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SSC DETECTOR SOLENOID DESIGN NOTE #112

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

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ABSTRACT: This Design Note contains the calculations for an outer vacuum shell made of solid aluminum plate, using CGA-341 as a design standard. With no stiffening rings between the flat annular heads the thickness of the shell must be 1 inch to meet the CGA standard (a collapse external pressure of 30 psid). The shell thickness is also calculated using stiffening rings, but in order to meet the CGA requirement on the moment of inertia the rings must be 6 to 10 inches in radial extent, making the use of rings an undesirable solution.

DESIGN STANDARDS

Vessel Under External Pressure

The standard used here in the design of the outer vacuum shell of the magnet cryostat 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,

$$P_c = 2.6E(t/D)^{2.5}/[(L/D) - 0.45(t/D)^{.5}]. \tag{1}$$

The CGA standard requires that P_c be at least 30 psid.

Given L, D and E (the modulus), Eq. 1 is usually solved by trial and error for t with $P_c = 30$. Equation (1) can be rearranged into a form that may be easier to solve,

$$2.6Et^{2.5} + 0.45P_cD^2t^{0.5} = P_cD^{1.5}L. \tag{2}$$

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

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

Vessel Under Internal Pressure

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

$$P(\text{circum.}) = SEt/(R + 0.6t) \tag{4}$$

$$P(\text{long.}) = 2SEt/(R - 0.4t) \tag{5}$$

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

DESIGN OF OUTER VACUUM SHELL WITHOUT STIFFENING RINGS

Normal Operating Conditions

The normal operating condition for the outer vacuum shell is with vacuum on the inside of the shell. This is equivalent to an external pressure of 15 psid.

For the aluminum alloy 5083-0, $E = 10.3 \times 10^6$ psi. For a cryostat outer radius of 2300 mm, as found in the EoI for the SDC, D = 181". Substituting these values and $P_c = 30$ into Eq. 2 we get,

$$26.78\ t^{2.5} + 0.442\ t^{0.5} = 0.073\ L. \tag{6}$$

For L = 9 m = 354", Eq. 6 becomes

$$26.78\ t^{2.5} + 0.442\ t^{0.5} = 25.842. \tag{7}$$

Solving Eq. 7 we find the minimum shell thickness to be 0.98" (24.89 mm); the design thickness would probably be 1 inch or 25 mm.

Upset or Fault Conditions

The fault condition for this shell occurs when the insulating vacuum space reaches a positive gauge pressure. This will occur following a rupture of a liquid helium or nitrogen line in the vacuum space.

For 5083-0 aluminum, S = 10,000 psi; for a double-filet weld with no radiography, E = 0.70. Substituting these values and R = 90.5" and t = 1" into Eqs. 4 and 5 we calculate

$$P(\text{circum.}) = 76.8 \text{ psi and } P(\text{long.}) = 155.4 \text{ psi.}$$

Therefore the MAWP(internal) = 76.8 psi, which is much greater than the MAWP(external) of the inner vacuum shell, as shown in Design Note #113.

Euler Buckling

The CGA standard does not specifically mention axial buckling of the shell due to the vacuum load on the heads; I conclude from this that either

Eq. 1 includes a factor to cover buckling or that buckling is not an issue. Just to be sure, I looked at axial buckling of the outer vacuum shell.

An equation for the theoretical buckling stress in a cylinder is found on page 495 of the sixth edition of "Roark's Formulas for Stress & Strain" by Warren C. Young (McGraw-Hill, 1989),

$$\sigma_{cr} = E(t/R)/[(1.732)(1-\nu^2)^{0.5}]. \quad (8)$$

Young notes that the actual buckling stress may be 40 to 60% of this value.

For 5083-0 aluminum, $E = 10.3 \times 10^6$ psi and $\nu = 0.33$; with these values and $R = 90.5"$,

$$\sigma_{cr} = (69.2 \text{ ksi/in})t \quad (9)$$

and for $t = 1"$, $\sigma_{cr} = 69.2$ ksi.

I estimated the axial vacuum load on the outer vacuum shell as,

$$F_a = 15 \text{ psi } (\pi D \, \Delta R) = \pi(15)(181)(12) \text{ lbs} = 102{,}353 \text{ lbs.} \quad (10)$$

I assumed that the outer shell carried all the axial load. The axial stress then is

$$\sigma_a = 15 \text{ psi } \pi D \, \Delta R/\pi D t = 15 \text{ psi } \Delta R/t = 180 \text{ psi}$$

which is far less than the critical buckling stress.

DESIGN OF OUTER VACUUM SHELL WITH STIFFENING RINGS

Using Eqs. 2 and 3, I calculated t and I for a number of values of L; the results are given in the table. To get an idea of the size of the stiffening ring required for each case I used

$$I = bh^3/12, \text{ where b is the axial and h the radial thickness.} \quad (11)$$

L (in)	t (in)	I (in^4)	b x h (in x in)
354	0.980	NA	NA
177	0.740	107	2 x 10
88.5	0.555	53.5	
48	0.430	29.0	1 x 7
24	0.315	16.3	
12	0.228	7.3	1 x 4.4, .5 x 5.5, .25 x 7
6	0.157	3.6	.5 x 4.4, .25 x 5.6

My conclusion from these calculations is that the stiffening rings are too large for this method to be feasible by itself. The idea of welding 6- or 12-inch aluminum channels together does not satisfy the CGA requirement on the moment of inertia.

ADDENDUM TO SDC SOLENOID DESIGN NOTE #112

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August 3, 1990

At our SDC solenoid engineering meeting on August 1, 1990, the question of tolerances on the out-of-roundness (OOR) of the vacuum shells came up. This is what I learned from the ASME Code on the subject.

PERMISSIBLE OOR OF OUTER VACUUM SHELL

Applicable Standards

The CGA standard (CGA-341) contains nothing on the maximum OOR permissible when using their equation for the vacuum shell (Eq. 1 of this Note). However, Section VIII, Division 1 of the ASME Code contains requirements on the maximum OOR for shells under external pressure. It seems reasonable to follow the ASME requirements on the OOR when designing a vacuum shell with CGAS-341.

Paragraph UG-80(b) of Section VIII, Division 1 (page 63 of the 1989 edition), gives the OOR permissible for cylindrical shells under external pressure. The first requirement [UG-80(b)(1)] is that "the difference between the maximum and minimum inside diameters at any cross section shall not exceed 1% of the nominal diameter at the cross section under consideration." The second requirement [UG-80(b)(2)] places a maximum on the local deviation from the true circular form.

Permitted OOR

From UG-80(b)(1): The nominal diameter of the outer vacuum vessel is 181", one percent of this is about 1 3/4" (45 mm).

From UG-80(b)(2): For $L/D = 354/181 = 1.96$ and $D/t = 181/1 = 181$, the maximum permissible deviation $e = 0.7t = 0.7"$ (Figure UG-80.1). From Fig. UG-29.2 we get an arc length = 0.180D = 32.88" and a chord length = 2 x arc length = 65.16". The requirement of UG-80(b)(2) is that in a chord length of 65.16", the maximum plus-or-minus deviation from the true circular form shall not exceed 0.7" (an example of this calculation is found in Appendix L, paragraph L-4).