Nozzle wall thickness required

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rev. A July 29, 2012 using 15 kg/s vent rate and current vent line with seismic loop

rev B, Mar 8 2013, revised for horizontal seismic loop

Internal radius of finished opening

$R_n := 4.4\text{cm}$

Thickness required for internal pressure:

$t_{rn} := \dfrac{P \cdot R_n}{S \cdot E - 0.6 \cdot P}$ $\quad t_{rn} = 0.495\,\text{mm}$

We set nozzle thickness

$t_n := 7\text{mm}$ we are limited by need to maintain CF bolt pattern which has typically a 4.0 inch OD pipe with room for outside fillet weld

$D_{on} := 2(R_n + t_n)$ $\quad D_{on} = 4.016\,\text{in}$ $\quad D_{on} = 102\,\text{mm}$

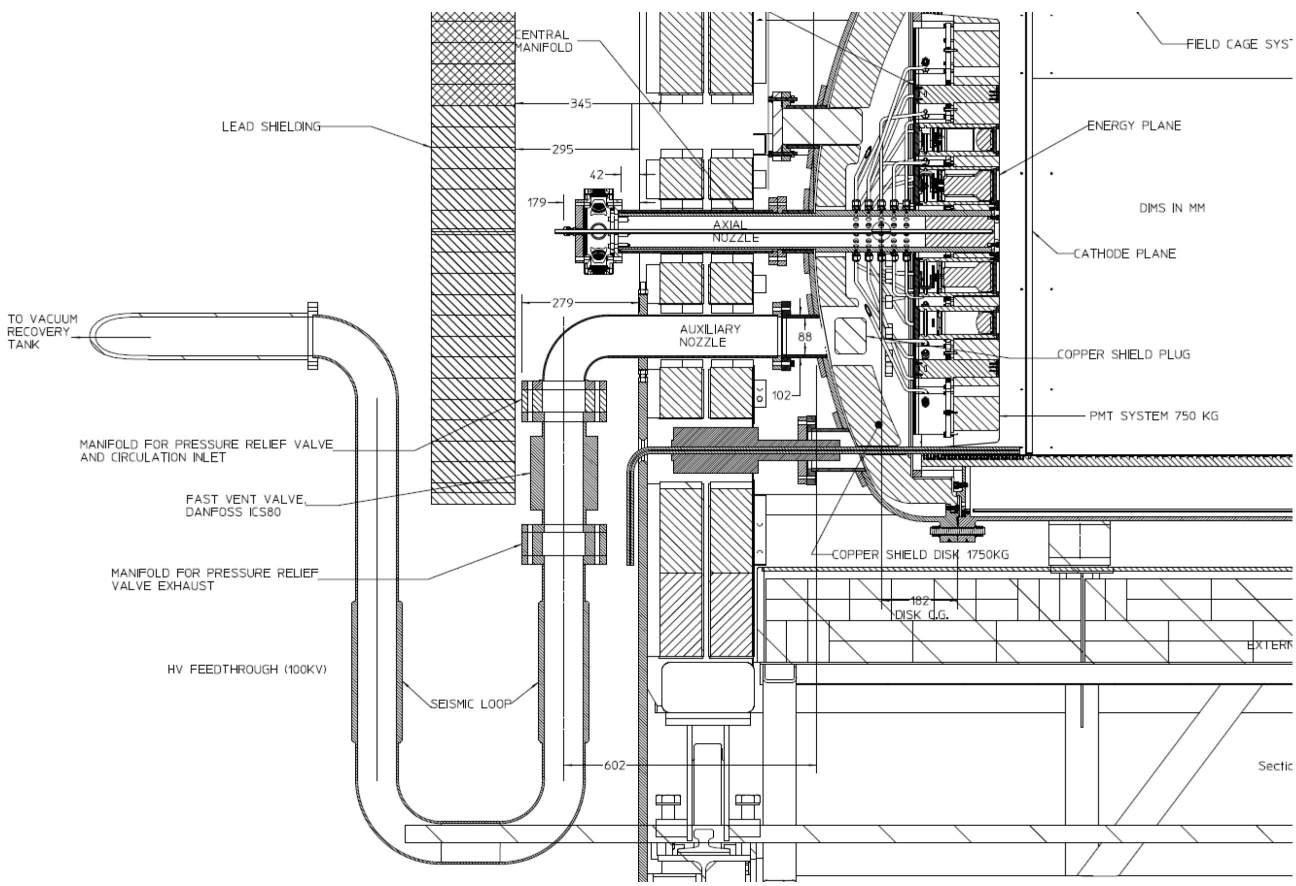
Thickness required for external loading

Nozzles on head may be subject to several possible non-pressure loads, simultaneously:

1. Reaction force from pressure relief, (fire) or fast depressure (auxiliary nozzle only)
2. Weight of attached components, including valves, expansion joints, high voltage feedthrough.

The nozzles may all have nozzle extensions rigidly attached which create to possibility of high moments being applied to the nozzles, not just shear loads. We consider the direction and location of center of gravity for these loads.

Current plan to use a straight through solenoid valve, Danfoss ICS80, aimed downward, to minimize shielding plug width. A seismic loop will be plumbed in so as to make reaction force in line with the vessel nozzle, under steady state vent conditions. However, a transient force will be present until the vent pipe fills, so we plan for this reaction force.



Component masses:

from CAD mass measurements 7 27/12

pres relief valve	fast vent valve	flange, each	manifold, each	seismic loop w/flanges	nozzle extension pipe and elbow
$m_{prv} := 2kg$	$m_{fvv} := 15kg$	$m_{fl} := 5kg$	$m_{manifold} := 10kg$	$m_{seis\_loop} := 70kg$	$m_{pipe} := 4kg$

total mass of vent line supported by nozzle

$$m_{vent\_line} := m_{pipe} + 3\cdot m_{fl} + m_{fvv} + m_{prv} + 2m_{manifold} + 0.5\cdot m_{seis\_loop}$$

$$m_{vent\_line} = 91\,kg$$

distance horizontal, from nozzle-head junction to valve and vessel half of seismic loop (other end of seimic loop supported by work platform

$$l_{valve} := 60cm$$

Vent rate, Danfoss ICS80 valve and associated vent piping, as calculated elsewhere

$$15\frac{kg}{s} = 1.19\times 10^5\frac{lb}{hr}$$

Fast vent reaction force, as calculated below, from Anderson Greenwood Technical Seminar Manual, pg 49

6.3.1 Reactive Force for GASES

On larger orifice, higher pressure valves, the reactive forces during valve relief can be substantial. External bracing might be required. Refer to Figure 6-7.

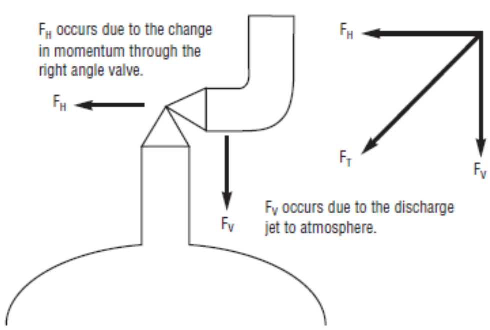


Figure 6-7. Reactive Force

API RP 520, Part 2 gives the following formula for calculating this force.

$$F_T = \frac{W\sqrt{\frac{kT}{(k+1)\,M}}}{366} + (A_o \times P_2) = F_H + F_V \qquad (1)$$

Where:

- F_T = Reactive force at the point of discharge to the atmosphere (lbs.)
- W = Flow of any gas or vapor (lb./hr.)
- k = Ratio of specific heats (C_p/C_v)
- T = Inlet temperature, absolute (°F + 460)
- M = Molecular weight of flowing media
- A_o = Area of the outlet at the point of discharge (in²)
- P_2 = Static pressure at the point of discharge (psig)

$$F_H := \frac{W\cdot\sqrt{\frac{k\cdot T}{(k+1)\cdot M_a}}}{366} \qquad F_H := \frac{119000\cdot\sqrt{\frac{1.67\cdot 535}{(1.67+1)\cdot 136}}}{366}\cdot lbf \qquad F_H = 510\,lbf \qquad F_H = 2.269\times 10^3\,N$$

$$A_0 := \frac{\pi}{4}80mm^2 \qquad P_2 := 15bar$$

$$F_V := A_0\cdot P_2 \qquad F_V = 21.2\,lbf \qquad F_V = 94.2\,N$$

$$F_T := F_H + F_V \qquad F_T = 2363\,N$$

Note : As mentioned above, the seismic should substantially redirect the momentum force (F_h) to be inline with the nozzle by relocating it to the loop exit port, or perhaps even to the recovery tank where the flow dissipates. This make F_h a force in line with the nozzle axis and essentially negates the static pressure load (axial tensile stress). The extent of this redirection is not clear, especially during the transient when the valve first opens, however and so we assume no redirection occurs, as the worst case.

Moments:

$$M_{vent\_weight} := -g \cdot l_{valve} \cdot \left(0.5m_{pipe} + 3 \cdot m_{fl} + m_{fvv} + m_{prv} + 2m_{manifold} + 0.5 \cdot m_{seis\_loop}\right)$$

$$M_{vent\_weight} = -523.675\,N \cdot m$$

$M_{fv} := F_T \cdot l_{valve}$ $\quad$ $M_{fv} = 1418\,N \cdot m$

Total moment, fast vent:

$M_n := M_{fv} + M_{vent\_weight}$ $\quad$ $M_n = 894\,N \cdot m$

For seismic loop oriented horizontally, moments will add in quadrature:

$M_{n\_hsl} := \sqrt{M_{fv}^2 + M_{vent\_weight}^2}$ $\quad$ $M_{n\_hsl} = 1.511 \times 10^3\,N \cdot m$

Total moment, someone sitting on nozzle extension

$M_{ps} := M_{vent\_weight} - 90kg \cdot g \cdot l_{valve}$ $\quad$ $M_{ps} = -1053\,N \cdot m$ will not happen during fast vent, so worst case is fast vent, above

Moment of Inertia, bending, of nozzle section

$I_n := \pi \cdot \left(R_n + 0.5t_n\right)^3 \cdot t_n$ $\quad$ $I_n = 235.7\,cm^4$ $\quad$ $t_n = 7\,mm$

Stress, bending (longitudinal)

$\sigma_{n\_l} := \dfrac{M_{n\_hsl} \cdot \left(R_n + t_n\right)}{I_n}$ $\quad$ $\sigma_{n\_l} = 32.7\,MPa$

Stress, circumferential (hoop)

$\sigma_{n\_c} := \dfrac{P \cdot R_n}{t_n}$ $\quad$ $\sigma_{n\_c} = 9.68\,MPa$

Criterion for acceptable stress - use maximum shear stress theory:

Maximum shear stress (min. stress is in third direction, = zero on outside of nozzle):

$\tau_n := \sqrt{\left(\dfrac{\sigma_{n\_l} - 0MPa}{2}\right)^2}$ $\quad$ $\tau_n = 16.4\,MPa$ OK $\quad$ (J. Shigley, Mech.Eng. 3rd ed., eq. (2-9)

Compare with maximum shear stress from minimum thickness nozzle (pressure only, no applied moments)

$t_{rn} = 0.495\,mm$

$\sigma_{rn} := \dfrac{P \cdot R_n}{t_{rn}}$ $\quad$ $\sigma_{rn} = 137\,MPa$

$\tau_{rn} := \sqrt{\left(\dfrac{\sigma_{rn} - 0MPa}{2}\right)^2}$ $\quad$ $\tau_{rn} = 68.5\,MPa$

Additional Factor of Safety, over ASME factor of safety:

$FS_n := \dfrac{\tau_{rn}}{\tau_n}$ $\quad$ $FS_n = 4.2$ OK