Project: PS-TGT-12-001 Hall A Tritium Target

Tittle: General target cell calculations

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Code(s) of Record:

ASME B31.3 2012

Reference Codes and Sources:

- ASME BPVC VIII D2 2013
- Antonetti, V.W., Whittle, T.D. and Simons, R.E., An approximate thermal contact conductance correlation, ASME J. Electronic Packaging, 115 (1993) 131-134.
- Helicoil TB 68-2 Tensile Strength of Threaded Insert Assembly

Description:

General target cell calculations

Reference Drawing(s):

TGT-103-1000-0013	Cell assembly with shipping covers
TGT-103-1001-0000	P&ID

General:

Temperatures:

$T_{op} := 45 \cdot K$ operating temperature

$T_{room} := 300 \cdot K$ Room temperature

Pressures:

$P_{fill} := 200 \cdot psi$ assumed fill pressure

Helicoils:

The tensile strength of a single threaded insert assy is given in Helicoil Tech Bulletin 68-2. The aluminum 7075-T6 base metal has a higher shear strength than that of Al 2024 which shall be used. The strengths given below are for inserts that are 2x nom bolt diameter

$T_{164} := 6000 \cdot lbf$ tensile strength for 8-32 assy

$T_{25} := 11000 \cdot lbf$ tensile strength of 1/4-28 assy

To accurately model the effects of the beam heating in the entrance and exit windows the thermal contact resistance at cell block to heat sink interface must be modeled. The heat generated in by the beam in the cell is conservatively estimated at 15W. The model for contact conductance is from Antonetti et. al.

$h_{cont} := 2.788 \cdot in$ $w_{cont} := 3 \cdot in$

$A_{cont} := w_{cont} \cdot h_{cont} = 0.005\ m^2$ Area of contact

$N_{bolt} := 4$ number of bolts fixing

$\tau := 150 \cdot in \cdot lbf$ torque on bolts

$d_{bolt} := 0.25 \cdot in$ nominal diameter

$F_{bolt} := \dfrac{\tau}{0.2 \cdot d_{bolt}} = \left(3 \cdot 10^{3}\right)\ lbf$ force per bolt

$P_{cont} := \dfrac{N_{bolt} \cdot F_{bolt}}{A_{cont}} = \left(1.435 \cdot 10^{3}\right)\ psi$ total pressure on contact area (avg)

$R_{surf} := 1.6$ surface roughness in microns

$k_{copper} := 401 \cdot \dfrac{W}{m \cdot K}$ cond of copper

$k_{alum} := 77 \cdot \dfrac{W}{m \cdot K}$ cond of 7075 AL

$k_s := \dfrac{2 \cdot k_{copper} \cdot k_{alum}}{k_{copper} + k_{alum}}$ mean harmonic cond

$H_{cont} := 878 \cdot MPa$ hardness of contact surface

Contact conductance is shown below to be high s.t. we need not consider it in the thremal model

$$h_{cont} := 4200 \cdot k_s \cdot R_{surf}^{-0.257} \cdot \left(\frac{P_{cont}}{H_{cont}}\right)^{0.95} = \left(6.78 \cdot 10^{3}\right)\ \frac{W}{m \cdot K}$$

Pressure design of the tritium cell:

The design pressure of the tritium cell assembly is determined below:
Given the geometry and materials of cell the most applicable ASME Pressure Code is ASME B31.3 2012. The geometry is abnormal and Section 304.7.2 shall be applied. This section requires that the design be substantiated through one of several methods. The methods chosen for the tritium cell are as follows:

1. extensive experience
2. proof test meeting the requirements of ASME BPVC D1 VIII UG-101
3. detailed analysis consisting of both hand calculations and FEA.

The cell material with the exception of the fill tube assembly is frabricated from ASTM B209 Aluminum 7075-T651 Plate. This material is unlisted; thus, an allowable stress must be determined. All cells shall be made from a single billet (with MTR). This material conforms to a listed specification as well. The allowable stress is determined below using values from B209 where applicable:

$S_{ut} := 72 \cdot ksi$ B209 listed minimum ultimate tensile

$S_y := 61 \cdot ksi$ B209 listed minimum yield 0.2%

Note that from material certification (MTR):

Sut = 77.6 ksi and Sy = 65.6

Material is domestic from Kaiser Aluminum Lot#107954B0

$Sut_{MTR} := 77.6 \cdot ksi$ $Sy_{MTR} := 65.6 \cdot ksi$

$S_a := min\left(\frac{1}{3} \cdot S_{ut}, \frac{2}{3} \cdot S_y\right) = 24\ ksi$ max allowable stress AL7075 T651

The shear strength assumed for AL 7075-T651 is

$S_{shear} := 0.65 \cdot S_{ut}$ shear of Al 7075

Main Body:

For simplicity we assume that the cell main body TGT-103-1000-0101 is a cylinder with a hemishperical end cap with a thinner section. The design wall thickness is 0.018 for the cylinder and 0.011 for the endcap. The machine tolerance is +/- 0.001 inch which is verified by measurement for each cell on entire body.

$t_{cap} := 0.010 \cdot in$	endcap min thickness
$t_{wall} := 0.017 \cdot in$	wall min thickness
$D_{in} := 0.5 \cdot in$	inner diameter of cell
$D_o := D_{in} + 2 \cdot 0.020 \cdot in$	OD of shell with max wall thickness
$E := 1$	quality factor for machined tooling plate
$W := 1$	weld factor (no welds)
$Y := 0.0$	Factor Y=0 for conservatism

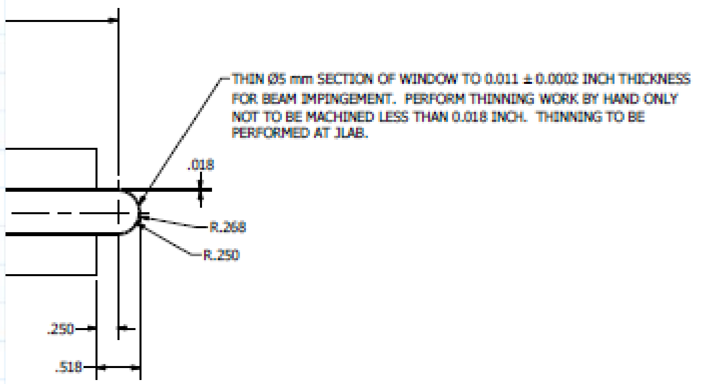
The maximum design pressure for the cylinder part of the main body is:

$$P_{main} := \frac{2 \cdot t_{wall} \cdot S_a \cdot E \cdot W}{D_o - 2 \cdot t_{wall} \cdot Y} = \left(1.511 \cdot 10^3\right) \, psi$$

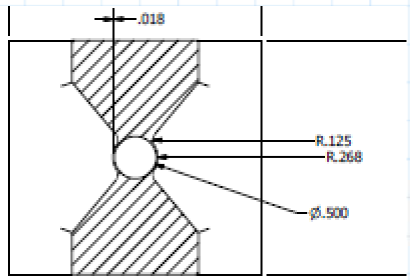
The endcap may be treated similarly; stress in hemi is 1/2 that of cylinder

$$P_{cap} := \frac{4 \cdot t_{cap} \cdot S_a \cdot E \cdot W}{D_o - 2 \cdot t_{wall} \cdot Y} = \left(1.778 \cdot 10^3\right) \, psi$$

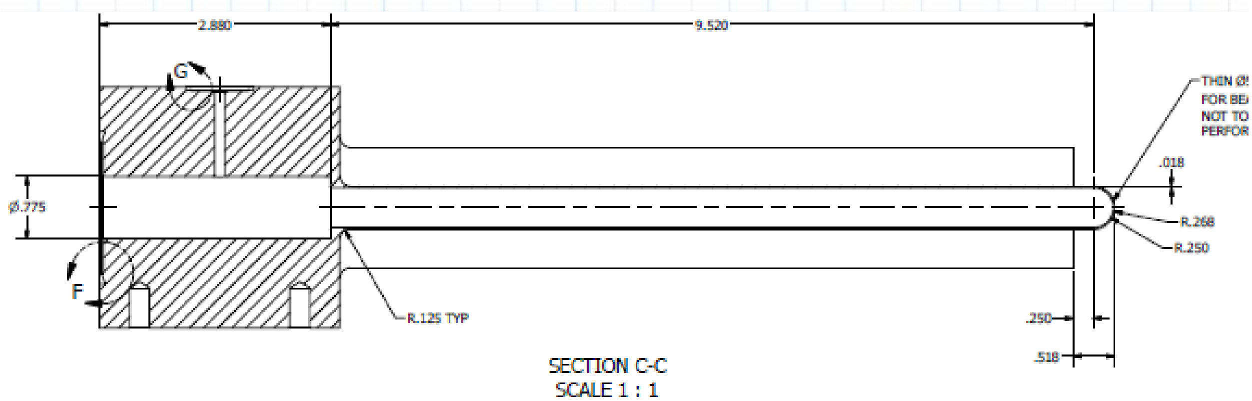
Note that the thickness of the head increases to the wall thickness of the cylinder. See figures below for ref.



section view of beam exit region of main body



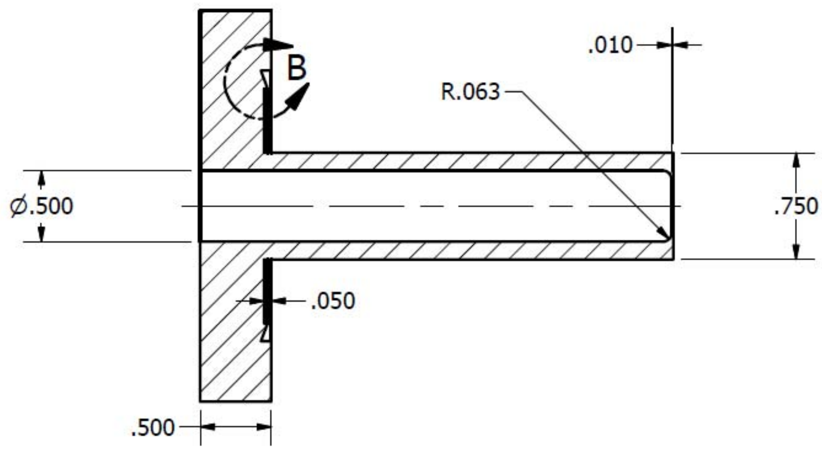
section view of main body from downstream viewpoint



Section view of main body from side

Entrance Window:

The entrance window TGT-103-1000-0100 is machined from a single piece of B209 7075-T651 AL from the same billet as the cell main body. A number of windows of this type have been used successfully for 10 years. A proof test was performed on this design. The entrance window thicknesses are measured carefully using a MagnaMike (accuracy < 0.03 mm). Windows that are below the required thickness of 0.010 in are discarded or tested to destruction. Entrance window is shown in the figure as it is machined. The window is hydrofromed such that a dished head has a radius of 0.5 inch as shown in the second fig.



SECTION A-A
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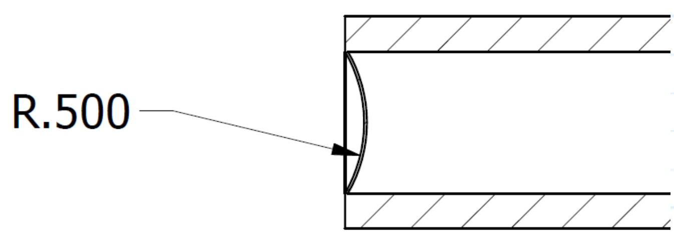


window after forming

The following analysis for the thin disk section is from Roarks Formulas for Stress and Strain 6th ed (1989) section 10.11.

Radius of the disk $a := \frac{0.5}{2} \cdot in$

Elastic modulus $E := 1.02 \cdot 10^{7} \cdot psi$

Piosons ratio $\nu := 0.33$

Pressure $q := 1000 \cdot psi$

Thickness of section machined $t_{pre} := 0.010 \cdot in$

Thickness after forming $t := t_{pre} \cdot 0.93 = 0.009\ in$

From ASME BPVC VIII D1 UG-32 for dished toraspherical heads

$R_h := 0.5 \cdot in$ inner radius of head

$E := 1$ quality factor for machined window

maximum design pressure:

$$P_{win} := \frac{2 \cdot S_a \cdot E \cdot t}{0.885 \cdot R_h + 0.2 \cdot t} = \left(1.005 \cdot 10^{3}\right)\ psi$$

Entrance tube:

The entrance tube is made of 7075 with properties given below. Analysis follows VIII D2 4.4 using a tangent modulus of 0. Entrance tube is under external pressure.

$E_y := 1.02 \cdot 10^{7} \cdot psi$ lesser of tensile and flexural modulus

$E_t := 0 \cdot ksi$ tangent modulus (see below for torlon)

$\nu_{trn} := 0.45$ poisons ratio

$E_t := 0 \cdot ksi$ tanget modulus assuming bilinear model

$D_0 := 0.75 \cdot in$ OD of tube

$R_0 := 0.5 \cdot D_0$ outter radius of tube

Step1 : unsupported length and thickness

$L := 3.329 \cdot in$ Length of tube (unsupported)

$t := 0.125 \cdot in$ wall thickness of tube

step 2: predicted eleastic buckling stress

$M_x := \dfrac{L}{\sqrt{R_0 \cdot t}} = 15.376$ elastic moment

$$C_h := \begin{array}{l} \text{if } M_x \ge 2 \cdot \left(\dfrac{D_0}{t}\right)^{0.94} \\ \quad \text{return } 0.55 \cdot \dfrac{t}{D_0} \\ \text{else if } 13 < M_x < 2 \cdot \left(\dfrac{D_0}{t}\right)^{0.94} \\ \quad \text{return } 1.12 \cdot M_x^{-1.058} \\ \text{else if } 1.5 < M_x \le 13 \\ \quad \text{return } \dfrac{0.92}{M_x - 0.579} \\ \text{else} \\ \quad \text{return } 1.0 \end{array}$$

$C_h = 0.092$

The predicted elastic buckling stress is

$$F_{he} := \frac{1.6 \cdot C_h \cdot E_y \cdot t}{D_0} = \left(2.493 \cdot 10^{5}\right)\ psi$$

step 3: buckling stress

$$F_{ic} := \begin{array}{l} \text{if } \dfrac{F_{he}}{S_y} \ge 2.439 \\ \quad \| S_y \\ \text{else if } 0.552 < \dfrac{F_{he}}{S_y} < 2.439 \\ \quad \| 0.7 \cdot S_y \cdot \left(\dfrac{F_{he}}{S_y}\right)^{0.4} \\ \text{else} \\ \quad \| F_{he} \end{array}$$

Predicted buckling stress

$$F_{ic} = \left(6.1 \cdot 10^{4}\right)\ psi$$

step 4: design factor

$$FS := \begin{array}{l} \text{if } F_{ic} \le 0.55 \cdot S_y \\ \quad \| 2 \\ \text{else if } 0.55 \cdot S_y < F_{ic} < S_y \\ \quad \| 2.407 - 0.741 \cdot \left(\dfrac{F_{ic}}{S_y}\right) \\ \text{else} \\ \quad \| 1.667 \end{array}$$

Required design factor

$$FS = 1.667$$

step 5: allowable pressure

$$F_{ha} := \frac{F_{ic}}{FS} = \left(3.659 \cdot 10^{4}\right)\ psi$$

maximum allowable pressure for the entrance tube

$$P_{tube} := 2 \cdot F_{ha} \cdot \frac{t}{D_0} = \left(1.22 \cdot 10^4\right)\ psi$$

This is more than adequate given the very conservative assumption of an elastic plastic model.

CF flange stress:

The entrance window CF flange is a 2.75 in CF design. The material is AL7075. This is an unlisted material. The minimum tensile and yield strengths are given below. The gasket is aluminum The tensile and yield strengths of AL7075-T6 are both larger than that of SST 304. The values for SST 304 are assumed. The flange is 2.75" OD with a 0.75" diameter tube. The thickness of the flange is 0.43". Bolts - 6 each are1/4-28 A286 Alloy. To determine the suitability of this flange, we will use the rules of BPVC VIII D1 Appendix 2. Load on single bolt to seat the gasket was **measured** to be 340 lbf.

$N_{bolt} := 6$ The number of bolts is

$F_{bolt} := 340 \cdot lbf$ Load on each bolt

$P := 1000 \cdot psi$ The pressure is

$A := 2.74 \cdot in$ The outer diameter of the flange is

$t_f := 0.495 \cdot in$ The thickness of the flange is

$C := 2.312 \cdot in$ The bolt circle diameter is

$B := 0.5 \cdot in$ The inner or bore diameter is

The diameter at the gasket load reaction is chosen to be the knife edge of the flange

$$G := 1.65 \cdot in$$

The relevant moment arms for this flange are (notation same as Appendix 2)

$$h\_d := \frac{(C - B)}{2} \qquad h\_t := \frac{(C - G)}{2} \qquad h\_g := \frac{(C - G)}{2}$$

The total hydrostatic end force is

$$H := \frac{\pi}{4} \cdot G^2 \cdot P = 2138.246\ lbf$$

$$H\_p := N_{bolt} \cdot F_{bolt}$$

The hydrostatic end force and the inside of the flange is

$$H\_d := \frac{\pi}{4} \cdot B^2 \cdot P = 196.35\ lbf$$

H_t is the difference between H and H_d

$$H\_t := H - H\_d = \left(1.942 \cdot 10^3\right)\ lbf$$

The gasket load is

$$H\_g := H\_p - H = -98.246\ lbf$$

The total moment acting on the flange

$$M\_o := H\_t \cdot h\_t + H\_d \cdot h\_d + H\_g \cdot h\_g = 788.141\ in \cdot lbf$$

The ratio of outside to inside diameter is

$$K_a := \frac{A}{B} = 5.48$$

The factor Y is given

$$Y := \frac{1}{K_a - 1} \cdot \left(0.66845 + 5.7169 \cdot K_a^{\ 2} \cdot \frac{\log\left(K_a\right)}{K_a^{\ 2} - 1}\right) = 1.124$$

We are assuming that the CF flange meets the conditions of sketch 4a of Fig 2-4
For loose type flanges of this type we may use 2-7 formula (9) for the stress in the flange

$$S\_T := \frac{Y \cdot M\_o}{\left(t_f\right)^2 \cdot B} = 7.234\ ksi$$

This is less than the maximum allowable stress the design is acceptable

Flange Bolt Stress

The bolt material is A286. The max allowable stress is unlisted and must be determined:

$S_{ut} := 130 \cdot ksi$ listed UT for A286

$S_y := 85 \cdot ksi$ listed Yield for A286

$S_a := min\left(\frac{1}{3} \cdot S_{ut}, \frac{2}{3} \cdot S_y\right) = 43.333\ ksi$ max allowable stress for a286

The root bolt area is

$$A_{root} := \frac{\pi}{4} \cdot 0.205^2 \cdot in^2$$

The tensile stress in the bolt is

$$S_t := \frac{F_{bolt} + \frac{H}{N_{bolt}}}{A_{root}} = 21.098\ ksi$$

This is less than the max allowable

Helicoil pull out:

$\tau := 140 \cdot in \cdot lbf$ specified torque on 1/4-28

$K_f := 0.19$ measured coef

$d := 0.25 \cdot in$ nom diam

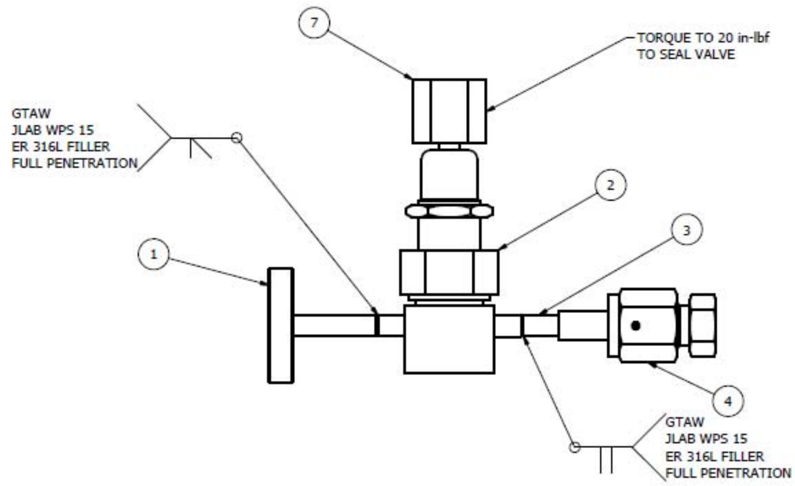
$F := F_{bolt} + \frac{\tau}{K_f \cdot d} = \left(3.287 \cdot 10^3\right)\ lbf$ Force on bolt

$F_a := \frac{T_{25}}{3} = \left(3.667 \cdot 10^3\right)\ lbf$ Max Allowable force

$F < F_a = 1$ acceptable

Valve stem assy:

The assembly is described in TGT-103-1000-0011 and is shown below. The valve is a Swagelok SS-4BW-BW4-C5-SC11 with the handle replaced with a nut for more acurate torque. The stem has a VCR fitting and is but welded to the valve. The mini CF fitting is machined from ASTM A240 with an integral tube stub.



Swagelok has mad available testing documentation and calculations showing justification (using ASME B31.3) for the catalog pressure ratings of their valves and fittings. Some of this documentation is available on JLAB DocuShare. The catalog working pressures shall be accepted without further calculations.

Valve:

Jefferson lab has successfully used the BW series valves in cryogenic conditions for many years. These valves were not cycled in the cryogenic environment only at room temp. The same will be true of the valve on the stem assy. The catalog rating is 1000 psi and shal be accepted.

VCR fitting:

These fittings have been used for many years at JLAB in cryogenic service. The catalog rating of more than 3000 psi working pressure shall be accepted. Nikle gaskets shall be used in this assembly.

$S_a := 20 \cdot ksi$ max allowable stress

$E := 1$ Casting factor

$W := 0.9$ weld eff factor

$d := 0.25 \cdot in$ diameter of tube

$t_{nom} := 0.065 \cdot in$ nominal wall

$t := t_{nom} \cdot (1 - 0.125)$ wall with mill toll

$P_{design} := 1000 \cdot psi$ design pressure of fill station

$Y := 0.4$ factor fot SST

$$t_{min} := \frac{P_{design} \cdot d}{2 \cdot \left(S_a \cdot E \cdot W + P_{design} \cdot Y\right)} = 0.007\ in$$

The tubing is clearly adequate as is the butt joint with a 0.9 eff factor. The joint is 100% in process VT and 100% RT.

CF flange stress:

The entrance window CF flange is a 1.33 in CF design. The material is SST 304. The gasket is aluminum. The flange is 1.33" OD with a 0.75" diameter tube. The thickness of the flange is 0.43". Bolts - 6 each are 8-32 A286 Alloy. To determine the suitability of this flange, we will use the rules of BPVC VIII D1 Appendix 2. Load on single bolt to seat the gasket was **measured** to be 203 lbf.

$N_{bolt} := 6$ The number of bolts is

$F_{bolt} := 180 \cdot lbf$ Load on each bolt

$P := 1000 \cdot psi$ The pressure is

$A := 1.33 \cdot in$ The outer diameter of the flange is

$t_f := 0.235 \cdot in$ The thickness of the flange is

$C := 1.062 \cdot in$ The bolt circle diameter is

$B := 0.125 \cdot in$ The inner or bore diameter is

The diameter at the gasket load reaction is chosen to be the knife edge of the flange

$$G := 0.72 \cdot in$$

The relevant moment arms for this flange are (notation same as Appendix 2)

$$h\_d := \frac{(C-B)}{2} \qquad h\_t := \frac{(C-G)}{2} \qquad h\_g := \frac{(C-G)}{2}$$

The total hydrostatic end force is

$$H := \frac{\pi}{4} \cdot G^2 \cdot P = 407.15 \ lbf$$

$$H\_p := N_{bolt} \cdot F_{bolt}$$

The hydrostatic end force and the inside of the flange is

$$H\_d := \frac{\pi}{4} \cdot B^2 \cdot P = 12.272 \ lbf$$

H_t is the difference between H and H_d

$$H\_t := H - H\_d = 394.879 \ lbf$$

The gasket load is

$$H\_g := H\_p - H = 672.85 \ lbf$$

The total moment acting on the flange

$$M\_o := H\_t \cdot h\_t + H\_d \cdot h\_d + H\_g \cdot h\_g = 188.331 \ in \cdot lbf$$

The ratio of outside to inside diameter is

$$K_a := \frac{A}{B} = 10.64$$

The factor Y is given

$$Y := \frac{1}{K_a - 1} \cdot \left(0.66845 + 5.7169 \cdot K_a^{\ 2} \cdot \frac{\log(K_a)}{K_a^{\ 2} - 1}\right) = 0.684$$

We are assuming that the CF flange meets the conditions of sketch 4a of Fig 2-4
For loose type flanges of this type we may use 2-7 formula (9) for the stress in the flange

$$S\_T := \frac{Y \cdot M\_o}{\left(t_f\right)^2 \cdot B} = 18.655\ ksi$$

This is less than the maximum allowable stress

Flange Bolt Stress

The bolt material is A286. The max allowable stress is unlisted and must be determined:

$S_{ut} := 130 \cdot ksi$ listed UT for A286

$S_y := 85 \cdot ksi$ listed Yield for A286

$S_a := min\left(\frac{1}{3} \cdot S_{ut}, \frac{2}{3} \cdot S_y\right) = 43.333\ ksi$ max allowable stress for A286

The root bolt area is

$$A_{root} := \frac{\pi}{4} \cdot 0.1257^2 \cdot in^2$$

The tensile stress in the bolt is

$$S_t := \frac{F_{bolt} + \frac{H}{N_{bolt}}}{A_{root}} = 19.973\ ksi$$

This is less than the max allowable

Helicoil pull out:

$\tau := 50 \cdot in \cdot lbf$ specified torque on 8-32

$K_f := 0.19$ measured coef

$d := 0.164 \cdot in$ nom diam

$F := F_{bolt} + \frac{\tau}{K_f \cdot d} = (1.785 \cdot 10^3)\ lbf$ Force on bolt

$F_a := \frac{T_{164}}{3} = (2 \cdot 10^3)\ lbf$ Max Allowable force

$F < F_a = 1$ acceptable

Pressure testing:

Proof testing of the entrance and main body was performed in compliance with UG-101.

$S_{ut} := 72 \cdot ksi$ B209 listed minimum ultimate tensile

$S_y := 61 \cdot ksi$ B209 listed minimum yield 0.2%

Note that from material certification (MTR):

Sut = 77.6 ksi and Sy = 65.6

Material is domestic from Kaiser Aluminum Lot#107954B0

$Sut_{MTR} := 77.6 \cdot ksi$ $Sy_{MTR} := 65.6 \cdot ksi$

Entrance window:

$P_{burst} := 2900 \cdot psi$ Lowest burst pressure of entrance window

From UG-101

$$P_a := \frac{P_{burst}}{4} \cdot \frac{S_{ut}}{Sut_{MTR}} = 672.68\ psi$$ maximum design pressure for window

Main body:

$P_{burst} := 3500 \cdot psi$ burst pressure main body

From UG-101

$$P_a := \frac{P_{burst}}{4} \cdot \frac{S_{ut}}{Sut_{MTR}} = 811.856\ psi$$ maximum design pressure for main body

These are higher than the design pressure needed for operations (in beam) but not for filling which is 1000 psi. A removable support has been developed that shall be installed when filling and shipping. The calculations however do indicate that the design is acceptable for filling with a design pressure of 1000 psi. The conservatism built into the Code for proof testing addresses the uncertainties of that method. Proof tests with the shipping covers installed have also been performed an the full assembly (without valve stem). The lowest burst pressure for the assy was 5646 psi.

$P_{burst} := 5000 \cdot psi$ conservative assy burst pressure

$P_a := \frac{P_{burst}}{4} \cdot \frac{S_{ut}}{Sut_{MTR}} = (1.16 \cdot 10^3)\ psi$ maximum design pressure for main body

With the shipping covers even the more conservative proof testing requirements indicate a design pressure above the fill line design pressure.

Cyclic loading screening criteria:

These criteria follow the method given in ASME BPVC VIII D2 5.5.2. The cell will be in place no longer than 12 calender months; 4 months are consecutively inactive. The period is assumed to have shutdowns for holidays and maintenance as well. The target system has a number of solid and 4 gas targets of which only one will be in the beam at any given time. Thus the duty factor for the tritium cell is not 100%. The experiment group has 75 approved PAC days. The maximum current allowed on the tritium cell is 20 μA.

$N_{pac} := 75 \cdot day$ number of PAC days for T2 experiments

$N_{cal} := 2 \cdot N_{pac}$ number of calendar days

The cell will experience several beam trip cycles per hour. The following is a conservative estimate of the number of cycles.

$\varepsilon := 0.33$ duty factor for T2 cell

$R_{trip} := \frac{15}{hr}$ expected number of beam trips per hour

$N_{trip} := N_{cal} \cdot \varepsilon \cdot R_{trip} = 17820$ total estimated number of trips

The beam trip will result in a temperature cycle from the beam heating effects on hte entrance and exit windows and the heat sink temperature maintained at 50K. The target fluid will also cool reducing cell pressure. The beam heating for the steady state was calculated using ANSYS (see TGT-CALC-103-004).

$\Delta T_{trip} := 68\ K$ see TGT-CALC-103-004

The target will operate for 8 to 9 months of calendar time. Assuming 1 ESR trip per month and scheduled downs for holidays and maintenance a conservative estimate for the number of full cryogenic temperature cycles is.

$N_{cryo} := 20$ conservative estimate for number of full temperature cycles from 300K to 50K

$\Delta T_{cryo} := 300\ K - 50\ K = 250\ K$ full temperature cycle allowable stress for AL7075 -T651

$S_{ut} := 72 \cdot ksi$ B209 listed minimum ultimate tensile

$S_y := 61 \cdot ksi$ B209 listed minimum yield 0.2%

Note that from material certification (MTR):

Sut = 77.6 ksi and Sy = 65.6

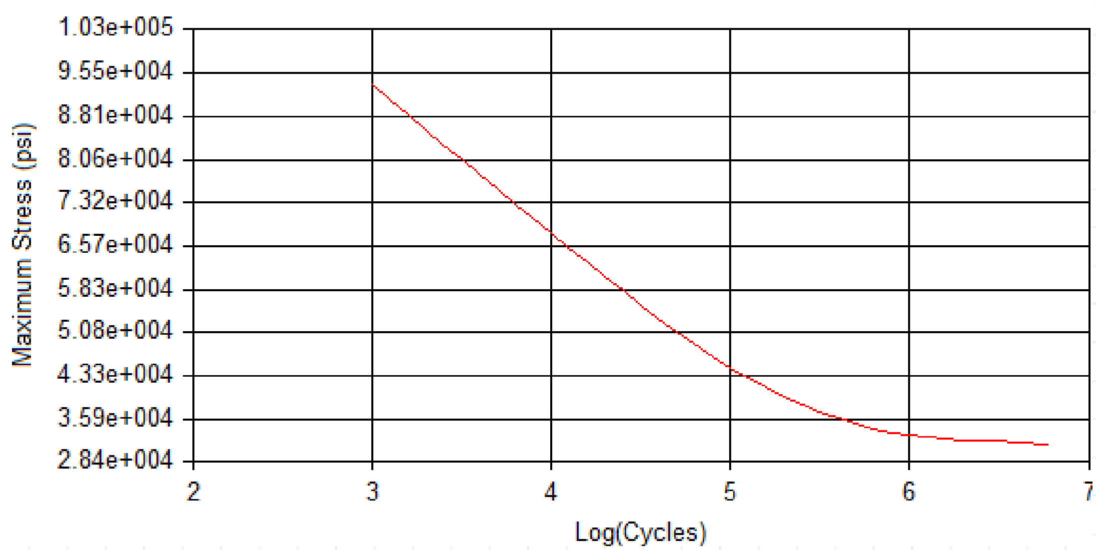
Material is domestic from Kaiser Aluminum Lot#107954B0

$Sut_{MTR} := 77.6 \cdot ksi$ $Sy_{MTR} := 65.6 \cdot ksi$

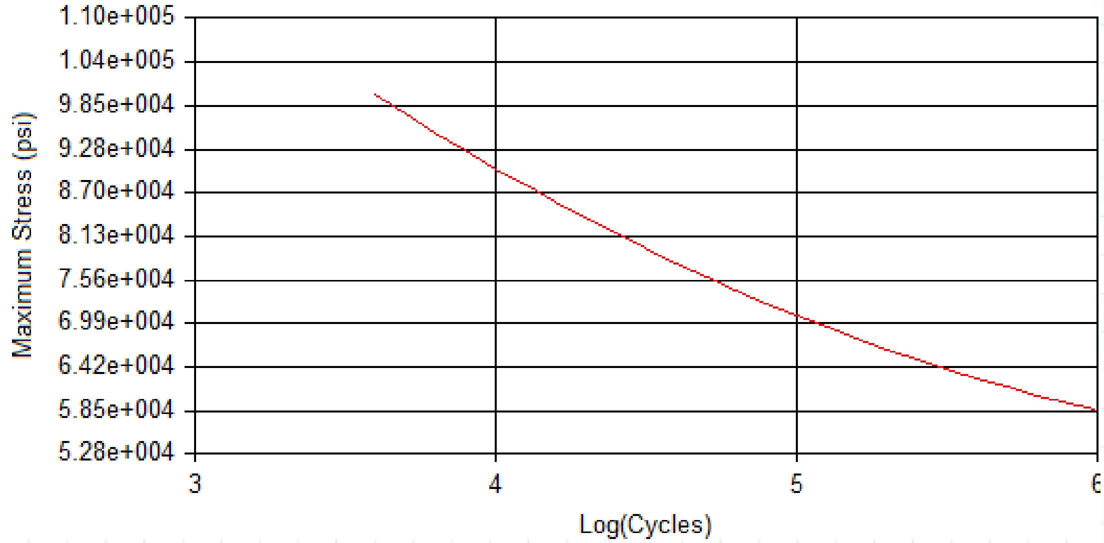
$S := min\left(\frac{1}{3} \cdot S_{ut}, \frac{2}{3} \cdot S_y\right) = 24\ ksi$ max allowable stress AL7075 T651

The following curves are from the MPDB as the fatigue curves for aluminum alloys are not available in Anex 3-F of Div 2.

Fatigue S-N curve for 7075-T6 at 68 F



Fatigue S-N curve for 7075-T6 at -321 F



The S-N curve for 68F shall be used for conservatism and Code compliance.

Screening method from Div 2: 5.5.2.4

Step 1:

$N_{\Delta FP} := 20$ number of cycles for full warm up/cool down

$N_{trip} = 1.782 \cdot 10^4$ number of beam trips this has both temperature and pressure load cycles

step 2:

we have integral construction with loading away from nozzles etc.

from table 5.10

$C_1 := 3$ pressure screening factor

$C_2 := 2$ temp screen factor

step 3:

from S-N curve

$N_{C1S} := 5000$ number of cycles for 3xSa

$N_{\Delta FP} \le N_{C1S}$ true

step 4:

$N_{\Delta P} := N_{trip}$ number of pressure cycles from trips

$Sa_N := 60 \cdot ksi$ Stress for cycles from S-N

$\Delta P_N := 175 \cdot psi$ pressure swing assumed is full load

$P := 1000 \cdot psi$ design pressure

$$R_p := \frac{P}{C_1} \cdot \frac{Sa_N}{S} = 833.333\ psi$$

The following must be true

$$\Delta P_N \le R_p = 1$$

The pressure cycles are not deep enough to require fatigue analysis

step 5:

$\Delta T_N := 50 \cdot K$ max temp difference

$T_{mean} := 170 \cdot K$ mean temp between points

$E_{ym} := 1. \cdot 10^7 \cdot psi$ Ey at 74K

$S_a := 88 \cdot ksi$ allow stress for N cycles

$\alpha := 1.8 \cdot 10^{-5} \cdot \frac{1}{K}$ coef therm expan

$$\Delta T_{max} := \frac{S_a}{C_2 \cdot E_{ym} \cdot \alpha} = 244.444\ K$$ Max temp difference for startup etc.

$$\Delta T_N \le \Delta T_{max} = 1$$

step 6

$$\Delta T_R := 68 \cdot K$$ max temp difference

$$T_{mean} := 74 \cdot K$$ mean temp between points

$$E_{ym} := 1.2 \cdot 10^7 \cdot psi$$ Ey at 74K

$$S_a := 60 \cdot ksi$$ allow stress for N cycles

$$\alpha := 1.8 \cdot 10^{-5} \cdot \frac{1}{K}$$ coef therm expan

$$\Delta T_{max} := \frac{S_a}{C_2 \cdot E_{ym} \cdot \alpha} = 138.889\ K$$ Max temp difference for startup etc.

$$\Delta T_R \le \Delta T_{max} = 1$$

step 7 and step 8

these do not apply

Based on the screening analysis, a detailed fatigue analysis is not required.