Maximum Pressure in the Tohoku Bubble Chamber Magnet System

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Table of Contents

INTRODUCTION

The Tohoku Bubble Chamber magnet system consists of two vacuum independent superconducting magnet/LHe storage dewar systems. An estimated 80 liters of LHe is in each magnet cryostat while the bulk of the reservoir is in two 1300 liter storage dewars above the coils. Four possible heat sources exist.

1. Condensing air on the cryostat and storage dewar during a catastrophic vacuum failure.

2. Gaseous helium conduction heat load after a liquid helium leak into the vacuum space.

3. Liberation of all or part of the total 11.5 MJ of stored energy into one or both of the magnets.

4. Freezing LN2 on the cryostat or storage dewar from a rupture in the LN2 system.

Three relief devices are provided for unusual helium venting requirements. They are listed below.

1. 1" Circle Seal Check Valve, 280T-8PP-5, set at 5 psig; $C_v = 14.2$.

2. 1-1/4" Circle Seal Check Valve, 249B-10PP-8, set at 8 psig; $C_v = 23.2$.

3. 4" Fike Rupture Disk, set at 20 psig.

Both check valves are plumbed off a common 1" pipe. All three relief devices empty into a 6" schedule 5 vent pipe which extends outside the building. The two check valves are intended to vent excesses of helium gas generated during filling and normal magnet discharge so that the 4" rupture disk is not continually being replaced. The smaller 1" stainless steel check valve is specifically designed for cryogenic use, while the larger 1-1/4" brass valve serves as a backup and provides full relieving up to the capacity of the 1" pipe.

HEAT LOADS INTO HELIUM

Condensation:

The best available information regarding a condensing air heat load on a liquid helium surface is found in Fig. 6.3 of the NBS Technology of Liquid Helium Monograph (#111). For a vessel wrapped with 1" of superinsulation the heat flux = 400 BTU/hr-ft^2 = 117 watts/ft^2. It is assumed that our magnet/dewar system has at least a 1" equivalent of superinsulation. All liquid helium surfaces are surrounded by a LN_2 shield which has only small holes in it. The areas are calculated below.

Magnet; $2\pi(23.75+36.07)\times 10.5+2\pi(36.07^2-23.75^2) = 8580 \text{ in}^2$

Dewar; $2\pi x 24 x 50 + 2\pi x 24^2 = 11160$ in^2

Interconnecting line; 20ft x 12 x π x 2.375 = 1791 in^2

Sum of areas = 21530 in^2

To take into account the area of gussets, small pipes, etc., assume

Total area = 25000 in^2

Total Condensation Heat Load Per System = 117 w/ft^2 x $\frac{25000}{144}$ = 20,000 watts

1 watt = 1.4 liter/hr

1 liter LHe = 24.7 ft^3 (0°C)

26.6 ft^3 (70°C)

20,000 watts = 28000 liter/hr per coil

= 12410 SCFM (70°F)

= 3 minutes to vaporize all 1380 liters in one system.

Gaseous Helium Conduction Head Load:

$$Q = UA(300 - 4.2)$$

is the general formula for conduction between two surfaces at 300°K and 4.2°K where

Q = watts

U = thermal conductivity/gap watts/cm$^{2\circ}$K

A = area (cm^2)

From the NBS Monograph 631 the thermal conductivity of helium at 300°K and 1 atm is $k = 1.55 \times 10^{-3}$ watt/cm$^\circ$K. Thermal conductivity is virtually independent of pressure in this range but decreases rapidly with temperature. Thus our assumed value of k is overestimated for all temperature and pressure combinations under consideration. An average distance of 2-1/2 inches between the 4.2 K cold surface and the warm outer wall of the vacuum jacket was used.

The gaseous conduction heat load is given by

$$A = \frac{1.55 \times 10^{-3}}{2.5 \times 2.54} \times 25000 \text{ in}^2 \times 2.54^2 \ (300-4.2)$$

Q = 11600 watts = conduction of heat load.

Magnetic Stored Energy:

Total stored energy = 11.5 MJ

One can only make estimates for a worse case release of stored energy. The coil is designed to never quench, but a backup protection circuit has been provided which will discharge the magnet through a large dump resistor in approximately 50 sec such that the adiabatic hot spot temperature of the conductor is less than 300°K. The most likely outcome of a quench due to conductor motion would be an energy dump. At some point during the discharge it is expected that the normal section would "cold end" recover and only a small fraction of the total energy would be released into the liquid helium. It is anticipated, however, that loss of vacuum would cause a normal magnet trip rather than a quench due to excessive lead flow. The thermal temperature reserve of the superconductor is greater than the temperature rise in the helium caused by a pressure rise before the rupture disk opened.

The cooling channels between coil layers are narrow (0.050") and horizontal and are, therefore, highly susceptable to vapor locking. This fact not only reduces our confidence in the actual cold end recovery current value, but it makes any prediction of quench propagation exceedingly difficult. Therefore, only crude estimates of power generation in the coil can be made and are given below. In any event, these models would be invalid if a massive catastrophic arc were to occur.

Each coil has 43,500 ft of cable

43500 ft x 12 x 2.54 x 0.014 in^2 = 7300 in^3 = 12×10^5 cm^3

Weight of cable = 0.754 gm/cm

Total weight of wire in one coil = 1.0×10^6 gm

Assuming the wire was all copper (8.95 gm/cm^3) the coil weight = 1.07×10^6 gm

$$\Delta H = \frac{11.5 \times 10^6}{1.0 \times 10^6} = 11.5 \text{ j/gm}$$

Using the enthalpy of copper found in the Compendium of Material Properties at Low Temperature,

ΔH_{vap} (He) = 20.42 j/gm = 2.55 j/cm^3

ΔH_{freeze} (N2) = 27.6 j/gm = 22.36 j/cm^3

1 gram of freezing nitrogen boils 1.35 grams of LHe

or

1 liter of freezing nitrogen boils 8.8 liter of LHe

The maximum possible failure is complete rupture of the 3/4" O.D. x 0.049" wall tubing which surrounds both the magnet cryostat and the storage dewar. Any failure in the LN2 dewar itself is not expected to pose any significant threat to the helium system. The LN2 dewar vacuum is isolated by valve MV-08-N and is only connected through the long LN2 supply line. Most all of the LN2 would be vaporized by this line and vented through the vacuum reliefs.

From the section on maximum pressure in the LN2 system we find 40 psig is greatest possible pressure. Flow through the LN2 return line is thought to be the line with minimum impedance. The supply line is much longer and is made from a smaller diameter flex line.

Using the formula from Crane

$$w(\text{lbs/sec}) = 0.525\, d^2 \left(\frac{\Delta P \rho}{K}\right)^{1/2}$$

d = 0.652"

ΔP = 40 psi

ρ = 50.4 lbs/ft^3

$K = \frac{fL}{D}$ = total resistance coefficient

f_{min} = 0.026 from the Moody Diagram

L = 8 ft = 96"

K = (0.026 x 96)/0.652 = 3.8	for the straight run
K = 5 x 30 x 0.026 = 3.9	for five elbows
K = 1.0	for the exit
K = 0.5	for the entrance
K = 340 x 0.026 = 8.8	for the globe valve MV-08-N

$T_{max} = 103^oK$

All the energy dumped into one coil produces at most an average temperature of 103^oK. This neglects any heat absorbed by helium, G-10, or stainless steel.

At ~ 100 K the coil resistance ~ 35 Ω x 0.21 = 7.4 Ω

Maximum power generation of the entire coil when normal at its maximum average temperature =

$$P = 700^2 \times 7.4 = 3.6 \times 10^6 \text{ watts}$$

If the coil were at a uniform temperature of 10^oK, the power generation at 700 amps would be roughly

$$P \sim 700^2 \times \frac{35}{100} \times 2 = 3.4 \times 10^5 \text{ watts}$$

where 2 is an approximate value to the correction for magnetoresistivity of the copper with a RRR=100. Note that the resistance of a 10^oK normal coil under full field is roughly the same a the 0.85 Ω dump resistor.

Assume the more likely event of low liquid level causing a quench in the top outer layers. The outer few layers "see" an average magnetic field of ~ 1B1 = 15 Kg. Using a 50% increase for magnetoresistivity, one layer which is normal for 1 foot of circumference has a resistance of

$$R = \frac{35\ \Omega}{43500\ \text{ft}} \times \frac{1}{100} \times \frac{45\ \text{turns}}{\text{layer}} \times 1\ \text{ft} \times 1.5 = 5.4 \times 10^{-4}\ \Omega$$

$$\text{Power} = 700^2 \times 5.4 \times 10^{-4} = 266 \text{ watts}$$

Quench voltage sensitivity is between 50 mv and 500 mv.

At 700 amps for 10^oK conductor in the low field outer layers we need,

6 ft normal for a 50 mv sensitivity trip

60 ft normal for a 500 mv sensitivity trip

The magnet would be, therefore, normally expected to start its discharge with the normal zone producing only small levels of power.

LN2 Leak:

The liquid nitrogen system holds 70 liters. If this were to rupture, nitrogen would freeze on the outer wall of the liquid helium system causing a potentially large heat load.

$K_{total} = 18.0$

Then,

$w = 2.4$ lbs/sec = 1070 gm/sec = 1.3 liter/sec

This will boil 11.7 liter/sec of LHe = 1460 gm/sec

Equivalent heat input to LHe = 2.97×10^4 watts
= 18400 SCFM (70^oF)

70 liters total / 1.3 liter/sec = 53 sec

Thus under worse case it will take almost a minute to empty the LN2 dewar into the vacuum shell. In actuality this heat load would certainly be less because even this postulated worse case leak would only spray on a limited surface area. The film boiling heat transfer coefficient for LHe with a $\Delta T = 73^oK$ is ~ 4 watts/cm^2. See the attached curve. To obtain this heat load of 3×10^4 watt over 8 sq. ft. would need to be covered.

SIMPLE CALCULATIONS AND ESTIMATES

1. 3.52×10^6 joules are required to vaporize all 1380 liters of liquid helium at 1 atm constant pressure.

2. Only 2.0×10^5 are required to vaporize all 80 liters in the magnet cryostat. This is 1.7% of the total stored energy.

3. At 1 atm the ratio of specific volume of saturated gas to saturated liquid is 7.4. 1 cm^3 of vaporizing liquid displaces 6.4 cm^3 of liquid. The energy required to expel all liquid helium from the magnet cryostat

   $= 2.0 \times 10^5/7.4 = 2.7$ x10 10^4 (0.023% of the total energy)

4. Calculate the equilibrium temperature and pressure for the total stored energy released into a completely sealed system (i.e. all relief valves as plugged)

   $11.5 \times 10^6 = (M\Delta H)_{metal} + M\Delta U_{He}$

Each system has

1.73×10^5 gm of helium

1.0×10^6 gm of copper

8.0×10^5 gm of stainless steel in the magnet cryostat

4.8×10^5 gm of stainless steel in the helium dewar

Assume all the metal is stainless steel. Copper has a somewhat larger enthalpy than stainless steel for the temperature range of interest. Guess a final temperature for the metal, calculate ΔU_{He}, then compare the helium temperature with the metal temperatures. Repeat the process until both temperatures are the same. Thermodynamic equilibrium is approximately 90 atm (1300 psi) and 26^{o}K. At this temperature the metal has absorbed only slightly more than 1% of the total energy.

COMPUTER ANALYSIS

Otto Davidson has made several computer runs to estimate pressures and flow rates using a program he developed which utilizes the NBS helium properties subroutines. At the present time this program is not in its final form and does not directly simulate our conditions. Two types of models were used.

1. Constant pressure venting from a vessel subject to a specified heat input.

2. Venting from a magnet cryostat into a sealed buffer dewar volume: The heating rate is unimportant since we are interested in the pressure rise versus the total energy input. The magnet mass is in thermodynamic equilibrium with the helium in the magnet cryostat and both vessels are at nearly identical pressures. This model will closely approximate the case of a catastrophic release of energy into the magnet. Depending upon initial conditions the helium in the cryostat is quickly expelled because rapidly rising temperature (specific volume) and forces the helium in the storage dewar into a supercritical or compressed liquid state.

Case #1 Constant 20 psig Pressure Venting with 20,000 Watt Heat Load:

Initial conditions

Helium Mass = 173 Kg (1380 liters)

Mass of Copper = 980 Kg (2160 lbs)

Specific Volume of Helium = 8.18 cm^3/gm

Temperature = 4.68^{o}K

Pressure = 34.7 psia (20 psig)

The specific volume was chosen to be the average specific volume when the 30 liter ullage space is included. Pressure is actually derived from the initial temperature and specific volume.

Results

Initial mass flow rate = 760 gm/sec

Peak mass flow rate = 1470 gm/sec (after 72 sec or 1.44×10^6 j total energy)

Case #2 Cryostat with Sealed Storage Dewar:

Initial Conditions

Mass of helium in cryostat = 10 Kg (80 liter)

Mass of copper = 1070 Kg (2360 lbs)

Temperature = 4.2 K

Specific volume of helium = 8 cm^3/gm

Mass of helium in storage dewar = 163 Kg (1300 liter)

Specific volume of helium in dewar = 8.2 cm^3/gm
(This takes into account the 30 liter ullage space)

Temperature of storage dewar = 4.2 K

Mass of metal in storage dewar = 0

Initial pressure for both vessels = 14.4 psia ~ 1 atm

Results

Final pressure 65 psia = 50 psig
ΔP = 51 psia

P = 21.8 psig (> rupture disk setting) with 1.0×10^5 joules released (0.8% of the total). At this point 81% of the helium mass in the cryostat has been expelled.

P = 58 psig after 5.75×10^5 joules (5% of the total) of energy has been released.

Case #3 Cryostat with Sealed Storage Dewar

Initial Conditions

Same as run #2 except

Temperature = 5.3^oK

Specific volume of helium = 9.62 cm^3/gm

Calculated initial pressue = 44 psia (29 psig)

Results

Final pressure = 97 psia (82 psig)
ΔP = 53 psia

Note that the pressure rise for case 2 and 3 is ~ 52 psi. Any initial condition between these two cases would also be expected to have this 52 psi pressure rise.

Conclusion:

The worse case scenario would be a heat input of 20,000 watts from loss of vacuum raising the pressure up to the rupture disk setting followed by a very unlikely massive quench. The maximum pressure possible would be 20 + 52 = 72 psig. A 30,000 watt freezing nitrogen load would produce the same result since the rupture disk will pop at 20 psig. Following sections show that the vent system is more than adequate for the 30,000 watt case.

For the helium system,

$$P_{max} \leq 72 \text{ psig} = 87 \text{ psia} \sim 6 \text{ atm}$$

This pressure rating assumes a small pressure drop in the vent system which will be verified in the next section.

VENTING FLOW RATE

The main helium vent is a 4" Fike rupture disk connected to a 6" schedule 5 pipe. For details of this piping assembly see Bob Wands note "Mechanical Analysis of the Tohoku Bubble Chamber Magnet Helium Dewar Vent Piping". A worse case 100 psig 10°K gas reservoir was used in this model to estimate maximum piping forces and stresses.

Calculation of Pressure Drop for 20,000 Watt Condensation Load:

20,000 watts = 1030 gm/sec vaporized helium
at 1 atm
(0 psig) 12410 SCFM (70°F)

Otto Davidson's program has demonstrated that 20,000 watts relieving at 20 psig requires a maximum mass flow of 1470 gm/sec (18800 SCFM, 70°F) at some point in the venting process. Thus we must size the venting for this higher mass flow rate. A worse caase 30,000 watt freezing nitrogen load would require a peak mass flow rate of 1.5 x 1470 = 2200 gm/sec.

Size the Fike rupture disk without the vent line as a first approximation. Use Fike formulas

$$\text{Critical Pressure Ratio} = \left[\frac{2}{K+1}\right]^{[k/(k-1)]}$$

$$= 0.487$$

where $k = C_p/C_v = 1.67$ which is a very good value down to ~ $15^{o}K$.

$$\text{Pressure ratio} = \frac{14.7}{20+14.7} = 0.424$$

The flow is sonic since this pressure ratio is less than the critical pressure ratio. Sonic flow simplifies to

$$Q_s = (22772\ aKC_2\ P_o)/(TM)^{1/2}$$

where

Q_s = flow rate in SCFM $60^{o}F$

a = flow areas = $\pi/4 \times 4^2 = 12.57\ in^2$

P_o = relieving pressure (psia)

M = molecular weight = 4

T = temperature ^{o}R

K = ASME factor = 0.62

$$C_2 = \frac{520}{3600}\left[k\ \frac{2}{k+1}^{(k+1)/(k-1)}\right]^{1/2} = 0.105$$

$k = C_p/C_v = 1.67$

This simplifies to

$$Q_s = \frac{9317\ P_o}{(T)^{1/2}}\ \text{SCFM}$$

4" Fike Rupture Disk Flow Rate

P_o (psia)	Temp. (^oK)	Temp. (^oR)	Q_s (SCFM, 60^oF)	Q Liquid Liter Equivalent per sec	Q gm/sec
34.7	293	527	14100	8.8	1100
34.7	200	360	17000	10.7	1335
34.7	100	180	24100	15.1	1890
34.7	77	139	27400	17.2	2150
34.7	25	45	48190	30.3	3782
34.7	10	18	76200	47.7	5970
44.7	293	527	18160	11.3	1417
44.7	200	360	21950	13.8	1720
44.7	77	139	35300	22.2	2770
44.7	25	45	62080	38.9	4860
44.7	10	18	98160	61.5	7690
87	293	527	35350	22.1	2760
87	200	360	42720	26.8	3350
87	77	139	68700	43.1	5390
87	25	45	1.21×10^5	75.7	94610
87	10	18	1.91×10^5	120	14960

Pressure Drop of Entire Helium Vent Line

The vent line drawings and component summary have been reproduced here from B. Wands note on vent line forces. The calculations will be repeated here for different pressure and temperature conditions. Only the "A" vent will be considered because this one has the greatest pressure drop.

The expression for compressible flow with a relatively large pressure drop is given by the modified Darcy equation. Refer to Cranes Flow of Fluids, Technical Paper 410, equation 3-22. The flow is assumed to be adiabatic which is reasonable for sonic flow.

$$q'_m = 678\ Y d^2\ (\Delta P\ P'_1 / K\ T_1\ S_g)^{1/2}$$

where

$K = f \frac{L}{D}$ the total resistance coefficient = 8.0

P'_1 = inlet pressure in psia

d = inside pipe diameter = 6.407 inches

S_g = specific gravity relative to air = 0.137

T_1 = inlet temperature ^{O}R

Y = net expansion factor for piping system

q'_m = flow rate SCFM ($60^{O}F$)

ΔP = pressure drop (psi)

Use example 4-21 in Crane as a reference if desired. See appendix 4 for values of K, f, and Y. The flow is assumed to be turbulent so that the fraction factor f = 0.015. All resistance coefficient are calculated in terms of the larger 6" pipe. For example the value of K for the opening is given by 0.5 x $(6.407/4.334)^4$ = 2.4. From the previous section on Fike rupture disks without piping we found the critical pressure ratio to be 0.487. This value changes for our piping system. Unfortunately, we must use the charts in Crane for k = 1.4 rather than 1.67. This is not expected to make a large difference, however.

From page A-22 for k = 1.4 and K = 8.0

$\frac{\Delta P}{P'_1}$ = 0.762 = limiting sonic factor

Y = 0.685

Thus, the expression for flow rate simplifies to

$q'_m = 18210\ (\Delta P\ P'_1/T_1)^{1/2}$ SCFM for subsonic flow

$q'_m = 15900\ P'\ (1/T_1)1/2$ SCFM for sonic flow

If $\Delta P/P'_1 = (P_{inlet} - P_{outlet})/P_{inlet} > 0.762$, the flow is sonic

Flow Rates Through the Helium Vent Line

P'_1 (psia)	T_1 ^oK	T_1 ^oR	Flow	Q SCFM 60^oF	Q Liquid Liter per sec	Q gm/sec	Q* SCFM 60^oF
34.7	293	527	subsonic	20900	13.1	1640	12960
34.7	200	360	subsonic	25280	15.8	1980	15670
34.7	100	180	subsonic	35760	22.4	2800	22170
34.7	77	139	subsonic	40690	25.5	3190	25230
34.7	10	18	subsonic	1.13×10^5	70.8	8850	70060
44.7	293	527	subsonic	29050	18.2	2275	18010
44.7	77	139	subsonic	56560	35.4	4430	35070
87	293	527	sonic	60260	37.8	4720	37360
87	77	139	sonic	1.17×10^5	73.3	9160	72540
87	10	18	sonic	3.26×10^5	204	25530	2.02×10^5

*Note: As can be seen, the calculated flow rates for the piping system are greater than the flow rates for the Fike rupture disk. This difference is due to the required ASME 0.62 derating factor for rupture disks caused by the broken disk partially blocking the flow passage.

$$Q^* = 0.62\ Q$$

and is thought to be an overly conservative correction to the flow rate in the piping system.

When compared to flow rates for a stand alone rupture disk our piping system flow predictions Q* are 8% lower to 6% higher. Higher piping flow rates occur because 4" was used in the rupture disk formula, and 4.33" was used in the piping system calculation.

Conclusion

The piping system adds negligible resistance to the 4" Fike rupture disk venting directly to atmosphere.

TEMPERATURE RISE DURING VENTING

Vent rates are dependent upon the temperature of the exiting gas. It will be assumed that the helium leaving the dewar is at least 10^oK during a loss of vacuum or quench. It is instructive to calculate the initial mass of gas required to cool the vent pipe to liquid nitrogen temperature and also to calculate the temperature rise in the helium itself.

Required Helium Mass for Pipe Cooldown to 77^oK:

Jacobs presents a very simple method for cooldown requirements in Vol. 8 of Advances in Cryogenic Engineering.

Mass of stainless steel = 57' x 9.3 lbs/ft = 530 lbs

Add another 150 lbs for flanges

Total mass of stainless steel = 680 lbs = 3.1×10^5 gm

Jacobs Fig. 1 gives a specific mass requirement of 0.1-0.015 = 0.085 to cool stainless steel from 300 to 80°K where the specific mass is the ratio of gm of He required/gm of stainless cooled. Thus

$$3.1 \times 10^5 \times 0.085 = 2.6 \times 10^4 \text{ gm} = 211 \text{ liquid liters}$$

are required to cool the entire pipe to 80°K. This method assumes the heat of vaporization is available for cooling which in our case is not true. But,

$$\Delta H\ (4.2 \text{ liquid} \rightarrow 10^\circ K) = 55 \text{ j/gm}$$

$$\Delta H\ (10^\circ K \text{ to } 300^\circ K) = 1510 \text{ j/gm}$$

so that the error in using this method is very small.

Helium Temperature Rise During Venting:

The Seider Tate equation is used to calculate the heat transfer coefficient.

$$N_u = 0.023\, R_e^{0.8}\, P_r^{0.4}\, (\mu_w/\mu)^{0.14}$$

where

N_u = Nusselt number = hD/k

R_e = Reynolds number = $DV\rho/\mu = 4\,\dot{m}/\pi D\mu$

P_r = Prandtl number = $C_p\, \mu/k$

All properties are evaluated at their average bulk temperature except μ_w which is evaluated at the wall. The factor $(\mu_w/\mu)^{0.14}$ is a modification to the more famous Dittus Boelter equation for cases with large temperature differences across the film.

P_r = 0.71 For all cases this is a close approximation

D = 6.407" = 16.27 cm

Rearranging and simplifying,

$$h = 1.233 \times 10^{-3}\, kRe^{0.8}\, (\mu_w/\mu)^{0.14}$$

Helium Properties

Temperature ^{o}K	Pressure	μ(poise) gm/cm-sec	ρ gm/cm^3	k w/cm ^{o}K
$10^{o}K$	14.7	22.6×10^{-6}	5.02×10^{-3}	1.75×10^{-4}
$10^{o}K$	44.7	24.2×10^{-6}	1.60×10^{-2}	1.84×10^{-4}
$10^{o}K$	88.2	27×10^{-6}	3.5×10^{-2}	2.05×10^{-4}
80 K	14.7	85×10^{-6}	6.09×10^{-4}	6.35×10^{-4}
80 K	44.7	85×10^{-6}	1.82×10^{-3}	6.35×10^{-4}
80 K	88.2	85×10^{-6}	3.62×10^{-3}	6.35×10^{-4}
$300^{o}K$	14.7	199×10^{-6}	1.63×10^{-4}	1.55×10^{-3}
$300^{o}K$	44.7	199×10^{-6}	4.87×10^{-4}	1.55×10^{-3}
$300^{o}K$	88.2	199×10^{-6}	9.73×10^{-4}	1.55×10^{-3}

The total heat transfer rate is given by

$$Q = hA\ (T_w - T_f)$$

$A = 8.88 \times 10^4\ cm^2$

T_w = Wall temperature

T_f = Fluid temperature

An approximation for the outlet temperature is $Q/\dot{m}$.

For large temperature increases the model is inaccurate. Flow rates decrease with temperature as $T^{-1/2}$ and the thermal conductivity of helium rises with increasing temperature. Nevertheless, the following table is instructive as a guide to the temperature rise.

Fluid Temp ^{o}K	Pipe Temp ^{o}K	Inlet Pres psia	$\dot{m}$ gm/sec	R_e	h $w/cm^{2\,o}K$	Q MW	$\Delta H_{\circ}$ Q/m	T_{final} ^{o}K
10	300	87	14960	4.68×10^7	0.429	11.0	739	150
10	300	44.7	7690	2.41×10^7	0.239	6.2	800	160
77	300	87	5390	4.96×10^6	0.201	4.0	739	150
77	300	44.7	2770	2.55×10^6	0.118	2.3	844	170
10	80	87	14960	4.68×10^7	0.381	2.4	158	40
10	80	44.7	7690	2.41×10^7	0.213	1.3	172	42
25*	80	87	9460	4.4×10^7	0.565	2.8	298	65
25*	80	44.7	4860	2.3×10^7	0.327	1.6	329	75

*Note: These numbers are based on a second iteration using average fluid properties and a mass flow rate of 25°K fluid for the preceding two cases of 10°K inlet and 40°K outlet properties.

It is easily seen that the heat transfer rates between the pipe and the venting helium are enormous (several megawatts). Condensation heat transfer rates are calculated next. Figure 6.3 in the NBS Monograph 111 gives condensing heat transfer rates of 2 w/cm^2 to 6 w/cm^2. Using the upper value, $Q_{max} = 6 \times 8.9 \times 10^4 = 5.3 \times 10^5$ watts.

Thus we get

Initial Fluid Temp °K	Avg. Fluid Temp °K	Inlet Pres. psia	$\dot{m}$ gm/sec	ΔH j/gm	T_{final} °K
10	25	87	9460	56	20
10	25	44.7	4860	109	30

Conclusions:

1. ~ 210 liters of helium are required to cool the vent pipe to 80°K.

2. Heat transfer rates to the helium are limited by the air condensation heat loads and not the convective heat transfer to the venting helium.

3. If one assumes a 10°K inlet temperature of helium, a mass flow rate based on 25°K properties is conservative.

4. Venting can be visualized as a two step process.

   First, vent 210 liters to cool pipe to 77°K, assume 200°K helium properties.

   Second, vent the remaining 1170 liters using 25°K helium properties.

   Total time to vent all 1380 liters = 45 sec at 44.7 psia (3 atm)
   23 sec at 87 psia (6 atm)

5. The vent system is adequate for all heat sources. The 30,000 watt freezing nitrogen heat load could vent at 34.7 psia inlet using a worse case 77°K average gas temperature assumption.

CIRCLE SEAL RELIEF VALVE VENT RATES

These two helium relief valves are used to protect the rupture disk against small perturbations. Use the formulas as found in the Swajlok Manual.

$$Q = 16.05\ C_v\ (P_1^2 - P_2^2 / S_g\ T)^{1/2} \qquad \text{non choked}$$

$$Q = 13.61\ C_v\ P\ (1 / S_g\ T)^{1/2} \qquad \text{choked}$$

Q = SCFM

P = psia

$T = {}^{o}R$

S_g = specific gravity relative to air = 0.137

From the previous section the critical pressure ratio = 0.487. The flow is choked for all inlet pressures > 7.5 psig. The two valves are plumbed off a common 1" pipe and relieved into the 6" vent line.

Valve RV-03-H: 1", 5 psig, C_v = 14.2, cryogenic, stainless steel

RV-02-H: 1-1/4", 8 psig, C_v = 23.2, brass

Calculate flows based on only the capacity of the larger valve at the rupture disk set pressure, 34.7 psia.

$$Q = \frac{2.96 \times 10^4}{T^{1/2}} \quad \text{SCFM @ } 70^{o}F$$

Flow Capacity of Circle Seal Relief Valve

Temperature ^{o}K	Q SCFM(70^{o}F)	Q liter/sec	Q gm/sec
293	1290	0.81	101
77	2510	1.57	197
10	6980	4.37	546
4	14800	9.27	1160

Conclusions:

The Circle Seal relief valves probably would not keep the rupture disk from blowing under a loss of vacuum condition (20,000 watts = 12410 SCFM). They would be very effective, however, if an "ordinary" (small) quench were to occur and the magnet started a normal discharge.

BACK PRESSURE IN VENT LINE

A 20 psig relief pressure is predicated on the assumption of inconsequential back pressure in the vent line. Examine the worse case of 4^{o}K helium gas and 77^{o}K nitrogen gas venting through the Circle Seal relief valves and instantly warming to 293^{o}K. Use the standard Darcy Weisback formula.

From Crane's Paper #410

$$\Delta P = 7.26 \times 10^{-9} \, \frac{f \, L \, T \, (q'_h)^2 \, S_g}{d^5 \, P'} \qquad \text{(Eq 3-5)}$$

S_g = 0.137 for helium; 0.97 for nitrogen

d = 6.407 inches

T = 527 ^oR

L = 57 ft x 3.5 = 200 ft
where 3.5 is an estimate for the equivalent effective length due to bends, exits, etc.

f = friction factor = 0.016 for helium; 0.018 for nitrogen

$q'_h = 8.9 \times 10^5$ SCFH (1160 gm/sec) for helium

$= 5.22 \times 10^4$ SCFH (476 gm/sec) for nitrogen

$P_1' = 19.7$ psia

Nitrogen flow rates are based on 50 psig, 77^oK gas flowing through relief valve CV-02-N ($C_v = 11.4$). This is an absolute worse case. The first relief valve on the nitrogen dewar is actually set at 20 psig.

For helium,

$$R_e = \frac{4\dot{m}}{\pi D \mu} = \frac{4 \times 1160 \text{ gm/sec}}{\pi \times 6.407 \times 2.54 \times 199 \times 10^{-6}} = 4.56 \times 10^5$$

For nitrogen,

$$R_e = \frac{4\dot{m}}{\pi D \mu} = \frac{4 \times 476 \text{ gm/sec}}{\pi \times 6.407 \times 2.54 \times 1.8 \times 10^{-4}} = 2.07 \times 10^5$$

Refering to the Moody diagram the values; the friction factors are confirmed.

Results and Conclusions:

Pressure Drop = Back Pressure = 6.3 psi for helium
= 0.17 psi for nitrogen

No significant back pressure can occur for the nitrogen exhaust. Using worse case calculations a 6 psi back pressure can develop for the helium exhaust. If a more reasonable 10^oK helium gas were assumed, the back pressure would be reduced to 1.4 psi. In any event, the safety of the magnet is completely unaffected by a change of rupture disk pressure to 26 psig.

MAXIMUM PRESSURE IN THE LN_2 SYSTEM

It is possible to valve off the normal Circle Seal vent valve, CV-02-H. Then two relief valves which vent to atmosphere, RV-06-N and RD-04-N, still protect the system. RV-07-N protects the fill line.

RV-06-N	1" Circle Seal Relief Valve	533T-8M-20 @ 20 psig
RD-04-N	3/4" Fike Rupture Disk	Set @ 50 psig
RV-07-N	3/4" Circle Seal Relief Valve	533B-6M-30 @ 30 psig

Loss of vacuum results in a gaseous conduction heat load for helium or air condensation plus conduction for a air leak. The LN_2 system is based on the thermal syphon principle. The supply line is essentially insulated and feeds the shield/supports from the very bottom. Thus no vapor locking can be expected. This means that the effective area for heat transfer is in fact the area of the nitrogen shield rather than only the area of the LN_2 dewar.

Effective area = 25000 x 1.2 = 30,000 in^2

where 25000 in^2 was the surface area of the helium system and the factor 1.2 compensates for the slightly larger size of the nitrogen shield.

Helium Conduction Loads:

$Q = UA\,\Delta T$ where $U = K/L$

$k = 1.55 \times 10^{-3}$ watt/cm°K 300°K worse case

$\Delta T = 300-77$

$A = 30{,}000 \times 2.54^2 = 1.94 \times 10^5\ cm^2$

$L = 1.5" \times 2.54 = 3.8$ cm

$Q = 1.77 \times 10^4$ watts

Air Conduction:

Use $k = 2.62 \times 10^{-4}\ w/cm^2\ °K$ 300°K worse case

$Q = 2980$ watts

Air Condensation:

To my knowledge this heat transfer coefficient does not appear in the literature. Estimates can be made using Nusselts method, but they do not take into account superinsulation or the fact that only oxygen is condensing. The following values can be found, however

$\frac{BTU}{hrft^2}$	$\frac{Watt}{ft^2}$	Source

400	117	NBS Mono 111	Total heat flux to 4.2^oK surface surrounded by 1" of superinsulation, NBS Monograph #111
~10000	2930	NBS Mono 111	Total heat flux to a bare 4.2 K surface
575	169	Barron's p. 504	Condensation load to an uninsulated LOX transfer line
3500	1026	Barron's p. 504	Condensation load to an uninsulated LH_2 transfer line

Clearly superinsulation and low enough temperatures for condensation have major impact on the overall heat flux. The best guess for heat flux would be 400 x 0.2 = 80 BTU/hrft^2 where 0.2 represents the 20% fraction of condensable oxygen. However, 400 BTU/hrft^2 = 117 w/ft^2 will be used.

$$1 \text{ liter } LN_2 = 22.8 \text{ ft}^3 \text{ at } 0^oC$$
$$24.6 \text{ ft}^3 \text{ at } 70^oF$$

Heat of vaporization = 1.607×10^5 j/liter

1 watt = 0.0224 liter/hr = 9.18×10^{-3} SCFM (70^oF)

$$Q = \frac{117 \text{ w/ft}^2}{144} \times 30000 \text{ in}^2 \times 9.18 \times 10^{-3}$$

$$= 233 \text{ SCFM } 70^oF$$

Flow through the Fike rupture disk (RD-04-N):

Use previous Fike rupture disk formula

$$Q = 22772\ aK\ C_2\ P_o\ /\ (TM)^{1/2}$$

$a = 0.442$ in^2

$K = 0.62$

$P_o = 65$ psia (50 psig)

$T = {}^oR$

$M = 28$

C_2 = gas flow constant for sonic flow = 0.0989
($k = C_p/C_v = 1.4$)

Flow through rupture disk,

$Q = 330$ SCFM @ $293^{O}K$
632 SCFM @ $80^{O}K$

Flow through the Circle Seal LN_2 Dewar Relief Valve (RV-06-N):

C_v coefficients are not available for this valve, but a flow chart is provided and is reproduced here. With a 20 psig cracking pressure, 200 SCFM is achieved with a 30 psig at $70^{O}F$. From the slope characteristics off the other curves > 250 SCFM should be expected with a 40 psig pressure drop. At the same pressure this Circle Seal valve has nearly the same flow rates as a rupture disk.

Conclusion: The Circle Seal valve should limit the LN_2 dewar pressure to 40 psig for the loss of vacuum condition. The rupture disk provides redundancy and extra flow capacity but is never expected to burst.

160 Liter LN_2 Supply Dewar and Connection Fill Line with Relief Valve (RV-07-N):

The permanent LN_2 is supplied from 160 liter dewars. The connecting line is protected against the trapped volume situation by relief valve RV-07-N. This valve also provides protection against overpressurization of the LN_2 system from a failure in the supply dewar. If the 22 psig relief on the supply dewar were to fail closed, the supply dewar pressure could rise to 170 psig. Upon opening the supply valve solenoid a large volume of high pressure LN_2 could be forced into the LN_2 dewar. The flow rate through the solenoid valve (EV-01-N) is given by,

$$Q = C_v \, (\Delta P/S_g)^{1/2}$$

where

Q = gal/min

ΔP = psi = 170

S_g = specific gravity relative to water = 0.8

C_v = 12.4 gal/min = 47 liter/min = 1160 SCFM ($70^{O}F$)

The normal LN_2 dewar gaseous reliefs could not handle this flow rate. Therefore, the 30 psig relief was selected to bypass most of this flow. Again from the Circle Seal charts we can extrapolate to find a flow rate of >10 gal/minute with a 10 psi pressure rise.

Summary:

The LN_2 system should never rise above 40 psig.

MAXIMUM INTERNAL PRESSURE OF THE VACUUM SYSTEM

The vacuum system has 4 separate reliefs all set at ~ 0 psig which vent into Lab F.

1. PP-01-V 6" Parallel Plate Relief
2. PP-02-V 2" Parallel Plate Relief
3. MV/RV-03-V 1" Cryolab Pump Out/Relief
4. MV/RV-06-V 1" Cryolab Pump Out/Relief

Introduction:

The maximum internal pressure of the vacuum system during a fault condition is by far the most complex to estimate. The pressure depends upon

1. The initial helium pressure.
2. The size of the leak.
3. The location of the leak.
4. How much helium is in the system prior to failure.

It is postulated that a major rupture occurs in the magnet cryostat instantly dumping all the helium into the vacuum shell of magnet and dewar. If the failure were to occur up in the dewar, the magnet vacuum shell would probably vapor lock and its surface area would not be available for heating the helium.

Instantaneous Dumping:

Assume that all the helium is instantly distributed throughout the system such that heat transfer has not yet come into play. Then by knowing the initial conditions we can calculate the final state since we can calculate the change in specific volume, and internal energy remains constant. The volume between the helium surface and the vacuum shell is calculated below.

Volume around the O.D. of the helium dewar,

$$38 \times \pi \times 48 \times 2.5 = 1.43 \times 10^4 \text{ in}^3 = 234 \text{ liter}$$

Volume between the heads of helium dewar,

$$2 \times (53.1 \text{ gal} - 37.3 \text{ gal}) = 31.6 \text{ gal} = 120 \text{ liter}$$

Volume of magnet cryostat,

$$10.5 \times \pi (36.07^2 - 23.75^2) = 2.43 \times 10^4 \text{ in}^3$$

Inside volume of magnet vacuum shell,

$$\sim 13 \times \pi \,(44.63^2 - 22.66^2) = 6.04 \times 10^4 \text{ in}^3$$

Use only 70% of the calculated volume between the cryostat and vacuum shell inner wall to account for supports, etc.

Volume between magnet cryostat and vacuum shell =

$$0.7 \times [6.04 \times 10^4 - 2.43 \times 10^4] = 2.53 \times 10^4 \text{ in}^3 = 414 \text{ liter}$$

Total volume = 234 + 120 + 414 = 768 liter

$$\text{Fractional Increase in Volume} = \frac{1300 + 768}{1300} = 1.59$$

Two cases are examined. Both sets of initial conditions are obtained from Otto Davidson computer output.

Case 1: Initial condition obtained after a "sealed volume" quench starting with 1 atm, 4.2°K helium.

$T_i = 4.87\ ^\circ K$	$T_f \sim 4.2$
$P_i = 65$ psia (4.4 atm)	$P_f = 14$ psia ~ 1 atm
$V_i = 7.74\ cm^3/gm$	$V_f = 7.74 \times 1.59 = 12.3$
$U_i = 10.1$ j/gm	

Case 2: Initial conditions obtained after a "sealed volume" quench starting with 44 psia (3 atm), 5.3°K helium.

$T_i = 6.27^\circ K$	$T_f \sim 5^\circ K$
$P_i = 97$ psia (66 atm)	$P_f = 1.8$ atm $= 26.5$ psia
$V_i = 9.09\ cm^3/gm$	$V_f = 9.09 \times 1.59 = 14.45$
$U_i = 16.1$ j/gm	$U_f = 16.1$

Conclusion: A large depressurization occurs for an intantaneous uniform release of helium in the vacuum. The worse case maximum pressure is 1.8 atm = 12 psig.

Heating of Helium and Required Flow Rates:

If the helium were assumed to be dumped instantly and uniformly in the vacuum shell, an extremely large heat flux into the helium occurs. The assumed area of heat transfer = 30,000 in^2 (the area of LN_2) although the area of the vacuum shell is a little larger. The LN_2 will be the first to cool.

$30{,}000\ in^2 \times 1/16 = 1875\ in^3 = 2.75 \times 10^5$ gm = 600 lbs of copper.

Using Jacobs article the required volume of helium to cool the shield = 650 liters using only the latent heat or 43 liters when including the sensible heat. The smaller value is assumed. This means the nitrogen shield cools quickly before the helium has a chance to escape to the vacuum shell walls so that heat transfer is based on 30,000 in^2 rather than 60,000 in^2.

Heat transfer coefficients can be enormous. Based on the review article by R.V. Smith in the BNL 1968 Summer Study

$h \sim 30\ watts/cm^2$ $\quad \Delta T = 300^oK$
$h \sim 4\ watts/cm^2$ $\quad \Delta T = 80^oK$

These are based on pool boiling helium. This is not quite accurate but gives us a starting point.

$Q_{max} = 30{,}000 \times 2.54^2 \times 30 = 5.8 \times 10^6$ watts

With this heat flux all helium must be vented in approximately 1/2 sec (4 seconds for 80^oK surface) To maintain a 1 atm internal pressure rating.

Flow Capacity of 6" Rupture Disk:

Other than pressure testing, internal pressure can only occur from a failure in the helium system. It will be asumed that a parallel plate relief has the same flow capacity as a Fike rupture disk with the same diameters. Using the same formula as before and simplifying for 14.7 psig internal pressure and considering only the 6" rupture disk.

$Q\ (70^oF,\ 14.7\ psig) = 6.17 \times 10^5 / T^{1/2}$ SCFM

where T = ^oR

Helium Exhaust Temp ^oK	Q SCFM	Q liter/sec	Time to Exhaust 1300 liters (sec)
293	2.69×10^4	16.9	77
77	5.24×10^4	32.8	40
10	1.45×10^5	91.1	14
6	1.88×10^5	118	11

Flow rates are linear with absolute pressure (e.g. multiply by 1.5 for 29.4 psig.)

Maximum Allowable Opening of an Intermediate Size Leak:

1. Helium system is at 72 psig, $6^{o}K$ (highest possible pressure).

2. Vacuum system is allowed to rise 19 psig (pressure rating of vacuum system).

3. Flow out the vacuum 6" relief = flow into vacuum system.

Then equate flow rates using Fike rupture disk formula simplifying,

$$\frac{d^2\,(72+14.7)}{(6)^{1/2}} = \frac{6^2\,(19+14.7)}{{}^{o}K^{1/2}}$$

where d is the maximum allowable size of a circular rupture into the vacuum system.

Temperature of Exhaust from Vacuum System ^{o}K	Maximum Size of Circular Opening (inches)
77	2.0
25	2.6
10	3.3

Conclusion:

Vent times are much larger than that required for a worse case catastrophic helium release. Assuming this condition the pressure will be limited by the driving pressure in the helium system, 72 psig maximum. However, this would require the following all occuring simultaneously.

1. Vacuum failure.

2. Massive quench: A vacuum failure is not expected to cause a quench.

3. Failure of major helium system component. All helium system components will have been tested at room temperature to 1.25 x 87 psid. The ultimate strength of 304 stainless steel is ~3 times greater at $4.2^{o}K$ than room temperature.

This postulated failure mode is assumed to be virtually impossible.

A rupture of any of the helium piping or any other opening up to ~3" diameter does not exceed the MAWP of the vacuum system.

MAXIMUM PRESSURE IN THE VACUUM SYSTEM DURING A LN2 SYSTEM FAILURE

If a catastrophic rupture in the LN2 system occured where the spill was outside the LN2 shield, a rapid vaporization of the LN2 would occur. The only possible concern would be a catastrophic rupture of the inner LN2 vessel. Although the 6" rupture disk on the LHe dewar is directly connected to this space, it is considered to be ineffective due to the long length of flexible pipe which is filled with superinsulation. Two reliefs exist on the vacuum shell of the LN2 dewar

1) 1" Pump Out/Relief MV/RV-06-V
2) 2" Parallel Plate PP-02-V

Attached to this document is a boiling curve for liquid nitrogen. We find Q/A = 3 watt/cm^2 for flat plates with a ΔT = 220 K. The surface area of the vacuum shell = 2400 in^2 = 1.56×10^4 cm^2.

Max Heat Load = 4.68×10^4 watts

Rate of LN2 Vaporization = 0.29 liter/sec
= 430 SCFM (70^{o}F)

Using the previous Fike rupture disk formula,

$Q = 22772\ a\ K\ C_2\ P_o\ /\ (TM)^{1/2}$

a = 3.14 in^2 (only the 2" parallel plate relief)

K = 0.62

P_o = 30 psia (1 atm differential)

T = 293^{o}K = 527^{o}R

M = 28

C_2 = sonic gas flow constant = 0.0989

Q = 1080 SCFM (70^{o}F)

Conclusion:

The parallel plate relief on the LN2 is oversized by 2-1/2 times for a worse case LN2 rupture. The internal pressure would be much less than a 1 atmosphere differential.

PRESSURIZATION OF HELIUM SUPPLY DEWAR

A quench will force helium back through the flexible transfer line if valve MV-02-H is open. Calculate the mass flow and pressure rise in the Airco helium supply dewar.

The helium transfer line has the following dimensions.

35 feet of 0.444" I.D. line
14 feet of 0.194" I.D. line

The maximum pressure difference in the line is 72 psig - 0 psig = 72=psid. Using the same equation from Crane as before,

$$q'_m = 678 \ Y \ d^2 \ (\Delta P \quad P'_1 \ / \ K \ T_1 \ S_g)^{1/2}$$

Neglect the resistance of the valve, elbows, transition between 1/2" and 0.194" diameters, the length of 1/2" line, and the extra impedance of the convolutions.

$f \;=\; 0.035$ from the Moody diagram

$$K = \frac{f\ L}{D} = \frac{0.035 \times (14 \times 12)}{0.194} = 30.3$$

$K_{entrance} = 0.5$

$K_{exit} = 1.0$

$K_{total} = 32$

From page A-22 in Crane for $k = 1.4$ and $K = 32$

$\frac{\Delta P}{P'_1} = 0.86 =$ limiting sonic factor

$Y = 0.710$

$\Delta P/P'_1 = 72/87 = 0.83 < 0.86$

The flow is subsonic due to the large resistance of the line.

$d = 0.194"$

$S_g = 0.137$

$T_1 = {}^oR$

$q'_m = 685/T_1^{\,1/2}$

T_1 ^oK	T_1 ^oR	Q SCFM, 60^oF	Q Liquid liter per sec	Q gm/sec
6	11	207	0.13	16
293	527	40	0.025	3.1

For the warm gas flow the bottom 45" of uninsulated 1/4" line in the storage dewar will require no cooldown. The flow rate would only increase by 16% neglecting this length of tube. This is more than compensated by the resistances which were neglected.

From a previous section we found that all helium could be vented in 23 seconds by a pressure of 72 psig. At the most we could, therefore, force only 370 grams of 6^oK supercritical helium, 71 gm of warm gas, or a combination of both into the Airco supply dewar. Both the internal 1/2" O.D. and 1/4" O.D. tubes have a 0.028" wall. Using Jacobs article we find that 270 grams of liquid helium are required to cool the transfer line to 4.2^oK. The amount of hot gas entering the Airco dewar will be limited by flow rates rather than the transfer line getting cold. 370 grams of supercritical helium presents no problem since the ullage space of the dewar can easily accomodate this extra volume. Hot gas is more troublesome in that it can vaporize a much larger quantity of gas. Both constant pressure and constant volume models will be used.

Constant pressure model:

71 grams of 300^oK gas enters the Airco dewar in 23 seconds. At 1 atm the mass of helium vaporized is given by

$$m = \frac{71 \times h}{\Delta h_v} = \frac{71 \times 1573 \text{ j/gm}}{20.4 \text{ j/gm}} = 5474 \text{ gm} = 44 \text{ liquid liters}$$

$$\dot{m} = 3053 \text{ SCFM } (60^oF \text{ helium})$$

Several Airco dewars were examined and found to have different relief valve combinations. Most dewars had two Circle Seal reliefs, model 559B-6M-10. Assume the minimum relief is a single Circle Seal valve. From the flow sheets we find a capacity of ~ 70 SCFM air. Flow rates are increased for 10^oK helium gas by $1/(0.137 \times 10/293)^{1/2} = 14.6$. Thus the maximum flow rate for cold helium = 70 x 14.6 = 1020 SCFM. Using this model the relief is not adequate.

Constant volume model:

Assume 71 grams of 300^oK gas enters a 500 liter Airco dewar at 1 atm and then mixes to thermodynamic equilibrium in a sealed system. The final state can be calculated from the final internal energy with no change in specific volume.

$$U_{total} = (um)_{liquid} + (um)_{hot\ gas}$$

$$U_{total} = 6.25 \times 10^4 \times 8.9 \text{ j/gm} + 71 \times 950 \text{ j/m} = 6.24 \times 10^5$$

$$U_{final} = 6.24 \times 10^5 / (6.25 \times 10^4 + 71) = 9.97 \text{ j/gm}$$

Following the constant specific volume line of 8 cm^3/gm from the saturated liquid line at u = 8.9 j/gm to 9.97 j/gm, the final pressure is found to be ~ 2.4 atm = 21 psig. In practice the final pressure will be significantly lower due to the ullage space in the dewar.

Loss of vacuum:

The loss of vacuum accident must also be considered. Flow rates are much lower but last for 3 minutes. Use 20 psig in the storage dewar as the driving pressure although the pressure drops way off once the 4" rupture disk pops. From the same formula as before with $\Delta P = 20$ and $P_1' = 34.7$, flow rates are reduced by a factor of 0.33.

T ^oK	Q SCFM, 60^{o}F	Q gm/sec
6	68	5.3
293	13	1.0
*293	1000	80

*NOTE: This last entry is the flow rate out through the Airco dewar relief and includes the helium vaporized from 1.0 gm/sec of 293^{o}K flow rate into the dewar.

Conclusion:

Airco buys dewars from Cryofab and MVE. Cryofab dewars are rated for 35 psig and MVE dewars are rated for 33 psig based on DOT 4L standards. Normal operating procedures require that fill valve MV-02-H be opened only when the magnet is not energized. If this procedure were violated, a worse case analysis shows that the Airco dewar could be pressurized to 21 psig. Loss of vacuum could possibly result in a 25 psig pressure rise until the line is cooled. Simultaneous violation of our operating procedures, a major accident, and a dewar with a single relief valve still would not exceed the pressure rating of the Airco supply dewars.

From Advances in Cryogenic
Engineering Vol. 10
or 1968 BNL Summer Study

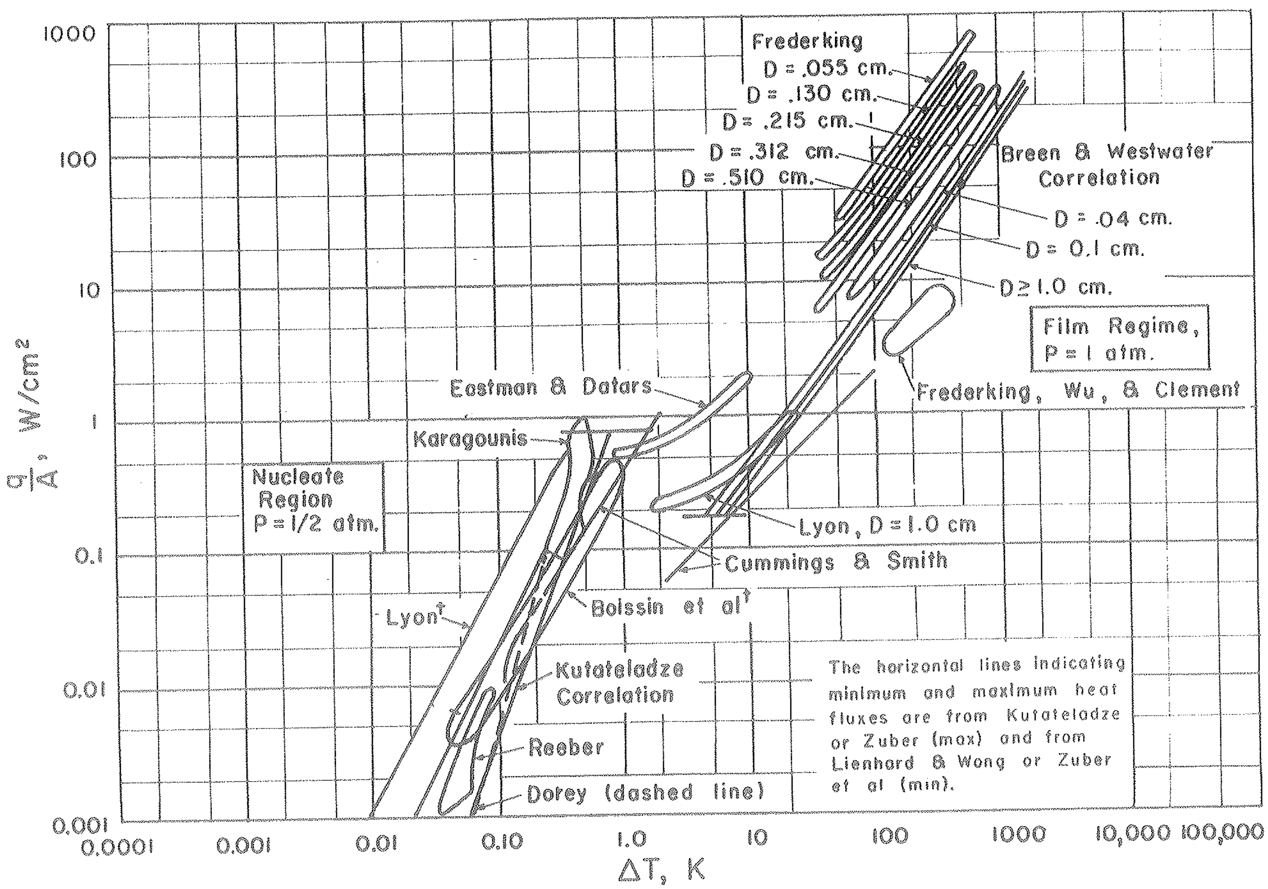

Fig. 13. Pool boiling of helium compared with results from correlations. Data identified by † has been corrected to values for the ½ atm case by use of the pressure dependence implied from the Kutateladze correlation.

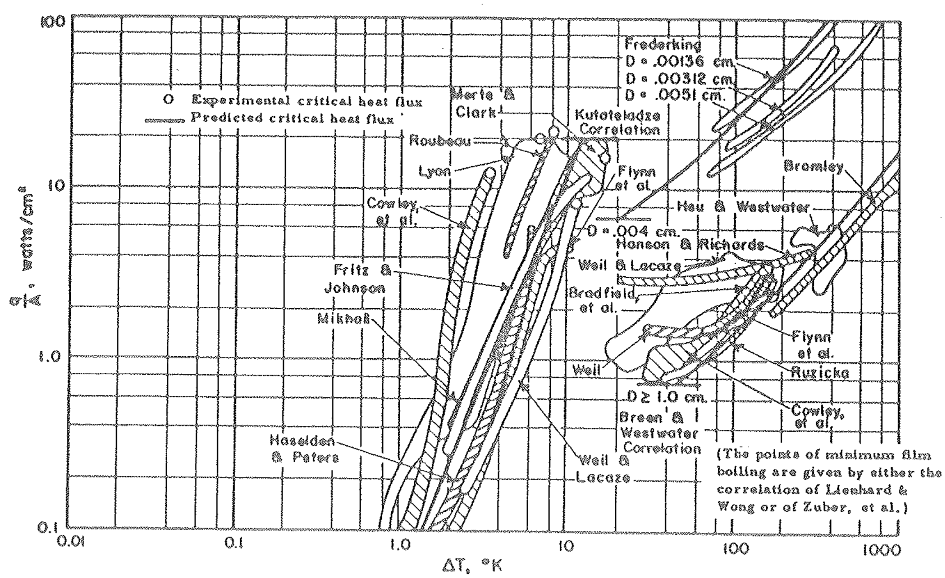

Fig. 3. Experimental nucleate and film pool boiling of nitrogen at 1 atm compared with the predictive correlations of Kutateladze, and Breen and Westwater.

variables were formulated into groups which could be evaluated from available theoretical and experimental property data. Finally, these groups were rearranged into the conventional form of heat flux as a function of difference between the wall and the saturated liquid temperature

$$\frac{q}{A} = Cf(\Delta T) = 4.87 \times 10^{-11}\left[\frac{(C_p)_l}{\lambda \rho_v}\right]^{1.50}\left[\frac{k_l \rho_l^{1.282} P^{1.750}}{\sigma^{0.906} \mu_l^{0.626}}\right](\Delta T)^{2.50} \qquad (1)$$

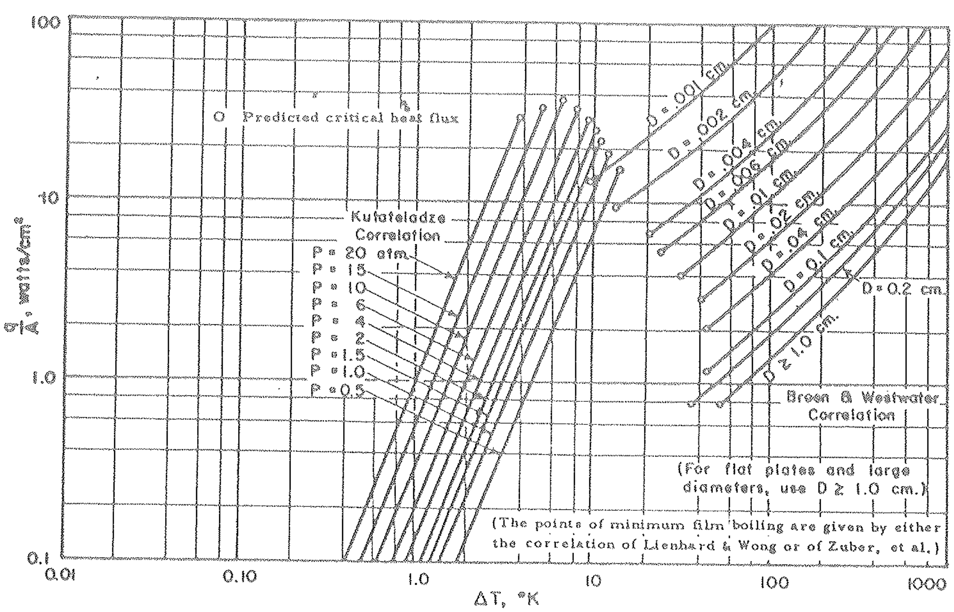

Fig. 4. Predictive nucleate and film pool boiling correlations for nitrogen.

From Advances in Cryogenic
Engineering Vol 10

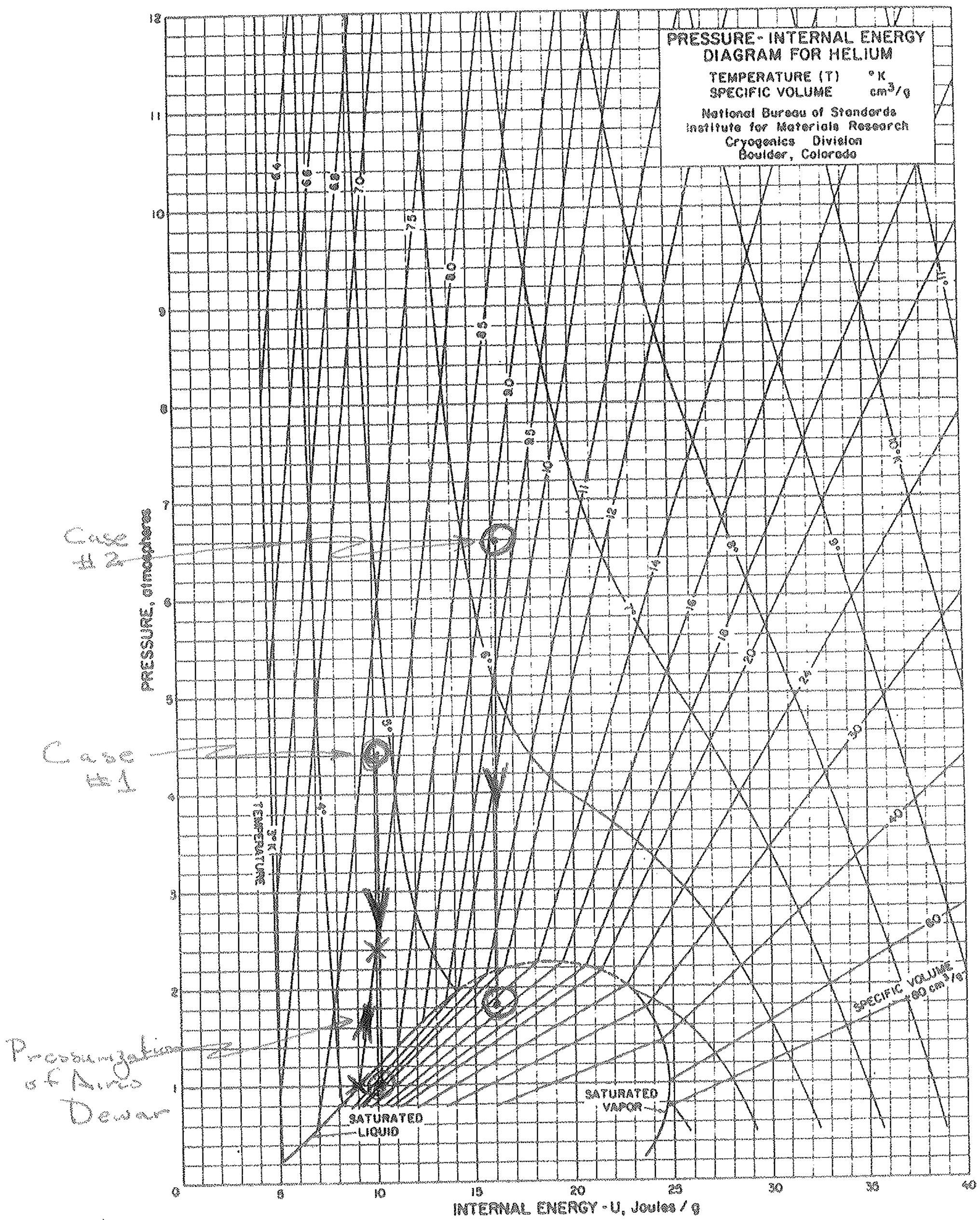

FIGURE 2.7. *Pressure-internal energy diagram.*