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Test Cryostat Nozzle

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5/22/86

D-Zero Engineering Note 3740.710-EN-92

Approved Richard D. Smith

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This report contains the results of calculations for a resized nozzle on the D-Zero test cryostat. The nozzle neck in the inner vessel will be 6"ø schedule 10S pipe and 8"ø schedule 10S will be used for the outer vessel. On top of the nozzle will be placed the D-Zero Endcap Calorimeter signal board feedthru dewar box (see dwg. 3740.223-ME-223636). This box weighs approximately 250 lb, but the weight was conservatively taken to be 500 lb. Refer to Dwg. 3740-ME-209058 for details of the test cryostat.

Calculations

Vessel pressure = 45 psid
Allowable stress = 18800 X 0.8(FERMI stress reduction factor)
= 15000 psi

6" Pipe (Ref. 1 Par. UG-27)

$t_{r\ shell} = PR/(SE-0.6P)$

$= (45)(24.75)/[(0.6)(15000)(1)-(0.6)(45)] = 0.124$ in

$t_{r\ nozzle} = PR_{nozzle}/(SE-0.6P)$

$= (45)(3.1785)/[(0.6)(15000)(1)-(0.6)(45)] = 0.016$ in

Area of Reinforcement (Ref. 1, App. L)

$A_{req.} = d \times t_{r\ shell} \times F$

$= (6.357)(0.124)(1) = 0.79\ in^2$

$A_{avail.} = (E_1 t - F t_r)d + (t_{nozzle} - t_{r\ nozzle})5t f_r$

$= [(1)(0.25)-(1)(0.124)](6.375)+(0.148-0.016)(5)(0.25)(1)$

$= 0.97\ in^2$

$A_{avail.} > A_{req.}$ therefore no reinforcement is required.

<u>8" Pipe (Ref.1 Par. UG-28)</u>

L/D_0 = 72/55 = 1.31

Let t_r = 0.25/1.5
= 0.1667 in ($t = 1.5t_r$ ensures that any isolated opening in an externally pressurized vessel is self-reinforced.)

D_0 = 55"

D_0/t_r = 330

$P_a = 4B/3(D_0/t_r)$

A = 0.00018 (Fig. 5-UGO-28.0)

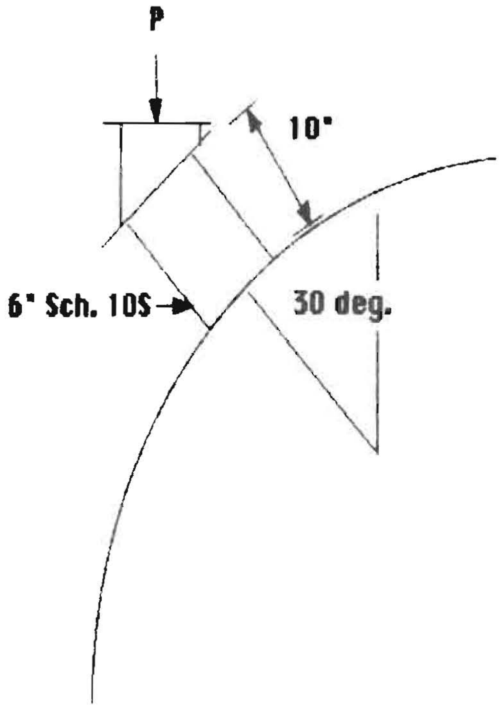
B = 2400 (Fig. 5-UGO-28.1)

P_a = 9.7 psi

Conclusion: The allowable pressure from above does not exceed 15 psig per ASME, but it does exceed 7.5 psig required by CGA-341 therefore reinforcement is not required. Note that the reinforcement provided by the nozzle neck in combination with the excess shell thickness is probably sufficient to satisfy the ASME requirements.

Stresses in Nozzle and Shell (due to feedthru box)

For conservatism, the 6"ø schedule 10S pipe was assumed to carry the full load of the feedthru box.



Max. bending stress in nozzle neck = Mc/I

$I = 15.3 \text{ in}^4$
$c = 3.3 \text{ in}$
$M = 500 \text{ lb}(10 \text{ in})\cos 60^0 = 2500 \text{ in-lb}$

Stress = 540 psi, negligible

Stresses on shell due to radial load (Ref. 2, 4.3.1)

$T = 0.25 \text{ in}$
$r_o = 3.3 \text{ in}$
$R_m = 27.5 \text{ in}$
$P = 500 \sin 60^0 = 433 \text{ lb}$
$\beta = 0.875\, r_o/R_m = 0.105$
$\text{Gamma} = R_m/T = 110$

$M\phi/P)_t$ - 0.08 $Mx/P)_l$ - 0.084

$Mx/P)_t$ = 0.05 $M\phi/P)_l$ = 0.05

$N\phi/(P/R_m))_t = Nx/(P/R_m)_l = 12$

$N\phi/(P/R_m))_l = Nx/(P/R_m)_t = 16$

<u>Longitudinal Axis</u>

$Stress_\phi = [N\phi/(P/R_m)_l][P/R_mT] + [M\phi/P)_l][6P/T^2]$

$= 3086$ psi

$Stress_x = [Nx/(P/R_m)_l][P/R_mT] + [Mx/P)_l][6P/T^2]$

$= 4248$ psi

<u>Transverse Axis</u>

$Stress_\phi = [N\phi/(P/R_m)_t][P/R_mT] + [M\phi/P)_t][6P/T^2]$

$= 4081$ psi

$Stress_x = [Nx/(P/R_m)_t][P/R_mT] + [Mx/P)_t][6P/T^2]$

$= 3086$ psi

Stresses in Shell Due to Circumferential Moment (Ref.2. 4.3.2)

$M\phi/(Mc/R_m\beta) = 0.08$ $Mx/(Mc/R_m\beta) = 0.044$

$N\phi/(Mc/R_m{}^2\beta) = 4.3$ $Nx/(Mc/R_m{}^2\beta) = 7.5$

$Stress_\phi = [N\phi/(Mc/R_m{}^2\beta)][Mc/R_m{}^2\beta T]+[M\phi/(Mc/R_m\beta)][6Mc/R_m\beta T^2]$

$= 7190$ psi

$Stress_x = [Nx/(Mc/R_m{}^2\beta)][Mc/R_m{}^2\beta T]+[Mx/(Mc/R_m\beta)][6Mc/R_m\beta T^2]$

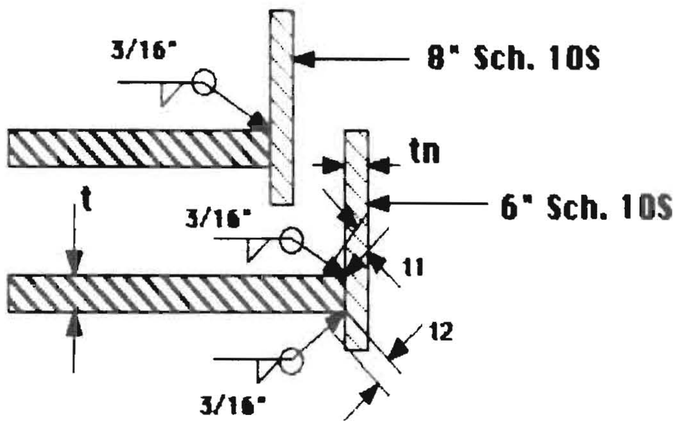
$= 4601$ psi

Pressure Bending Stress

The pressure bending stress can be assumed to be 1.5 times the membrane stress (PR/t). The membrane stress in the shell is approximately 5000 psi, therefore the pressure bending stress is approximately 7500 psi.

Conclusion: The largest total stress (pressure bending stress, stress due to radial load, and stress due to circumferential moment) occurs on the transverse axis and is equal to 4080 + 7190 + 7500 = 18770 psi. The bending stress limit is 3 X S or 45120 psi. The total stress is less than half of the limit.

<u>Weld Detail (Fig. UW-16.1(i))</u>



$t_1 + t_2 \geq 1.25t_{min}$
$t_1, t_2 > 0.7t_{min}$

$t_{min} = tn = 0.135"$

$.1875" + .1875" \geq 0.185"$ ($1.25t_{min}$) **OK**

$0.1875" > 0.1036"$ ($0.7t_{min}$) **OK**

References

1. "ASME Boiler and Pressure Vessel Code, Section VIII-Division1", The American Society of Mechanical Engineers, 1983.

2. "Local Stresses in Spherical and Cylindrical Shells Due to External Loading", Welding Research Council Bulletin #107, March, 1979.