Tutorial on

Cooling

by

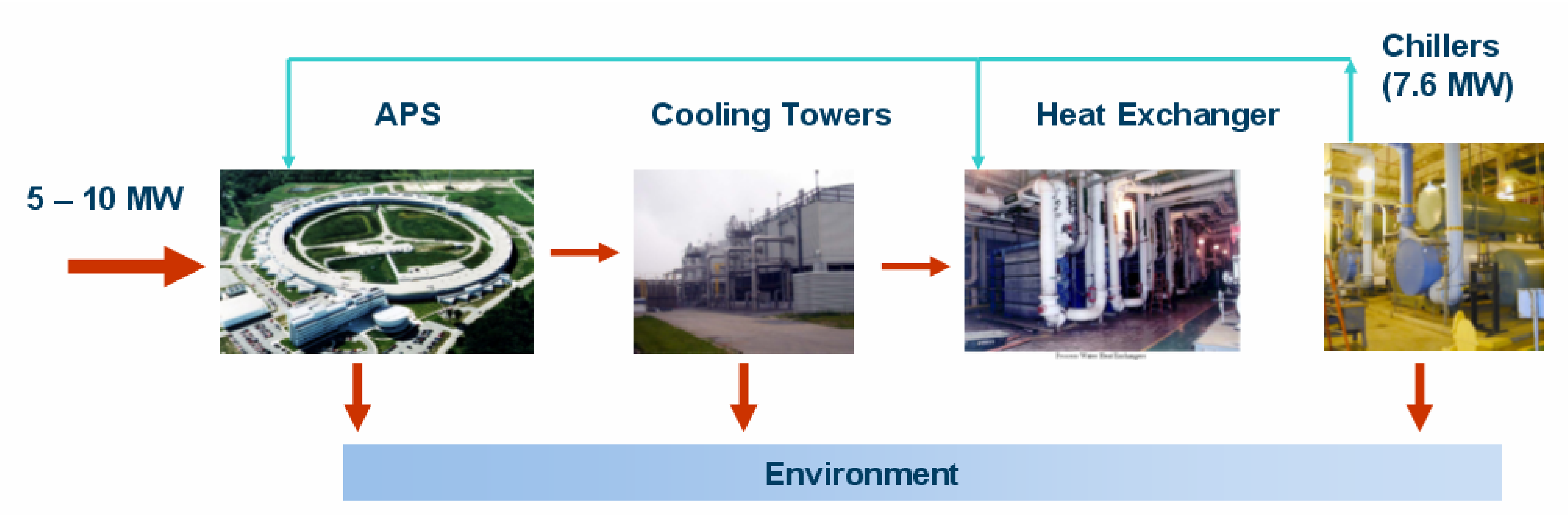
Sushil Sharma

Advanced Photon Source

Outline

- Utility System
- Heat Transfer
  - Conduction – Convection – Radiation
- Cryogenic Cooling
  - Liquid Nitrogen – Gaseous Helium
- Cooling Enhancement
- Nucleate Boiling and Critical Heat Flux
- Peltier Cooling and Heat Pipe
- Miscellaneous Design Considerations
- Summary

Cooling – Utility Engineer's Perspective



- 5-10 MW of power is input into the facility
- Most of the power is carried to the utility building by process (DI) water
- The power is rejected to the environment via cooling towers and/or chillers

Utility System - Requirements

Requirements at Component Level

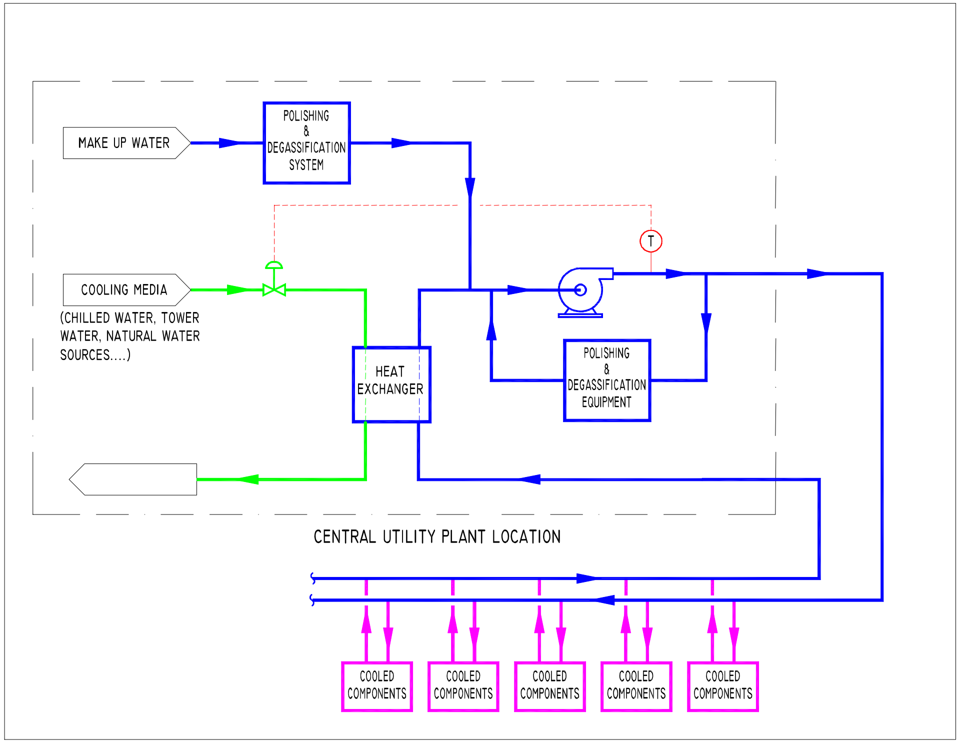
	APS Nominal Values
Base Temperature, ΔT	25.6 ˚C, < 4 °C
Temperature stability	± 0.1 °C (originally ± 1°C)
ΔP (components)	100 psi (0.69 MPa)

Requirements at System Level

Total heat rejection	5 -10 MW
Temperature stability	± 1.0 °C
Resistivity	> 6 MΩ-cm
Dissolved oxygen	< 10 ppb
Separation of Cu and Al water systems	

System's total flow, base temperature, ΔT and ΔP depend on the type of system:
(1) Primary Distribution, or (2) Primary-Secondary Distribution

Utility System - Central Distribution System



Advantages

Single centralized plant

Low installed cost

Low maintenance cost

Limitations

Water distributed at one temperature

Water distributed at one pressure

Energy inefficient

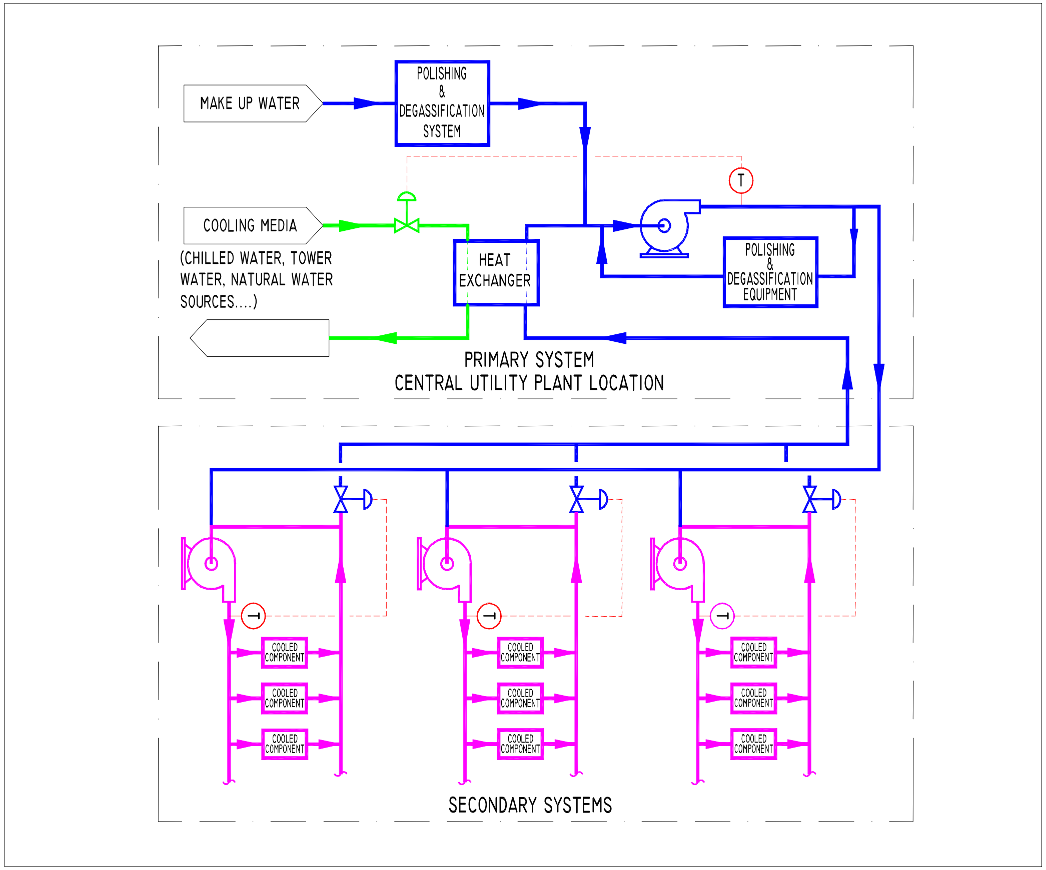
Limited flexibility

$$\text{Flow}\,(\text{L}/\text{min}) = \frac{14.40\,.\text{Heat(kw)}}{\Delta T(^{0}C)}$$

$$\text{Flow}\,(\text{GPM}) = \frac{3.805\,.\text{Heat(kw)}}{\Delta T(^{0}C)}$$

Total Flow → Pipe Size (based on admissible velocity) → System ΔP → Pump Power

Utility System - Primary-Secondary Distribution System



Advantages

Flexible

Local temperature control

Local pressure control

Energy efficient

Limitations

High first installed cost

More complex than central system

Secondary systems are not chemically independent

Higher maintenance cost than a central system

APS Primary System

Flow = 10,000 GPM

P_S = 33 psig

P_R = 20 psig

T_S = 72 deg F

T_R = 79 deg F

Utility System – APS Water System

Chillers (7.6 MW)

Heat Exchanger

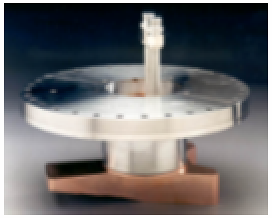
Primary Pumps (200 HP)

Degassifier

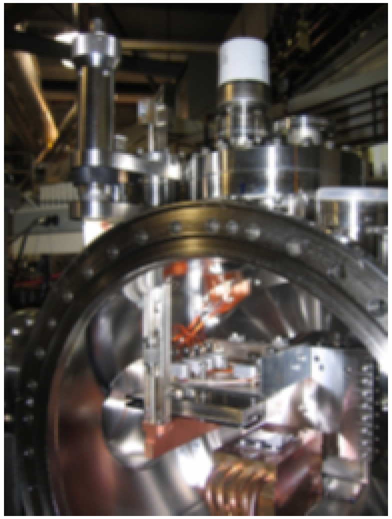
Secondary Pumps (75 HP)

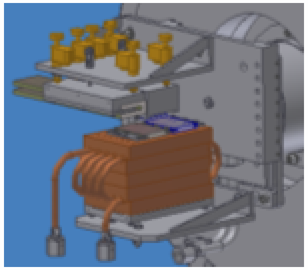
Secondary Headers in SR

Heat Transfer

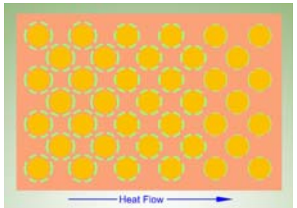


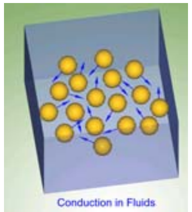
APS SR Crotch Absorber

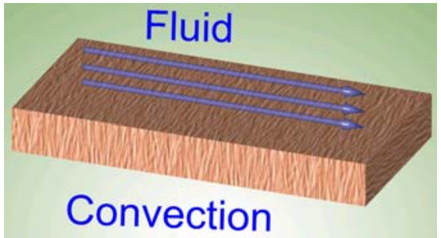


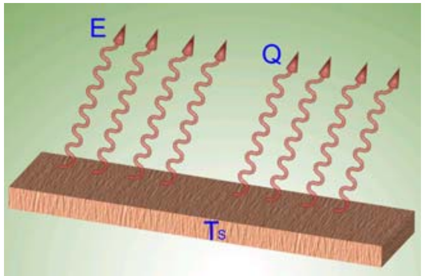


NSLS He-Cooled Si Monochromator









Conduction: Transfer of thermal energy due to lattice vibration (solids) or random molecular motion (fluids)

Convection: Transfer of thermal energy by motion of a fluid

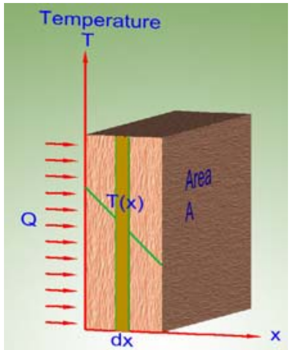
Radiation: Transfer of thermal energy by electromagnetic radiation

Heat Transfer - Conduction

Fourier Law of Conduction:

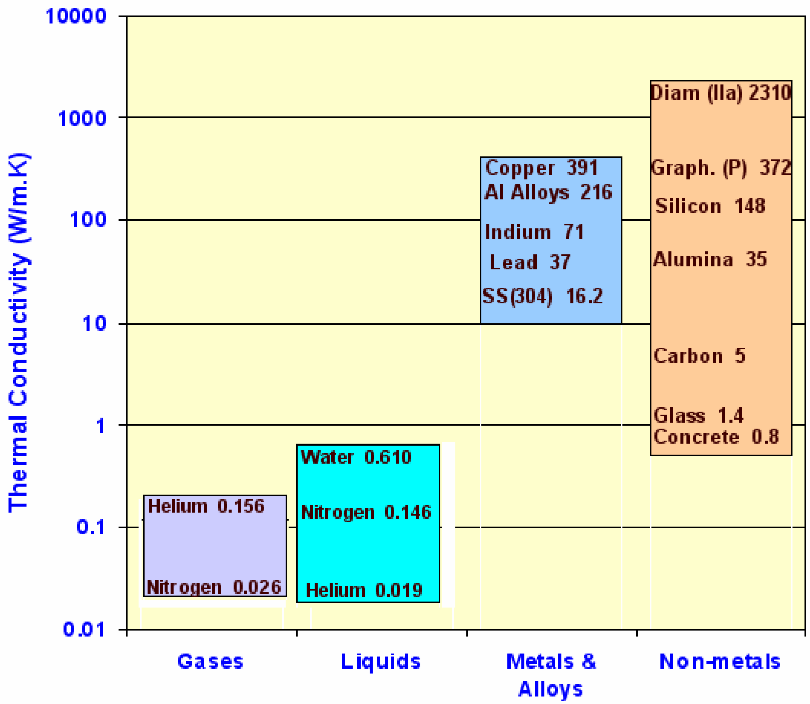
$$Q = -k\,A\,\frac{dT}{dx}$$

Q = rate of heat, A = surface area
dT/dx = temperature gradient



General equation

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+q=\rho c\frac{\partial T}{\partial t}$$



(Values at STP except for liquids N2 and He)

Material	k (W/m.K)
Diamond (Type IIa)	2310
Silver	419
Copper (oxygen free)	391
Glidcop (AL-15)	365
Gold	298
Beryllium Copper	260
Aluminum Alloys (6000 series)	216
Tungsten	167
Silicon (single crystal)	140
Indium	70.1
Silicon Carbide (CVD)	67
Carbon steel (ASTM A36)	46.7
Titanium	22
Stainless steel (304)	16.2
Inconel (625)	9.8
Zerodur	1.6
Concrete	0.8

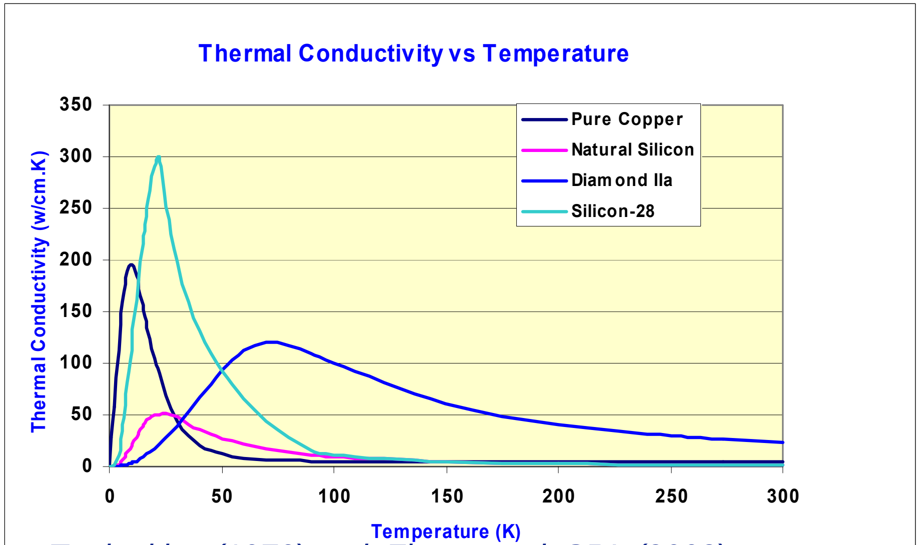
Heat Transfer- Conduction – Design Options

Fourier Law of Conduction

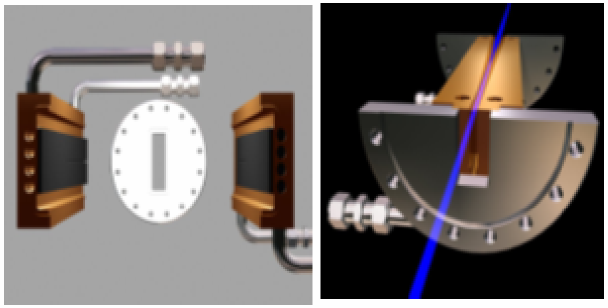
$$Q = -k\,A\,\frac{dT}{dx}$$

To handle high power densities (> 100 kw/mrad^2), decrease dT/dx by

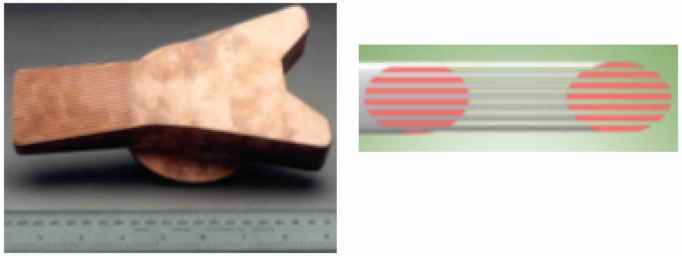
- **Using materials with high k values**
  → copper, Glidcop, cryogenic cooling
- **Increasing surface area**
  → inclined surfaces, external fins
- **Decreasing Q**
  → slits, pinholes, filters,



Touloukian (1970) and Zhang et al. SRI (2003)



Inclined Surface of an APS Mask



External Fins of APS Crotch Absorber

A Pinhole Absorber

Heat Transfer - Convection - Equation

Newton's Law of Cooling

$$Q = h\ A\ (T_w - T_f)$$

Q = Heat transfer rate (W)
h = Convection film coefficient (W/m^2.K)
A = Surface area (m^2)

Film coefficient, h, depends on

- Fluid density, velocity, viscosity
- Fluid specific heat, conductivity
- Fluid expansion coefficient
- Channel geometry
- Wall temperature

Nomenclature:

D	Hydraulic diameter
V	Fluid velocity (m/s)
ρ	Fluid density (m^3/kg)
c_p	Fluid specific heat (J/kg)
μ	Fluid viscosity (Pa.s)
k	Fluid thermal conductivity (W/m.K)
T_f	Fluid (bulk) temperature (K)
T_w	Wall temperature (K)
R_e	Reynolds number
P_r	Prandtl number
N_u	Nusselt number

These quantities define terms as natural and forced convection, laminar or turbulent flow, single phase or two phase (boiling) convection.

Heat Transfer - Convection – Dimensionless Numbers

Reynolds Number

$$R_e = \frac{\rho . V . D}{\mu} = \frac{G . D}{\mu}$$

G is mass flow rate (kg/s)

Prandtl Number

$$P_r = \frac{C_p . \mu}{k}$$

Nusselt Number

$$N_u = \frac{h\, D}{k}$$

Physical properties are evaluated at T_f for fluids but $(T_f + T_w)/2$ for gases

V = Average fluid velocity for internal flow, or far field velocity for external flow

D = Hydraulic diameter or characteristics length for flat plates

D = diameter of a circular channel

= 4A / P_w for an arbitrary channel, where A is the cross-sectional area and P_w is the wetted perimeter

Web Resources

http://webbook.nist.gov/chemistry/fluid/   *http://www.processassociates.com/process/dimen/dn_all.htm*

Heat Transfer- Convection – Film Coefficient

Forced Turbulent Flow

Dittus-Boelter	$N_u = 0.023\, R_e^{0.8}\, P_r^{0.4}$	$2500 < R_e < 124{,}000$ $0.7 < P_r < 120$
Sieder -Tate	$N_u = 0.027\, R_e^{0.8}\, P_r^{1/3}\, (\mu/\mu_w)^{0.14}$	$R_e > 10{,}000$ $0.5 < P_r < 1E06$

μ_w is fluid viscosity at wall temperature in Sieder-Tate equation

Forced Laminar Flow (Sieder – Tate)

$$N_u = 1.86\left(R_e P_r \frac{D}{L}\right)^{1/3} (\mu/\mu_w)^{0.14}$$

$R_e P_r \frac{D}{L} > 10$

$0.014 < \mu/\mu_w < 14$

Petukhov Equation

$$N_u = \frac{\left(\frac{f}{8}\right) R_e P_r}{1.07 + 12.7\left(\frac{f}{8}\right)^{1/2} (P_r^{2/3} - 1)} (\mu/\mu_w)^n$$

$$n = \begin{cases} 0.11 & \text{for } T_w > T_f \\ 0.25 & \text{for } T_w < T_f \\ 0 & \text{for constant heat flux} \end{cases}$$

f = friction factor

$$= (1.82 \log_{10} R_e - 1.64)^{-2}$$

http://users.wpi.edu/~chslt/courses/es3003/lect22_6.pdf

Heat Transfer - Convection – Film Coefficient

Flat Plate, Forced Convection:

http://lyre.mit.edu/3.185/2001/handout-nusselt.doc

$$N_u = (1/\sqrt{\pi}) R_e^{1/2} P_r^{1/2}$$

Laminar, Low P_r

$$N_u = 0.026 R_e^{0.8} P_r^{1/2}$$

Laminar, High P_r

$$N_u = 0.037 R_e^{0.8} P_r^{1/2}$$

Turbulent, High P_r

Flat Plate, Free Convection:

Goldstein et al, J. Heat and Mass Transfer, 16(1973)
http://lyre.mit.edu/3.185/2001/handout-nusselt.doc
http://www.cheresources.com/convection.shtml

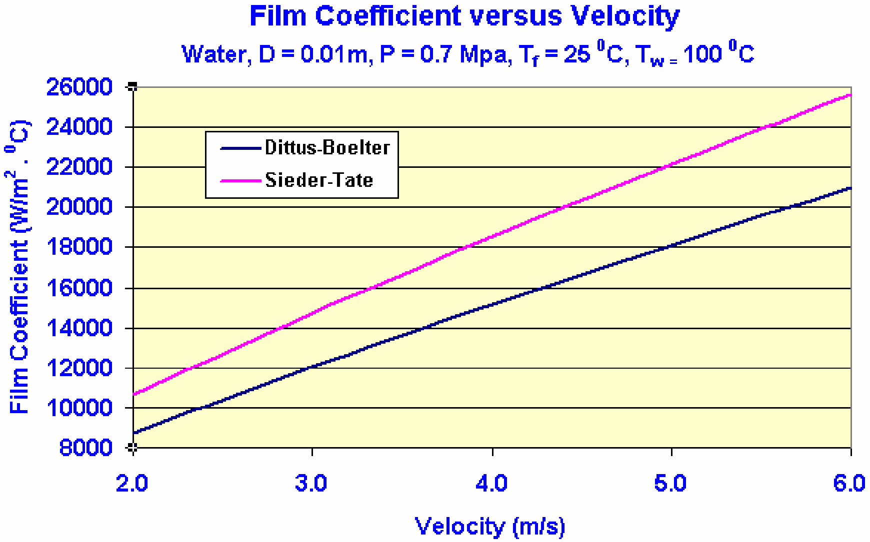
Grashof Number

$$G_r \equiv \frac{g\beta(T_w - T_f)L^3}{\mu^2}$$

APS Exit Port – Al Finned Tube

Typical values of convection film coefficient:

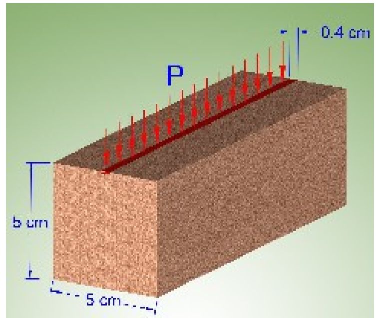
Type of convection	Medium	Film coefficient ($W/m^2.K$)
Free	gases	2 - 25
Forced	gases	25 - 250
Free	liquids	10 - 1000
Forced	liquids	50 – 20,000
Boiling	liquids	2,500 – 100,000

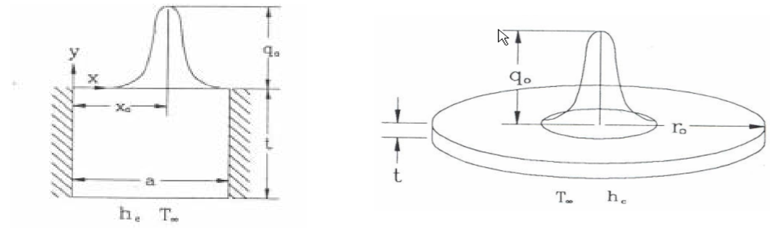


Heat Transfer - Convection – Cooling Efficiency

Example – Long Copper Block

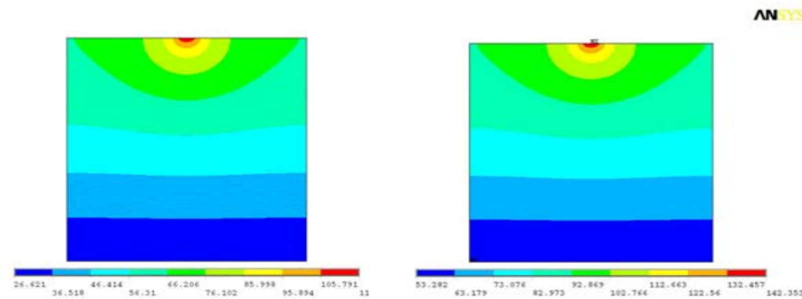
$k = 395$ W/m.K
$P = 500$ W/cm^2





Closed-Form Solutions
Sheng et al., SPIE, Vol. 1997 (1993)

T = temperature above ambient
ΔT_{mat} = temperature difference within the material
Deformations (Thermal bump and bending) = $f\,(\Delta T_{mat}, \alpha)$
Thermal stresses = $f\,(T, \alpha, E)$



h = 1500 W/m^2.K   h =750 W/m^2.K

1. Effect of Cooling efficiency

(Assumption: α, k are constants)

- No significant effect on thermal deformations.
- Thermal stresses are increased considerably

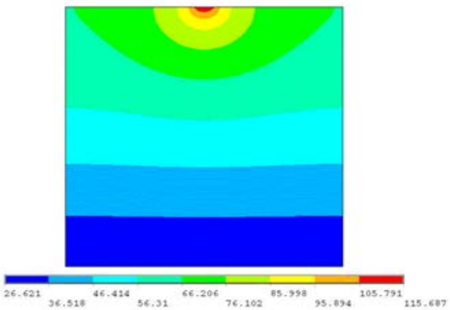
Temp. (^{0}C)	h = 1500 (W/m^2.K)	h =750 (W/m^2.K)	Δ (K)
T_{max}	115.69	142.35	26.66
T_{side}	51.31	77.98	26.67
T_{min}	26.62	26.62	26.66

→ No significant change in ΔT_{mat}

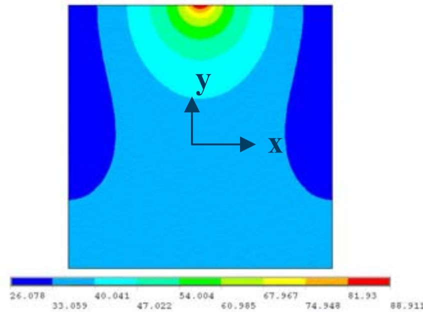
(1) When α, k are temperature-dependent (e.g. cryogenic cooling of Si), cooling efficiency can lead to material temperature being in an undesirable range, (2) Film boiling can be a factor at higher temperatures

Heat Transfer- Convection – Cooling Geometry

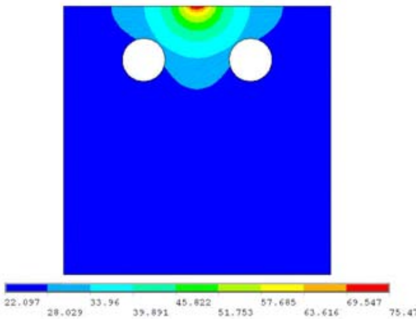
$h = 1500\ W/m^2K$



Bottom-Cooled



Side-Cooled



Channel-Cooled

Cooling Geometry	T_{max} (^{0}C)	ΔT_{mat} (^{0}C)	Thermal Moment (^{0}C.cm^3)
Bottom-Cooled	**115.7**	**89.1**	**527.2**
Side-Cooled	**88.9**	**62.8**	**99.1**
Channel-Cooled	**75.5**	**53.4**	**34.0**

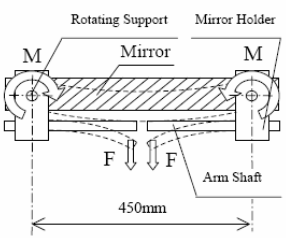
Thermal moment

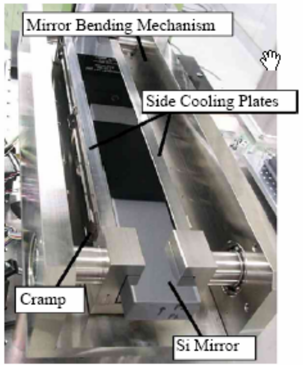
$$= \iint T(x,y)\, y\, dxdy$$

Effect of Cooling Geometry:

(Assumption: α and k are constants)

- **Considerable effect on thermal bump and thermal stresses.**
- **Large effect on bending deformations**





Kamachi et al., MEDSI 2002

Compensation for Bending Deformations

Heat Transfer- Radiation - Equation

Radiative heat transfer can occur in vacuum.

$$Q = \sigma \varepsilon A \left(T^4 - T_0^4\right)$$

where

Q = Net energy radiated (W)

σ = Stefan-Boltzmann constant = 5.67×10^{-8} W/m^2

ε = Constant, emissivity of the object

A = Surface area (m^2)

T = Absoulte temperature of the object (K)

T_0 = Absolute temperature of environment (K)

Typical Emissivity Values

Material	Clean	Oxidized
Aluminum	0.02	0.1
Copper	0.03	0.8
Gold	0.02	
Iron and steel	0.1	0.85
Nickel	0.5	
Silver	0.02	0.1
Ceramics	0.65	
Graphite	0.8	

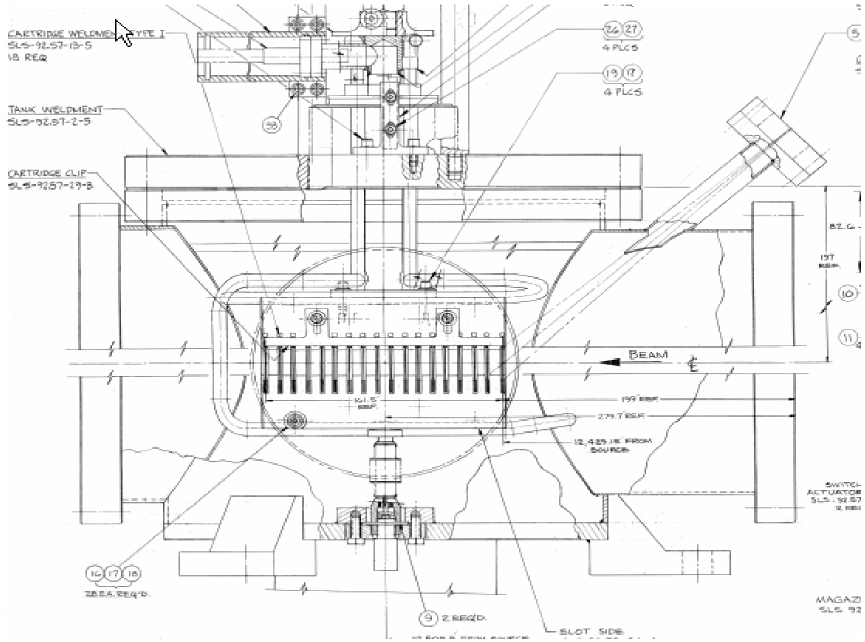
Reference: Inframetric ThermaCam Manual

Heat Transfer - Radiation – Example

Example: A 5 cm cube in vacuum => surface area = 0.0150 m^2 , assume ε = 0.5, T_0 = 300 K

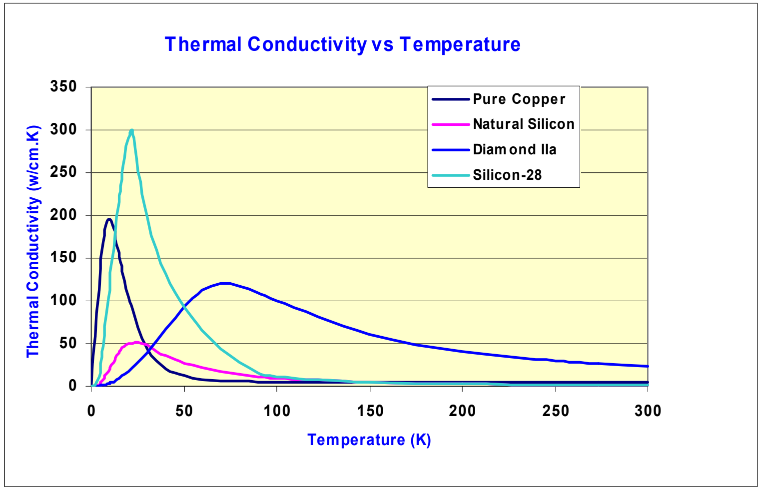
T (K)	77	200	400	600	800	1000
Q (W)	- 3.4	- 2.8	7.4	51.6	170.7	421.8

- NSLS X-17 graphite filter assembly uses radiative heat transfer.
- Radiated power from mirrors and monochromators can cause thermal drift of the drive mechanisms. Heat shields are often used to reduce thermal drifts.



NSLS X-17 Graphite Filter Assembly
Courtesy – S. Pjerov (NSLS)

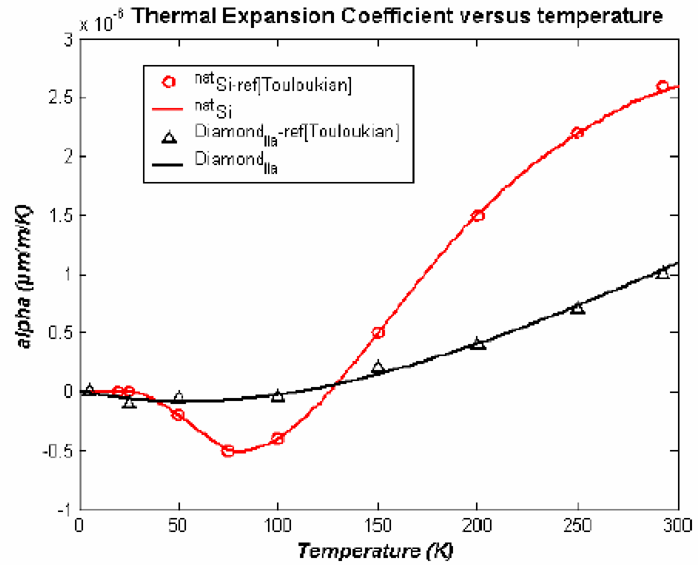
Cryogenic Cooling – Figure of Merit

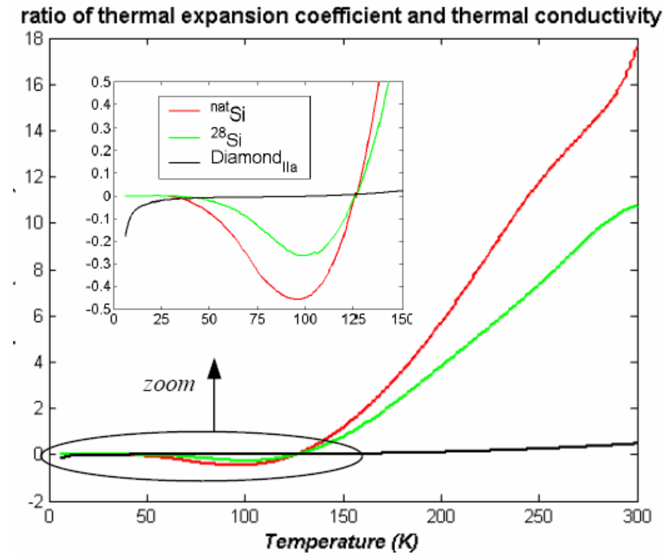


Reference: Touloukian (1970) and Zhang et al (2003)

- Thermal conductivity of silicon and diamond is one order of magnitude higher at cryogenic temperatures.
- Thermal coefficient of expansion, α, is one order of magnitude lower.
- Figure-of-merit, α/k, is superior at cryogenic temperatures.

- α for silicon has a cross-over point at 125 K.
- Diamond has good figure of merit even at room temperature.





Zhang et al. SRI (2003)

Cryogenic Cooling – LN2

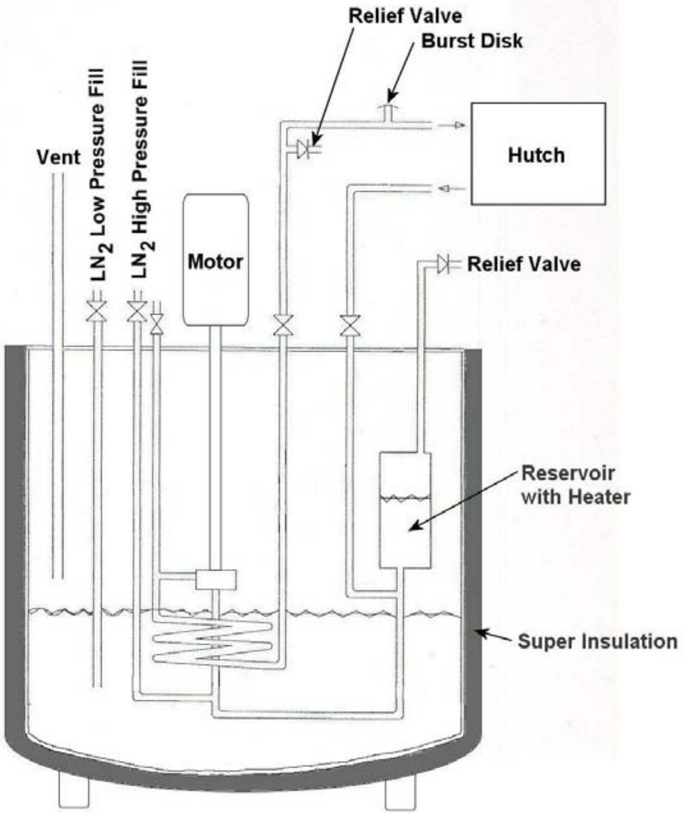
Two Options are in general use for cryogenic cooling under high heat load: (1) LN2, (2) gaseous helium

Options under consideration: gaseous N2, liquid helium, and Peltier

LN2 Cooling

- Reasonable temperature range for cooling Si
- Reasonable film coefficient for heat transfer
- Inexpensive
- Widely used

Limitation: Installation and maintenance cost of distribution system and low boiling heat flux



Schematic of an LN_2 Cryocooler

Properties of Liquid Nitrogen

T (K)	Pressure (MPa)	Density (kg/m3)	Cp (J/kg*K)	Viscosity (Pa*s)	k (W/m*K)	Prandtl Number
77	0.1	807.7	2039.8	0.000163	0.14657	*2.2678*
77	0.7	809.2	2033.3	0.000165	0.14723	*2.2732*

Source: http://webbook.nist.gov/chemistry/fluid/

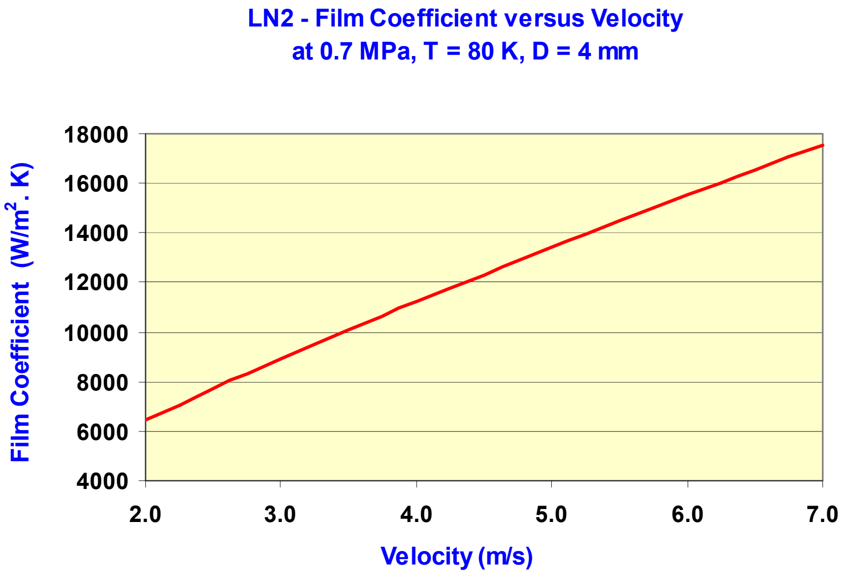
Saturation Temperature (K)

P (MPa)	Water	LN2
0.1	372.76	77.24
0.7	438.10	98.49

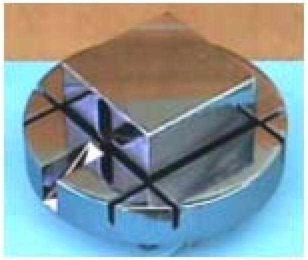
Cryogenic Cooling – LN2- Direct Cooling

Pictures below, courtesy of A.Macrander

Direct Cooling:



C.S. Rogers, et al SPIE, 2885 (1996)

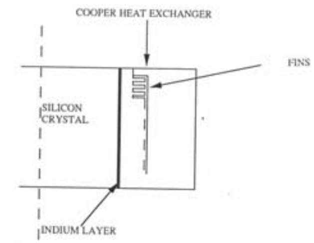



I. Ivanov et al. SRI (2000)
AIP Conf. Proc., Vol. CP521

Using fins or pins the effective film coefficient may be increased by a factor of 2 or 3.

Cryogenic Cooling – LN2 – Indirect Cooling

Indirect Cooling

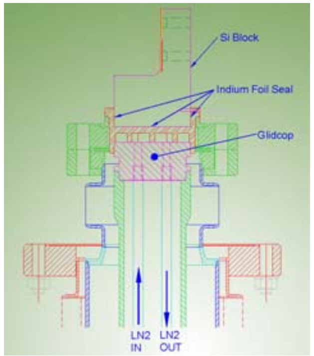


Marot et al. SPIE, Vol. 1739, (1992)

Effective film coefficient, h_{cv} (*Zhang et al, JSR, 2003*):

$$h_{cv} = \left[\frac{1}{h_{cv0}\, f_a} + R_c + R(Cu) + R(In) \right]$$

h_{cv0} = film coefficient in the cooling channel,
f_a = channel surface area / contact area
R_c, R(Cu) and R(In) = thermal conduction resistances of Si-Cu-In interface, Cu block and indium foil, respectively

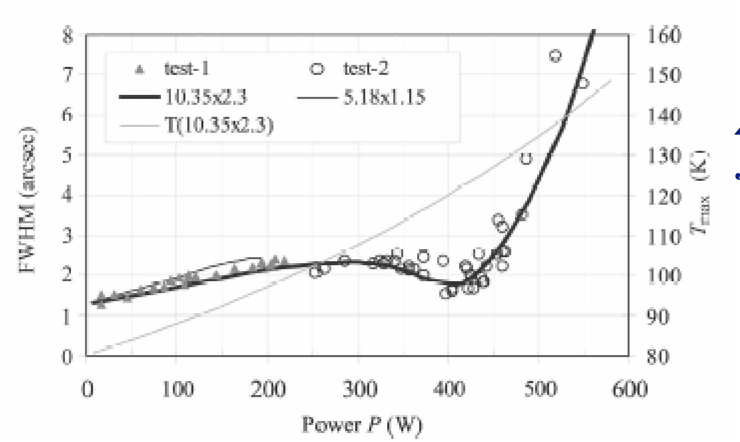


APS Sector-35 Monochromator Subassembly

H_{cv} was estimated to be in 3000 – 8000 W/m^{2}K range.

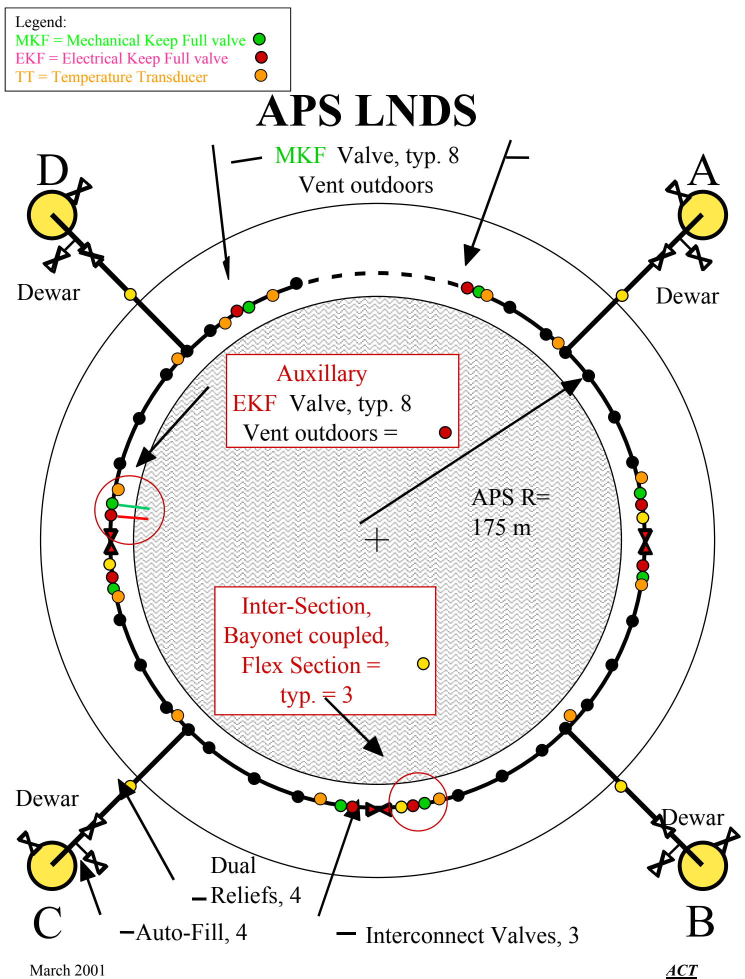
Main Results:

- Ideal performance up to a heat load of 400 W (peak power of 210 W/mm^{2}) , and an acceptable performance at 900 W.
- Better performance than predictions.
  (Chumakaov et al. JSR 2004)
- Deviation from linear response at ~ 425W.
  (Zhang et al. JSR 2003)



Zhang et al., JSR 2003

Cryogenic-Cooling - APS Liquid Nitrogen Distribution System



Courtesy – W. Wesolowski (APS)

3000 Gallons Dewar

Local PLC Panel

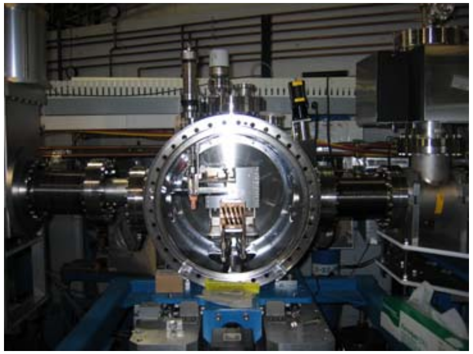
LN2 Cryocooler

EPICS Screen

Nominal primary system pressure = 40 psi
Nominal usage rate ~ 2,500 gallons/day

Cryogenic cooling – Helium Gas

NSLS X-25 Helium-Cooled Monochromator



Cooling by Helium Gas

Advantages:

- Allows cryogenic cooling of Si ~ 100 K
- Cryogen remains in single (gaseous) phase
- Closed-loop systems are commercially available
- Eliminates the need of continuous supply of cryogen

Successful operation has been demonstrated at ESRF and NSLS

Limitations:

- High flow rates are necessary to remove heat of a couple of hundred watts.
- Vibration isolation may be necessary.

Properties of Helium Gas

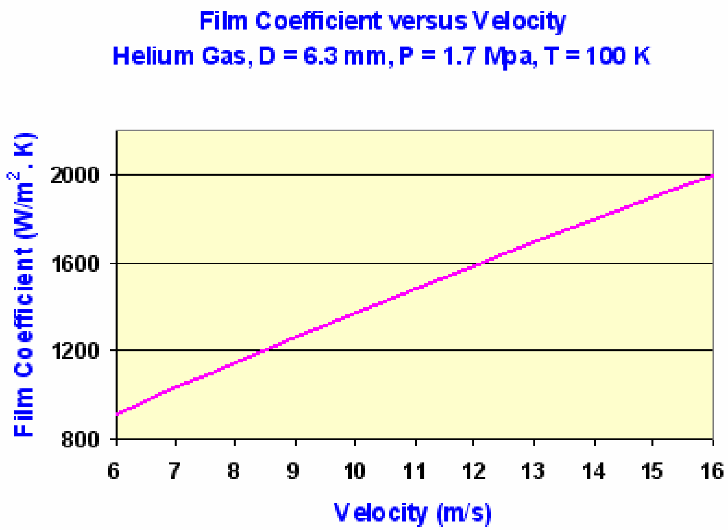
T (K)	Pressure (MPa)	Density (kg/m3)	Cp (J/g*K)	Viscosity (Pa*s)	Thermal Cond. (W/m*K)	Prandtl Number
40	0.1	1.2004	5.2061	5.54E-06	0.040444	0.713
40	1.7	19.566	5.3935	5.89E-06	0.043575	0.729
100	0.1	0.48071	5.1943	9.78E-06	0.073713	0.689
100	1.7	7.9857	5.2146	9.98E-06	0.075393	0.690

Helium gas has comparatively high C_p (on mass basis) and k, but low ρ .

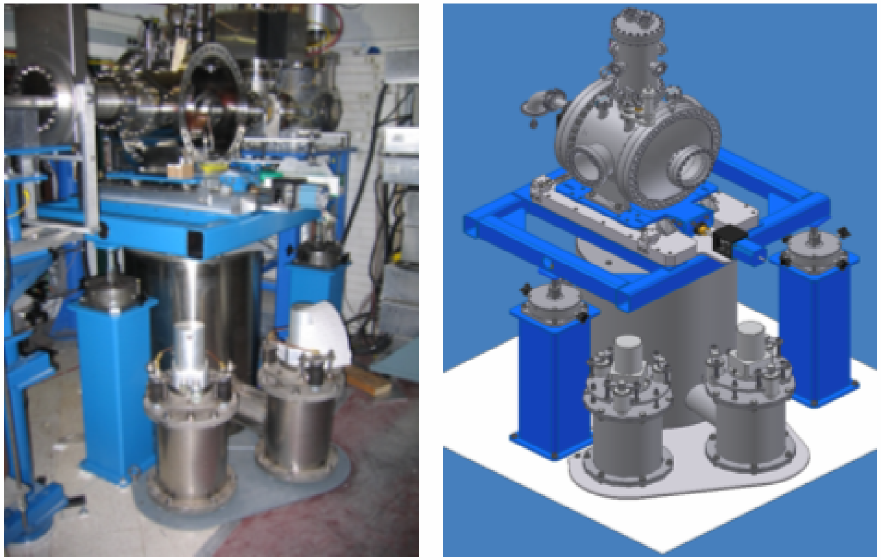
Reference: http://webbook.nist.gov/chemistry/fluid/

Cryogenic cooling – Helium Gas – NSLS X13, X21 Monochromators

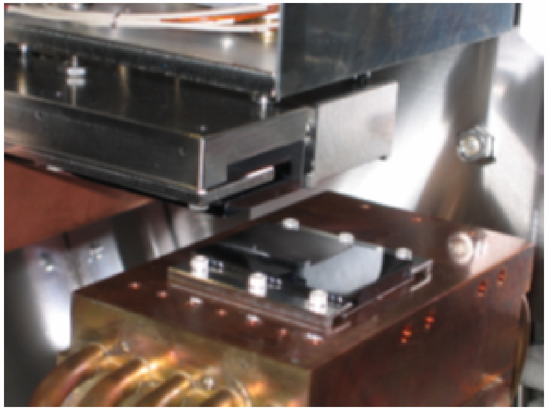
Courtesy – L. Berman (NSLS)



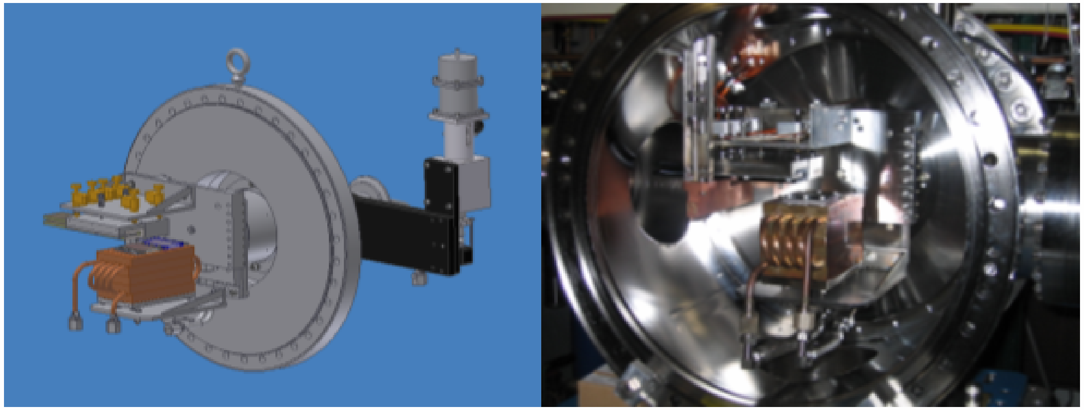
Nominal gas flow rate (for X-13) = 470 cm^3/s



NSLS X-21 Helium Cooling System



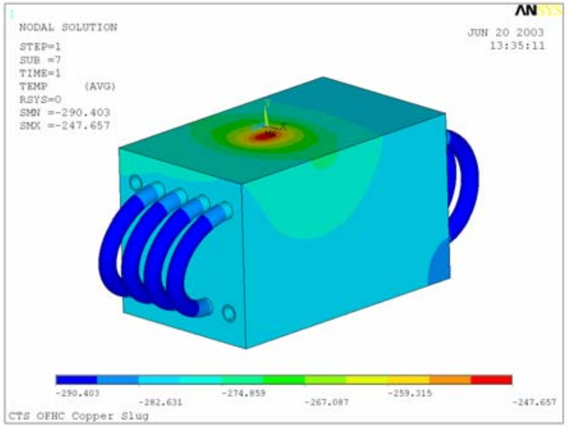
X-21 Si Crystal, and Cu Cooling Block



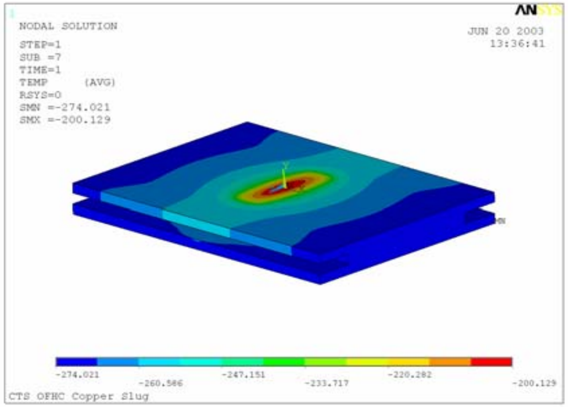
X-21 Monochromator Subassembly

Cryogenic cooling – Helium Gas – NSLS X13, X21 Monochromators

Courtesy –
P. Montanez (NSLS)



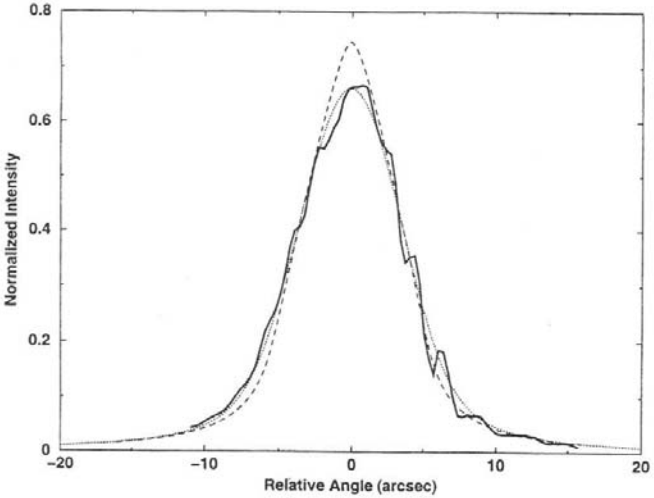
Temperature Contours (F) – X21 Cu Cooling Block



Temperature Contours (F) – Si Crystal

- Almost Ideal performance was obtained at low power of X-13 beamline: total power = 65 W, peak power density = 2 W/mm^2
- Upgraded helium cooler in X-21can handle up to 400 W of total power allowing a peak power density of > 10 W/mm^2

Web reference: http://www.cryomech.com/



X-13 Rocking Curve – Si (111)
Berman et al., RSI, 73(2002)

Cooling Enhancement

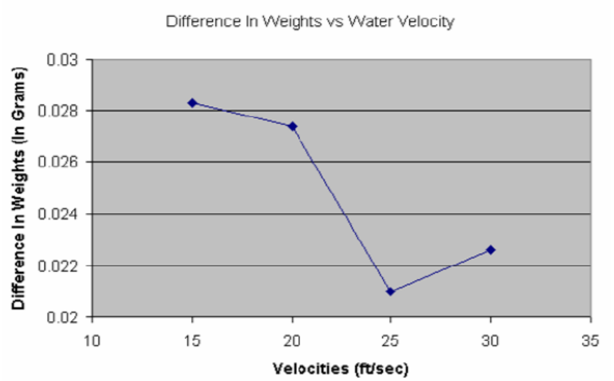
The goals of enhanced cooling are to:

- decrease maximum temperature rise of high-heat-load components → *increase fatigue life*
- for cryogenic cooling, optimize k, α → *decrease thermal deformations*
- decrease channel wall temperature, T_w→ *prevent film boiling*

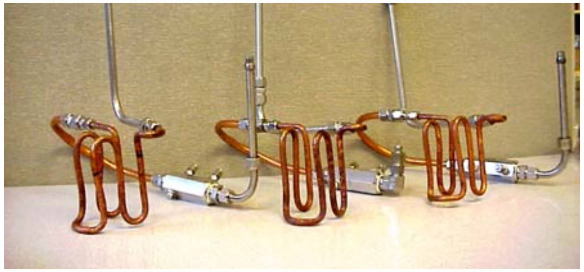
Techniques

- Increase fluid velocity → *higher Reynolds number*
- Channel inserts → *generate turbulence*
- Internal fins → *increase surface area for heat transfer*

Concerns about erosion above 3 m/s flow velocity of DI water have led to the use of channel inserts at APS. Recent (ongoing) erosion tests have shown little erosion even at much higher velocities.



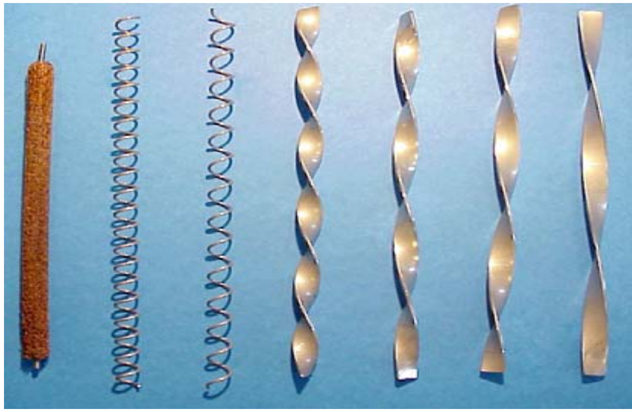
Erosion in Glidcop Tubes – Vemuri, MS Thesis, IIT, 2003

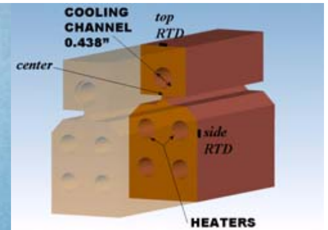


Flow	Flow	Straight	Bent
gpm	Ft/sec	Thickness (in)	Thickness (in)
1.85	12.1	0.063	0.058
3.43	22.4	0.063	0.057
4.89	32.0	0.063	0.057

Erosion in OFHC 3/8” copper tube after 4 years. Sharma et al., MEDSI 2002

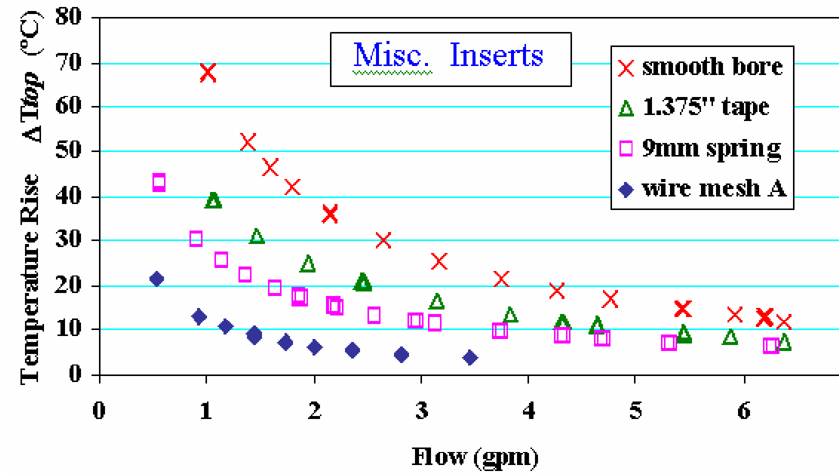
Cooling Enhancement – Channel Inserts

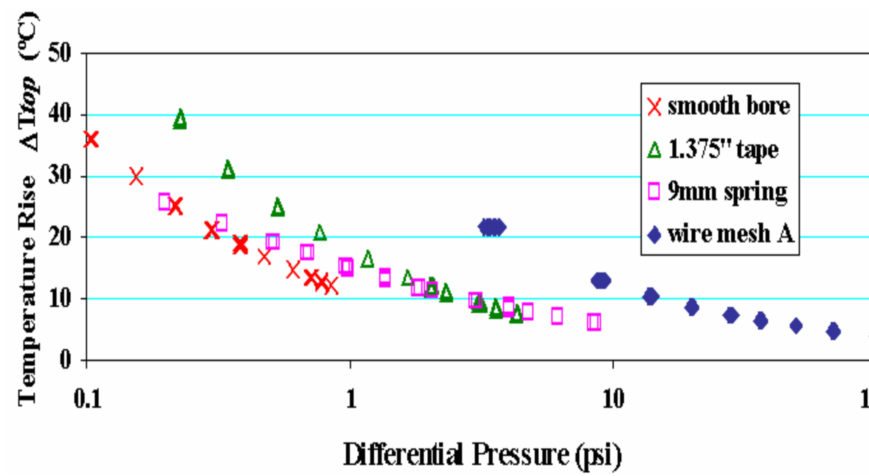




0.438” 9mm 16mm 1.375” 2.0” 2.5” 3.0”

Channel Inserts





Sharma et al. MEDSI 2002

See also, Collins et al., MEDSI 2002

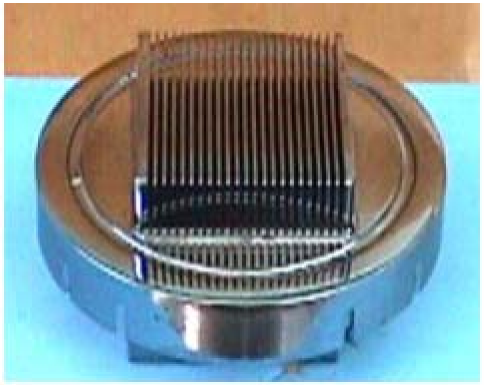
Results:

- Because of diminishing return, there was no significant advantage in increasing h_f above 1500 W/m^2.K (equivalent to 6 GPM in the smooth-bore channel).
- For the same ΔT, smooth bore channel yielded the lowest ΔP.
- With low pressure drops, components can be connected in series, leading to substantial reduction in the cost of flow instrumentation.

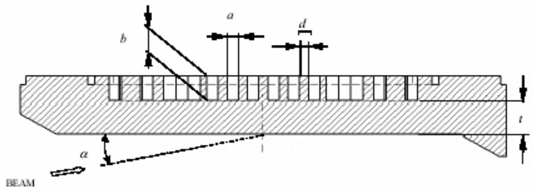
Cooling Enhancement – Internal Fins

APS Beam Dump

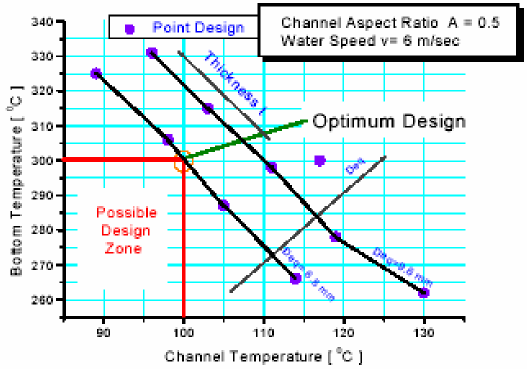
APS Crotch Absorber

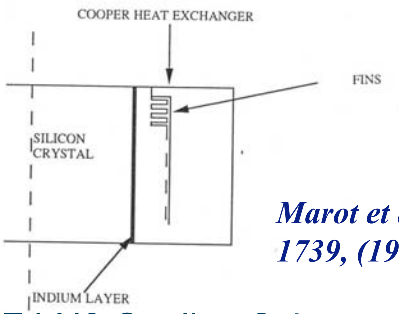


Si Crystal



ELETTRA Wiggler Shutter
Gambitta et al. MEDSI 2000





Marot et al. SPIE, Vol. 1739, (1992)

ESRF LN2 Cooling Scheme

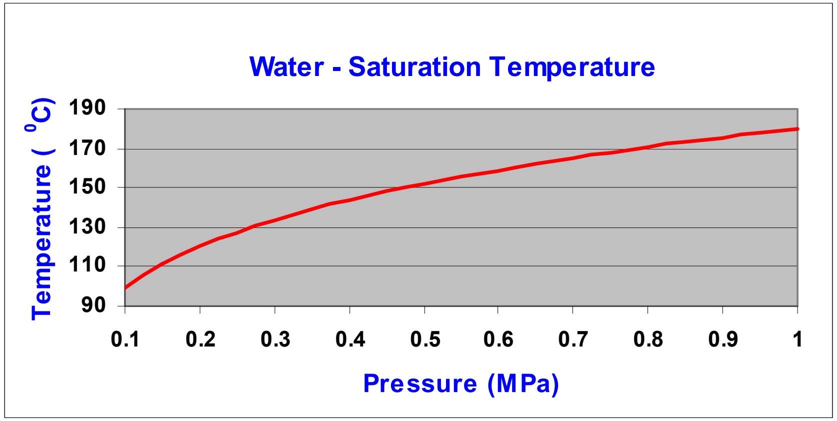
Fins increase cooling efficiency by increasing surface area for heat transfer.

Microchannels (*Tuckerman, UCRL-53515*) can increase the effective film coefficient by an order of magnitude. Limitations: (1) Require large ΔP → impractical for absorbers, masks and shutters
→ film boiling can be an issue for LN2 cooling
(2) Channels can clog in DI water-cooled components.

Nucleate Boiling and Critical Heat Flux

A common practice is to avoid nucleate boiling for the following reasons:

1. **To avoid flow-induced vibration**
2. **To provide margin-of-safety against burnout**

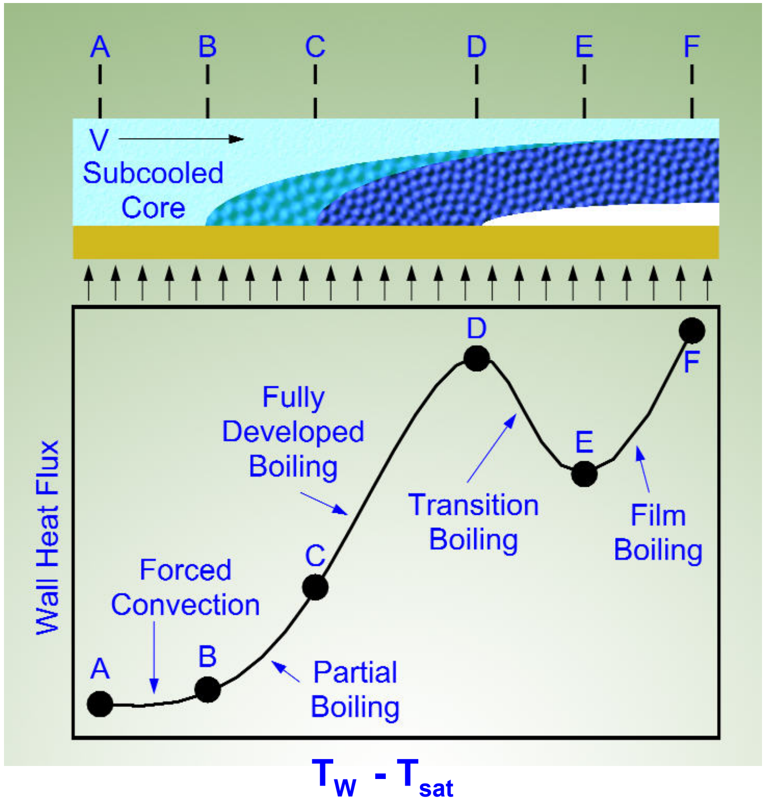


The maximum wall temperature is often required to be below the saturation temperature of the fluid at the given pressure.

- **Stress-sensitive components → too conservative**
- **Deformation-sensitive components**

  Water-cooled → no impact

  LN2-cooled → conservative



Nukiyama Curve - JSME (1934)

Nucleate Boiling (Point B) $T_w > T_{sat}$

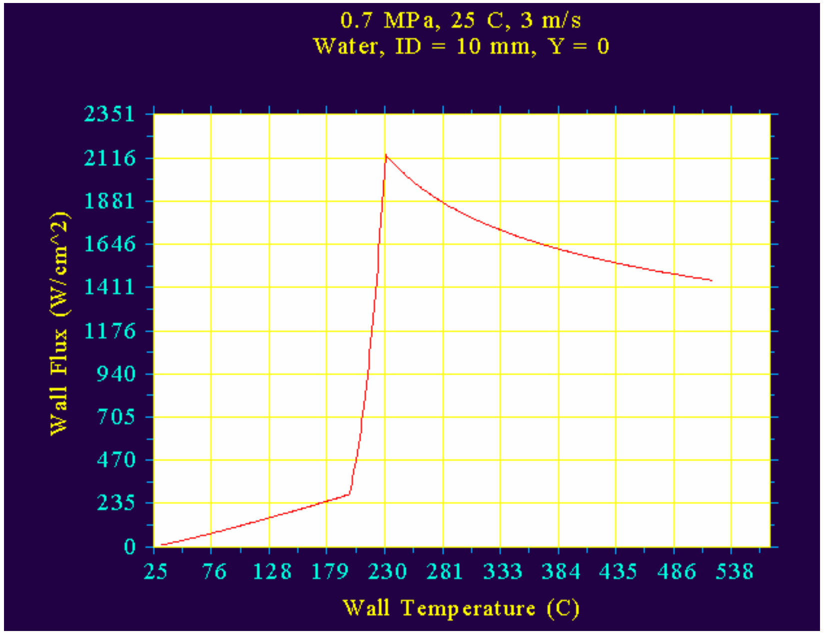
Nucleate Boiling and Critical Heat Flux

Φ_{bi}	= Incipient boiling heat flux (MW/m^2)
Φ_{CHF}	= Critical heat flux (MW/m^2)

P	= Pressure, Fluid (MPa)
ρ_L	= Density, fluid (kg/m^3)
ρ_G	= Density, vapor (kg/m^3)
σ_L	= Surface tension (N/m)
H_{fg}	= Latent heat of vaporization (J/kg)
T_{sat}	= Saturation temperature (K)
g	= Gravity acceleration (m/s^2)
g_c	= Conversion factor = 1 $kg.m/N.s^2$

$$J_a = \text{Jacob number} = \frac{C_p(T_W - T_{sat})}{H_{fg}}$$

Water



Reference: http://film2000.free.fr/TOFE.pdf

Bergles-Rohsenow (1964)

$$\Phi_{bi} = 15496\, P^{1.156}\left[1.799\left(T_w - T_{sat}\right)\right]^{\frac{2.0465}{P^{0.0234}}}$$

Tong (1975)

$$\Phi_{CHF} = 0.023\, f_0\, V \rho_L H_{fg}\left[1 + 0.00216\, P_{ratio}^{1.8}\, R_e^{0.5}\, J_a\right]$$

$$P_{ratio} = P/P_{crit}\ ;\ P_{crit} = \text{Critical pressure of water} = 22.809\ \text{MPa}$$

$$f_0 = 8.0\, R_e^{-0.6}\left(D/D_0\right)^{0.32}\ ;\ D_0 = \text{reference inner dia.} = 0.0127\ \text{m}$$

Nucleate Boiling and Critical Heat Flux- Cryogens

Rohsenow (1952) correlation for nucleate boiling:

$$\Phi_{bi} = \mu H_{fg} \left(\frac{J_a}{C_{sf} P_r^{1.7}} \right)^3 \left[\frac{g(\rho_L - \rho_G)}{g_c \sigma_L} \right]^{1/2}$$

where C_{sf} is a surface-fluid coefficient
C_{sf} = 0.013 for all cryogen except helium, 0.169 for helium

Lienhard, Dhir and Riherd (1973) developed the following correlation for Critical Heat Flux:

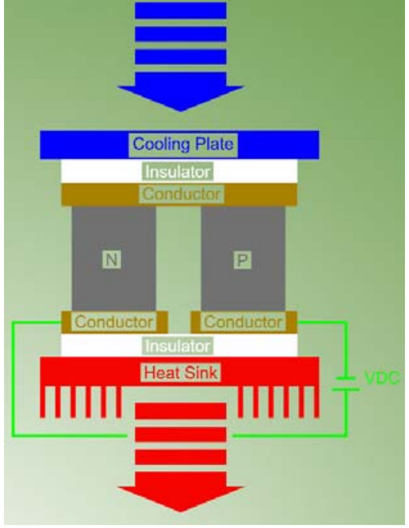
$$\Phi_{CHF} = 0.1492 \rho_G H_{fg} \left[\frac{g g_c (\rho_L - \rho_G) \sigma_L}{\rho_G^2} \right]^{1/4}$$

Liquid Nitrogen

Pressure (MPa)	Φ_{bi} (W/cm^2)	Φ_{CHF} (W/cm^2)
0.1	2.5	18.4
0.7	6.9	28.5

Peltier Cooling

A current passing through two dissimilar metals or semiconductors (n- or p-type) that are connected at two junctions (Peltier junctions) causes a heat transfer from one junction to the other. The effect was discovered by jean Peltier in 1834.



Peltier Effect

Benefits:

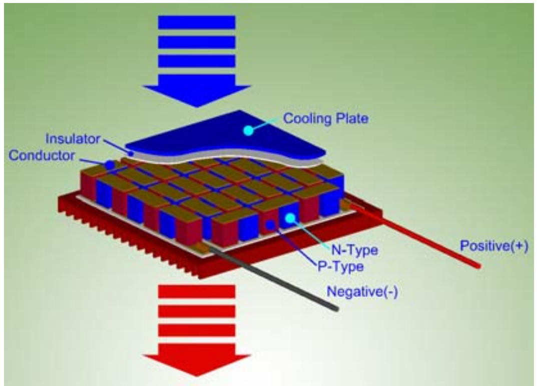
- No moving parts, low maintenance and high reliability
- Lightweight
- Controllable (by voltage / current)
- Small size
- Fast, dynamic response
- Cooling below ambient temperature is possible

Limitation:

- Low cooling capacity (relatively)

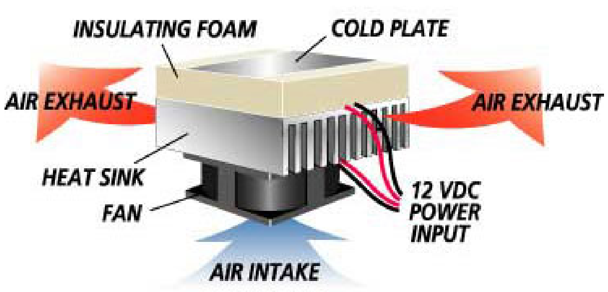
Thermoelectric Module (Peltier chip)

The P-N pairs can be connected in series electrically on metalized ceramic substrate. The pairs act in parallel thermal transferring heat from one side to the other.

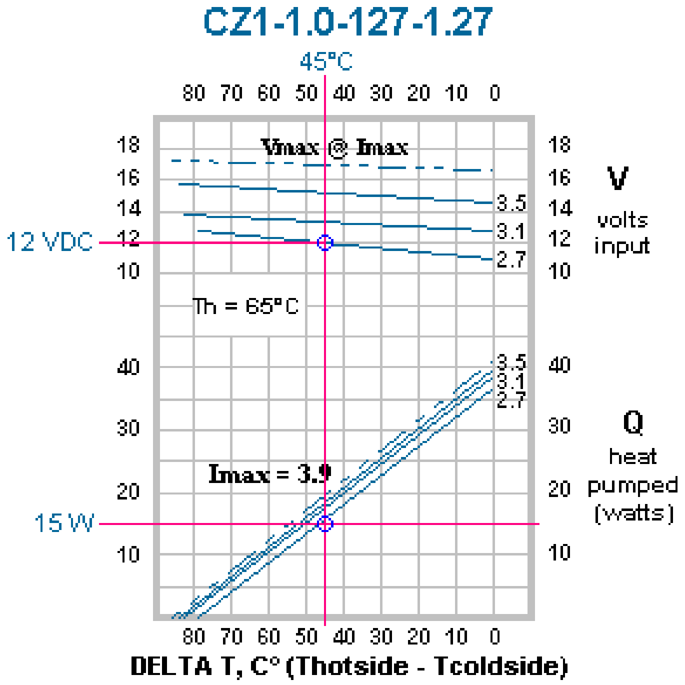


Peltier Chip

Peltier Cooling – System Design



Reference: http://www.tellurex.com/cthermo.html



System Design

$\Delta T = Th - Tc$

$Th = Ta + (V.I + Q)\, R_\theta$

where,

Tc = Cold side temperature

Th = Hot side temperature

Ta = Maximum ambient temperature

Q = Heat pumped

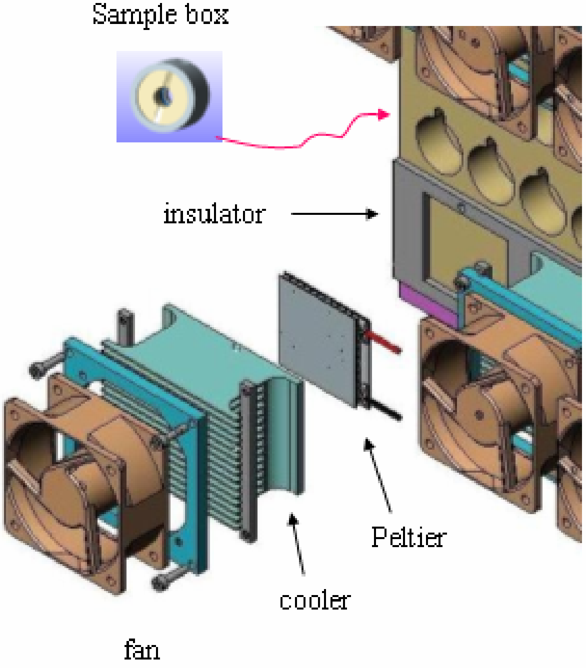
R_θ = Thermal resistance of the heat sink, ^0C / W

Peltier Cooling – An Application

High throughput sample changer

Specifications

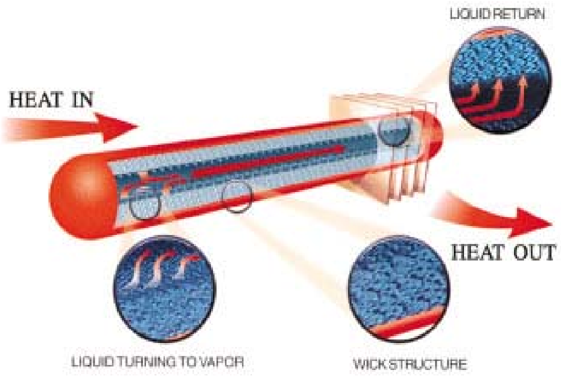
- Range : 10 °C to 150 °C
- Rapidity : 10 °C per min
- Size : 300 x 200 mm
- Weight : 2.4 kg
- Power : 500 W
- Current max : 12 A
- Samples : 30
- Regulation : Peltier element
- Regulator : Neocera LTC11/21
- Set point : +- 0.5 °C
- Accuracy : 0.1 °C



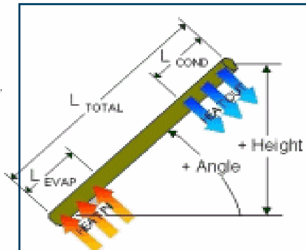
08/05/2004

J. GORINI
ESRF ID2 SAXS WAXS

Heat Pipes



http://www.thermacore.com/



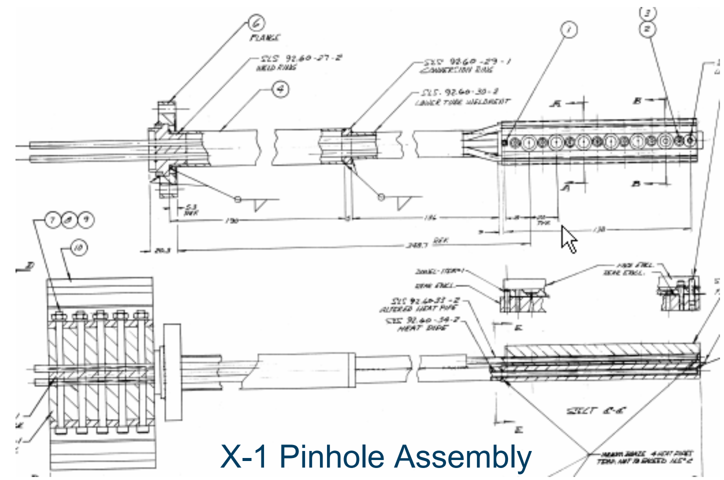
L_{TOTAL} = 100 millimeter
L_{EVAP} = 40 millimeter
L_{COND} = 40 millimeter
Height = 70 millimeter
ANGLE = 45 degrees

Heat Pipe	Number Required
0.118 in [3 mm]	13
0.157 in [4 mm]	7
0.1875 in [4.76 mm]	7
0.197 in [5 mm]	5
0.236 in [6 mm]	3
0.25 in [6.35 mm]	2
0.375 in [9.53 mm]	2
0.5 in [12.7 mm]	1
0.625 in [15.88 mm]	1
0.75 in [19.05 mm]	1
1.00 in [25.4 mm]	1

Heat Dissipation: 100 W

Advantages:

- Passive and cost-effective device
- No maintenance required
- Highly reliable, MTTF of 100,000 hours
- Heat pipes can be bent and formed in flexible configurations



X-1 Pinhole Assembly

Courtesy- P. Mortazavi (NSLS)

Miscellaneous Design Considerations

- **Water Chemistry -** Copper corrosion (clogging of small channels) , galvanic corrosion, microbiological corrosion (Dortwegt, PAC 2003, MEDSI 2002).

  DI water is too aggressive for tungsten.

- **Temperature Stability** – Important consideration for the design of diagnostics components.

- **Pressure Deformations –** Should be verified for mirrors and monochromators.
- **Pressure Drop –** Large pressure drops in the channels may lead to low flow and/or film boiling.

- **Thermal Design Criteria -** Maximum temperature rise of 150 ^{0}C for copper and 300 ^{0}C for Glidcop → too conservative

  An investigation is underway at ESRF and APS to develop more realistic criteria.

- **Radiation Damage -** Avoid mid-plane crossing of plastic hoses and instrumentation wiring.

  Hoses and wire insulation must be radiation resistant.

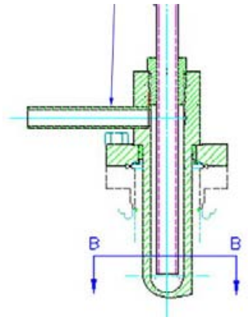
- **Flow Instrumentation -** Instrumentation should be outside the accelerator tunnels or experimental hutches.

  Components should be designed so that they can be connected in series.

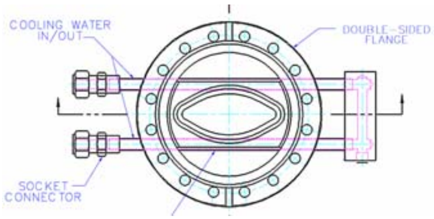
  Use reliable sensors and instruments (at APS most of the flowmeters are being replaced by Yokogawa flowmeters).

Miscellaneous Design Considerations (contd.)

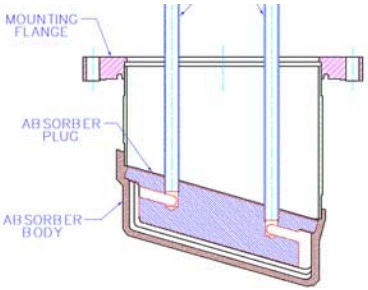
- **No Water-to-Vacuum Joints**



Counter-Flow Design



Tube In-Out Design



Double-Joint Design

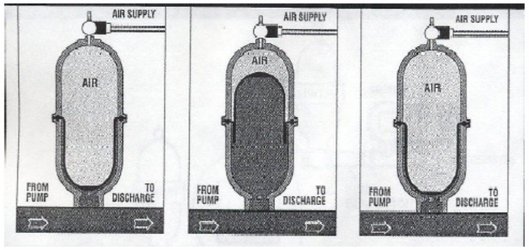
EDM Channels

- **Flow–Induced Vibrations**

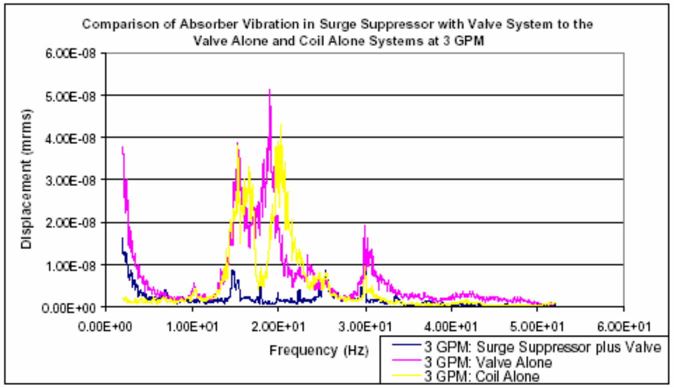
Vibration Damping – Viscoelastic Damping Pads

- Surge Suppressors

http://www.aps.anl.gov/asd/me/Publications/pdf/Lindsay.pdf



Surge Suppressor



Summary

Cooling: **Removal of heat from components while optimizing their thermal performance.**

Overview of utility system - design and specifications

Heat transfer to minimize temperature rise and thermal deformations

Cryogenic cooling – LN2 and gaseous helium

Cooling enhancements – benefits and limitations

Nucleate boiling and critical heat flux – estimated values versus design criteria

Other cooling schemes (Peltier, heat pipe) – advantages and limitations

Miscellaneous design considerations for high reliability and performance

Acknowledgments

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NSLS: L. Berman, R. Bowman, P. Montanez, P. Mortazavi, S. Pjerov