

Head Nozzle and Flange Calculation, DN40 (CF) size Nozzle

Internal radius, nozzle

$$R_n := 15\text{mm}$$

Nozzle wall thickness

Required nozzle wall thickness, for internal pressure is:

$$E_w := 1$$

$$t_{rn} := \frac{\text{MAWP} \cdot (R_n)}{S_{\text{max_304L_div1}} \cdot E_w - 0.6 \cdot \text{MAWP}} \quad t_{rn} = 0.205 \text{ mm}$$

We set wall thickness to be:

$$t_n := 3\text{mm}$$

$$t_n > t_{rn} = 1$$

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 heads to maintain full tightness on CF gasket 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 304L or 316L (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected. 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 to use one Helicoflex 2mm gasket (smallest size possible) with aluminum facing (softest) loaded to the minimum force required to achieve helium leak rate.

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

$$S_f := S_{\text{max_316Ti}} \quad S_f = 137.9 \text{ MPa}$$

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

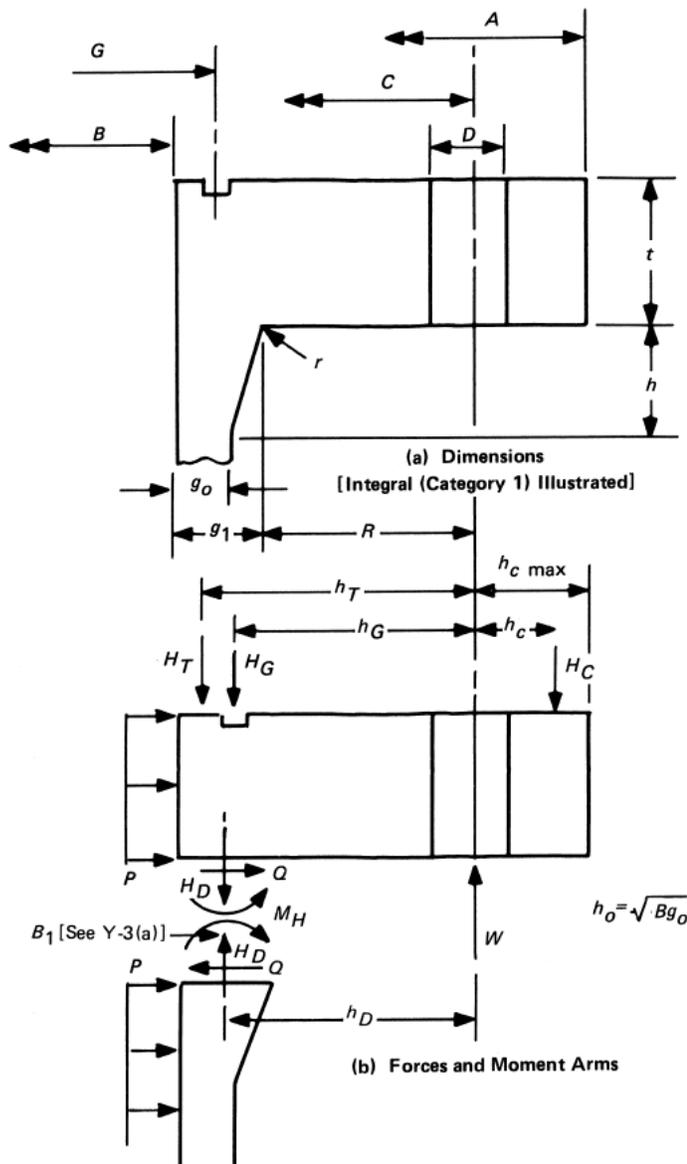
| | | |
|---|---|---|
| Inconel 718 (UNS N07718) | 316 condition/temper 2 (SA-193, SA-320) | temper 1 |
| $S_{\text{max_N07718}} := 37000\text{psi}$ | $S_{\text{max_316_bolt2}} := 22000\text{psi}$ | $S_{\text{max_316_bolt1}} := 18800\text{psi}$ |

Bolt, flange materials (we use MathCAD's parallel calculation capability, with vectors)

$$S_b := \begin{pmatrix} S_{\text{max_316_bolt1}} \\ S_{\text{max_N07718}} \end{pmatrix} \quad S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} \text{MPa}$$

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:

$$g_0 := t_n \quad g_1 := t_n \quad g_0 = 3 \text{ mm} \quad g_1 = 3 \text{ mm} \quad r_1 := \min(.5g_1, 5\text{mm}) \quad r_1 = 1.5 \text{ mm}$$

Flange OD

$$A := 8 \text{ cm}$$

Flange ID

$$B := 2R_n \quad B = 3 \text{ cm}$$

define:

$$B_1 := B + 2g_1 \quad B_1 = 3.6 \text{ cm}$$

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

$$C := 5.9 \text{ cm} \quad \text{VACOM}$$

Gasket dia

$$G := 2(R_n + .5\text{cm}) \quad G = 4 \text{ cm}$$

Force of Pressure on head

$$H := .785G^2 \cdot \text{MAWP} \quad H = 1.96 \times 10^3 \text{ N}$$

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 equiv. values of Y for the ASME force term F and gives several possible values for 3mm HN200 with aluminum jacket:

$$Y_1 := 8 \frac{\text{lb}_f}{\text{in}} \quad \text{min value for our pressure and required leak rate (He)} \quad Y_2 := 175 \frac{\text{N}}{\text{mm}} \quad \text{recommended value for large diameter seals, regardless of pressure or leak rate}$$

for gasket diameter $D_j := G \quad D_j = 0.04 \text{ m}$

Force is then either of:

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

$$F_m = 176.057 \text{ N} \quad F_j = 2.199 \times 10^4 \text{ N}$$

Helicoflex recommends using Y2 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 in through the outer O-ring as well.

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

$$n := 12 \quad d_b := 6 \text{ mm} \quad p_t := 0.8 \text{ mm} \quad h_3 := .614 p_t$$

root dia.

$$d_3 := d_b - 2h_3 \quad d_3 = 5.018 \times 10^{-3} \text{ m}$$

$$A_b := n \cdot \frac{\pi}{4} \cdot d_3^2 \quad A_b = 2.373 \text{ cm}^2$$

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

$$\pi C - 2.0n \cdot d_b \geq 0 = 1 \quad \pi \frac{C}{n \cdot d_b} = 2.574$$

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 :

$$D_{tmin} := d_b + 0.5 \text{ mm} \quad D_{tmin} = 6.5 \text{ mm}$$

Set:

$$D_t := 6.5 \text{ mm} \quad D_t \geq D_{tmin} = 1$$

Compute Forces on flange:

$$H_G := \begin{pmatrix} F_m \\ F_j \end{pmatrix} \quad H_G = \begin{pmatrix} 176.057 \\ 2.199 \times 10^4 \end{pmatrix} \text{N}$$

from Table 2-6 Appendix 2, Integral flanges

$$h_G := 0.5(C - G) \quad h_G = 0.95 \text{ cm}$$

$$H_D := .785 \cdot B^2 \cdot \text{MAWP} \quad H_D = 1.103 \times 10^3 \text{ N}$$

Unlike the other nozzles we will have any external forces or moments applied to these nozzles, while under pressure

$$R := 0.5(C - B) - g_1 \quad R = 1.15 \text{ cm}$$

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

$$h_D := R + 0.5g_1 \quad h_D = 1.3 \text{ cm}$$

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

$$H_T := H - H_D \quad H_T = 857.679 \text{ N}$$

$$h_T := 0.5(R + g_1 + h_G) \quad h_T = 12 \text{ mm}$$

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

Total Moment on Flange (maximum value)

$$M_P := (H_D) \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad M_P = \begin{pmatrix} 26.3 \\ 233.5 \end{pmatrix} \text{N}\cdot\text{m}$$

Appendix Y Calc

$$P := \text{MAWP} \quad P = 15.4 \text{ bar}$$

Choose values for plate thickness and bolt hole dia:

$$t := 1.5 \text{ cm} \quad D := D_t \quad D = 0.65 \text{ cm}$$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \quad \beta = 1.319 \quad h_C := 0.5(A - C) \quad h_C = 0.011 \text{ m}$$

$$a := \frac{A + C}{2B_1} \quad a = 1.931 \quad AR := \frac{n \cdot D}{\pi \cdot C} \quad AR = 0.421 \quad h_0 := \sqrt{B \cdot g_0}$$

$$r_B := \frac{1}{n} \left(\frac{4}{\sqrt{1 - AR^2}} \operatorname{atan} \left(\sqrt{\frac{1 + AR}{1 - AR}} \right) - \pi - 2AR \right) \quad r_B = 0.036 \quad h_0 = 9.487 \times 10^{-3} \text{ m}$$

We need factors F and V, 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)

$$J_S := \frac{1}{B_1} \left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a} \right) + \pi r_B \quad J_S = 0.813 \quad J_P := \frac{1}{B_1} \left(\frac{h_D}{\beta} + \frac{h_C}{a} \right) + \pi r_B \quad J_P = 0.539$$

$$(5a) \quad F' := \frac{g_0^2 (h_0 + F \cdot t)}{V} \quad F' = 3.783 \times 10^{-7} \text{ m}^3 \quad M_P = \begin{pmatrix} 26.3 \\ 233.544 \end{pmatrix} \text{N}\cdot\text{m}$$

$$A = 8 \text{ cm} \quad B = 3 \text{ cm}$$

$$K := \frac{A}{B} \quad K = 2.667 \quad Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 1.327$$

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 = 3.3 cm strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

Elastic constants

<http://www.hightempmetals.com/techdata/hitemplInconel718data.php>

$$E := E_{SS_aus} \quad E = 193 \text{ GPa} \quad E_{Inconel_718} := 208 \text{ GPa}$$

$$E_{bolt} := \begin{pmatrix} E_{SS_aus} \\ E_{Inconel_718} \end{pmatrix}$$

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \quad M_S = \begin{pmatrix} -1.5 \\ -12.9 \end{pmatrix} \text{ J} \quad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E \cdot \pi t^3} (J_S \cdot M_S + J_P \cdot M_P) \quad \theta_B = \begin{pmatrix} 3.468 \times 10^{-5} \\ 3.08 \times 10^{-4} \end{pmatrix} \quad E \cdot \theta_B = \begin{pmatrix} 6.694 \\ 59.444 \end{pmatrix} \text{ MPa} \quad (7)$$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \quad H_C = \begin{pmatrix} 2.366 \times 10^3 \\ 2.101 \times 10^4 \end{pmatrix} \text{ N} \quad (8)$$

Bolt Load at operating condition:

$$W_{m1} := H + H_G + H_C \quad W_{m1} = \begin{pmatrix} 4.502 \times 10^3 \\ 4.496 \times 10^4 \end{pmatrix} \text{ N} \quad (9)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \quad \sigma_b = \begin{pmatrix} 19 \\ 189.5 \end{pmatrix} \text{ MPa} \quad S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} \text{ MPa} \quad (10)$$

$$r_E := \frac{E}{E_{bolt}} \quad r_E = \begin{pmatrix} 1 \\ 0.928 \end{pmatrix} \quad \text{elasticity factor}$$

Design Prestress in bolts

$$S_i := \left[\sigma_b - \frac{1.159 \cdot h_C^2 \cdot (M_P + M_S)}{a \cdot t^3 \cdot r_E \cdot B_1} \right] \quad S_i = \begin{pmatrix} 18.6 \\ 185.6 \end{pmatrix} \text{ MPa} \quad (11)$$

Radial Flange stress at bolt circle

$$S_{R_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \quad S_{R_BC} = \begin{pmatrix} 6.2 \\ 54.8 \end{pmatrix} \text{ MPa} \quad (12)$$

Radial Flange stress at inside diameter

$$S_{R_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_{R_ID} = \begin{pmatrix} 0.411 \\ 3.65 \end{pmatrix} \text{MPa} \quad (13)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_T = \begin{pmatrix} 2.8 \\ 24.89 \end{pmatrix} \text{MPa} \quad (14a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \quad S_H = \begin{pmatrix} 3.524 \\ 31.292 \end{pmatrix} \text{MPa}$$

Y-7 Flange stress allowables:

$$S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} \text{MPa} \quad S_f = 137.9 \text{MPa}$$

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

(b) (1) $S_H < 1.5S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ 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

$$F_{\text{bolt}} := \sigma_b \cdot .785 \cdot d_b^2 \quad F_{\text{bolt}} = \begin{pmatrix} 120.552 \\ 1.204 \times 10^3 \end{pmatrix} \text{lbf}$$

Bolt torque required

$$T_{\text{bolt_min}} := 0.2F_{\text{bolt}} \cdot d_b \quad T_{\text{bolt_min}} = \begin{pmatrix} 0.6 \\ 6.4 \end{pmatrix} \text{N} \cdot \text{m} \quad T_{\text{bolt_min}} = \begin{pmatrix} 0.5 \\ 4.7 \end{pmatrix} \text{lbf} \cdot \text{ft} \quad \text{for pressure test use 1.5x this value}$$