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Fundamentals of Cryomodule Design -- Theory and Practice

Part I -- Cryogenic Considerations

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Large-scale cooling of superconducting devices

- Physicists and engineers designing a large-scale liquid helium system typically must design the cooled components (magnets or RF cavities, their containers, and the interfaces to them)
- Cooling mode, heat transfer, pressure drops, cool-down, warm-up and non-steady or upset system operations all must be considered as part of the component design
- The cooled devices must be viewed as part of the cryogenic system

Outline

- Part I -- Tom Peterson
  - Cooling modes and cryogenic issues in cryostat design
- Part II -- Tom Nicol
  - Key components in cryostat design

Cooling modes in large-scale cryogenic systems recently in operation

- Pool boiling helium I (SRF for HERA, LEP, KEKB, CESR)
- Forced flow of subcooled or supercritical helium I (Tevatron, HERA, SSC, RHIC)
- Stagnant, pressurized helium II (the Tore Supra tokamak in France demonstrated the technology, LHC)
- Saturated helium II (CEBAF, TTF at DESY, SNS at Oak Ridge, (and others), foreseen for FRIB, Project X, ILC)
- This list also illustrates the extent to which superconductivity and cryogenics have become standard technology for accelerators

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Helium phase diagram

(S. W. VanSciver, Helium Cryogenics, p. 54)

- Critical point
  - 5.2 K, 2.245 atm
- Lambda transition at 1 atm
  - 2.172 K

- SRF -- HERA, LEP, KEKB, CESR
- Magnets -- HERA, Tevatron
- Magnets -- SSC
- Magnets -- Tore Supra, LHC
- SRF -- CEBAF, TTF, SNS, ILC

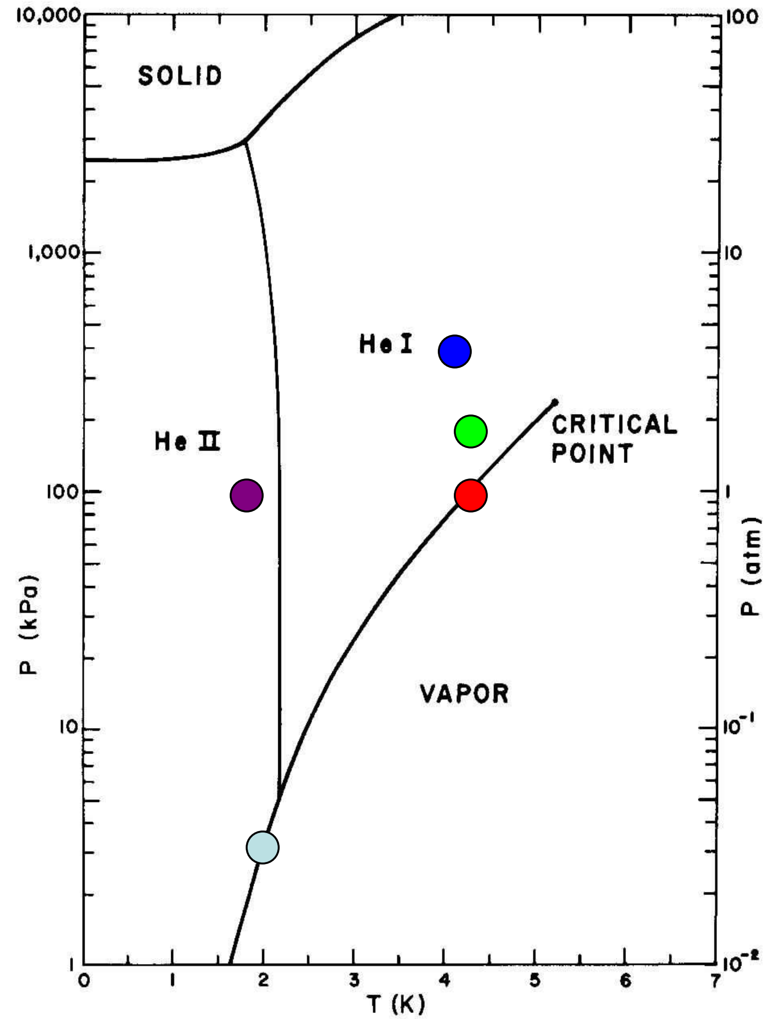

Fig. 3.1. ^{4}He phase diagram.

Cooling modes -- magnets vs RF

- Accelerator magnets are often cooled with subcooled liquid
  - Typically working near the limit of the superconductor with large stored energy
  - Ensure complete liquid coverage and penetration
- Superconducting RF cavities are generally cooled with a saturated bath
  - Large surface heat transfer in pool boiling for local "hot spots"
  - Very stable pressures, avoid impact of pressure variation on cavity tune

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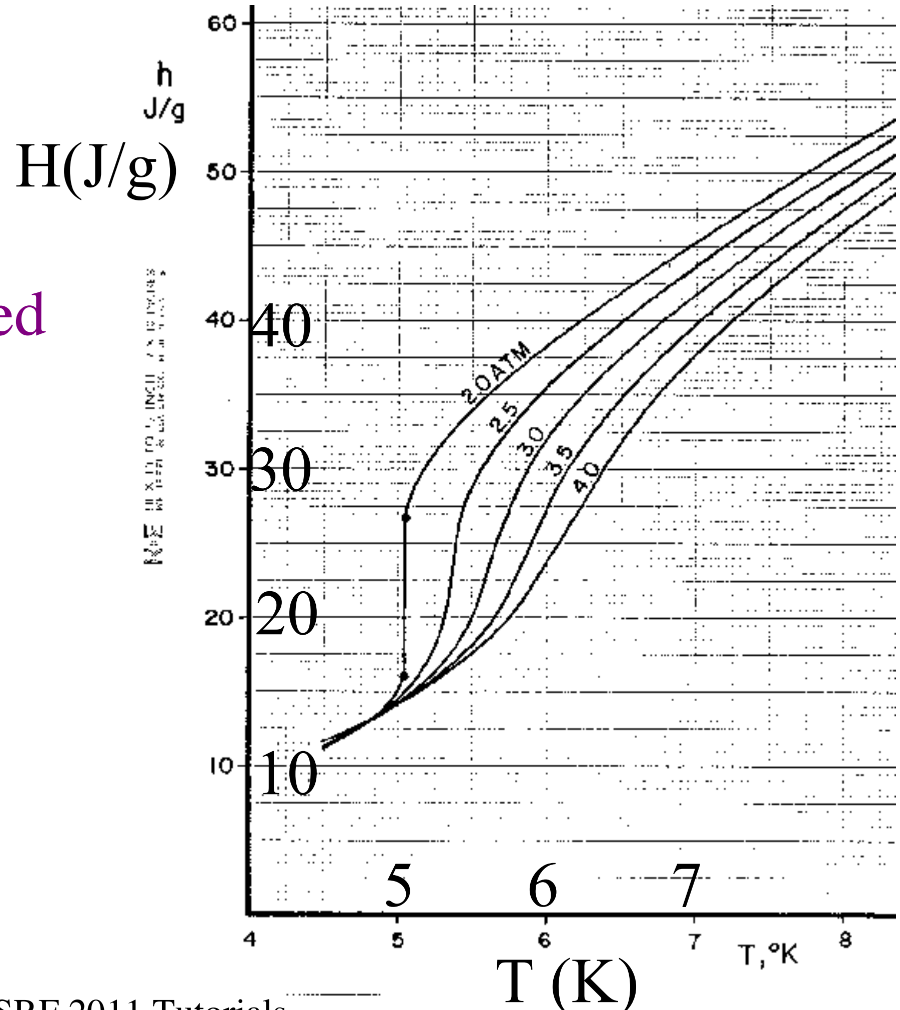
Heat transport through channels-- pressurized normal helium

- This plot of helium enthalpy versus T illustrates the large amount of heat absorbed (20+ J/g) if one can tolerate 6.5 K or even more
- Nominally “5 K” thermal intercept flow may take advantage of this heat capacity



Pool boiling and 2-phase flow

- Considerations for pool boiling systems
  - Control of liquid levels, long time constants, inventory management
  - Forced convection for warm-up and cool-down
- Two-phase flow
  - Liquid and vapor phases separate with any acceptably low pressure drop
  - Baker Plot does not apply!

Provisions for cool-down and warm-up

- Cool-down
  - Return vapor may block liquid supply flow in the same channel; a simple fill from the top or one end might not work. A cool-down vent and/or a bottom-fill port may be required.
- Warm-up
  - Flow will stratify. Local electric heat, a bottom vent port, or other feature to force heat down to the lower parts of a cold mass may be required.
    - The small “capillary” tubes connected to a manifold and providing helium to the bottoms of helium vessels in TESLA-style cryomodules were included primarily with warm-up in mind

Helium II heat transport

Conduction through ordinary materials is written as $q = k\frac{dT}{dx}$, where q is heat flux, T is temperature, and k is thermal conductivity. Heat transport through the pressurized superfluid with constant cross-section and constant heat flux obeys $q^m = \frac{1}{f(T)}\frac{dT}{dx}$ where m ≅ 3 and q is the heat flux in W/cm^2.

He II heat function

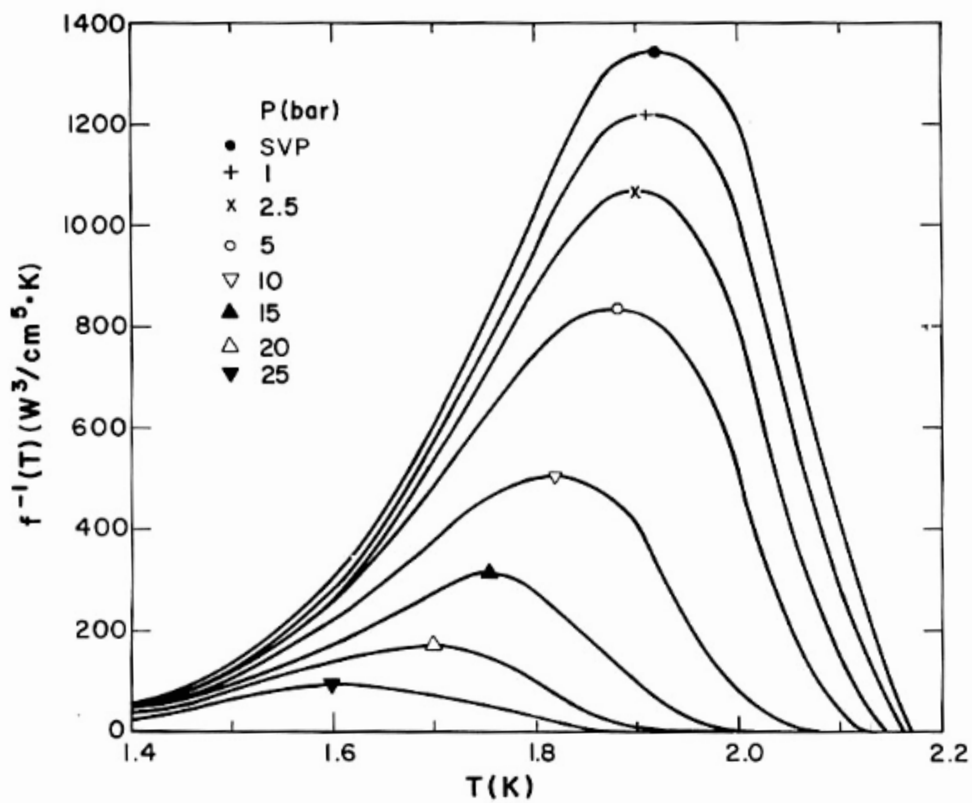


Fig. 5.3. Heat conductivity function for turbulent He II. Symbols indicate the location of the peak value.

Figure from
Steven Van Sciver,
Helium Cryogenics,
Plenum Press,
New York, 1986.

Helium II heat transport -- near saturation pressure

For the small temperature differences which we consider in heat transport from below the surface up to the surface of the saturated helium liquid, differences of at most 10's of mK, nominally all at 2.0 K, we may take constant $\frac{1}{f(T)} = 1200$ (W³/cm⁵-K), with units for q of W/cm². With constant heat flux through a channel (heat added at one end of the channel), m = 3, and constant $\frac{1}{f(T)} = 1200$, we have

$$\frac{1200\Delta T}{L} = q^3 \qquad (2)$$

or

$$q = \left(\frac{1200\Delta T}{L}\right)^{1/3} \qquad (3)$$

where L is distance in cm, q is the heat flux in W/cm², and delta-T is the temperature difference through the conduit in K.

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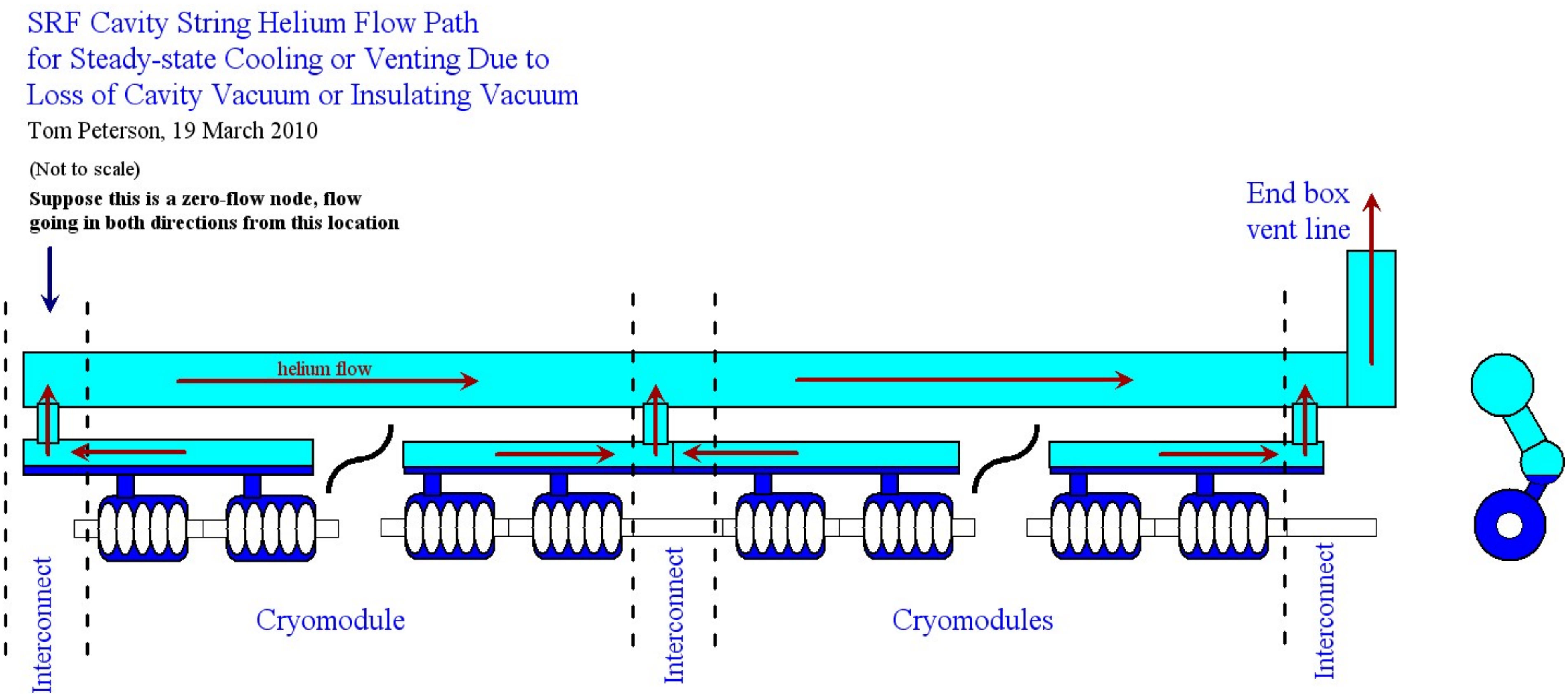
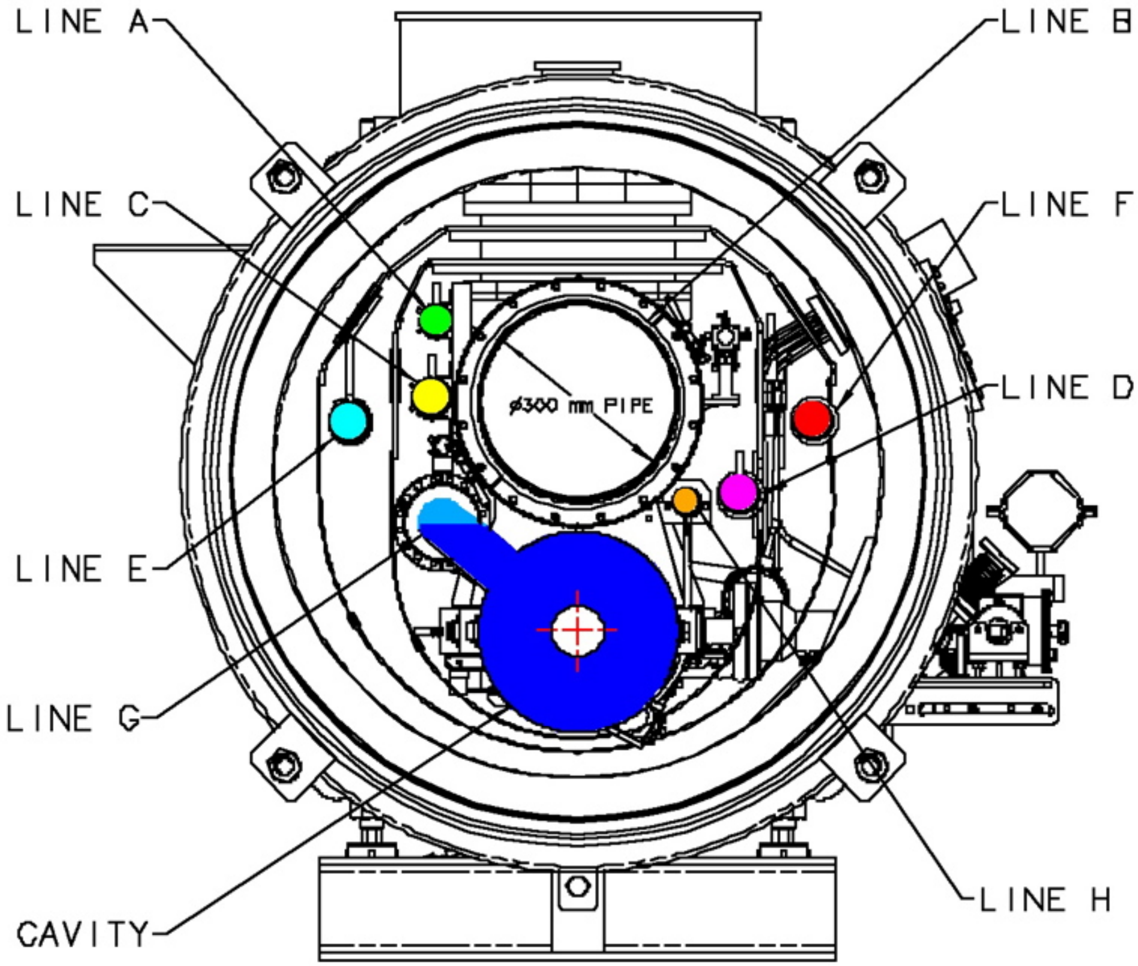
2-pipe 2 Kelvin vapor system

650 MHz cryomodule a modified TESLA-type



TTF cryomodule cross-section



Heat transport vertically to surface

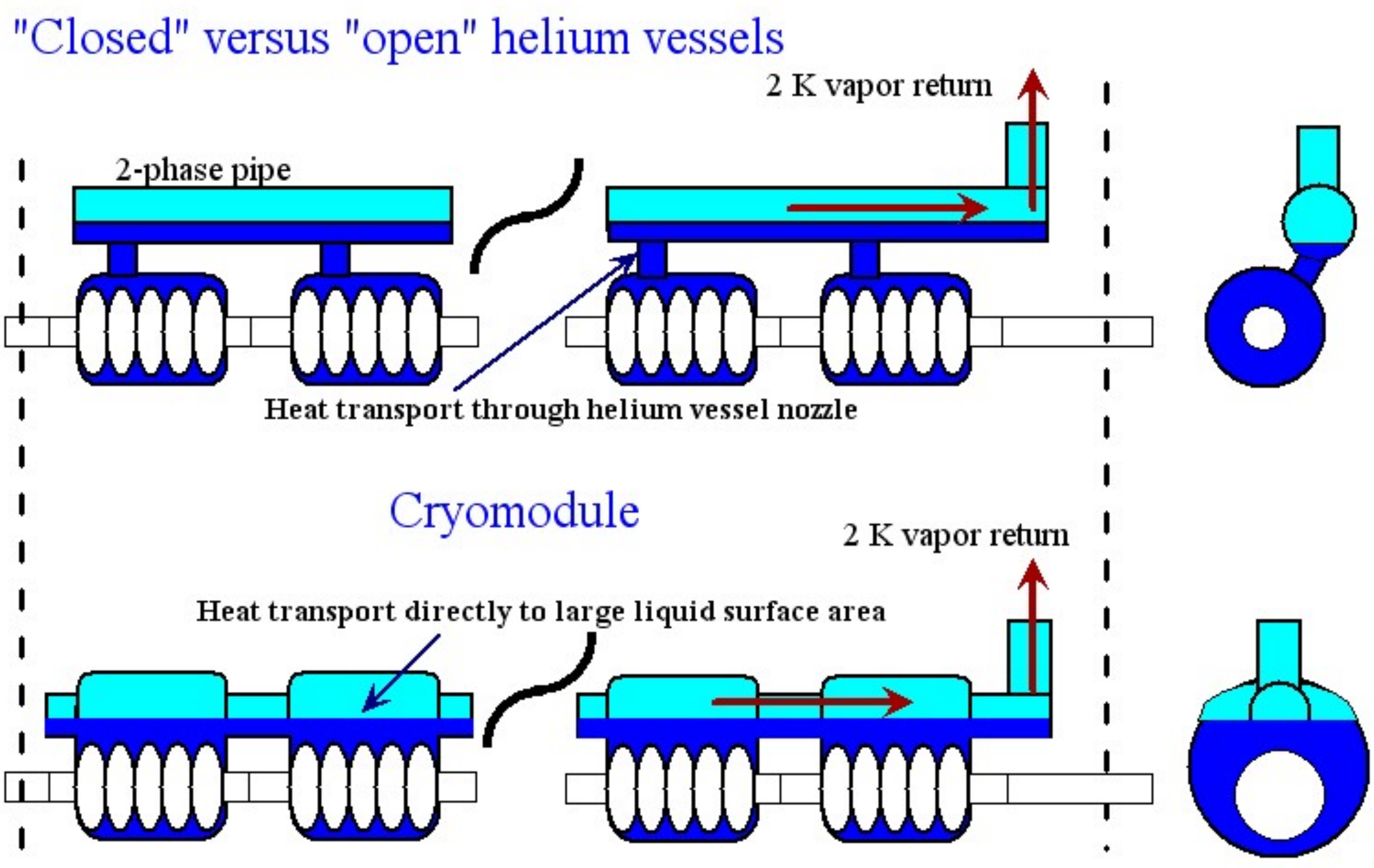
The ΔT available is determined by the depth below the surface through which heat transport occurs. Slight subcooling is provided by the pressure due to liquid weight (the liquid head). The head pressure $P = \rho g h$ where P is pressure, ρ is density, and h is height of the liquid. For helium at 2.0 K, 31 mbar, $P/h = \rho g = 0.142$ mbar/cm which is equivalent in saturation temperature to 1.5 mK/cm. Thus, the boiling temperature increases by 1.5 mK/cm of depth below the surface, providing a maximum of 1.5 mK/cm of temperature difference for heat transport.

Substituting 0.0015 K/cm into the previous equation results in 1.2 W/cm^2 heat flux vertically to the surface of the helium II. Note that this heat flux is essentially independent of depth for the depths of 10's of cm typical of RF helium vessels. I use 1 W/cm^2 as a limit for sizing a conduit for heat transport to the surface of a saturated helium II bath.

Helium vessel style

- Helium vessel style (open vs. closed) is independent of support style (hung from 300 mm pipe, space frame, rails)
- High heat loads and tight pressure stability ==>
  - Large liquid-vapor surface area for liquid-vapor equilibrium
  - Acts as thermal/pressure buffer with heat and pressure changes
- Linac is short enough that total helium inventory not an issue ==>
  - Open helium vessel is feasible
- For the stand-alone CW cryomodule, a closed TESLA-type helium vessel may be favored by
  - Tuner design
  - Input coupler design
  - And allowed by reduced pressure sensitivity

Helium vessel style and pressure stability

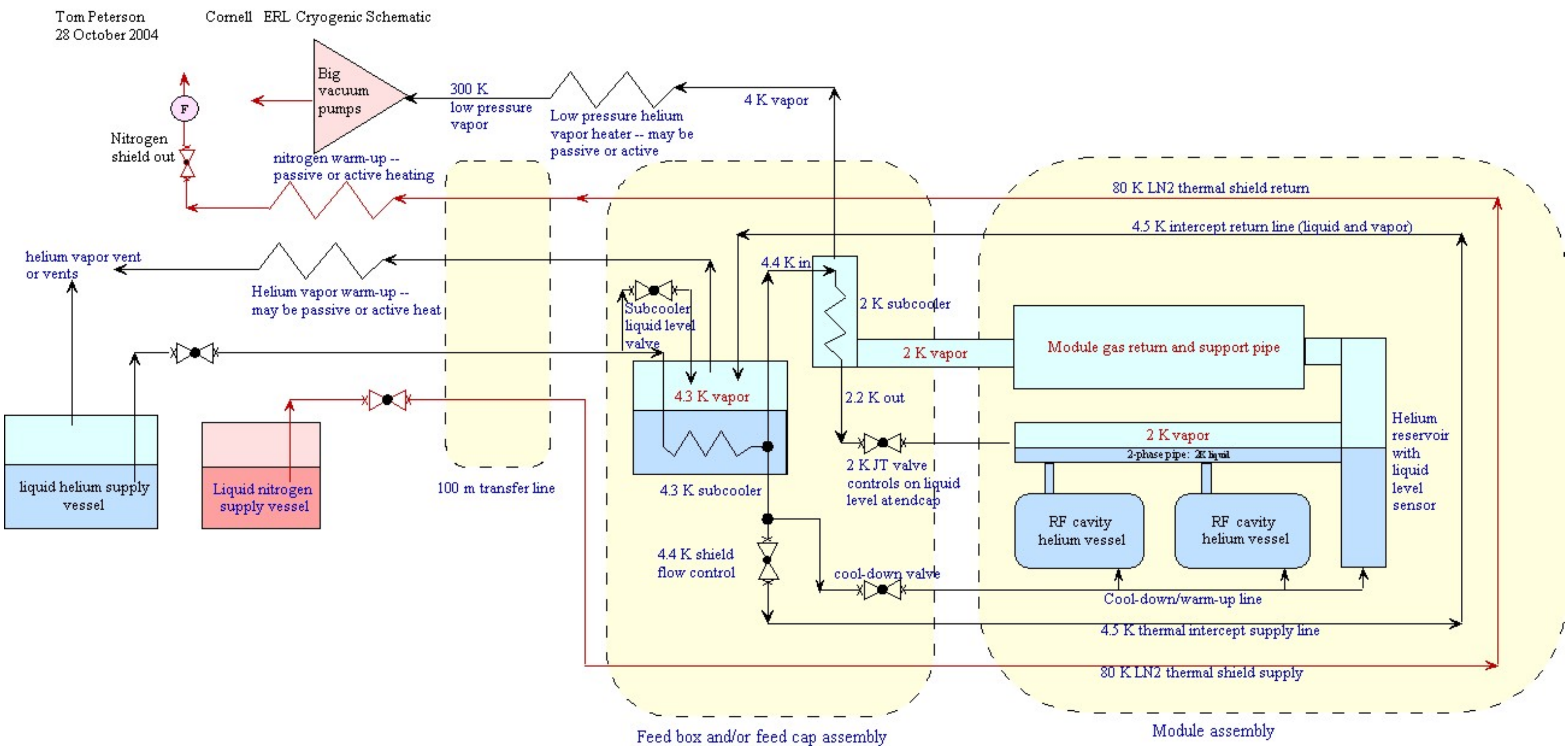


Rate of return to vapor-liquid equilibrium following a pressure change is limited by the rate of heat transport through the "chimney" for the closed helium vessel. Rate based on surface areas is about a factor 60 higher for the "open" vessel.

Providing 2 K on a test stand

- Test stand refrigeration requirements are typically small
  - A large, 2 K cryoplant will not be available
  - 4.5 K helium from either a small liquefier or storage dewars will provide refrigeration
  - Room-temperature vacuum pumps provide the low pressure for the low temperature helium
  - Small heat exchangers may be incorporated for continuous fill duty

Horizontal SRF test stand



Cryomodule requirements -- major components

- Dressed RF cavities
- RF power input couplers
- One intermediate temperature thermal shield
- Cryogenic valves
  - 2.0 K liquid level control valve
  - Cool-down/warm-up valve
  - 5 K thermal intercept flow control valve
- Pipe and cavity support structure
- Instrumentation -- RF, pressure, temperature, etc.
- Heat exchanger for 4.5 K to 2.2 K precooling of the liquid supply flow
- Bayonet connections for helium supply and return

Further considerations

- Support structure
  - Stiffness of pipe if used as support backbone
  - Or other support structure options
- Emergency venting scenarios drive pipe sizes and influence segmentation
  - Cold MAWP may be low, driving up pipe sizes and/or reducing spacing between relief vent ports
  - Trade-off of pipe size with vent spacing
  - Thermal shield pipe may also require frequent venting
    - 5 K may have large surface area for large heat flux
    - 70 K helium typically starts at a high pressure
- Liquid management length
  - May want to subdivide strings for liquid management due to large specific liquid flow rate per cavity

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Cryomodule requirements -- typical vessel and piping pressures

Region	Warm MAWP (bar)	Cold MAWP (bar)
2 K, low pressure space	2.0	4.0
2 K, positive pressure piping (separated by valves from low P space)	20.0	20.0
5 K piping	20.0	20.0
70 K piping	20.0	20.0
Insulating vacuum space	1 atm external with full vacuum inside 0.5 positive differential internal	
Cavity vacuum	2.0 bar external with full vacuum inside 0.5 positive differential internal	4.0 bar external with full vacuum inside 0.5 positive differential internal
Beam pipe vacuum outside of cavities	1 atm external with full vacuum inside 0.5 positive differential internal	1 atm external with full vacuum inside 0.5 positive differential internal

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Cryomodule string segmentation

- Various degrees of possible segmentation
  - Total isolation and warm-up with adjacent cryomodule cold
  - Total separation of vacuum and cryogenic circuits but no provisions for maintaining segments cold which are adjacent to warm cryomodules
    - Limits extent of control lengths and vacuum
    - No end buffer from adjacent cold segment for vacuum let-up of warm segment. E.g., one must warm three segments to let up one.
    - Segments downstream of warm one may not be held cold
  - Vacuum isolation and/or cryo circuit extent of various lengths, not all equal
    - For example, liquid flow path shorter than thermal shield circuits
    - Some valves distributed more finely than others. May have small "feed boxes" or valves in cryomodules for 2 K liquid supply.
  - Relief valve frequency -- low MAWP may force frequent vents
- Pipe sizes trade off with segmentation lengths

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Cryomodule Pipe Sizing Criteria

- Heat transport from cavity to 2-phase pipe
  - 1 Watt/cm^2 is a conservative rule for a vertical pipe (less heat flux with horizontal lengths)
- Two phase pipe size
  - 5 meters/sec vapor "speed limit" over liquid
  - Not smaller than nozzle from helium vessel
- Gas return pipe (also serves as the support pipe in TESLA-style CM)
  - Pressure drop < 10% of total pressure in normal operation
  - Support structure considerations
- Loss of vacuum venting P < cold MAWP at cavity
  - Path includes nozzle from helium vessel, 2-phase pipe, may include gas return pipe, and any external vent lines

Evaluating the relief venting flow rate and conditions

These may determine pipe sizing and segmentation, rather than steady-state considerations

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

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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

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

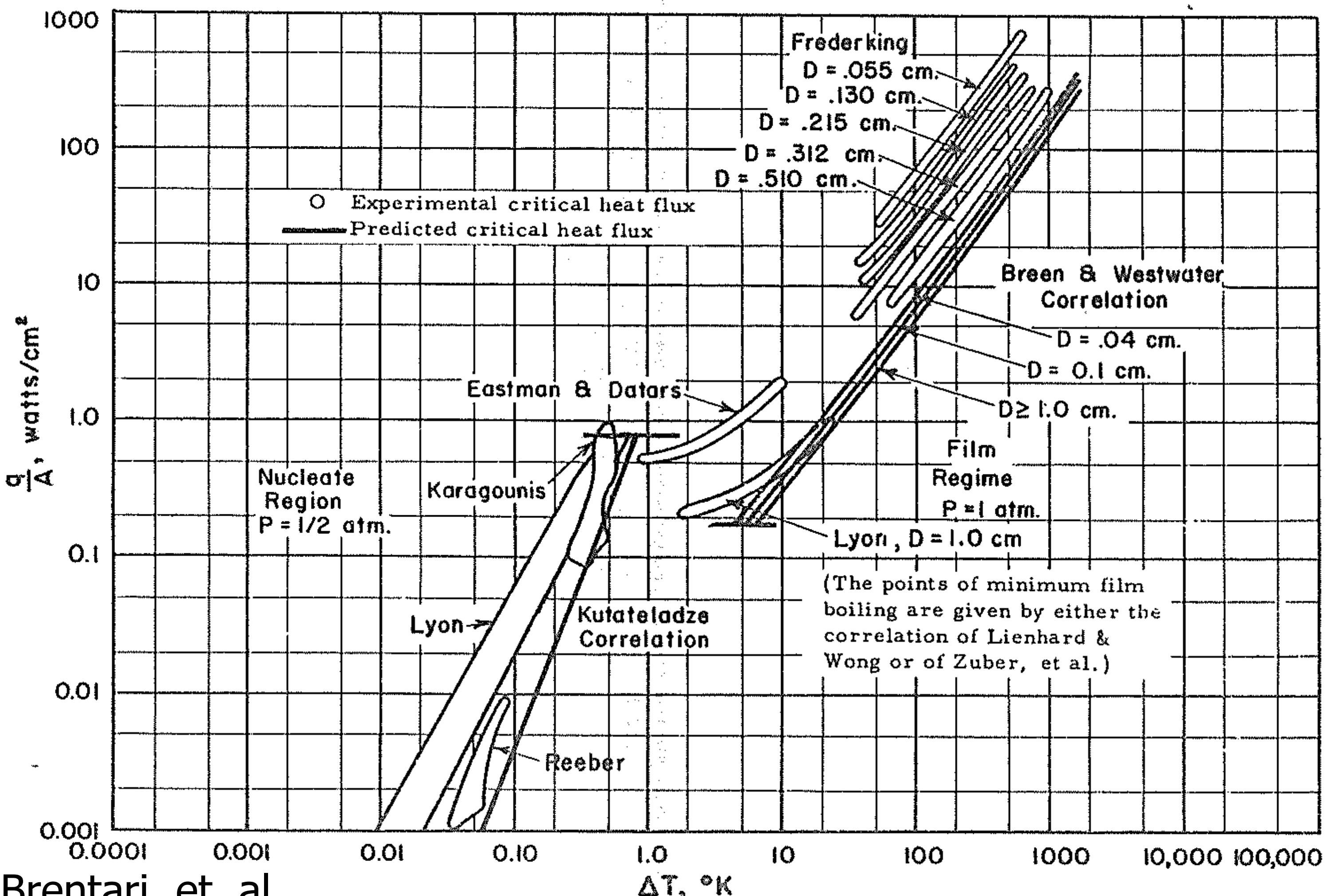
- W. Lehman and G. Zahn, “Safety Aspects for LHe Cryostats and LHe Transport Containers,” ICEC7, London, 1978
- G. Cavallari, et. al., “Pressure Protection against Vacuum Failures on the Cryostats for LEP SC Cavities,” 4th Workshop on RF Superconductivity, Tsukuba, Japan, 14-18 August, 1989
- M. Wiseman, 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

- 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

Other heat flux comments

- 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 slide)
- I use 4 W/cm^2 for bare metal surfaces



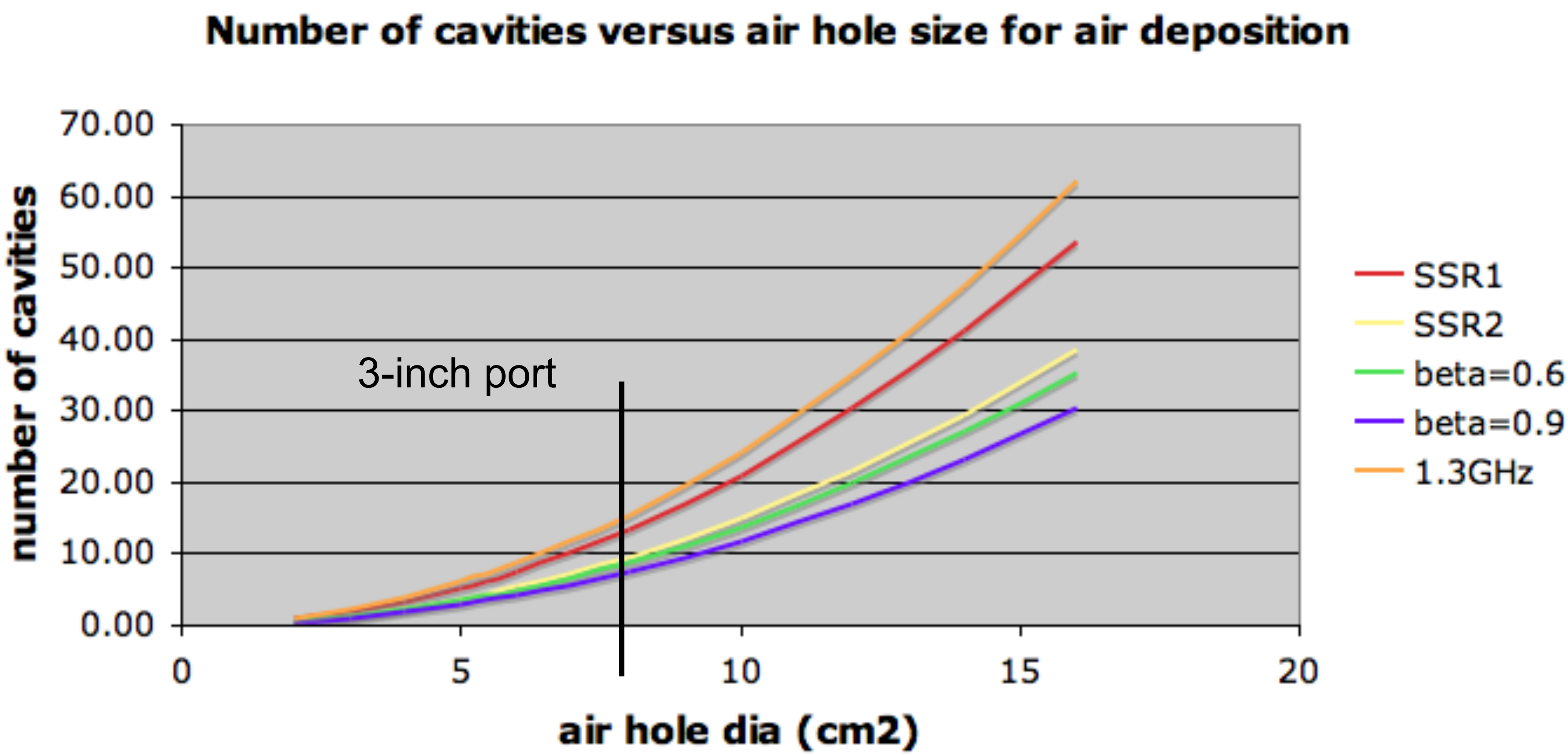
E. G. Brentari, et. al.,
NBS Technical Note 317

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

Air inflow heat flux limit

- Atmospheric air flowing into a vacuum via a round hole
  - ~23 grams/sec per cm2 hole size
- Heat deposition by air condensing on cold surface
  - ~470 J/g, so 10.8 kW per cm2 hole size
- Helium heat input per gram ejected for typical (2.5 - 4 bar) pressures
  - ~13 J/g
- Helium mass flow per air inlet area
  - ~830 grams/sec helium per cm2 hole size

For a 3-inch (76 mm) diameter opening, air flow becomes the limiting factor in heat deposition after a few cryomodules



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*
- 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.

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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

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

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Constraints and assumptions

- For ODH analysis
  - Pipe size may not be a free parameter
    - Analyses are often done for existing systems
  - A flow estimate is based on worst-case pressures and rupture or open valve assumptions
    - Worst-case in terms of maximum flow of inert gas

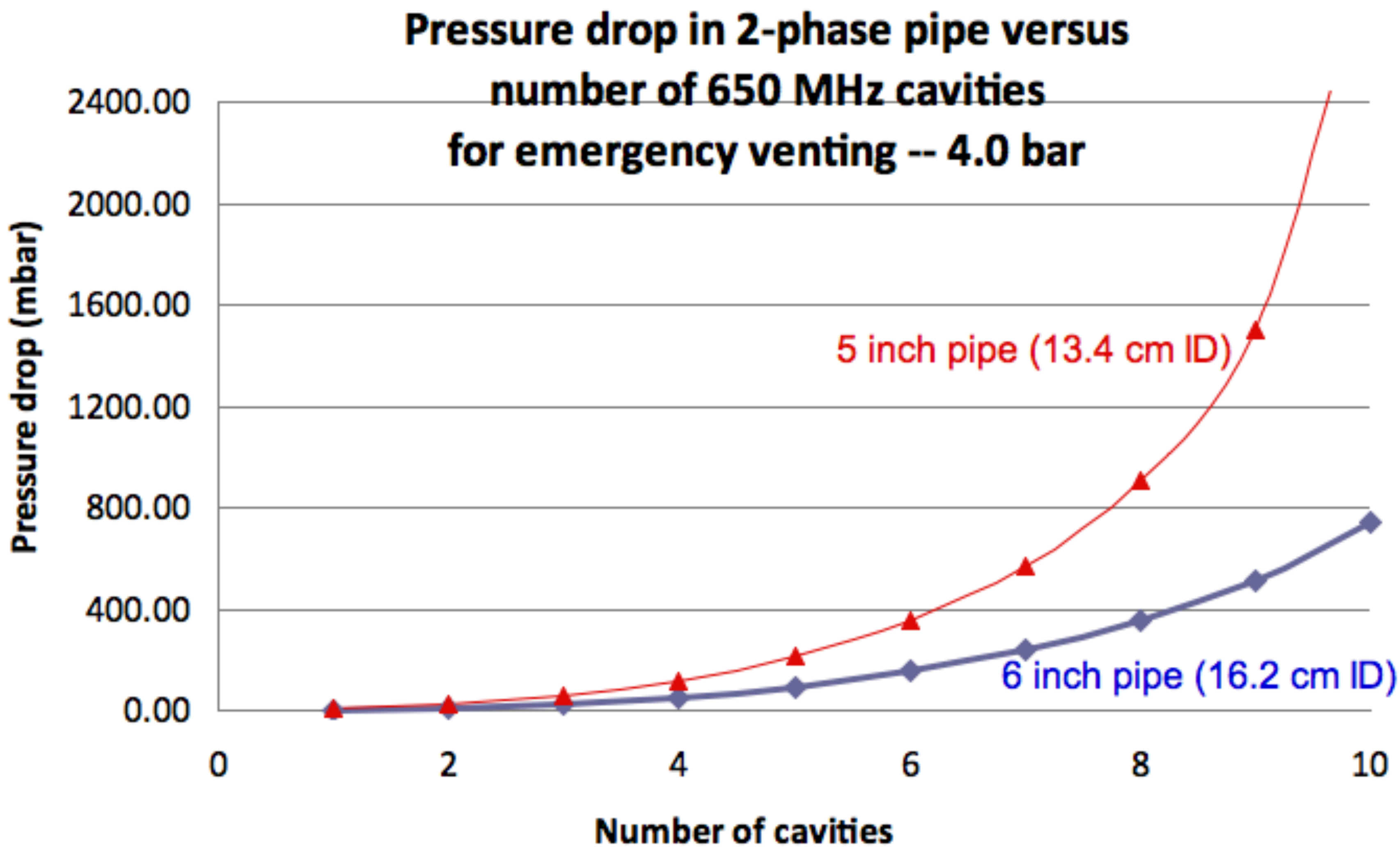
Venting and relief sizing analysis

- 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 severe constraints)

Vent sizing vs ODH flow estimate

- Vent sizing goal is to show that venting system (piping and relief devices) carry flow which starts at or below MAWP
  - So pressure drop estimate may be conservatively high so as to end with a conservatively low flow rate and verify safely large vent system size
- ODH venting analysis may be to estimate flow of inert gas into a space
  - So pressure drop estimate may be conservatively low so as to end with a conservatively high flow rate and verify safely large room ventilation

Example: 650 MHz cavities



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Liquid management length

- The 2 K to 4.5 K heat exchanger needs to be divided (not one large heat exchanger) in order to be a practical size, which means distributing multiple heat exchangers in the tunnel.
- 2 K to 4.5 K heat exchanger size which fits in the Project X tunnel will be roughly 10 - 15 grams/sec (about 200 - 300 Watts of 2 K heat)
- With 650 MHz and 1.3 GHz CW heat loads of nearly 200 W per cryomodule, this implies liquid management lengths of one or two cryomodules as limited by JT heat exchanger practical size limits

Various cryomodule concepts

- Single Spoke Resonator (SSR) cryostat concept using support posts under the cavities and magnets
- TESLA/ILC
- SNS/Jlab 12 GeV upgrade style "space frame" supports
- Closed-ended "TESLA" style
- BESSY/HZB CW cryomodule string rather than stand-alone cryomodules
- Cornell's ERL cryomodule
- ANL/FRIB/TRIUMF-style top-loading rectangular cryostats

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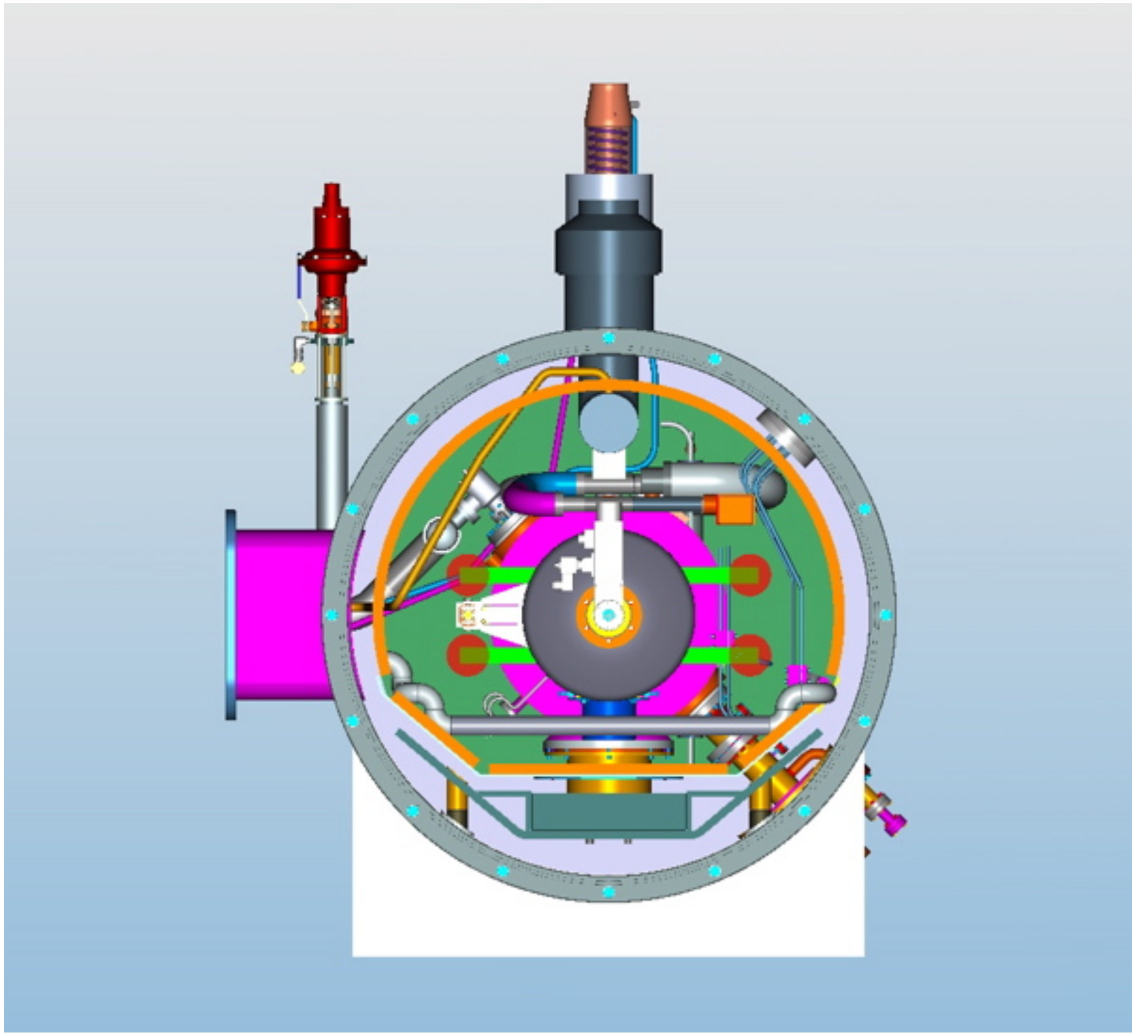
SSR cryomodule concept



Vacuum vessel end removed

Support structure is a cradle on which magnets and cavities are supported

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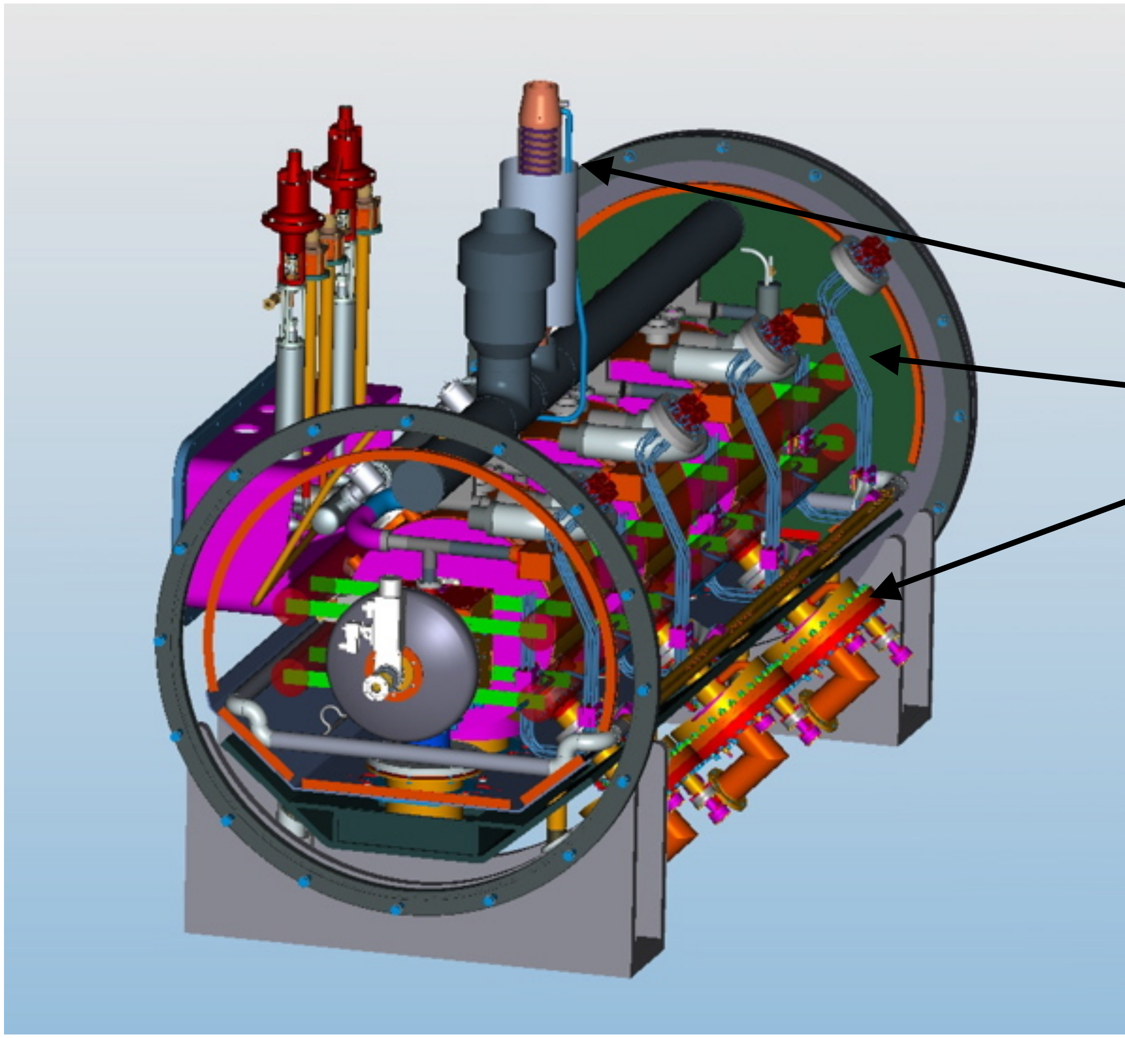
SSR cryomodule concept



Heat exchanger

Current leads

Input couplers

48 inch diameter vacuum vessel

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SSR cryomodule concept

Still working on conceptual design as 3-D model, but becoming a complete model

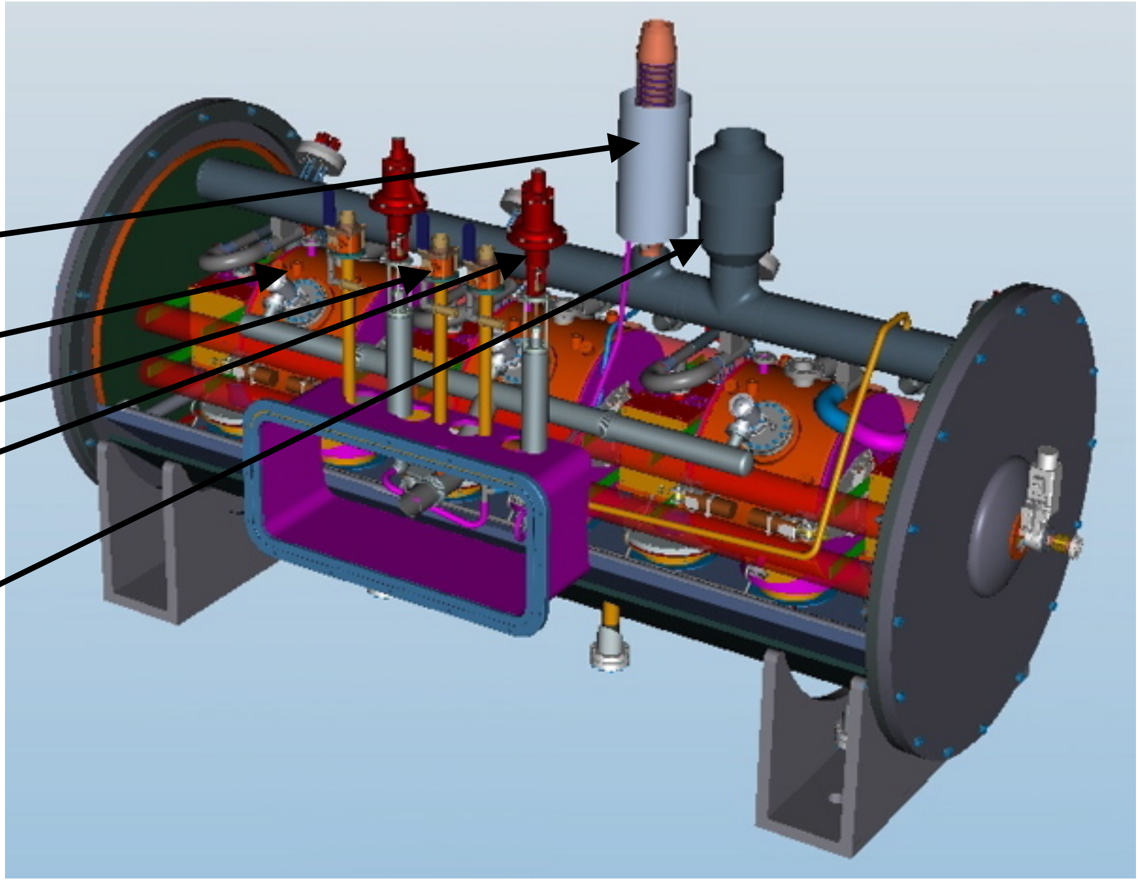
Heat exchanger

Cavity helium vessel

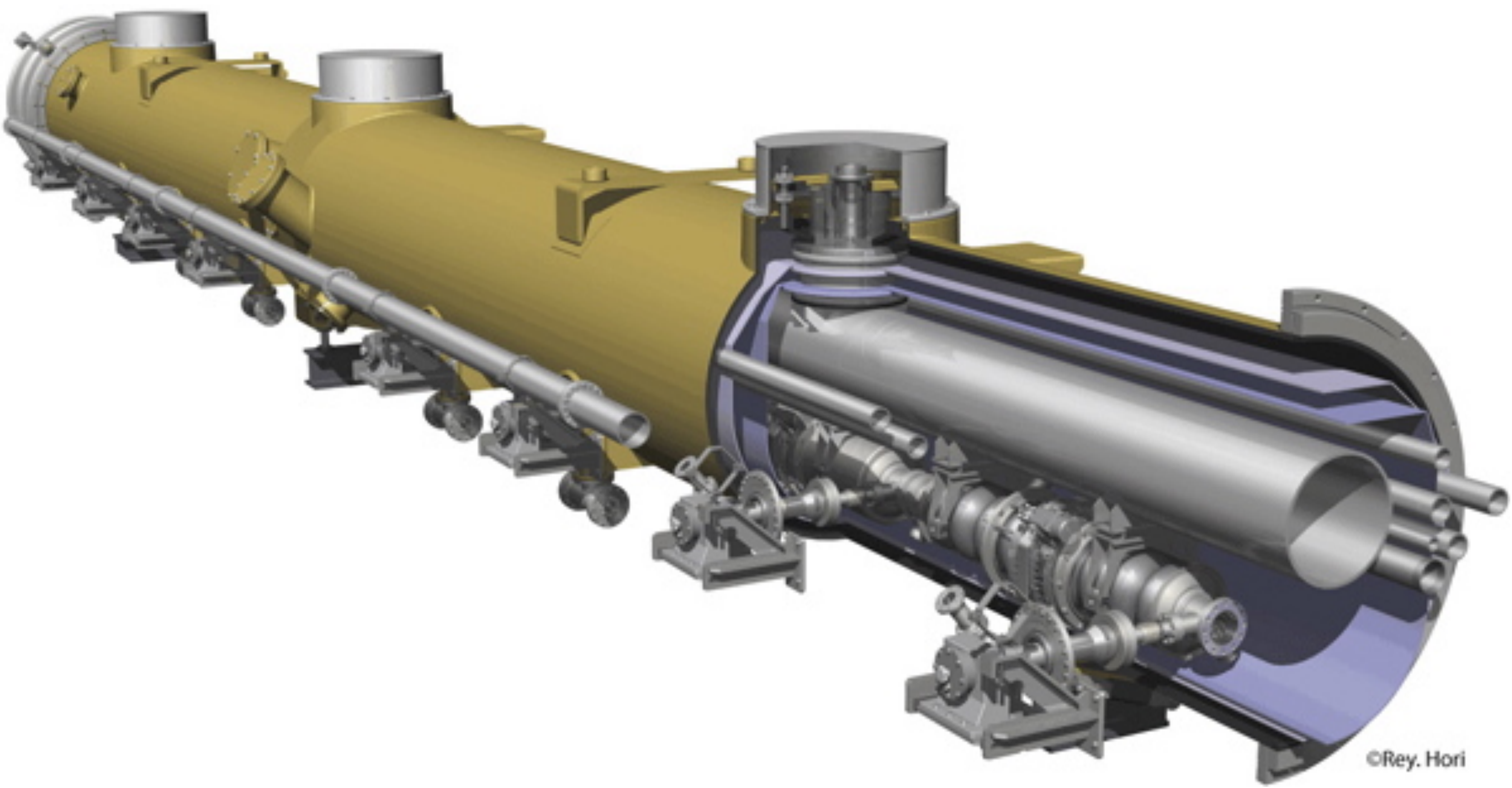
Bayonet connection

Control valve

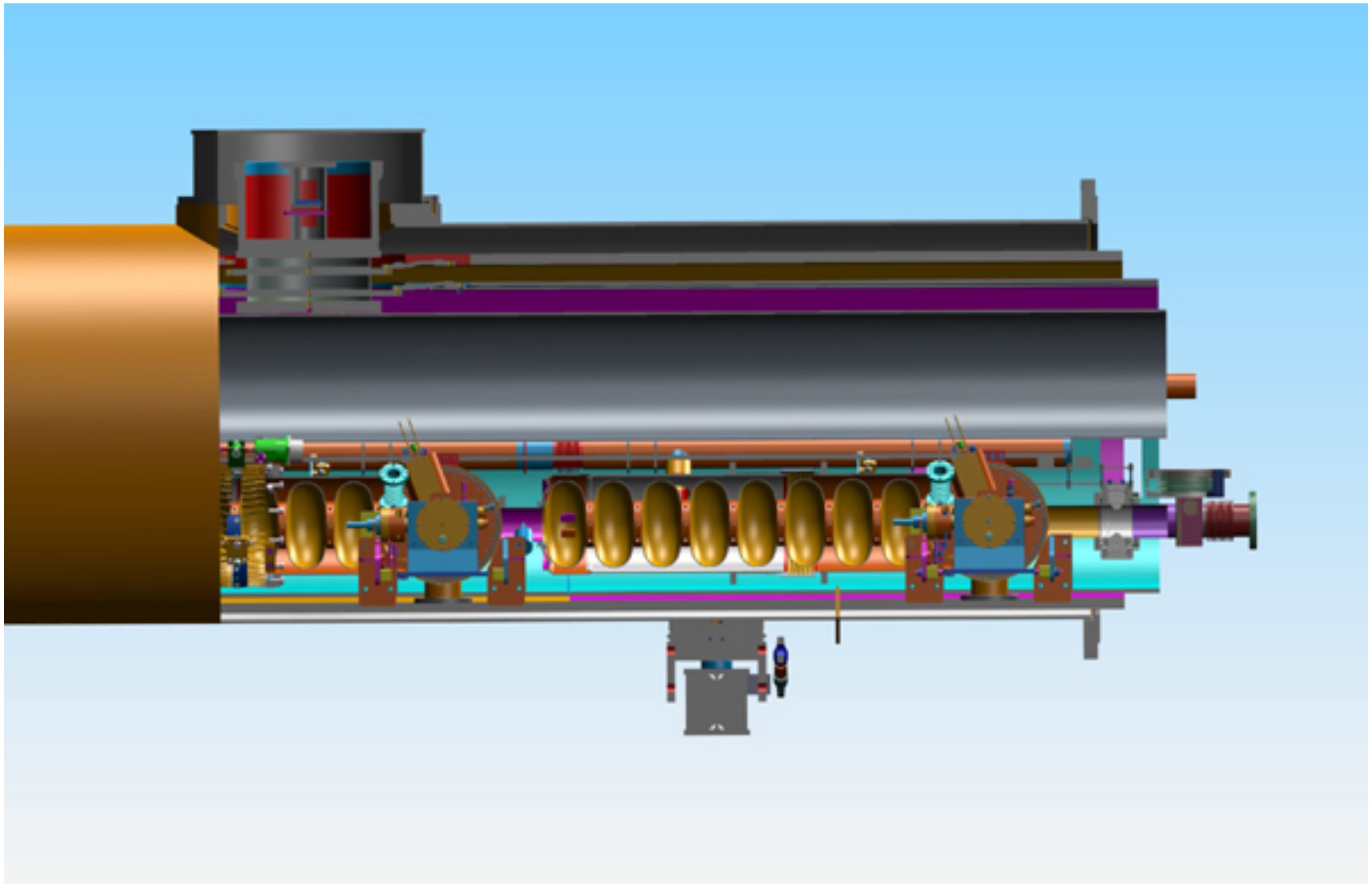
Vent line with check valve



TESLA-style Cryomodule



Cutaway view of cavity within a cryomodule



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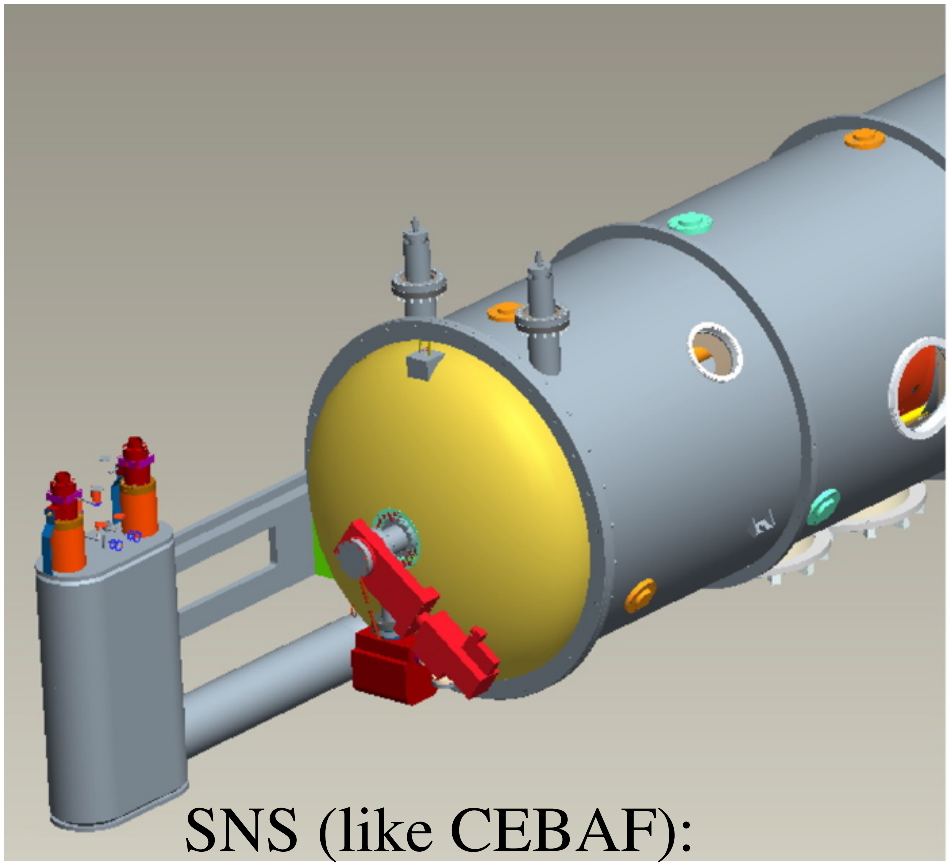
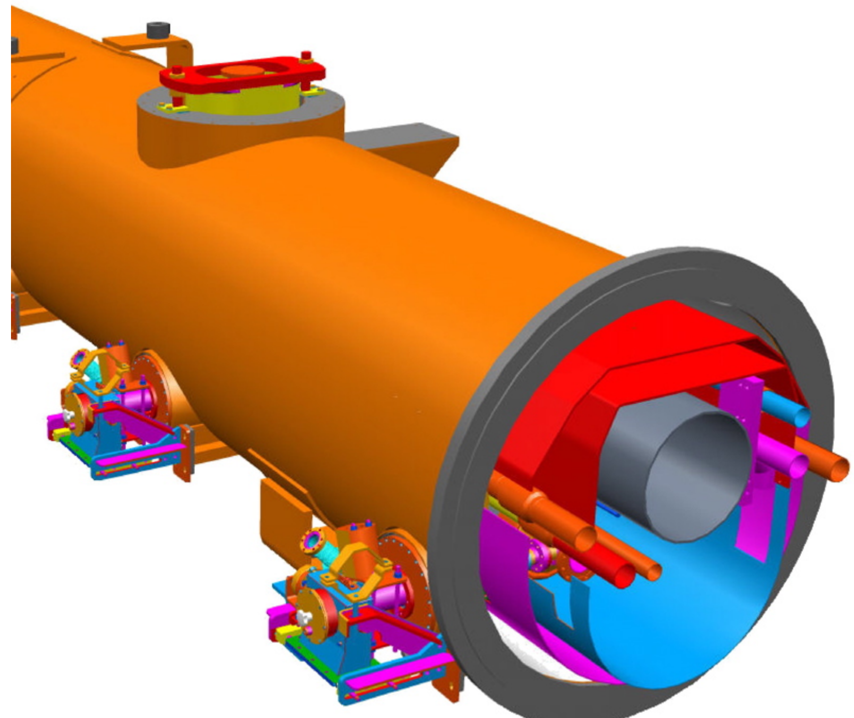
SNS vs TTF cryomodule

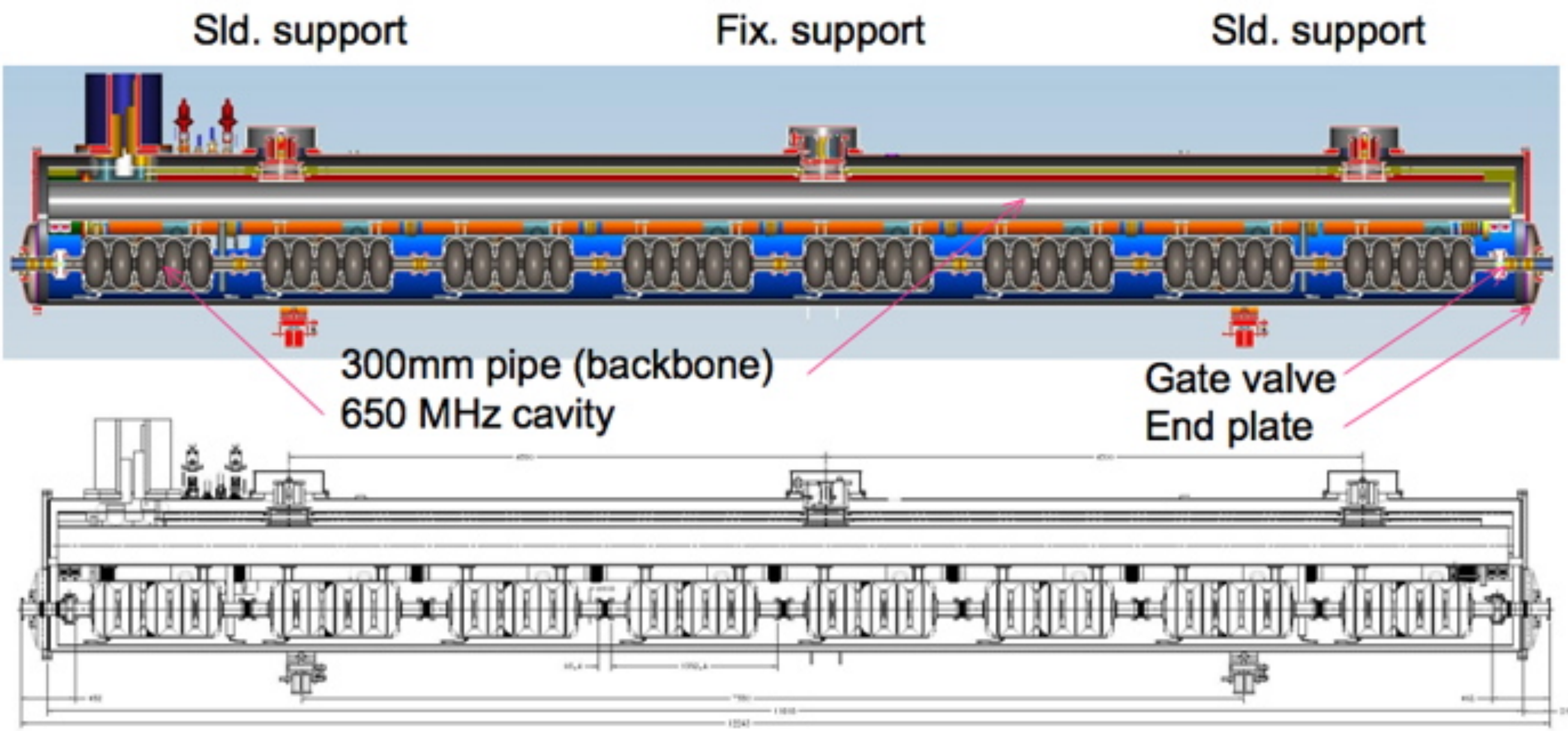


SNS (like CEBAF):
self-contained vacuum vessel
“stand-alone” style



TTF: vacuum vessel string.
End boxes and bellows
would become part of
vacuum/pressure closure

Closed ended "TESLA" style



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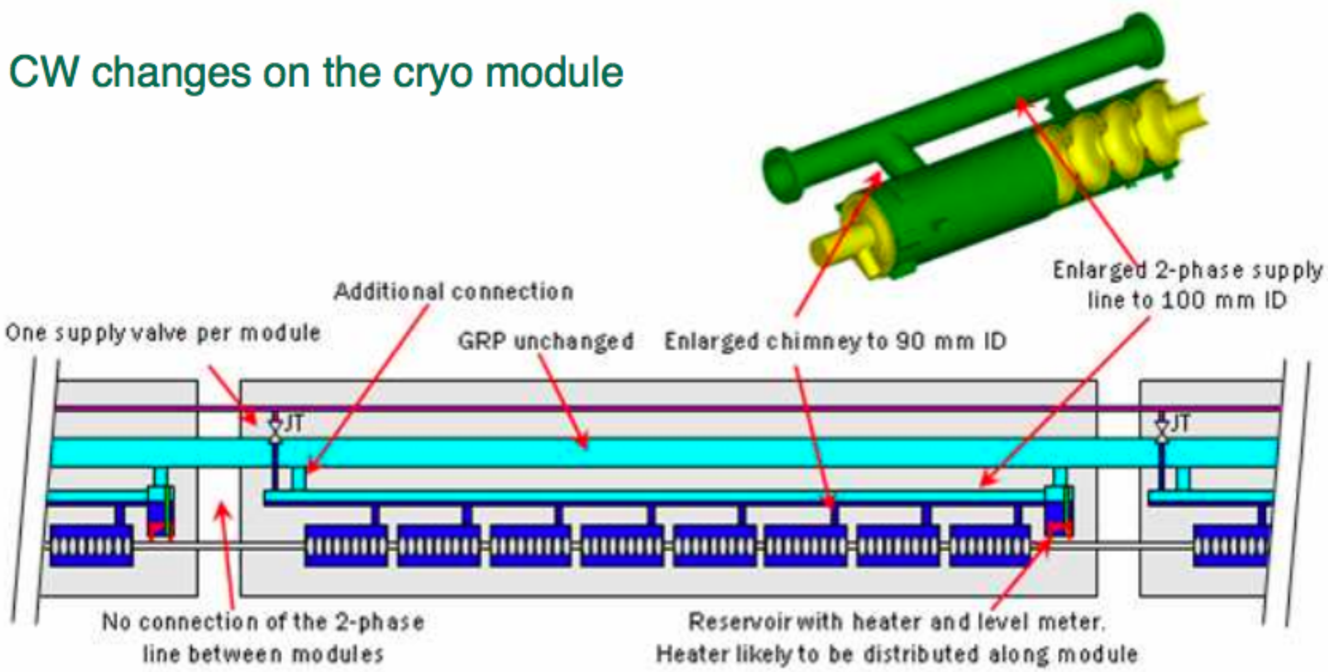
CW Cryo Module

BESSY

CW changes on the cryo module



W. Anders, BESSY CW Operation of SC TESLA Cavities ESLS RF Meeting 5.10.2007

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ERL cryomodule features

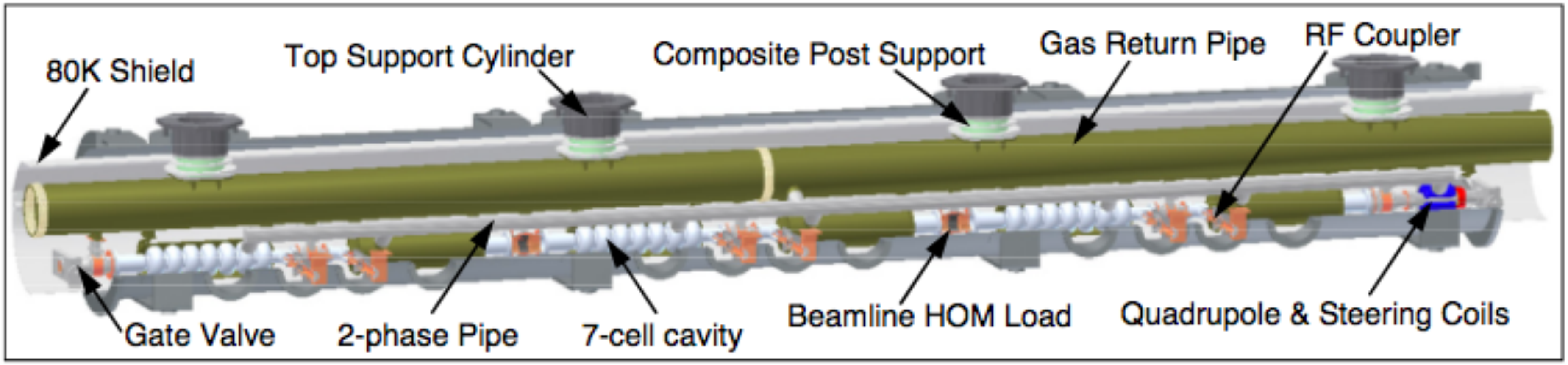


Figure 1: A cut-away CAD model showing the main features of the ERL Linac cryomodule.

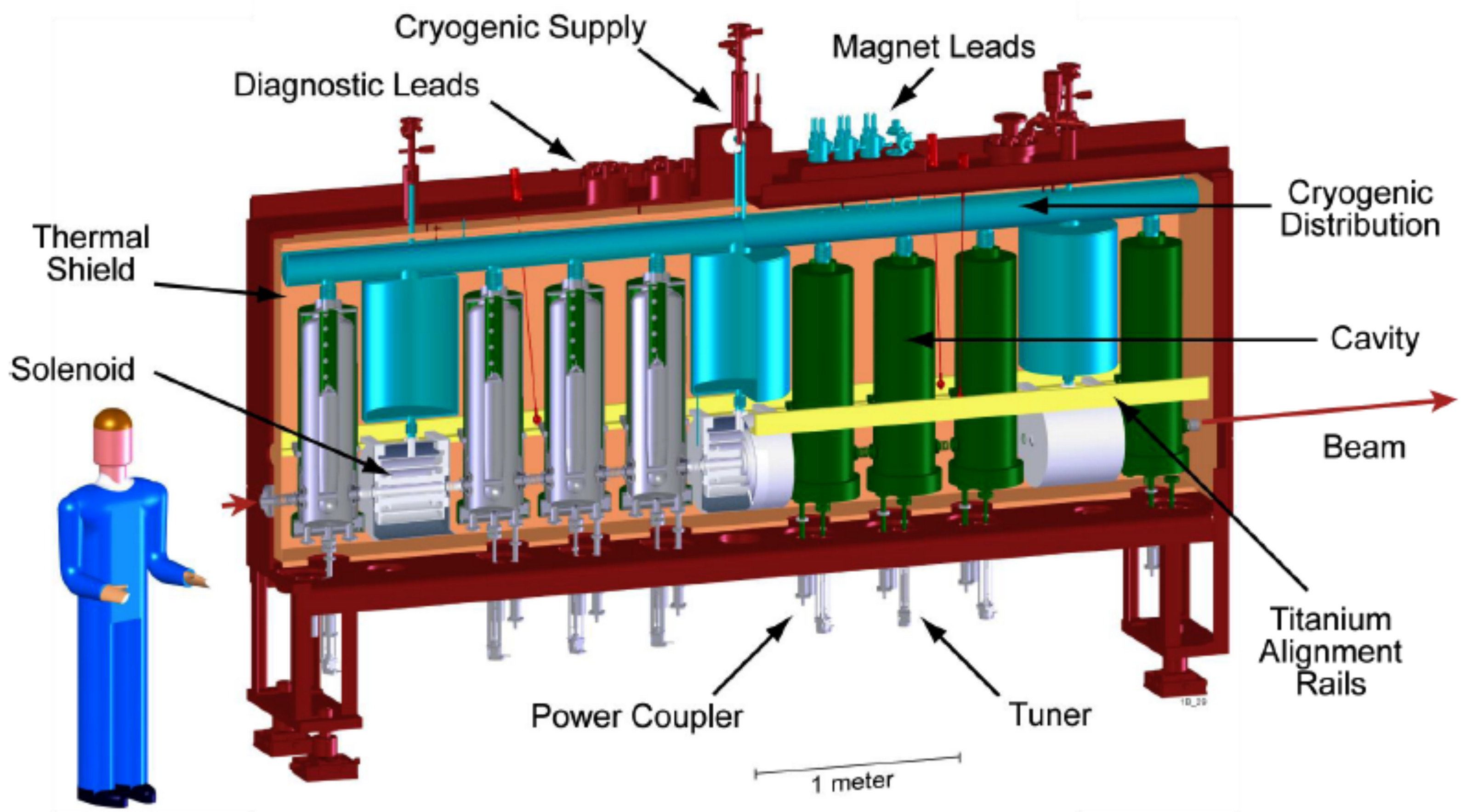
Figure 1 from CRYOGENIC HEAT LOAD OF THE CORNELL ERL MAIN LINAC CRYOMODULE, by E. Chojnacki, E. Smith, R. Ehrlich, V. Veshcherevich and S. Chapman, Cornell University, Ithaca, NY, U.S.A.

Published in Proceedings of SRF2009, Berlin, Germany

ERL cryomodule features

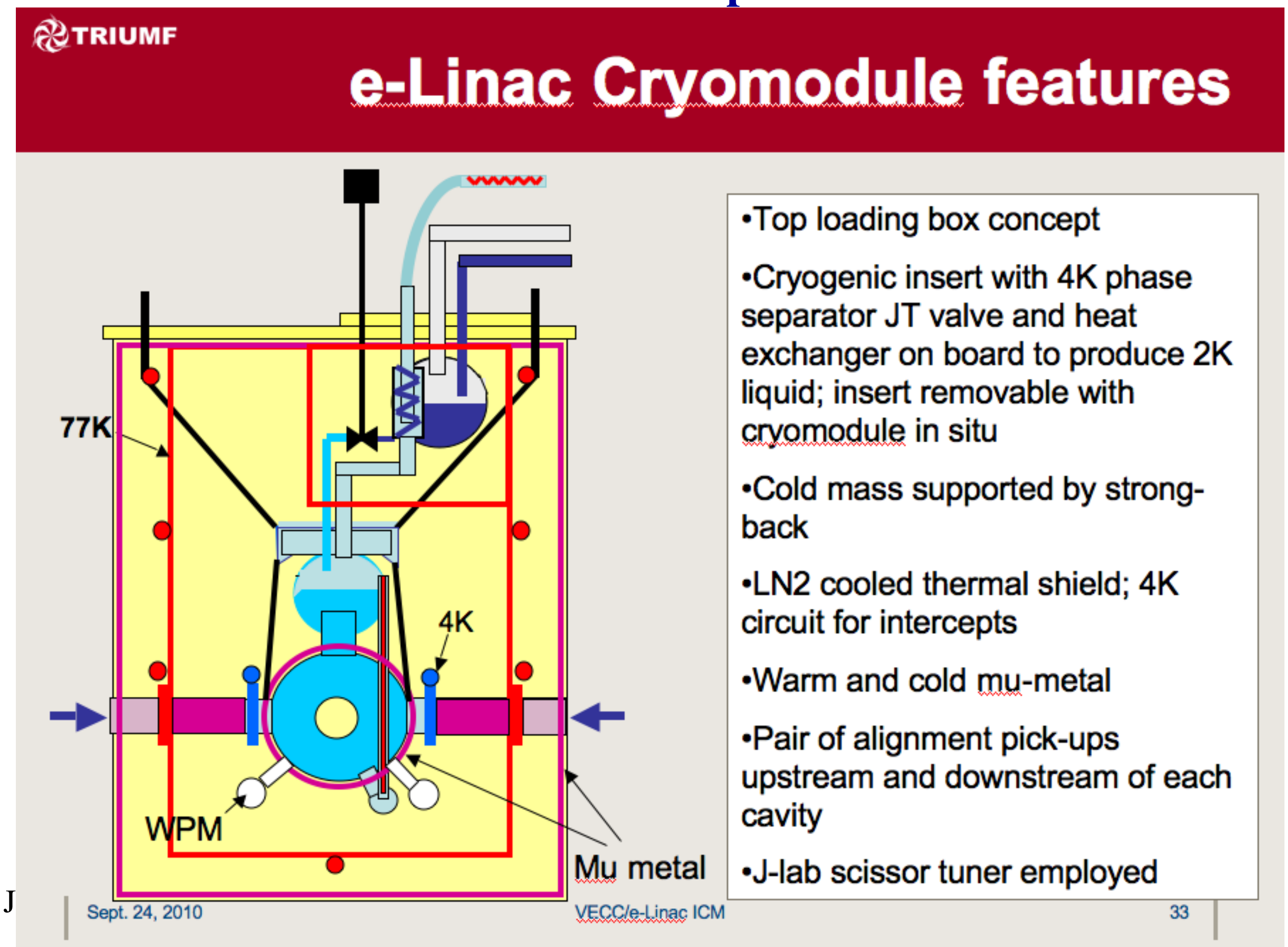
- TESLA-style support structure -- dressed cavities hang from gas return pipe (GRP), but
  - Titanium GRP
  - No invar rod, no rollers
  - 6 cavities per CM, 9.8 m total CM length
  - HOM absorbers at 40 - 100 K between cavities
  - GRP split with bellows at center, 4 support posts
  - Helium vessels pinned to GRP
  - Some flexibility in the input coupler
  - De-magnetized carbon-steel shell for magnetic shielding (this is like TTF)
  - 2-phase pipe closed at each CM end, JT valve on each CM (like BESSY design)
  - String rolls into vacuum vessel on rails

FRIB cryomodule
“bathtub” style for comparison



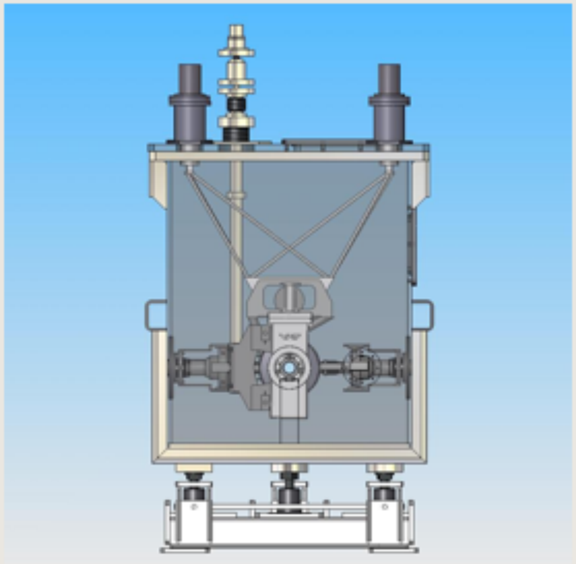
TRIUMF cryomodule concept

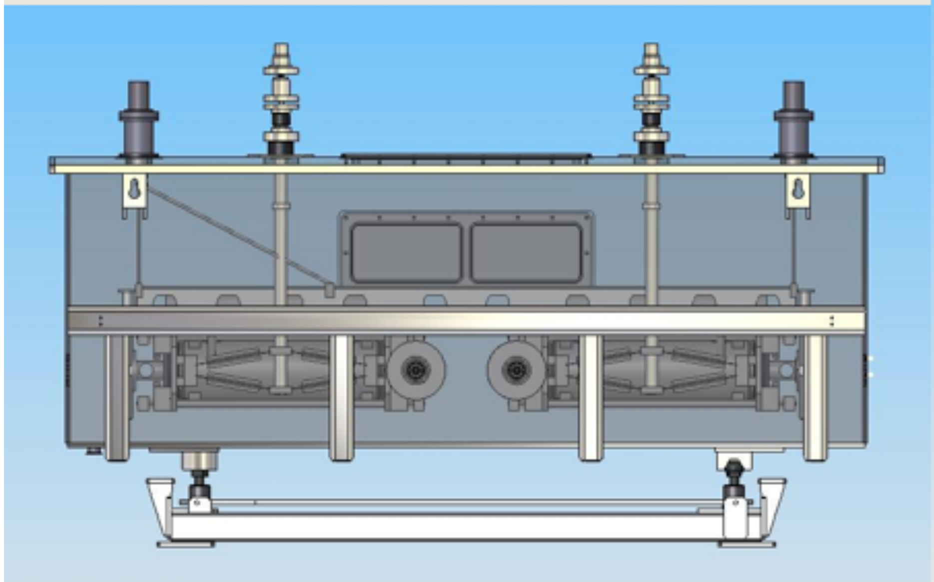
"bathtub" style with 1.3 GHz cavities for comparison

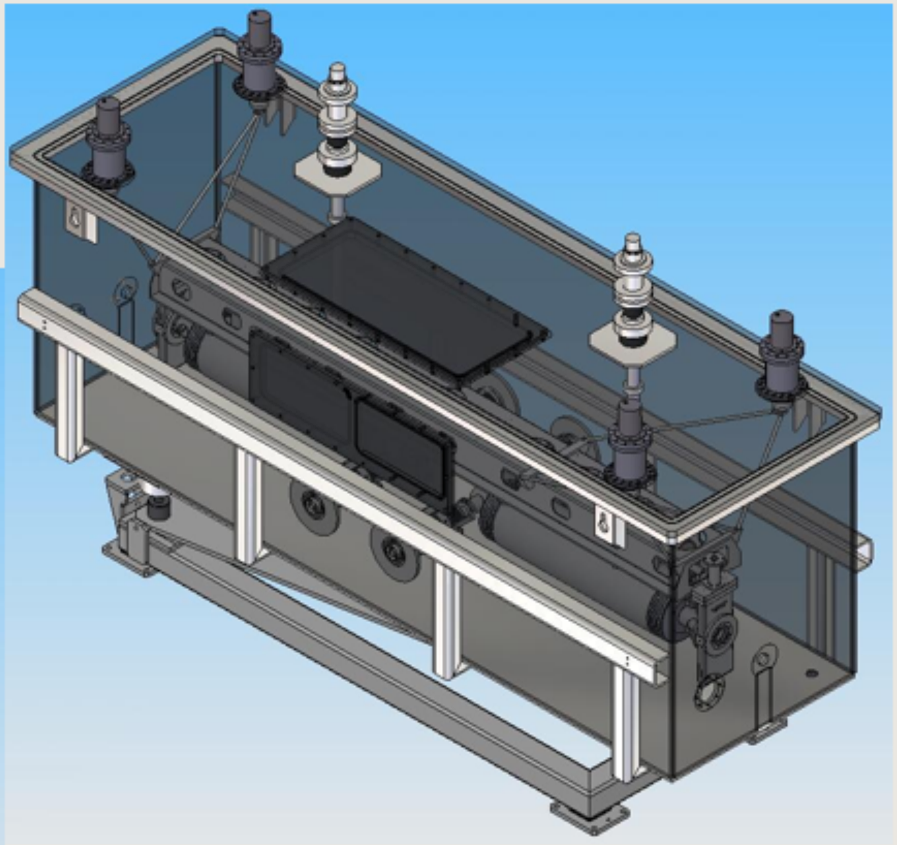


e-Linac top-load box concept

- Cold mass (cavity string, tuners) supported from strongback
- Strongback held in place by support posts strung from the lid







Conclusions

- Cryomodule must be designed as part of the larger cryogenic system
- Vacuum and cryogenic line segmentation choices depend on heat loads as well as various mechanical, alignment, and reliability considerations
- Off-design requirements such as emergency venting may determine line sizes and design features
- There is no single best design configuration. A variety of quite different designs already exist with features appropriate to the various specific applications

Extra slides

and references

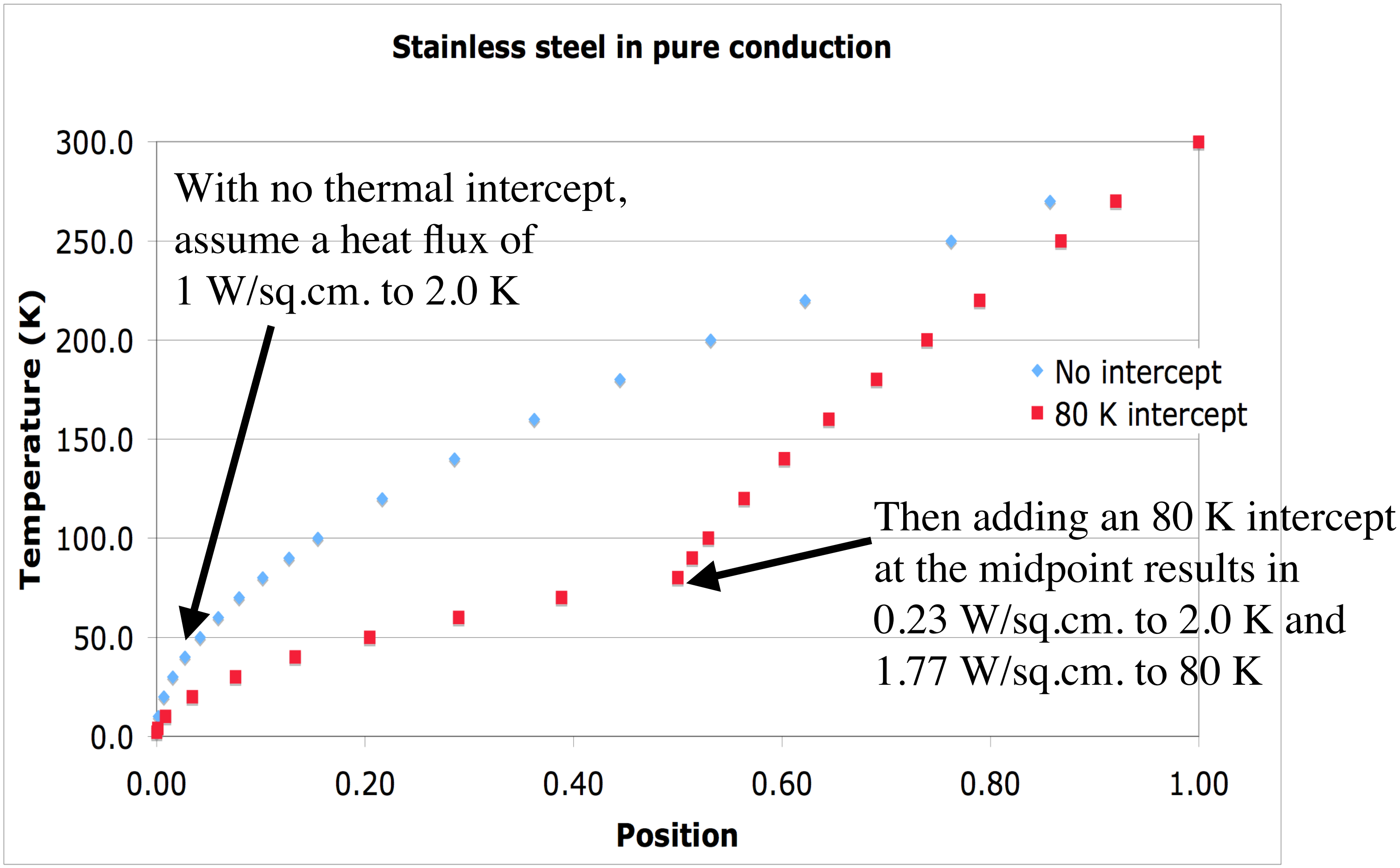
Room-temperature power

Cryoplant coefficient of performance (W/W)

	40 K - 80 K	5 K - 8 K	2 K
TESLA TDR:	17	168	588
XFEL:	20	220	870
Industrial est:	16.5	200	700
ILC assumption:	16.4	197.9	703.0

- Above numbers were compiled for the ILC RDR. A good estimate for isothermal cooling would be
  - 250 W/W @ 4.5 K
  - 12 W/W @ 80 K

Thermal intercepts



Thermal intercept benefit

- From previous illustration:
  - No thermal intercept:
    - 1 W/sq.cm. to 2 K x 700 W/W = 700 W of room temperature power per sq.cm. of rod between 2 K and 300 K
  - With thermal intercept:
    - 0.23 W/sq.cm. to 2 K x 700 W/W + 1.77 W/sq.cm. to 80 K x 12 W/W = 161 W + 21 W = 182 W of room temperature power per sq.cm. of rod between 2 K and 300 K

CW cryomodule requirements

1 Fermilab's baseline design concept includes cryomodules closed at each end, individual insulating vacuums, with warm beam pipe and magnets in between cryomodules such that individual cryomodules can be warmed up and removed while adjacent cryomodules are cold.
2 Provide the required insulating and beam vacuum reliably
3 Minimize cavity vibration and coupling of external sources to cavities
4 Provide good cavity alignment (<0.5 mm)
5 Allow removal of up to 250 W at 2 K per cryomodule
6 Protect the helium and vacuum spaces including the RF cavity from exceeding allowable pressures.
7 Intercept significant heat loads at intermediate temperatures above 2.0 K to the extent possible in full CW operation
8 Provide high reliability in all aspects of the cryomodule (vacuum, alignment stability, mechanics, instrumentation) including after thermal cycles
9 Provide excellent magnetic shielding for high Q0
10 Minimize cost (construction and operational)

Cryomodule requirements -- major interfaces

- Bayonet connections for helium supply and return
- Vacuum vessel support structure
- Beam tube connections at the cryomodule ends
- RF waveguide to input couplers
- Instrumentation connectors on the vacuum shell
- Alignment fiducials on the vacuum shell with reference to cavity positions.

Design considerations

- Cooling arrangement for integration into cryo system
- Pipe sizes for steady-state and emergency venting
- Pressure stability factors
  - Liquid volume, vapor volume, liquid-vapor surface area as buffers for pressure change
    - Evaporation or condensation rates with pressure change
- Updated heat load estimates
- Options for handling 4.5 K (or perhaps 5 K - 8 K) thermal intercept flow
- Alignment and support stability
- Thermal contraction and fixed points with closed ends
- Etc.

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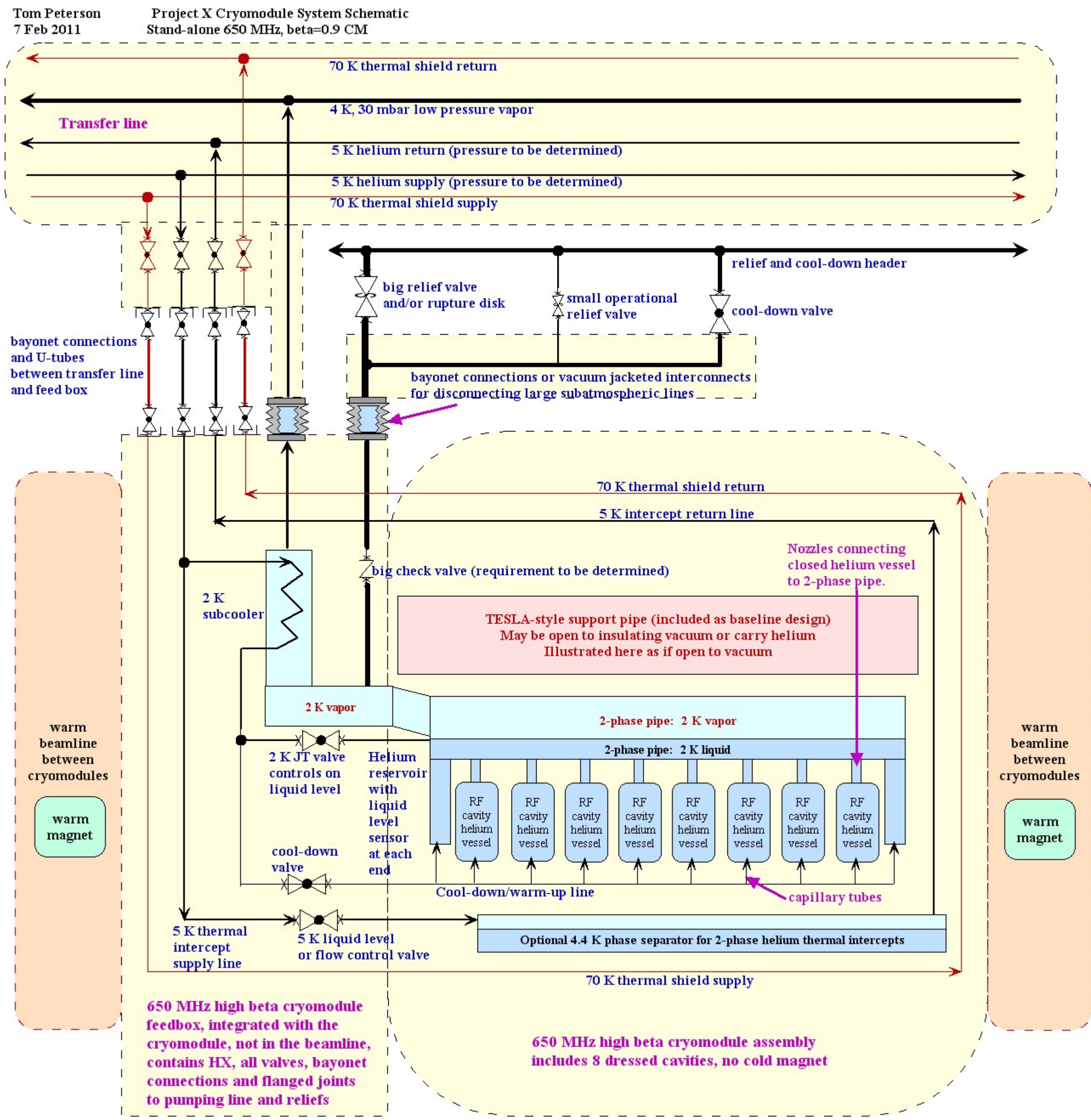
CW cryomodule style

- Very high heat flux (200 W per CM) and relatively short linac (not large quantity production nor several km long strings) ==>
  - Need separated liquid management
  - Prefer small heat exchangers, distributed with cryomodules
  - Prefer stand-alone cryomodules, warm magnets and instrumentation between cryomodules like at SNS
- Stand-alone CM ==>
  - "300 mm" pipe is unnecessary for helium flow
- Not need 300 mm pipe for helium flow ==>
  - Empty 300 mm pipe as support 'backbone" or
  - Different support structure (space frame or posts)

Stand-alone cryomodule schematic



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

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)*

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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.

An example from CGA S-1.3—2005 for evaluation of the discharge temperature and effective latent heat (or “pseudo latent heat”)

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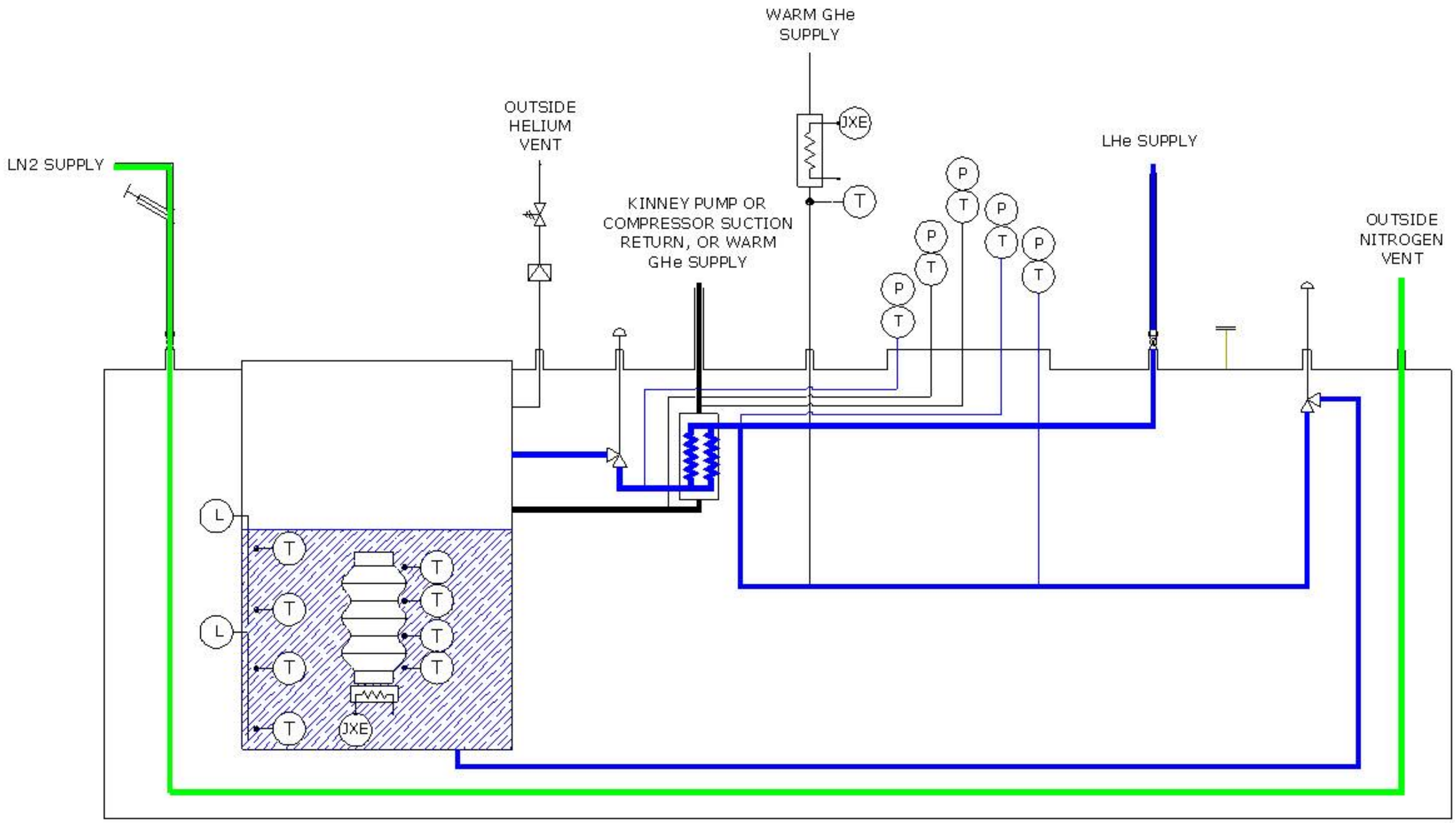
Example for flow just above the helium critical pressure				
P=2.4 atm	(data from NBS Technical Note 631)			
			$v\left(\frac{\partial h}{\partial v}\right)_P$	$\frac{\sqrt{v}}{v\left(\frac{\partial h}{\partial v}\right)_P}$
T (K)	v	sqrt (v)	v(dh/dv) const P	(sqrt (v))/(v(dh/dv))
	cc/gram		J/gram	
5.10	9.55	3.09	18.7	0.165
5.20	10.52	3.24	16.5	0.197
5.30	15.72	3.96	13.8	0.287
5.40	22.14	4.71	15.1	0.312
5.50	25.35	5.03	16.3	0.309
5.60	27.81	5.27	17.4	0.303
5.70	29.91	5.47	18.4	0.297
5.80	31.76	5.64	19.4	0.290
5.90	33.50	5.79	20.3	0.285
6.00	35.11	5.93	21.1	0.281

Maximum, so venting temperature is about 5.40 K and effective latent heat is 15.1 J/g

Example from an engineering note analysis for a superconducting RF cavity vertical test dewar

Vertical Test System (VTS)



4. Secondary relief sizing – air condensation

On possible air condensation event is loss of cavity vacuum to atmosphere during active pumping of cavities in VTS-2 & 3, resulting in condensation of air on the inside surfaces of the cavity and the pumping line. The maximum surface area is 23,276 cm^2, which includes the 17,500 cm^2 cavity surface area of two ILC-style 9-cell cavities [1] and the 5,776 cm^2 cavity pumping line surface area. The heat flux of air condensation on a bare cavity surface has been estimated to be 4 W/cm^2 by Cavallari et al. The resulting heat rate is 93.1 kW to the liquid helium bath.

Note comparison of loss of cavity vacuum with condensation on smaller area of bare metal to loss of insulating vacuum with smaller heat flux on larger area.

A second possible air condensation event is loss of cryostat insulating vacuum to atmosphere. The MLI-insulated helium vessel has a surface area of approximately 1.48×10^5 cm^2. At a heat flux 0.6 W/cm^2, the heat rate is 89.0 kW to the liquid helium bath.

Loss of cavity vacuum to atmosphere will have a larger heat rate and will be used in relief device sizing.

From an analysis like on slide 77 to obtain effective latent heat

At 65 psig = 80 psia, the 93.1 kW heat input from loss of cavity insulating vacuum will result in a 3,878 g/s flow rate. Equation 3 converts this helium mass flow rate to a standard volumetric flow rate of air.

$$Q_a = \frac{13.1 W C_a}{60 C}\sqrt{\frac{Z T M_a}{Z_a T_a M}}, \qquad (3)$$

From CGA S-1.3

where W is the helium mass flow rate (30,784 lbm/hr), C_a is a gas constant for air (356), C is the gas constant for helium (378), Z_a is the air compressibility (1), Z is the helium compressibility (0.60 at 7.3 K/80 psia), T_a is the air temperature (520 R), T is the helium temperature (13.1 R), M_a is the molecular weight of air (28.97), and M is the molecular weight of helium (4).

The calculated required secondary relief capacity for the air condensation condition is 2094 scfm air at 65 psig.

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- R. Byron Bird, Warren E. Stewart, Edwin N. Lightfoot, “Transport Phenomena,” John Wiley &Sons, 1960.
- S. W. VanSciver, “Helium Cryogenics,” Plenum Press, 1986.
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- Fermilab's ES&H Manual (FESHM) pressure vessel standard, FESHM 5031
  - http://esh-docdb.fnal.gov/cgi-bin/ShowDocument?docid=456
- FESHM 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
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- "Flow of Fluids through Valves, Fittings, and Pipes," Crane Technical Paper #410.
- R.H. Kropschot, et. al., "Technology of Liquid Helium," NBS Monograph 111
- Ascher H. Shapiro, "The Dynamics and Thermodynamics of Compressible Fluid Flow," Wiley, 1953.