Fermilab

SSC DETECTOR SOLENOID DESIGN NOTE #113

TITLE: Design of Inner Vacuum Shell for SDC Solenoid using Solid Plate

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ABSTRACT: This Design Note contains the calculations for an inner vacuum shell made of solid aluminum plate, using the ASME Pressure Vessel Code and CGA-341 as design standards. Using a very low-pressure relief valve the thickness of the shell could be 1/4" (6.4 mm).

DESIGN STANDARDS

Vessel Under Internal Pressure

The ASME Pressure Vessel Code (Section VIII, Division 1, UG-27) was used to determine the MAWP of the inner vacuum vacuum shell in the normal operating condition of internal pressure. UG-27 contains equations for the MAWP considering both circumferential and longitudinal stresses,

$$P(\text{circum.}) = SEt/(R + 0.6t) \qquad t(\text{circum.}) = PR/(SE - 0.6P) \tag{1}$$

$$P(\text{long.}) = 2SEt/(R - 0.4t) \qquad t(\text{long.}) = PR/(2SE + 0.4P) \tag{2}$$

where E is the weld efficiency found in Table UW-12 and S is the allowable stress found in Table UNF-23.1.

Vessel Under External Pressure

The standard used here in the design of the inner vacuum shell of the magnet cryostat under upset conditions is CGA-341--1987, "Standard for Insulated Cargo Tank Specification for Cryogenic Liquids". This standard contains an equation for the critical collapsing pressure of a cylindrical shell under external pressure,

$$P_c = 2.6E(t/D)^{2.5}/[(L/D) - 0.45(t/D)^{.5}]. \tag{3}$$

The CGA standard requires that P_c be at least twice the maximum expected external pressure.

Equation 3 can be rearranged into

$$P_c = 2.6Et^{2.5}/(LD^{1.5} - 0.45D^2t^{0.5}). \tag{4}$$

If stiffening rings are used on a vacuum shell the CGA standard requires that the minimum moment of inertia about the centroid of the stiffening rings be

$$I = 1.05D^3L/E. \tag{5}$$

DESIGN OF INNER VACUUM SHELL WITHOUT STIFFENING RINGS

Normal Operating Conditions

The normal operating condition for the inner vacuum shell is with vacuum on the outside of the shell. This is equivalent to an internal pressure of 15 psid, which places a tensile circumferential or hoop stress on any longitudinal welds in the shell. The vacuum load also places a longitudinal or axial compressive stress on the shell which places any circumferential joints in compression and which may cause Euler buckling.

The internal pressure on the shell will make the shell less susceptible to Euler buckling due to the axial vacuum load. However, for this discussion I shall ignore this coupling effect and consider the hoop stress and the buckling requirements independently.

For the aluminum alloy 5083-0, S = 10,000 psi. I used a weld efficiency of 0.70. The inner radius of the inner vacuum vessel is 1850 mm, as found in the EoI for the SDC; I used D = 146" and R = 73". Substituting these values into Eq. 1 I got the MAWPs for the shell thicknesses given in the table.

I used Roark's formula for the buckling stress of a thin cylinder

$$\sigma_{cr} = E(t/R)/[(1.732)(1-\nu^2)^{0.5}] = (86.4 \text{ ksi/in})t. \tag{6}$$

With the assumption that the inner cylinder carries all the axial vacuum load the actual longitudinal stress in the shell is (180 psi-in)/t. The table indicates whether the longitudinal stress is less than the critical buckling stress.

t (in)	t (mm)	MAWP (psi)	Axial stress (psi)	Crit. stress (psi)	OK for buckling?
5/32	4	15	1154	13500	yes
3/16	4.8	18	960	16200	yes
1/4	6.4	24	720	21600	yes
5/16	8	30	576	27000	yes
3/8	9.5	36	480	32400	yes

This calculation shows that the inner vacuum shell must be at least 5/32" (4 mm) thick to satisfy the ASME Code with regard to the hoop stress and that this thickness is satisfactory for axial buckling.

Upset or Fault Conditions

The fault condition for this shell occurs when the insulating vacuum space reaches a positive gauge pressure following a rupture of a liquid helium or nitrogen line in the vacuum space. This is like an external pressure on the shell, tending to collapse it. The shell must be designed for some collapse pressure and the vacuum space relieved so that the pressure in it does not exceed about half this collapse pressure.

Substituting $E = 10.3 \times 10^6$ psi, L = 354" and D = 146" and into Eq. 4 gives

$$P_c = 267.8t^{2.5}/(6.245 - 0.096t^{0.5})$$

The critical pressure and maximum allowable pressure are given below for several thicknesses.

t		P_c	Maximum pressure in vacuum space	
in	mm	psi	psi	" water
5/32	4.	0.416	0.208	5.8
3/16	4.8	0.657	0.328	9.1
1/4	6.4	1.350	0.675	18.7
5/16	8.	2.361	1.181	32.7
3/8	9.5	3.728	1.864	51.6
7/16	11.1	5.485	2.742	76.0

The collapse pressure of the inner vacuum shell of the CDF cryostat is 3.6 psi, using the CGA standard. The relief device on the insulating vacuum space is the "toilet seat" located at the top of the vertical chimney. It has a cracking pressure of about 1 psi.

It seems clear that, using a relief similar in location and style to that at CDF, the inner vacuum shell of the SDC solenoid cryostat would need to be no thicker than 3/8". The ideas and calculations in the last section suggest that a thickness of 1/4" would probably be acceptable and maybe 3/16.

DESIGN OF INNER VACUUM SHELL WITH STIFFENING RINGS

I concluded that stiffening rings were not an acceptable solution for the outer vacuum shell of the SDC solenoid cryostat (SDC Design Note #112) because of the size required to satisfy the CGA standard. I did not consider the use of stiffening rings for the inner vacuum shell.

RELIEVING AND VENTING OF INSULATING VACUUM SPACE

This section contains a number of almost-random thoughts that I had on the general subject of relieving a very low pressure vessel. I was fishing for ways to justify from a safety point of view my assertion above about a 1/4-inch inner vacuum shell.

No-Vent Situation

In this sub-section I calculated the volume of LHe that could be dumped into the vacuum space and warmed to room temperature without pressurizing the vacuum space.

First, I estimated the volume of the insulation vacuum space:

Total inside ΔR of vacuum annulus = 20 cm

Radial thickness of coil and outer support cylinder = 6 cm
Radial thickness of MLI, shields, etc = 4 cm
Radial thickness of empty space = 20 cm

Length of vacuum space = 700 cm

Volume of empty space = $\pi D \Delta R L$ = π(400 cm)(20 cm)(700 cm) = 17.6 m^3

Then I calculated the mass of helium that would fill this volume to one atmosphere,

$$m = \rho(\text{GHe at 1 atm, 300 K}) \times \text{volume of vacuum space}$$

$$= 1.625\ \text{kg/m}^3 \times 17.6\ \text{m}^3 = 2.86\ \text{kg}$$

Then I found the volume of LHe at 4.2 K which has this mass,

$$V(\text{LHe}) = 2.86\ \text{kg} \times 8\ \text{L/kg} = 22.9\ \text{L}$$

The volume of the liquid in the CDF LHe circuit was 45 L; the SDC solenoid could be 500 to 1000 L. A low-pressure relief device on the vacuum space will therefore open at some temperature substantially colder than 300 K.

Venting Flow Rate

One way to make the venting of the vacuum space easier and with less pressure rise from the relief valve to the vessel is to shut off the LHe and LIN flows into the cryostat in case of a leak into the vacuum space. That way only the inventory in the piping would be vented, not the steady-state flows into the cryostat. I recognize that the maximum flow rate out the relief valve may not be very different in these two cases, but it seems intuitively that you are better off venting less gas.

Set Pressure of Relief Device

One would surely choose a relief device with the lowest set (or cracking) pressure possible. Anderson-Greenwood makes pilot-operated relief valves with set pressures down to 5 inches of water. There was not enough head room for an AGCO valve at CDF so we invented the "toilet seat" relief valve, which has a set pressure around 1 psi. If we want to make the inner vacuum shell very thin then I believe we should demand enough space somewhere for an AGCO pilot-op valve.

The lowest set pressure is probably attainable with a up-side down fall-off plate like we used on the superconducting analysis magnets of 1.5 decades ago. This type of relief device must include some method for holding it in place during pump-down of the vacuum space.

Location of Relief Device

I would like to find a place on the bottom of the cryostat for a fall-off plate, but I suspect that this is really not possible. The relief valve will probably be located at some point in the chimney, as it is at CDF. The cross section of the chimney available for venting and the length of the vent path will determine the pressure rise between the relief valve and the cryostat. We already know that there will be great pressure to make the chimney small in diameter, which is just what you don't want if you are trying to keep the pressure rise small.

CONCLUSION

All things considered I think at this point that a 1/4" shell, with a relief valve set for 5 to 10 inches of water and a pressure rise between 9 and 14 inches of water, is the thinnest we should consider using solid aluminum. We need to do the venting calculations to size the relief valve and determine the inlet and outlet pressure losses. A 3/16" shell would be acceptable only if the pressure rise were 4 inches of water or less.

We have not really thought too much about using aluminum honeycomb between two aluminum skins for the inner vacuum shell, but I think we should consider it as a means of increasing the collapsing overpressure.

ADDENDUM

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PERMISSIBLE OUT-OF-ROUNDNESS OF INNER VACUUM SHELL

Applicable Standard

Paragraph UG-80 of Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code gives the out-of-roundness (OOR) permissible for cylindrical shells under either internal or external pressure. For shells under internal pressure, UG-80(a) says that the "difference between the maximum and minimum inside diameters. . . shall not exceed 1% of the nominal diameter. UG-80(b)(1) makes the same requirement for shells under external pressure, while UG-80(b)(2) places a maximum on the local deviation from circularity.

OOR Permitted for 1/4" Shell

From UG-80(a) and (b)(1): The nominal diameter of the vessel is 146", 1% of this is about 1 1/2".

From Ug-80(b)(2): For $L/D = 354/146 = 2.42$ and $D/t = 146/0.25 = 584$, the maximum permissible deviation $e = 1.0t = 0.25"$ (Figure UG-80.1). From Fig UG-29.2 we get an arc length $= 0.165D = 24.1"$ and a chord length $= 2 \times$ arc length $= 48.2"$. The requirement of UG-80(b)(2) is that in a chord length of 48.2", the maximum plus-or-minus deviation from the true circular form shall not exceed 0.25".