

**Active volume dimensions**, from earlier analyses:

$$r_{Xe} = 0.53 \text{ m} \quad l_{Xe} = 1.3 \text{ m}$$

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.

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

**Pressure Vessel Inner Radius:**

$$R_{i\_pv} := r_{Xe} + t_{fc} + t_{Cu} \quad R_{i\_pv} = 0.68 \text{ m}$$

**Temperatures:**

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

**Vessel wall thicknesses**

**Material:**

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

**Design Rules: division 1**

316Ti is not an allowed material under section VIII, division 2, so we must use **division 1** rules. The saddle supports are however, designed 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:

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

Youngs modulus

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

**color scheme for this document**

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

$$xx := 1 \quad xx > 0 = 1$$

Choose material:

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

Maximum Operating Pressure (MOP), gauge:

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

Minimum Pressure, gauge:

$$P_{min} = -1.5 \text{ bar} \quad \text{the extra 0.5 atm maintains an upgrade path to a water or scintillator tank}$$

Maximum allowable pressure, gauge (from LBNL Pressure Safety Manual, PUB3000)  
at a minimum, 15% over max operating pressure; this is design pressure at LBNL:

$$MAWP_{pv} := 1.1 MOP_{pv} \quad MAWP_{pv} = 15.4 \text{ bar}$$

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

$$E_w := 1 \quad \text{<---note: it is not clear if butt welds using ceramic backing strips will qualify as type (1)}$$

Minimum wall thickness is then:

$$t_{pv\_d1\_min\_ip} := \frac{MAWP_{pv} \cdot (R_{i\_pv})}{S_{max} \cdot E_w - 0.6 \cdot MAWP_{pv}} \quad t_{pv\_d1\_min\_ip} = 7.75 \text{ mm}$$

We set wall thickness to be:

$$t_{pv} := 10 \text{ mm}$$

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

### Maximum Allowable External Pressure

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

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

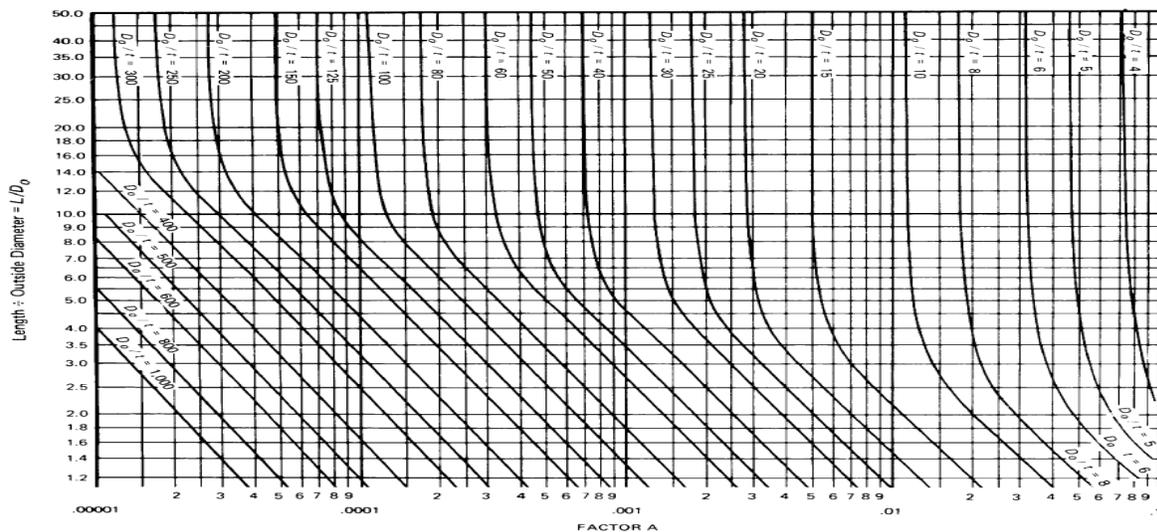
The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

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

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

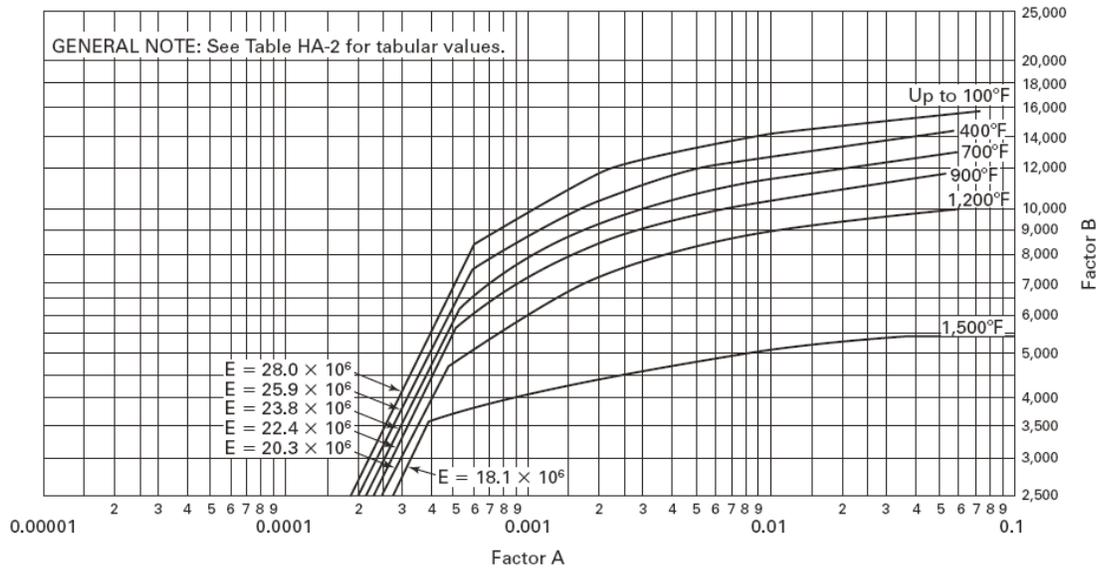
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$$A := 0.0005$$

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



$$B := 6200 \text{ psi} \quad @ 400 \text{ F}$$

The maximum allowable working external pressure is then given by :

$$P_a := \frac{4B}{3 \left( \frac{2R_{i\_pv}}{t_{pv}} \right)} \quad P_a = 4.1 \text{ bar} \quad -P_{\min} = 1.5 \text{ bar}$$

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

**Flange thickness:**

inner radius                      max. allowable pressure  
 $R_{i\_pv} = 0.68 \text{ m}$                $MAWP_{pv} = 15.4 \text{ bar}$  (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 usin a Helicoflex 5mm gasket (smallest size possible) specially designed with a maximum sealing ofrcce 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 2B:

Maximum allowable design stress for flange

$S_f := S_{\text{max\_316Ti\_div1}}$                        $S_f = 137.9 \text{ MPa}$

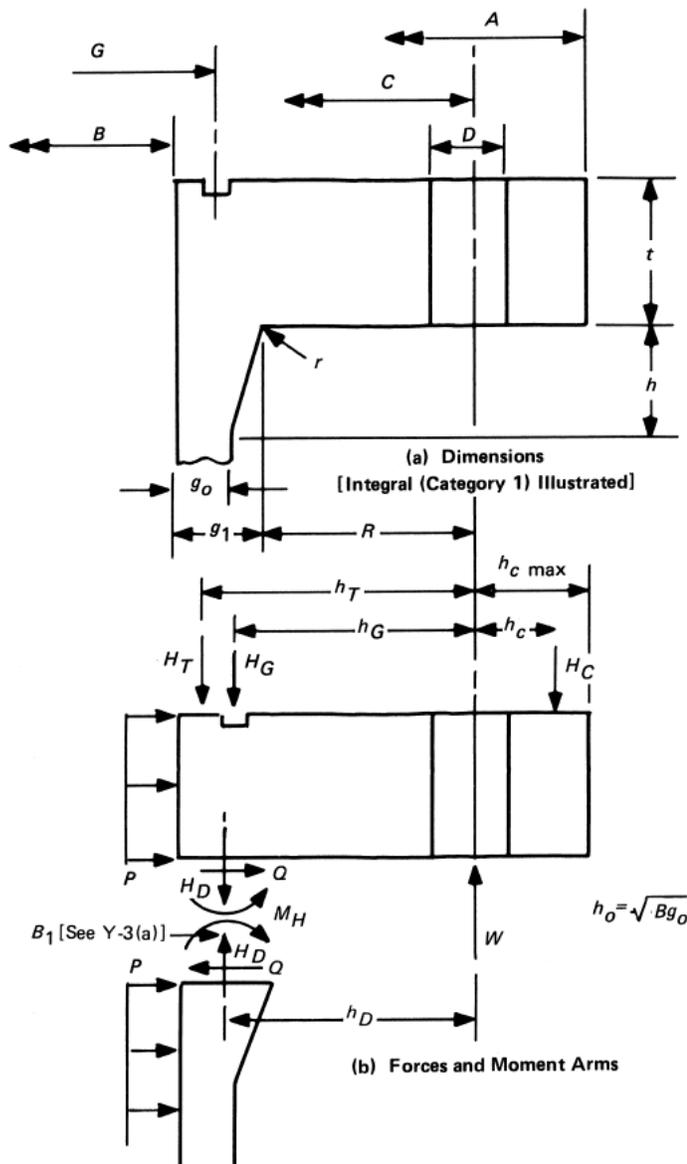
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

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

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

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:

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

Flange OD

$$A := 1.47 \text{ m}$$

Flange ID

$$B := 2R_{i_{pv}} \quad B = 1.36 \text{ m}$$

define:

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

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

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

Gasket dia

$$G := 2(R_{i\_pv} + .75\text{cm}) \quad G = 1.375\text{m}$$

Force of Pressure on head

$$H := .785G^2 \cdot \text{MAWP}_{pv} \quad H = 2.316 \times 10^6 \text{N}$$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70  $F = \sim 5$  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:

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

$$\text{for gasket diameter} \quad D_j := G \quad D_j = 1.375\text{m}$$

Force is then either of:

$$F_m := \pi D_j \cdot Y_1 \quad \text{or} \quad F_j := \pi \cdot D_j \cdot Y_2$$

$$F_m = 3.024 \times 10^5 \text{N} \quad F_j = 9.503 \times 10^5 \text{N}$$

Helicoflex recommends using Y2 ( 220 N/mm) for large diameter seals, even though for small diameter one can use the greater of Y1 or  $Y_m = (Y_2 \cdot (P/P_u))$ . For 15 bar Y1 is greater than  $Y_m$  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.

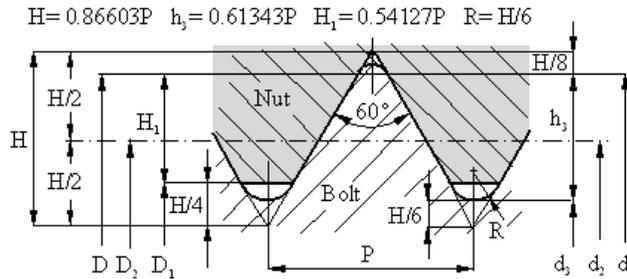
Start by making trial assumption for number of bolts, nominal bolt dia., pitch, and bolt hole dia D,

$$n := 120 \quad d_b := 16\text{mm}$$

Choosing ISO fine thread, with pitch; thread depth is:

$$p_t := 1.0\text{mm} \quad h_3 := .614 \cdot p_t$$

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



metric screw threads ISO 724 (DIN 13 T1)

Nominal diameter d = D	Pitch P	root radius r	pitch diameter d2=D2	minor diameter d3	D1	thread height h3	H1	drill diameter mm
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
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
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

<--- use h3 for 1.0 mm pitch

<--- use H1 for 1.5mm pitch

Bolt root dia. is then:

$$d_3 := d_b - 2h_3 \quad d_3 = 14.772 \text{ mm}$$

Total bolt cross sectional area:

$$A_b := n \cdot \frac{\pi}{4} d_3^2 \quad A_b = 205.66 \text{ cm}^2$$

Check bolt to bolt clearance, here we use narrow thick washers (28mm OD) under the 24mm 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) :

$$\pi C - 2.0n \cdot d_b \geq 0 = 1$$

Check nut, washer clearance:  $OD_w := 2d_b$  this covers the nut width across corners

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

Flange hole diameter, minimum for clearance :

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

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

$$H_1 := .812 \text{ mm} \quad \text{http://www.tribology-abc.com/calculators/metric-iso.htm}$$

$$d_{\min\_20\_1.5} := 20\text{mm} - 2 \cdot H_1 \quad d_{\min\_20\_1.5} = 1.838\text{cm} \quad d_{\min\_20\_1.5} \geq D_{\min} = 1$$

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

Set:

$$D_t := D_e$$

$$D_t > D_{\min} = 1$$

note: actual clearance holes will be 18mm but some holes will be tapped for M20x1 instead (19.4mm avg dia), so as to allow bolting up the head retraction fixture. These holes must contain thin (0.5mm max thk.) metal sleeves when not in use to avoid thread interference with flange bolts.

Compute Forces on flange:

We use a unit gasket seating force of Y1 above

$$H_G := F_m \quad H_G = 3.024 \times 10^5 \text{ N}$$

$$h_G := 0.5(C - G) \quad h_G = 2.75 \text{ cm}$$

$$H_D := .785 \cdot B^2 \cdot \text{MAWP}_{pv} \quad H_D = 2.266 \times 10^6 \text{ N}$$

$$h_D := D_t \quad h_D = 1.919 \text{ cm}$$

$$H_T := H - H_D \quad H_T = 5.027 \times 10^4 \text{ N}$$

$$h_T := 0.5(C - B) \quad h_T = 35 \text{ mm}$$

Total Moment on Flange

$$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad M_P = 5.356 \times 10^4 \text{ J}$$

## Appendix Y Calc

$$P := \text{MAWP}_{pv} \quad P = 1.561 \times 10^6 \text{ Pa}$$

Choose values for plate thickness and bolt hole dia:

$$t := 3.25\text{cm} \quad D := D_t \quad D = 1.919\text{cm}$$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \quad \beta = 1.022 \quad h_C := 0.5(A - C) \quad h_C = 0.02 \text{ m}$$

$$a := \frac{A + C}{2B_1} \quad a = 1.058 \quad AR := \frac{n \cdot D}{\pi \cdot C} \quad AR = 0.513 \quad h_0 := \sqrt{B \cdot g_0}$$

$$r_B := \frac{1}{n} \left( \frac{4}{\sqrt{1 - AR^2}} \operatorname{atan} \left( \sqrt{\frac{1 + AR}{1 - AR}} \right) - \pi - 2AR \right) \quad r_B = 6.212 \times 10^{-3} \quad h_0 = 0.117 \text{ m}$$

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

$$\text{since } \frac{g_1}{g_0} = 1 \quad \text{these values converge to} \quad F := 0.90892 \quad V := 0.550103$$

## 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)

$$J_S := \frac{1}{B_1} \left( \frac{2 \cdot h_D}{\beta} + \frac{h_C}{a} \right) + \pi r_B \quad J_S = 0.061 \quad J_P := \frac{1}{B_1} \left( \frac{h_D}{\beta} + \frac{h_C}{a} \right) + \pi \cdot r_B \quad J_P = 0.047$$

$$(5a) \quad F' := \frac{g_0^2 (h_0 + F \cdot t)}{V} \quad F' = 2.657 \times 10^{-5} \text{ m}^3 \quad M_P = 5.356 \times 10^4 \text{ N}\cdot\text{m}$$

$$A = 1.47 \text{ m} \quad B = 1.36 \text{ m}$$

$$K := \frac{A}{B} \quad K = 1.081 \quad Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 12.883$$

$$f := 1$$

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

$$l := 2t + t_s + 0.5d_b \quad l = 7.3 \text{ cm}$$

sec Y-6.2(a)(3)

Elastic constants

<http://www.hightempmetals.com/tech-data/hitemplnconel718data.php>

$$(7-13) \quad E := E_{SS\_aus} \quad E_{Inconel\_718} := 208 \text{ GPa} \quad E_{bolt} := E_{Inconel\_718}$$

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \quad M_S = -1.9 \times 10^3 \text{ N}\cdot\text{m}$$

$$\theta_B := \frac{5.46}{E \cdot \pi t^3} (J_S \cdot M_S + J_P \cdot M_P) \quad \theta_B = 6.309 \times 10^{-4} \quad E \cdot \theta_B = 121.763 \text{ MPa}$$

$$H_C := \frac{M_P + M_S}{h_C} \quad H_C = 2.585 \times 10^6 \text{ N}$$

$$W_{m1} := H + H_G + H_C \quad W_{m1} = 5.204 \times 10^6 \text{ N}$$

### Compute Flange and Bolt Stresses

$$\sigma_b := \frac{W_{m1}}{A_b} \quad \sigma_b = 253 \text{ MPa} \quad S_b = 255.1 \text{ MPa}$$

$$r_E := \frac{E}{E_{bolt}} \quad r_E = 0.928$$

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^2 \cdot (M_P + M_S)}{a \cdot t^3 \cdot r_E \cdot B_1} \quad S_i = 245.9 \text{ MPa}$$

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2 (\pi \cdot C - n \cdot D)} \quad S_{R\_BC} = 134.1 \text{ MPa} \quad S_f = 137.9 \text{ MPa}$$

$$S_{R\_ID1} := - \left( \frac{2F \cdot t}{h_0 + F \cdot t} + 6 \right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_{R\_ID1} = 2.622 \text{ MPa}$$

$$S_{T1} := \frac{t \cdot E \cdot \theta_B}{B_1} + \left( \frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8 \right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_{T1} = 1.49 \text{ MPa}$$

$$S_{T3} := \frac{t \cdot E \cdot \theta_B}{B_1} \quad S_{T3} = 2.889 \text{ MPa}$$

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left( \frac{g_1}{g_0} \right)^2 B_1 \cdot V} \quad S_H = 20.705 \text{ MPa}$$

**Y-7 Flange stress allowables:**  $S_f = 137.9 \text{ MPa}$

- (a)  $\sigma_b < S_b = 1$
- (b) (1)  $S_H < 1.5S_f = 1$   $S_n$  not applicable  
 (2) not applicable
- (c)  $S_{R\_BC} < S_f = 1$   
 $S_{R\_ID1} < S_f = 1$
- (d)  $S_{T1} < S_f = 1$   
 $S_{T3} < S_f = 1$
- (e)  $\frac{S_H + S_{R\_BC}}{2} < S_f = 1$   
 $\frac{S_H + S_{R\_ID1}}{2} < S_f = 1$
- (f) not applicable

Bolt force

$$F_{\text{bolt}} := \sigma_b \cdot .785 \cdot d_b^2 \quad F_{\text{bolt}} = 1.143 \times 10^4 \text{ lbf}$$

Bolt torque required, minimum:

$$T_{\text{bolt\_min}} := 0.2F_{\text{bolt}} \cdot d_b \quad T_{\text{bolt\_min}} = 162.7 \text{ N} \cdot \text{m} \quad T_{\text{bolt\_min}} = 120 \text{ lbf} \cdot \text{ft} \quad \text{for pressure test use 1.5x this value}$$

This is the minimum amount of bolt preload needed to assure joint does not open under pressure. and additional amount for bolt preload is needed to maintain a minimum shear resistance to assure head does not slide downward from weight. 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, in which, for properly designed joints where connection stiffness is much greater than bolt stiffness,

Additional Calculations:

Shear stress in inner flange lip from shield (could happen only if flange bolts come loose, are left loose, or if joint