The NPDGamma experiment is going to run at BL13 at SNS and will use an isolating vacuum chamber around its LH₂ target. The target isolation vacuum chamber was approved and used in runs with the neutron beam at LANSCE in 2006. The design parameters and safety calculations of the vacuum chamber used at LANSCE are discussed in the “NPDGamma Liquid Hydrogen Target Engineering Document” version 1.00 March 07, 2006, available in the web site www.iucf.indiana.edu/U/lh2target/export-files/. For the SNS experiment the following modifications to the chamber and the chamber operation are carried out compared to the LANSCE run:

1. The burst pressure of the two parallel rupture discs is lowered from 29 psid to 7 psid (see figure 4). This will change the vacuum chamber from a pressure vessel to a non-pressure vessel.

2. The beam entrance window thickness is reduced from 0.125 inch to 0.063 inch.

3. The coaxial vacuum/ fill/vent line will be lengthened to extend from the target position to outside of the beam line enclosure, thus increasing the volume of the isolation vacuum from 0.08 m³ to 0.22 m³.

The changes will be discussed in the new engineering document “SNS NPDGamma Target Design and Safety Document”.

This document describes the design and safety calculations of the new vacuum windows of the chamber. These windows are an integral part of the vacuum chamber that contains the LH₂ vessel, which will be filled with 16 liter of liquid hydrogen and operated at temperatures down to 17 K (-256° C), with occasional thermal cycles between this temperature and room temperature during a period of approximately 3 years. The temperature of the vacuum windows will always be at room temperature. In addition to design, this document also analyzes the worst-case accident scenario, namely, the rupture of the LH₂ vessel to the isolation vacuum and consequently a rapid boil off of the liquid hydrogen and pressure rise in the vacuum chamber.

**Description of the Vacuum Windows**

Figure 1 shows the target vacuum chamber, which has the beam entry and exit window. Both of these windows are double windows, i.e. the windows are formed by two parallel windows with a small gap between. The windows are bolted with 36 0.25”-20
size brass bolts on the bolt circle of 13.25” for the inner window and 15.0” for the outer window giving axial sealing forces of 2300 lb/inch and 2100 lb/inch on the bolt circles, respectively.

Since the exiting beam diameter is only 2 inch and beam flux is decreased by the attenuation of the LH$_2$, the exit window thickness will not be changed and will be 0.125 inches per window. All the windows are fabricated from 6061-T6 aluminum alloy.

The required thickness of the entry windows (the convex head) by the experiment in neutron beam for reduction of gamma backgrounds in the detector caused by neutron capture on Al is 0.063 inch per window. The windows are curved into an ellipsoidal dome configuration for increased strength. The localized stresses in an ellipsoidal shape is considered to be “self-resolving”: the loads create strains which allow the shape to yield slightly, increasing the strength proportionally, while also making the shape more spherical, improves its resistance to further loads. Figs 2 and 3 show the design drawings of the new inner and outer beam entry vacuum windows.

Helium gas is introduced into the space between the double windows, through gas channels machined into the chamber; the vacuum seals of the windows are exposed to helium gas to catch any leaks into the isolation vacuum. The double vacuum seals of the windows are formed by the inner indium-seal and outer viton o-ring seal. The indium in the inner seal prevents helium diffusion through the viton o-ring seal to the isolation vacuum. The inside surfaces of the windows are polished to a mirror finish to reduce emissivity of heat radiation.
Fig. 1: Assembly of the LH$_2$ target main isolation vacuum chamber, showing the location of the vacuum windows. In the drawing is shown the thickness of the Al box section of the vacuum chamber, with the removable rear panel. This drawing # SNEUT-E-112-002 is available in www.iucf.indiana.edu/U/lh2target/export-files/.
Fig. 2: Dimensions of the new inner vacuum window at the beam entry, drawing number of SNEUT-E-111-002
Fig. 3: Dimension of the new outer vacuum window at beam entry, drawing number of SNEUT-E-111-003
Fig. 4. The LH$_2$ target system, cryostat, vacuum/fill/vent line that penetrates the BL13 experimental enclosure and connects to the vent isolation box located outside the enclosure.
Method to Design the Vacuum Windows

DOE has adapted 10 CFR 851 requirements for the design of a vacuum vessel and components whose failure impacts equipment and personnel safety. These requirements apply also to windows in vacuum vessels defined as “scientific equipment”. At SNS the vacuum window design should follow NFDD-ENG-008, “SNS Vacuum Window Requirements”. According to NFDD-ENG-007, “SNS Vacuum System Requirements” the vacuum windows shall be designed using one of the following methods:

1. a finite element analysis, FEA or
2. cyclic pressure testing of a representative prototype or
3. another engineering method selected by the designer and acceptable to the facility.

We have selected the FEA method since the design of the windows is simple enough to permit a credible design evolution, the design can be verified with ASME codes, and the resulting safety factors are large.

Strength Calculation of the Windows

The vacuum chamber is protected by two parallel rupture disks shown in Fig. 4. Each has the burst pressure of 7 psid. Based on the large conductance of the outer pipe of the coaxial vacuum/fill/vent line, the internal pressure in the vacuum chamber under any accident scenario does not exceed 15+7=22 psia=7 psid for our design. The pressure on the downstream side is atmospheric pressure. Since the other sides of the rupture disks are immersed in inert gas atmosphere at a pressure of 15 psi, in the worst-case accident scenario the pressure corresponds to an internal pressure of 22 psia. The calculations show, see below, that the failure pressure of the 0.063 inch thick window exceeds significantly the 7 psid burst pressure of the rupture disks.

In performing the strength calculations for the windows, we approached the problem in two methods. First, we performed evaluations based on standard formulas in ASME Boiler and Pressure Vessel Code Section II, part D with assumptions about the strength of the 6061-T6 aluminum. Although it is expected that the hydro-forming process proposed to fabricate the windows should not decrease the T6 temper, it is not easy to verify the temper after fabrication. We, therefore, wanted to investigate what happens to the strength of the windows with their design geometry as a function of the temper of the aluminum. Therefore, we performed a FEA analyses to study the effect of the T0 temper on the strength of the windows. The windows will be fabricated from 6061-T6.

Symbols and Formulae

The ASME Code, Section VIII, and Cryogenic Process Engineering provide the design equations to calculate design pressure for the vacuum windows when the thickness $t$ is known.
List of symbols and their values used in the calculations:

- \( t \) = minimum thickness of window = 0.063 inch
- \( p \) = design internal pressure [psi]
- \( p_c \) = critical pressure [psi]
- \( D \) = diameter of the window [inch]
- \( S_u \) = ultimate tensile strength for 6061-T6 Al = 49600 psi
- \( S \) = allowable stress for 6061-T6 Al = 12400 psi = \( \frac{1}{4} \times S_u \)
- \( Y \) = modulus of elasticity (Young’s modulus) = \( 1.0 \times 10^7 \) psi
- \( \mu \) = Poisson’s ratio = 0.33
- \( E \) = weld joint efficiency factor = 1.0

(a) Elliptical window under pressure on concave side of the window; internal pressure on vacuum window

\[
p = \frac{2SEt}{DK + 0.2t},
\]

where the constant \( K \) is given by
\[
K = \frac{1}{6} \left[ 2 + \left( \frac{D}{2h} \right)^2 \right] \quad \text{with} \quad \frac{D}{2h} = \text{ratio of the major to the minor axis of elliptical heads.}
\]

For inner window when pressure is inside the LH\(_2\) vessel and \( D=12 \) inch, \( h=2.28 \), \( D/2h=2.63 \), and \( K=1.49 \)

\[
p = \frac{2 \times 12400 \times 1 \times 0.063}{12 \times 1.5 + 0.2 \times 0.063} = 87 \text{ psi}
\]

(b) Elliptical window under pressure on convex side; external pressure on the vacuum window

\[
p_c = \left( \frac{R}{R_0^*} \right)^2 \frac{YE}{2[3(1-\mu^2)]^{1/2}},
\]

with \( R_0^* = K_1D \). \( K_1 \) is given by table UG–33.1 in the ASME Code as a function of \( \frac{D}{2h} \).

For outer window when pressure is external on the LH\(_2\) vessel and we have \( D=12 \) inch, \( h=1.9 \), \( D/2h=3.32 \), \( K_1=1.44 \), and \( R_0^*=17.28 \), then the critical pressure is

\[
p_c = \left( \frac{0.063}{17.28} \right)^2 \cdot \frac{1.0 \times 10^7 \times 1}{2[3(1-0.33^2)]^{1/2}} = 41 \text{ psi}
\]

The ASME Code specifies that the critical pressure \( p_c \) shall be four times the maximum allowable (external) working pressure (MAWP) on the chamber defined by 7 psid rupture disk. This would give \( p_c=4\times7=28 \) psid,
Results of the Finite Element Analysis for the Inner and Outer Windows of the Vacuum Chamber

Since a hydro-forming process used to fabricate the windows, involves no application of heat, there is no reason to suspect that the T6 properties will be diminished. Nevertheless to be sure that any possible weakening of the T6 temper does not affect the safety of the proposed design, we performed an FEA analysis using as a worst-case scenario: a T0 temper.

The finite element program ALGOR was used to analyze the inner and outer windows of the vacuum isolation chamber. The model for the inner window was imported from IUCF drawing 311218-001 and the model for the outer window from drawing 311219-000. Figures 2 and 3 above show the models of these windows. The models were meshed with a mix of bricks, wedges, pyramids, and tetrahedra. The total number of elements for the outer window was 16970 with a total number of nodes of 20432. To make the mesh more uniform and to speed up the calculations, the holes around the edge were neglected. The flat edges were assumed to be fully clamped and held, i.e. fixed in x, y, and z.

Calculations with the following assumptions were performed.

1. An external pressure of 17 psia and internal pressures of 30 psia and 50 psia were applied to the window models.
2. The material is 6061-T0 aluminum which has the following properties:
   - Modulus of elasticity (Young’s modulus): $Y = 1.0 \times 10^7$ psi
   - Allowable stress for 6061-T0 Al: $S = 8.0 \times 10^3$ psi
   - Ultimate tensile strength for 6061-T0 Al: $S_u = 18.0 \times 10^3$ psi
   - Poisson’s ratio: $\mu = 0.330$
   - Density of Al: $\rho = 0.098$ lb/in$^3$
3. The x-axis is the symmetry axis.

This analysis is an elastic analysis where the stresses in the model change linearly. Thus the calculated maximum allowable working pressure (MAWP) can be determined by the factor between the 6061-T0 aluminum tensile yield strength of 8000 psi and the calculated maximum stress intensity for the particular internal and/or external pressures.

The results of the finite element analysis for the inner and outer window at 17 psid external pressure and at 30 and 50 psid internal pressures are shown in Figs. 5-10. All calculations show that the maximum stress occurs at the outer edge of the window where the elliptical section starts.

Inner window: The maximum stress intensity at 17 psid external pressure is 4440 lbf/in$^2$. It was concluded that the inner window can withstand a maximum pressure 1.8 times the applied pressure of 17 psid. Therefore the maximum external design pressure of the inner window is 30.6 psid. The maximum stress intensity at 30 psid internal pressure is 7835
lbf/in². Here the ratio between tensile yield strength of 8000 and calculated maximum stress intensity is 1.02 yielding the maximum internal design pressure of 30.6 psid. The calculations for an internal pressure of 50 psid yield the maximum internal design pressure of $\frac{8000}{13058 \times 50} = 30.6$ psid.

Outer window: The maximum stress intensity at 17 psid external pressure is 4390 lbf/in². It was concluded that the inner window can withstand an maximum external pressure 1.8 times the applied pressure of 17 psid. Therefore the maximum external design pressure of the outer window is $\frac{8000}{4390 \times 17} = 31$ psid. The maximum stress intensity at 30 psid internal pressure is 7745 lbf/in². Here the factor is 1.0 yielding again an $\frac{8000}{7745 \times 30} = 31$ psid maximum internal design pressure. The calculations for an internal pressure of 50 psid yield the maximum internal design pressure of $\frac{8000}{12909 \times 50} = 31$ psid.

**Summary and Conclusions**

The maximum internal design pressure for both inner and outer windows is 31 psid. Since the vacuum vessel is protected by two parallel rupture disks as mentioned above, each with a set point of 7 psid, the actual internal pressure cannot exceed 7 psig thus yielding a safety factor of 4.3 for the worst-case accident scenario.
Fig. 5. Stress results for the inner window at 17 psid external pressure.
Fig. 6. Stress results for the inner window at 30 psid internal pressure.
Fig. 7. Stress results for the inner window at 50 psid internal pressure.
Fig. 8. Stress results for the outer window at 17 psia external pressure.
Fig. 9. Stress results for the outer window at 30 psid internal pressure.
Fig. 10. Stress results for the outer window at 50 psid internal pressure.
Possible Deviations from the FEA calculation predictions

Since the hydro-forming process itself will produce some strain hardening, the material strength in the final product will be improved relative to the calculations. This strain hardening is the greatest where the strains are the greatest, especially in the knuckle region and the center of the dome for our ellipsoidal shaped windows. Consequently, these areas should see some of the greatest increases in ultimate strength from this effect, which is good since these areas are also where the FEA calculation some of the highest localized load strains.

However, these same localized strains can tend to cause the material to thin out slightly due fiber elongation from the forming strains. This tends to decrease the strength somewhat. In an attempt to set a limit on the possible size of this effect, we measured the deviations away from the nominal thickness of the existing (thicker) aluminum windows of 0.125 inch thickness, which were also hydro-formed by the same vendor for the new windows. The results of the measurements are shown below in Fig. 11. In no place was the deviation away from the nominal thickness larger than 8% of the design thickness of 0.125 inch, and this largest deviation is a thickening in the area of the rim, thereby again increasing the strength of the window in an area of higher stress under load. The upper limit on the amount of thinning was 0.001 inch in 0.125 inch, a less than a 1% fractional decrease. Even if the same absolute size of a thickness deviation should occur on our thinner windows, this produces a negligible change in the FEA calculations.

Hazard Analysis

The worst-case accident scenario is a rupture of the LH2 target vessel or piping inside the isolation vacuum releasing the LH2 into the vacuum space consequently causing a fast phase transition from liquid to gas that leads to a rapid increase in pressure in the vacuum chamber. When overpressure through the rapid boil off occurs, a pressure relief system with two parallel rupture disks are designed to safely release the hydrogen gas through the vent isolation box to the vent line which leads to the outside of the Target building while maintaining the pressure within the vessels at a safe level. The conductance of each safety relief system has been calculated based on the Bates Internal Report # 90-02 and the Crane Technical Paper No. 410 to be large enough that a pressure rise will not lead to a rupture of the weakest component in the system. The maximum pressure in the vacuum chamber for the case of a rupture of the target vessel is 22 psia=7 psid for an inner diameter of the pressure relief piping of 6 inch in diameter and a boil-off rate of 0.50 lb/s. The boil-off rate and with it the maximum pressure depends on the heat transfer rate to the liquid hydrogen. The rates in different situations were confirmed by measurements at Indiana where nitrogen was used as the working fluid in the target vessel along with the appropriate scaling of the results for the thermodynamic differences between liquid nitrogen and liquid hydrogen. In addition, we were able to use data from the venting of the hydrogen target chamber at LANSCE to determine relevant parameters of the system such as heat transfer coefficients. All of these calculations are presented in detail in the document “Designereliefvent. doc” which is available at www.iucf.indiana.edu/U/lh2target/export-files/.
Conclusions
Based on the presented calculations we conclude that the 0.063 inch thick beam entry vacuum windows as designed meet the safety requirements.

Fig. 11. Measurements of thickness variations in a 0.125 inch thick hydro-formed window made by the same vendor and procedure proposed for the new 0.06 inch windows. Red Area: thickness variation is about 0.001 inch. Green Area: thickness of this area is about 0.005 inch thicker than the red area. Blue Area: thickness of this area is about 0.01 inch thicker than the red area. Cyan Area: the mounting edge of the window.