Maximum operating pressure MOP

$P_{MOPa} := 15bar$ 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),

$OD := 3.8in \quad ID := 80mm$

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

have the following nominal dimensions

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

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: $P_e := -P_{MAWPa} \quad P_e = -16.5\,bar$

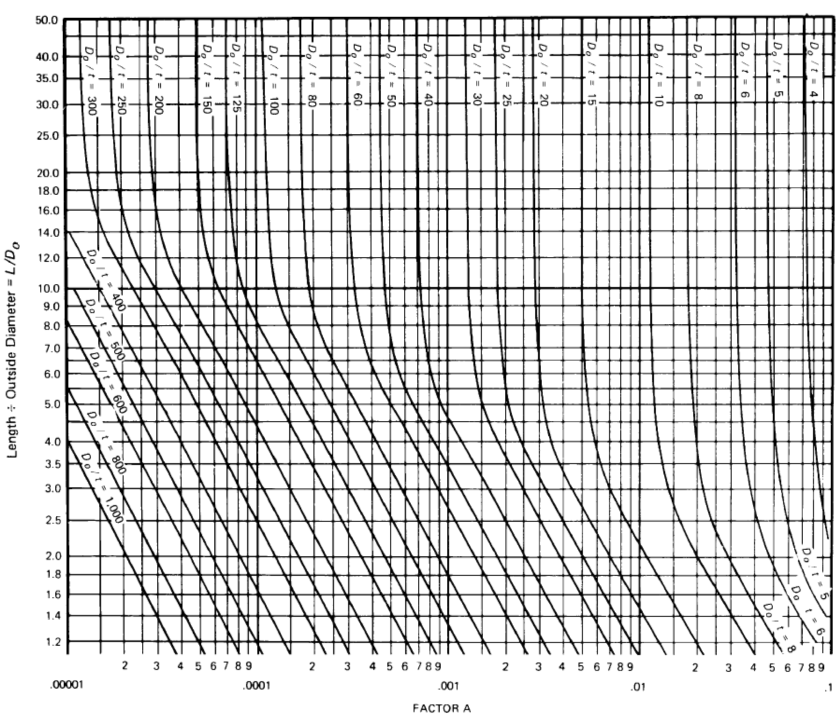
The maximum allowable working external pressure is determined by the following procedure:

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:

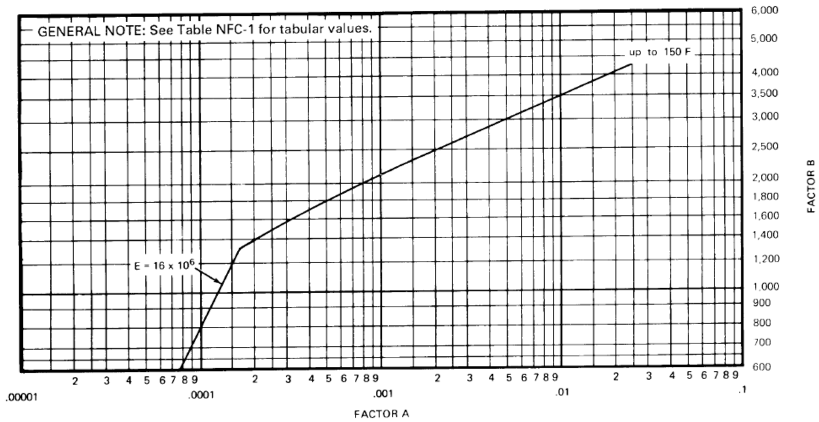
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$A := 0.02$

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

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

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

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:

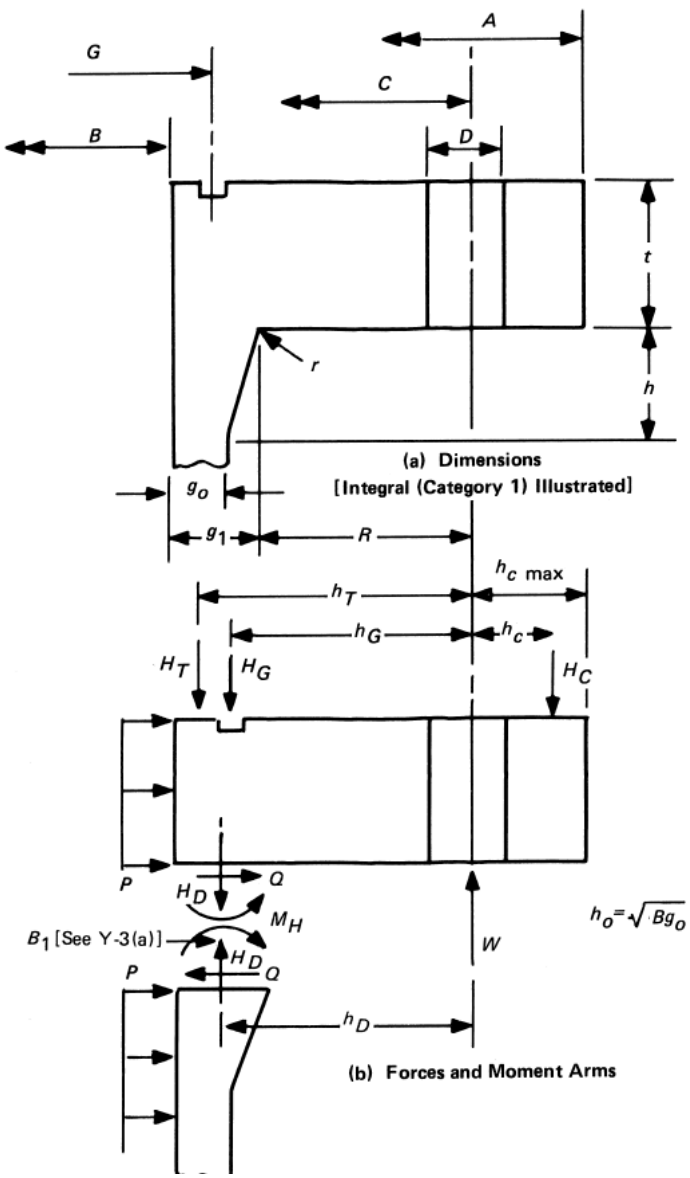
$S_{y\_50} := 210MPa$ $S_{y\_70} := 450MPa$ $S_{y\_80} := 600MPa$

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:

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

Flange OD

$A := 9.6cm$

Flange ID

$B := 7.6\text{cm}$

define:

$B_1 := 7.6\text{cm}$

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

$C := 8.8\text{cm}$

Gasket dia

$G := 8.15\text{cm}$

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

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

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:

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

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

Force is then either of:

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

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

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,

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

Check thread clearance on window bore

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

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

root dia.

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

$A_b := n \cdot \dfrac{\pi}{4} \cdot d_3^{\,2}$ $\qquad A_b = 0.953\,\text{cm}^2$

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

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

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

$pct_{th} := .75$

$d_{drill} := d_b - \dfrac{pct_{th} \cdot p_t}{1.4706}$ $\qquad$ $d_{drill} = 2.271\,\text{mm}$

Flange hole diameter, minimum for clearance :

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

Set:

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

Compute Forces on flange:

$H_G := F_j$ $\qquad$ $H_G = 1.664 \times 10^4\,\text{N}$ $\qquad$ from Table 2-6 Appendix 2, Integral flanges

$h_G := 0.5(C - G)$ $\qquad$ $h_G = 0.325\,\text{cm}$

$H_D := .785 \cdot B^2 \cdot 0\text{bar}$ $\qquad$ $H_D = 0\,\text{N}$

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

$h_D := R + 0.5g_1$ $\qquad$ $h_D = 0.4\,\text{cm}$ $\qquad$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D$ $\qquad$ $H_T = 0\,\text{N}$

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

Total Moment on Flange (maximum value)

$M_P := (H_D) \cdot h_D + H_T \cdot h_T + H_G \cdot h_G$ $\qquad$ $M_P = 54.1\,\text{N} \cdot \text{m}$

Appendix Y Calc

$P_0 := 0\text{Pa}$ $\qquad$ $P_0 = 0\,\text{bar}$

Choose values for plate thickness and bolt hole dia:

$t := .5\text{cm}$ $\qquad$ $D := D_t$ $\qquad$ $D = 0.275\,\text{cm}$

Going back to main analysis, compute the following quantities:

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

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

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

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

since $\dfrac{g_1}{g_0} = 1$ 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

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

Y-6.1, Class 1 Assembly Analysis

Elastic constants $E_{SS\_aus} := 193GPa \quad E_{Cu} := 115GPa$ 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

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

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

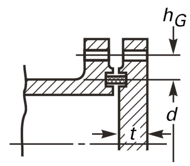
$T_{bolt\_min} := 0.2F_{bolt} \cdot d_b$ $\quad T_{bolt\_min} = 0.4\,N \cdot m$ $\quad T_{bolt\_min} = 0.3\,lbf \cdot ft$

		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 is weld efficiency (=1) P is pressure and C is a factor from fig. UG-34 (k) shown below:

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



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

$S := S_f$

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

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

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

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

using Wm1 of above window clamp calculation:

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

Pressure Load on Window

$H_w := \frac{\pi}{4} G^2 \cdot P$ $\quad H_w = 8.608 \times 10^3\,N$

Compare to gasket preload

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

we see that gasket preload is much higher than pressure, and any further movement of the window under pressure would relieve the prestress from the pressure ring, so the effect is that there is a true prestress condition, and the window will not move or compress gasket any further under pressure application. Thus there are no concerns with losing the electrical connection between the ITO and the pressure ring. It is a different story with O-rings however, as described below

Ledge shear stress, approximate

For helicoflex, the gasket force is directed through the wall thickness of the enclosure, and the only concern is for brinelling of the copper. Tests on the IFIC prototype gaskets show a flat area on the gaskets of ~1mm

$\sigma_{c\_hf} := \frac{Y_2}{1mm}$ $\quad \sigma_{c\_hf} = 65\,MPa$ $\quad$ compare to ASME max allowable $S_f = 68.948\,MPa$ $\quad$ OK

For O-ring operation, the ledge takes the full pressure load:

Ledge thickness in axial direction (shear width)

$w_{l\_s} := 3mm$

$\tau_1 := \frac{H_w}{\pi G \cdot w_{l\_s}}$ $\tau_1 = 11.206\,MPa$ very low, no concern here

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 Kapton which may be radiopure.

Gasket diameter

$$G = 8.15\,\text{cm}$$

Gasket compression required:

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

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

Some possible materials (elastic moduli and strengths)

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

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

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

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

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

$$E_{PEEK} := 500000\text{psi}$$

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

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

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

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

Pressure area:

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

Compressive Stress

$$\sigma_{pr} := \frac{2H_p}{A_{pr}} \qquad \sigma_{pr} = 33.958\,\text{MPa}$$ 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

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

Compression distance required:

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

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 Helicoflex, not being clamped tightly against ledge (kapton or PEEK window seat). The window seat now has a groove in the ledge to hold this set in place, 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 the preferred option. Alternatively a few soft fine gold wires would work, they will flatten out and into the pressure ring.

For O-ring operation, the window may move somewhat under pressure, so this PEEK pressure ring can be replaced with a conductive elastomer of some sort to maintain electrical conductivity.

Window Seat Design

For helicoflex gasket, this ring is not stressed, other than that from tolerances (the nominal clearnace is zero, in the unstressed condition). For O-ring use this seat ring reacts full gas pressure on window.

$r_{i\_ws} := 3.8cm \qquad r_{o\_ws} := 3.95cm \qquad w_{ws} := r_{o\_ws} - r_{i\_ws} \qquad w_{ws} = 1.5\,mm$

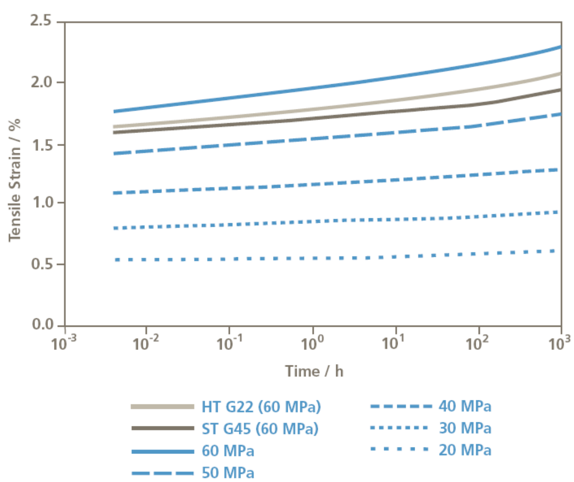
$$\sigma_{ws} := \frac{H_w}{2\pi \cdot r_{o\_ws}\, w_{ws}} \qquad \sigma_{ws} = 23.122\,MPa$$

$G = 0.082\,m$

$\sigma_{ws} = 3.354 \times 10^{3}\,psi$

We need a material that has negligible creep at this stress. Delrin (acetal) literature shows creeep curves only to 20 MPa. PEEK is a good candidate; from the Victrex design guide:

Figure 7: Tensile creep of PEEK 450G, HT and ST at 23°C



We see that at strain growth at 30 MPa is negligible