Project 2
Convection Cooling with Extended Surfaces

Prof. Kurabayashi

ME 495

Upcoming Events

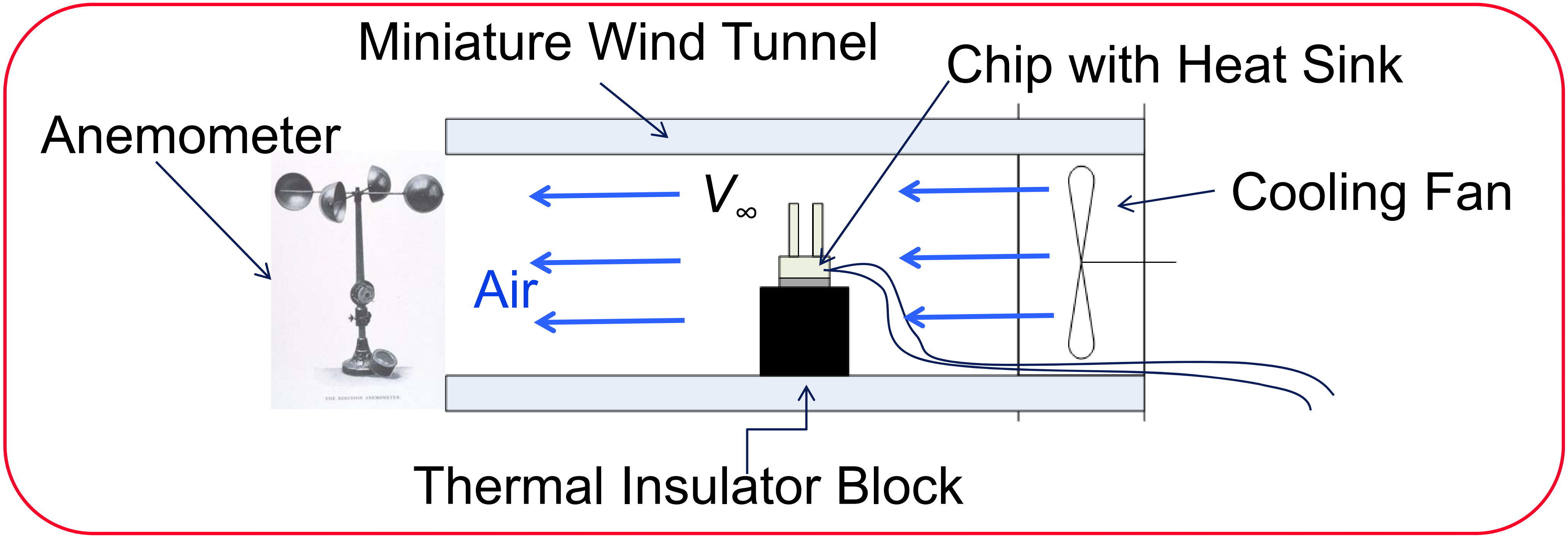
- Tuesday-Thursday 2/12-14: Lab 1 Due in section
- Friday, 2/15: Convection Cooling with Extended (Kurabayashi)
- Monday, 2/18: Convection Cooling with Extended Surfaces (Kurabayashi)
- Friday, 2/22: Uncertainly Analysis (Kurabayashi)

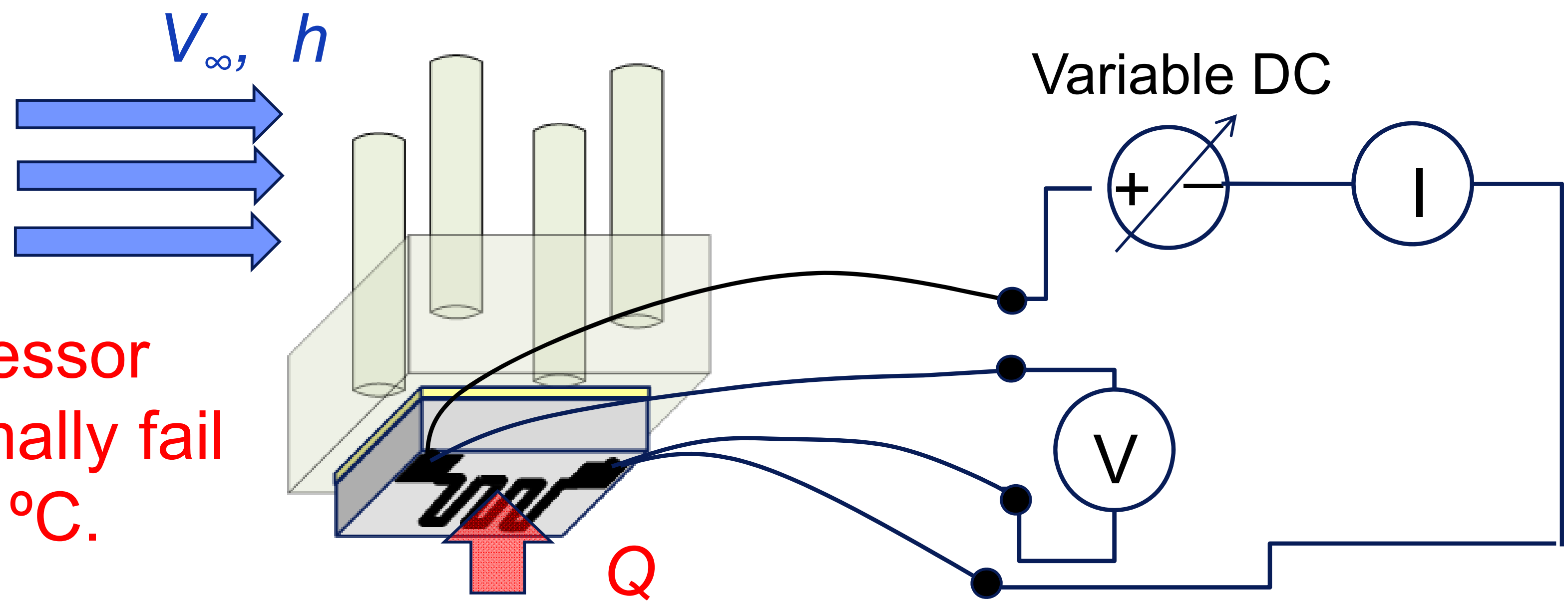
Week 2 & 3 : Microprocessor Cooling System Design

Objectives

- Conduct experimental studies to evaluate the cooling capacity of the conventional microprocessor chip packaging and air-cooling technology.
- Develop a heat transfer model to analyze the performance of the microprocessor air chip cooling system and validate the model by experiments.
- Provide a thermal management strategy (fin design, heat sink orientation, air velocity, and etc.).
- Comment on a new method(s) for achieving cooling of high-speed, high-power microprocessor chips for mobile devices as your future study.

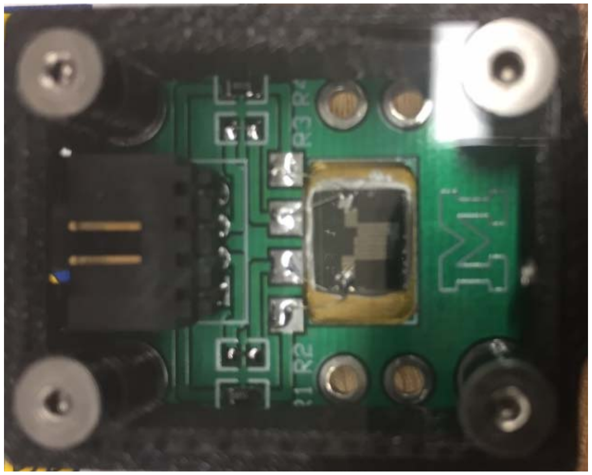
Microprocessor Chip Cooling System

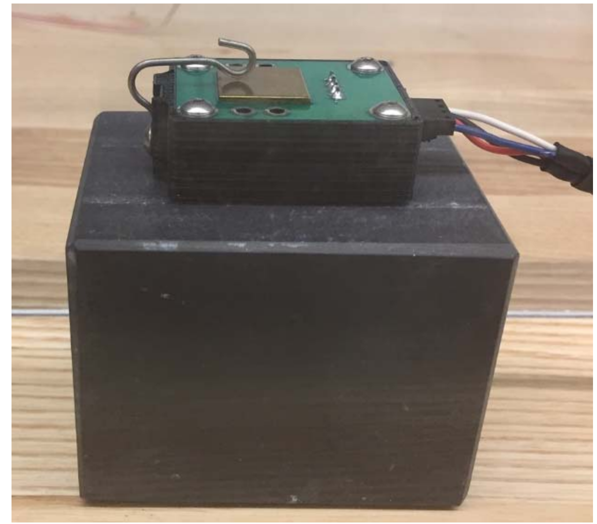


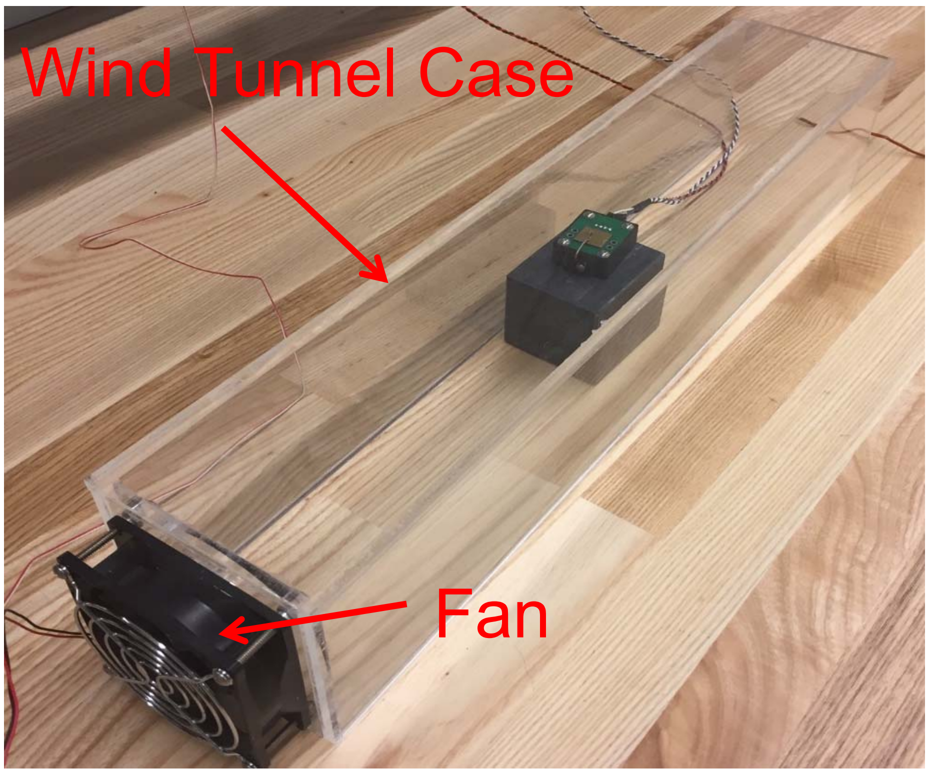


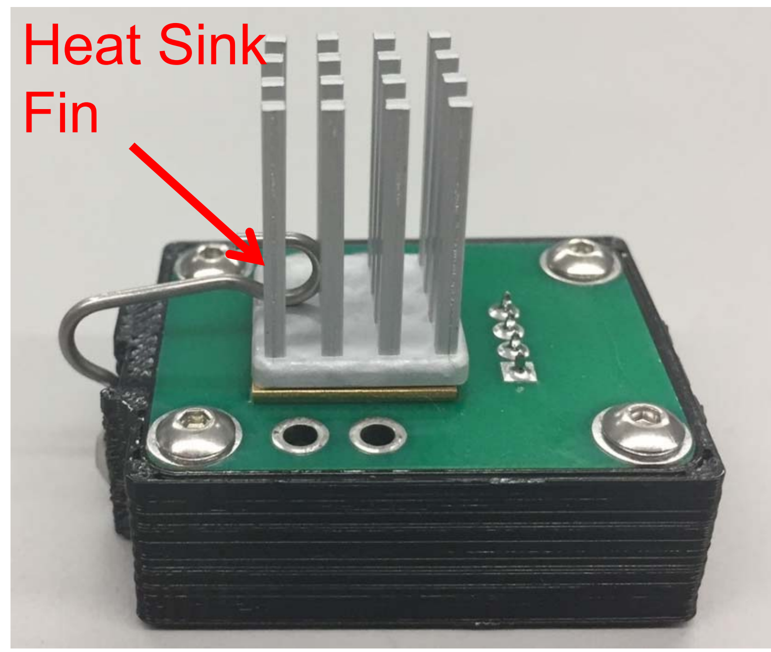
Microprocessor chips normally fail at T = 150 ºC.

Chip Packaging



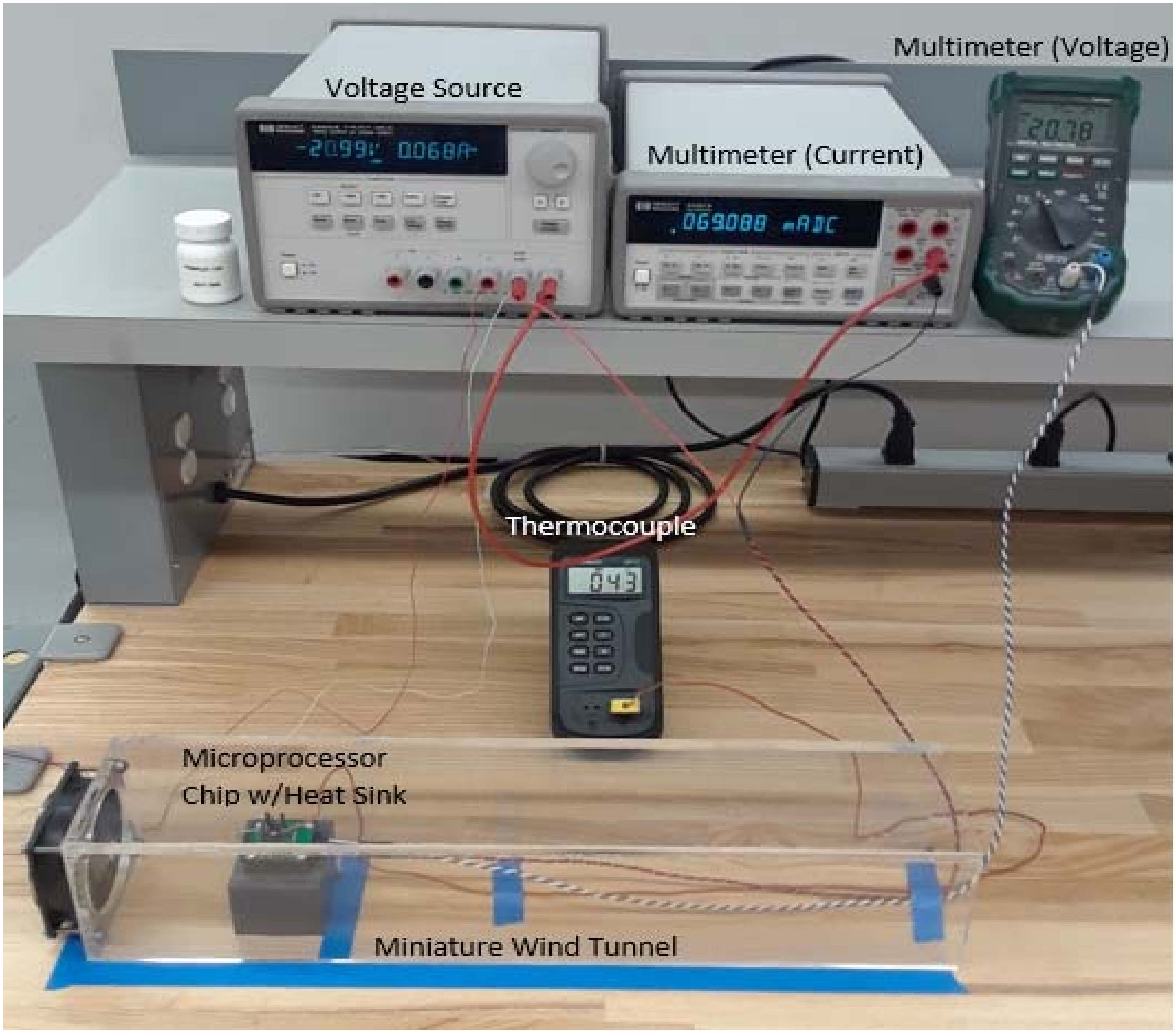




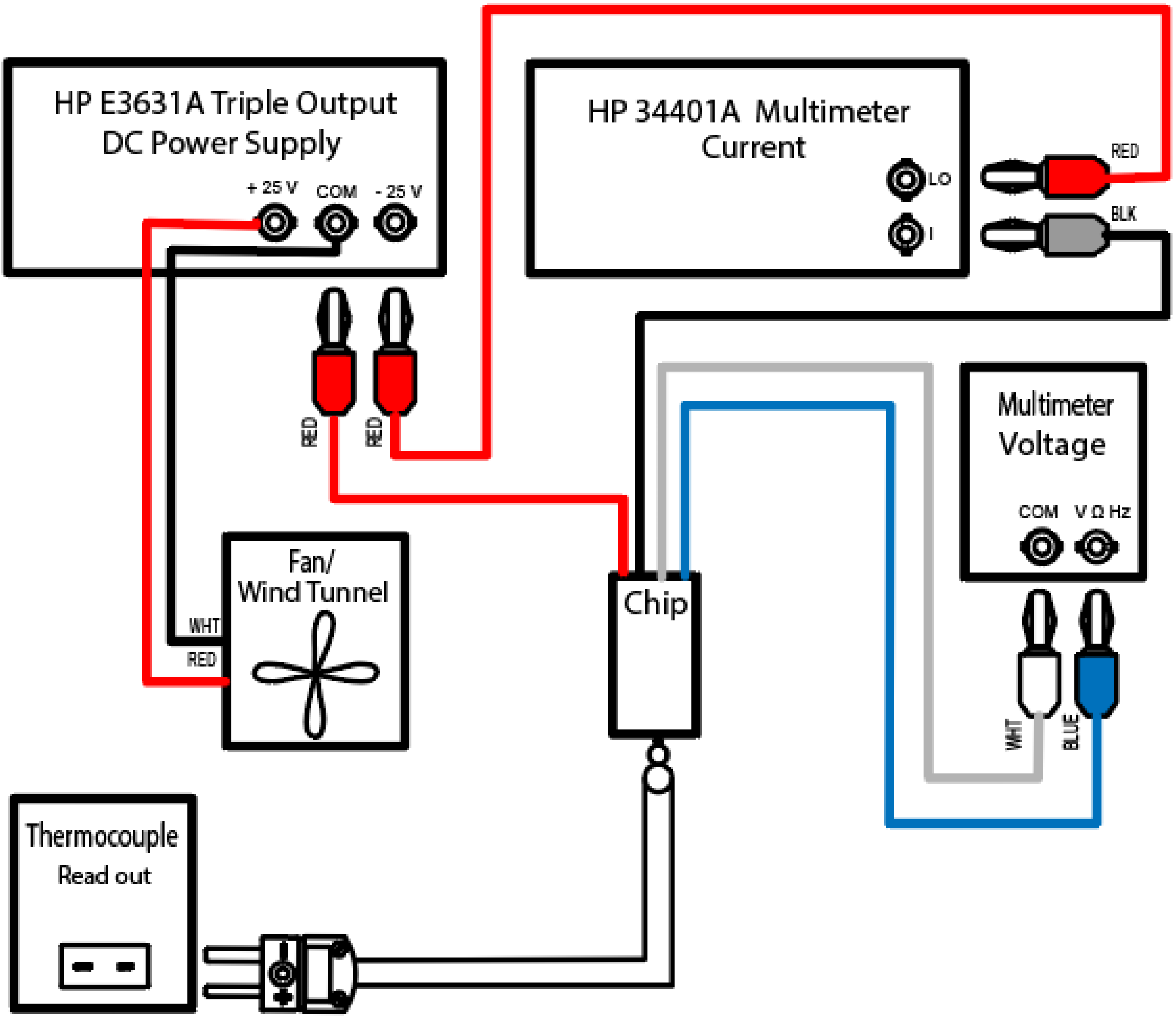


Katsuo Kurabayashi / ME
University of Michigan College of Engineering

Chip Packaging



Wiring Diagram



Tasks for the next two weeks

Experiments:

1. Calibrate the air velocity V_∞ as a function of power input to the cooling fan $P_{fan} = V_{fan} * I_{fan}$ using the anemometer.
2. Measure the chip temperature T_{chip} for varying cooling fan power P_{fan} (V_{fan} = 6, 8, 10, and 12 V) with V = 25 V applied to the on-chip heater.
3. Measure thermal contact resistance.
4. Plot the chip-to-ambient thermal resistance = $(T_{chip} - T_\infty)/Q_H$ vs. the air velocity V_∞ .

Modeling and Validation of Model:

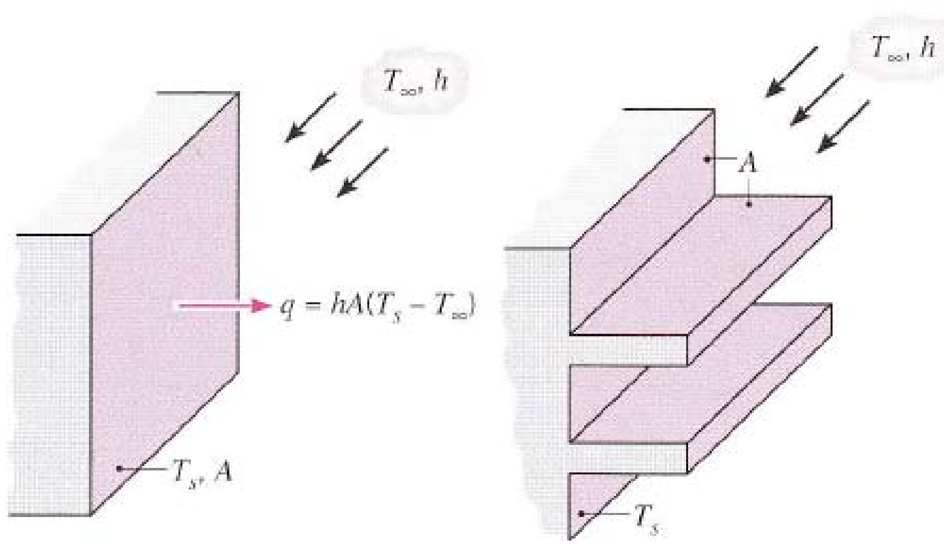
1. Develop a heat transfer model for the microprocessor cooling system employed in this lab and predict the chip-to-ambient thermal resistance as a function of the air velocity V_∞ and compare the theoretical prediction with the experimental results. Does your model agree well with experiments? Can you use the validated model for predictions? Can you perform additional simulations?

* Note: Perform uncertainty analysis for all the measurements in 1, 2, and 3.

Heat Transfer over Extended Surfaces

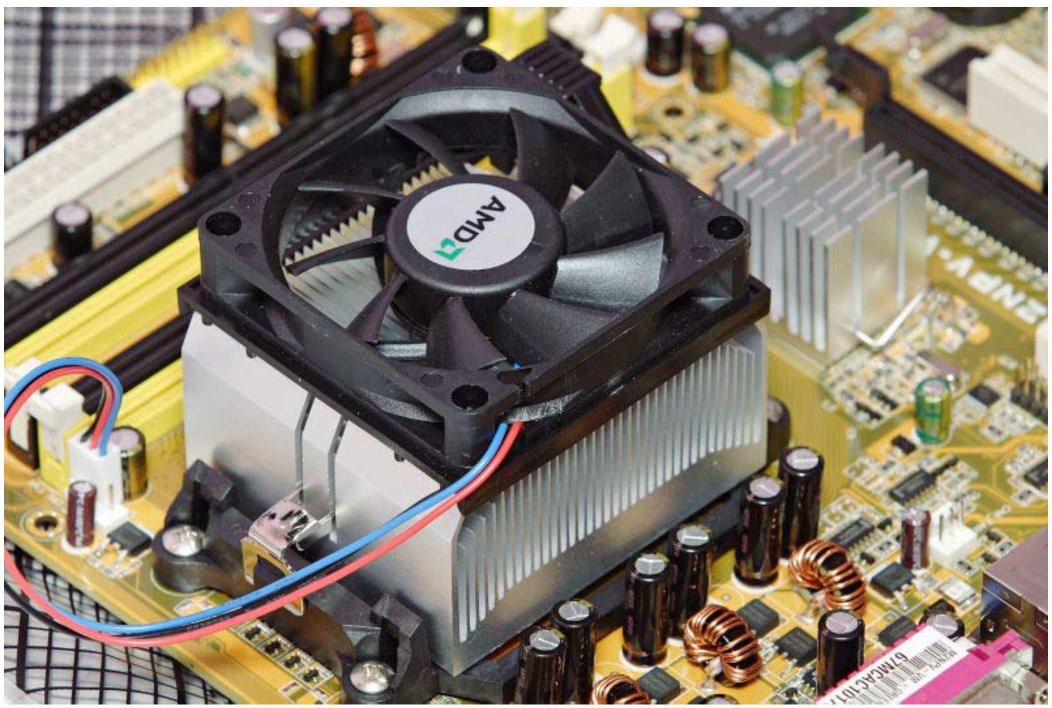
Heat Transfer Enhancement with Extended Surfaces

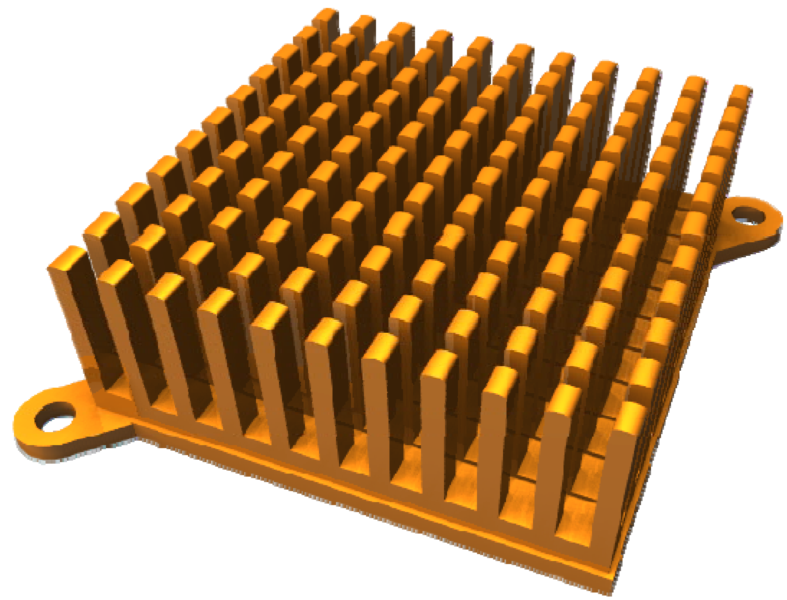
- There are many situations that require *h* to be maximized to the maximum possible value.
- But this requires increasing a flow rate of fluid at the expense of more pumping power.
- The heat transfer rate may be increased by increasing the surface area across which the convection occurs .
- Employing fins that extended from a flat surface is one of the most economical ways to increase the heat transfer rate.



Microprocessor Chip Cooling Using Heat Sinks

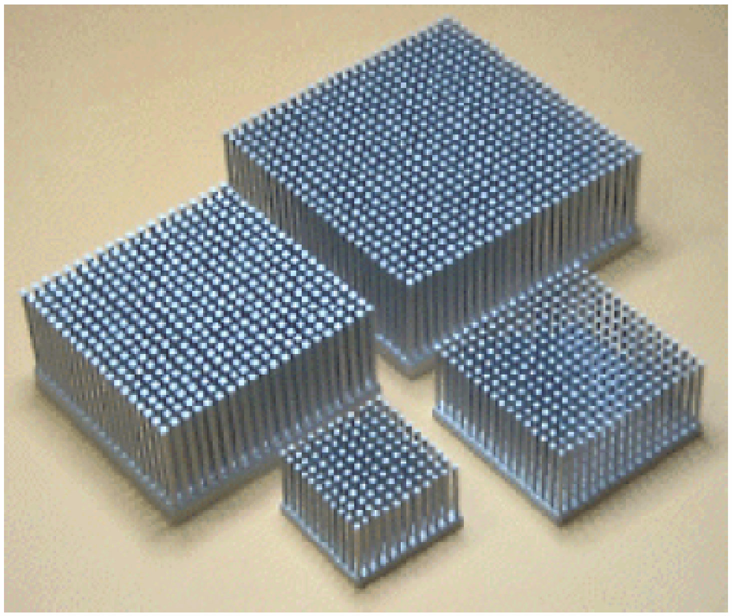
- Heat sinks with fins are used in a wide range of applications wherever efficient heat dissipation is required.
- Heat sinks are widely used in electronics, and have become almost essential to modern integrated circuits for their thermal management.

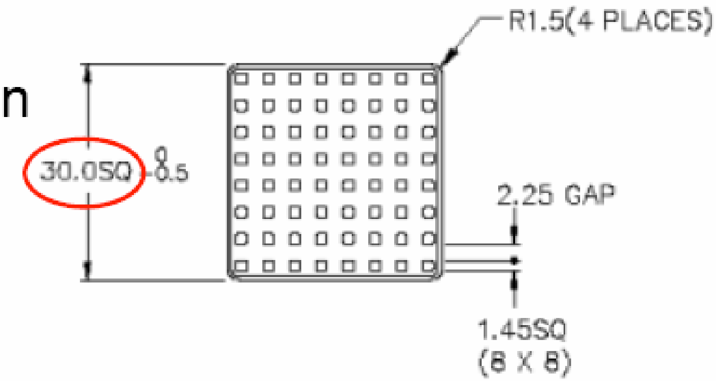


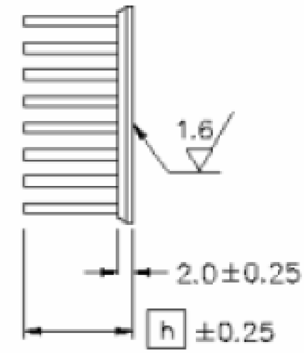


Alpha Novatech S-Series Heat Sinks

- Dense pin fin heat sinks
- Suitable for forced convection



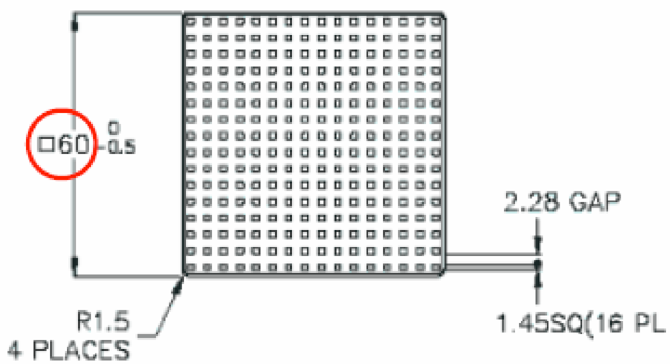




HEAT SINK
MATERIAL : A 6063
FINISH : CLEAR ANODIZE

MODEL	HEIGHT h	WEIGHT (grams)
S1530–7W	7	6.8
S1530–10W	10	7.6
S1530–15W	15	9.4
S1530–20W	20	11.2
S1530–25W	25	13.0

Dimensions : mm

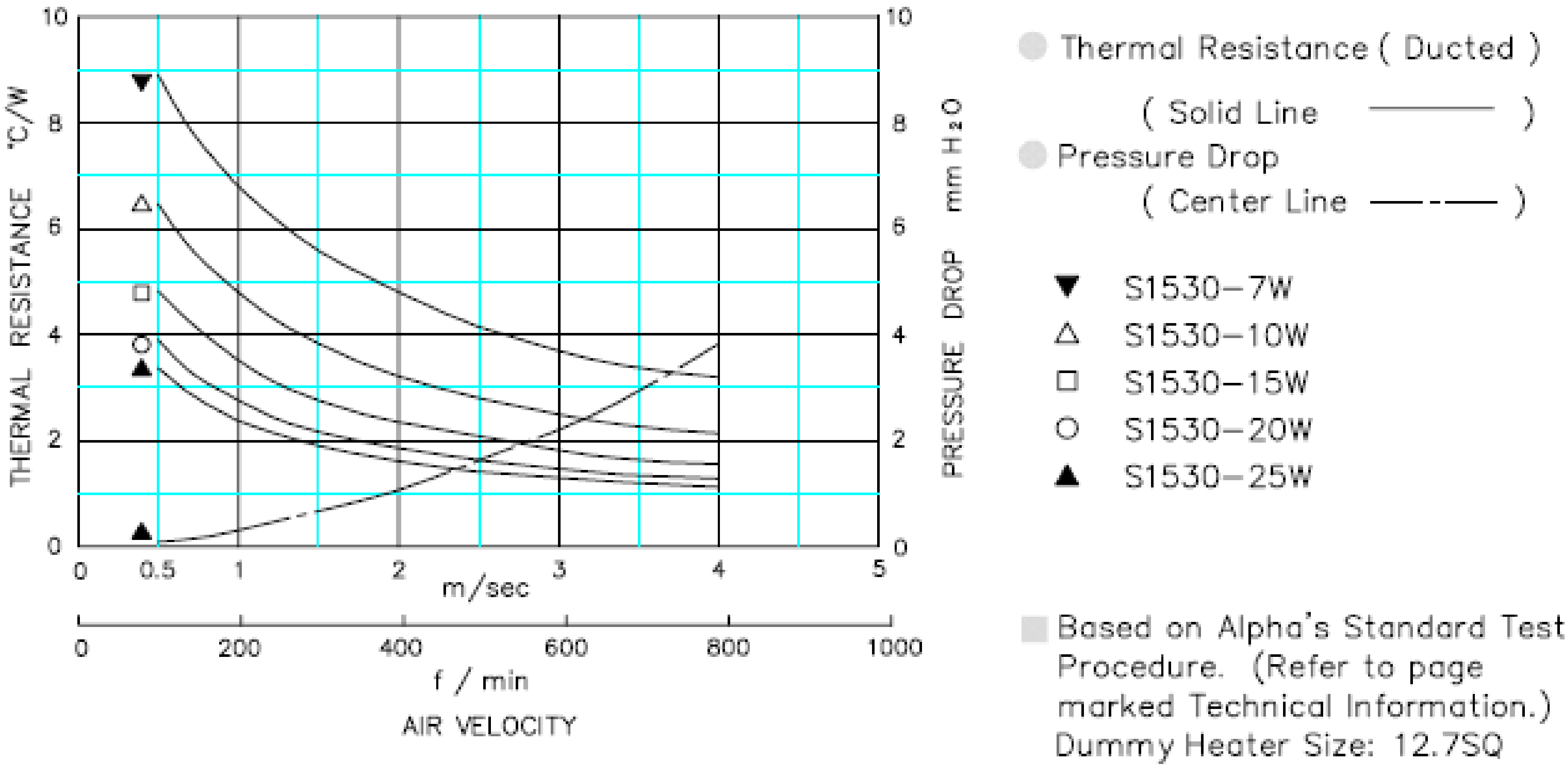


MODEL	HEIGHT h	Thickness t	WEIGHT (grams)
S1560–15W	15	3.0	44.5
S1560–20W	20	3.0	50.5
S1560–25W	25	3.5	61.0
S1560–30W	30	4.0	70.5

Dimensions : mm

See web site for prices: http://www.alphanovatech.com/

Alpha Novatech S-Series Heat Sinks



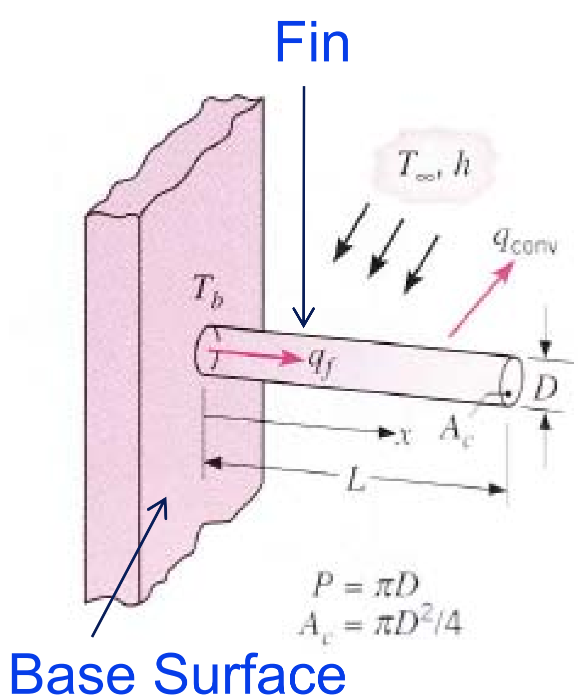
Heat Sink Material (Aluminum Alloy 6063) Material Properties

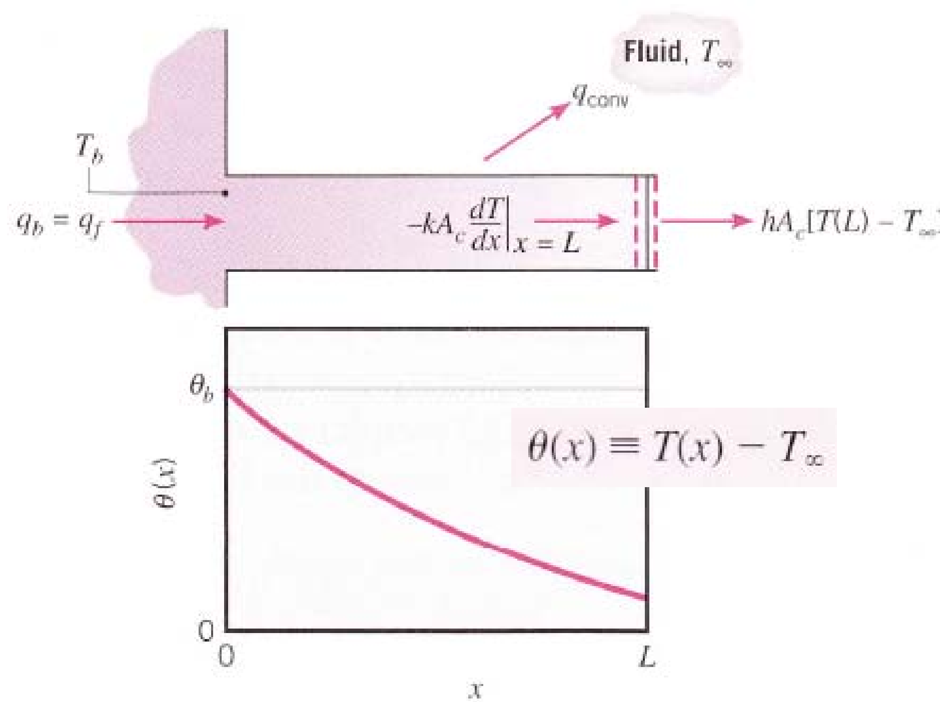
Property	Value
Density	2.71 g/cm^3
Melting Point	600 ºC
Electrical Resistivity	0.035 x 10^{-4} W m
Thermal Conductivity	180 W/m K
Young's Modulus	67 GPa
Thermal Expansion	2.3 x 10^{-6} /K

Heat Transfer from a Surface with Fins

- **There are two heat transfer mechanisms from a surface with a fin that is subjected to a bulk motion of cooling fluid:**
  - **(1) Convection from the base surface**
  - **(2) Conduction along the fin and convection from the fin surface**

$$\frac{d^2T}{dx^2} - \frac{hP}{kA_c}(T - T_\infty) = 0$$

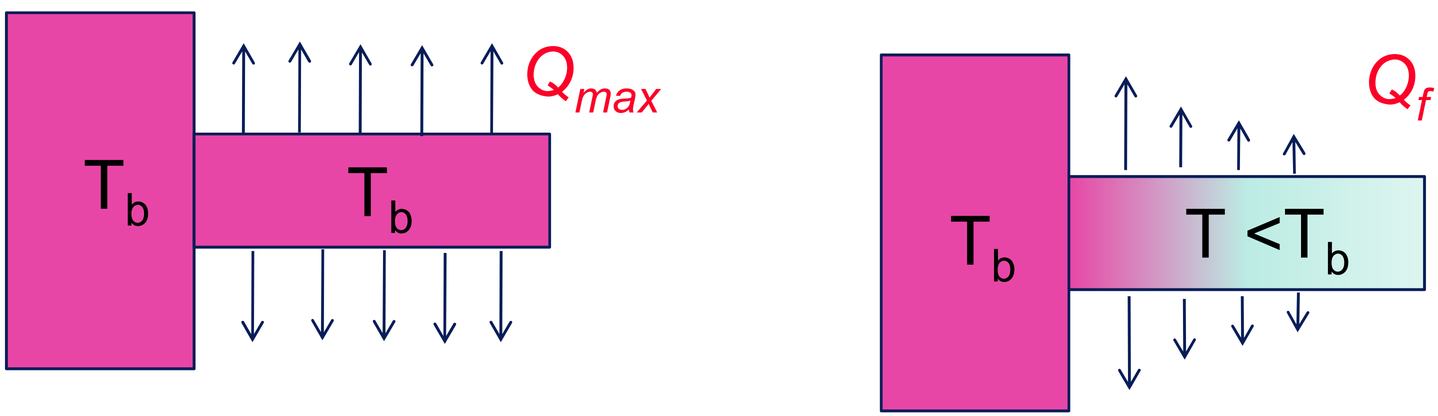




Fin Efficiency (1)

- **Fin efficiency is defined as the ratio of the actual heat energy rate dissipated from the fin surface Q_f to the maximum possible heat energy rate dissipated from the fin surface with the thermal conductivity of the fin material assumed to be infinite Q_{max}.**

$$\eta_f \stackrel{\text{def}}{=} \frac{Q_f}{Q_{max}} = \frac{Q_f}{hA_f[T_b - T_\infty]}$$

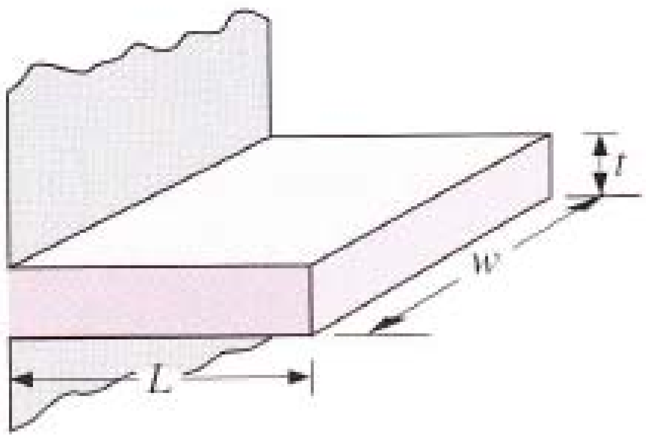


Fin Efficiency (2)

$$\eta_f = \frac{\tanh(mL_c)}{mL_c} \quad \text{where } m = \sqrt{\frac{hP}{kA_c}}$$

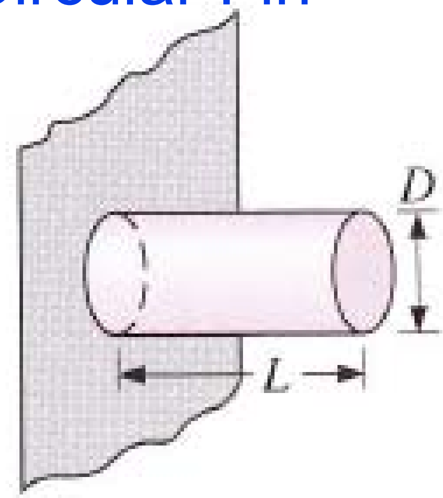
(Adiabatic Tip)

Straight Fin



$$P = 2(w + t)$$
$$A_c = wt$$
$$A_f = 2wL_c$$
$$L_c = L + t/2$$

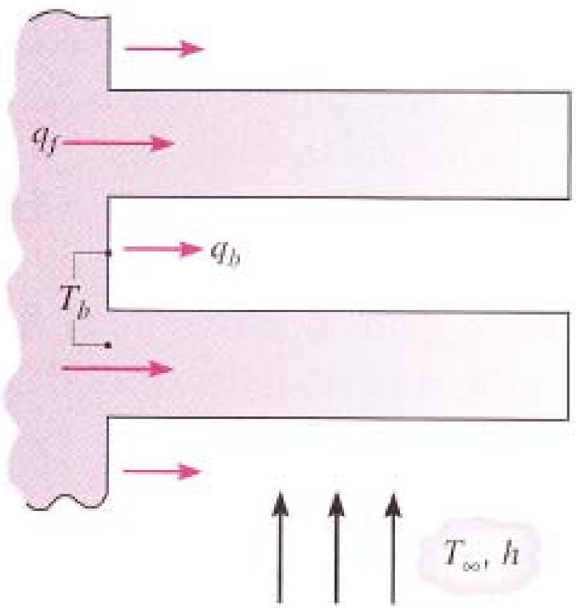
Circular Fin



$$P = \pi D$$
$$A_c = \pi D^2/4$$
$$A_f = \pi D L_c$$
$$L_c = L + D/4$$

Thermal Circuit Model for Heat Sink Surfaces

- **Thermal circuit model accounting for heat transfer from both (1) the base surface and (2) the fin surfaces.**



Parallel Circuit Model

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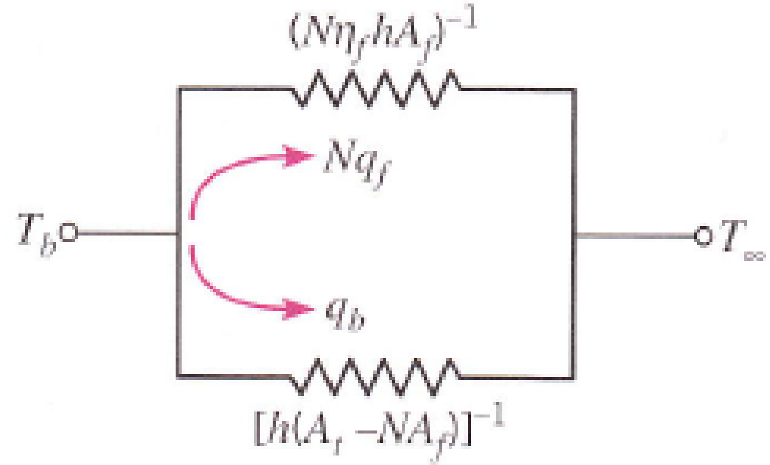

Single Resistance Model

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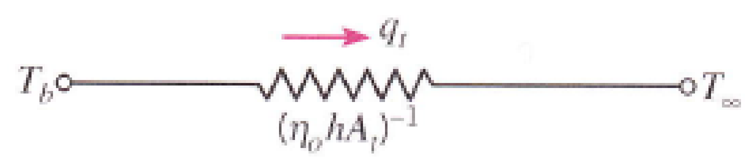


$$\eta_o = 1 - \frac{NA_f}{A_t}(1 - \eta_f)$$

N: total number of fins

$q_t = q_b + Nq_f$:total dissipated heat power

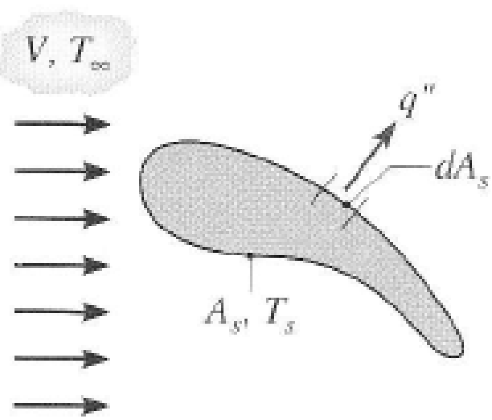
$A_t = A_b + NA_f$:total surface area

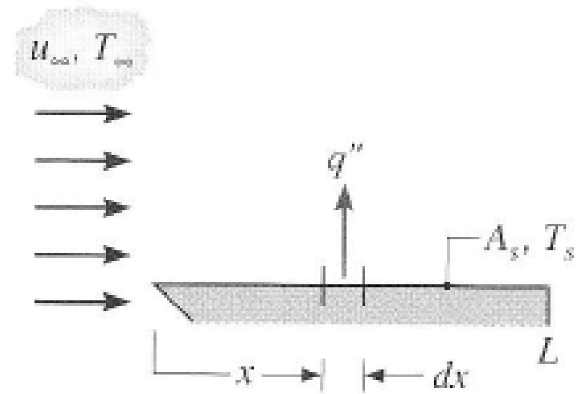
Convection over a Body Surface

- Fluid motion over a body surface can vary from position to position.
- As a result, convection coefficient h can be a function of position.
- Total heat transfer rate is given by

$$q'' = h(T_s - T_\infty)$$

$$q = \int_{A_s} q''\, dA_s \qquad q = (T_s - T_\infty)\int_{A_s} h\, dA_s$$





- Average convection coefficient

$$\bar{h} = \frac{1}{L}\int_0^L h\, dx$$

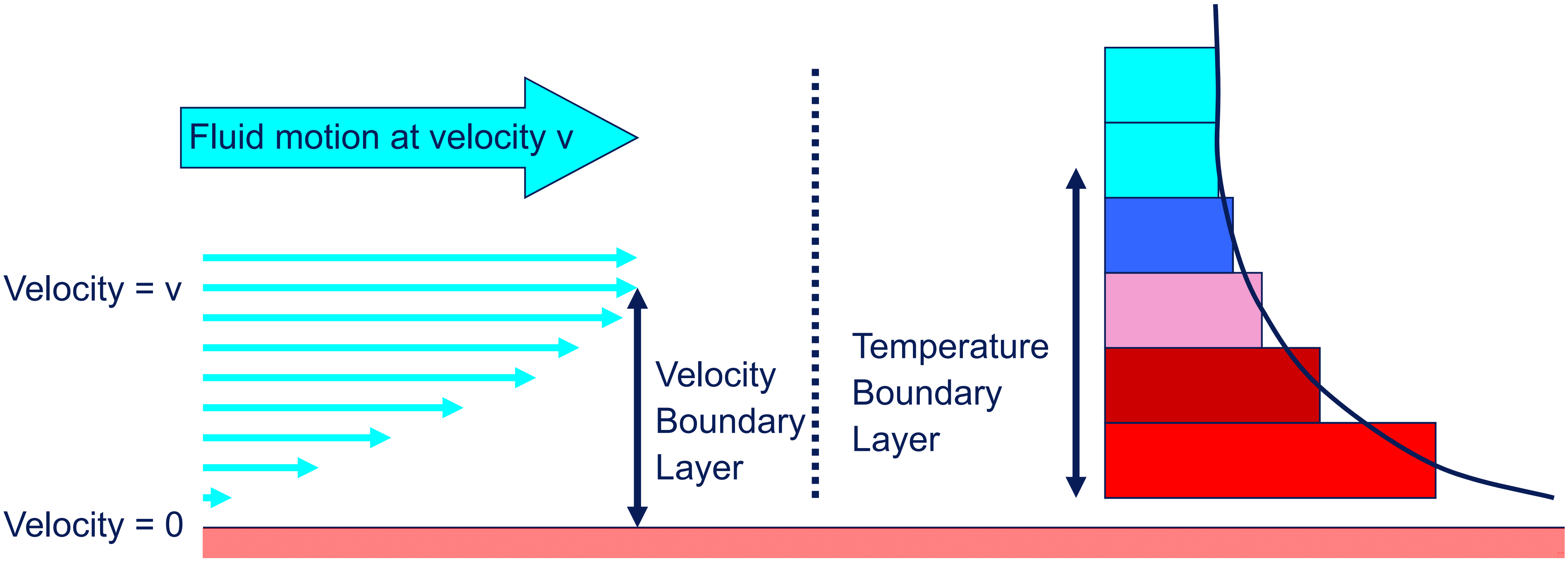
$$q = \bar{h}A_s(T_s - T_\infty)$$

Boundary Layers in Convection

Velocity Boundary Layer

Thermal Boundary Layer

COLD FLUID, TEMP = T_c



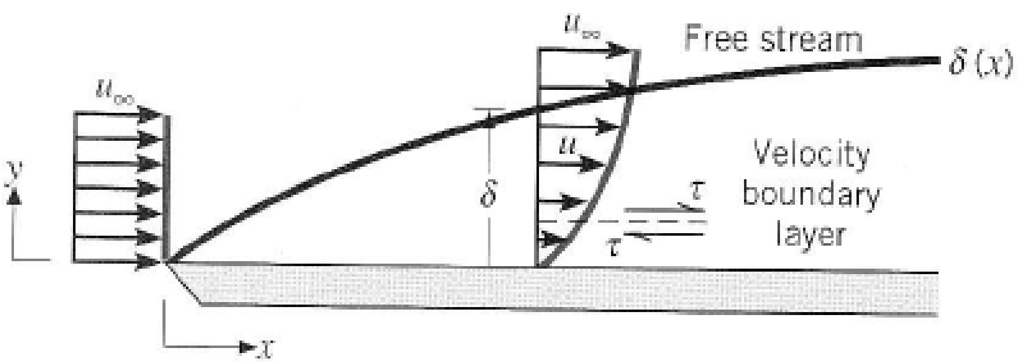
HOT SURFACE, TEMP = T_H

Heat transfer at the surface takes place by conduction, not convection, because the fluid velocity is zero. Convection becomes more important away from the surface.

The change in temperature is largest close to the surface. The temperature boundary layer may not be the same thickness as the velocity boundary layer, but the rate of change of temperature depends on the rate of change of fluid velocity.

Key Parameters for Convection Boundary Layers

- **Velocity Boundary Layer**



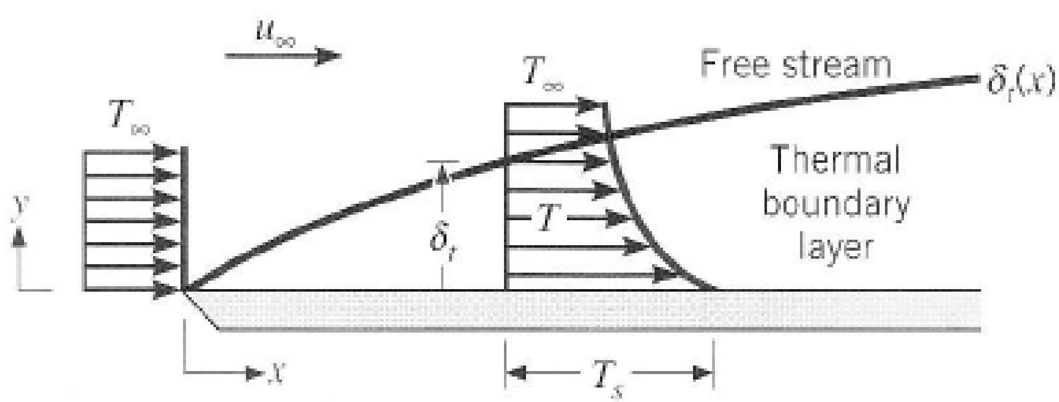
$$\tau_s = \mu \frac{\partial u}{\partial y}\bigg|_{y=0}$$

Shear Stress

$$C_f \equiv \frac{\tau_s}{\rho u_\infty^2/2}$$

Friction Coefficient

- **Thermal Boundary Layer**



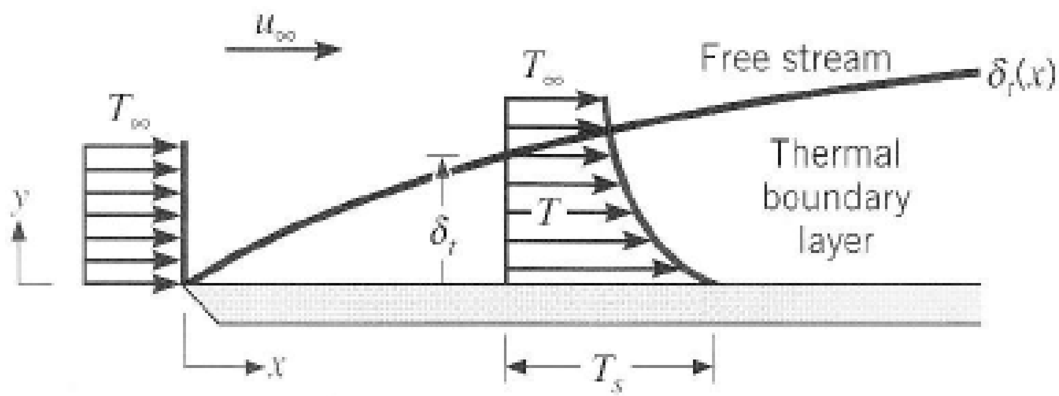
$$q_s'' = -k_f \frac{\partial T}{\partial y}\bigg|_{y=0}$$

Fourier's Law

$$h = \frac{-k_f \,\partial T/\partial y\big|_{y=0}}{T_s - T_\infty}$$

Convection Heat Transfer Coefficient

Laminar Flow Over a Flat Plate



Heat Transfer from an Isothermal plate

$$\overline{Nu}_L = \frac{\bar{h}L}{k_f} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \qquad \mathrm{Re}_L = \frac{\rho V L}{\mu};\ \mathrm{Pr} = \frac{\upsilon}{\alpha}$$

Heat Transfer from a Plate with a Constant Heat Flux

$$\overline{Nu}_L = \frac{\bar{h}L}{k_f} = 0.680\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3}$$

The Nusselt number is the non-dimensional temperature gradient at the surface, providing a measure of the convection heat transfer occurring at the surface.

Fundamentals of Heat and Mass Transfer, Incropera and Dewitt, Seventh Edition

Laminar Flow Over a Cylinder

$$\overline{Nu}_D = \frac{\bar{h}D}{k_f} = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{1/3}$$

$$\mathrm{Re}_L = \frac{\rho V D}{\mu};\ \ \mathrm{Pr} = \frac{\upsilon}{\alpha}$$

TABLE 7.2 Constants of Equation 7.52 for the circular cylinder in cross flow [11, 12]

Re_D	C	m
0.4–4	0.989	0.330
4–40	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.027	0.805

Heat Correction factors for other shapes

$$\overline{Nu}_D = \frac{\bar{h}D}{k_f} = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{1/3}$$

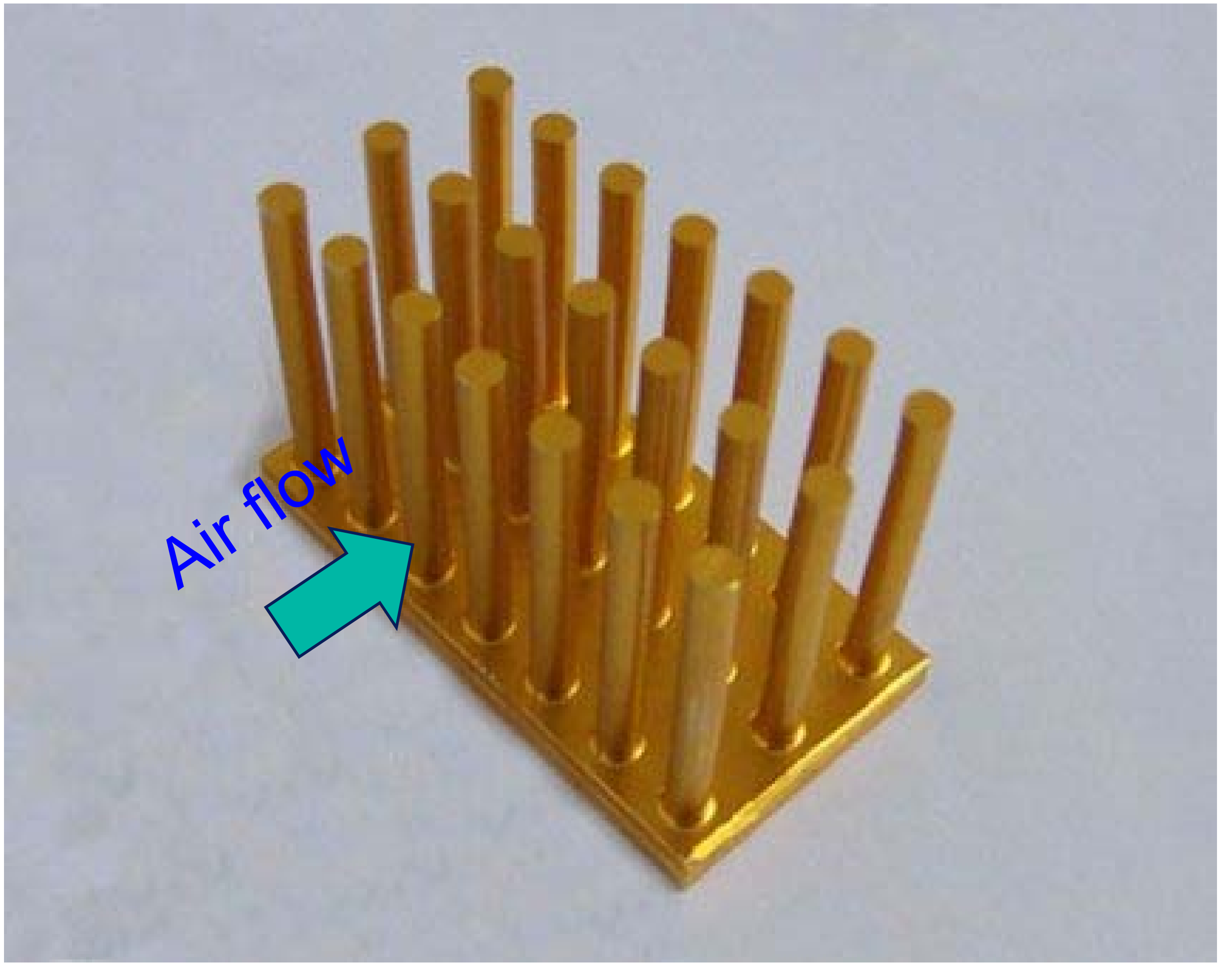
TABLE 7.3 Constants of Equation 7.52 for noncircular cylinders in cross flow of a gas [13]

Geometry	Re_D	C	m
Square ($V \rightarrow$, D)	5×10^3–10^5	0.246	0.588
Square ($V \rightarrow$, D)	5×10^3–10^5	0.102	0.675
Hexagon ($V \rightarrow$, D)	5×10^3–1.95×10^4	0.160	0.638
	1.95×10^4–10^5	0.0385	0.782
Hexagon ($V \rightarrow$, D)	5×10^3–10^5	0.153	0.638
Vertical plate ($V \rightarrow$, D)	4×10^3–1.5×10^4	0.228	0.731

Fundamentals of Heat and Mass Transfer, Incropera and Dewitt, Sixth Edition

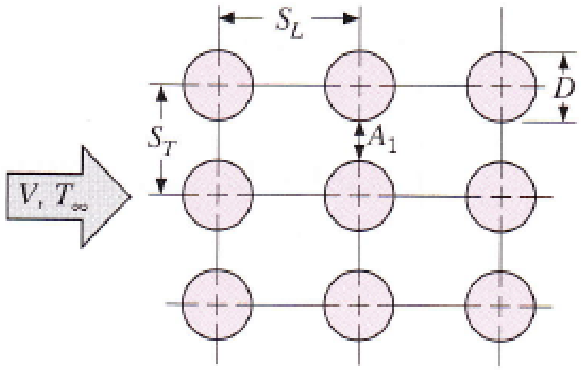
Heat Transfer Coefficient for a collection of Fins

Let's consider a flow across banks of tubes to model the flow across the heat sink fins.

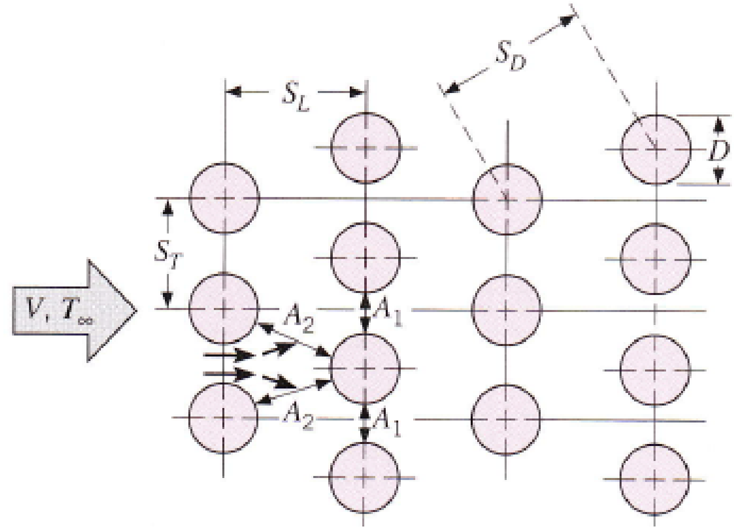


How can we predict the value of *h*?

Maximum Reynolds Number for flow across fins



Aligned arrangement



Staggered arrangement

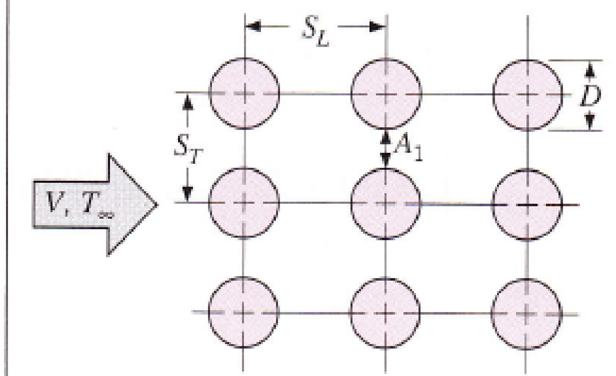
$$V_{max} = \frac{S_T}{S_T - D} V$$

$$V_{max} = \frac{S_T}{2(S_D - D)} V$$

$$Re_{D,max} \equiv \frac{\rho V_{max} D}{\mu}$$

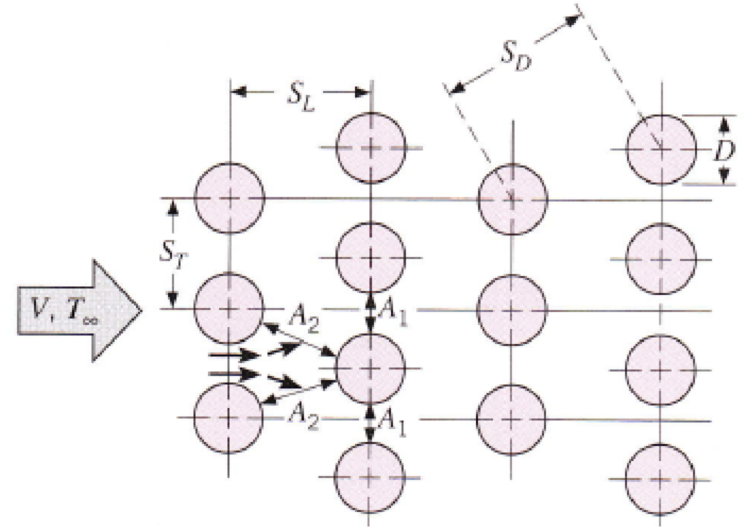
How can we predict the value of *h*?

Maximum Reynolds Number for flow across fins



Aligned arrangement

$$V_{\max} = \frac{S_T}{S_T - D} V$$



Staggered arrangement

$$V_{\max} = \frac{S_T}{2(S_D - D)} V$$

$$Re_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu}$$

How can we predict the value of *h*?

We can obtain Nusselt number from empirical relations.

$$\overline{Nu}_D = C\, Re_{D,\max}^m\, Pr^{0.36} \left(\frac{Pr}{Pr_s}\right)^{1/4}$$

$$\left[\begin{array}{l} N_L \geq 20 \\ 0.7 < Pr < 500 \\ 1000 < Re_{D,\max} < 2 \times 10^6 \end{array}\right]$$

N_L = total number of fins

Pr_s = Prandtl Number at the average fin temperature T_s.

All properties except Pr_s to be evaluated at the mean temperature $T_m = (T_i + T_O)/2$

Configuration	$Re_{D,\max}$	C	m
Aligned	10–10^2	0.80	0.40
Staggered	10–10^2	0.90	0.40
Aligned	10^2–10^3	Approximate as a single	
Staggered	10^2–10^3	(isolated) cylinder	
Aligned ($S_T/S_L > 0.7$)[a]	10^3–2×10^5	0.27	0.63
Staggered ($S_T/S_L < 2$)	10^3–2×10^5	$0.35(S_T/S_L)^{1/5}$	0.60
Staggered ($S_T/S_L > 2$)	10^3–2×10^5	0.40	0.60
Aligned	2×10^5–2×10^6	0.021	0.84
Staggered	2×10^5–2×10^6	0.022	0.84

[a]For $S_T/S_L < 0.7$, heat transfer is inefficient and aligned tubes should not be used.

T_i : air temperature at the inlet of the tube bank

T_o : air temperature at the exit of the tube bank

Fundamentals of Heat and Mass Transfer, Incropera and Dewitt, Seventh Edition

How can we predict the value of *h*?

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$$\overline{Nu}_D = C\,Re_{D,\max}^m\,Pr^{0.36}\left(\frac{Pr}{Pr_s}\right)^{1/4}$$

$$\left[\begin{array}{l} N_L \geq 20 \\ 0.7 < Pr < 500 \\ 1000 < Re_{D,\max} < 2 \times 10^6 \end{array}\right]$$

If $N_L < 20$

$$\overline{Nu}_D\big|_{(N_L<20)} = C_2\,\overline{Nu}_D\big|_{(N_L\geq 20)}$$

C_2 value

N_L	1	2	3	4	5	7	10	13	16
Aligned	0.70	0.80	0.86	0.90	0.92	0.95	0.97	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.92	0.95	0.97	0.98	0.99

Fundamentals of Heat and Mass Transfer, Incropera and Dewitt, Seventh Edition

How can we predict the value of *h*?

We can obtain Nusselt number from empirical relations.

Note: In your calculation, you need to perform iterative calculations to obtain the true value of *h*.

Use properties of air at $T_s = T_f$ $(T_b + T_\infty)/2$ for Pr_s

(1) Make a guess for the value of T_m to calculate $Re_{D,max}$ and Pr

(2) Calculate *h* from Nu and obtain the new value of T_m from

$$T_o = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m} C_p}\right)$$

(3) Repeat the process until the value of h converges.

How can we predict the value of *h*?

We can obtain Nusselt number from empirical relations.

Note: In your calculation, you need to perform iterative calculations to obtain the true value of *h*.

Use properties of air at $T_s = T_f \ (T_b + T_\infty)/2$ for Pr_s

(1) Make a guess for the value of T_m to calculate $Re_{D,max}$ and Pr

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$$T_o = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m} C_p}\right)$$

(3) Repeat the process until the value of h converges.

Alternatively,
instead of going through the iterative calculation, you could simply take temperature $T_m = (T_i + T_O)/2 \approx T_\infty$ → Verify it!

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Safety

1) The chip can get pretty hot

2) Do not supply more than 12 V to the fan