Particle Physics Division

Mechanical Department Engineering Note

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Project: LArTPC
Title: ASME Calculations for the “Bo” cryostat top flange

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Reviewer(s):

Key Words:

Abstract/Summary:

Applicable Codes:
ASME DIVISION I SECTION VIII
ASME DIVISION II SECTION VIII

Introduction

A 27.5 inch OD 304 stainless steel flange assembly is used to cap the “Bo” 250 L liquid argon cryostat. The “Bo” cryostat is ASME code stamped and has a MAWP of 35 psig. The flange is 1.5 inches thick and has several penetrations shown in the following three figures. The nozzles attached to the penetrations terminate in either conflat flanges or VCR fittings.

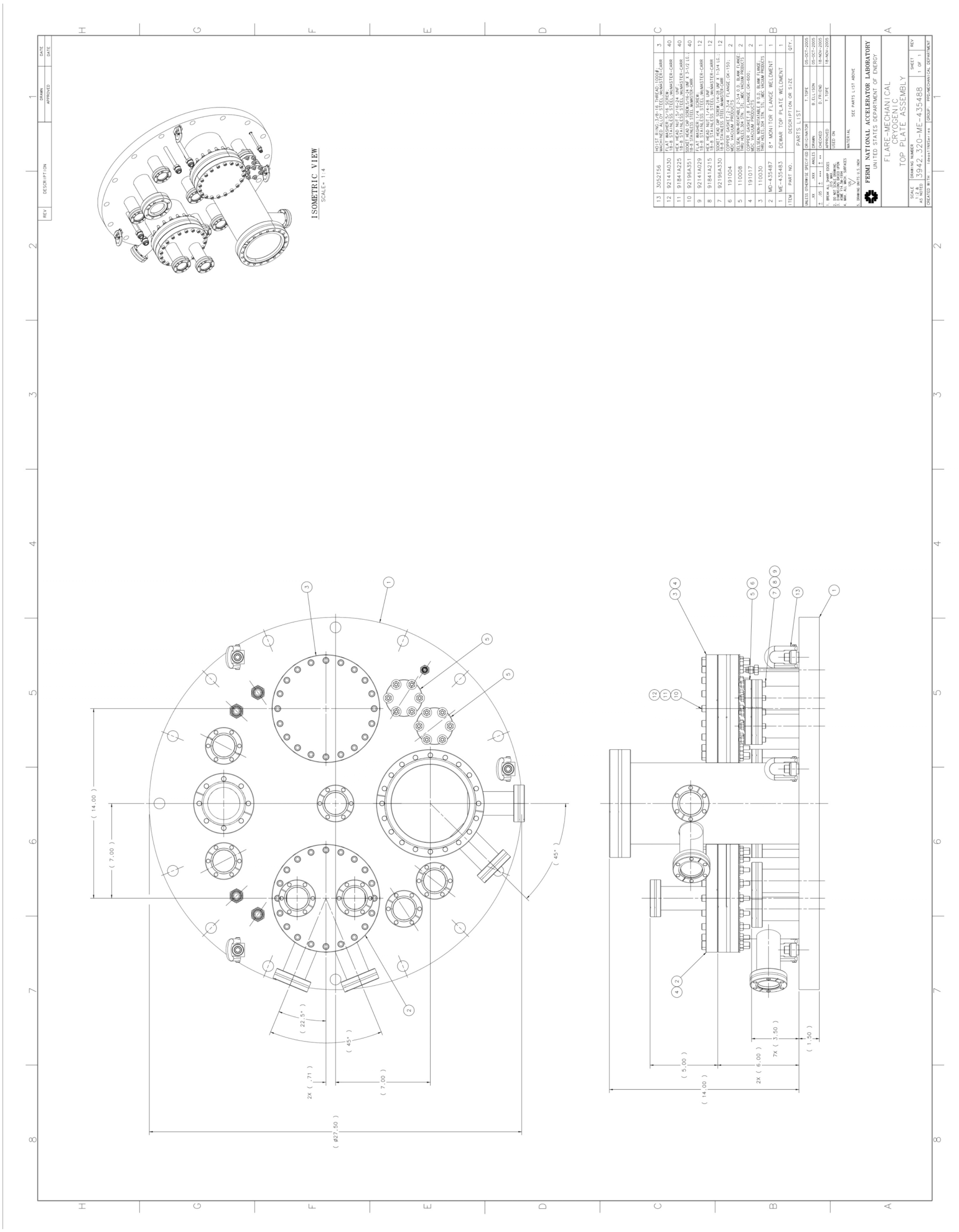

Figure 1: Flange assembly drawing.

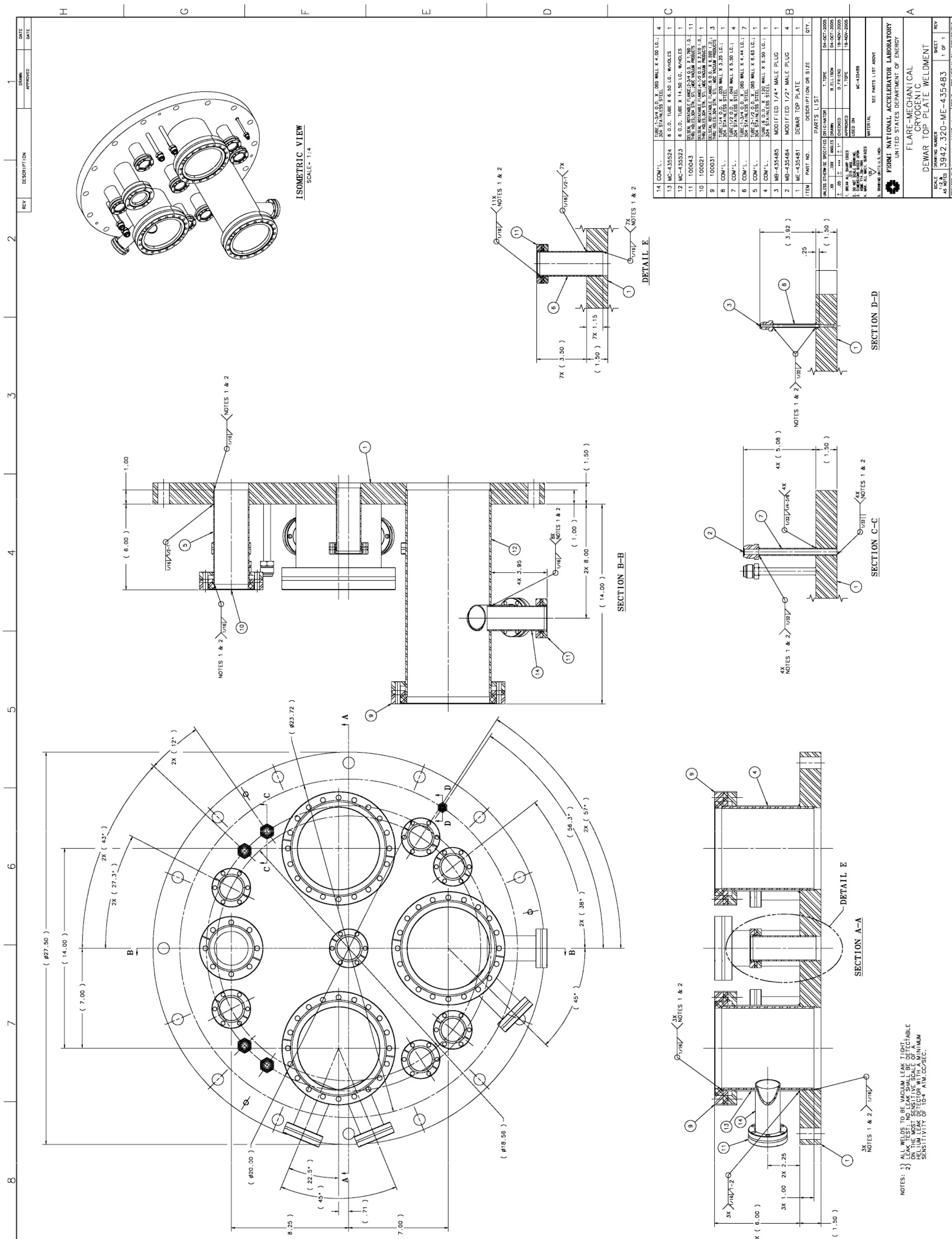

Figure 2: Flange weldment drawing.

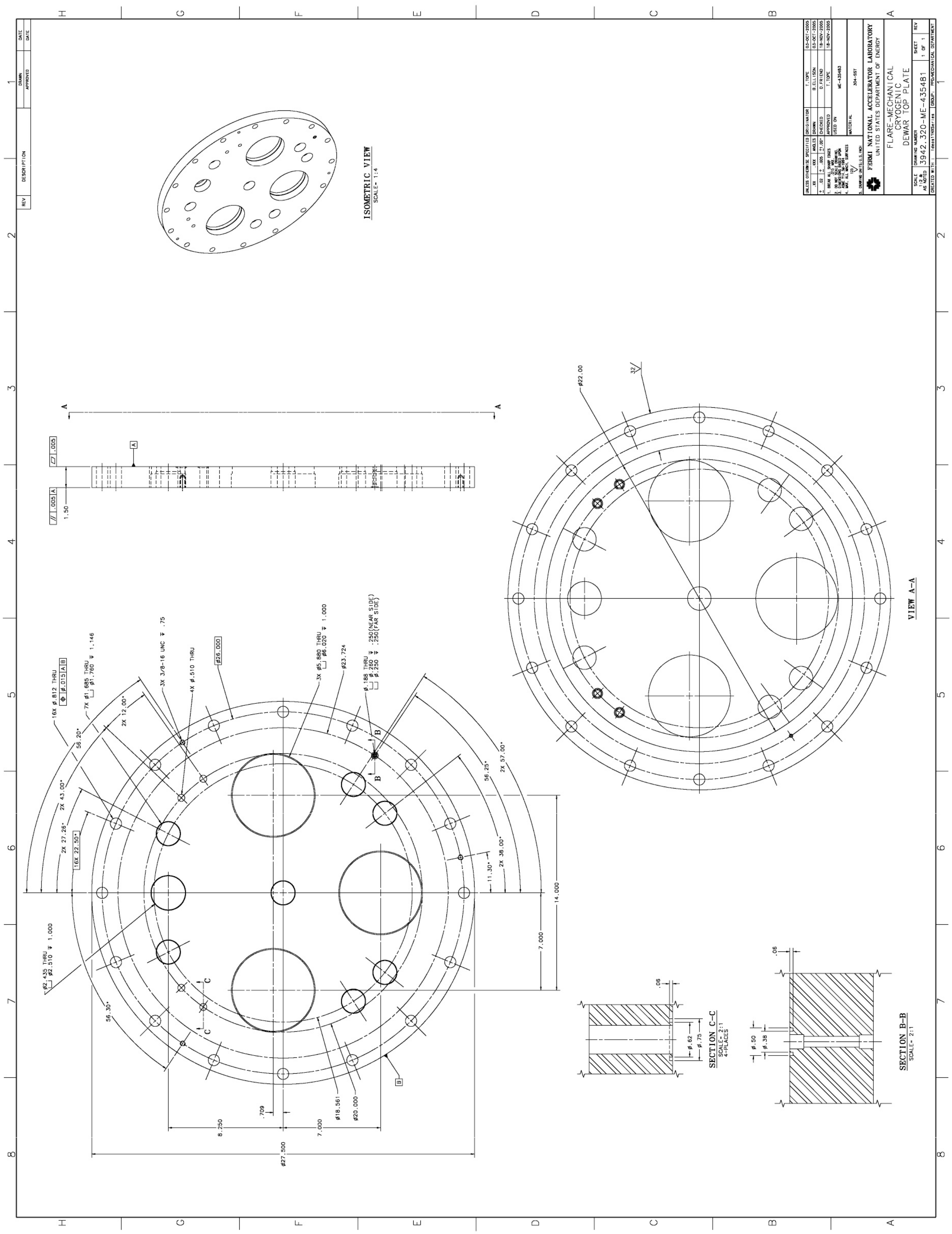

Figure 3: Flange drawing.

Flange Thickness and Bolt Loading

The required flange thickness is computed with a 0.8 multiplier on the allowable stress per UG-34 case (k) which is for a bolted flat head.

$$t = d\sqrt{CP/SE + 1.9Wh_G/SEd^3}$$

t	= minimum head thickness
d	= 22.126 inches, inside flange diameter
C	= 0.3 flange attachment factor
S	= 0.8 * 16,700 psi = 13,360 psi (lowest value for 304 SS plate found in Section II, Part D, Table 1A multiplied by 0.8 for in house flange construction)
E	= 1, welded joint efficiency
W	= 14,728 lbs, total bolt load
h_G	= 2.847 inch, gasket moment arm, center of gasket reaction to center of bolt hole
P	= 35 psig, internal design pressure
t	= 0.80 inch, required thickness

The actual flange thickness is 1.5 inches.

Appendix 2-5(e) is utilized to calculate the flange design bolt Load *W*.

The bolt loads used in the design of the flange shall be the values obtained from W = W_{m1} such that

$$W_{m1} = H + H_p = 0.785G^2P + (2b \times 3.14Gmp)$$

where

H = total hydrostatic end force

H_p = total joint-contact surface compression load.

m = gasket factor, obtain from Table 2-5.1 in Mandatory Appendix 2. For o-ring seals, the gasket factor is 0.

$$W_{m1} = 0.785(23.15)^2 35 = 14{,}728\ lbs$$

The *Load* on each bolt is then

Load = W_{ml} / # of bolts
= 14,728 lbs/ 16 bolts
= 920.5 lbs per bolt

Bolt *Stress*

Stress = *Load* / *area* of ¾-10 bolt
= 920.5 lbs/ 0.3345 inch^2
= 2752 psi

Required *Torque*

Torque = *k* x *D* x *F*

$k = 0.2$ steel fastener
$D = 0.750$ inch bolt diameter
$F = 920.5$ lb clamping load
$Torque = 0.2 \times 0.750$ inch x 920.5 lbs = 138 in-lbs
= 11.5 foot lbs

The bolts are 18-8 Stainless Steel with a minimum tensile strength rating of 70,000 psi. Nuts are brass with a minimum tensile strength of 55,000 psi. Thus the bolts and nuts have adequate strength with respect to the internal pressure load.

Reinforcement of Openings

Due to the pair of 6 inch penetrations having an average diameter greater than 25% of the head diameter, the reinforcement requirements of Section VIII DIV I could not be applied (see UG-39(b)(2)). Thus U-2(g) is invoked which states that "…….it is intended that the Manufacturer, subject to the acceptance of the Inspector, shall provide details of design and construction which will be as safe as those provided by the rules of this Division."

To analyze the reinforcement, a FEA model of the flange was created by Bob Wands and is available in the Appendix. If the peak stress reported by the FEA model is less than the ASME Section VIII DIV I allowable stress, then the level of safety is consistent with that of the Division. The rules of the 2007 Edition of the ASME Code, Section VIII, Div. II, Part 5, "Design by Analysis Requirements" were chosen as appropriate to the analysis of the openings in the flat head, and satisfactory in the context of U-2(g).

The model shows a peak Von Mises stress of 7,772 psi. The flange is constructed of 304 SS plate. The lowest allowable stress found in Section II Part D for this material is 16,700 psi. Because the flange was constructed at Fermilab without material control, this allowable stress is further reduced to 13,360 psi by applying a 0.8 factor. The latest edition of the pressure vessel code accepts Von Mises stresses after decades of utilizing stress intensity (Section VII Div 2 5.2.2.1). Because the peak Von Mises stress is less than the allowable stress, no further analysis is required. See Bob Wand's explanation of the reported stress in the Appendix after the FEA analysis.

Thickness of the Nozzles Under Internal Pressure

The nozzles attached to the flange are shells under internal pressure. The minimum wall thickness required in the nozzles is calculated per UG-27.

Circumferential Stress

$T_{required} = P\,R\,/\,(S\,E - 0.6\,P)$

Where,

$P = 35$ psi MAWP
R = inside radius of the nozzle
$S = 0.8 \times 14{,}200$ psi = 11,360 psi (lowest value for welded 304 SS tube found in Section II, Part D, Table 1A multiplied by 0.8 for in house flange construction).
$E = 0.5$ joint efficiency

Longitudinal Stress

$T_{required} = P R / (2 S E + 0.4 P)$

Where,

P = 35 psi MAWP

R = inside radius of the nozzle

S = 0.8 x 14,200 psi = 11,360 psi (lowest value for welded 304 SS tube found in Section II, Part D, Table 1A multiplied by 0.8 for in house flange

E = 0.5 joint efficiency

The wall thickness required for the nozzles is listed in Table 1.

Table 1: Nozzle wall thicknesses required for 35 psig internal pressure.

Nozzle OD (inch)	Nozzle Wall Thickness (inch)	Nozzle Inside Radius (inch)	Required Nozzle Wall Thicknesses	
			Circumferential stress t_min (inch)	Longitudinal stress t_min (inch)
6.000	0.120	2.9400	0.0178	0.0089
1.750	0.065	0.8425	0.0050	0.0025
0.250	0.035	0.1075	0.0006	0.0003
0.500	0.049	0.2255	0.0012	0.0006
2.500	0.065	1.218	0.0073	0.0036

Nozzle External Pressure

Because “Bo” can be evacuated, the method prescribed in part UG-28 of the ASME Section VIII Division 1 code is used to calculate the maximum allowable external pressure for the nozzles.

The longest length of 6 inch OD tube has a length of 14.5 inches.

First D_o / t is calculated where D_o is the pipe outside diameter of 6.000 inches and t is the wall thickness of 0.120 inches. $D_o / t = 6.000 / 0.120 = 50.0$.

Because Do / t is greater than 4, calculate L / D_o where L is the length of the cylinder which is 14.5 inches. $L / D_o = 14.5 / 6.00 = 2.42$.

Because L / D_o is greater than 0.05 and less than 50, enter Figure G in ASME Section II Part D and locate the value for Factor A. With a $L / D_o = 2.42$ and $D_o / t = 50.0$, Factor A = 0.0015.

From Figure HA-1 in Section II Part D, Factor B is 10,200 based on Factor A equal to 0.0015 and the 100 ^{o}F modulus curve.

Because D_o / t is greater than 10, the maximum allowable external pressure is calculated using

$$P = \frac{4B}{3\frac{D_o}{t}} = \frac{4 \times 10{,}200}{3\frac{6.000}{0.120}} = 272\,psi.$$

Thus the 6 inch OD tubes can with stand the external pressure due to vacuum.

The longest length of 2 1/2 inch OD tube is 6.63 inches long.

First D_o / t is calculated where D_o is the pipe outside diameter of 2.500 inches and t is the wall thickness of 0.065 inches. $D_o / t = 2.500 / 0.065 = 38.5$.

Because Do / t is greater than 4, calculate L / D_o where L is the length of the cylinder which is 6.63 inches. $L / D_o = 6.63 / 2.50 = 2.65$.

Because L / D_o is greater than 0.05 and less than 50, enter Figure G in ASME Section II Part D and locate the value for Factor A. With a $L / D_o = 2.65$ and $D_o / t = 38.5$, Factor A = 0.0018.

From Figure HA-1 in Section II Part D, Factor B is 11,000 based on Factor A equal to 0.0018 and the 100 ^{o}F modulus curve.

Because D_o / t is greater than 10, the maximum allowable external pressure is calculated using

$$P = \frac{4B}{3\frac{D_o}{t}} = \frac{4 \times 11{,}000}{3\frac{2.500}{0.065}} = 381\,psi.$$

Thus the 2 1/2 inch OD tubes can with stand the external pressure due to vacuum.

The longest length of 1 ¾ inch OD tube is 4.44 inch long. Because the 1 ¾ inch OD tube has the same wall thickness as the 2 ½ inch OD tube and is shorter, it can withstand a higher external pressure.

Nozzle Penetrations

Two of the three 6 inch OD nozzles have a pair of closely spaced 1.76 inch diameter penetrations as shown in Figure 4. Due to the close spacing of these holes, they are analyzed as an assumed opening that encloses both openings as suggested in UG-42(c). The nozzle walls of the two 1.75 inch OD tubes are considered to have no reinforcing value. Because the assumed opening diameter of 3.82 inches exceeds 50% of the 6 inch OD nozzle diameter, the rules of Appendix 1-10 are used as suggested in UG-36(b)(1).

The calculations were performed using the EES software and the details are available in the Appendix. A maximum internal pressure of 107.6 psi was computed for this penetration such that the design is adequate for the 35 psig MAWP of "Bo."

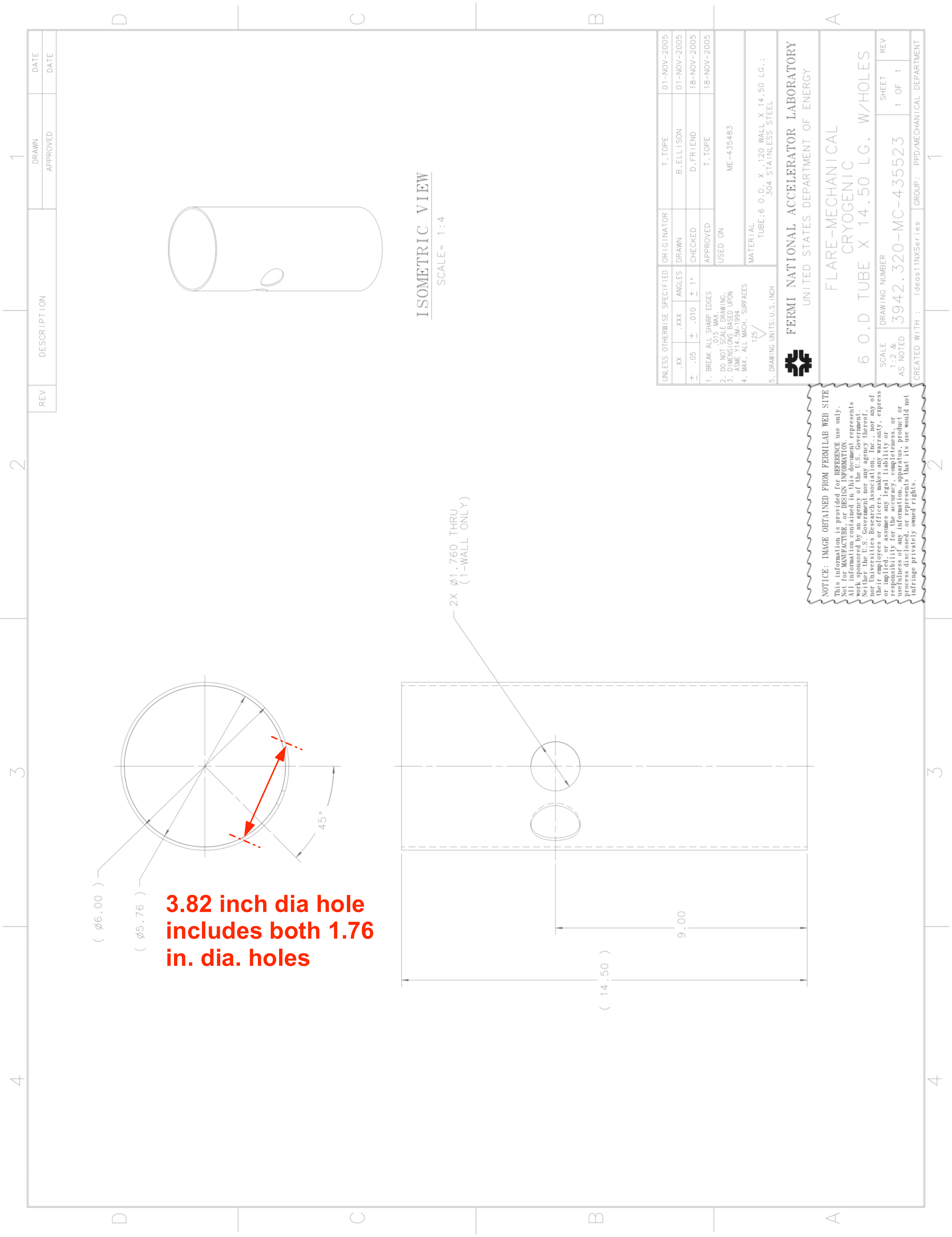

Figure 4: Two closely spaced penetrations on the 6 in. OD nozzle.

Summary

The peak stress in the flange is only 58% of a conservative allowable stress for 304 SS. The nozzles attached to the flange have adequate wall thicknesses. The bolts used to attach the head are properly sized to resist the maximum pressure loading the flange will see.

Appendix

FEA Model of Flange

July 2, 2008

Finite Element Analysis of Reinforcement of Openings in Flare Cryogenic Dewar Top Plate

Bob Wands

Summary

A finite element analysis of the Flare cryogenic dewar top plate is used to verify that the openings in the plate are adequately reinforced per the requirements of the 2007 edition of the ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1, para. U-2(g).

The head is constructed of 304 stainless steel. The maximum allowable stress is taken as that of the weakest 304 material listed in the Code tables, that of SA-403 welded fittings. This stress is 17 ksi, compared to the highest strength listed, which is 20 ksi for SA-240 plate. A factor of 0.8 is applied to the 17 ksi, resulting in a maximum allowable stress of 13.6 ksi.

The finite element model is shown in Fig. 1. Due to symmetry, only half of the head is modeled. A mesh of 20-node brick elements is used. The perimeters of the bolt holes are constrained to react the pressure loading. A uniform pressure of 35 psi is applied over the head. Additional nodal forces are applied to the perimeters of the holes to represent the pressure force from the missing material, which must appear in thrust through the tube walls which connect to the holes.

A reaction force check shows that the model agrees with the expected reaction value (= plate area x 35 psi) to within 1%.

The highest stresses (disregarding the concentrations around the constrained bolt holes) occur at the perimeter of the central hole. The stresses in all openings are shown in Fig. 2. No stress exceeds 8000 psi, well below the maximum allowable value of 13.6 ksi.

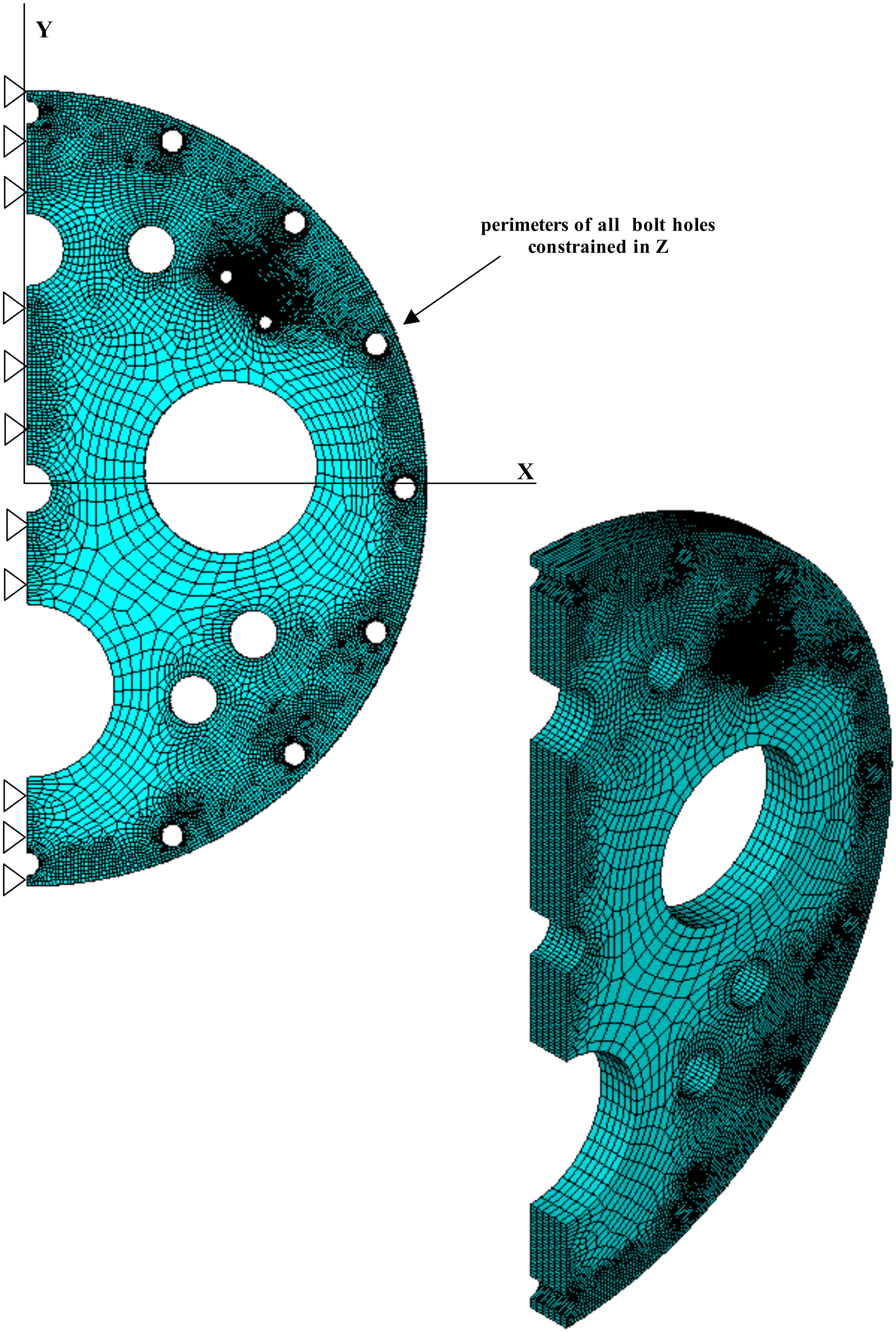

Figure 1. Finite Element Model

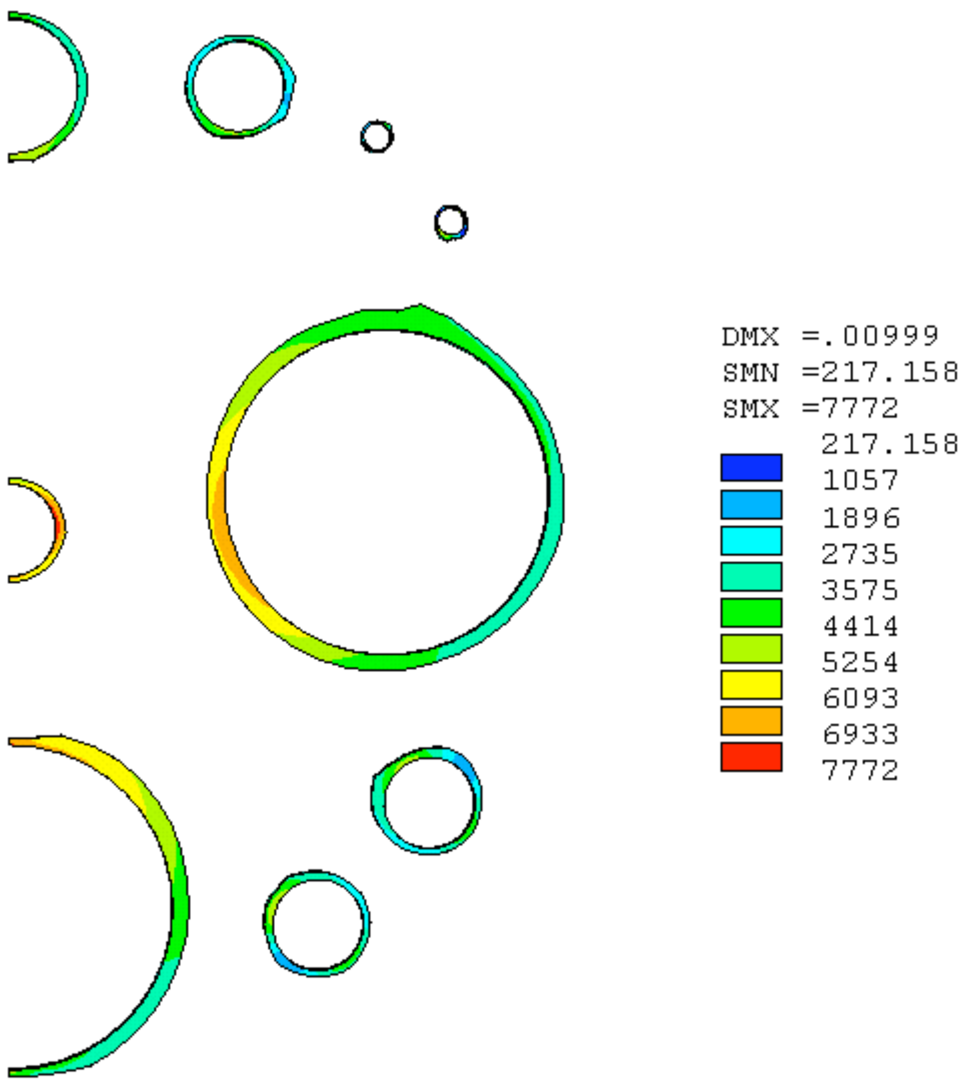

Figure 2. Von Mises stress at perimeters of openings

Stress Analysis of the Flare Flat Head - Explanation

1) It was decided that, according to the 2007 Edition of the ASME Code, Section VIII, Div. I, UG-39(b)(2), the number and spacing of 6 inch openings in the flat head requires that the reinforcement of these openings be treated by the rules of U-2(g).

2) U-2(g) states: This Division of Section VIII does not contain rules to cover all details of design and construction. Where complete details are not given, it is intended that the Manufacturer, subject to the acceptance of the Inspector, shall provide details of design and construction which will be as safe as those provided by the rules of this Division.

3) The rules of the 2007 Edition of the ASME Code, Section VIII, Div. II, Part 5, "Design by Analysis Requirements" were chosen as appropriate to the analysis of the openings in the flat head, and satisfactory in the context of U-2(g).

4) Part 5 concerns protection of the vessel against four possible types of failures:

   a) Plastic collapse (gross yielding in tension or compression)
   b) Local failure (yielding at structural discontinuities)
   c) Buckling (collapse due to geometric instability under compressive loading)
   d) Failure from cyclic loading (fatigue and ratcheting)

5) For the flat head with penetrations, only a) is considered.

6) Part 5 allows three approaches to stress analysis:

   a) Elastic stress analysis method
   b) Limit-load method
   c) Elastic-plastic stress analysis method

7) Approach a) was chosen for the analysis of the flat head. It is very similar to the Section VIII, Div II Appendix 4 rules of previous Code editions.

8) For plastic collapse, 5.2.2.1 states: "To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value."

9) Continuing at 5.2.2.1 (a): "A quantity known as the equivalent stress is computed at locations in the component and compared to an allowable value of equivalent stress to determine if the component is suitable of the intended design conditions."

10) Continuing at 5.2.2.1 (b): "The maximum distortion energy yield criterion shall be used to establish the equivalent stress. In this case, the equivalent stress is equal to the **von Mises equivalent stress** given by Equation (5.1)"

11) Stresses from the elastic analysis are classified as primary membrane, primary local membrane, and primary local membrane plus bending.

12) From 5.2.2.4, the following limits on these three categories of stresses are established as:

a) Primary membrane stress $\leq$ S
b) Primary local membrane stress $\leq$ 1.5S
c) Primary local membrane plus bending stress $\leq$ 1.5 S

where S is defined as the "allowable stress based on the material of construction and design temperature." This stress is typically found in Section II of the Code.

13) The value of S (allowable stress) for the flat head was chosen from the stresses of Section II conservatively as 13.6 ksi

14) A finite element model of the flat head was created which included all penetrations. The model showed that the maximum equivalent stress calculated at the inner radius of any penetration was less than 8 ksi. (Runs with various mesh densities confirmed convergence on this stress.)

15) According to Table 5.6, "Examples of Stress Classification," the stresses in a typical ligament in a uniform pattern of a perforated head or shell are classified as primary membrane plus bending, as are the stresses in the center of an unperforated flat head. Stresses in isolated or atypical ligaments, which is actually the case for this head, are classified as secondary and peak stresses, and are considered for cyclic loading effects only.

16) The most conservative approach is to classify the head stresses as primary. Since no membrane stresses exist, the stress is entirely primary bending, with an allowable stress of 1.5S, or 20.4 ksi.

17) Even if classified as primary membrane stress, with an allowable of S = 13.6 ksi, all points in the regions of the penetrations satisfy the limits of 5.2.2.4.

EES Analysis of 6 in. OD tube penetrations

{This calculation sheet looks at the two closely spaced 1.76" thru holes in the 6 inch OD tubes on "Bo"}
{it combines the two holes into one 3.82 inch OD hole that includes both 1.76" OD holes}

{From ASME Section VIII Division 1 - Appendix 1-10}

{Step 1}
{Calculate the limit of reinforcement along the vessel wall}
{For integrally reinforced nozzles (NO reinforcing pads) }
L_R = 8 * t {effective length of the vessel wall}
t = 0.120 {nominal thickness of the vessel wall, 6 inch OD tube, 0.120 wall}

{Step 2}
{Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface}
L_H1 = t + 0.78*SQRT(R_n*t_n)
L_H2 = L_pr1 + t
L_H3 = 8*(t + t_e)
L_H = MIN(L_H1,L_H2,L_H3) {effective length of nozzle wall outside the vessel}
R_n = 3.82 {nozzle inside radius, nozzle wall - thickness does NOT contribute for combined holes}
t_n = 0 {nominal thickness of nozzle wall - thickness taken to be ZERO for combined holes}
L_pr1 = 4 {nozzle projection from outside of the vessel wall}
t_e = 0 {thickness of the refinforcing pad - NO reinforcement pad available}

{Step 3}
{Calculate the limit of reinforcement along the nozzel wall projecting inside the vessel surface}
L_I1 = 0.78*SQRT(R_n*t_n)
L_I2 = L_pr2
L_pr2 = 0 {nozzle projection from inside the vessel wall, nozzle does not extend into 6 in OD tube nor can the wall reinforcing values be used}
L_I3 = 8*(t + t_e)
L_I = min(L_I1, L_I2, L_I3) {effective length of wall inside the vessel}

{Step 4}
{Determine the total available area near the nozzle opening}
A_T = A_1 +A_2 + A_3 + A_41+ A_42 + A_43 + A_5 {Total area within the assumed limits of reinforcement}
A_1 = t*L_R*MAX(lambda/4,1) {Area contributed by the vessel wall}
lambda = min(((d_n+t_n)/SQRT((D_i+t_eff)*t_eff)),10)
d_n = 3.82 {ID of nozzle}
D_i = 6 - 0.120*2 {ID of shell}
t_eff = 0.12 {effective thickness used in the calculation of pressure stress near the nozzle opening}
A_2 = t_n*L_H {area contributed by the nozzle outside the vessel wall}
A_3 = t_n * L_I {area contributed by the nozzle inside the vessel wall}
A_41 = 0 {area contributed by the outside nozzle fillet weld}
A_42 = 0 {area contributed by the pad to vessel fillet weld}
A_43 = 0 {area contributed by the inside nozzel fillet weld}
A_5a = W*t_e
W = 0 {width of reinforcing pad}
A_5b = L_R*t_e
A_5 = min(A_5a,A_5b) {area contributed by the reinforcing pad}

{Step 5}
{Determine the effective radius of the shell}
R_eff = D_i/2 {effective pressure radius}

{Step 6}
{Determine the applicable forces}
f_N = P*R_n*(L_H - t) {force from internal pressure in the nozzle outside of the vessel}

Equations with comments (1/2)

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f_S = P*R_eff*(L_R + t_n)          {force from internal pressure in the shell}
f_Y = P*R_eff*R_nc                 {discontinuity froce from internal pressure}
R_nc = R_n                         {radius of the nozzle opening in the vessel along the chord}
P = 35                             {MAWP of Bo}

{Step 7}
{Determine the effective thickness for nozzles in cylindrical or conical shells as follows}
{previously defined in Step 4}

{Step 8}
{Determine the average local primary membrane stress and the general primary membrane stress in the vessel}
sigma_avg = (f_N+f_S+f_Y)/A_T    {average primary membrane stress}
sigma_circ = P*R_eff/t_eff       {general primary membrane stress}

{Step 9}
{Determine the maximum local primary membrane stress at the nozzle intersection}
P_L = MAX(2*sigma_avg - sigma_circ, sigma_circ)   {nozzle maximum allowable stress}

{Step 10}
{The calculated maximum local primary membrane stress should satisfy the following}
S_allow = 1.5*S*E         {allowable local membrane stress}
S = 16700*0.8             {allowable for SS304}
E = 1.0                   {the weld joint factor, 1.0 if it does not intersect a seam}
P_max_1 = S_allow/(  2*(A_p/A_t) - (R_eff/t_eff) )
P_max_2 = S * (t/R_eff)
A_p = R_n * (L_H -  t) + R_eff*(L_R + t_n + R_nc)    {area resisting pressure, used to determine the nozzle opening discontinuity, use
   4 for actual L_H value}
P_max = MIN(P_max_1,P_max_2)                 {nozzle maximum allowable pressure}
```

Equations with comments (2/2)

$L_R = 8 \cdot t$

$t = 0.12$

$L_{H1} = t + 0.78 \cdot \sqrt{R_n \cdot t_n}$

$L_{H2} = L_{pr1} + t$

$L_{H3} = 8 \cdot [t + t_e]$

$L_H = \mathbf{Min}[L_{H1}, L_{H2}, L_{H3}]$

$R_n = 3.82$

$t_n = 0$

$L_{pr1} = 4$

$t_e = 0$

$L_{I1} = 0.78 \cdot \sqrt{R_n \cdot t_n}$

$L_{I2} = L_{pr2}$

$L_{pr2} = 0$

$L_{I3} = 8 \cdot [t + t_e]$

$L_I = \mathbf{Min}[L_{I1}, L_{I2}, L_{I3}]$

$A_T = A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5$

$A_1 = t \cdot L_R \cdot \mathbf{Max}\left[\frac{\lambda}{4}, 1\right]$

$\lambda = \mathbf{Min}\left[\frac{d_n + t_n}{\sqrt{(D_i + t_{eff}) \cdot t_{eff}}}, 10\right]$

$d_n = 3.82$

$D_i = 6 - 0.12 \cdot 2$

$t_{eff} = 0.12$

$A_2 = t_n \cdot L_H$

$A_3 = t_n \cdot L_I$

$A_{41} = 0$

$A_{42} = 0$

$A_{43} = 0$

$A_{5a} = W \cdot t_e$

Formatted equations (1/2)

$W = 0$

$A_{5b} = L_R \cdot t_e$

$A_5 = \mathbf{Min}\left[A_{5a}, A_{5b}\right]$

$R_{eff} = \dfrac{D_i}{2}$

$f_N = P \cdot R_n \cdot \left[L_H - t\right]$

$f_S = P \cdot R_{eff} \cdot \left[L_R + t_n\right]$

$f_Y = P \cdot R_{eff} \cdot R_{nc}$

$R_{nc} = R_n$

$P = 35$

$\sigma_{avg} = \dfrac{f_N + f_S + f_Y}{A_T}$

$\sigma_{circ} = P \cdot \dfrac{R_{eff}}{t_{eff}}$

$P_L = \mathbf{Max}\left[2 \cdot \sigma_{avg} - \sigma_{circ}, \sigma_{circ}\right]$

$S_{allow} = 1.5 \cdot S \cdot E$

$S = 16700 \cdot 0.8$

$E = 1$

$P_{max,1} = \dfrac{S_{allow}}{2 \cdot \dfrac{A_p}{A_T} - \dfrac{R_{eff}}{t_{eff}}}$

$P_{max,2} = S \cdot \dfrac{t}{R_{eff}}$

$A_p = R_n \cdot \left[L_H - t\right] + R_{eff} \cdot \left[L_R + t_n + R_{nc}\right]$

$P_{max} = \mathbf{Min}\left[P_{max,1}, P_{max,2}\right]$

Formatted equations (2/2)

A_1 = 0.131	A_2 = 0	A_3 = 0	A_{41} = 0	A_{42} = 0
A_{43} = 0	A_5 = 0	A_{5a} = 0	A_{5b} = 0	A_p = 13.77
A_T = 0.131	D_i = 5.76	d_n = 3.82	E = 1	f_N = 0
f_S = 96.77	f_Y = 385.1	λ = 4.548	L_H = 0.12	L_{H1} = 0.12
L_{H2} = 4.12	L_{H3} = 0.96	L_I = 0	L_{I1} = 0	L_{I2} = 0
L_{I3} = 0.96	L_{pr1} = 4	L_{pr2} = 0	L_R = 0.96	P = 35
P_L = 6518	P_{max} = 107.6	$P_{max,1}$ = 107.6	$P_{max,2}$ = 556.7	R_{eff} = 2.88
R_n = 3.82	R_{nc} = 3.82	S = 13360	σ_{avg} = 3679	σ_{circ} = 840
S_{allow} = 20040	t = 0.12	t_e = 0	t_{eff} = 0.12	t_n = 0
W = 0				

Solution printout (1/1)