COOLING TECHNOLOGIES RESEARCH CENTER

Thermal Management of Electronics

Justin A. Weibel
Research Associate Professor, School of Mechanical Engineering
Associate Director, Cooling Technologies Research Center

Power and Energy, Onshore and Afloat
Course Number: CNIT 58100 (CRN: 18530) / IT 58100 (CRN:18534)
Nov 20 (3:30-4:45 PM)

Lecture Topics

Background & Motivation

- Thermal Management Landscape
- Computing Trends and Challenges
- Energy Implications

Thermal Management Basics

- Thermal Packaging Architecture
- Review of Heat Transport Resistances

Advanced Thermal Management Strategies

- Vapor Chamber Heat Spreaders
- Microchannel Heat Sinks

Further Textbook Reading

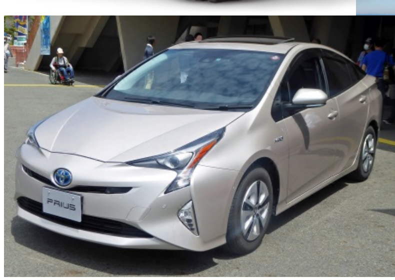
- Heat Transfer Fundamentals
  - Incropera and DeWitt, *Fundamentals of Heat and Mass Transfer*
  - A.F. Mills, *Basic Heat and Mass Transfer*, 1995
- Electronics Cooling
  - L.-T. Yeh, *Thermal Management of Microelectronic Equipment*, 2016
- Heat Pipes:
  - A. Faghri, *Heat Pipe Science and Technology*, 1995
  - G.P. Peterson, *An Introduction to Heat Pipes*, 1994
  - D. Reay and P. Kew, *Heat Pipes*, 5th Edition, 2005
- Microchannels
  - *Heat Transfer and Fluid Flow in Minichannels and Microchannels*, 2006
  - *Microchannel Heat Sinks for Electronics Cooling*, 2013

Thermal Management Landscape

Cooling Technologies Research Center

- Information and communication technology
  - High-performance computing
  - Data centers
  - Telecommunications

- Electrified transportation systems
  - Hybrid/electric powertrains
  - Aerospace
  - Watercraft



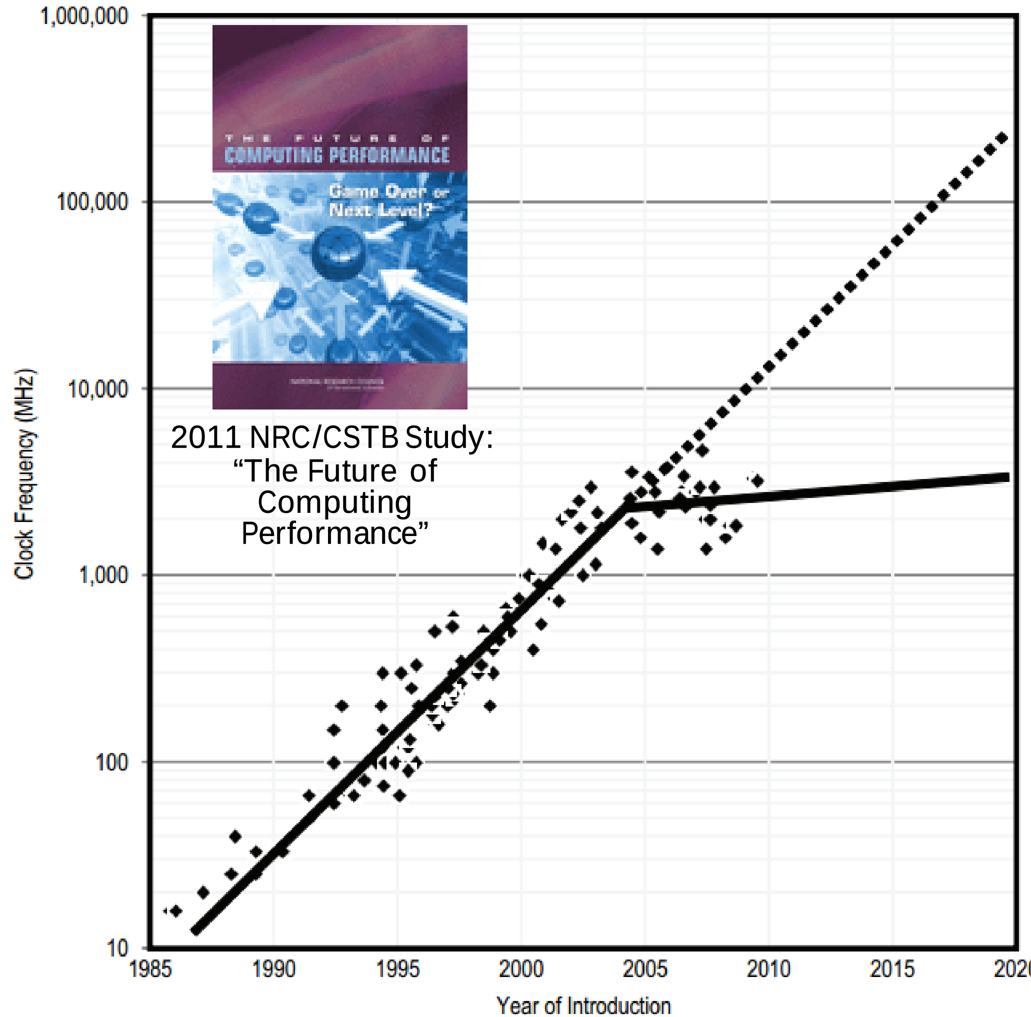
- Renewable energy
  - Smart grid
  - Energy conversion processes
  - Photovoltaics
  - Batteries

Computing Trends

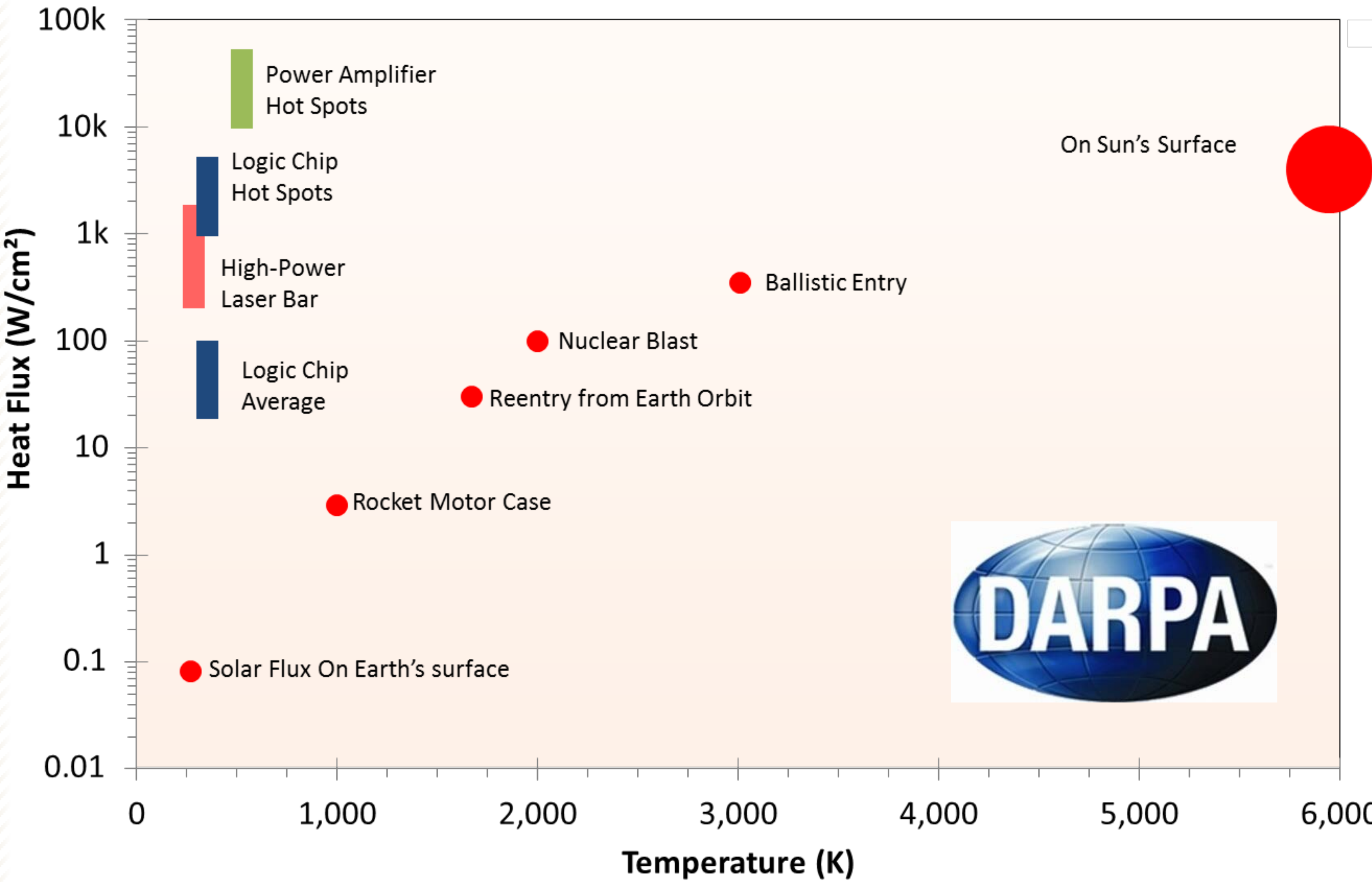
National Research Council Study by the
Committee on Sustaining Growth in Computing Performance

"Thermal-power challenges and increasingly expensive energy demands pose threats to the historical rate of increase in processor performance."

"…growth in the performance of computer systems will become limited by their power and thermal requirements within the next decade"

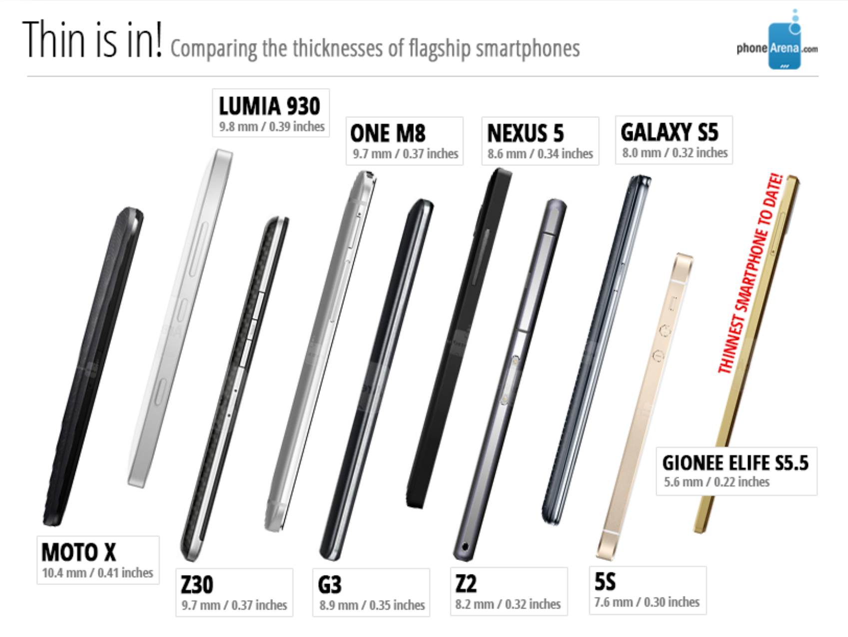


Heat Flux Challenge



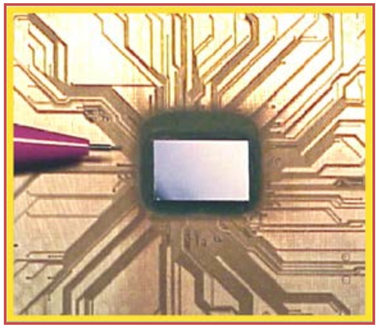
Other Challenges

- Multi-core processing: more efficient and lower heat fluxes…. but causes local hot spots that must be accommodated
- Thermal management in extremely space constrained devices
- System-level volumetric heat generation density
- Thermo-mechanical and Electro-thermal co-design constraints
- Cost, consumer perception & human factors, market trends, etc.



Challenge of Multiple Scales

Silicon Die (10^{-2})

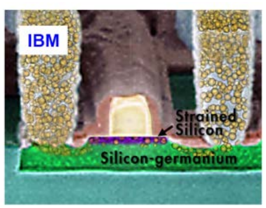


Package (10^{-2})

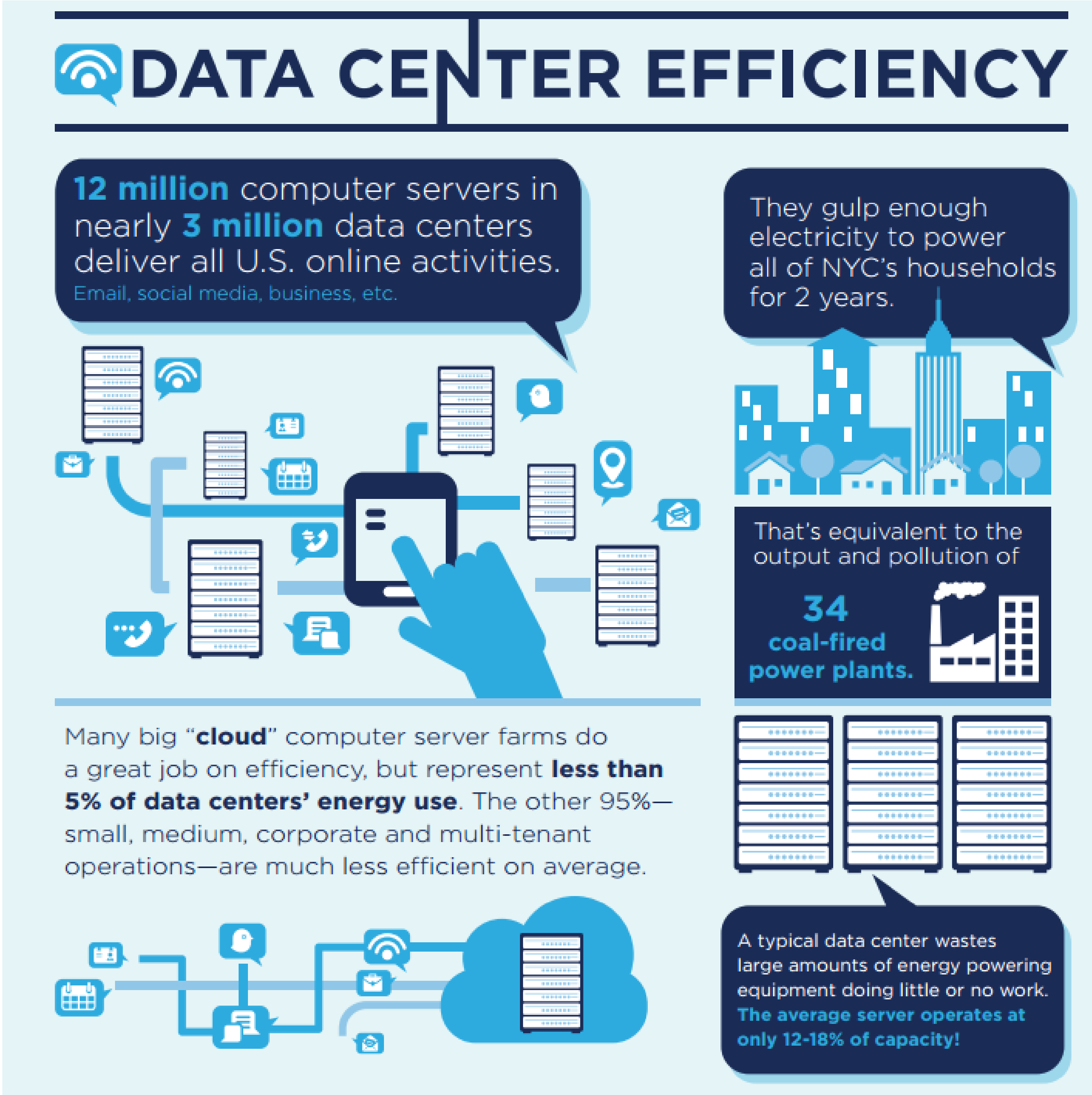
Heat Sink (10^{-1})

System (10^{0})

Facility (10^{2})



Transistor (10^{-9} m)



Natural Resources Defense Council (NRDC) Data Center Efficiency Assessment, August 2014

Waterlogged

A midsize data center uses roughly as much water as about 100 acres of almond trees or three average hospitals, and more than two 18-hole golf courses.

Approximate annual water usage, in gallons*

Data Center
(15 megawatt)
130M

Hospital
(3 buildings)
130M

Almond orchard
(100 acres)
115M

Golf Courses
(2 18-hole courses)
100M

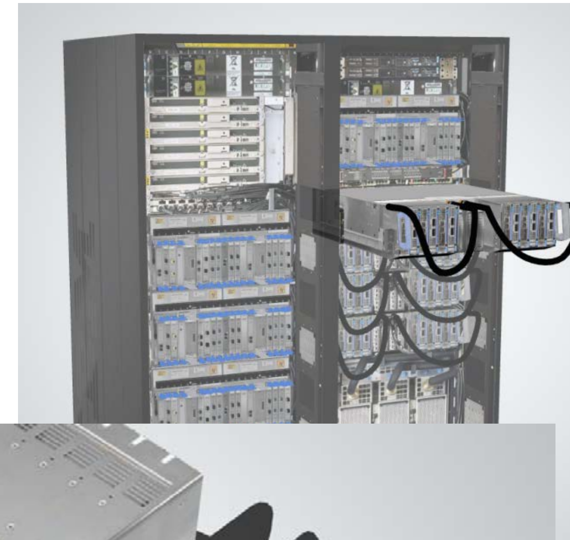
*Use varies depending on climate and other factors

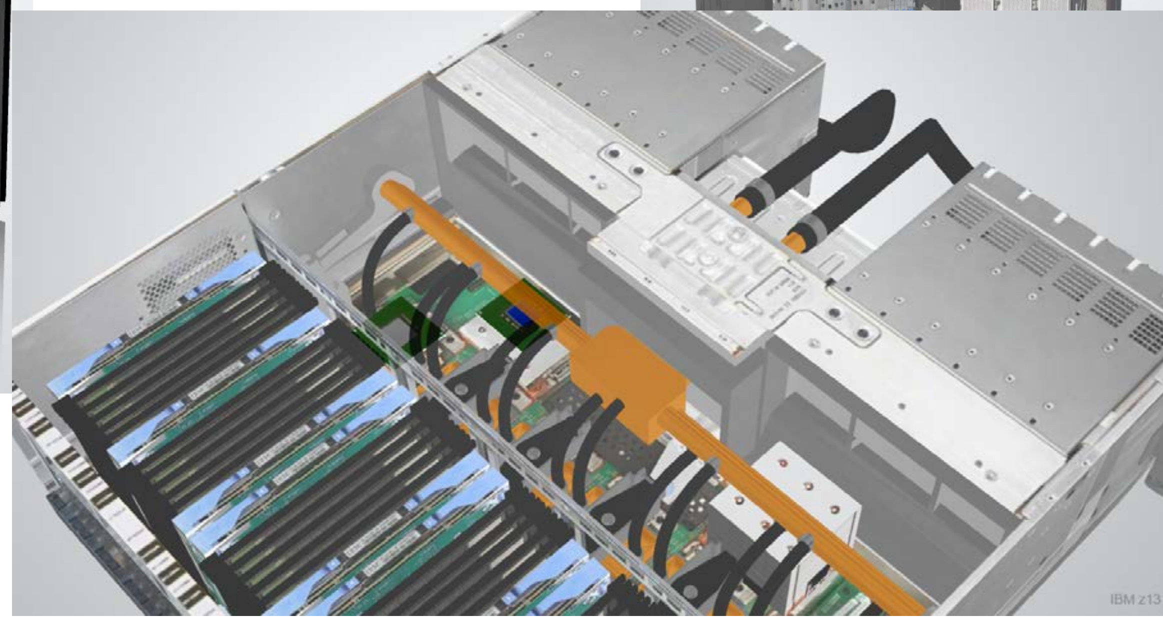
Sources: California Department of Water Resources (orchards); James Hamilton (data centers); U.S. Department of Energy (hospitals); Golf Course Superintendents Association of America (golf courses)

THE WALL STREET JOURNAL.

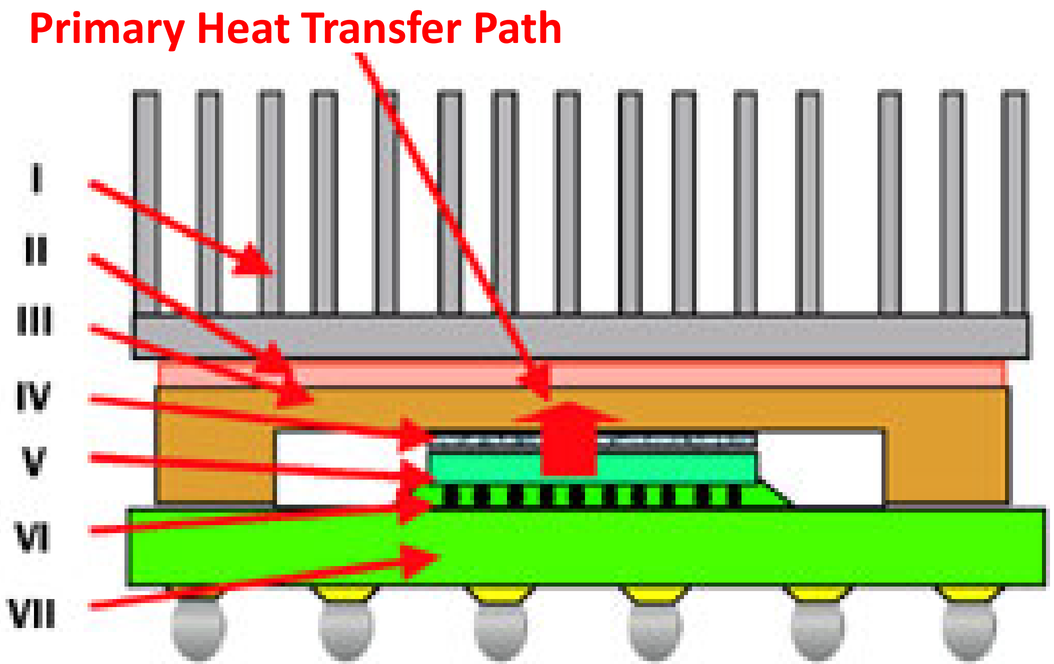
IBM Z13 Server

https://apps.kaonadn.net/4882011/product.html#10/1066;C178





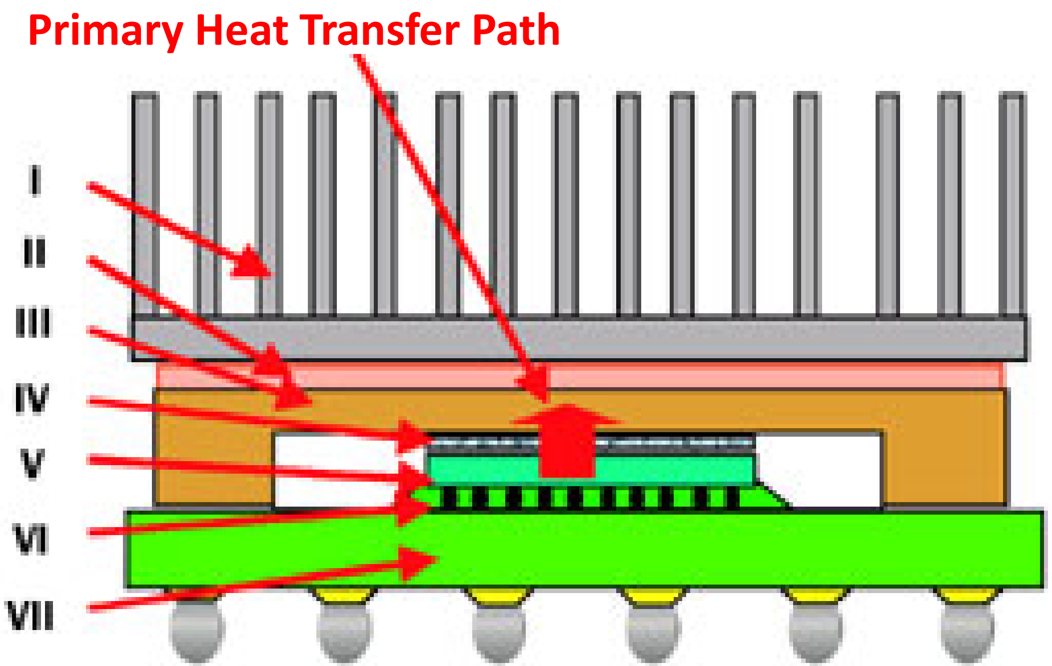
Thermal Management Architectures

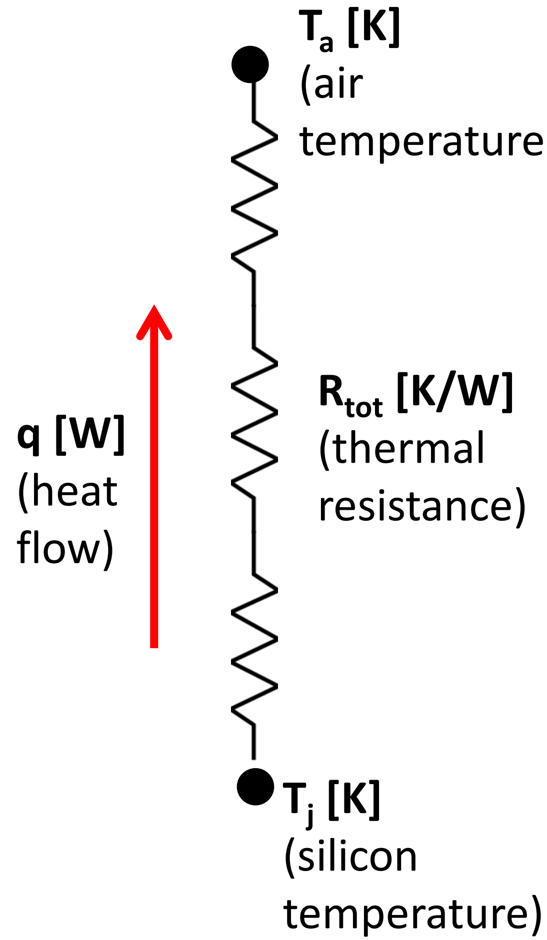


- I – Heat Sink
- II – Thermal Interface Material
- III – Integrated Heat Spreader
- IV – Thermal Interface Material
- V – Die
- VI – Underfill
- VII – Package Substrate

Mahajan, "Thermal Interface Materials: A Brief Review of Design Characteristics and Materials", *Electronics Cooling, Feb 2004*

Simple Resistance Model



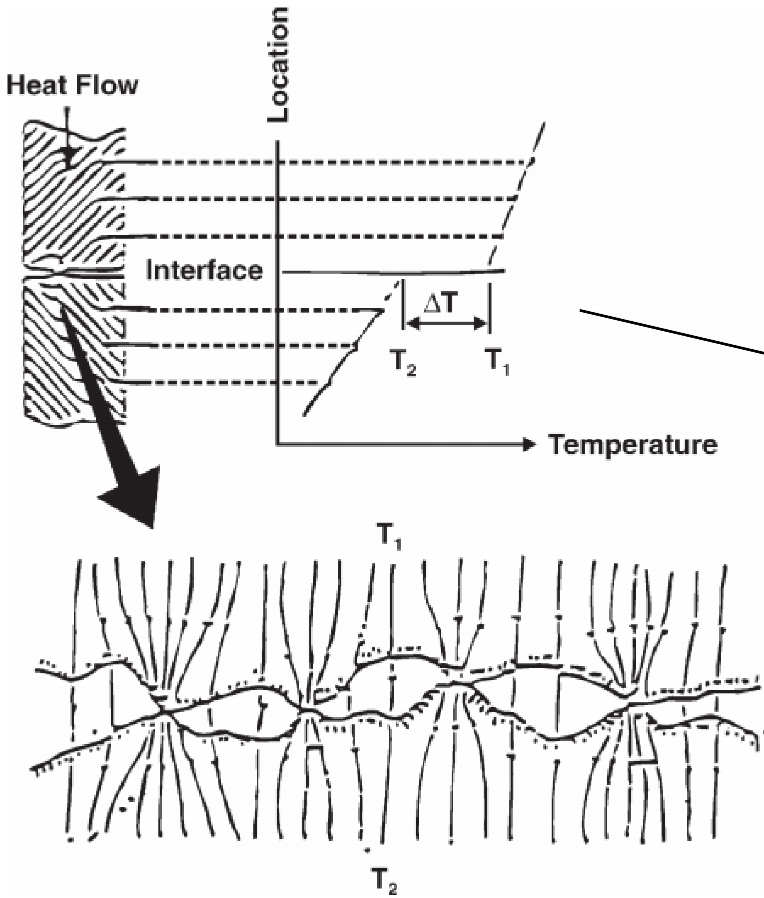


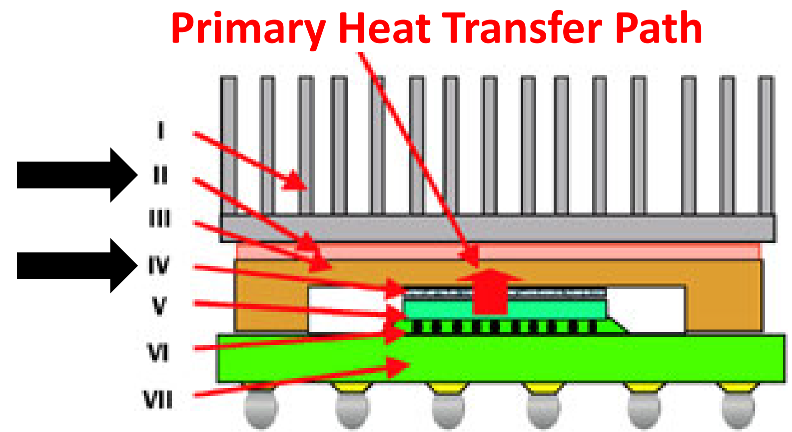
Total temperature rise of the chip described by the junction-to-ambient thermal resistance

$$q = \frac{\left(T_j - T_a\right)}{R_{tot}}$$

Interfacial Resistances

Interfaces pose large resistances due to microscale surface asperities





$$R_c = \frac{1}{h_c} = \frac{(\Delta T)}{q}$$

Dry Contact Conductance

$$h_c = 1.25\left(\frac{k_s m}{\sigma}\right)\left(\frac{P}{H}\right)^{0.95}$$

k_s = harmonic mean conductivity, W/(m·°C) $= \frac{2k_1k_2}{k_1+k_2}$

m = RMS absolute surface asperity slope $= (m_1^2 + m_2^2)^{0.5}$

σ = RMS surface roughness, m $= (\sigma_1^2 + \sigma_2^2)^{0.5}$

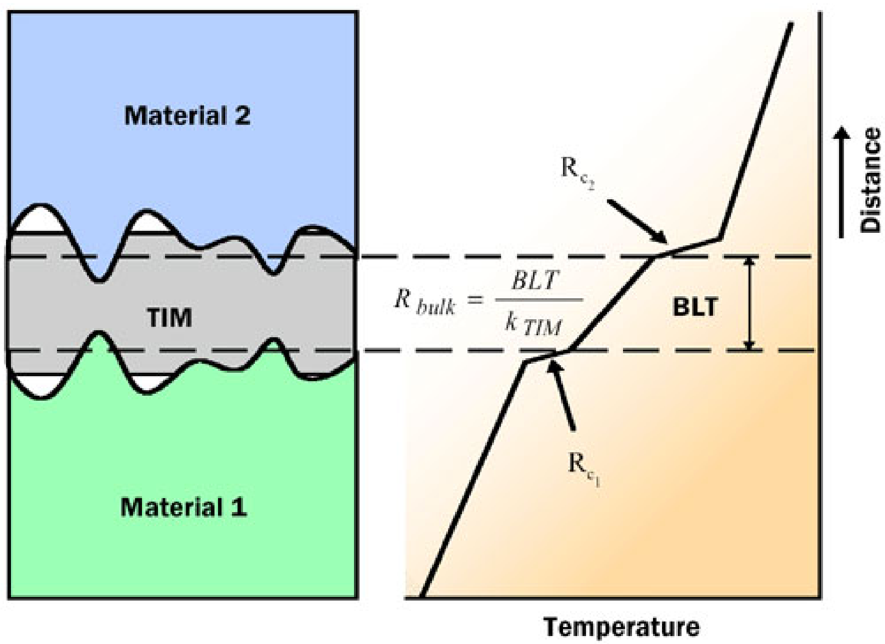
P = apparent contact pressure, N/m²

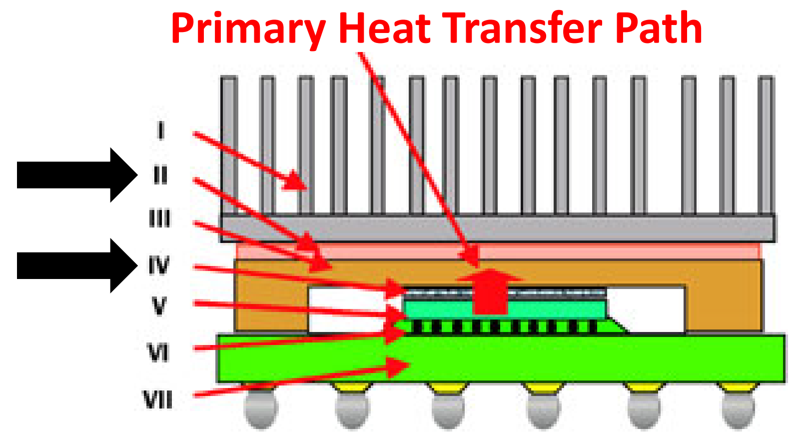
H = hardness of the softer solid, N/m²

L.-T. Yeh, Thermal Management of Microelectronic Equipment, 2016

Thermal Interface Materials (TIM)

Materials introduced into the air gap to reduce the thermal resistance





$$R_{TIM} = R_{bulk} + R_{c1} + R_{c2}$$

See reference below for calculation of individual resistances

Mahajan, "Thermal Interface Materials: A Brief Review of Design Characteristics and Materials", *Electronics Cooling, Feb 2004*

Thermal Interface Materials (TIM)

Cooling Technologies Research Center

TIM Type	General Characteristics	Advantages	Disadvantages
Greases	Typically silicone based matrix loaded with particles (typically AlN or ZnO) to enhance thermal conductivity	•High bulk thermal conductivity •Thin BLT with minimal attach pressure •Low viscosity enables matrix material to easily fill surface crevices •TIM curing not required •TIM delamination is not a concern	•Susceptible to grease pump-out and phase separation •Considered messy in a manufacturing environment due to a tendency to migrate
Phase Change Materials	Polyolefin, epoxy, low molecular weight polyesters, acrylics typically with BN or Al_2O_3 fillers	•Higher viscosity leads to increased stability and hence less susceptible to pumpout •Application and handling is easier compared to greases •No cure required •Delamination is not a concern	•Lower thermal conductivity than greases •Surface resistance can be greater than greases. Can be reduced by thermal pre-treatment •Requires attach pressure to increase thermal effectiveness hence can lead to increased mechanical stresses
Gels	Al, Al_2O_3, Ag particles in silicone, olefin matrices that require curing	•Conforms to surface irregularity before cure •No pump out or migration concerns	•Cure process needed •Lower thermal conductivity than grease •Lower adhesion than adhesives; delamination can be a concern
Adhesives	Typically Ag particles in a cured epoxy matrix	•Conform to surface irregularity before cure •No pump out •No migration	•Cure process needed •Delamination post reliability testing is a concern •Since cured epoxies have high post cure modulus, CTE mismatch induced stress is a concern

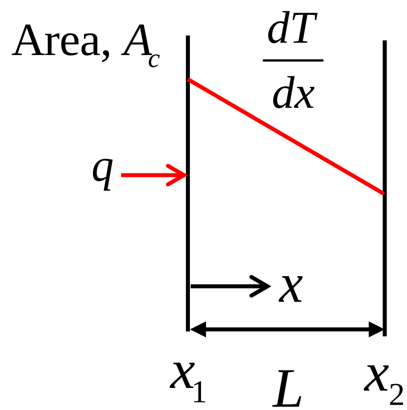
Mahajan, "Thermal Interface Materials: A Brief Review of Design Characteristics and Materials", *Electronics Cooling, Feb 2004*

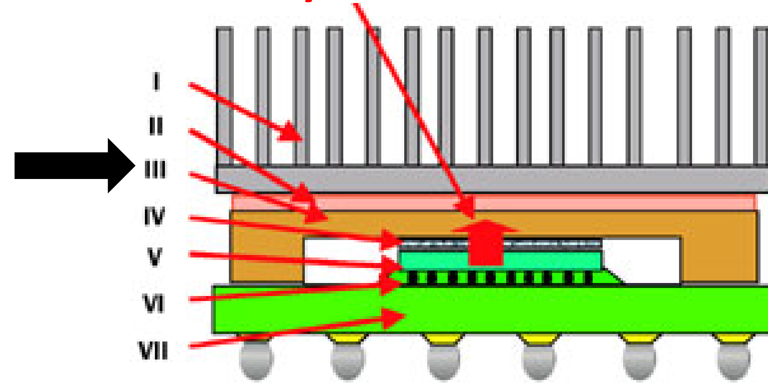
PURDUE UNIVERSITY

Principles of Conduction

Fourier's Law (1D):

$$q = -kA_c \frac{dT}{dx}$$





k – thermal conductivity [W/mK] is a material property

A_c – cross sectional area perpendicular to heat flow

dT/dx is the rate of change of temperature with distance (K/m)

1D Conduction Resistance

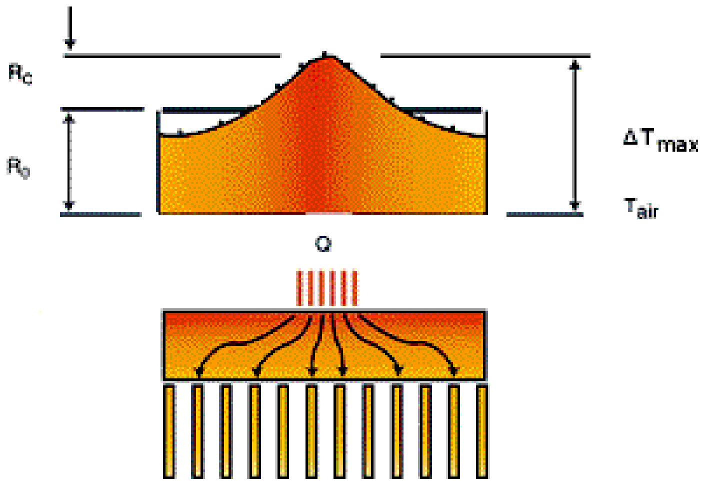
$$q = \frac{\Delta T}{R_{cond,1D}}$$

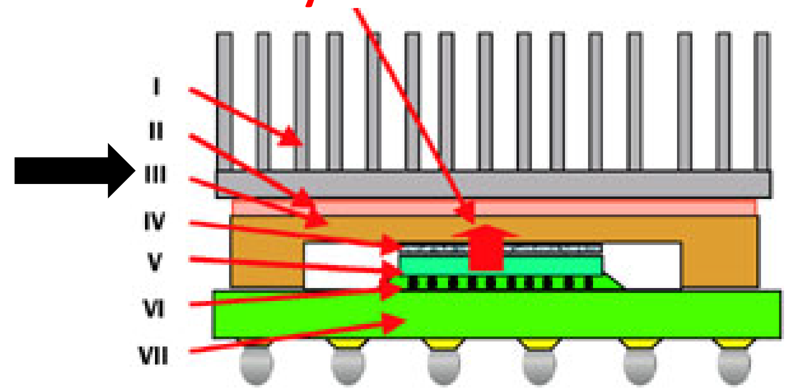
$$R_{cond,1D} = \frac{L}{kA_c}$$

L.-T. Yeh, Thermal Management of Microelectronic Equipment, 2016

Integrated Heat Spreader Resistance

Heat flow is not 1D when heat sink is larger than the chip area; 'constriction' resistance associated with heat spreading to different area.





The spreading resistance can be determined from the following set of parameters:

- footprint or contact area of the heat source, A_s
- footprint area of the heat sink base-plate, A_p
- thickness of the heat sink base-plate, t
- thermal conductivity of the heat sink base-plate, k
- average heat sink thermal resistance, R_0

$$R_c = \left(\frac{\sqrt{A_p} - \sqrt{A_s}}{k\sqrt{\pi A_p A_s}}\right)\left(\frac{\pi k A_p R_o + \tanh(\lambda t)}{1 + \pi k A_p R_o \tanh(\lambda t)}\right)$$

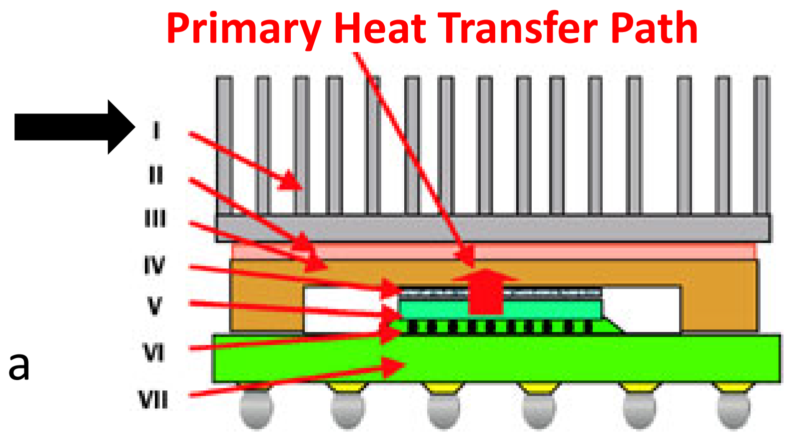
$$\lambda = \frac{\pi^{3/2}}{\sqrt{A_p}} + \frac{1}{\sqrt{A_s}}$$

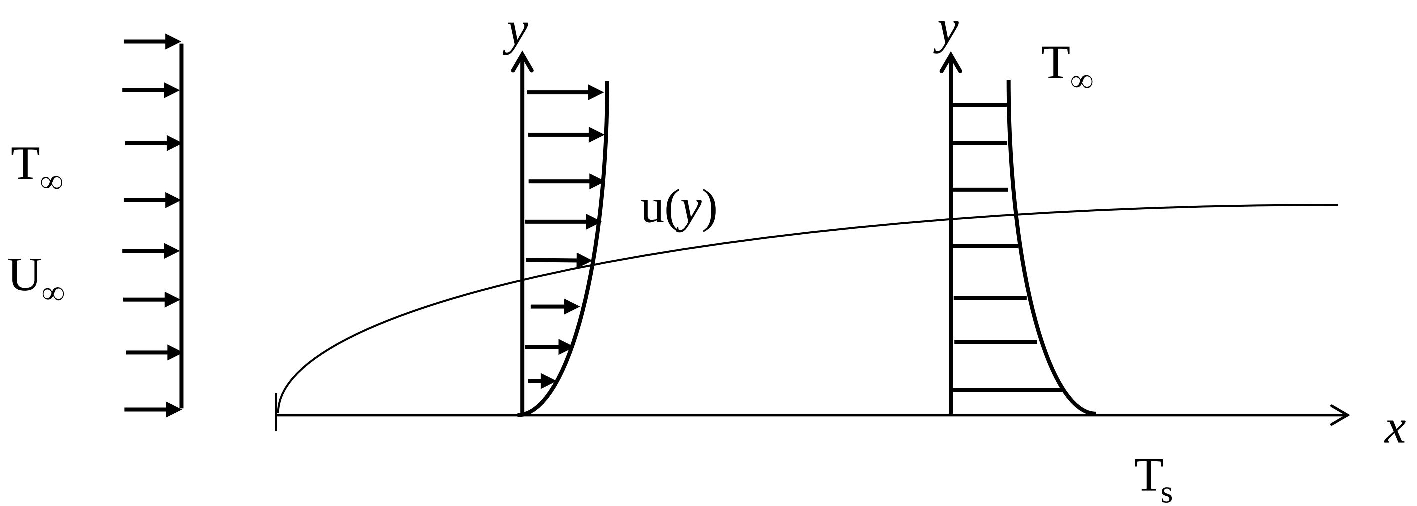
Principles of Convection

Newton's Law of Cooling:

$$q = \bar{h}A(T_s - T_\infty) \qquad \bar{h} = \frac{1}{L}\int_0^L h\,dx$$

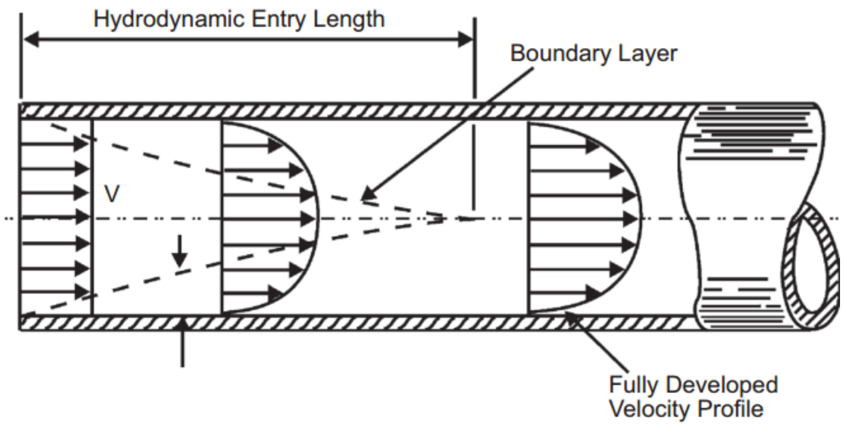
h – heat transfer coefficient [W/m^{2}K] is not a material property, but depends on the flow conditions and geometry



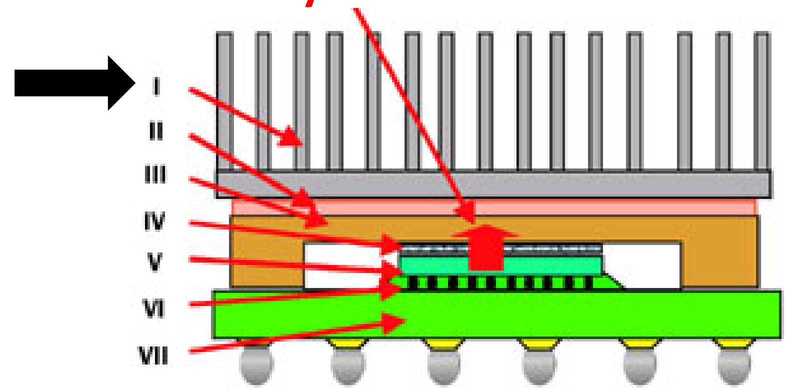


Principles of Convection

COOLING TECHNOLOGIES RESEARCH CENTER



Primary Heat Transfer Path



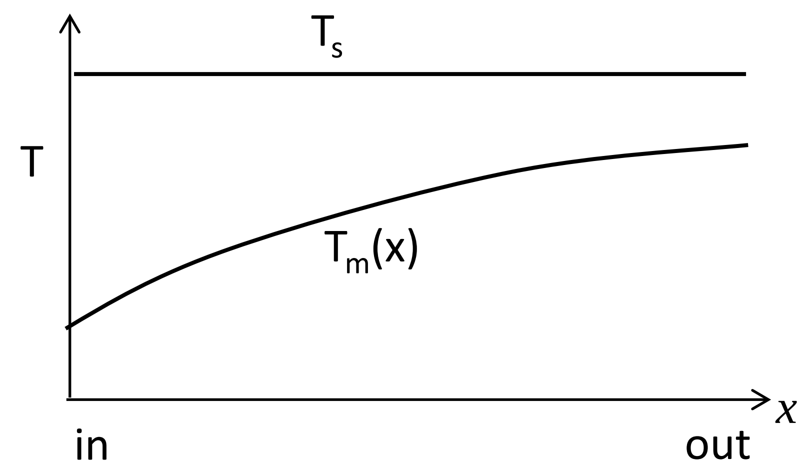
L.-T. Yeh, Thermal Management of Microelectronic Equipment, 2016

Internal Flow (<u>Constant Surface Temperature</u>):

$$q = \bar{h}A(\Delta T_{lm}) = \dot{m}c_p(T_{out} - T_{in})$$

$$\Delta T_{lm} = \frac{(T_s - T_{out}) - (T_s - T_{in})}{\ln[(T_s - T_{out})/(T_s - T_{in})]}$$

$$T_m(x) = T_s + (T_{m,in} - T_s)\exp\left(\frac{hPx}{\dot{m}c_p}\right)$$



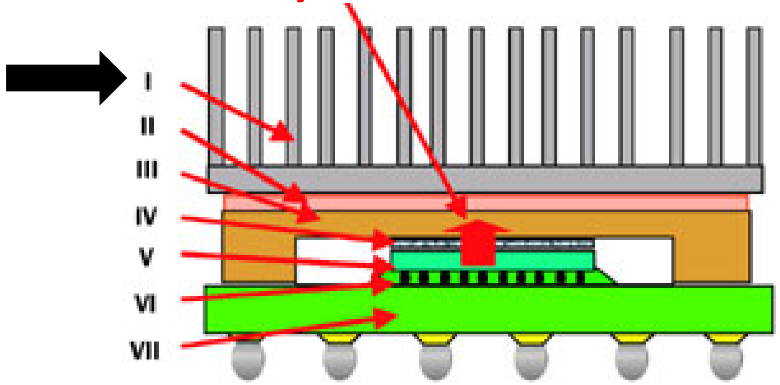
PURDUE UNIVERSITY

Fin Heat Sink Resistance

Cooling Technologies Research Center

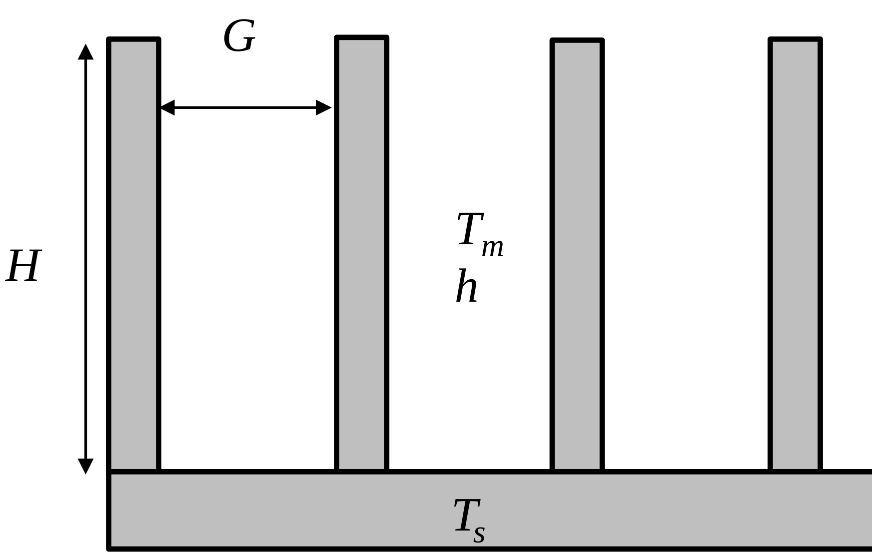
Assumptions

- Uniform heat sink base temperature
- Fully developed flow between fins (conservative)
- Uniform flow
- Ambient air temperature constant around fins



$$q = \eta_o \bar{h} A_t (\Delta T_{lm})$$

$$R_t = \frac{1}{\eta_o \bar{h} A_t}$$



R_t – effective resistance that accounts for the parallel paths by conduction in the fins and convection to the air

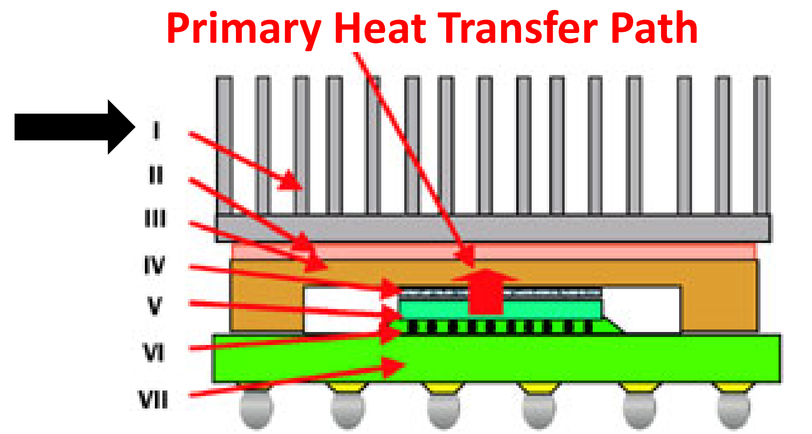
η_o - surface efficiency

Purdue University

Fin Heat Sink Resistance

Assumptions

- Uniform heat sink base temperature
- Fully developed flow between fins (conservative)
- Uniform flow
- Ambient air temperature constant around fins



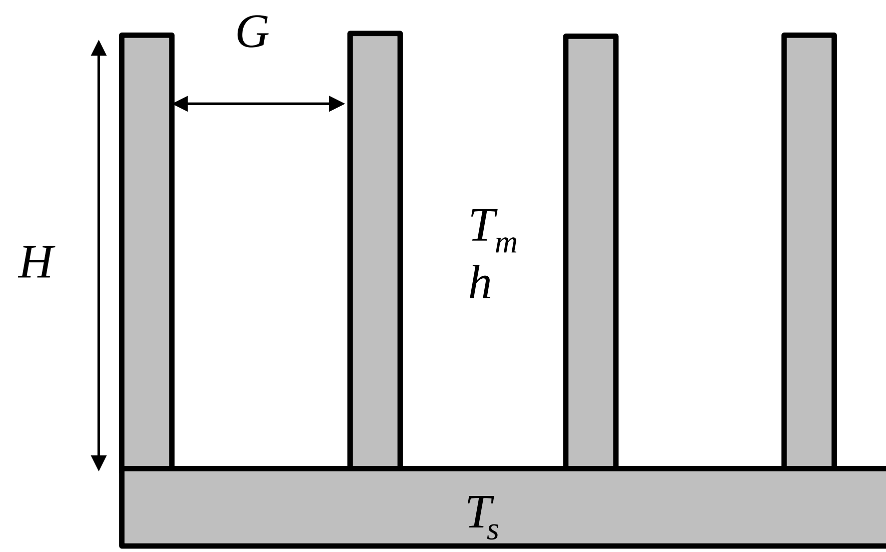
$$\bar{h} = \frac{k_{fluid} Nu}{d_c}$$

Nu – nondimensional Nusselt number is a constant for fully developed laminar flow

d_c - Hydraulic diameter [m] is the characteristic dimension

$$d_c = \frac{4GH}{2H + G}$$

G – fin gap; H – fin height



Fin Heat Sink Resistance

Total Heat Sink Resistance :

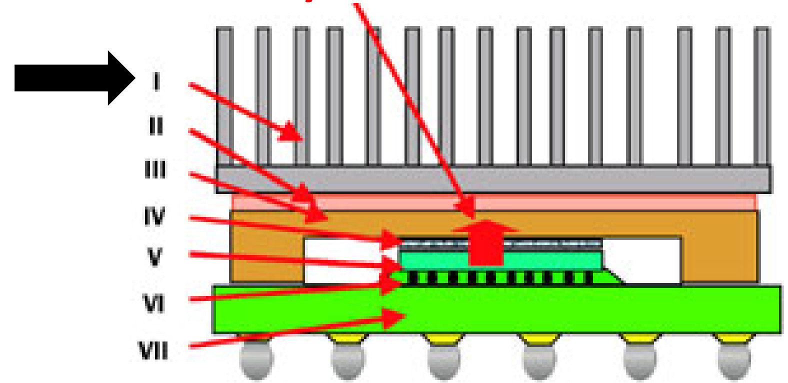
$$R_{HS} = \frac{(T_s - T_a)}{q}$$

$$q = \eta_o \bar{h} A_t (\Delta T_{lm}) = \dot{m} c_p (T_{out} - T_a)$$

$$\Delta T_{lm} = \frac{(T_s - T_{out}) - (T_s - T_a)}{\ln[(T_s - T_{out})/(T_s - T_a)]}$$

$$T_{out} = T_s - (T_s - T_a)\exp\left(-\frac{\eta_o \bar{h} A_t}{\dot{m} c_p}\right)$$

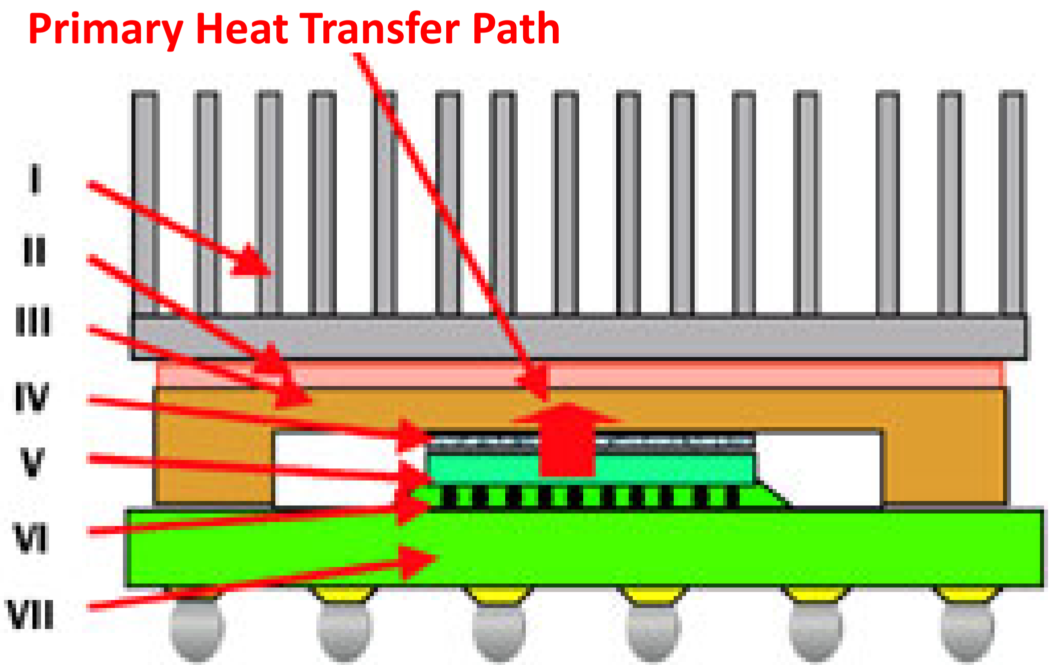
Primary Heat Transfer Path

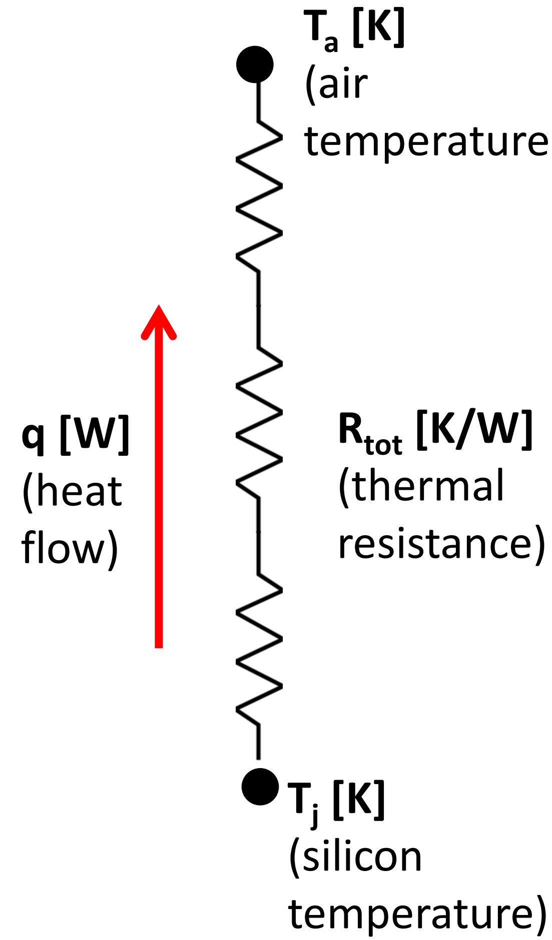


$$R_{HS} = \frac{(T_s - T_a)}{q} = \frac{1}{\dot{m} c_p \left[1 - \exp\left(-\frac{\eta_o \bar{h} A_t}{\dot{m} c_p}\right)\right]}$$

Determines the required flow rate of air

Simple Resistance Model





Total temperature rise of the chip described by the junction-to-ambient thermal resistance

$$q = \frac{\left(T_j - T_a\right)}{R_{tot}}$$

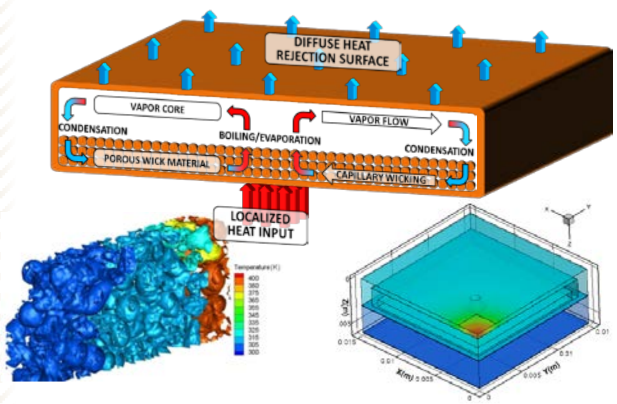
Advanced Thermal Management Strategies

Next-Gen Thermal Management

Naval vessels and aircraft are equipped with increasingly advanced electronic systems that require high-density cooling strategies to enable reductions in size, weight, and power without compromising performance.

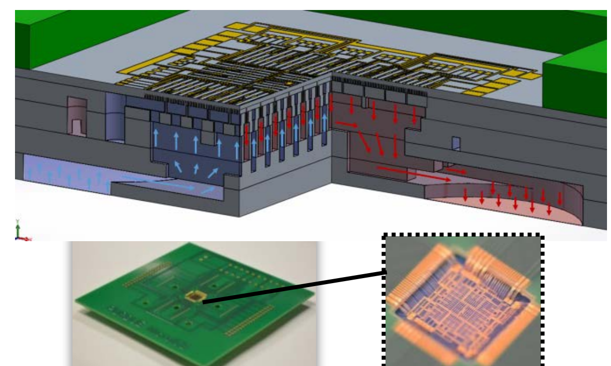
Active electronically scanned arrays (AESA) house hundreds of high-power amplifiers (HPA); next-gen GaN HPA devices require transformative high-heat-flux embedded cooling strategies.

DARPA *'Thermal Ground Planes'*



Radio Frequency TGP
(Raytheon, **Purdue**, GTRI, Thermacore)

DARPA *'ICECool Fundamentals'*

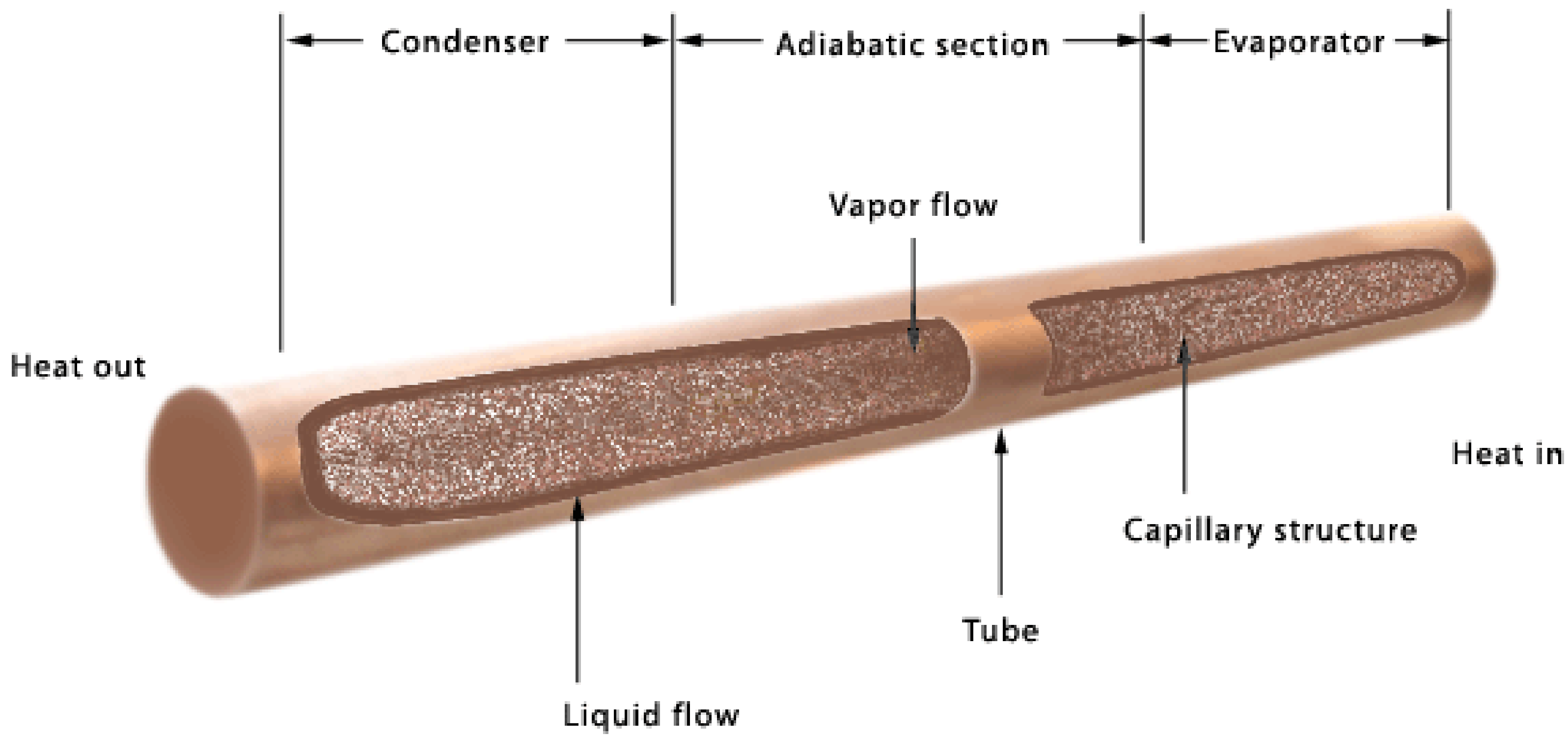


Hierarchical Manifold Microchannel Array
(**Purdue**, Raytheon, IBM)

Vapor Chamber and Heat Pipe Spreaders

Justin A Weibel: jaweibel@purdue.edu
https://engineering.purdue.edu/CTRC

Principles of Heat Pipe Operation



Passive Operation

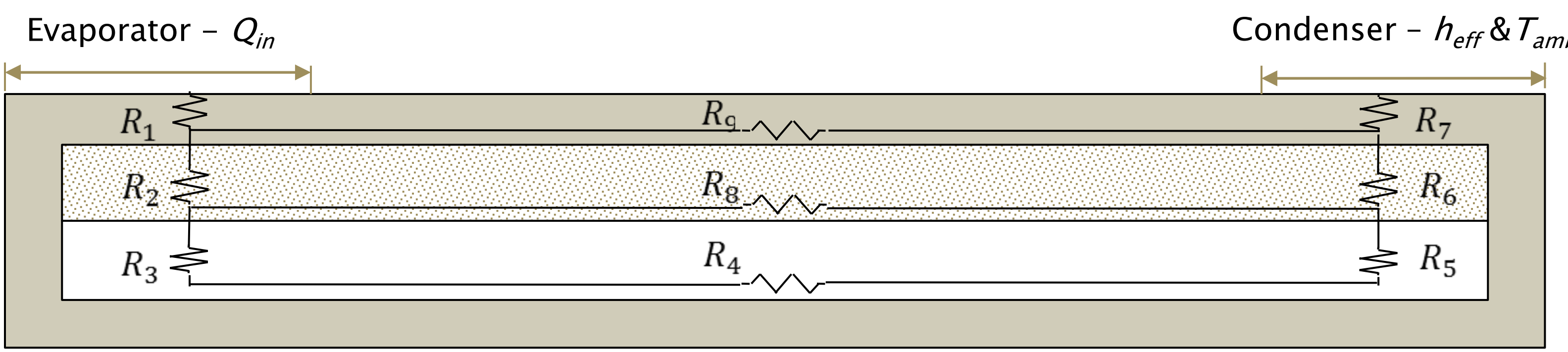
Low ΔT

Operate against gravity

Reliable

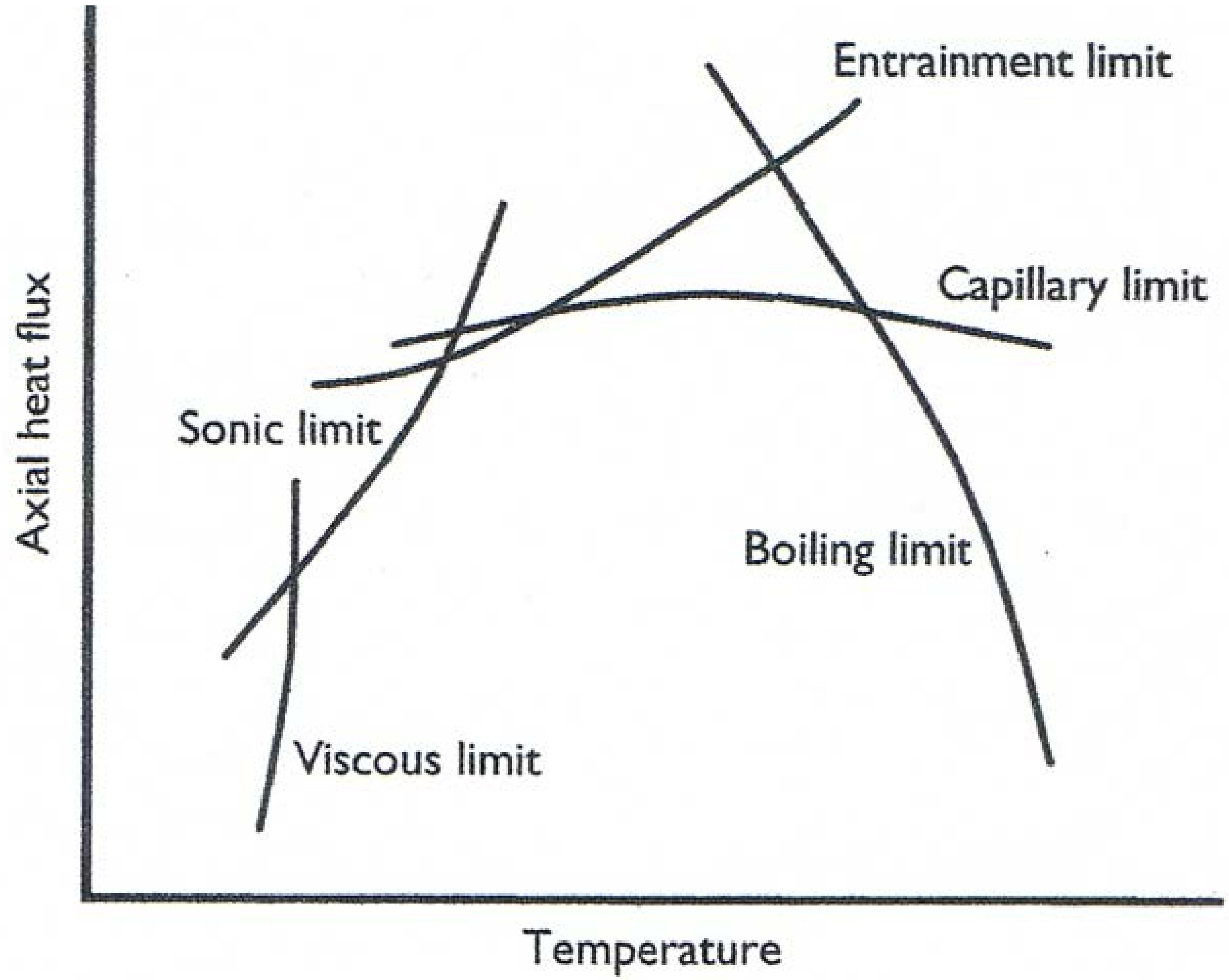
http://www.aavid.com/product-group/heatpipe/operate

Heat Transfer Performance



- Thermal Resistances
  - R_1, R_7 - Conduction through the heat pipe wall
  - R_2, R_6 - Conduction through the wick
  - R_3, R_5 - Liquid-vapor phase change resistance
  - R_4 -Lateral vapor saturation pressure/temperature drop
  - R_8,R_9 - Lateral conduction in wall/wick
- Effective Conductivity
  - 100,000 W/mK possible
  - Not an intrinsic property; does not account for operation limits; meant for higher level system models after experimental validation

Heat Pipe Transport Limits



Reay and Kew, Heat Pipes (2006)

Heat Pipe Transport Limits

COOLING TECHNOLOGIES RESEARCH CENTER

Transport Limit	What it is	Why it happens	Possible Solution
Capillary	Sum of gravitational, liquid and vapor flow pressure drops exceed the capillary pumping head of the heat pipe wick structure	Heat pipe input power exceeds the design heat transport capacity of the heat pipe	Modify heat pipe wick structure design or reduce power input
Sonic	Vapor flow reaches sonic velocity (choking) when exiting heat pipe evaporator resulting in a constant heat pipe transport power and large temperature gradients	Power/temperature combination, too much power at low operating temperature	Typically only a problem at start-up. The heat pipe carries a set power and the large ΔT self-corrects as the heat pipe warms up. Increase vapor core area.
Entrainment /Flooding	High velocity vapor flow prevents condensate from returning to evaporator (liquid tear-off)	Heat pipe operating above designed power input or at too low an operating temperature	Increase vapor space diameter or operating temperature
Boiling	Film boiling in heat pipe evaporator initiates	High radial heat flux causes film boiling resulting in heat pipe dryout and large thermal resistances	Use a wick with a higher heat flux capacity or spread out the heat load
Viscous	Viscous forces prevent vapor flow in the heat pipe, vapor pressure drop can’t exceed absolute vapor pressure	Heat pipe operating below recommended operating temperature, operating near freezing point	Increase heat pipe operating temperature or find alternative working fluid

Adapted from S. D. Garner, Heat Pipes for Electronics Cooling Applications (1996)

Capillary Limit

COOLING TECHNOLOGIES RESEARCH CENTER

$$P_c \geq \Delta P_l + \Delta P_v + \Delta P_g$$

- At steady operation, the following pressures balance
  - Capillary pumping pressure, P_c
  - Liquid pressure drop, ΔP_l
  - Vapor pressure drop, ΔP_v
  - Gravitational pressure drop, ΔP_g
- There is a maximum capillary pressure that can be developed by a given liquid-wick combination. The required P_C must never exceed $P_{C,max}$ that can be provided by the wick. Else, dryout is caused in the evaporator.

PURDUE UNIVERSITY

Fluid Figures of Merit

- Neglecting gravitational and vapor pressure drop

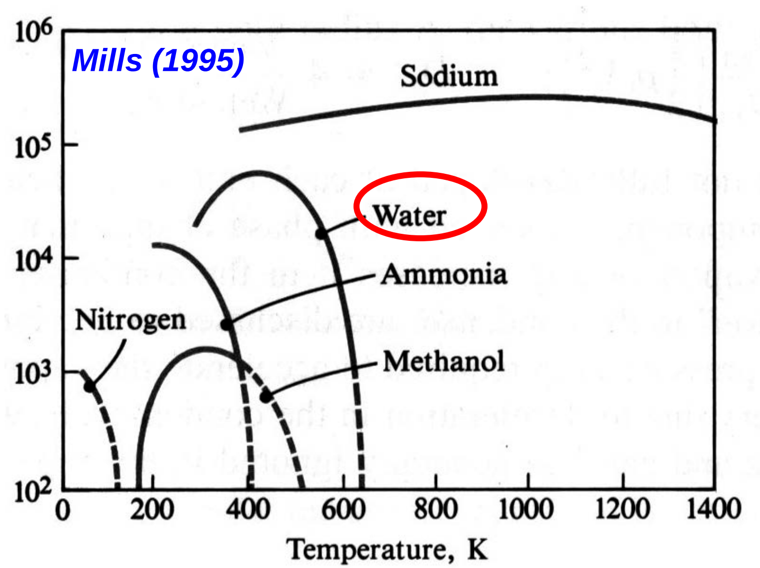
$$P_c \geq \Delta P_l + \cancel{\Delta P_v} + \cancel{\Delta P_g}$$

- The maximum heat transport becomes

$$\dot{Q}_{max} = \left[\boxed{\frac{\rho \sigma h_{fg}}{\mu}}\right]\left[\frac{KA}{L_{eff}}\right]\left[\frac{2}{r_{eff}}\right]$$

- Fluid property dependent Figure of Merit

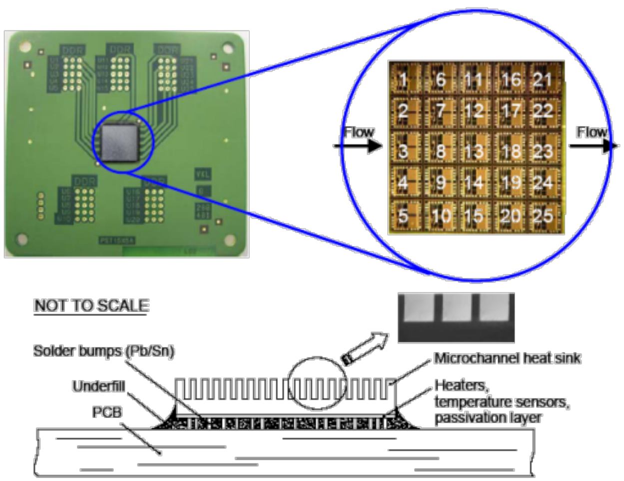
$$M = \frac{\rho \sigma h_{fg}}{\mu}$$



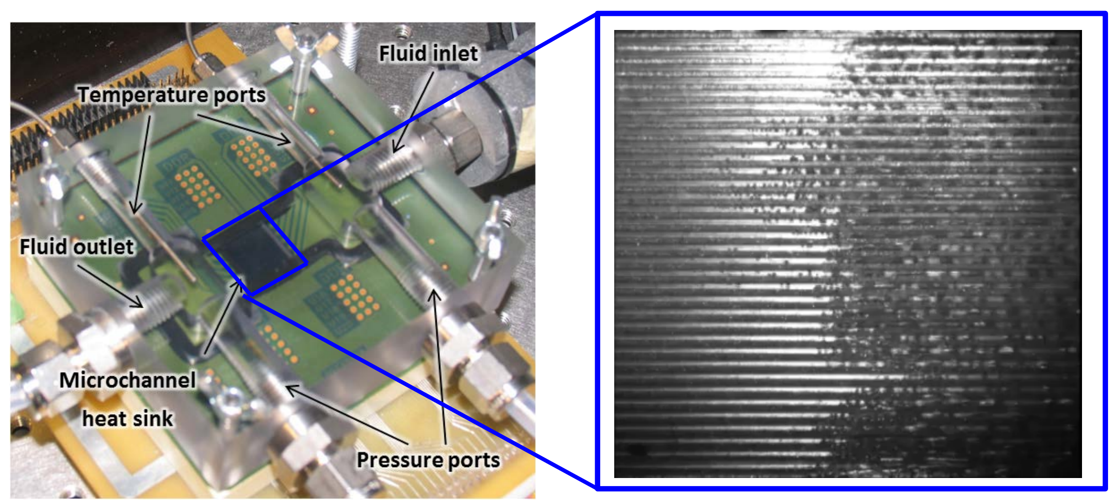
Microchannel Heat Sinks

Why Microchannels?

- Amongst the most effective of the high-heat-flux heat transfer technologies
- Flow boiling operation has promise for: (1) lower pumping power, (2) improved fluid temperature uniformity, and (3) higher heat transfer coefficients (compared to single-phase liquid cooling)
- Why now? → new manufacturing techniques and emerging application needs



Microchannel Test Chip



High-Speed Visualization of Flow Boiling in a Microchannel

A Simple Calculation

As revealed by simple scaling, large convection coefficients are associated with small channel dimensions ($h \propto 1/d$).

Example:

Consider a microchannel of width 60 μm and depth 300 μm (i.e., D_h = 100 μm). With a uniform wall temperature assumption, in fully developed laminar flow, the Nu_{Dh} is 4.8. Even for a dielectric liquid with k = 0.06 W/mK,

$$h = (Nu_{Dh}\, k_f) / D_h = 2880\ W/m^2K$$

For water ($k \approx 0.6$ W/mK) the value of h is 10 times higher

Single-Phase Heat Transfer

COOLING TECHNOLOGIES RESEARCH CENTER

Reference	Correlations	Conditions: Geometry	Conditions: Flow regime	Valid range
Kays and Crawford	$Nu_{fd} = 8.235\left(1-1.883/\alpha+3.767/\alpha^2-5.814/\alpha^3+5.361/\alpha^4-2/\alpha^5\right)$ (1)	Rectangular	Fully developed flow (Both hydrodynamically and thermally)	Re < 2200
Incropera and DeWitt	Sieder-Tate correlation $Nu = 1.86\left(\mathrm{Re}\,\mathrm{Pr}\,D/L\right)^{1/3}\left(\frac{\mu_f}{\mu_w}\right)^{0.14}$ (2)	Circular	Simultaneously developing flow	Re < 2200
Stephan and Preußer	Stephan correlation $Nu = 3.657+\frac{0.0677\left(\mathrm{Re}\,\mathrm{Pr}\,D/L\right)^{1.33}}{1+0.1\mathrm{Pr}\left(\mathrm{Re}\,D/L\right)^{0.3}}$ (3)	Circular	Simultaneously developing flow (Constant wall temperature)	0.7 < Pr < 7 or Re Pr *D/L* < 33 for Pr > 7
Stephan and Preußer	Stephan correlation $Nu = 4.364+\frac{0.086\left(\mathrm{Re}\,\mathrm{Pr}\,D/L\right)^{1.33}}{1+0.1\mathrm{Pr}\left(\mathrm{Re}\,D/L\right)^{0.83}}$ (4)	Circular	Simultaneously developing flow (Constant wall heat flux)	0.7 < Pr < 7 or Re Pr *D/L* < 33 for Pr > 7
Incropera and DeWitt	Hausen correlation $Nu = 3.66+\frac{0.19\left(\mathrm{Re}\,\mathrm{Pr}\,D/L\right)^{0.8}}{1+0.117\left(\mathrm{Re}\,\mathrm{Pr}\,D/L\right)^{0.467}}$ (5)	Circular	Hydrodynamically developed, but thermally developing laminar flow (Constant wall temperature)	Re < 2200
Shah and London	$Nu = \begin{cases} 1.953\left(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L}\right)^{1/3} & \left(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L}\right)\geq 33.3 \\ 4.364+0.0722\,\mathrm{Re}\,\mathrm{Pr}\frac{D}{L} & \left(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L}\right)<33.3 \end{cases}$ (6)	Circular	Hydrodynamically developed, but thermally developing laminar flow (Constant wall heat flux)	–
Kacac et al.	Hausen correlation $Nu = 0.116\left(\mathrm{Re}^{2/3}-125\right)\mathrm{Pr}^{1/3}\left[1+\left(D/L\right)^{2/3}\right]\left(\frac{\mu_f}{\mu_w}\right)^{0.14}$ (7)	Circular	Transitional flow	2200 < Re < 10000
Incropera and DeWitt	Dittus-Boelter correlation $Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}$ (8)	Circular	Fully developed turbulent flow	Re > 10000
Incropera and DeWitt	Petukhov correlation $Nu = \frac{\left(f/8\right)\mathrm{Re}\,\mathrm{Pr}}{K+12.7\left(f/8\right)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)}$ (9) $K = 1.07+\frac{900}{\mathrm{Re}}-\frac{0.63}{1+10\mathrm{Pr}}$	Circular	Fully developed turbulent flow	Re > 10000
Gnielinski	Gnielinski correlation $Nu = \frac{\left(f/8\right)\left(\mathrm{Re}_D-1000\right)\mathrm{Pr}}{1+12.7\left(f/8\right)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)}$ (10)	Circular	Both the transitional and fully developed turbulent flows	3000 < Re < 5 × 10^6

PURDUE UNIVERSITY

Single-Phase Conclusions

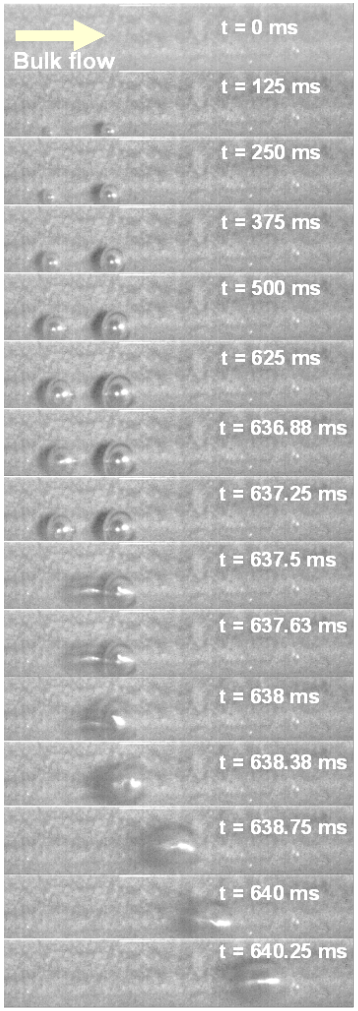
Cooling Technologies Research Center

- Conventional theory offers reliable predictions for the flow characteristics in microchannels
- Heat transfer measurements are conducted with an emphasis on correctly matching the entrance and thermal conditions when comparing to conventional correlations developed for large-sized channels
- Numerical predictions obtained based on a classical, continuum approach with carefully matched entrance and boundary conditions are in good agreement with the data

Two Phase Flow in Microchannels

- Flow boiling is attractive because it
  - can dissipate higher heat fluxes
  - requires lower coolant mass
  - has more uniform wall temperature

- Flow boiling mechanisms in microchannels are not fully understood
- Most available flow maps based on macroscale channel flow are not applicable
- Results obtained for single microchannel cannot be simply extrapolated for multiple microchannels
- Lack of generally accepted models/correlations



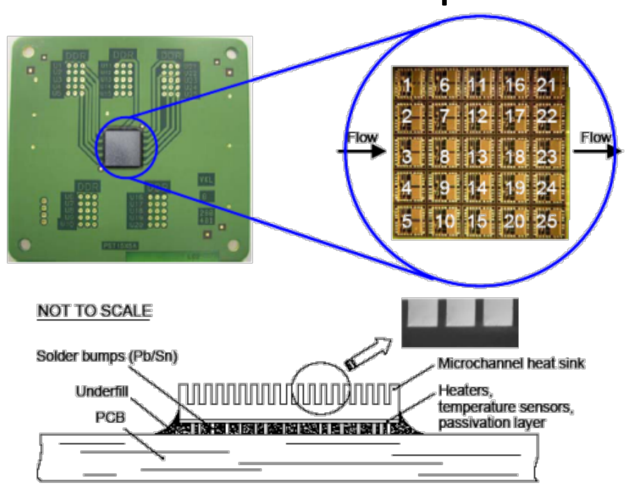
Liu et al., *IJHMT* **48**, 2005; *JHT* **127**, 2005

Two Phase Flow in Microchannels

Cooling Technologies Research Center

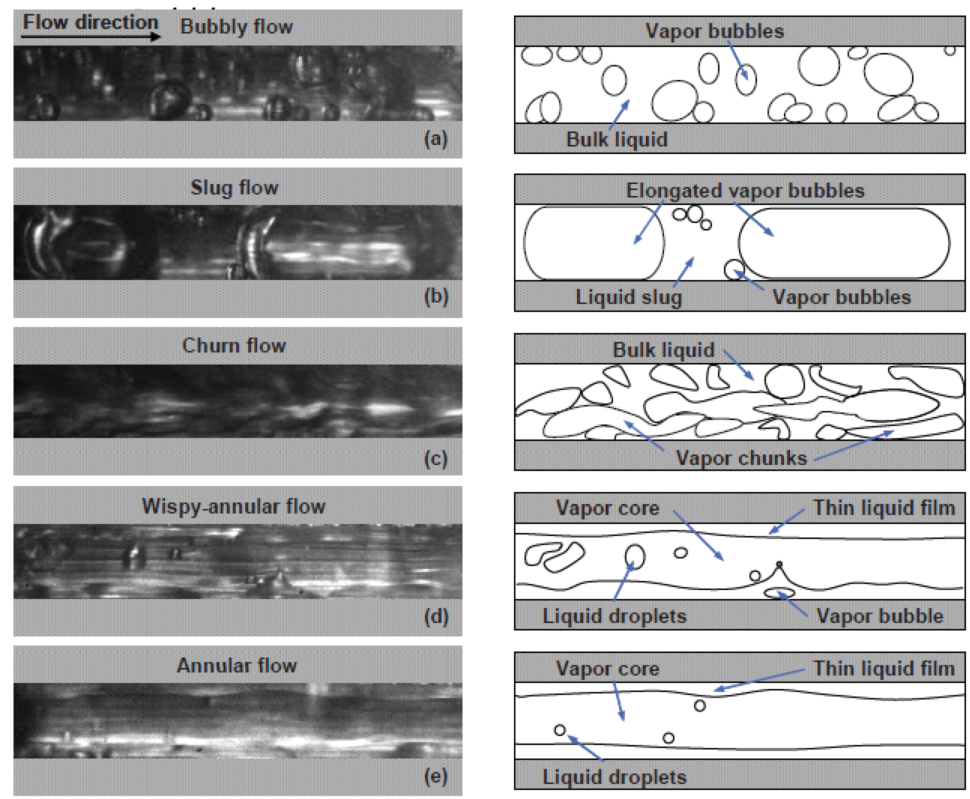
- In-situ measurement of local wall temperature & flux, high-speed visualizations
- Flow regime determination and regime-based modeling for dielectric fluids
- Instability, hysteresis, surface roughness, geometry, operating condition effects

Microchannel Test Chip



Channel width:	100-5850 µm
Channel depth:	100-400 µm
Hydraulic diameter:	100-750 µm
Mass flux:	225-1450 kg/m^{2}s

Flow Boiling Regimes in Microchannels



Chen, Garimella, *IEEE CPT* 2007; *IJMF* 2006

Bertsch, Groll, Garimella, *IJHMT* 2008; *NMTE* 2008

Harirchian, Garimella, *IJHMT* 2008, 2010; *IJMF* 2008, 2009; *JEP* 2010

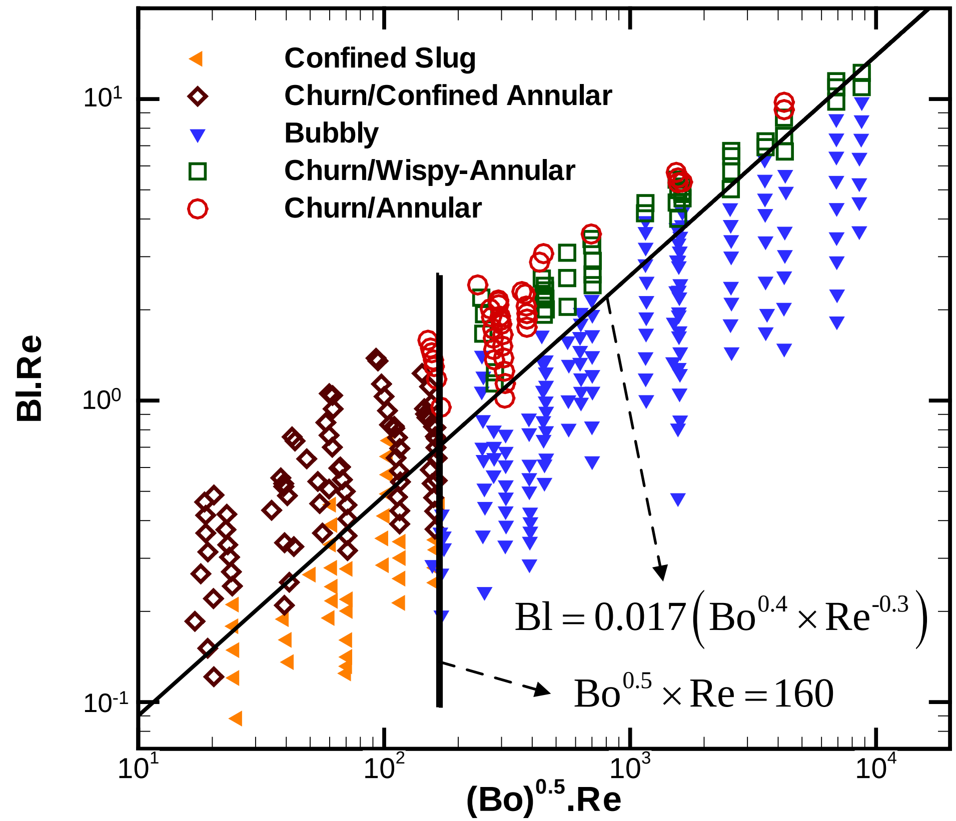
Patel, Harirchian, Garimella, *IPACK* 2011

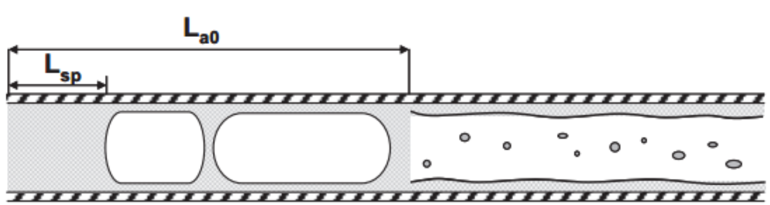
Purdue University

Flow Regime Prediction and Models

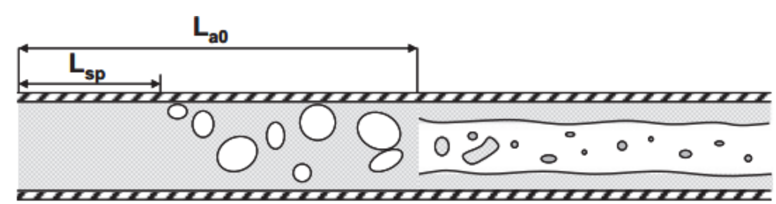
- $G - x$ and $j_f - j_g$ flow regime maps depend on channel dimensions
- Comprehensive four-region flow regime map represents data for a wide range of channel dimensions, mass fluxes, and heat fluxes
- Validation of flow-regime based theoretical heat transfer and pressure drop models against empirical data

Comprehensive Flow Regime Map





"Microchannels": Confined Flow Regimes



"Macrochannels": Unconfined Flow Regimes

Harirchian, Garimella, *IJHMT* 2010, *IJMF* 2012

Current Challenges/Needs

COOLING TECHNOLOGIES RESEARCH CENTER

- ✓ Embedded sensors for system control
- ✓ Measurement techniques for characterization of thin liquid films that govern heat transport in small channels
- ✓ Development of numerical methods to allow tractable direct simulation of flow boiling in small channels
- ✓ Electro-thermal co-design for development of high-performance, intrachip heat sink technologies
- ✓ Characterization of the impact of static/dynamic instabilities and flow maldistribution on performance

PURDUE UNIVERSITY