Active volume dimensions, from earlier analyses:

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

We consider using a field cage solid insulator/light tube of 3 cm total thk., and a copper liner of 12 cm thickness, including all tolerances and necessary gaps.

$t_{fc} := 3cm \qquad t_{Cu} := 12cm$

Pressure Vessel Inner Radius, Diameter:

$R_{i\_pv} := r_{Xe} + t_{fc} + t_{Cu} \qquad R_{i\_pv} = 0.68\,m \qquad D_{i\_pv} := 2R_{i\_pv} \qquad D_{i\_pv} = 1.36\,m$

Pressure vessel length:

main cyl. vessel — overall, inside

$L_v := 1.6m \qquad L_o := 2.2m$

Temperatures:

For pressure operation, the temperature range will be 10C-30C. For vacuum operation, the temperature range will be 10C to 150C (bakeout).

Maximum Operating Pressure (MOP), gauge:

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

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)
From LBNL PUB3000, recommended minimum is, 10% over max operating pressure; this is design pressure at LBNL. This is for spring operated relief valves, to avoid leakage. Use of pilot operated relief valves can reduce this to as little as 2%, as they seal tighter when approaching relief pressure:

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

$P := MAWP_{pv}$

Mass supported internally by pressure vessel

Internal copper shield (ICS)

$M_{ICS\_cyl} := 6000kg \qquad M_{ICS\_eh} := 1500kg \qquad M_{ICS\_tp} := 2500kg$

Detector subsystems, est.

$M_{ep} := 750kg \qquad M_{tp} := 200kg \qquad M_{fc} := 350kg$

Length inside vessel of copper, total

$L_{Cu} := 2.0m$

Mass total of internal copper shielding:

$M_{ICS} := M_{ICS\_cyl} + M_{ICS\_eh} + M_{ICS\_tp} \qquad M_{ICS} = 10000\,kg$

Maximum mass supported on internal flange of each head:

$M_{fl\_h} := M_{ICS\_tp}$

Maximum mass supported on each internal flange of the main cylindrical vessel:

$M_{fl\_v} := 0.5(M_{ICS\_cyl} + M_{fc}) + M_{ep} \qquad M_{fl\_v} = 3925\,kg$ this mass will be present when heads are not mounted

Estimated approximate total vessel mass carried on supports (numbers from calcs below):

vessel heads flanges

$$M_v := \rho_{SS} \cdot \left(2\pi R_{i\_pv} \cdot 10mm \cdot L_o + \pi R_{i\_pv}^{\;2} \cdot 12mm + 4 \cdot 2\pi R_{i\_pv} \cdot 4.2cm \cdot 5cm\right) \quad M_v = 1179\,kg \qquad \rho_{SS} := 8\,\frac{gm}{cm^3}$$

Total detector mass:

$$M_{det} := M_{ICS} + M_{ep} + M_{fc} + M_{tp} + M_v \qquad M_{det} = 1.248 \times 10^4\,kg$$

Vessel wall thicknesses

Material:

We use 316Ti for vessel shells and flanges due to its known good radiopurity and strength.

Design Rules:

ASME Boiler and Pressure Vessel code section VIII, Rules for construction of Pressure vessels division 1 (2010)

316Ti is not an allowed material under section VIII, division 2, so we must use **division 1** rules. The saddle supports are however, deisgned using the methodology given in div. 2, as div. 1 does not provide design formulas (nonmandatory Appendix G)

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

$S_{max\_316Ti\_div1} := 20000psi$ -20F - 100F

Youngs modulus

$E_{SS\_aus} := 193GPa$

color scheme for this document

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

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

Choose material, then maximum allowable strength is:

$S := S_{max\_316Ti\_div1}$

Vessel wall thickness, for internal pressure is then (div. 1), Assume all welds are type (1) as defined in UW-12 , are double welds, fully radiographed, so weld efficiency:

$E := 1$

Minimum wall thickness is then:

$$t_{pv\_d1\_min\_ip} := \frac{P \cdot R_{i\_pv}}{S \cdot E - 0.6 \cdot P}$$

$t_{pv\_d1\_min\_ip} = 7.75\,mm$

We set wall thickness to be:

$t_{pv} := 10mm$ $t_{pv} > t_{pv\_d1\_min\_ip} = 1$

Maximum Allowable External Pressure

ASME PV code Sec. VIII, Div. 1- UG-28 Thickness of Shells under External Pressure

Maximum length between flanges $L_{ff} := 1.6m$

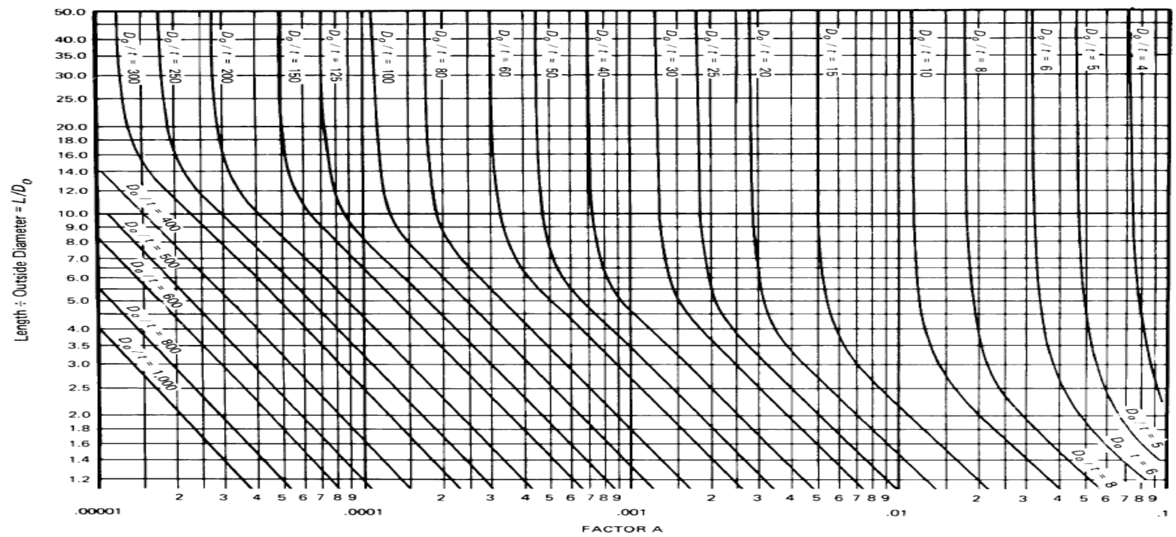
The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{ff}}{2R_{i\_pv}} = 1.2 \qquad \frac{2R_{i\_pv}}{t_{pv}} = 136$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor A:

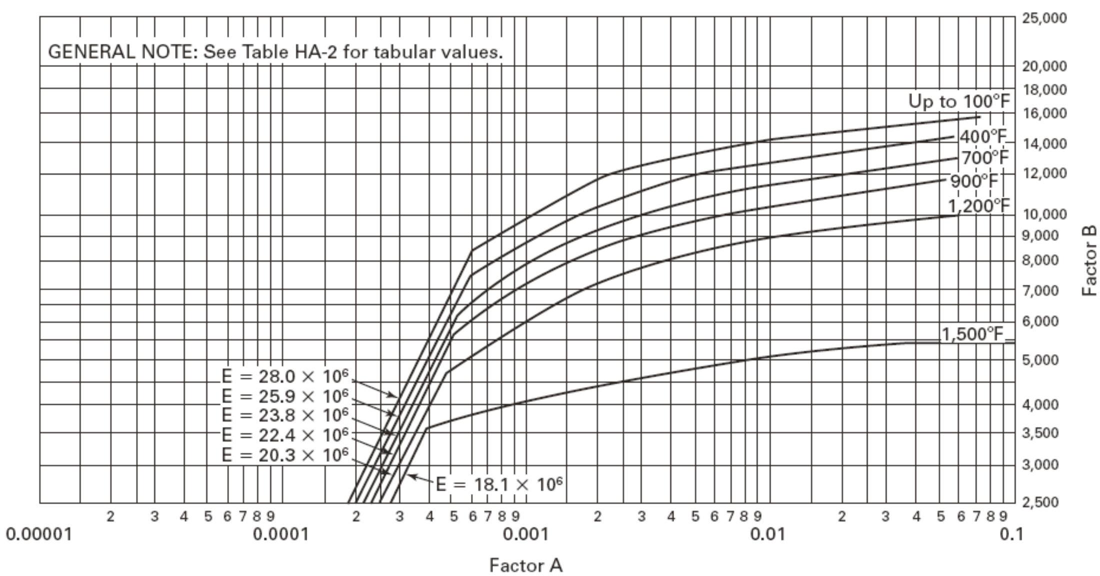
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$A_{.} := 0.0005$

Using the factor A in chart (HA-2) in Subpart 3 of Section II, Part D, we find the factor B (@ 400F, since we may bake while pulling vacuum):

FIG. HA-2 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE DEVELOPED FOR AUSTENITIC STEEL 16Cr–12Ni–2Mo, TYPE 316



$B_{.} := 6200\mathrm{psi}$ @ 400F

The maximum allowable working external pressure is then given by :

$$P_a := \frac{4B_{.}}{3\left(\frac{2R_{i\_pv}}{t_{pv}}\right)} \qquad P_a = 4.1\,\mathrm{bar} \qquad -P_{min} = 1.5\,\mathrm{bar}$$

$$P_a > -P_{min} = 1$$

Flange thickness, head to vessel main flanges:

inner radius max. allowable pressure

$R_{i\_pv} = 0.68\,m$ $P = 15.4\,bar$ (gauge pressure)

The flange design for O-ring sealing (or other self energizing gasket such as helicoflex) 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. The rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the allowable stresses of division 1. Flanges and shells will be fabricated from 316Ti (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected for flat laminar flaws which may create leak paths. 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 with enough flange strength to accomodate using a Helicoflex 5mm gasket (smallest size possible) specially designed with a maximum sealing force of 70 N/mm.

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

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti\_div1}$ $S_f = 137.9\,MPa$ $S_f = 2 \times 10^4\,psi$

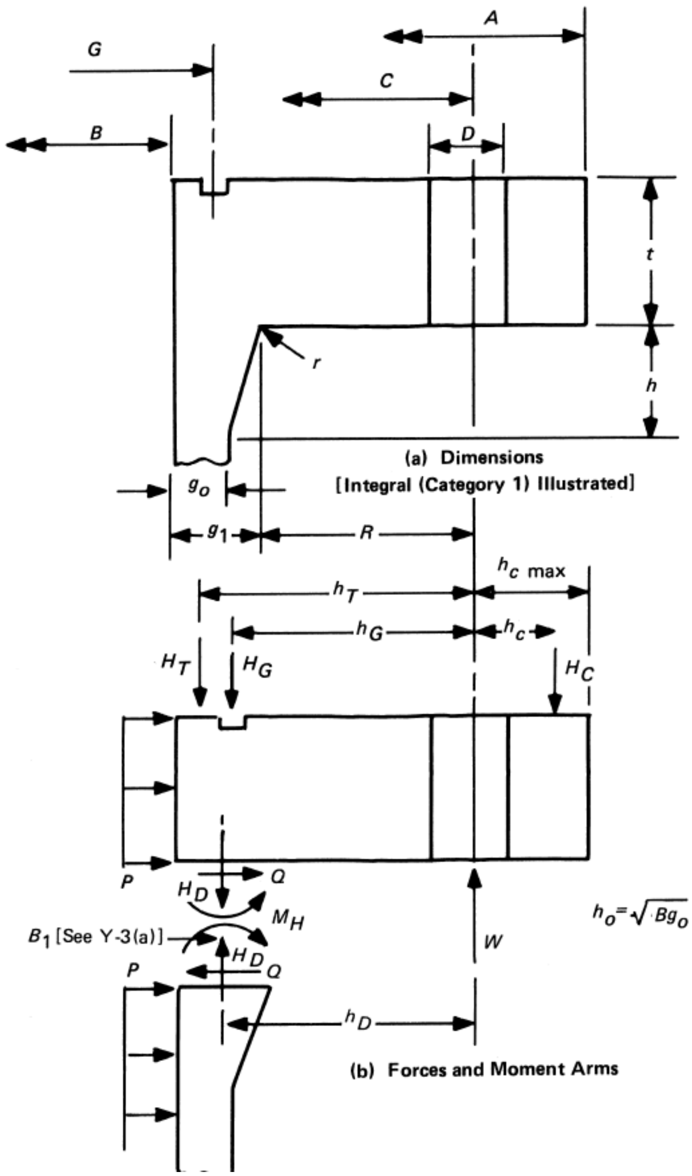
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

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

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

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) corner radius:

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

Flange OD

$A := 1.48\text{m}$

Flange ID

$B := 2R_{i\_pv}$ $B = 1.36\,\text{m}$

define:

$B_1 := B + g_1$ $B_1 = 1.37\,\text{m}$

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

$C := 1.43 \cdot \text{m}$

Gasket dia

$G := 2(R_{i\_pv} + .65\text{cm})$ $G = 1.373\,\text{m}$ O-ring mean radius as measured in CAD model: $68.65 \cdot 2 = 137.3$

Note: this diameter will be correct for Helicoflex gasket, but slightly higher for O-ring, which is fluid and "transmits pressure" out to its OD, howqever the lower gasket unit force of O-ring more than compensates, as per below:

Force of Pressure on head

$H := .785G^2 \cdot MAWP_{pv}$ $\quad H = 2.31 \times 10^6 \, N$

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 equivalent formulas using Y as the unit force term and gives several possible values.

for 5mm HN200 with aluminum jacket:

$Y_1 := 70 \frac{N}{mm}$ min value for our pressure and required leak rate (He) $\quad Y_2 := 220 \frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter $D_j := G \quad D_j = 1.373 \, m$

Force is then either of:

$F_m := \pi D_j \cdot Y_1$ or $F_j := \pi \cdot D_j \cdot Y_2$

$F_m = 3.019 \times 10^5 \, N \quad F_j = 9.489 \times 10^5 \, N$

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 in between, 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. Note: in the cold trap one will get water and N2, O2, that permeates through the outer O-ring as well.

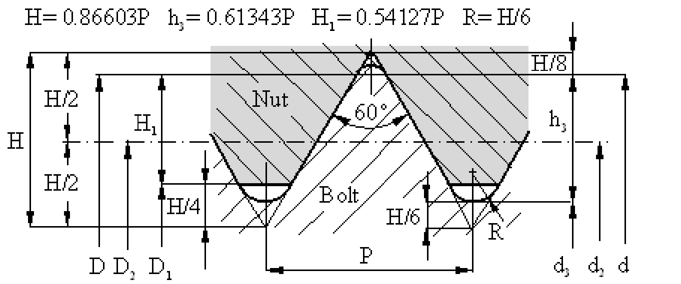
Start by making trial assumption for number of bolts, nominal bolt dia., pitch, and bolt hole dia D,

$n := 132 \quad d_b := 16mm$ maximum number of bolts possible, using narrow washers: $n_{max} := \mathrm{trunc}\left(\frac{\pi C}{2.0 d_b}\right) \quad n_{max} = 140$

Choosing ISO fine thread, to maximize root dia.; thread depth is:

$p_t := 1.0mm \quad h_3 := .6134 \cdot p_t$

using nomenclature and formulas from this chart at http://www.tribology-abc.com/calculators/metric-iso.htm



Nominal diameter d = D	Pitch P	root radius r	pitch diameter d2=D2	minor diameter d3	D1	thread height h3	H1	drill diameter mm	
metric screw threads ISO 724 (DIN 13 T1)									
M 1.00	0.25	0.036	0.838	0.693	0.729	0.153	0.135	0.75	
M 1.10	0.25	0.036	0.938	0.793	0.829	0.153	0.135	0.85	
M 1.20	0.25	0.036	1.038	0.893	0.929	0.153	0.135	0.95	
M 1.40	0.30	0.043	1.205	1.032	1.075	0.184	0.162	1.10	
M 1.60	0.35	0.051	1.373	1.171	1.221	0.215	0.189	1.25	
M 1.80	0.35	0.051	1.573	1.371	1.421	0.215	0.189	1.45	
M 2.00	0.40	0.058	1.740	1.509	1.567	0.245	0.217	1.60	
M 2.20	0.45	0.065	1.908	1.648	1.713	0.276	0.244	1.75	
M 2.50	0.45	0.065	2.208	1.948	2.013	0.276	0.244	2.05	
M 3.00	0.50	0.072	2.675	2.387	2.459	0.307	0.271	2.50	
M 3.50	0.60	0.087	3.110	2.764	2.850	0.368	0.325	2.90	
M 4.00	0.70	0.101	3.545	3.141	3.242	0.429	0.379	3.30	
M 4.50	0.75	0.108	4.013	3.580	3.688	0.460	0.406	3.80	
M 5.00	0.80	0.115	4.480	4.019	4.134	0.491	0.433	4.20	
M 6.00	1.00	0.144	5.350	4.773	4.917	0.613	0.541	5.00	<---use h3 for 1.0 mm pitch
M 7.00	1.00	0.144	6.350	5.773	5.917	0.613	0.541	6.00	
M 8.00	1.25	0.180	7.188	6.466	6.647	0.767	0.677	6.80	
M 9.00	1.25	0.180	8.188	7.466	7.647	0.767	0.677	7.80	
M 10.00	1.50	0.217	9.026	8.160	8.376	0.920	0.812	8.50	<--- use H1 for 1.5mm pitch
M 11.00	1.50	0.217	10.026	9.160	9.376	0.920	0.812	9.50	
M 12.00	1.75	0.253	10.863	9.853	10.106	1.074	0.947	10.20	
M 14.00	2.00	0.289	12.701	11.546	11.835	1.227	1.083	12.00	
M 16.00	2.00	0.289	14.701	13.546	13.835	1.227	1.083	14.00	
M 18.00	2.50	0.361	16.376	14.933	15.394	1.534	1.353	15.50	
M 20.00	2.50	0.361	18.376	16.933	17.294	1.534	1.353	17.50	

Bolt root dia. is then:

$d_3 := d_b - 2h_3$ $\qquad\qquad$ $d_3 = 14.7732\,\text{mm}$

Total bolt cross sectional area:

$A_b := n\cdot\frac{\pi}{4}d_3^{\,2}$ $\qquad\qquad$ $A_b = 226.263\,\text{cm}^2$

Check bolt to bolt clearance, here we use narrow thick washers (28mm OD) under the 24mm wide (flat to flat) nuts (28mm is also corner to corner distance on nut), we adopt a minimum bolt spacing of 2x the nominal bolt diameter (to give room for a 24mm socket) :

$\pi C - 2.0n\cdot d_b \geq 0 = 1$ $\qquad$ actual bolt to bolt distance: $\frac{\pi C}{n} = 34.034\,\text{mm}$

Check nut, washer, socket clearance: $OD_w := 2d_b$ $\qquad$ this is for standard narrow washers, and for wrench sockets which more than cover the nut width across corners

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

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 2\text{mm}$ $\qquad\qquad$ $D_{tmin} = 18\,\text{mm}$

We will thread some of these clearance holes for M20-1.5 bolts to allow the head retraction fixture to be bolted up the the flange. The effective diameter of these holes will be the average of nominal and minimum diameters. To avoid thread interference with flange bolts, the studs will be machined to root diameter per **UG-12(b)**.in between threaded ends of 1.5x diameter in length. The actual clearance holes will be 18mm, depending on achievable tolerances, so as to allow threading where needed.

$H_1 := .812\text{mm}$ from chart above

$d_{min\_20\_1.5} := 20\text{mm} - 2 \cdot H_1$ $d_{min\_20\_1.5} = 1.838\,\text{cm}$ this will be max bolt hole size or least material condition (LMC)

$d_{min\_20\_1.5} \geq D_{tmin} = 1$

$D_e := 0.5(20 \cdot \text{mm} + d_{min\_20\_1.5})$ $D_e = 1.919\,\text{cm}$

Set:

$D_t := D_e$ $D_t > D_{tmin} = 1$

Compute Forces on flange:

We use a unit gasket seating force of Y1 above

$H_G := F_m$ $H_G = 3.019 \times 10^5\,\text{N}$

$h_G := 0.5(C - G)$ $h_G = 2.85\,\text{cm}$ from Table 2-6 Appendix 2, Integral flanges

$H_D := .785 \cdot B^2 \cdot P$ $H_D = 2.266 \times 10^6\,\text{N}$

$R_. := 0.5(C - B) - g_1$ $R_. = 2.5\,\text{cm}$ radial distance, B.C. to hub-flange intersection, int fl..

$h_D := R_. + 0.5 g_1$ $h_D = 3\,\text{cm}$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D$ $H_T = 4.353 \times 10^4\,\text{N}$

$h_T := 0.5 \cdot (R_. + g_1 + h_G)$ $h_T = 31.75\,\text{mm}$ from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange

$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G$ $M_P = 7.797 \times 10^4\,\text{J}$

Appendix Y Calculation

$P = 15.4\,\text{bar}$

Choose values for plate thickness and bolt hole dia:

$t := 4.15\text{cm}$ $D := D_t$ $D = 1.919\,\text{cm}$

Going back to main analysis, compute the following quantities:

$\beta := \dfrac{C + B_1}{2B_1}$ $\beta = 1.022$ $h_C := 0.5(A - C)$ $h_C = 2.5\,\text{cm}$

$a := \dfrac{A + C}{2B_1}$ $a = 1.062$ $AR := \dfrac{n \cdot D}{\pi \cdot C}$ $AR = 0.564$ $h_0 := \sqrt{B \cdot g_0}$ $h_0 = 11.662\,\text{cm}$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}}\,\text{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 7.462 \times 10^{-3}$$

We need factors F and V, 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)

$$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \qquad J_S = 0.083 \qquad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B \qquad J_P = 0.062$$

(5a) $$F' := \frac{{g_0}^2 (h_0 + F \cdot t)}{V} \qquad F' = 2.806 \times 10^{-5}\,\mathrm{m}^3 \qquad M_P = 7.797 \times 10^4\,\mathrm{N \cdot m}$$

$A = 1.48\,\mathrm{m}$ $B = 1.36\,\mathrm{m}$

$$K := \frac{A}{B} \qquad K = 1.088 \qquad Z := \frac{K^2 + 1}{K^2 - 1} \qquad Z = 11.854$$

$f := 1$ hub stress correction factor for integral flanges, use f =1 for g1/g0=1 (fig 2-7.6)

$t_s := 0\mathrm{mm}$ no spacer between flanges

$l := 2t + t_s + 0.5d_b$ $l = 9.1\,\mathrm{cm}$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

http://www.hightempmetals.com/techdata/hitempInconel718data.php

Elastic constants:

$E := E_{SS\_aus}$ $E = 193\,\mathrm{GPa}$ $E_{Inconel\_718} := 208\mathrm{GPa}$ $E_{bolt} := E_{Inconel\_718}$

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \qquad M_S = -1.8 \times 10^3\,\mathrm{N \cdot m} \qquad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right) \qquad \theta_B = 5.903 \times 10^{-4} \qquad (8)$$

$E \cdot \theta_B = 113.924\,\mathrm{MPa}$

opening half gap = $\theta_B \cdot 3\mathrm{cm} = 0.018\,\mathrm{mm}$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = 3.045 \times 10^6\,\mathrm{N} \qquad (9)$$

Bolt Load at operating condition:

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = 5.657 \times 10^6\,\mathrm{N} \qquad (10)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = 250\,\mathrm{MPa} \qquad S_b = 255.1\,\mathrm{MPa} \qquad (11)$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = 0.928$$ elasticity factor

Design Prestress in bolts

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot (M_P + M_S)}{a \cdot t^3 \, l \cdot r_E \cdot B_1} \qquad S_i = 243.7\,MPa \qquad (12)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \qquad S_{R\_BC} = 135.4\,MPa \qquad (13)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID} = 1.61\,MPa \qquad (14a)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_T = 2.46\,MPa \qquad (15a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = 19.372\,MPa \qquad (16a)$$

Y-7 Bolt and Flange stress allowables: $S_b = 255.1\,MPa$ $\quad S_f = 137.9\,MPa$

(a) $\sigma_b < S_b = 1$

(b)

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

(2) not applicable

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

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

(d) $S_T < S_f = 1$

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

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

(f) not applicable

Bolt force

$$F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2} \qquad F_{bolt} = 5.024 \times 10^4\,N$$

Bolt torque required, minimum:

$T_{bolt\_min} := 0.2F_{bolt} \cdot d_b$ $\quad T_{bolt\_min} = 160.8\,N \cdot m \quad T_{bolt\_min} = 118.6\,lbf \cdot ft$ for pressure test use 1.5x this value

This is the minimum amount of bolt preload needed to assure joint does not open under pressure. An additional amount of bolt preload is needed to maintain a minimum frictional shear resistance to assure head does not slide downward from weight; we do not want to depend on lip to carry this. Non-mandatory Appendix S of div. 1 makes permissible higher bolt stresses than indicated above when needed to assure full gasket sealing and other conditions. This is consistent with proper preloaded joint practice, for properly designed joints where connection stiffness is much greater than bolt stiffness, and we are a long way from the yield stress of the bolts

$M_{head} := 2500kg$ $\quad \mu_{SS\_SS} := .7$ typ. coefficient of friction, stainless steel (both) clean and dry

$V_{head} := M_{head} \cdot g$ $\quad V_{head} = 2.452 \times 10^4\,N$

$F_n := \frac{V_{head}}{\mu_{SS\_SS}}$ $\quad F_n = 3.502 \times 10^4\,N$ this is total required force, force required per bolt is:

$F_{n\_bolt} := \frac{F_n}{n}$ $\quad F_{n\_bolt} = 265.331\,N$ this is insignificant compared to that required for pressure.

Let bolt torque for normal operation be then 25% greater than minimum:

$T_{bolt} := 1.25T_{bolt\_min}$ $\quad T_{bolt} = 201\,N \cdot m \quad T_{bolt} = 148\,ft \cdot lbf$

It is recommended that a pneumatic torque wrench be used for tightening of bolts. Anti-seize lubricant (checked for radiopurity) should be used on threads and washers. Bolts should be tightened in 1/3 full torque increments, but there is no specific tightening pattern to be used, as gasket compression is not determined by bolt tightness. The head lift fixture may be retracted once all bolts not occupied by lift fixture have been tightened to the first 1/3 torque increment; there will be adequate frictional shear resistance to eliminate head slippage while detaching lift fixture. Bolts should be run up uniformly to fully close gap before proceeding with tightening. Do not forget to install sleeves in all threaded holes after removing lift fixture.

Additional Calculations for Shielding Weight:

Shear stress in inner flange lip from shield (could happen only if flange bolts come loose, are left loose, or if joint opens under pressure, otherwise friction of faces will support shield, given additional tension, as permissible under non-mandatory Appendix S above)

Masses of Copper shielding in cyl and heads (maybe extra in tracking head)

$t_{Cu} = 0.12\,m \quad t_{Cu\_h} := 20cm \quad L_{ff} = 1.6\,m \quad \rho_{Cu} = 9 \times 10^3 \frac{kg}{m^3}$

$M_{sh\_head} := \rho_{Cu} \cdot \pi R_{i\_pv}^2 \cdot t_{Cu\_h}$ $\quad M_{sh\_head} = 2.615 \times 10^3\,kg$

$M_{sh\_cyl} := \rho_{Cu} \cdot 2\pi \cdot R_{i\_pv} \cdot t_{Cu} \cdot L_{ff}$ $\quad M_{sh\_cyl} = 7.383 \times 10^3\,kg$

$M_{sh} := M_{sh\_cyl} + 2M_{sh\_head}$ $\quad M_{sh} = 1.261 \times 10^4\,kg$ slightly less than this, due to gaps

$t_{lip} := 3mm$

Shear stress in lip (projected force):

$\tau_{lip} := \frac{M_{sh\_head} \cdot g}{R_{i\_pv} \cdot t_{lip}}$ $\quad \tau_{lip} = 12.57\,MPa$

Shear stress on O-ring land (section between inner and outer O-ring), from pressurized O-ring. This is assumed to be the primary stress. There is some edge moment but the "beam" is a very short one. This shear stress is not in the same direction as the nominal tangential (hoop) stress of the flange.

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

$F_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot w_{land\_axial} \cdot P$

$A_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot t_{land\_radial}$

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

Bolt loads from Cu bars

The internal copper shield bars are attached to the inside flanges with M6-1 bolts. The worst case for attachment ar the bars with collimation holes; these are narrow where they attach. For a flange hole pattern of 240 bolts, there are 5 attachment holes at each end.

$d_{root\_M6} := 4.77mm$

On the tracking side, the bars will be pulled up tight to the inside flange. On the energy side they must float axially, this is done using a special shoulder bolt which provides a loose double shear connection. Worst case would be single shear, where the tracking side bolts are left loose.

$M_{cubar\_vfan} := 225kg$

$\tau_{bolt\_cubar} := \frac{0.5M_{cubar\_vfan} \cdot g}{5 \cdot \frac{\pi}{4} d_{root\_M6}^{2}} \qquad \tau_{bolt\_cubar} = 12.347\,MPa$

This stress is inconsequential, as bolts will be ASME SB-98 silicon copper UNS C65500 - HO2 (half hard cond); this material should be radiopure and has > 20% elongation in the hard condition. Shear strength in yield is 50% Sy.

$S_{y\_65500\_H2} := 38000psi \qquad S_{y\_65500\_H2} = 262.001\,MPa$

$S_{sy\_65500\_H2} := 0.5S_{y\_65500\_H2} \qquad S_{sy\_65500\_H2} = 131\,MPa$

O-Ring groove dimensions

the Recommmended range of compression for static face seals is 21-30% in the Parker O-ring handbook; Trelleborg recommend 15-30%. For each nominal size, there are several cross sections, metric, JIS and A-568. Ity ios recommended by this author to design a groove which can accomodate all these cross sections with squezze in the acceptable range, so as to give the most flexibility.

For large diameter O-rings, Parker recommends using one size smaller to avoid sag. This is feasible for the inner O-ring, as the undercut lip is on the ID of the groove, but will not work on the outer vacuum O-ring as the undecut must be on the OD (otherwise the undercut may reduce seal effectiveness). Using an O-ring 1 or 2 sizes larger on the outer O-ring may develop enough compressive stress to retain O-ring in groove, but this should be tested. Stiffer compounds may help here if there is a problem Regardless, the groove dimensions should account for the stretch or compression of the O-ring which changes its effective cross section diameter. There are several close sizes that Trelleborg makes unspliced O-rings from (these are strongly preferred) and a stiffer than normal compound could be used for the vacuum O-ring, if needed

Inner (pressure bearing) O-ring:

Groove wall radii (average), depth, inner corner radii:

$R_{Ogpo} := 688.7\text{mm}$ $R_{Ogpi} := 682.25\text{mm}$ $d_{Opg} := 3.8\text{mm}$ $r_{ip} := 1\text{mm}$

O-ring inner radius, cross section diameter, unstretched

$R_{Opi} := 660\text{mm}$ $d_{Op} := \begin{pmatrix} 5 \\ 5.34 \end{pmatrix}\text{mm}$ metric size
AS - 568 size

O-ring elongation (tangential direction, normal to cross section)

$\varepsilon_{Opt} := 1 - \dfrac{R_{Opi}}{R_{Ogpi}}$ $\varepsilon_{Opt} = 3.261\,\%$ recommeded less than 3% (Trelleborg); 3% is our min. target

Bulk Modulus of most rubber polymers is very high, material is essentially incompressible (Poisson's ratio = -0.5)

Strain, O-ring cross section, in axial direction

$\varepsilon_{Opa} := -0.5\varepsilon_{Opt}$ $\varepsilon_{Opa} = -0.016$

O-ring dia., stretched:

$d_{Ops} := d_{Op}\cdot\left(1 + \varepsilon_{Opa}\right)$ $d_{Ops} = \begin{pmatrix} 4.918 \\ 5.253 \end{pmatrix}\text{mm}$

Resulting squeeze (using the vectorize operator to continue parallel calculations)

$sq_p := \overrightarrow{\left(\dfrac{d_{Ops} - d_{Opg}}{d_{Ops}}\right)}$ $sq_p = \begin{pmatrix} 22.74 \\ 27.659 \end{pmatrix}\%$ $15\% < sq_p < 30\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

O-ring groove cross sectional area,

$A_{Opg} := \overrightarrow{\left[d_{Opg}\cdot\left(R_{Ogpo} - R_{Ogpi}\right) - \left(\dfrac{1}{2} - \dfrac{\pi}{2}\right)\cdot r_{ip}^{\ 2}\right]}$ $A_{Opg} = 2.558 \times 10^{-5}\,\text{m}^2$

Trelleborg recommends no more than 85% fill ratio

$R_{fp} := \dfrac{\overrightarrow{\frac{\pi}{4}d_{Ops}^{\ 2}}}{A_{Opg}}$ $R_{fp} = \begin{pmatrix} 74.274 \\ 84.718 \end{pmatrix}\%$ $R_{fp} < 85\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

Outer (vacuum) O-ring:

Groove wall radii (average), depth, inner corner radii:

$R_{Ogvo} := 697.66mm$ $R_{Ogvi} := 692.93mm$ $d_{Ovg} := 2.6mm$ $r_{iv} := 0.6mm$

O-ring inner radius, cross section diameter, unstretched

$R_{Ovi} := 730mm$ $d_{Ov} := \begin{pmatrix} 3 \\ 3.55 \end{pmatrix} mm$ metric size / metric/JIS size note: there are several intermediate sizes

O-ring elongation (tangential direction, normal to cross section)

$\varepsilon_{Ovt} := 1 - \frac{R_{Ovi}}{R_{Ogvi}}$ $\varepsilon_{Ovt} = -5.35\,\%$ recommended less than 3% (Trelleborg); we go for ~5% here as compression should not compromise integrity

Bulk Modulus of most rubber polymers is very high, material is essentially incompressible (Poisson's ratio = -0.5)

Strain, O-ring cross section, in axial direction

$\varepsilon_{Ova} := -0.5\varepsilon_{Ovt}$ $\varepsilon_{Ova} = 0.027$

O-ring dia., stretched:

$d_{Ovs} := d_{Ov} \cdot \left(1 + \varepsilon_{Ova}\right)$ $d_{Ovs} = \begin{pmatrix} 3.08 \\ 3.645 \end{pmatrix} mm$

Resulting squeeze

$sq_v := \overrightarrow{\frac{d_{Ovs} - d_{Ovg}}{d_{Ovs}}}$ $sq_v = \begin{pmatrix} 15.591 \\ 28.669 \end{pmatrix} \%$ $15\% < sq_v < 30\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

O-ring groove cross sectional area,

$A_{Ovg} := \overrightarrow{\left[d_{Ovg} \cdot \left(R_{Ogvo} - R_{Ogvi}\right) - \left(\frac{1}{2} - \frac{\pi}{2}\right) \cdot r_{iv}^{\,2} \right]}$ $A_{Ovg} = 1.268 \times 10^{-5}\, m^2$

Fill ratio; Trelleborg recommends no more than 85%:

$R_{fv} := \overrightarrow{\frac{\frac{\pi}{4} d_{Ovs}^{\,2}}{A_{Ovg}}}$ $R_{fv} = \begin{pmatrix} 58.752 \\ 82.269 \end{pmatrix} \%$ $R_{fv} < 85\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ We should have a comfortable margin here

Support Design using rules of div 2, part 4.15:

From the diagram below the rules are only applicable to flange attached heads if there is a flat cover or tubesheet inside, effectively maintaining the flanges circular. Since the PMT carrier plate and shielding is firmly bolted in, it serves this purpose and we may proceed. We must also compute the case with the heads attached, as there will be additional load

a) Design Method- although not specifically stated, the formulas for bending moments at the center and at the supports are likely based on a uniform loading of the vessel wall from the vessel contents. In this design, the internal weight (primarily of the copper shield) is applied at the flanges; there is no contact with the vessel shell. We calculate both ways and take the worst case.

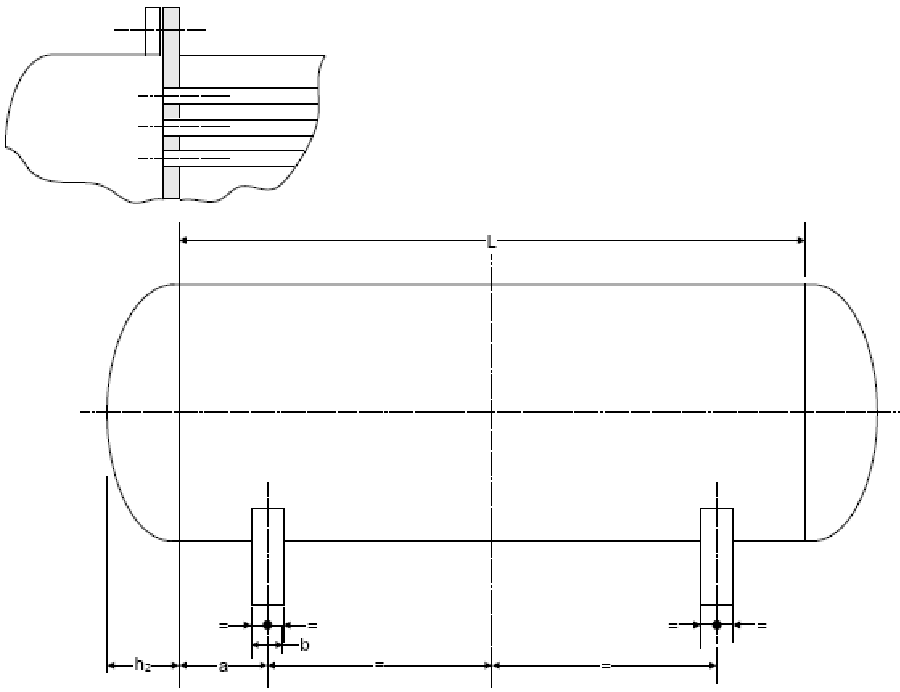


Figure 4.15.1 – Horizontal Vessel on Saddle Supports

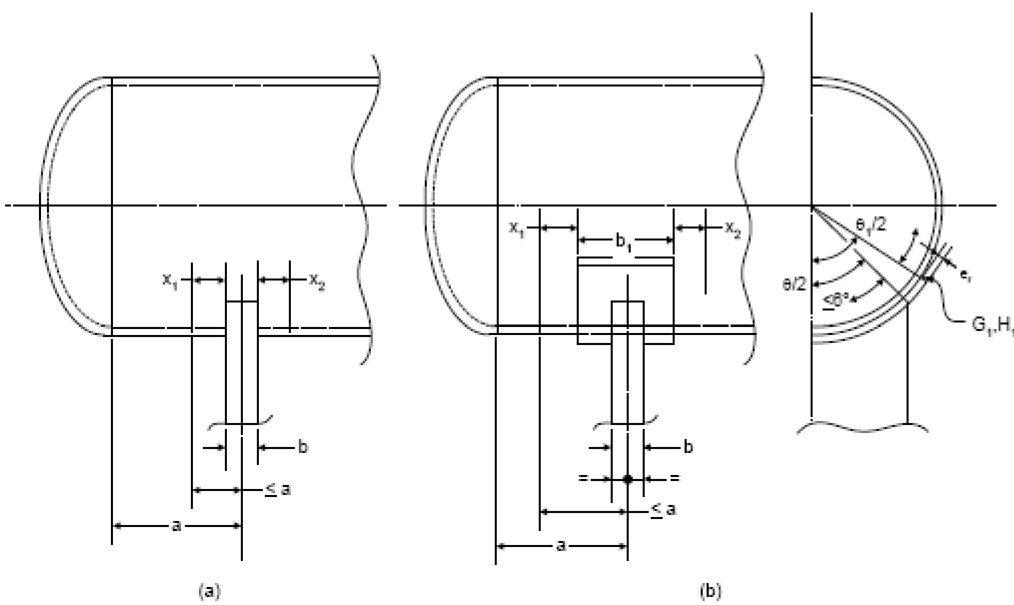


Figure 4.15.2 – Cylindrical Shell Without Stiffening Rings

$L := L_{ff}$ $\quad M_{tot} := 12000kg$ $\quad L = 1.6\,m$

$b := 1.5cm$ $\quad a_{min} := .18L_{ff}$ $\quad a_{min} = 28.8\,cm$ $\quad a := 29cm$ $\quad \theta := 120deg$ $\quad R_m := R_{i\_pv} + 0.5t_{pv}$

$b_1 := \min\left[\left(b + 1.56\cdot\sqrt{R_m\cdot t_{pv}}\right), 2\cdot a\right]$ $\quad b_1 = 14.411\,cm$ $\quad h_2 := 20cm$ $\quad k := 0.1$

$\theta_1 := \theta + \dfrac{\theta}{12}$ $\quad \theta_1 = 130\,deg$

maximum reaction load at each support:

$Q := 0.5M_{tot}\cdot g$ $\quad Q = 5.884\times 10^4\,N$

$$M_1 := -Q\cdot a\cdot\left(1 - \frac{1 - \frac{a}{L} + \frac{R_m^2 - h_2^2}{2\cdot a\cdot L}}{1 + \frac{4h_2}{3L}}\right) \qquad M_1 = 1.676\times 10^3\,N\cdot m \qquad Q\cdot a = 1.706\times 10^4\,J$$

$$M_2 := \frac{Q\cdot L}{4}\cdot\left[\frac{1 + \frac{2\cdot\left(R_m^2 - h_2^2\right)}{L^2}}{1 + \frac{4\cdot h_2}{3L}} - \frac{4a}{L}\right] \qquad M_2 = 9.875\times 10^3\,N\cdot m$$

$M_{1'} := Q\cdot a$ $\quad M_{1'} = 1.706\times 10^4\,N\cdot m$

$M_{2'} := M_{1'}$ $\quad M_{2'} = 1.706\times 10^4\,N\cdot m$

$$T := \frac{Q\cdot(L - 2a)}{L + \frac{4h_2}{3}} \qquad T = 3.215\times 10^4\,N$$

4.15.3.3 - longitudinal stresses

distributed load (ASME assumption) | end load (actual)

$$\sigma_1 := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_2}{\pi R_m^2 t_{pv}} \qquad \sigma_1 = 52.789\,MPa \qquad \sigma_{1'} := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_{2'}}{\pi R_m^2 t_{pv}} \qquad \sigma_{1'} = 52.301\,MPa$$

$$\sigma_2 := \frac{P\cdot R_m}{2t_{pv}} + \frac{M_2}{\pi R_m^2 t_{pv}} \qquad \sigma_2 = 54.128\,MPa \qquad \sigma_{2'} := \frac{P\cdot R_m}{2t_{pv}} + \frac{M_{2'}}{\pi R_m^2 t_{pv}} \qquad \sigma_{2'} = 54.616\,MPa$$

same stress at supports, since these are stiffened, as a<0.5Rm and close to a torispheric head $\quad a < 0.5R_m = 1$

$$\sigma_3 := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_1}{\pi R_m^2 t_{pv}} \qquad \sigma_3 = 53.345\,MPa \qquad \sigma_{3'} := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_{1'}}{\pi R_m^2 t_{pv}} \qquad \sigma_{3'} = 52.301\,MPa$$

$$\sigma_4 := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_1}{\pi R_m^{\,2} t_{pv}} \qquad \sigma_4 = 53.572\,\text{MPa} \qquad\qquad \sigma_{4'} := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_{1'}}{\pi R_m^{\,2} t_{pv}} \qquad \sigma_{4'} = 54.616\,\text{MPa}$$

4.15.3.4 - Shear stresses

$$\Delta := \frac{\pi}{6} + \frac{5\theta}{12} \qquad \Delta = 1.396$$

$$\alpha := 0.95\left(\pi - \frac{\theta}{2}\right) \qquad \alpha = 1.99$$

$$K_2 := \frac{\sin(\alpha)}{\pi - \alpha + \sin(\alpha)\cos(\alpha)} \qquad K_2 = 1.171$$

here we use c), formula for cyl. shell with no stiffening rings and which is not stiffened by a formed head, flat cover or tubesheet. This is worst case, as we havea flange, which can be considered as one half of a stiffening ring pair for each support.

c) $$\tau_1 := \frac{K_2 \cdot T}{\pi R_m \cdot t_{pv}} \qquad \tau_1 = 1.749\,\text{MPa} \qquad (4.15.14)$$

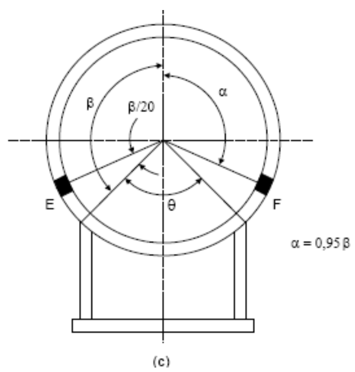
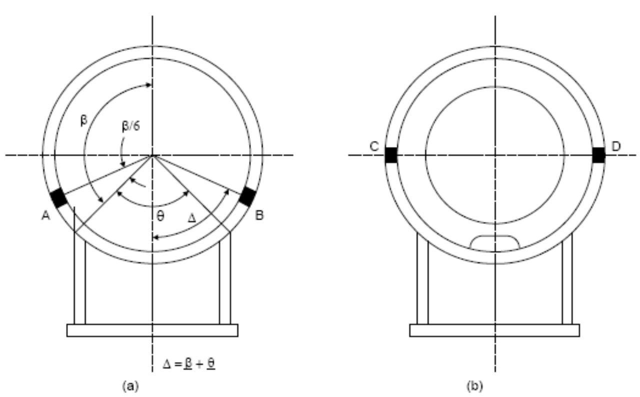


Figure 4.15.5 – Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder



4.15.3.5 Circumferential Stress

$$K_5 := \frac{1 + \cos(\alpha)}{\pi - \alpha + \sin(\alpha) \cdot \cos(\alpha)} \qquad K_5 = 0.76$$

$$\beta := \pi - \frac{\theta}{2} \qquad \beta = 2.094$$

$$K_6 := \frac{\frac{3\cdot\cos(\beta)}{4}\cdot\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{5\cdot\sin(\beta)\cdot\cos(\beta)}{4\cdot\beta} + \frac{\cos(\beta)^3}{2} - \frac{\sin(\beta)}{4\cdot\beta} + \frac{\cos(\beta)}{4} - \beta\cdot\sin(\beta)\cdot\left[\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin(2\cdot\beta)}{4\cdot\beta}\right]}{2\cdot\pi\cdot\left[\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin(2\cdot\beta)}{4\cdot\beta}\right]}$$

$$K_6 = -0.221$$

$$\frac{a}{R_m} < 0.5 = 1$$

$$K_7 := \frac{K_6}{4} \qquad K_7 = -0.055$$

a) Max circ bending moment

1) Cyl shell without a stiffening ring

$$M_\beta := K_7\cdot Q\cdot R_m \qquad M_\beta = -2.223\times 10^3\ N\cdot m$$

c) Circ. stress in shell, without stiffening rings

$$x_1 := 0.78\sqrt{R_m\cdot t_{pv}} \qquad x_1 = 6.456\,cm \qquad x_2 := x_1 \qquad k = 0.1$$

$$\sigma_6 := \frac{-K_5\cdot Q\cdot k}{t_{pv}\cdot(b + x_1 + x_2)} \qquad \sigma_6 = -3.104\,MPa$$

$$L < 8R_m = 1 \qquad L = 1.6\,m \qquad b_1 = 14.411\,cm$$

$$\sigma_7 := \frac{-Q}{4t_{pv}\cdot(b + x_1 + x_2)} - \frac{12K_7\cdot Q\cdot R_m}{L\cdot t_{pv}^2} \qquad \sigma_7 = 156.484\,MPa \qquad (4.15.25)$$

too high; we need a reinforcement plate of thickness;

$t_r := t_{pv}$ strength ratio: $\eta := 1$ (4.15.29)

$$\sigma_{7r} := \frac{-Q}{4(t_{pv} + \eta\cdot t_r)\cdot b_1} - \frac{12K_7\cdot Q\cdot R_m}{L\cdot(t_{pv} + \eta\cdot t_r)^2} \qquad \sigma_{7r} = 36.569\,MPa \qquad (4.15.28)$$

3) f) Acceptance Criteria

$$S = 1.379\times 10^8\,Pa \qquad S = 2\times 10^4\,psi$$

$$|\sigma_{7r}| < 1.25S = 1$$

4) this section not applicable as $t_r > 2t_{pv} = 0$

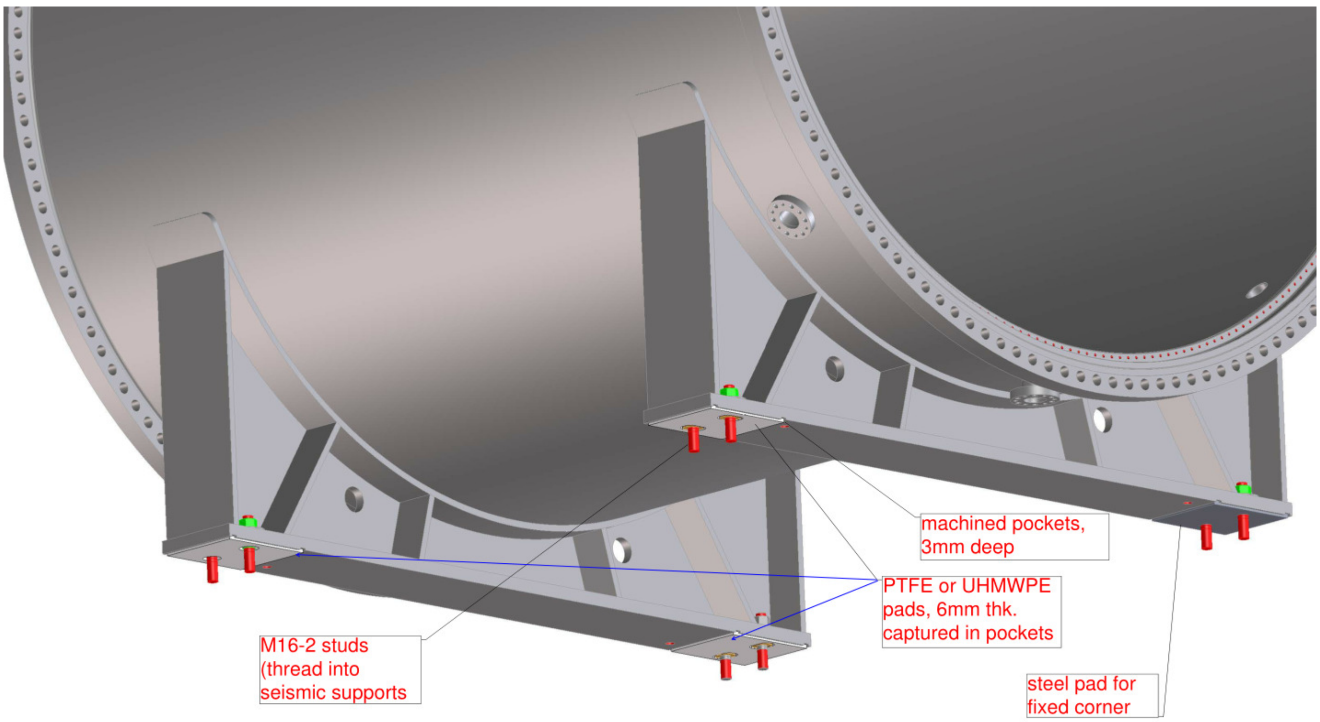
4.15.3.6 - Saddle support, horizontal force given below must be resisted by low point of saddle (where height = h_s)

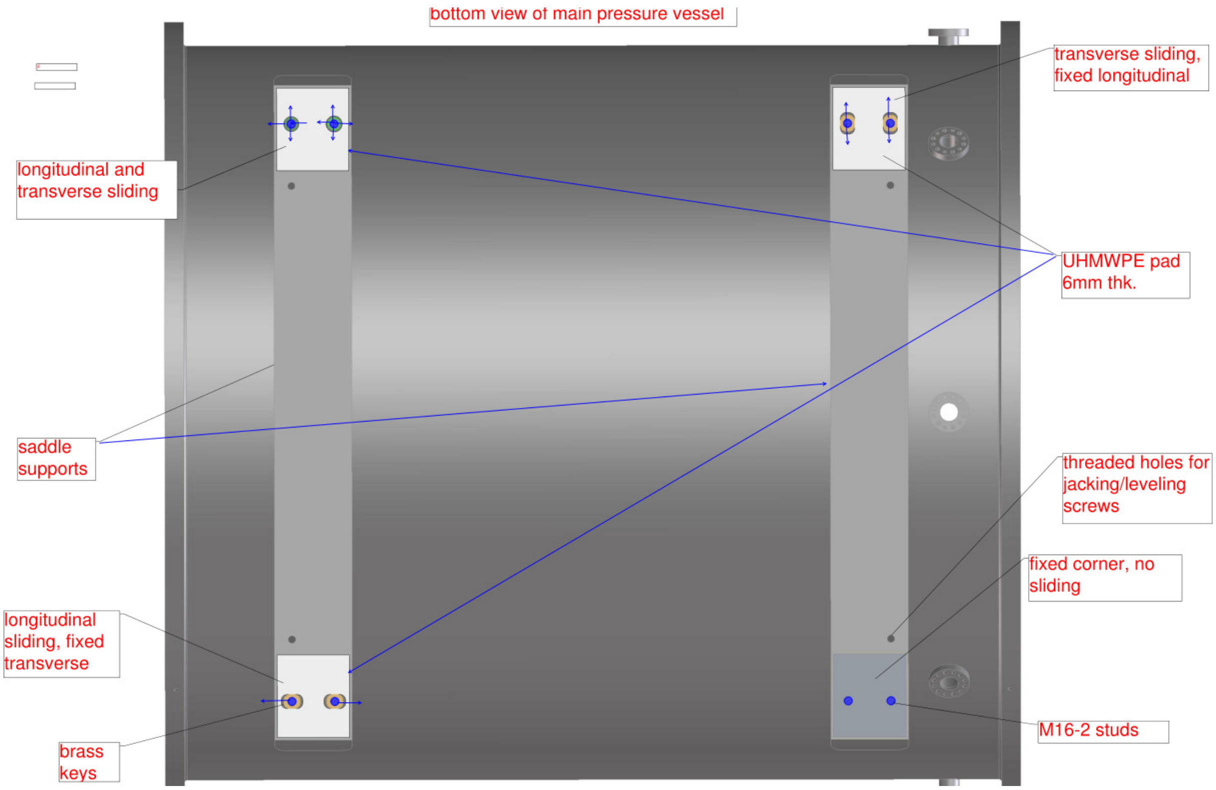
$$F_h := Q \cdot \left(\frac{1 + \cos(\beta) - 0.5 \cdot \sin(\beta)^2}{\pi - \beta + \beta \cdot \sin(\beta) \cos(\beta)} \right) \qquad F_h = 5.242 \times 10^4 \, N \qquad h_s := 9cm$$

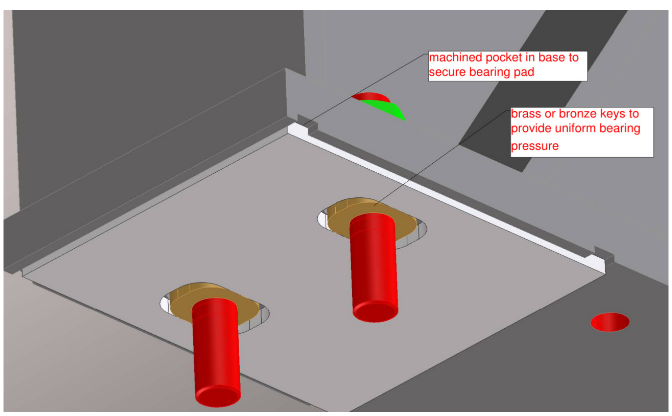
$$\sigma_h := \frac{F_h}{b \cdot h_s} \qquad \sigma_h = 38.833 \, MPa$$

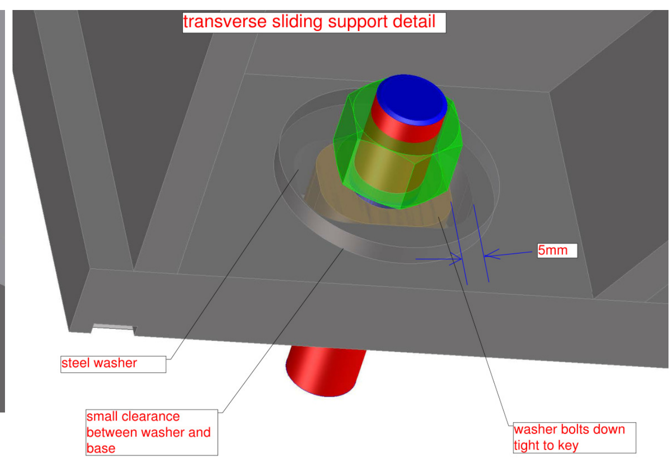
Support on, and Attachment to floor

The vessel has four points of connection ("corners") to the seismic platform, two on each saddle support; each connection point consisting of two M16mm studs. Other possibilities exist. Pressurization or thermal excursion (bakeout, cryogen spill from ArDM) will result in dimensional changes of the vessel, so it is required to use low friction pads under the supports to constrain the vessel in a 2D kinematic fashion. One corner is fixed, two others are slotted to allow sliding in one direction (orthogonal to each other), and a full clearance hole pattern at the fourth corner allows sliding in both directions.









Vessel length and width change under pressurization and heating:

length between saddle supports:

$L_s := L_{ff} - 2a \quad L_s = 1.02\,m$ $\qquad R_{i\_pv} = 0.68\,m \qquad t_{pv} = 10\,mm$

Pressure load, longitudinal

$H_D = 2.266 \times 10^6\,N$

saddle support width, transverse

$w_s := 1.2m$

stresses in vessel shell, longitudinal and tangential (hoop):

$$\sigma_{long} := \frac{H_D}{2\pi R_{i\_pv} \cdot t_{pv}} \qquad \sigma_{long} = 53.041\,MPa \qquad \sigma_{hoop} := \frac{P \cdot R_{i\_pv}}{t_{pv}} \qquad \sigma_{hoop} = 106.137\,MPa$$

length width changes from pressure:

$E_{SS\_aus} = 193\,\text{GPa}$

$\delta L_s := \dfrac{\sigma_{long} \cdot L_s}{E_{SS\_aus}}$ $\delta L_s = 0.28\,\text{mm}$

$\delta w_s := \dfrac{\sigma_{hoop} \cdot w_s}{E_{SS\_aus}}$ $\delta w_s = 0.66\,\text{mm}$ in reality, the support itself will restrain a significant portion of this deflection, since the saddle is welded to the vessel shell

thermal growth, 150C bakeout

$\alpha_{SS} := 16 \cdot 10^{-6} K^{-1}$ up to 100C

$\Delta T_v := 150K - 20K$

$\varepsilon_{th\_SS} := \alpha_{SS} \cdot \Delta T_v$

$\delta_{v\_t} := \varepsilon_{th\_SS} \cdot w_s$ $\delta_{v\_t} = 2.496\,\text{mm}$

$\delta_{v\_l} := \varepsilon_{th\_SS} \cdot L_s$ $\delta_{v\_l} = 2.122\,\text{mm}$

bakeout will only be performed under vacuum condition.

These deflections (from either pressure or thermal excursion) are substantial enough to warrant the use of low friction pads under three of the four supports, which will allow the vessel to slide both lengthwise and widthwise when pressurizing/depressurizing or baking. In addition there is a remote possibility of cryogen spillage, perhaps from ArDM which may chill the vessel, so a capacity for contraction equal to the above expansion should be designed in. Bolt holes should be slotted, with sliding keys to give uniform bearing pressure on slots under transverse loads, as described above. In addition, each corner should have one large tapped hole for a leveling/jacking screw that will allow bearing pad replacement, in situ.

Bolt shear stress from seismic acceleration

The maximum horizontal acceleration from a seismic event is expected to be much less than 1 m/s^2; we use a design value here of:

$a_{horiz} := 2\,\dfrac{m}{s^2}$

$F_{horiz} := M_{tot} \cdot a_{horiz}$ $F_{horiz} = 2.4 \times 10^4\,\text{N}$

Bolt area required:

We calculate for all horizontal load taken on two corners only, since we will have sliding supports. We calculate for austenitic stainless steel bolts:

$S_{sup\_bolt} := S_f$ $S_{sup\_bolt} = 137.895\,\text{MPa}$

maximum shear stress:

$S_{s\_sup\_bolt} := 0.5 S_{sup\_bolt}$

bolt area required, per corner

$A_{sup\_bolts} := \dfrac{0.5 F_{horiz}}{S_{s\_sup\_bolt}}$ $A_{sup\_bolts} = 1.74\,\text{cm}^2$

assume 2 bolts per corner, for redundancy and symmetry about support web. with 2 bolts, the only critical dimension to match between the holes in the support and the holes in the seismic frame are the distance between the hole pairs (hole pattern rotation need not be matched). The sliding keys can be custom machined if needed to compensate for mismatch.

$$d_{sup\_bolt} := \sqrt{\frac{4}{\pi} \cdot 0.5 A_{sup\_bolts}} \qquad d_{sup\_bolt} = 10.526\,mm$$ this is required minimum root diameter

Support uses (2) M16-2.0 bolts on each corner, root diameter is 12mm

Bearing design

Assume a full square contact patch under each corner;accounting for bolts and keys:

$$A_{bearing} := b_1^{\;2} - 4A_{sup\_bolts} \qquad A_{bearing} = 200.724\,cm^2$$

Bearing pressure is then (assuming a non-leveled condition where full weight is supported on two diagonal corners):

$$P_{bearing} := 0.5\frac{M_{tot} \cdot g}{A_{bearing}} \qquad P_{bearing} = 425.162\,psi$$

Maximum allowable bearing pressures and temperatures (we may bake vessel at 150C with copper shielding inside)

frrom Slideways bearing catalogue (similar to table 10-4 in J. Shigley, Mech. Engin. 3rd ed.)

Physical Properties of Various Materials

Physical Properties	UHMW	OF/UHMW	Wood	MD-Nylon	Nylon	PTFE	Acetal
PV Capacity (psi-fpm)	2,000	6,000	15,000	3,500	2,700	1,000	3,000
Max Pressure (psi)	1,200	600	1,000	2,000	2,000	500	1,000
Max Velocity (fpm)	100	500	500	150	100	400	100
Max Continuous Temp (˚F)	180	160	160	220	180	500	200
Dynamic Coefficient of Friction vs. Steel (dry)	.15-.20	.13-.16	0.09	.15-.35	.16-.43	.04-.10	.15-.35

Material for Bearing Pad

We choose only unfilled plastics, as most fillers are not radiopure (possible exception: bronze filled PTFE). PTFE (unfilled), @500 psi , has little margin for stability, but any creep flow will act to equalize pressure over all 4 supports, resulting in a lower, stable pressure. Furthermore it is the only material that can withstand 150C, although the temperature at the supports will be substantially less than 150C, due to the poor thermal conductivity of SS. Cooling of supports should be performed in case of bakeout, regardless. Bronze-filled PTFE, UHMWPE (non-oil-filled), nylon, or acetal may also be used; cooling of support pads during bakeout would be mandatory.

Jacking screw diameter

Each jacking screw must be able to lift half the entire weight of the detector. We look for a low grade bolt that can support this force

$$F_{js} := 0.5M_{tot} \cdot g \qquad F_{js} = 5.884 \times 10^4\,N \qquad F_{js} = 1.323 \times 10^4\,lbf$$

Use 90% yield strength as allowable stress (non critical)

$$S_{y\_316Ti} := 30000psi$$

$$A_{js} := \frac{F_{js}}{0.9 \cdot S_{y\_316Ti}} \qquad A_{js} = 3.161\,cm^2$$

$$d_{js\_root} := \sqrt{\frac{4}{\pi} A_{js}} \qquad d_{js\_root} = 20.061\,mm$$

Use an M24-2 bolt at each corner. Lubricate or PTFE coat (preferred)

Saddle support bending stress

Cross section of saddle support is an I-beam, with a central "web" connecting two "flanges" We check bending stress in support at bottom of vessel, where cross section height is a minimum.

Vessel axis height (axis above floor) $R_{o\_pv} := R_{i\_pv} + t_{pv}$ $R_{o\_pv} = 69\,\text{cm}$

$h_v := 80\text{cm}$

flange and web thicknesses, widths:

$t_{fll} := 2\text{cm}$ $w_{fll} := b_1$ $w_{fll} = 14.411\,\text{cm}$ $t_w := 1.5\text{cm}$ $t_r = 1\,\text{cm}$

I-beam web height, not including flanges:

$h_w := h_v - \left(R_{o\_pv} + t_r + t_{fll}\right)$ $h_w = 8\,\text{cm}$

I-beam Area Moment of Inertia:

Parallel axis theorem

sum moments of flanges and web about axis thru top surface , then divide by total area to find neutral axis

$A_r := b_1 \cdot t_r$ $A_w := t_w \cdot h_w$ $A_{fll} := w_{fll} \cdot t_{fll}$

$$c_1 := \frac{A_r \cdot (0.5 \cdot t_r) + A_w \cdot (t_r + 0.5 \cdot h_w) + A_{fll} \cdot (t_r + h_w + 0.5 \cdot t_{fll})}{A_r + A_w + A_{fll}}$$

$c_1 = 6.435\,\text{cm}$ down from top surface

$I_r := \dfrac{b_1 \cdot t_r^3}{12}$ $I_w := \dfrac{t_w \cdot h_w^3}{12}$ $I_{fll} := \dfrac{w_{fll} \cdot t_{fll}^3}{12}$

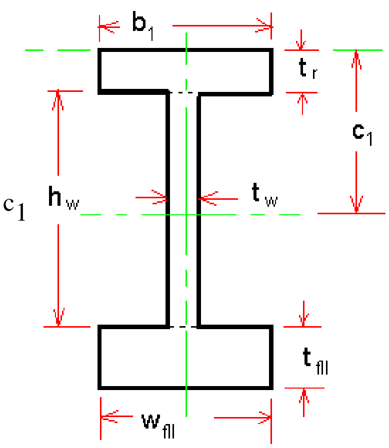
$I_r = 1.201\,\text{cm}^4$ $I_w = 64\,\text{cm}^4$ $I_{fll} = 9.608\,\text{cm}^4$

$d_r := 0.5 t_r - c_1$ $d_w := (t_r + 0.5 \cdot h_w) - c_1$ $d_{fll} := (t_r + h_w + 0.5 \cdot t_{fll}) - c_1$

$d_r = -5.935\,\text{cm}$ $d_w = -1.435\,\text{cm}$ $d_{fll} = 3.565\,\text{cm}$

$$I_{s\_min} := \left(I_r + A_r \cdot d_r^2\right) + \left(I_w + A_w \cdot d_w^2\right) + \left(I_{fll} + A_{fll} \cdot d_{fll}^2\right)$$

$I_{s\_min} = 973.459\,\text{cm}^4$



Consider as a uniformly loaded beam, simply supported on each end

load per unit width (along the long dimension; transverse to vessel axis)

$\omega := \dfrac{0.55 M_{tot} \cdot g}{w_s}$ $\omega = 539.366 \dfrac{\text{N}}{\text{cm}}$ $M_{tot} = 1.2 \times 10^4\,\text{kg}$

Moment at center:

$M_{sup\_max} := \dfrac{\omega \cdot w_s^2}{8}$ $M_{sup\_max} = 9.709 \times 10^3\,\text{N} \cdot \text{m}$

Maximum stress, tensile in flange under vessel

$\sigma_{sup\_max} := \dfrac{M_{sup\_max} \cdot 0.5 h_w}{I_{s\_min}}$ $\sigma_{sup\_max} = 39.893\,\text{MPa}$

This is low enough to allow support only at corners; we do not need to support under the full width of the support feet.

ANGEL Torispheric Head Design, using (2010 ASME PV Code Section VIII, div. 1, UG-32 Formed heads and sections, Pressure on Concave Side, Appendix 1-4 rules eq 3

$P = 1.561 \times 10^{6}\,Pa \quad E = 1 \quad S = 2 \times 10^{4}\,psi$

I.D.

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

O.D.

$D_o := D_i + 2t \qquad D_o = 1.443\,m$

Crown radius:

$L_{cr} := 1D_i \quad L_{cr} = 1.36\,m$

Knuckle radius:

$r_{kn} := 0.1D_i \qquad r_{kn} = 0.136\,m$

$E = 1 \quad S_{div1} := 20000psi \qquad S_{div1} = 1.379 \times 10^{8}\,Pa \qquad S_{y\_316Ti} = 206.843\,MPa$

Appendix 1-4 mandatory Supplemental Design Formulas

UG-32 does not give equations for a range of crown and knuckle radii; these are found in **App 1-4**

$$\frac{L_{cr}}{r_{kn}} = 10$$

$$M := \frac{1}{4}\left(3 + \sqrt{\frac{L_{cr}}{r_{kn}}}\right) \qquad M = 1.541$$

Minimum shell thickness:

$$t_{min} := \frac{P \cdot L_{cr} \cdot M}{2S \cdot E - 0.2P} \qquad t_{min} = 11.871\,mm \qquad (3)$$

note: we will need full weld efficiency for the above thickness to be permissible, as per UG-32(b)

this formula is only valid if the following equation is true (1-4(a))

$$\frac{t_{min}}{L_{cr}} \geq 0.002 = 1 \qquad \frac{t_{min}}{L_{cr}} = 8.729 \times 10^{-3}$$

Set head thickness:

$t_h := 12mm$

Note: under EN_13455-3 rules for 316Ti, a thinner thickness of 10.25 mm is possible, due to a higher maximum allowable strength at the knuckle. Below is an analysis from Sara Carcel

Torispherical heads, VIII, Div 2			DIN 28011			EN 13445-3 (316Ti, para D	
	KORBBOGEN		r=0,1L			f	166.666667
D	1360		1360	Diámetro interior		X	0.1
t	10		10			t	10
De	1380		1380			Y	0.00735294
L	1104		1360	Diámetro interior corona		Z	2.13353891
ri	212.52		136			N	0.84954918
L/D	0.81176471	Ok	1	Entre 0,7 y 1, ver 4-49		β0,1	0.86799204
ri/D	0.15626471	Ok	0.1	Mayor de 0,06		β0,2	0.51421113
Li/t	110.4	Ok	136	Entre 20 y 2000		β	0.86799204
β	1.01880199		1.11024234			P	1.52
φ	0.49440713		0.85749293			**eb**	8.66383993
R	752.567792		697.850818	Si φ<β		**ey**	10.2275849
C1	0.71313518	r/D>0,08	0.6742			**es**	6.21577196
C2	1.05371176		1.2			**Thickness**	**10.2275849**
Peth	64.2498476	E=117000	44.2387943				
C3	206	Sy=206MPa	206				
Py	3.37117586		1.57121205				
G	19.0585868		28.1558395				
Pck	6.73919067		3.14731728	G>1			

Nozzle wall thickness required

Internal radius of finished opening

$R_n := 4.4cm$

Thickness required for internal pressure:

$t_{rn} := \dfrac{P \cdot R_n}{S \cdot E - 0.6 \cdot P}$ $\quad t_{rn} = 0.501\,mm$

We set nozzle thickness

$t_n := 7mm$ we are limited by need to maintain CF bolt pattern which has typically a 4.0 inch OD pipe with room for outside fillet weld

$D_{on} := 2(R_n + t_n)$ $\quad D_{on} = 4.016\,in$

Thickness required for external load

Nozzles on head may be subject to several possible non-pressure loads, simultaneously:

1. Reaction force from pressure relief, (fire) or fast depressure (auxiliary nozzle only)
2. Weight of attached components, including valves, expansion joints, copper or lead shielding plugs, high voltage feedthrough.

The nozzles may all have nozzle extensions rigidly attached which create to possibility of high moments being applied to the nozzles, not just shear loads. We consider the direction and location of center of gravity for these loads:

$L_{ne} := 58cm$ $\quad \rho_{Pb} := 11.3 \dfrac{gm}{cm^3}$

Forces and centers of gravity (l):

$F_{shp} := \pi R_n^{\,2} \cdot L_{ne} \cdot \rho_{Pb} \cdot g$ $\quad F_{shp} = 391\,N$ $\quad l_{shp} := 0.5 L_{ne}$

$W_{ne} := 2 \cdot (2\pi R_n \cdot L_{ne} \cdot t_n \cdot \rho_{SS}) \cdot g$ $\quad W_{ne} = 176\,N$ $\quad l_{Wne} := l_{shp}$

(factor of 2 to account for flange weights)

Fast vent reaction force, as calculated below

$F_{fv} := 3700N$ worst case is venting upward, at right angles to nozzle axis (we plan to use a straight through valve, regardless, for which reaction force will not produce a bending moment and will simply reduce longitudinal stress from pressure)

Moments:

$M_{shp} := F_{shp} \cdot l_{shp}$ $\quad M_{shp} = 113\,N \cdot m$

$M_{Wne} := W_{ne} \cdot l_{Wne}$ $\quad M_{Wne} = 51.1\,N \cdot m$

$M_{fv} := F_{fv} \cdot L_{ne}$ $\quad M_{fv} = 2146\,N \cdot m$

Total moment:

$M_n := M_{fv} + M_{shp} + M_{Wne}$ $\quad M_n = 2310\,N \cdot m$

Moment of Inertia, bending

$I_n := \pi \cdot (R_n + 0.5 t_n)^3 \cdot t_n$ $\quad I_n = 235.7\,cm^4$

Stress, bending (longitudinal)

$\sigma_{n\_l} := \dfrac{M_n \cdot (R_n + t_n)}{I_n}$ $\quad \sigma_{n\_l} = 50\,MPa$

Stress, circumferential (hoop)

$$\sigma_{n\_c} := \frac{P \cdot R_n}{t_n} \qquad \sigma_{n\_c} = 9.811\,\text{MPa}$$

Criterion for acceptable stress - use maximum shear stress theory:

Maximum shear stress (min. stress is in third direction, = zero on outside of nozzle):

$$\tau_n := \sqrt{\left(\frac{\sigma_{n\_1} - 0\text{MPa}}{2}\right)^2} \qquad \tau_n = 25\,\text{MPa}$$ OK (J. Shigley, <u>Mech.Eng.</u> 3rd ed., eq. (2-9)

Compare with maximum shear stress from minimum thickness nozzle (pressure only, no applied moments)

$$\sigma_{rn} := \frac{P \cdot R_n}{t_{rn}} \qquad \sigma_{rn} = 137\,\text{MPa}$$

$$\tau_{rn} := \sqrt{\left(\frac{\sigma_{rn} - 0\text{MPa}}{2}\right)^2} \qquad \tau_{rn} = 68.5\,\text{MPa}$$

Additional Factor of Safety, over ASME factor of safety:

$$FS_n := \frac{\tau_{rn}}{\tau_n} \qquad FS_n = 2.7$$ OK

External pressure:

Nozzles on head are very short; no analysis needed. Nozzle extensions are longer:

$$L_{ne} = 58\,\text{cm} \qquad t_{ne} := 7\text{mm}$$

$$\frac{L_{ne}}{2R_n} = 6.591 \qquad 2\frac{R_n}{t_{ne}} = 12.571$$

From charts HA-1 and HA-2 above:

$$A_{ne} := .02 \qquad B_{ne} := 13000\text{psi}$$

$$P_{a\_ne} := \frac{4B_{ne}}{3\left(\frac{2R_n}{t_{ne}}\right)} \qquad P_{a\_ne} = 93.795\,\text{bar}$$ OK

UG-37 Reinforcement Required for Openings in Shells and heads

Reinforcement is not required for the DN40 and DN75 flanged nozzles welded to the main cylindrical vessel as per **UG-36** below:

UG-36 (c) (3) Strength and Design of finished Openings:

(3) Openings in vessels not subject to rapid fluctuations in pressure do not require reinforcement other than that inherent in the construction under the following conditions:

<--no rapid fluctuations, condition met

(a) welded, brazed, and flued connections meeting the applicable rules and with a finished opening not larger than:

$3\frac{1}{2}$ in. (89 mm) diameter — in vessel shells or heads with a required minimum thickness of $\frac{3}{8}$ in. (10 mm) or less;

<-- applicable to cyl. vessel, condition met for DN40; DN75 nozzles

$2\frac{3}{8}$ in. (60 mm) diameter — in vessel shells or heads over a required minimum thickness of $\frac{3}{8}$ in. (10 mm);

<-- not applicable to cyl. vessel, but is applicable to heads; condition not met for DN100 nozzles, reinforcement needed

(b) threaded, studded, or expanded connections in which the hole cut in the shell or head is not greater than $2\frac{3}{8}$ in. (60 mm) diameter;

<--not applicable

(c) no two isolated unreinforced openings, in accordance with (a) or (b) above, shall have their centers closer to each other than the sum of their diameters;

<-- condition met

(d) no two unreinforced openings, in a cluster of three or more unreinforced openings in accordance with (a) or (b) above, shall have their centers closer to each other than the following: for cylindrical or conical shells,

<-- condition met

$$(1 + 1.5 \cos \theta)(d_1 + d_2);$$

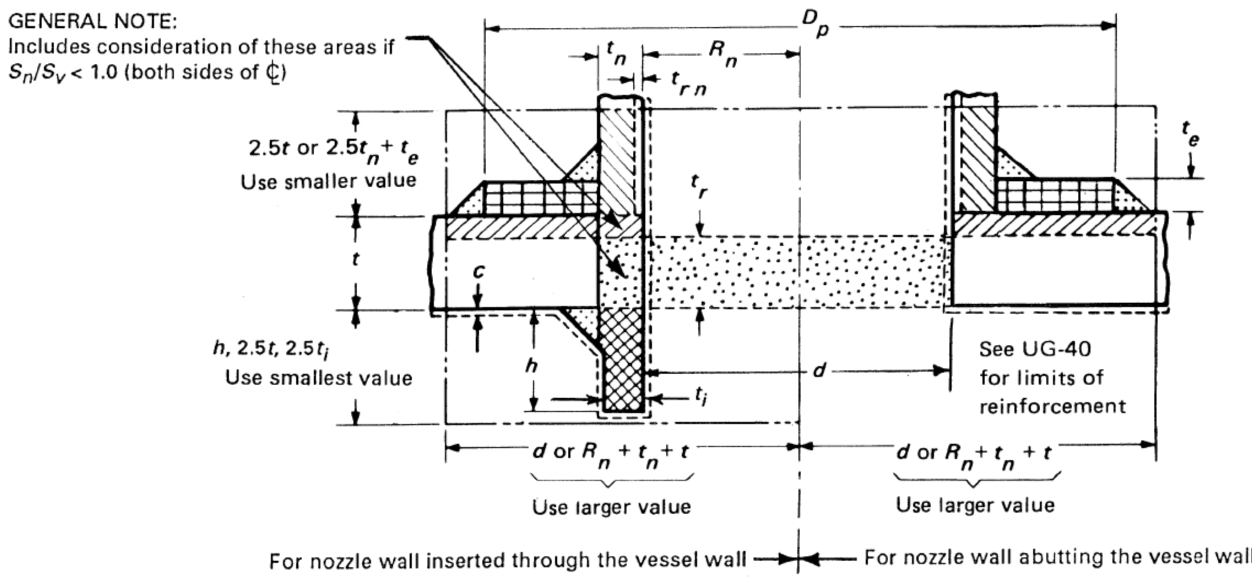
for doubly curved shells and formed or flat heads,

$$2.5(d_1 + d_2)$$

In addition, there are no significant external loads on the radial nozzles of the vessel, only the weight of an HV feedthrough at 45 deg angle; this is insignicant compared to the maximum loads and moments possible on the head nozzles (which are similar in size and thickness). Proceeeding with the head nozzle reinforcement:

Reinforcement for the DN100 flanged nozzles welded to the torispheric heads is required and calculated according to **UG-37** :

FIG. UG-37.1 NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS



nozzle and reinforcement are also 316Ti

$t_{rn} = 0.501\,\mathrm{mm}$ $\quad d := 2R_n \quad d = 8.8\,\mathrm{cm} \quad F := 1.0 \quad f_r := 1.0 \quad f_{r1} := 1.0 \quad E_1 := 1 \quad f_{r2} := 1$

$t_n := t_h \quad t_r = 12\,\mathrm{mm} \quad t_i := 0\mathrm{mm} \quad h := 0\mathrm{mm} \quad leg_i := 0\mathrm{mm} \quad t := 12\mathrm{mm} \quad leg_n := 1.4t_n \quad leg_n = 0.98\,\mathrm{cm}$

We will need a reinforcing pad, as the head is already minimum thickness and the nozzle is much thinner. Note: this is for ASME; sec VIII, div. 1. European Codes allow thinner thickness for 316Ti.

$t_e := 12\mathrm{mm} \quad D_p := 1.8d \quad D_p = 0.158\,\mathrm{m}$ (from UG-40 Limits of Reinforcement) $f_{r4} := 1 \quad leg_e := .71\cdot t_e$

Area or reinforcement required:

$A_{req} := d\cdot t_r\cdot F + 2t_n\cdot t_r\cdot F\cdot(1 - f_{r1}) \qquad A_{req} = 1.056\times 10^3\,\mathrm{mm}^2$

Area available in shell:

$$A_{1a} := d\cdot(E_1\cdot t - F\cdot t_r) - 2\cdot t_n\cdot(E_1\cdot t - F\cdot t_r)\cdot(1 - f_{r1})$$
$$A_{1b} := 2\cdot(t + t_n)\cdot(E_1\cdot t - F\cdot t_r) - 2\cdot t_n\cdot(E_1\cdot t - F\cdot t_r)\cdot(1 - f_{r1})$$

$A_1 := \max(A_{1a}, A_{1b}) \qquad A_1 = 0\,\mathrm{mm}^2$

Area available in nozzle projecting outwards

$$A_{2a} := 5(t_n - t_{rn})\cdot f_{r2}\cdot t \qquad A_{2a} = 389.914\,\mathrm{mm}^2$$

$$A_{2b} := 2\cdot(t_n - t_{rn})\cdot(2.5t_n + t_e)\cdot f_{r2} \qquad A_{2b} = 383.415\,\mathrm{mm}^2$$

$A_2 := \min(A_{2a}, A_{2b}) \qquad A_2 = 383.415\,\mathrm{mm}^2$

Area available in nozzle projecting inwards

$A_{3a} := 5t \cdot t_i \cdot f_{r2}$ $\quad A_{3a} = 0\,mm^2$

$A_{3b} := 5t_i \cdot t_i \cdot f_{r2}$ $\quad A_{3b} = 0\,mm^2$

$A_{3c} := 2 \cdot h \cdot t_i \cdot f_{r2}$ $\quad A_{3c} = 0\,mm^2$

$A_3 := \min(A_{3a}, A_{3b}, A_{3c})$ $\quad A_3 = 0\,mm^2$

Area available in weld, outward

$A_{41} := {leg_n}^2 \cdot f_{r2}$ $\quad A_{41} = 96.04\,mm^2$

Area available in outer element weld

$A_{42} := {leg_e}^2 \cdot f_{r4}$ $\quad A_{42} = 72.59\,mm^2$

Area available in weld, inward

$A_{43} := {leg_i}^2 \cdot f_{r2}$ $\quad A_{43} = 0\,mm^2$

Area available in reinforcement

$A_5 := (D_p - d - 2t_n) \cdot t_e \cdot f_{r4}$ $\quad A_5 = 676.8\,mm^2$

Total Area available

$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 = 1229\,mm^2$

Area required:

$A_{req} = 1056\,mm^2$

$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A_{req} = 1$

Torispheric Head, per DIN

A thinner head thickness of 10.58mm is calculated by S. Carcel to DIN formula; this is acceptable. It is not yet clear whether or not reinforcement pads are needed.

Pressure Relief Capacity requirements

There are two possible conditions 1. regulator failure and 2 external fire

$L_{pv} := 2.1m$ length of vessel , inside average $\quad R_{o\_pv} = 0.69\,m$ outer radius

Pressure vessel outer area:

$A_{pv} := 2\pi {R_{o\_pv}}^2 + 2\pi R_{o\_pv} \cdot L_{pv}$ $\quad A_{pv} = 12.096\,m^2$

From Anderson Greenwood Technical Seminar Manual, fire sizing is:

$A_{orif} := \frac{F' \cdot A'}{\sqrt{P_1}} \cdot in^2$ $\quad F' := .045$ $\quad A' := \frac{A_{pv}}{ft^2}$ $\quad P_1 := \frac{MAWP_{pv}}{psi}$ $\quad A' = 130.198$

$P_1 = 226.38$

$A_{orif} = 0.389\,in^2$ $\quad k := 1.667$ $\quad K_D := 1$

$d_{orif} := \frac{4}{\pi} \cdot \sqrt{A_{orif}}$ $\quad d_{orif} = 0.795\,in$

However we will want to use the higher value which gives a fast vent , so as to safe Xe in case of leak

$$A_{vent} := \pi \cdot (30mm)^2 \qquad A_{vent} = 4.383\, in^2$$

Mass flow:

$$P' := \frac{P}{psi} \qquad T := 535$$ R, ambient

$$M_a := M_{a\_Xe} \cdot \frac{mol}{gm}$$

$$C_g := 520 \cdot \sqrt{k \cdot \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \qquad C_g = 377.641$$

$$A_o := \frac{A_{vent}}{in^2} \qquad Z_c := .95$$

$$W := \frac{A_o \cdot C_g \cdot K_D \cdot P' \cdot \sqrt{M_a}}{\sqrt{T \cdot Z_c}} \cdot \frac{lb}{hr} \qquad W = 24.419 \frac{kg}{s} \qquad W = 1.938 \times 10^5 \frac{lb}{hr}$$

Reactive force, from same ref. (pg. 49)

$$W' := W \cdot \frac{hr}{lb} \qquad P_2 := 0$$

$$F_T := \frac{W' \cdot \sqrt{\frac{k \cdot T}{(k+1) \cdot M_a}}}{366} \cdot lbf \qquad F_T = 3.694 \times 10^3\, N$$

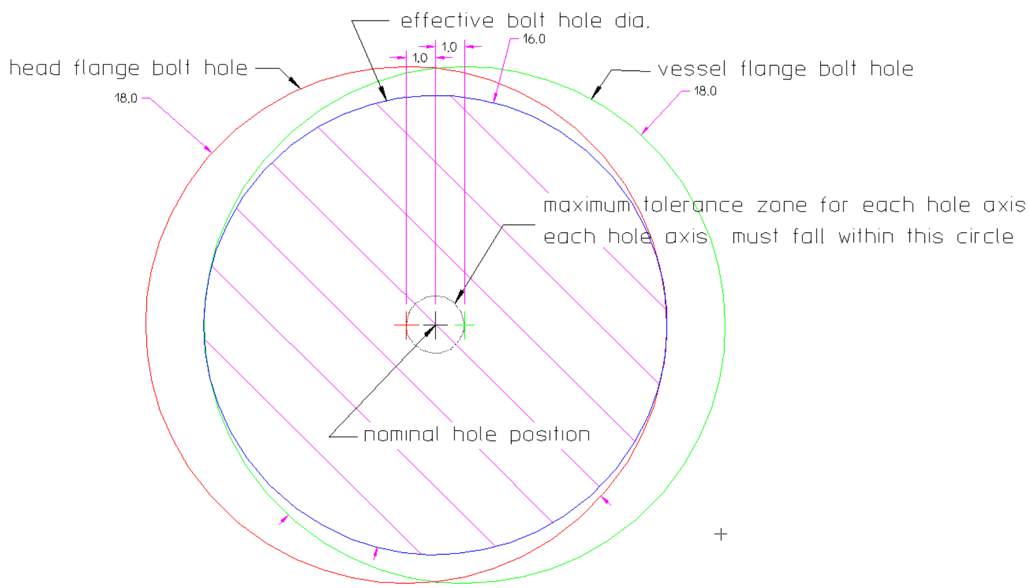
Tolerance analysis

Flange bolt holes:

First consideration is to realize that flanges absolutely must mate and bolt up without interference. This means that tolerances must not be considered to add up in any statistical manner, all features must be considered as being both at their limits of positional tolerance (oppositional) and in their maximum material condition (MMC). That is, all holes and female features are as small as the tolerances allow, and all bolts and male features are as large as the tolerances allow. Materials are all similar, so temperature ranges need not be considered, but part deformations under load must be factored in.

Heads will be assembled to the vessel by first mounting them to an adjustable cradle support which allows precise motions in all 6 degrees of freedom (translations in x,y, and z, plus pitch, yaw and roll about the center axes). This lift fixture is mounted on roller slides that move along the central vessel axis. The head is not assembled to the vessel by hanging it loosely from a crane hook, though this, and other methods are acceptable during construction.

A desirable, but not mandatory, design goal here is to assure that once the shear lip is assembled to the vessel ID, the bolts will all insert without further translational alignment (rotation may still be needed). Thus, if the vessel ID and mating shear lip are at MMC, and there is no remaining clearance between them, then the total bolt hole tolerance is equal to the hole to bolt diametral clearance, at MMC. This must be shared between the head and flange holes so the tolerance for each will be half the diametral tolerance. That is: for a 2 mm diametral clearance, each bolt hole axis may be as much as 1.0 mm off its nominal position; the total will be no more than 1 mm which produces an effective aperture 2 mm smaller than the hole diameter, = 16 mm and bolts will still assemble. In other words, the hole axis must be within a 1 mm radius (2 mm diameter) circle, thus the true position tolerance for the bolt holes is (a cylinder of) 2 mm dia. this is illustrated below:



$d_{cl\_mmc} := 18mm$

$d_b = 16\,mm$ note root diameter is less, but threaded portion is 25 mm long and must pass through both holes simultaneously

We need to account for vessel flange deflection under load which will distort the hole pattern. Maximum deflection, in the vertical direction is:

$$\delta_{fl} := 0.1mm$$

This is from an ANSYS workbench model with a 60000N load applied to each vessel internal flange, no head present. Assume head flange is undistorted

$$t_{bh\_max} := d_{cl\_mmc} - \left(d_b + \delta_{fl}\right)$$

$$t_{bh\_max} = 1.9\,mm$$

This is total maximum positional tolerance diameter for each flange hole, assuming the nominal hole positions of head and vessel flanges are in alignment.

There are two ways to specify bolt hole positional tolerance, either with respect to themselves as a pattern (the pattern otherwise unconstrained) or each hole individually, with respect to the specified datums. The former is a precisional tolerance, the latter an accuracy tolerance, (which is more difficult to achieve).

For the case where there is no remaining clearance between the shear lip and the vessel ID, when both are at MMC, the requirements for accuracy and precision are the same, and t_{bh_max} is the maximum allowable positional tolerance with respect to the datums A/B,C/D, and E/F; that is we have only an absolute accuracy requirement for bolt hole positional tolerance.

Any radial clearance between the shear lip and the vessel ID, with both at MMC, allows the two hole patterns to shift, as an ensemble, with respect to each other. This allows a larger maximum allowable positional tolerance of the holes with respect to the datums A/B,C/D, and E/F (accuracy requirement), but still requires that the hole pattern still be toleranced to t_{bh_max} or smaller, with respect to itself (a local precision or repeatability requirement). This is accomplished with two tolerance blocks on the drawing. The drawback is that the shear lip and vessel id may assemble, but in a shifted condition, such that bolts will not assemble. Additional alignment will then be needed. The cure for this is to tighten the tolerance (from t_{bh_max}) on the hole pattern in reference to itself, by the maximum shift that can occur when the shear lip and vessel ID are in LMC condition. Given that these are large features, their tolerance will necessarily be large.

Set:

$t_{bh} := 1.5mm$ $\qquad$ $t_{bh} < t_{bh\_max} = 1$

$R_{i\_pv} = 680\,mm$ $\qquad$ $\Delta R_{i\_pv} := 0.25mm$ $\qquad$ (+/-)

$R_{sl} := 679mm$ $\qquad$ $\Delta R_{sl} := 0.25mm$ $\qquad$ (+/-)

Nominal radial clearance between shear lip and vessel, both at MMC:

$$r_{cl} := \left(R_{i\_pv} - \Delta R_{i\_pv}\right) - \left(R_{sl} + \Delta R_{sl}\right) \qquad r_{cl} = 0.5\,mm$$

Check: $r_{cl} > 2\delta_{fl} = 1$ Head will assemble to flange with full detector mass loading (safety factor>2)

The radial clearance between shear lip and vessel ID (both at MMC) is represents an additional tolerance that we can add to the accuracy tolerance, because we can use it to shift the patterns to match. Since tolerances are specified on a diameter basis, we add 2x the radial offset (minus 2x the deflection):

$$t_{bh\_acc\_max} := t_{bh} + 2\left(r_{cl} - \delta_{fl}\right) \qquad t_{bh\_acc\_max} = 2.3\,mm$$

Using this value might require a very high alignment precision to find the proper bolt alignment so we use a slightly smaller value

$t_{bh\_acc} := 2mm$

Will head and bolts "auto-assemble" (assemble without further translational alignment) for shear lip and vessel ID at MMC?

Check: $t_{bh\_acc} \le t_{bh\_max} = 0$

If false, additional shift of head relative to vessel may be necessary, even though shear lip assembles to vessel ID. If true, we can proceed to check for the case of shear lip and vessel ID at LMC, below:

Maximum offset of shear lip and vessel ID axes (both at LMC)

$$\Delta r_{cl} := \left(R_{i\_pv} + \Delta R_{i\_pv}\right) - \left(R_{sl} - \Delta R_{sl}\right) \qquad \Delta r_{cl} = 1.5\,mm$$

Maximum offset of bolt holes for flanges at LMC, bolts and holes at MMC

$$\Delta r_{cl} + t_{bh} = 3\,mm$$

check if bolts will assemble with vessel ID and shear lip assembled at LMC (fully misaligned), without further translation alignment:

$$\Delta r_{cl} + t_{bh} \le t_{bh\_max} = 0$$

We conclude that we may need to further translate and rotate the head relative to the vessel in order to align the bolt holes, even though the shear lip assembles. Since no "autoassembly is possible, we can loosen the accuracy requirement conditionally by specifying the true position tolerance for circular datum C or D at MMC condition; this allows the final bolt hole accuracy tolerance to increase by the amount datum C or D are from MMC.

The head must be retracted for this operation as the actual clearance between the shear lip and the vessel ID will not be known. Furthermore, the adjustment of the struts is not performed simultaneously, and large intermediate translations or rotations of the headmay take place prior to achieving the final small alignment. This would cause an interference of the shear lip with the ID, with possible damage. Internal components may require further retraction of head prior to alignment. LBNL Engineering Note 10182B, D. Shuman, provides a general method and MathCAD worksheet for determining the needed strut adjustments to align a component. The 6 strut head alignment and assembly fixture designed for the heads has Cartesian motions which are largely uncoupled and should be simple enough to adjust intuitively without needing this methodology.

Equivalent maximum radial, angular misalignments of bolt holes (given here as +/- values) for all parts at MMC. These describe square tolerance zones inscribed within the circular tolerance zones

With respect to each other (precisional tolerance)

$$\Delta r_{bh} := \frac{.71}{2} t_{bh} \qquad \Delta r_{bh} = 0.532\,mm \quad (+/-)$$

$$\Delta \theta_{bh} := \frac{.71}{2} \frac{t_{bh}}{R_{i\_pv}} \qquad \Delta \theta_{bh} = 0.045\,deg \quad (+/-)$$

With respect to datums A/B,C/D,E/F (accuracy tolerance)

$$\Delta r_{bh\_acc} := \frac{.71}{2} t_{bh\_acc} \qquad \Delta r_{bh\_acc} = 0.71\,mm \quad (+/-)$$

$$\Delta \theta_{bh\_acc} := \frac{.71}{2} \frac{t_{bh\_acc}}{R_{i\_pv}} \qquad \Delta \theta_{bh\_acc} = 0.06\,deg \quad (+/-)$$