Note: under EN_13455-3 rules for 316Ti, a thinner thickness of 10.25 mm is possible, due to a higher maximum allowable strength at the knuckle. Below is an analysis from Sara Carcel

Torispherical heads, VIII, Div 2			**DIN 28011**			**EN 13445-3 (316Ti, para D**	
	KORBBOGEN		r=0,1L			f	166.666667
D	1360		1360	Diámetro interior		X	0.1
t	10		10			t	10
De	1380		1380			Y	0.00735294
L	1104		1360	Diámetro interior corona		Z	2.13353891
ri	212.52		136			N	0.84954918
L/D	0.81176471	Ok	1	Entre 0,7 y 1, ver 4-49		β0,1	0.86799204
ri/D	0.15626471	Ok	0.1	Mayor de 0,06		β0,2	0.51421113
Li/t	110.4	Ok	136	Entre 20 y 2000		β	0.86799204
β	1.01880199		1.11024234			P	1.52
φ	0.49440713		0.85749293			**eb**	8.66383993
R	752.567792		697.850818	Si φ<β		**ey**	10.2275849
C1	0.71313518	r/D>0,08	0.6742			**es**	6.21577196
C2	1.05371176		1.2			**Thickness**	**10.2275849**
Peth	64.2498476	E=117000	44.2387943				
C3	206	Sy=206MPa	206				
Py	3.37117586		1.57121205				
G	19.0585868		28.1558395				
Pck	6.73919067		3.14731728	G>1			

Nozzle wall thickness required

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

Internal radius of finished opening

$$R_n := 4.4\text{cm}$$

Thickness required for internal pressure:

$$t_{rn} := \frac{P \cdot R_n}{S \cdot E - 0.6 \cdot P} \qquad t_{rn} = 0.501\,\text{mm}$$

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

$$D_{on} := 2\left(R_n + t_n\right) \qquad D_{on} = 4.016\,\text{in}$$

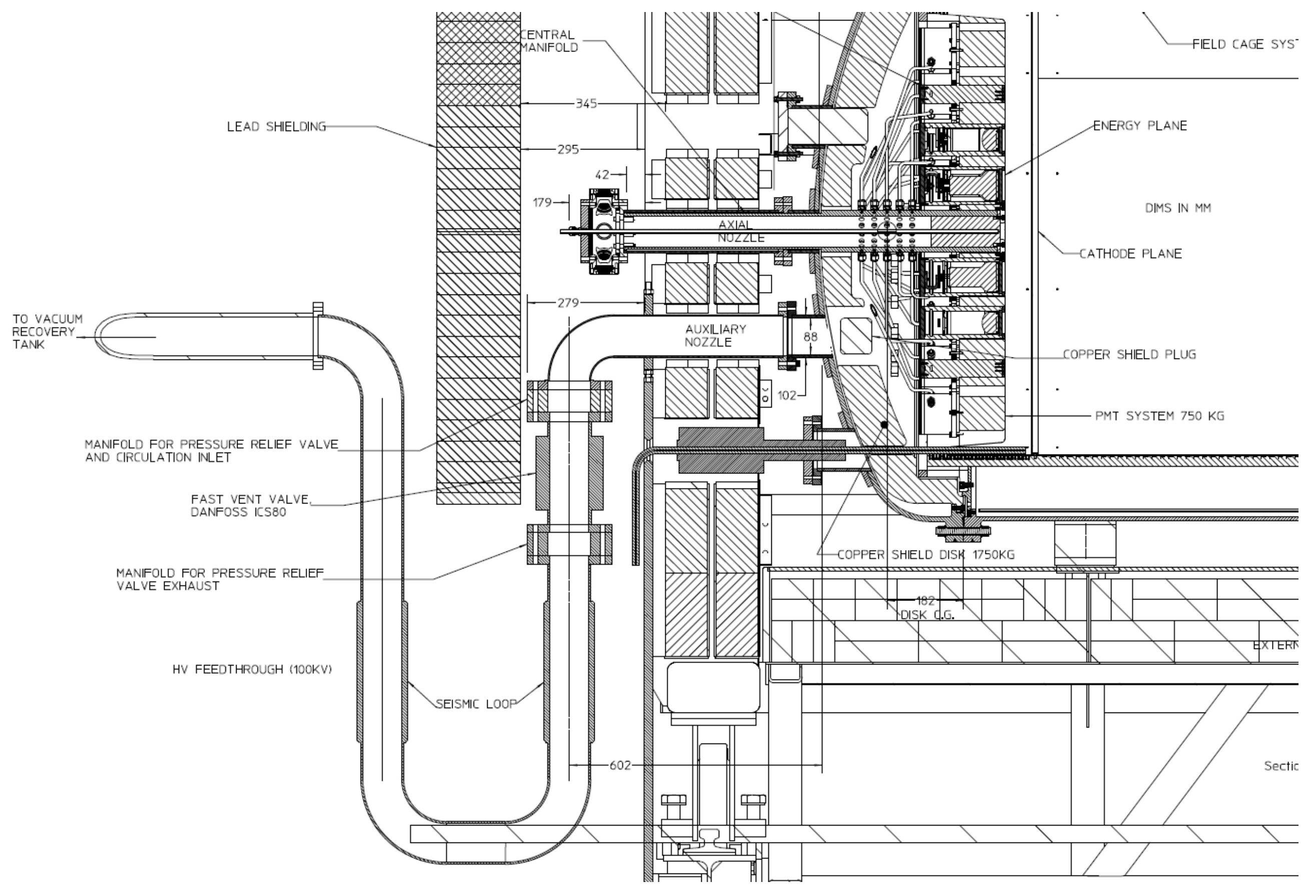
Thickness required for external loading

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, which will be vertically upward, counteracting weight of valve and other components on the nozzle extension.



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.

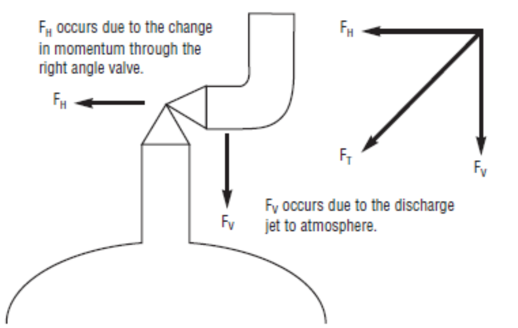


Figure 6-7. Reactive Force

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 (in^2)
- P_2 = Static pressure at the point of discharge (psig)

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

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

$$F_V := A_0\cdot P_2 \qquad F_V = 21.5\,lbf \qquad F_V = 95.5\,N$$

$$F_T := F_H + F_V \qquad F_T = 2364\,N$$

Moments:

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