CAE 331/513
Building Science
Fall 2013

Lecture 4: September 16, 2013
Finishing solar radiation and windows
Psychrometrics and thermal comfort

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Deliverables

- HW 2 due today
  - Any questions?

- HW 1 everyone got 2 additional points for 1.19
  - MBtu vs. MMBtu

Scheduling

- Lectures I will miss
  - September 30th (conference)
  - October 7 (no class; Fall Break Day)
  - October 14th (conference) → will likely schedule exam for this period

- Final project guidelines forthcoming (for grad students)
  - Hopefully next week

- Plug for ASHRAE IIT Chapter membership
  - http://ashraeiit.wix.com/ashraeiit

CAE 463/524 Building Enclosure Design

- Design of building exteriors, including the control of heat flow, air and moisture penetration, building movements, and deterioration. Study of the principle of rain screen walls and of energy conserving designs. Analytical techniques and building codes are discussed through case studies and design projects.

Instructor's Course Objectives and Learning Outcomes

- To introduce students to the design of building enclosures, elements of which include walls, floors, roofs, and intentional openings. By taking this course students will be able to:
  - Design and assess building enclosure elements for heat transfer, airflow, and moisture control
  - Be proficient in current building codes and standards as they pertain to building enclosure design
  - Critically analyze designs for advanced building enclosures for their impacts on energy use, airflow, and potential moisture issues
  - Be proficient with several software tools used in building enclosure design

For those interested: Spring 2014 or Fall 2014?

Last time

- Finished up basics of heat transfer in buildings
  - Finished convection
  - Radiation
    - Short-wave
      - Solar radiation calculations and data
    - Long-wave
  - Performed some example problems

Today's objectives

- Finish solar radiation
- Heat transfer through windows
- Building energy balances

- Begin psychrometrics and thermal comfort

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

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

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

Radiation

Long-wave

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

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

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

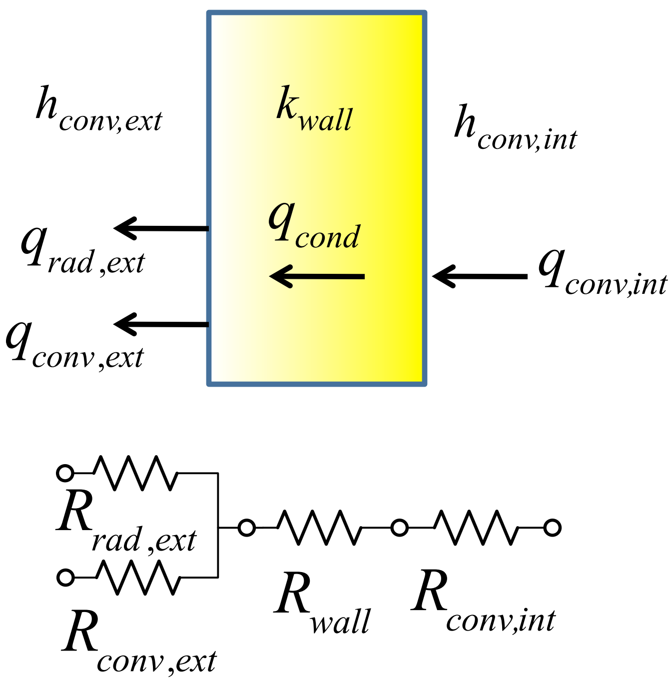
$$q_{1\to 2} = \varepsilon_{surf}\sigma F_{12}\left(T^4_{surf,1} - T^4_{surf,2}\right)$$

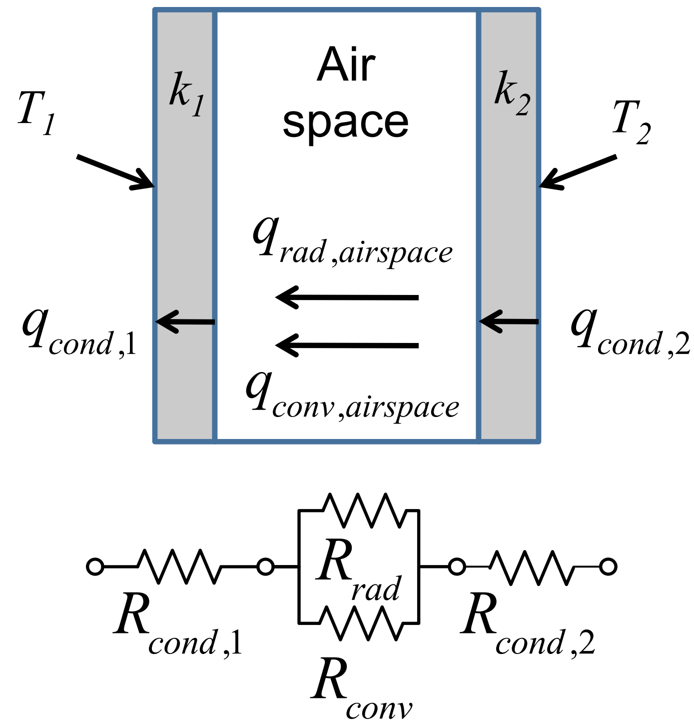
Solar radiation: (opaque surface) $q_{solar} = \alpha I_{solar}$

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

Combined heat transfer

- When more than one mode of heat transfer exists at a location (usually convection + radiation), we add the heat flow from each node to get the total
  - Resistances get placed in parallel
    - Example: Heat transfer to/from exterior wall or in a cavity





SOLAR RADIATION

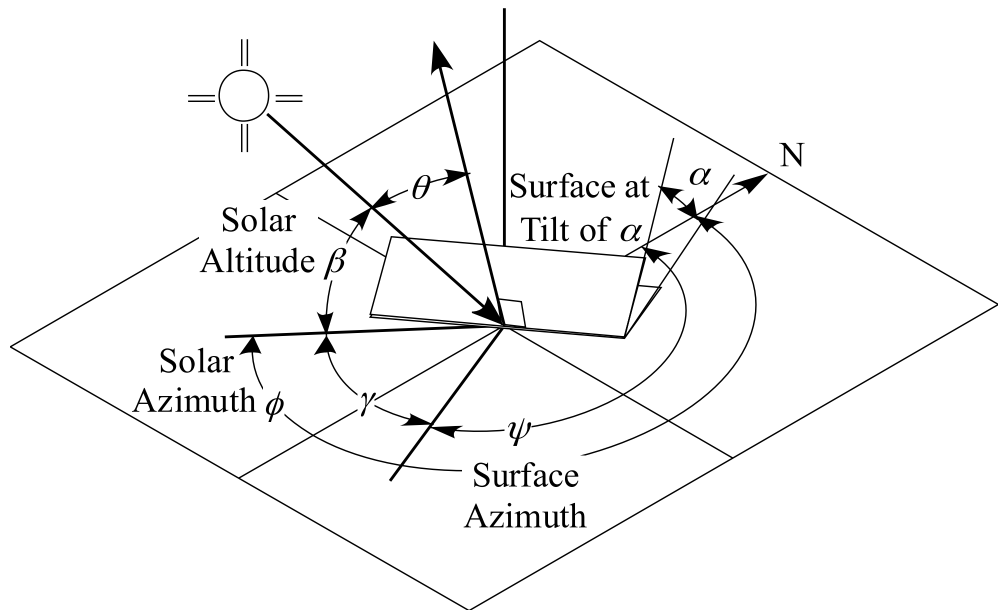
Solar radiation

- Solar radiation is an important term in the "energy balance" of a building
  - One 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
  - But will discuss basic relationships and where to download data
- CAE 463/524 Building Enclosure Design goes into more detail

Solar radiation striking an exterior surface

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

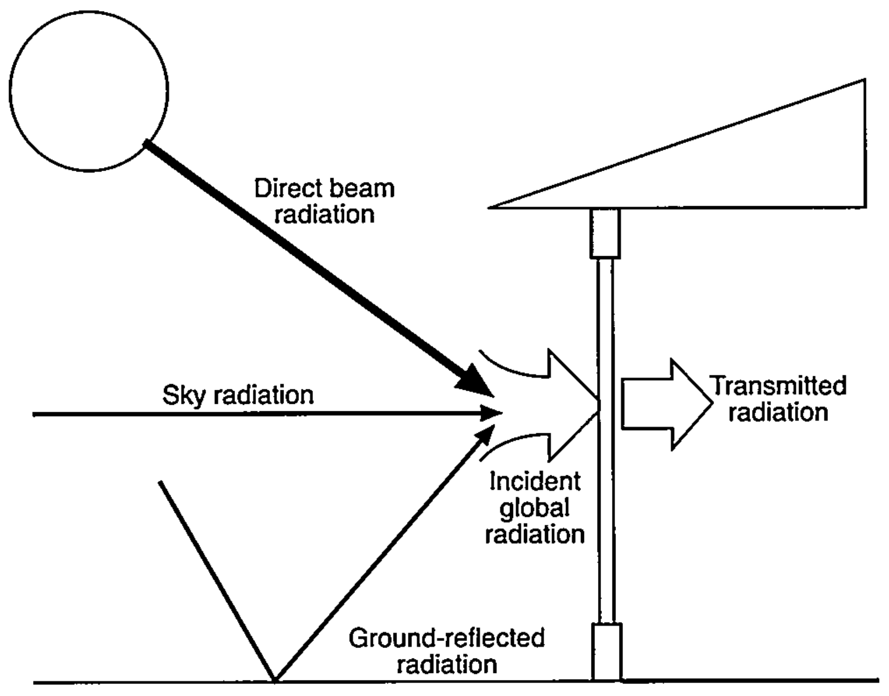
- This is a function of:
  - Solar geometry
    - Location
    - Time
  - Surface geometry
  - Shading/obstacles
  - Level of cloudiness



Components of solar radiation

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

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



Incident global solar radiation includes direct beam, sky, and ground-reflected radiation

Components of solar radiation

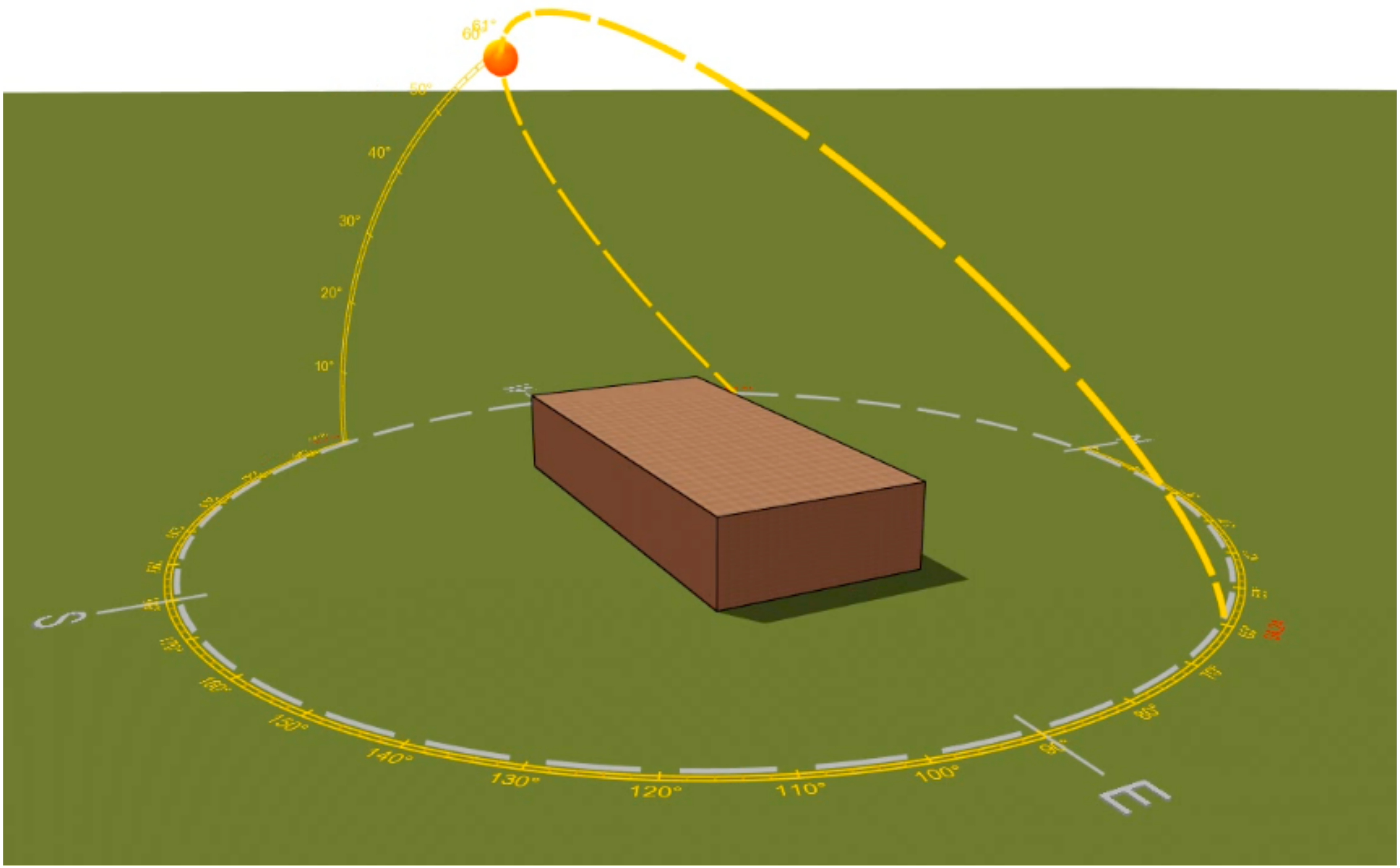
- Direct solar radiation (I_{direct}) is a function of the "normal incident irradiation" (I_{ND}) on the earth's surface and the solar incidence angle of the surface of interest, θ
  - Where I_{ND} is a function of day of the year and atmospheric properties

- Diffuse solar radiation ($I_{diffuse}$) is the irradiation that is scattered by the atmosphere
  - Function of I_{ND}, 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_{ND}, solar geometry, ground reflectance, and surface tilt angle

Visualizing solar relationships

- For visualizing geometry, using something like IES-VE



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 hourly solar *actual* 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 annual hourly load calculations and building energy simulations we often rely on a collection of weather data for a specific location

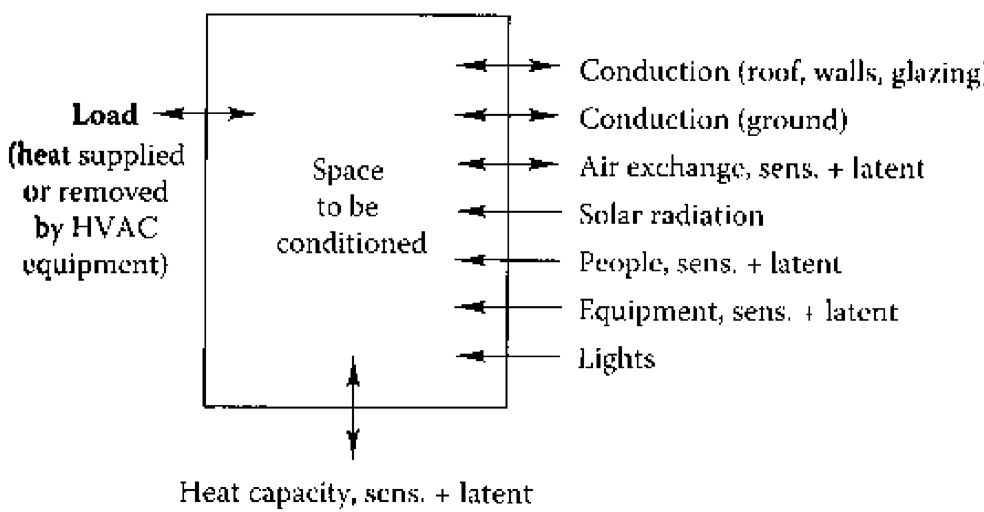
- We generate this data to be representative of more than just the previous year
  - Represents a wide range of weather phenomena for our location
  - TMY3: Data for 1020 locations from 1960 to 2005
    - Composed of 12 typical meteorological months
    - Each month is pulled from a random year in the range
    - Actual time-series climate data
    - Mixture of measured and modeled solar values

What to do with solar data once you have it?

- Solar data can be used on exterior opaque surfaces to help determine external surface temperatures

- Solar data can also be used on exterior transparent surfaces (e.g. windows and skylights) to determine how much solar radiation enters an indoor environment

- Both are used for a building's overall "energy balance"



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:

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

Sol-air temperatures

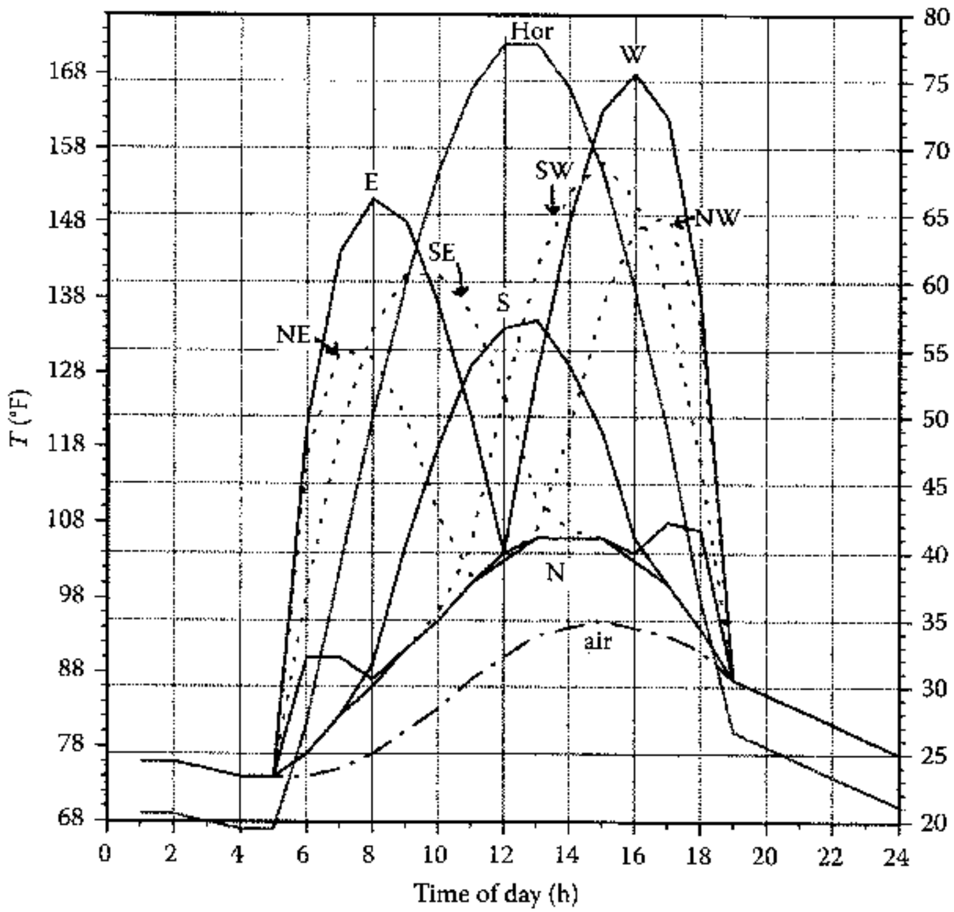


FIGURE 6.17
Sol-air temperature for horizontal and vertical surfaces as a function of time of day for summer design conditions, July 21 at 40° latitude, assuming $\alpha/h_o = 0.30$ (h · ft^2 · °F)/Btu [0.052 (m^2 · K)/W]. The curves overlap when there is no direct radiation on a surface. (Courtesy of ASHRAE, *Handbook of Fundamentals*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1989, Table 26.1.)

Solar radiation and external surface temperatures

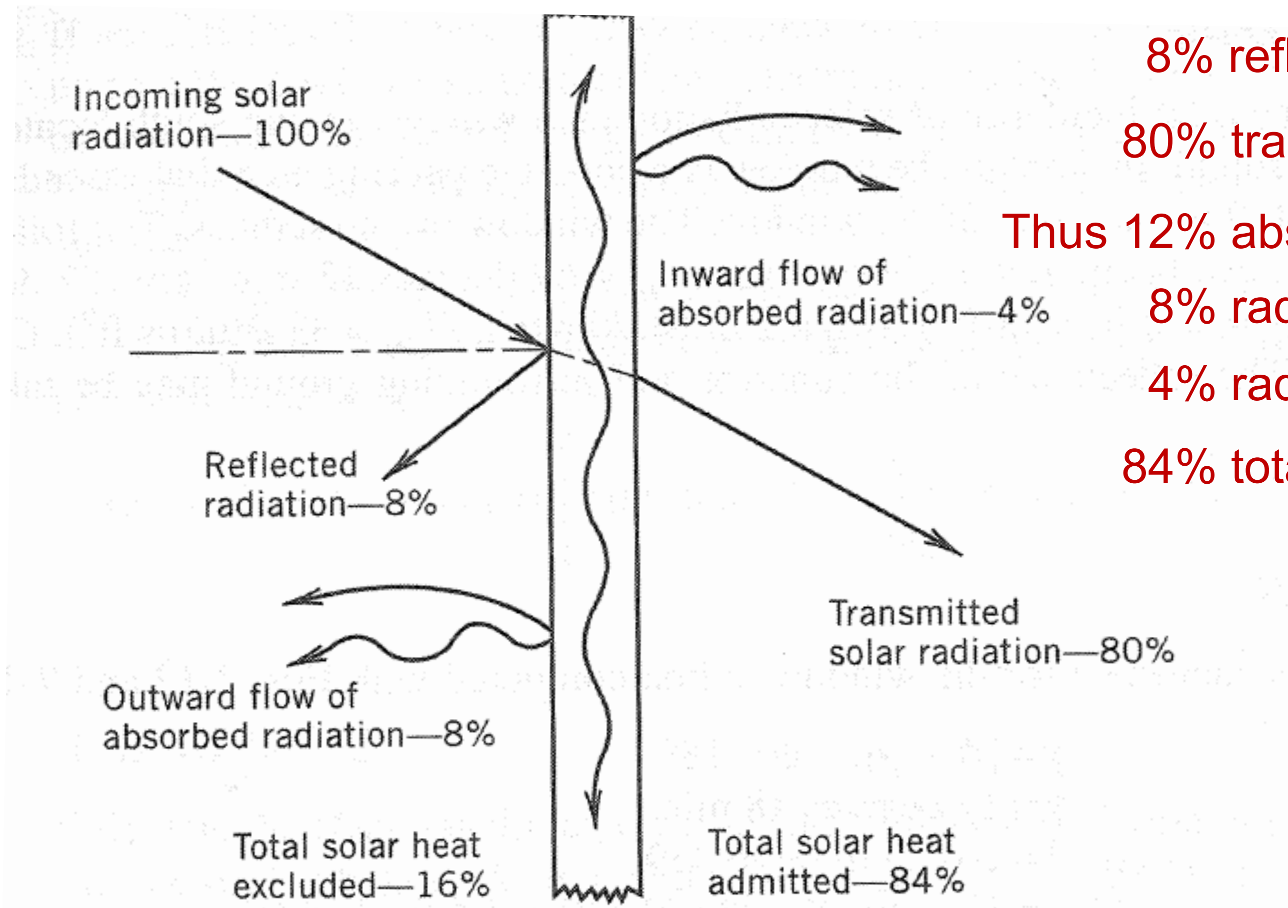
- We can also use air temperatures and material properties (emissivity and absorptance) to estimate exterior surface temperatures that are exposed to radiation
  - These are not perfectly accurate but provide a reasonable estimate for use in simple conduction

Situation	Thermally massive	Thermally lightweight
Roofs: direct sun	$t_a + 42\ \alpha$	$t_a + 55\ \alpha$
Roof: sun + reflected /emitted radiation	$t_a + 55\ \alpha$	$t_a + 72\ \alpha$
Roof exposed to night sky	$t_a - 5\ \varepsilon$	$t_a - 10\ \varepsilon$
Walls: winter sun	$t_a + 35\ \alpha$	$t_a + 48\ \alpha$
Walls: summer sun	$t_a + 28\ \alpha$	$t_a + 40\ \alpha$
Walls exposed to night sky	$t_a - 2\ \varepsilon$	$t_a - 4\ \varepsilon$

Source: Straube and Burnett

Solar radiation and windows (i.e., fenestration)

- Solar radiation through a single glaze



100% incident

8% reflected

80% transmitted

Thus 12% absorbed

8% rad/conv outward

4% rad/conv inward

84% total transmitted

Windows and total heat gain

- The total heat gain of a window is the sum of two terms:
  - The solar radiation heat gain from solar irradiation (transmittance)
  - Conductive/convective/radiative thermal heat gain from the temperature difference between the interior and exterior

- In the summer, both terms are positive towards the interior and add heat gains

- In the winter, solar is positive inwards but the other is negative towards the exterior
  - Net heat gain may vary in direction

Heat gain through windows

- Calculating the **thermal** heat gain through a window is easy

$$Q = UA\Delta T$$

- Accounting for **solar** heat gain is more complicated
  - Need to include spectral and angular characteristics of radiation and glazing
  - Need to include absorption of solar energy and re-radiation of thermal energy

- We can do this with a simplified metric
  - The solar heat gain coefficient (SHGC):

$$Q_{solar,window} = \left(I_{solar} A\right) SHGC$$

Solar heat gain coefficient, SHGC

$$Q_{solar,window} = \left(I_{solar} A\right) SHGC$$

- For a single pane of glass:

$$SHGC = \tau + \alpha \frac{U}{h_{ext}} \qquad \frac{1}{U} = \frac{1}{h_{int}} + \frac{1}{R_{glass}} + \frac{1}{h_{ext}}$$

*R_{glass} is negligible

- For double glazing with a small air space:

$$SHGC = \tau + \alpha_{outer\ pane} \frac{U}{h_{ext}} + \alpha_{inner\ pane} U \left(\frac{1}{h_{ext}} + \frac{1}{h_{airspace}} \right)$$

$$\frac{1}{U} = \frac{1}{h_{int}} + \frac{1}{R_{outer\ pane}} + \frac{1}{h_{airspace}} + \frac{1}{R_{inner\ pane}} + \frac{1}{h_{ext}}$$

*$R_{outer\ pane}$ and $R_{inner\ pane}$ are negligible

Multiple layers of glazing

- We can separate glass panes with air-tight layers of air or other gases

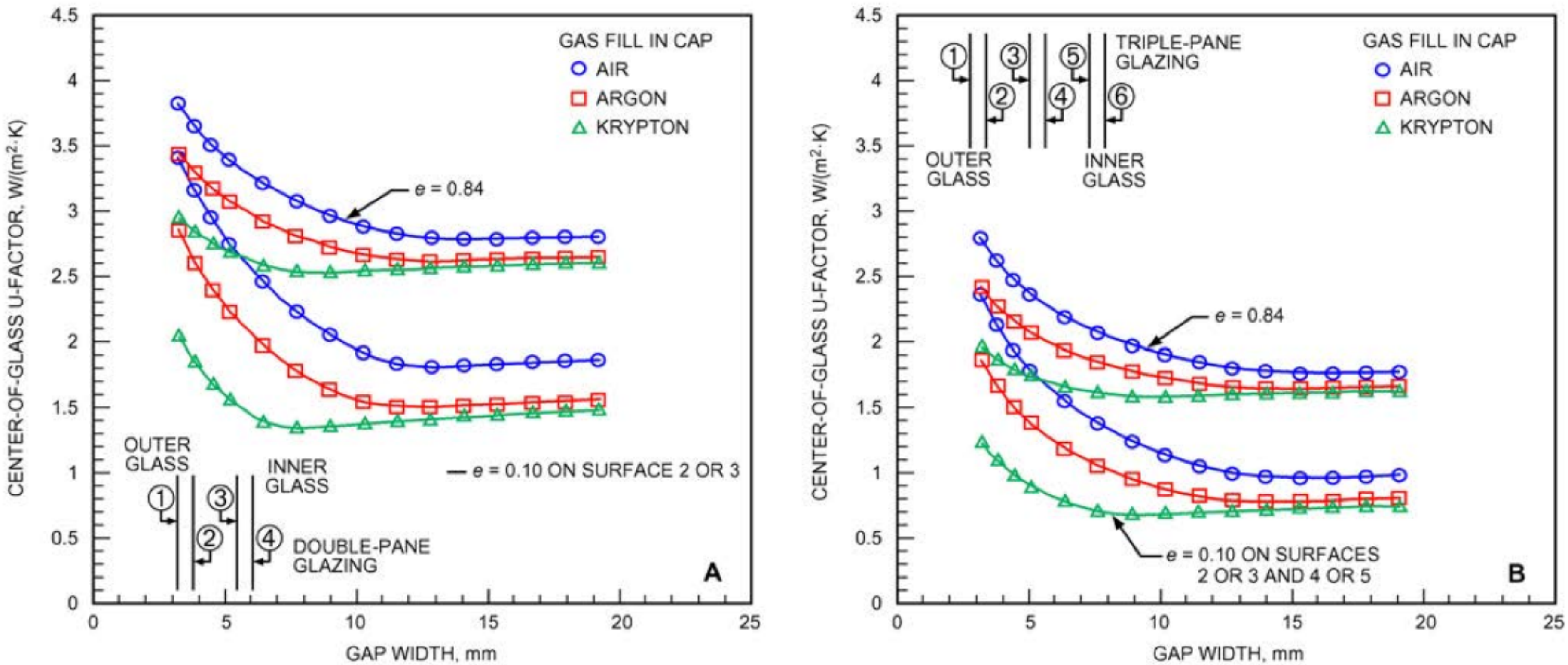
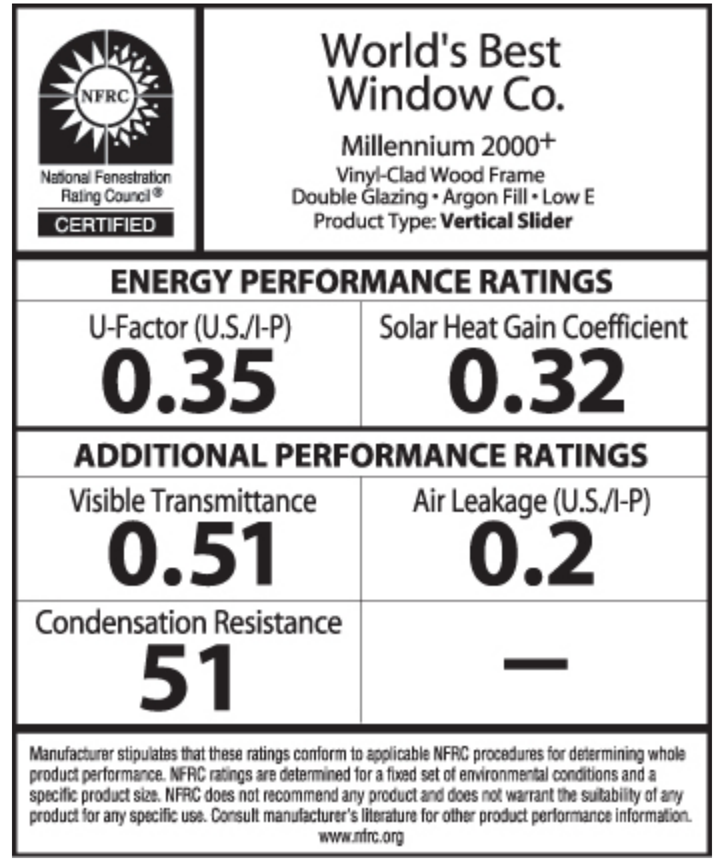


Fig. 3 Center-of-Glass U-Factor for Vertical Double- and Triple-Pane Glazing Units

2013 ASHRAE Handbook of Fundamentals: Chapter 15

Manufacturer supplied SHGC

- Glazing manufacturers will measure and present SHGC for normal incidence according to the methods of NFRC 200
  - National Fenestration Rating Council has developed methods for rating and labeling SHGC, U factors, air leakage, visible transmittance and condensation resistance of fenestration products
- In reality, SHGC is a function of incidence angle (θ)



$$Q_{solar,window} = I_{direct} SHGC(\theta) A + (I_{diffuse+reflected}) SHGC_{diffuse+reflected} A$$

Complex SHGC

- SHGC, solar transmittance, reflectance, and absorptance properties for glazing all vary with incidence angles of solar radiation
- The ASHRAE Handbook of Fundamentals 2013 Chapter 15 provides data for a large variety of glazing types

Table 10 Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance ($\mathcal{A}_n^f$) for Glazing and Window Systems

	Glazing System				Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
					Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
Uncoated Single Glazing																			
1a	3	CLR	0.90	SHGC	0.86	0.84	0.82	0.78	0.67	0.42	0.78	0.78	0.79	0.70	0.76	0.80	0.81	0.72	0.79
				T	0.83	0.82	0.80	0.75	0.64	0.39	0.75								
				R^f	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				R^b	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				$\mathcal{A}_1^f$	0.09	0.10	0.10	0.11	0.11	0.11	0.10								
1b	6	CLR	0.88	SHGC	0.81	0.80	0.78	0.73	0.62	0.39	0.73	0.74	0.74	0.66	0.72	0.78	0.79	0.70	0.77
				T	0.77	0.75	0.73	0.68	0.58	0.35	0.69								
				R^f	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				R^b	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				$\mathcal{A}_1^f$	0.16	0.17	0.18	0.19	0.19	0.17	0.17								

What about 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

Window U-factors

- U-values (or U-factors) for windows include all of the elements of the fenestration system
  - Center of glass properties (*cg*)
  - Edge of glass properties (*eg*)
  - Frame properties (*f*)

- The overall U-factor is estimated using area-weighted U-factors for each:

$$U = \frac{U_{cg} A_{cg} + U_{eg} A_{eg} + U_f A_f}{A_{pf}}$$

Combined U-factor data: ASHRAE 2013

Table 4 U-Factors for Various Fenestration Products in W/(m²·K)

Product Type		Glass Only		Vertical Installation: Operable (including sliding and swinging glass doors)					Vertical Installation: Fixed				
Frame Type													
ID	**Glazing Type**	**Center of Glass**	**Edge of Glass**	**Aluminum Without Thermal Break**	**Aluminum With Thermal Break**	**Reinforced Vinyl/ Aluminum Clad Wood**	**Wood/ Vinyl**	**Insulated Fiberglass/ Vinyl**	**Aluminum Without Thermal Break**	**Aluminum With Thermal Break**	**Reinforced Vinyl/ Aluminum Clad Wood**	**Wood/ Vinyl**	**Insulated Fiberglass/ Vinyl**
	Single Glazing												
1	3 mm glass	*5.91*	*5.91*	7.01	6.08	5.27	5.20	4.83	6.38	6.06	5.58	5.58	5.40
2	6 mm acrylic/polycarb	*5.00*	*5.00*	6.23	5.35	4.59	4.52	4.18	5.55	5.23	4.77	4.77	4.61
3	3.2 mm acrylic/polycarb	*5.45*	*5.45*	6.62	5.72	4.93	4.86	4.51	5.96	5.64	5.18	5.18	5.01
	Double Glazing												
4	6 mm airspace	*3.12*	*3.63*	4.62	3.61	3.24	3.14	2.84	3.88	3.52	3.18	3.16	3.04
5	13 mm airspace	*2.73*	*3.36*	4.30	3.31	2.96	2.86	2.58	3.54	3.18	2.85	2.83	2.72
6	6 mm argon space	*2.90*	*3.48*	4.43	3.44	3.08	2.98	2.69	3.68	3.33	3.00	2.98	2.86
7	13 mm argon space	*2.56*	*3.24*	4.16	3.18	2.84	2.74	2.46	3.39	3.04	2.71	2.69	2.58
	Double Glazing, e = 0.60 on surface 2 or 3												
8	6 mm airspace	*2.95*	*3.52*	4.48	3.48	3.12	3.02	2.73	3.73	3.38	3.04	3.02	2.90
9	13 mm airspace	*2.50*	*3.20*	4.11	3.14	2.80	2.70	2.42	3.34	2.99	2.67	2.65	2.53
10	6 mm argon space	*2.67*	*3.32*	4.25	3.27	2.92	2.82	2.54	3.49	3.13	2.81	2.79	2.67
11	13 mm argon space	*2.33*	*3.08*	3.98	3.01	2.68	2.58	2.31	3.20	2.84	2.52	2.50	2.39
	Double Glazing, e = 0.40 on surface 2 or 3												
12	6 mm airspace	*2.78*	*3.40*	4.34	3.35	3.00	2.90	2.61	3.59	3.23	2.90	2.88	2.77
13	13 mm airspace	*2.27*	*3.04*	3.93	2.96	2.64	2.54	2.27	3.15	2.79	2.48	2.46	2.35
14	6 mm argon space	*2.44*	*3.16*	4.07	3.09	2.76	2.66	2.38	3.30	2.94	2.62	2.60	2.49
15	13 mm argon space	*2.04*	*2.88*	3.75	2.79	2.48	2.38	2.11	2.95	2.60	2.29	2.27	2.16
	Double Glazing, e = 0.20 on surface 2 or 3												
16	6 mm airspace	*2.56*	*3.24*	4.16	3.18	2.84	2.74	2.46	3.39	3.04	2.71	2.69	2.58
17	13 mm airspace	*1.99*	*2.83*	3.70	2.75	2.44	2.34	2.07	2.91	2.55	2.24	2.22	2.12
18	6 mm argon space	*2.16*	*2.96*	3.84	2.88	2.56	2.46	2.19	3.05	2.70	2.38	2.36	2.26
19	13 mm argon space	*1.70*	*2.62*	3.47	2.53	2.24	2.14	1.88	2.66	2.30	2.00	1.98	1.88
	Double Glazing, e = 0.10 on surface 2 or 3												
20	6 mm airspace	*2.39*	*3.12*	4.02	3.05	2.72	2.62	2.34	3.25	2.89	2.57	2.55	2.44
21	13 mm airspace	*1.82*	*2.71*	3.56	2.62	2.32	2.22	1.96	2.76	2.40	2.10	2.08	1.98
22	6 mm argon space	*1.99*	*2.83*	3.70	2.75	2.44	2.34	2.07	2.91	2.55	2.24	2.22	2.12
23	13 mm argon space	*1.53*	*2.49*	3.33	2.40	2.12	2.02	1.76	2.51	2.16	1.86	1.84	1.74

A note on air cavities

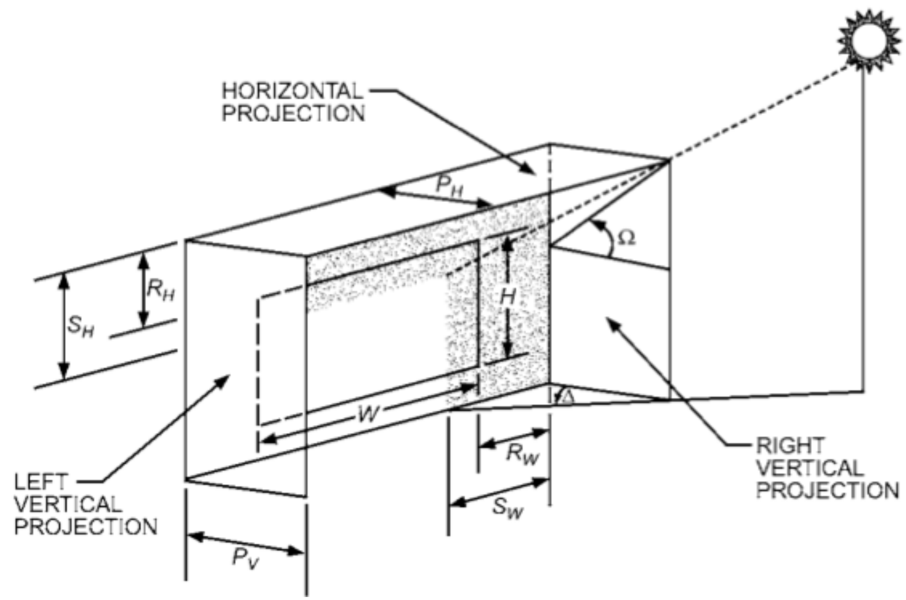
Table 3 Thermal Resistances of Plane Air Spaces,[a,b,c] $(m^2 \cdot K)/W$

Position of Air Space	Direction of Heat Flow	Air Space Mean Temp.[d], °C	Air Space Temp. Diff.,[d] K	13 mm Air Space[c] 0.03	0.05	0.2	0.5	0.82	20 mm Air Space[c] 0.03	0.05	0.2	0.5	0.82
				Effective Emittance ε_{eff}[d,e]									
Horiz.	Up ↑	32.2	5.6	0.37	0.36	0.27	0.17	0.13	0.41	0.39	0.28	0.18	0.13
		10.0	16.7	0.29	0.28	0.23	0.17	0.13	0.30	0.29	0.24	0.17	0.14
		10.0	5.6	0.37	0.36	0.28	0.20	0.15	0.40	0.39	0.30	0.20	0.15
		−17.8	11.1	0.30	0.30	0.26	0.20	0.16	0.32	0.32	0.27	0.20	0.16
		−17.8	5.6	0.37	0.36	0.30	0.22	0.18	0.39	0.38	0.31	0.23	0.18
		−45.6	11.1	0.30	0.29	0.26	0.22	0.18	0.31	0.31	0.27	0.22	0.19
		−45.6	5.6	0.36	0.35	0.31	0.25	0.20	0.38	0.37	0.32	0.26	0.21
45° Slope	Up ↗	32.2	5.6	0.43	0.41	0.29	0.19	0.13	0.52	0.49	0.33	0.20	0.14
		10.0	16.7	0.36	0.35	0.27	0.19	0.15	0.35	0.34	0.27	0.19	0.14
		10.0	5.6	0.45	0.43	0.32	0.21	0.16	0.51	0.48	0.35	0.23	0.17
		−17.8	11.1	0.39	0.38	0.31	0.23	0.18	0.37	0.36	0.30	0.23	0.18
		−17.8	5.6	0.46	0.45	0.36	0.25	0.19	0.48	0.46	0.37	0.26	0.20
		−45.6	11.1	0.37	0.36	0.31	0.25	0.21	0.36	0.35	0.31	0.25	0.20
		−45.6	5.6	0.46	0.45	0.38	0.29	0.23	0.45	0.43	0.37	0.29	0.23
Vertical	Horiz. →	32.2	5.6	0.43	0.41	0.29	0.19	0.14	0.62	0.57	0.37	0.21	0.15
		10.0	16.7	0.45	0.43	0.32	0.22	0.16	0.51	0.49	0.35	0.23	0.17
		10.0	5.6	0.47	0.45	0.33	0.22	0.16	0.65	0.61	0.41	0.25	0.18
		−17.8	11.1	0.50	0.48	0.38	0.26	0.20	0.55	0.53	0.41	0.28	0.21
		−17.8	5.6	0.52	0.50	0.39	0.27	0.20	0.66	0.63	0.46	0.30	0.22
		−45.6	11.1	0.51	0.50	0.41	0.31	0.24	0.51	0.50	0.42	0.31	0.24
		−45.6	5.6	0.56	0.55	0.45	0.33	0.26	0.65	0.63	0.51	0.36	0.27

2005 ASHRAE Handbook of Fundamentals: Chapter 25; 2013 ASHRAE Handbook of Fundamentals: Chapter 26

What about shading?

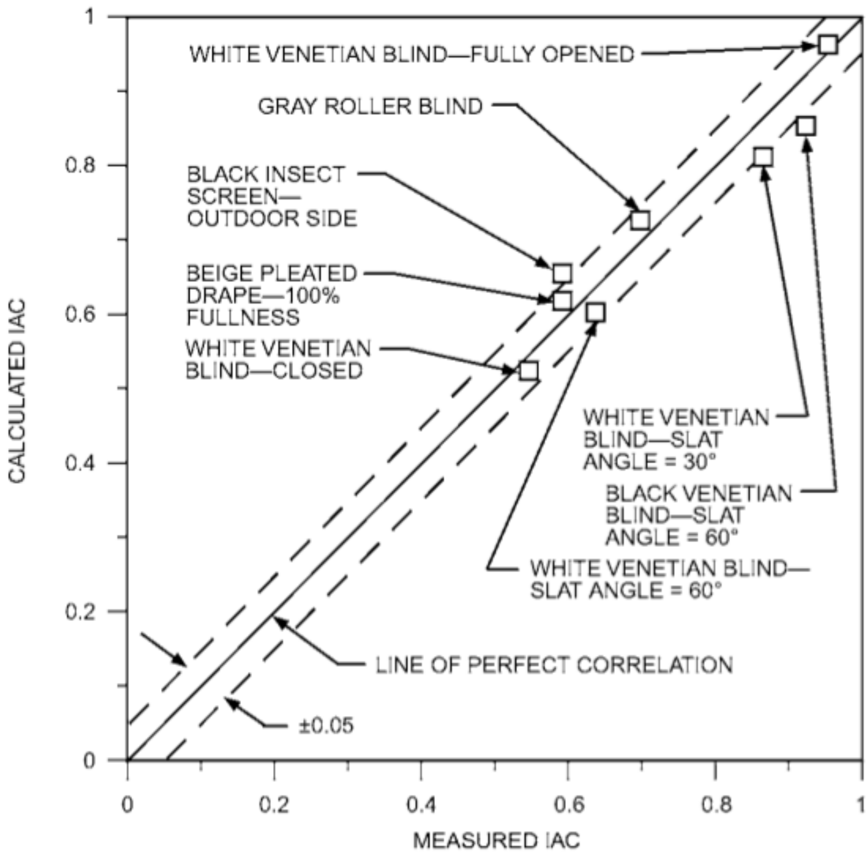
- Shading devices, including drapes and blinds, can mitigate some solar heat gain
- We can attempt to describe this with an **indoor solar attenuation coefficient (IAC)**
- Heat gain through a window can be modified as follows:

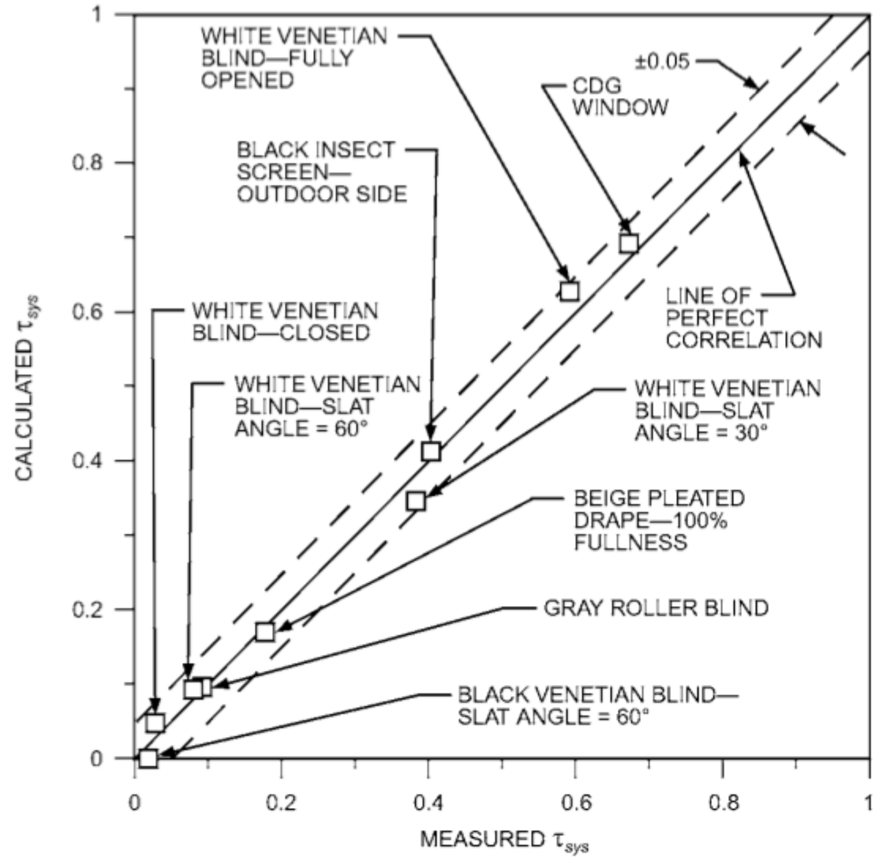


$$Q_{window} = UA_{pf}\left(T_{out} - T_{in}\right) + I_{direct} A_{pf} SHGC(\theta) IAC(\theta,\Omega) + (I_{diffuse+reflected}) A_{pf} SHGC_{diffuse+reflected} IAC_{diffuse+reflected}$$

IAC is a function of incidence angle, θ, and the angle created by a shading device

Blinds and drapes: ASHRAE Handbook





Combined thermal transmittance for walls + fenestration

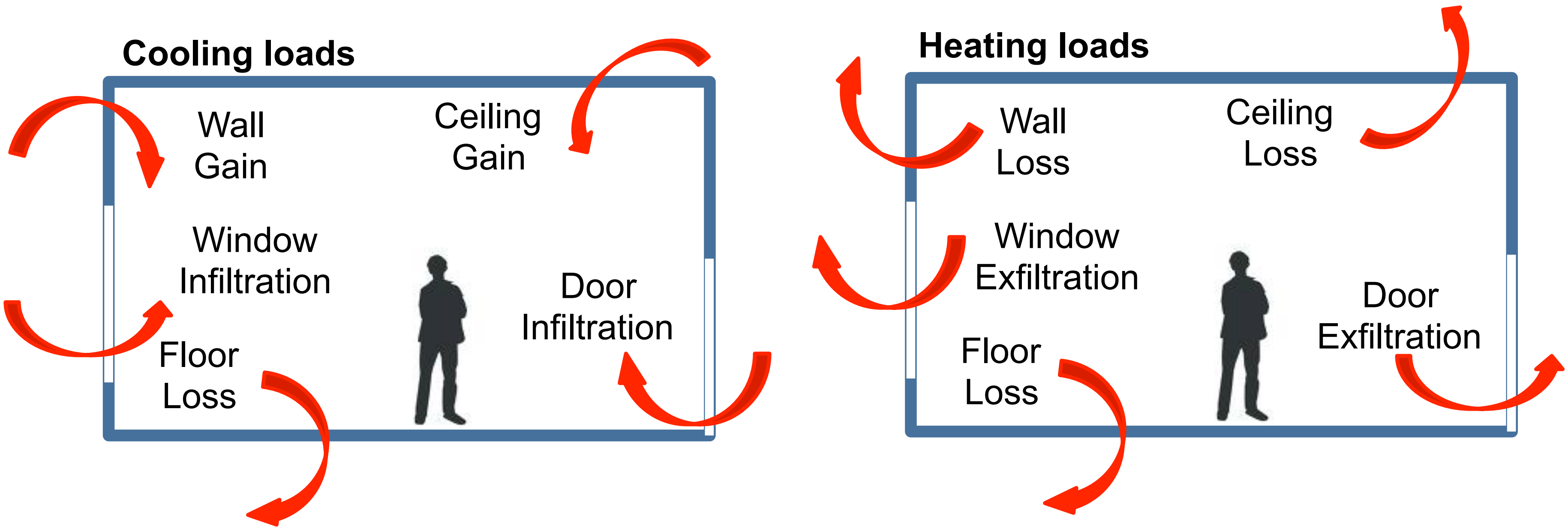
- Single assemblies of walls, windows, doors, etc. can be combined into an overall U-value for a building's enclosure
  - **Combined thermal transmittance**: U_o or U_{total}
    - Area-weighted average U-value
    - Just like center of glass, edge of glass, frame analysis for windows

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

We will use this later for calculating heating and cooling loads

Building energy balances

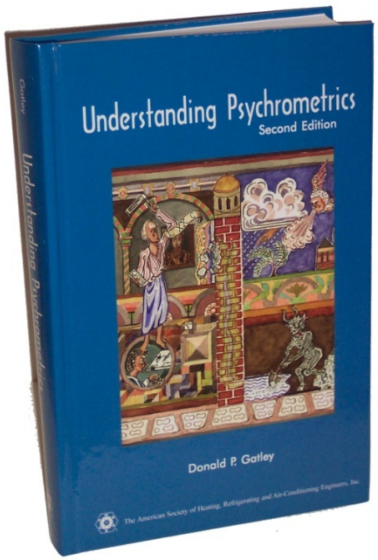
- Taken altogether, each of the heat transfer modes we've discussed can be combined with inputs for climate data, material properties, and geometry to make up a building's **energy balance**
  - We will revisit this for heating and cooling load calculations



PSYCHROMETRICS AND THERMAL COMFORT

Kreider Ch. 4

Psychrometrics



Psychrometrics is the science and engineering of air/vapor mixtures

- For architectural engineers the vapor is usually water
- The term psychrometry comes from the Greek *psuchron* meaning cold and *metron* meaning measure
- In building engineering we use psychrometrics to relate the thermodynamic and physical properties of moist air

Applying psychrometrics

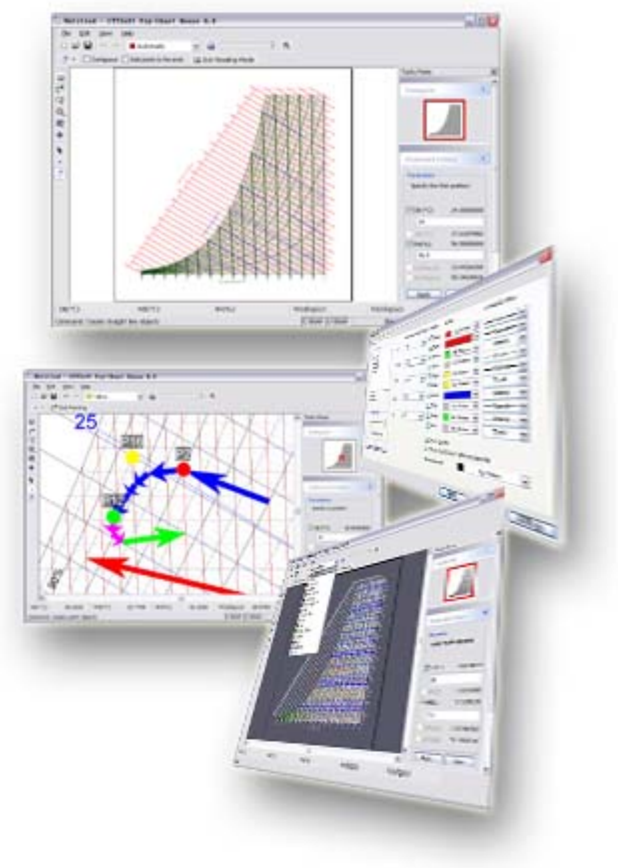
- We need to understand air temperature and moisture content to understand human thermal comfort

- In hot, humid weather we remove moisture by dehumidification/cooling

- In dry, cold weather, we add moisture by humidifiers

- We are also concerned about moisture for structural, aesthetic, and indoor air quality

Applying psychrometrics



- Most engineers look up properties of moist air in tables, on a psychrometric chart or with software
  - We will learn that later

- Psychrometrics also involves learning how to use and combine those quantities to determine things like heating and cooling loads

Composition of dry and moist air

- **Atmospheric air** contains:
  - Many gaseous components
  - Water vapor
  - Contaminants (smoke, pollen, gaseous pollutants)

- **Dry air** is atmospheric air with all of the water vapor and contaminants removed

- **Moist air** is a two-component mixture of dry air and water vapor

Standard composition of dry air

Gas	Molecular weight (g/mol)	Volume %
Nitrogen (N_2)	32.000	78.084
Oxygen (O_2)	28.016	20.946
Argon (Ar)	39.444	0.9340
Carbon Dioxide (CO_2)	44.010	0.03697
Neon (Ne)	20.179	0.00182
Helium (He)	4.002	0.00052
Methane (CH_4)	16.042	0.00014
Krypton	83.800	0.00010

Where does water fit in?

Standard composition of moist air

Gas	Molecular weight (g/mol)	Volume %
Nitrogen (N_2)	32.000	78.084%
Oxygen (O_2)	28.016	20.946%
Water (H_2O)	18.015	0 to 4%
Argon (Ar)	39.444	0.9340%
Carbon Dioxide (CO_2)	44.010	0.03697%
Neon (Ne)	20.179	0.00182%
Helium (He)	4.002	0.00052%
Methane (CH_4)	16.042	0.00014%
Krypton	83.800	0.00010%

Where does water fit in?

Treating air as an ideal gas

- At typical temperatures and pressures within buildings, air and its constituents act approximately as ideal gases

- Each gas i in the mixture, as well as the entire mixture, will follow the ideal gas law:

$$pV = nRT$$

or

$$pv = \frac{p}{\rho} = RT$$

p = pressure (Pa)

V = volume (m^3)

n = number of moles (#)

R = gas constant ($Pa \cdot m^3/(mol\ K)$)

T = absolute temperature (K)

v = specific volume ($= 1/\rho = m^3/kg$)

ρ = density (kg/m^3)

Universal gas constant

- The universal gas constant relates energy and temperature
- Takes many forms depending on units

Value of R	**Units ($V\ P\ T^{-1}\ n^{-1}$)**
8.314	J/(K·mol)
8.314	m^3·Pa/(K·mol)
0.08206	L·atm/(K·mol)
8.206×10^{-5}	m^3·atm/(K·mol)
10.731	ft^3·psi/(R·lb-mol)
1.986	Btu/(lb-mol·R)

$$pv = \frac{p}{\rho} = RT$$

Mass-specific gas constants

- To work with air and water vapor we need gas-specific gas constants (which are functions of molecular weight)

$$R_i = \frac{R}{MW_i}$$

- Dry air (no water vapor): MW_{da} = 28.965 g/mol

$$R_{da} = \frac{R}{MW_{da}} = \frac{8.314 \dfrac{\mathrm{J}}{\mathrm{K \cdot mol}}}{28.965 \dfrac{\mathrm{g}}{\mathrm{mol}}} \frac{1000\mathrm{g}}{\mathrm{kg}} = 287 \frac{\mathrm{J}}{\mathrm{kg \cdot K}}$$

- Water vapor alone: MW_w = 18.015 g/mol

$$R_w = \frac{R}{MW_w} = \frac{8.314 \dfrac{\mathrm{J}}{\mathrm{K \cdot mol}}}{18.015 \dfrac{\mathrm{g}}{\mathrm{mol}}} \frac{1000\mathrm{g}}{\mathrm{kg}} = 462 \frac{\mathrm{J}}{\mathrm{kg_w \cdot K}}$$

$$pv = \frac{p}{\rho} = RT$$

Air pressure variations

- The barometric (atmospheric) pressure and temperature of air vary with both altitude and local weather conditions
- But there are standard values for pressure as a function of altitude that are normally used
- At sea level, the standard temperature is 15°C and the standard pressure is 101.325 kPa (1 atm)
- Temperature is assumed to decrease linearly with altitude
  - Pressure is more complicated

$$T_{air} = 15 - 0.0065Z \qquad p = 101.325\left(1 - \left(2.25577 \times 10^{-5}\right)Z\right)^{5.2559}$$

$$pv = \frac{p}{\rho} = RT$$

T = temperature (°C)

Z = altitude (m)

p = barometric pressure (kPa)

Air pressure variations

Table 1 Standard Atmospheric Data for Altitudes to 10 000 m

Altitude, m	Temperature, °C	Pressure, kPa
–500	18.2	107.478
0	15.0	101.325
500	11.8	95.461
1000	8.5	89.875
1500	5.2	84.556
2000	2.0	79.495
2500	–1.2	74.682
3000	–4.5	70.108
4000	–11.0	61.640
5000	–17.5	54.020
6000	–24.0	47.181
7000	–30.5	41.061
8000	–37.0	35.600
9000	–43.5	30.742
10 000	–50	26.436

Source: Adapted from NASA (1976).

Dalton's law of partial pressures

- In an ideal gas, the total pressure can be considered to be the sum of the partial pressures of the constituent gases

$$p = p_{N_2} + p_{O_2} + p_{H_2O} + p_{CO_2} + p_{Ar} + \dots$$

- We can consider moist air as dry air combined with water vapor and break the pressure into only two partial pressures

$$p = p_{da} + p_w$$

Dalton's law of partial pressures

- We can analyze the dry air, the water vapor, and the mixture of each gas using the ideal gas law and assuming they are all at the same temperature

$$p_{da} v_{da} = R_{da} T \quad \& \quad p_w v_w = R_w T \quad \& \quad pv = RT$$

- For each individual gas, a mole fraction (Y_i) can be defined as the ratio of the partial pressure of gas i to the total pressure

$$\frac{n_i}{n} = \frac{p_i}{p} = Y_i$$

Describing moist air

- To describe and deal with moist air, we need to somehow discuss the fraction 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

Saturation

- Air can hold moisture (water vapor)
- The amount of moisture air can hold in vapor form before condensation occurs is dependent on temperature
- We call the limit ***saturation***

Saturation vapor pressure, p_{ws}

- The saturation vapor pressure at a given temperature is a very important quantity that we often need for additional computations
  - a.k.a. the partial pressure of water vapor at saturation (p_{ws})

- We can look up p_{ws} in tables
  - Table 3 in Ch.1 of 2003 ASHRAE Fundamentals (Ch. 6 of 2005)

- We can also use empirical equations
  - NOTE: Eqns give ln(p_{ws}) get p_{ws} by: $$p_{ws} = e^{\ln(p_{ws})}$$

p_{ws} for -100°C< T <0°C

For p_{ws}, the saturation pressure over ice:

$$\ln p_{ws} = \frac{C_1}{T} + C_2 + C_3 T + C_4 T^2 + C_5 T^3 + C_6 T^4 + C_7 \ln T$$

where

$C_1 = -5.674\,535\,9\text{ E}+03$
$C_2 = 6.392\,524\,7\text{ E}+00$
$C_3 = -9.677\,843\,0\text{ E}-03$
$C_4 = 6.221\,570\,1\text{ E}-07$
$C_5 = 2.074\,782\,5\text{ E}-09$
$C_6 = -9.484\,024\,0\text{ E}-13$
$C_7 = 4.163\,501\,9\text{ E}+00$

Note:
These constants are only for SI units
IP units are different

Units:

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

p_{ws} for 0°C< T <200°C

For p_{ws}, the saturation pressure over liquid water:

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

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

Note:
These constants are only for SI units
IP units are different

*Will use this equation for most conditions in building science

Units:

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

Humidity ratio, *W*

- The humidity ratio, *W*, is ratio of the mass of water vapor to mass of dry air in a given volume
  - We use *W* when finding other mixture properties
  - Note 1: *W* is small (*W* < 0.04 for any real building conditions)
  - Note 2: *W* is sometimes expressed in grains/lb where 1 lb = 7000 grains (try not to ever use this)

UNITS

$$W = \frac{m_w}{m_{da}} = \frac{MW_w p_w}{M_{da} p_{da}} = 0.622 \frac{p_w}{p_{da}} = 0.622 \frac{p_w}{p - p_w} \quad [\frac{\mathrm{kg_w}}{\mathrm{kg_{da}}}]$$

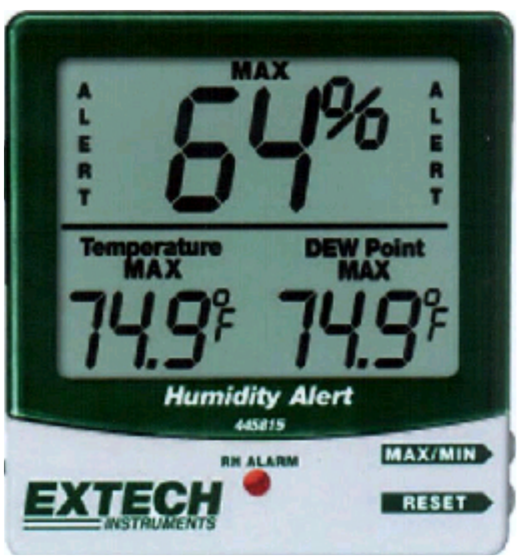
Saturation humidity ratio, W_s

- At a given temperature T and pressure P there is a maximum W that can be obtained
- If we try to add any more moisture, it will just condense out
  - It is when the partial pressure of vapor has reached the saturation pressure
- This maximum humidity ratio is called the saturation humidity ratio, W_s
  - From our previous equation we can write:

$$W_s = 0.622 \frac{p_s}{p_{da}} = 0.622 \frac{p_s}{p - p_s}$$

Relative humidity, ϕ (RH)

- The relative humidity ratio, ϕ, is the mole fraction of water vapor (x_w) relative to the water vapor that would be in the mixture if it were saturated at the given T and P (x_{ws})
- Relative humidity is a common measure that relates well to how we perceive moisture in air



$$\phi = \left[\frac{x_w}{x_{ws}}\right]_{T,P} = \frac{p_w}{p_{ws}}$$

Degree of saturation, μ

- The degree of saturation, μ, is the ratio of the humidity ratio W to that of a saturated mixture W_s at the same T and P
  - Note that μ and ϕ are not quite the same
  - Their values are very similar at lower temperatures but may differ a lot at higher temperatures

$$\mu = \left[\frac{W}{W_s}\right]_{T,P}$$

$$\mu = \frac{\phi}{1+(1-\phi)W_s/(0.6295)}$$

$$\phi = \frac{\mu}{1-(1-\mu)p_{ws}/p}$$

Specific volume, v

- The specific volume of moist air (or the volume per unit mass of air) can be expressed as:

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

where

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

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

- If we have v then we can also find moist air density, ρ:

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

Enthalpy, *h*

- The enthalpy of a mixture of perfect gases equals the sum of the individual partial enthalpies of the components
- Therefore, the enthalpy (h) for moist air is:

$$h = h_{da} + W h_g$$

h = enthalpy for moist air [kJ/kg]
h_g = specific enthalpy for saturated water vapor (i.e., h_{ws}) [kJ/kg$_w$]
h_{da} = specific enthalpy for dry air (i.e., h_{ws}) [kJ/kg$_{da}$]

- Some approximations:

$$h_{da} \approx 1.006T \qquad h_g \approx 2501 + 1.86T$$

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

*where T is in °C

Specifying the state of moist air

In order to specify the state of moist air, we need pressure, p, the air temp, T, and one other property

- W, ϕ, or h would be fine, but these cannot always be measured easily

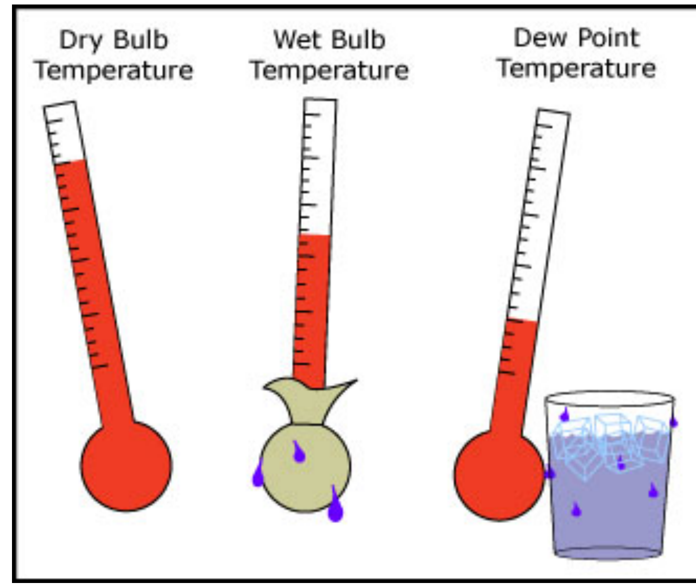
We need another property, easily measured but related to the moisture content that we can use

- Temperatures are easy to measure so let's look at another way relate temperature and humidity

Four different temperatures: T, T_{dew}, T_{wb}, T^*

The standard temperature, T, we have been using is called the "dry-bulb" temperature, or T_d

- It is a measure of internal energy



We can also describe the following:

- Dew-point temperature, T_{dew}
- Adiabatic saturation temperature, T^*
- Wet-bulb temperature, T_{wb}

Dew-point temperature, T_{dew}

The dew point temperature, T_{dew}, is the air temperature at which the current humidity ratio W is equal to the saturation humidity ratio W_s at the same temperature

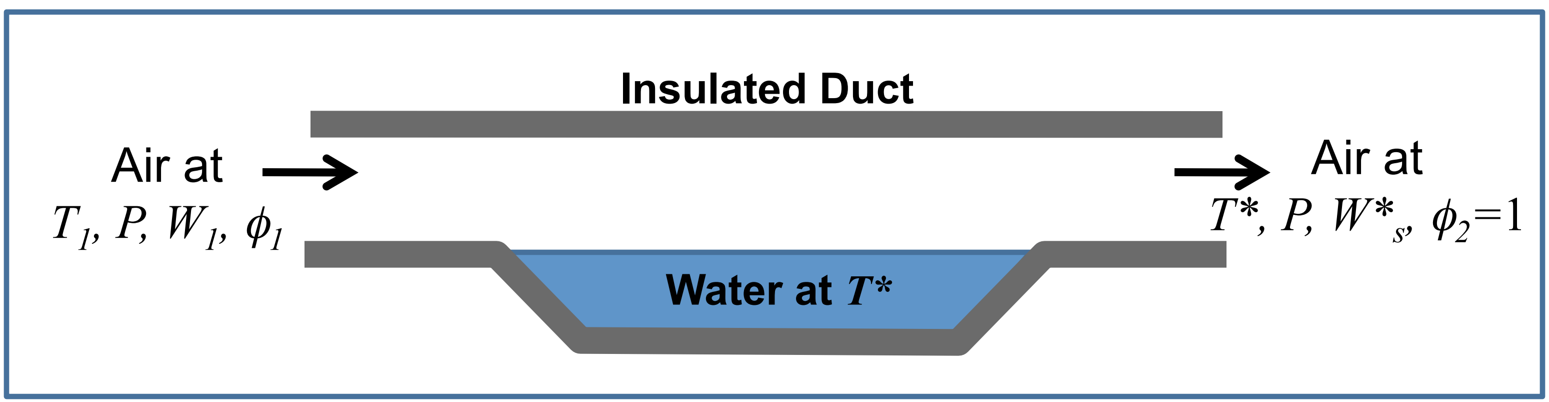
i.e. $W_s(p, T_{dew}) = W$

When the air temperature is lowered to the dew-point at constant pressure, the relative humidity rises to 100% and condensation occurs

T_{dew} is a direct measure of the humidity ratio W since $W = W_s$ at $T = T_{dew}$

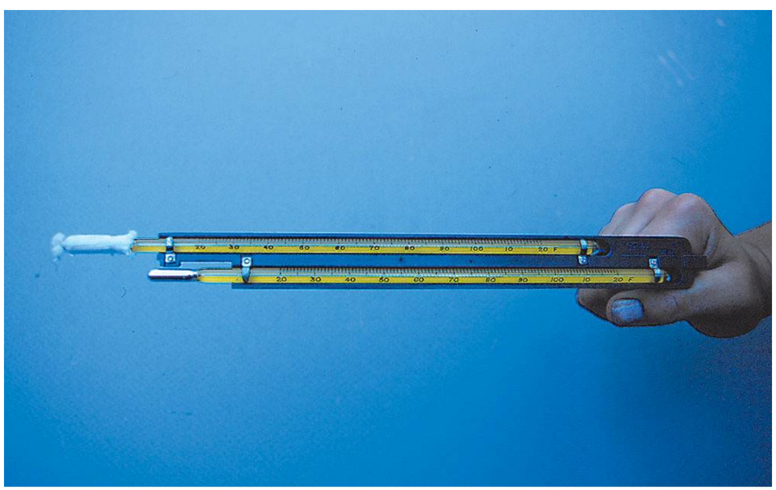
Adiabatic saturation temperature, T^*

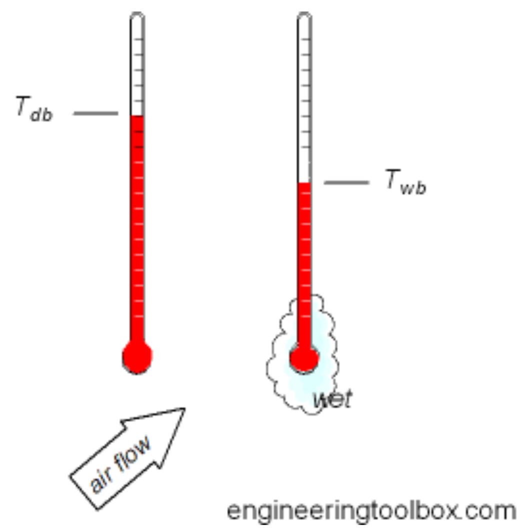
- The adiabatic saturation temperature, T^*, is the temperature at which evaporating water also at T^* will bring the air to saturation (ϕ =1)
  - T^* is an indirect measure of the current humidity ratio, W
  - T^* is rarely used alone, but we use it to get other temperatures



Wet-bulb temperature, T_{wb}

- Measuring T^* is not easy, but we can approximately measure T^* by measuring the air temp using a moving thermometer covered with a wet wick
  - We call this the wet-bulb temperature, T_{wb}
- T^* and T_{wb} differ by very little
  - So we usually use T_{wb} in place of T^*
- You can measure T and T_{wb} at the same time with a device called the Sling Psychrometer





Equations for T_{dew}

- Dew-point temperature, T_{dew}

Between dew points of 0 and 93°C,

$$t_d = C_{14} + C_{15}\alpha + C_{16}\alpha^2 + C_{17}\alpha^3 + C_{18}(p_w)^{0.1984} \quad (39)$$

Below 0°C,

$$t_d = 6.09 + 12.608\alpha + 0.4959\alpha^2 \quad (40)$$

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

Note:
These constants are only for SI units
IP units are different

Equations for T_{wb}

- Wet-bulb temperature, T_{wb}
- Requires iterative solver… find the T_{wb} that satisfies the following equation (above freezing):

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

- And below freezing:

$$W = \frac{(2830 - 0.24T_{wb})W_{s@T_{wb}} - 1.006(T - T_{wb})}{2830 + 1.86T - 2.1T_{wb}}$$

Obtaining these data from ASHRAE Tables

ASHRAE Ch. 1 (2013) Table 2 gives us W_s, v_{da} , and v_s directly at different temperatures:

Table 2 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressu

Temp., °C t	Humidity Ratio W_s, kg_w/kg_{da}	Specific Volume, m^3/kg_{da} v_{da}	v_{as}	v_s	Specific Enthalpy, kJ/kg_{da} h_{da}	h_{as}	h_s
15	0.010694	0.8159	0.0140	0.8299	15.087	27.028	42.115
16	0.011415	0.8188	0.0150	0.8338	16.093	28.873	44.966
17	0.012181	0.8216	0.0160	0.8377	17.099	30.830	47.929
18	0.012991	0.8245	0.0172	0.8416	18.105	32.906	51.011
19	0.013851	0.8273	0.0184	0.8457	19.111	35.107	54.219
20	0.014761	0.8301	0.0196	0.8498	20.117	37.441	57.558
21	0.015724	0.8330	0.0210	0.8540	21.124	39.914	61.037
22	0.016744	0.8358	0.0224	0.8583	22.130	42.533	64.663

Obtaining these data from ASHRAE Tables

ASHRAE Ch. 1 (2013) Table 3 gives us p_{ws}, v_{da} , and h_g directly at different temperatures:

Table 3 Thermodynamic Properties of Water at Saturation

Temp., °C t	Absolute Pressure p_{ws}, kPa	Specific Volume, m^3/kg_w			Specific Enthalpy, kJ/kg_w		
		Sat. Liquid v_i/v_f	Evap. v_{ig}/v_{fg}	Sat. Vapor v_g	Sat. Liquid h_i/h_f	Evap. h_{ig}/h_{fg}	Sat. Vapor h_g
3	0.7581	0.001000	168.013	168.014	12.60	2493.80	2506.40
4	0.8135	0.001000	157.120	157.121	16.81	2491.42	2508.24
5	0.8726	0.001000	147.016	147.017	21.02	2489.05	2510.07
6	0.9354	0.001000	137.637	137.638	25.22	2486.68	2511.91
7	1.0021	0.001000	128.927	128.928	29.43	2484.31	2513.74
8	1.0730	0.001000	120.833	120.834	33.63	2481.94	2515.57
9	1.1483	0.001000	113.308	113.309	37.82	2479.58	2517.40
10	1.2282	0.001000	106.308	106.309	42.02	2477.21	2519.23

Some examples

Moist air exists at 30°C dry-bulb temperature with a 15°C dew point temperature

Find the following:

(a) the humidity ratio, W

(b) degree of saturation, μ

(c) relative humidity, ϕ

(d) enthalpy, h

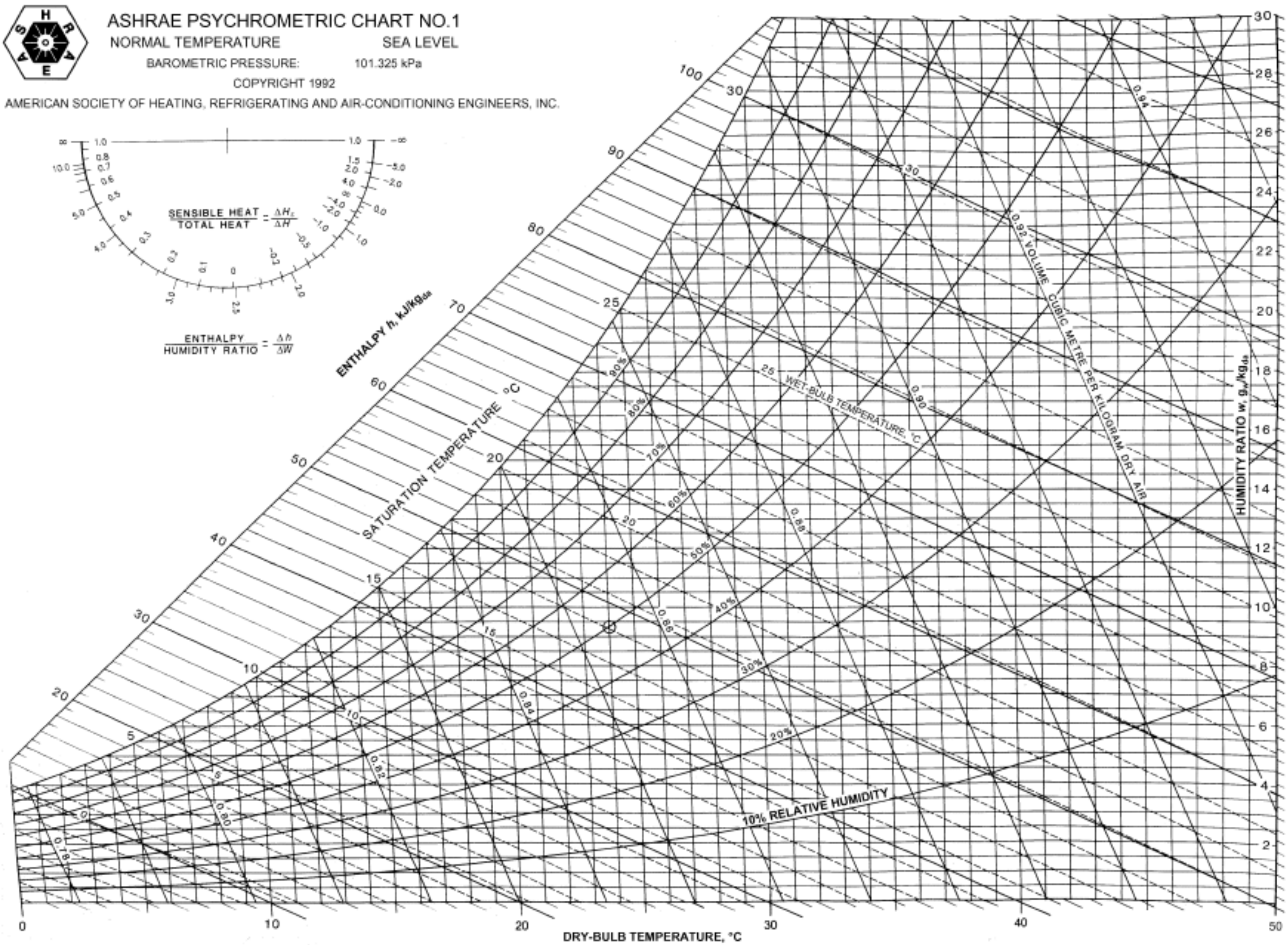
(e) specific volume, v

(f) density, ρ

(g) wet-bulb temperature, T_{wb}

The Psychrometric Chart

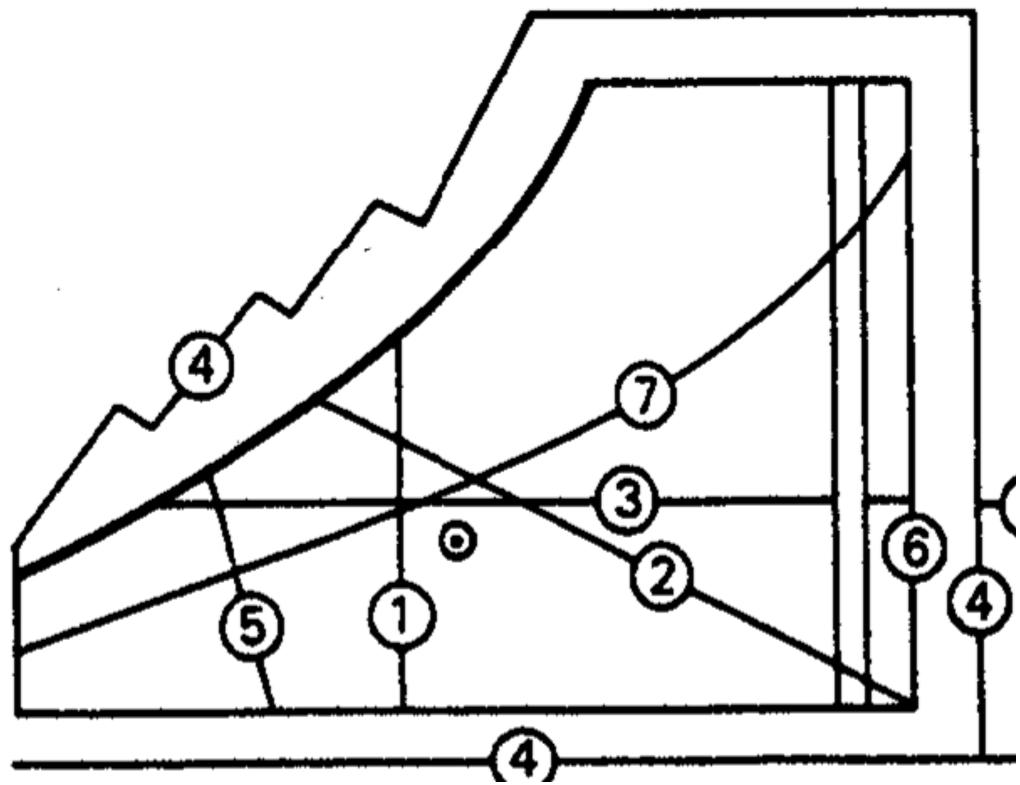
- Looking up properties and manipulating the equations quickly becomes tedious
  - Since we usually only need to have an accuracy of 5-10% for most building science applications, we can often rely on graphs instead of tables and equations
- The solution: the Psychrometric Chart
  - Plots dry bulb temperature on the x-axis and humidity ratio on the y-axis
    - Shows relationships between T and W and relative humidity, wet-bulb temperature, vapor pressure, specific volume
  - Charts are unique at each value of atmospheric pressure (p)
- Both SI and IP versions are on BB in the materials folder



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

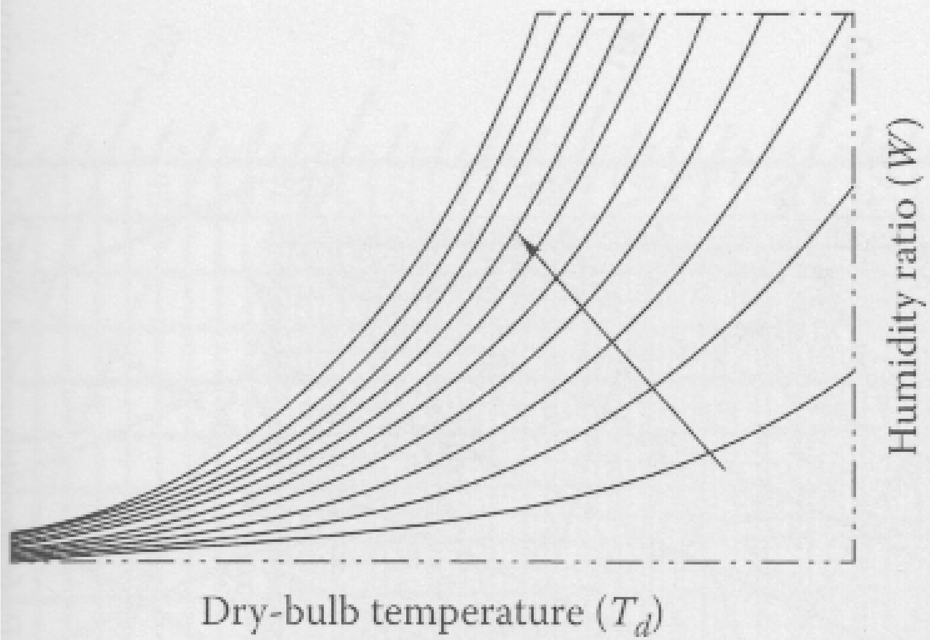
Deciphering the psychrometric chart

1. Dry Bulb Temp
2. Wet Bulb Temp
3. Humidity Ratio
4. Enthalpy
5. Specific Volume
6. Dew Point Temp
7. Relative Humidity

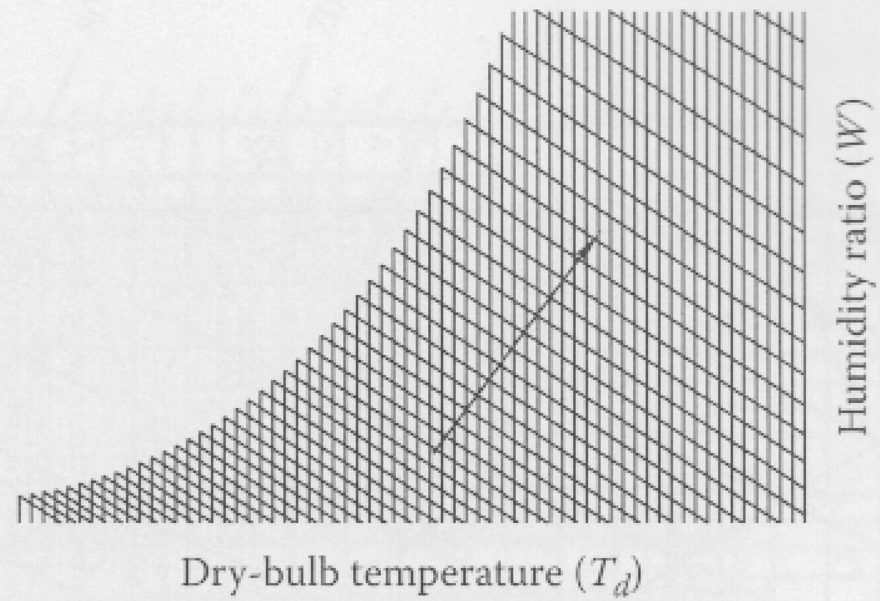


Deciphering the psychrometric chart

Lines of constant RH

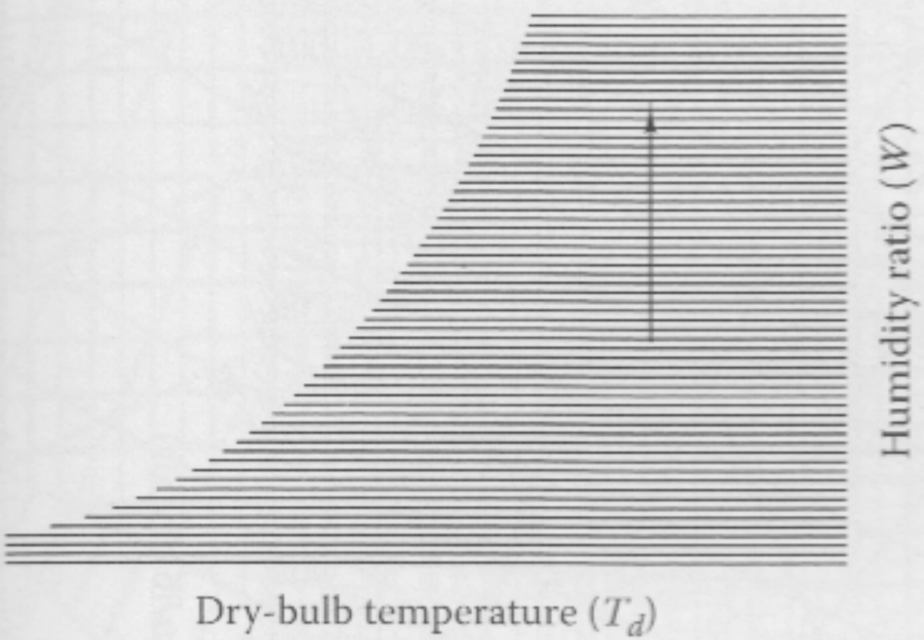


Lines of constant wet-bulb and dry-bulb

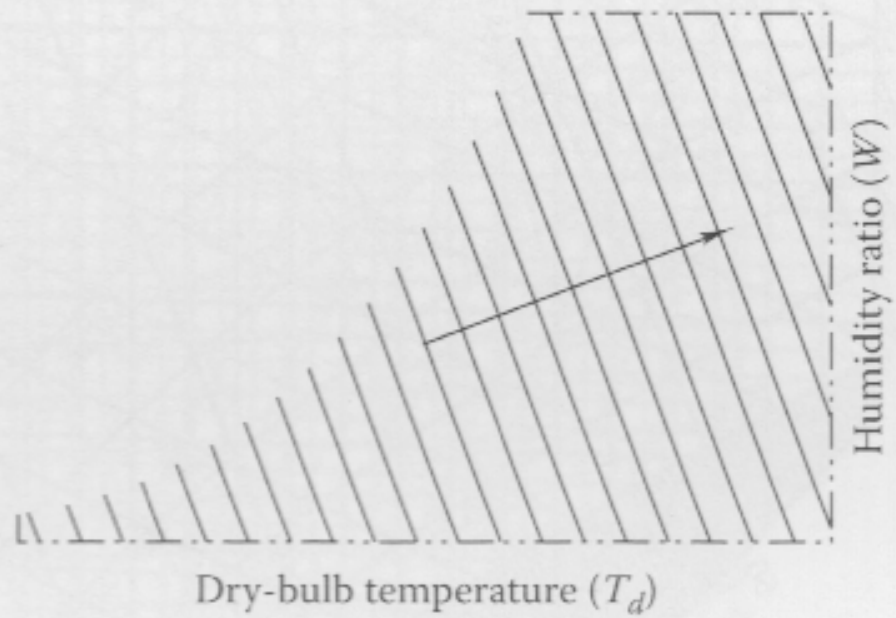


Deciphering the psychrometric chart

Lines of constant humidity ratio

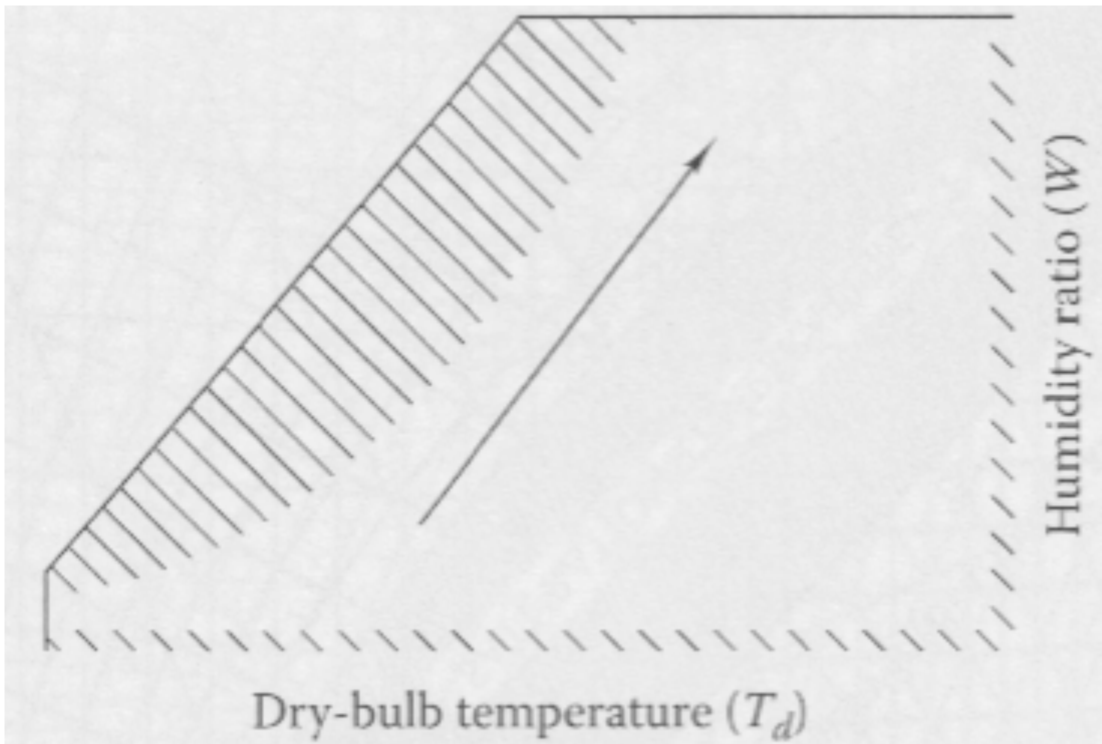


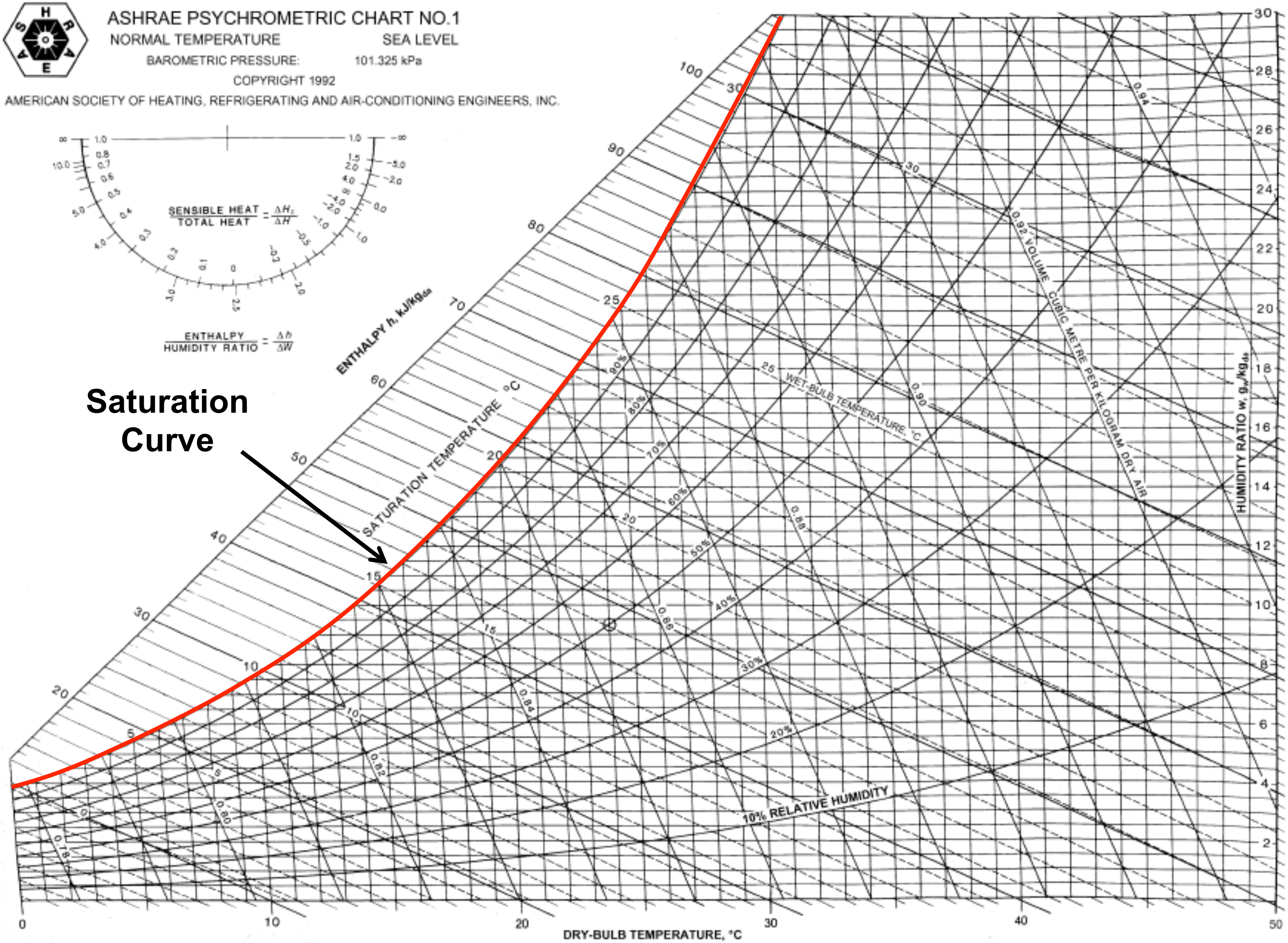
Lines of constant specific volume



Deciphering the psychrometric chart

Lines of constant enthalpy





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NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE:
101.325 kPa
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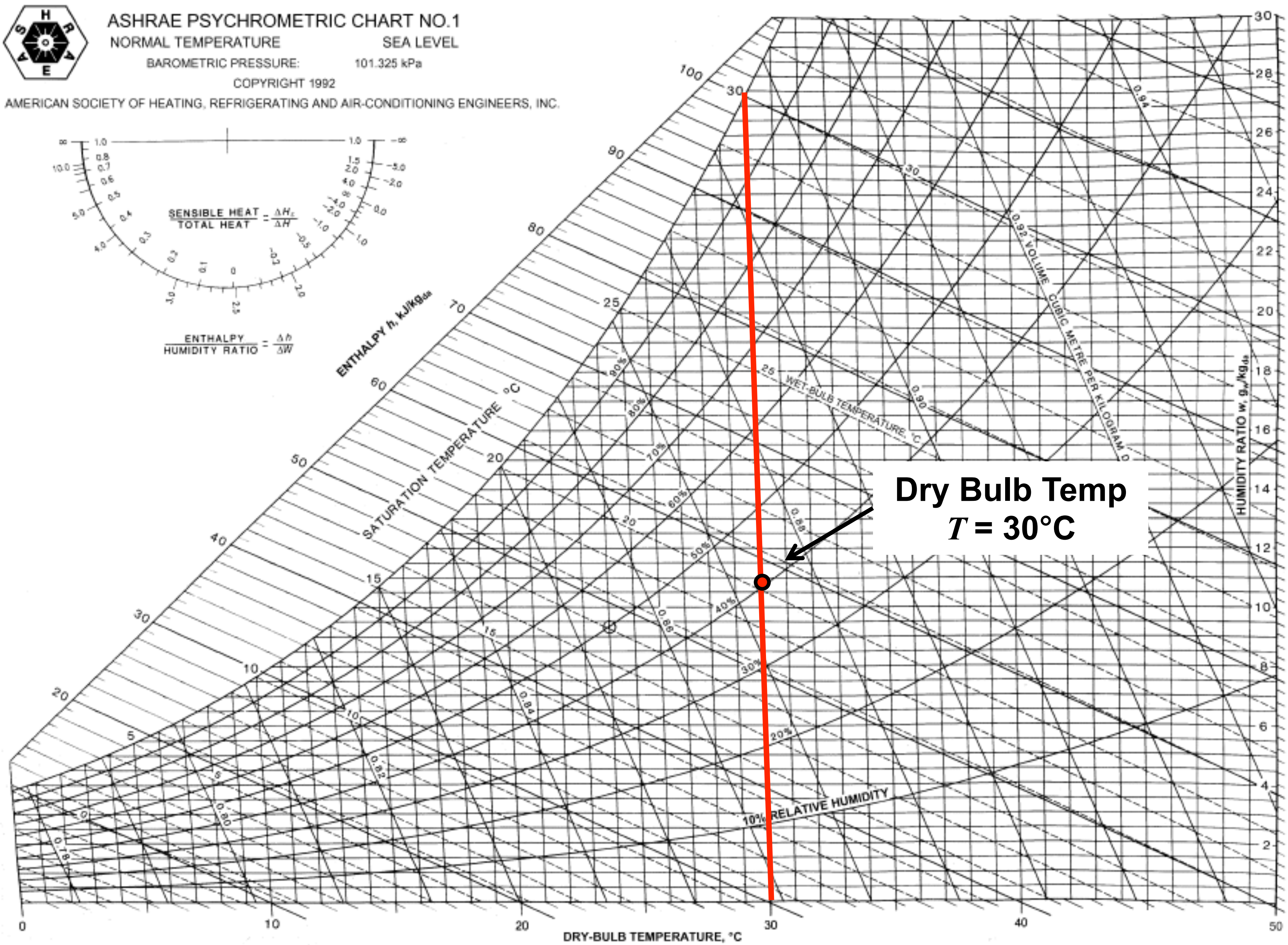
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔH
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kg$_{da}$
SATURATION TEMPERATURE °C
Saturation Curve
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, g$_w$/kg$_{da}$
DRY-BULB TEMPERATURE, °C



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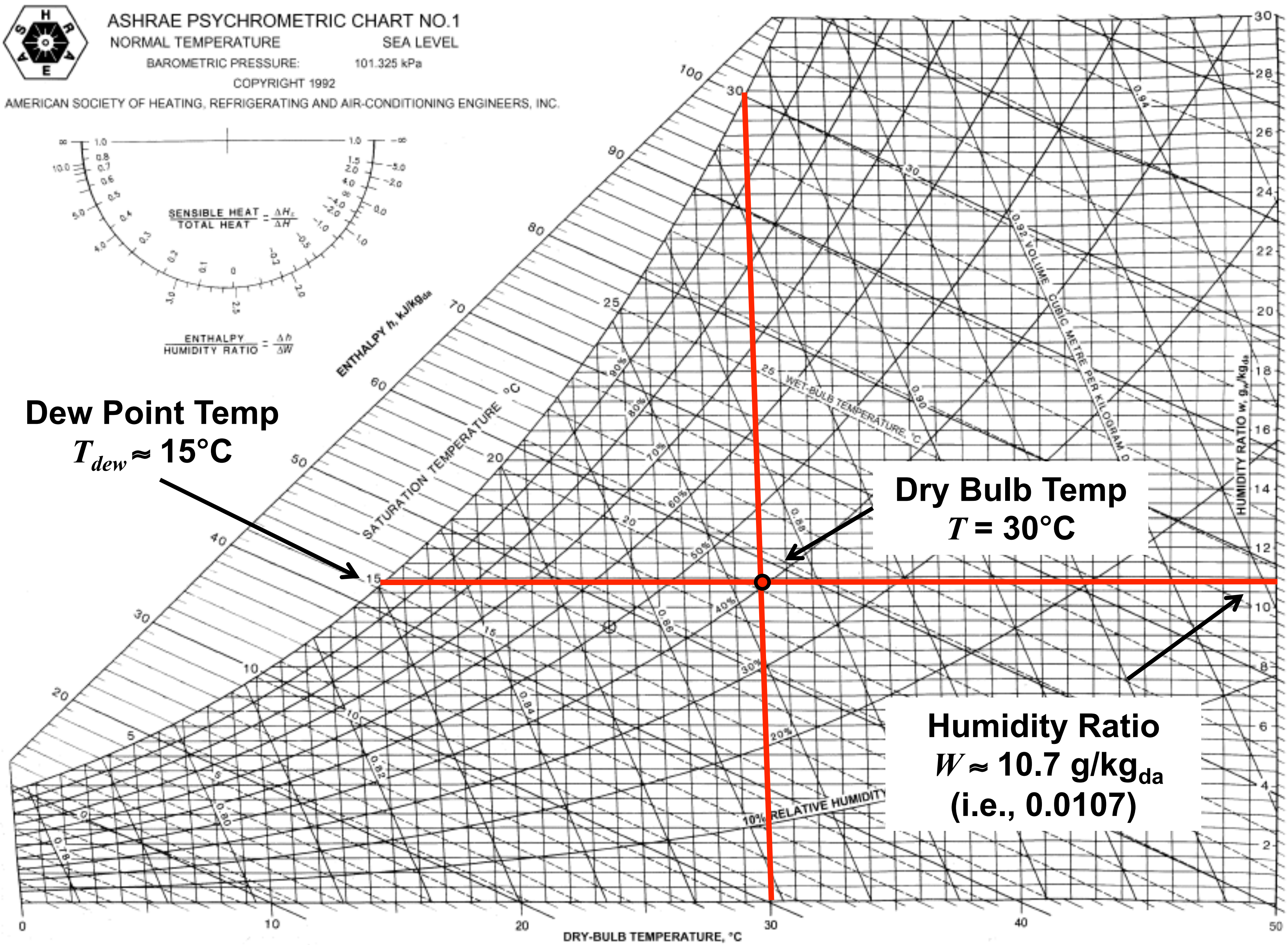
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kg da
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, gw/kg da
DRY-BULB TEMPERATURE, °C
Dry Bulb Temp
T = 30°C



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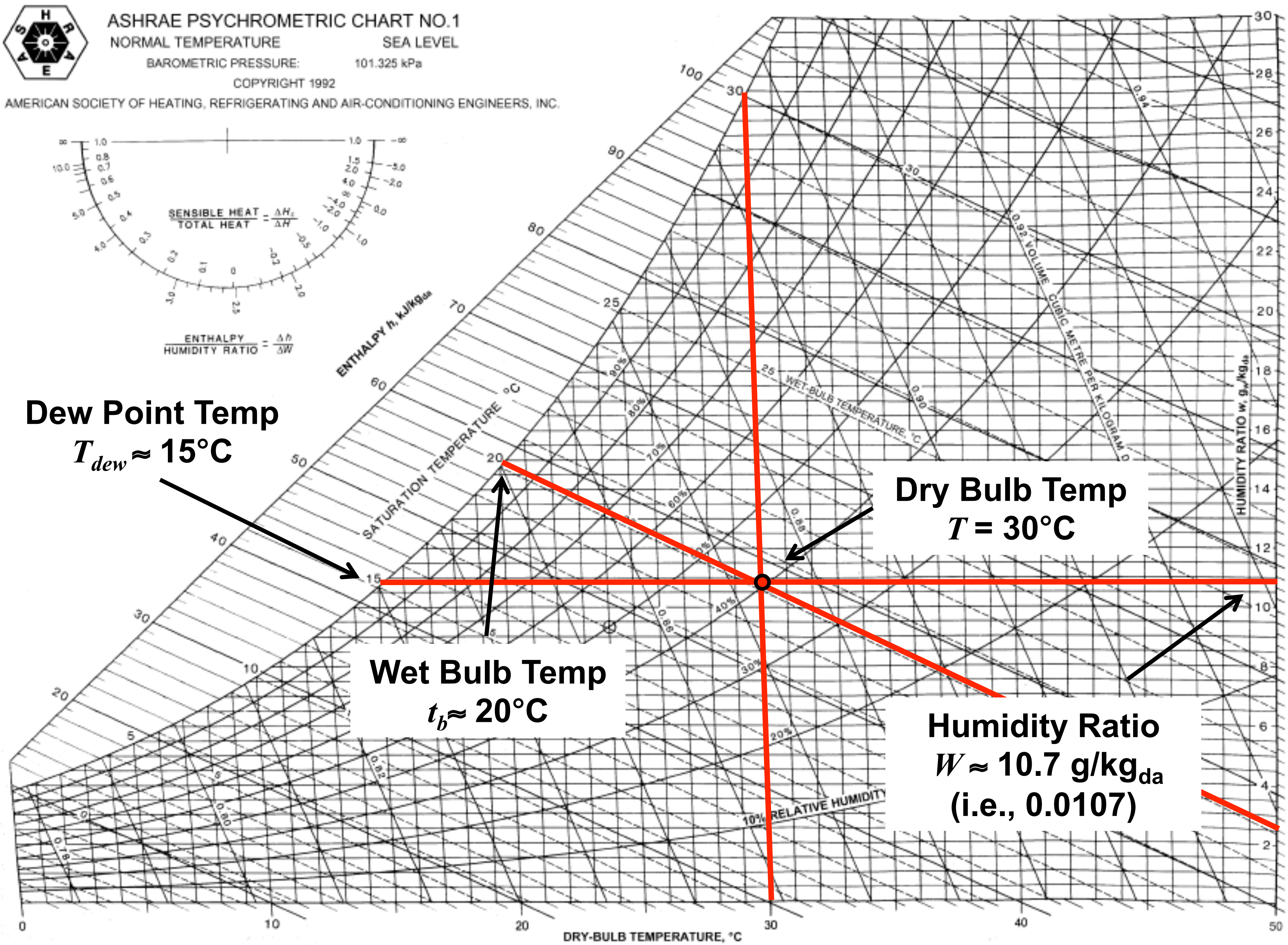
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kg_da
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM
HUMIDITY RATIO W, g_w/kg_da
10% RELATIVE HUMIDITY
DRY-BULB TEMPERATURE, °C
Dew Point Temp
$T_{dew} \approx 15°C$
Dry Bulb Temp
$T = 30°C$
Humidity Ratio
$W \approx 10.7\ g/kg_{da}$
(i.e., 0.0107)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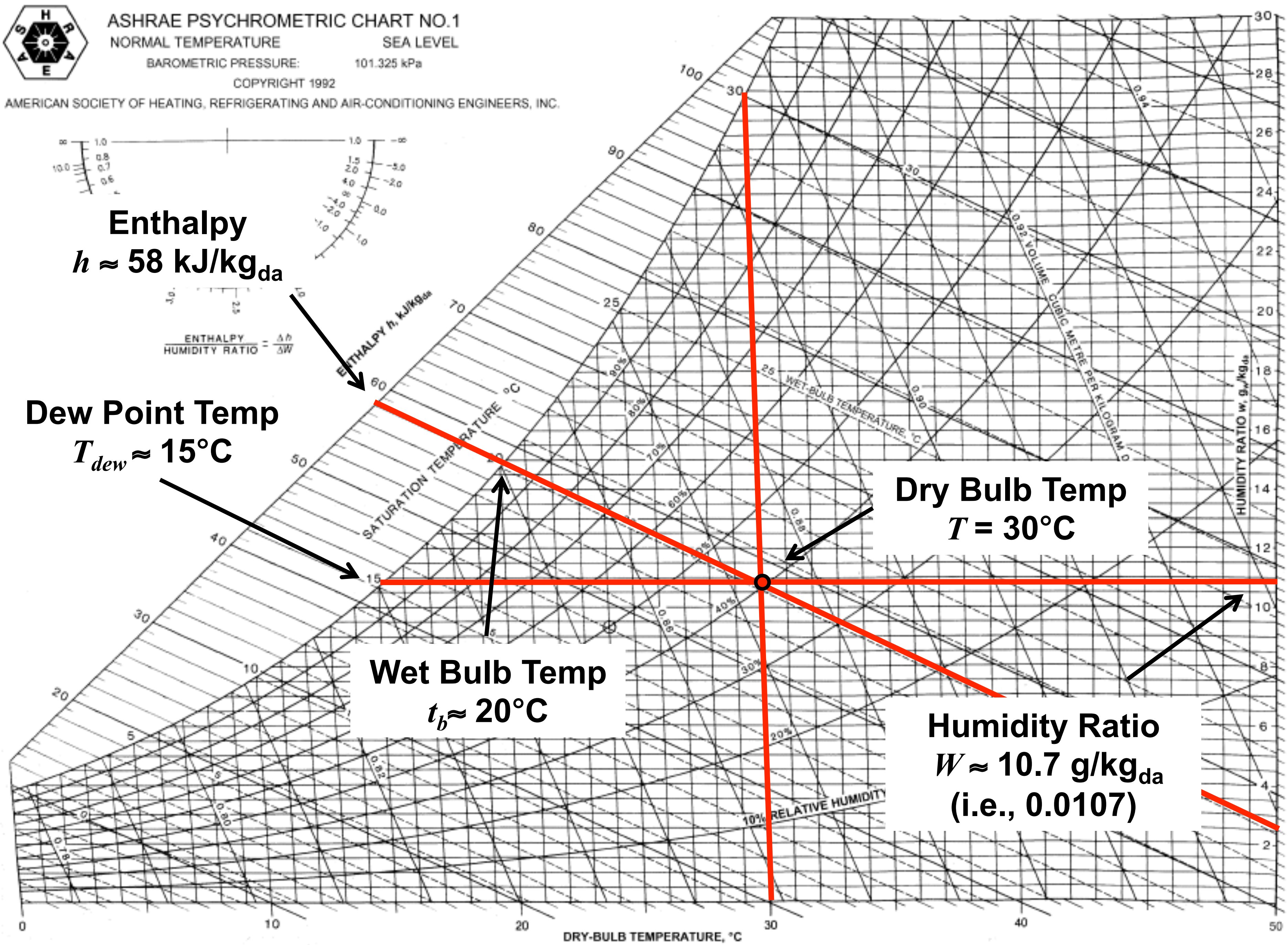
SENSIBLE HEAT / TOTAL HEAT = $\Delta H_s / \Delta H$
ENTHALPY / HUMIDITY RATIO = $\Delta h / \Delta W$
ENTHALPY h, kJ/kg$_{da}$
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, g$_w$/kg$_{da}$
DRY-BULB TEMPERATURE, °C
Dew Point Temp
$T_{dew} \approx$ 15°C
Wet Bulb Temp
$t_b \approx$ 20°C
Dry Bulb Temp
T = 30°C
Humidity Ratio
$W \approx$ 10.7 g/kg$_{da}$
(i.e., 0.0107)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SEA LEVEL
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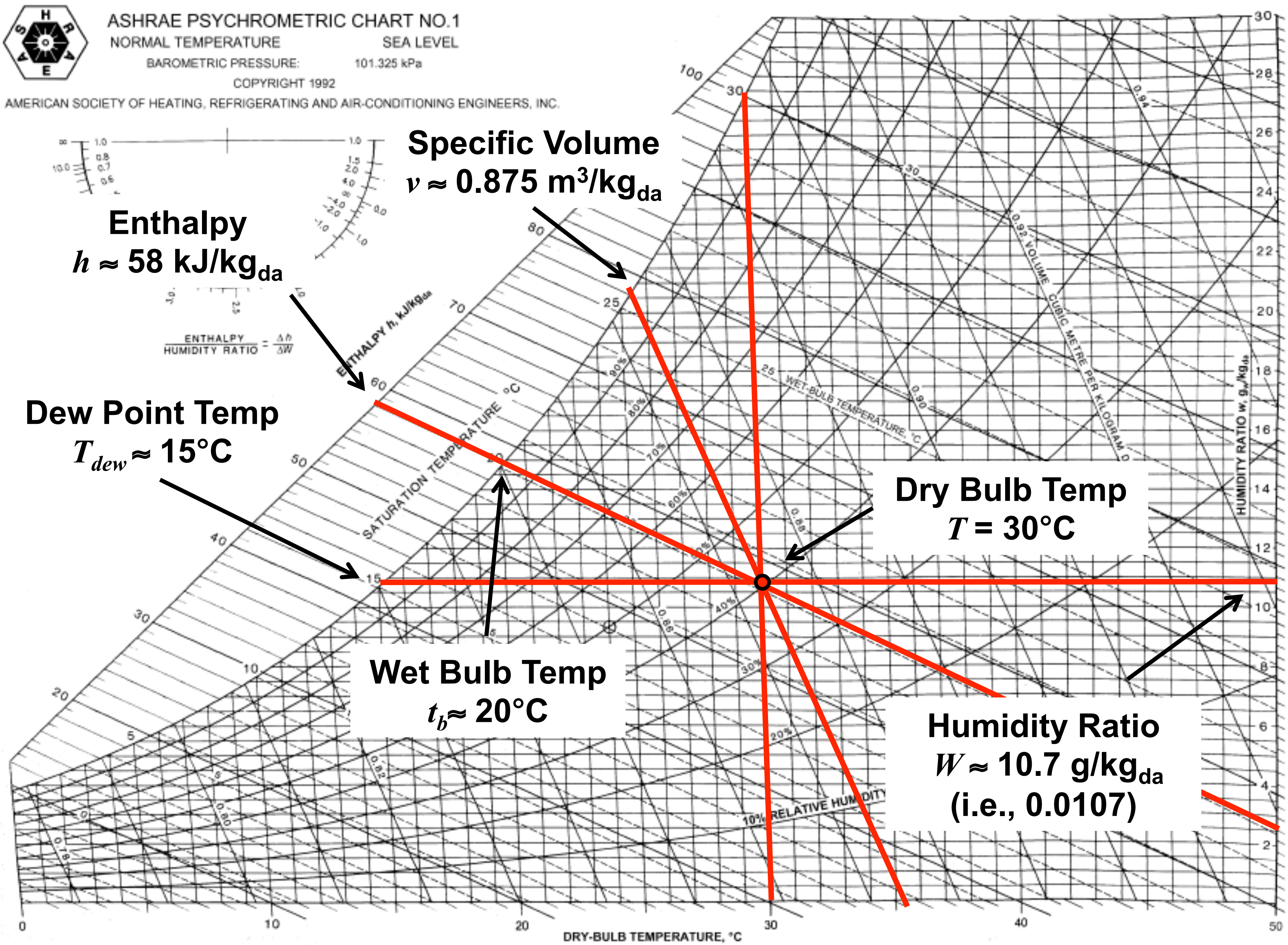
Enthalpy
$h \approx 58$ kJ/kg$_{da}$
Dew Point Temp
$T_{dew} \approx 15$°C
Wet Bulb Temp
$t_b \approx 20$°C
Dry Bulb Temp
$T = 30$°C
Humidity Ratio
$W \approx 10.7$ g/kg$_{da}$
(i.e., 0.0107)
ENTHALPY / HUMIDITY RATIO = $\Delta h / \Delta W$
ENTHALPY h, kJ/kg$_{da}$
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
0.92 VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, g$_w$/kg$_{da}$
DRY-BULB TEMPERATURE, °C



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SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
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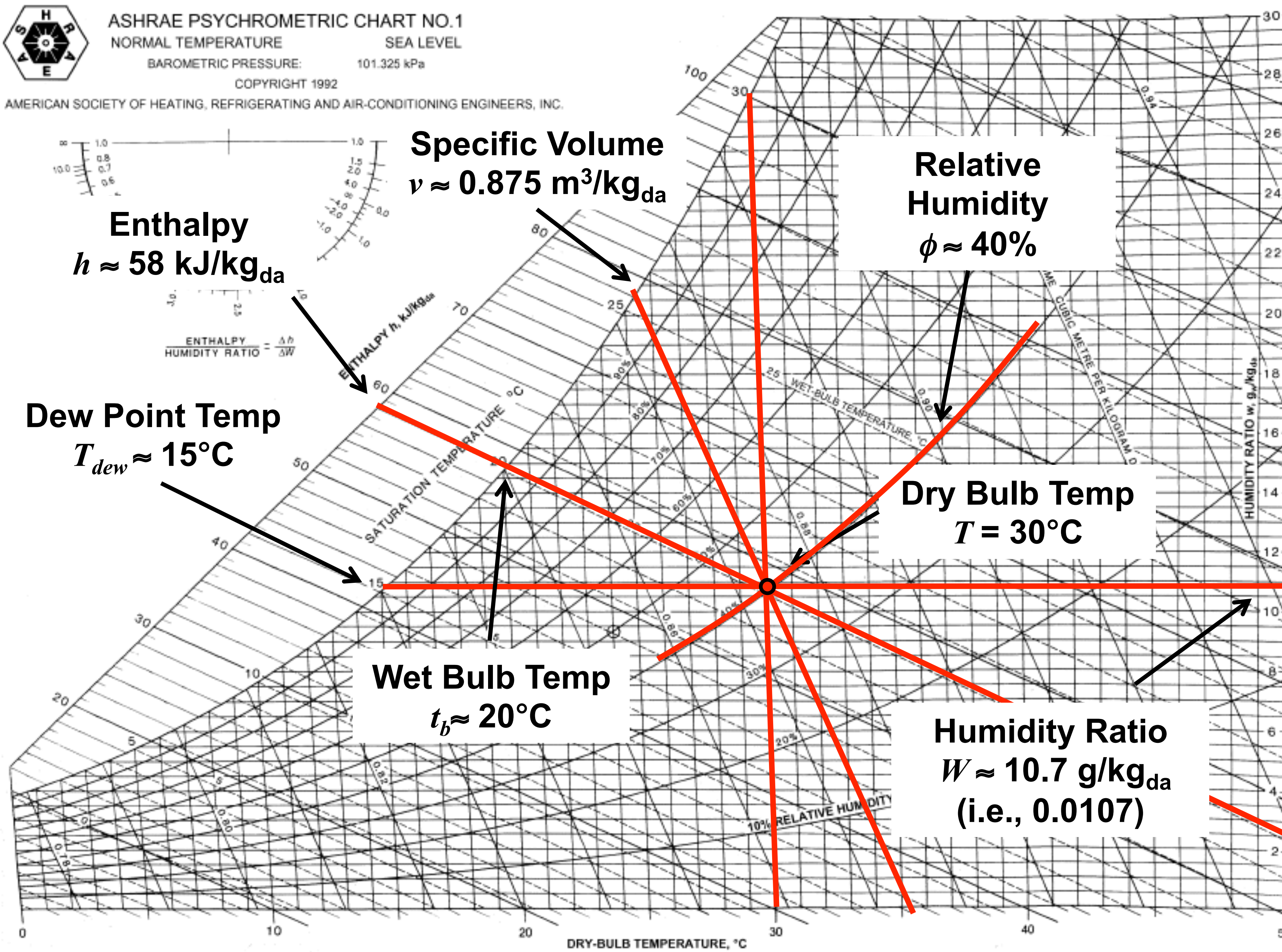
Specific Volume
$v \approx 0.875$ m^3/kg$_{da}$
Enthalpy
$h \approx 58$ kJ/kg$_{da}$
Dew Point Temp
$T_{dew} \approx 15$°C
Dry Bulb Temp
$T = 30$°C
Wet Bulb Temp
$t_b \approx 20$°C
Humidity Ratio
$W \approx 10.7$ g/kg$_{da}$
(i.e., 0.0107)
ENTHALPY h, kJ/kg$_{da}$
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE, °C
0.92 VOLUME CUBIC METRE PER KILOGRAM D
HUMIDITY RATIO w, g$_w$/kg$_{da}$
10% RELATIVE HUMIDITY
DRY-BULB TEMPERATURE, °C
ENTHALPY / HUMIDITY RATIO = Δh / ΔW



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SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
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Specific Volume
v ≈ 0.875 m³/kg_da
Relative Humidity
φ ≈ 40%
Enthalpy
h ≈ 58 kJ/kg_da
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
Dew Point Temp
T_dew ≈ 15°C
Dry Bulb Temp
T = 30°C
Wet Bulb Temp
t_b ≈ 20°C
Humidity Ratio
W ≈ 10.7 g/kg_da
(i.e., 0.0107)
ENTHALPY h, kJ/kg_da
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE °C
CUBIC METRE PER KILOGRAM
HUMIDITY RATIO w, g_w/kg_da
10% RELATIVE HUMIDITY
DRY-BULB TEMPERATURE, °C

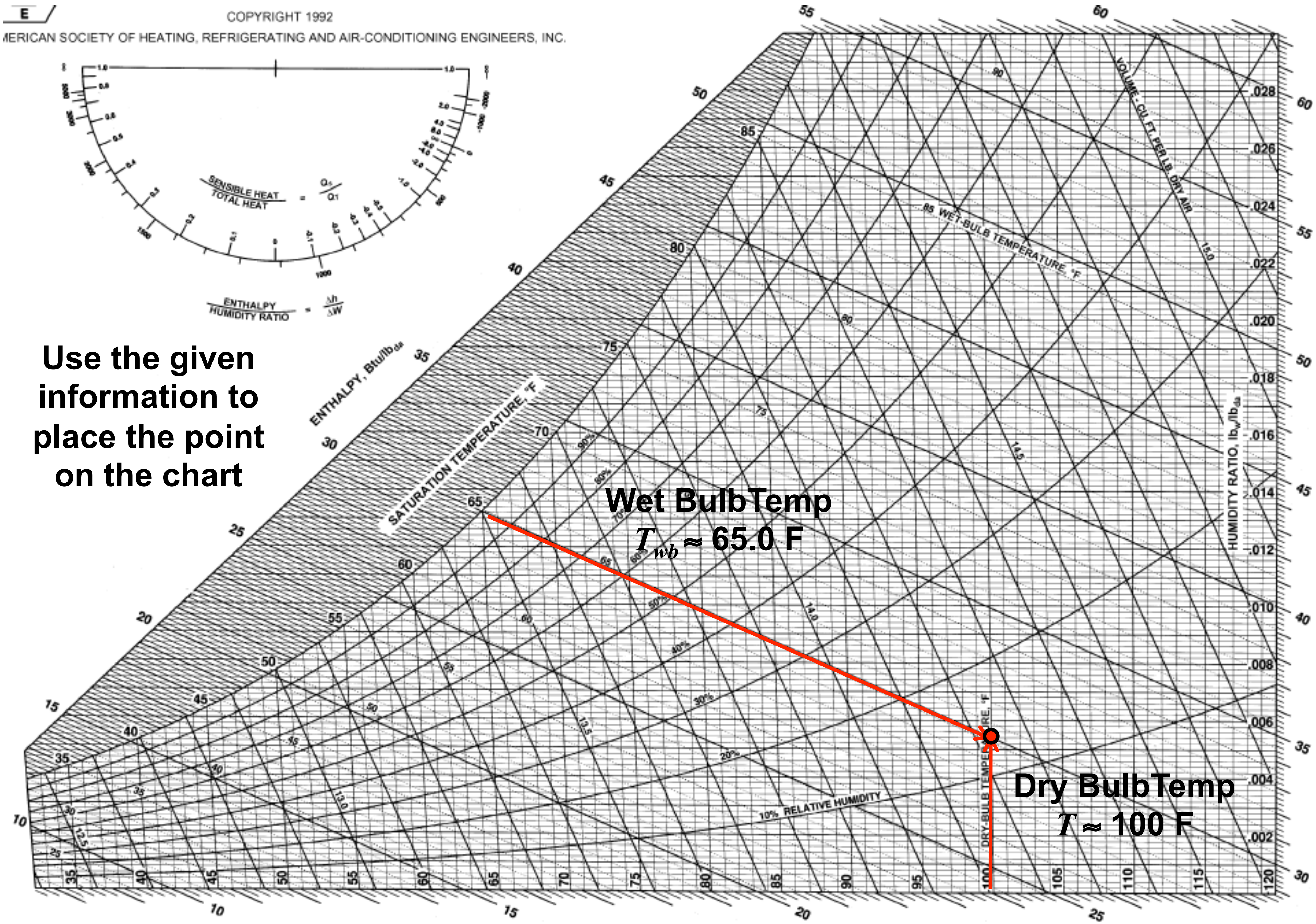
IP example

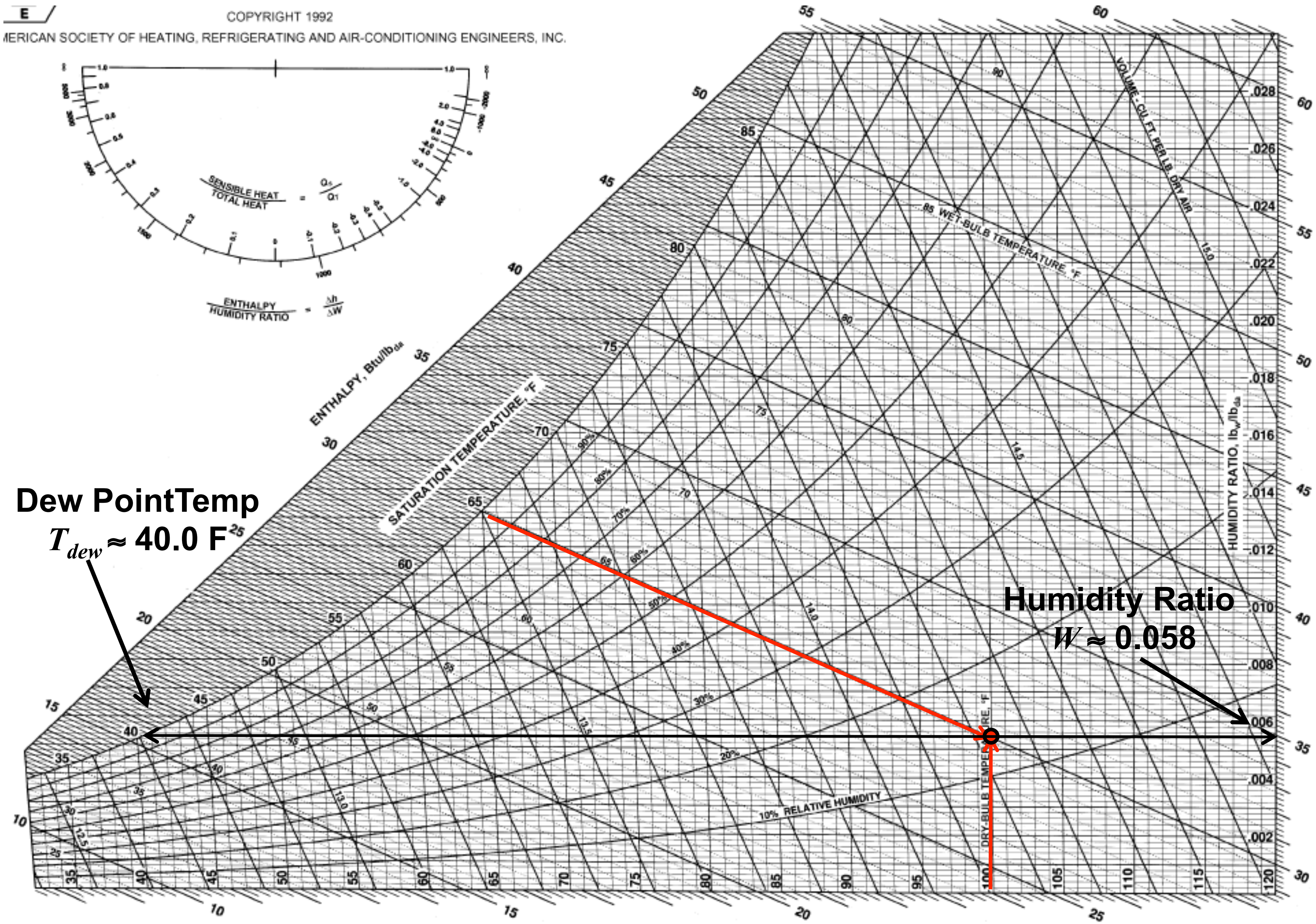
- Moist air exists at 100°F dry bulb, 65°F wet bulb and 14.696 psia

  Find:

  a) Humidity ratio
  b) Enthalpy
  c) Dew-point temperature
  d) Relative humidity
  e) Specific volume

Use the given information to place the point on the chart





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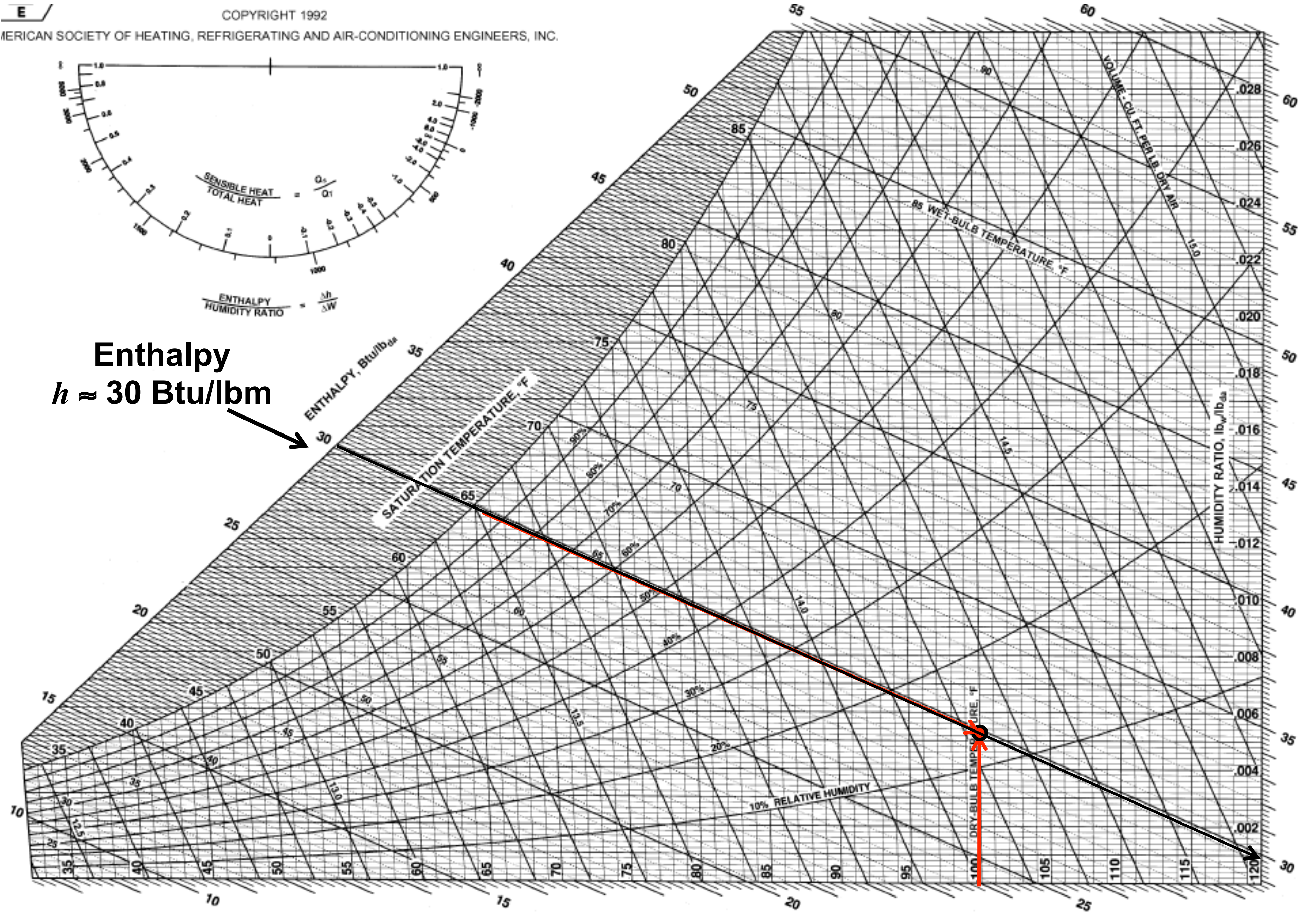
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY, Btu/lb da
SATURATION TEMPERATURE, °F
WET-BULB TEMPERATURE, °F
VOLUME - CU. FT. PER LB. DRY AIR
HUMIDITY RATIO, lbw/lbda
10% RELATIVE HUMIDITY
DRY BULB TEMPERATURE, °F
Dew PointTemp
T_{dew} ≈ 40.0 F
Humidity Ratio
W ≈ 0.058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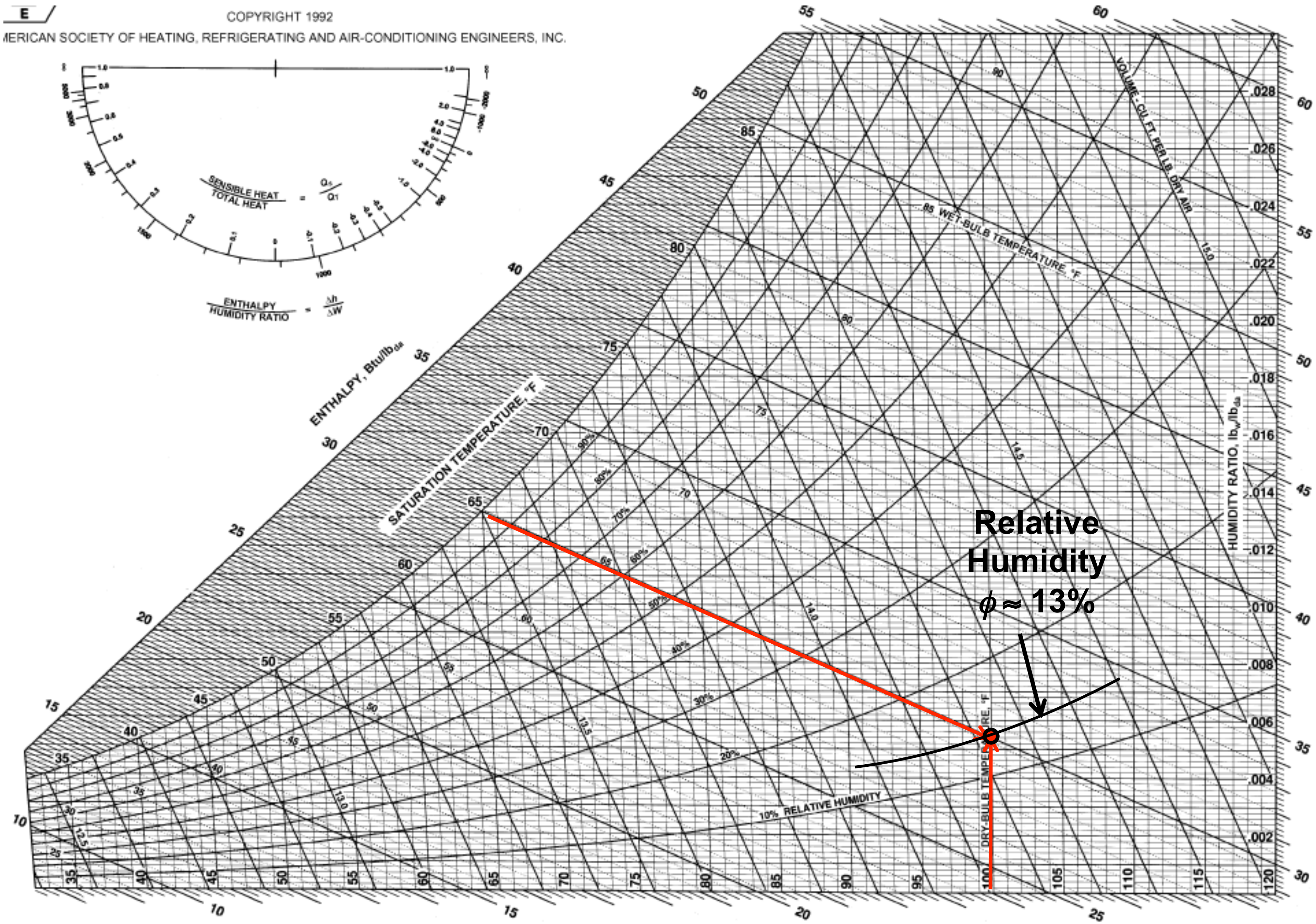
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
Enthalpy
h ≈ 30 Btu/lbm
ENTHALPY, Btu/lbda
SATURATION TEMPERATURE, °F
85 WET-BULB TEMPERATURE, °F
VOLUME - CU.FT. PER LB. DRY AIR
HUMIDITY RATIO, lbw/lbda
10% RELATIVE HUMIDITY
DRY-BULB TEMPERATURE, °F



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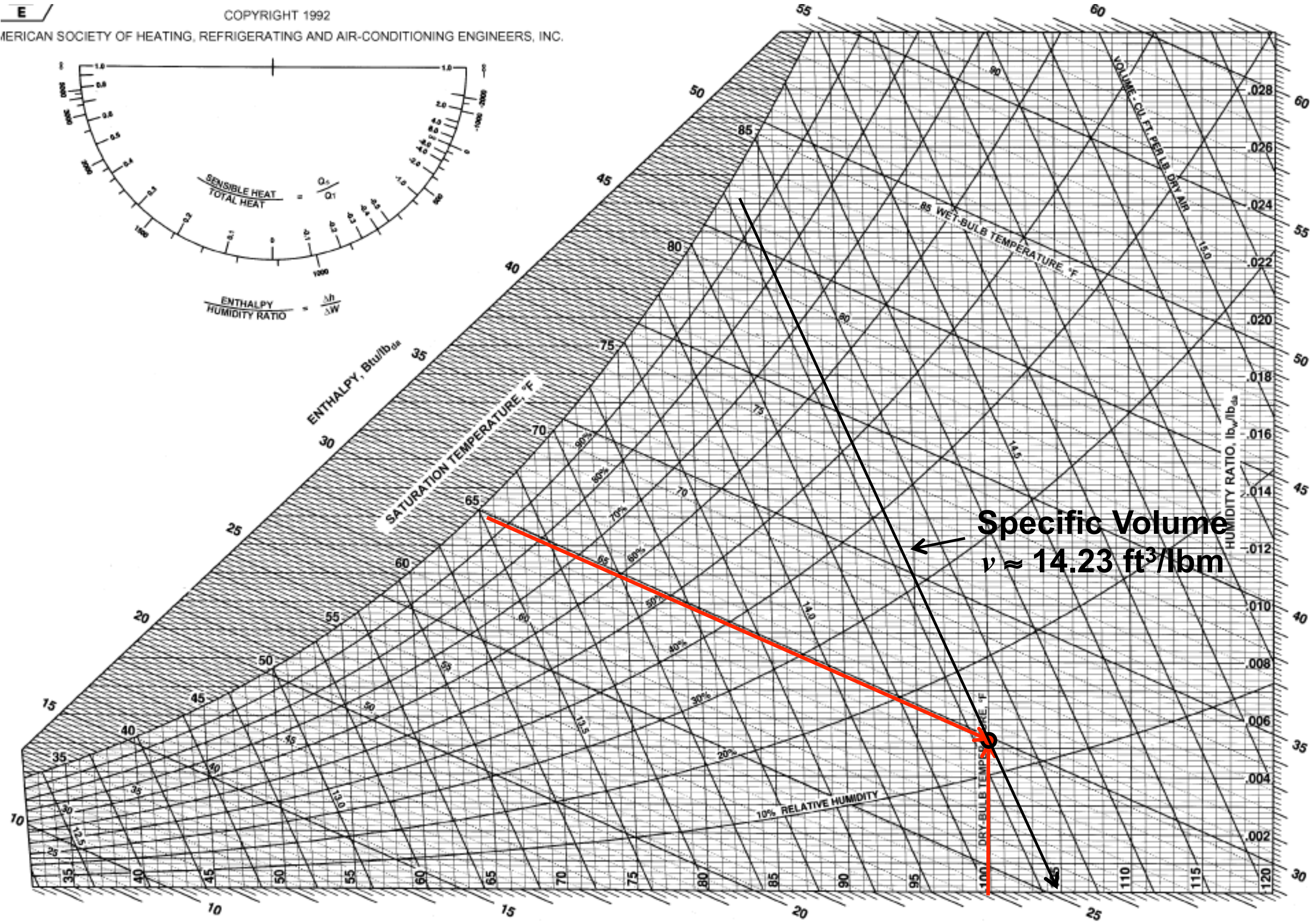
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY, Btu/lb_da
SATURATION TEMPERATURE, °F
85 WET-BULB TEMPERATURE, °F
VOLUME-CU.FT. PER LB. DRY AIR
HUMIDITY RATIO, lb_w/lb_da
DRY-BULB TEMPERATURE, °F
10% RELATIVE HUMIDITY
Relative Humidity
φ ≈ 13%



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SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY, Btu/lb da
SATURATION TEMPERATURE, °F
85 WET-BULB TEMPERATURE, °F
VOLUME - CU.FT. PER LB. DRY AIR
HUMIDITY RATIO, lbw/lbda
10% RELATIVE HUMIDITY
DRY-BULB TEMPERATURE, °F
Specific Volume
v ≈ 14.23 ft³/lbm

Psychrometric software

- Psych and Psychpro
  - Very popular psych chart and analysis software
  - I think at least one of these is in the AM 217 lab
- There are a bunch of online calculators as well
  - http://www.sugartech.co.za/psychro/
- And smart phone apps too
- You can also make your own (i.e., in Excel)
  - You will have a HW problem where you have to do this

Use of the psychrometric chart for *processes*

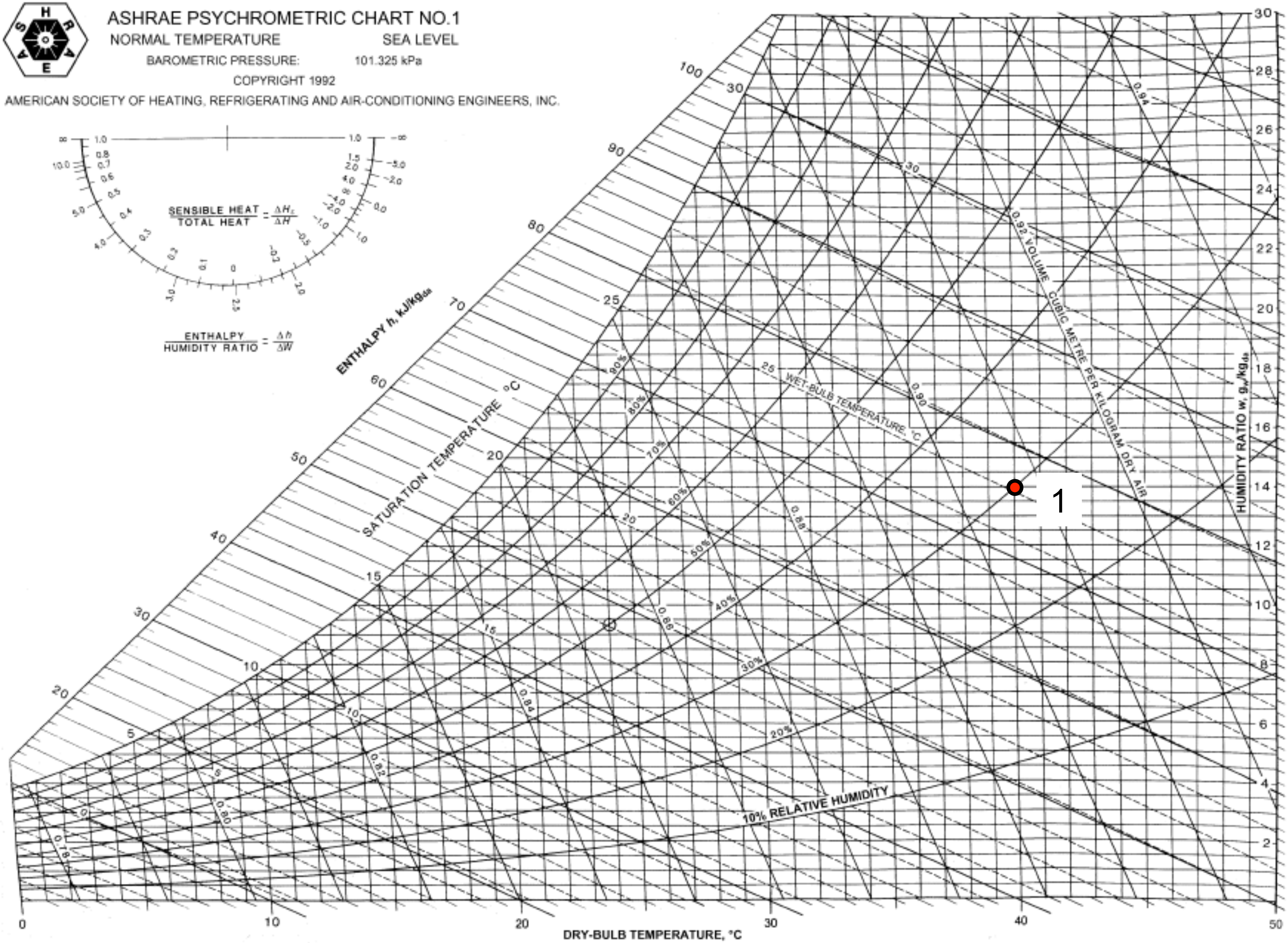
- We can use the psychrometric chart not only to describe states of moist air, but for a number of processes that are important for building science

Examples:

- Sensible cooling
- Sensible + latent cooling
- Adiabatic saturation
- Warming and humidification of cold, dry air
- Cooling and dehumidification of warm, humid air
- Evaporative cooling
- Mixing of airstreams

Example: Sensible cooling

- Moist air is cooled from 40°C and 30% RH to 30°C
- Does the moisture condense?
- What are values of RH at W at the process end point?



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NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
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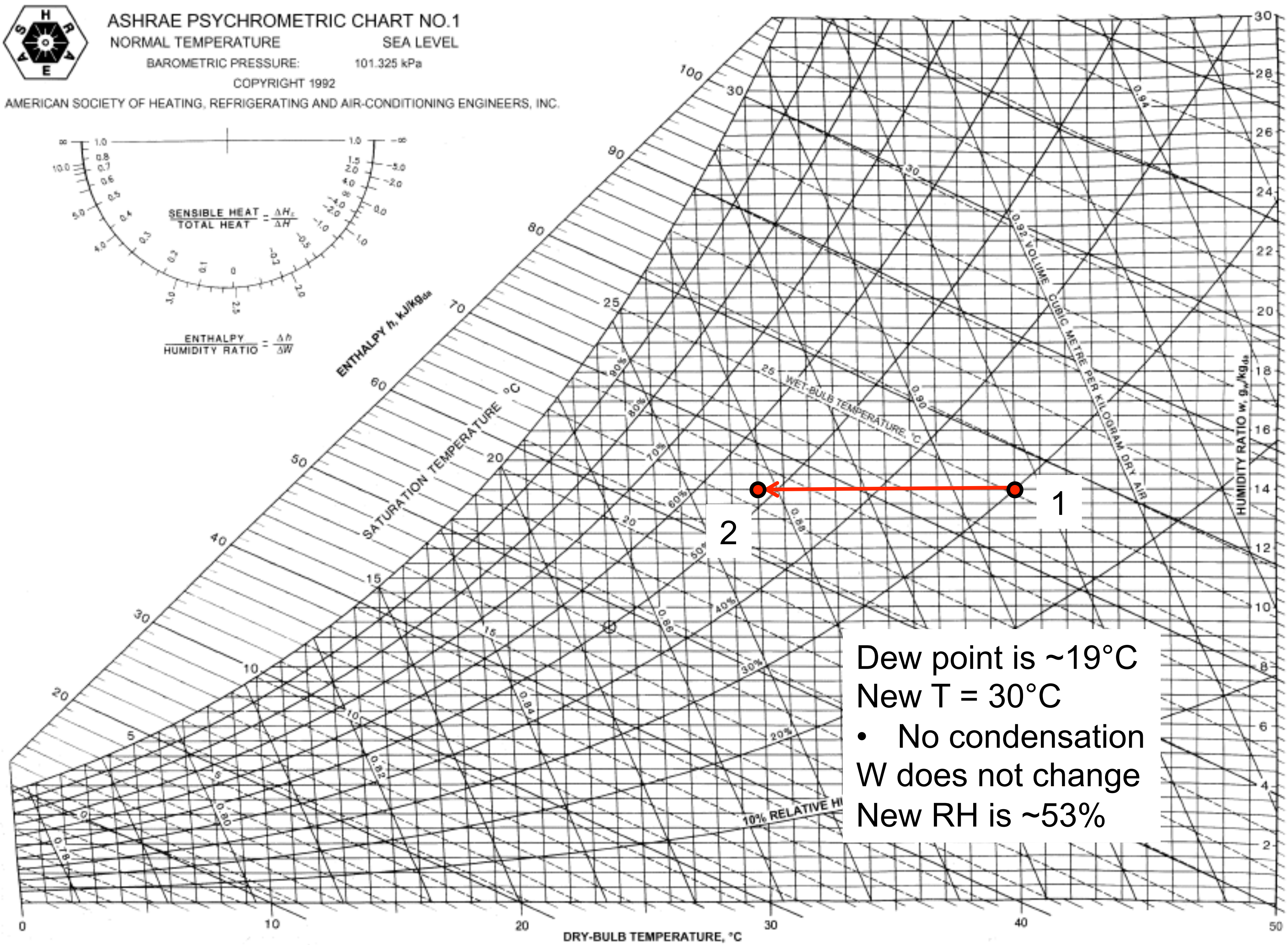
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔH
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kgda
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, gw/kgda
DRY-BULB TEMPERATURE, °C
1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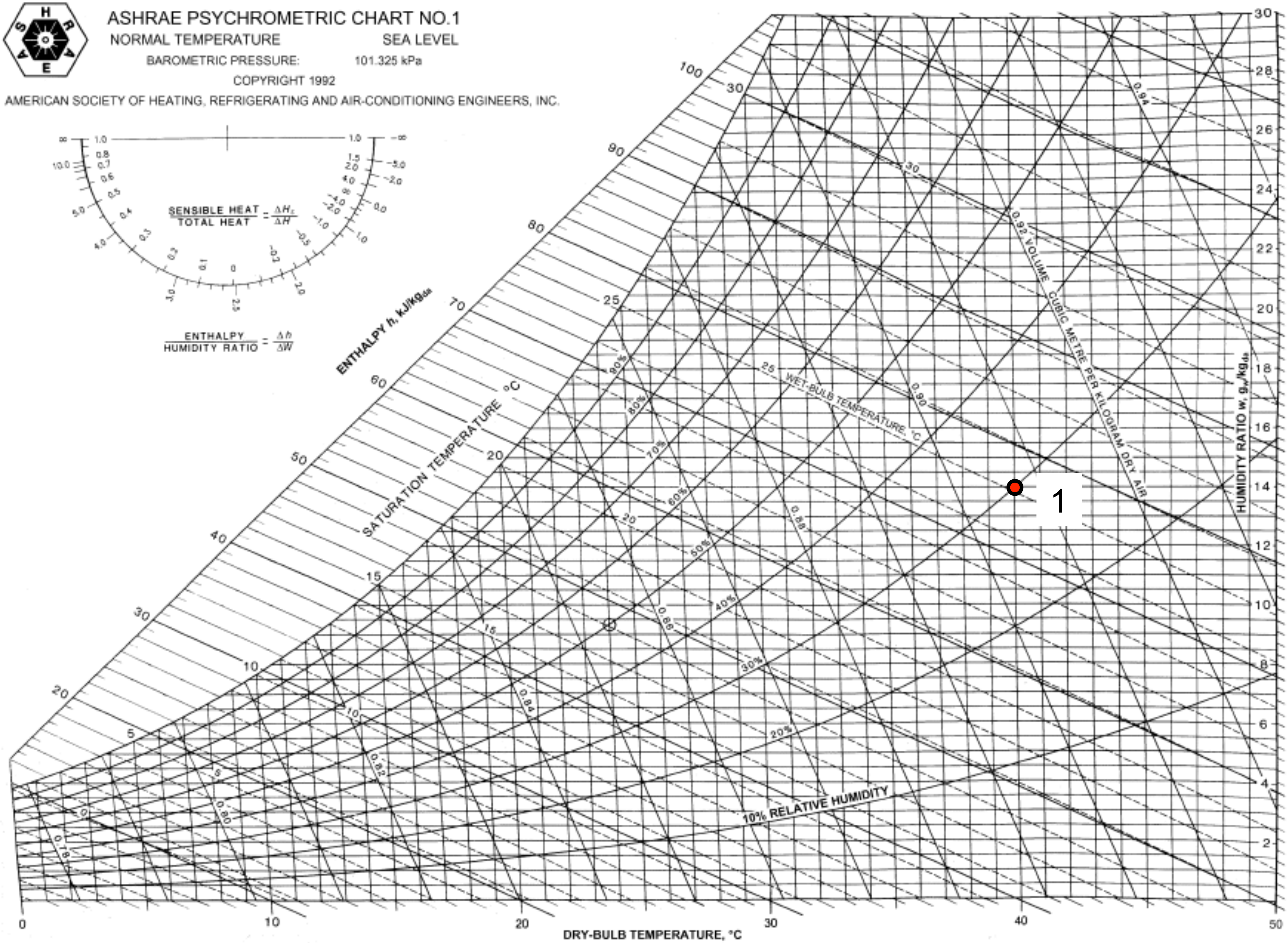
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY h, kJ/kg_da
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
0.92 VOLUME CUBIC METRE PER KILOGRAM DRY AIR
HUMIDITY RATIO w, g_w/kg_da
10% RELATIVE HI
DRY-BULB TEMPERATURE, °C
1
2
Dew point is ~19°C
New T = 30°C
• No condensation
W does not change
New RH is ~53%

Example: Sensible + latent cooling

- Moist air is cooled from 40°C and 30% RH to 15°C
- Does the moisture condense?
- What are values of RH at W at the process end point?



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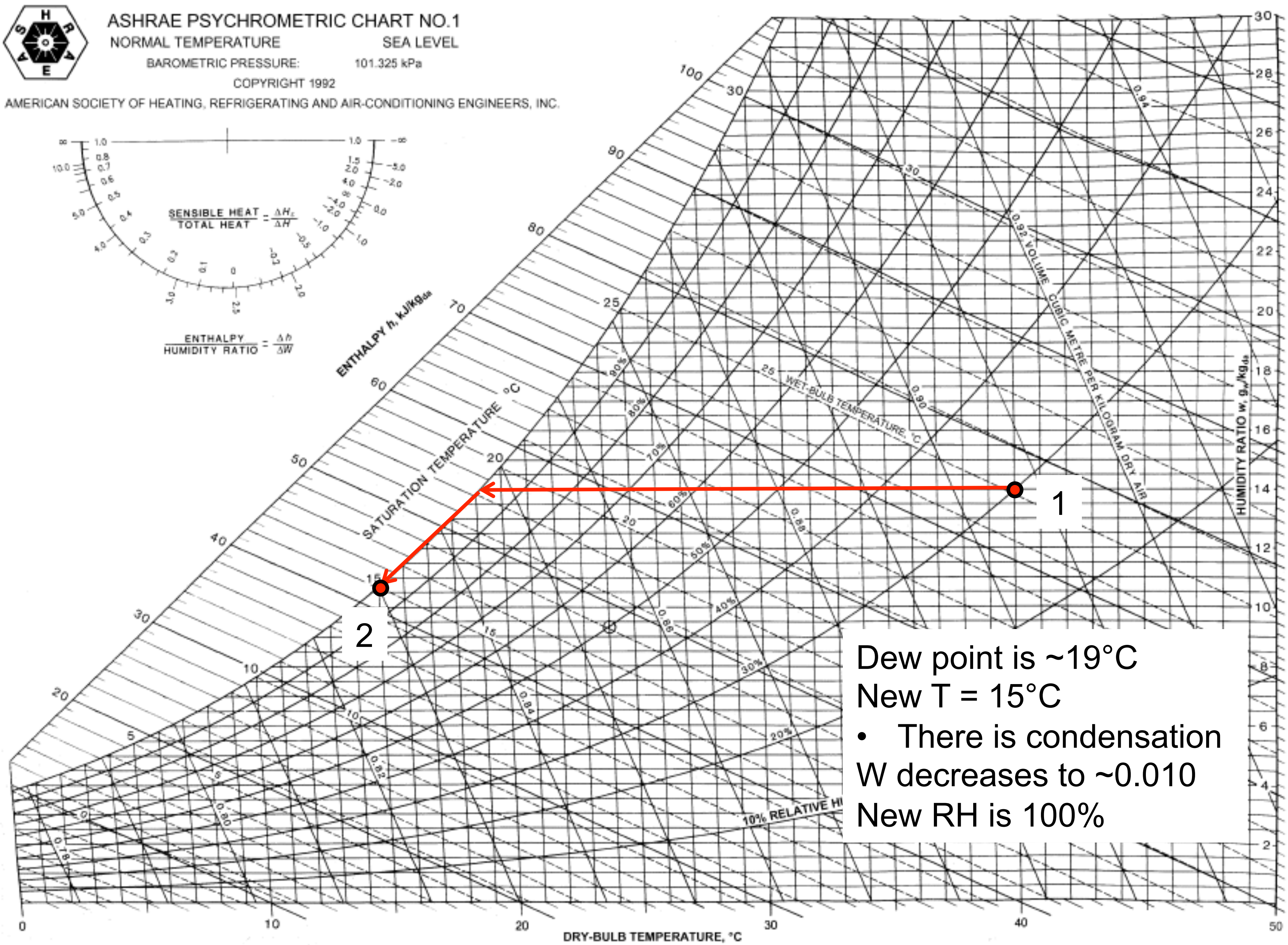
ENTHALPY h, kJ/kgda
SATURATION TEMPERATURE °C
WET-BULB TEMPERATURE, °C
VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO w, gw/kgda
DRY-BULB TEMPERATURE, °C
1



Dew point is ~19°C

New T = 15°C

- There is condensation

W decreases to ~0.010

New RH is 100%

Psychrometrics and human thermal comfort

- We will do more process-related examples next lecture
- First we will focus on human thermal comfort
  - And how it relates to these moist air properties

Human thermal comfort

- Our ultimate desire in designing a building and its HVAC system is to provide a suitably comfortable environment for the occupants
- One important consideration is thermal comfort
- In general, thermal comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized
- Something else we want to be able to do is quantify the amount of discomfort that a space might present to people and what fraction of occupants are dissatisfied with a space

Thermal balance of body and effective temperature

- The heat produced by the body's metabolism dissipates to the environment
  - Otherwise we would overheat

- Roughly, 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

Assessing thermal comfort

- To develop equations and guidelines for thermal comfort, we have to have some idea of what perceive to be comfortable
- Comfort analysis is usually done through surveys of users in real spaces and through controlled human experiments and a questionnaire that rates comfort on a seven point scale
- The result of the questionnaire is the Mean Vote (MV)
  - If we predict the results of a questionnaire through equations we generate a predicted mean vote (PMV)

Predicted Mean Vote (PMV)

- The PMV is an estimate of the mean value that would be obtained if a large number of people were asked to vote on thermal comfort using a 7 point scale:

-3	-2	-1	0	+1	+2	+3
cold	cool	slightly cool	neutral	slightly warm	warm	hot

The environment is considered acceptable when: $-0.5 < PMV < 0.5$

Percent People Dissatisfied (PPD)

- Once we have the PMV (which are average results) we need to estimate how many people are satisfied with the thermal conditions for that PMV
  - We quantify that as the percent of people dissatisfied (PPD)

- Our design goal usually is to achieve a PPD<10%

- After lots of testing, researchers have found that PPD is a fairly nonlinear function of PMV

Percent People Dissatisfied (PPD)

A plot of the PPD Function is shown below:

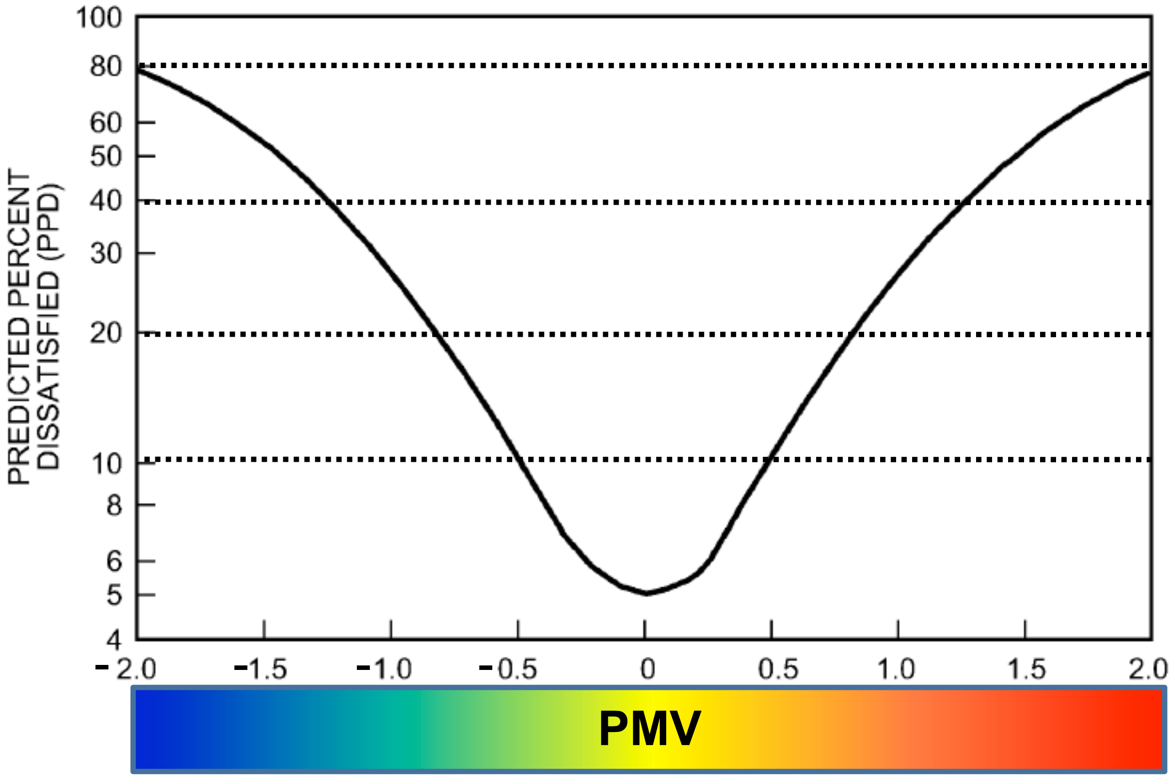
Since we want

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



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)

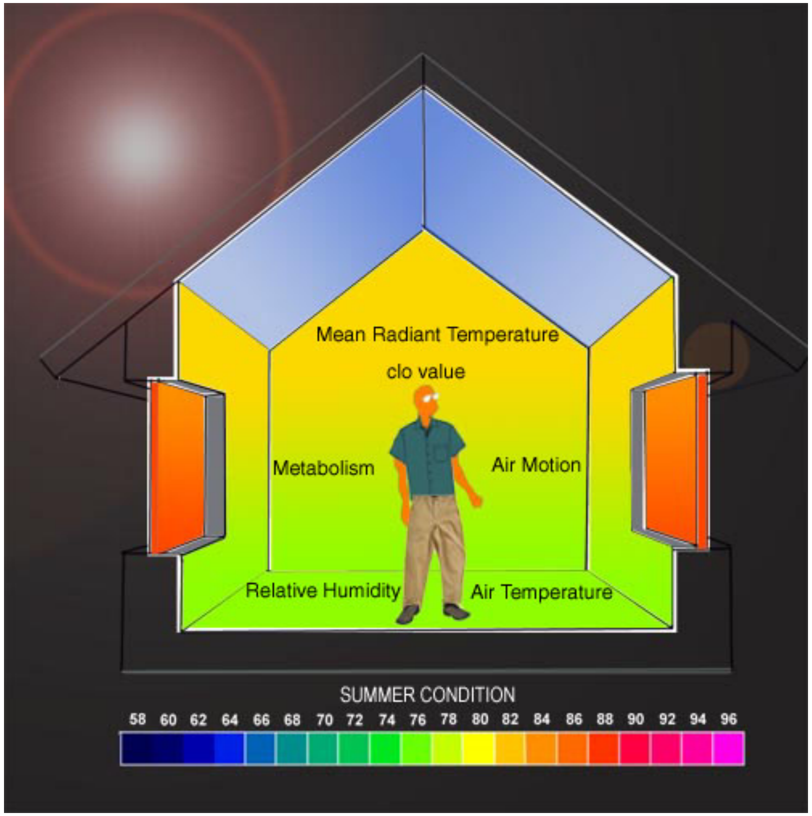
Local Air Speed (v)

Some are probably not:

Metabolic Rate (M)

Clothing Insulation (I_{cl})

Mean Radiant Temp. (T_r)



Metabolic energy production

- The total energy production rate of the human body is the sum of the production rates of heat (Q) and work (W):

$$\dot{Q} + \dot{W} = MA_{skin}$$

where
M = rate of metabolic energy production per surface area of skin (W/m^2)
A_{skin} = total surface area of skin (m^2)
(work, W, is typically neglected)

$$1 \text{ met} = 18.4 \frac{\text{Btu}}{\text{h} \cdot \text{ft}^2} = 58 \frac{\text{W}}{\text{m}^2}$$

Metabolic Rates for Typical Tasks

Activity	Met Units	Metabolic Rate W/m²	(Btu/h·ft²)
Resting			
Sleeping	0.7	40	(13)
Reclining	0.8	45	(15)
Seated, quiet	1.0	60	(18)
Standing, relaxed	1.2	70	(22)
Walking (on level surface)			
0.9 m/s, 3.2 km/h, 2.0 mph	2.0	115	(37)
1.2 m/s, 4.3 km/h, 2.7 mph	2.6	150	(48)
1.8 m/s, 6.8 km/h, 4.2 mph	3.8	220	(70)
Office Activities			
Seated, reading, or writing	1.0	60	(18)
Typing	1.1	65	(20)
Filing, seated	1.2	70	(22)
Filing, standing	1.4	80	(26)
Walking about	1.7	100	(31)
Lifting/packing	2.1	120	(39)
Driving/Flying			
Automobile	1.0-2.0	60-115	(18-37)
Aircraft, routine	1.2	70	(22)
Aircraft, instrument landing	1.8	105	(33)
Aircraft, combat	2.4	140	(44)
Heavy vehicle	3.2	185	(59)

Metabolic rates (continued)

Activity	Met Units	Metabolic Rate W/m^2	(Btu/h·ft^2)
Miscellaneous Occupational Activities			
Cooking	1.6-2.0	95-115	(29-37)
House cleaning	2.0-3.4	115-200	(37-63)
Seated, heavy limb movement	2.2	130	(41)
Machine work			
sawing (table saw)	1.8	105	(33)
light (electrical industry)	2.0-2.4	115-140	(37-44)
heavy	4.0	235	(74)
Handling 50 kg (100 lb) bags	4.0	235	(74)
Pick and shovel work	4.0-4.8	235-280	(74-88)
Miscellaneous Leisure Activities			
Dancing, social	2.4-4.4	140-255	(44-81)
Calisthenics/exercise	3.0-4.0	175-235	(55-74)
Tennis, single	3.6-4.0	210-270	(66-74)
Basketball	5.0-7.6	290-440	(92-140)
Wrestling, competitive	7.0-8.7	410-505	(129-160)

What about A_{skin}?

- For an adult, the area of our skin is typically on the order of 16-22 ft^2 (1.5 to 2 m^2)

- So for a typical adult doing typical indoor activities, their heat production rate will be:

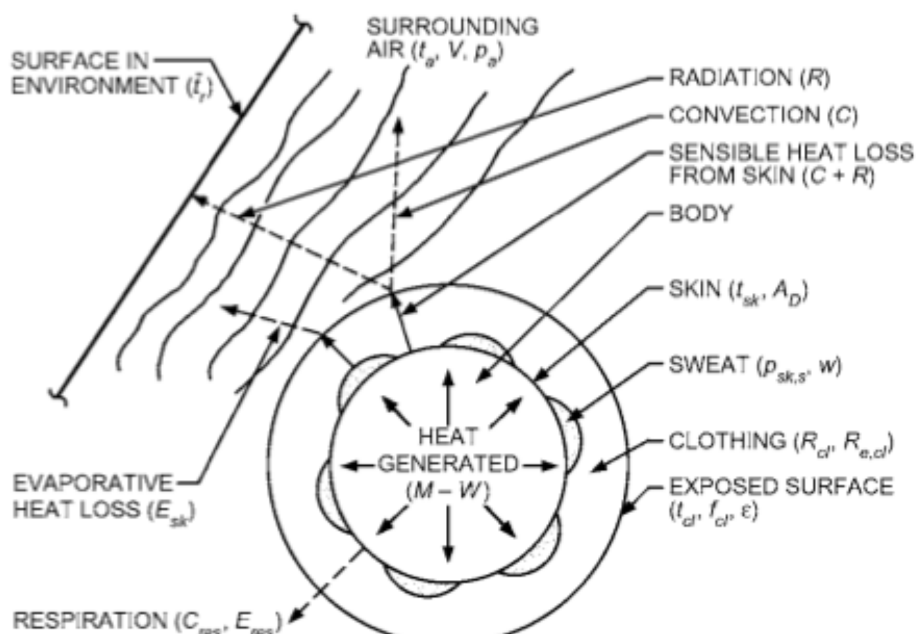
$$\dot{Q} + \dot{W} = MA_{skin} \approx (1\ \mathrm{met})(1.5 - 2\ \mathrm{m}^2)$$

$$\approx (58.2\ \frac{\mathrm{W}}{\mathrm{m}^2})(1.5 - 2\ \mathrm{m}^2) \approx 100\ \mathrm{W}$$

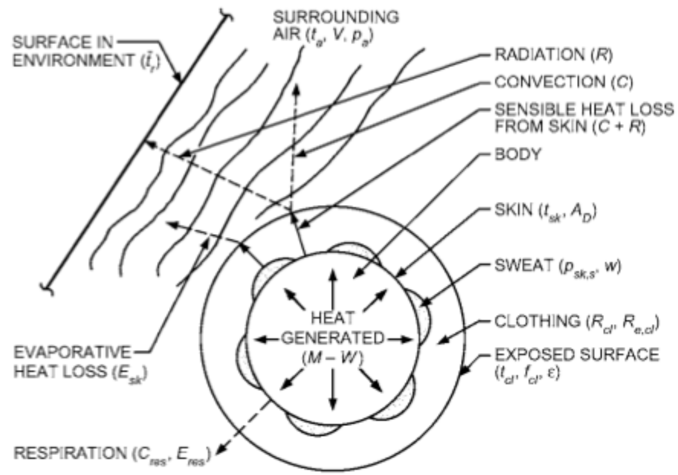
Body energy balance in a space

- Our internal body temperatures are consistent around 36-37°C
- We can set our heat production rate equal to the instantaneous heat flow to the environment (no storage):

$$\dot{Q} = MA_{skin} = \dot{Q}_{conv} + \dot{Q}_{rad} + \dot{Q}_{evap} + \dot{Q}_{resp,sens} + \dot{Q}_{resp,latent}$$
$$q = M = q_{conv} + q_{rad} + q_{evap} + q_{resp,sens} + q_{resp,latent}$$



Body energy balance in a space



$$M - W = q_{sk} + q_{res} + S$$
$$= (C + R + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr})$$

where

M = rate of metabolic heat production, W/m^2
W = rate of mechanical work accomplished, W/m^2
q_{sk} = total rate of heat loss from skin, W/m^2
q_{res} = total rate of heat loss through respiration, W/m^2
$C + R$ = sensible heat loss from skin, W/m^2
E_{sk} = total rate of evaporative heat loss from skin, W/m^2
C_{res} = rate of convective heat loss from respiration, W/m^2
E_{res} = rate of evaporative heat loss from respiration, W/m^2
S_{sk} = rate of heat storage in skin compartment, W/m^2
S_{cr} = rate of heat storage in core compartment, W/m^2

Sensible heat loss from skin

$$C = f_{cl} h_c (t_{cl} - t_a) \qquad (5)$$

$$R = f_{cl} h_r (t_{cl} - \bar{t}_r) \qquad (6)$$

where

h_c = convective heat transfer coefficient, $W/(m^2 \cdot K)$
h_r = linear radiative heat transfer coefficient, $W/(m^2 \cdot K)$
f_{cl} = clothing area factor A_{cl}/A_D, dimensionless

The coefficients h_c and h_r are both evaluated at the clothing surface.

Evaporative heat loss from skin

$$E_{sk} = \frac{w(p_{sk,s} - p_a)}{R_{e,cl} + 1/(f_{cl} h_e)} \qquad (12)$$

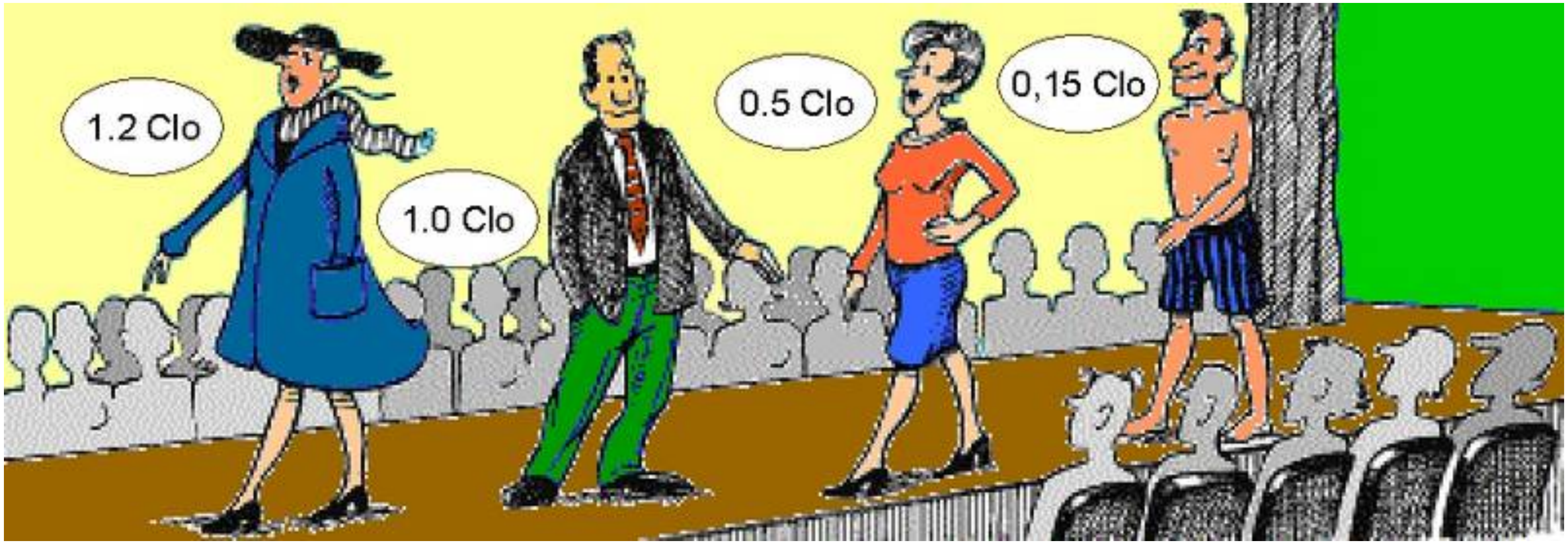
where

w = skin wettedness, dimensionless
$p_{sk,s}$ = water vapor pressure at skin, normally assumed to be that of saturated water vapor at t_{sk}, kPa
p_a = water vapor pressure in ambient air, kPa
$R_{e,cl}$ = evaporative heat transfer resistance of clothing layer (analogous to R_{cl}), $(m^2 \cdot kPa)/W$
h_e = evaporative heat transfer coefficient (analogous to h_c), $W/(m^2 \cdot kPa)$

These equations get complex quickly…

Thermal insulation, I_{cl}

- The thermal insulating effects of clothes are measured in **clos** (1 clo = 0.88 h·ft^{2}·˚F/Btu)
- Insulating values for various garments are found in ASHRAE Fundamentals and Appendix B of Std 55



Mean radiant temperature, T_r

- Radiation to/from occupants is a primary form of energy exchange
  - We can estimate its effects using the **mean radiant temperature**
- The mean radiant temperature is the temperature of an imaginary uniform black box that results in the same radiation heat loss to the occupant as the current room
- This is particularly important for environments with drastically different surface temperatures
  - e.g. a poorly insulated window on a winter day has a surface temperature much lower than most other surfaces around it
  - e.g. a concrete slab warmed by the sun may have a higher temperature than its surroundings

$$\bar{T}_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \cdots + T_N^4 F_{p-N}$$

where

$\bar{T}_r$ = mean radiant temperature, K
T_N = surface temperature of surface N, K
F_{p-N} = angle factor between a person and surface N

ASHRAE comfort zone

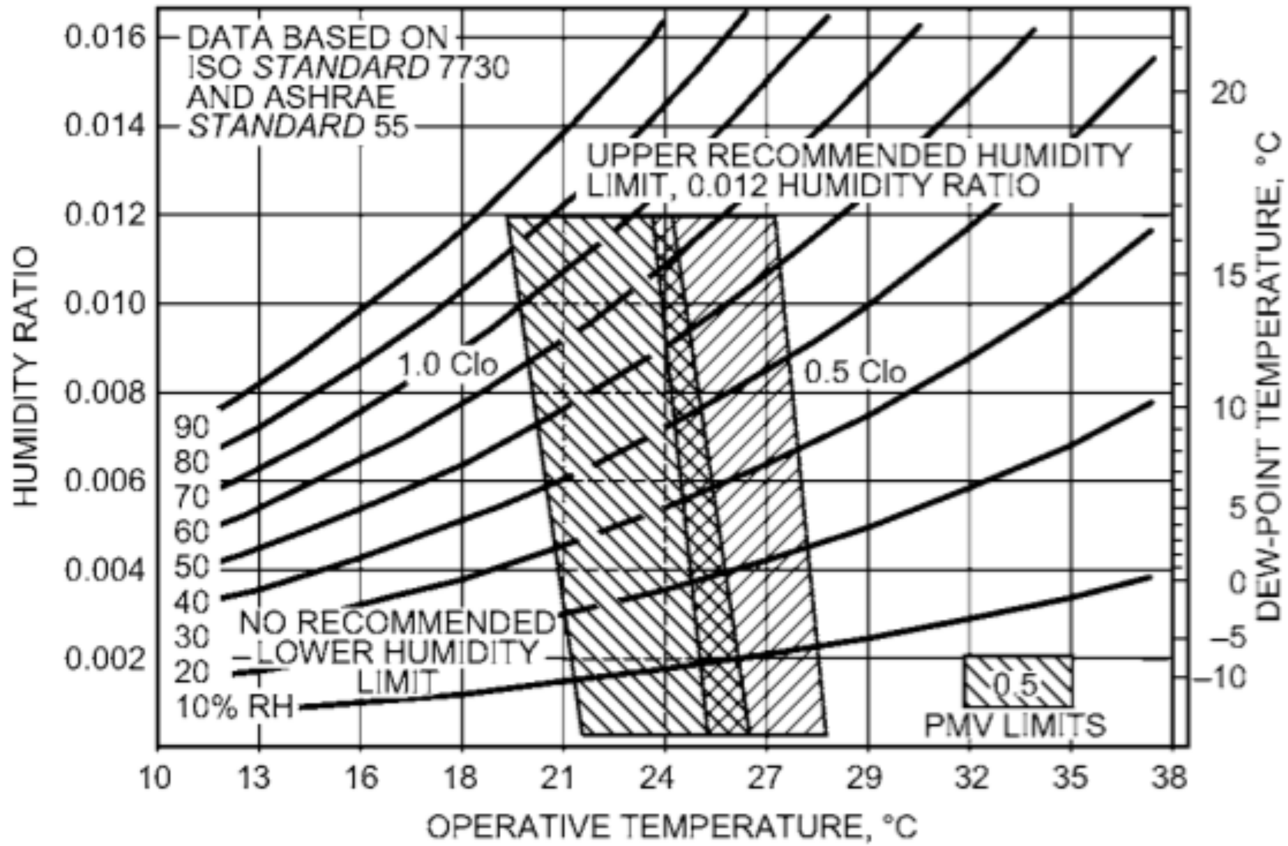


Fig. 5 ASHRAE Summer and Winter Comfort Zones
[Acceptable ranges of operative temperature and humidity with air speed ≤ 0.2 m/s for people wearing 1.0 and 0.5 clo clothing during primarily sedentary activity (≤1.1 met)].

Operative temperature?

- The operative temp is basically the average value between the air temperature and the mean radiant temperature, adjusted for air velocity effects:

$$T_{operative} = AT_a + (1 - A)T_r$$

T_a = ambient temp, T_r=mean radiant temp

$$A = \begin{cases} 0.5 \text{ for } v < 0.2 \text{ m/s} \\ 0.6 \text{ for } 0.2 < v < 0.6 \text{ m/s} \\ 0.7 \text{ for } 0.6 < v < 1.0 \text{ m/s} \end{cases}$$

where v is the air velocity

Finding T_r from globe temperature

- We can measure the temperature of the interior of a black globe as well as the ambient air temperature to estimate T_r
  - The black globe acts as a perfectly round black body radiator

$$T_r = \left[(T_{globe} + 273)^4 + \frac{1.1 \times 10^8 v_{air}^{0.6}}{\varepsilon D^{0.4}} (T_{globe} - T_{air}) \right]^{1/4} - 273$$

T_{globe} = temperature inside globe (°C)
T_{air} = air temperature (°C)
v_{air} = air velocity (m/s)

D = globe diameter (m)
ε = emissivity of globe (-)

Next time

- HW #3 is assigned today
  - Due next week

- Next time we will cover psychrometric processes in building science (particularly in HVAC systems)