ASME Code Calculations for the CC Cryostat

This engineering note contains the ASME Code calculations for the CC Cryostat prepared by the manufacturer, Richmond-Lox Equipment Company. Most of these were taken from calculations initially prepared by Fermilab personnel and published in Eng. Note 68.

Issued by: R. D. Luther
Nov. 4, 1987
DESIGN CALCULATIONS

CUSTOMER: FERMILAB NATIONAL ACCELERATOR LABORATORY

P.O. NO. 

ITEM NO: DΦ CENTRAL CALORIMETER

PREPARED BY: J. L. James DATE 2/18/87

CHECKED BY: AmB DATE 2/18/87
SUMMARY OF FERMI CALCULATION RESULTS

I. INNER VESSEL
   ASME Code Sec. VIII Div. 1
   \( E = 0.65 \) (No radiography)

A. Outer Cylinder

1. Internal Pressure
   Design pressure = 40 psi;
   Reg'd thick = 0.3163 in.
   If actual thick \( \frac{5}{8} \) allow. int. pres. = 79.15 psi

2. External Pressure
   Design pres. = 15 psi
   \[ \text{For } t = \frac{5}{8} \text{ in. } P_a = 20.14 \text{ psi} \]
   \[ \text{For } t = \frac{7}{16} \text{ in. } P_a = 15.03 \text{ psi} \]

B. 8" Nozzles in Outer Cylinder (N1P)
   8" NPS (8.625" O.D.)
   \( P = 40 \text{ psi} \) \( E = 0.65 \)

1. Nozzle Section in Longitudinal Plane
   a. Inward Extension of Nozzle [Fig. W-16.1(2)]
   (1) Internal Pressure
   \[ t_{\text{in min}} = 0.2570 \text{ in.} \]
   \[ t_r = 0.2570 \text{ in.} \]
   \[ t_{nn} = 0.011 \text{ in.} \]
   \[ t_s \geq 0.4190 \text{ in. reg'd shell thickness} \]
   \[ \text{reg'd shell thickness} \leq \frac{5}{8} \text{ in. OK} \]
   (2) External Pressure
   \[ t_{\text{out min}} = 0.2570 \text{ in.} \]
   \[ t_r = 0.9375 \text{ in.} \]
   \[ t_{nn} = 0.050 \text{ in.} \]
   \[ t_s \geq 0.5673 \text{ in. reg'd shell thickness} \]
   \[ \text{reg'd shell thickness} \leq \frac{5}{8} \text{ in. OK} \]
b. W/O Inward Projection of Nozzle [Fig UW-16.1(c)]

(i) Internal Pressure
\[ t_{n_{\text{min}}} = 0.2570 \text{ in.} \]

(ii) External Pressure
\[ t_{n_{\text{min}}} = 0.2570 \text{ in.} \]
\[ t_r = 0.4375 \text{ in.} \]
\[ t_m = 0.050 \text{ in.} \]
\[ t_S \geq 0.616 \text{ in.} \]

Consider the steel thickness to avoid adding a reinforced element: \( \frac{5}{8} \) in. OK.

2. Nozzle Section in Transverse Plane

Limits of reinforcement between adjacent 6'' & 8'' nozzles do not overlap.

a. Integral Reinforcement \( (F = 0.5) \) [Fig UG-37]

(i) Internal Pressure
\[ A_{\text{reg}} = 1.562 \text{ in.}^2 < 2.085 \text{ in.}^2 \]
\[ t_r = 0.2570 \text{ in.} \]

b. Non-Integral Reinforcement \( (F = 1.0) \) [UG-37(a)]

(i) External Pressure
\[ t_r = 0.4375 \text{ in.} \]
\[ A_{\text{reg}} = 2.659 \text{ in.}^2 > 1.774 \text{ in.}^2 \]
\[ t_S \geq 0.5970 \text{ in.} \]

Consider the steel thickness to avoid adding a reinforced element: \( \frac{5}{8} \) in. OK.
C. Heads

1. Outer Knuckle
   a. Check as an equivalent regular torispherical head
      Internal pressure \( (P = 40) \) \( E = 0.65 \)
      \[ t = 0.593 \text{ inch} \quad [U6-32(e)] \]
      \(< \frac{5}{8} \text{ in.} \quad \text{OK} \)
   b. Size knuckle thickness based on equivalent internal pressure
      Eqv. internal pressure based on plastic collapse criterion is \( 0.7 \times 40 = 28 \text{ ps} : \quad (E = 0.65) \)
      \[ t = 0.415 \text{ inch} \quad [U6-32(e)] \]
      \(< \frac{5}{8} \quad \text{OK} \)
   c. Check knuckle thickness/critical buckling pressure/allow pressure based on:
      (1) Elastic Buckling Theory
      For \( t = 0.415 \text{ inch} \)
      \[ P_{cr} = 632 \text{ psi} \]
      and for S.F. = 4, \( P_a = \frac{632}{4} = 158 \text{ psi} \) \( > 28 \text{ or 28} \)
      (2) Elastic-Plastic-No Strain Hardening
      For \( t = 0.415 \text{ inch} \)
      \[ P_{cr} = 49.4 \text{ psi} \] \( > 40 > 28 \text{ psi} \)
      (3) Plastic Collapse
      For \( t = 0.625 \text{ inch} \)
      \[ P_{cr} = 42.2 \text{ psi} \] \( > 40 > 28 \text{ psi} \)
C. Heads (cont'd)

2. Crown ("Spherical Surface")

a. Internal Pressure

Estimate of reg'd thickness based on detailed computer analysis of membrane stresses

\[ t_r = 0.256 \text{ in} \]

Check per Appendix 1-6 (e),

\[ t = \frac{5}{6} \frac{P_L}{T} = \frac{5}{6} \frac{(40)(14)}{18800} = 0.156 \text{ in} \]

b. External Pressure (Governed by yielding/collapse in knuckles)

(1) For \( t = \frac{5}{6} \), \( P_{cr} = 42.8 \text{ psi} > 15 \text{ psi} \) des. p [WRC-227] per Specs.

(2) For \( t = 0.4375 \), \( P_{cr} = 40 \text{ psi} \)

Unsymmetrical buckling


(3) For critical buckling pressure,

\[ t = 0.422 \text{ inch} < \frac{5}{6} \] [App O-5 (b), Code]

(4) Check as curved panel, simply supported:

For \( P_{cr} = 30 \text{ psi} \),

\[ t = 0.435 \text{ inch} < \frac{5}{6} \] [Remark, 4th, p.359, Case 33]

(5) Check as curved panel, clamped

For \( P_{cr} = 30 \text{ psi} \),

\[ t = 0.344 \text{ inch} < \frac{5}{6} \] [Remark, 4th, p.359, Case 34]
C. Heads (cont'd)

3. Inner Knuckle

Check as an equivalent torispherical shell

with an $L_2 = 155.7$ in., $\theta = 10.55^\circ = \tan^{-1}\left(\frac{155.7}{20.879}\right)$

$$\frac{r}{L} = \frac{4.5}{155.7} = 0.029 \neq 0.9 \Rightarrow \text{(hence UTG-32(c) doesn't apply)}$$

$$\left(\frac{r}{D_2} = \frac{4.5}{44.47} = 0.090 \quad \frac{t}{L_2} = \frac{0.025}{155.7} = 0.0040\right)$$

a. Internal Pressure ($P = 40$ psi)

$t = \frac{D_2}{8}$ OK based on Battelle analysis of max membrane stress in knuckle, $\sim 18,000$ psi < 1.5 $SE = 28,310$

b. External Pressure

For $t = \frac{D_2}{8}$ in., $P_{ex} = 312$ psi > 15 psi design $P$ per specs

Check with Drucker formula: for $t = \frac{D_2}{8}$ in.

$$P_{ex} = \frac{1}{2} \rho (\frac{t^2}{D}) \left(\frac{L}{t} - 0.0006\right)$$

$$= 30,000 \left[(0.33 + 5.5 \times 0.070) \times 0.0040 + 28(1 - 2.2 \times 0.070) \times 0.0040\right]$$

$$= 30,100 \geq 118 \text{ psi design: old}$$

$P_{ex} = \frac{15 - 29 = 118 \text{ psi} > 15 \text{ psi design: old}}{10}$
D. Inner Cylinder

Vessel internal pressure is pressure on convex surface of this cylinder; vessel external pressure is pressure on concave surface of this cylinder.

1. Vessel Internal Pressure (Cyl. "External")

Design pressure = 15+15+ head = 37 psi

\[ t = \frac{3}{8} \text{ inch} \text{, for which } P_a = 37.1 \text{ psi} \]  

\[ [Ug-28(c)] \]

2. Vessel External Pressure (Cyl. Internal"")

Check axial compression under 15 psi pressure

\[ \sigma = \frac{P R}{2 t} = \frac{15 \times 32.525}{2 \times \frac{3}{8}} = 651 \text{ psi} \]  

\[ [Ug-23(b)] \]

Max. allowable compressive stress = \( B = 10,000 \text{ psi} < 651 \text{ psi} \text{ OK} \)

Circumferential Stress \( [Ug-27(c)(i)] \):

\[ t = \frac{P R}{S E - 0.6 P} = \frac{15 \times 32.525}{18,300 \times 0.65 - 0.6 \times 15} = 0.0395 \text{ in} < \frac{5}{8} \text{ on} \]

E. Support Structure

1. Stiffening Rings

Supported wt. = 175,000 lbs

\[ \text{wt/ring} = 87,500 \text{ lbs} \]

Ring loading, \( p_r \) = 145.7 lbs/inch

Ref: Rosen, 4th ed., p. 172, Case 19

Bending moments, axial loads, shear loads in ring

On cylinder wall side of ring, hoop stress due to pressure
At nozzles, hoop stress due to internal pressure is higher because of local stresses; assumed to be \( 1.5 \times 172 = 920 \text{ psi} \), per ASME-23(c).

Allow. @ nozzles = \( 1.5 \times 18,500 \times 0.8 = 23,500 \text{ psi} \).

Allow away from openings = \( 18,500 \times 0.8 = 15,000 \text{ psi} \).

Allow a Category A. walls = \( 18,500 \times 1.5 = 27,750 \text{ psi} \).

Allow. stress level - hoop membrane due to internal pressure = stress that can be allocated to support loads only.

On cover plate side of ring, allow stresses due to bending limited to \( 0.66 \times y.p. \) per ASME 1.5.1.4.1;

due to shear limited to \( 0.4 \times y.p. \) per ASME 1.5.1.2;

and weld shear stresses limited to \( 0.3 \times t.s. \) of weld metal or \( 0.3 \times y.p. \) of base metal per ASME 1.5.3, Table 15.3.

Cover plate thickness = \( 3/4 \) inch.

Wells = \( 2 - 1 1/2 \times 1 1/2 \times 1/4 \) ft.

(Vessel wall thickness = \( 3/8 \) inch.)

Rings carried @ supports, @ 5 ft. from bottom (location of max bending moment), \( f @ \text{signal port NIP} \).
### Design for Int. Pressure

- $P = 40,261$
- $A_0 = 19.35$
- $R_0 = 96.75$
- $S = 18800$

\[
P = 40,261 \quad A_0 = 19.35 \quad R_0 = 96.75 \quad S = 18800
\]

\[
t = \frac{P R_0}{3E + 0.4P} = \frac{40(96.75)}{18800 + 4(40)} = \frac{3870}{18800 + 16}
\]

\[
E = 1.0
\]

\[
t = 0.2057
\]

\[
E = 0.9
\]

\[
t = 0.2285
\]

\[
E = 0.85
\]

\[
t = 0.2419
\]

\[
E = 0.80
\]

\[
t = 0.2570
\]

\[
E = 0.65
\]

\[
t = 0.3143 \quad \text{Int. Press.}
\]

\[
E = 0.65
\]

\[
t = 0.6285
\]

\[
\begin{align*}
\text{Int. } P_o &= 18900(1.65)(0.625) \\
&= 79.15 \text{ psi}
\end{align*}
\]

### Design for Ext. Pressure

- $P = 15.4\text{ psi}$
- $D_o = 193.5$

\[
P = 15.4 \quad D_o = 193.5
\]

\[
L = 117.25 - \frac{2}{3}(117.25 - 77.12)
\]

\[
L = 90.491
\]

\[
L/D_o = \frac{90.491}{193.5} = 0.468
\]

\[
t = 0.625
\]

\[
\theta = 30.24°
\]

\[
L/D_o = 1.0
\]

\[
A_z = 0.433 \times 10^{-3}
\]

\[
A_{az} = 0.364 \times 10^{-3}
\]

\[
A_{az} = 0.544 \times 10^{-3}
\]

\[
A_+ = 0.66 \times 10^{-3}
\]

\[
DA = 1.0
\]

\[
DA = 0.0
\]
\[ 2.2 + 0.7 = 2.9 \]

**CC, INNER**

\[ A_5 = A_1 \left( \frac{\log(4/1.468)}{\log(4/1.6)} \right) \]
\[ = 0.266 \times 10^{-3} \times \left( \frac{0.433}{0.466} \right) \]
\[ = 0.531 \times 10^{-3} \]

**Outer Cyl. Ext P**

\[ A_6 = A_1 \left( \frac{\log(0.9/1.468)}{\log(0.9/1.468)} \right) \]
\[ = 0.364 \times 10^{-3} \]

\[ A = A_6 \left( \frac{\log(300/10000)}{\log(300/10000)} \right) \]
\[ = 0.538 \times 10^{-3} \]

**Fig 5-UNA-28.1**

\[ B = \frac{A_1}{A_2} \left( \frac{\log(4/1.468)}{\log(4/1.2)} \right) \]
\[ A_1 = 0.463 \times 10^{-3} \]
\[ B = 2.590 \]
\[ A_2 = 1.5 \times 10^{-3} \]
\[ B_2 = 10600 \]

\[ B = \frac{2.590}{10600} \left( \frac{\log(4/1.468)}{\log(4/1.58)} \right) \]
\[ = 7002 \]

\[ P = \frac{4}{3} \left( 9002 \times 30 \% \right) = 30.16 \text{ psi} \]

**try t = 0.4375**

\[ D_{1/4} = 422.3 \]

\[ L_{0.5} = 15 \]
\[ A_0 = 14.9 \times 10^{-3} \]
\[ L_{1.0} = 259 \times 10^{-3} \]
\[ A_1 = 3.0 \times 10^{-3} \]
\[ L_{1.5} = 422.3 \]
\[ A_2 = 0.429 \times 10^{-3} \]

\[ A = A_7 \left( \frac{\log(4/1.468)}{\log(4/1.468)} \right) \]
\[ = 0.259 \times 10^{-3} \]

\[ A = A_2 \left( \frac{\log(400/422.3)}{\log(400/500)} \right) \]
\[ = 0.335 \times 10^{-3} \]

\[ A_1 = 0.01 \times 10^{-3} \]
\[ A_2 = 0.463 \times 10^{-3} \]
\[ B = 140 \left( \frac{2590}{400} \right) \left( \frac{\log(0.1/1.335)}{\log(0.1/1.468)} \right) \]
\[ = 4741.5 \]

\[ P = \frac{4}{3} \left( 4741.5 \times 422.3 \right) = 15.03 \text{ psi} \]

\[ \eta = 0.4375 \]

**Ext Pressure**
Determine the maximum ID for the 8" x Signal Port's NIP.

Assume steel pipe will be used (8.625" OD) with the bore enlarged to the maximum allowed by the ASME Code. Reinforcement requirements for this geometry will then be determined.

Longitudinal plane

\[ d = 8.625 - 2t_n \]
\[ = 8.625 - 2(0.2570) \]
\[ = 8.111" \quad E=0.65 \]
\[ = 8.214" \quad E=1.0 \]

\[ t_n = 0.2570" \]

\[ t_{n\min} = (UG-45) \]
\[ E=1.0 \text{ or } 0.85 \]
\[ t_n = \frac{P_{ho}}{SE+0.4P} \]
\[ = 40(19.35) \]
\[ 18800(16) + 0.4(40) \]
\[ = 0.2057 \]
\[ E=0.65 \]
\[ t_n = 0.2570" \]

**Area Available in Nozzle**

\[ A_{rg} = 0.5 d_t e F \]
\[ = 0.5 (8.111 \times 0.4375) \]
\[ = 1.774 \text{ in}^2 \quad E=0.65 \]
\[ = 1.794 \text{ in}^2 \quad E=1.0 \]

**Outside**

\[ t_{n\min} = \frac{d_n}{n} \]
\[ \text{Assume } L_{max} = 32" \]

\[ \frac{L}{d_n} = 4.6 \]
\[ \text{let } \frac{L}{d_n} = 0.050 \]

\[ \frac{d_n}{L} = \frac{172.5}{0.050} = 3450 \]

\[ \text{UG-28} \]
\[ \frac{P_o}{3} = 2(0.0004 \times 28 \times 10^6) \]
\[ = 15.15 \text{ kC} \]
\[ t_{n\min} = 0.050 \]
\[ A_2 = (t_{\text{p}} - t_{\text{n}}) \delta^2\]

\[ A_3 = 2(t_{\text{n}}) \frac{2}{3} t_{\text{n}} = 5t_{\text{n}}^2 \]

**Weld Areas**

\[ \Delta \quad \Delta \]

\[ \Delta \quad \Delta \]

Assume 1/4" minday length

\[ A_w = 2 (\frac{d}{4})^2 \]

\[ = 2(0.25)^2 \]

\[ = 0.125 \text{ in}^2 \]

**Area Exclusive of Shell**

\[ A_L + A_3 + A_w \]

\[ E = 0.65 \]

\[ t_{\text{n}} = 0.2570 \]

\[ \delta = 0.050 \times 5(0.2570) + 5(0.2570)^2 + 0.125 \]

\[ = 0.721 \text{ in}^2 \]

**Area required in shell**

\[ A = A_0 - A \]

\[ = 1.774 - 0.721 \]

\[ = 1.053 \text{ in}^2 \]

**Weld Shell thickness:**

\[ A_3 = 1.053 + (t_{\text{s}} - t_{\text{n}}) \delta \]

\[ 1.053 < (t_{\text{s}} - 0.4375)(1.111) \]

\[ t_{\text{s}} > 0.5673 \]

For \( E = 10 \)

\[ t_{\text{n}} = 0.157 \]

\[ A = A_2 + A_3 + A_w = (1.2057 - 0.5)(0.2057) + 5(0.2057)^2 + 0.125 \]

\[ = 1.160 + 0.212 + 0.125 \]

\[ = 0.497 \text{ in}^2 \]

\[ A_1 = 1.796 - 1.497 = 1.299 \text{ in}^2 \]

\[ 1.299 < (t_{\text{s}} - 0.4375)(9.24) \]

\[ t_{\text{s}} > 0.595 \text{ in} \]

\[ E = 1.0 \text{ in} \]

\[ t_{\text{n}} = 0.157 \text{ in} \]

**Internal Pressure**

\[ A_{\text{int}} = \frac{d}{2} t_{\text{n}} F \]

\[ = 0.065(0.2570)(1.0) \]

\[ = 0.109 \text{ in}^2 \]

\[ E = 0.65 \]

\[ t_{\text{n}} = 0.011 \text{ in} \]

\[ E = 0.65 \]

\[ = 0.009 \text{ in} \]

\[ E = 1.0 \]
Area Excl. Surr. 

\[ \varepsilon = 0.65 \]

\[ A = (0.2570 - 0.11)(3)(2.570) + 1.330 + 1.325 \]

\[ = 0.316 + 330 + 1.325 \]

\[ = 0.771 \text{ in}^2 \]

\[ A_8 = (0.315 - 0.771) \leq (t_8 - 0.75)(x_{111}) \]

\[ t_8 = 0.419'' < 0.5673'' \]

\[ \therefore \text{ Ext P. Controls } \]

\[ t = 1.0 \]

\[ A = (0.2057 - 0.009)(3)(2.057) + 0.122 + 1.125 \]

\[ = 0.202 + 0.122 + 1.125 \]

\[ = 1.539 \]

\[ A_1 = (0.190 - 0.537) \leq (t_1 - 0.205)(x_{111}) \]

\[ t_1 = 0.346 < 0.595'' \]

\[ \therefore \text{ Ext P. Controls } \]

Transverse Plane

\[ d_8^* = \frac{8.11}{\cos 48.14^\circ} = 15 \times 1.11 \]

\[ = 12,155'' = \frac{12,309}{\cos 1.10} \]

\[ d_6 = 6.25 - 2(2.057) \]

\[ = 6.214'' \]

\[ \alpha = 25^\circ = \frac{56}{94} \]

\[ = 48.69^\circ \]

\[ d_7^* = \frac{6.214}{\cos 35.69^\circ} \]

\[ = 7.765'' \]

\[ C = \frac{96(1245/\pi)}{140} \]

\[ = 20.86'' \]

\[ d_{10a} = \frac{12.155 + 2.351}{2} \]

\[ = 7.903'' \]

\[ d_{10a}^* = 19.806 \]

19.806 < 20.86

* Limits of Radius, Do Not Overlap.
**Trans. Plane (Cont'd)**

**Integral Reinforcement** $F = 0.5$

**End. Press**

\[
egin{align*}
\text{Area} &= 0.5 \times t \times F \\
&= 0.5 \times (12.155 \times 4.875) \times 1.5 \\
&= 13.29 \text{ in}^2 \quad E = 165 \\
&= 13.46 \text{ in}^2 \quad E = 10
\end{align*}
\]

This area is less than the area required for the longitudinal plane in which the limits of reinforcement are smaller. Therefore, the longitudinal plane controls.

**Int. Pressure**

\[
egin{align*}
\text{Area} &= d \times F \\
&= 12.155 \times (0.257) \times (E) \\
&= 1.562 \text{ in}^2 \quad E = 165 \\
&= 1.260 \text{ in}^2 \quad E = 10
\end{align*}
\]

These are less than the corresponding values for the longitudinal plane. Therefore, the longitudinal plane controls.

**Non-Integral Reinforcement** $F = 1.0$

**End Pressure**

\[
egin{align*}
\text{Area} &= 0.5 \times d \times F \\
&= 0.5 \times (12.155 \times 4.875) \times 1.0 \\
&= 26.59 \text{ in}^2 \quad E = 165 \\
&= 24.93 \text{ in}^2 \quad E = 10
\end{align*}
\]

\[
A_i = \text{Area} - (A_2 + A_3 + A_4)
\]

\[
= 26.59 - 0.721 \\
= 19.38 \text{ in}^2
\]

**Summary** 8' Equal parts only

\[
E = 0.65
\]

\[
\begin{align*}
t_{\text{min}} &= 0.2257'' \\
t_{3\text{min}} &= 0.5675'' \text{ Full penetration} \\
t_{3\text{min}} &= 0.5970'' \text{ Non-Integral Welds}
\end{align*}
\]

\[
E = 1.0
\]

\[
\begin{align*}
t_{\text{min}} &= 0.2057'' \\
t_{3\text{min}} &= 0.546'' \text{ Full penetration} \\
t_{3\text{min}} &= 0.614'' \text{ Non-Integral Welds}
\end{align*}
\]

**Note:** Internal Projection Requisite.
Look at 8" nozzle w/ internal projection
(fill pen weld w/ backing strip)

\[ t_{\text{min}} = 0.2570 \ (E=1.0) \]
\[ = 0.257 \ (E=1.0) \]

**Ent. Pressure Controls**
\[ E = 0.4375 \]
\[ A_{\text{sh}} = 1.774 \text{ in}^2 \ F (E = 0.65) \]
\[ = 1.794 \ E = 1.0 \]
\[ t_{\text{min}} = 0.050 \]

**Area Excluding Shell**
\[ E = 0.65 \]
\[ A = (2570 - 0.050)(5)(2570) + \left(\frac{1}{4}\right)^2 \]
\[ = 1246.0025 \]
\[ = 0.328 \sqrt{ } \]

**Shell & Rod**
\[ (1.774 - 0.328) < (1.4375)(8.11) \]
\[ t_5 = 0.616 \text{ in} \quad \text{5/8" OK} \]

\[ E = 1.0 \]
\[ A = (2057 - 0.15)(5)(2057) + 0.025 \]
\[ = 0.223 \]
\[ (1.794 - 0.223) < (1.4375)(8.214) \]
\[ t_5 \geq 0.629 \ 	ext{" . Can use main nozzle / 1/4" wall OK.} \]

**Summary**

For nozzles welded from OS only (fill pen weld)
\[ E = 0.65 \]
\[ t_{\text{min}} = 1.2570 \]
\[ t_{\text{min}} = 0.616 \text{ in} \]
\[ E = 1.0 \]
\[ t_{\text{min}} = 0.257 \]
\[ t_{\text{min}} = 0.625 \text{ in} \text{ approx.} \]
Pages 7 thru 11

Not Applicable
do I.V. present design
CHECK AS A REG. TORISPERICAL HELD PER DM. 1 RULES:

\[ R = 12" \]
\[ R = 9.125 \quad L = 204.41 \]

\[ q = 0.40 \text{ psi} \quad \psi = 0.65 \]

\[ E = 0.0586 = 6\% \]

\[ t = 0.885 \frac{qL}{5E - 0.1P} = 0.885 \frac{(40)(204.41)}{18800(65) - 1(40)} = 0.593" \]

\[ \text{by full spot x-ray weld; no x-ray on attack weld (E=0.80)} \]
\[ t = 0.482 \quad (E=0.80) \]

\[ \text{spot on full x-ray weld, spot x-ray attack weld (E=0.85)} \]
\[ t = 0.453 \quad (E=0.85) \]

\[ \text{full x-ray weld & attack weld (E=1.0)} \]
\[ t = 0.385 \quad (E=1.0) \]

Because the inner cylinder will carry a portion of the axial pressure load, the outer knuckle is not as highly stressed as a D1.5 802 knuckle would be. The magnitude of the load in the knuckle is estimated below:
Compare stresses in CC knuckles w/ EC knuckles:
(using results from Bottle NONLYN runs.)

**CC**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hoop Mem.</td>
<td>-14.50</td>
</tr>
<tr>
<td>Hoop Bnd.</td>
<td>-4650 (ps)</td>
</tr>
<tr>
<td>Mer. Mem.</td>
<td>2900</td>
</tr>
<tr>
<td>Mer. Bnd.</td>
<td>-15850 (ps)</td>
</tr>
</tbody>
</table>

**EC**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hoop Mem.</td>
<td>-20000</td>
</tr>
<tr>
<td>Hoop Bnd.</td>
<td>-7800 (ps)</td>
</tr>
<tr>
<td>Mer. Mem.</td>
<td>4300</td>
</tr>
<tr>
<td>Mer. Bnd.</td>
<td>-25000 (ps)</td>
</tr>
</tbody>
</table>

Comparisons:

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hoop Mem.</td>
<td>14530</td>
<td>0.721</td>
</tr>
<tr>
<td>Mer. Mem.</td>
<td>2900</td>
<td>0.674</td>
</tr>
<tr>
<td>Mer. Bnd.</td>
<td>15150</td>
<td>0.634</td>
</tr>
</tbody>
</table>

Summary:

Ratios vary from 63 to 73%: hence, a 4D resist.
Pressure in CC produces stresses in the knuckle equivalent to a pressure of about 27 psi. Since plastic collapse will probably control the design, and plastic collapse is related to meridional bending stresses for which the ratios are all less than 70%, an equivalent pressure of 0.7 x 4D = 28 psi will be used to size the knuckle for initial press.

Stress Intensities:

```
CC
S.I. = (Mer M+B) - (Hoop M+B)
= (2900 + 15850) - (-4650 -14530)
= 18750 + 19180
= 37930
```

```
EC
S.I. = (4300 + 25000) - (-7800 -20000)
= 29300 + 27300
= 56600
```

Ratio = 37930 / 56600 = 0.670
Determine \( R_0, a, t \) for int. pressure of 28.5 psi.

**Per UG-32:**

\[
R = 96.125 \quad L = 204.61 \quad t = 12 \quad P = 28 \quad E = \text{varies}
\]

\[
S = 18800
\]

\[
\frac{t}{S} = 0.06
\]

\[
t = 0.885 \cdot \frac{P L}{S E - 0.11 P} = 0.885 \cdot \frac{(28) \cdot 204.61}{18800 - 0.1(28)}
\]

\[
\begin{array}{c|c|c}
E & t & \text{Calc.}\n\hline
0.65 & 0.415 & \\
0.30 & 0.337 & \\
0.85 & 0.317 & \\
1.0 & 0.270 & \\
\end{array}
\]

Because the heads are relatively thin (\( Dt = 193.125 = 310 \) there is a possibility of buckling or collapse of the knuckle under internal pressure. This is checked on the following pages for operating and test equivalent pressure.
Check Outer Knuckle for Buckling:


Elastic Buckling

\[ P_{cr} = 100 \times \left[ -\frac{3.75}{5} + 0.48 \right] \left( \frac{t}{D} \right)^{2/5} \]

\[ R = D_0 = 205.0 \quad r = 12.635 \]

\[ D = 0.04 \]

let \( t = 0.415 \quad \frac{D}{t} = 494 \]

\[ P_{cr} = 100 \times \left[ 28 \times 10^6 \right] \left[ 3.7(06) + 0.68 \right] \left( 0.0202 \right)^{2/5} \]

\[ = 632 \text{ psi} \]

For a factor of safety of 4.0

\[ Pa = \frac{632}{4} = 158 \text{ psi} > 42 \text{ psi} \]

Elastic-Plastic - 1.5 Strain Hardening

\[ P_{cr} = \frac{0.285}{4} \left( 1 - 0.055(0.5) \right) \left( \frac{r}{b} \right)^{1.84} \]

\[ \left( D/t \right)^{1.53} \left( R_3/D \right)^{1.1} \]

\[ = 30000 \times 285 \times \left( 1 - 0.055(0.5) \right) \left( \frac{0.02}{28000} \right)^{1.84} \]

\[ \left( 494 \right)^{1.53} \left( 1.06 \right)^{1.1} \]

\[ = 49.4 \text{ psi} \]

Find thicknesses 0.5 in. for various factors of safety:

<table>
<thead>
<tr>
<th>F.S</th>
<th>( t )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.717</td>
</tr>
<tr>
<td>1.0</td>
<td>0.587</td>
</tr>
<tr>
<td>0.75</td>
<td>0.486</td>
</tr>
</tbody>
</table>
Plastic Collapse

\[ P_f = \frac{12.607 \left(1 + 2\frac{d}{b}\right) \left(\frac{d}{b}\right)^{1.04}}{E} \left(\frac{d}{b}\right)^{-1.09} \left(\frac{d}{b}\right)^{-1.09} \]

for \( t = 0.625 \)

\[ d/t = 523.3 \]

\[ P_f = 12.6 \left(30000 \times 1 + \frac{20000}{20000}\right) \left(1.06\right)^{1.04} \]

\[ = 43.21 \mu \text{in}^2 \]

\[ F.S. = \frac{43.21}{80} = 0.54 \text{ min.} \]

\[ \text{Test } P = 315 \times 30 + 1347 = 62.5 \mu \text{in} \]

\[ F.S. = \frac{43.21}{28 \left(\frac{525}{40}\right)} = 118 \text{ OK} \]

Check \( t = 0.415 \)

\[ d/t = 494 \]

\[ P_f = 27.68 \mu \text{in} \]

F.S. = 1.0 on operating. !!!!

Find \( t/80 \) for rod F.S.

<table>
<thead>
<tr>
<th>Rod</th>
<th>F.S.</th>
<th>( t/80 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.0</td>
<td>12</td>
<td>15.0</td>
</tr>
<tr>
<td>3.0</td>
<td>84</td>
<td>1.15</td>
</tr>
<tr>
<td>2.0</td>
<td>56</td>
<td>0.792</td>
</tr>
<tr>
<td>1.5</td>
<td>63</td>
<td>0.663</td>
</tr>
</tbody>
</table>

Use 5/8" outer knuckle
Not Applicable because it is
Head size calculated by ASME Section VIII Division 2 rules
CROWN - INT. PRESSURE

Under internal pressure the crown behaves like a torus. The hoop stress in a torus can be calculated by:

\[ \sigma = \frac{Pr}{2t} \left( \frac{2R_0 + r \sin \theta}{r_0 + r \sin \theta} \right) \]

\[ \sigma = \frac{40(86875)}{2(51.75 - 37.85)} \]

\[ = \frac{40(86875)}{2(51.75)} (4.72) < 0.65 \]

\[ r = \frac{30}{2} \]

\[ R_0 = 51 \frac{3}{4} \]

\[ \Theta = 360 - 26 = 334^0 \]

\[ r \sin \Theta = 13 \frac{785}{5} \]

\[ = 2 \cdot 0 \frac{1}{9} \]

This formula appears to be overly conservative based on Barineau's stress results which indicate that membrane hoop and meridional stresses are both on the order of 5000 psi. The required thickness can be estimated by allowing these stresses to approach SE, the basic O.D. allowable.

\[ \frac{Pr}{2t} \leq 15(165) = 12220 \]

\[ t_r = t \left( \frac{5000}{12220} \right) \]

\[ = 0.25 \left( \frac{5000}{12220} \right) \]

\[ = 0.256^" \]

Check Per 1-6.9 for dished heads:

\[ t = \frac{5PL}{65} = \frac{5(40785)}{65} = 0.156^" \]

\[ t = 0.256^" \]

For Int. Pressure

\[ t_r = 0.256^" \]
While not Code Calculations in the strict sense, the following 6 sheets (not numbered) of calculations by RDL dated 8/14/81 serve to establish the integrity of the Code vessel head subjected to external pressure.
Crown - External Pressure

The most likely mode of failure of the head's under external pressure is asymmetric buckling. Per WRC Bulletin 119 and 227 (below) this mode of failure is related to knuckle stresses and deformations. Table 7 of WRC-227 indicates that an equivalent torospherical head (D = 193.5, t = 5/8", L = 204") with "B" = 0.06 would probably collapse at the onset of yielding in the knuckle region. The Bulletin uses the Von Mises criterion. For convenience we will use the maximum shear stress theory which will be slightly conservative.

From the Battelle report (Table 5, p18) the maximum stress in the body is 32.6ksi for an internal pressure of 40 psi. Assuming linearity, the external pressure required to produce yielding in the knuckle (S_y = 30 ksi = 57) is:

\[
P_{ext} = 40 \left(\frac{30}{57.5}\right) = 31.2 \text{ksi} = \text{Crown Buckling Pressure.}
\]

Note that this collapse pressure is less than that predicted by Equation 8 of WRC-227 where a value of \( L = 204\) " is used:

\[
P_c = 30000 \times 10
\]

\[
(1.158 - 1.437 \log_{0.031} + 3.55 \log_{0.06})
\]

\[
= 42.8 \text{ psi}
\]

The minimum critical pressure is greater than 30 psi which is the tab minimum for safe. Therefore, a knuckle thickness of \( 5/8 \)" will ensure that yielding does not occur at 30 psi, promoting asymmetric buckling at the crown. The crown lining also be used to prevent collapse of the crown assuming a clamped edge. This is done in Battelle's report on sheet 2P-CC where a critical pressure of 80 psi is determined. Since there appears to be some excess thickness in the crown, Battelle's procedure


will be used to minimize the crown thickness:


\[
\lambda_2 = 2 \left( \frac{1}{\sin \theta_2} \right)
\]

\[
\theta_2 = \sin^{-1} \left( \frac{2 \lambda_2}{\pi} \right) = \sin^{-1} \left( \frac{121.3}{23.4} \right) = 17.3^\circ
\]

\[
H_2 = R_2 \left( 1 - \cos \theta_2 \right) = 204 \left( 1 - \cos 17.3^\circ \right) = 9.23''
\]

\[
\lambda_2 = 2 \left[ 3 \left( 1 - \nu^2 \right) \right]^{1/4} \left( \frac{R}{t} \right)^2
\]

\[
= 2 \left[ 3.91 \right] \left( \frac{25}{2.570} \right) \sqrt{9.23}
\]

\[
= 814.4 \text{ in}^2
\]

By \( t = 0.256 \)

\[
\lambda_2 = 15.43 \quad \text{Too large,} \quad \frac{q_{cr}}{q_0} \approx 0.15
\]

\[
q_{cr} = 0.15 (814.4)(1.26 c)^2
\]

\[
= 20 \text{ psi} \quad \text{No Good}
\]

\[
t = 0.375
\]

\[
\lambda_2 = 12.75 \quad \frac{q_{cr}}{q_0} \approx 0.2
\]

\[
q_{cr} = 0.2 (814.4)(0.375)^2
\]

\[
= 23 \text{ psi} \quad \text{No Good}
\]

\[
t = 0.4375
\]

\[
\lambda_2 = 11.8 \quad \frac{q_{cr}}{q_0} \approx 0.25
\]

\[
q_{cr} = 0.25 (814.4)(0.4375)^2
\]

\[
= 40 \text{ psi} \quad \text{OK} \quad > 30 \text{ psi}
\]

Also check per ASME for spherical shells.
App. D of the Code states that the critical buckling pressure for a spherical shell is equal to:

\[ P_{cr} = \frac{Pr_0}{R_0} = 0.125 \frac{Et}{R_0} \]

\[ P = 204.6 \]

\[ E = 28 \times 10^6 \]

for \( P_{cr} = 30 \text{ psi} \), solve for \( t \)

\[ \frac{30(204)}{2t} = 112.5(28 \times 10^6) t \]

\[ t^2 = \frac{30(204)^2}{2(112.5)(28 \times 10^6)} \]

\[ t = 0.422'' \sqrt{ } \]

Check stability of a curved panel:

Simple Support

\[ \alpha = 87^\circ \]

\[ \frac{37^\circ}{2} = 2t \]

\[ t = \sqrt{\left[ \frac{30(12)(87)^2(1-3^2)}{28 \times 10^6 (1329^2 - 1)} \right]} \]

\[ t = 0.323 \text{ rad} \]

\[ \alpha = \frac{37^\circ}{2} = 0.323 \text{ rad} \]

\[ t = 0.435'' \sqrt{ } \]

for a Safety Factor of 4.0 (ASME), \( P_{cr} = 4(15) = 60 \)

\[ t = 0.525'' \]

Use 5/8" Crown.
Check a clamped curved panel (Quarter, case 34):

\[
\begin{array}{cc}
\theta = 15.5^\circ & k \\
13.2 & 2.135 \\
10 & 36.91 \\
13.5 & 1.687 \\
13.8 & 1.71 \\
14.0 & 1.91 \\
20^\circ & 12 & 0.9 \\
12 & 1.89 \\
1 & 3 \\
4 \\
\end{array}
\]

\[
\frac{25 \times 10^3 (3.8^2 - 1)}{12 (723^3 (91))} = 30
\]

Since the buckle will allow a small amount of edge rotation, the critical pressure will likely be closer to the hinged case. Therefore, the min t for the crown will be set at 0.344".
The knuckle is very sharp which will lead to high bending and circumferential stresses. Battelle's stress results show a max stress intensity of 38.5 ksi in the knuckle for an internal pressure of 40 psi. This value is less than the guaranteed 45E (41.65 ksi) and 35M (31.2 ksi) and is considered acceptable. The maximum membrane stress intensity (which is equal to the max membrane stress in this instance) from Battelle's report is about 18 ksi (18.3 ksi) and is also acceptable. Since both values are close to the diff. allowable (especially the membrane stress) it does not appear that reducing the thickness of this knuckle is advisable.

For external pressure it was shown above that the critical pressure for buckling of the crown is not at the pressure at which the knuckle yields. For the 5/8" knuckle this was shown to be 31.2 psi which is just above the minimum critical p of 30 psi required by the code.
Buckling of the inner knuckle will be checked using Druker's formula. First the equivalent pressure for the fictitious load (shown on the previous page) must be determined.

From Battelle's Report the axial stress in the inner cylinder under 40 psi external pressure is 2200 psi. The axial force in the inner cylinder under 15 psi external pressure would be:

\[
N_f = 2200 \text{ psi} \times \frac{3.1416}{14} \times \left( \frac{15}{40} \right) \\
= 361 \text{ lb/in}
\]

The equivalent internal pressure in the fictitious cylinder is:

\[
\frac{P_2}{P_2^*} = \frac{N_f}{10} \\
P_2^* = 361 \times \frac{10}{14} = 10.94 \text{ psi} \quad \text{use 11 psi.}
\]

Druker's Formula:

\[
\frac{P_2^*}{P_2} = 30000 \left[ 1.33 + 5.5 \left( 0.09 \right) \right] 0.04 + 28 \left[ 1 - 2 \times (0.09)^2 \right] \frac{0.04}{-0.006}
\]

\[
P_2^* = 79 \text{ psi}
\]

The corresponding external collapse pressure for the inner knuckle is:

\[
P_2 = 79 \left( \frac{15}{14} \right) = 88 \text{ psi}
\]

\[
S = \text{factor of safety} = \frac{88}{11.8} = 7.4 \quad \text{OK}
\]
Acme @ bottom of cyl

\[ P = 15 + \text{head} + 15 \]
\[ = 30 + \text{head} \]

\[ \text{head} = \left( \frac{13}{8} \right) \times \text{depth} \times \frac{62.4}{1728} \]
\[ = \left( \frac{1}{2} \times 128.94 + 64.375 \right) \times \frac{62.4}{1728} \]
\[ = 62.4 \times \frac{62.4}{1728} \]
\[ = 0.52 \text{ psi} \Rightarrow 7 \text{ psi} \]

\[ P = 37 \text{ psi} \]

\[ \tan t = \frac{3}{8} \]

\[ \frac{t}{d} = \frac{109.935}{64.375 + 3} = 1.687 \]

\[ \frac{d}{t} = \frac{64.375}{375} = 172.7 \]

\[ A_1 = 0.0033 \]

\[ A_2 = 0.1 \times 10^{-3} \]

\[ A_3 = 140 \]

\[ A_4 = 0.469 \times 10^{-3} \]

\[ A_5 = 0.227 \]

\[ \frac{A_5}{A_4} = \frac{0.469}{0.227} \]

\[ \frac{A_5}{A_4} = \frac{11.691}{4.69} \]

\[ \frac{A_5}{A_4} = 0.271 \times 10^{-3} \]

\[ A = \frac{A_1 + A_2 + A_3 + A_4 + A_5}{5} \]

\[ A = \frac{0.0033 + 0.469 \times 10^{-3} + 140 + 0.227}{5} \]

\[ A = \frac{1.917 \times 10^{-3}}{5} \]

\[ A = 0.383 \times 10^{-3} \]

\[ A = 0.417 \times 10^{-3} \]

\[ \frac{d}{t} = \frac{172.7}{4804} \]

\[ P_d = \frac{4}{3} \times \frac{4804}{172.7} \]

\[ = 37.1 \text{ psi} > 37 \text{ ok} \]

Check axial comp units & press.

\[ \sigma = \frac{2P}{t} = 15(3.3) \]

\[ = 660 \text{ psi} \]

\[ A = \frac{1.25t}{2A} \]

\[ B = 10000 > 660 \text{ ok} \]

\[ \frac{3}{8} \text{" Rate OK} \]
CC Inner Vessel = 23000 (EN. 39)

![Diagram showing vessel dimensions and calculations]

\[ V_0 = \frac{\pi}{4} (192.5^2 - 4.375^2) 116 = 1730 \text{ ft}^3 \]

\[ \omega = \frac{8661}{1.4(624)(1730)} = 15100 \text{ #} \]

Total \( \omega \):
- Shell: 23000
- Argon: 15100
- Final, etc.: 100
- Total: 175000 #

Look at the stiffening rings with a length of shell. Most of the load is transferred into the ring from the remainder of the vessel by shear. Initially (and conservatively) look at a ring loaded by its own weight (Ref. Roark, 4th Ed. p. 176 Case 19). (Subsequent analysis indicated that results for shear only are only a few percent lower.) For each of 2 rings:

\[ \omega_{\text{shell}} = \frac{175000 \text{ #}}{2\pi R} \]

\[ \omega = \sin^{-1} 73.332 \]

\[ \omega = 47.72^\circ = 0.848 \text{ rad} \]

\[ C = \cos \theta = 0.763 \]

\[ 2 = \sin \theta \quad \theta = \cos \theta \]

\[ R = \frac{192.25}{1.1} = 96.125 \]

\[ M = M_{x=0} = \omega R^2 \left( \frac{3}{2} + C + 3 - \pi - \theta - \frac{2}{3} \right) \]

\[ T = \omega R^2 (s^2 - \frac{1}{2}) \]

Previous work showed that max values occur at the load and at about 93°.

\[ \theta = \arcsin \left( M_y = M - \frac{1}{2} R (\text{load}) + \omega R^2 (x_2 + \mu - 1) \right) \]

\[ \frac{2}{3} \omega R^2 \left( \frac{3}{2} + C + \theta - \frac{3}{2} + x_2 + \mu - 1 - \frac{1}{2} \right) \]

\[ \omega R^2 \left( C + \theta S + x_2 - \pi - x_2 + \frac{1}{2} - \frac{2}{3} u + \frac{4}{3} \right) \]
\[ M_x = \omega R^2 \left[ c + \frac{1}{2} u + xz - \pi z + C_2 + u s^2 \right] \]
\[ = \omega R^2 \left[ 0.47 + \frac{1}{2} (2.00 \pi) + x \sin x - \pi \sin x + 0.62 + 0.58 \cos x \right] \]
\[ = \omega R^2 \left[ 1.309 + 0.082 \cos x - (\pi - x) \sin x \right] \]

At load \((x = \Theta)\)

\[ M = \omega R^2 \left( 1.309 + 0.082 \cos 93^\circ - (\pi - 165^\circ) \sin 93^\circ \right) \]
\[ = \omega R^2 (0.2743) \]

\( x = 93^\circ = 1.623 \text{ rad.} \)

\[ M = \omega R^2 \left( 1.309 + 0.082 \cos 93^\circ - (\pi - 165^\circ) \sin 93^\circ \right) \]
\[ = \omega R^2 (-24.4) \]

At \( x = 0^\circ \), \( z = 5 \sin x = 0 \), \( u = \cos x = 1 \)

\[ M = M_r = \omega R^2 \left( \frac{1}{2} (x + 0.67 + 1.68(1.763) - \pi (1.763) + (1.763)^2 \right) \]
\[ = \omega R^2 (-0.0058) \]

Apical Loads

\( 0 \leq x \leq \pi/2; \)
\[ T = \omega R \left[ u(\frac{x}{2} - \frac{z}{2}) - (\pi - x) \right] \]
\[ = \omega R \left[ 0.62 \cos x - (\pi - x) \sin x \right] \]

At load \( x = 49.72^\circ \)
\[ T = \omega R (-1.682) \]

At \( x = 93^\circ \)
\[ T = \omega R (-1.521) \]

At \( x = 0^\circ \)
\[ T = t = \omega R (0.082) \]

For \( \omega = 144.9 \)
\[ R = 97.625 \]
\( x = 49.72^\circ \)
\[ T = -23793 \text{ ft} \]
\( x = 93^\circ \)
\[ T = -21516 \text{ ft} \]
\( x = 0^\circ \)
\[ T = 1160 \text{ ft} \]
For the supports, internal pressure is likely to be the controlling condition since the vessel will be empty when it is evacuated.

The membrane hoop stress in the cylinder due to pressure is:

\[
\sigma_p = \frac{P \cdot t}{t} = \frac{40(96.125 + 3.125)}{.625} = 6172 \text{ psi}
\]

At nozzle locations the stress is higher due to dilation of the opening. Since the code design attempts to limit local stresses to 155, assume that the hoop stress at openings is 15 x 6172 = 9260 psi. The allowable membrane hoop stress is given in 6G-23(b) as 8. In addition the weld joint efficiency must be considered. Since the stress will vary significantly around the circumference, the value of 8 will be dependent on the location. At Category A welds a value of 8E = 18800 x .85 = 12200 psi will be used. At other locations away from openings a value of 18800 x .85 = 15040 psi will be used. At openings the stress will be considered to be local and the allowable value will be taken as 15 x 18800 x .8 = 22560 psi.

Allowable stresses for the support loads only will be equal to the above values less the membrane pressure stress:

<table>
<thead>
<tr>
<th>Location</th>
<th>A + Support</th>
<th>Support Only</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nozzles</td>
<td>22560</td>
<td>22560 - 9260 = 13300</td>
</tr>
<tr>
<td>Shell</td>
<td>15040</td>
<td>15040 - 6180 = 8860</td>
</tr>
<tr>
<td>Cat. A. Prod</td>
<td>12200</td>
<td>12200 - 6200 = 6000</td>
</tr>
</tbody>
</table>

* These values reflect no radiography.
At the Support Nozzle ($\theta = 0$)

The moment is such that the shell side is in tension. Therefore, the increased membrane stress in the shell must be less than

$$22560 - 9260 = 13300 \text{ psi}.$$ 

$$\frac{M}{S} \leq 13300 \text{ psi}.$$ 

$$\sigma > \frac{\omega R^2 (2.743)}{13300}$$

$$\omega = 144.9 \text{ #/in}^2$$

$$R = 96.125 + 1.5 \text{ (conservative)}$$

$$S_{yg} \geq 144.9(97.625)^2 (2.743)$$

$$\frac{S_{yg}}{S_h} \geq \frac{375805}{13300}$$

$$S_h \geq 28.5 \text{ in}^3$$

Cover Plate:

Stress in the cover plate due to bending will be limited to 0.66 $\sigma_y$ per AISC. $\sigma_y (30) = 20 \text{ ksi} \times 1200 \mu$$

$$= 19.8 \text{ ksi per axial comp.}$$

(See next sh.)

$$S_c \geq \frac{M}{S} = \frac{375805}{18800} = 20.15 \text{ in}^3$$

Determine the allowable stress the width/thickness ratio for the compression flange (cover plate) must be less than $190/1.5 \sqrt{t}$ (Sec. 15.1.4.1 of AISC) — $w = 12\text{ in}$, $S_h = 15.75$.

$$\frac{w}{t} < \frac{190}{1.5 \sqrt{t}}$$

$$t \geq 0.154$$

:. 1/2" cover plate req'd near support nozzles.
at $\theta = 90^\circ$ (Max) $M = 0.2641 \omega R^2$

The moment is such that the shell will be in compression. Hence, the stress will be maximized when the internal pressure is zero. The resulting shell allowable stress will be $15040$ psi. (Note $15040 - 6183$)

$$M = 0.2641 \omega R^2 \rho (\omega^2)$$

$$= 0.2641 (\pi 4.9)(97.625)^2$$

$$= 364720 \text{ in.}^2 \text{lb}$$

$$S_h = \frac{364720}{15040}$$

$$S_h = 24.25 \text{ in.}^3 \text{ (See below)}$$

For the cover plate, $S_{16} = 0.4(32) = 20.65$:

$$S_c = \frac{364720}{20000}$$

$$\geq 18.24 \text{ in.}^3$$

Assume an area for the stiffener of $21 \text{ in.}^2$

Allowable stress due to axial load $T = \frac{T}{A} = -\frac{21516}{21} = -1025 \text{ psi}$

Since this is compression, it must be reduced in the cover plate and increase the stress in the shell.

$$S_h = \frac{364720}{15040 - 1025}$$

$$= 26.02 \text{ in.}^3 \text{ LOD}$$
TRY RING AS DESIGNED 14" ID

\[ \Delta = 19.35 \]

\[ t = 0.08 \]

\[ 12.75 + 1.125^2 = 18.75 + 12.1 = 30.85^2 \]

(Note that the additional stiffness provided by the head knuckle has not been considered except that the shell has been assumed to extend beyond the joint for the calculations below.)

<table>
<thead>
<tr>
<th>Shell</th>
<th>( A )</th>
<th>( y )</th>
<th>( A_y )</th>
<th>( Adz )</th>
<th>( T_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angle</td>
<td>2.75</td>
<td>1.375</td>
<td>1.080</td>
<td>0.381</td>
<td>0.141</td>
</tr>
<tr>
<td>Ring</td>
<td>6.25</td>
<td>2.0</td>
<td>1.25</td>
<td>5.657</td>
<td>0.021</td>
</tr>
<tr>
<td>Cover Pl</td>
<td>2.75</td>
<td>2.75</td>
<td>18.703</td>
<td>13.978</td>
<td>0.164</td>
</tr>
<tr>
<td>Total</td>
<td>24.780</td>
<td>25.838</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ G = \frac{15.99}{20.843} = 0.767 \]

\[ S_y = \frac{14.816}{0.767} = 19.317 \ (Shell \ stress) \]

\[ S_x = \frac{14.816}{9.241} = 1.624 \ (Cover \ Pl \ stress) \]

No Good!!

TRY 3/4" COVER PLATE

<table>
<thead>
<tr>
<th>Shell</th>
<th>( A )</th>
<th>( y )</th>
<th>( A_y )</th>
<th>( Adz )</th>
<th>( T_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angle</td>
<td>2.75</td>
<td>1.375</td>
<td>1.080</td>
<td>0.381</td>
<td>0.141</td>
</tr>
<tr>
<td>Fl.</td>
<td>6.25</td>
<td>2.0</td>
<td>1.25</td>
<td>5.657</td>
<td>0.021</td>
</tr>
<tr>
<td>Cover Pl</td>
<td>3.175</td>
<td>3.175</td>
<td>18.903</td>
<td>13.978</td>
<td>0.164</td>
</tr>
<tr>
<td>Total</td>
<td>24.780</td>
<td>25.838</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ G = \frac{25.838}{24.780} = 1.0427 \]

\[ S_{sh} = \frac{23.725}{1.7302} = 13.706 \ (Shell \ stress) \]

\[ S_o = \frac{23.725}{2.625-10.427} = 14.99 \ (Shell \ stress) \]

No Good
**TRY 5/8” Cover R.**

<table>
<thead>
<tr>
<th></th>
<th>A</th>
<th>H</th>
<th>Ag</th>
<th>Ad^2</th>
<th>( \bar{f} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>SHELL</td>
<td>15.625</td>
<td>5125</td>
<td>4.883</td>
<td>11.242</td>
<td>0.5</td>
</tr>
<tr>
<td>WELDS</td>
<td>0.75</td>
<td>1375</td>
<td>3.031</td>
<td>1.034</td>
<td></td>
</tr>
<tr>
<td>FLANGES</td>
<td>2.5</td>
<td>2.5</td>
<td>4.8</td>
<td>1.440</td>
<td></td>
</tr>
<tr>
<td>COVER P.</td>
<td>9.842</td>
<td>2.438</td>
<td>25994</td>
<td>16060</td>
<td>0.3</td>
</tr>
</tbody>
</table>

\[ \bar{f} = \frac{31158}{26844} = 1.1607 \]

\[ S_u = \frac{28.576}{11607 \times 3.75} = 33.69 \text{ OK} > 34.21 \]

\[ S_2 = \frac{28.576}{2.75 \times 11607} = 17.98 < 17.24 \]

\[ \text{400,000} \]

\[ \text{30,000} \]

\[ \text{20,000} \]

\[ \text{Method} \]

\[ 20.5^2 = 0.19 \]

\[ \frac{20.5^2}{4} \approx 21 \text{ in}^3 \]

\[ S \approx 21 \text{ in}^3 \]

\[ 3” \text{ WILL HAVE} \]

\[ 3” \text{ CANTILEVER} \]

\[ \text{in area of support} \]

\[ \text{use 3/4” cover R.} \]

\[ \text{around to a point above the} \]

\[ \text{support and around} \]

\[ \text{to a point above the} \]

\[ \text{where the deck is adequate.} \]

\[ \text{See next page} \]
Plot Moments vs. Adj. Angle to determine cover R thickness limits:

\[ M_x = \frac{1}{4} (97.625)^2 \left[ 1.082 \cos x + x \sin x - 1.082 \right] \]

\[ M_x = \frac{1}{4} (97.625)^2 \left[ 1.309 + 1.082 \cos x - (\pi - x) \sin x \right] \]

<table>
<thead>
<tr>
<th>x</th>
<th>M</th>
<th>x</th>
<th>M</th>
<th>x</th>
<th>M</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-8240</td>
<td>65</td>
<td>196790</td>
<td>120</td>
<td>-19120</td>
</tr>
<tr>
<td>5</td>
<td>-3447</td>
<td>60</td>
<td>50800</td>
<td>125</td>
<td>-13550</td>
</tr>
<tr>
<td>10</td>
<td>10800</td>
<td>65</td>
<td>-72900</td>
<td>130</td>
<td>-75950</td>
</tr>
<tr>
<td>15</td>
<td>34160</td>
<td>70</td>
<td>-172650</td>
<td>135</td>
<td>-15810</td>
</tr>
<tr>
<td>20</td>
<td>60660</td>
<td>75</td>
<td>-250110</td>
<td>140</td>
<td>-43350</td>
</tr>
<tr>
<td>25</td>
<td>105720</td>
<td>80</td>
<td>-306480</td>
<td>145</td>
<td>-99850</td>
</tr>
<tr>
<td>30</td>
<td>15220</td>
<td>85</td>
<td>-343110</td>
<td>150</td>
<td>152130</td>
</tr>
<tr>
<td>35</td>
<td>204078</td>
<td>90</td>
<td>-341540</td>
<td>155</td>
<td>198830</td>
</tr>
<tr>
<td>40</td>
<td>260225</td>
<td>95</td>
<td>-344560</td>
<td>160</td>
<td>231725</td>
</tr>
<tr>
<td>45</td>
<td>319020</td>
<td>100</td>
<td>-363460</td>
<td>165</td>
<td>270825</td>
</tr>
<tr>
<td>49.72</td>
<td>375440</td>
<td>105</td>
<td>-350690</td>
<td>170</td>
<td>294330</td>
</tr>
<tr>
<td>50</td>
<td>367890</td>
<td>110</td>
<td>-325140</td>
<td>175</td>
<td>308670</td>
</tr>
</tbody>
</table>

(Plotted on next page)

For a 1/4" cover R: S = 9.0

\[ 20000 \leq \frac{M}{9.0} \]

\[ M \leq 180,000 \]

Plot on next page shows that the 1/4" cover plate could be used on small areas only. Therefore, the cover plate will be the same thickness all around.
USE 3/4" COVER PLATE.

\[ A = \frac{3}{4} \times 15.75 \]

\[ h = 12 \times 1/2 \]

\[ 5/8 \times 25" \]

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>WEBS</td>
<td>.75</td>
<td>1.375</td>
<td>1031</td>
<td>1008</td>
<td>—</td>
</tr>
<tr>
<td>FL.</td>
<td>.625</td>
<td>2.0</td>
<td>125</td>
<td>1330</td>
<td>—</td>
</tr>
<tr>
<td>COVER R</td>
<td>11.813</td>
<td>2.50</td>
<td>29.533</td>
<td>17.767</td>
<td>.55</td>
</tr>
<tr>
<td></td>
<td>28.813</td>
<td></td>
<td>36.697</td>
<td></td>
<td>I = 33.588</td>
</tr>
</tbody>
</table>

\[ \bar{y} = \frac{36.697}{28.813} = 1.273c \]

\[ R = 96.125 + 1.274 = 97.4" \]

**Check Shear**

The maximum shear occurs at the load,

\[ x = 0 \]

\[ V = -T \varepsilon + \omega R (x u - \pi u) \]

\[ = -\omega R (5^2 - \frac{1}{2}) \varepsilon + \omega R (x u - \pi u) \]

\[ = \omega R \left[ (\pi - \pi) \cos \theta - \sin \theta (\sin^2 \theta - \frac{1}{2}) \right] \]

\[ = \omega R (-1534) \]

\[ = 144.9 (96.125 + 1.274) (-1534) \]

\[ = -21650 \] # OK

\[ \bar{v} = \frac{-21650}{2(1/2)(2.875)} = 15/100 \text{ psi} \]

<table>
<thead>
<tr>
<th>Allowable Stress:</th>
</tr>
</thead>
<tbody>
<tr>
<td>( V = 12000 (2)(2.5)(2.875) )</td>
</tr>
<tr>
<td>= 19250</td>
</tr>
</tbody>
</table>

\[ f_{v} = 0.4 f_{l} = 0.4(30000) = 12000 < 15100 \] No Good!

**Final Width Req:**

\[ w = \frac{21650}{2(2.875)(1200)} = 0.3125 \Rightarrow 5/16" \]

Plot the variation of the shear force & determine where design is good.
For \( 0 \leq x \leq \theta \),
\[
V = \omega R (xu - z(s^2 - \frac{1}{2})) \\
= \omega R (x \cos x - \sin x (\sin^2 x - \frac{1}{2})) \\
= \omega R (x \cos x - .082 \sin x) \\
= (144.9)(90.125 + 125)(x \cos x - .082 \sin x)
\]

For \( \theta \leq x \leq \pi \),
\[
V = \omega R \left[ (xu - \pi u) - z(s^2 - \frac{1}{2}) \right] \\
= \omega R \left[ (x - \pi) \cos x - .082 \sin x \right] \\
= (144.9)(90.125 + 125)(x - \pi) \cos x - .082 \sin x
\]

<table>
<thead>
<tr>
<th>( K )</th>
<th>( V )</th>
<th>( \theta )</th>
<th>( V )</th>
<th>( \pi )</th>
<th>( V )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>65</td>
<td>-1302</td>
<td>135</td>
<td>7018</td>
</tr>
<tr>
<td>5</td>
<td>1126</td>
<td>70</td>
<td>-10352</td>
<td>140</td>
<td>6802</td>
</tr>
<tr>
<td>10</td>
<td>2224</td>
<td>75</td>
<td>-7810</td>
<td>145</td>
<td>6377</td>
</tr>
<tr>
<td>15</td>
<td>3249</td>
<td>80</td>
<td>-5414</td>
<td>150</td>
<td>5820</td>
</tr>
<tr>
<td>20</td>
<td>4232</td>
<td>85</td>
<td>-3192</td>
<td>155</td>
<td>5091</td>
</tr>
<tr>
<td>25</td>
<td>5091</td>
<td>90</td>
<td>-1157</td>
<td>160</td>
<td>4232</td>
</tr>
<tr>
<td>30</td>
<td>5826</td>
<td>95</td>
<td>472</td>
<td>165</td>
<td>3269</td>
</tr>
<tr>
<td>35</td>
<td>6397</td>
<td>100</td>
<td>2282</td>
<td>170</td>
<td>2224</td>
</tr>
<tr>
<td>40</td>
<td>6802</td>
<td>105</td>
<td>3463</td>
<td>175</td>
<td>1126</td>
</tr>
<tr>
<td>45</td>
<td>7018</td>
<td>110</td>
<td>4809</td>
<td>180</td>
<td>0</td>
</tr>
<tr>
<td>49.72</td>
<td>-2162</td>
<td>115</td>
<td>571</td>
<td>180</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>-21464</td>
<td>120</td>
<td>6384</td>
<td>180</td>
<td></td>
</tr>
<tr>
<td>55</td>
<td>-13604</td>
<td>125</td>
<td>6821</td>
<td>180</td>
<td></td>
</tr>
<tr>
<td>60</td>
<td>-15776</td>
<td>130</td>
<td>7028</td>
<td>180</td>
<td></td>
</tr>
</tbody>
</table>

Additional Strength is required only for \( \pm 10^\circ \) (\( \pm 18^\circ \)) above the support nozzle.


Allowable Shear Force for Current Ring Design.
Check weld size for gage R.

Allowable weld size, per AISC \( \S \) 15.3, Table 15.3

\[ T_{\text{allow}} = 0.3 \times \text{Nom. Tensile Strength of the weld metal} \]
or

\[ T_{\text{allow}} = 0.4 T_{\text{Fy}} \text{ of base metal.} \]

For a 1" fillet:

\[ \text{Shear/Force} \quad \frac{T_{\text{allow}}}{\text{length}} = \frac{0.707 \times \text{length}}{0.3 \times \text{T.S.}} \]

For E508/E509 Welds - \( T_{\text{Fy}} = 50 \times 10^6 \)

\[ \text{Shear} < 0.707(3)(30) \text{(Length)} < 17 \text{ k} \]

\[ \text{Bonded joint:} \quad \frac{T_{\text{allow}}}{\text{length}} = \frac{\text{Shear}}{1" \times \text{length}} \leq 0.4(30) \]

\[ \text{Shear} < 12 \text{ k}, \text{ Controls} \]

\[ f = 120 \text{ kips/in of weld size} \]

\[ W = \frac{VQ}{n^2} \]

\[ W = \frac{1.450(3/4)(15.75)(1.226)}{(2)(33.59)} = 4.67 \text{ kips/in} \]

\[ \Delta W = 4.67 = 0.389" \]

Use 3/8" fillets since welds need not be larger than the attached pieces.

Away from the support where the loads are never more than 1/3 as large as those. Therefore use 3/4" fillets elsewhere.
SIZE STIFF RING FOR STANCHION WHICH DOES NOT PIERCE INNER VESSEL:

\[
\begin{align*}
\text{Shell} & : \frac{A}{y} = 7.97 \quad \frac{y}{y_0} = 3.716 \quad A_y = 24.9 \quad A_{1.0} = 9.202 \quad \Gamma = 0.26 \\
\text{Angle webs} & : 0.375 \quad 2.0 \quad 2.0 \quad 0.311 \quad 1.07 \\
\text{Flanges} & : 0.375 \quad 2.0 \quad 2.0 \quad 0.141 \quad 1.07 \\
\text{Cover PL} & : 5.875 \quad 2.875 \quad 16.71 \quad 12.870 \quad 40.9 \\
\end{align*}
\]

\[y = \frac{20.07}{14.49} = 1.387\]

\[I = 47.89 \times 1.387 = 66.26 \times 1.387 = 34.5 \text{ in}^3 \quad > 28.5 \text{ in}^3 \quad \text{OK}\]

\[S_{op} = \frac{47.89}{34.5} = 1.40 \text{ in}^3 \quad > 20.45 \text{ in}^3 \quad \text{OK}\]

**Check Shear**

\[V = 22 \text{ kip}\]

\[f_s = \frac{V}{A} = \frac{22}{2(3.625)25} = 0.121 \text{ ksi} \quad F_s = 0.4F_y = 12 \text{ ksi} \quad \text{OK}\]

USE 8" CAR. MUST BE WELDED TO SHELL & COVER FLANGE TO PREVENT OVER-STRESSING SHELL. IF RAIN IS 50 COW NUTRED, USE 4" EXTENSION TO TRANSFER SHEAR.
Check Rings at Signal Ports NIP

<table>
<thead>
<tr>
<th>Shell</th>
<th>A</th>
<th>Y</th>
<th>A_y</th>
<th>A_d^2</th>
<th>I_0</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.07</td>
<td>.3125</td>
<td>1.58</td>
<td>3.36</td>
<td>.16</td>
<td></td>
</tr>
<tr>
<td>Neck</td>
<td>1.44</td>
<td>1.25</td>
<td>1.8</td>
<td>.02</td>
<td>2.43</td>
</tr>
<tr>
<td>Angle</td>
<td>.395</td>
<td>.395</td>
<td>.43</td>
<td>.02</td>
<td>.07</td>
</tr>
<tr>
<td>&quot; Al</td>
<td>3.13</td>
<td>2.0</td>
<td>2.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cover A.</td>
<td>2.67</td>
<td>2.5</td>
<td>6.68</td>
<td>5.04</td>
<td>0.13</td>
</tr>
</tbody>
</table>

\[ g = \frac{11.11}{2.868} = 3.912 \]
\[ I = 11.47 \text{ in}^4 \quad \text{ONE S100 ONLY!} \]
\[ I_{tot} = 22.94 \text{ in}^4 \]

\[ S_{shell} = \frac{22.94}{20.37} = 1.126 \]
\[ S_{cover} = \frac{22.94}{13.12} = 1.74 \]

From Sn 22 the stress limit in the shell is 13300 psi.

\[ \frac{M}{S} < 13300 \]

Min allowable \[ M < 13300 (20.4) \]

\[ M < 271000 \text{ in-lb} \]

From Sn 25 the stress limit in cover rate is 18800 psi.

Max allowable \[ M < 18800 (13.12) \]

\[ M < 247000 \text{ in-lb} \]

Other Nozzles

For the smaller nozzles the section modulus will be substantially larger and are deemed OK by inspection.
An ANSYS model of the beam was run for internal pressure. Element 247 was a beam element between nodes 999 and 3534. The resulting axial force in the beam is 10225 lb.

\[ \sqrt{(8.62^2 + 13.0^2)} = 82.65'' \]
\[ \theta = \sin^{-1} \left( \frac{29.87}{96.7} \right) = 20.3^\circ \]

\[ p = 10.3^\circ \cos 20.3^\circ \]
\[ = 9.67^\circ \]

\[ u = 10.3 \sin 20.3^\circ \]

\[ = 3.6^\circ \]

The table on the following page (from WREC-107) demonstrates that shell stresses at the bypass tube are not excessive.
Table 3—Computation Sheet for Local Stresses in Spherical Shells (Hollow Attachment)

from WRC-107

1. Applied Loads

<table>
<thead>
<tr>
<th>Term</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P$</td>
<td>10000 lb</td>
</tr>
<tr>
<td>$V_1$</td>
<td>3000 lb</td>
</tr>
<tr>
<td>$V_2$</td>
<td>1 lb</td>
</tr>
</tbody>
</table>

2. Geometry

<table>
<thead>
<tr>
<th>Term</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
<tr>
<td>$I$</td>
<td>$\frac{1}{12} \pi b^4$</td>
</tr>
<tr>
<td>$R_m$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
</tbody>
</table>

3. Geometric Parameters

<table>
<thead>
<tr>
<th>Term</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I$</td>
<td>$\frac{1}{12} \pi b^4$</td>
</tr>
<tr>
<td>$M$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
</tbody>
</table>

4. Stress Concentration Factors

<table>
<thead>
<tr>
<th>Term</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_n$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
<tr>
<td>$K_b$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
</tbody>
</table>

HOLLOW ATTACHMENT

<table>
<thead>
<tr>
<th>Term</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
<tr>
<td>$N_m$</td>
<td>$\frac{1}{2} \pi b^2 t$</td>
</tr>
</tbody>
</table>

From Fig. Read curves for Compute absolute values of stress and enter result:

<table>
<thead>
<tr>
<th>Stress Intensity - $S$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$S = \sqrt{\sigma_x^2 + \sigma_y^2}$</td>
</tr>
</tbody>
</table>

When $T \neq 0$, $S$ = largest absolute magnitude of either $\sigma_x$, $\sigma_y$, or $(\sigma_x - \sigma_y)^2 + 4T^2$.

When $T = 0$, $S$ = largest absolute magnitude of either $\sigma_x$, $\sigma_y$, or $(\sigma_x - \sigma_y)$.
RICHMOND-LOX EQUIPMENT COMPANY
NOZZLE REINFORCEMENT CALCULATIONS

JOB #: 30093  CALC BY: DATE 02/17/97
CUSTOMER NAME: FERMI LAB  CHK'D BY: DATE 2-19-97

Vessel Material Type: 304 S/S  Max. Design Temperature = 100 °F

Nozzle Designation: 6" N2P  Located: 1 (0=Heads/l=Shell)

Nozzle O.D. (Do) = 6.625 in  Design Pressure (P) = 30.0 psig
Nozzle I.D. (d) = 6.065 in  Des Press w/Stat Hd(Ps) = 30.9 psig
Nozzle Wall Nom (tn) = 0.280 in  Allow. Vessel Stress(Sva) = 18,800 psi
Vessel Thk Nom (t) = 0.625 in  Allow. Nozzle Stress(Sna) = 18,800 psi

Vessel Thk Reqd (tr):  Corrosion Allowance (CA) = 0.000 in
For Internal Pr = 0.314 in  Liq. Depth Above Nozzle = 18,000 in
For External Pr = 0.331 in  Nozzle Proj Outward (L) = 18,875 in
Dist CL to Knuckle = 11.055 in  Nozzle Proj Inward (h) = 0.625 in

Thickness required for Internal Pressure:
\[ trn = \frac{Ps - Ro}{(Sna + 4 \cdot Ps)} \]

Minimum Thickness For Full Vacuum:  Assuming \( trn = 0.034 \) in
L/Do = 2.849 and Do/t = 194.9

Per Fig. UGO-28.0, A = 0.00016  \( Pa = 4 \cdot B/(3 \cdot Do/t) \)
Per Matl Chart, UHA 28.1; B = 2309  \( Pa = 15.8 \) psi

WELD SIZING:  \( t_{min} = 0.280 \) in  Per UW-16(b)
Outside Fillet (W1) = \( t_{min} = 0.280 \); Use: 5/16
If Int Proj, Inside Fillet (W2) = \( t_{min} = 0.280 \); Use: 5/16
If No Int Proj, Inside Bevel (W2) = 0.7 \( t_{min} = 0.196 \); Use: 1/4

Strength Reduction (frl):  [No Greater Than 1.0]
Allowable Nozzle Stress(Sna)/Allowable Vessel Stress(Sva) = 1.000

Reinforcement Area Required:
\[ A = d \cdot tr + 2 \cdot tn \cdot tr \cdot (1-frl) = 1.903 \text{ in}^2 \]
(1/2 of above for External Pressure)

Reinforcement Area Available:
\[ A1 = d \cdot (t-tr-CA) - 2 \cdot tn \cdot (t-tr-CA) \cdot (1-frl) = 1.888 \text{ in}^2 \]
\[ A2 = \text{Smaller of the following} = 0.384 \text{ in}^2 \]
\[ 5 \cdot (tn-trn-CA) \cdot frl \cdot t = 0.769 \]
\[ 5 \cdot (tn-trn-CA) \cdot frl \cdot tn = 0.344 \]
\[ A3 = 2 \cdot (tn-CA) \cdot frl \cdot h = 0.350 \text{ in}^2 \]
\[ A41 = W1^2 = 0.098 \text{ in}^2 \]
\[ A43 = W2^2 = 0.098 \text{ in}^2 \]

Total Avail(A')=A1 + A2 + A3 + A41 + A43 = 2.817 \text{ in}^2
A'(Av) > A(Reqd) = Reinf OK

A'(Av) > A(Reqd) = Reinf OK

Reinforcement Area Required: INTERNAL = 1.903 \text{ in}^2  EXTERNAL = 1.004 \text{ in}^2

Reinforcement Area Available: INTERNAL = 1.888 \text{ in}^2  EXTERNAL = 1.783 \text{ in}^2

Reinforcement Area Available: INTERNAL = 1.888 \text{ in}^2  EXTERNAL = 1.783 \text{ in}^2

\[ A' = A' \text{ (Av) > A' (Reqd)} \]

Reinf OK
RICHMOND-LOX EQUIPMENT COMPANY
NOZZLE REINFORCEMENT CALCULATIONS

JOB #: 30093
CUSTOMER NAME: FERMI LAB

---

Vessel Material Type: 304 S/S
Max. Design Temperature = 100 °F

Noz Designation: 6" N3P
Located: 1 (0=Heads/1=Shell)

---

Nozzle O.D. (Do) = 6.625 in
Noz I.D. (Corr) (d) = 5.625 in
Nozzle Wall Nom(tn) = 0.500 in
Vessel Thk Nom (t) = 0.625 in

---

Design Pressure (P) = 30.0 psig
Des Press w/Stat Hd(Ps) = 30.0 psig
Allow. Vessel Stress(Sva) = 18,800 psi
Allow. Nozzle Stress(Sna) = 18,800 psi

---

Vessel Thk Reqd (tr):
For Internal Pr = 0.314 in
For External Pr = 0.331 in
Dist CL to Knuckle = 30.805 in

---

Thickness required for Internal Pressure:
trn = Ps-Ro / (Sna + 4*Ps) = 0.017 in

---

Minimum Thickness For Full Vacuum: Assuming trn = 0.017 in
L/Do = 0.585 and Do/t = 385.2

---

WELD SIZING: tmin = 0.500 in

---

Strength Reduction (fr1): [No Greater Than 1.0]
Allowable Nozzle Stress(Sna)/Allowable Vessel Stress(Sva) = 1.000

---

Reinforcement Area Required:

---

Reinforcement Area Available:

---

Total Avail(A\prime)= A1 + A2 + A3 + A41 + A43 = 2.745 in²
A\prime (Av) > A(Reqd) Reinf OK
A\prime (Av) > A(Reqd) Reinf OK
JOB #: 30093

CUSTOMER NAME: FERMI LAB

Vessel Material Type: 304 S/S  Max. Design Temperature = 100 °F

Noz Designation: 4" N11P Located: 1 (0=Heads/1=Shell)

Nozzle O.D. (Do) = 4.500 in  
Nozzle I.D. (Corr) (d) = 4.026 in  
Nozzle Wall Nom(tn) = 0.237 in  
Vessel Thk Nom (t) = 0.625 in  

Design Pressure (P) = 30.0 psig  
Design Pressure w/Stat Hdl(Ps) = 30.9 psig  
Allow. Vessel Stress (Sva) = 18,800 psi  
Allow. Nozzle Stress (Sna) = 18,800 psi  

Vessel Thk Reqd (tr):  
For Internal Pr = 0.314 in  
For External Pr = 0.331 in  

L/Do = 4.250 and  
Per Fig. UG0-28.0, A = 0.00013
Per MatI Chart, UHA 28.1; B= 1886

Minimum Thickness For Full Vacuum: Assuming trn = 0.027 in
Assuming trn = 0.027 in  
L/Do = 4.250 and  
P = 4-B/(3-Do/t)  
Pa = 4-B/(3-Do/t)  

WELD SIZING:  
Outside Fillet (W1) = tmin = 0.237 "; Use: 1/4  
If Intl Proj, Inside Fillet (W2) = tmin = 0.237 "; Use: 1/4  
If No Intl Proj, Inside Bevel (W2) = .7 tmin = 0.166 "; Use: 3/16

Strength Reduction (frl): [No Greater Than 1.0]
Allowable Nozzle Stress (Sna)/Allowable Vessel Stress (Sva) = 1.000

Reinforcement Area Required:

A = d·tr + 2·tn·tr·(1-frl) = 1.263 in²
(1/2 of above for External Pressure)

Reinforcement Area Available:

A1 = d·(t-tr-CA) - 2·tn·(t-tr-CA)·(1-frl) 1.253 in²
A2 = Smaller of the following = 0.276 in²  
5·(tn-trn-CA)·frl·t = 0.729  
5·(tn-trn-CA)·frl·tn = 0.276
A3 = 2·(tn-CA)·frl·h = 0.296 in²
A41 = W1² = 0.063 in²
A43 = W2² = 0.063 in²

Total Avail (A') = A1 + A2 + A3 + A41 + A43 = 1.951 in²

A'(Av) > A(Reqd)  
Reinf OK

A'(Av) > A(Reqd)  
Reinf OK
NOZZLE REINFORCEMENT CALCULATIONS (I.V.)

BEAM BYPASS TUBE

Refs: PEC Doc. C87-0271
Fermi Doc. 3740.214-1E-213235 Sat. 2 Rev. A
" " " " Sat. 2 Rev. A

\[ t = \frac{5}{8} \]

UG-37(a)

Def. (1) for \( t_r \) (opening & reinf. entirely w/ spherical portion of torispherical head)

\[ t_r = \frac{PLM}{25E - 0.2P} \]

where \( E = 1 \) & \( M = 1 \)

\[ P = 15 + 15 + \left[ \frac{1935 - 2x^5}{2} \right] \times \frac{0.02}{1728} \]

\[ P = 30.7 \text{ psi} \]

\[ t_r = \frac{30.7 \times 60.125 \times 1}{2 \times 18.89 \times 1 - 0.2 \times 30.7} = 0.0703 \text{ inch} \]

UG-37(c): Internal Pressure

Reg'd reinf. area, \( A \):

\[ A = d \cdot \frac{1}{t} , t_r \cdot F \left( 1 - \frac{1}{t_r} \right) \]

\[ \frac{1}{t_r} = 1 = \frac{15,800}{15,800} ; \quad 1 - \frac{1}{t_r} = 0 \]

Nozzle; 10" 528 8.3 pipe - Assume

\[ d = 9.562 \text{ inches} \]

\[ t_n = 0.394 \]

\[ t_n = \frac{10}{16} \times \frac{0.394}{0.6} = \frac{30.7}{18.89 \times 1 - 0.6 \times 30.7} = 0.0703 \text{ in.} \]

\[ A = 9.562 \times 0.0703 \times 1 = 0.672 \text{ in.}^2 \]

Avail. reinf. area in shell, \( A_i \):

\[ A_i = d \left( E_i + F_i \right) - 2t_n \left( E_i + t_n \right) \left( 1 - \frac{1}{t_n} \right) \]

\[ A_i = 9.522 \left( 0.625 - 0.394 \right) = 5.364 \text{ in.}^2 \gg A = 0.672 \text{ in.}^2 \]

OK
NOZZLE RENFORCEMENT CALCULATIONS
BEM BYPASS TUBE
VACUUM VESSEL

\[
A = \frac{1}{2} \left[ d t_r F + 2 t_n t_r F (1 - f_r) \right]
\]

\[
R_0 = 87.875 + 0.5 = 88.375
\]

Try \( t_r = 0.250 \)

\[
\frac{R_0}{t} = 353.5
\]

\[
A = \frac{0.125}{353.5} = 0.000354
\]

\[
B = 5500
\]

\[
\rho_n = \frac{5500}{353.5} = 15.5 \text{ psi}
\]

\[ t_r < 2t \text{ Reinforcement is adequate} \]
UG-37(d)(1): External Pressure

Rig'd rein. area = \( A = \frac{1}{t} [d + F + 2l, t + F(1-t)] \)

where \( t \) determined per UG-33(e)

\[ R_0 = 86.125 + 0.625 = 86.75 \]

let \( t = 0.040 \)

\[ \frac{R_0}{t} = 2168.75 \]

\[ A = \frac{0.175}{t} = 0.0000576 \quad \text{to left of curve in Fig. 5-UNH-24.1} \]

\[ P_a = \frac{0.0025 t}{(\frac{R_0}{t})^2} = \frac{0.0025x28x10^4}{(2.16875)^2x10^6} = 0.372 \text{ psi; } \]

Try \( t = 0.125 \) 

\[ \frac{R_0}{t} = 694 \quad A = \frac{0.175}{694} = 0.00018 \]

Fig. 5-UNH-24.1, \( B = 2600 \)

\[ P_a = \frac{B}{(R_0/t)} = \frac{2600}{694} = 3.746 \text{ psi; } \]

Try \( t = 0.250 \)

\[ \frac{R_0}{t} = 347 \quad A = \frac{0.175}{347} = 0.00051 \]

\[ B = 5100 \]

\[ P_a = \frac{5100}{347} = 14.7 \text{ psi; } \]

\( t \approx 0.0455 \)

Rig'd rein. area:

\[ A = \frac{1}{t} (9.562x0.185 + 1) = 1.218 \text{ in}^2 \]

avail. area \( A' : \)

\[ A' = 0.133 (1 + 0.125) - 0 = 9.52 (1 + 0.255 = 0.255) \]

\[ A = 3.538 \text{ in}^2 > A' = 1.218 \text{ in}^2 \quad \text{OK} \]
NOZZLE REINFORCEMENT CALCULATIONS

By virtue of ASME PV. Code, Section VIII, Div. 1, UG-36(c)(3)(4):

Where for head/shell thicknesses > 2 3/8 inch, nozzles of 2" NPS and smaller are exempted from "... reinforcement other than that inherent in the construction..."

The following nozzles penetrating the 5/8 inch thick outer cylinder of the inner vessel do not require reinforcements:

<table>
<thead>
<tr>
<th>Nozzle</th>
<th>Description</th>
<th>Size</th>
<th>Fermi Dwg. Where Shown</th>
</tr>
</thead>
<tbody>
<tr>
<td>N6P</td>
<td>Argon Vent</td>
<td>2&quot; SCH 40</td>
<td>3740.210-ME-222361</td>
</tr>
<tr>
<td>N8P</td>
<td>Argon Gas Line</td>
<td>1 1/2&quot; SCH 40</td>
<td>&quot;</td>
</tr>
<tr>
<td>N9P1</td>
<td>LN2 Cooldown XR</td>
<td>1 1/2&quot; SCH 40</td>
<td>&quot;</td>
</tr>
<tr>
<td>N9P2</td>
<td>LN2 Operating XR</td>
<td>1 1/2&quot; SCH 40</td>
<td>&quot;</td>
</tr>
<tr>
<td>N10P</td>
<td>LN2 Return</td>
<td>2&quot; SCH 40</td>
<td>&quot;</td>
</tr>
<tr>
<td>N16P</td>
<td>Thermal Siphon P</td>
<td>1&quot; Soc. Wld. Union</td>
<td>3740.214-ME-223235 244.7</td>
</tr>
<tr>
<td>N18P</td>
<td>Pressure Tap</td>
<td>1/2&quot; Soc. Wld. Union</td>
<td>3740.210-ME-222361</td>
</tr>
</tbody>
</table>
(UG-100)

PNEUMATIC TEST

\[ P_{sw} = 192.4 \times \frac{87.02}{1724} = 9.7 \text{ psi} \]

\[ P_T = 1.25 \left( \text{MAWP} + P_{sh} \right) \]

\[ P_T = 1.25 \left( \frac{15 + 15}{2} + 9.7 \right) \]

\[ P_T = 49.625 \text{ psig} \hspace{1cm} \text{[USE 50 psig]} \]

Gradually increase pressure to 25 psig (50% of test pressure)

Step 2: Increase pressure to 30 psig (60% of test pressure)

Step 3: Increase pressure to 35 psig (70% of test pressure)

Step 4: Increase pressure to 40 psig (80% of test pressure)

Step 5: Increase pressure to 45 psig (90% of test pressure)

Step 6: Increase pressure to 50 psig (100% of test pressure)

All welds to be examined per UW-50
DESIGN CALCULATIONS
SUPPLEMENT NO. 1

CUSTOMER  FERMILAB
P.O. NO.  925530
ITEM NO.  0\ø CRYOSTAT

TO CHECK REINFORCEMENT AREA AT N1P
(NOZZLE WALL IS SCH 10 BEYOND VESSEL.)

PREPARED BY:  D. D. Holmgren  DATE  6/23/87
CHECKED BY:  J. A. Stagg  DATE  6/24/87
NOZZLE REINFORCEMENT CALCULATIONS (Sec VIII Div 1, UG-37)

JOB #: 30093  
CALC BY: [Signature]  
DATE: 06/23/87  

CUSTOMER NAME: FERMILAB  
CHKD BY: [Signature]  
DATE: 6/24/87  

Vessel Material Type: 304 S/S Max. Design Temperature 100 °F  

Noz Designation: 8" SIGNAL PORT Located: 2 (1=Head/2=Shell)

---

Nozzle O.D. (Do) = 8.625 in  
Noz I.D. (Corr) (d) = 7.981 in  
Nozzle Wall Nom (tn) = 0.322 in  
Vessel Thk Nom (t) = 0.625 in

---

Vessel Thk Reqd (tr):  
For Internal Pr = 3.143 in  
For External Pr = 4.315 in

---

Corrosion Allow. (CA) = 0.000 in  
Noz Radius (Hds Only) = 8.625 in

---

Design Pressure (P) = 40.0 psig  
Des Press w/Stat Hd (Ps) = 40.6 psig

---

Allow. Vessel Stress (Sva) = 18,800 psi  
Allow. Nozzle Stress (Sna) = 18,800 psi

---

Liq. Depth Above Nozzle = 30.000 in  
Distance Noz CL-Knuckle = NA in

---

Minimum Thickness For Full Vacuum: Assuming trn = 0.050 in

L/Do = 3.710 and Do/t = 172.5

Per Fig. UGO-28.0, A = 0.00015  
Per Matl Chart, UHA 28.1; B = 2095

---

WELD SIZING: tmin = 0.322 in

Outside Fillet (W1) = tmin = 0.322 "; Use: 3/8

If Int Proj, Inside Fillet (W2) = tmin 0.322 "; Use: 3/8

If No Int Proj, Inside Bevel (W2) = .7 tmin = 0.225 "; Use: 1/4

---

Strength Reduction (fr1): [No Greater Than 1.0]  
Allowable Nozzle Stress (Sna)/Allowable Vessel Stress (Sva) = 1.000

---

Reinforcement Area Req'd: Long Axis, External Pressure

\[ F_{r1} = \frac{A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5}{2} \]

---

\[ A_1 = 5.223 \]

\[ A_2 = 2.238 \] (Large) \[ A_3 = 0.873 \] (Small)

\[ A_{41} = 0.431 \]

\[ A_{42} = 0.141 \]

\[ A_{43} = 0.141 \]

---

\[ A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A/2 \]

\[ 3.024 \geq 5.223/2 \]

---

Nozzle Reinforcement: OK for External
Calculation of \( d \) for internal pressure

\[ R_m = R_i + \frac{d}{2} = 96.75 - .625 + .3163/2 = 96.283 \]

\[ L = 71.5'' \]

\[ \alpha_1 = \cos^{-1} \left( \frac{L+R_n}{R_m} \right) = \cos^{-1} \left( \frac{71.5 + 3.991}{96.283} \right) = 38.367'' \]

\[ \alpha_2 = \cos^{-1} \left( \frac{L-R_n}{R_m} \right) = \cos^{-1} \left( \frac{71.5 - 3.991}{96.283} \right) = 48.481'' \]

\[ \alpha = \alpha_2 - \alpha_1 = 7.114'' \]

\[ d = 2 \times R_m \times \sqrt{1 - \cos^2 (\alpha/2)} = 11.946'' \]
NOZZLE REINFORCEMENT CALCULATIONS (Sec VIII Div 1, UG-37)

RICHMOND-LOX EQUIPMENT COMPANY

PAGE 2

JOB #: 30093

CUSTOMER NAME: FERMILAB

NOZZLE DESIGNATION: 8" SIGNAL PORT

Located: 2 (1=Head/2=Shell)

Vessel Material Type: 304 S/S Max. Design Temperature = 100 °F

Nozzle O.D. (Do) = 8.625 in

Design Pressure (P) = 40.0 psig

Noz I.D. (Corr) (d) = 7.981 in

Des Press w/Stat Hd(Ps) = 40.6 psig

Nozzle Wall Nom.(tn) = 0.322 in

Allow. Vessel Stress(Sva) = 18,800 psi

Vessel Thk Nom (t) = 0.625 in

Allow. Nozzle Stress(Sna) = 18,800 psi

Vessel Thk Reqd (tr): Liq. Depth Above Nozzle = 30.000 in

For Internal Pr = 0.317 in

Nozzle Proj Outward (L) = 32.000 in

For External Pr = 0.4375 in

Nozzle Projection Inward = 0.670 in

Corrosion Allow. (CA)= 0.000 in

Allow. Inward Proj (h) = 0.670 in

Noz Radius(Hds Only) = 0.000 in

Distance Noz CL-Knuckle = NA in

THICKNESS REQUIRED FOR INTERNAL PRESSURE:

trn = Ps·Ro / (Sna + 0.4·Ps) = 0.011 in

MINIMUM THICKNESS FOR FULL VACUUM: Assuming trn = 0.050 in

L/Do = 3.710 and Do/t = 172.5

Per Fig. UGO-28.0, A = 0.00015

Pa = 4 B / (3 Do/t) = 2095

Per MatI Chart, UHA 28.1; B = 16.2 psi

WELD SIZING: tmin = 0.322 in

Per UW-16(b)

Outside Fillet (W1) = tmin = 0.322 "; Use: 3/8

If Int Proj, Inside Fillet (W2) = tmin = 0.322 "; Use: 3/8

If No Int Proj, Inside Bevel (W2) = .7 tmin = 0.225 "; Use: 1/4

STRENGTH REDUCTION (fr1): [No Greater Than 1.0]

Allowable Nozzle Stress(Sna)/Allowable Vessel Stress(Sva) = 1.000

REINFORCEMENT AREA REQ'D: LONG AXIS, INTERNAL PRESSURE

\[ t_r = 0.3163 \]

\[ d = 11.946 \]

\[ F = 1.0 \]

\[ E_1 = 1.0 \]

\[ S_{r1} = 1.0 \]

\[ S_{r2} = 1.0 \]

\[ f_{r1} = 1.0 \]

\[ f_{r2} = 0.0 \]

\[ h = 1.0(\sin 42°) = 0.67 \]

\[ A_1 = 2(t_n - f_{r1}/2) \]

\[ A_2 = 2(t_n - f_{r2}/2)^2 \]

\[ A_3 = 2(t_n - c/l_{f_{r2}}) \]

\[ A_{41} = \text{outward nozzle weld} = (\text{leg})^2 f_{r3} \]

\[ A_{43} = \text{inward nozzle weld} = (\text{leg})^2 f_{r2} \]

\[ A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 > A \]

4.502 > 3.779 : NOZZ REINF OK FOR INTERNAL
**Calculation of \( d \)**

**For external pressure**

\[
R_m = R_i + \frac{t}{2} = 96.75 + 0.625 + 0.4375/2 = 96.344
\]

\[
L = 71.5"\]

\[
\alpha_1 = \cos^{-1} \left( \frac{L + R_m}{R_m} \right) = 38.412^\circ
\]

\[
\alpha_2 = \cos^{-1} \left( \frac{L - R_m}{R_m} \right) = 45.516^\circ
\]

\[
\alpha = \alpha_2 - \alpha_1 = 7.104^\circ
\]

\[
d = \frac{2}{3} R_m \sqrt{1 - \cos^2 \left( \frac{\alpha}{2} \right)} = 11.938"
\]

**Reference:** App. L, Ex. 7