Tutorial on

Cooling

by

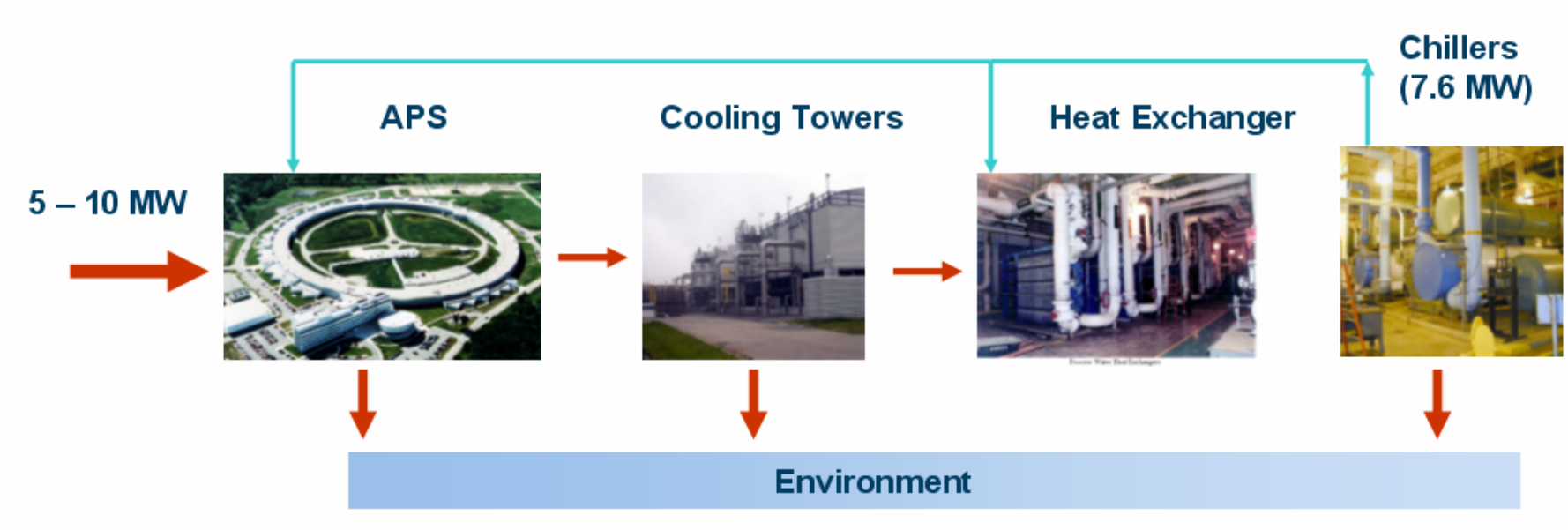
Sushil Sharma

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

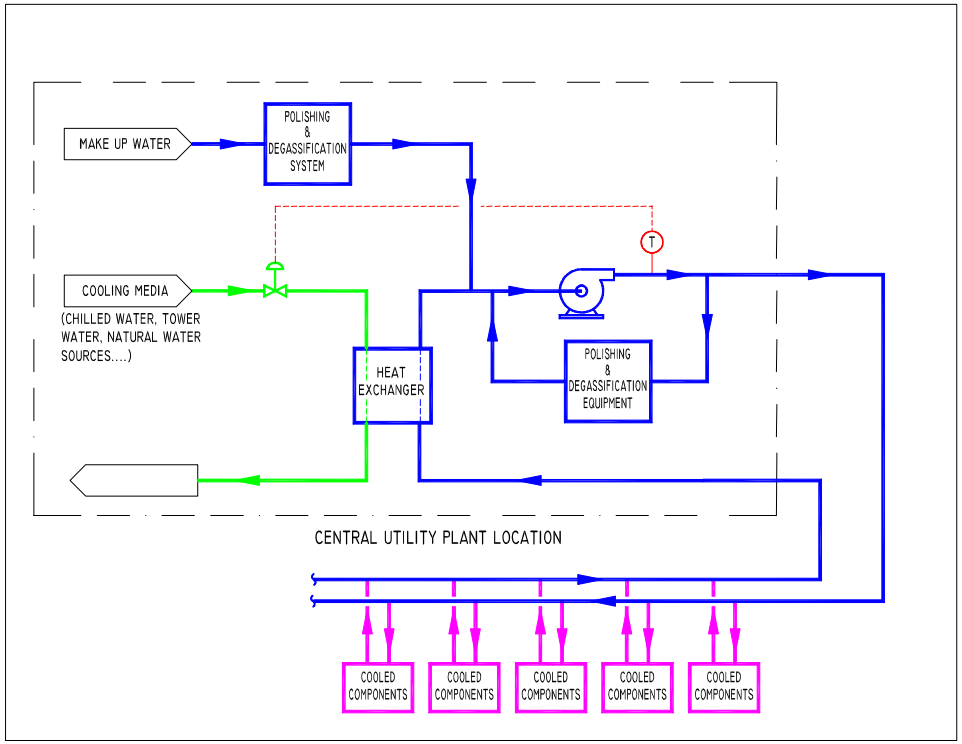
	APS Nominal Values
Base Temperature, ΔT	25.6 ˚C, < 4 °C
Temperature stability	± 0.1 °C (originally ± 1°C)
ΔP (components)	100 psi (0.69 MPa)

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

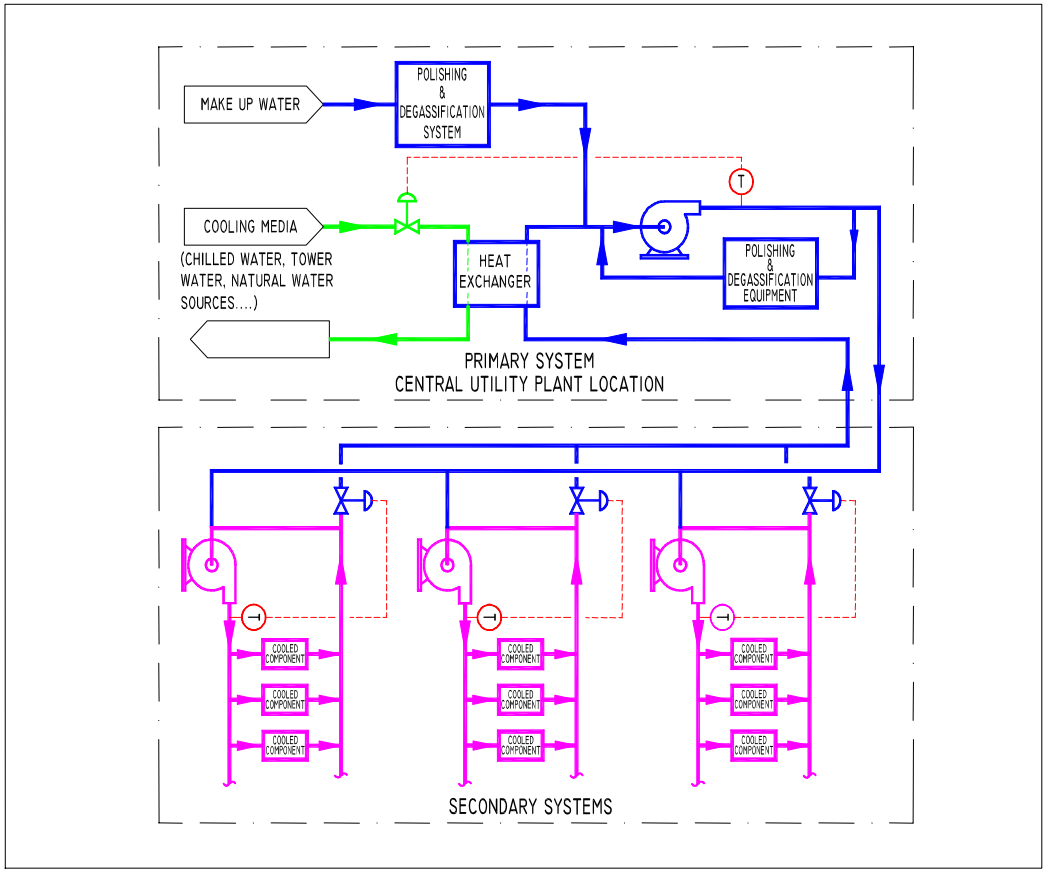
Limited flexibility

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

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

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)

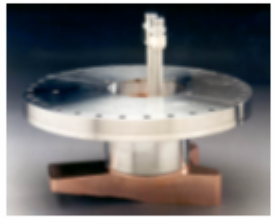
Degassifier

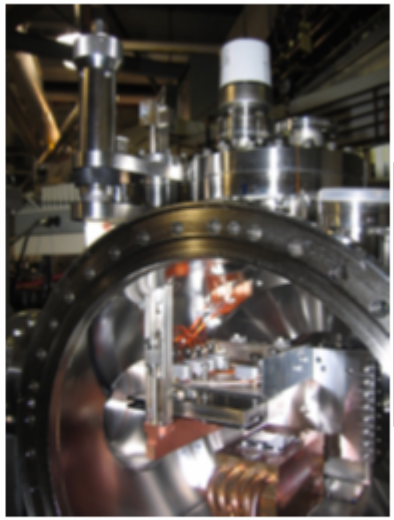
Secondary Pumps (75 HP)

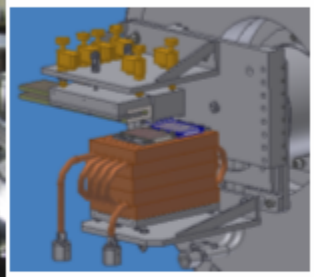
Secondary Headers in SR

Heat Transfer

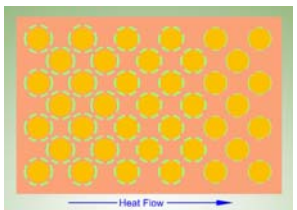
APS SR Crotch Absorber

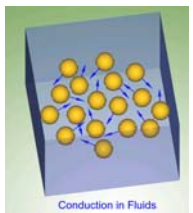


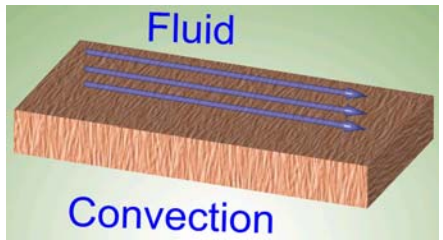


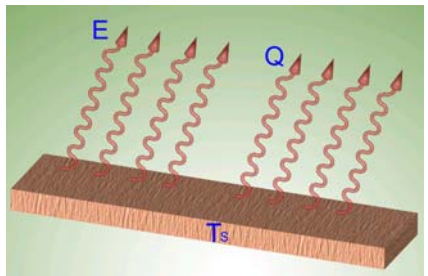


NSLS He-Cooled Si Monochromator









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

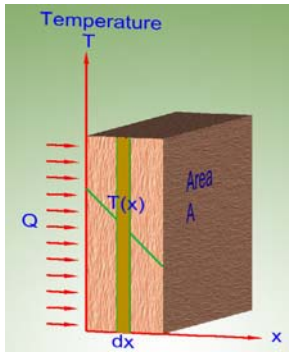
Radiation: Transfer of thermal energy by electromagnetic radiation

Heat Transfer - Conduction

Fourier Law of Conduction:

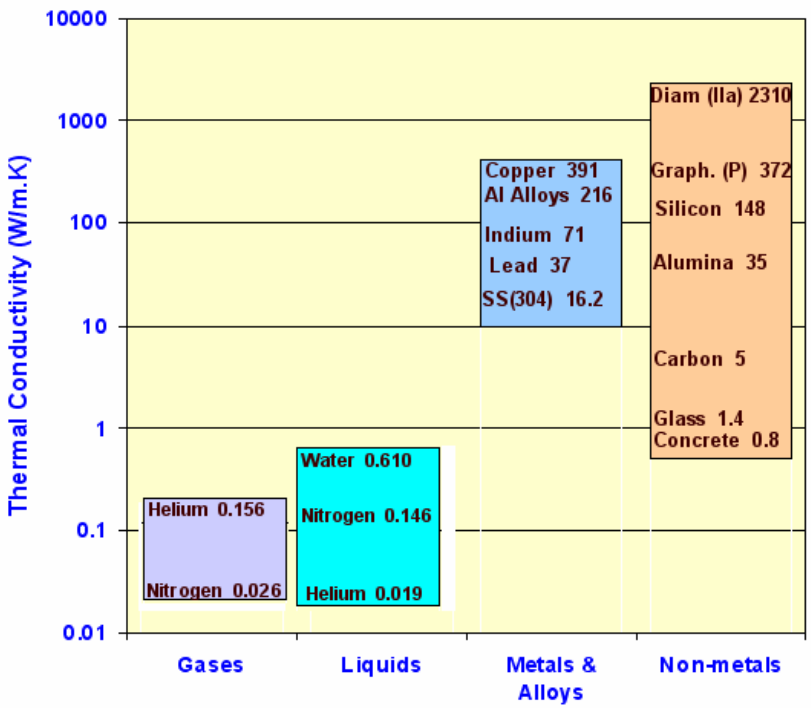
$$Q = -k\,A\frac{dT}{dx}$$

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

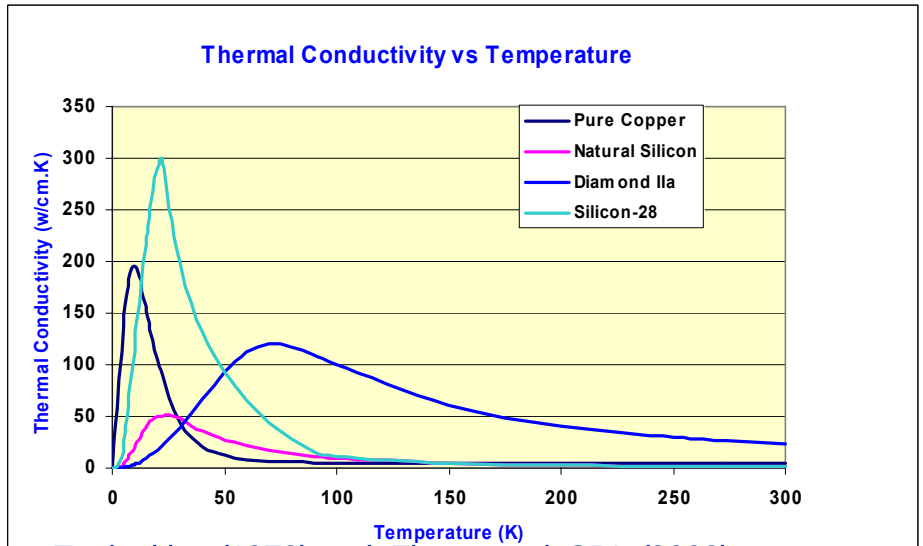
Heat Transfer- Conduction – Design Options

Fourier Law of Conduction

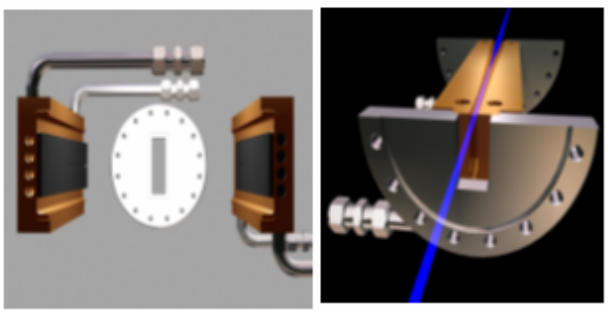
$$Q = -k\,A\,\frac{dT}{dx}$$

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

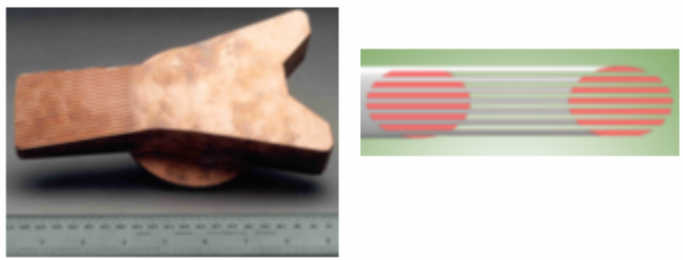
- **Using materials with high k values**
  - → **copper, Glidcop, cryogenic cooling**
- **Increasing surface area**
  - → **inclined surfaces, external fins**
- **Decreasing Q**
  - → **slits, pinholes, filters,**



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

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

n = 0.11 for $T_w > T_f$
0.25 for $T_w < T_f$
0 for constant heat flux

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

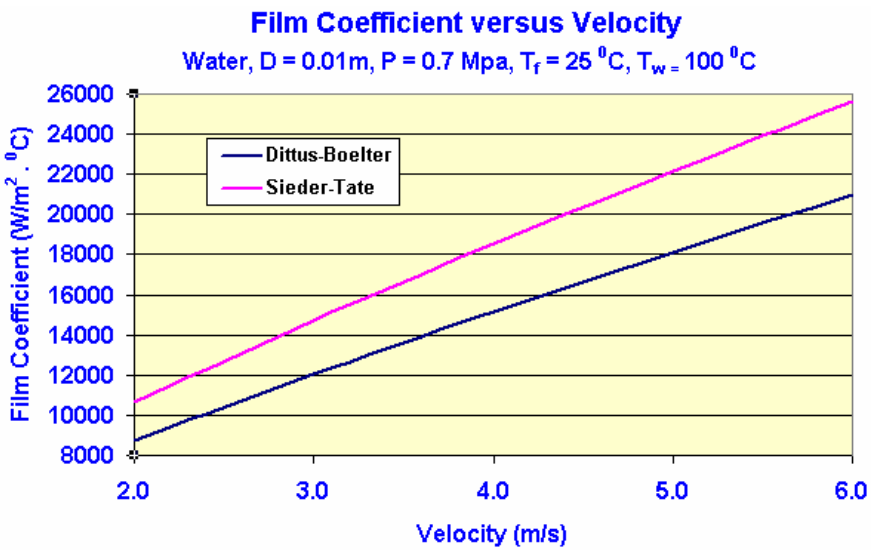
Grashof Number

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

APS Exit Port – Al Finned Tube

Typical values of convection film coefficient:

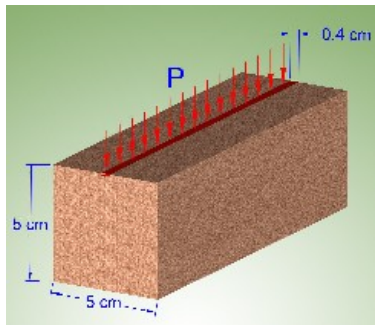
Type of convection	Medium	Film coefficient ($W/m^2.K$)
Free	gases	2 - 25
Forced	gases	25 - 250
Free	liquids	10 - 1000
Forced	liquids	50 – 20,000
Boiling	liquids	2,500 – 100,000

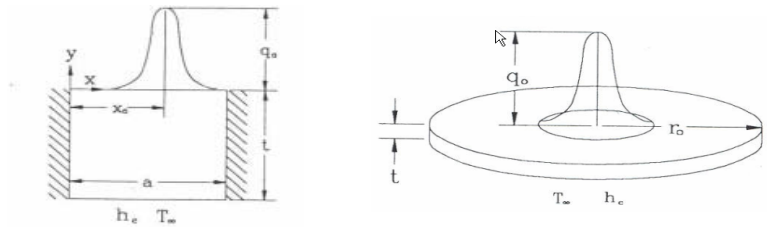


Heat Transfer - Convection – Cooling Efficiency

Example – Long Copper Block

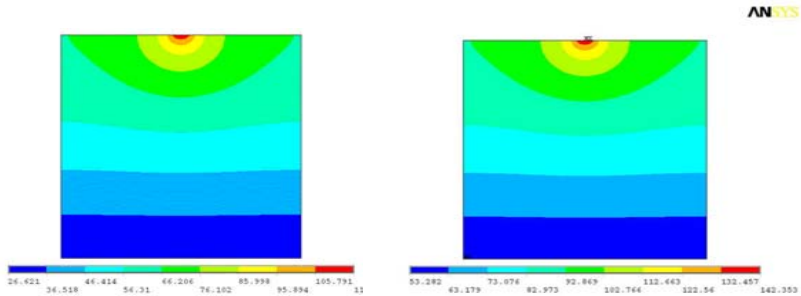
k = 395 W/m.K
P = 500 W/cm²





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 (ΔT_{mat}, α)
Thermal stresses = f (T, α, E)



h = 1500 W/m².K h =750 W/m².K

1. Effect of Cooling efficiency

(Assumption: α, k are constants)

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

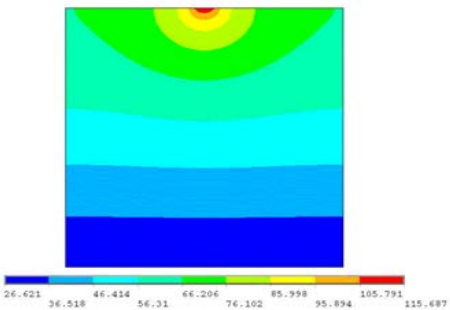
Temp. (⁰C)	h = 1500 (W/m².K)	h =750 (W/m².K)	Δ (K)
T_{max}	115.69	142.35	26.66
T_{side}	51.31	77.98	26.67
T_{min}	26.62	26.62	26.66

→ No significant change in ΔT_{mat}

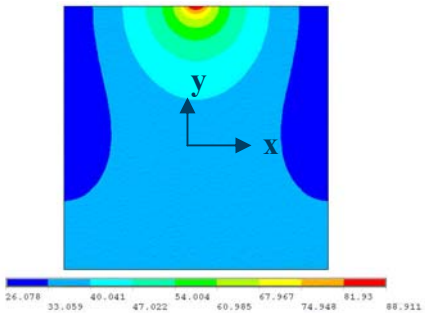
(1) When α, k are temperature-dependent (e.g. cryogenic cooling of Si), cooling efficiency can lead to material temperature being in an undesirable range, (2) Film boiling can be a factor at higher temperatures

Heat Transfer- Convection – Cooling Geometry

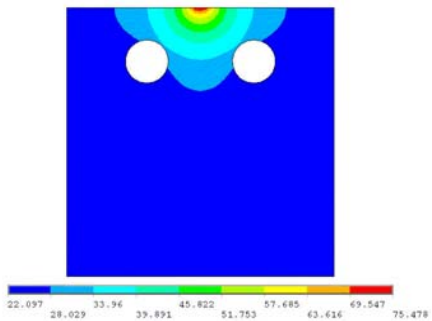
$h = 1500\ W/m^2K$



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**

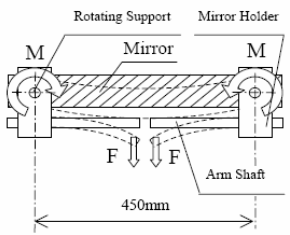
Thermal moment

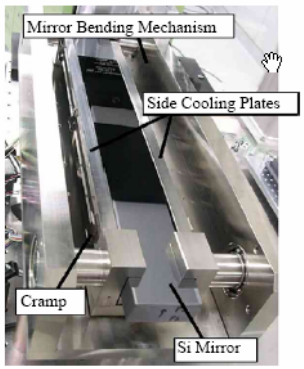
$$= \iint T(x,y)\, y\, dxdy$$

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	

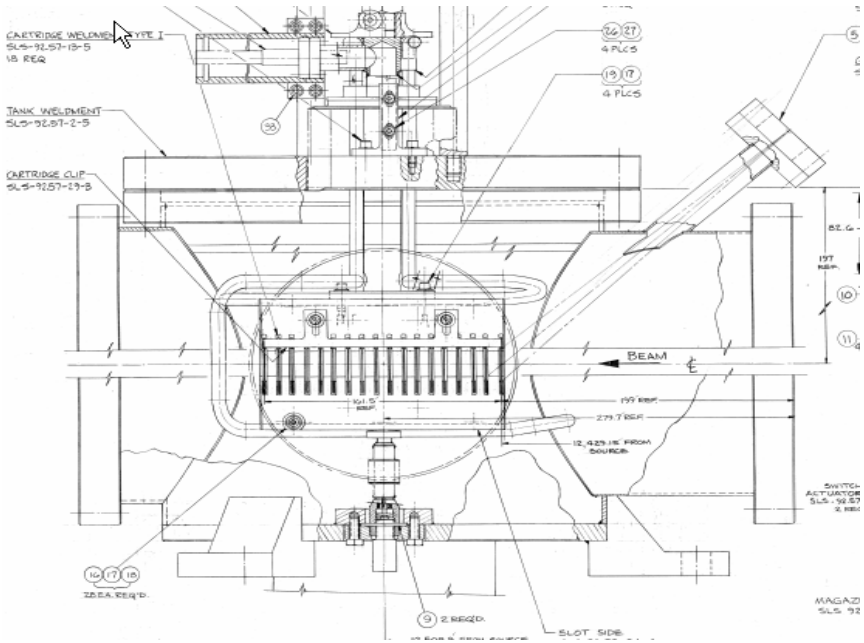
Reference: Inframetric ThermaCam Manual

Heat Transfer - Radiation – Example

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

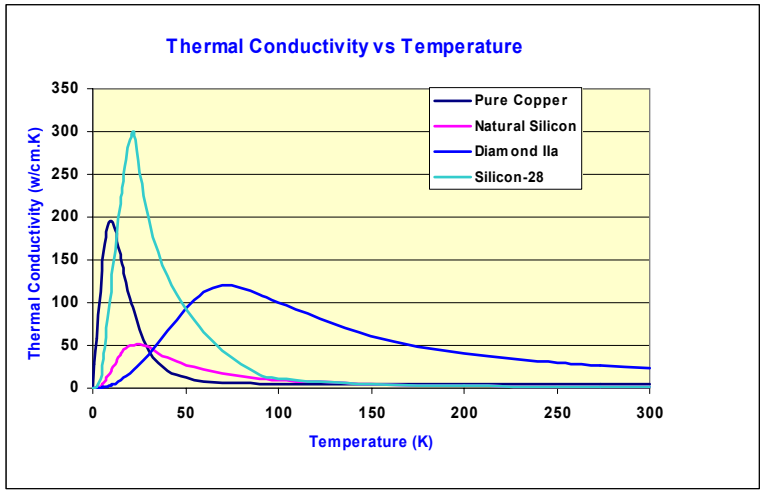
T (K)	77	200	400	600	800	1000
Q (W)	- 3.4	- 2.8	7.4	51.6	170.7	421.8

- 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)

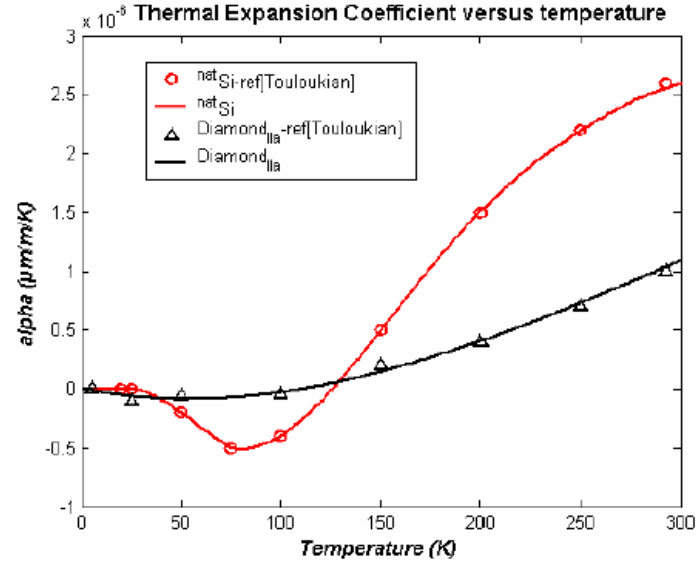
Cryogenic Cooling – Figure of Merit

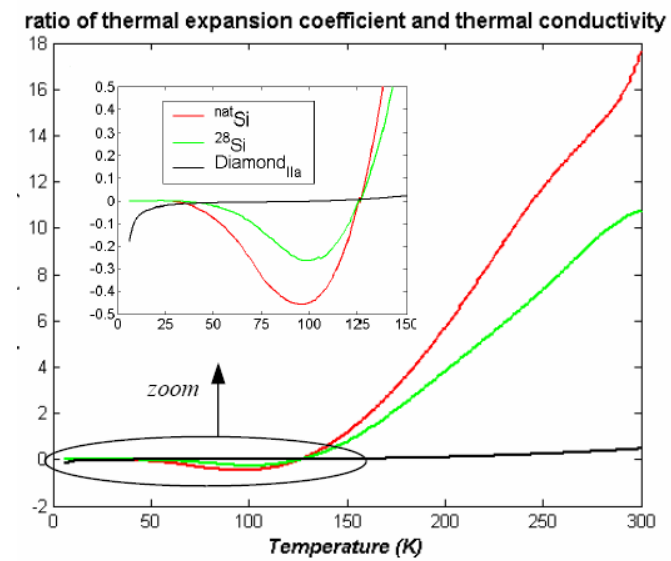


Reference: Touloukian (1970) and Zhang et al (2003)

- 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

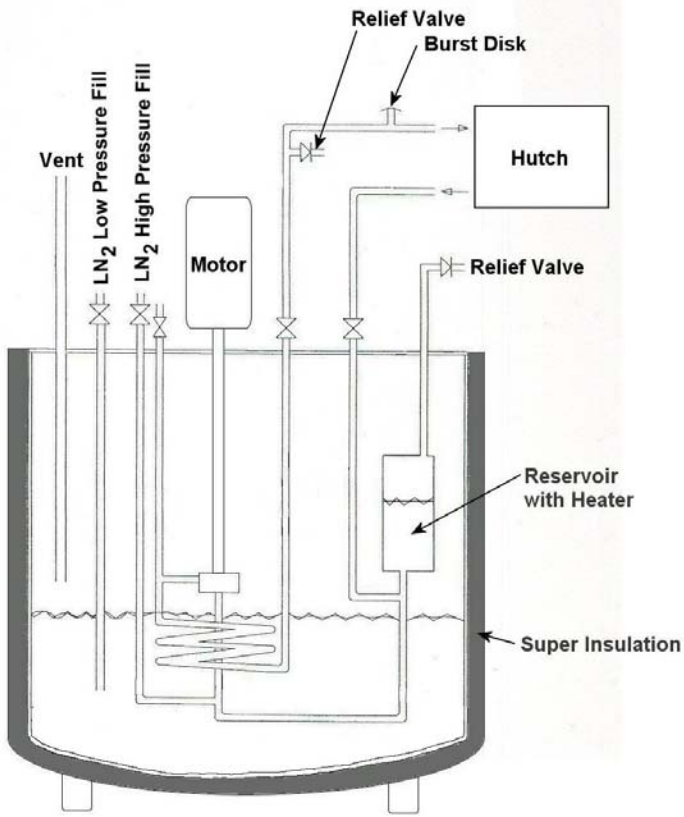
Two Options are in general use for cryogenic cooling under high heat load: (1) LN2, (2) gaseous helium

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