Flange thickness, head to vessel main flanges, using Movesa suggested design of Nov 26, 2012:

inner radius $R_{i\_pv} = 0.68\,m$ max. allowable pressure $P = 15.4\,bar$ (gauge pressure)

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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 for a metal C-ring gasket seal will be inconel 718, (UNS N77180) as this is the highest strength non-corrosive material allowed for bolting. For O-ring sealing we can use 304 bolts, temper B. We design the flanges for both cases, using the parallel calculation mode of MathCAD in which the possible values for a parameter are expressed as a matrix. Calculations are then performed in parallel for each row index. Where necessary (multiple vectors in an expression) an arrow over the expression enforces this parallel calculation mode.

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2A (division 1 only):

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti\_div1}$ $S_f = 137.9\,MPa$ $S_f = 2 \times 10^4\,psi$

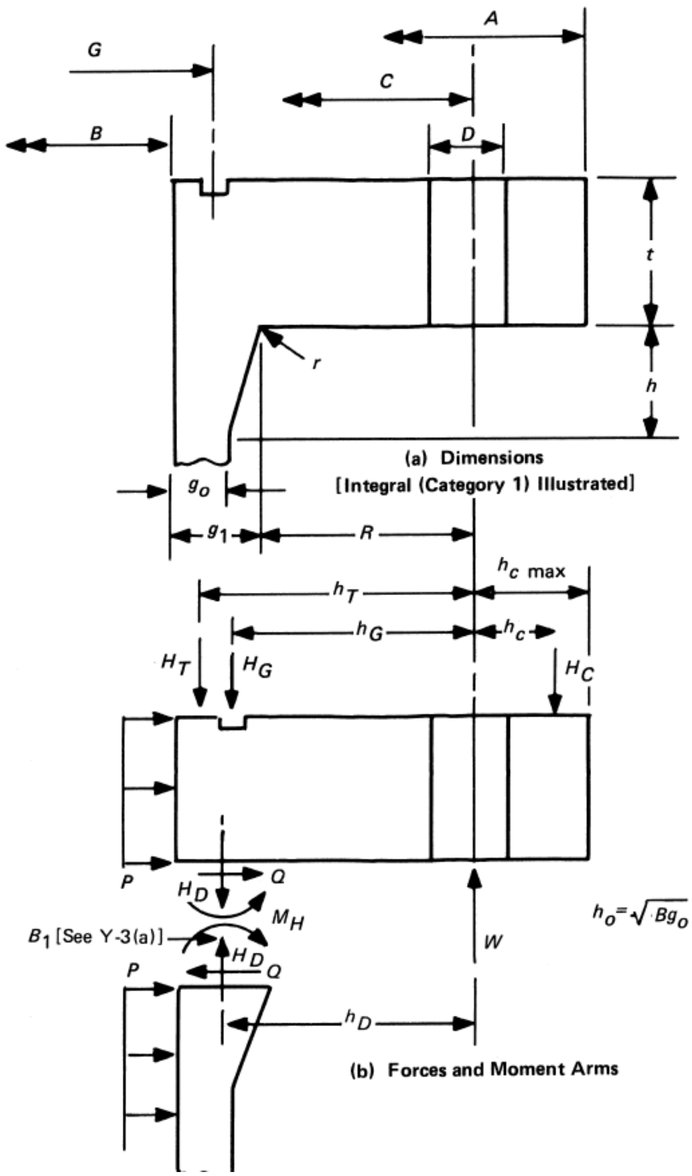
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

Inconel 718 (UNS N07718) $S_{max\_N07718} := 37000psi$ $S_{max\_SA\_574} := 33800psi$ or bolts => 5/8 in

$S_b := \begin{pmatrix} S_{max\_SA\_574} \\ S_{max\_N07718} \end{pmatrix}$ $S_b = \begin{pmatrix} 233 \\ 255.1 \end{pmatrix} MPa$ $S_{max\_316\_2} := 22000psi$ for bolts less than 3/4 in

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)

$g_0 := t_{pv}$ $g_1 := t_{pv}$ $g_0 = 10\,mm$ $g_1 = 10\,mm$

corner radius:

$r_1 := \max(.25g_1, 5mm)$ $r_1 = 5\,mm$

note: corner radius is not included in g_1 as shown in top drawing of Fig. Y-3.2, above

Flange OD

$A := \begin{pmatrix} 1.47 \\ 1.6 \end{pmatrix} m$

Steel bolts (A-574 SHCS, O-ring seals

Inconel 718 bolts, helicoflex low force C-ring

Flange ID

$B := 2R_{i\_pv}$ $B = 1.36\,m$

define:

$B_1 := B + g_1$ $B_1 = 1.37\,m$

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

$C := 1.43 \cdot m$

Appendix Y refers to Appendix 2-5 (c) regarding how to treat self-energizing gaskets such as O-rings. Paragraph 2-5 (c)(3) states that for self-energizing gaskets, gasket compression load H_P is to be considered as = 0 (except for certain special types not applicable here) and that the bolt load W_{m1} be computed using the outer gasket diameter. For Helicoflex average diameter is used:

Gasket width

$b := 5mm$

Gasket diameter:

$G := \begin{pmatrix} 1.373 \\ 1.3755 \end{pmatrix} m$ O-ring mean radius as measured in CAD model: $68.65 \cdot 2 = 137.3$

Force of Pressure on head

$H := .785 G^2 \cdot P$ $\quad H = \begin{pmatrix} 2.31 \times 10^6 \\ 2.318 \times 10^6 \end{pmatrix} N$ $\quad H = \begin{pmatrix} 2.355 \times 10^5 \\ 2.364 \times 10^5 \end{pmatrix} kgf$

note: 1 bar is slightly larger than 0.1MPa- PVElite is underestimating pressure by this difference

Sealing force, per unit length of circumference:

for 4.78mm C-ring, M surface hardness:

$Y_2 := \begin{pmatrix} 0 \\ 65 \end{pmatrix} \frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

Gasket Load:

$H_G := \overrightarrow{(\pi G \cdot Y_2)}$ $\quad H_G = \begin{pmatrix} 0 \\ 2.809 \times 10^5 \end{pmatrix} N$

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

$n := 103$ $\quad d_b := 24mm$

maximum number of bolts possible, using narrow washers (OD=2x bolt dia):

$n_{max} := \mathrm{trunc}\left(\frac{\pi C}{2 d_b}\right)$ $\quad n_{max} = 93$ $\quad n_{max} > n = 0$

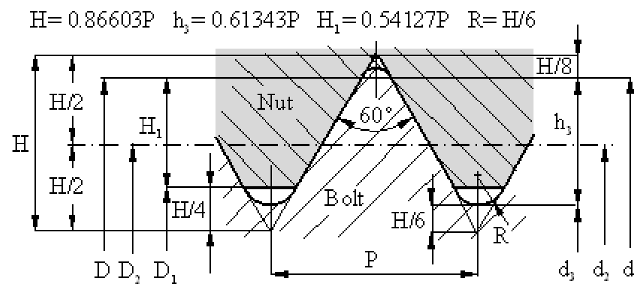
Check strength restriction: d_b =>5/8in

$d_b \geq 0.625in = 1$

Choosing ISO fine thread for CS, thread depth is:

$$p_t := \begin{pmatrix} 3 \\ 3 \end{pmatrix} mm \qquad h_3 := .6134 \cdot p_t \qquad h_3 = \begin{pmatrix} 1.84 \\ 1.84 \end{pmatrix} mm$$

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:

$$d_3 := d_b - 2h_3 \qquad d_3 = \begin{pmatrix} 20.3196 \\ 20.3196 \end{pmatrix} mm$$

Total bolt cross sectional area:

$$A_b := n \cdot \frac{\pi}{4} d_3^{\,2} \qquad A_b = \begin{pmatrix} 334.008 \\ 334.008 \end{pmatrix} cm^2$$

Check bolt to bolt clearance, here we use narrow thick washers (2x bolt dia) under the 24mm wide (flat to flat) 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) :

$$d_w := 2d_b \qquad d_w = 48\,mm$$

$\pi C - n \cdot d_w \geq 0 = 0$ actual bolt to bolt distance: $\dfrac{\pi C}{n} = 43.616\,mm$

Check nut, washer, socket clearance: $OD_w := 2d_b$ this is for standard narrow washers, and for wrench sockets which more than cover the nut width across corners

$$0.5C - \left(0.5B + g_1 + r_1\right) \geq 0.5OD_w = 0$$

Check minimum bolt circle

$0.5B + g_1 + r_1 + 0.5 \cdot d_w \le 0.5C = 0$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 1.4mm$ $\qquad$ $D_{tmin} = 25.4\,mm$

We will thread some of these clearance holes for lift fixture bolts of size (db+4mm) 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. To avoid thread interference with flange bolts, the flange studs will be machined to root diameter per **UG-12(b)**.in between threaded ends of 1.5x diameter in length. The actual clearance holes will be db+2mm, depending on achievable tolerances, so as to allow threading where needed.

$d_{lfb} := d_b + 4mm$

$H_1 := .812mm$ $\qquad$ from chart above, for 1.5mm thread pitch

$d_{min\_lfb} := d_{lfb} - 2 \cdot H_1$ $\qquad$ $d_{min\_lfb} = 2.638\,cm$ $\qquad$ this will be max bolt hole size or least material condition (LMC)

$d_{min\_lfb} \ge D_{tmin} = 1$

effective threaded clearance hole diameter:

$D_e := 0.5\left(d_{lfb} + d_{min\_lfb}\right)$ $\qquad$ $D_e = 2.719\,cm$

Set:

$D_t := D_{tmin}$ $\qquad$ $D_t \ge D_{tmin} = 1$ $\qquad$ here we skip the above section; we can weld on support tabs that will allow retraction using the hexapod.

Compute Forces on flange:

$H_P := 0N$ $\qquad$ $H_P = 0\,N$ $\qquad$ $H_P = 0\,kgf$

$H_G = \begin{pmatrix} 0 \\ 2.809 \times 10^5 \end{pmatrix} N$ $\qquad$ $H_G = \begin{pmatrix} 0 \\ 2.864 \times 10^4 \end{pmatrix} kgf$

$h_G := 0.5(C - G)$ $\qquad$ $h_G = \begin{pmatrix} 2.85 \\ 2.725 \end{pmatrix} cm$ $\qquad$ from Table 2-6 Appendix 2, Integral flanges

$H_D := .785 \cdot B^2 \cdot P$ $\qquad$ $H_D = 2.266 \times 10^6\,N$

$R_. := 0.5(C - B) - g_1$ $\qquad$ $R_. = 2.5\,cm$ $\qquad$ radial distance, B.C. to hub-flange intersection, int fl..

$h_D := R_. + 0.5g_1$ $\qquad$ $h_D = 3\,cm$ $\qquad$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D$ $\qquad$ $H_T = \begin{pmatrix} 4.353 \times 10^4 \\ 5.195 \times 10^4 \end{pmatrix} N$

$h_T := 0.5 \cdot \left(R_. + g_1 + h_G\right)$ $\qquad$ $h_T = \begin{pmatrix} 31.75 \\ 31.125 \end{pmatrix} mm$ $\qquad$ from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange

$M_P := \overrightarrow{(H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G)}$ $\quad M_P = \begin{pmatrix} 6.937 \times 10^4 \\ 7.726 \times 10^4 \end{pmatrix} J$ $\quad M_P = \begin{pmatrix} 7.074 \times 10^3 \\ 7.878 \times 10^3 \end{pmatrix} kgf \cdot m$

Appendix Y Calculation

$P = 15.4\,bar$

Choose values for plate thickness and bolt hole dia:

$t := 4.0cm$ $\quad D := D_t$ $\quad D = 2.54\,cm$

Going back to main analysis, compute the following quantities:

$\beta := \dfrac{C + B_1}{2B_1}$ $\quad \beta = 1.022$ $\quad h_C := 0.5(A - C)$ $\quad h_C = \begin{pmatrix} 2 \\ 8.5 \end{pmatrix} cm$

$a := \dfrac{A + C}{2B_1}$ $\quad a = \begin{pmatrix} 1.058 \\ 1.106 \end{pmatrix}$ $\quad AR := \dfrac{n \cdot D}{\pi \cdot C}$ $\quad AR = 0.582$ $\quad h_0 := \sqrt{B \cdot g_0}$ $\quad h_0 = 11.662\,cm$

$r_B := \dfrac{1}{n}\left(\dfrac{4}{\sqrt{1 - AR^2}} \operatorname{atan}\left(\sqrt{\dfrac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right)$ $\quad r_B = 0.011$

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 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)

$J_S := \overrightarrow{\left[\dfrac{1}{B_1}\left(\dfrac{2 \cdot h_D}{\beta} + \dfrac{h_C}{a}\right) + \pi r_B\right]}$ $\quad J_S = \begin{pmatrix} 0.09 \\ 0.132 \end{pmatrix}$ $\quad J_P := \overrightarrow{\left[\dfrac{1}{B_1}\left(\dfrac{h_D}{\beta} + \dfrac{h_C}{a}\right) + \pi \cdot r_B\right]}$ $\quad J_P = \begin{pmatrix} 0.068 \\ 0.111 \end{pmatrix}$

(5a) $\quad F' := \dfrac{g_0^{\,2}(h_0 + F \cdot t)}{V}$ $\quad F' = 2.781 \times 10^{-5}\,m^3$ $\quad M_P = \begin{pmatrix} 6.937 \times 10^4 \\ 7.726 \times 10^4 \end{pmatrix} N \cdot m$

$A = \begin{pmatrix} 1.47 \\ 1.6 \end{pmatrix} m$ $\quad B = 1.36\,m$

$K := \dfrac{A}{B}$ $\quad K = \begin{pmatrix} 1.081 \\ 1.176 \end{pmatrix}$ $\quad Z := \overrightarrow{\dfrac{K^2 + 1}{K^2 - 1}}$ $\quad Z = \begin{pmatrix} 12.883 \\ 6.207 \end{pmatrix}$

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

$t_s := 0mm$ no spacer between flanges

$l := 2t + t_s + 0.5d_b$ $\quad l = 9.2\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

http://www.hightempmetals.com/techdata/hitempInconel718data.php

Elastic constants:

$E := E_{SS\_aus}$ $\quad E = 193\,GPa$ $\quad E_{Inconel\_718} := 208GPa$ $\quad E_{bolt} := \begin{pmatrix} E_{CS} \\ E_{Inconel\_718} \end{pmatrix}$

Flange Moment due to Flange-hub interaction

$$M_S := \overrightarrow{\frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'}} \qquad M_S = \begin{pmatrix} -2 \times 10^3 \\ -3.5 \times 10^3 \end{pmatrix} N \cdot m \tag{7}$$

Slope of Flange at I.D.

$$\theta_B := \overrightarrow{\left[\frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right)\right]} \quad \theta_B = \begin{pmatrix} 6.425 \times 10^{-4} \\ 1.138 \times 10^{-3} \end{pmatrix} \tag{8}$$

opening half gap =

$$\theta_B \cdot 3cm = \begin{pmatrix} 0.019 \\ 0.034 \end{pmatrix} mm$$

$$E \cdot \theta_B = \begin{pmatrix} 123.996 \\ 219.648 \end{pmatrix} MPa$$

Contact Force between flanges, at h_C:

$$H_C := \overrightarrow{\frac{M_P + M_S}{h_C}} \qquad H_C = \begin{pmatrix} 3.369 \times 10^6 \\ 8.676 \times 10^5 \end{pmatrix} N \tag{9}$$

Bolt Load at operating condition:

$$W_{m1} := \overrightarrow{\left(H + H_G + H_C\right)} \qquad W_{m1} = \begin{pmatrix} 5.679 \times 10^6 \\ 3.467 \times 10^6 \end{pmatrix} N \tag{10}$$

Operating Bolt Stress

$$\sigma_b := \frac{\overrightarrow{W_{m1}}}{\overrightarrow{A_b}} \qquad \sigma_b = \begin{pmatrix} 170 \\ 103.8 \end{pmatrix} MPa \qquad S_b = \begin{pmatrix} 233 \\ 255.1 \end{pmatrix} MPa \tag{11}$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = \begin{pmatrix} 0.965 \\ 0.928 \end{pmatrix}$$ elasticity factor

Design Prestress in bolts

$$S_i := \overrightarrow{\left[\sigma_b - \frac{1.159 \cdot h_C{}^2 \cdot \left(M_P + M_S\right)}{a \cdot t^3 \, l \cdot r_E \cdot B_1}\right]} \qquad S_i = \begin{pmatrix} 166.2 \\ 29.2 \end{pmatrix} MPa \tag{12}$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \overrightarrow{\frac{6\left(M_P + M_S\right)}{t^2\left(\pi \cdot C - n \cdot D\right)}} \qquad S_{R\_BC} = \begin{pmatrix} 134.7 \\ 147.4 \end{pmatrix} MPa \tag{13}$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := \overrightarrow{\left[-\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2}\right]} \qquad S_{R\_ID} = \begin{pmatrix} 1.866 \\ 3.305 \end{pmatrix} MPa \tag{14a}$$

Tangential Flange stress at inside diameter

$$S_T := \overrightarrow{\left[\frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2}\right]} \qquad S_T = \begin{pmatrix} 2.37 \\ 5.83 \end{pmatrix} MPa \tag{15a}$$

Longitudinal hub stress

$$S_H := \overrightarrow{\frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V}} \qquad S_H = \begin{pmatrix} 21.085 \\ 37.35 \end{pmatrix} \text{MPa} \qquad \text{(16a)}$$

Y-7 Bolt and Flange stress allowables: $S_b = \begin{pmatrix} 233 \\ 255.1 \end{pmatrix} \text{MPa} \quad S_f = 137.9\,\text{MPa}$

(a) $\overrightarrow{(\sigma_b \le S_b)} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(b)

(1) $\overrightarrow{(S_H \le 1.5 S_f)} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ S_n not applicable

(2) not applicable

(c) $\overrightarrow{(S_{R\_BC} \le S_f)} = \begin{pmatrix} 1 \\ 0 \end{pmatrix}$

$\overrightarrow{(S_{R\_ID} \le S_f)} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(d) $\overrightarrow{(S_T \le S_f)} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(e) $\overrightarrow{\frac{S_H + S_{R\_BC}}{2} \le S_f} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$\overrightarrow{\frac{S_H + S_{R\_ID}}{2} \le S_f} = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(f) not applicable

Bolt force

$$F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2} \qquad F_{bolt} = \begin{pmatrix} 7.688 \times 10^4 \\ 4.693 \times 10^4 \end{pmatrix} \text{N}$$

Bolt torque required, minimum:

$$T_{bolt\_min} := 0.2 F_{bolt} \cdot d_b \qquad T_{bolt\_min} = \begin{pmatrix} 369 \\ 225.3 \end{pmatrix} \text{N} \cdot \text{m} \quad T_{bolt\_min} = \begin{pmatrix} 272.2 \\ 166.1 \end{pmatrix} \text{lbf} \cdot \text{ft}$$

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. An additional amount of bolt preload is needed to maintain a minimum frictional shear resistance to assure head does not slide downward from weight; we do not want to depend on lip to carry this. 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, for properly designed joints where connection stiffness is much greater than bolt stiffness, and we are a long way from the yield stress of the bolts

$M_{head} := 2500\text{kg}$ $\qquad \mu_{SS\_SS} := .7$ typ. coefficient of friction, stainless steel (both) clean and dry

$V_{head} := M_{head} \cdot g \qquad V_{head} = 2.452 \times 10^4\,\text{N}$

$F_n := \frac{V_{head}}{\mu_{SS\_SS}} \qquad F_n = 3.502 \times 10^4\,\text{N}$ this is total required force, force required per bolt is:

$F_{n\_bolt} := \frac{F_n}{n} \qquad F_{n\_bolt} = 340.036\,\text{N}$ this is insignificant compared to that required for pressure.

Let bolt torque for normal operation be then 25% greater than minimum: