Maximum operating pressure MOP

$P_{MOPa} := 15\text{bar}$ this is absolute, and so applies to cans with vacuum inside

Maximum allowable pressure MAWP, = 110% MOP, at minimum

$P_{MAWPa} := 1.1P_{MOPa}$

The cans, fabricated from Cu pipe (4in OD x .625" wall),

$OD := 3.8\text{in} \quad ID := 80\text{mm}$

tolerances (+/-) $t_{can} := 0.5(OD - ID) \qquad t_{can} = 8.26\,\text{mm}$

have the following nominal dimensions

$l_{can} := 150\text{mm} \qquad r_{i\_can} := 0.5ID$

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

Here we use division 1 rules, since wall thickness will likely be fairly thin

External pressure, maximum: $P_e := -P_{MAWPa} \qquad P_e = -16.5\,\text{bar}$

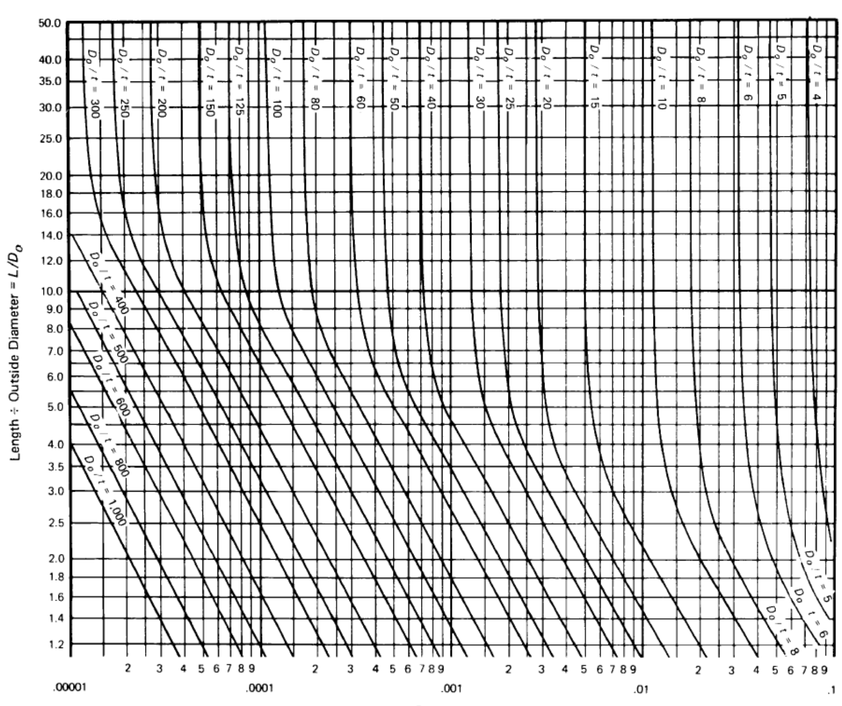
The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$\frac{l_{can}}{OD} = 1.6 \qquad \frac{OD}{t_{can}} = 12$

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

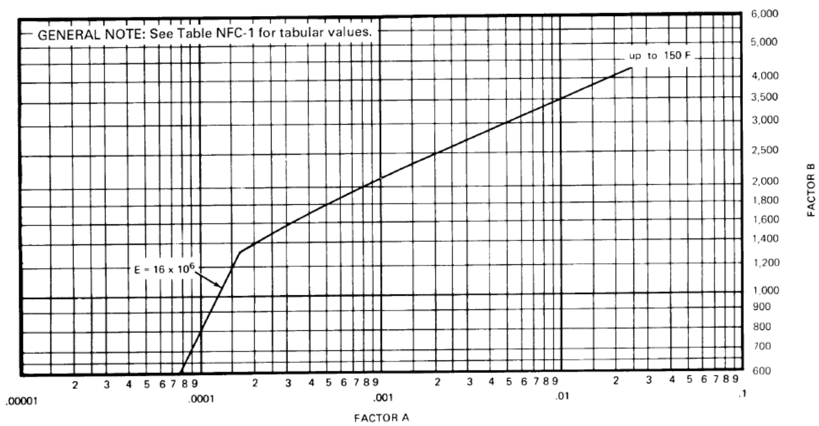
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$A := 0.02$

Using the factor A in the applicable material (Cu) chart (NFC-1) in Subpart 3 of Section II, Part D, we find the factor B:

FIG. NFC-1 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE DEVELOPED FOR ANNEALED COPPER, TYPE DHP



$B := 4000\text{psi}$

The maximum allowable working external pressure is then given by :

$$P_{max\_e} := \frac{4B}{3\left(\frac{2r_{i\_can}}{t_{can}}\right)} \qquad P_{max\_e} = 38\,\text{bar} \qquad P_{MAWPa} = 16.5\,\text{bar}$$

$P_{max\_e} > P_{MAWPa} = 1$ so the can is safe from buckling under external pressure, for wall thickness: $t_{can} = 8.26\,\text{mm}$ or greater

$P := P_{MAWPa}$

Sapphire thickness required: this is described in a separate document : sapphire.mcd

Window Clamp Design

We use ASME Appendix Y formulas (sec VIII, div 1) to calculate flange thickness and required number of screws for both O-ring and Helicoflex gasket options.

Flange Design for enclosure window clamp (same for backplate)

We assume here that the flanges are symmetric, that is, there is zero angular deflection at the flange interface. This assumption is made by noting that the enclosure itself will be very stiff, imposing a near-zero edge deflection

color scheme for this document

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

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

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.

We will design to use one Helicoflex 1.6mm 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

$S_{max\_C10200} := 6700psi$ $S_{y\_C10200} := 10000psi$

$S_f := S_{y\_C10200}$ $S_f = 68.9\,MPa$

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

Inconel 718 (UNS N07718) 316 condition/temper 2 (SA-193, SA-320) temper 1

$S_{max\_N07718} := 37000psi$ $S_{max\_316\_bolt2} := 22000psi$ $S_{max\_316\_bolt1} := 18800psi$

Excerpt from DIN EN ISO 3506-1:2010

Using european strength class designations the yield strengths are:

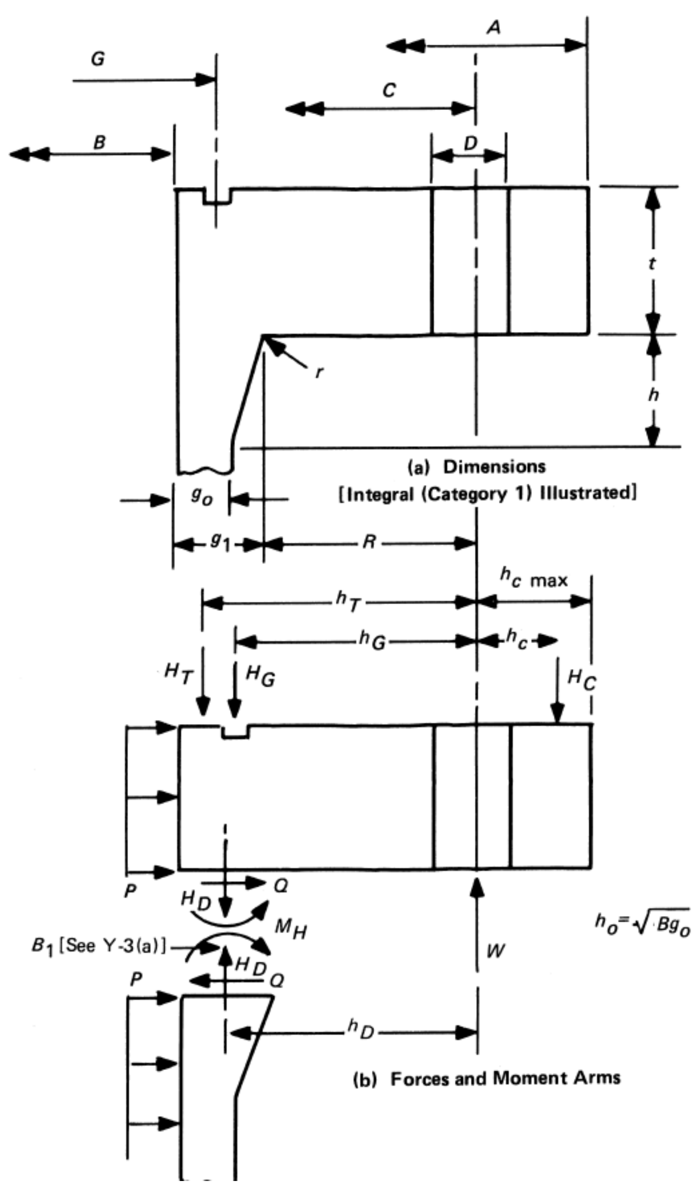
$S_{y\_50} := 210MPa$ $S_{y\_70} := 450MPa$ $S_{y\_80} := 600MPa$

Bolt material allowable stresses (here we do not use ASME allowables, as pressure is external):

$S_b := 0.85S_{y\_70}$ $S_b = 382.5\,MPa$

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:

$g_0 := 4mm \quad g_1 := g_0 \qquad g_0 = 4\,mm \qquad g_1 = 4\,mm \qquad r_1 := \min(.5g_1, 5mm) \quad r_1 = 2\,mm$

Flange OD

$A := 9.6cm$

Flange ID

$B := 7.6cm$

define:

$B_1 := 7.6cm$

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

$C := 8.8cm$

Gasket dia

$G := 8.15cm$

Force of Pressure on flange   Pressure acts only on window, not flange

$H := .7854\pi B^2 \cdot 0bar \qquad H = 0\,N$

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 equiv. values of Y for the ASME force term F and gives several possible values for 3mm HN200 with aluminum jacket:

$Y_1 := 8\frac{lbf}{in}$ min value for our pressure and required leak rate (He) $Y_2 := 65\frac{N}{mm}$ low force seal, available from MHTS

for gasket diameter $D_j := G \qquad D_j = 0.082\,m$

Force is then either of:

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

$F_m = 358.716\,N \qquad F_j = 1.664 \times 10^4\,N$

Helicoflex recommends using Y2 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 in through the outer O-ring as well.

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

$n := 32 \qquad d_b := 2.5mm \qquad p_t := 0.45mm \qquad h_3 := .614p_t$

Check thread clearance on window bore

$R_{wb} := 4.21cm \qquad D_{wb} := 2R_{wb} \qquad D_{wb} = 8.42\,cm$

$C - (D_{wb} + d_b) = 1.3\,mm$ OK, but for roll tap threads, threads should be made first then window bore cut

root dia.

$d_3 := d_b - 2h_3 \qquad d_3 = 1.947 \times 10^{-3}\,m$

$A_b := n \cdot \frac{\pi}{4} \cdot d_3^{\,2}$ $\quad A_b = 0.953\,cm^2$

Check bolt to bolt clearance, head dia. is twice bolt dia:

$\pi C - 2.0 n \cdot d_b \geq 0 = 1$ $\quad \pi \frac{C}{n \cdot d_b} = 3.456$

roll formtap drill size, http://www.kar.ca/pdf/catalog/H4.pdf

$pct_{th} := .75$

$d_{drill} := d_b - \frac{pct_{th} \cdot p_t}{1.4706}$ $\quad d_{drill} = 2.271\,mm$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 0.25mm$ $\quad D_{tmin} = 2.75\,mm$

Set:

$D_t := D_{tmin}$ $\quad D_t \geq D_{tmin} = 1$

Compute Forces on flange:

$H_G := F_j$ $\quad H_G = 1.664 \times 10^4\,N$ — from Table 2-6 Appendix 2, Integral flanges

$h_G := 0.5(C - G)$ $\quad h_G = 0.325\,cm$

$H_D := .785 \cdot B^2 \cdot 0bar$ $\quad H_D = 0\,N$

$R := 0.5(C - B) - g_1$ $\quad R = 0.2\,cm$ — radial distance, B.C. to hub-flange intersection, int fl..

$h_D := R + 0.5 g_1$ $\quad h_D = 0.4\,cm$ — from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D$ $\quad H_T = 0\,N$

$h_T := 0.5(R + g_1 + h_G)$ $\quad h_T = 4.625\,mm$ — from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange (maximum value)

$M_P := (H_D) \cdot h_D + H_T \cdot h_T + H_G \cdot h_G$ $\quad M_P = 54.1\,N \cdot m$

Appendix Y Calc

$P_0 := 0Pa$ $\quad P_0 = 0\,bar$

Choose values for plate thickness and bolt hole dia:

$t := .5cm$ $\quad D := D_t$ $\quad D = 0.275\,cm$

Going back to main analysis, compute the following quantities:

$\beta := \frac{C + B_1}{2B_1}$ $\quad \beta = 1.079$ $\quad h_C := 0.5(A - C)$ $\quad h_C = 4 \times 10^{-3}\,m$

$a := \frac{A + C}{2B_1}$ $\quad a = 1.211$ $\quad AR := \frac{n \cdot D}{\pi \cdot C}$ $\quad AR = 0.318$ $\quad h_0 := \sqrt{B \cdot g_0}$

$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}} \operatorname{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right)$ $\quad r_B = 6.85 \times 10^{-3}$ $\quad h_0 = 0.017\,m$

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 do not have identical (class 1 assembly) integral (category 1) flanges, however the can is very stiff and acts like a fixed rotation boundary, similar to a pair of identical flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13), (14a), (15a), (16a)

$$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \qquad J_S = 0.163 \qquad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B \qquad J_P = 0.114$$

(5a) $$F' := \frac{g_0^{\,2}(h_0 + F \cdot t)}{V} \qquad F' = 6.393 \times 10^{-7}\,m^3 \qquad M_P = 54.088\,N \cdot m$$

$A = 9.6\,cm$ $\qquad B = 7.6\,cm$

$$K := \frac{A}{B} \qquad K = 1.263 \qquad Z := \frac{K^2 + 1}{K^2 - 1} \qquad Z = 4.358$$

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

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b$ $\quad l = 1.125\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

Elastic constants $\quad E_{SS\_aus} := 193GPa \quad E_{Cu} := 115GPa$ http://www.hightempmetals.com/techdata/hitempInconel718data.php

$E := E_{Cu}$ $\qquad E = 115\,GPa \qquad E_{Inconel\_718} := 208GPa \qquad E_{Inconel\_x750} := 213GPa$

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

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \qquad M_S = -17.2\,J \qquad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right) \qquad \theta_B = 4.063 \times 10^{-4} \qquad E \cdot \theta_B = 46.721\,MPa \qquad (7)$$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = 9.226 \times 10^3\,N \qquad (8)$$

Bolt Load at operating condition:

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = 2.587 \times 10^4\,N \qquad (9)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = 271.4\,MPa \qquad S_b = 382.5\,MPa \qquad (10)$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = 0.596$$ elasticity factor

Design Prestress in bolts

$$S_i := \overrightarrow{\left[\sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot (M_P + M_S)}{a \cdot t^3 \cdot l \cdot r_E \cdot B_1}\right]} \qquad S_i = 262.5\,\text{MPa} \qquad (11)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \qquad S_{R\_BC} = 47\,\text{MPa} \qquad (12)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID} = 18.466\,\text{MPa} \qquad (13)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_T = 3.07\,\text{MPa} \qquad (14a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = 21.412\,\text{MPa}$$

Y-7 Flange stress allowables: $S_b = 382.5\,\text{MPa}$ $S_f = 68.9\,\text{MPa}$

(a) $\sigma_b < S_b = 1$

(b)

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

(2) not applicable

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

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

(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

$$F_{bolt} := \frac{W_{m1}}{n} \qquad F_{bolt} = 808.38\,\text{N}$$

check chart below:

Bolt torque required

$T_{bolt\_min} := 0.2F_{bolt} \cdot d_b \quad T_{bolt\_min} = 0.4\,N \cdot m \qquad T_{bolt\_min} = 0.3\,lbf \cdot ft$

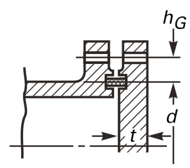
for pressure test use 1.5x this value

		Yield strength load for A2 and A4 in N		
Thread	Stress area	Strength 50	Strength 70	Strength 80
M 1.6	1,27	266,7	571,5	762
M 2	2,07	434,7	931,5	1242
M 2.5	3,39	711,9	1525,5	2034
M 3	5,03	1056,3	2263,5	3018

Backplate thickness required

We use ASME formula for flat heads (sec VIII, div 1), where S is strength, max allowable, E s weld efficiency (=1) P is pressure and C is a factor from fig. UG-34 (k) shown below:

$$t := d \cdot \sqrt{\frac{C \cdot P}{S \cdot E} + \frac{1.9W \cdot h_G}{S \cdot E \cdot d^3}} \quad \text{(eq 2)}$$



C = 0.3
[Use Eq. (2) or (5)]
(k)

$S := S_f$

$h_G := 2mm$

$G_{bp} := 8.35cm \qquad C_{bp} := 9.1cm$

$h_{G\_bp} := 0.5\left(C_{bp} - G_{bp}\right)$

$h_{G\_bp} = 3.75\,mm$

$d_{bp} := G_{bp}$

using Wm1 of above window clamp calculation:

$$t_{bp} := d_{bp} \cdot \sqrt{\frac{0.3 \cdot P}{S} + \frac{1.9W_{m1} \cdot h_{G\_bp}}{S_f \cdot d_{bp}^{\;3}}} \qquad t_{bp} = 9.059\,mm$$

Effect of Pressure on

$H_w := .7854\pi B^2 \cdot P \qquad H_w = 2.352 \times 10^4\,N$

Compare to gasket preload

$H_G = 1.664 \times 10^4\,N$

we see that additional gasket compression will occur at high pressures, once total force exceeds the preload. This will not affect bolts as they are tightened to a closed joint condition, however electrical contact between ITO and pressure ring may be compromised. We need to minimize the clearance between the window seat and the window and we may need to design in some compliance into the pressure ring.

Ledge shear stress, approximate

$$\frac{H_w}{(3mm \cdot 6 \cdot 40mm)} = 4.737 \times 10^3\,psi$$

Pressure Ring Design

Requirements:

1. Compatible with both O-ring and Helicoflex gasket
2. Long term maintenance of preload
3. Minimal outgassing and water absorption
4. Radiopure to <1mBq (all 60)
5. electrical contact between ITO and can, compatible

There are two design philosophies one can use here, either:

A. use a low elastic modulus elastomer with enough preload to compress the gasket and build some addtional clamping stress in the window between the pressure ring and the window seat (thin Kapton) or:

B. use high modulus materials to constrain the window movement on both sides, but only enough to compress the window gasket fully.

In both methods, compression is displacement controlled; the window clamp flange is screwed fully down onto the enclosure surface; screw torque is used to maintain flush contact. This allows screws to be tightened enough to maintain tightness of the copper to copper joint, decoupling it from the gasket preload.

In method A, an elastomer pressure ring must be used, and these materials seem to have high radioactivities. In method B is a low modulus material such as UHMW-PE might be a possibility but, as shown below, stress is too high, so a high strength polymer is indicated. PEI (ULTEM-1000) is a clear polymer similar to KaptcGasket diameterradiopure.

$G = 8.15\,cm$

Gasket compression required:

$Y_2 = 65\,\frac{N}{mm}$ low force Helicoflex HN100 design, from HTMS

$H_p := \pi G \cdot Y_2$ $H_p = 1.664 \times 10^4\,N$

Some possible materials (elastic moduli and strengths)

$E_{c\_PEI} := 480000psi$ $S_{c\_PEI} := 22000psi$ Boedecker, ULTEM-1000, unfilled

$E_{UHMW} := 125000psi$ note: UHMW creeps under load $E_{UHMW} = 861.845\,MPa$

$E_{UHMW\_1000hr} := 200MPa$ creep modulus, 1000hr, 23C from GUR datasheet

$E_{acetal} := 450000psi$ all from Boedecker plastics

$E_{nylon\_6\_6} := 400000psi$

$E_{PEEK} := 500000psi$

$E_{c\_PEEK\_30pC} := 1250000psi$ $S_{c\_PEEK\_30pC} := 29000psi$

Pressure ring dimensions and material selection

radius width thickness, should be > 1.5x gasket compression distance to allow ring to fit into bore prior to

$R_{pr} := 4cm$ $w_{pr} := 4.19cm - 3.8cm$ $w_{pr} = 0.39\,cm$ $t_{pr} := 1mm$

$E_{pr} := E_{c\_PEI}$

Pressure area:

$A_{pr} := 2\pi R_{pr} \cdot w_{pr}$ $A_{pr} = 9.802\,cm^2$ $M_{pr} := 1\,\frac{gm}{cm^3} \cdot A_{pr} \cdot t_{pr} \cdot 60$ $M_{pr} = 58.811\,gm$ all

Compressive Stress

$\sigma_{pr} := \frac{2H_p}{A_{pr}}$ $\sigma_{pr} = 33.958\,MPa$ this is high and rules out HDPE, UHMW, acetal, etc; we need high strength

$S_{c\_PEI} = 151.685\,MPa$ OK, good margin of safety

$S_{c\_PEEK\_30pC} = 199.948\ MPa$

Strain

$\varepsilon_{pr} := \frac{\sigma_{pr}}{E_{pr}}$ $\quad \varepsilon_{pr} = 1.026\%$

Compression distance required:

$\delta_{pr} := \varepsilon_{pr} \cdot t_{pr}$ $\quad \delta_{pr} = 0.01\ mm$ essentially zero $\quad .04in - 1mm = 0.016\ mm$

The only materials with sufficient strength to compress and maintain an Helicoflex gasket are PEI , PEEK, etc, but they are very stiff. Compression strain is essentially nothing, so we design flange height to give flush surfaces, with no O-ring or gasket present (window seat present). Gasket compression is thus determined by the dimensional tolerances and we give up the desire to obtain a repeatable and well defined clamping pressure between the pressure ring and window seat ring (this requires a low modulus, low creep material like an elastomer which tend to be radioactive. Window will then "float" on O-ring or Helicoflex, not being clamped tightly against ledge (kapton window seat). The problem is then that the window seat will not stay centered, and will interfere with the O-ring or Helicoflex gasket. One solution is to machine a groove in the ledge and use a thicker window seat that fits into it. The copper must still support the O-ring but can be made thinner. One thing to possibly investigate is whether the plastic will swell under Xenon permeation, leading to high compression of the window.

PEI may prove radiopure, but we need some electrical connection between ITO and flange, so 30% carbon filled PEEK is an option. Alternatively a few soft fine gold wires would work, they will flatten out and into the pressure ring.