Cryogenic Pressure Safety

with Emphasis on Overpressure Protection of Low Temperature Helium Vessels

Part 1

Pressure Safety Fundamentals

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CSA Webinar

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Introduction

- The purpose of this lecture is to provide a review of cryogenic pressure safety and pressurized gas hazards
- A lecture in two parts
  - **Pressure safety fundamentals**
  - Impact of pressure safety on design
- Pressure safety in cryogenics involves all the usual pressurized gas concerns, plus:
  - The possibility of pressurizing a closed volume via warm-up
  - Possible embrittlement and resulting fracture of materials
- We start with some examples of incidents and "lessons learned"

Topics in cryogenic pressure safety

- Part 1, this week – fundamentals
  - Compliance with consensus standards, exceptional vessel methodology
  - Sources of pressure
  - Thermodynamics of cryogen expansion and venting
  - Analytical methods for vent line and relief sizing
  - Relief devices
  - Conclusions and references

- Part 2, next week, impact on design
  - Example of a venting system analysis
  - Examples of the impact on cryostat design
  - Conclusions and references

Two MRI system event videos

- http://www.youtube.com/watch?v=1R7KsfosV-o
- http://www.youtube.com/watch?v=sceO38idjic&feature=related

Risks of excessive pressure

- Vessel rupture, explosion
- Freezing, burns, due to plume of cold gas or liquid
- Discharge of fittings, pipe caps, valves

Over Pressurization or explosion due to rapid expansion

- Without adequate venting or pressure-relief devices on the containers, enormous pressures can build up which can cause an explosion.
- Unusual or accidental conditions such as an external fire, or a break in the vacuum which provides thermal insulation, may cause a very rapid pressure rise.
- The pressure relief valve must be properly installed and free from obstruction.

Lessons Learned

- The following are a few "lessons learned" which have been compiled from various sources. One source of examples is the American Industrial Hygiene Association, which has a section of their website describing accidents and "lessons learned" including several cryogenic accidents: https://www.aiha.org/get-involved/VolunteerGroups/LabHSCommittee/Pages/Lessons-Learned.aspx

Lessons Learned

- Empty 55 gallon drum (1999)
  - At the Nevada Test Site, a waste handler was opening new, empty 55 gallon open-top drums. Upon removing the bolt from the drum lid clamp, the ring blew off and the lid was ejected approximately 5 to 10 feet in the air, just missing the Waste Handler's face. The drum did not hiss or show signs of pressurization.
  - Because the Waste Handler had been properly trained to stand away from the drum while opening it, he was not injured.
  - The event was caused by the drums being manufactured and sealed at sea level in Los Angeles and subsequently shipped to a much higher elevation of approximately 6,000 feet at the Nevada Test Site. The increased elevation, combined with the midday heat, created sufficient pressure buildup to cause the lid to blow off when the ring was being released.
  - Lesson -- large force with small pressure times large area

Lessons learned (continued)

- 50 liter LN2 laboratory dewar explosion
  - Transfer of LN2 from 160 liter dewar to 50 liter "laboratory" dewar
  - Flex hose end from 160 l dewar would not fit in lab dewar neck (normally a "wand" is inserted for filling), so a connection was made with rubber hose over the OUTSIDE of the lab dewar neck and transfer hose end
  - "Slot" cut in rubber hose for vent
  - Failure not initially caused by overpressure, but by cooling of upper part of neck during fill! Seal between neck and vacuum jacket broke due to differential thermal contraction.
  - Seal to vacuum jacket broke after lab dewar nearly full, subsequent overpressure with lack of sufficient vent caused explosion of lab dewar
  - One person badly injured
  - Lesson -- rupture of insulating vacuum with restricted venting resulted in explosion

50 liter LN2 dewar explosion

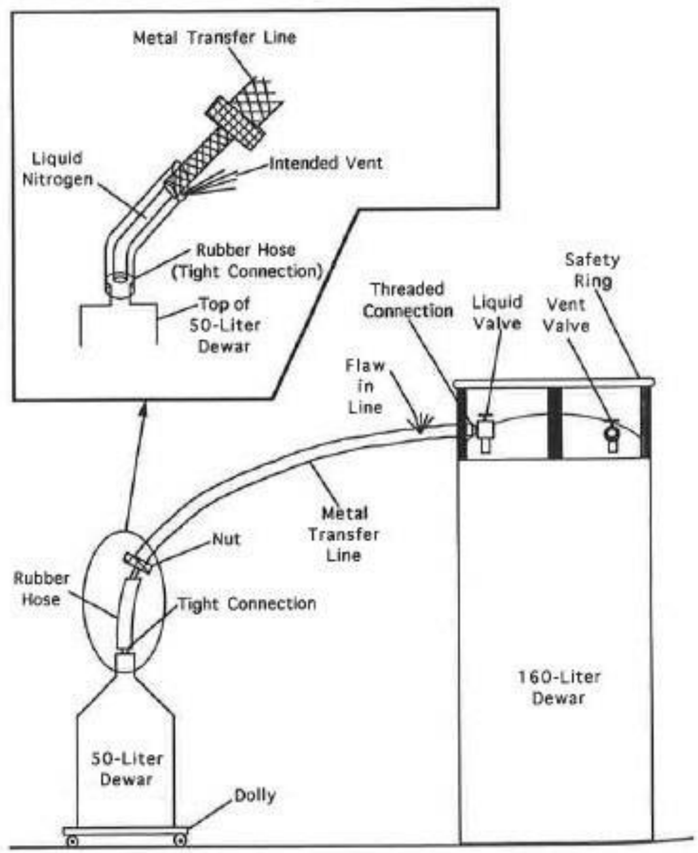


Fig. 13.1a. Schematic of liquid-nitrogen transfer arrangement.

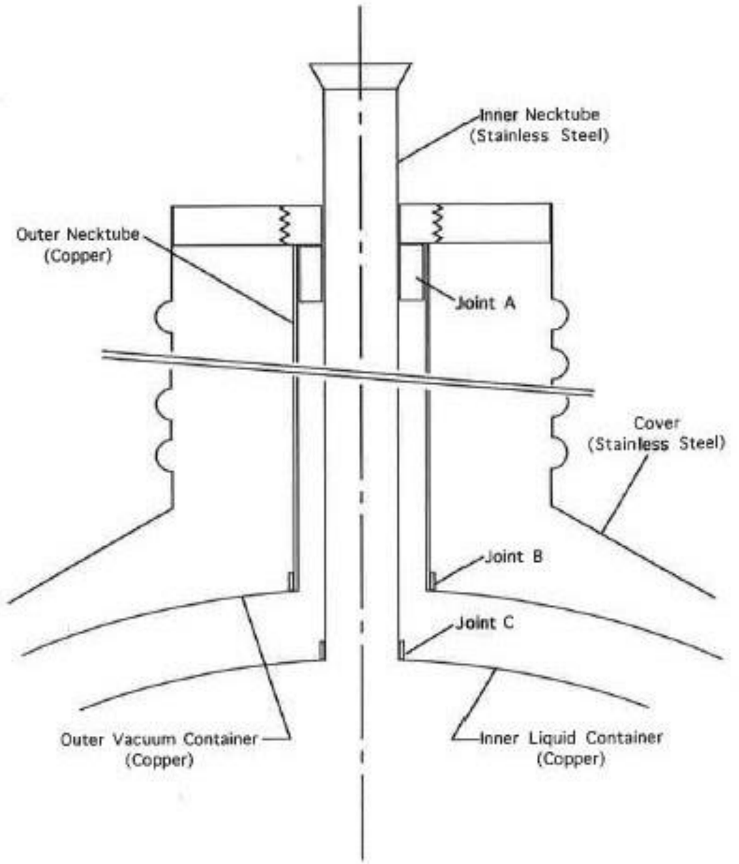


Fig. 13.1b. Detail of the top of the 50-l Dewar shown in Fig. 13.1a.

Figures from "Safety in the Handling of Cryogenic Fluids," by F. J. Edeskuty and W. F. Stewart, Plenum Press, NY, 1996.

Another LN2 dewar explosion, for the report, see http://www.tdi.texas.gov/fire/documents/fmred022206.pdf

At approximately 3:00 a.m. on Thursday, January 12, 2006, an explosion occurred in a state university chemistry building laboratory, causing substantial building damage. The explosion resulted from a rupture in a liquid nitrogen (Dewar) cylinder. The cylinder was originally constructed and tested in December 1980.

The State Fire Marshal's Office, in cooperation with the university's environmental health & safety office, conducted an investigation that included an assessment of the building damage and reconstruction of the events leading to the explosion. The resulting examination revealed catastrophic failure of the cylinder. The failure permitted rapid expansion of the nitrogen gas, blowing out the bottom of the tank and propelling the cylinder upwards.

The examination revealed that the cylinder's pressure release valve and rupture disc had been replaced by two brass plugs. Without these two features in place, the cylinder's rupture-prevention function became compromised. During the investigation, lab students related that the bottom portion of the cylinder had been frosting for approximately twelve to eighteen months, suggesting to them that the cylinder was "leaking".

State Fire Marshal's Alert

February 22, 2006

University Campus Liquid Nitrogen Cylinder Explosion

Incident Specifics

Figure 1 -Effect of Explosion on Dewar Cylinder Compared to unaffected cylinder

Figure 2 - Hallway Outside Laboratory Showing Explosion Damage

Figure 3 - Inside the Laboratory after Explosion

Pressure vessel and piping codes and national laboratory standards

- ASME Boiler and Pressure Vessel Code and ASME B31 Piping Codes
  - In general, we try to purchase vessels built to the code from code-authorized shops
  - Where code-stamping is not possible, we design (or specify designs) to the intent of the code and note implications of exceptions to the code
- 10 CFR 851 requirements for pressure systems for DOE contractors
- SLAC's ES&H Manual Chapter 14, "Pressure Safety"
- Fermilab's ES&H Manual (FESHM) pressure vessel standards
  - FESHM 5031 (general pressure vessel standard)
  - FESHM 5031.6 (dressed cavity standard)

10 CFR part 851

Appendix A to Part 851—Worker Safety and Health Functional Areas

This appendix establishes the mandatory requirements for implementing the applicable functional areas required by § 851.24.

4. Pressure Safety

(a) Contractors must establish safety policies and procedures to ensure that pressure systems are designed, fabricated, tested, inspected, maintained, repaired, and operated by trained and qualified personnel in accordance with applicable and sound engineering principles.

(b) Contractors must ensure that all pressure vessels, boilers, air receivers, and supporting piping systems conform to:

(1) The applicable American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (2004); sections I through section XII including applicable Code Cases (incorporated by reference, see § 851.27)

(2) The applicable ASME B31 (Code for Pressure Piping) standards as indicated below; and or as indicated in paragraph (b)(3) of this section:

10 CFR part 851 Appendix A (4) (c)

(c) When national consensus codes are not applicable (because of pressure range, vessel geometry, use of special materials, etc.), contractors must implement measures to provide equivalent protection and ensure a level of safety greater than or equal to the level of protection afforded by the ASME or applicable state or local code. Measures must include the following:

(1) Design drawings, sketches, and calculations must be reviewed and approved by a qualified independent design professional (*i.e.*, professional engineer). Documented organizational peer review is acceptable.

(2) Qualified personnel must be used to perform examinations and inspections of materials, in-process fabrications, non-destructive tests, and acceptance test.

(3) Documentation, traceability, and accountability must be maintained for each pressure vessel or system, including descriptions of design, pressure conditions, testing, inspection, operation, repair, and maintenance.

If codes to not apply, we must provide a level of safety greater than or equal to the applicable code.

From SLAC's ES&H Manual Chapter 14, "Pressure Safety"

2.2 Design Standards

The following is an overview. See the Pressure Systems Design Manual (forthcoming) for technical detail.

2.2.1 Conventional Pressure Systems

All pressure vessels, boilers, and air receivers and supporting piping systems must be designed in accordance with applicable ASME code, which includes the *Boiler and Pressure Vessel Code (BPVC)* Sections I though XII, including applicable code cases and applicable ASME B31 (*Pressure Piping Code*) standards.

2.2.2 Scientific Pressure Systems

10 CFR 851 specifies that when national consensus codes are not applicable (because of pressure range, vessel geometry, use of special materials), measures must be implemented to provide equivalent protection to ensure a level of safety greater than or equal to the level of protection afforded by the ASME code.

Required measures include

- Applying a basic minimum design margin[1] (safety factor) of 3.5 for any pressure system(s) unless a lower design margin can be justified by applicable codes or stress analysis or engineering calculations
- Design drawings, sketches, and calculations must be reviewed and approved by a qualified independent design professional such as a professional engineer. Documented organizational peer review of such design is acceptable.

Issues for code compliance for SRF dressed cavities

- Cavity design that satisfies level of safety equivalent to that of a consensus pressure vessel code is affected by
  - use of the non-code material (niobium),
  - complex forming and joining processes,
  - a shape that is determined entirely by cavity RF performance,
  - a thickness driven by the cost and availability of niobium sheet,
  - and a possibly complex series of chemical and thermal treatments.
- Difficulties emerge pressure vessel code compliance in various areas
  - Material not approved by the pressure vessel code
  - Loadings other than pressure
    - Thermal contraction
    - Tuning
  - Geometries not covered by rules
  - Weld configurations difficult to inspect

Safety/compliance issue summary

- In the U.S., Europe, and Japan, SRF helium containers and part or all of the RF cavity fall under the scope of the local and national pressure vessel rules.
- Thus, while used for its superconducting properties, niobium ends up also being treated as a material for pressure vessels.
- For various reasons, it is not possible to completely follow all the rules of the pressure vessel codes for most of these SRF helium vessel designs
- Thus, we have to invoke the "equivalent level of safety" allowed by 10 CFR 851.

General solution

- In applying ASME code procedures, key elements demonstrating the required level of design safety are
  - the establishment of a maximum allowable stress
  - And for external pressure design, an accurate approximation to the true stress strain curve
- Apply the ASME Boiler and Pressure Vessel Code as completely as practical
  - Exceptions to the code may remain
  - We have to show the risk is minimal
- Satisfy the requirement for a level of safety greater than or equal to that afforded by ASME code.
- Fermilab, Brookhaven, Jefferson Lab, Argonne Lab, and others in the U.S. have taken a similar approach

Fermilab developed a standard and guidelines for vessels which cannot fully meet the pressure vessel code (FESHM 5031.6)

- Design drawings, sketches, and calculations are reviewed and approved by qualified independent design professionals.
- Only qualified personnel must be used to perform examinations and inspections of materials, in-process fabrications, non-destructive tests, and acceptance tests.
- Documentation, traceability, and accountability is maintained for each pressure vessel and system, including descriptions of design, pressure conditions, testing, inspection, operation, repair, and maintenance.

FESHM 5031.6

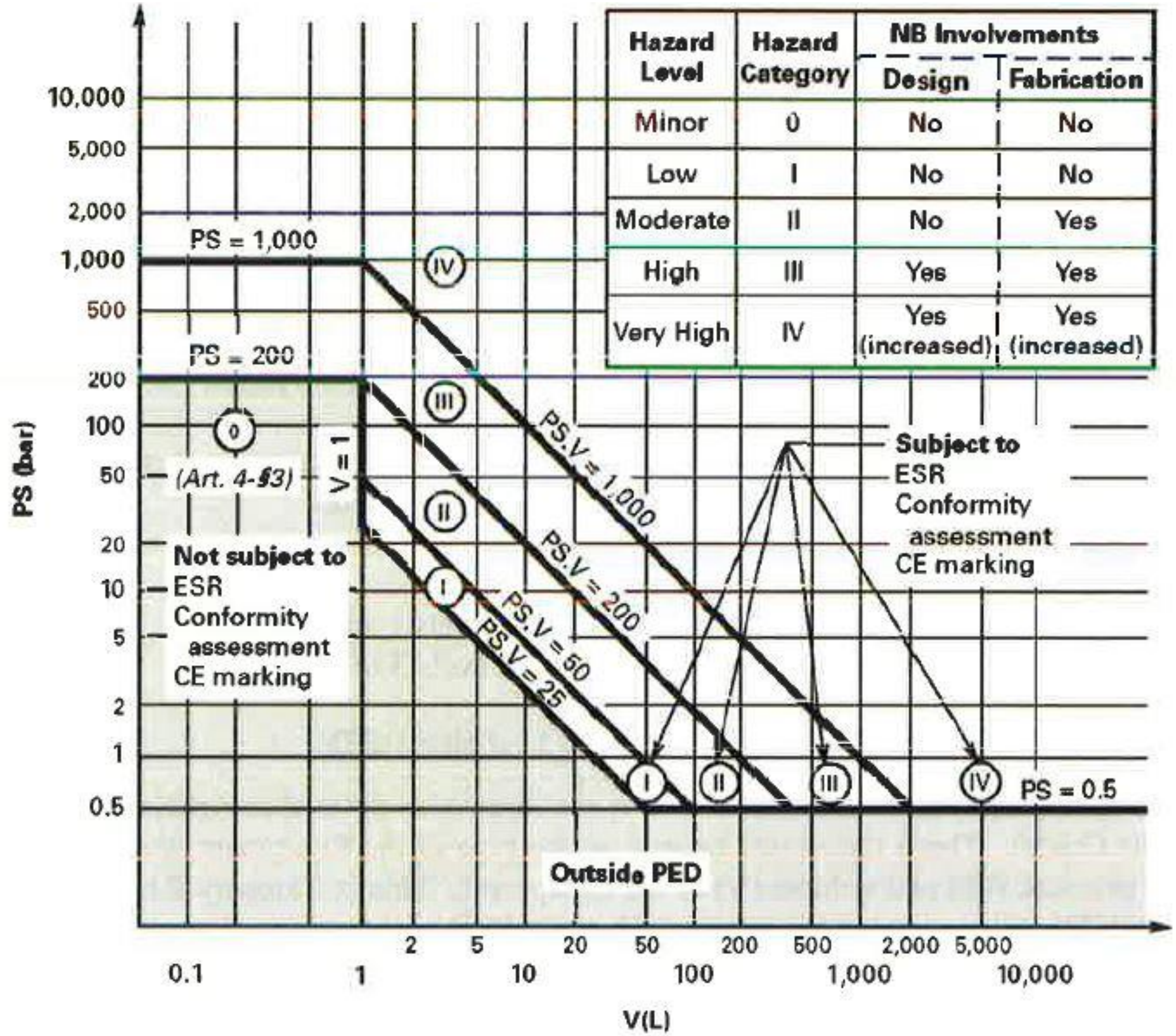
- The chapter applies to any Dressed SRF Cavity that is designed or used at Fermilab
    - "Dressed SRF Cavity –An integrated assembly wherein a niobium cavity has been permanently joined to a cryogenic containment vessel, such that niobium is part of the pressure boundary and the cavity is surrounded by cryogenic liquid during operation."
- The chapter references specially developed engineering guidelines
- An Engineering Note is prepared for all Dressed SRF cavities
    - I will describe such an engineering note in detail in the next talk
- A panel specifically assigned to SRF cavity engineering note reviews ensures uniformity in preparation and review

Pressure Vessel Code Scope

ASME Section VIII, Division 1, describes scope in terms of what is excluded. Key general exclusions are copied here from U-1(c)(2)(-h) and U-1(c)(2)(-i):

> *(-1)* vessels having an internal or external pressure not exceeding 15 psi (100 kPa);
>
> *(-2)* combination units having an internal or external pressure in each chamber not exceeding 15 psi (100 kPa) and differential pressure on the common elements not exceeding 15 psi (100 kPa) [see UG-19(a)];
>
> *(-i)* vessels having an inside diameter, width, height, or cross section diagonal not exceeding 6 in. (152 mm), with no limitation on length of vessel or pressure;

Hence the fundamental rule for the scope (with some specific exceptions, e.g., pumps, compressors, piping systems, water tanks):
15 psi or more, and 6 inches cross section or more.



PED Classification Chart from "Guide for ASME Section VIII, Division 1 Stamp Holders"

ASME Section VIII, Division 1 versus PED version 2014/68/EU

- Note that the scopes of the ASME BPVC and PED version 2014/68/EU codes differ
- Differences between PED requirements and ASME requirements are described the document, "Guide for ASME Section VIII, Division 1 Stamp Holders“, which compares ASME Section VIII, Division 1 with PED version 2014/68/EU
- The comparisons indicate equivalent or higher level of requirements for the EU standards for Category III vessels.
- In general, material requirements, design requirements including formal design review and approval, allowable stresses, construction, inspection, and test requirements are similar or sometimes include more stringent requirements than ASME and also sometimes more responsibility placed on the manufacturer.

Use of CE stamped vessel at SLAC

- We have an electron microscope pressure vessel of 283.6 liters volume with CE stamp
- The gas is sulfur hexafluoride at a pressure up to 5 bar
- Temperature range is ambient (5 C to 40 C).
- The pressure-volume product of 1418 bar-liters places this vessel in PED Hazard Category III
- A review concluded that this Category III CE-stamped vessel has a level of safety equivalent to what would have been provided by ASME pressure vessel codes for this vessel.

Pressure protection

- Vessel and piping have a Maximum Allowable Working Pressure (MAWP) defined by the design of the vessel or system
  - A venting system and relief devices must be in place to prevent any event from pressurizing the vessel or piping above the MAWP (plus whatever code allowance may be available)
- Evaluate all pressure sources and possible mass flow rates
- Size the vent line to the relief device
  - Temperature and pressure of flow stream
  - Typically a pressure drop analysis for turbulent subsonic flow
- Size the relief device
- Size downstream ducting, if any
  - Downstream piping may be necessary to carry inert gas safely away from an occupied area or sensitive equipment

ASME pressure vessel code -- relief devices

- Section VIII of the ASME Code provides fundamental guidance regarding pressure relief requirements.
  - ASME Section VIII, Division 1, UG-125 through UG133, for general selection, installation and valve certification requirements
  - ASME Section VIII, Appendix 11 for flow capacity conversions to SCFM-air
- For Div. 2, relevant information is found in Part 9.

Vessel pressures and relief set pressures allowed per ASME Section VIII, Division 1

% of MAWP	Maximum vessel pressure as % of MAWP for relief configuration or purpose (UG-125)	Maximum relief set pressure as % of MAWP (UG-134)
100%		Maximum set pressure for single relief device
105%		Maximum set pressure for second and subsequent reliefs
110% (or 3 psi, whichever is greater)	Vessel has single relief device	Relief for "fire" (or unexpected external heat source)
116% (or 4 psi, whichever is greater)	Vessel has multiple relief devices	
121%	Vessel ". . . exposure to fire or other unexpected sources of external heat . . . " I include loss of vacuum in this category.	

Compressed Gas Association publication, CGA S-1.3, "Pressure Relief Device Standards"

- Extensive guidance on requirements for relief devices consistent with ASME code
  - Applicable where MAWP and venting pressure exceed 15 psig
- I will not provide a detailed discussion of CGA S-1.3, but rather just point to a few key issues and most useful elements of the standard

Note: we take exception to paragraph 2.2 in CGA S-1.3

- "CGA believes that reclosing PRDs on a container shall be able to handle all the operational emergency conditions except fire, for which reclosing or nonreclosing PRDs shall be provided. The operational emergency conditions referred to shall include but not be limited to loss of vacuum, runaway fill, and uncontrolled operation of pressure buildup devices."
- Exception: we treat loss of vacuum to air, with the very high heat flux resulting from condensation on the liquid helium temperature surface of a container, like the fire condition and may use nonreclosing relief devices for that situation
- This interpretation is consistent with wording in the ASME code, Section VIII, Division I, UG-125, which refers to ". . . exposure to fire or other unexpected sources of external heat.“

Compressed Gas Association publication, CGA S-1.3, "Pressure Relief Device Standards"

- From CGA S-1.3: Among the particular issues which must be addressed for low temperature vacuum jacketed helium containers are
  - the temperature at which liquid-to-gas evolution should be estimated for the supercritical fluid at its venting pressure *(CGA S-1.3 is very useful here; I'll discuss this)*
  - the warming of the cold fluid passing through a long vent line *(CGA S-1.3 also provides useful practical approximation methods here which I will discuss)*
  - the volume generated per unit heat added *(we have data from lab tests about this which provide useful numbers)*

Sources of pressure -- mechanical

- Compressors, pumps
  - Screw compressors are positive displacement devices
  - Worst case flow may be with high suction pressure as limited by inlet-side reliefs or pump/compressor motor power
    - Calculate worst-case flow as highest inlet density combined with known displacement volume
    - Or consider power limitations of pump or compressor motor
- Connection to a higher pressure source, such as a tube trailer
  - Evaluate the mass flow as determined by the pressure drop from the highest possible source pressure to the MAWP of vessel to be protected

Sources of pressure -- heat

- Trapped volume, slow warm-up and pressurization with normal heat inleak
  - All possible volumes which may contain "trapped" (closed off by valves or by other means) cold fluid require small reliefs
  - Rate of warm-up may be evaluated, generally slow enough that trapped volume reliefs are not individually analyzed.
- Loss of vacuum to helium with convection and conduction through helium gas
- Sudden large heat influx to a liquid-helium temperature container due to condensation of nitrogen or air on the surface
  - Either through MLI or, worst-case, on a bare metal surface
- Stored energy of a magnetic field
  - May provide a larger flow rate than loss of insulating vacuum
- Fire, with heat transport through the gas-filled insulation space

Nominal heat loads

- Working numbers for making heat load estimates
  - ~1.5 W/m^2 from 300 K to MLI-insulated (typically about 30 layers) cold surface
  - ~50 mW/m^2 from 80 K to MLI-insulated (typically about 10 layers) 4.5 K or 2 K surface
- Note that support structures and "end effects" may dominate the total heat load

Heat flux due to loss of insulating vacuum as a source of pressure

- W. Lehman and G. Zahn, "Safety Aspects for LHe Cryostats and LHe Transport Containers," ICEC7, London, 1978
- G. Cavallari, et. al., "Pressure Protection against Vacuum Failures on the Cryostats for LEP SC Cavities," 4th Workshop on RF Superconductivity, Tsukuba, Japan, 14-18 August, 1989
- M. Wiseman, et. al., "Loss of Cavity Vacuum Experiment at CEBAF," *Advances in Cryogenic Engineering*, Vol. 39, 1994, pg. 997.
- T. Boeckmann, et. al., "Experimental Tests of Fault Conditions During the Cryogenic Operation of a XFEL Prototype Cryomodule," DESY.

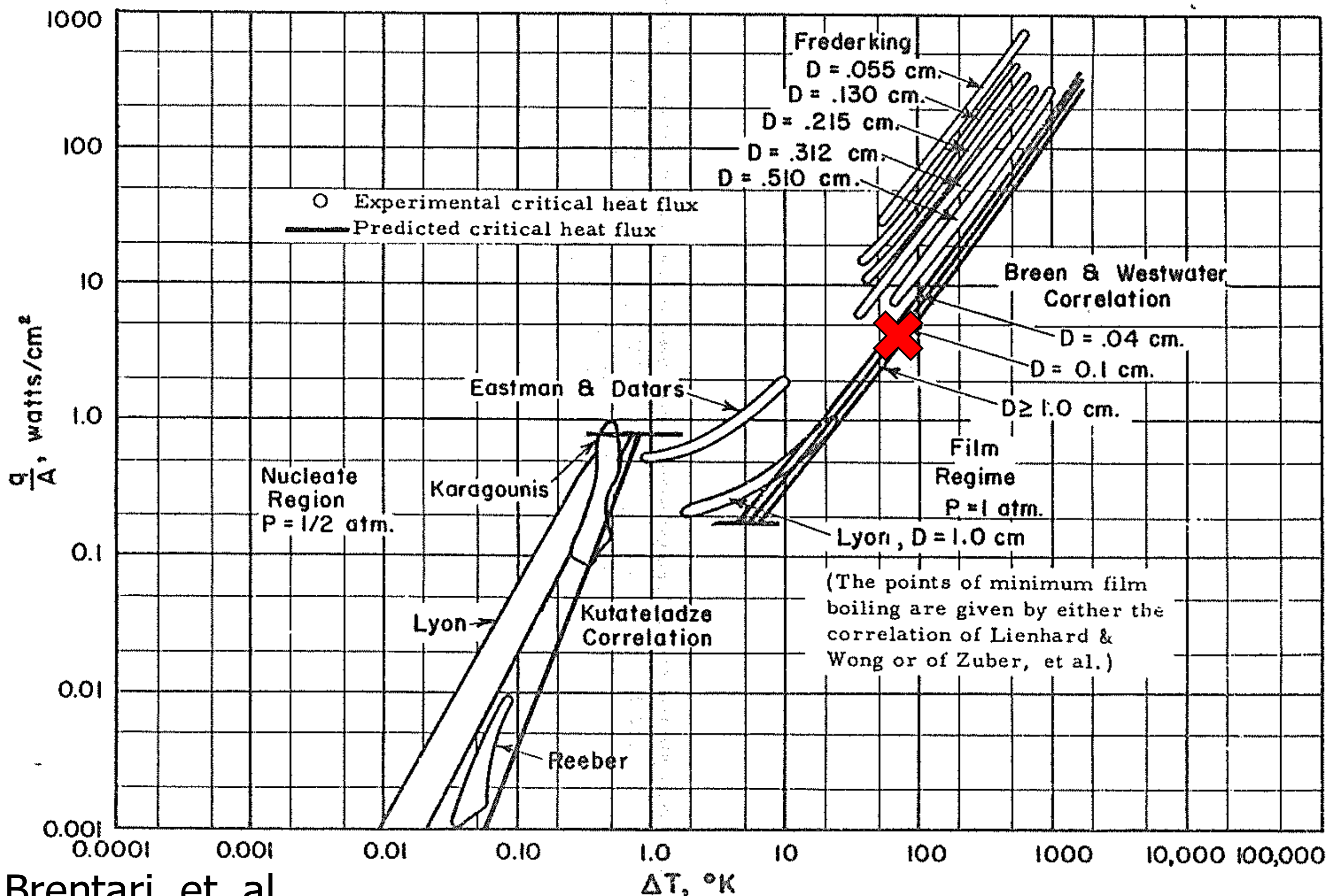
Heat flux conclusions

Reference	Heat flux to nominally 4.2 Kelvin helium
Lehman and Zahn	0.6 W/cm^2 for the superinsulated tank of a bath cryostat 3.8 W/cm^2 for an uninsulated tank of a bath cryostat
Cavallari, et al.	4 W/cm^2 maximum specific heat load with loss of vacuum to air
Wiseman, et al.	3.5 W/cm^2 maximum peak heat flux 2.0 W/cm^2 maximum sustained heat flux

- T. Boeckmann, et. al. (DESY)
  - Air inflow into cavity beam vacuum greatly damped by RF cavity structures
- Various authors also comment about layer of ice quickly reducing heat flux
- Heat flux curves for liquid helium film boiling with a delta-T of about 60 K agree with these heat flux numbers (next slides)
- I use 4 W/cm^2 for bare metal surfaces

Underlying thermodynamics

- Nitrogen freezes at 63 K
- Oxygen freezes at 55 K
- So heat flux for air condensation begins to become large with condensing surface temperatures under 63 K
- Delta-T to liquid helium will be around 58 K or less
- Film boiling of helium with 50 K to 60 K delta-T will provide about 4 W/cm^2



E. G. Brentari, et. al.,
NBS Technical Note 317

FIGURE 2.4
Experimental Nucleate and Film Pool Boiling of Helium Compared with the Predictive Correlation of Kutateladze and Breen and Westwater

Atmospheric air rushing into a vacuum space and condensing on a surface deposits about 11 kW per cm^2 of air hole inlet area. In many cases, heat flux will be limited by this air hole inlet size rather than low-temperature surface area.

Inputs in blue	
air inlet hole diameter (in)	**3.00**
air inlet hole diameter (mm)	**76.200**
hole area (cm2)	45.581
discharge coefficient	0.700
k (Cp/Cv)	1.400
air, 1 atm, 290 K density (g/cc)	0.00118
1 atm pressure (g/cm-sec2)	1.00E+06
sqr root k	0.685
sqr root (2 x rho x P) (g/s-cm2)	48.580
air mass flow (g/sec)	1061.3
air mass flux (g/sec-cm2)	23.3
heat from 290 K to solid air (J/g)	470.0
power deposited (W)	498829.1
power deposited per air inlet area (W/cm2)	**10943.9**

Conversion of heat to mass flow

- Low pressures, below the critical pressure
  - Latent heat of vaporization
  - Net flow out is vapor generated by the addition of heat minus the amount of vapor left behind in the volume of liquid lost
  - For helium at pressures approaching the critical pressure (2.3 bar), the density and mass of vapor "left behind" in the volume formerly occupied by the boiled liquid can be significant, so this may be an important factor in reducing mass flow to a net mass flow out.
- High pressures, above the critical pressure
  - Heat added results in fluid expelled
  - A "pseudo latent heat" can be evaluated

Supercritical fluid -- energy added per unit mass expelled

The pressure of a liquid helium container during venting will often exceed the critical pressure of helium (2.3 bar)

From CGA S-1.3, paragraph 6.1.3, for a volume of helium (or another fluid) at or above its critical pressure, heat added results in expulsion of the fluid at some rate which is a function of pressure. The heat added per unit mass of fluid expelled from the volume (the "pseudo latent heat") is $v\left(\frac{\partial h}{\partial v}\right)_P$ (with units for example, J/g). Values for pseudo latent heat for helium are tabulated in NBS Technical Note 631, "Thermophysical Properties of Helium-4 from 2 to 1500 K with Pressures to 1000 Atmospheres", 1972. These values are also available from the equation of state programs such as HEPAK (by Cryodata, Inc.) And also, "Technology of Liquid Helium," (NBS Monograph 111, by R.H. Kropschot, et. al.) contains a chart of "Heat absorbed per pound efflux for a helium container relieving above the critical pressure", Figure 6A-2.

Pseudo latent heat heat – 4 atm helium

Temperature (K)	Density (g/liter)	$v(dh/dv)_p$ (J/g) (heat aborbed per unit of mass expelled)
5.0	124.2	29.8
5.5	109.4	24.6
6.0	82.6	19.8
6.5	55.0	21.4
7.0	42.7	24.8

Venting occurs at a temperature near (not exactly, but close, more later) where $v(dh/dv)_p$ is at a minimum, so for our 4.1 bar venting with loss of vacuum, around 6 K.

Relief venting

- Up to now, we have discussed estimation of the venting flow rate
- In summary
  - We have a vessel or piping MAWP
  - We have a mass flow rate provided either by compressors/pumps or heating of low temperature fluid which must be removed from that vessel at or below the MAWP

Berkeley MRI magnet quench

- https://www.youtube.com/watch?v=QRahBusouRs

Venting flow analyses

- Size piping to the relief device
- Size the relief device
  - Typically using the vendor-provided or standard relief device formulas and charts
- Size piping downstream of the relief device
- A somewhat different venting flow analysis -- estimate flow from a rupture or open valve into a room for an ODH analysis

Constraints and assumptions

- For relief and vent pipe sizing
  - Typically flow driven by a Maximum Allowable Working Pressure (MAWP, as defined by code requirements) at the vessel
  - Pipe size and relief device size are the free parameters
    - Perhaps also pipe routing
  - Flow rate may be determined by a compressor or pump capacity or heat flux to a low temperature vessel

Venting and relief sizing analysis

- Conservative, err on the safe side
  - Venting is typically not steady-state, very dynamic
  - Make simplifying assumptions on the safe side
    - For example, flow rate estimate should be safely on the high side for relief sizing
- Reviewable
  - Simplest and most straightforward analysis which demonstrates requirement
  - Of course, more sophisticated analysis (such as FEM fluid dynamic simulation may be necessary for a system with sever constraints)

Vent line flow temperature

The temperature of the expelled fluid for analysis of the flow out the vent line is where the quantity

$$\frac{\sqrt{v}}{v\left(\frac{\partial h}{\partial v}\right)_P}$$

is a maximum for the specified venting pressure.

This exit temperature will typically be 5 K - 6 K for a liquid helium container venting at a somewhat supercritical pressure.

The temperature into the relief device may be higher than the exit temperature due to heat transfer to the flow via the vent pipe. For very high flow rates and a relatively short vent line, this temperature rise may be insignificant. A simple energy balance on the flow and stored energy in the vent line, with an approximate and conservatively large convection coefficient may provide a safely conservative estimate of the temperature rise. For a long vent line, a more detailed analysis may be required in sizing the relief device. CGA S1.3, paragraph 6.1.4 and following, provides some guidance for this analysis.

Vent line pressure drop evaluation

General form of Bernoulli Equation

For pressure drop analysis of isothermal turbulent flow of a compressible or incompressible fluid through a piping system upstream or downstream of a relief device, we may start with the following general form of the Bernoulli equation. (Equation and friction factor definition follows that in "Transport Phenomena," R. Byron Bird, Warren E. Stewart, Edwin N. Lightfoot, John Wiley & Sons, Inc., New York, 1960.)

$$\int_{P_1}^{P_2} \frac{dP}{\rho} + \sum_i \left(\frac{v^2}{2} \frac{L}{R_h} f \right)_i + \sum_i \left(\frac{v^2}{2} k \right)_i = 0 \quad \text{(Equation 1)}$$

P_1 is pressure in.

P_2 is pressure out.

ρ is fluid density, here only a function of pressure since temperature assumed constant.

v is average fluid velocity within the i-th section of conduit or downstream of the i-th fitting.

L is conduit section length.

R_h is channel hydraulic radius, defined as flow area divided by wetted perimeter, which implies $R_h = D/4$ for round pipes.

f is friction factor based on hydraulic radius.

k is the resistance factor for fittings such as elbows, tees, sudden flow area changes, etc.

Pressure drop analysis, working formula for round pipes

This is a form of the D'Arcy-Weisbach formula. With pressure drop expressed as head loss, this is sometimes called simply the Darcy formula. (Note that delta-P changed signs here, to a positive number.)

Pressure drop for turbulent flow in a pipe is

$\Delta P = \frac{\rho v^2}{2}\frac{4L}{D} f$ where ρ is average fluid density, v is average fluid velocity, L is pipe length, D is pipe inner diameter, and f is friction factor based on hydraulic radius (which is D/4 for circular pipes).

Substituting $\dot{m} = \rho v\left(\pi \frac{D^2}{4}\right)$ where $\dot{m}$ is mass flow

gives $\Delta P = (0.811)\frac{\dot{m}^2}{\rho D^5} L \times 4 \times f$

Crane Technical Paper #410

- Crane Technical Paper #410 "Flow of Fluids through Valves, Fittings, and Pipes"
- A classic reference, still available in updated forms
- Contains many forms of Bernoulli Equation and other formulas for both compressible and incompressible flow
- Relief valve and rupture disk catalogue formulas often reference Crane Technical Paper #410
- My only criticism (and strictly my personal opinion) -- I do not like the incorporation of unit conversions into formulas, which is too common in these engineering handbooks

Crane Technical Paper #410 “Flow of Fluids through Valves, Fittings, and Pipes”

Summary of Formulas — continued

• Head loss and pressure drop through valves and fittings

Head loss through valves and fittings is generally given in terms of resistance coefficient K which indicates static head loss through a valve in terms of "velocity head", or, equivalent length in pipe diameters L/D that will cause the same head loss as the valve.

From Darcy's formula, head loss through a pipe is:

$$h_L = f \frac{L}{D} \frac{v^2}{2g} \qquad \text{Equation 3-5}$$

and head loss through a valve is:

$$h_L = K \frac{v^2}{2g} \qquad \text{Equation 3-14}$$

therefore:

$$K = f \frac{L}{D} \qquad \text{Equation 3-15}$$

To eliminate needless duplication of formulas, the following are all given in terms of K. Whenever necessary, substitute $(f\,L/D)$ for (K).

$$h_L = \frac{522\,Kq^2}{d^4} = 0.002\,59\,\frac{KQ^2}{d^4} \qquad \text{Equation 3-14}$$

$$h_L = 0.001\,270\,\frac{KB^2}{d^4} = 0.000\,0403\,\frac{KW^2\bar{V}^2}{d^4}$$

$$\Delta P = 0.000\,1078\,K\rho v^2 = 0.000\,000\,0300\,K\rho V^2$$

$$\Delta P = 3.62\,\frac{K\rho q^2}{d^4} = 0.000\,017\,99\,\frac{K\rho Q^2}{d^4}$$

$$\Delta P = 0.000\,008\,82\,\frac{K\rho B^2}{d^4}$$

$$\Delta P = 0.000\,000\,280\,\frac{KW^2\bar{V}}{d^4}$$

$$\Delta P = 0.000\,000\,000\,605\,\frac{K(q'_h)^2\,T\,S_g}{d^4\,P'}$$

$$\Delta P = 0.000\,000\,001\,633\,\frac{K\,(q'_h)^2\,S_g^2}{d^4\,\rho}$$

• Head loss and pressure drop with laminar flow ($R_e < 2000$) through valves; Darcy's formula

$$h_L = 0.003\,28\left(\frac{L}{D}\right)\frac{\mu Q}{d^4\rho} \qquad \text{Equation 3-17}$$

$$h_L = 1.470\left(\frac{L}{D}\right)\frac{\mu q}{d^4\rho} = 0.008\,01\left(\frac{L}{D}\right)\frac{\mu v}{d\rho}$$

$$h_L = 0.000\,408\left(\frac{L}{D}\right)\frac{\mu W\bar{V}^2}{d^4}$$

$$\Delta P = 0.000\,0557\left(\frac{L}{D}\right)\frac{\mu v}{d} = 0.010\,21\left(\frac{L}{D}\right)\frac{\mu q}{d^4}$$

$$\Delta P = 0.000\,0228\left(\frac{L}{D}\right)\frac{\mu Q}{d^4}$$

$$\Delta P = 0.000\,015\,93\left(\frac{L}{D}\right)\frac{\mu B}{d^4}$$

$$\Delta P = 0.000\,002\,84\left(\frac{L}{D}\right)\frac{\mu W\bar{V}}{d^4}$$

• Equivalent length correction for laminar flow with $R_e < 1000$

$$\left(\frac{L}{D}\right)_a = \left(\frac{L}{D}\right)_t \frac{R_e}{1000} \qquad \text{Equation 3-18}$$

See pages 3-11 and A-30. Minimum $(L/D)_a$ = length of center line of actual flow path through valve or fitting. Subscript a refers to equivalent length with $R_e < 1000$. Subscript t refers to equivalent length with $R_e > 1000$.

• Discharge of fluid through valves, fittings, and pipe; Darcy's formula

Equation 3-19

Liquid flow:

$$q = 0.0438\,d^2\sqrt{\frac{h_L}{K}} = 0.525\,d^2\sqrt{\frac{\Delta P}{K\rho}}$$

$$Q = 19.65\,d^2\sqrt{\frac{h_L}{K}} = 236\,d^2\sqrt{\frac{\Delta P}{K\rho}}$$

$$w = 0.0438\,\rho d^2\sqrt{\frac{h_L}{K}} = 0.525\,d^2\sqrt{\frac{\Delta P\rho}{K}}$$

For example from previous list

$$\Delta P = 0.000\,000\,280 \frac{K W^2 \bar{V}}{d^4}$$

Where ΔP is pressure drop in psi, V is the specific volume (in^3/lbm), K is the total resistance coefficient = fL/d so is dimensionless, W is the mass flow rate (lbm/hr), and d is the pipe inner diameter (in).

Compare to

$$\Delta P = (0.811)\frac{\dot{m}^2}{\rho D^5} L \times 4 \times f$$

from slide 34 -- no unit conversions, and a different definition of friction factor. Note! Some sources define f based on hydraulic radius and some on diameter, a factor 4 difference for pipes!

Rupture disk and relief valve sizing

- Flow will typically be choked (sonic) or nearly choked in a relief valve or rupture disk
  - Inlet pressure is at least 15 psig (1 atm gauge) for ASME approved relief devices
  - Discharge is to atmosphere
- This makes analysis relatively simple
  - Relief valve catalogues and rupture disk catalogues have good, practical working formulas and charts for sizing relief devices

Choked flow in a nozzle

Choked (sonic) flow of an ideal gas through a nozzle (from Ascher H. Shapiro, "The Dynamics and Thermodynamics of Compressible Fluid Flow) is

$$\dot{m} = C_D A \left(\sqrt{k \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} \right) \left(\sqrt{\rho P} \right)$$

where $k = \frac{c_p}{c_v} = 5/3$ for helium, C_D is the coefficient of discharge, A is minimum area, and P is the absolute inlet pressure.

Thus, for helium, $\dot{m} = 0.726 C_D A \sqrt{\rho P}$

Relief devices

- For cracking pressures of 15 psig or higher, ASME-approved (UV- or UD-stamped) pressure relief devices may be used.
- For vessels with a differential pressure of more than 15 psid within the vacuum jacket but a gauge pressure of less than 15 psig, ASME-approved reliefs are not available.

The "granddaddy" of all metal disks is the solid design shown in Figure 11. This disk design has been around for over sixty years and has maintained a position of leadership because it is available in a greater range of sizes and pressure ratings than are disks in other designs.

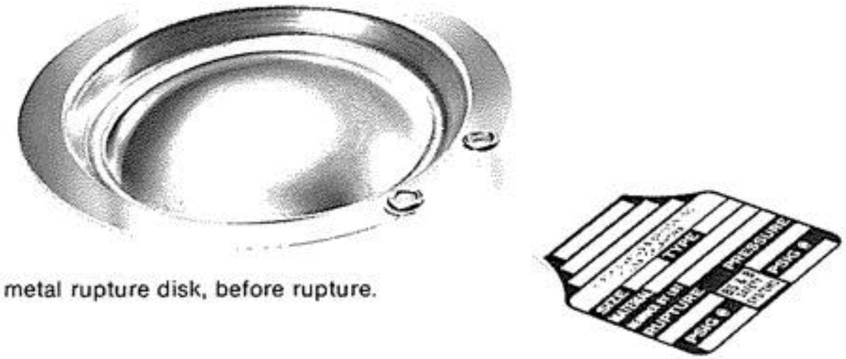


Figure 11: Solid metal rupture disk, before rupture.

A solid metal disk should retain its initial contour during exposure to the normal system pressure. An overpressure buildup to the rating of the disk will cause a thinning out of the metal. Failure will then take place at the center of the crown. When the flowing media is a gas, the opening pattern will be as shown in Figure 12.

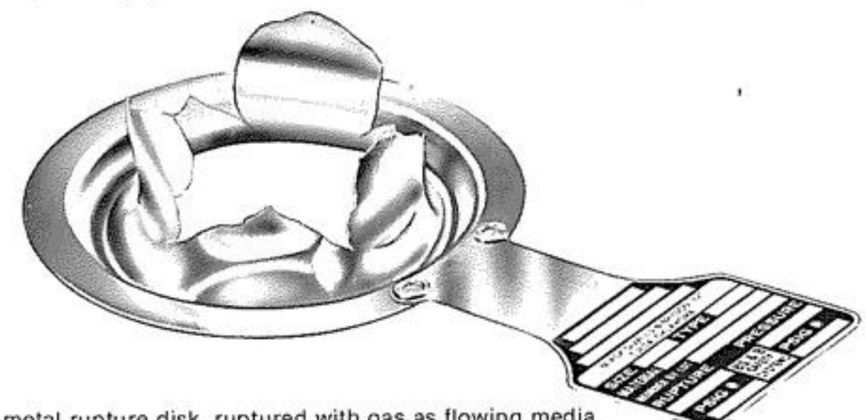


Figure 12: Solid metal rupture disk, ruptured with gas as flowing media.

From the BS&B rupture disk catalogue

Rupture disks

- Various types, some pre-etched or with knife edge, or failure in collapse (pressure on the dome) and other designs and materials
  - Difficult to set a precise opening pressure
- A last resort device since they do not close
  - You don't want these opening in normal operations
  - Switching valves available for dual disks such that one can be replaced while the other holds pressure and provides protection
- Inexpensively provide very large capacity, so typical for the worst-case loss of vacuum
  - Operational reclosing relief valves set at a safely lower pressure (80% of RD or less) prevent accidental opening of the rupture disk

Relief valves

Image from
Rockwood Swendeman
brochure

- Even though valve at room temperature, will cool upon relieving, so need cold-tolerant material and design
- Take care to provide ASME UV-stamped valves for code-stamped vessels

Relief valves

- Sizing best done via valve manufacturer information
  - Shape of valve body, type of plug make sizing unique to the valve design
  - Manufacturers certify flow capacity for UV-stamped (ASME approved) valves

Gas and Vapor Sizing
10% Overpressure (kg/hr)

The following formula is used for sizing valves for gases and vapor (except steam) when required flow is expressed as a mass flow rate, kilograms per hour. Correction factors are included to account for the effects of back pressure, compressibility and subcritical flow conditions. For steam applications use the formula on page 6-6.

$$A = \frac{13160 W \sqrt{TZ}}{C\ K\ P_1\ K_b\ \sqrt{M}}$$

Where:

- A = Minimum required effective discharge area, square millimeters.
- C = Coefficient determined from an expression of the ratio of specific heats of the gas or vapor at standard conditions (see Table T7-7 on page 7-26). Use C = 315 if value is unkown.
- K = Effective coefficient of discharge. K = 0.975
- K_b = Capacity correction factor due to back pressure. For standard valves with superimposed (constant) back pressure exceeding critical see Table T7-1 on page 7-3. For bellows or Series BP valves with superimposed or variable back pressure see Figure F7-2 on page 7-5. For pilot operated valves see discussion on page 7-4.
- M = Molecular weight of the gas or vapor obtained from standard tables or Table T7-7 on page 7-26.
- P_1 = Relieving pressure, kiloPascals absolute. This is the set pressure (kPa) + overpressure (kPa) + atmospheric pressure (kPaa).
- T = Absolute temperature of the fluid at the valve inlet, degrees Kelvin (°C + 273).
- W = Required relieving capacity, kilograms per hour.
- Z = Compressibility factor (see Figure F7-1 on page 7-2). Use Z = 1.0 if value is unknown.

Conclusions for piping and emergency venting

- Cryogenic vessels and piping generally fall under the scope of the ASME pressure vessel and piping codes
- Protection against overpressure often involves not only sizing a rupture disk or relief valve but sizing vent piping between those and the vessel, and also perhaps further ducting downstream of the reliefs
- Loss of vacuum to air with approximately 4 W/cm^2 heat flux on bare metal surfaces at liquid helium temperatures can drive not only the design of the venting system but pipe sizes within the normally operational portions of the cryostat
- Piping stability due to forces resulting from pressure around expansion joints is sometimes overlooked and may also significantly influence mechanical design

References – Standards and Codes

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- ASME B31.3-2014, Process Piping, ASME Code for Pressure Piping
- CGA S-1.3, "Pressure Relief Device Standards", Compressed Gas Association, 2005.
- EUD 2014/68/EU DIRECTIVE — Directive 2014/68/EU of the European Parliament and of the Council of 15 May 2014 on the harmonisation of the laws of the Member States relating to the making available on the market of pressure equipment
- ASME Document PTB-10-2015, "Guide for ASME Section VIII, Division 1 Stamp Holders"

References – National Lab Standards

- SLAC ES&H Manual, Chapter 14: Pressure Systems
  - http://www-group.slac.stanford.edu/esh/hazardous_activities/pressure/
- SLAC ES&H Manual, Chapter 36: Cryogenic and Oxygen Deficiency Hazard Safety
  - http://www-group.slac.stanford.edu/esh/hazardous_substances/cryogenic/
- Fermilab ES&H Manual, Chapter 5031: Pressure Vessels
  - http://esh-docdb.fnal.gov/cgi-bin/ShowDocument?docid=456
- Fermilab ES&H Manual, Chapter 5031.6: Dressed Niobium SRF Cavity Pressure Safety
  - And associated document: "Guidelines for the Design, Fabrication, Testing and Installation of SRF Nb Cavities," Fermilab Technical Division Technical Note TD-09-005
  - http://esh-docdb.fnal.gov/cgi-bin/ShowDocument?docid=1097
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- E. G. Brentari, et. al., "Boiling Heat Transfer for Oxygen, Nitrogen, Hydrogen, and Helium," NBS Technical Note 317, 1965.
- NBS Technical Note 631, "Thermophysical Properties of Helium-4 from 2 to 1500 K with Pressures to 1000 Atmospheres", 1972.
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