Fermilab

Particle Physics Division
Mechanical Department Engineering Note

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Project Internal Reference:

Project:

Title: Decay Tank Flange

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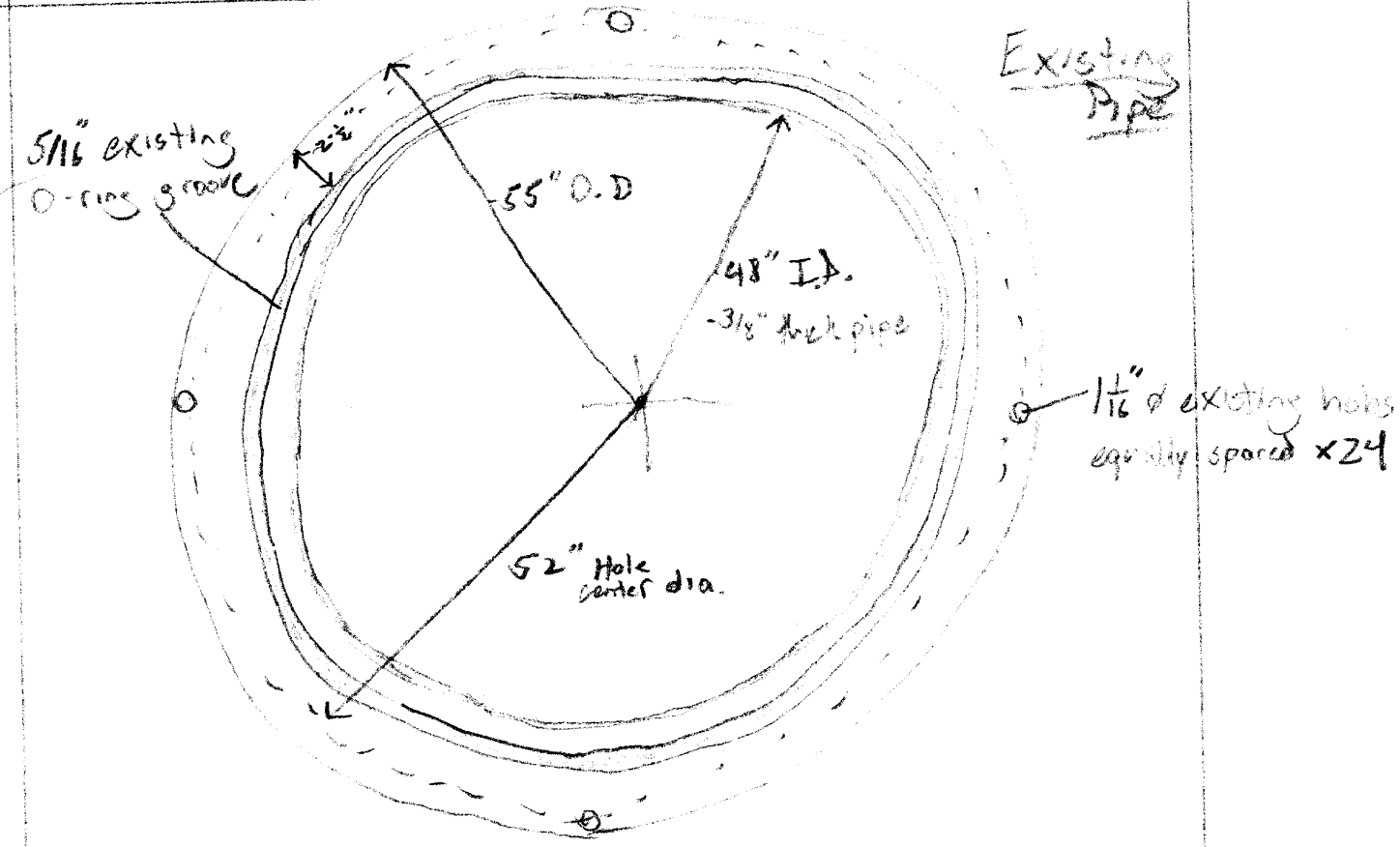
Reviewer(s):

Key Words: Decay Tank, Flange

Applicable Codes:

Abstract Summary:

This experiments beamline is taking the place of the KTev experiment and will be located in the same building using some of the same parts. This note discusses the flange on the 48 inch dia. vacuum sealed decay pipe, sec. M.C.-6, analyzed for minimum thickness, and minimum bolt diameter. All analysis for external pressure only.



Discussion

- Flange will be considered a hubless loose type flange with only external pressure applied due to vacuum sealing.
- Tangential stress on the flange will be the dominating factor; bolt loads and stresses will be negligable but will be calculated anyway.
- Both the flange and bolts will be composed of mild carbon steel
  - → Flange max allowable stress (SA-283 Grd. A)
    $S_f = 12.9$ ksi (ASME sec. II Pt. D Tb 1A)
  - → Bolt max allowable stress (SA-449, Alloy K04200)
    $S_a = 20.2$ ksi (ASME sec. II Pt. D Tb. 3)

Variable Def. (ASME Sec. VIII div. 1, Man. App. 2)

- $P_e = 14.69$ psi → External design pressure
- $N = 5/16''$ → Gasket contact width
- $b_o = \frac{N}{2} = (5/16)/2 = 5/32'' = b_o$ → Basic gasket seating width
- G ($b_o < \frac{1}{4}''$) → mean gasket contact dia → $G = 49\frac{11}{16}'' \rightarrow 49.6875''$
  dia. at gasket load reaction
- b ($b_o < \frac{1}{4}''$) → $b = b_o = 5/32'' = b$ → .15625"
  - → $m = 1$ - gasket factor (Table 2-5.1)
  - → $y = 200$ psi - gasket unit seating load (Table 2-5.1)
- $B = 48''$ → Inner dia. of flange; $A = 55''$ → outer flange dia.
- $C = 52''$ → Dia. of bolt circles
- $h_D = \frac{C-B}{2} = \frac{52''-48''}{2} \Rightarrow h_D = 2''$ → dis. from bolt circle to circle which the hydrostatic end force (H_D) acts.
- $h_G = \frac{C-G}{2} = \frac{52''-49.6875''}{2} \Rightarrow h_G = 1.15625''$ → dis. from bolt circle to gasket load reaction
- $h_T = \frac{h_D + h_G}{2} = \frac{2''+1.15625''}{2} \Rightarrow h_T = 1.57813''$

Moment arms for loose type flanges (To 2-6)

- $K = \frac{A}{B} = \frac{55''}{48''} \Rightarrow K = 1.15$
- Y (@ $K=1.15$) ⇒ $Y = 15$ (Fig. 2-7.1)

→ See page 5 for flange & variable illustrations

Forces

- $H = 0.785 G^2 P_e$ → Hydrostatic end force
  $= 0.785(49.6875'')^2(14.69\ psi) \Rightarrow \boxed{H = 28{,}469.9\ lbs}$

- $H_p = 2b \times 3.14 G m P_e$ → Joint-contact surface comp. force
  $= 2(.15625'') \times 3.14(49.6875'')(1)(14.69\ psi) \Rightarrow \boxed{H_p = 716.2\ lbs}$

- $H_D = 0.785 B^2 P_e$ → Inner flange hydrostatic force
  $= 0.785(48'')^2(14.69\ psi) \Rightarrow \boxed{H_D = 26{,}569\ lbs}$

- $H_T = H - H_D = 28{,}469.9 - 26{,}569 \Rightarrow \boxed{H_T = 1901\ lbs}$

Loads

bolt design load

- $W_{m_1} = H + H_p$ → Min Bolt load (op. cond.)
  $= 28{,}469.9 + 716.2 \Rightarrow \boxed{W_{m_1} = 29{,}186.1}$
  → Min. bolt dia - Load per bolt $= \frac{W_{m_1}}{24} = 1216\ lbs/bolt$
  $S_{bolt,max} = 20{,}200\ psi = \frac{F}{A_{min}} = \frac{1216}{\frac{1}{4}\pi d_b^2}$; $d_{bolt,min} = .277''$
  → $1\frac{1}{16}''$ is sufficient for bolt dia.

Initial load to seat gasket

- $W_{m_2} = 3.14 G b y$ → Initial Load to be applied to seat the O-ring
  $= 3.14(49.6875'')(.15625'')(200\ psi) \Rightarrow \boxed{W_{m_2} = 4878.6\ lbs}$

Load at operating conditions

- $W_{op} = W_{m_1} \Rightarrow \boxed{W_{op} = 29{,}186.1\ lbs}$

Load for gasket seating

- $W_{Gas} = \frac{A_{m_2} + A_b}{2} S_a$ → $A_{m_2} = \frac{W_{m_2}}{S_a} = \frac{4878.6\ lbs}{20{,}200\ psi} = \boxed{A_{m_2} = .24\ in^2}$ → min. bolt dia. for gasket seating
  → $A_b = \frac{1}{4}\pi(1\frac{1}{32})^2 = .835\ in^2 = A_b$ → bolt area w/o $\frac{1}{16}''$ threading
  $= \frac{(0.24 + 0.835)\ in^2}{2}(20{,}200\ psi) \rightarrow \boxed{W_{Gas} = 10{,}857.5\ lbs}$

Moments (external pressure only sec. 2-11)

- $M_{O_F} = H_D(h_D - h_G) + H_T(h_T - h_G)$ → Flange Moment
  $= 26{,}569 \text{ lbs}(2'' - 1.15625'') + 1901 \text{ lbs}(1.57813'' - 1.157813'') \Rightarrow \boxed{M_{O_F} = 23{,}217 \text{ in}\cdot\text{lbs}}$

- $M_{O_G} = W_{gas} h_G$ → Gasket seating moment
  $= (10{,}857.5 \text{ lbs})(1.15625'') \Rightarrow \boxed{M_{O_G} = 12{,}554 \text{ in}\cdot\text{lbs}}$

→ Moment on the flange under operating conditions is greatest and will be used in stress calculations. $M_O = 23{,}217$ in·lbs

Stresses (for loose type, hubless flanges)

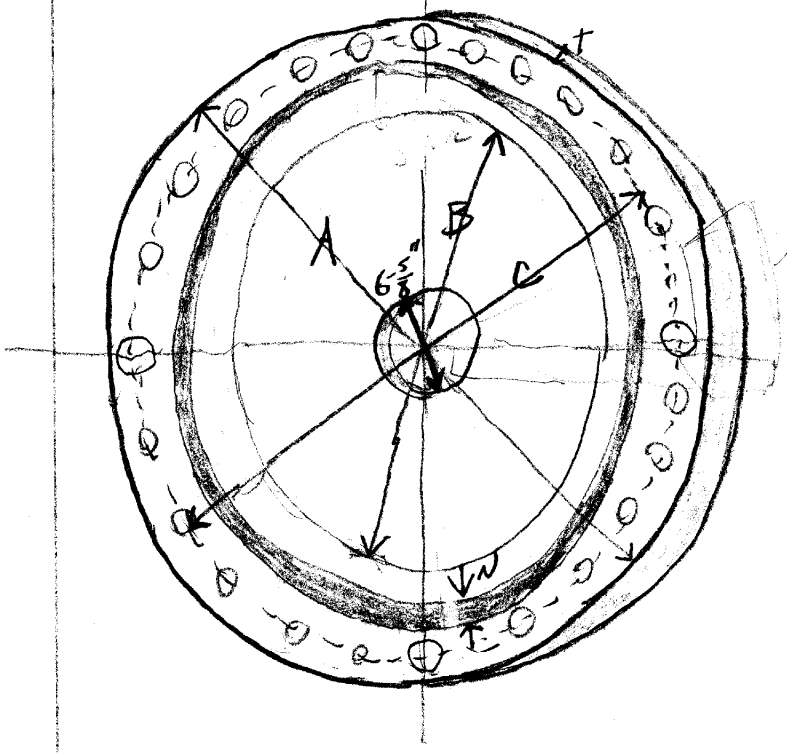
- Longitudinal hub stress, S_H, $= 0$ (no hub)
- Radial flange stress, S_R, $= 0$ (negligable w/o hub)
- Tangential flange stress, S_T, is dominant

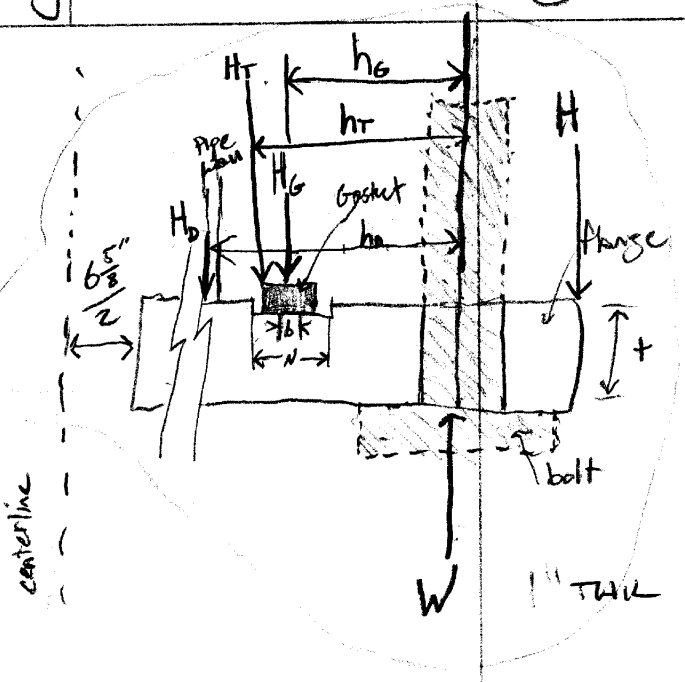
→ $S_{T_{max}} = \dfrac{YM_O}{t^2 B}$; $S_{T_{max}} = S_f = 12.9$ ksi

$$\rightarrow t_{min} = \left[\frac{YM_O}{S_{T_{max}} B}\right]^{\frac{1}{2}} = \left[\frac{15(23{,}217 \text{ in}\cdot\text{lbs})}{(12{,}900 \text{ psi})(48'')}\right]^{\frac{1}{2}} \Rightarrow \boxed{t_{min} = 0.7499''}$$

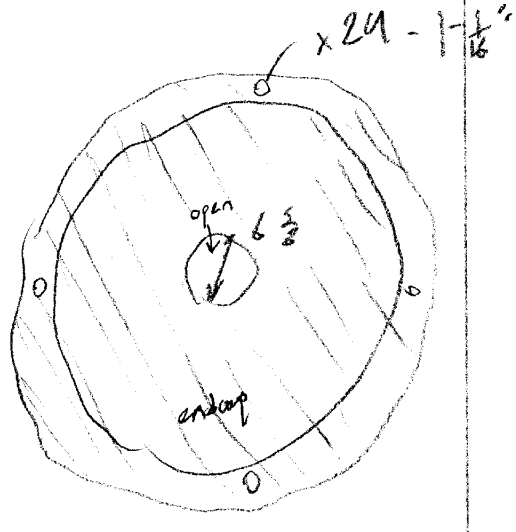
→ Flange must be a minimum of 3/4" thick to withstand stresses at operating conditions

→ With flange thickness set a 1", and the existing 1-1/4" thick flange on the 48" ID pipe, bolts should have a length no less than $\boxed{2\text{-}3/4'' = L_{Bolt,min}}$





N = 5/16 O-RING
(24) Φ 1 1/16"



Gary,
There's an error in my drawing of the flange... There is actually NO O-ring groove in it since there's one on the existing pipe.
— Mike