Vessel wall thicknesses **DRAFT** D. Shuman, Jul 13, 2011

Pressure vessel inner radius: $R_{i\_pv} = 57\,cm$

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.

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

Grade 2 Titanium $S_{max\_Ti\_g2\_div2} := 20800psi$

Grade 3 Titanium $S_{max\_Ti\_g3\_div2} := 27100psi$

Choose material:

$S_{max} := S_{max\_Ti\_g3\_div2}$

color scheme

input	check result
$xx := 1$	$xx > 0 = 1$

Maximum Operating Pressure (MOP), gauge:

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

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 is then:

$$t_{pv\_d2\_min} := R_{i\_pv} \cdot \left(e^{\frac{MAWP_{pv}}{S_{max}}} - 1 \right) \qquad t_{pv\_d2\_min} = 4.781\,mm$$

possible values

$t_{pv\_g2\_d2} := 6.35mm$ $t_{pv\_g3\_d2} := 5mm$

choose:

$t_{pv} := t_{pv\_g3\_d2}$

Flange thickness:

inner radius max. allowable pressure

$R_{i\_pv} = 0.57\,m$ $MAWP_{pv} = 15.4\,bar$ (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 ASME grade 3 CP titanium. The flange bolts and nuts will be inconel x750, (UNS N07750) as no titanium bolting materials are allowed. Inconel x750 is the highest strength allowed bolting material which does not contain Mo (which appears to be highly radioimpure). Inconel 718 is preferred, if proven sufficiently radiopure.

We will design to use one helicoflex 5mm gasket (smallest size possible) with aluminum facing (softest) loaded for helium leak rate.

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

Flange:

Grade 2 Titanium (UNS R50400) $S_{max\_Ti\_g2\_div1} := 14300psi$

Grade 3 Titanium (UNS R50500) $S_{max\_Ti\_g3\_div1} := 18600psi$

Bolting:

Inconel x750 (UNS N07750) $S_{max\_N07750} := 28700psi$

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

both in solution annealed condition

We choose grade 2 Ti , as grade 3 seems difficult to procure, and we choose Inconel x750 as it does not contain Molybdenum, which is suspected to have high background radioactivity

Maximum allowable design stress for flange

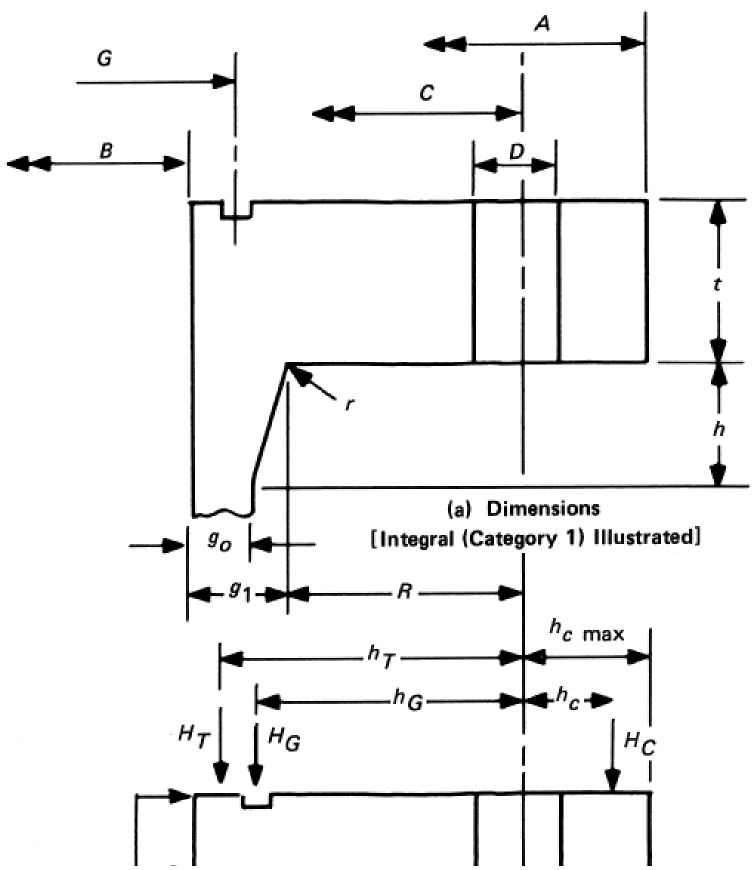
$S_f := S_{max\_Ti\_g3\_div1}$ $\qquad S_f = 128.2\,MPa$

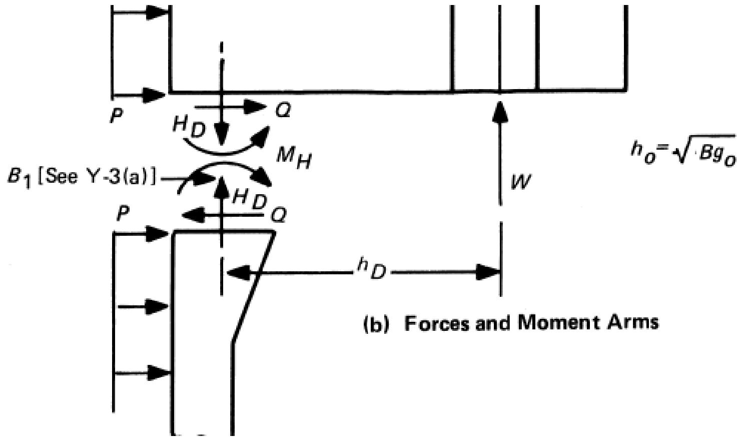
Maximum allowable design stress for bolts

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

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





(b) Forces and Moment Arms

hub thickness at flange (no hub)

$g_0 := t_{pv}$ $g_1 := t_{pv}$

corner radius

$r_1 := \max(.25g_1, 5mm)$ $r_1 = 5\,mm$

Flange OD

$A := A_{718\_g3}$

possible values

$A_{x750\_g2} := 1.37m$ $A_{x750\_g3} := 1.36m$ $A_{718\_g2} := 1.3m$ $A_{718\_g3} := 1.3m$

Flange ID

$B := 2R_{i\_pv}$ $B = 1.14\,m$

define:

$B_1 := B + g_1$ $B_1 = 1.145\,m$

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

$C := C_{718\_g3}$

$C_{x750\_g3} := 1.235m$ $C_{x750\_g2} := 1.23m$ $C_{718\_g2} := 1.22m$ $C_{718\_g3} := 1.22m$

Gasket dia

$G := 1.1475m$

Force of Pressure on head

$H := .785G^2 \cdot MAWP_{pv}$ $H = 1.613 \times 10^6\,N$

Helicoflex gasket, 5mm xsec dia.

aluminum jacket, from Helicoflex lit.:

Groove width, depth

$G_g := .627cm$ $F_g := .41cm$ $F_g = 0.161\,in$ $Y_2 := 220\dfrac{N}{mm}$

Sealing force, unit, assume O-ring, 0.275" dia., shore A 70 = ~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)

$Y_2 := 8\dfrac{lbf}{in}$ $D_j := G$ $D_j = 1.147\,m$ $F_j := 2\pi \cdot D_j \cdot Y_2$ $F_j = 1.586 \times 10^6\,N$ $F_g = 4.1 \times 10^{-3}\,m$

Start by making trial assumption (see sec Y.9 below) for number of bolts, root dia., pitch, bolt hole dia D, and check for hydraulic tensioner compatibility (wrench clearance similar but wrench use not recommended)

$n_{x750} := 68$ $n_{718} := 72$ $d_{b\_x750} := 22mm$ $d_{b\_718} := 20mm$

$n := n_{718}$ $d_b := d_{b\_718}$

Choosing ISO fine thread, with pitch; thread depth:

$p_t := 1.5\text{mm}$ $d_t := .614 \cdot p_t$

Nominal bolt dia is then;

$d_{b\_nom\_min} := d_b + 2d_t$ $d_{b\_nom\_min} = 21.842\,\text{mm}$

set: $d_{b\_nom\_x750} := 24\text{mm}$ $d_{b\_nom\_718} := 22\text{mm}$

$d_{b\_nom} := d_{b\_nom\_718}$

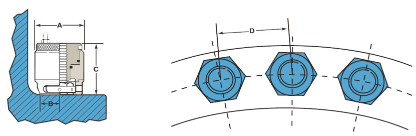
to provide clearance we will need :

$D_{tmin} := d_{b\_nom} + 1.5\text{mm}$ $D_{tmin} = 23.5\,\text{mm}$

set: $D_{t\_x750} := 26\text{mm}$ $D_{t\_718} := 24\text{mm}$

$D_t := D_{t\_718}$

Hydraulic tensioner dimension chart



A minimum of one thread diameter of thread engagment is recommended.
Stud Load is directly proportional to the hydraulic pressure applied.
A Nut Rotating Socket can be provided or the Nuts can be drilled.
Allow 9 inches (23 cm) above the Tensioner for Hose assembly.

Inch Tool Model	Metric Tool Model	Thread Size		Maximum Force		Hydraulic Area		Weight		- A - Tool Diameter		- B - Bridge Chamfer		- C - Overall Height		- D - Min. Stud Pitch	
		Inch	Metric	lbf	kN	in²	cm²	lb	kg	in	mm	in	mm	in	mm	in	mm

the following matrix contains these tabulated values:

$$
\begin{array}{cc}
 & \begin{array}{cccc} d_{b\_nom} & A & B & D \end{array} \\
ht := & \begin{pmatrix} 20 & 58 & 22 & 46 \\ 20 & 58 & 22 & 45 \\ 22 & 66 & 24 & 53 \\ 24 & 66 & 27 & 57 \\ 27 & 66 & 27 & 57 \\ 30 & 83 & 33 & 66 \\ 33 & 83 & 38 & 72 \\ 36 & 97 & 39 & 79 \\ 39 & 97 & 41 & 83 \end{pmatrix} \text{mm}
\end{array}
$$

$d_{b\_nom} = 22\,\text{mm}$

set index to match d_{b_nom} in first column:

$i := 2$

note: matrix indices start at zero, e.g.:

$ht_{0,0} = 20\,\text{mm}$ $ht_{2,3} = 53\,\text{mm}$

Then, define the following:

$$R_{.} := \frac{C - B}{2} - g_1 \qquad R_{.} = 3.5\,\text{cm}$$

Check if tensioner clears:

main shell $R_{.} > 0.5 \cdot ht_{i,1} = 1$

fillet $R_{.} - r_1 > ht_{i,2} = 1$

adjacent bolt $C \cdot n^{-1} \pi > ht_{i,3} = 1$

Compute the following Loads:

$H_G := F_j \qquad H_G = 1.586 \times 10^6 \, N$

$h_G := 0.5(C - G) \qquad h_G = 3.625 \, cm$

$H_D := .785 \cdot B^2 \cdot MAWP_{pv} \qquad H_D = 1.592 \times 10^6 \, N$

$h_D := D_t \qquad h_D = 2.4 \, cm$

$H_T := H - H_D \qquad H_T = 2.102 \times 10^4 \, N$

$h_T := 0.5(C - B) \qquad h_T = 40 \, mm$

Total Moment on Flange

$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \qquad M_P = 9.656 \times 10^4 \, J$

Appendix Y Calc Class 1 assembly, use eqs 5a, 7-13,14a, 15a 16 a

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

From Y-9, Trial bolt area required

$H = 1.613 \times 10^6 \, N$

$$A_{b'} := \frac{H + \frac{2M_P}{(A - C)}}{S_b} \qquad A_{b'} = 157.867 \, cm^2$$

$$d_{b'} := \frac{4}{\pi} \cdot \sqrt{\frac{A_{b'}}{n}} \qquad d_{b'} = 1.885 \, cm \qquad d_{b'} < d_b = 1$$

From Y-9, Trial flange thickness

First compute the following:

$$t_a := 2.45 \sqrt{\frac{M_P}{(\pi C - n \cdot D_t) S_f}} \qquad t_a = 4.634 \, cm$$

$$t_b := 0.56 B_1 \cdot \sqrt{\frac{MAWP_{pv}}{S_f}} \qquad t_b = 7.074 \, cm$$

$t_c := \max(t_a, t_b) \qquad t_c = 0.071 \, m$ (choose greater of t_a; t_b)

$t_{718\_g2} := 6cm \qquad t_{718\_g3} := 5cm$

$t_{x750\_g2} := 6cm \qquad t_{x750\_g3} := 5cm$

choose values for plate thickness and bolt hole dia:

$t := t_{718\_g3} \qquad D := D_t \qquad D = 2.4 \, cm$

going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \qquad \beta = 1.033 \quad h_C := 0.5(A - C) \qquad h_C = 0.04\,m$$

$$a := \frac{A + C}{2B_1} \quad a = 1.1 \qquad AR := \frac{n \cdot D}{\pi \cdot C} \quad AR = 0.451 \qquad h_0 := \sqrt{B \cdot g_0} \qquad h_0 = 0.075\,m$$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}}\,\mathrm{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 7.282 \times 10^{-3}$$

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)

$$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \qquad J_S = 0.095 \qquad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B \qquad J_P = 0.075$$

(5a)
$$F' := \frac{g_0^{\,2}(h_0 + F \cdot t)}{V} \qquad F' = 5.496 \times 10^{-6}\,m^3 \qquad M_P = 9.656 \times 10^4\,N \cdot m$$

$A = 1.3\,m \qquad B = 1.14\,m$

$$K := \frac{A}{B} \qquad K = 1.14 \qquad Z := \frac{K^2 + 1}{K^2 - 1} \; Z = 7.658$$

$f := 1$

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b \qquad l = 11\,cm \qquad A_b := n \cdot .785 d_b^{\,2}$

sec Y-6.2(a)(3) Elastic constants

(7-13) $E_{Ti\_g3} := 103.4 \cdot GPa \quad E := E_{Ti\_g3} \qquad E_{Inconel\_x750} := 213GPa \quad E_{bolt} := E_{Inconel\_x750}$

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \qquad M_S = -316.8\,J$$

$$\theta_B := \frac{5.46}{E \cdot \pi t^3}(J_S \cdot M_S + J_P \cdot M_P) \qquad \theta_B = 9.687 \times 10^{-4} \qquad E \cdot \theta_B = 100.159\,MPa$$

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = 2.406\times 10^6\,N$$

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = 5.606\times 10^6\,N$$

Compute Flange and Bolt Stresses

Bolt force total

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = 247.9\,MPa \qquad S_b = 255.1\,MPa \qquad F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2}$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = 0.485 \qquad F_{bolt} = 1.75\times 10^4\,lbf$$

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot (M_P + M_S)}{a \cdot t^3 \cdot l \cdot r_E \cdot B_1} \qquad S_i = 226.7\,MPa$$

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \qquad S_{R\_BC} = 109.7\,MPa \qquad S_f = 128.2\,MPa$$

$$S_{R\_ID1} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID1} = 0.238\,MPa$$

$$S_{T1} := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{T1} = 4.23\,MPa$$

$$S_{T3} := \frac{t \cdot E \cdot \theta_B}{B_1} \qquad S_{T3} = 4.374\,MPa$$

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = 13.193\,MPa$$

Y-7 Flange stress allowables:

(a) $\sigma_b < S_b = 1$

(b)

(1) $S_H < 1.5 S_f = 1$ $\qquad S_n$ 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

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

Torispheric head with standard dimensions. First step determine the following:

thickness: $t_{ts\_g2} := 6.35\text{mm}$ $t_{ts\_g3} := 5\text{mm}$

$t_{ts} := t_{ts\_g3}$

I.D.

$D_i := 1.14\text{m}$

O.D.

$D := D_i + 2t_{ts}$ $D = 1.15\,\text{m}$

Crown radius: Knuckle radius:

$L_{cr} := 1D_i$ $L_{cr} = 1.14\,\text{m}$ $r_{kn} := 0.1D_i$ $r_{kn} = 0.114\,\text{m}$

(b) Step 2- Compute the follwoing 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

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

$$\phi_{th} := \frac{\sqrt{L_{cr} \cdot t_{ts}}}{r_{kn}} \qquad \phi_{th} = 0.662\,\text{rad}$$

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

$R_{th} = 0.62\,\text{m}$

(d) Step 4 compute:

$$C_{1ts} := \begin{vmatrix} \left[9.31\left(\dfrac{r_{kn}}{D_i}\right) - 0.086\right] & \text{if } \dfrac{r_{kn}}{D_i} \le 0.08 \\ \left[0.692\left(\dfrac{r_{kn}}{D_i}\right) + 0.605\right] & \text{if } \dfrac{r_{kn}}{D_i} > 0.08 \end{vmatrix} \qquad \frac{r_{kn}}{D_i} \le 0.08 = 0 \qquad \frac{r_{kn}}{D_i} > 0.08 = 1$$

$$C_{1ts} = 0.674$$

$$C_{2ts} := \begin{vmatrix} 1.25 \ \text{if } \dfrac{r_{kn}}{D_i} \le 0.08 \\ 1.46 - 2.6\cdot\left(\dfrac{r_{kn}}{D_i}\right) \ \text{if } \dfrac{r_{kn}}{D_i} > 0.08 \end{vmatrix} \qquad \frac{r_{kn}}{D_i} \le 0.08 = 0 \qquad \frac{r_{kn}}{D_i} > 0.08 = 1$$

$$C_{2ts} = 1.2$$

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

$$P_{eth} := \frac{C_{1ts}\cdot E\cdot t_{ts}^{\ 2}}{C_{2ts}\cdot R_{th}\cdot\left(0.5R_{th} - r_{kn}\right)} \qquad P_{eth} = 118\,\text{bar}$$

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

time independent

$$C_{3ts} := S_{y\_Ti\_g3} \qquad S_{y\_Ti\_g2} := 40000\text{psi} \qquad S_{y\_Ti\_g3} := 55000\text{psi}$$

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

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

for: $G_{th} := \dfrac{P_{eth}}{P_y}$ $\quad G_{th} = 8.063$

$$P_{ck} := \left(\frac{0.77508\cdot G_{th} - 0.20354\cdot G_{th}^{\ 2} + 0.019274\cdot G_{th}^{\ 3}}{1 + 0.19014G_{th} - 0.089534G_{th}^{\ 2} + 0.0093965G_{th}^{\ 3}}\right)\cdot P_y \qquad P_{ck} = 28\,\text{bar}$$

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

$$P_{ak} := \frac{P_{ck}}{1.5} \qquad P_{ak} = 18.58\,\text{bar}$$

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

$$P_{ac} := \frac{2S_{max} \cdot 1}{\frac{L_{cr}}{t_{ts}} + 0.5} \qquad P_{ac} = 16.1\,\text{bar}$$

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

$$P_a := \min(P_{ak}, P_{ac}) \qquad P_a = 16.1\,\text{bar} \qquad P_a > P = 1$$

4.5.10.1 Radial Nozzle in formed head

$R_n := 5.1\text{cm}$

$t_n := 7\text{mm}$

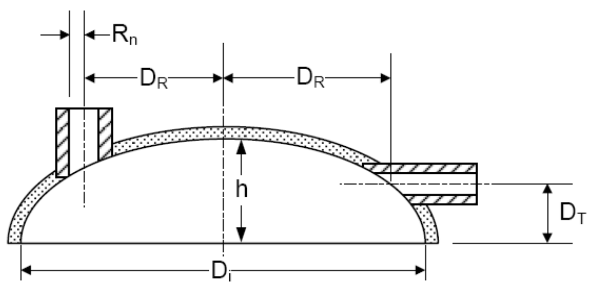
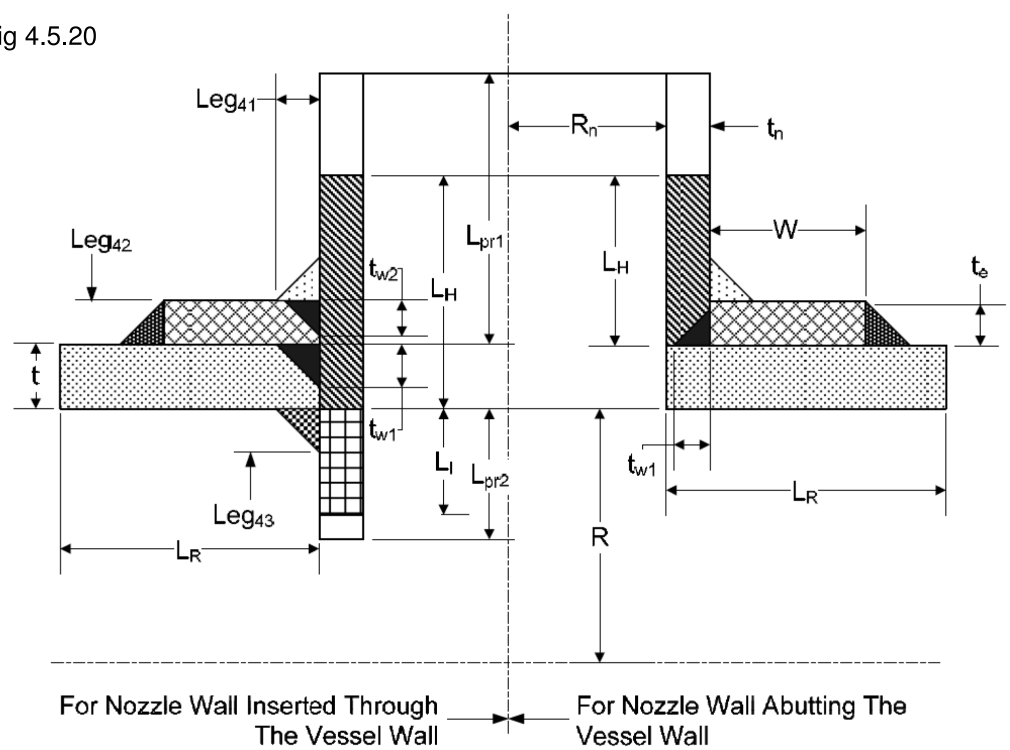


Fig 4.5.11

4.5.10.1 Procedure for Radial Nozzle in a Spherical or Formed head

Fig 4.5.20



a) Step 1

$R_{eff} := L_{cr} \qquad R_{eff} = 1.14\,\text{m}$ (4.5.64)

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

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

Possible limits of reinforcment:

$$L_{R1} := 0.5 \cdot D_i - (D_R + R_n + t_n) \qquad L_{R1} = \begin{pmatrix} 51.2 \\ 41.2 \end{pmatrix}\text{cm} \tag{4.5.67}$$

$$\sqrt{R_{eff} \cdot t_{ts}} = 7.55\,\text{cm} \qquad 2R_n = 10.2\,\text{cm}$$

$$L_{R2} := \min\left(\sqrt{R_{eff} \cdot t_{ts}}, 2R_n\right) \qquad L_{R2} = 7.55\,\text{cm} \tag{4.5.67}$$

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

$$L_R := \min(L_{R1}, L_{R2}) \qquad L_R = 7.55\,\text{cm} \tag{4.5.68}$$

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

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

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

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

where:

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

$$C_p := e^{\frac{0.35D_i - X_o}{8t_{ts}}} \qquad C_p = \begin{pmatrix} 5.039 \times 10^3 \\ 413.642 \end{pmatrix} \tag{4.5.78}$$

$$C_n := \min\left[\left(\frac{t_{ts} + t_e}{t_n}\right)^{0.35}, 1.0\right] \qquad C_n = 0.889 \tag{4.5.81}$$

$$F_p := \min(C_n, C_p) \qquad F_p = 0.889 \tag{4.5.80}$$

note that this is true for both values of C_p, so we can drop the parallel calculation

$$L_H := \min\left[\left(t_{ts} + t_e + F_p \cdot \sqrt{R_n \cdot t_n}\right), L_{pr1} + t_{ts}\right] \qquad L_H = 2.18\,\text{cm} \tag{4.5.73}$$

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)

$A_1 := t_{ts} \cdot L_R \qquad A_1 = 3.775\,cm^2$ (4.5.84)

$A_2 := t_n \cdot L_H \qquad A_2 = 1.526\,cm^2$ (4.5.86)

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

$L_{41} := 0.7cm$

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

$L_{42} := 0cm$

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

$L_{43} := 0.7cm$

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

$t_e = 0\,cm$

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

$A_T := A_1 + f_{rn} \cdot (A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp} \cdot A_5 \qquad A_T = 5.791\,cm^2$ (4.5.83)

f) Step 6 - determine applicable forces

$t_{eff} := t_{ts} \cdot \left(\frac{t_{ts} \cdot L_R + A_5 \cdot f_{rp}}{t_{ts} \cdot L_R}\right) \qquad t_{eff} = 5\,mm$ (4.5.100)

$R_{xn} := \frac{t_n}{\ln\left(\frac{R_n + t_n}{R_n}\right)} \qquad R_{xn} = 5.442\,cm \qquad R_{xs} := \frac{t_{eff}}{\ln\left(\frac{R_{eff} + t_{eff}}{R_{eff}}\right)} \qquad R_{xs} = 1.142\,m$ (4.5.98)

$f_N := P \cdot R_{xn} \cdot (L_H - t_{ts}) \qquad f_N = 1.427 \times 10^3\,N$ (4.5.95)

$f_S := \frac{P \cdot R_{xs} \cdot (L_R + t_n)}{2} \qquad f_S = 7.356 \times 10^4\,N$ (4.5.96)

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

$f_T := \frac{P \cdot R_{xs} \cdot R_{nc}}{2} \qquad f_T = 4.547 \times 10^4\,N$ (4.5.97)

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

$t_{eff} = 0.5\,cm$ 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

$$\sigma_{avg} := \frac{f_N + f_S + f_T}{A_T} \qquad \sigma_{avg} = 208\,\text{MPa} \tag{4.5.101}$$

$$\sigma_{circ} := \frac{P \cdot R_{xs}}{2t_{eff}} \qquad \sigma_{circ} = 178.3\,\text{MPa} \tag{4.5.102}$$

I) Step 9 Determine maximum local primary membrane stress

$$P_L := \max\left[\left(2\sigma_{avg} - \sigma_{circ}\right), \sigma_{circ}\right] \qquad P_L = 237.7\,\text{MPa} \tag{4.5.103}$$

$$E_w = 1$$

$$S_{allow} := 1.5 S_{max} \cdot E_w \qquad S_{allow} = 280.3\,\text{MPa} \tag{4.5.43}$$

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

$$P_L \le S_{allow} = 1 \tag{4.5.104}$$

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

$$A_p := R_{xn} \cdot \left(L_H - t_{ts}\right) + \frac{R_{xs} \cdot \left(L_R + t_n + R_{nc}\right)}{2} \qquad A_p = 771.7\,\text{cm}^2 \tag{4.5.108}$$

$$P_{max1} := \frac{S_{allow}}{\left(\frac{2A_p}{A_T} - \frac{R_{xs}}{2t_{eff}}\right)} \qquad P_{max1} = 18.2\,\text{bar} \tag{4.5.105}$$

$$S := S_{max} \qquad S = 186.8\,\text{MPa}$$

$$P_{max2} := 2 \cdot S \cdot \left(\frac{t_{ts}}{R_{xs}}\right) \qquad P_{max2} = 16.1\,\text{bar} \tag{4.5.106}$$

$$P_{max} := \min\left(P_{max1}, P_{max2}\right) \qquad P_{max} = 16.1\,\text{bar} \tag{4.5.107}$$

$$P_{max} > P = 1$$