E_1 \cdot t_{ovc} - f_{r1} \cdot t_{r\_ovc}) - 0 \quad ; \; A_{1a\_ovc\_l} = 7567\,\text{mm}^2$ <<<

$A_{1b\_ovc\_l} := 2 \cdot (t_{ovc} + t_{n\_l}) \cdot (E_1 \cdot t_{ovc} - f_{r1} \cdot t_{r\_ovc}) - 0 \quad ; \; A_{1b\_ovc\_l} = 714.4\,\text{mm}^2$

Total area of reinforcement:

$A_{t\_ovc\_l} := A_{1a\_ovc\_l}$; $A_{t\_ovc\_l} = 7567\,mm^2$;

compare with required area>> $A_{rnf\_ovc\_l} = 80.4\,mm^2$

$A_{t\_ovc\_l} > A_{rnf\_ovc\_l}$ so these openings have sufficient reinforcement area (in excess vessel wall thickness).

UG-42 Reinforcement of Multiple Openings

For multiple openings whose centers are spaced less than twice the average of their diameters, this section describes additional requirements. The cryocooler openings have a section between them of width:

$w_l := 1249.9mm - (747.4 + 402.5)mm$ $w_l = 0.1\,m$ which is less than twice the average of their diameters

Available reinforcement area:

$A_{a\_ovc\_l} := w_l \cdot (E_1 \cdot t_{ovc} - f_{r1} \cdot t_{r\_ovc})$ $A_{a\_ovc\_l} = 1880\,mm^2$

It is required that half the required reinforcement area be present on each side of each opening. For the large lead hole we need:

$A_{r\_l} := 0.5 \cdot A_{rnf\_ovc\_l}$ $A_{r\_l} = 40.2196\,mm^2$

and for the adjacent cryocooler opening of width: $d_{h\_c} := 235.0mm$

$A_{r\_c} := A_{r\_l} \cdot \frac{d_{h\_c}}{d_{h\_l}}$ $A_{r\_c} = 23.4822\,mm^2$

$A_{a\_ovc\_l} > A_{r\_l} + A_{r\_c}$ OK, multiple openings for leads and cryocoolers in outer vacuum cylinder have sufficient reinforcement area between them in excess wall thickness.

Inner Vacuum Cylinder

This simple cylinder is 304 S.S., 3mm thick (assumed welded) and has no penetrations or intermediate reinforcing ribs. It is welded on each end to the endplates. It is subject to both internal pressure and longitudinal compression. It might be subject to external pressure, should a rupture of the He vessel system occur.

Dimensions (from dwgs. MICE-3100 and MICE-3400):

wall thickness: $t_{ivc} := 3.0\text{mm}$

length: $L_{ivc} := L_{ovc}$; $L_{ivc} = 2.691\,\text{m}$

inner radius: $R_{i\_ivc} := 0.2005\text{m}$

outer radius: $R_{o\_ivc} := R_{i\_ivc} + t_{ivc}$; $R_{o\_ivc} = 0.2035\,\text{m}$

inner diameter $D_{i\_ivc} := 2R_{i\_ivc}$; $D_{i\_ivc} = 0.401\,\text{m}$

Material: 304 S.S. , welded. Yield strength: $S_{y\_304} = 3.0\times 10^{4}\,\text{psi}$ (Section II Part D,Table 2A)

Young's modulus: $E_{304} = 2.83\times 10^{7}\,\text{psi}$ (Section II Part D)

ASME Maximum allowable Stress: $S_{304} = 1.88\times 10^{4}\,\text{psi}$ (Section II Part D,Table 2A)

Maximum allowable Stress: $S_{ivc} := K_{sr}\cdot S_{304}$; $S_{ivc} = 1.504\times 10^{4}\,\text{psi}$ (Section VIII, Table ULT-23)

Weld joint efficiency: $E_j = 0.6$ (Section VIII, Table UW-12)

Axial compression

In addition to external pressure, there is an axial compressive load on the thin inner vacuum shell from the inner periphery of the end plates. Roark's Formulas for Stress and Strain give the following reactions for a uniform pressure load distribution:

Unit load distribution (force per unit length of inner radius):

$Q_b := K_Q\cdot q\cdot a$ $a_{vv} := R_{o\_ovc}$ $q_{vv} := P_{max\_vv}$ $b_{vv} := R_{o\_ivc}$

for: $\dfrac{b_{vv}}{a_{vv}} = 0.29$ $K_{Q\_vv} := 0.6$ (Roark's Formulas for Stress and Strain, 6th ed. Table 24, case 2c)

$Q_{b\_vv} := K_{Q\_vv}\cdot q_{vv}\cdot a_{vv}$; $Q_{b\_vv} = 73\,\dfrac{\text{lbf}}{\text{in}}$

Compressive stress:

$$\sigma_{c\_ivc} := \frac{Q_{b\_vv}}{t_{ivc}} \qquad \sigma_{c\_ivc} = 617.76\,\text{psi}$$

UG-23b Maximum Allowable Longitudinal Compressive Stress

This section covers stress from longitudinal compression; under subsection b) first compute:

$$A := \frac{0.125}{\left(\frac{R_{o\_ivc}}{t_{ivc}}\right)} \qquad A = 1.8428\times 10^{-3}$$

On chart HA-1, find, for the above value of A :

$B := 11500\text{psi}$

check:

$\sigma_{c\_ivc} < B$ OK

Factor of safety:

$$FS_{c\_ivc} := \frac{B}{\sigma_{c\_ivc}} \quad ; \quad FS_{c\_ivc} = 18.6$$ OK

As a comparison, from Roark's Formulas: $E_{ivc} := E_{304} \quad E_{ivc} = 2.83 \times 10^{7}\,psi \quad \nu_{ivc} := 0.3$

Buckling stress limit:

$$\sigma_{cr\_ivc} := \frac{1}{\sqrt{3}} \cdot \frac{E_{ivc}}{\sqrt{1 - \nu_{ivc}^{2}}} \cdot \frac{t_{ivc}}{R_{o\_ivc}} \quad \sigma_{cr\_ivc} = 2.525 \times 10^{5}\,psi$$

Roark's Formulas for Stress and Strain, 6th ed. table 35, case 15

Factor of Safety:

$$FS_{ab\_ivc} := \frac{\sigma_{cr\_ivc}}{\sigma_{c\_ivc}} \quad FS_{ab\_ivc} = 409$$

this gives an indication of how conservative the ASME code is regarding longitudinal compression

UG-27 Thickness of shells under Internal Pressure

Internal pressure, maximum: $P_{i\_ivc} := -P_{min\_vv}$; $P_{i\_ivc} = 15\,psi$

Minimum required thickness based on circumferential stress:

$$t_{min\_ivc\_cs} := \frac{P_{i\_ivc} \cdot D_{i\_ivc}}{2S_{ivc} \cdot E_{j} - 0.6P_{i\_ivc}} \quad ; \quad t_{min\_ivc\_cs} = 0.3334\,mm$$

Minimum required thickness based on longitudinal stress:

$$t_{min\_ivc\_ls} := \frac{P_{i\_ivc} \cdot D_{i\_ivc}}{4S_{ivc} \cdot E_{j} - 0.4P_{i\_ivc}} \quad ; \quad t_{min\_ivc\_ls} = 0.1667\,mm$$

; $t_{ivc} = 3\,mm$

$t_{ivc} > t_{min\_ivc\_cs}$ therefore the inner vacuum cylinder is safe from maximum possible internal pressure.

Maximum allowable working internal pressure loading:

$$P_{mawi\_ivc} := P_{i\_ivc} \cdot \frac{t_{ivc}}{t_{min\_ivc\_cs}} \quad ; \quad P_{mawi\_ivc} = 135\,psi$$

UG-28 Thickness of Shells under External Pressure

External pressure, maximum: $P_{e\_ivc} := P_{bd\_vv}$; $P_{e\_ivc} = 4.4\,psi$

The maximum allowable working external pressure is determined by the following procedure:

Determine the following two dimensionless constants:

$$\frac{L_{ivc}}{2R_{o\_ivc}} = 6.6 \qquad \frac{2R_{o\_ivc}}{t_{ivc}} = 136$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor:

$$A_{ivc} := 0.00012$$

Using the factor A in the applicable material (304) chart (HA-1) in Subpart 3 of Section II, Part D, we (try to) find the factor B, but since factor A falls to the left of the appropriate material curve, we must compute the allowable external pressure as follows (including the FNAL factor K_{sr}):

Modulus of Elasticity (Young's Modulus):

$$E_{ivc} = 2.83 \times 10^{7}\,\text{psi}$$

Maximum allowable working external pressure on inner vacuum cylinder (maximum internal gauge pressure of vacuum vessel):

$$P_{mawe\_ivc} := \frac{2A_{ivc}\cdot E_{ivc}\cdot K_{sr}}{3\left(\frac{D_{i\_ivc}}{t_{ivc}}\right)} \quad ; \quad P_{mawe\_ivc} = 13.6\,\text{psi} \qquad P_{e\_ivc} = 4.4\,\text{psi}$$

(internal positive gauge pressure in vacuum vessel)

$P_{mawe\_ivc} > P_{e\_ivc}$ OK

Endplates, vacuum vessel

These are simple single thickness endplates, one front and one back endplate of same dimensions. They are welded to the inner and outer vacuum cylinders. There are two screw hole patterns which slightly reduce the available cross sectional area (by a few percent); we assume these are negligible if a substantial safety factor exists.

Dimensions (from dwgs. MICE-3200 and MICE-3300):

wall thickness: $t_{vep} := 19.1\text{mm}$

inner diameter: $D_{i\_vep} := 0.407\text{m}$

outer diameter: $D_{o\_vep} := 1.362\text{m}$

Material: 304 S.S. plate, welded at both periphery (weld joint efficiency may be unity for these plates, but we use 0.6 here.

UG-34 Unstayed Flat heads and Covers

Minimum thickness is given by:

$$t_{min\_fhc} := d\cdot\sqrt{\frac{C\cdot p}{S\cdot E_j}}$$

where :

Internal pressure, maximum: $P_{i\_vep} := -P_{min\_vv}$; $P_{i\_vep} = 15\,\text{psi}$

head attachment factor: $C_{vep} := 0.33$ (figs UG-34 e-i)

Stress limit: $S_{vep} := K_{sr}\cdot S_{304}$ $\quad S_{vep} = 1.504 \times 10^{4}\,\text{psi}$

Weld joint efficiency: $E_j = 0.6$ (welds are considered part of shell)

$$t_{min\_vep} := D_{i\_vep}\cdot\sqrt{\frac{C_{vep}\cdot P_{i\_vep}}{S_{vep}\cdot E_j}} \qquad t_{min\_vep} = 9.5\,\text{mm}$$

Factor of safety:

Since plate stiffness constant D, is proportional to t^3 :

$$FS_{vep} := \left(\frac{t_{vep}}{t_{min\_vep}}\right)^{3} \qquad FS_{vep} = 8$$

OK, bolt holes negligible

4. Pressure Relief Devices

Fire

From NBS-111, Technology of Liquid Helium , pages 268-269, the following procedure (from CGA S-1.3, section 5.3) modified here for helium is:

Where :

Gas factor for insulated container of LHe:

$G_i := 52.5$

Total outside surface area of vacuum Vessel (not including bore) $L_{ovc} = 2.691\,m$

$A_{vv} := L_{ovc} \cdot 2\pi \cdot R_{o\_ovc} + 2\pi\left(R_{o\_ovc}^{\;2} - R_{o\_ivc}^{\;2}\right)$

$A_{vv} = 158.2901\,ft^2$

Correction factor for long relief manifolds

$F := 1$

Thermal conductivity of Helium at STP:

We consider the possibility exists that a fire condition is accompanied by (and may even be caused by) some other abnormal condition, such as earthquake, tornado, terrorist attack, etc., which has cause the He vessel to rupture internally with the result that He gas has filled the insulating space. (He has a higher thermal conductivity than air at STP).

$k_{He\_STP} := 0.0871 \dfrac{BTU}{hr \cdot ft \cdot R}$ $\qquad k_{He\_STP} = 0.1507 \dfrac{W}{m \cdot K}$

for:

$R_{i\_ovc} = 0.683\,m$; $R_{o\_Hev} := 0.3630m$

Insulation thickness is:

$x_{ins} := R_{i\_ovc} - R_{o\_Hev}$; $x_{ins} = 0.32\,m$ (we ignore the end area which has reduced insulation thickness)

Heat transfer coefficient is:

$U := \dfrac{k_{He\_STP}}{x_{ins}}$ $\qquad U = 0.083 \dfrac{BTU}{hr \cdot ft^2 \cdot R}$

Minimum flow capacity (in units of SCFM air) is then given as:

$Q_a := F \cdot G_i \cdot U \cdot A^{0.82}$ (SCFM air)

$Q_{a\_f} := 1 \cdot (52.5) \cdot (158) \cdot (0.078)$

$Q_{a\_f} = 647$ SCFM (air) (Note that this method gives the resulting flow rate for air, even though He is the gas being vented. Further below, we calculate burst disk flow for He, then convert result to equivalent air flow rate)

Loss of vacuum

$Q_a := 4.5 U \cdot A$ from NBS-111, pg. 269

$Q_{a\_lv} := 4.5 \cdot 0.078 \cdot 158$

$Q_{a\_lv} = 55.5$ SCFM (air)

Sizing of Helium Vessel Rupture Disk

The helium Vessel has a 1" dia. rupture disk set to open at 32-44 psid (positive pressure)

$d_{vent\_Hev} := 25mm$ from spec 10154

From Fike technical bulletin TB8102, eq (11):

$$Q_s := \frac{22772 \cdot a \cdot K \cdot C_2 \cdot P_o}{\sqrt{(t + 460)M}}$$

atomic weight: $M_{He} := 4.0 \frac{gm}{mole}$

temperature: $t := 20R$

$C_2 := 0.1048$

$K := 0.62$

$P_0 := 44psig + 15psi$ $P_0 = 59\,psia$

$a_{vent\_Hev} := \frac{\pi}{4} d_{vent\_Hev}^{\ 2}$; $a_{vent\_Hev} = 0.7609\,in^2$

$$Q_{s\_Hev} := \frac{22772 \cdot (0.761) \cdot (0.62) \cdot (0.1048) \cdot (59)}{\sqrt{(20)4.0}}$$

$Q_{s\_Hev} = 7428$ SCFM (He)

converting He flow rate to an equivalent air flow rate

$$Q_{air} := Q_{He} \cdot \sqrt{\frac{M_{he}}{M_{air}}}$$ $M_{air} := 29 \frac{gm}{mole}$

$$Q_{air\_Hev} := Q_{s\_Hev} \cdot \sqrt{\frac{M_{He}}{M_{air}}}$$

$Q_{air\_Hev} = 2759$ SCFM (air) compare to required ---> $Q_{a\_f} = 647$ SCFM (air)

compare to required ---> $Q_{a\_lv} = 55.5$ SCFM (air)

$Q_{air\_Hev} > Q_{a\_f} > Q_{a\_lv}$ OK

Sizing of Vacuum Vessel Rupture Disk

The Vacuum Vessel has a 1" dia. rupture disk set to open at 4.4 psid (positive pressure in vacuum space). We assume that any overpressure is due to rupture of either the He vessel system or the LN2 shield cooling lines.

$d_{vent\_vv} := 25mm$ (from Specification 10154)

From Fike technical bulletin TB8102, eq (11):

$$Q_s := \frac{22772 \cdot a \cdot K \cdot C_2 \cdot P_o}{\sqrt{(t + 460)M}}$$

for He

$M_{He} = 4\,\frac{gm}{mole}$

$t_{.} := 20R$

$C_2 = 0.1048$

$K = 0.62$

$P := P_{max\_vv} + 15psi \qquad P = 19.4\,psi$

$a_{vent\_vv} := \frac{\pi}{4} d_{vent\_vv}^{\ 2} \quad ; \quad a_{vent\_vv} = 0.7609\,in^2$

$$Q_{s\_vv} := \frac{22772\cdot(0.761)\cdot(0.62)\cdot(0.1048)\cdot(19.4)}{\sqrt{(20)4.0}}$$

$Q_{s\_vv} = 2442$ SCFM (He)

converting He flow rate to an equivalent air flow rate

$Q_{air} := Q_{He}\cdot\sqrt{\frac{M_{He}}{M_{air}}} \qquad M_{air} = 29\,\frac{gm}{mole}$

$Q_{air\_vv\_He} := Q_{s\_vv}\cdot\sqrt{\frac{M_{He}}{M_{air}}}$

$Q_{air\_vv\_He} = 907$ SCFM (air)

for N2:

$M_{N2} := 28\,\frac{gm}{mole}$

$t_{i2} := 77K \quad ; \quad t_{i2} = 138.6\,R$

$C_2 = 0.1048$

$K = 0.62$

$P = 19.4\,psi$

$a_{vent\_vv} = 0.7609\,in^2$

$$Q_{s\_vv\_N2} := \frac{22772\cdot(0.761)\cdot(0.62)\cdot(0.105)\cdot(19.4)}{\sqrt{(138)28}}$$

$Q_{s\_vv\_N2} = 352$ SCFM (N2)

converting N2 flow rate to an equivalent air flow rate

$Q_{air} := Q_{He}\cdot\sqrt{\frac{M_{N2}}{M_{air}}} \qquad M_{air} = 29\,\frac{gm}{mole}$

$Q_{air\_vv\_N2} := Q_{s\_vv\_N2}\cdot\sqrt{\frac{M_{N2}}{M_{air}}}$

$Q_{air\_vv\_N2} = 346$ SCFM (air)

It is not clear what the required vent rate would be for a rupture of either the He or LN2 system inside the insulating space.

Neck Tube Flow Capacity

We calculate the He gas flow rate for a modest pressure drop of several psi

for :

$\Delta P_{nt} := 5\,\text{psi}$ allowable pressure drop

$P_r := 29\,\text{psi}$ relief valve pressure

$S_{g\_He} := .1381$

$T_r := 20\,R$ average quench gas temp

$f_f := .02$ from chart on pg 3-19 of Crane TP-410M

$K_{entrance} := 0.5$

$LovrD := \dfrac{l_{nt}}{d_{i\_nt}}$ $LovrD = 15.2042$ $d_{i\_nt} := d_{o\_nt} - 2t_{nt}$

$K_{check} := 55 f_f$

$K_{exit} := 1$

$K_{tot} := K_{entrance} + f_f \cdot LovrD + K_{check} + K_{exit}$ $K_{tot} = 2.9041$

$Y := .7$ from chart A-22 of Crane TP-410M

$d_i := 1.5$ in.

$$q_{He} := 40700 \cdot Y d_i^{\,2} \sqrt{\frac{\Delta P_{nt} \cdot P_r}{K_{tot} \cdot T_r \cdot S_{g\_He}}}$$; $q_{He} = 2.7255 \times 10^{[illegible]}$ SCFM eq 3-20 Crane TP-410 (english units)

in SI: $q_{He\_b} := q_{He} \cdot \dfrac{ft^3}{min}$; $q_{He\_b} = 129\,\dfrac{m^3}{s}$

equivalent air flow

$$q_{air} := q_{He} \cdot \sqrt{\frac{M_{He}}{M_{air}}}$$; $q_{air} = 1.0122 \times 10^{[illegible]}$ SCFM

at working pressure: $q_{He\_b} \cdot \dfrac{14.7\,psi}{P_{rv\_hev}} = 65\,\dfrac{m^3}{s}$

5. Summary of Calculations (controlling parameters)

2. Helium Vessel

Outer Helium cylinder: thickness required: $t_{min\_oHec\_cs} = 7.2\,mm$
thickness available: $t_{oHec} = 12.7\,mm$ OK

Inner Helium Cylinder: thickness required: $t_{min\_iHec\_cs} = 0.67\,mm$
thickness available: $t_{iHec} = 12\,mm$ OK
opening reinforcements adequate

Neck Tube (.015" thk.): MAWEP (maximum allowable working external pressure): $P_{mawe\_nt} = 56\,psi$
MWOP (maximum working external pressure): $P_{e\_nt} = 19.4\,psi$ OK

Cryocooler sleeves

300K-80K: MAWEP: $P_{mawe\_sl\_300\_80} = 17.4\,psid$
MWOP: $P_{e\_sl} = 4.4\,psid$ OK, must be stabilized for any low pressure condition in He Vessel (P>15psia)

80K-4K: MAWEP: $P_{mawe\_sl\_80\_4} = 6.1\,psid$
MWOP: $P_{e\_sl} = 4.4\,psid$ OK, must be stabilized for any low pressure condition in He Vessel (P>15psia)

3. Vacuum Vessel:

Outer Vacuum Cylinder: thickness required: $t_{min\_ovc\_cs} = 0.33\,mm$
thickness available: $t_{ovc} = 19\,mm$ OK
opening reinforcements adequate

Inner Vacuum Cylinder: thickness required: $t_{min\_ivc\_cs} = 0.3334\,mm$
thickness available: $t_{ivc} = 3\,mm$ OK

MAWEP: $P_{mawe\_ivc} = 13.6\,psi$
MWOP: $P_{e\_ivc} = 4.4\,psi$ OK

Vacuum Endplates: thickness required: $t_{min\_vep} = 9.5323\,mm$
thickness available: $t_{vep} = 19.1\,mm$ OK

4. Relief Devices

Helium vessel burst disk: flow rate required, fire condition: $Q_{a\_f} = 647$ SCFM (equiv. air)
flow rate required, loss of insulating vacuum condition: $Q_{a\_lv} = 55$ SCFM (equiv air)
flow rate available: $Q_{air\_Hev} = 2759$ SCFM (equiv air) OK

Vacuum vessel burst disk: flow rate available, He: $Q_{air\_vv\_He} = 907$ SCFM (equiv air)
flow rate available, N2: $Q_{air\_vv\_N2} = 346$ SCFM (equiv air)

6. Appendix

a. Helium quench pressure estimate (very simple, and possibly not conservative):

from M.A.Green corresp. 12/13/07:

areal heat transfer coefficient from cold mass into helium:

$h_{film\_boil} := 670 \frac{W}{m^2 K}$ for film boiling (at atmospheric pressure)

$h_{free\_conv} := \frac{1}{45} h_{film\_boil}$ $h_{free\_conv} = 14.9 \frac{W}{m^2 K}$ for free convection with no boiling (at supercritical pressure of 2.2 bar)

Maximum cold mass temperature after quench initiation:

$\Delta T_{max\_quench} := 75K$ note: this is higher than the uniform value calculated below:

Heat transfer area available (use average of inner and outer helium cylinder radii):

$A_{ht} := \pi(R_{o\_iHec} + R_{i\_oHec}) \cdot 1.93m$; $A_{ht} = 3.6774\,m^2$ $R_{o\_iHec} = 0.257\,m$ $R_{i\_oHec} = 0.3495\,m$

Heat flow into helium:

$P_{fc} := h_{free\_conv} \cdot A_{ht} \cdot \Delta T_{max\_quench}$; $P_{fc} = 4.1064\,kW$

helium heat of vaporization:

$h_{vaporization\_He} := 82.9 \frac{J}{mole}$

Resulting mass flow of He vapor: $M_{He} = 4 \frac{gm}{mole}$

$M_{vap} := \frac{P_{fc} \cdot M_{He}}{h_{vaporization\_He}}$; $M_{vap} = 0.1981 \frac{kg}{s}$

specific volume of He vapor (from NBS-111, TABLE 2.1):

$v_{He\_29psia} := 25.67 \frac{cm^3}{gm}$ @5K 1.96 atm

Volumetric vapor flow:

$V_{vap} := M_{vap} \cdot v_{He\_29psia}$; $V_{vap} = 0.0051 \frac{m^3}{s}$

Flow velocity: ; $d_{i\_nt} = 24.638\,mm$

$v_{quench} := \frac{V_{vap}}{\frac{\pi}{4} \cdot d_{i\_nt}^2}$; $v_{quench} = 11 \frac{m}{s}$

Pressure drop through neck tube

$\rho_{He} := v_{He\_29psia}^{-1}$ $f_f = 0.02$

$\Delta P_{nt\_quench} := \frac{1}{2} \cdot f_f \cdot \rho_{He} \cdot \frac{l_{nt}}{d_{i\_nt}} \cdot v_{quench}^2$ (not including entrance and exit losses)

$\Delta P_{nt\_quench} = 0.0978\,psi$ negligible pressure drop across neck tube

b. Coil temp rise (for quench) estimate (assume all stored energy dissipated instantaneously into coil masses)

Volume of coils (from Final Design Report, table I-1):

$V_{coils} := 2\pi(44.7 \cdot 280 \cdot 201 + 29.8 \cdot 273 \cdot 199.5 + 59.6 \cdot 288 \cdot 110.6 + 21.3 \cdot 268 \cdot 1314.3 + 63.9 \cdot 290 \cdot 110.6)\,mm^3$

$V_{coils} = 0.098\,m^3$

Mass of coils (assume all copper, at 60% fill ratio)

$M_{mag} := 0.6 V_{coils} \cdot \rho_{Cu}$ $\qquad \rho_{Cu} := 8890 \dfrac{kg}{m^3}$

$M_{mag} = 522.5\,kg$

Stored magnetic energy of coils

$U_{mag2} := (0.4 + 0.3 + 0.48 + 1.83 + 0.59) \cdot 10^6 J$ from Final Design Report, table I-1

$U_{mag2} = 3.6 \times 10^6 J$

Energy density

$u_{mag2} := \dfrac{U_{mag2}}{M_{mag}}$; $u_{mag2} = 6.89 \dfrac{kJ}{kg}$

Temperature rise $\qquad c_{P\_Cu} := 0.385 \dfrac{J}{gm \cdot K}$

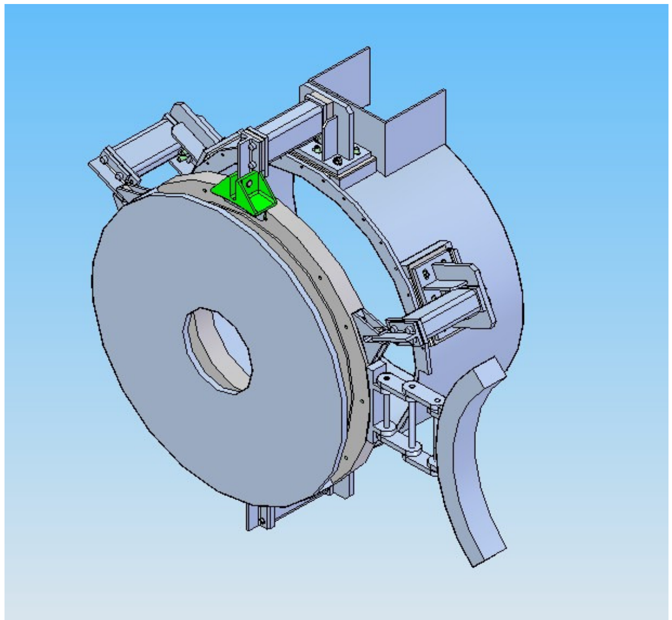
$\Delta T_{mag} := \dfrac{u_{mag2}}{c_{P\_Cu}}$; $\Delta T_{mag} = 17.9\,K$ this temperature rise is much less than the maximum temp rise estimated above (75K), so the above pressure drop calculation can be considered to be conservative.

c. TOF Shielding Support Stress Analysis:

The following 4 pages provide a summary of the TOF shielding support stress analysis.

SPECTROMETER SOLENOID IRON SHIELD SUPPORT

The support for the TOF, cage and shield have been re-designed to take account of the manufacturing method of the solenoid vessel i.e. non-machined interfaces for the support.



TD-1129-6010

Each support position is identical except for the spacer thicknesses introduced between the mating faces. Any one of the supports can carry the total weight of the TOF, cage and shield. The all up weight of the TOF, cage and shield is taken as 3tonnes. Any three supports can carry the axial magnetic force of 10tonnes.

The beam section of a support is 100mm x 150mm x 10mm wall thickness rectangular hollow section and is 370mm long, the material is 304L stainless steel.(I x-x = 1310cm4)

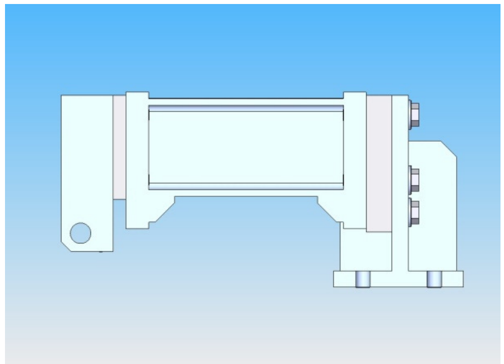
With spacers the beam can be considered as a cantilever 400mm long:-

Max BM = WL = 3000Kg x 10m/sec2 x 0.4m = 12000Nm

Max. Stress = (Mmax. x D/2)/I (section modulus) = 12000 x 0.15/2 x 1/(13.1 x 10-6) = 68.7mPa. Less than half the acceptable level.

The beam is bolted to the vertical part of the base and the top support will try and rotate about its bottom edge.

SPECTROMETER SOLENOID IRON SHIELD SUPPORT



The distance of each pair of bolts from the edge are 25mm, 75mm and 175mm. Let u = tensile load in a bolt at unit distance therefore (2u x 0.025m x 0.025m) + (2u x 0.075m x 0.075m) + (2u x 0.175mm x 0.175mm) = 30000N x 0.4m

0.00125u + 0.01125u + 0.06125u = 12000Nm

U = 12000/0.07375 = 162.71 x 1000 N

Load on bolts furthest from edge = 162.71 x 1000 x 0.175 = 28000N

Tensile stress in bolts furthest from edge = 28000N/245mm2 = 114.28N/mm2 (54% of the 0.2% proof stress for A1 fasteners, A2 are normally used)   (245 is tensile stress area for M20)

The attachment support lug has to be capable of carrying the total all up weight of 3 tonnes. Three lugs have to carry the axial magnetic force of 10 tonnes so any lug can have a total combined force of 6.33 tonnes. The shear pin dia. Is 30mm, c.s.a = 0.7854 x 30 x 30 = 706.86mm2. Shear stress = 63300N/706 = 90N/mm2 (50% of the 0.2% proof stress for 304L)

Lug cross sectional area in tension = 24 x 80 = 1920mm2, stress = 63300N/1920 = 33N/mm2(20% of the 0.2% proof stress for 304L)

Lug section in bending = 24 x 80 2nd moment of area = 0.024 x 0.080^3/12 = 1.024 x 10^-6 m4

Treat as cantilever therefore max BM = WL = 63300N x 0.052m = 3291.6Nm

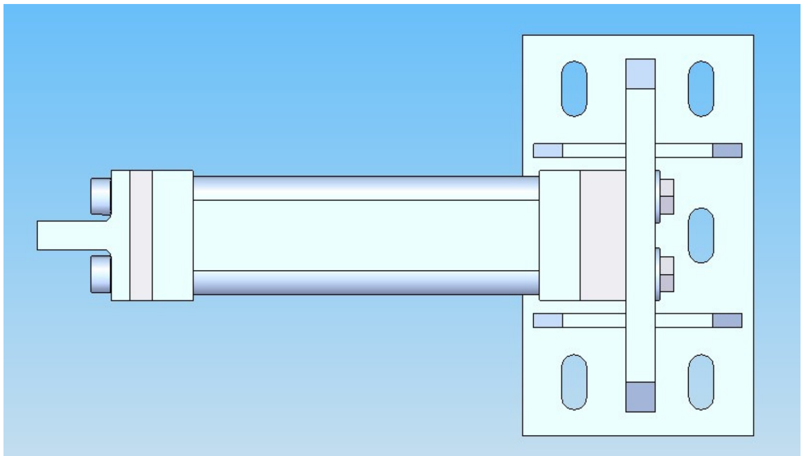
Max stress = (Mmax x D/2)/I(section modulus) = 3291.6 x 0.08/2 x 1/(1.024 x 10^-6) = 128.6MPa

Area of shear pin carrying bearing stress = 30 x 24 = 720mm2. Bearing stress = 63300N/720 = 88N/mm2.

Direct load of 3tonnes carried by 4x M20 St. Steel cap heads, assume 2 carry load. Shear stress area = 245mm2, shear stress in 1 fastener = 30000N/2 x 245 = 61N/mm2.

Tearing out of shear pin in lug hole, area in shear = 2 x 28 x 24 = 1344mm2, shear stress = 30000N/1344 = 47N/mm2.

Applying bending moment to vertical base, I N - A = 1.30 x 10^-5m4 (see below). From above Max BM = 12000Nm, max stress = (Mmax x D/2)/Isection modulus) = 12000 x 0.185/2 x 1/(1.3 x 10^-5) = 85.24MPa



The top bracket carrying the weight and 33% of the magnetic load will want to rotate about the front edge this will be reacted by the M24 studs in the interface plates and base of the bracket. Let u = tensile load in bolt at unit distance from edge and taking moments:-

(3u x 0.045 x 0.045) + (3u x 0.155 x 0.155) = 63300N x 0.4m

0.006075u + 0.072075u = 25320Nm

U = 25320/0.07815 = 323992.32N/m

Load on studs furthest from edge = 32399.32 x 0.155 = 50218.8N

Tensile stress in studs furthest from edge = 50218.8/353 = 142.3N/mm2(68% of the 0.2% proof stress for A1 fasteners normally A2 are used.(353 is the tensile stress for an M24)

The M20 hex. Soc. Cap head fasteners holding the support lug onto the beam will also be subject to the magnetic load as well as the weight of the TOF cage and shield.

Let u = tensile load in fastener at unit distance from edge of lug and taking moments:-

(2u x 0.03 x 0.03) + (2u x 0.13 x 0.13) = 63300 x 0.052

0.0018u + 0.0338u = 3291.6Nm

U = 3291.6/0.0356 = 92460.67N/m

Tensile load in fasteners at furthest distance from edge = 92460.67 x 0.13 = 12019.89N

Tensile stress in fasteners furthest from edge = 12019.89/245 = 49N/mm2

Vertical base I value – above NA

SPECTROMETER SOLENOID IRON SHIELD SUPPORT

Igg of web = (0.012 x 0.08^3)/12 = 5.12 x 10^-7m4

Ay2 = 0.012 x 0.08 x 0.0525^2 = 2.646 x 10^-6m4

Ina = (5.12 x 10^-7) + (2.646 x 10^-6) = 3.158 x 10^-6m4

Total for two webs = 6.316 x 10^-6m4

Ina plate = (0.3 x 0.025^3)/12 = 3.9 x 10^-7m4

Total for base =(2 x 6.316 x 10^-6) + (3.9 x 10^-7) = 1.3 x 10^-5m4

(check 2 webs only = 2 x 0.012 x 0.185^3/12 = 1.266 x 10^-5m4)

Ver.2 23/09/11