Head Nozzle and Flange Calculation, DN100 (CF) size Nozzle

Internal radius, nozzle

$R_n := 44mm$

Nozzle wall thickness

Required nozzle wall thickness, for internal pressure is:

$E_w := 1$

$$t_{rn} := \frac{MAWP \cdot (R_n)}{S_{max\_304L\_div1} \cdot E_w - 0.6 \cdot MAWP} \qquad t_{rn} = 0.601\,mm$$

We set wall thickness to be:

$t_n := 7mm$ $\qquad t_n > t_{rn} = 1$

Flange thickness:

Note: we design, if possible for standard CF bolt pattern so as to allow possibility of using CF flanges prebolted to adapter plates, on extra long screws. This allows CF flange/adapter plate to be preassembled and tested for both pressure and leak tightness prior to installing as an assembly onto pressure vessel flange. This will require utmost care to tighten nuts without loosening the CF joint. It is recommended that a torque wrench be used on the bolt head to maintain full tightness while tightening nut on opposite side.

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.

Flanges and shells will be fabricated from 316Ti (UNSS31635, EN 1.4571, ASME spec SA-240) stainless steel plate. The flange bolts and nuts will be either 316 SS, or Inconel 718, (UNS N77180), if required, as this is the highest strength non-corrosive material allowed for bolting.

We will design to use one Helicoflex 3mm thick gasket with aluminum facing (softest).

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

$S_f := S_{max\_316Ti}$ $\qquad S_f = 137.9\,MPa$

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

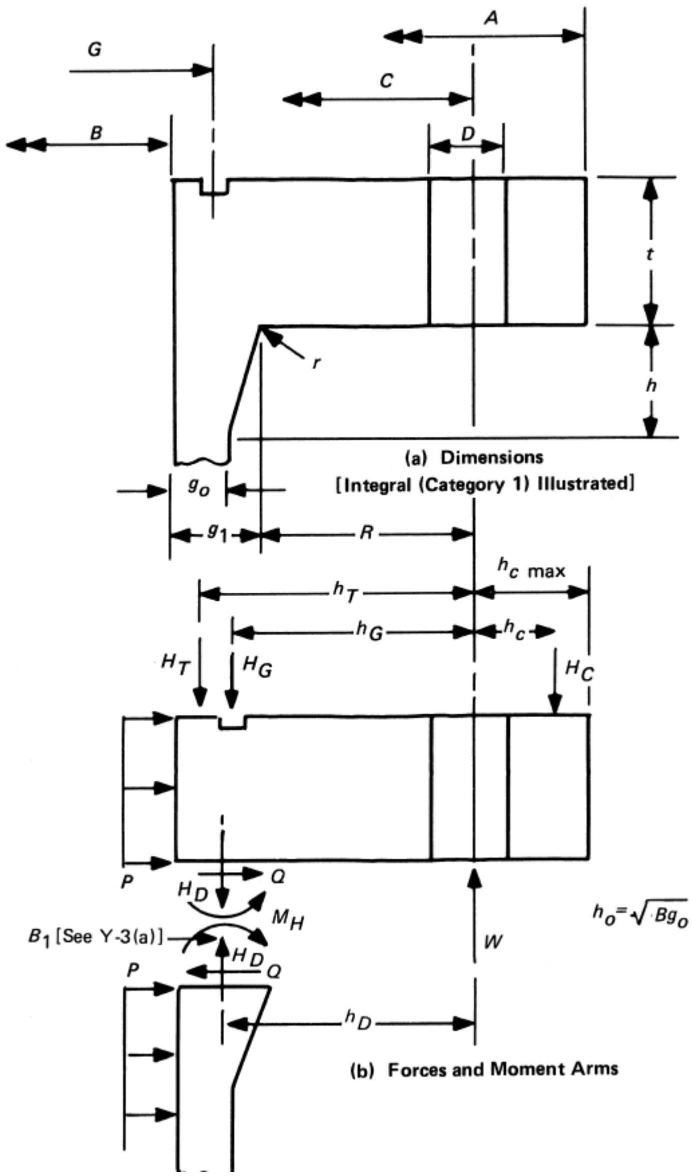
Inconel 718 (UNS N07718) $\qquad$ 316 condition/temper 2 (SA-193, SA-320)

$S_{max\_N07718} := 37000psi$ $\qquad S_{max\_316\_bolt} := 22000psi$

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

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)

$g_0 := t_n \quad g_1 := t_n \quad g_0 = 7\,\text{mm} \quad g_1 = 7\,\text{mm}$

corner radius:

$r_1 := \min(.5g_1, 5\text{mm}) \quad r_1 = 3.5\,\text{mm}$

Flange OD

$A := 16.5\text{cm}$

Flange ID

$B := 2R_n \qquad B = 8.8\,\text{cm}$

define:

$B_1 := B + 2g_1 \quad B_1 = 10.2\,\text{cm}$

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

$C := 13.0\text{cm}$

Gasket dia

$G := 2(R_n + .5\text{cm}) \qquad G = 0.098\,\text{m}$

Force of Pressure on head

$H := .785G^2 \cdot MAWP$ $\qquad H = 1.177 \times 10^4 N$

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:

$Y_1 := 25 \frac{N}{mm}$ min value for our pressure and required leak rate (He) $\qquad Y_2 := 185 \frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter $\quad D_j := G \qquad D_j = 0.098\,m$

Force is then either of:

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

$F_m = 7.697 \times 10^3 N$ $\qquad F_j = 5.696 \times 10^4 N$

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.

Number of bolts, root dia., pitch, bolt hole dia D, (these are from DN75 CF standard dimensions (VACOM catalog)

$n := 16 \qquad d_b := 8mm \qquad p_t := 1mm \qquad h_3 := .614p_t$

root dia.

$d_3 := d_b - 2h_3 \qquad d_3 = 6.772 \times 10^{-3} m$

$A_b := n \cdot \frac{\pi}{4} \cdot d_3^{\,2} \qquad A_b = 5.763\,cm^2$

Check bolt to bolt clearance, for box wrench b2b spacing is ~ 1.2 in for 1/2in bolt twice bolt dia ($2.4xd_b$):

$\pi C - 2.4n \cdot d_b \geq 0 = 1$ $\qquad \pi \frac{C}{n \cdot d_b} = 3.191$

Check nut, washer clearance: $\qquad OD_w := 2d_b$ $\qquad$ 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 :

$D_{tmin} := d_b + 0.5mm \qquad D_{tmin} = 8.5\,mm$

Set:

$D_t := 9mm \qquad D_t \geq D_{tmin} = 1$

Compute Forces on flange:

$$H_G := \begin{pmatrix} F_m \\ F_j \end{pmatrix} \qquad H_G = \begin{pmatrix} 7.697 \times 10^3 \\ 5.696 \times 10^4 \end{pmatrix} N$$

from Table 2-6 Appendix 2, Integral flanges

$$h_G := 0.5(C - G) \qquad h_G = 1.6\,cm$$

$$H_D := .785 \cdot B^2 \cdot MAWP \qquad H_D = 9.488 \times 10^3\,N$$

Here we add in the force on the nozzle from externally applied moment of:

$M_n := 2310 N \cdot m$ from shielding, venting, etc as calculated elsewhere)

This force acts through the nozzle and can be thought of as an an additional force to be added with H_D. To calculate, we compare longitudinal stresses in nozzle from pressure to that from moment:

ID OD Moment of Inertia, nozzle

$$d_{ni} := 2R_n \qquad d_n := 2(R_n + t_n) \qquad I_n := \frac{\pi}{64}\left(d_n^{\,4} - d_{ni}^{\,4}\right) \qquad I_n = 236.963\,cm^4$$

max. bending stress:

$$\sigma_M := \frac{M_n \cdot (R_n + t_n)}{I_n} \qquad \sigma_M = 50\,MPa$$

this is somewhat high, we compare with allowable longitudinal stress from pressure with minimum thickness nozzle (as calculated above)

$$\sigma_{D\_min} := \frac{H_D}{2\pi(R_n + 0.5t_n) \cdot t_{rn}} \qquad \sigma_{D\_min} = 53\,MPa \qquad \sigma_M < \sigma_{D\_min} = 1$$

OK

max long stress from pressure:

$$\sigma_D := \frac{H_D}{2\pi(R_n + 0.5t_n) \cdot t_n} \qquad \sigma_D = 4.54\,MPa$$

then let the equivalent force be:

$$F_n := \frac{\sigma_M}{\sigma_D} \cdot H_D \qquad F_n = 1.039 \times 10^5\,N$$

$$R := 0.5(C - B) - g_1 \qquad R = 1.4\,cm$$

radial distance, B.C. to hub-flange intersection, int fl.

$$h_D := R + 0.5 g_1 \qquad h_D = 1.75\,cm$$

from Table 2-6 Appendix 2, Int. fl.

$$H_T := H - H_D \qquad H_T = 2.279 \times 10^3\,N$$

$$h_T := 0.5(R + g_1 + h_G) \qquad h_T = 18.5\,mm$$

from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange (maximum value)

$$M_P := (H_D + F_n) \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \qquad M_P = \begin{pmatrix} 2.1 \times 10^3 \\ 2.9 \times 10^3 \end{pmatrix} N \cdot m$$

Appendix Y Calc

$$P := MAWP \qquad P = 15.4\,bar$$

Choose values for plate thickness and bolt hole dia:

$$t := 2.1cm \qquad D := D_t \qquad D = 0.9\,cm$$

Going back to main analysis, compute the following quantities:

$\beta := \frac{C + B_1}{2B_1}$ $\beta = 1.137$ $h_C := 0.5(A - C)$ $h_C = 0.018\,m$

$a := \frac{A + C}{2B_1}$ $a = 1.446$ $AR := \frac{n \cdot D}{\pi \cdot C}$ $AR = 0.353$ $h_0 := \sqrt{B \cdot g_0}$

$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}} \operatorname{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right)$ $r_B = 0.018$ $h_0 = 0.025\,m$

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

since $\frac{g_1}{g_0} = 1$ 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)

$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B$ $J_S = 0.475$ $J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B$ $J_P = 0.325$

(5a) $F' := \frac{g_0^2 (h_0 + F \cdot t)}{V}$ $F' = 3.911 \times 10^{-6}\,m^3$ $M_P = \begin{pmatrix} 2.149 \times 10^3 \\ 2.937 \times 10^3 \end{pmatrix} N \cdot m$

$A = 16.5\,cm$ $B = 8.8\,cm$

$K := \frac{A}{B}$ $K = 1.875$ $Z := \frac{K^2 + 1}{K^2 - 1}$ $Z = 1.795$

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

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b$ $l = 4.6\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

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

$E := E_{SS\_aus}$ $E = 193\,GPa$ $E_{Inconel\_718} := 208GPa$ $E_{bolt} := E_{Inconel\_718}$

Flange Moment due to Flange-hub interaction

$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'}$ $M_S = \begin{pmatrix} -245.3 \\ -335.3 \end{pmatrix} J$ (7)

Slope of Flange at I.D.

$\theta_B := \frac{5.46}{E \cdot \pi t^3}(J_S \cdot M_S + J_P \cdot M_P)$ $\theta_B = \begin{pmatrix} 5.649 \times 10^{-4} \\ 7.72 \times 10^{-4} \end{pmatrix}$ $E \cdot \theta_B = \begin{pmatrix} 109.018 \\ 149.001 \end{pmatrix} MPa$ (7)

Contact Force between flanges, at h_C:

$H_C := \frac{M_P + M_S}{h_C}$ $H_C = \begin{pmatrix} 1.088 \times 10^5 \\ 1.487 \times 10^5 \end{pmatrix} N$ (8)

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = \begin{pmatrix} 1.282 \times 10^5 \\ 2.174 \times 10^5 \end{pmatrix} N \qquad (9)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = \begin{pmatrix} 222.5 \\ 377.2 \end{pmatrix} MPa \qquad S_b = 255.1\,MPa \qquad (10)$$

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

elasticity factor

Design Prestress in bolts

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot (M_P + M_S)}{a \cdot t^3 \cdot l \cdot r_E \cdot B_1} \qquad S_i = \begin{pmatrix} 210.9 \\ 361.4 \end{pmatrix} MPa \qquad (11)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \qquad S_{R\_BC} = \begin{pmatrix} 98 \\ 133.9 \end{pmatrix} MPa \qquad (12)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID} = \begin{pmatrix} 11.925 \\ 16.299 \end{pmatrix} MPa \qquad (13)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_T = \begin{pmatrix} 22.86 \\ 31.24 \end{pmatrix} MPa \qquad (14a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = \begin{pmatrix} 52.991 \\ 72.426 \end{pmatrix} MPa$$

Y-7 Flange stress allowables: $S_b = 255.106\,MPa \quad S_f = 137.9\,MPa$

(a) $\sigma_b < S_b = \begin{pmatrix} 1 \\ 0 \end{pmatrix}$

<-- we cannot use full Y2 load, even with Inconel 718 bolts (unless we assure that fast vent valve is a straight through design

(b)

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

(2) not applicable

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

$S_{R\_ID} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

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

(e)

$$\frac{S_H + S_{R\_BC}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$$

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

(f) not applicable

Bolt force

$F_{bolt} := \sigma_b \cdot .785 \cdot {d_b}^2$ $\qquad F_{bolt} = \begin{pmatrix} 2.513 \times 10^3 \\ 4.261 \times 10^3 \end{pmatrix} lbf$

Bolt torque required

$T_{bolt\_min} := 0.2 F_{bolt} \cdot d_b$ $\qquad T_{bolt\_min} = \begin{pmatrix} 17.9 \\ 30.3 \end{pmatrix} N \cdot m$ $\; T_{bolt\_min} = \begin{pmatrix} 13.2 \\ 22.4 \end{pmatrix} lbf \cdot ft$ for pressure test use 1.5x this value