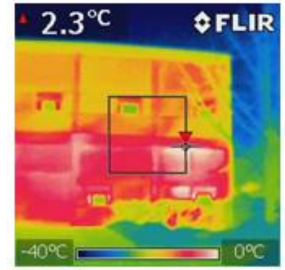
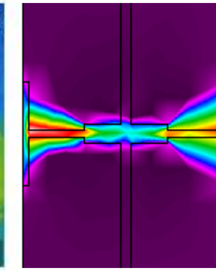
CAE 331/513
Building Science
Fall 2014





Week 7: October 9, 2014
Review for Exam 1

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Illinois Institute of Technology
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Undergraduate exam (CAE 331)

- In-class Tuesday October 14
  - 1:50 pm to 3:05 pm
  - Come a few minutes early, we will start on time
  - Exam will be proctored by our TA Liz Mullin
- Closed book and closed notes
  - Bring a pencil and calculator
  - Equations sheet and unit conversion sheet will be provided
  - Scratch paper will also be given

Graduate exam (CAE 513)

- Longer, more difficult take-home exam
- Exam will be available on Blackboard Tues Oct 14 at 12 pm
- Must complete and return:
  - Either online on BB by **Friday Oct 17 at 4 pm**
  - Or hard copy in person in my office by **Friday Oct 17 at 4 pm**
  - If you turn it in early and I’m not there, just slide under the door
    - Or arrange with our TA Liz to drop off <emullin@hawk.iit.edu>

Topics to prepare for:

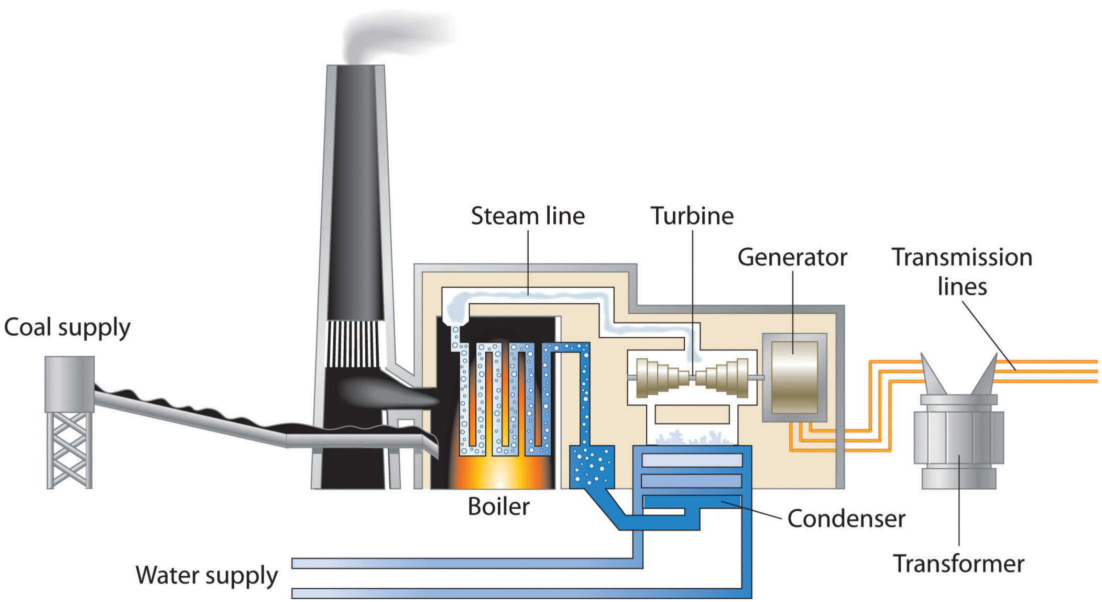
- Buildings and energy use
- Energy conversion
- Unit conversions
- Heat transfer in buildings
  - Conduction
  - Convection
  - Radiation
  - Solar radiation
- Human thermal comfort
- Psychrometrics
  - Chart
  - Equations
  - Processes
- Introduction to HVAC systems

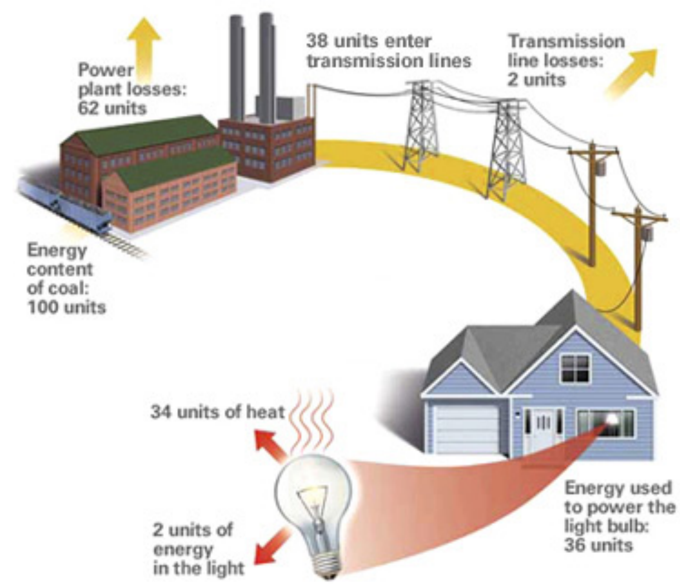
ENERGY BASICS

Conversion efficiency for electric generators

1st law of thermodynamics: Energy can be transformed from one form to another, but cannot be created or destroyed

- What is a typical electric power plant efficiency? **30 - 45%**



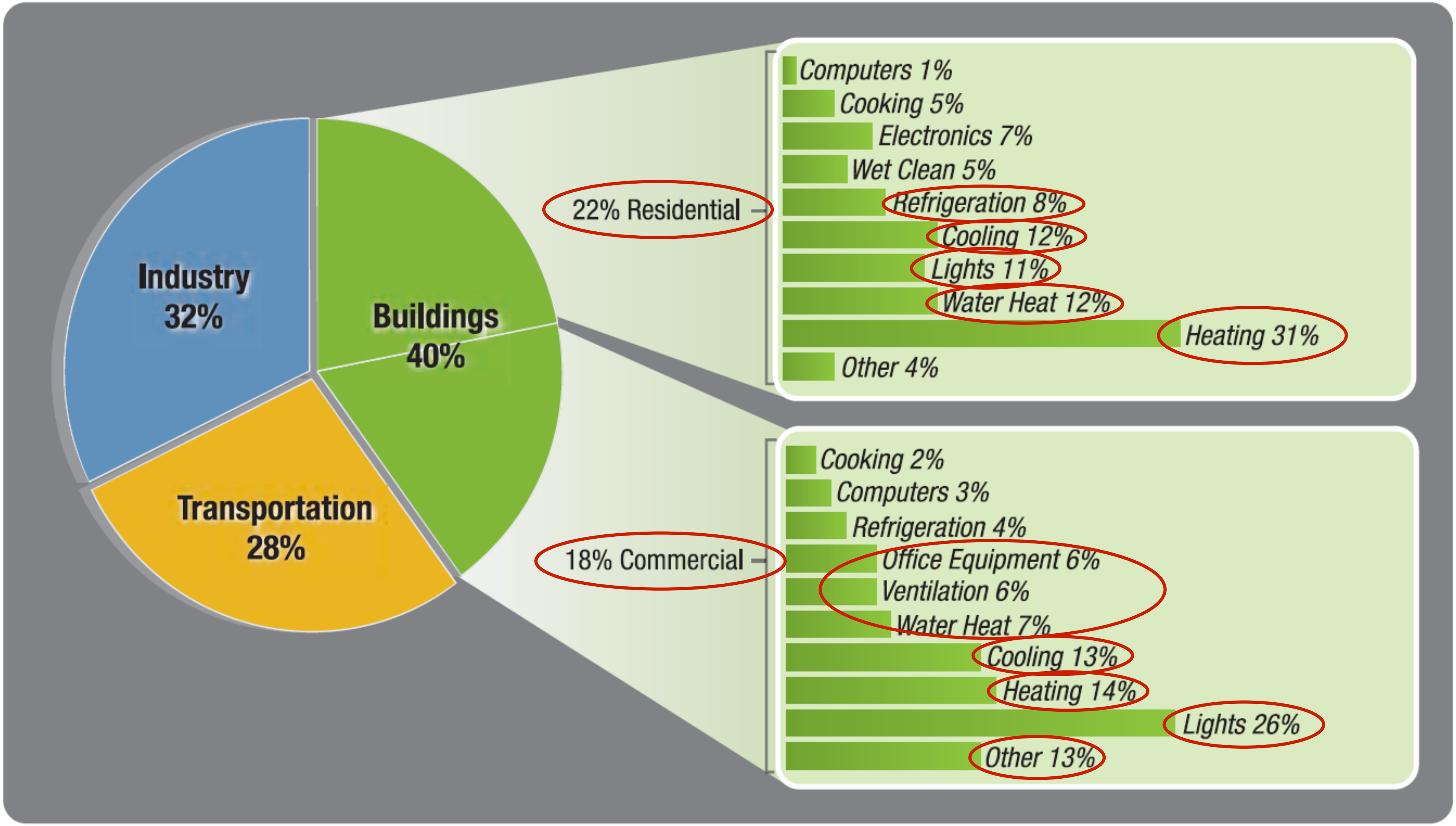


Natural gas used directly in buildings is typically **80 - 90%** efficient or more

http://www.eia.gov/tools/faqs/faq.cfm?id=107&t=3

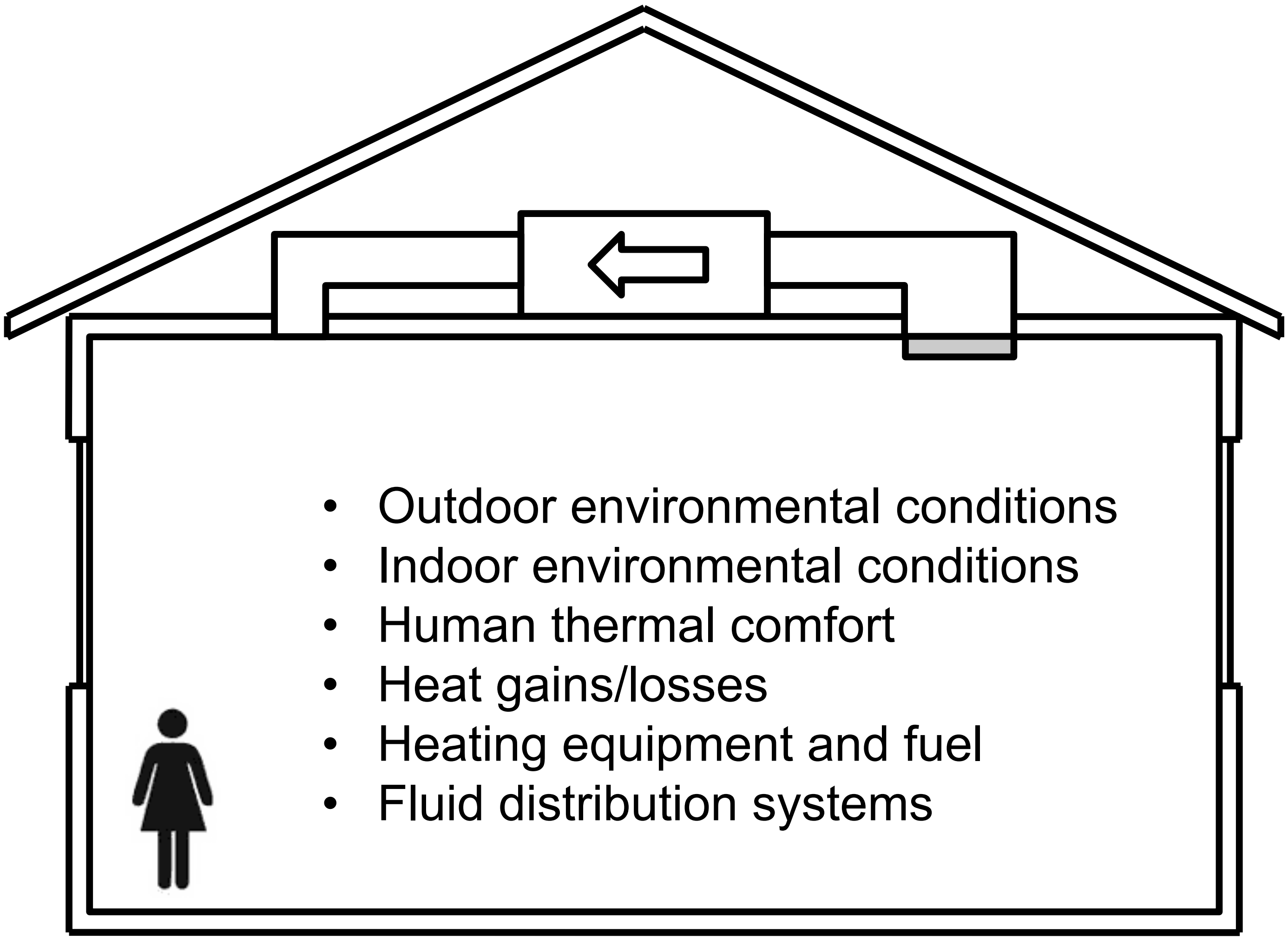
How do buildings use energy?

Figure 1-1 U.S. Primary Energy Consumption, 2005[2]



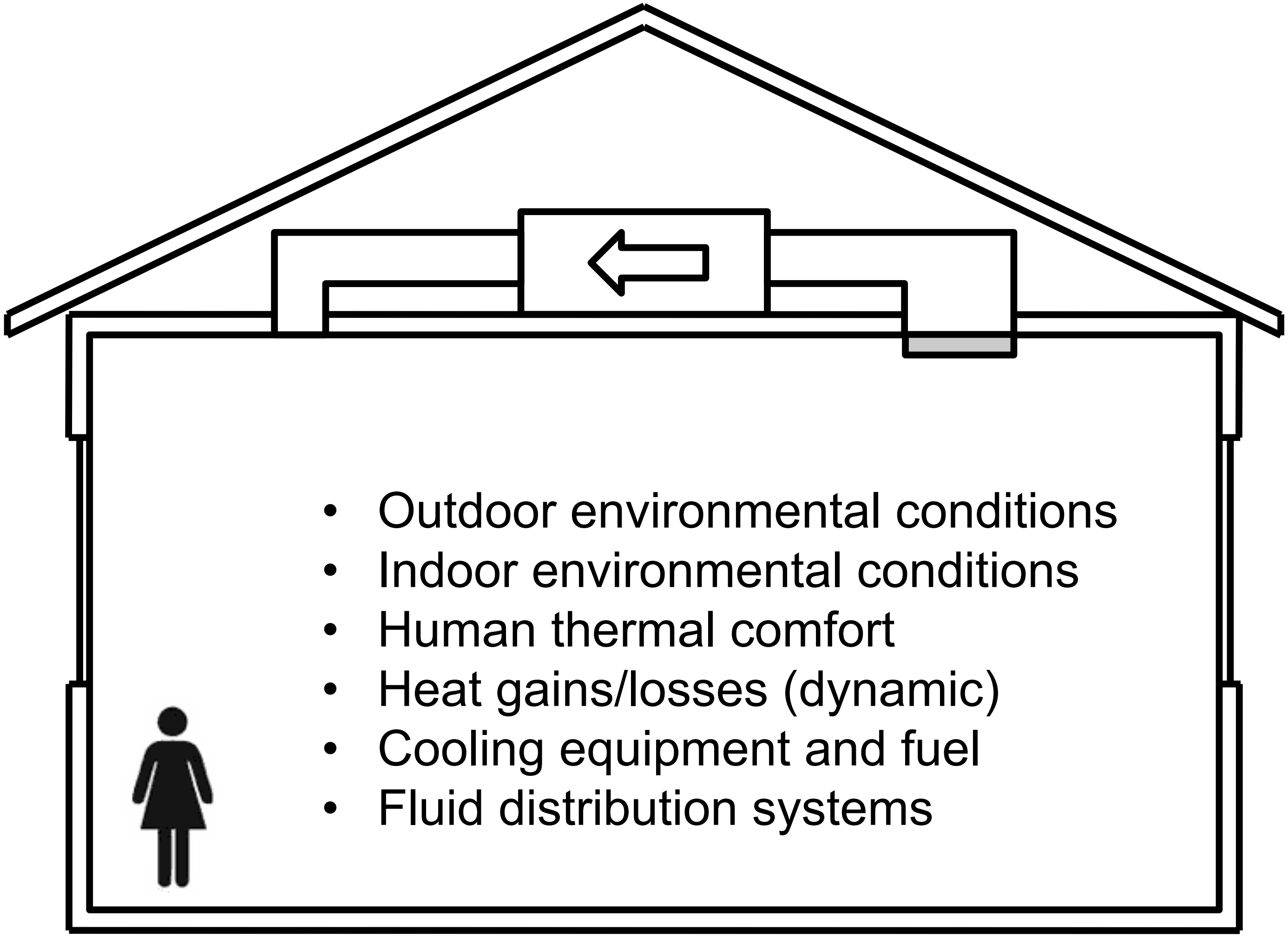
Building energy use: Heating

What do we need to know to understand how buildings heat?



Building energy use: Cooling

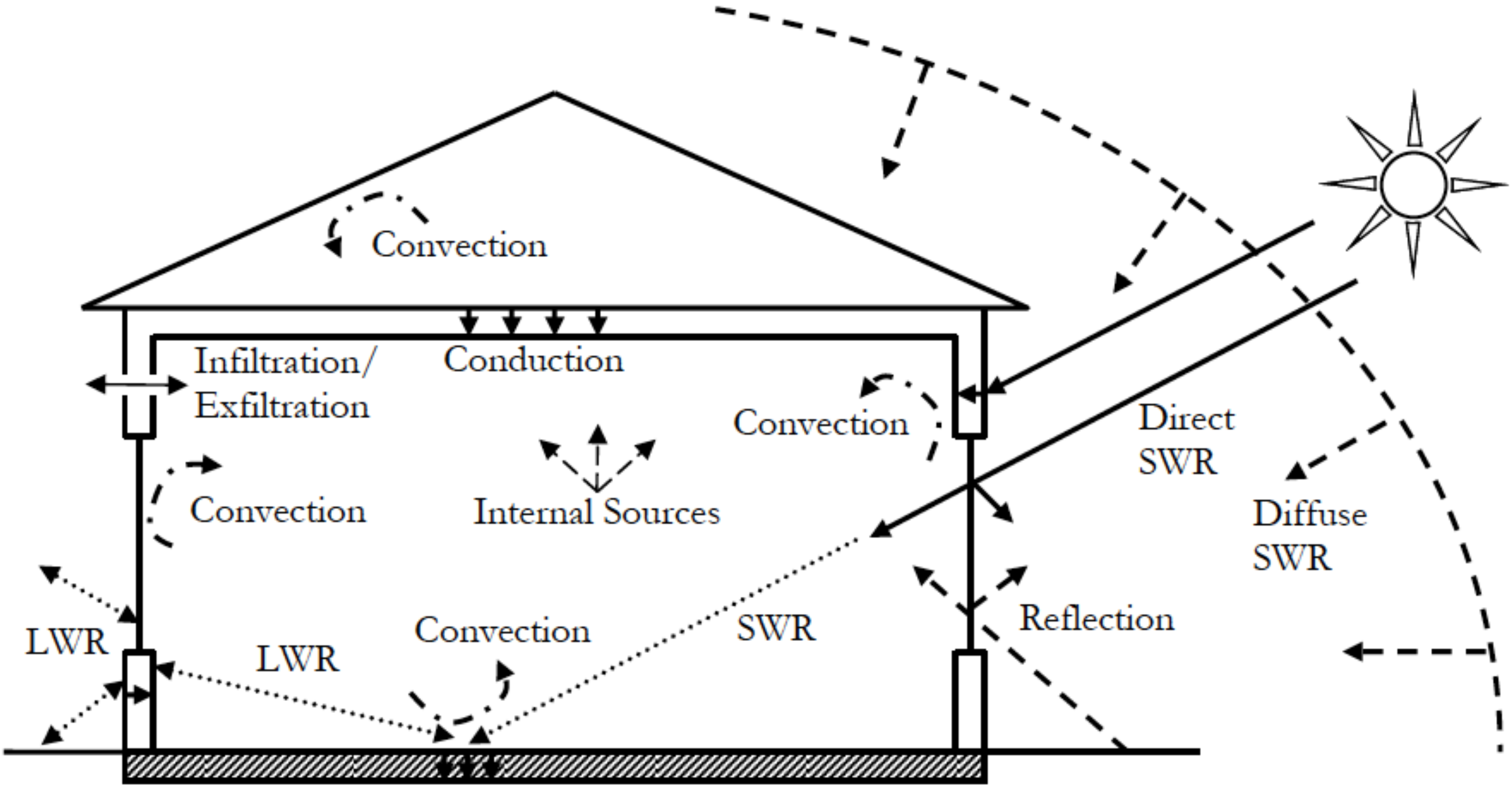
What do we need to know to understand how buildings cool?



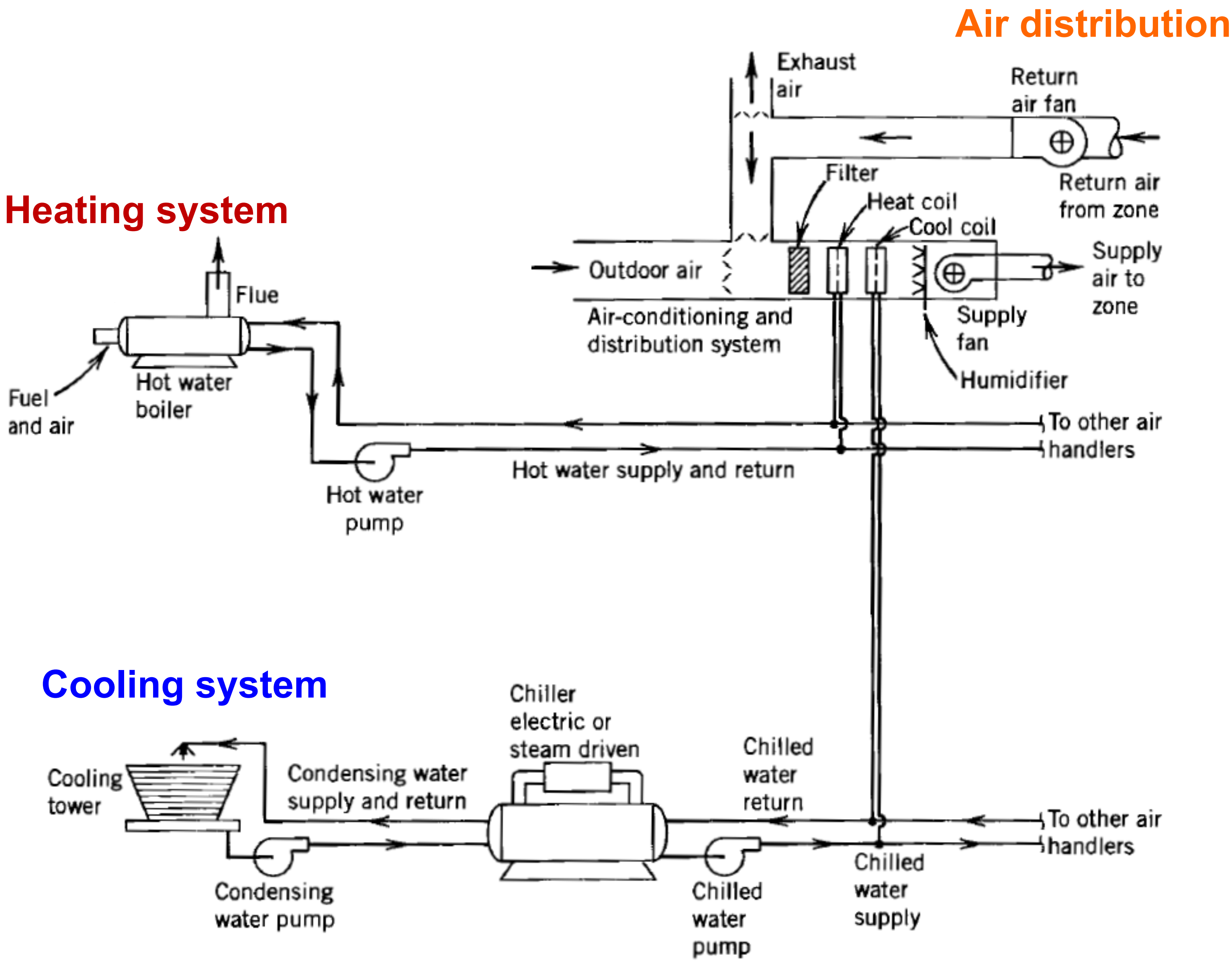
- Outdoor environmental conditions
- Indoor environmental conditions
- Human thermal comfort
- Heat gains/losses (dynamic)
- Cooling equipment and fuel
- Fluid distribution systems

HEAT TRANSFER

Primary modes of heat transfer in a building



Heat transfer in HVAC systems



Heat transfer in building science: Summary

Conduction

$$q = \frac{k}{L}\left(T_{surf,1} - T_{surf,2}\right)$$

$$\frac{k}{L} = U = \frac{1}{R}$$

$$R_{total} = \frac{1}{U_{total}}$$

$$R_{total} = R_1 + R_2 + R_3 + \ldots$$

For thermal bridges and combined elements:

$$U_{total} = \frac{A_1}{A_{total}} U_1 + \frac{A_2}{A_{total}} U_2 + \ldots$$

Convection

$$q_{conv} = h_{conv}\left(T_{fluid} - T_{surf}\right)$$

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

$$Q_{bulk} = \dot{m} C_p (T_{fluid,1} - T_{fluid,2})$$

Radiation

Long-wave

$$q_{1\to 2} = \frac{\sigma\left(T^4_{surf,1} - T^4_{surf,2}\right)}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{A_1}{A_2}\frac{1-\varepsilon_2}{\varepsilon_2} + \frac{1}{F_{12}}}$$

$$q_{rad,1\to 2} = h_{rad}\left(T_{surf,1} - T_{surf,2}\right)$$

$$h_{rad} = \frac{4\sigma T^3_{avg}}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad R_{rad} = \frac{1}{h_{rad}}$$

$$q_{1\to 2} = \varepsilon_{surf}\sigma F_{12}\left(T^4_{surf,1} - T^4_{surf,2}\right)$$

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

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

Thermal conductivity of building materials

TABLE 2.3

Values of Thermal Conductivity for Building Materials

Material	k, Btu/(h · ft · °F)	T, °F	k, W/(m · K)	T, °C
Construction materials				
Asphalt	0.43–0.44	68–132	0.74–0.76	20–55
Cement, cinder	0.44	75	0.76	24
Glass, window	0.45	68	0.78	20
Concrete	1.0	68	1.73	20
Marble	1.2–1.7	—	2.08–2.94	—
Balsa	0.032	86	0.055	30
White pine	0.065	86	0.112	30
Oak	0.096	86	0.166	30
Insulating materials				
Glass fiber	0.021	75	0.036	24
Expanded polystyrene	0.017	75	0.029	24
Polyisocyanurate	0.012	75	0.020	24
Gases at atmospheric pressure				
Air	0.0157	100	0.027	38
Helium	0.0977	200	0.169	93
Refrigerant 12	0.0048	32	0.0083	0
	0.0080	212	0.0038	100
Oxygen	0.00790	−190	0.0137	−123
	0.02212	350	0.0383	175

Source: Courtesy of Karlekar, B. and Desmond, R.M., *Engineering Heat Transfer*, West Publishing, St. Paul, MN, 1982. With permission.

Thermal properties of building materials

Table 1 Building and Insulating Materials: Design Values[a]

Description	Density, kg/m³	Conductivity[b] k, W/(m·K)	Resistance R, (m²·K)/W	Specific Heat, kJ/(kg·K)	Reference[l]
Insulating Materials					
Blanket and batt[c,d]					
Glass-fiber batts				0.8	Kumaran (2002)
	7.5 to 8.2	0.046 to 0.048	—	—	Four manufacturers (2011)
	9.8 to 12	0.040 to 0.043	—	—	Four manufacturers (2011)
	13 to 14	0.037 to 0.039	—	—	Four manufacturers (2011)
	22	0.033	—	—	Four manufacturers (2011)
Rock and slag wool batts	—	—	—	0.8	Kumaran (1996)
	32 to 37	0.036 to 0.037	—	—	One manufacturer (2011)
	45	0.033 to 0.035	—	—	One manufacturer (2011)
Mineral wool, felted	16 to 48	0.040	—	—	CIBSE (2006), NIST (2000)
	16 to 130	0.035	—	—	NIST (2000)
Board and slabs					
Cellular glass	120	0.042	—	0.8	One manufacturer (2011)
Cement fiber slabs, shredded wood with Portland cement binder	400 to 430	0.072 to 0.076	—	—	
with magnesia oxysulfide binder	350	0.082	—	1.3	
Glass fiber board	—	—	—	0.8	Kumaran (1996)
	24 to 96	0.033 to 0.035	—	—	One manufacturer (2011)
Expanded rubber (rigid)	64	0.029	—	1.7	Nottage (1947)
Extruded polystyrene, smooth skin	—	—	—	1.5	Kumaran (1996)
aged per Can/ULC *Standard* S770-2003	22 to 58	0.026 to 0.029	—	—	Four manufacturers (2011)
aged 180 days	22 to 58	0.029			One manufacturer (2011)
European product	30	0.030			One manufacturer (2011)
aged 5 years at 24°C	32 to 35	0030	—	—	One manufacturer (2011)
blown with low global warming potential (GWP) (<5) blowing agent		0.035 to 0.036	—	—	One manufacturer (2011)
Expanded polystyrene, molded beads	—	—	—	1.5	Kumaran (1996)
	16 to 24	0.035 to 0.037	—	—	Independent test reports (2008)
	29	0.033	—	—	Independent test reports (2008)

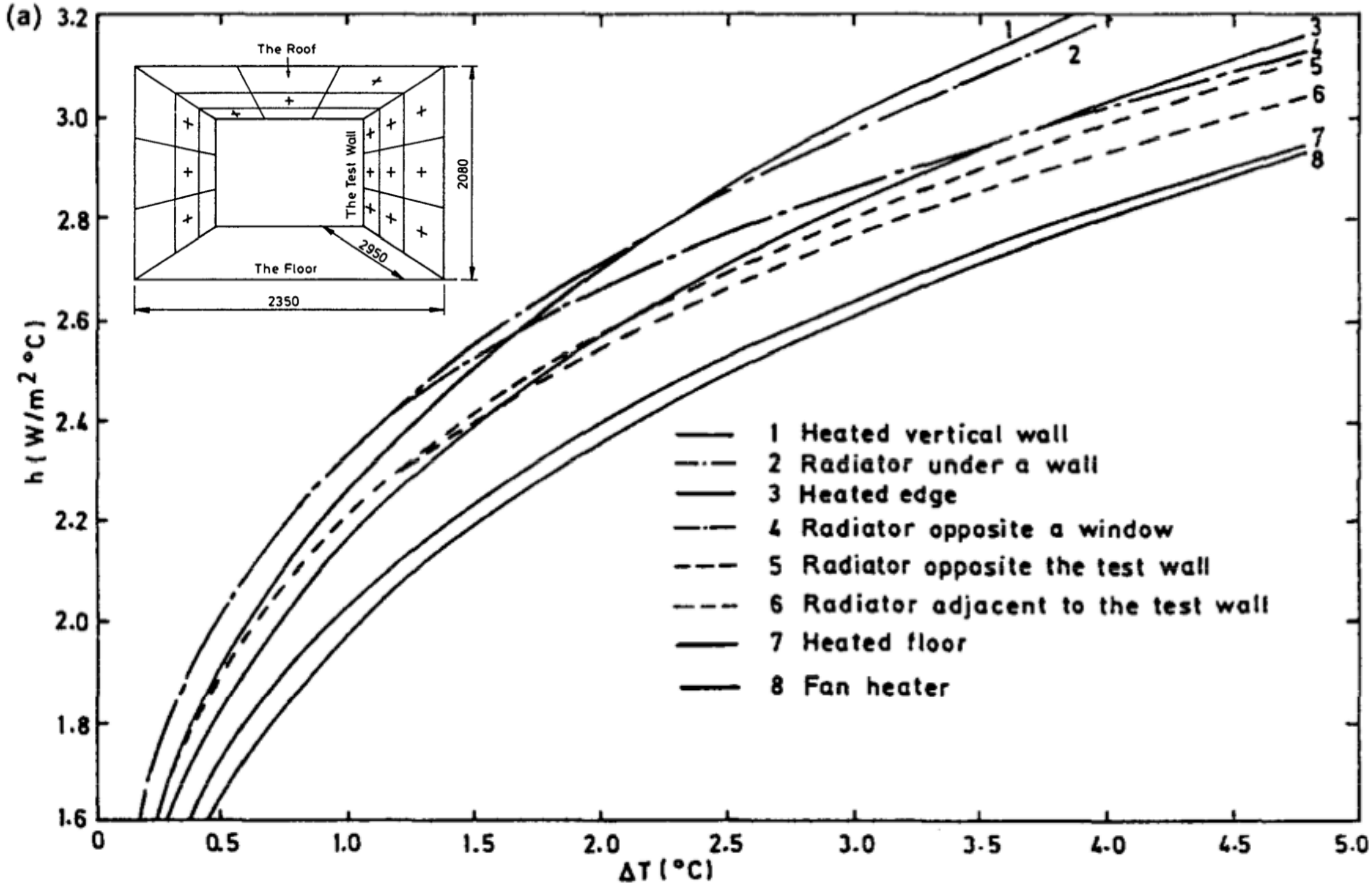
Combined thermal transmittance for walls + fenestration

- Single assemblies of walls, windows, doors, etc. can be combined into an overall U-value for a building's enclosure
  - **Combined thermal transmittance**: U_o or U_{total}
    - Area-weighted average U-value
    - Just like center of glass, edge of glass, frame analysis for windows

$$U_{total} = \frac{U_{wall}A_{wall} + U_{windows}A_{windows} + U_{doors}A_{doors}}{A_{total}}$$

We will use this later for calculating heating and cooling loads

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



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

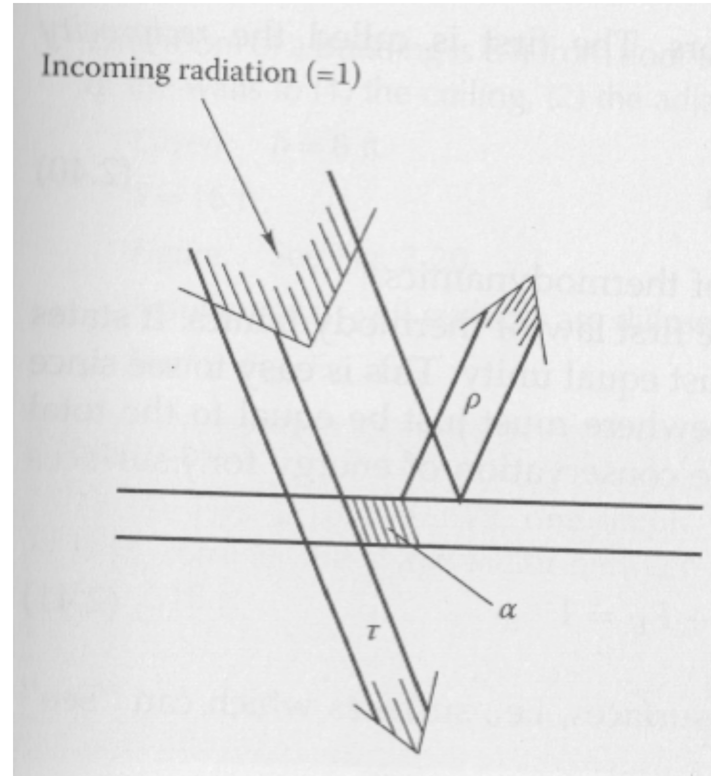
Typical convective surface resistances

- 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)		

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

Emissivity (ε) of common building materials

TABLE 2.11

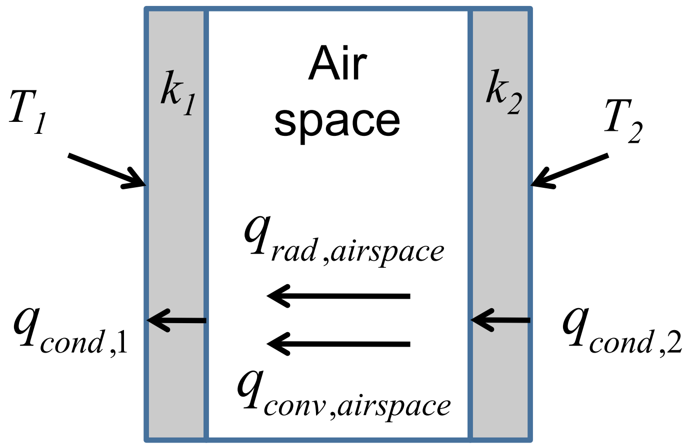
Emissivities of Some Common Building Materials at Specified Temperatures

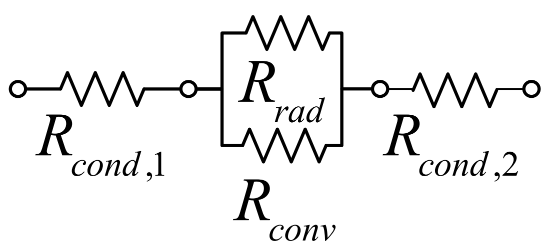
Surface	Temperature, °C	Temperature, °F	ϵ
Brick			
Red, rough	40	100	0.93
Concrete			
Rough	40	100	0.94
Glass			
Smooth	40	100	0.94
Ice			
Smooth	0	32	0.97
Marble			
White	40	100	0.95
Paints			
Black gloss	40	100	0.90
White	40	100	0.89–0.97
Various oil paints	40	100	0.92–0.96
Paper			
White	40	100	0.95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

Combined heat transfer

- When more than one mode of heat transfer exists at a location (usually convection + radiation), resistances get placed in parallel
  - Example: Heat transfer in a cavity

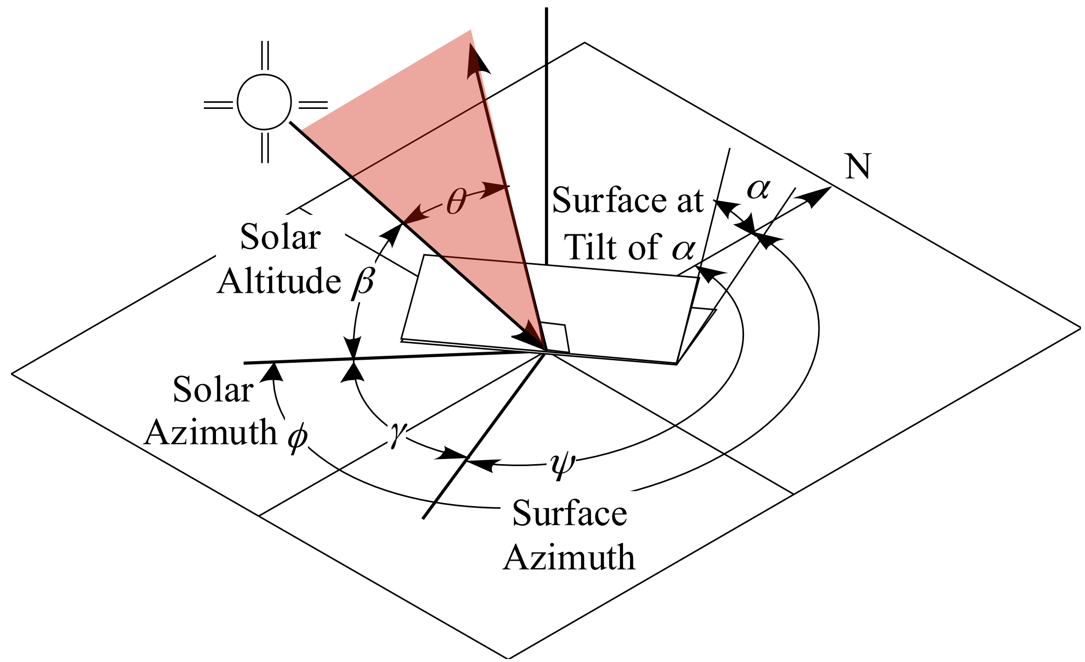




Solar radiation striking an exterior surface

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

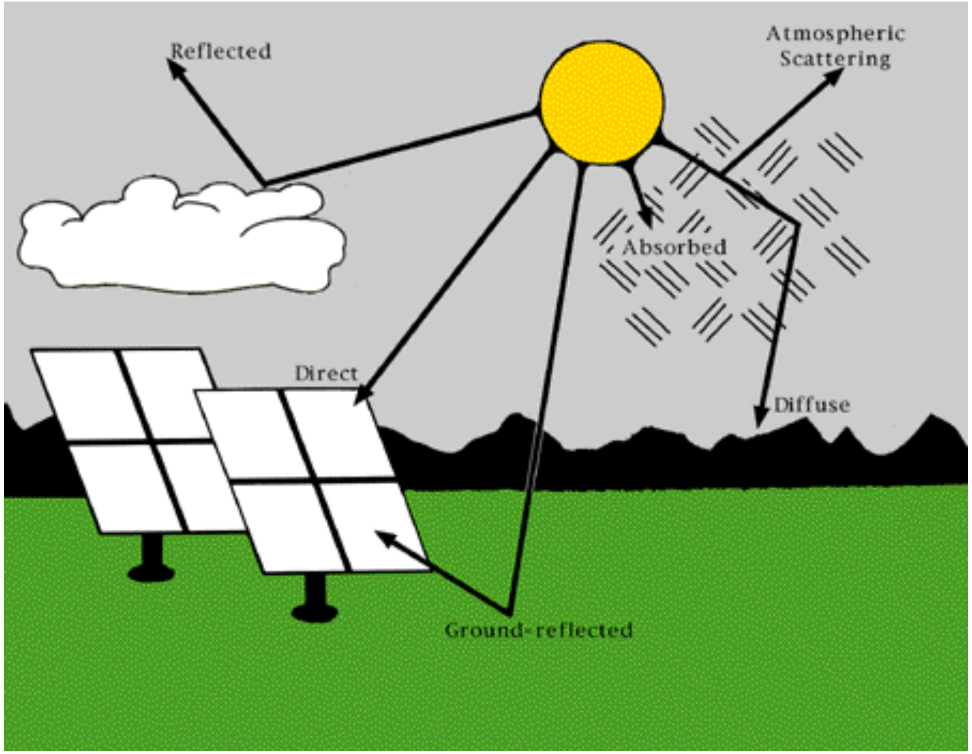
- This is a function of:
  - Solar geometry
    - Location
    - Time
  - Surface geometry
  - Shading/obstacles



Components of solar radiation

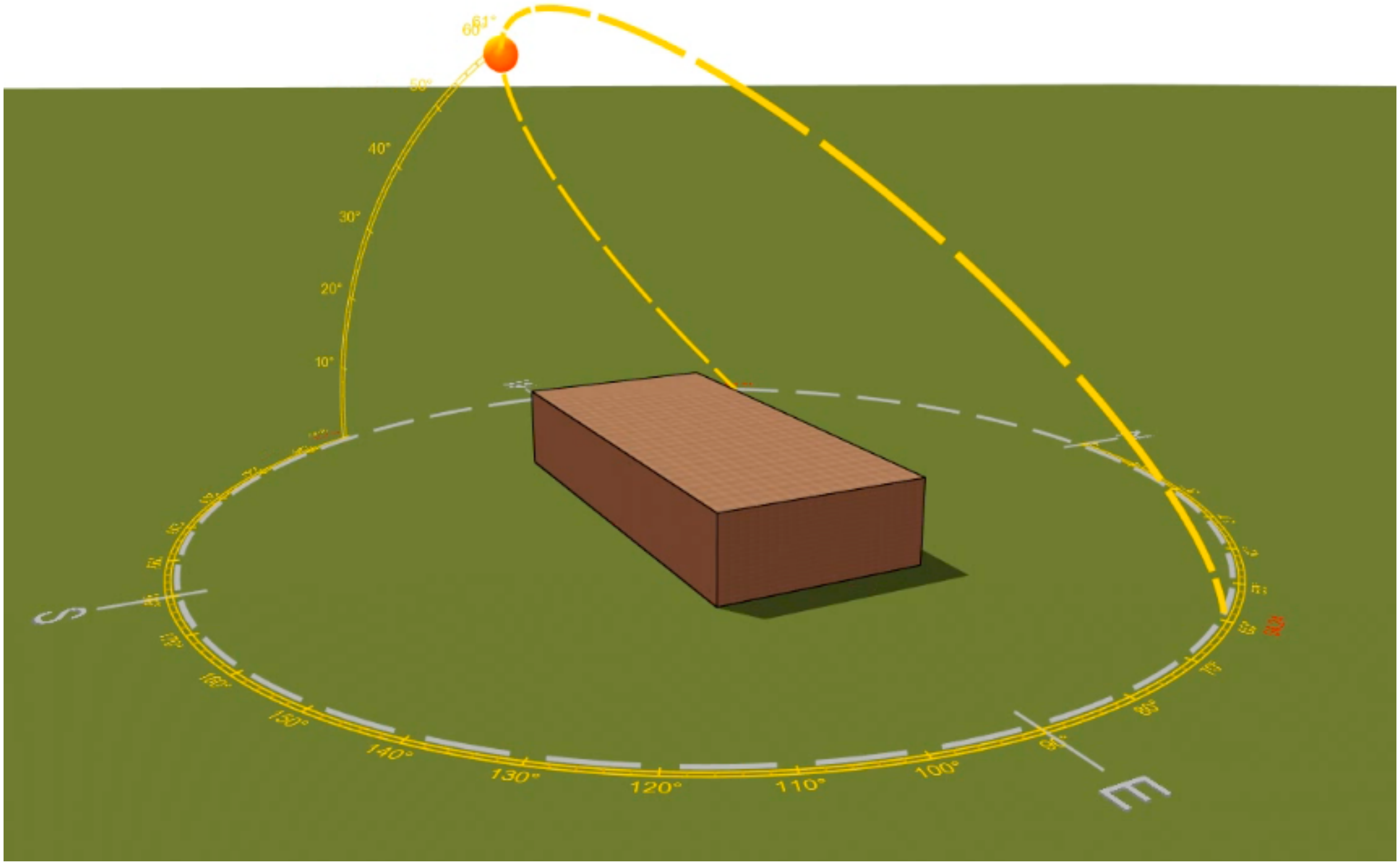
- Solar radiation striking a surface consists of three main components:

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



Visualizing solar relationships

- For visualizing geometry, using programs like IES-VE



Sol-air temperatures

- If we take an external surface with a combined convective and radiative heat transfer coefficient, $h_{conv+rad}$

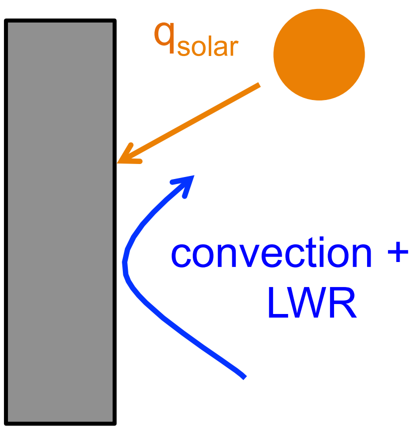
$$q_{conv+rad} = h_{conv+rad}\left(T_{air} - T_{surf}\right)$$

- If that surface now absorbs solar radiation (αI_{solar}), the total heat flow at the exterior surface becomes:

$$q_{conv+rad} = h_{conv+rad}\left(T_{air} - T_{surf}\right) + \alpha I_{solar}$$

- To simplify our calculations, we can define a "**sol-air**" temperature that accounts for all of these impacts:

$$T_{sol-air} = T_{air} + \frac{\alpha I_{solar}}{h_{conv+rad}}$$



- Now we can describe heat transfer at that surface as:

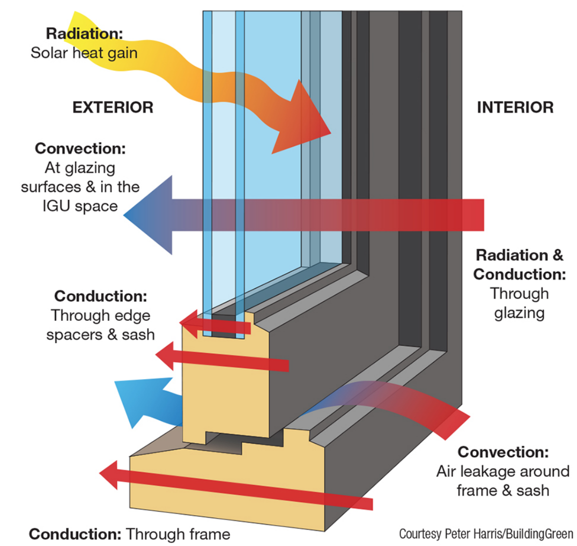
$$q_{total} = h_{conv+rad}\left(T_{sol-air} - T_{surf}\right)$$

Heat gains/losses through window assemblies

- In addition to glazing material, windows also include framing, mullions, muntin bars, dividers, and shading devices
  - These all combine to make **fenestration systems**
- Total heat transfer through an assembly:

$$Q_{window} = UA_{pf}\left(T_{out} - T_{in}\right) + I_{solar}A_{pf}SHGC$$

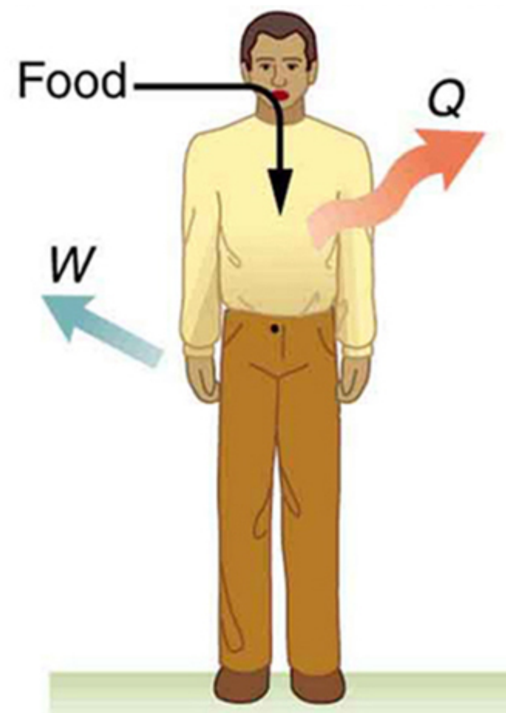
Where:
U = overall coefficient of heat transfer (U-factor), W/m^2K
A_{pf} = total *projected* area of fenestration, m^2
T_{in} = indoor air temperature, K
T_{out} = outdoor air temperature, K
SHGC = solar heat gain coefficient, -
I_{solar} = incident total irradiance, W/m^2



THERMAL COMFORT

Thermal balance of body and effective temperature

- The heat produced by the body's metabolism dissipates to the environment
  - Otherwise we would overheat
- If the rate of heat transfer is higher than the rate of heat production, the body cools down and we feel cold
  - If heat transfer is lower than production, we feel hot
- This is a complex problem in transient heat transfer, involving radiation, convection, conduction, and evaporation, and many variables including skin wetness and clothing composition
  - We can simplify a lot of this



Percent People Dissatisfied (PPD)

Since we want:

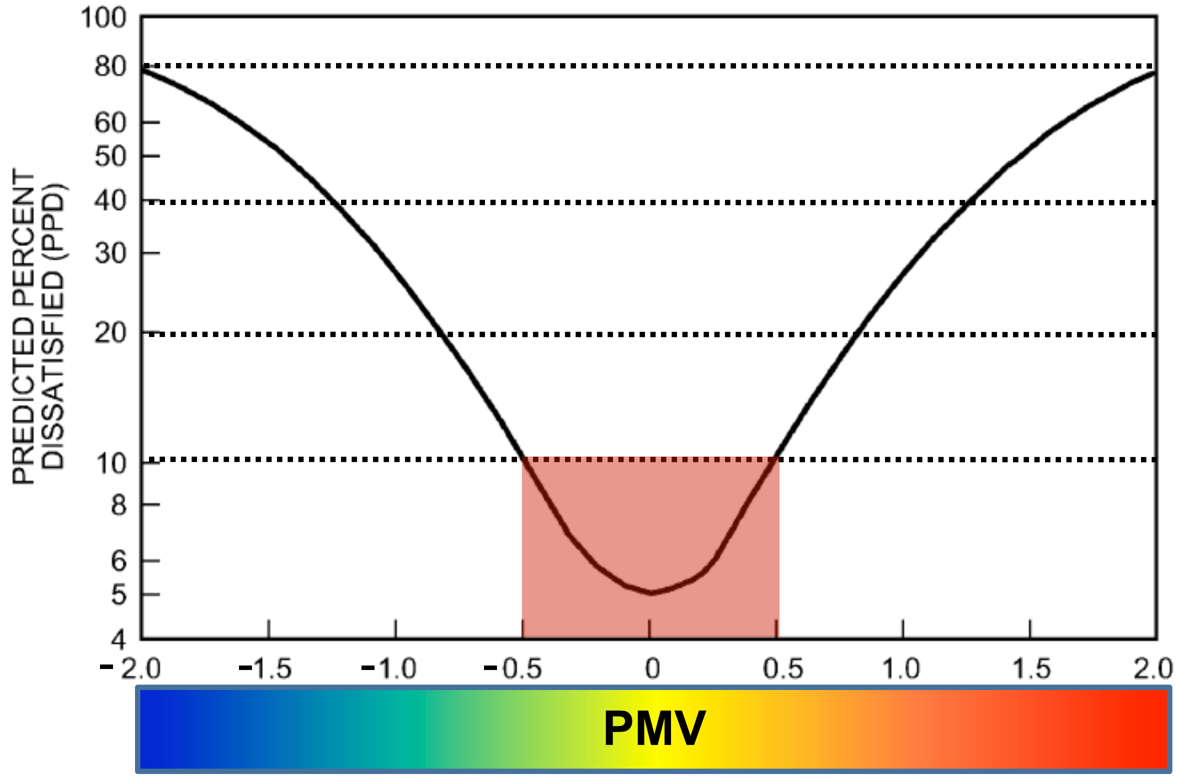
PPD<10%

we can see that:

-0.5<PMV<0.5

Notice that the minimum PPD is 5% showing that you cannot satisfy everyone at the same time

The PPD Function:



Variables affecting thermal comfort

- ASHRAE Standard 55 considers 6 parameters important for thermal comfort

Some are familiar:

Ambient Air Temp (T)

Humidity (W or RH)

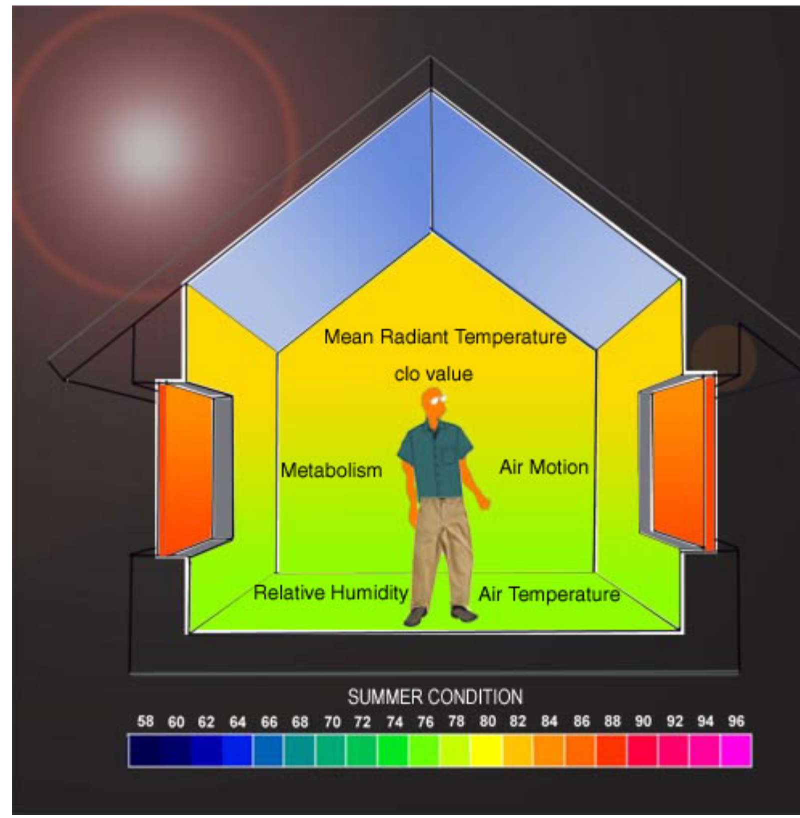
Local Air Speed (v)

Some are probably not:

Metabolic Rate (M)

Clothing Insulation (I_{cl})

Mean Radiant Temp. (T_r)



ASHRAE comfort zone

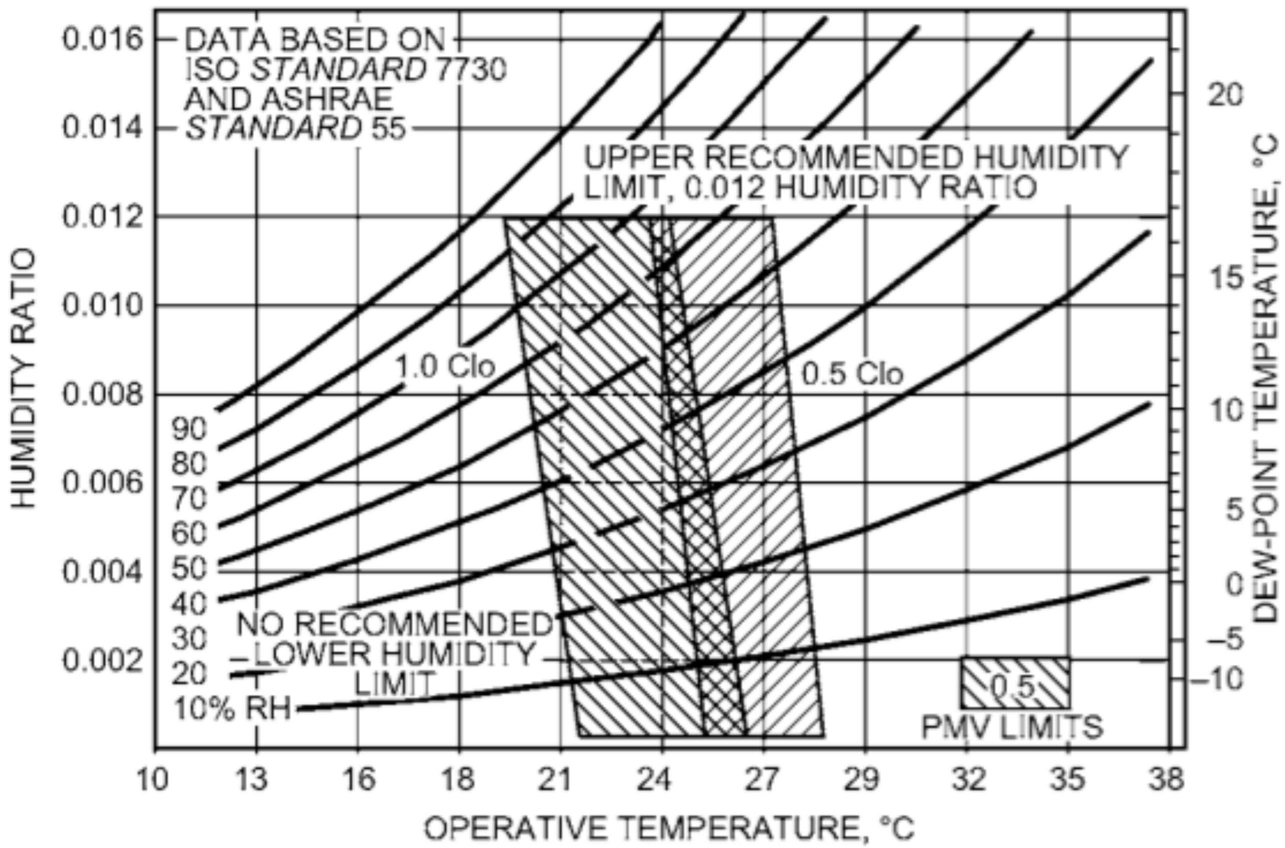
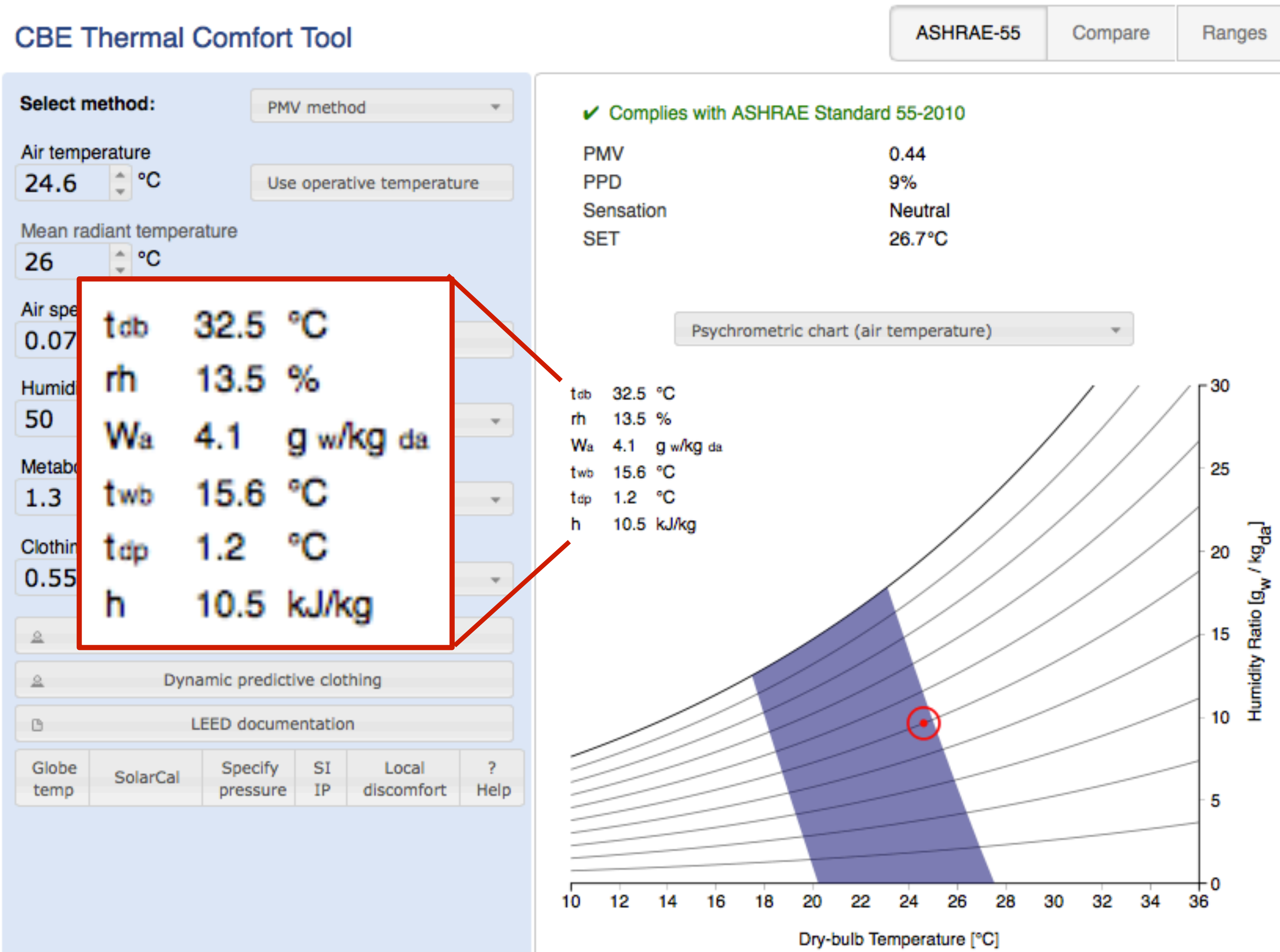


Fig. 5 ASHRAE Summer and Winter Comfort Zones
[Acceptable ranges of operative temperature and humidity with air speed ≤ 0.2 m/s for people wearing 1.0 and 0.5 clo clothing during primarily sedentary activity (≤1.1 met)].

ASHRAE comfort zone: CBE Thermal Comfort Tool



http://smap.cbe.berkeley.edu/comforttool

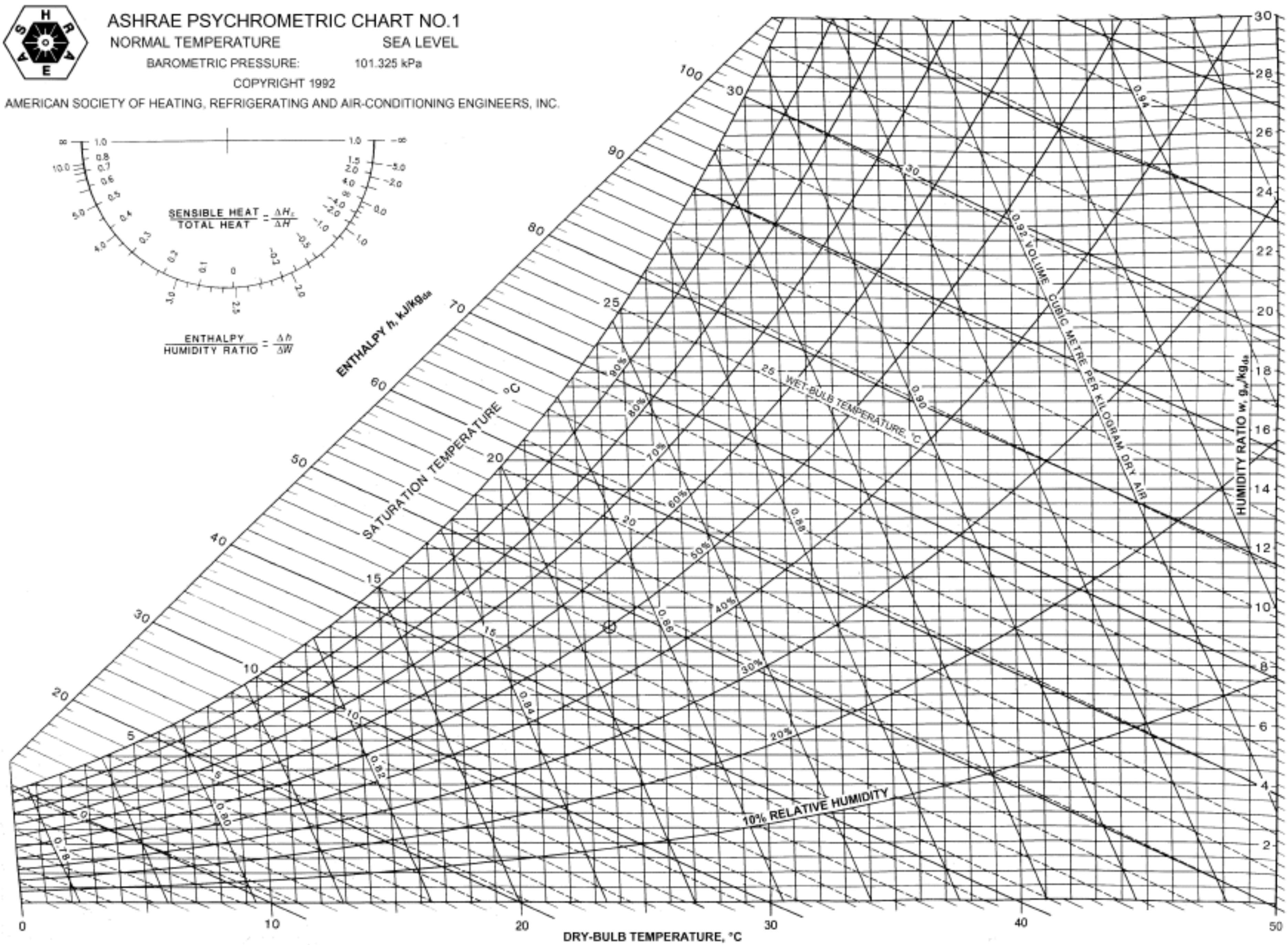
PSYCHROMETRICS

Psychrometrics: Terms for describing moist air

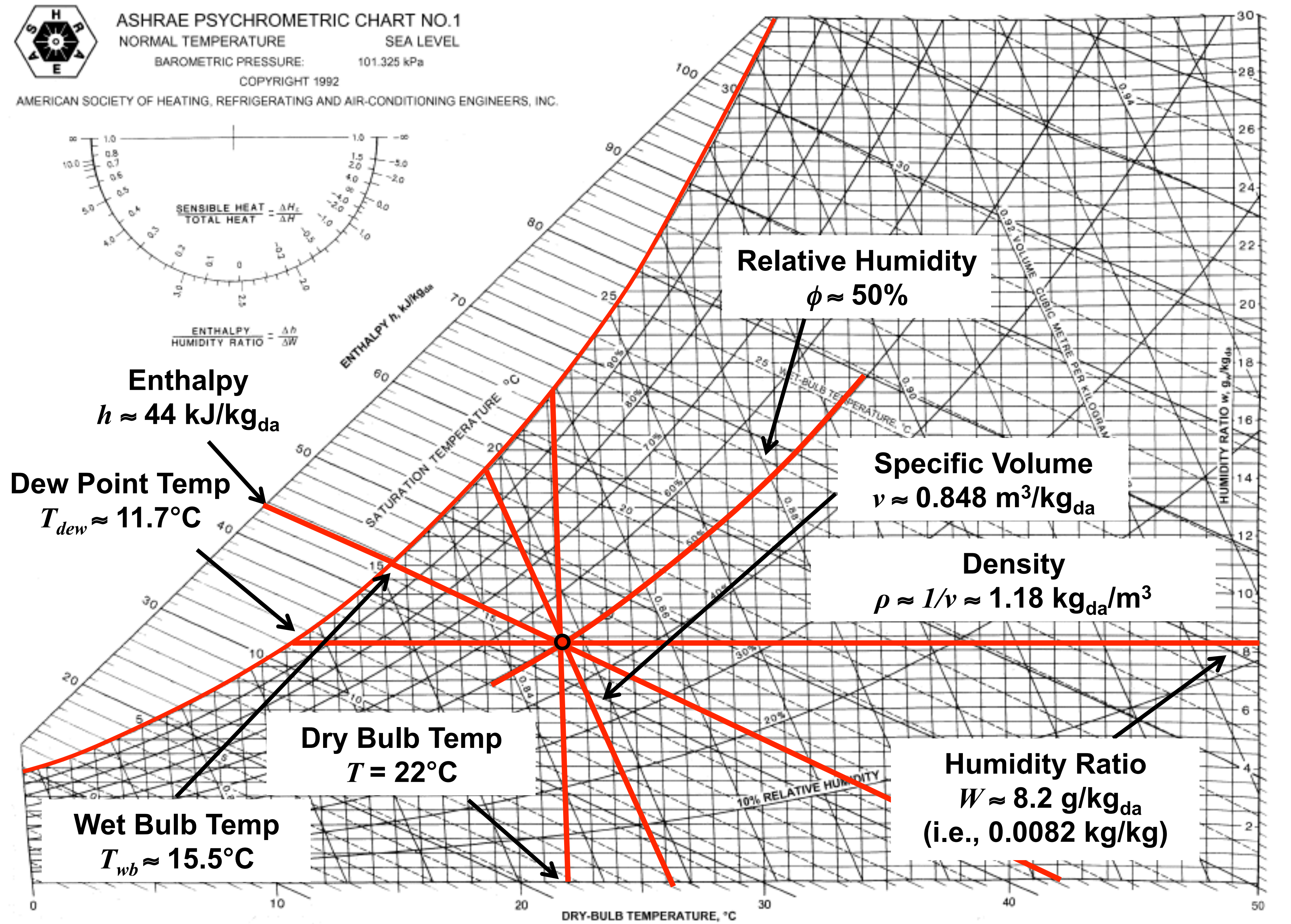
- To describe and deal with moist air, we need to be able to describe the fractions of dry air and water vapor
- There are several different equivalent measures
  - Which one you use depends on what data you have to start with and what quantity you are trying to find

Key terms to know:

- Dry bulb temperature
- Vapor pressure
- Saturation
- Relative humidity
- Absolute humidity (or humidity ratio)
- Dew point temperature
- Wet bulb temperature
- Enthalpy
- Density
- Specific volume



ASHRAE PSYCHROMETRIC CHART NO.1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔH
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kg da
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, gw/kg da
DRY-BULB TEMPERATURE, °C



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SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔH
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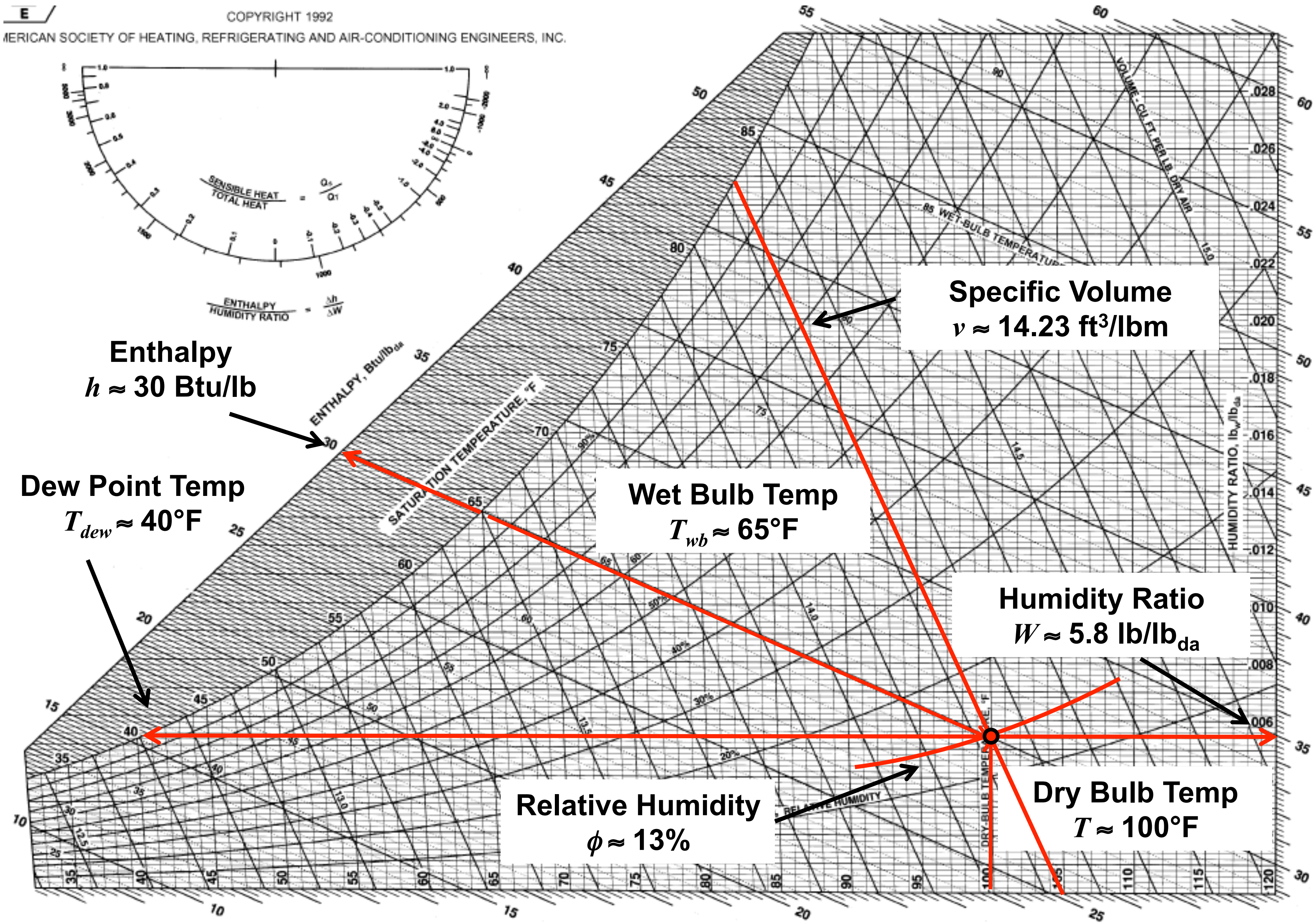
ENTHALPY h, kJ/kg da
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE °C
VOLUME CUBIC METRE PER KILOGRAM
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, gw/kgda
DRY-BULB TEMPERATURE, °C
Relative Humidity
φ ≈ 50%
Enthalpy
h ≈ 44 kJ/kgda
Dew Point Temp
Tdew ≈ 11.7°C
Specific Volume
v ≈ 0.848 m3/kgda
Density
ρ ≈ 1/v ≈ 1.18 kgda/m3
Dry Bulb Temp
T = 22°C
Wet Bulb Temp
Twb ≈ 15.5°C
Humidity Ratio
W ≈ 8.2 g/kgda
(i.e., 0.0082 kg/kg)



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SENSIBLE HEAT / TOTAL HEAT = Q_s / Q_t
ENTHALPY / HUMIDITY RATIO = $\Delta h / \Delta W$
ENTHALPY, Btu/lb$_{da}$
SATURATION TEMPERATURE, °F
WET-BULB TEMPERATURE
VOLUME, CU FT PER LB DRY AIR
HUMIDITY RATIO, lb$_w$/lb$_{da}$
RELATIVE HUMIDITY
DRY BULB TEMPERATURE, °F
Specific Volume
$v \approx 14.23$ ft^3/lbm
Enthalpy
$h \approx 30$ Btu/lb
Dew Point Temp
$T_{dew} \approx 40$°F
Wet Bulb Temp
$T_{wb} \approx 65$°F
Humidity Ratio
$W \approx 5.8$ lb/lb$_{da}$
Relative Humidity
$\phi \approx 13\%$
Dry Bulb Temp
$T \approx 100$°F

Psychrometric equations

$$\phi = \frac{p_w}{p_{ws}} \qquad W = 0.622\frac{p_w}{p - p_w} \qquad \mu = \frac{W}{W_s}$$

$$\ln p_{ws} = \frac{C_8}{T} + C_9 + C_{10}T + C_{11}T^2 + C_{12}T^3 + C_{13}\ln T$$

where

C_8 = −5.800 220 6 E+03
C_9 = 1.391 499 3 E+00
C_{10} = −4.864 023 9 E−02
C_{11} = 4.176 476 8 E−05
C_{12} = −1.445 209 3 E−08
C_{13} = 6.545 967 3 E+00

p_{ws} = saturation pressure, Pa
T = absolute temperature, K = °C + 273.15

Dew point temperature:

Between dew points of 0 and 93°C,

$$t_d = C_{14} + C_{15}\alpha + C_{16}\alpha^2 + C_{17}\alpha^3 + C_{18}(p_w)^{0.1984}$$

Below 0°C,

$$t_d = 6.09 + 12.608\alpha + 0.4959\alpha^2$$

where

t_d = dew-point temperature, °C
α = $\ln p_w$
p_w = water vapor partial pressure, kPa
C_{14} = 6.54
C_{15} = 14.526
C_{16} = 0.7389
C_{17} = 0.09486
C_{18} = 0.4569

Psychrometric equations

Wet bulb temperature (iterative solver):

$$W = \frac{(2501 - 2.326T_{wb})W_{s@T_{wb}} - 1.006(T - T_{wb})}{2501 + 1.86T - 4.186T_{wb}} \quad = \text{actual } W$$

$$v = \frac{R_{da}T}{p - p_w} = \frac{R_{da}T(1 + 1.6078W)}{p}$$

$$v \approx 0.287042(T + 273.15)(1 + 1.6078W) / p$$

where

v = specific volume, m^3/kg_{da}
t = dry-bulb temperature, °C
W = humidity ratio, kg_w/kg_{da}
p = total pressure, kPa

$$\rho = \frac{m_{da} + m_w}{V} = \frac{1}{v}(1 + W)$$

$$h \approx 1.006T + W(2501 + 1.86T)$$

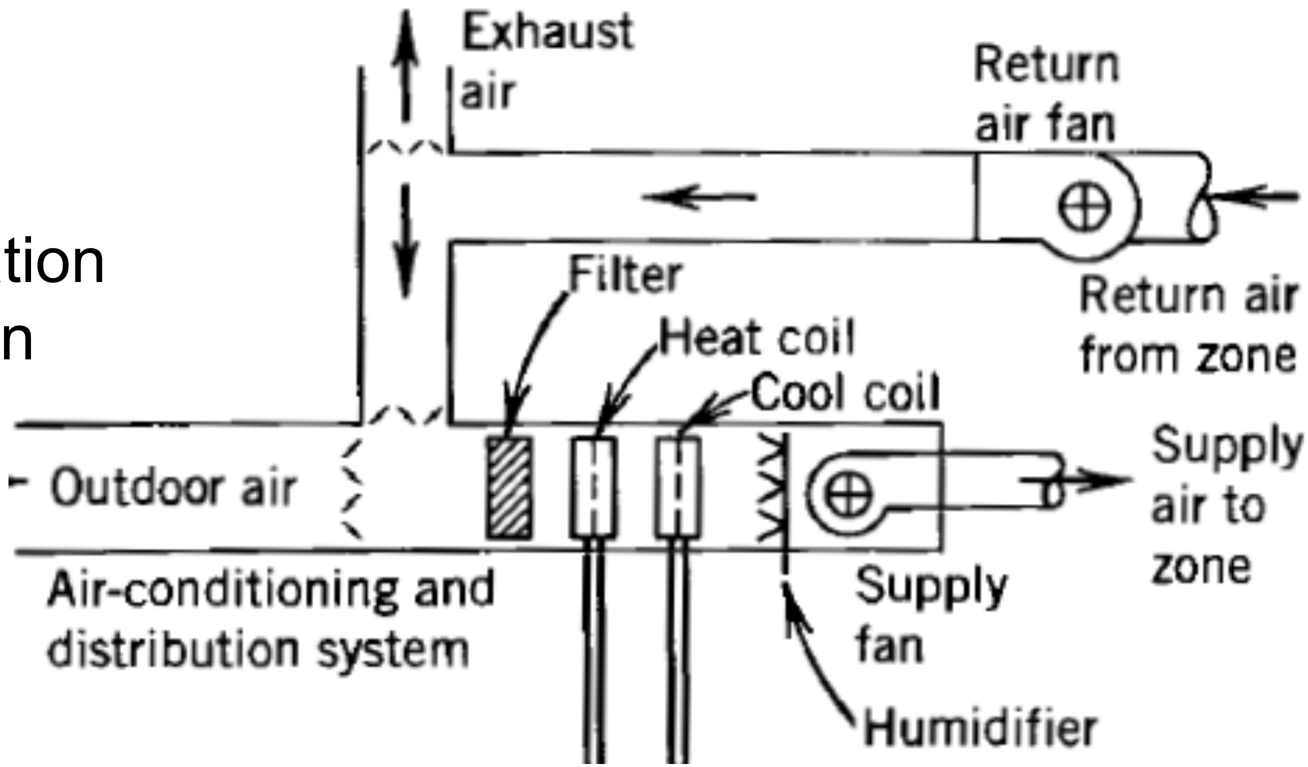
*where T is in °C

PSYCHROMETRIC PROCESSES

Typical forced air distribution system

Common processes:

- Air mixing
- Heating
- Cooling
- Dehumidification
- Humidification



Psychrometric processes and heat transfer equations

$$Q_{total} = Q_{sensible} + Q_{latent}$$

$$Q_{total} = \dot{m}_{air}(h_{exit} - h_{inlet})$$

$$Q_{sens} = \dot{m}c_p(T_{exit} - T_{inlet}) = \dot{V}\rho c_p(T_{exit} - T_{inlet})$$

$Q_{sens} > 0$ for heating

$Q_{sens} < 0$ for cooling

$$Q_{lat} = \dot{m}_w h_{fg}$$

h_{fg} = latent heat of vaporization

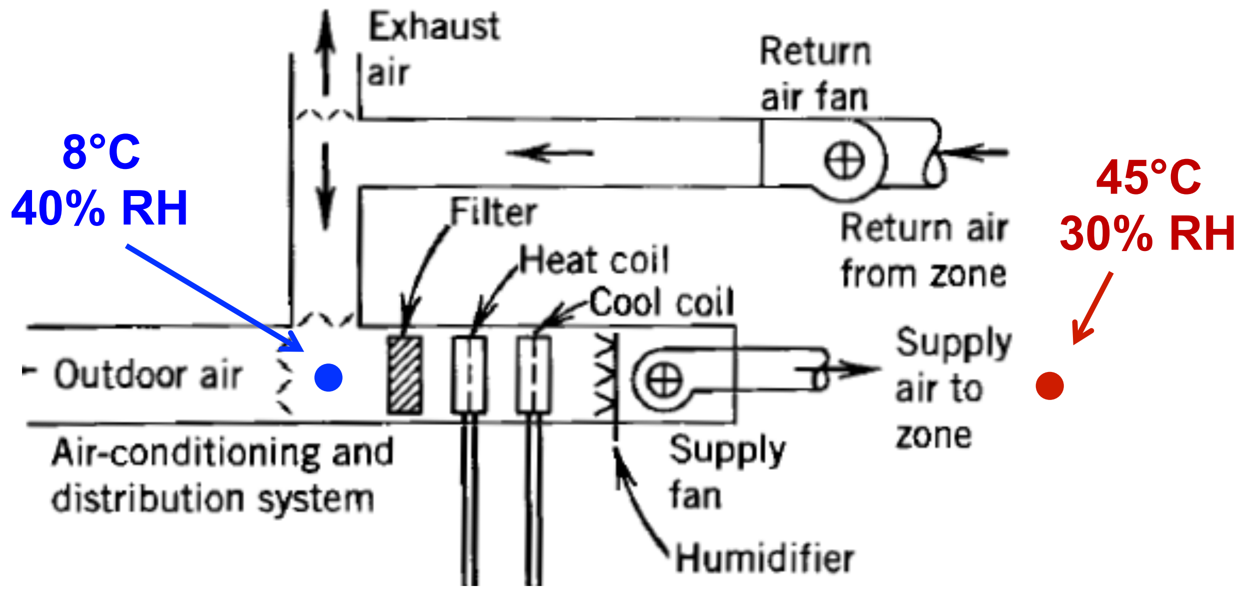
(h_{fg} = 2260 kJ/kg for water)

$$SHR = \frac{Q_{sens}}{Q_{total}} = \frac{\Delta h_{sens}}{\Delta h_{total}}$$

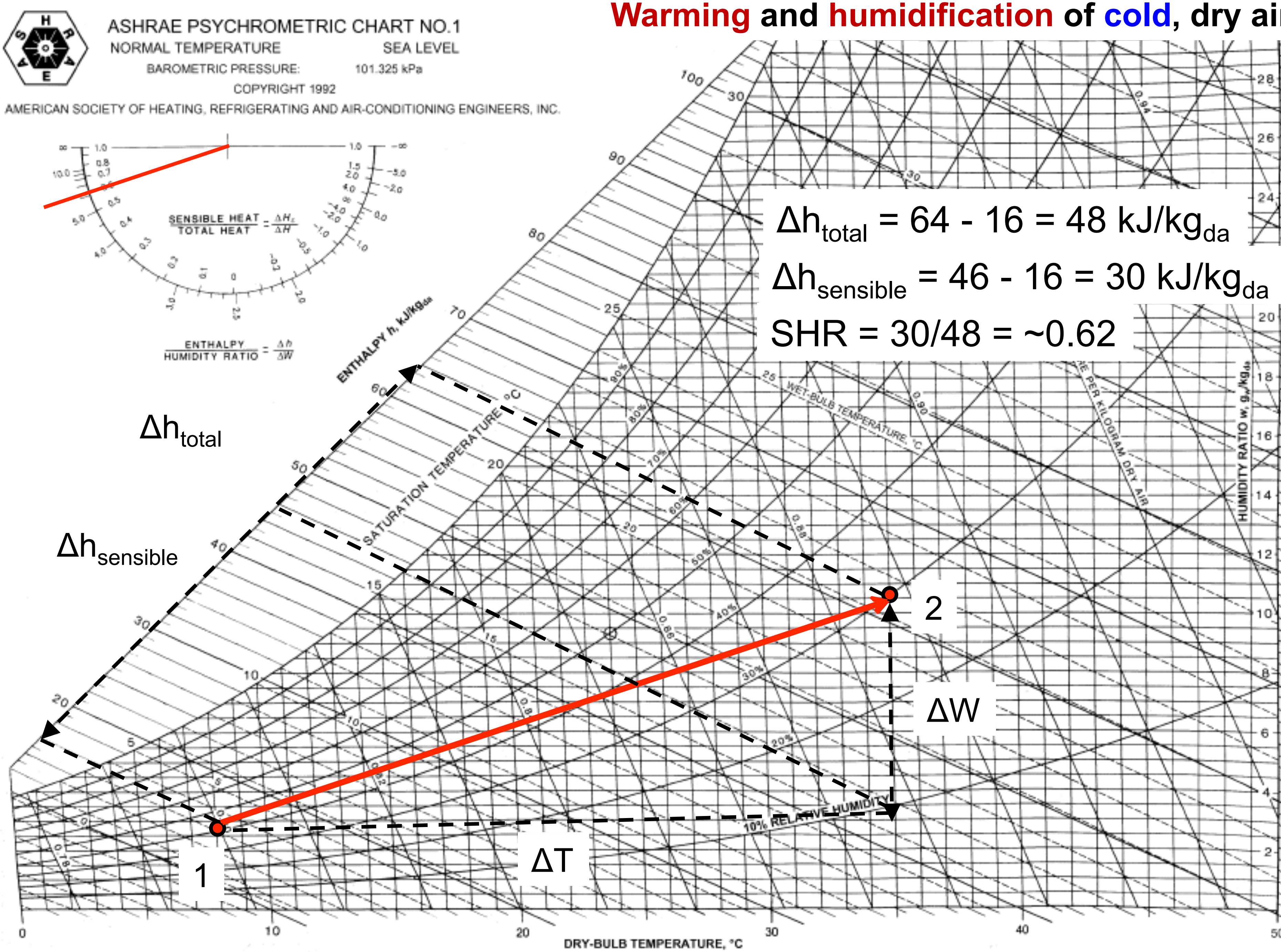
Heating and **humidification** of **cold**, dry air

- Example: Heating and humidifying coils
  - Process: Adding moisture and heat (sensible + latent heating)

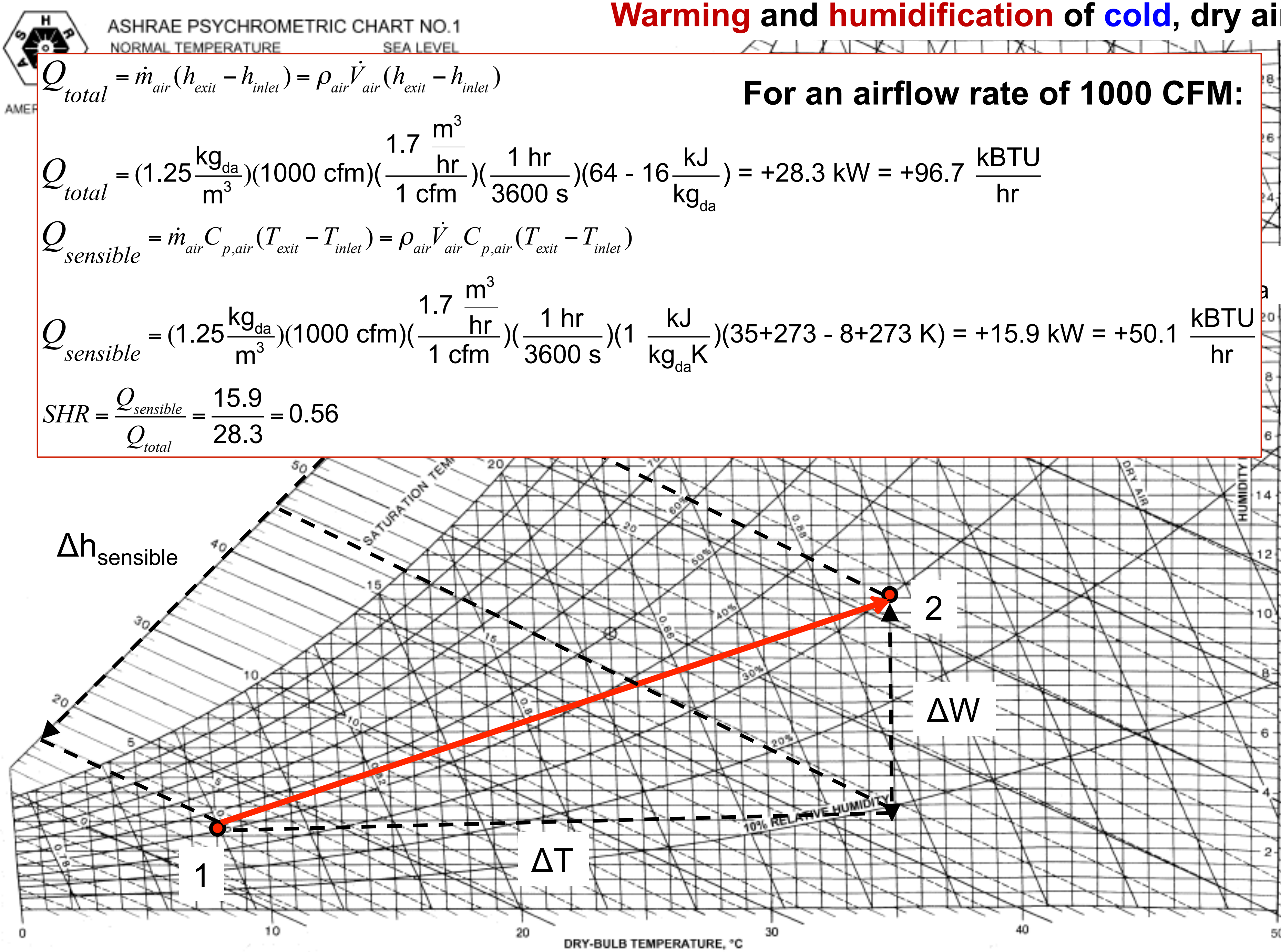
Q1: What is the enthalpy change required?
Q2: What is the split between sensible and latent load?



Warming and humidification of cold, dry air



Warming and humidification of cold, dry air



For an airflow rate of 1000 CFM:

$$Q_{total} = \dot{m}_{air}(h_{exit} - h_{inlet}) = \rho_{air}\dot{V}_{air}(h_{exit} - h_{inlet})$$

$$Q_{total} = (1.25\,\frac{\text{kg}_{\text{da}}}{\text{m}^3})(1000\text{ cfm})(\frac{1.7\,\frac{\text{m}^3}{\text{hr}}}{1\text{ cfm}})(\frac{1\text{ hr}}{3600\text{ s}})(64 - 16\,\frac{\text{kJ}}{\text{kg}_{\text{da}}}) = +28.3\text{ kW} = +96.7\,\frac{\text{kBTU}}{\text{hr}}$$

$$Q_{sensible} = \dot{m}_{air}C_{p,air}(T_{exit} - T_{inlet}) = \rho_{air}\dot{V}_{air}C_{p,air}(T_{exit} - T_{inlet})$$

$$Q_{sensible} = (1.25\,\frac{\text{kg}_{\text{da}}}{\text{m}^3})(1000\text{ cfm})(\frac{1.7\,\frac{\text{m}^3}{\text{hr}}}{1\text{ cfm}})(\frac{1\text{ hr}}{3600\text{ s}})(1\,\frac{\text{kJ}}{\text{kg}_{\text{da}}\text{K}})(35+273 - 8+273\text{ K}) = +15.9\text{ kW} = +50.1\,\frac{\text{kBTU}}{\text{hr}}$$

$$SHR = \frac{Q_{sensible}}{Q_{total}} = \frac{15.9}{28.3} = 0.56$$

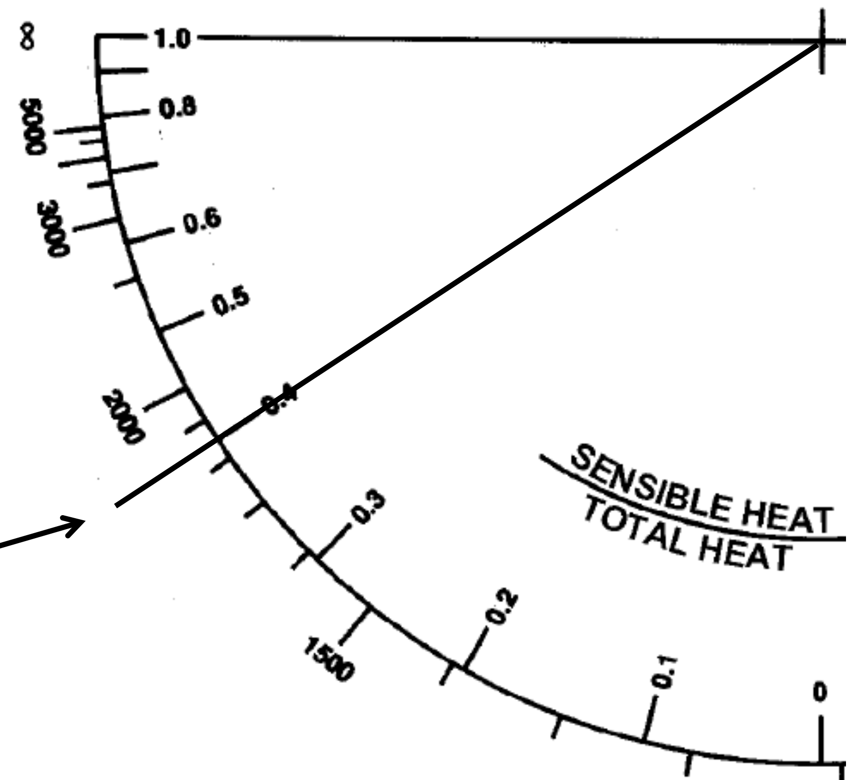
Sensible heat ratio (SHR)

- The sensible heat ratio is defined as:

$$SHR = \frac{\dot{q}_{sens}}{\dot{q}_{total}} = \frac{\dot{q}_{sens}}{\dot{q}_{sens} + \dot{q}_{latent}} = \frac{\Delta h_{sens}}{\Delta h_{total}}$$

- Allows for understanding sensible load relative to latent load

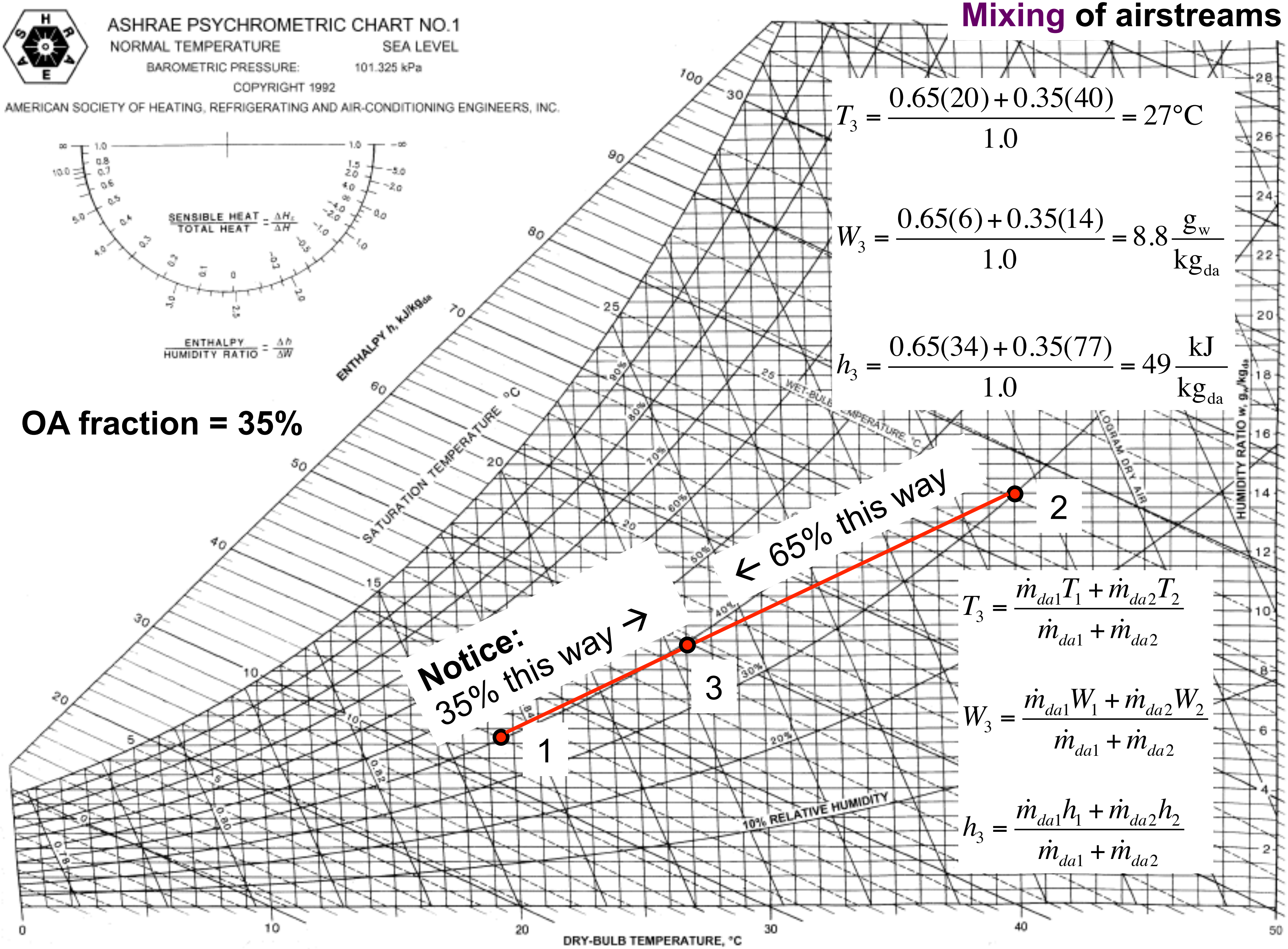
- Typical SHR: 0.6 to 0.9



Here is a process with an SHR ≈ 0.4

Mixing of airstreams

OA fraction = 35%



$$T_3 = \frac{0.65(20) + 0.35(40)}{1.0} = 27°\text{C}$$

$$W_3 = \frac{0.65(6) + 0.35(14)}{1.0} = 8.8\,\frac{\text{g}_\text{w}}{\text{kg}_\text{da}}$$

$$h_3 = \frac{0.65(34) + 0.35(77)}{1.0} = 49\,\frac{\text{kJ}}{\text{kg}_\text{da}}$$

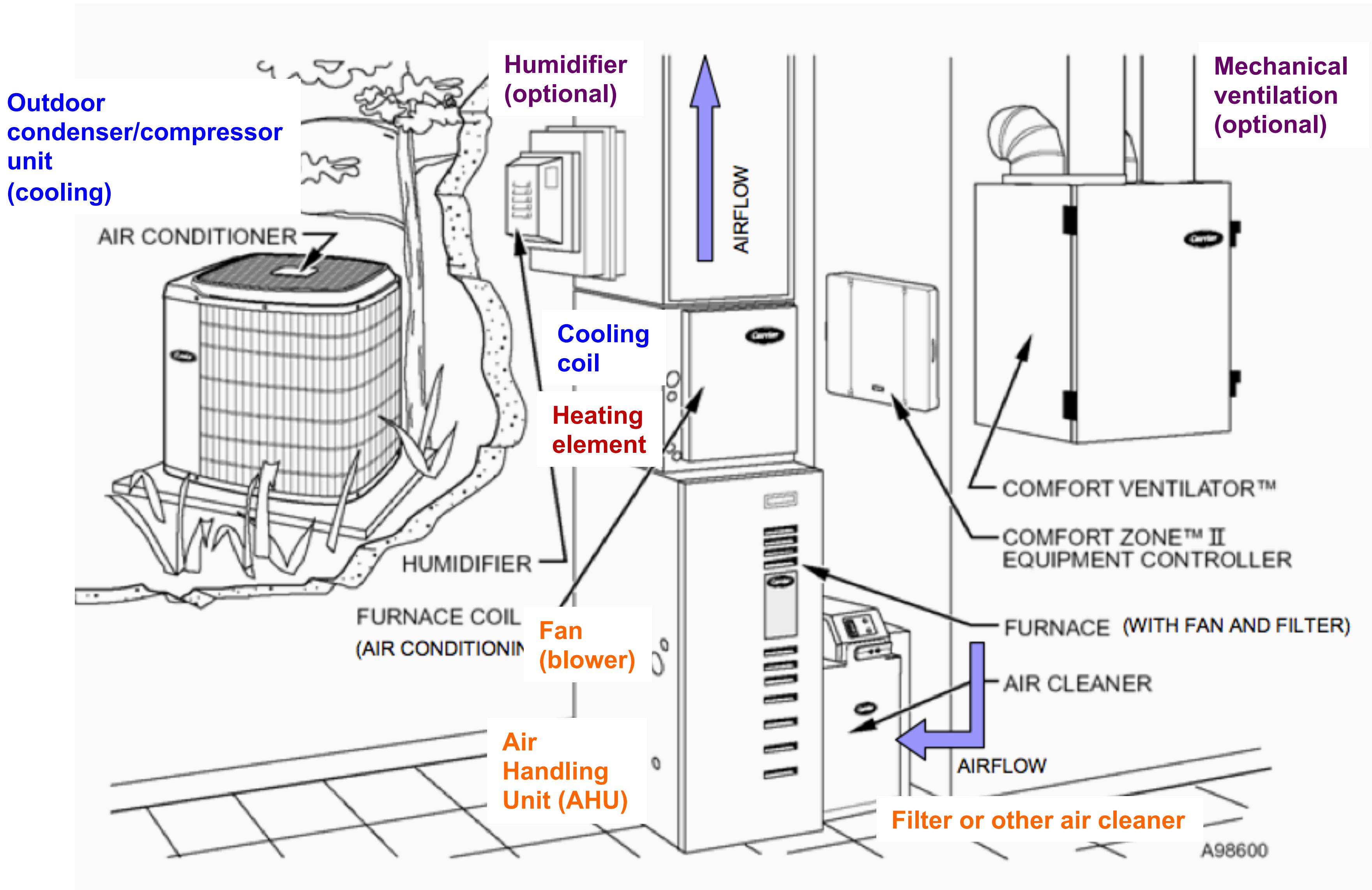
$$T_3 = \frac{\dot{m}_{da1}T_1 + \dot{m}_{da2}T_2}{\dot{m}_{da1} + \dot{m}_{da2}}$$

$$W_3 = \frac{\dot{m}_{da1}W_1 + \dot{m}_{da2}W_2}{\dot{m}_{da1} + \dot{m}_{da2}}$$

$$h_3 = \frac{\dot{m}_{da1}h_1 + \dot{m}_{da2}h_2}{\dot{m}_{da1} + \dot{m}_{da2}}$$

HVAC EQUIPMENT

Typical central residential system w/ upgrades



Typical large central commercial systems

Heating option 1: Steam boiler

AHU serves all rooms

Heating option 2: Hot water boiler

Cooling option 1: Air cooled chiller (chilled water)

Cooling option 2: Water cooled chiller w/ cooling tower (chilled water)