**NEXT-100 DIMENSIONS, ANGEL** D. Shuman 10/23/2011

Mass of Xenon in Vessel at Maximum Operating Pressure (absolute):

Assume we have 100 kg Xe in active volume of gas inside a cylinder inscribed within the field cage (TTX panels)

$M_{Xe\_100} := 100kg$

Maximum Operating pressure (absolute):

$P_{MOPa\_100} := 15bar$

Minimum Operating pressure (minus sign indicates external pressure)

$P_{min} := -1.5bar$ this is driven by the need to pull vacuum with a possible hydrostatic head of 0.4bar if water tank is used for shielding

Operating Temperature, physical constants:

$T_{amb} := 293K$ $R := 8.314J \cdot mol^{-1} \cdot K^{-1}$ $M_{a\_Xe} := 136gm \cdot mol^{-1}$

Critical Pressure, temperature of Xenon:

$P_{c\_Xe} := 58.40bar$ $T_{c\_Xe} := 15.6K + 273K$ $T_{c\_Xe} = 288.6K$

reduced pressure:

$P_{r\_100} := \frac{P_{MOPa\_100}}{P_{c\_Xe}}$ $P_{r\_100} = 0.257$ $P_{r\_8bar} := \frac{8bar}{P_{c\_Xe}}$ $P_{r\_8bar} = 0.137$

reduced temperature

$T_r := \frac{T_{amb}}{T_{c\_Xe}}$ $T_r = 1.015$

Compressibility Factor: from chart for pure gasses shown below

$Z_{Xe\_15bar} := .93$ $Z_{Xe\_8bar} := .96$

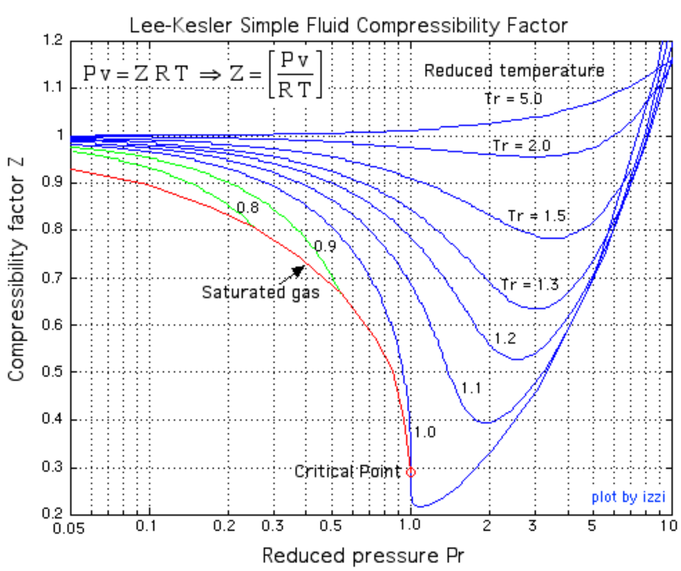



Fig. 6 Compressibility Factor, pure gasses

ref: <u>A Generalized Thermodynamic Correlation based on Three-Parameter Corresponding States</u>, B.I.Lee & M.G.Kesler, AIChE Journal, Volume 21, Issue 3, 1975, pp. 510-527' (secondary ref. from:http://www.ent.ohiou.edu/~-thermo/

Number of moles:

$n_{Xe\_100} := \frac{M_{Xe\_100}}{M_{a\_Xe}}$ $n_{Xe\_100} = 735.294\,mol$

Current vessel interior volume is 2.0 m3 ( vessel vol. - internal component vols). If we have 100kg total Xe then operating pressure is:

$$P_{100kg\_tot} := \frac{n_{Xe\_100} \cdot Z_{Xe\_8bar} \cdot R \cdot T_{amb}}{2m^3} \qquad P_{100kg\_tot} = 8.483\,bar$$

Volume required:

$$V_{Xe\_100} := \frac{n_{Xe\_100} \cdot Z_{Xe\_15bar} \cdot R \cdot T_{amb}}{P_{MOPa\_100}} \qquad V_{Xe\_100} = 1.096\,m^3$$

$n_{Xe} := n_{Xe\_100}$ $\quad V_{Xe} := V_{Xe\_100}$ $\quad P_{MOPa} := P_{MOPa\_100}$

molar, mass, volumetric density:

$$\rho_{mol} := \frac{n_{Xe}}{V_{Xe}} \qquad \rho_{mol} = 0.671\,\frac{mol}{L}$$

$$\rho_{Xe} := \rho_{mol} \cdot M_{a\_Xe} \qquad \rho_{Xe} = 0.091\,\frac{gm}{cm^3}$$

$$v_{Xe} := \rho_{Xe}^{-1} \qquad v_{Xe} = 10.957\,\frac{cm}{gm}$$

$P_h := 1.15P_{MOPa}$ $\quad \gamma := 1.666$

$P_l := 1bar$ $\quad V_h := 1.5V_{Xe}$

Stored Energy @ 15.4 bar MOP

$$U_v := \frac{P_h \cdot V_h}{\gamma - 1}\left[1 - \left(\frac{P_l}{P_h}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad U_v = 2.9\,MJ$$

We desire active length to be = 1.25x active diameter. then:

Active volume radius: $r_{Xe\_exact} := \sqrt[3]{\frac{V_{Xe}}{2.5\pi}}$ $\quad r_{Xe\_exact} = 0.5186425\,m$

then: $l_{Xe\_exact} := 2.5r_{Xe\_exact}$ $\quad l_{Xe\_exact} = 1.296606\,m$

Round up to:

$r_{Xe} := 0.53m$ $\quad l_{Xe} := 1.30m$ $\quad \pi r_{Xe}^2 \cdot l_{Xe} = 1.147\,m^3$

Field Cage Ring minor radius

$r_{fc\_minor} := 0.5cm$

**Pressure Vessel Design Calculations, DRAFT Stainless steel vessel with Copper Liner** D. Shuman, 10/14/2011

Active volume dimensions, from earlier analyses:

$r_{Xe} = 0.53\,m$ $l_{Xe} = 1.3\,m$

We consider using a field cage solid insulator/light tube of 2 cm thk. and a copper liner of 6 cm thickness

$t_{fc} := 3cm$ $t_{Cu} := 6cm$ note: copper will be notched at internal flange

Pressure Vessel inner radius is then:

$R_{i\_pv} := r_{Xe} + t_{fc} + t_{Cu}$ $R_{i\_pv} = 0.62\,m$

**Vessel wall thicknesses** **DRAFT**

Pressure vessel inner radius: $R_{i\_pv} = 62\,cm$

We choose to use division 2 rules, which allow thinner walls at the expense of performing more strict material acceptance and additional NDE post weld inspection, as we will be performing these steps regardless, due to the high value of the vessel contents. For flat-faced flanges we use div. 1, as no methodology exists in div 2. Div. 1 is more typically conservative than div. 2, when div. 2 quality control is implemented.

Maximum allowable material stresses, for sec VIII, division 2 rules from ASME 2009 Pressure Vessel code, sec. II part D, table 5B (div. 1 stress from table 1A):

Youngs modulus

$S_{max\_304L\_div2} := 16700psi$ $S_{max\_304L\_div1} := 16700psi$ $E_{SS\_aus} := 193GPa$

$S_{max\_316L\_div2} := 16700psi$ $S_{max\_316L\_div1} := 16700psi$ we use the L grades for weldability

**color scheme for this document**

input check result (all conditions should be true (=1)

$xx := 1$ $xx > 0 = 1$

Choose material:

$S_{max} := S_{max\_304L\_div2}$

Maximum Operating Pressure (MOP), gauge:

$MOP_{pv} := (P_{MOPa} - 1bar)$ $MOP_{pv} = 14\,bar$

Minimum Pressure, gauge:

$P_{min} = -1.5\,bar$ the extra 0.5 atm maintains an upgrade path to a water or scintillator tank

Maximum allowable pressure, gauge (from LBNL Pressure Safety Manual, PUB3000)
at a minimum, 10% over max operating pressure; this is design pressure at LBNL:

$MAWP_{pv} := 1.1MOP_{pv}$ $MAWP_{pv} = 15.4\,bar$

Vessel wall thickness, for internal pressure is then ( div 2):

$$t_{pv\_d2\_min\_ip} := R_{i\_pv}\cdot\left(e^{\frac{MAWP_{pv}}{S_{max}}} - 1\right) \qquad t_{pv\_d2\_min\_ip} = 8.462\,mm$$

Compare with div. 1 rules ( weld efficiency E=1) $E_w := 1$

$$t_{pv\_d1\_min\_ip} := \frac{MAWP_{pv}\cdot(R_{i\_pv})}{S_{max\_304L\_div1}\cdot E_w - 0.6\cdot MAWP_{pv}} \qquad t_{pv\_d1\_min\_ip} = 8.473\,mm$$

There seems to be little difference between using division 1 or 2 for a 304L stainless steel vessel. We continue with div 2 rules, since we have high quality assurrance standards anyway. We set wall thickness to be:

$t_{pv} := 10mm$ $\quad t_{pv} > t_{pv\_d2\_min\_ip} = 1$

**Maximum Allowable External pressure using ASME PV code Sec. VIII div 2 rules, 4.4.5**

Step 1- trial thickness, outer radius, diameter longest length between flanges:

$$R_{o\_pv} := R_{i\_pv} + t_{pv} \qquad D_o := 2R_{o\_pv} \qquad L_{ff} := 1.4m$$

material elastic modulus:

$E_y := E_{SS\_aus} \qquad E_y = 193\,GPa$

Step 2 compute the following:

or greater

$$M_x := \frac{L_{ff}}{\sqrt{R_{o\_pv} \cdot t_{pv}}} \qquad M_x = 17.638 \qquad S_{y\_Ti\_g2} := 40000psi$$

for $\quad 2\left(\frac{D_o}{t_{pv}}\right)^{.94} = 188.529 \qquad S_y := S_{y\_Ti\_g2}$

$$C_h := 1.12M_x^{-1.058} \qquad C_h = 0.054$$

$$F_{he} := \frac{1.6 \cdot C_h \cdot E_y \cdot t_{pv}}{2(D_o)} \qquad F_{he} = 65.878\,MPa \qquad \frac{F_{he}}{S_y} = 0.239$$

$$F_{ic} := F_{he} \qquad \frac{F_{he}}{S_y} \le .552 = 1$$

$$F_{ha} := \frac{F_{ic}}{2}$$

per 4.4.2 eq. (4.4.1)

$$FS := 2 \qquad P_{a\_div2} := 2F_{ha} \cdot \left(\frac{t_{pv}}{D_o}\right) \qquad P_{a\_div2} = 5.159\,bar \qquad P_{a\_div2} > -P_{min} = 1$$

**Flange thickness**:

inner radius $\quad R_{i\_pv} = 0.62\,m$

max. allowable pressure $\quad MAWP_{pv} = 15.4\,bar$ (gauge pressure)

The flange design for helicoflex or O-ring sealing is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness, even though the rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the lower allowable stresses of division 1.

Flanges and shells will be fabricated from 304L or 316L (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected. The flange bolts and nuts will be inconel 718, (UNS N77180) as this is the highest strength non-corrosive material allowed for bolting.

We will design to use one Helicoflex 5mm gasket (smallest size possible) with aluminum facing (softest) loaded to the minimum force required to achieve helium leak rate.

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2B:

Maximum allowable design stress for flange

$S_f := S_{max\_304L\_div1}$ $\qquad S_f = 115.1\,MPa$

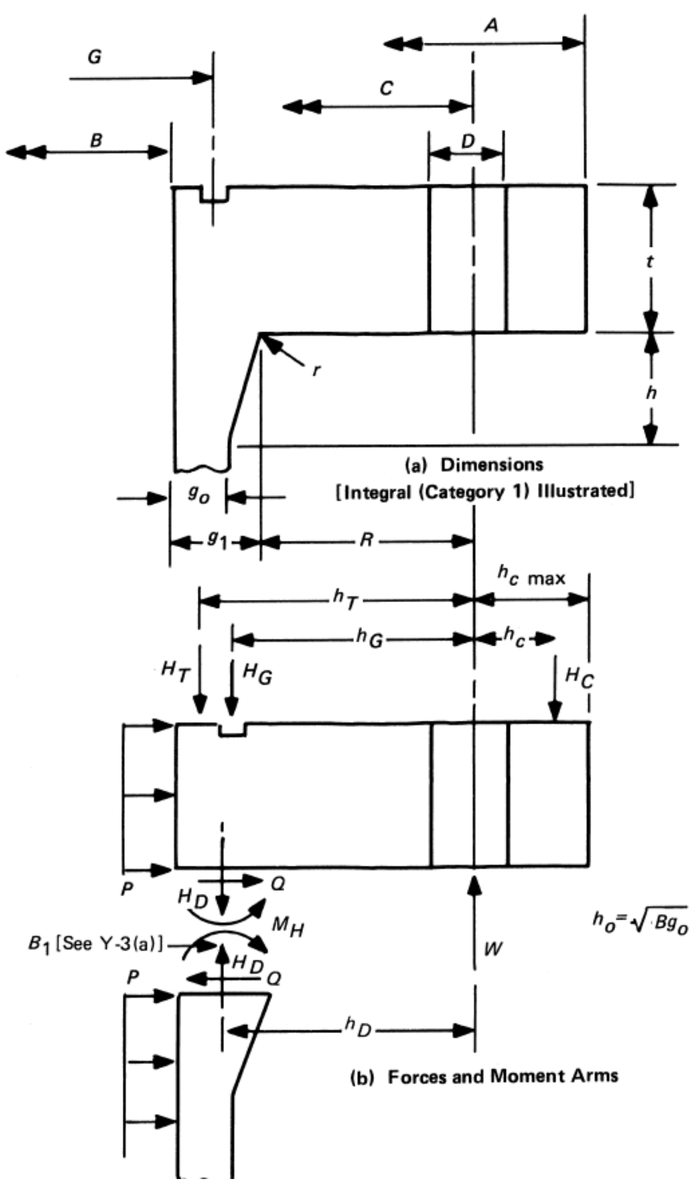
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

Inconel 718 (UNS N07718) $\qquad S_{max\_N07718} := 37000psi$

$S_b := S_{max\_N07718}$ $\qquad S_b = 255.1\,MPa$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES




hub thickness at flange (no hub) $\qquad$ corner radius:

$g_0 := t_{pv}$ $\quad g_1 := t_{pv}$ $\quad g_0 = 10\,mm$ $\quad g_1 = 10\,mm$ $\qquad r_1 := max(.25g_1, 5mm)$ $\quad r_1 = 5\,mm$

Flange OD

$A := 1.34m$

Flange ID

$B := 2R_{i\_pv}$  $B = 1.24\,m$

define:

$B_1 := B + g_1$  $B_1 = 1.25\,m$

Bolt circle (B.C.) dia, C:

$C := 1.3m$

Gasket dia

$G := 2\left(R_{i\_pv} + .75cm\right)$  $G = 1.255\,m$

Force of Pressure on head

$H := .785G^2 \cdot MAWP_{pv}$  $H = 1.93 \times 10^6\,N$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70 F= ~5 lbs/in for 20% compression, (Parker o-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equivaluses Y foas the frce teme and gives several v=possible values for 5mm HN200 with aluminum jacket:

$Y_1 := 30\frac{N}{mm}$ min value for our pressure and required leak rate (He)  $Y_2 := 150\frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter  $D_j := G$  $D_j = 1.255\,m$

Force is then either of:

$F_m := 2\pi D_j \cdot Y_1$  $F_m = 2.366 \times 10^5\,N$

or

$F_j := 2\pi \cdot D_j \cdot Y_2$  $F_j = 1.183 \times 10^6\,N$

Helicoflex recommends using Y2 ( 220 N/mm) for large diameter seals, even though for small diameter one can use the greater of Y1 or Ym=(Y2*(P/Pu)). For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port inbetween, so we thus design for Y1 (use $F_m$) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. A PCTFE O-ring may give lower permeability, but have a higher force requirement, than for an normal butyl or nitrile O-ring; designing to Y1 will account for this. Note: in the cold trap one will get water and N2, O2, that permeates through the outer O-ring as well.

Start by making trial assumption for number of bolts, root dia., pitch, bolt hole dia D,

$n := 120$  $d_b := 12.77mm$

Choosing ISO fine thread, with pitch; thread depth:

$p_t := 1.0mm$  $d_t := .614 \cdot p_t$

Nominal bolt dia is then;

$d_{b\_nom\_min} := d_b + 2d_t$  $d_{b\_nom\_min} = 13.998\,mm$

Set:

$d_{b\_nom} := 14mm$  $d_{b\_nom} > d_{b\_nom\_min} = 1$

Check bolt to bolt clearance, for box wrench b2b spacing is 1.2 in for 1/2in bolt twice bolt dia ( $2.4xd_b$ ):

$$\pi C - 2.4n \cdot d_{b\_nom} \geq 0 = 1$$

Check nut, washer clearance: $OD_w := 2d_{b\_nom}$ this covers the nut width across corners

$$0.5C - (0.5B + g_1 + r_1) \geq 0.5OD_w = 1$$

Flange hole diameter, minimum for clearance :

$$D_{tmin} := d_{b\_nom} + 0.5mm \qquad D_{tmin} = 14.5\,mm$$

Set:

$$D_t := 15mm \qquad D_t > D_{tmin} = 1$$

Compute Forces on flange:

$$H_G := F_m \qquad H_G = 2.366 \times 10^5\,N$$

$$h_G := 0.5(C - G) \qquad h_G = 2.25\,cm$$

$$H_D := .785 \cdot B^2 \cdot MAWP_{pv} \qquad H_D = 1.884 \times 10^6\,N$$

$$h_D := D_t \qquad h_D = 1.5\,cm$$

$$H_T := H - H_D \qquad H_T = 4.586 \times 10^4\,N$$

$$h_T := 0.5(C - B) \qquad h_T = 30\,mm$$

Total Moment on Flange

$$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \qquad M_P = 3.496 \times 10^4\,J$$

**Appendix Y Calc**

$$P := MAWP_{pv} \qquad P = 1.561 \times 10^6\,Pa$$

Choose values for plate thickness and bolt hole dia:

$$t := 3.0cm \qquad D := D_t \qquad D = 1.5\,cm$$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \qquad \beta = 1.02 \qquad h_C := 0.5(A - C) \qquad h_C = 0.02\,m$$

$$a := \frac{A + C}{2B_1} \qquad a = 1.056 \qquad AR := \frac{n \cdot D}{\pi \cdot C} \qquad AR = 0.441 \qquad h_0 := \sqrt{B \cdot g_0}$$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}}\,\mathrm{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 4.114 \times 10^{-3}$$

$$h_0 = 0.111\,m$$

We need factors F and G, most easily found in figs 2-7.2 and 7.3 (Appendix 2)

since $\frac{g_1}{g_0} = 1$ these values converge to $F := 0.90892$ $V := 0.550103$

**Y-5 Classification and Categorization**

We have identical (class 1 assembly) integral ( category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13),(14a),(15a),16a)

$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B$ $\quad J_S = 0.052$ $\quad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B$ $\quad J_P = 0.04$

(5a) $F' := \frac{g_0^{\,2}\left(h_0 + F \cdot t\right)}{V}$ $\quad F' = 2.52 \times 10^{-5}\,m^3$ $\quad M_P = 3.496 \times 10^4\,N \cdot m$

$A = 1.34\,m \quad B = 1.24\,m$

$K := \frac{A}{B} \quad K = 1.081 \quad Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 12.919$

$f := 1$

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b \quad l = 6.638\,cm \quad A_b := n \cdot .785d_b^{\,2}$

sec Y-6.2(a)(3)

Elastic constants

http://www.hightempmetals.com/tech-data/hitempInconel718data.php

$E := E_{SS\_aus} \quad E_{Inconel\_718} := 208GPa \quad E_{bolt} := E_{Inconel\_718}$

(7-13)

$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'}$ $\quad M_S = -1.2 \times 10^3\,J$

$\theta_B := \frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right)$ $\quad \theta_B = 4.432 \times 10^{-4}$ $\quad E \cdot \theta_B = 85.532\,MPa$

$H_C := \frac{M_P + M_S}{h_C}$ $\quad H_C = 1.686 \times 10^6\,N$

$W_{m1} := H + H_G + H_C$ $\quad W_{m1} = 3.852 \times 10^6\,N$

**Compute Flange and Bolt Stresses**

$\sigma_b := \frac{W_{m1}}{A_b}$ $\quad \sigma_b = 250.8\,MPa$ $\quad S_b = 255.1\,MPa$

$r_E := \frac{E}{E_{bolt}}$ $\quad r_E = 0.928$

$S_i := \sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot \left(M_P + M_S\right)}{a \cdot t^3\,l \cdot r_E \cdot B_1}$ $\quad S_i = 243.7\,MPa$

$S_{R\_BC} := \frac{6\left(M_P + M_S\right)}{t^2\left(\pi \cdot C - n \cdot D\right)}$ $\quad S_{R\_BC} = 98.4\,MPa$ $\quad S_f = 115.1\,MPa$

$S_{R\_ID1} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2}$ $\quad S_{R\_ID1} = 2.243\,MPa$

$$S_{T1} := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{T1} = 0.9\,\text{MPa}$$

$$S_{T3} := \frac{t \cdot E \cdot \theta_B}{B_1} \qquad S_{T3} = 2.053\,\text{MPa}$$

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = 15.221\,\text{MPa}$$

**Y-7 Flange stress allowables:** $S_f = 115.1\,\text{MPa}$

(a) $\sigma_b < S_b = 1$

(b)

(1) $S_H < 1.5S_f = 1$ $S_n$ not applicable

(2) not applicable

(c) $S_{R\_BC} < S_f = 1$

$S_{R\_ID1} < S_f = 1$

(d) $S_{T1} < S_f = 1$

$S_{T3} < S_f = 1$

(e) $\frac{S_H + S_{R\_BC}}{2} < S_f = 1$

$\frac{S_H + S_{R\_ID1}}{2} < S_f = 1$

(f) not applicable

Shear stress in inner flange lip from shield

$M_{sh} := 1000\text{kg} \qquad t_{lip} := 3\text{mm}$

$$\tau_{lip} := \frac{M_{sh} \cdot g}{R_{i\_pv} \cdot t_{lip}} \qquad \tau_{lip} = 5.272\,\text{MPa}$$

Shear stress on O-ring land (section between inner and outer O-ring)

$t_{land\_radial} := .36\text{cm} \qquad w_{land\_axial} := .41\text{cm}$

shear f $F_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot w_{land\_axial} \cdot P$ shear area: $A_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot t_{land\_radial}$

$$\tau_{land} := \frac{F_{O\_ring\_land}}{A_{O\_ring\_land}} \qquad \tau_{land} = 1.778\,\text{MPa}$$

Bolt force total

$F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2} \qquad F_{bolt} = 7.217 \times 10^3\,\text{lbf}$

Bolt torque required

$T_{bolt\_min} := 0.2F_{bolt} \cdot d_b \qquad T_{bolt\_min} = 82\,\text{N} \cdot \text{m} \qquad T_{bolt\_min} = 60.5\,\text{lbf} \cdot \text{ft}$ for pressure test use 1.5x this value

**ANGEL Torispheric Head Design, using (2010 ASME PV Code Section VIII, div. 2, part 4 rules)**
**2 nozzle head using standard dimension head** **DRAFT** D. Shuman, LBNL, July12, 2011

4.3.6.1Torispheric head with same crown and knuckle thicknessestandard dimensions.

(a) Step 1, determine I.D. and assume the following:

thickness:

$t_{ts} := t_{pv}$ $\quad t_{ts} = 1\,cm$

I.D.

$D_i := 2R_{i\_pv}$

O.D.

$D := D_i + 2t_{ts}$ $\quad D = 1.26\,m$

Crown radius: Knuckle radius:

$L_{cr} := 1D_i$ $\quad L_{cr} = 1.24\,m$ $\quad r_{kn} := 0.1D_i$ $\quad r_{kn} = 0.124\,m$

(b) Step 2- Compute the following ratios and check:

$$0.7 \le \frac{L_{cr}}{D_i} \le 1.0 = 1$$

$$\frac{r_{kn}}{D_i} \ge 0.06 = 1$$

$$20 \le \frac{L_{cr}}{t_{ts}} \le 2000 = 1$$

for all true, continue, otherwise design using part 5 rules

(c) Step 3 calculate:

thickness, this is an iterated value after going through part 4.5.10.1 (openings) further down in the document

$$\beta_{th} := \mathrm{acos}\left(\frac{0.5D_i - r_{kn}}{L_{cr} - r_{kn}}\right) \qquad \beta_{th} = 1.11\,rad$$

$$\phi_{th} := \frac{\sqrt{L_{cr} \cdot t_{ts}}}{r_{kn}} \qquad \phi_{th} = 0.898\,rad$$

$$R_{th} := \begin{cases} \dfrac{0.5D_i - r_{kn}}{\cos(\beta_{th} - \phi_{th})} + r_{kn} & \text{if } \phi_{th} < \beta_{th} \\ 0.5D_i & \text{if } \phi_{th} \ge \beta_{th} \end{cases} \qquad \phi_{th} < \beta_{th} = 1 \qquad \phi_{th} \ge \beta_{th} = 0$$

$$R_{th} = 0.631\,m$$

(d) Step 4 compute:

$$C_{1ts} := \begin{cases} \left[9.31\left(\frac{r_{kn}}{D_i}\right) - 0.086\right] & \text{if } \frac{r_{kn}}{D_i} \le 0.08 \\ \left[0.692\left(\frac{r_{kn}}{D_i}\right) + 0.605\right] & \text{if } \frac{r_{kn}}{D_i} > 0.08 \end{cases}$$

$\frac{r_{kn}}{D_i} \le 0.08 = 0$ (4.3.12)

$\frac{r_{kn}}{D_i} > 0.08 = 1$ (4.3.13)

$C_{1ts} = 0.674$

$$C_{2ts} := \begin{cases} 1.25 & \text{if } \frac{r_{kn}}{D_i} \le 0.08 \\ 1.46 - 2.6\cdot\left(\frac{r_{kn}}{D_i}\right) & \text{if } \frac{r_{kn}}{D_i} > 0.08 \end{cases}$$

$\frac{r_{kn}}{D_i} \le 0.08 = 0$ (4.3.14)

$\frac{r_{kn}}{D_i} > 0.08 = 1$ (4.3.15)

$C_{2ts} = 1.2$

(e) Step 5, internal pressure expected to cause elastic buckling at knuckle

$$P_{eth} := \frac{C_{1ts}\cdot E\cdot t_{ts}^{2}}{C_{2ts}\cdot R_{th}\cdot\left(0.5R_{th} - r_{kn}\right)} \qquad P_{eth} = 884\,\text{bar} \qquad (4.3.16)$$

(f) Step 6, internal pressure expected to result in maximum stress ($S_y$) at knuckle
time independent

$C_{3ts} := S_{y\_304L}$ $\qquad S_{y\_304L} := 25000\text{psi}$

$$P_y := \frac{C_{3ts}\cdot t_{ts}}{C_{2ts}\cdot R_{th}\cdot\left(0.5\frac{R_{th}}{r_{kn}} - 1\right)} \qquad P_y = 15\,\text{bar} \qquad (4.3.17)$$

(g) Step 7 - pressure expected to cause buckling failure of the knuckle

for: $G_{th} := \frac{P_{eth}}{P_y}$ $\qquad G_{th} = 60.879$

$$P_{ck} := \left(\frac{0.77508\cdot G_{th} - 0.20354\cdot G_{th}^{2} + 0.019274\cdot G_{th}^{3}}{1 + 0.19014G_{th} - 0.089534G_{th}^{2} + 0.0093965G_{th}^{3}}\right)\cdot P_y \qquad P_{ck} = 29\,\text{bar} \qquad (4.3.19)$$

(h) Step 8 - allowable pressure based on buckling failure of the knuckle

$$P_{ak} := \frac{P_{ck}}{1.5} \qquad P_{ak} = 19.574\,\text{bar}$$

(i) Step 9 - allowable pressure based on rupture of the crown

$$P_{ac} := \frac{2S_{max}\cdot 1}{\frac{L_{cr}}{t_{ts}} + 0.5} \qquad P_{ac} = 18.2\,\text{bar}$$

(j) Step 10 - maximum allowable internal pressure $\qquad P = 15.4\,\text{bar}$

$P_{a\_ip} := \min\left(P_{ak}, P_{ac}\right)$ $\qquad P_{a\_ip} = 18.2\,\text{bar}$ $\qquad P_{a\_ip} > P = 1$

### 4.5.10.1 Radial Nozzle in formed head

$R_n := 5.1cm$

$t_n := 7mm$

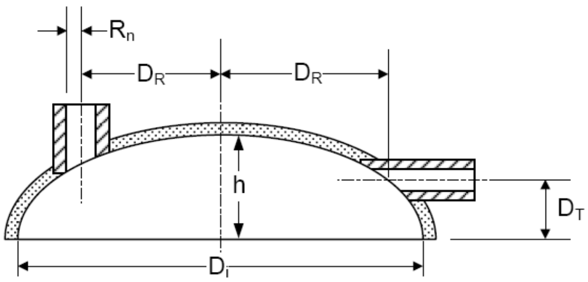
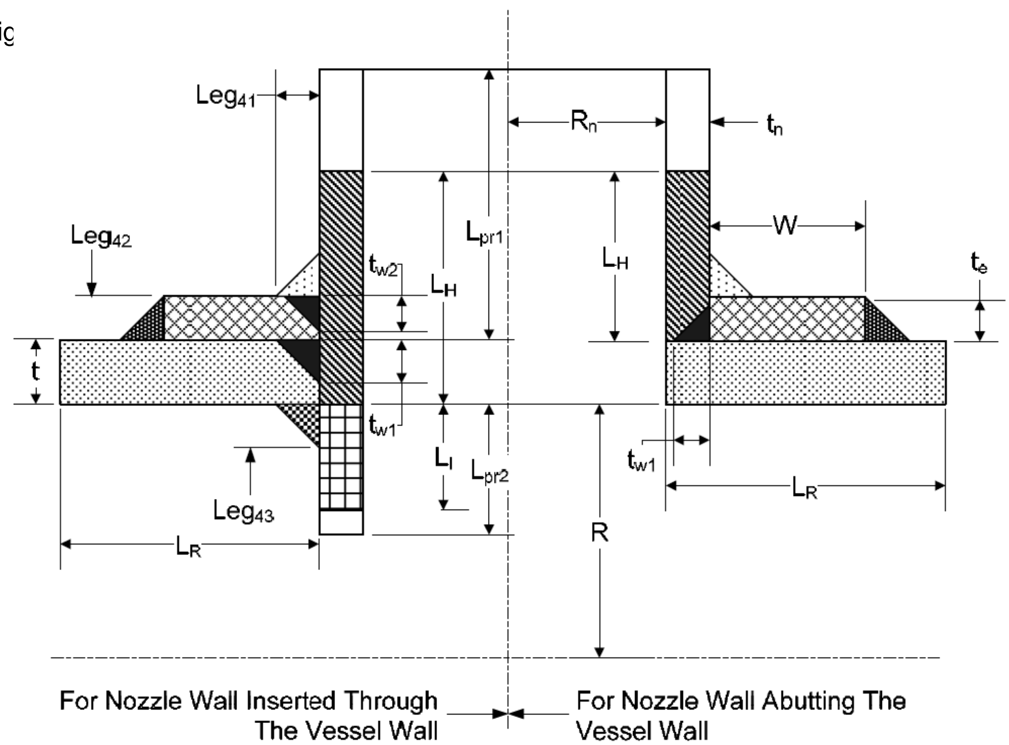



Fig 4.5.11

### 4.5.10.1 Procedure for Radial Nozzle in a Spherical or Formed head

Fig




a) Step 1

$R_{eff} := L_{cr} \qquad R_{eff} = 1.24\,m$ (4.5.64)

b) Step 2- limit of reinforcement along vessel wall. Here we compute for both nozzles using parallel calcs

$D_R := \begin{pmatrix} 0 \\ 10 \end{pmatrix} cm$ assume $R_n$, $t_n$ are same for both nozzles

Possible limits of reinforcment:

$L_{R1} := 0.5 \cdot D_i - (D_R + R_n + t_n) \qquad L_{R1} = \begin{pmatrix} 56.2 \\ 46.2 \end{pmatrix} cm$ (4.5.67)

$\sqrt{R_{eff} \cdot t_{ts}} = 11.136\,cm \qquad 2R_n = 10.2\,cm$

$L_{R2} := \min\left(\sqrt{R_{eff} \cdot t_{ts}}, 2R_n\right) \qquad L_{R2} = 10.2\,cm$ (4.5.67)

Final Limit of reinforcement along vessel wall (assume no pad reinforcement):

$L_R := \min(L_{R1}, L_{R2}) \qquad L_R = 10.2\,cm$ (4.5.68)

c) Step 3- limit of reinforcement along nozzle wall projecting outside vessel surface wall.

We have no pad reinforcement, and no inside nozzle so:

$t_e := 0mm \quad L_{pr1} := 30cm \quad L_{pr2} := 0cm$

$L_H := \min\left[\left(t_{ts} + t_e + F_p \cdot \sqrt{R_n \cdot t_n}\right), L_{pr1} + t_{ts}\right]$ (4.5.73)

where:

$X_o := D_R + R_n + t_n \qquad X_o = \begin{pmatrix} 0.058 \\ 0.158 \end{pmatrix} m$ (4.5.79)

$C_p := e^{\frac{0.35D_i - X_o}{8t_{ts}}} \qquad C_p = \begin{pmatrix} 109.947 \\ 31.5 \end{pmatrix}$ (4.5.78)

$C_n := \min\left[\left(\frac{t_{ts} + t_e}{t_n}\right)^{0.35}, 1.0\right] \qquad C_n = 1$ (4.5.81)

$F_p := \min(C_n, C_p) \qquad F_p = 1$ note that this is true for both values of $C_p$, so we can drop the parallel calculation (4.5.80)

$L_H := \min\left[\left(t_{ts} + t_e + F_p \cdot \sqrt{R_n \cdot t_n}\right), L_{pr1} + t_{ts}\right] \qquad L_H = 2.889\,cm$ (4.5.73)

d) Step 4 - limit of reinforcement along nozzle wall projecting inside vessel surface wall, if applicable

$L_I := \min\left(F_p \cdot \sqrt{R_n \cdot t_n}, L_{pr2}\right) \qquad L_I = 0\,cm$ (4.5.82)

e) Step 5 - determine total available area near nozzle opening

(material strength ratios)--> $f_{rn} := 1 \qquad f_{rp} := 1$ (4.5.30) (4.5.31)

$A_T := A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp} \cdot A_5$ (4.5.83)

$A_1 := t_{ts} \cdot L_R \qquad A_1 = 10.2\,cm^2$ (4.5.84)

$A_2 := t_n \cdot L_H \qquad A_2 = 2.023\,cm^2$ (4.5.86)

$L_{41} := 0.7cm$ $L_{42} := 0cm$ $L_{43} := 0.7cm$ $t_e = 0\,cm$

$A_3 := t_n \cdot L_I$ $A_3 = 0\,cm^2$ (4.5.83)

$A_{41} := 0.5 L_{41}^2$ $A_{41} = 0.245\,cm^2$ (4.5.88)

$A_{42} := 0.5 L_{42}^2$ $A_{42} = 0\,cm^2$ (4.5.89)

$A_{43} := 0.5 L_{43}^2$ $A_{43} = 0.245\,cm^2$ (4.5.90)

$A_5 := 0cm^2$ (4.5.94)

$A_T := A_1 + f_{rn} \cdot (A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp} \cdot A_5$ $A_T = 12.713\,cm^2$ (4.5.83)

f) Step 6 - determine applicable forces

$t_{eff} := t_{ts} \cdot \left( \frac{t_{ts} \cdot L_R + A_5 \cdot f_{rp}}{t_{ts} \cdot L_R} \right)$ $t_{eff} = 10\,mm$ (4.5.100)

$R_{xn} := \frac{t_n}{\ln\left( \frac{R_n + t_n}{R_n} \right)}$ $R_{xn} = 5.442\,cm$ $R_{xs} := \frac{t_{eff}}{\ln\left( \frac{R_{eff} + t_{eff}}{R_{eff}} \right)}$ $R_{xs} = 1.245\,m$ (4.5.98)

$f_N := P \cdot R_{xn} \cdot (L_H - t_{ts})$ $f_N = 1.605 \times 10^3\,N$ (4.5.95)

$f_S := \frac{P \cdot R_{xs} \cdot (L_R + t_n)}{2}$ $f_S = 1.059 \times 10^5\,N$ (4.5.96)

$R_{nc} := R_n$ (radius along chord= $R_n$ for radial nozzles)

$f_T := \frac{P \cdot R_{xs} \cdot R_{nc}}{2}$ $f_T = 4.955 \times 10^4\,N$ (4.5.97)

g) Step 7 - determine effective thickness for nozzles in spherical, ellipsoidal, or torispherical heads

$t_{eff} = 1\,cm$ same formula as above in step 6 (4.5.100)

h) Step 8 - Determine avg. local primary membrane stress and general primary membrane stress at nozzle intersection

$\sigma_{avg} := \frac{f_N + f_S + f_T}{A_T}$ $\sigma_{avg} = 123.5\,MPa$ (4.5.101)

$\sigma_{circ} := \frac{P \cdot R_{xs}}{2 t_{eff}}$ $\sigma_{circ} = 97.2\,MPa$ (4.5.102)

I) Step 9 Determine maximum local primary membrane stress

$P_L := \max\left[ (2\sigma_{avg} - \sigma_{circ}), \sigma_{circ} \right]$ $P_L = 149.9\,MPa$ (4.5.103)

$E_w = 1$

$S_{allow} := 1.5 S_{max} \cdot E_w$ $S_{allow} = 172.7\,MPa$ (4.5.43)

j) Step 10 - Maximum local primary membrane stress must be less than the allowable stress

$P_L \le S_{allow} = 1$ (4.5.104)

k) Step 11 - Determine max allowable working pressure of the nozzle

$$A_p := R_{xn} \cdot (L_H - t_{ts}) + \frac{R_{xs} \cdot (L_R + t_n + R_{nc})}{2} \qquad A_p = 1 \times 10^3 \, cm^2 \tag{4.5.108}$$

$$P_{max1} := \frac{S_{allow}}{\left(\frac{2A_p}{A_T} - \frac{R_{xs}}{2t_{eff}}\right)} \qquad P_{max1} = 17.7 \, bar \tag{4.5.105}$$

$$S := S_{max} \qquad S = 115.1 \, MPa$$

$$P_{max2} := 2 \cdot S \cdot \left(\frac{t_{ts}}{R_{xs}}\right) \qquad P_{max2} = 18.2 \, bar \tag{4.5.106}$$

$$P_{max} := \min(P_{max1}, P_{max2}) \qquad P_{max} = 17.7 \, bar \tag{4.5.107}$$

$P_{max} > P = 1$