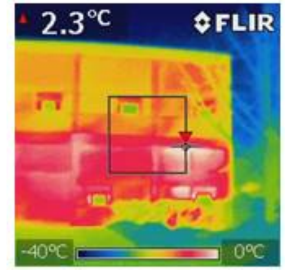
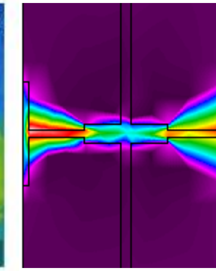
CAE 331/513
Building Science
Fall 2016





Week 3: September 6, 2016
Heat transfer in buildings: Finish convection, start radiation

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Last time

- Finished conduction
  - Transient conduction
  - Heat capacity
  - Thermal diffusivity

$$HCA = \rho LAC_p = \rho VC_p \qquad \alpha = \frac{k}{\rho C_p} \quad [\mathrm{m}^2/\mathrm{s}]$$

$$\boxed{\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{q}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t}}$$

- Convection
  - Natural vs. forced
  - Internal vs. external
  - Laminar vs. turbulent

$$q_{conv} = h_{conv}\left(T_{fluid} - T_{surface}\right) \qquad \left[\frac{\mathrm{W}}{\mathrm{m}^2}\right]$$

$$Q_{conv} = h_{conv}A\left(T_{fluid} - T_{surface}\right) \qquad [\mathrm{W}]$$

$$\mathrm{Nu} = \frac{hL_c}{k} \qquad \mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr}) \qquad \mathrm{Re}_x = \frac{\rho \mathcal{V} x}{\mu} = \frac{\mathcal{V} x}{\upsilon} \qquad \mathrm{Pr} = \frac{\mu C_p}{k}$$

Convective heat transfer coefficient, h_{conv}

- Convective heat transfer coefficients for buildings
  - From **Chapter 4** of the 2013 ASHRAE Handbook of Fundamentals:

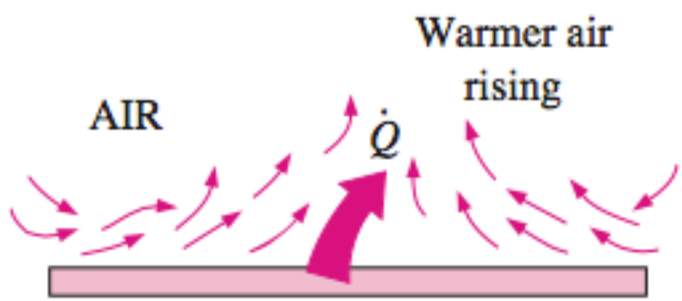
Table 8 Forced-Convection Correlations

I. General Correlation	$\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr})$		
II. Internal Flows for Pipes and Ducts: Characteristic length = D, pipe diameter, or D_h, hydraulic diameter.			
$\mathrm{Re} = \dfrac{\rho V_{avg} D_h}{\mu} = \dfrac{\dot{m} D_h}{A_c \mu} = \dfrac{Q D_h}{A_c \nu} = \dfrac{4\dot{m}}{\mu P_{wet}} = \dfrac{4Q}{\nu P_{wet}}$	where $\dot{m}$ = mass flow rate, Q = volume flow rate, P_{wet} = wetted perimeter, A_c = cross-sectional area, and ν = kinematic viscosity (μ/ρ).		
	$\dfrac{\mathrm{Nu}}{\mathrm{Re}\,\mathrm{Pr}^{1/3}} = \dfrac{f}{2}$	Colburn's analogy (turbulent)	(T8.1)
Laminar: Re < 2300	$\mathrm{Nu} = 1.86\left(\dfrac{\mathrm{Re}\,\mathrm{Pr}}{L/D}\right)^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14}$	$\dfrac{L}{D} < \dfrac{\mathrm{Re}\,\mathrm{Pr}}{8}\left(\dfrac{\mu}{\mu_s}\right)^{0.42}$	(T8.2)[a]
Developing	$\mathrm{Nu} = 3.66 + \dfrac{0.065(D/L)\mathrm{Re}\,\mathrm{Pr}}{1 + 0.04[(D/L)\mathrm{Re}\,\mathrm{Pr}]^{2/3}}$		(T8.3)
Fully developed, round	$\mathrm{Nu} = 3.66$	Uniform surface temperature	(T8.4a)
	$\mathrm{Nu} = 4.36$	Uniform heat flux	(T8.4b)
Turbulent:	$\mathrm{Nu} = 0.023\,\mathrm{Re}^{4/5}\mathrm{Pr}^{0.4}$	Heating fluid $\mathrm{Re} \geq 10\,000$	(T8.5a)[b]
Fully developed	$\mathrm{Nu} = 0.023\,\mathrm{Re}^{4/5}\mathrm{Pr}^{0.3}$	Cooling fluid $\mathrm{Re} \geq 10\,000$	(T8.5b)[b]
Evaluate properties at bulk temperature t_b except μ_s and t_s at surface temperature	$\mathrm{Nu} = \dfrac{(f_s/2)(\mathrm{Re} - 1000)\mathrm{Pr}}{1 + 12.7(f_s/2)^{1/2}(\mathrm{Pr}^{2/3} - 1)}\left[1 + \left(\dfrac{D}{L}\right)^{2/3}\right]$	$f_s = \dfrac{1}{(1.58 \ln \mathrm{Re} - 3.28)^2}$	(T8.6)[c]
	For fully developed flows, set $D/L = 0$.	Multiply Nu by $(T/T_s)^{0.45}$ for gases and by $(\mathrm{Pr}/\mathrm{Pr}_s)^{0.11}$ for liquids	
	$\mathrm{Nu} = 0.027\,\mathrm{Re}^{4/5}\mathrm{Pr}^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14}$	For viscous fluids	(T8.7)[a]
	For noncircular tubes, use hydraulic mean diameter D_h in the equations for Nu for an approximate value of h.		

FINISHING CONVECTION

Convective heat transfer coefficient, h_{conv}

- There are similar looking (albeit different) relationships for **natural convection**



(*b*) Free convection

$$\mathrm{Nu} = \frac{hL_c}{k} = f(\mathrm{Ra}_{Lc}, \mathrm{Pr})$$

Ra_{Lc} = Rayleigh number = $g\beta\,\Delta t L_c^3/\nu\alpha$
Δt = $|t_s - t_\infty|$
g = gravitational acceleration
β = coefficient of thermal expansion
ν = fluid kinematic viscosity = μ/ρ
α = fluid thermal diffusivity = $k/\rho c_p$
Pr = Prandtl number = ν/α

Ra = Gr Pr

Gr = Grashof # (relationship between buoyancy and viscosity in a fluid)

$$\mathrm{Gr}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}$$ for vertical flat plates

$$\mathrm{Gr}_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}$$ for pipes

Convective heat transfer coefficient, h_{conv}

- Equations for natural convection
  - From Chapter 4 of the 2013 ASHRAE Handbook of Fundamentals:

I. General relationships	$\text{Nu} = f(\text{Ra}, \text{Pr})$ or $f(\text{Ra})$		(T9.1)
Characteristic length depends on geometry	$\text{Ra} = \text{Gr Pr} \quad \text{Gr} = \dfrac{g\beta\rho^2\lvert\Delta t\rvert L^3}{\mu^2}$	$\text{Pr} = \dfrac{c_p\mu}{k} \quad \Delta t = \lvert t_s - t_\infty\rvert$	
II. Vertical plate			
t_s = constant	$\text{Nu} = 0.68 + \dfrac{0.67\text{Ra}^{1/4}}{[1+(0.492/\text{Pr})^{9/16}]^{4/9}}$	$10^{-1} < \text{Ra} < 10^9$	(T9.2)[a]
Characteristic dimension: L = height Properties at $(t_s + t_\infty)/2$ except β at t_∞	$\text{Nu} = \left\{0.825 + \dfrac{0.387\text{Ra}^{1/6}}{[1+(0.492/\text{Pr})^{9/16}]^{8/27}}\right\}^2$	$10^9 < \text{Ra} < 10^{12}$	(T9.3)[a]
q''_s = constant Characteristic dimension: L = height Properties at $t_{s,L/2} - t_\infty$ except β at t_∞	$\text{Nu} = \left\{0.825 + \dfrac{0.387\text{Ra}^{1/6}}{[1+(0.437/\text{Pr})^{9/16}]^{8/27}}\right\}^2$	$10^{-1} < \text{Ra} < 10^{12}$	(T9.4)[a]
Equations (T9.2) and (T9.3) can be used for vertical cylinders if $D/L > 35/\text{Gr}^{1/4}$ where D is diameter and L is axial length of cylinder			
III. Horizontal plate			
Characteristic dimension = $L = A/P$, where A is plate area and P is perimeter			
Properties of fluid at $(t_s + t_\infty)/2$			
Downward-facing cooled plate and upward-facing heated plate	$\text{Nu} = 0.96\,\text{Ra}^{1/6}$	$1 < \text{Ra} < 200$	(T9.5)[b]
	$\text{Nu} = 0.59\,\text{Ra}^{1/4}$	$200 < \text{Ra} < 10^4$	(T9.6)[b]
	$\text{Nu} = 0.54\,\text{Ra}^{1/4}$	$2.2 \times 10^4 < \text{Ra} < 8 \times 10^6$	(T9.7)[b]
	$\text{Nu} = 0.15\,\text{Ra}^{1/3}$	$8 \times 10^6 < \text{Ra} < 1.5 \times 10^9$	(T9.8)[b]
Downward-facing heated plate and upward-facing cooled plate	$\text{Nu} = 0.27\,\text{Ra}^{1/4}$	$10^5 < \text{Ra} < 10^{10}$	(T9.9)[b]

Convective heat transfer coefficient, h_{conv}

- Equations for natural convection
  - From Chapter 4 of the 2013 ASHRAE Handbook of Fundamentals:

IV. Horizontal cylinder Characteristic length = d = diameter Properties of fluid at $(t_s + t_\infty)/2$ except β at t_∞	$\mathrm{Nu} = \left\{0.6 + \dfrac{0.387\ \mathrm{Ra}^{1/6}}{[1+(0.559/\mathrm{Pr})^{9/16}]^{8/27}}\right\}^2$	$10^9 < \mathrm{Ra} < 10^{13}$	(T9.10)[c]
V. Sphere Characteristic length = D = diameter Properties at $(t_s + t_\infty)/2$ except β at t_∞	$\mathrm{Nu} = 2 + \dfrac{0.589\ \mathrm{Ra}^{1/4}}{[1+(0.469/\mathrm{Pr})^{9/16}]^{4/9}}$	$\mathrm{Ra} < 10^{11}$	(T9.11)[d]
VI. Horizontal wire Characteristic dimension = D = diameter Properties at $(t_s + t_\infty)/2$	$\dfrac{2}{\mathrm{Nu}} = \ln\left(1 + \dfrac{3.3}{c\mathrm{Ra}^n}\right)$	$10^{-8} < \mathrm{Ra} < 10^6$	(T9.12)[e]
VII. Vertical wire Characteristic dimension = D = diameter; L = length of wire Properties at $(t_s + t_\infty)/2$	$\mathrm{Nu} = c\,(\mathrm{Ra}\ D/L)^{0.25} + 0.763\ c^{(1/6)}(\mathrm{Ra}\ D/L)^{(1/24)}$ In both Equations (T9.12) and (T9.13), $c = \dfrac{0.671}{[1+(0.492/\mathrm{Pr})^{(9/16)}]^{(4/9)}}$ and $n = 0.25 + \dfrac{1}{10+5(\mathrm{Ra})^{0.175}}$	$c\,(\mathrm{Ra}\ D/L)^{0.25} > 2 \times 10^{-3}$	(T9.13)[e]
VIII. Simplified equations with air at mean temperature of 21°C: h is in W/(m²·K), L and D are in m, and Δt is in °C.			
Vertical surface	$h = 1.33\left(\dfrac{\Delta t}{L}\right)^{1/4}$	$10^5 < \mathrm{Ra} < 10^9$	(T9.14)
	$h = 1.26(\Delta t)^{1/3}$	$\mathrm{Ra} > 10^9$	(T9.15)
Horizontal cylinder	$h = 1.04\left(\dfrac{\Delta t}{D}\right)^{1/4}$	$10^5 < \mathrm{Ra} < 10^9$	(T9.16)
	$h = 1.23(\Delta t)^{1/3}$	$\mathrm{Ra} > 10^9$	(T9.17)

Simplifications of convective heat transfer coefficients

- For practical purposes in building science, we usually simplify convective heat transfer coefficients to common values for relatively common cases
  - Sometimes these are fundamentally estimated
  - Sometimes these are empirical (measured) in different scenarios

TABLE 2.9

Magnitude of Convection Coefficients

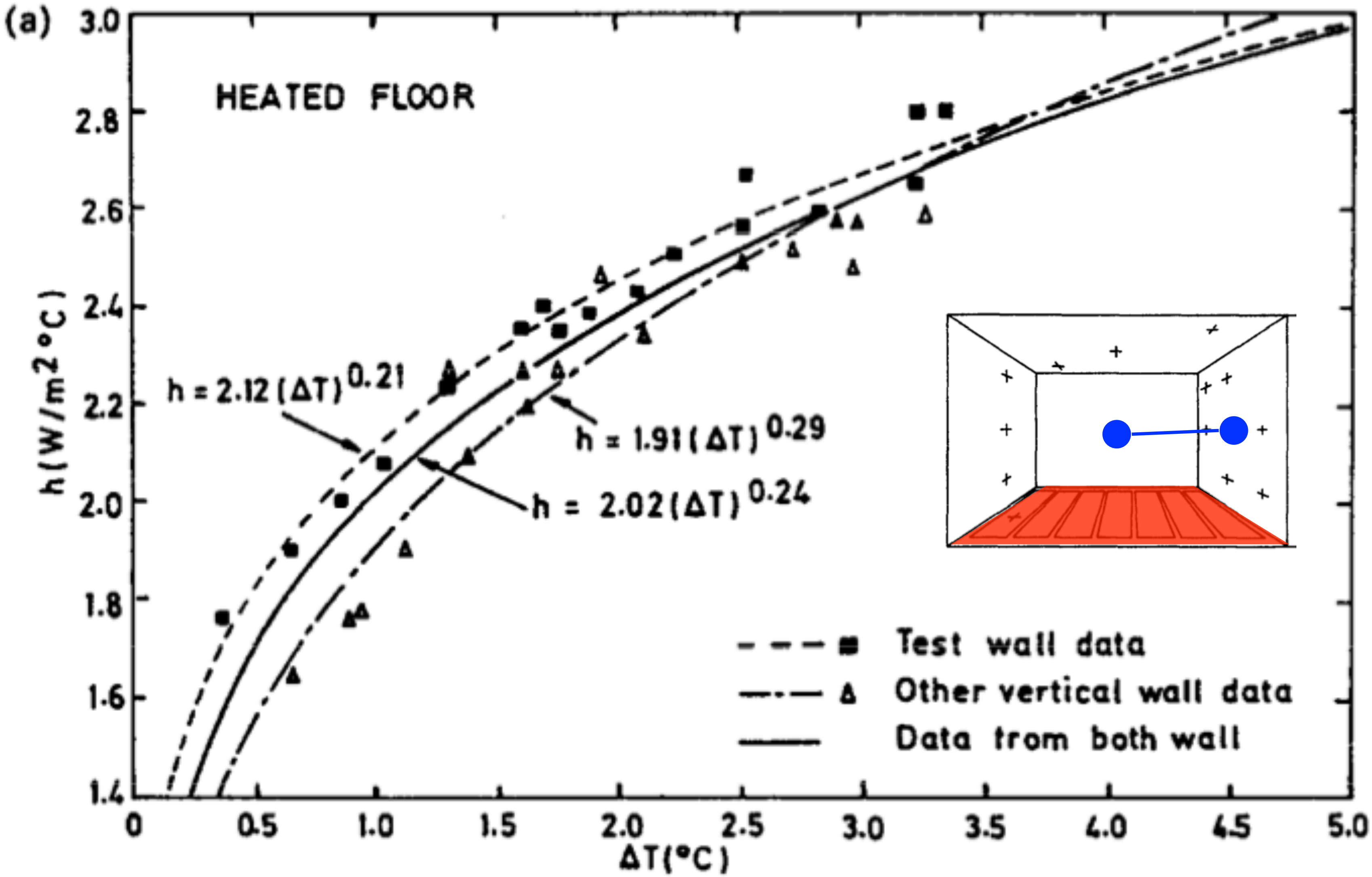
Arrangement	$W/(m^2 \cdot K)$	$Btu/(h \cdot ft^2 \cdot F)$
Air, free convection	6–30	1–5
Superheated steam or air, forced convection	30–300	5–50
Oil, forced convection	60–1800	10–300
Water, forced convection	300–6000	50–1000
Water, boiling	3000–60,000	500–10,000
Steam, condensing	6000–120,000	1000–20,000

The conversion between SI and USCS units is 5.678 $W/(m^2 \cdot K) = 1$ $Btu/(h \cdot ft^2 \cdot °F)$.

Simplifications of convective heat transfer coefficients

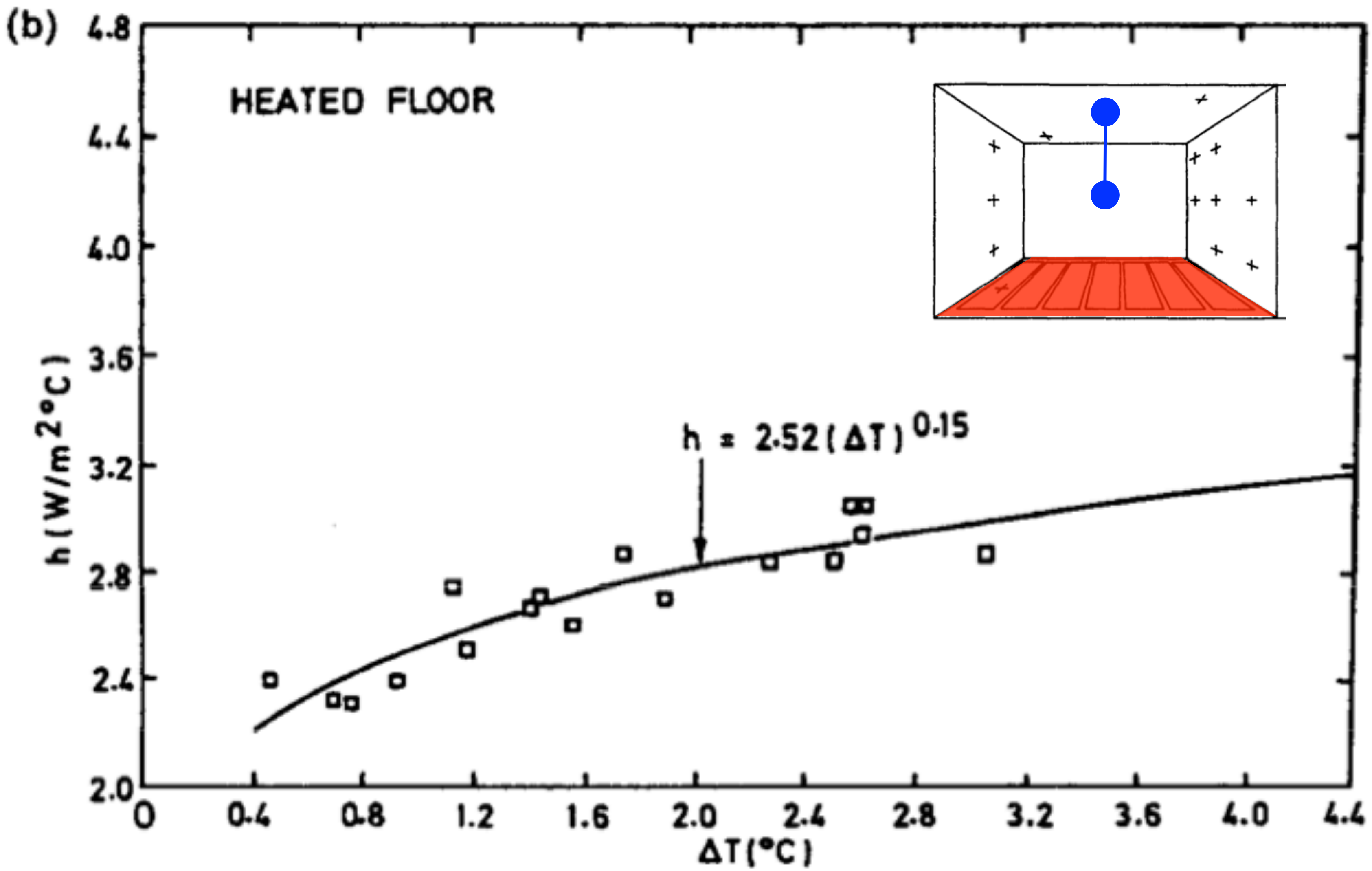
- Convective heat transfer coefficients can depend upon details of the surface-fluid interface
  - Rough surfaces have higher rates of convection
  - Orientation is important for natural convection
  - Convective heat transfer coefficients for natural convection can depend upon the actual fluid temperature and not just the temperature difference

Empirical: h_{conv} vs. ΔT for vertical walls and a heated floor



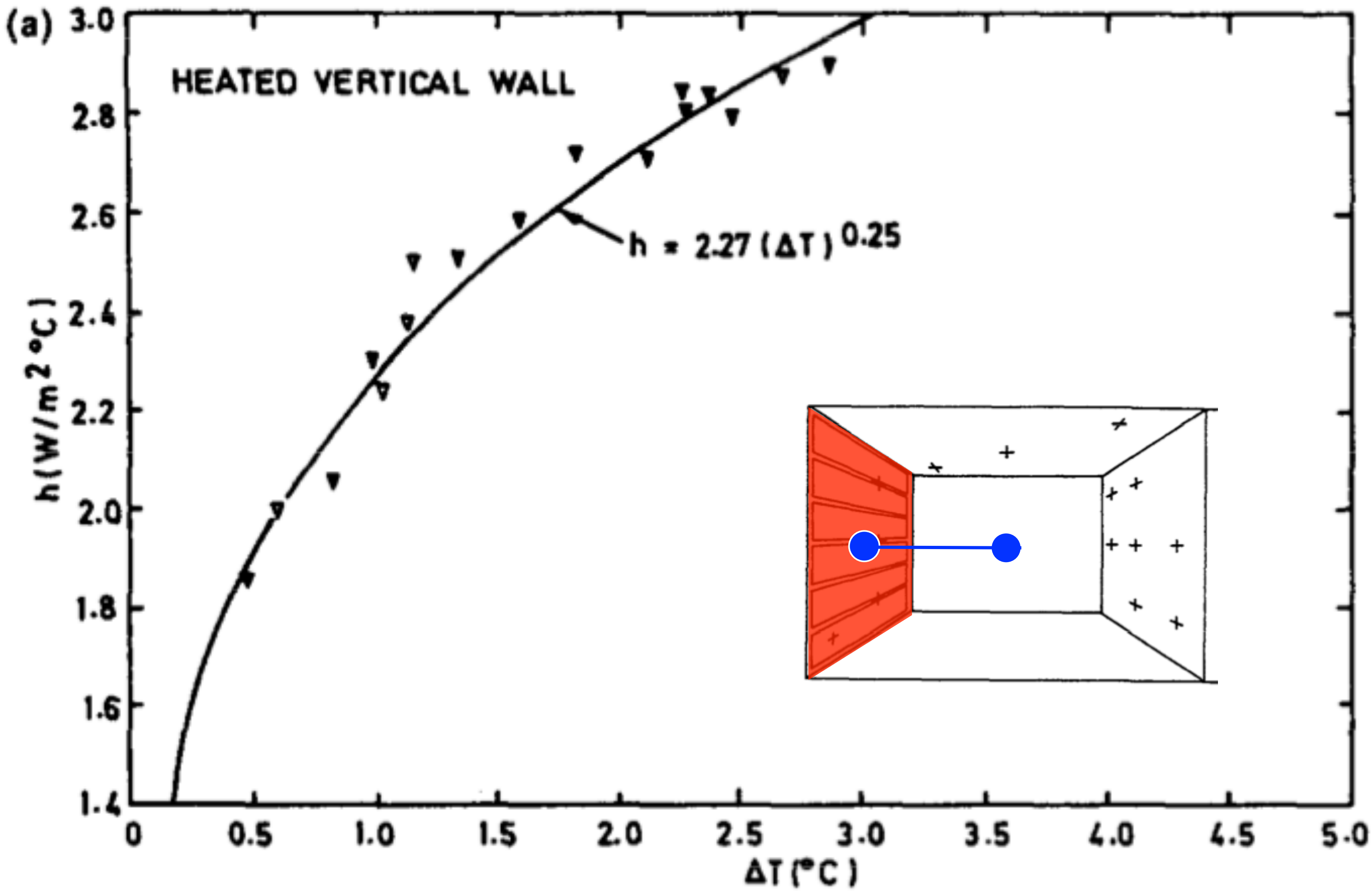
Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

Empirical: h_{conv} vs. ΔT for a ceiling and a heated floor



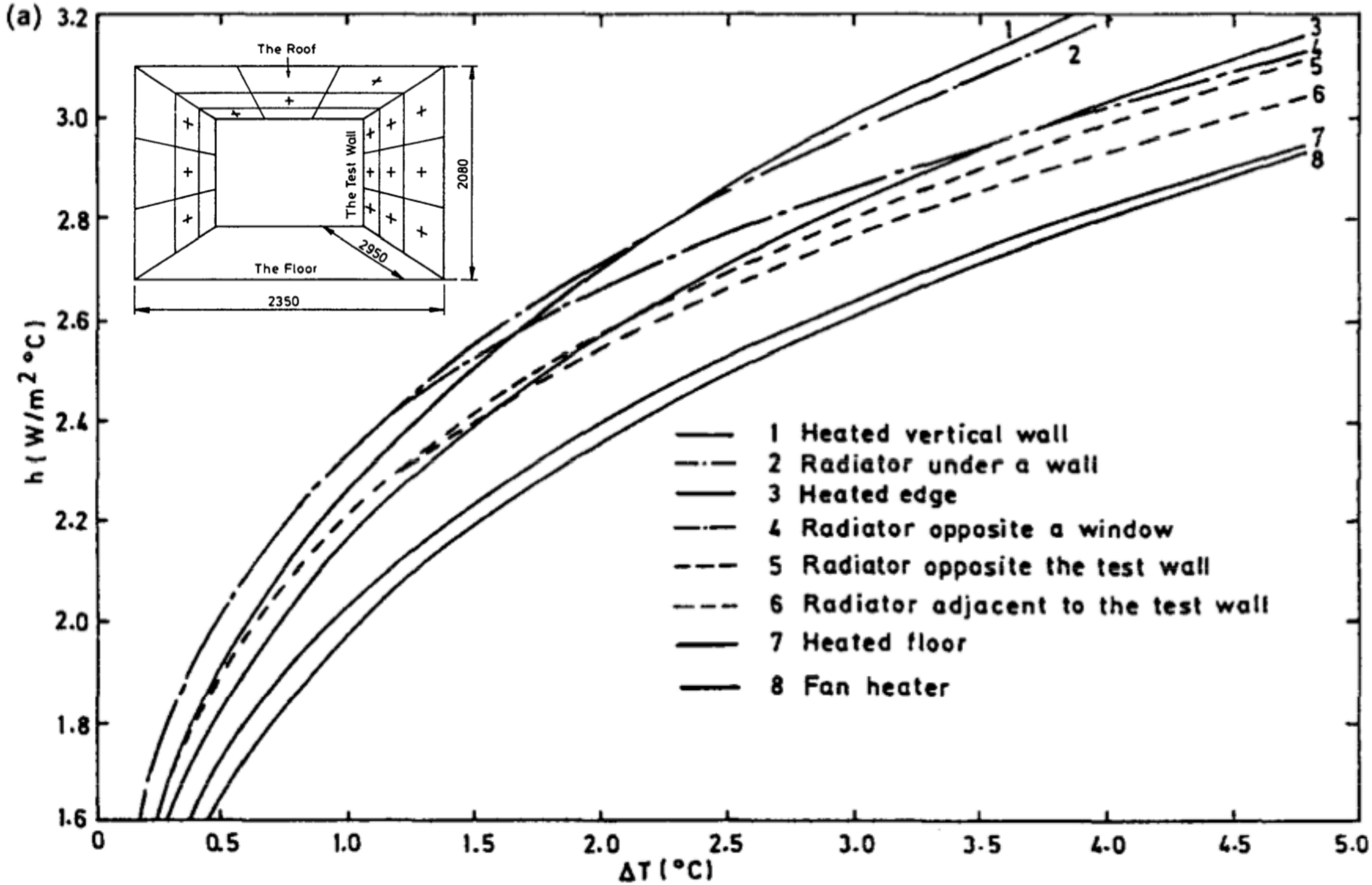
Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

Empirical: h_{conv} vs. ΔT for heated walls



Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

Empirical: h_{conv} vs. ΔT for interior walls



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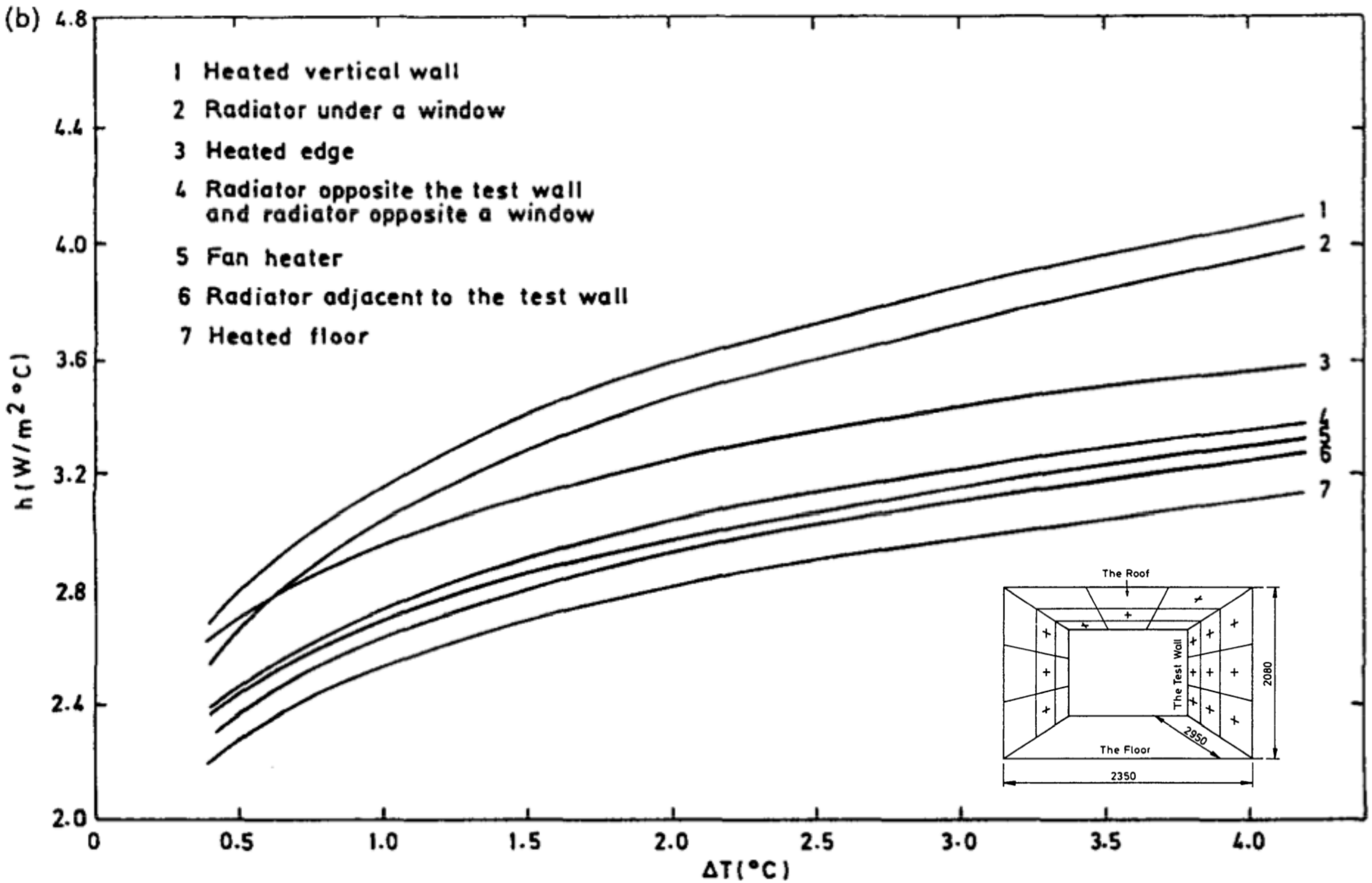
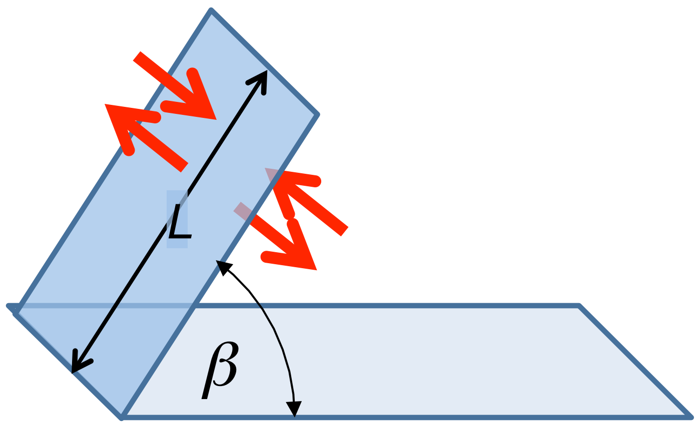
Empirical: h_{conv} vs. ΔT for interior ceilings



Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

Free convection in air from a tilted surface: Simplified



h_{conv} in [W/(m² K)]

For natural convection to or from either side of a vertical surface or a sloped surface with $\beta > 30°$

For laminar: $$h_{conv} = 1.42\left(\frac{\Delta T}{L}\sin\beta\right)^{\frac{1}{4}}$$ [Kreider 2.18SI]

For turbulent: $$h_{conv} = 1.31\left(\Delta T\sin\beta\right)^{\frac{1}{3}}$$ [Kreider 2.19SI]

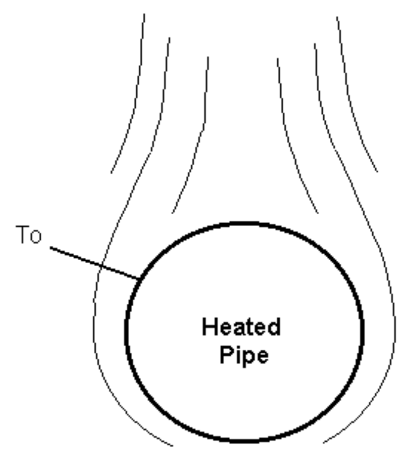
Note that these equations are ***dimensional***, so they are different for IP and SI

Free convection from horizontal pipes in air

- For cylindrical pipes of outer diameter, D, in [m]

For laminar: $$h_{conv} = 1.32\left(\frac{\Delta T}{D}\right)^{\frac{1}{4}}$$ [Kreider 2.20SI]

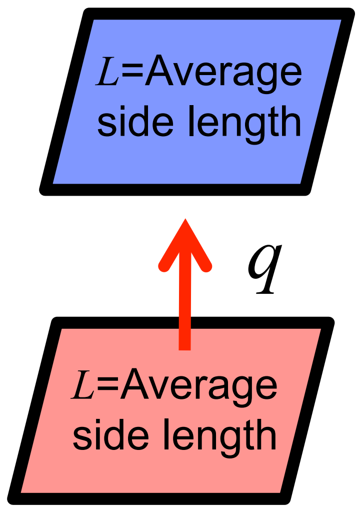
For turbulent: $$h_{conv} = 1.24\left(\Delta T\right)^{\frac{1}{3}}$$ [Kreider 2.21SI]



Free Convection Heat Transfer

Free convection for surfaces: Simplified

- Warm horizontal surfaces facing up
  - e.g. up from a warm floor to a cold ceiling

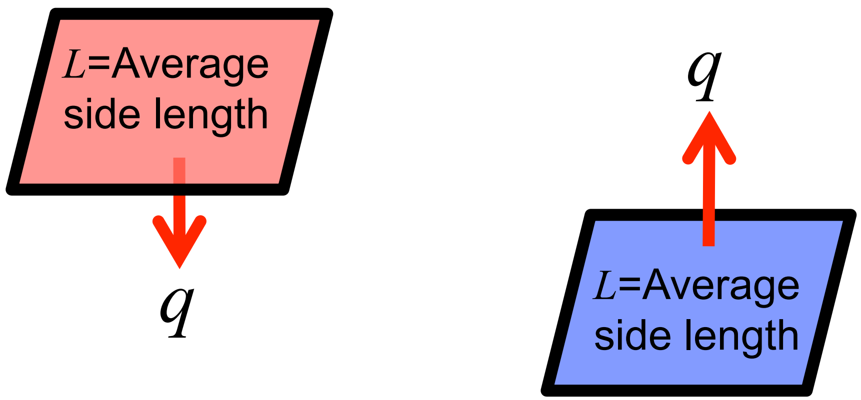


laminar: $h_{conv} \approx 1.32\left(\frac{\Delta T}{L}\right)^{1/4}$ [Kreider 2.22SI]

turbulent: $h_{conv} \approx 1.52\left(\Delta T\right)^{1/3}$ [Kreider 2.23SI]

Free convection for surfaces: Simplified

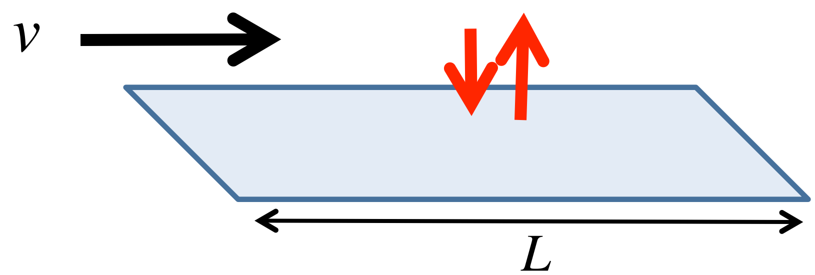
- Warm horizontal surface facing down
  - Convection is reduced because of stratification
    - e.g. a warm ceiling facing down (works against buoyancy)
    - Also applies for cooled flat surfaces facing up (like a cold floor)

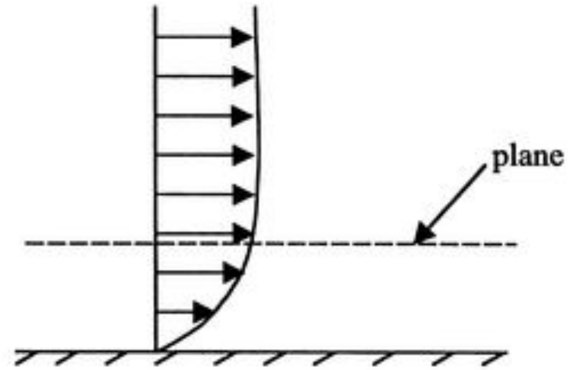


$$h_{conv} \approx 0.59\left(\frac{\Delta T}{L}\right)^{1/4}$$ both laminar and turbulent

Forced convection over planes: Simplified

- Does <u>not</u> depend on orientation





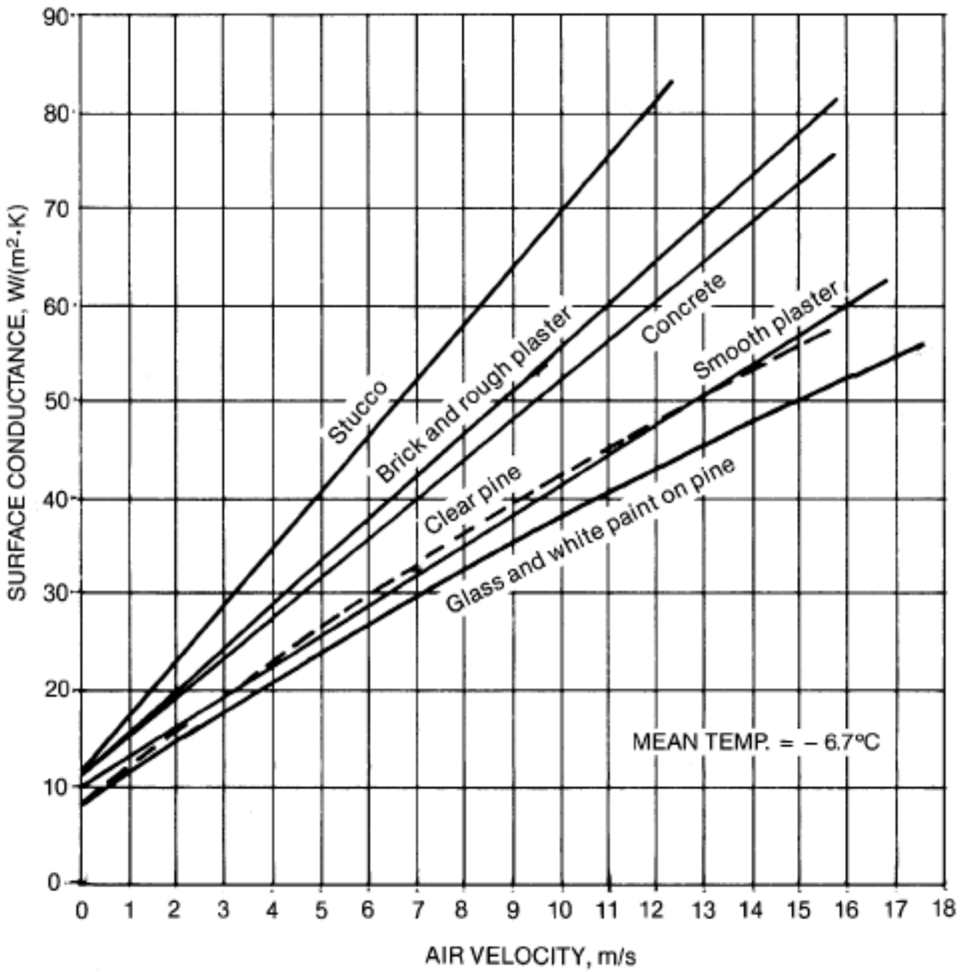
laminar: $$h_{conv} \approx 2.0\left(\frac{v}{L}\right)^{1/2}$$ [Kreider 2.24SI]

turbulent: $$h_{conv} \approx 6.2\left(\frac{v^4}{L}\right)^{1/5}$$ [Kreider 2.25SI]

*Velocity is in m/s

h_{conv} for exterior forced convection

- For forced convection, h_{conv} depends upon surface roughness and air velocity but not orientation



Most used h_{conv} for exterior forced convection

There are two relationships for h_{conv} (forced convection) which are commonly used, depending on wind speed:

- For $1 < v_{wind} < 5$ m/s

  $h_c = 5.6 + 3.9 v_{wind}$ [W/(m^2·K)] [Straube 5.15]

- For $5 < v_{wind} < 30$ m/s

  $h_c = 7.2 v_{wind}^{0.78}$ [W/(m^2·K)] [Straube 5.16]

*Good for use with external surfaces like walls and windows

Convective "R-value"

- Convective heat transfer can also be translated to an 'effective conductive layer' in contact with air
  - Allows us to assign an R-value to it

$$R_{conv} = \frac{1}{h_{conv}}$$

Typical convective surface resistances

- We often use the values given below for most conditions

Surface Conditions	Horizontal Heat Flow	Upwards Heat Flow	Downwards Heat Flow
Indoors: R_{in}	0.12 m^2K/W (SI) 0.68 $h{\cdot}ft^2{\cdot}°F/Btu$ (IP)	0.11 m^2K/W (SI) 0.62 $h{\cdot}ft^2{\cdot}°F/Btu$ (IP)	0.16 m^2K/W (SI) 0.91 $h{\cdot}ft^2{\cdot}°F/Btu$ (IP)
R_{out}: 6.7 m/s wind (Winter)		0.030 m^2K/W (SI) 0.17 $h{\cdot}ft^2{\cdot}°F/Btu$ (IP)	
R_{out}: 3.4 m/s wind (Summer)		0.044 m^2K/W (SI) 0.25 $h{\cdot}ft^2{\cdot}°F/Btu$ (IP)	

We can still sum resistances in series,
even if it involves different modes of heat transfer

Convection example

- Estimate the convective heat transfer coefficient between the back wall in the classroom and the indoor air, assuming either forced or natural convection as appropriate

- What is the convective resistance of the classroom wall?

- How does the convective thermal resistance compare to that of insulation in building walls and roofs?

Internal convection within building HVAC systems

- Flows of fluids confined by boundaries (such as the sides of a duct) are called internal flows

- Mechanisms of convection are different
  - And so are the equations for h_c

Forced convection for fully developed turbulent flow

- Airflow through ducts:

$$h_{conv} \approx 8.8\left(\frac{v^4}{D_h}\right)^{1/5} \qquad \text{[Kreider 2.26SI]}$$

D_h = the hydraulic diameter: 4 times the ratio of the flow conduit's cross-sectional area divided by the perimeter of the conduit

$$D_h = \frac{4\left(\frac{\pi D^2}{4}\right)}{\pi D} \qquad \text{[Kreider 2.27SI]}$$

- Water flow through pipes:

$$h_{conv} \approx 3580(1+0.015T)\left(\frac{v^4}{D_h}\right)^{1/5} \qquad \text{[Kreider 2.28SI]}$$

Bulk convective heat transfer: Advection

Finally, there is also one last type of convection:

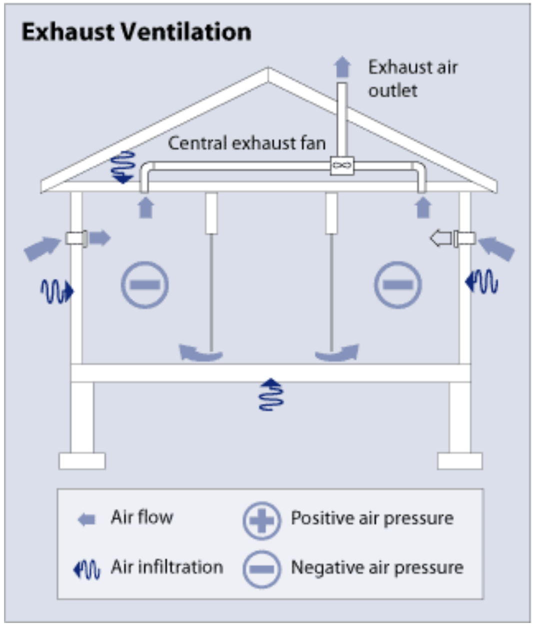
- Bulk convective heat transfer, or **advection**, is more direct than convection between surfaces and fluids
  - Bulk convective heat transfer is the transport of heat by fluid flow (e.g., air or water)
- Fluids, such as air, have the capacity to store heat, so fluids flowing into or out of a control volume also carry heat with it

$$Q_{bulk} = \dot{m} C_p \Delta T \qquad [\mathrm{W}]=[\frac{\mathrm{kg}}{\mathrm{s}} \cdot \frac{\mathrm{J}}{\mathrm{kg} \cdot \mathrm{K}} \cdot \mathrm{K}]$$

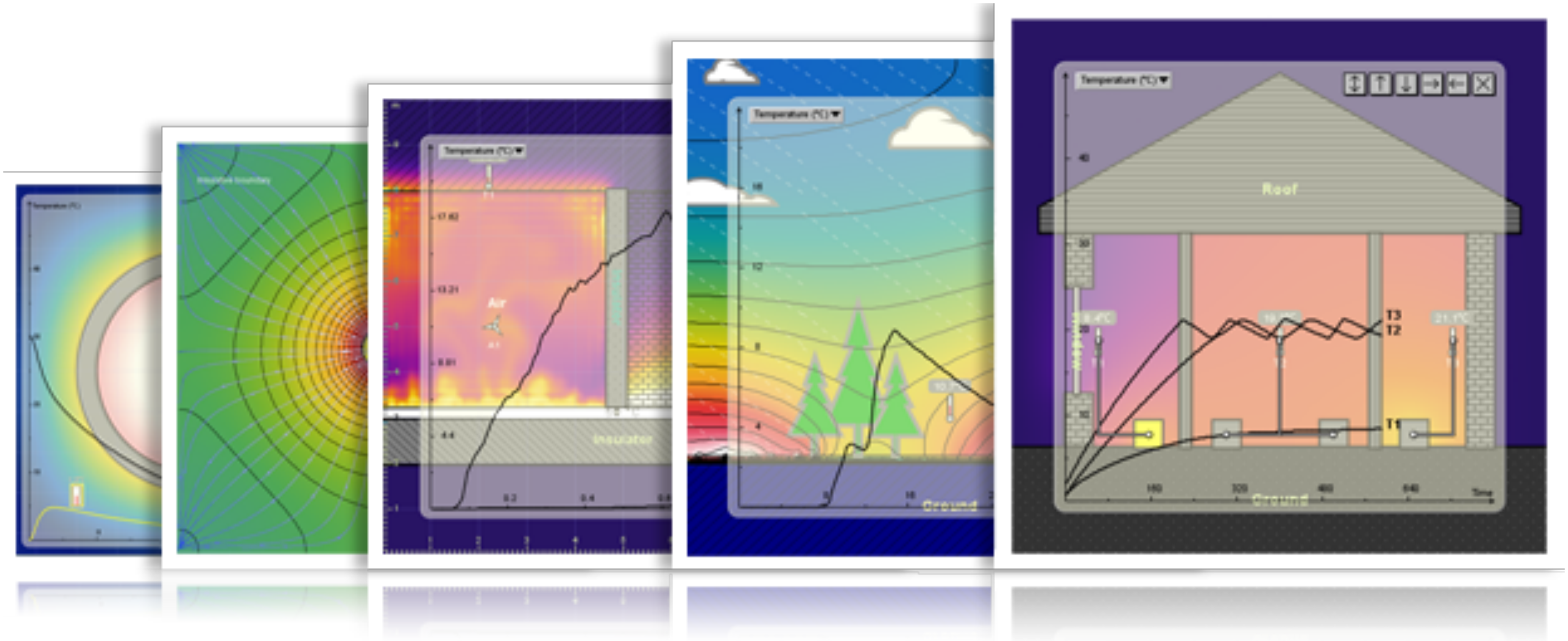
m "dot" = mass flow rate of fluid (kg/s)
C_p = specific heat capacity of fluid [J/(kgK)]

Bulk convective heat transfer: Advection

- Every time you take a shower at home, you use your bathroom exhaust fan to exhaust the hot/humid air generated by the shower
  - The fan operates at an airflow rate of 100 CFM
  - If it is 68°F inside the house and 10°F outside, what is the rate of heat loss via bulk convection during these conditions, assuming that the 100 CFM air comes in via infiltration through the building envelope

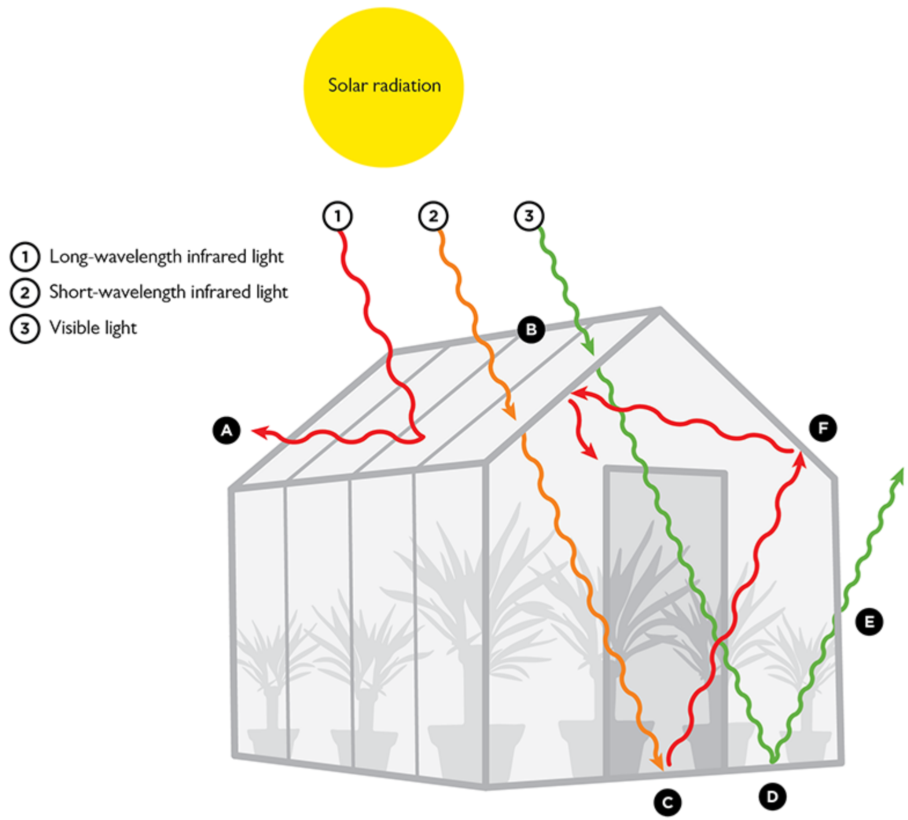


Convection visualizations



Energy2D

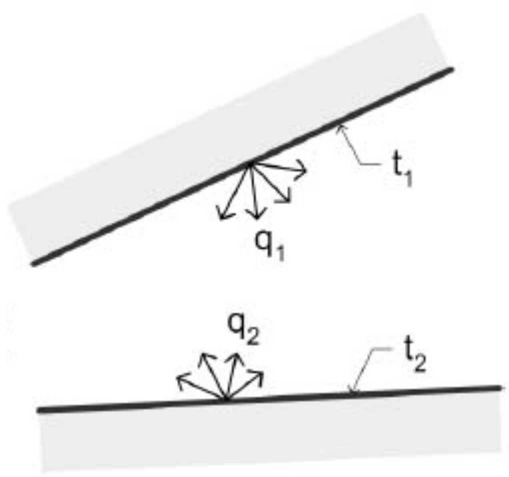
Interactive Heat Transfer Simulations for Everyone

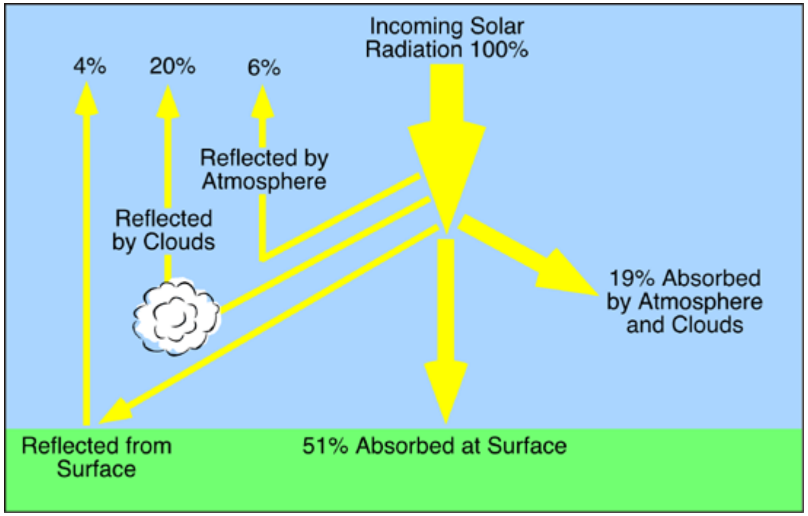

Solar radiation
1 Long-wavelength infrared light
2 Short-wavelength infrared light
3 Visible light
1
2
3
A
B
C
D
E
F

RADIATION

Radiation

- **Radiation** heat transfer is the transport of energy by electromagnetic waves
  - Oscillations of electrons that comprise matter
  - Exchange between matter at different temperatures
- Radiation must be absorbed by matter to produce internal energy; emission of radiation corresponds to reduction in stored thermal energy





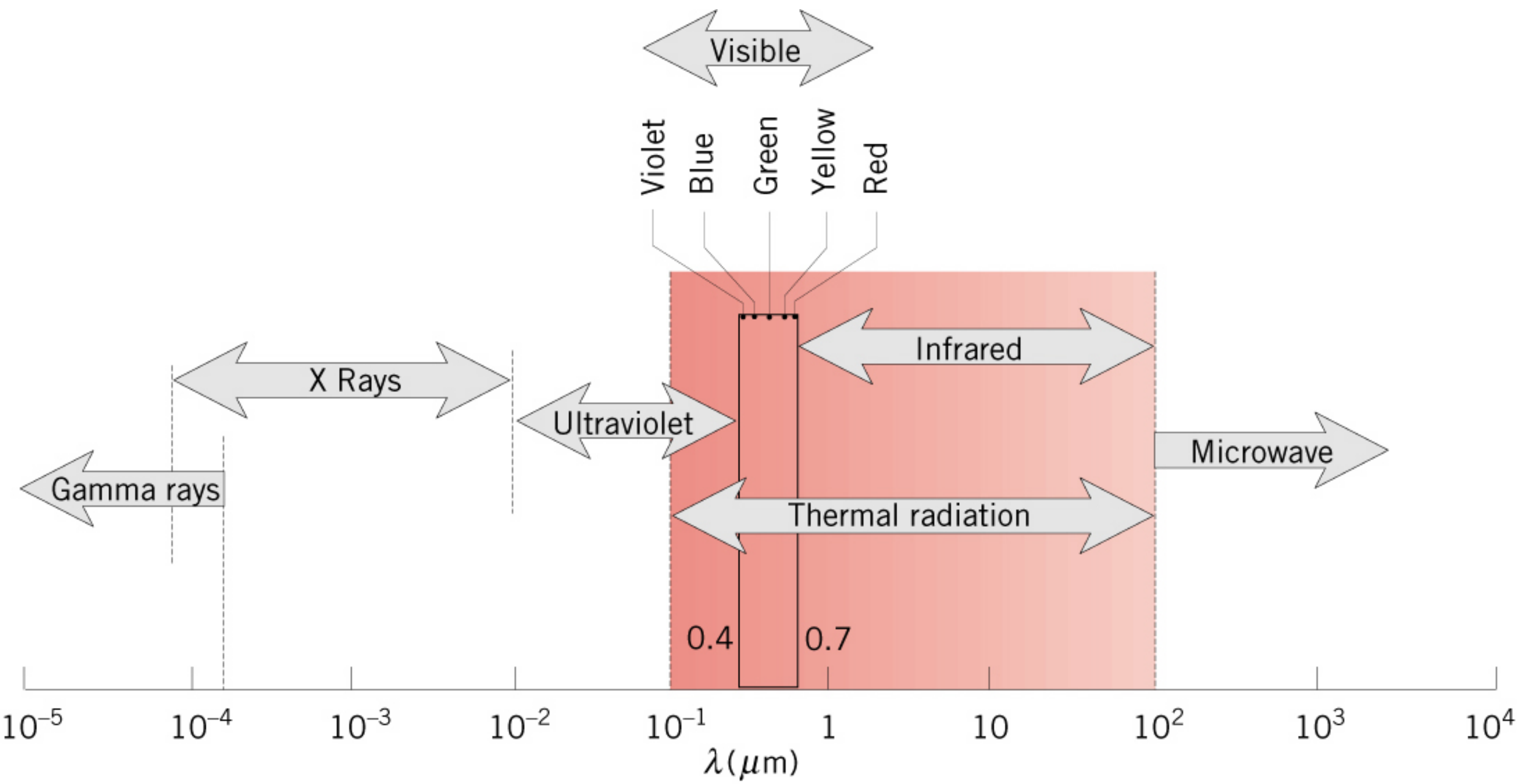
Radiation

- Radiation needs to be dealt with in terms of <u>wavelength</u> (λ)
  - Different wavelengths of solar radiation pass through the earth's atmosphere *more* or *less* efficiently than other wavelengths
  - Materials also *absorb* and *re-emit* solar radiation of different wavelengths with different efficiencies

- For our purposes, it's generally appropriate to treat radiation in two groups:
  - <u>Short-wave</u> (solar radiation)
  - <u>Long-wave</u> (emitted and re-emitted radiation)

Radiation: the electromagnetic spectrum

- <u>Thermal radiation</u> is confined to the infrared, visible, and ultraviolet regions ($0.1 < \lambda < 100\ \mu m$)



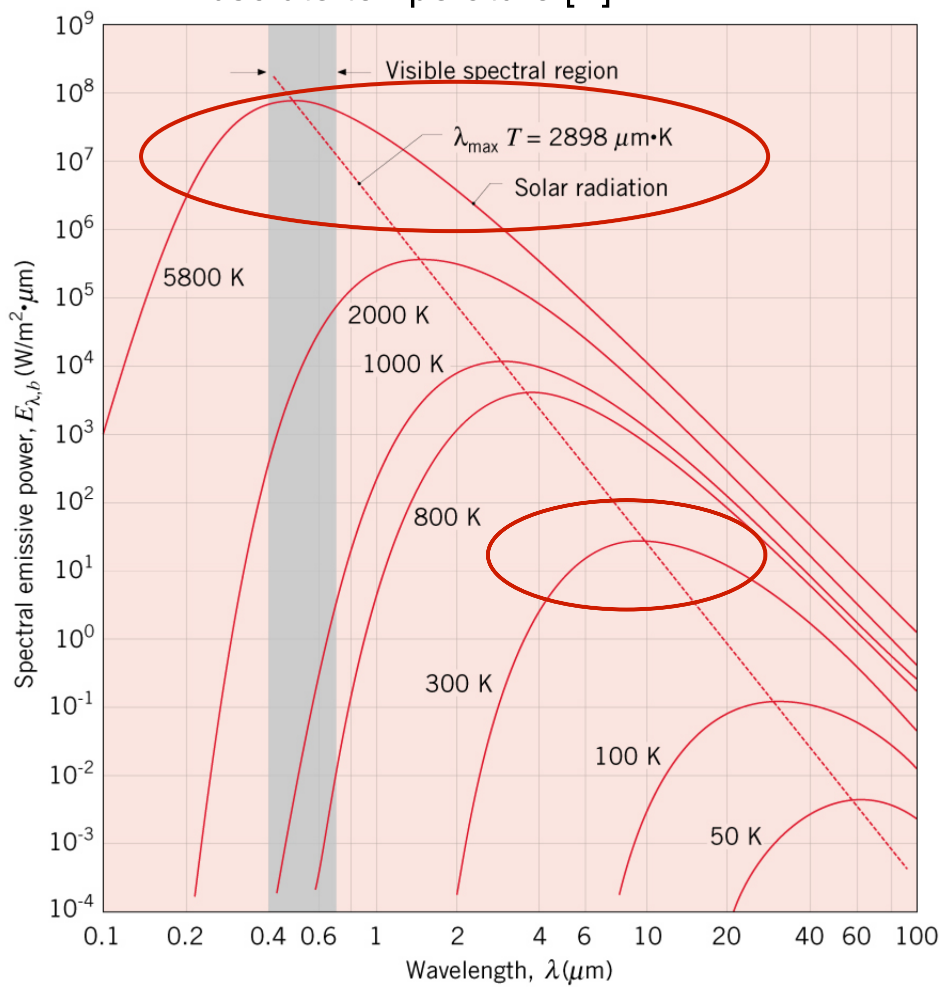
Black body radiation: Spectral (Planck) distribution

- Radiation from a perfect radiator follows the "black body" curve (ideal, black body *emitter*)
- The peak of the black body curve depends on the object's temperature
  - Lower T, larger λ peak
- Peak radiation from the sun is in the visible region
  - About 0.4 to 0.7 μm
- Radiation involved in building surfaces is in the infrared region
  - Greater than 0.7 μm

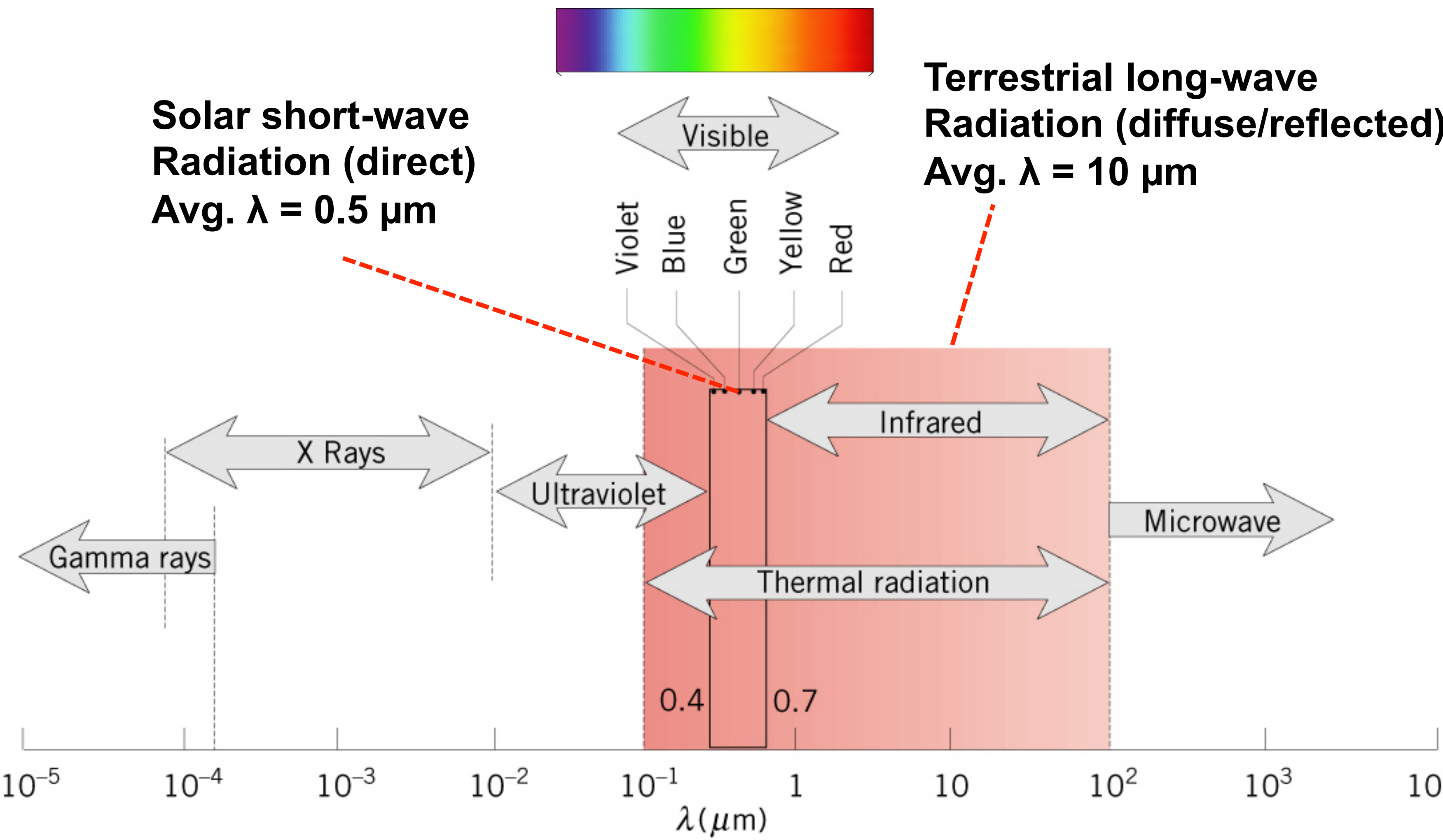
$q = \sigma T^4$

σ = Stefan-Boltzmann constant = $5.67 \times 10^{-8}\ \frac{W}{m^2 \cdot K^4}$

T = Absolute temperature [K]



Radiation: Short-wave and Long-wave



Solar radiation striking a surface (high temperature)

- Most solar radiation is at short wavelengths

Solar radiation striking a surface:

$$I_{solar} \quad \left[\frac{\text{W}}{\text{m}^2}\right]$$

Solar radiation: (opaque surface) $q_{solar} = \alpha I_{solar}$

Transmitted solar radiation: (transparent surface) $q_{solar} = \tau I_{solar}$

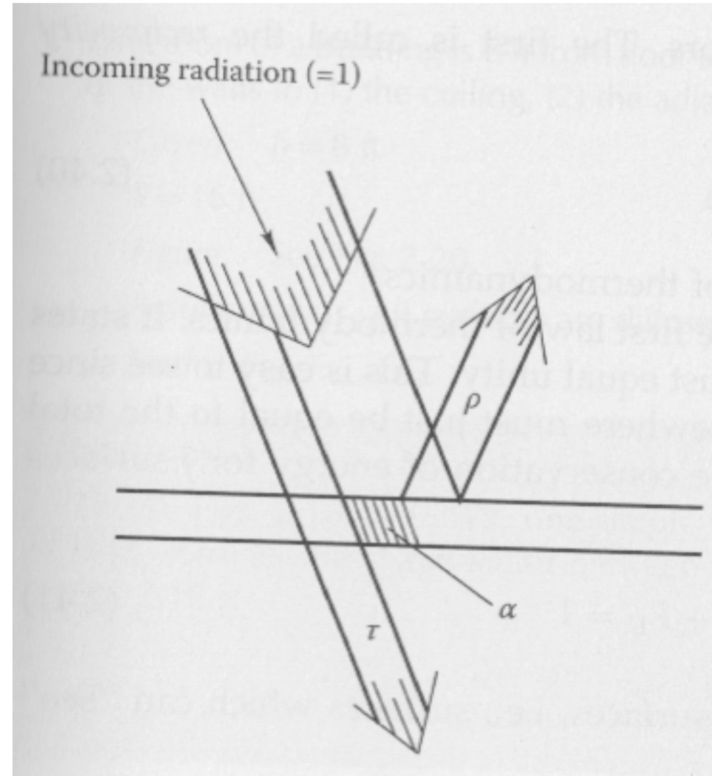
Absorptivity, transmissivity, and reflectivity

- The absorptivity, α, is the fraction of energy hitting an object that is actually absorbed

- Transmissivity, τ, is a measure of how much radiation passes through an object

- Reflectivity, ρ, is a measure of how much radiation is reflected off an object

- We use these terms primarily for solar radiation
  - For an opaque surface (τ = 0): $q_{solar} = \alpha I_{solar}$
  - For a transparent surface (τ > 0): $q_{solar} = \tau I_{solar}$



$$\alpha + \tau + \rho = 1$$

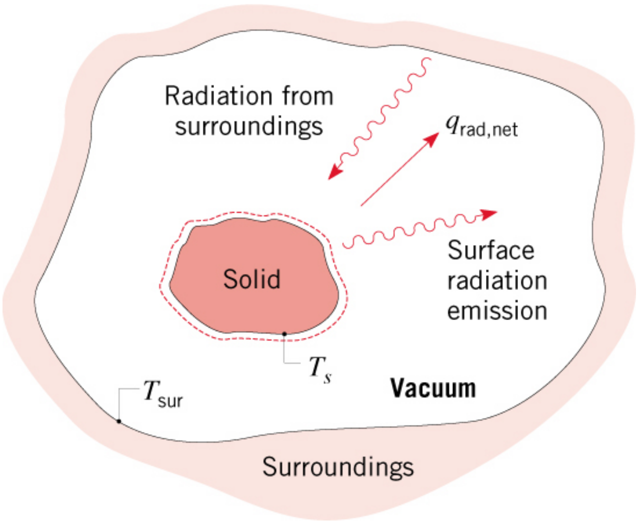
Absorptivity (α) for solar (short-wave) radiation

Surface	Absorptance for Solar Radiation
A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99
Black nonmetallic surfaces such as asphalt, carbon, slate, paint, paper	0.85 to 0.98
Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.65 to 0.80
Yellow and buff brick and stone, firebrick, fire clay	0.50 to 0.70
White or light-cream brick, tile, paint or paper, plaster, whitewash	0.30 to 0.50
Window glass	—
Bright aluminum paint; gilt or bronze paint	0.30 to 0.50
Dull brass, copper, or aluminum; galvanized steel; polished iron	0.40 to 0.65
Polished brass, copper, monel metal	0.30 to 0.50
Highly polished aluminum, tin plate, nickel, chromium	0.10 to 0.40

Surface radiation (lower temperature: long-wave)

- All objects above absolute zero radiate electromagnetic energy according to:

"Gray bodies"



$$q_{rad} = \varepsilon\sigma T^4$$

Where ε = emissivity

σ = Stefan-Boltzmann constant = $5.67\times10^{-8}\ \frac{\text{W}}{\text{m}^2\cdot\text{K}^4}$

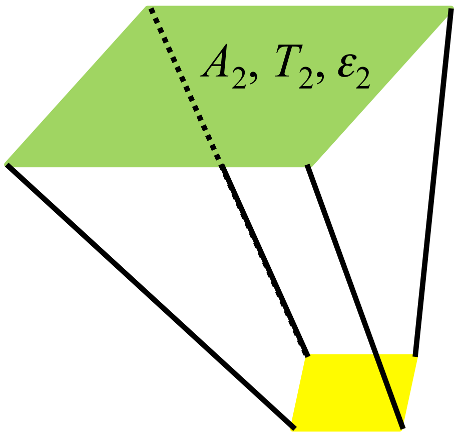
T = Absolute temperature [K]

- Net radiation heat transfer occurs when an object radiates a different amount of energy than it absorbs
- If all the surrounding objects are at the same temperature, the net will be zero

Radiation heat transfer (surface-to-surface)

- We can write the net thermal radiation heat transfer between surfaces 1 and 2 as:

$$Q_{1\to 2} = \frac{A_1\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{A_1}{A_2}\frac{1-\varepsilon_2}{\varepsilon_2} + \frac{1}{F_{12}}} \qquad q_{1\to 2} = \frac{Q_{1\to 2}}{A_1}$$



where ε_1 and ε_2 are the surface emittances,

A_1 and A_2 are the surface areas

and $F_{1\to 2}$ is the view factor from surface 1 to 2

$F_{1\to 2}$ is a function of geometry only

Emissivity (“gray bodies”)

- Real surfaces emit less radiation than ideal “black” ones
  - The ratio of energy radiated by a given body to a perfect black body at the same temperature is called the emissivity: ε
- ε is dependent on wavelength, but for most common building materials (e.g. brick, concrete, wood…), $\varepsilon = 0.9$ at most wavelengths

Emissivity (ε) of common materials

Surface	Emittance ε 50–100 °F
A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99
Black nonmetallic surfaces such as asphalt, carbon, slate, paint, paper	0.90 to 0.98
Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95
Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95
White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95
Window glass	0.90 to 0.95
Bright aluminum paint; gilt or bronze paint	0.40 to 0.60
Dull brass, copper, or aluminum; galvanized steel; polished iron	0.20 to 0.30
Polished brass, copper, monel metal	0.02 to 0.05
Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04

Emissivity (ε) of common building materials

TABLE 2.11

Emissivities of Some Common Building Materials at Specified Temperatures

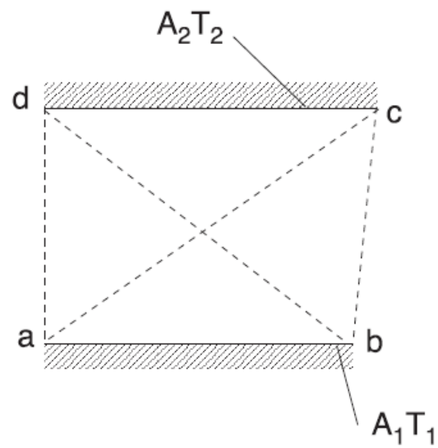
Surface	Temperature, °C	Temperature, °F	ϵ
Brick			
Red, rough	40	100	0.93
Concrete			
Rough	40	100	0.94
Glass			
Smooth	40	100	0.94
Ice			
Smooth	0	32	0.97
Marble			
White	40	100	0.95
Paints			
Black gloss	40	100	0.90
White	40	100	0.89–0.97
Various oil paints	40	100	0.92–0.96
Paper			
White	40	100	0.95
Sandstone	40–250	100–500	0.83–0.90
Snow	−12–−6	10–20	0.82
Water			
0.1 mm or more thick	40	100	0.96
Wood			
Oak, planed	40	100	0.90
Walnut, sanded	40	100	0.83
Spruce, sanded	40	100	0.82
Beech	40	100	0.94

Source: Courtesy of Sparrow, E.M. and Cess, R.D., *Radiation Heat Transfer*, augmented edn, Hemisphere, New York, 1978. With permission.

View factors, F_{12}

- Radiation travels in directional beams
  - Thus, areas and angle of incidence between two exchanging surfaces influences radiative heat transfer

Some common view factors:



$$A_1 F_{1\to 2} = 0.5((ac + bd) - (ad + bc))$$

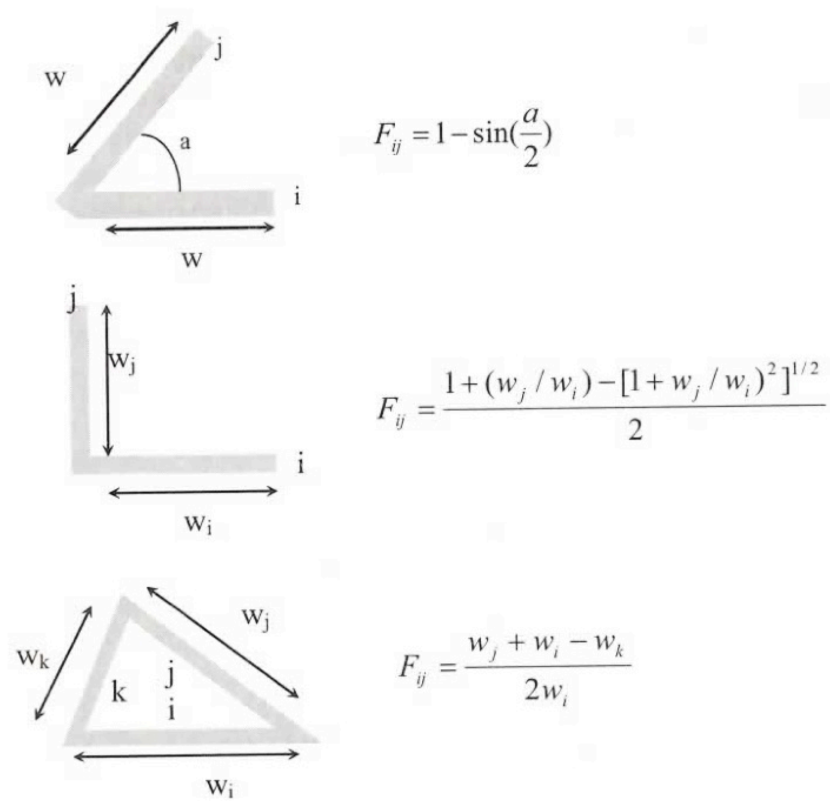
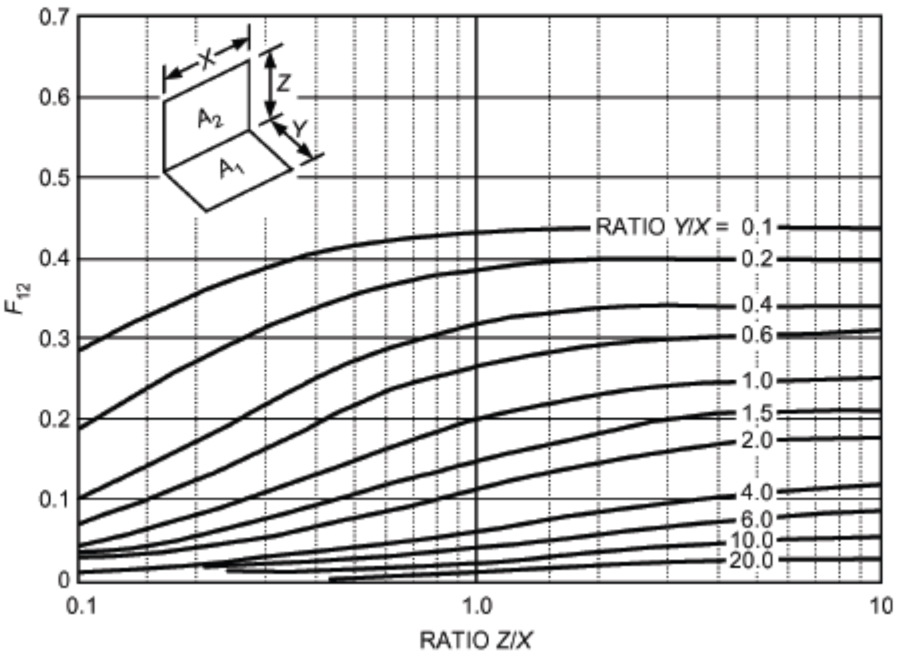


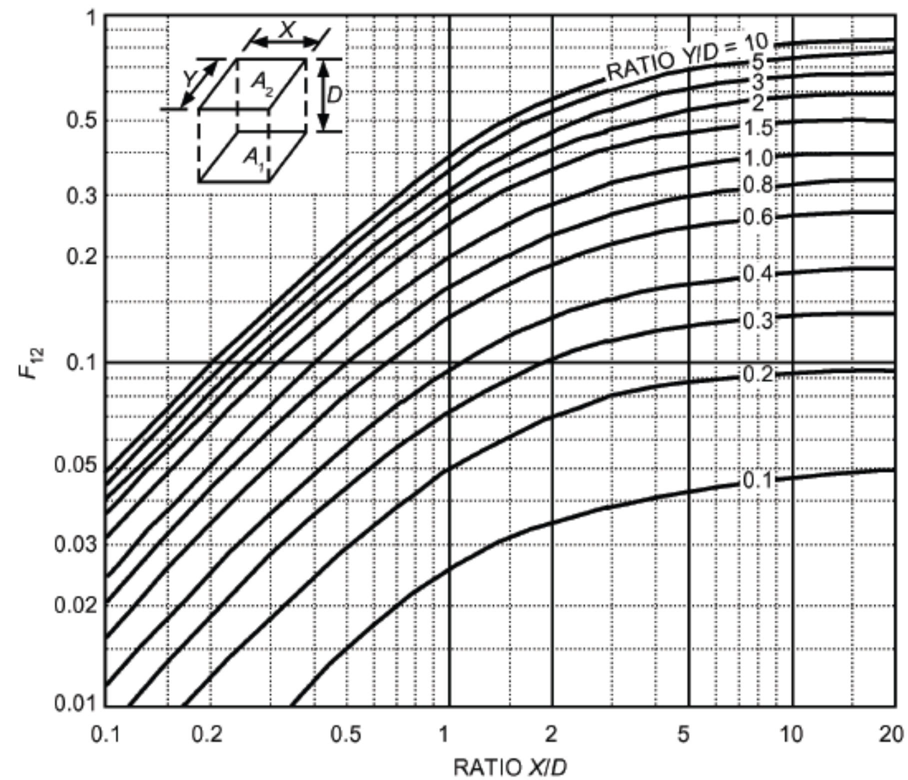
Figure 5.6: View factors for common situations in building enclosures [Hagentoft 2000]

Typical view factors

Other common view factors from the ASHRAE Handbook of Fundamentals:



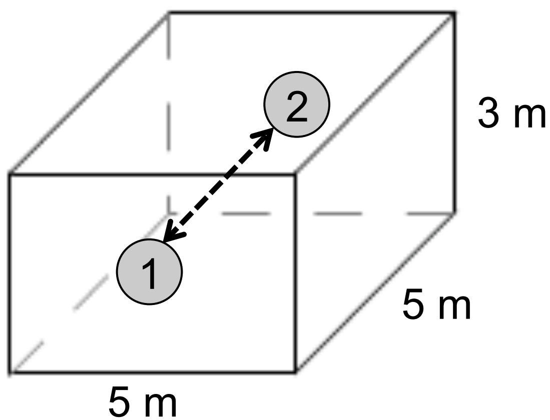
A. PERPENDICULAR RECTANGLES WITH COMMON EDGE

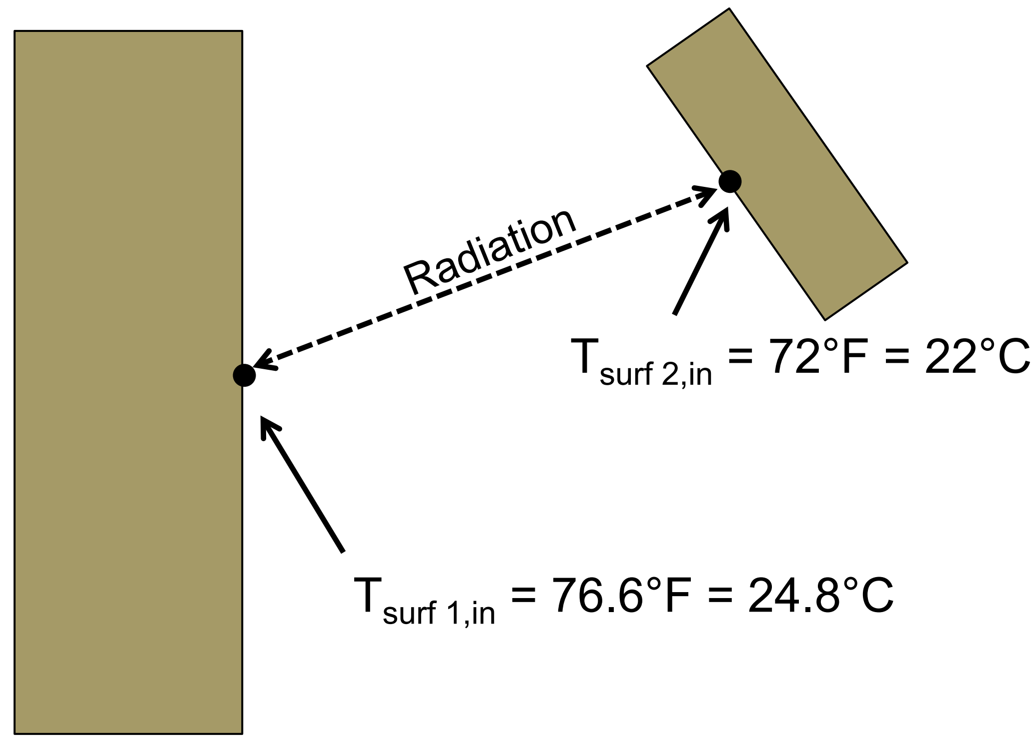


B. ALIGNED PARALLEL RECTANGLES

Long-wave radiation example

- What is the net radiative exchange between the two interior wall surfaces below end of the room if the room is 5 m x 5 m x 3 m?





Simplifying surface radiation

- We can also often simplify radiation from:

$$Q_{1\rightarrow 2} = \frac{A_1 \sigma \left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{A_1}{A_2}\frac{1-\varepsilon_2}{\varepsilon_2} + \frac{1}{F_{12}}}$$

- To: $Q_{1\rightarrow 2} = \varepsilon_{surf} A_{surf} \sigma F_{12} \left(T_1^4 - T_2^4\right)$

Particularly when dealing with large differences in areas, such as sky-surface or ground-surface exchanges

Simplifying radiation

- We can also define a radiation heat transfer coefficient that is analogous to other heat transfer coefficients

$$Q_{rad,1\rightarrow 2} = \boxed{h_{rad}} A_1 \left(T_1 - T_2\right) = \frac{1}{R_{rad}} A_1 \left(T_1 - T_2\right)$$

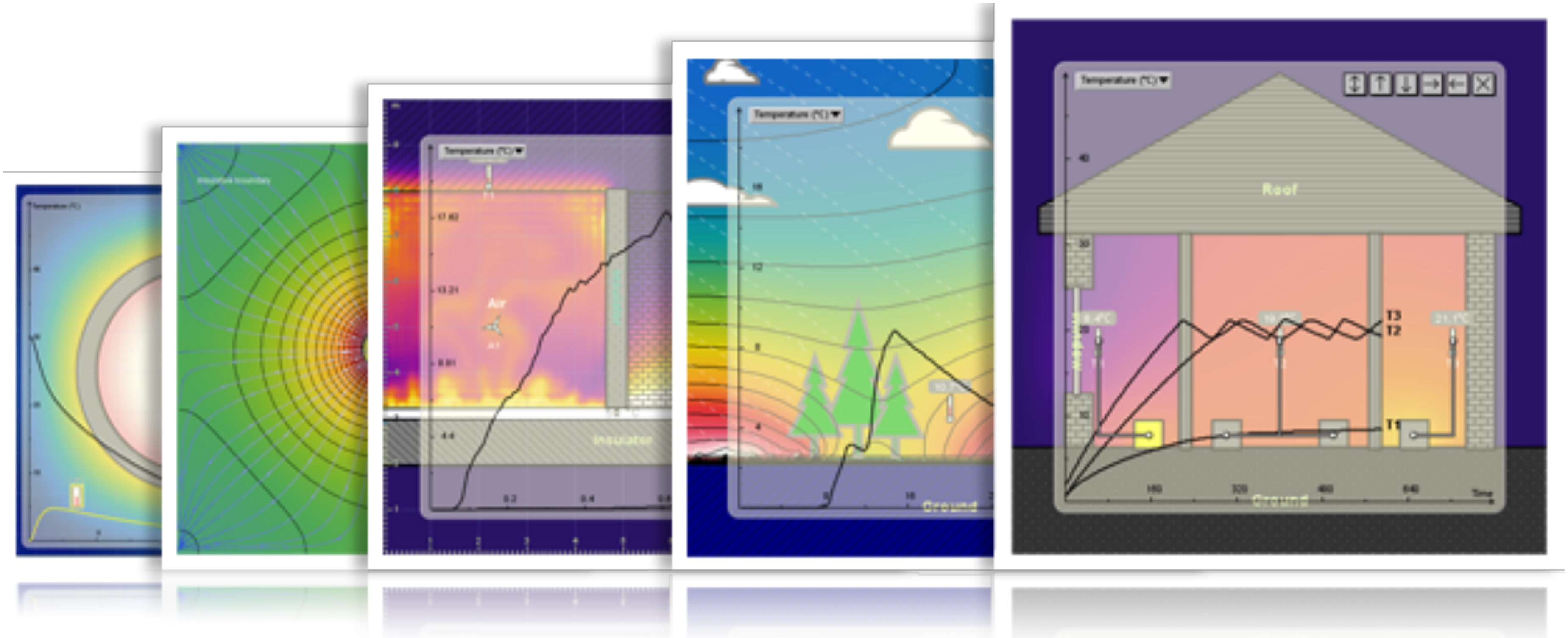
- When $A_1 = A_2$, and T_1 and T_2 are within ~50°F of each other, we can approximate h_{rad} with a simpler equation:

$$h_{rad} = \frac{4\sigma T_{avg}^3}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

where

$$T_{avg} = \frac{T_1 + T_2}{2}$$

Radiation visualizations



Energy2D

Interactive Heat Transfer Simulations for Everyone

Next class

- Solar radiation
- Heat transfer through windows