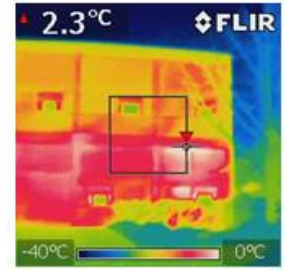
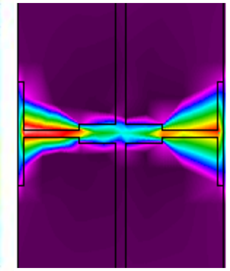
CAE 331/513
Building Science
Fall 2019





September 5, 2019
Radiation

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ASHRAE

Society Scholarships

38 ASHRAE Society Scholarships Available for 2020-2021

Applications due December 1 2019

- 13 Undergraduate Engineering Scholarships: $3,000 - $10,000 each. **Now accepting applications**
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- 5 Engineering Technology Scholarships: $5,000 each. **Now accepting applications**
- 1 Freshman Engineering Scholarship: $5,000
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Scholarships were presented at the September 18, 2018 MCA of Chicago Scholarship Awards. Congratulations to our winners! View photos from the event here.

Applications for 2019 scholarships must be submitted by September 13, 2019.

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https://www.mca.org/scholarships/

Last time: Convection

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

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

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

Forced:

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

Natural:

$$\text{Nu} = \frac{hL_c}{k} = f(\text{Ra}_{Lc}, \text{Pr}) \qquad \text{Gr}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}$$ for vertical flat plates

1. Theory
2. Empirical values
3. Simplified values

- **Advection**

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

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

Convective heat transfer coefficient, h_{conv}

- Equations for forced convection
  - From Chapter 4 of the ASHRAE Handbook of Fundamentals (SI):

Table 8 Forced-Convection Correlations

I. General Correlation $\text{Nu} = f(\text{Re}, \text{Pr})$

II. Internal Flows for Pipes and Ducts: Characteristic length = D, pipe diameter, or D_h, hydraulic diameter.

$$\text{Re} = \frac{\rho V_{avg} D_h}{\mu} = \frac{\dot{m} D_h}{A_c \mu} = \frac{Q D_h}{A_c \nu} = \frac{4\dot{m}}{\mu P_{wet}} = \frac{4Q}{\nu P_{wet}}$$

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).

	$\frac{\text{Nu}}{\text{Re Pr}^{1/3}} = \frac{f}{2}$	Colburn's analogy (turbulent)	(T8.1)
Laminar: Re < 2300	$\text{Nu} = 1.86\left(\frac{\text{Re Pr}}{L/D}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	$\frac{L}{D} < \frac{\text{Re Pr}}{8}\left(\frac{\mu}{\mu_s}\right)^{0.42}$	(T8.2)[a]
Developing	$\text{Nu} = 3.66 + \frac{0.065(D/L)\text{Re Pr}}{1 + 0.04[(D/L)\text{Re Pr}]^{2/3}}$		(T8.3)
Fully developed, round	$\text{Nu} = 3.66$	Uniform surface temperature	(T8.4a)
	$\text{Nu} = 4.36$	Uniform heat flux	(T8.4b)
Turbulent:	$\text{Nu} = 0.023\ \text{Re}^{4/5}\text{Pr}^{0.4}$	Heating fluid Re ≥ 10 000	(T8.5a)[b]
Fully developed	$\text{Nu} = 0.023\ \text{Re}^{4/5}\text{Pr}^{0.3}$	Cooling fluid Re ≥ 10 000	(T8.5b)[b]
Evaluate properties at bulk temperature t_b except μ_s and t_s at surface temperature	$\text{Nu} = \frac{(f_s/2)(\text{Re} - 1000)\text{Pr}}{1 + 12.7(f_s/2)^{1/2}(\text{Pr}^{2/3} - 1)}\left[1 + \left(\frac{D}{L}\right)^{2/3}\right]$	$f_s = \frac{1}{(1.58 \ln \text{Re} - 3.28)^2}$	(T8.6)[c]
	For fully developed flows, set $D/L = 0$.	Multiply Nu by $(T/T_s)^{0.45}$ for gases and by $(\text{Pr}/\text{Pr}_s)^{0.11}$ for liquids	
	$\text{Nu} = 0.027\ \text{Re}^{4/5}\text{Pr}^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	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.

Convective heat transfer coefficient, h_{conv}

- Equations for forced convection
  - From Chapter 4 of the ASHRAE Handbook of Fundamentals (SI):

III. External Flows for Flat Plate: Characteristic length = L = length of plate. Re = VL/ν.
All properties at arithmetic mean of surface and fluid temperatures.

Laminar boundary layer: Re < 5×10^5	$\mathrm{Nu} = 0.332\,\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3}$	Local value of h	(T8.8)
	$\mathrm{Nu} = 0.664\,\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3}$	Average value of h	(T8.9)
Turbulent boundary layer: Re > 5×10^5	$\mathrm{Nu} = 0.0296\,\mathrm{Re}^{4/5}\mathrm{Pr}^{1/3}$	Local value of h	(T8.10)
Turbulent boundary layer beginning at leading edge: All Re	$\mathrm{Nu} = 0.037\,\mathrm{Re}^{4/5}\mathrm{Pr}^{1/3}$	Average value of h	(T8.11)
Laminar-turbulent boundary layer: Re > 5×10^5	$\mathrm{Nu} = (0.037\,\mathrm{Re}^{4/5} - 871)\mathrm{Pr}^{1/3}$	Average value $\mathrm{Re}_c = 5 \times 10^5$	(T8.12)

IV. External Flows for Cross Flow over Cylinder: Characteristic length = D = diameter. Re = VD/ν.
All properties at arithmetic mean of surface and fluid temperatures.

Average value of h

$$\mathrm{Nu} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3}}{[1 + (0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}}{282\,000}\right)^{5/8}\right]^{4/5} \quad \text{(T8.14)}^{d}$$

V. Simplified Approximate Equations: h is in W/(m²·K), V is in m/s, D is in m, and t is in °C.

Flows in pipes Re > 10 000	Atmospheric air (0 to 200°C):	$h = (3.76 - 0.00497t)V^{0.8}/D^{0.2}$	(T8.15a)[e]
	Water (3 to 200°C):	$h = (1206 + 23.9t)V^{0.8}/D^{0.2}$	(T8.15b)[e]
	Water (4 4 to 104°C:	$h = (1431 + 20.9t)V^{0.8}/D^{0.2}$ (McAdams 1954)	(T8.15c)[g]

Flow over cylinders

Atmospheric air: 0°C < t < 200°C, where t = arithmetic mean of air and surface temperature.

$h = 2.755V^{0.471}/D^{0.529}$	35 < Re < 5000	(T8.16a)
$h = (4.22 - 0.002\,57t)V^{0.633}/D^{0.367}$	5000 < Re < 50 000	(T8.16b)

Water: 5°C < t < 90°C, where t = arithmetic mean of water and surface temperature.

$h = (461.8 + 2.01t)V^{0.471}/D^{0.529}$	35 < Re < 5000	(T8.17a)
$h = (1012 + 9.19t)V^{0.633}/D^{0.367}$	5000 < Re < 50 000	(T8.17b)[f]

Convective heat transfer coefficient, h_{conv}

- Equations for natural convection
  - From Chapter 4 of the ASHRAE Handbook of Fundamentals (SI):

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 ASHRAE Handbook of Fundamentals (SI):

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

<u>Simplifications</u> 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

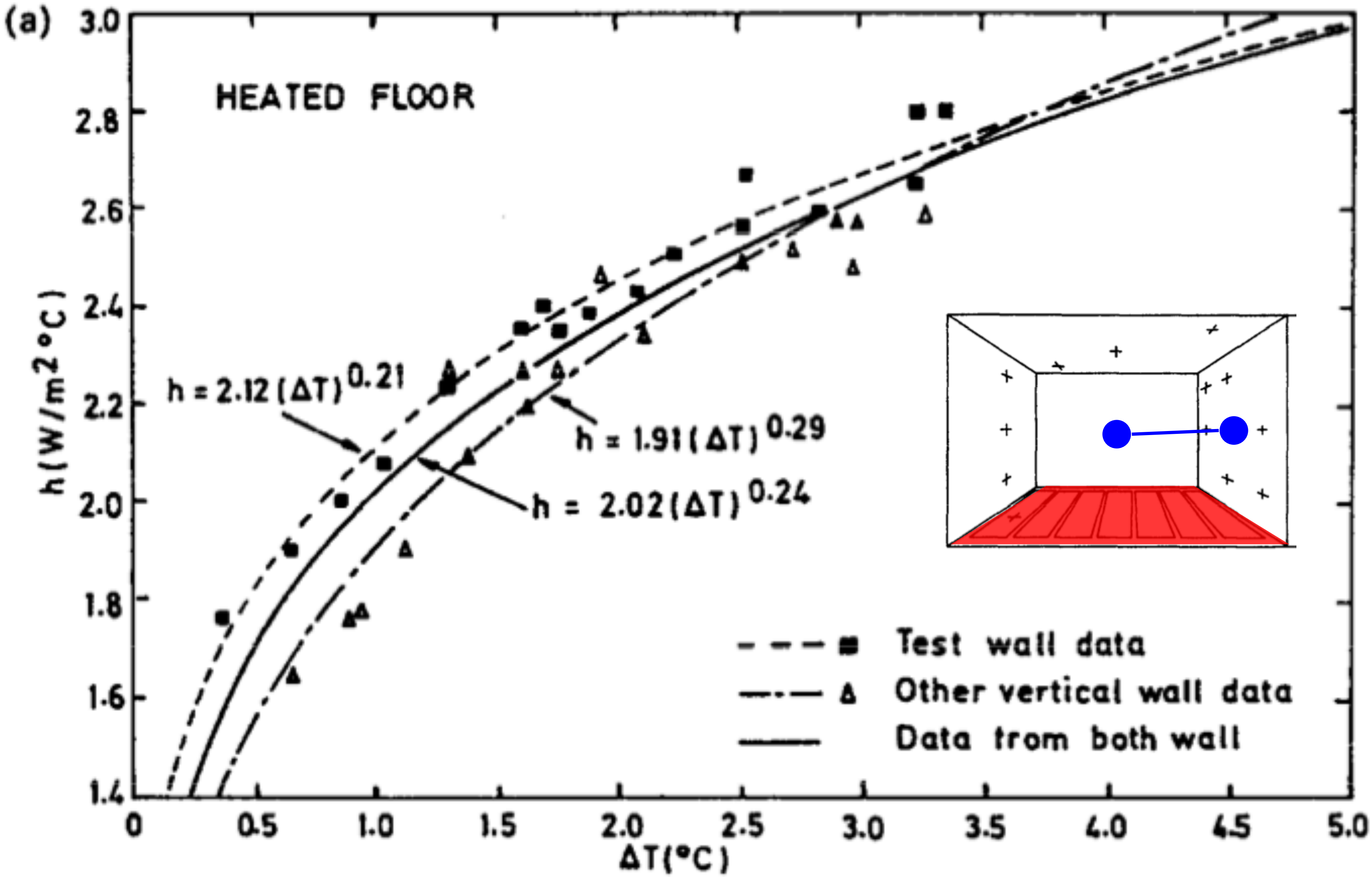
Arrangement	$W/(m^2 \cdot K)$	$Btu/(h \cdot ft^2 \cdot F)$
Air, free convection	6–30	1–5
Superheated steam or air, forced convection	30–300	5–50
Oil, forced convection	60–1800	10–300
Water, forced convection	300–6000	50–1000
Water, boiling	3000–60,000	500–10,000
Steam, condensing	6000–120,000	1000–20,000

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

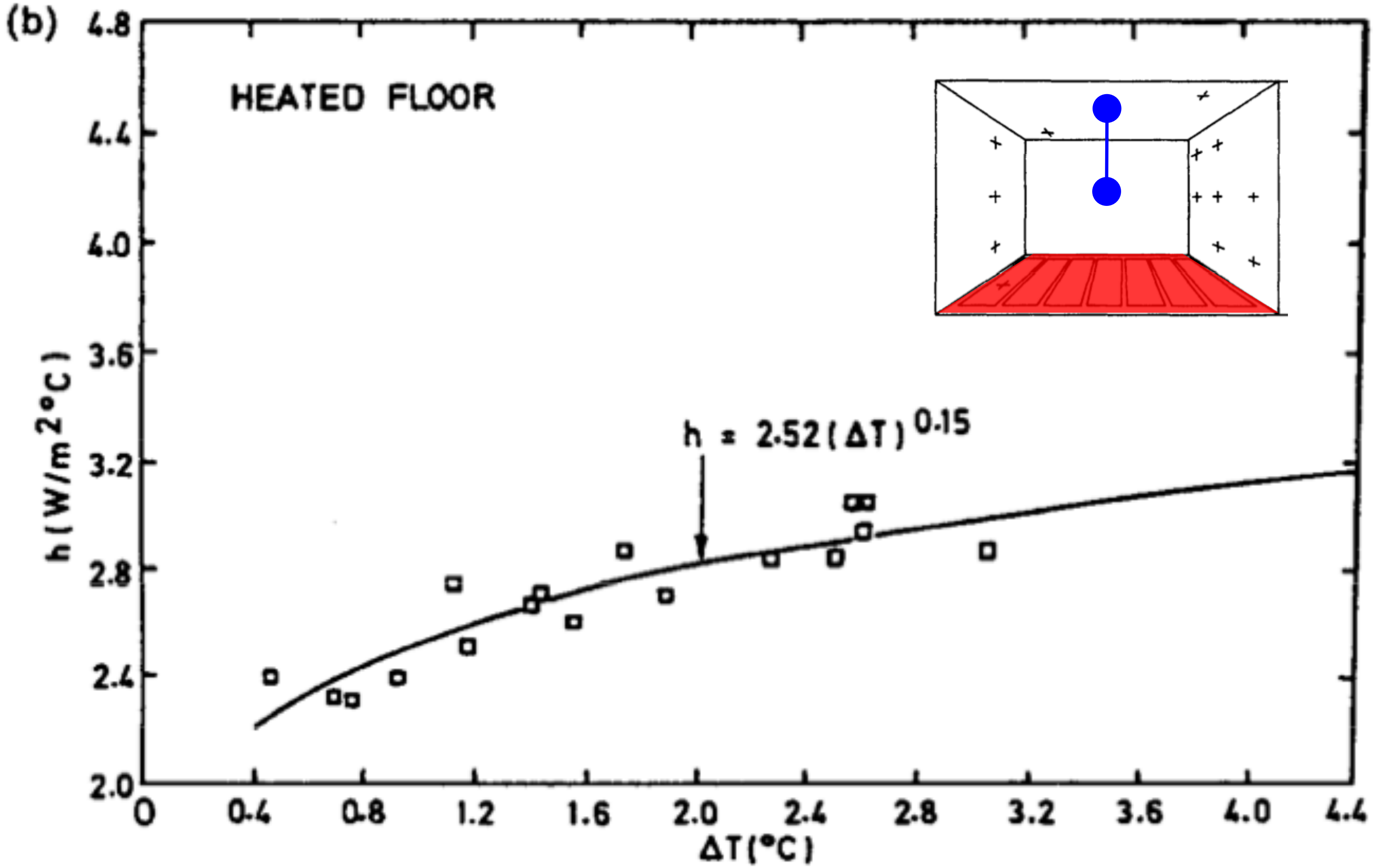
- Convective heat transfer coefficients can depend upon details of the surface-fluid interface
  - Rough surfaces have higher rates of convection
  - Orientation is important for natural convection
  - Convective heat transfer coefficients for natural convection can depend upon the actual fluid temperature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

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



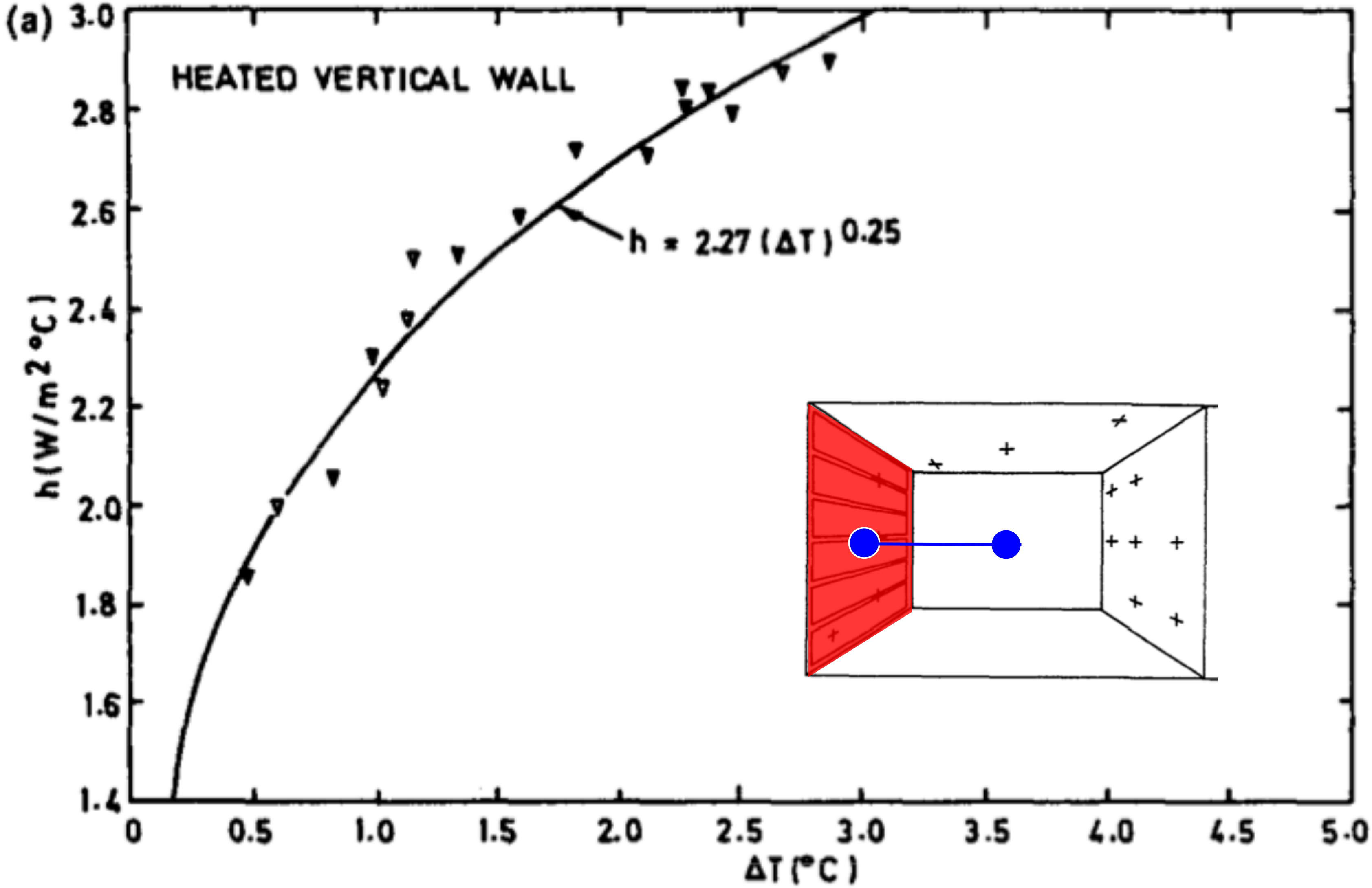
Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

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



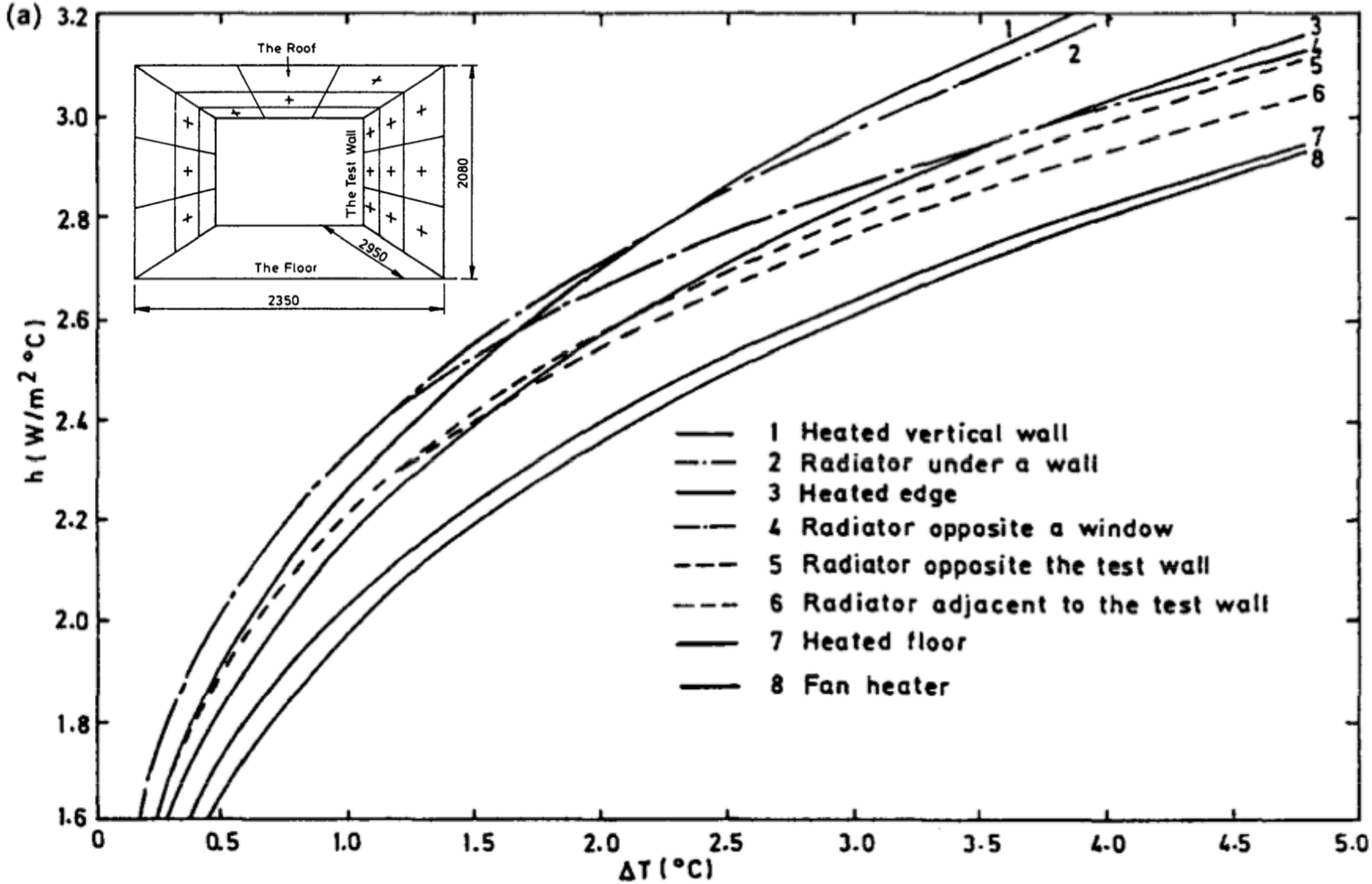
Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

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



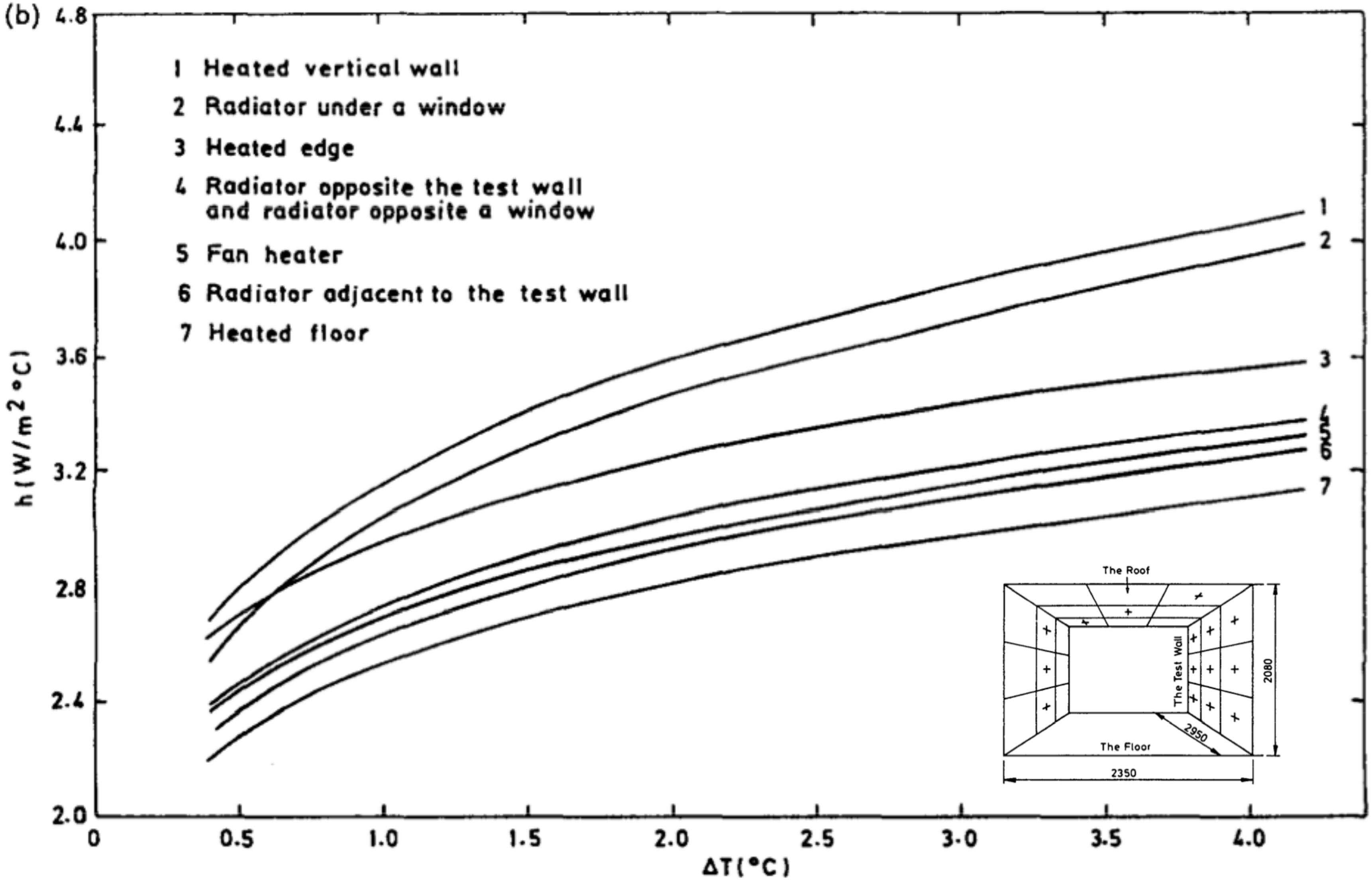
Khalifa and Marshall (1990) *Int J Heat Mass Transfer*

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



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

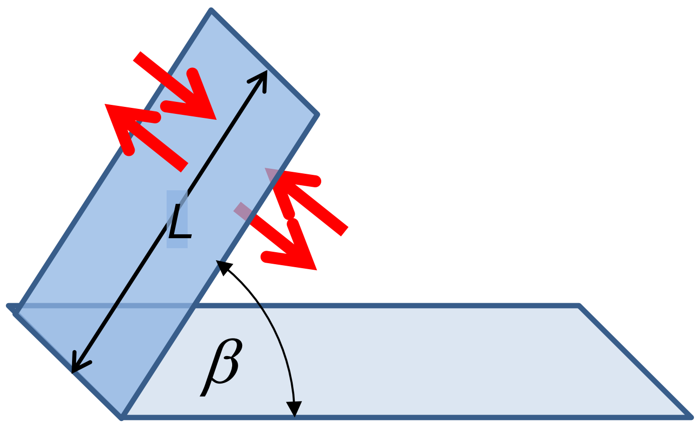
Empirical: h_{conv} vs. ΔT for interior **ceilings**



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

Free convection in air from a tilted surface: Simplified

SI units (IP equations are different! – see ASHRAE HOF Ch. 4)



h_{conv} in [W/(m^2 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]

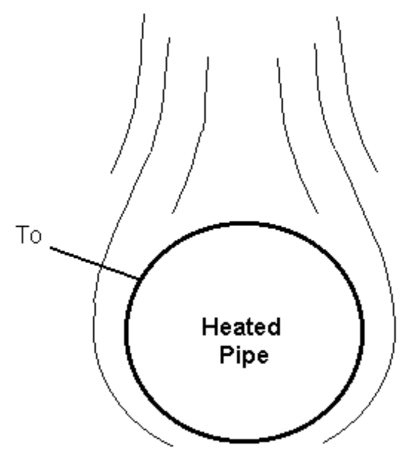
Note that these equations are ***dimensional***, so they are different for IP and SI

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]

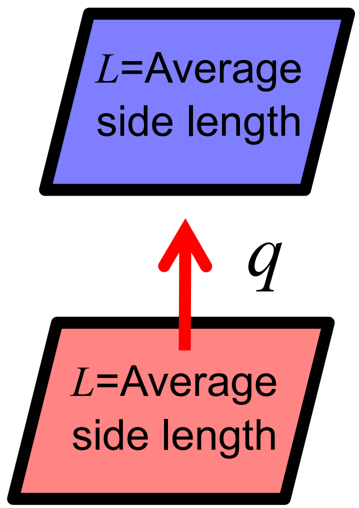


Free Convection Heat Transfer

SI units
(IP equations are different! – see ASHRAE HOF Ch. 4)

Free convection for surfaces: Simplified

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



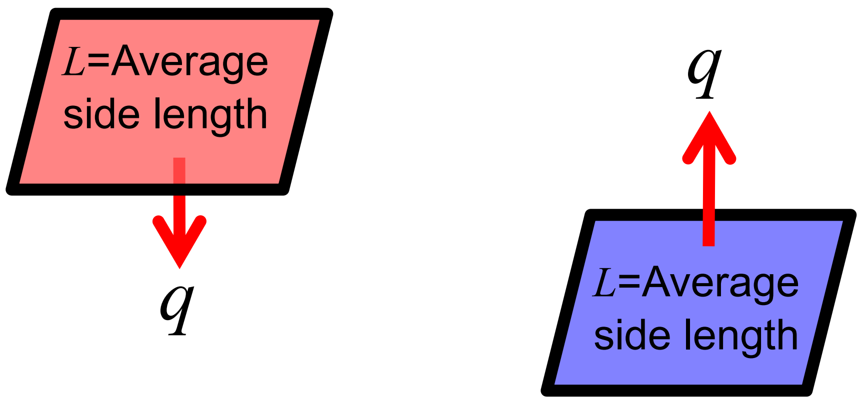
$$\text{laminar:}\quad h_{conv} \approx 1.32\left(\frac{\Delta T}{L}\right)^{1/4} \quad \text{[Kreider 2.22SI]}$$

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

SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

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)

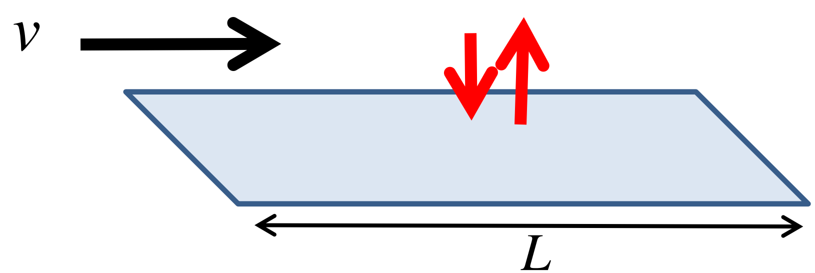


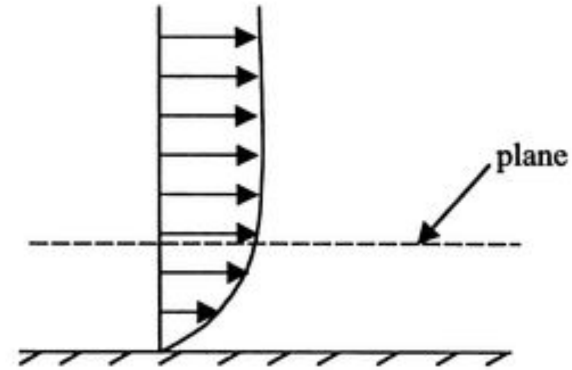
$$h_{conv} \approx 0.59\left(\frac{\Delta T}{L}\right)^{1/4}$$ both laminar and turbulent

SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

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]

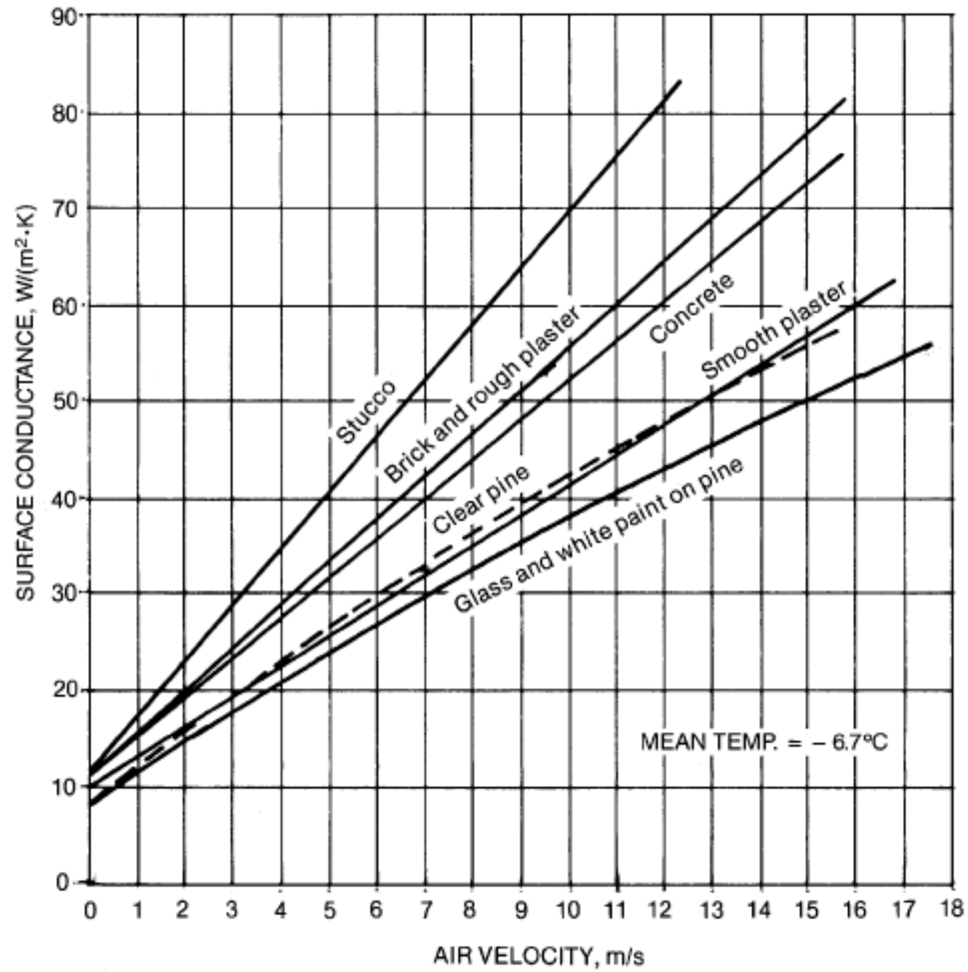
turbulent: $$h_{conv} \approx 6.2\left(\frac{v^4}{L}\right)^{1/5}$$ [Kreider 2.25SI]

*Velocity is in m/s

SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

h_{conv} for exterior forced convection

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



SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

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
  $h_c = 5.6 + 3.9v_{wind}$ [W/(m^2·K)] [Straube 5.15]
- For $5 < v_{wind} < 30$ m/s
  $h_c = 7.2v_{wind}^{0.78}$ [W/(m^2·K)] [Straube 5.16]

*Good for use with external surfaces like walls and windows

SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

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

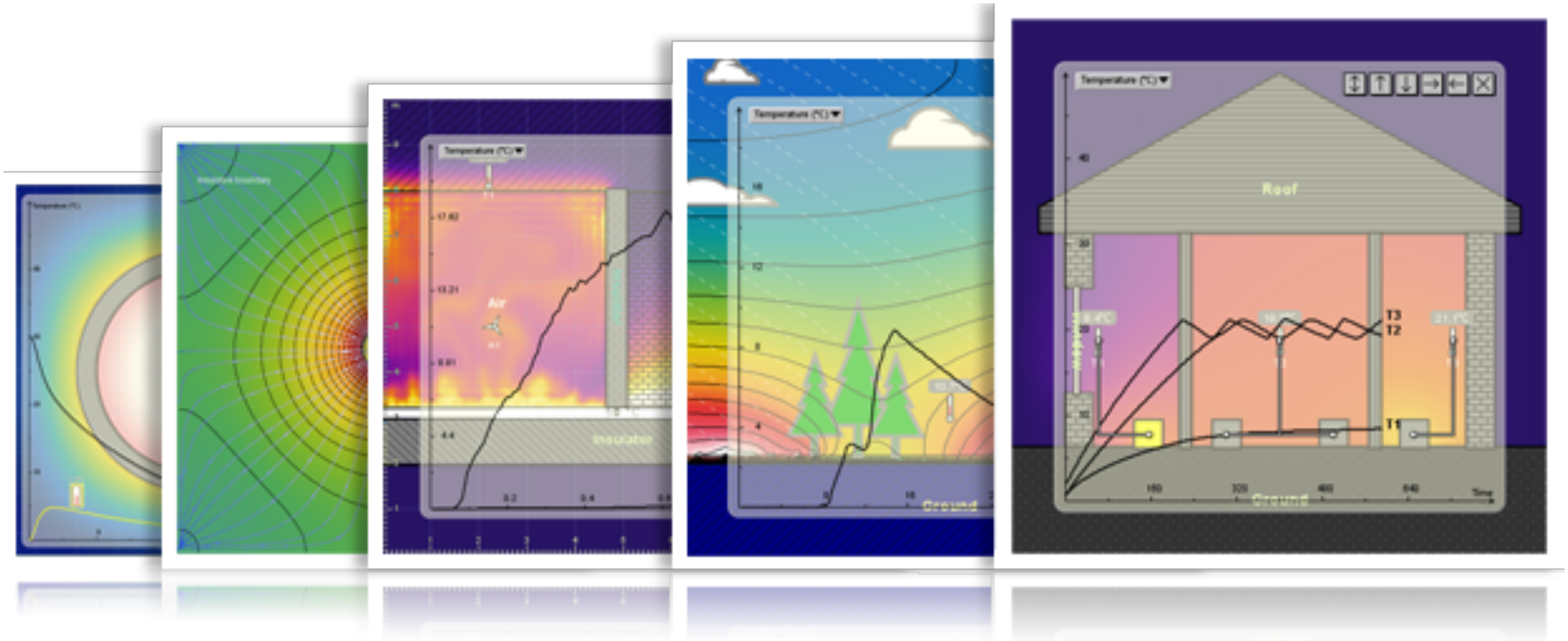
$$D_h = \frac{4\left(\frac{\pi D^2}{4}\right)}{\pi D} \qquad \text{[Kreider 2.27SI]}$$

- Water flow through pipes:

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

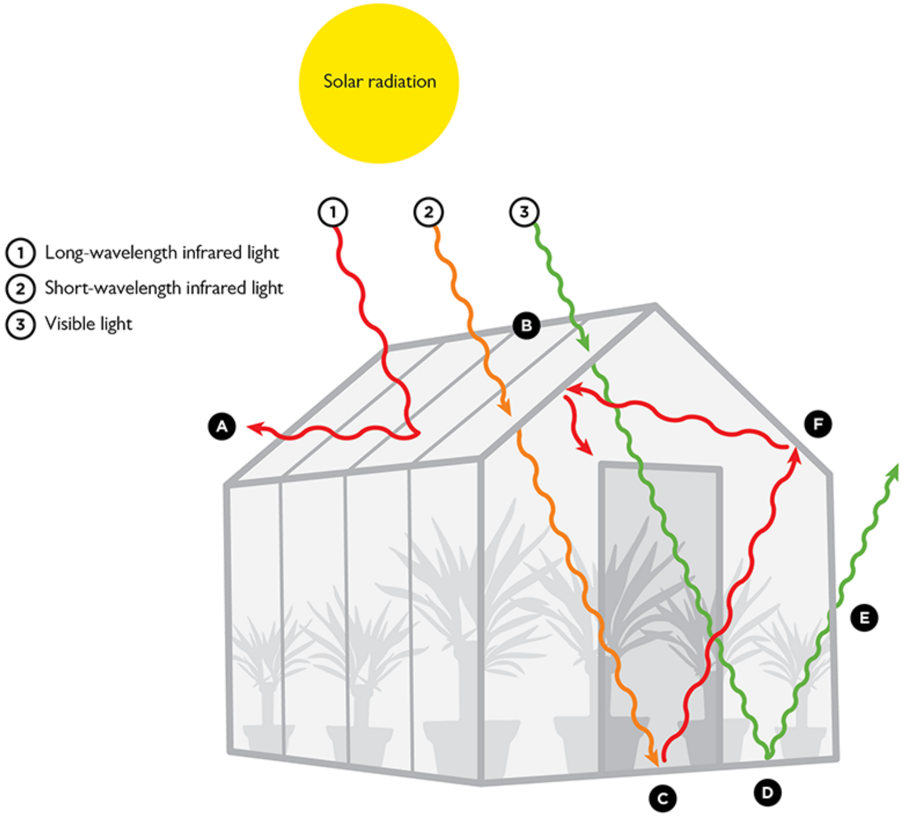
SI units (IP equations are different! – see ASHRAE HOF Ch. 4)

Convection visualizations



Energy2D

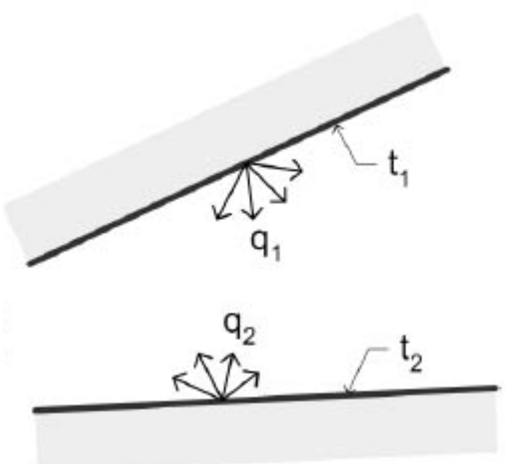
Interactive Heat Transfer Simulations for Everyone

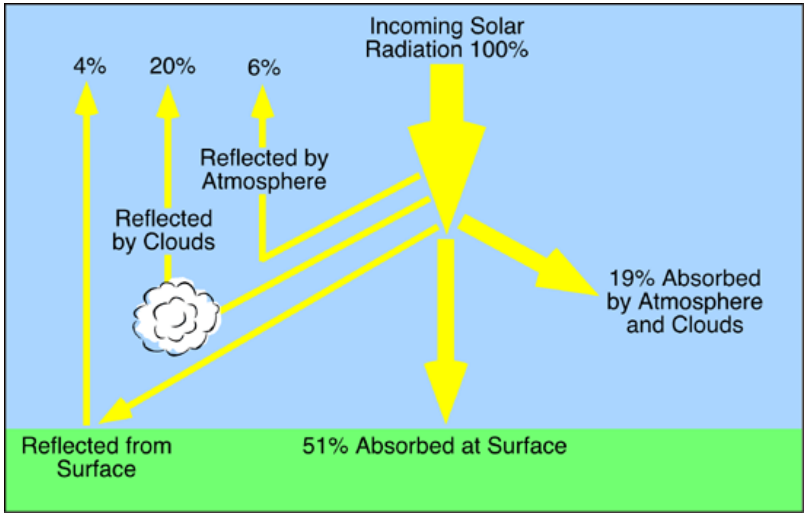

Solar radiation
1
2
3
1 Long-wavelength infrared light
2 Short-wavelength infrared light
3 Visible light
A
B
C
D
E
F

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





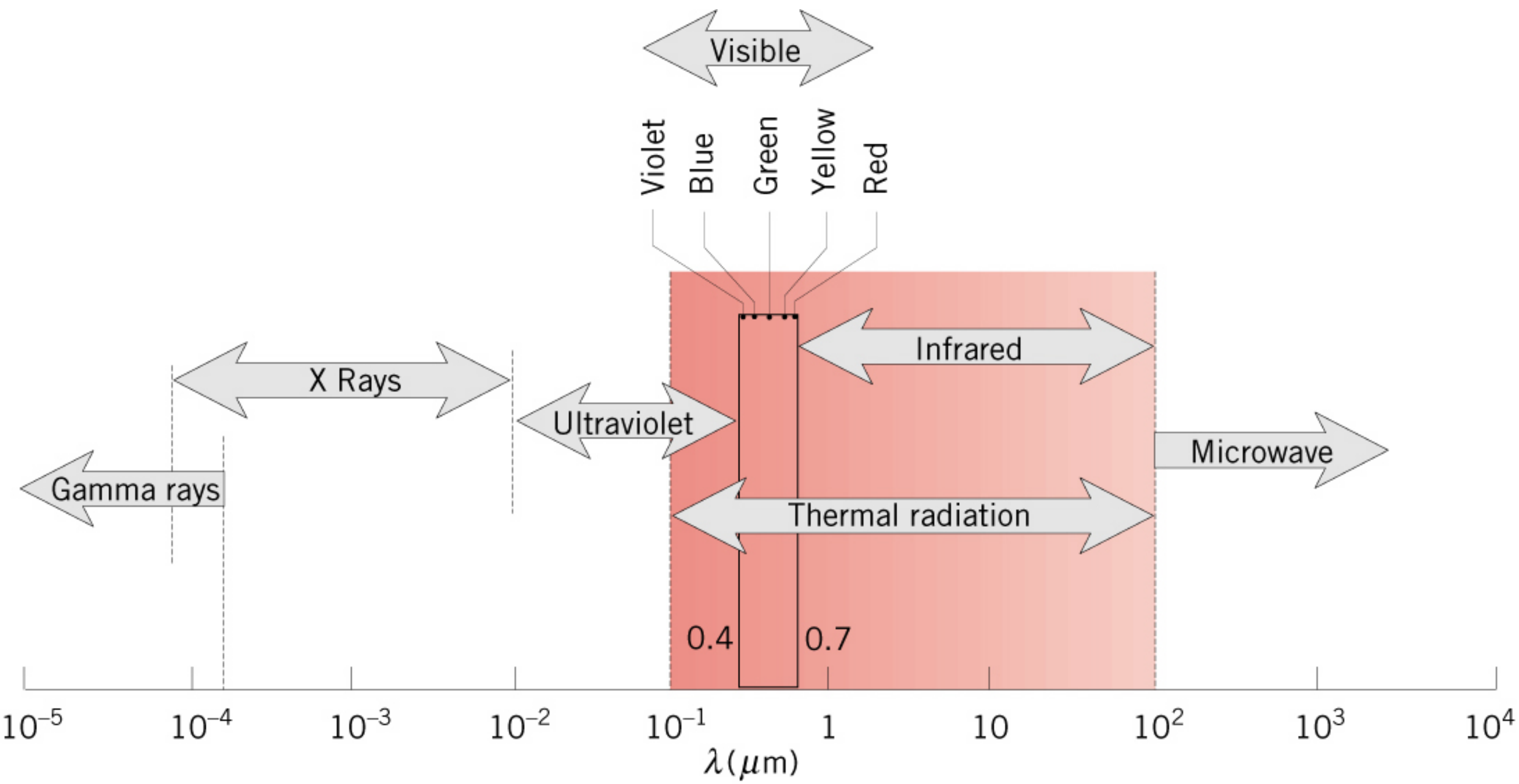
Radiation

- Radiation needs to be dealt with in terms of wavelength (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:
  - Short-wave (solar radiation)
  - Long-wave (emitted and re-emitted radiation)

Radiation: the electromagnetic spectrum

- Thermal radiation is confined to the infrared, visible, and ultraviolet regions ($0.1 < \lambda < 100$ μm)



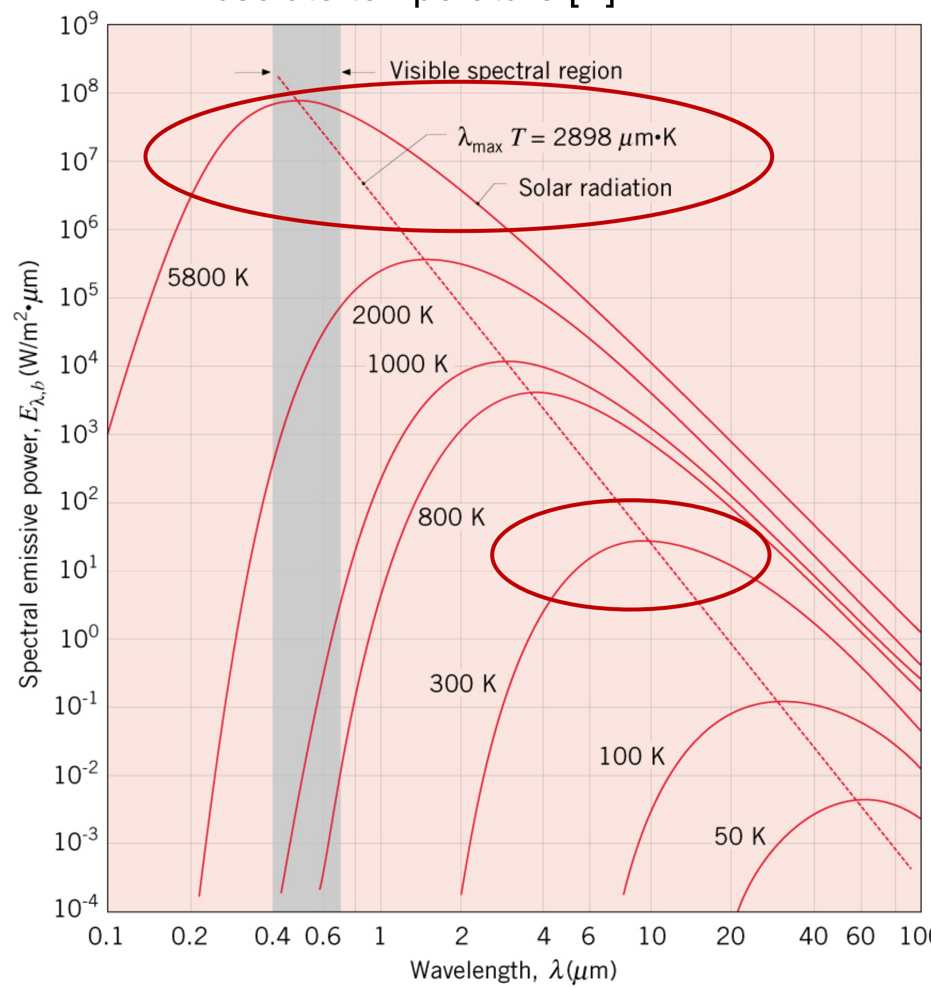
Black body radiation: Spectral (Planck) distribution

- 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 = $5.67 \times 10^{-8} \frac{\mathrm{W}}{\mathrm{m^2 \cdot K^4}}$

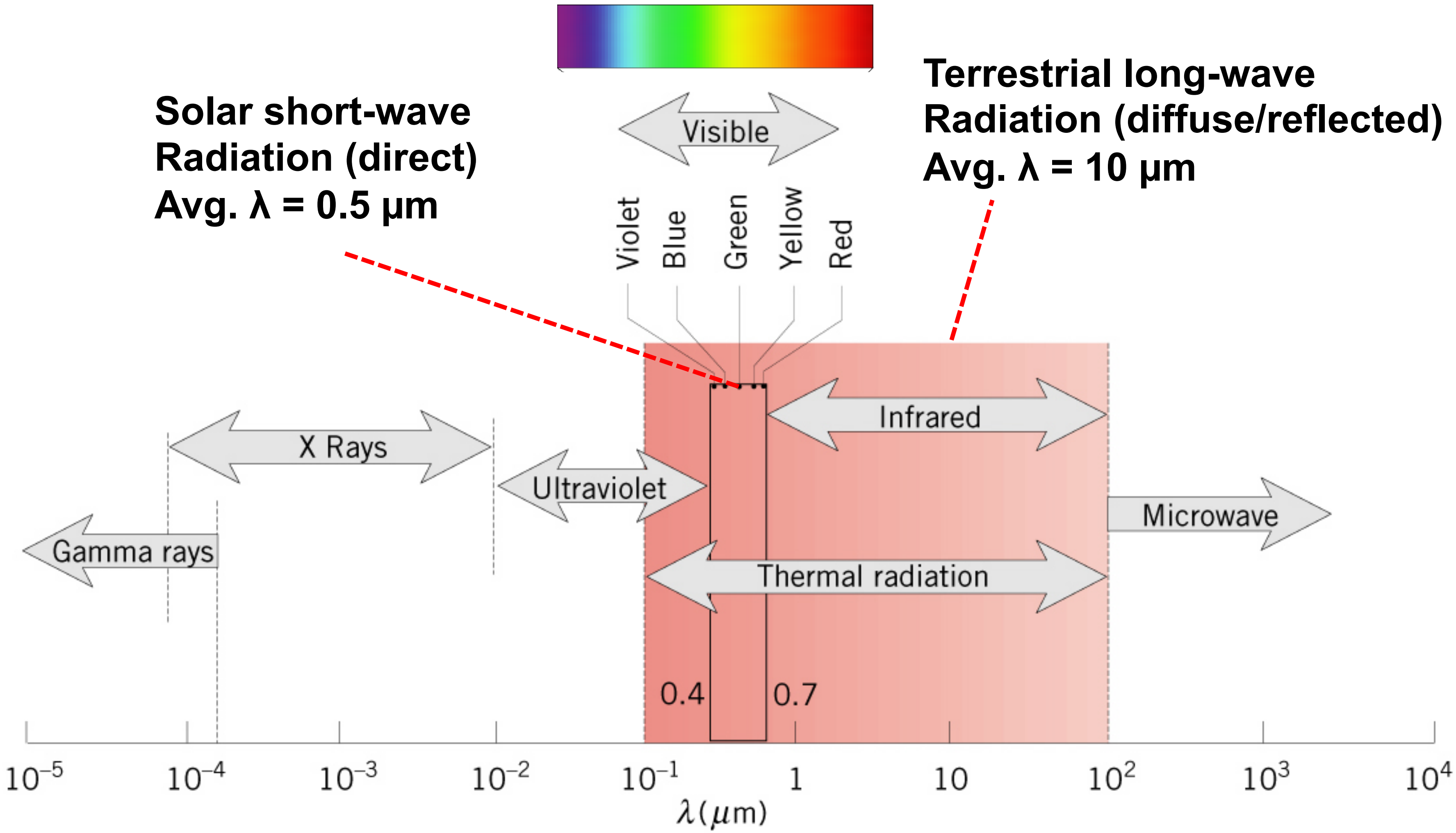
T = Absolute temperature [K]

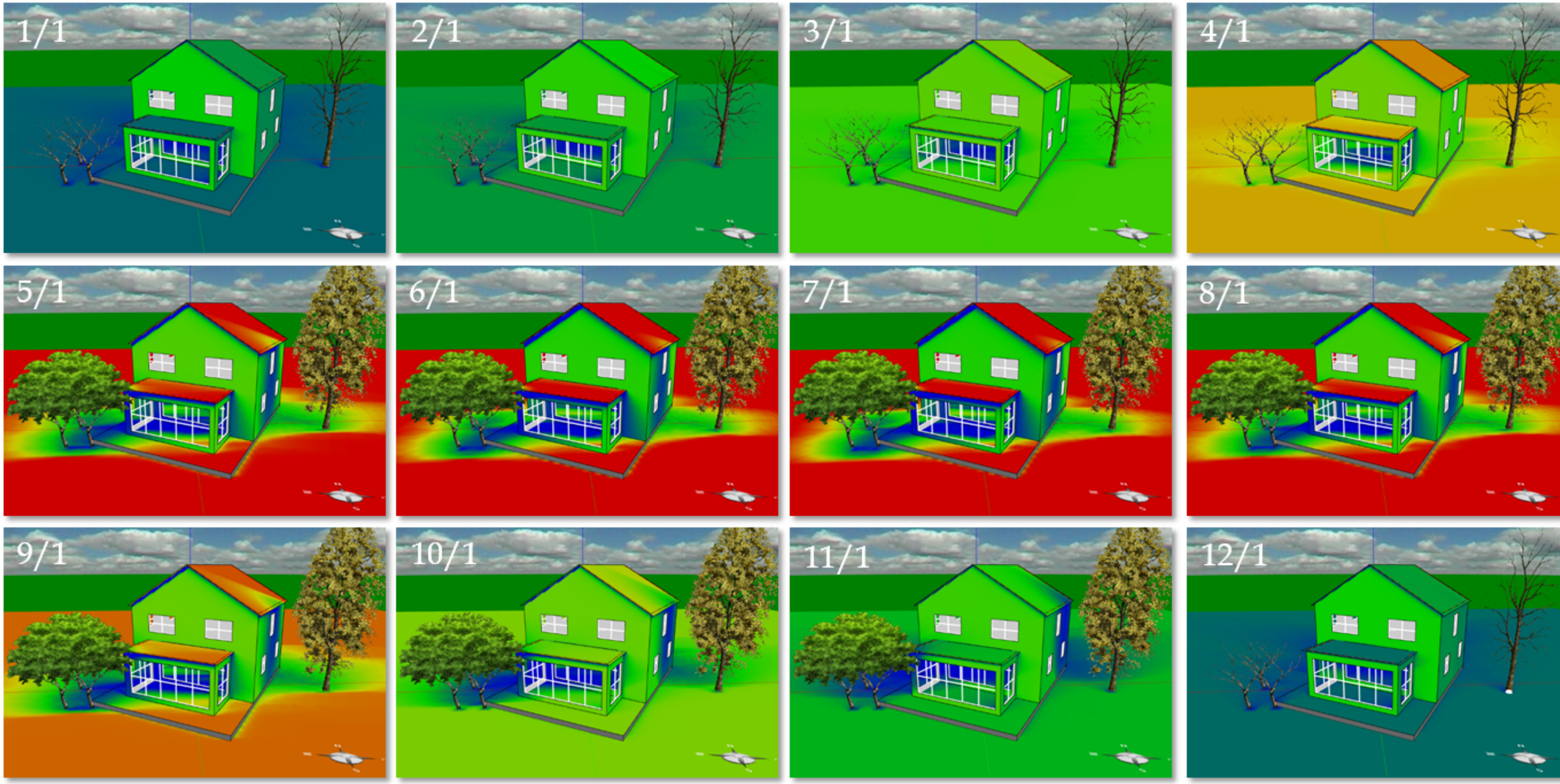


Radiation: Short-wave and Long-wave

Solar short-wave Radiation (direct) Avg. λ = 0.5 μm

Terrestrial long-wave Radiation (diffuse/reflected) Avg. λ = 10 μm





SOLAR (SHORT-WAVE) RADIATION

Solar radiation

- Solar radiation is a very important term in the energy balance of a building
  - We must account for it while calculating loads
  - This is particularly true for perimeter zones and for peak cooling loads
- Solar radiation is also important for daylighting design
- We won't cover the full equations for predicting solar geometry and radiation striking a surface in this class
  - CAE 463/524 Building Enclosure Design goes into more detail
  - But will discuss basic relationships and where to get solar data

Solar radiation striking a surface (high temperature)

- Most solar radiation is at short wavelengths

Solar radiation striking a surface:

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

Solar radiation striking a surface (high temperature)

- Solar radiation data (I_{solar}) can be used on **opaque surfaces** to help determine **surface temperatures**

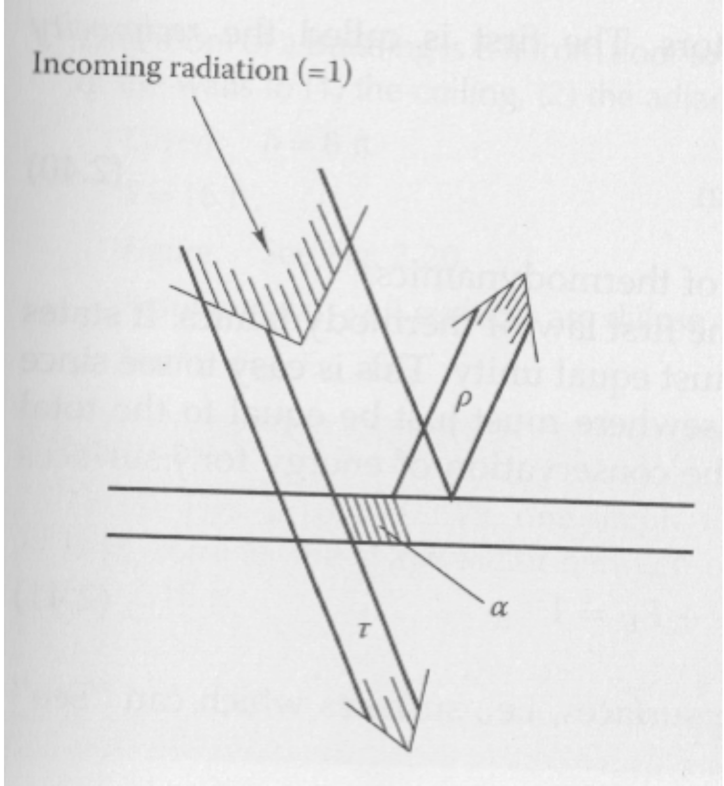
$$q_{solar} = \alpha I_{solar}$$

- Solar radiation data (I_{solar}) can also be used on exterior **fenestration** (e.g. **windows and skylights**) to determine how much solar radiation enters an indoor environment

$$q_{solar} = \tau I_{solar}$$

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 ($\tau = 0$): $q_{solar} = \alpha I_{solar}$
  - For a transparent surface ($\tau > 0$): $q_{solar} = \tau I_{solar}$



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

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

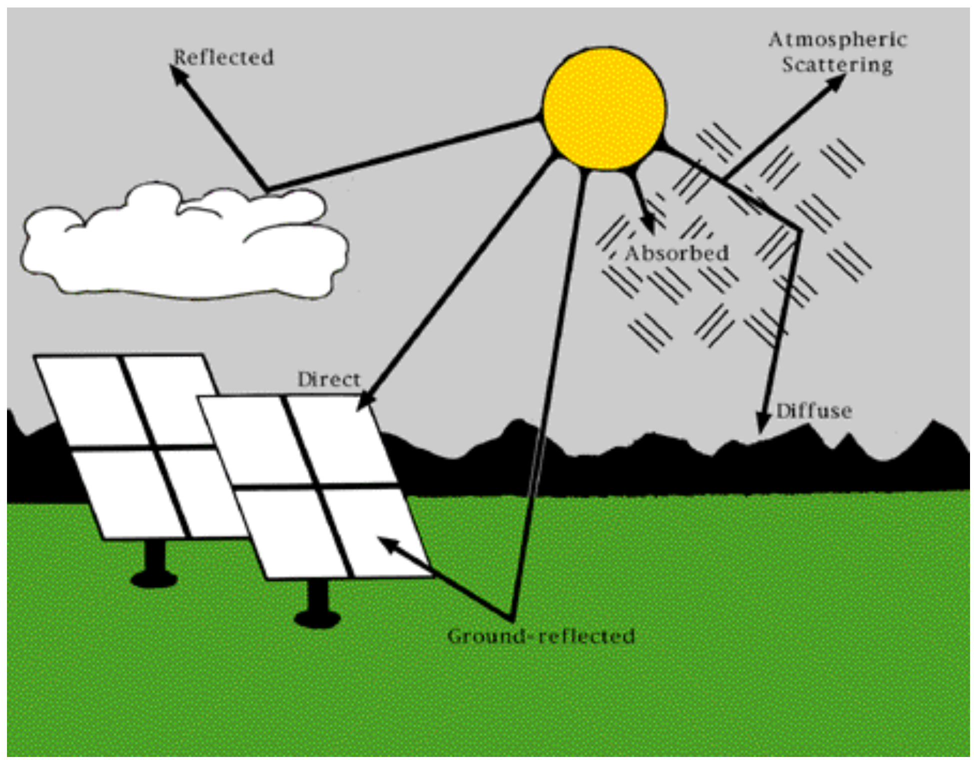
Surface	Absorptance for Solar Radiation
A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99
Black nonmetallic surfaces such as asphalt, carbon, slate, paint, paper	0.85 to 0.98
Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.65 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

Components of solar radiation (I_{solar})

- Solar radiation striking a surface consists of three main components:

$$I_{solar} = I_{direct} + I_{diffuse} + I_{reflected} \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2}\right]$$

- Direct
- Diffuse
- Reflected



Components of solar radiation

- **Direct solar radiation** (I_{direct}) is a function of the **normal incident irradiation** (I_{DN}) on the earth's surface and the solar incidence angle of the surface of interest, θ
  - Where I_{DN} is the amount of solar radiation received per unit area by a surface that is always perpendicular to the sun's direct rays
  - Function of day of the year and atmospheric properties

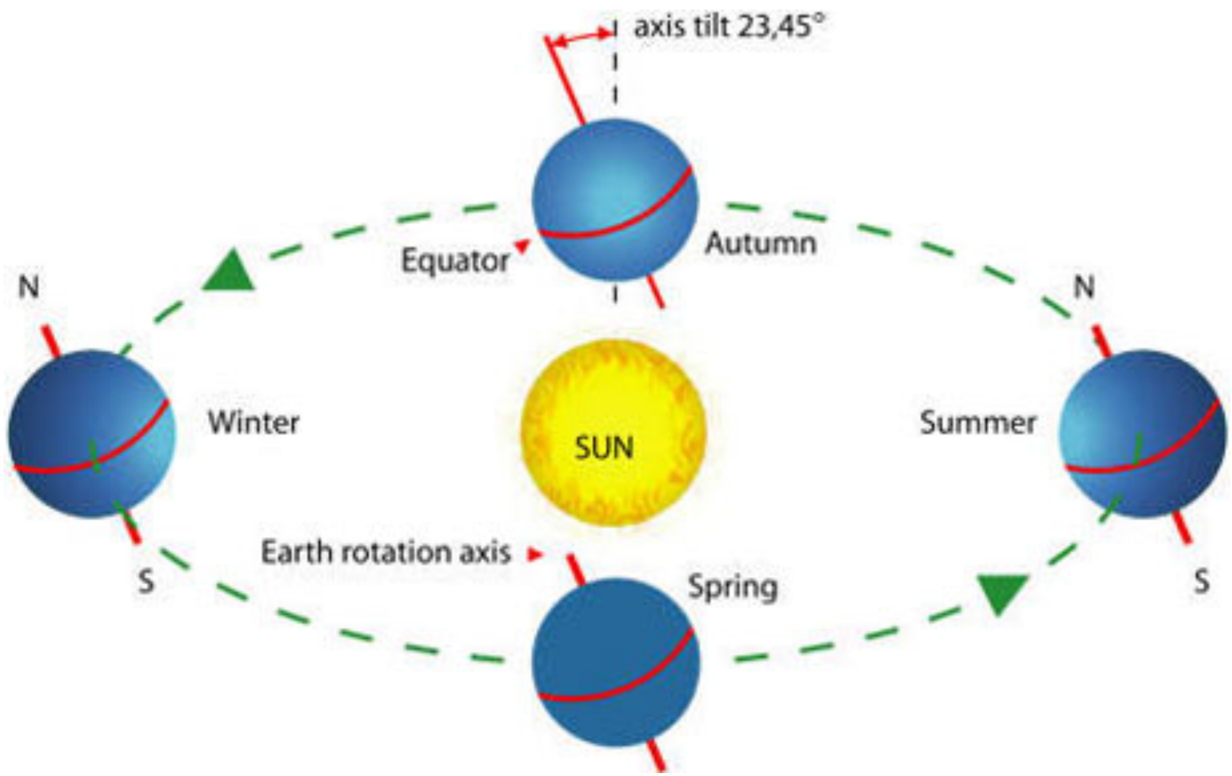
$$I_D = I_{DN} \cos\theta$$

- **Diffuse solar radiation** ($I_{diffuse}$) is the irradiation that is **scattered** by the atmosphere
  - Function of I_{DN}, atmospheric properties, and surface's tilt angle

- **Reflected solar radiation** ($I_{reflected}$) is the irradiation that is **reflected** off the ground (it becomes diffuse)
  - Function of I_{DN}, solar geometry, ground reflectance, and surface tilt angle

Solar radiation: earth-sun relationships

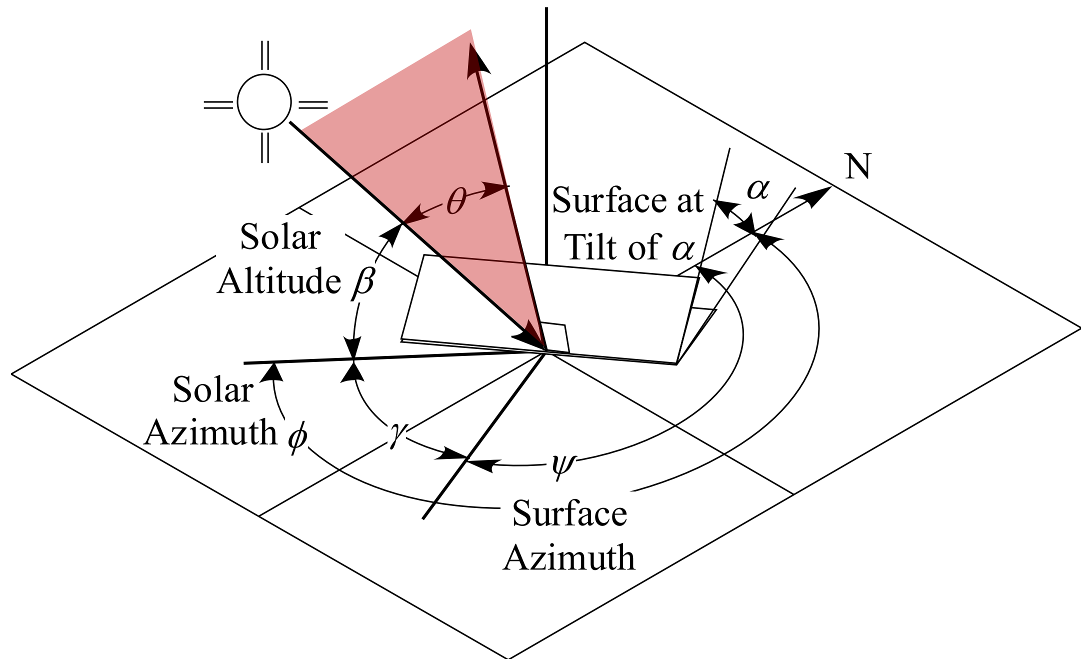
- Earth rotates about its axis every 24 hours
- Earth revolves around sun every 365.2425 days
- Earth is titled at an angle of 23.45°
  - Therefore, different locations on earth receive different levels of solar radiation during different times of the year (and different times of the day)



Solar radiation striking an exterior surface

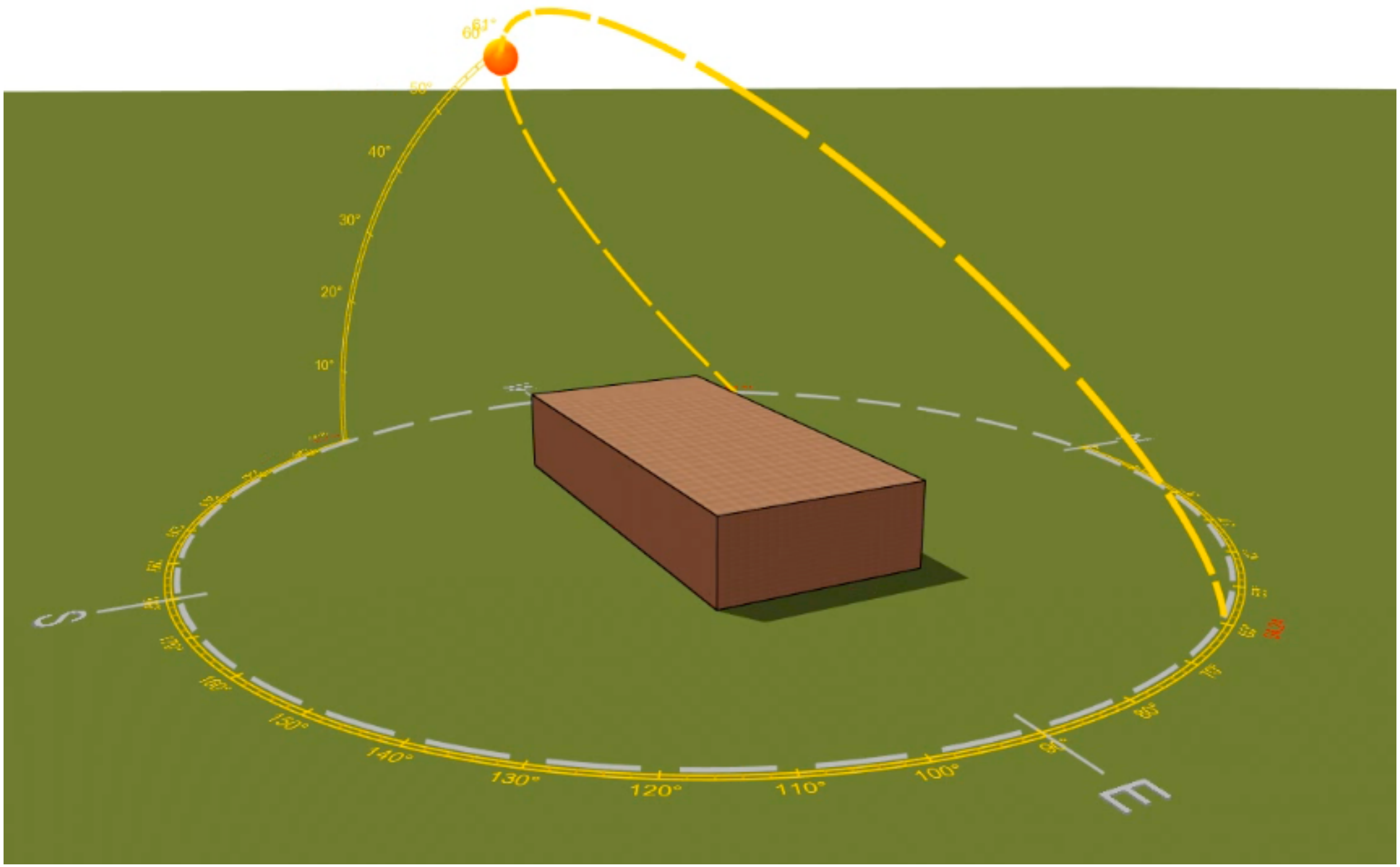
- The amount of solar radiation received by a surface depends on the **incidence angle**, θ

- This is a function of:
  - Solar geometry (I_{DN})
    - Location
    - Time
  - Surface geometry
  - Shading/obstacles



$$I_D = I_{DN} \cos\theta$$

Visualizing solar relationships



http://energy.concord.org/energy3d/

Downloading solar data

- For hourly sun positions, you can build a calculator or use one from the internet
  - http://www.susdesign.com/sunposition/index.php
  - http://www.esrl.noaa.gov/gmd/grad/solcalc/azel.html

- For solar position and intensity (from time and place)
  - http://www.nrel.gov/midc/solpos/solpos.html
  - Output of interest = "global irradiance on a tilted surface"

- For *actual* hourly solar data (direct + diffuse in W/m^2)
  - http://rredc.nrel.gov/solar/old_data/nsrdb/
  - Output of interest = "direct normal radiation" → adjust using $\cos\theta$
    - Note: "typical meteorological years"

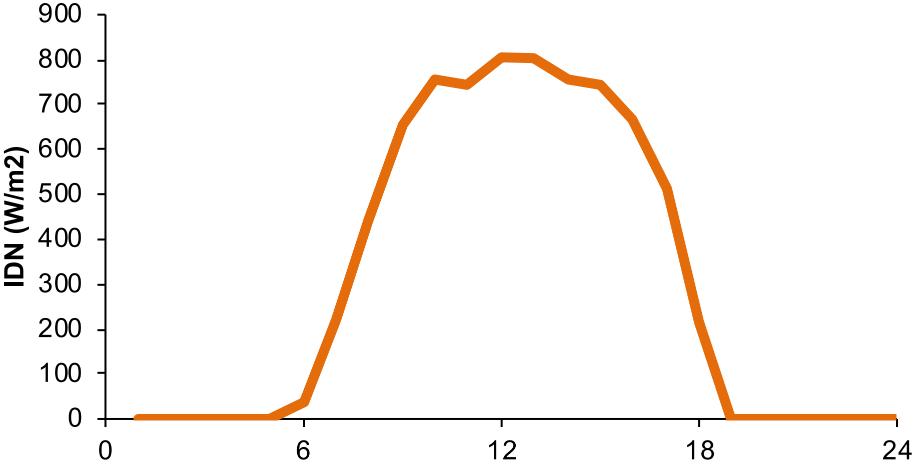
Typical meteorological year (TMY)

- For heating and cooling load calculations and for hourly building energy simulations, we often rely on a collection of weather data for a specific location
- We generate this data to be representative of more than just the previous year
  - Represents a wide range of weather phenomena for our location
  - TMY3: Data for 1020 locations from 1960 to 2005
    - Composed of 12 typical meteorological months
    - Each month is pulled from a random year in the range
    - Actual time-series climate data
    - Mixture of measured and modeled solar values
    - http://rredc.nrel.gov/solar/old_data/nsrdb/1991-2005/tmy3/
  - Variables include: outdoor temperature, direct normal radiation, wind speed, wind direction, outdoor RH, cloud cover, and more

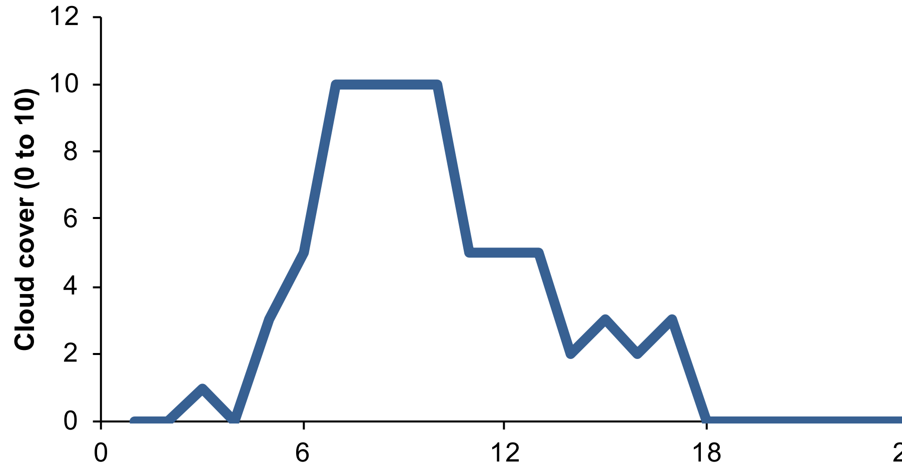
Typical meteorological year (TMY): Solar data

Data for typical September 10th at Midway, Chicago, IL

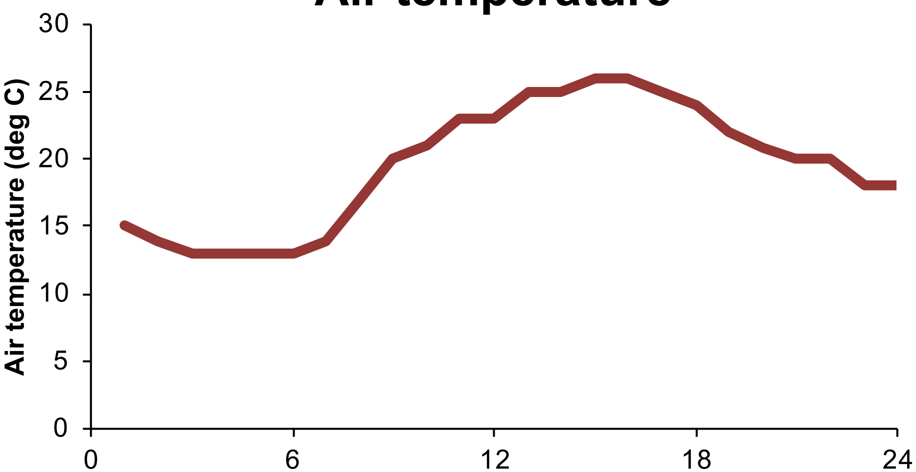
Direct Normal Irradiance



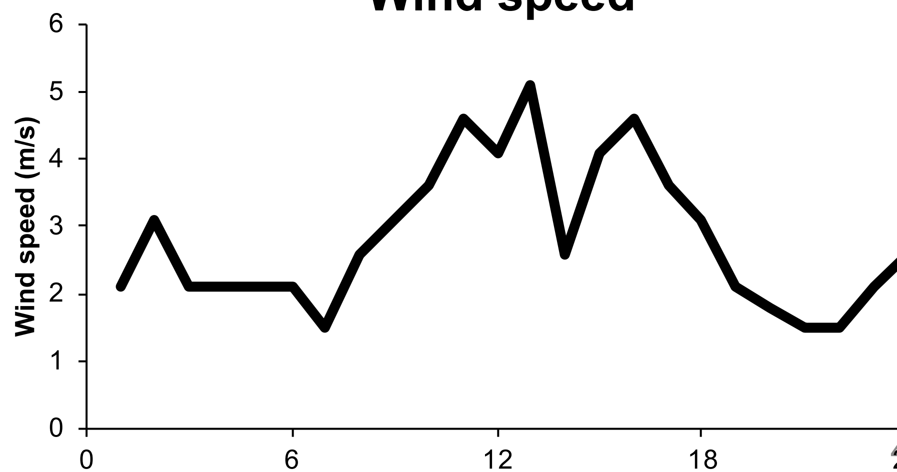
Cloud Cover



Air temperature



Wind speed

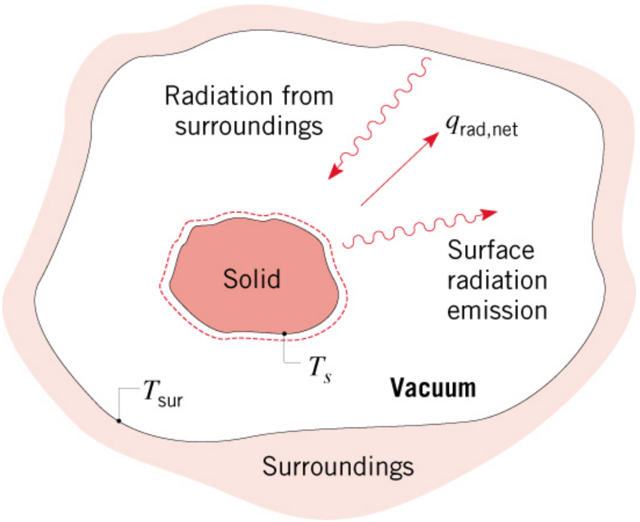


SURFACE (LONG-WAVE) RADIATION

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 = $5.67 \times 10^{-8} \frac{\text{W}}{\text{m}^2 \cdot \text{K}^4}$

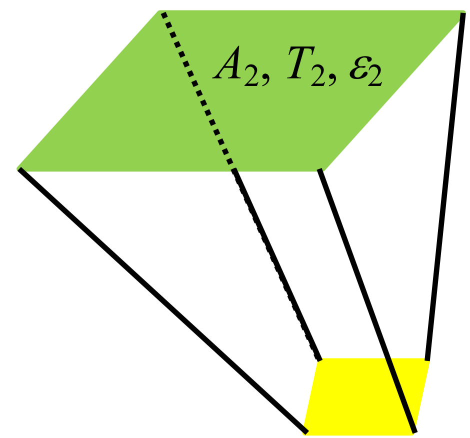
T = Absolute temperature [K]

- 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:

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



where ε_1 and ε_2 are the surface emittances,

A_1 and A_2 are the surface areas

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

$F_{1\rightarrow 2}$ 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

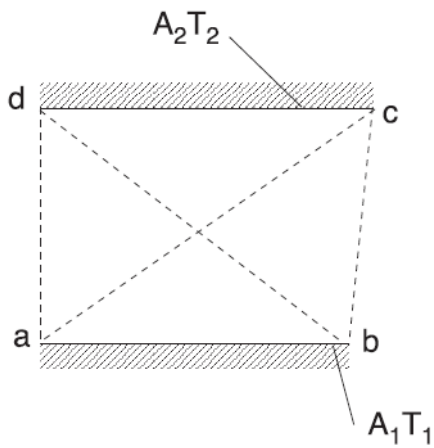
Surface	Temperature, °C	Temperature, °F	ϵ
Brick			
Red, rough	40	100	0.93
Concrete			
Rough	40	100	0.94
Glass			
Smooth	40	100	0.94
Ice			
Smooth	0	32	0.97
Marble			
White	40	100	0.95
Paints			
Black gloss	40	100	0.90
White	40	100	0.89–0.97
Various oil paints	40	100	0.92–0.96
Paper			
White	40	100	0.95
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:



$$A_1F_{1\to 2} = 0.5((ac+bd)-(ad+bc))$$

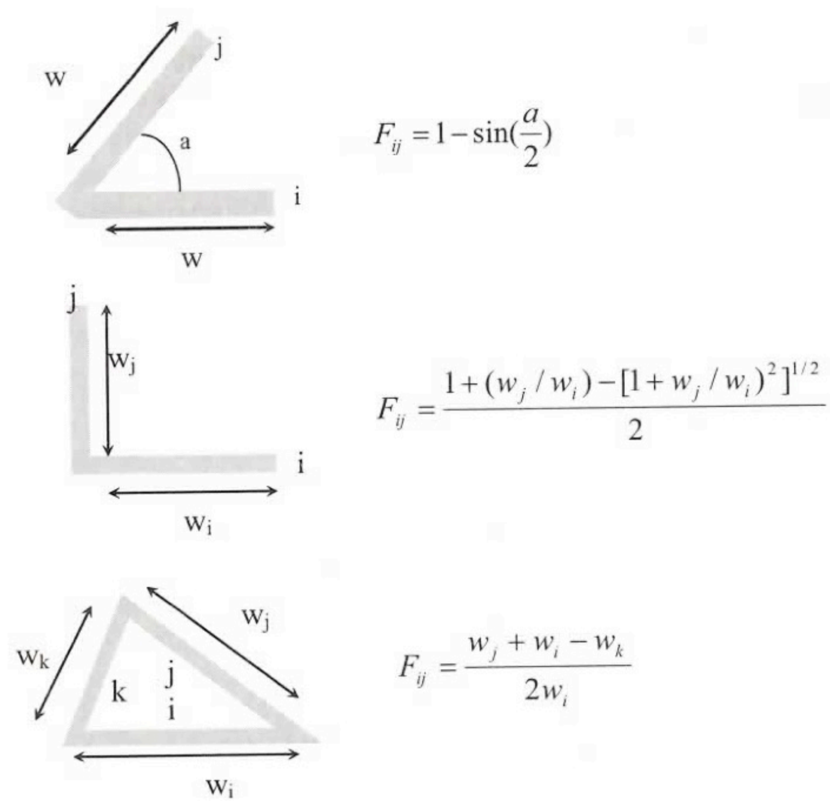
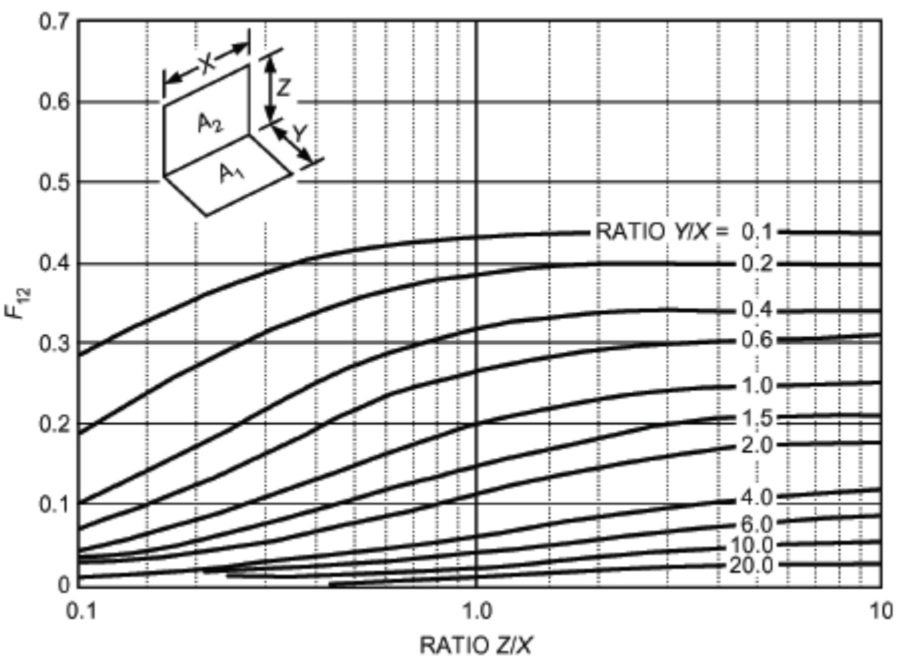


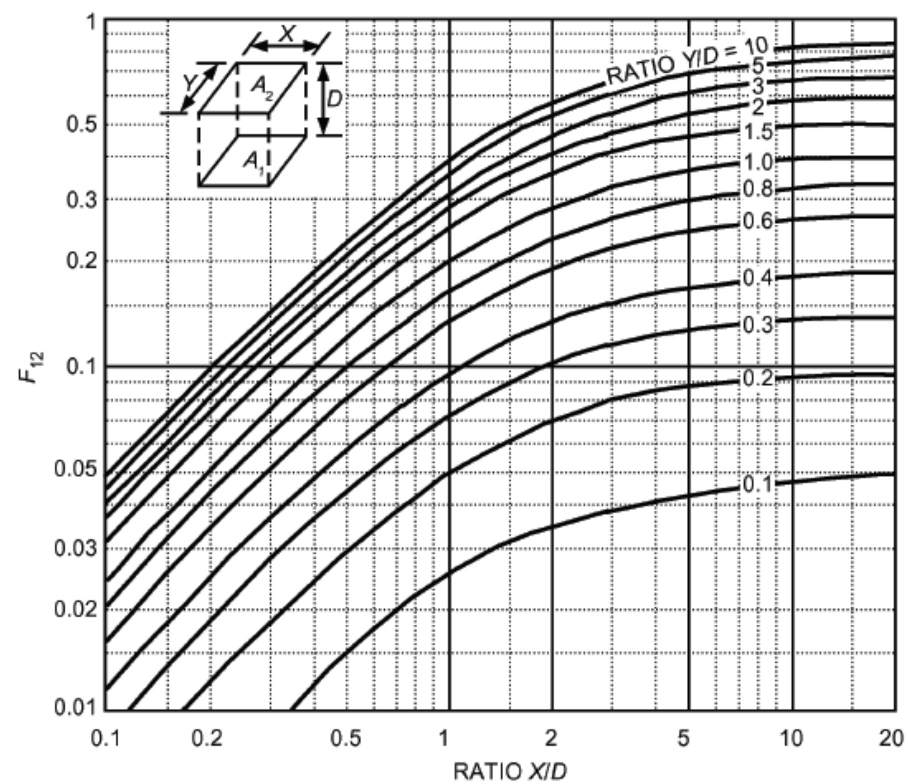
Figure 5.6: View factors for common situations in building enclosures [Hagentoft 2000]

Typical view factors

Other common view factors from the ASHRAE Handbook of Fundamentals:



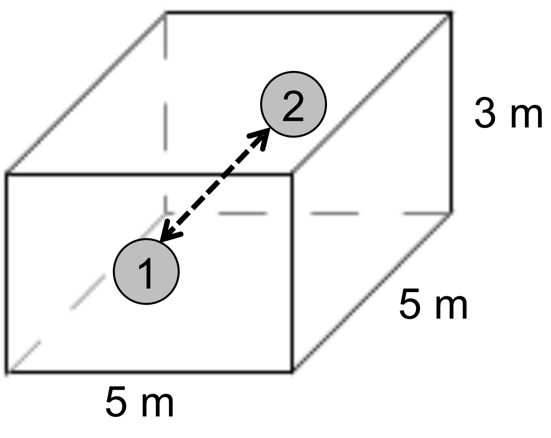
A. PERPENDICULAR RECTANGLES WITH COMMON EDGE

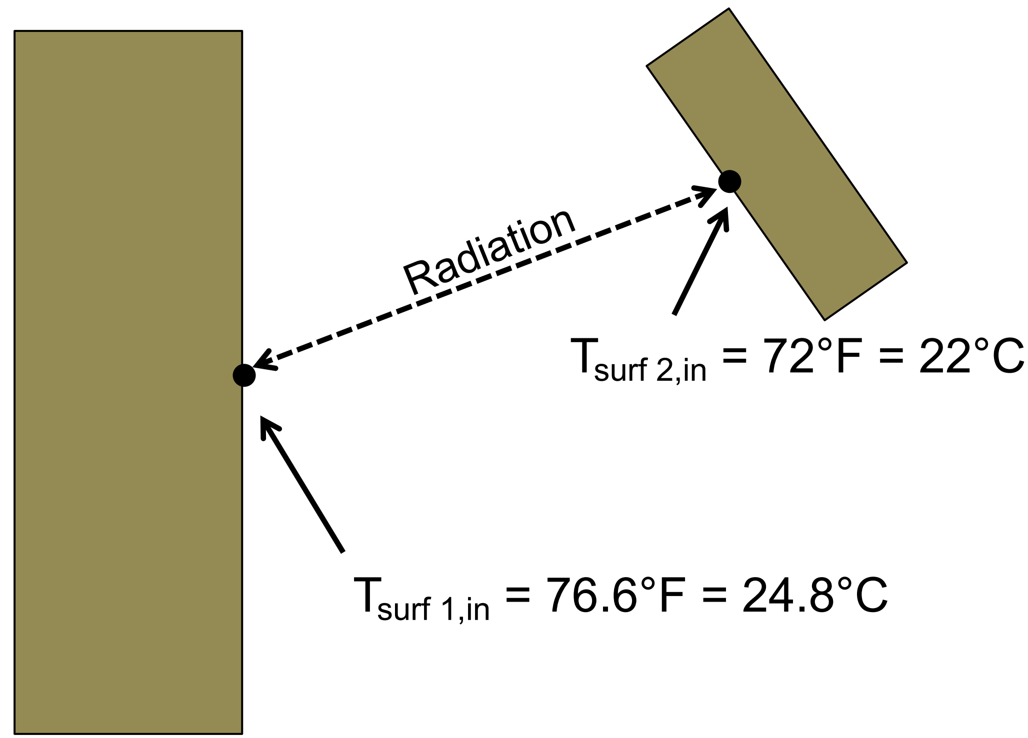


B. ALIGNED PARALLEL RECTANGLES

Long-wave radiation example

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





Q: What if $T_{surf1,in}$ dropped to 50°F (10°C)?

Simplifying surface radiation

- We can also often simplify radiation from:

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

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

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

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

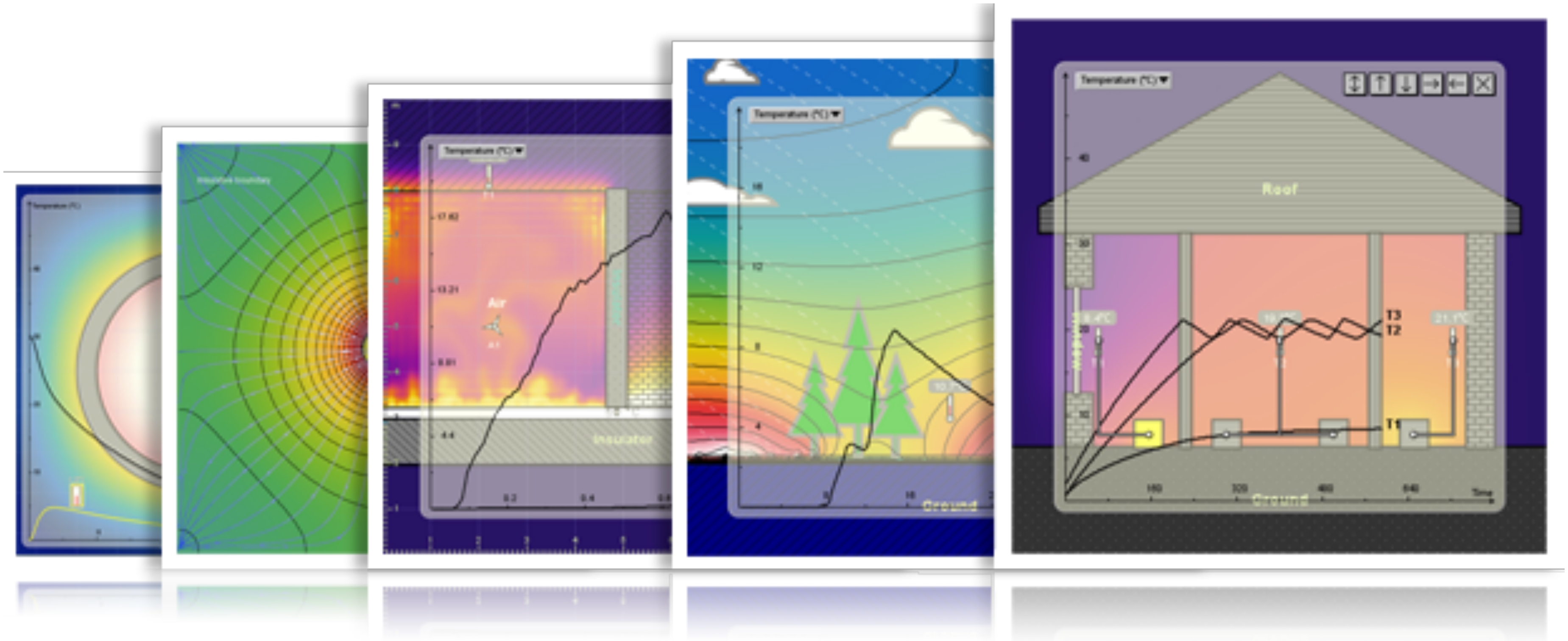
- When $A_1 = A_2$, and T_1 and T_2 are within ~50°F of each other, we can approximate h_{rad} with a simpler equation:

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

where

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

Radiation visualizations



Energy2D

Interactive Heat Transfer Simulations for Everyone

Moving forward

- HW #2 is assigned and available on BB – due Thurs Sep 12
- Next lecture:
  - Combined mode heat transfer