Project: E12-14-012 Argon/Ti

Tittle: General target cell calculations, fatigue screening, and testing analysis

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Code(s) of Record:
ASME B31.3 2014

Reference Codes and Sources:

- ASME BPVC VIII D1,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
- TGT-CALC-16-005 Elastic plastic analysis of cell
- TGT-CALC-16-004 Thermal analysis of cell
- Material properties are from MPDB

Description:
General target cell calculations

Reference Drawing(s):

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

Notes:

1. For comparison functions e.g. "less than" x<y if true MathCad returns a 1.
2. A thermal-stress analysis of the cell in the beam using ANSYS is given in TGT-CALC-16-004. This analysis is not required by Code due to the screening analysis given herein.

General:

Temperatures:

$T_{op} := 130 \cdot K$ operating temperature

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

Pressures:

$P_{fill} := 600 \cdot psi$ assumed fill pressure at 300K

Helicoils:

The tensile strength of a single threaded insert assy is given in Helicoil Tech Bulletin 68-2 for assemblies made in various Al alloys. The assemby is analyzed by considering Al 2024 -T6 listed in the bulletin and comparing to 7075. The aluminum 7075-T6 base metal has a higher shear strength than that of Al 2024. The strengths given below are for inserts that are 2x nom bolt diameter in length in Al 2024 are conservative values for assemblies in 7075.

$T_{164} := 6000 \cdot lbf$ 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 the cell block to heat sink interface must be modeled. The heat generated by the beam in the cell is conservatively estimated at 110W. 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 thermal 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 cell:

The design pressure of the 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 in ASME Codes; 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 ASTM B209. 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 tip section. The design wall thickness is 0.018 for the cylinder and 0.011 for the endcap. The machine tolerance is +0.002/- 0.001 inch which is verified by measurement for each cell on the entire cell 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

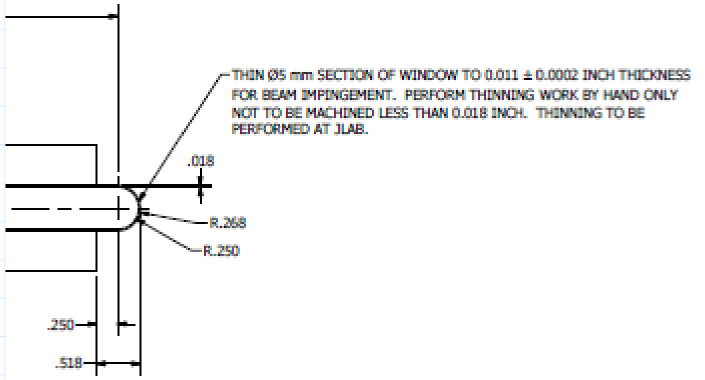
The maximum design pressure for the cylinder part of the main body is:

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

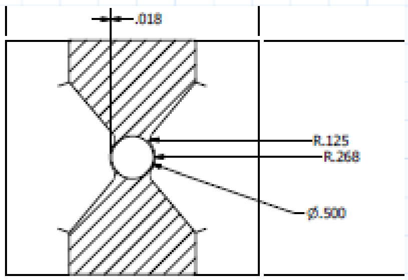
The endcap may be treated similarly; From ASME BPVC VIII D1 UG-32 for hemispherical heads

$$P_{cap} := \frac{2 \cdot S_a \cdot E \cdot t_{cap}}{0.5 \cdot D_{in} + 0.2 \cdot t_{cap}} = \left(1.905 \cdot 10^3\right)\ psi$$

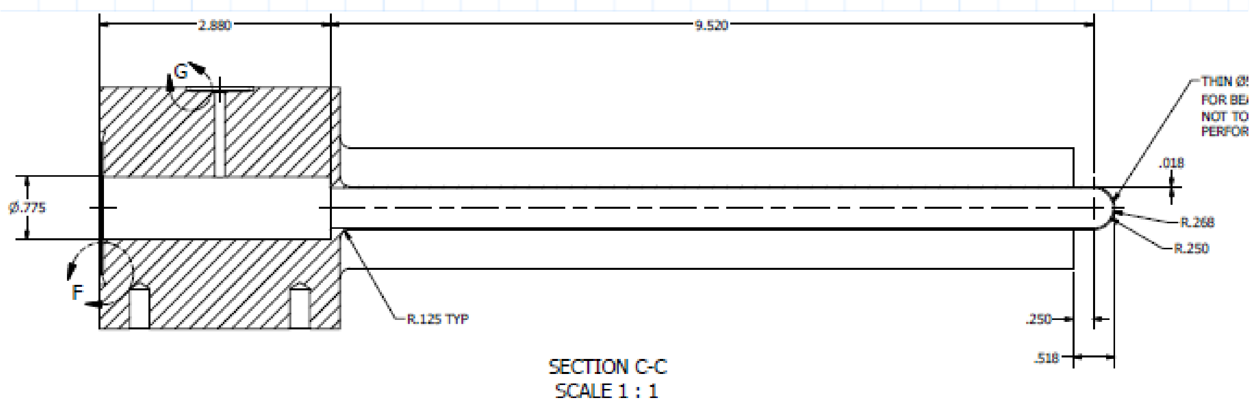
Note that the thickness of the head increases to the wall thickness of the cylinder. See figures below for ref.



section view of beam exit region of main body



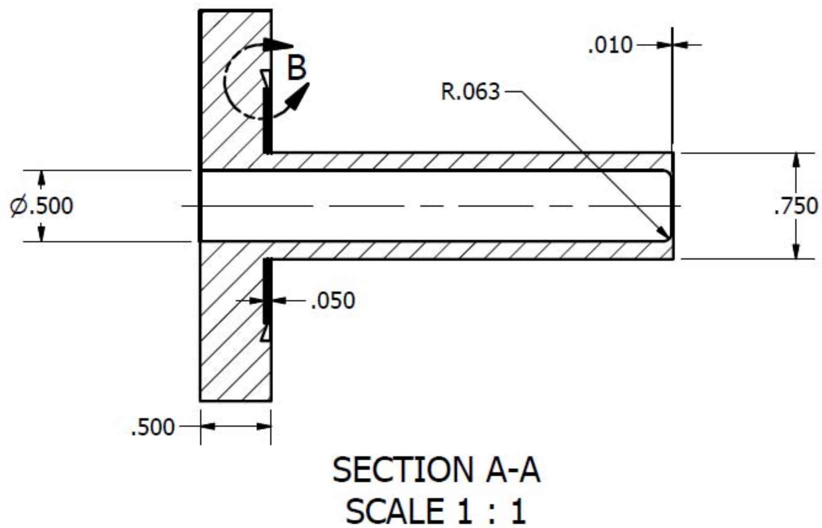
section view of main body from downstream viewpoint

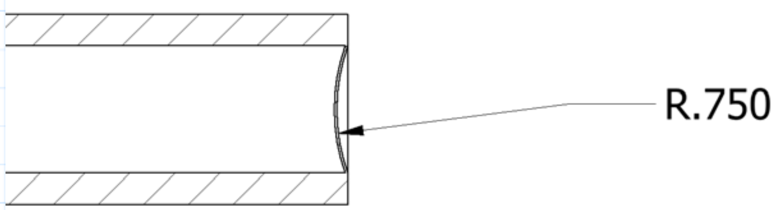


Section view of main body from side

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 more than 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 0.75 inch as shown in the second fig.





window after forming

Method of analysis for the thin section of the cell is from VIII UG-32

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

Thickness after forming $t := t_{pre} \cdot 0.97 = 0.0097\ in$

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

$R_h := 0.75 \cdot in$ inner radius of head

$E := 1$ quality factor for machined window

maximum design pressure:

$$P_{win} := \frac{2 \cdot S_a \cdot E \cdot t}{R_h + 0.2 \cdot t} = 619.198\ psi$$

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

$\nu_{trn} := 0.33$ poissons ratio

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

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

$R_0 := 0.5 \cdot D_0$ outer 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

$$M_x := \frac{L}{\sqrt{R_0 \cdot t}} = 15.376$$

elastic moment

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

$$C_h = 0.092$$

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

$$F_{ic} := \begin{cases} S_y & \text{if } \dfrac{F_{he}}{S_y} \geq 2.439 \\ 0.7 \cdot S_y \cdot \left(\dfrac{F_{he}}{S_y}\right)^{0.4} & \text{else if } 0.552 < \dfrac{F_{he}}{S_y} < 2.439 \\ F_{he} & \text{else} \end{cases}$$

Predicted buckling stress

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

step 4: design factor

$$FS := \begin{cases} 2 & \text{if } F_{ic} \leq 0.55 \cdot S_y \\ 2.407 - 0.741 \cdot \left(\dfrac{F_{ic}}{S_y}\right) & \text{else if } 0.55 \cdot S_y < F_{ic} < S_y \\ 1.667 & \text{else} \end{cases}$$

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 (not thin window)

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

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 previously. 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 0.495". 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)

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

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

$$M\_o := H\_t \cdot h\_t + H\_d \cdot h\_d + H\_g \cdot h\_g = \left(1.346 \cdot 10^3\right)\ in \cdot lbf$$

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

$$S\_T := \frac{Y \cdot M\_o}{\left(t_f\right)^2 \cdot B} = 12.358\ ksi$$

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

Flange Bolt Stress

The bolt material is ASTM A453 Grade 660 Class D Alloy A286. The max allowable stress is unlisted (in ASME) and must be determined using ASTM A453:

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

$S_y := 105 \cdot ksi$ 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

$F := F_{bolt} + \frac{H}{N_{bolt}} + \frac{\tau}{K_f \cdot d} = \left(3.644 \cdot 10^3\right)\ lbf$ total helicoil load

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

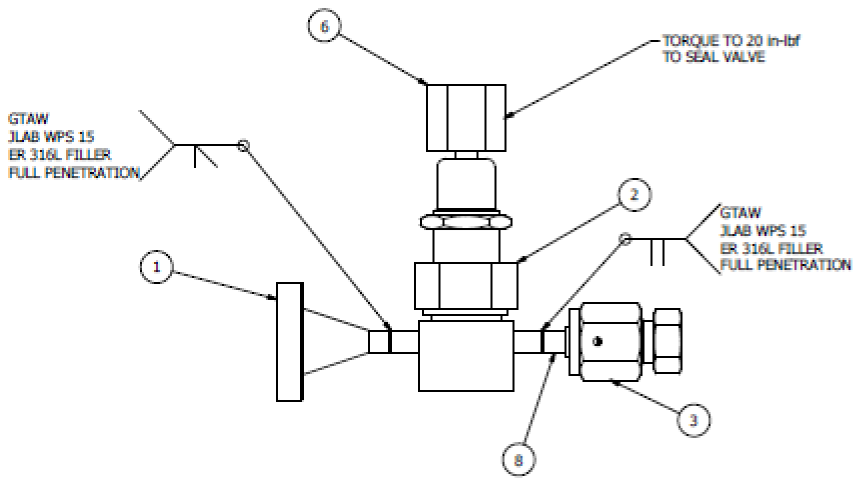
$F < F_a$ acceptable

Using the guidance of Appendix S it is allowed for that the bolt may be tightend beyond the max allowable from Appendix 2 to ensure a sound connection. The proof strength (yield) is not exceeded in this application.

$S_{full} := \frac{F}{A_{root}} = 110.395\ \boldsymbol{ksi}$ Which is less than S_y

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 butt welded to the valve. The mini CF fitting is machined from ASTM A276 with an integral tube stub.



Swagelok has made 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 are cycled/actuated 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 shall be accepted. Full valve part number is SS-4BG-BW4-C5-SC11.

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 and shall be accepted. Note nickle gaskets shall be used in this assembly.

The flange is custom made and has an integral hub. Modeling the conical shaped hub as a 1/4 inch tube is conservative.

$S_a := 20 \cdot ksi$ 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 exam requirements joint is 100% in process VT and 100% RT.

CF flange stress:

The fill valve assy 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 0.285". 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 160 lbf.

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

$F_{bolt} := 160 \cdot lbf$ Load on each bolt

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

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

$t_f := 0.285 \cdot in$ 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)

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

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

$$H\_g := H\_p - H = 552.85 \ lbf$$

The total moment acting on the flange

$$M\_o := H\_t \cdot h\_t + H\_d \cdot h\_d + H\_g \cdot h\_g = 226.549 \ in \cdot lbf$$

The ratio of outside to inside diameter is

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

The factor Y is given

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

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

$$S\_T := \frac{Y \cdot M\_o}{\left(t_f\right)^2 \cdot B} = 15.257\ ksi$$

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

$S_y := 105 \cdot ksi$ 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

$$S_t := \frac{F_{bolt} + \frac{H}{N_{bolt}}}{A_{root}} = 18.361\ ksi$$

This is less than the max allowable from appendix 2

Helicoil pull out:

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

$K_f := 0.19$ measured coef for properly lubed screw

$d := 0.164 \cdot in$ nom diam

$F := F_{bolt} + \frac{H}{N_{bolt}} + \frac{\tau}{K_f \cdot d} = \left(1.832 \cdot 10^3\right) lbf$ Force on bolt

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

$F < F_a$ 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 ksi

Material is domestic from Kaiser Aluminum Lot#107954B0

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

Entrance window:

$P_{burst} := 2950 \cdot psi$ Lowest burst pressure of entrance window

From UG-101

$$P_a := \frac{P_{burst}}{4} \cdot \frac{S_{ut}}{Sut_{MTR}} = 684.278\ psi$$ maximum design pressure for window

Main body:

$P_{burst} := 3500 \cdot psi$ lowest 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

Cyclic loading screening criteria:

These criteria follow the method given in ASME BPVC VIII D2 5.5.2. The target system is expected to be used for (9) days total. The maximum current allowed on the tritium cell is 25 μA.

$N_{cal} := 9 \cdot day$ number of callendar days for run

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

$\varepsilon := 1.0$ duty factor for T2 cell

$R_{trip} := \frac{20}{hr}$ expected number of beam trips per hour

$N_{trip} := N_{cal} \cdot \varepsilon \cdot R_{trip} = 4320$ total estimated number of trips

The beam trip will result in a temperature cycle from the beam heating effects on the entrance and exit windows and the heat sink temperature maintained at 130K. The target fluid will also cool reducing cell pressure. The beam heating for the steady state was calculated using ANSYS (see TGT-CALC-16-004).

$\Delta T_{trip} := (307 - 130)\ K$ see TGT-CALC-16-004

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.

$N_{cryo} := 3$ conservative estimate for number of full temperature cycles from 300K to 130K

$\Delta T_{cryo} := 300\ K - 130\ K = 170\ K$ 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

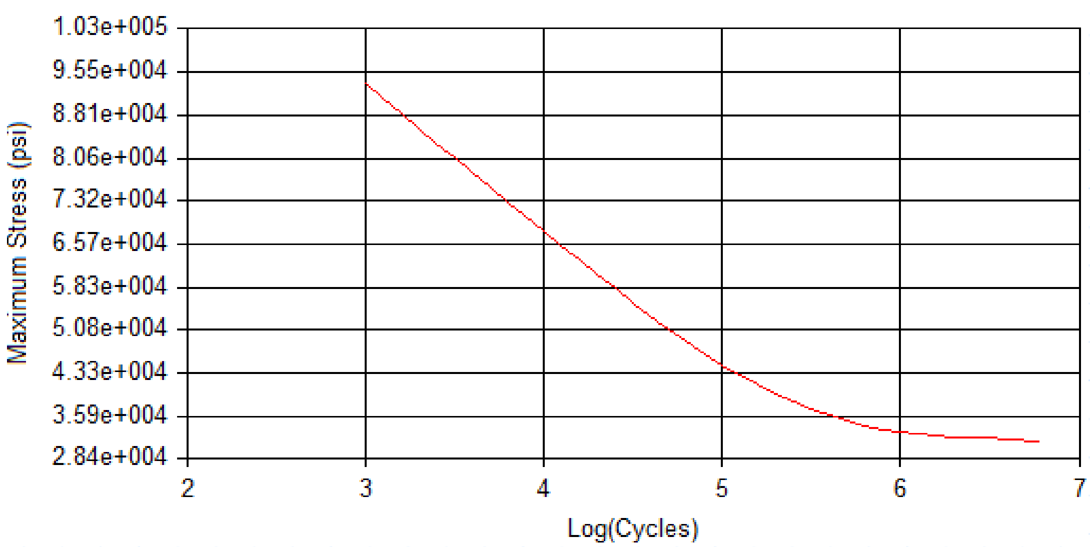
Material is domestic from Kaiser Aluminum Lot#107954B0

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

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



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:

$N_{\Delta FP} := 5$ number of cycles for full warm up/cool down

$N_{trip} = 4.32 \cdot 10^3$ 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

$\Delta P_N := 200 \cdot psi$ conservative pressure swing assumed

$P := 600 \cdot psi$ design pressure

$$R_p := \frac{P}{C_1} \cdot \frac{Sa_N}{S} = 500\ psi$$

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:

$\Delta T_N := (300 - 130) \cdot K$ max temp difference very conservative

$N_{\Delta TN} := N_{\Delta FP} = 5$ number of cycles for warm/cool down

$T_{mean} := 215 \cdot K$ mean temp between points

$E_{ym} := 1.1 \cdot 10^7 \cdot psi$ Ey at 220K

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

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

$\Delta T_{max} := \frac{S_a}{C_2 \cdot E_{ym} \cdot \alpha} = 222.222\ K$ Max temp difference for startup etc.

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

step 6

$\Delta T_R := 50 \cdot K$ max temp difference in metal from TGT-CALC-16-004

$N_{\Delta TR} := N_{trip} = 4.32 \cdot 10^3$ number of cycles

$T_{mean} := 219 \cdot K$ mean temp between points

$E_{ym} := 1.1 \cdot 10^7 \cdot psi$ Ey at 220K

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

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

$\Delta T_{max} := \frac{S_a}{C_2 \cdot E_{ym} \cdot \alpha} = 151.515\ K$ Max temp difference for startup etc.

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

step 7 and step 8

these do not apply

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

Conclusions:

The load conditions are difficult to determine acurately for beam on target case. Therefore, conservative estimates of the cell pressure during beam operations have been made.
Based on the above analysis and testing a design pressure of $P_{design} := 600 \cdot \mathbf{psi}$ is considered acceptable and conservative for the normal operation case. The cyclic screening analysis shows that cyclic loading need not be fully analyzed.

Given the low stored energy contained in the cell (see TGT-CALC-16-002) full Code compliance is not required by JLAB policy.