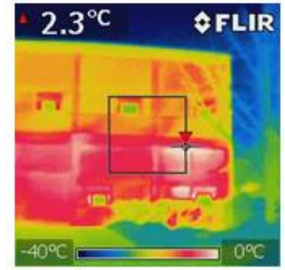
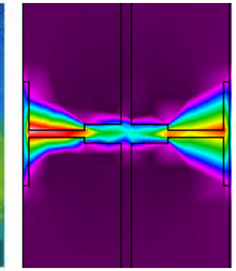
CAE 331/513
Building Science
Fall 2017





October 10, 2017
Psychrometric processes (part 1)

Built
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Dr. Brent Stephens, Ph.D.
Civil, Architectural and Environmental Engineering
Illinois Institute of Technology
brent@iit.edu

News from Illinois Chapter

Oct 10, 2017 Meeting

When

Tuesday, October 10, 2017
5:30 PM to 8:00 PM

Tickets

$45.00 Member Registration

$60.00 Non-Member Registration

$0.00 Student Member Registration

Register Now

Directions

Kaiser Tiger Bar & Grill
1415 W. Randolph St.
Chicago, IL 60601

Get Directions

The City of Chicago Code is one of the toughest codes in the country and can sometimes be difficult to understand and apply to keep yourself out of trouble. That fact along with the updates we have had over the past year can make it even more difficult. Our presenter for this meeting is a guru in this arena and will help give an overview of some of the more common pitfalls and topics of the Chicago code. He will explore the recent updates to topics like energy recovery, the use of toilet exhaust, and updates to the electrical code as well. Please join us for a very informational night on a topic that everyone can relate to!

Our presenters are **Darren B. Meyers, P.E., CEM, GBE, REP, BPI-BA/EP.** Darren is President of IECC_LLC, an energy codes and compliance services group, with engineers located in Champaign, Chicago and Tinley Park. IECC_LLC provides education, code consulting, plan review, HVAC system design, whole-building energy analysis and field assessment services. Darren is an Architectural Engineer with over 25 years of industry experience in energy-systems design and applications in the commercial and residential sectors, including 17-years as Technical Director for Energy Programs with the International Code Council.

Joining Darren at the podium will be **Grant Ullrich, Deputy Commissioner, City of Chicago Department of Buildings**. A graduate of the University of Illinois College of Law, he joined the City as a Law Clerk and moved up to Assistant Corporation Counsel before his current office.

We are excited to hold our first meeting at the **Kaiser Tiger Bar & Grill**, a new venue for us. Everyone is invited. Please pre-register in advance. The registration fee covers the meeting presentation, food and two drink tickets. Vegetarians, please indicate your preference for non-meat alternatives by entering *"Vegetarian"* in the *Special Instruction* box while making your registration. **PDHs** will be provided for the meeting presentation.

ASHRAE Scholarship Program

Deadline: December 1st

APPLY

Each year the ASHRAE Foundation awards scholarships of up to $10,000 each to qualified students.

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33 Society scholarships available for 2018-2019

https://www.ashrae.org/membership--conferences/student-zone/scholarships-and-grants/ashrae-scholarship-program-and-stem

HW 3

- Due today – psychrometric state calculations

Psychrometric equations summary (SI units)

$$pV = nRT$$

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

$$W = 0.622\frac{p_w}{p - p_w} \quad [\frac{kg_w}{kg_{da}}]$$

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

$$\phi = \frac{p_w}{p_{ws}}$$

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

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

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

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 summary (SI units)

Wet bulb temperature (iterative solver):

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

*Where T_{wb} and T are in Celsius

Specific volume:

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

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

where

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

Specific enthalpy:

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

*where T is in °C

Obtaining these data from ASHRAE Tables

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

Table 2 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressu

Temp., °C	Humidity Ratio	Specific Volume, m^3/kg_{da}			Specific Enthalpy, kJ/kg_{da}		
t	W_s, kg_w/kg_{da}	v_{da}	v_{as}	v_s	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 HoF Ch. 1 (2013) Table 3 gives us p_{ws} 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 Sat. Liquid v_i/v_f	Specific Volume, m^3/kg_w Evap. v_{ig}/v_{fg}	Specific Volume, m^3/kg_w Sat. Vapor v_g	Specific Enthalpy, kJ/kg_w Sat. Liquid h_i/h_f	Specific Enthalpy, kJ/kg_w Evap. h_{ig}/h_{fg}	Specific Enthalpy, kJ/kg_w 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

Why is this stuff helpful?

PSYCHROMETRIC PROCESSES

Using the psychrometric chart

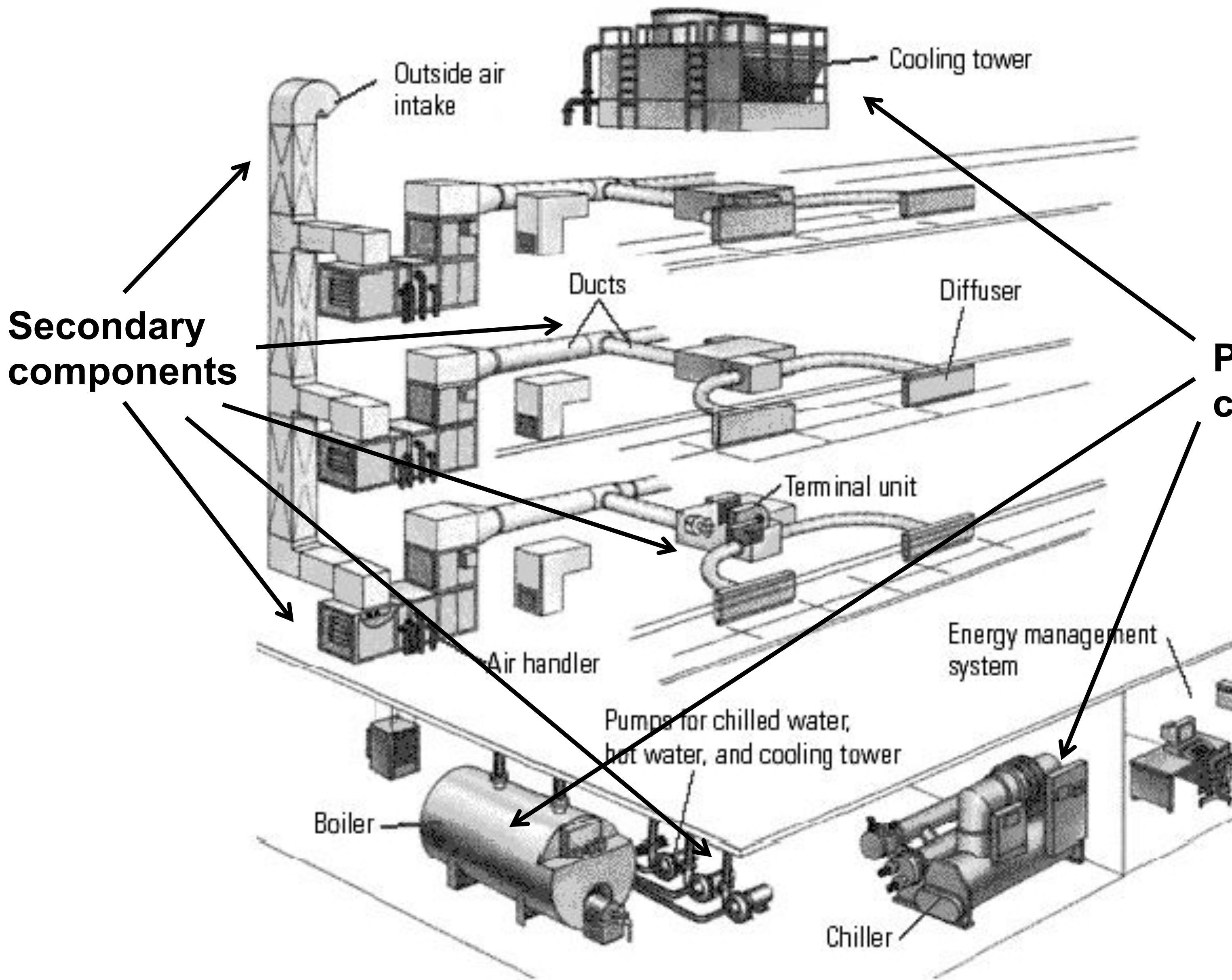
Use of the psychrometric chart for *processes*

We can use the psychrometric chart (and equations) not only to describe states of moist air, but for a number of **processes** that are important for building science and HVAC applications

Examples:

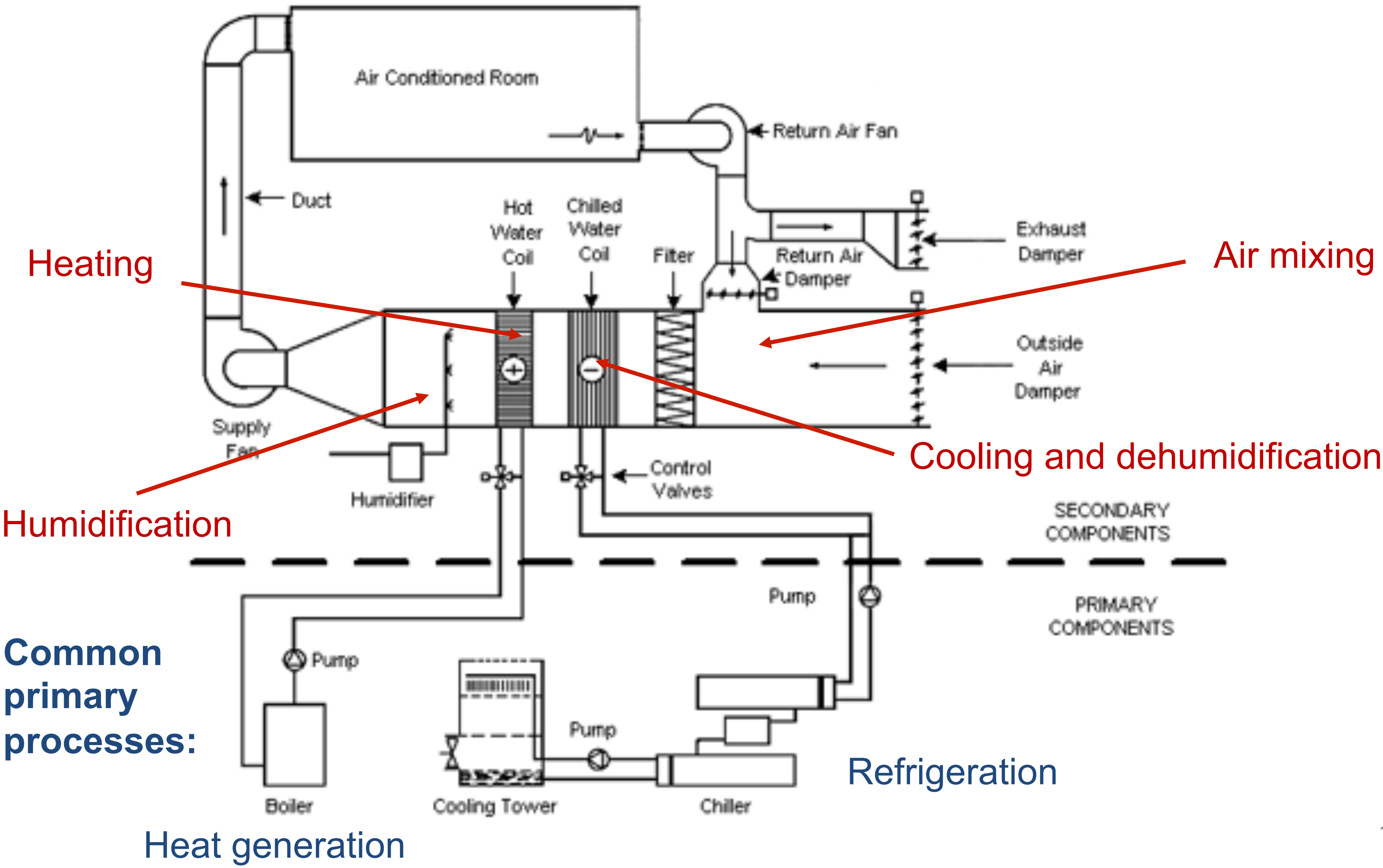
- Sensible cooling or heating
- Warming and humidification of cold, dry air
- Cooling and dehumidification of warm, humid air
  - Sensible + latent cooling
- Evaporative cooling
- Mixing of airstreams

Typical components of an HVAC system



Typical HVAC *processes*

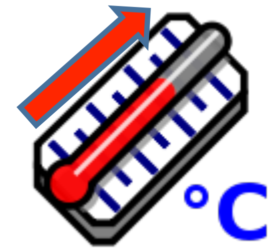
Some common psychrometric processes:



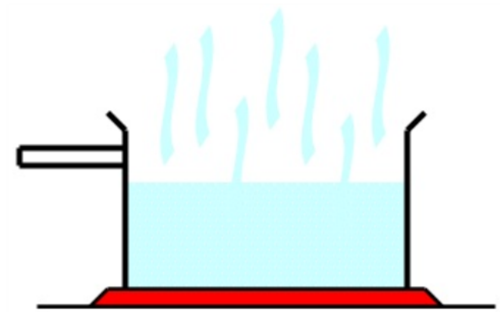
Common primary processes:

Definitions: Sensible and latent heat

- **Sensible heat transfer**
  - Increase or decrease in temperature of a substance *without* undergoing a phase change



- **Latent heat transfer**
  - Heat transfer required to change the phase of a substance (e.g., heat required to change liquid to vapor)



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

Units of [W], [BTU/hr], or [ton]

Sensible and latent heat transfer equation

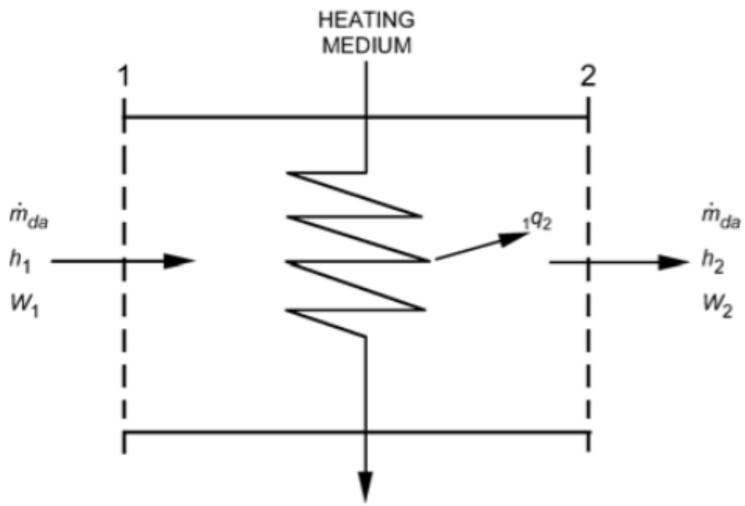


Fig. 2 Schematic of Device for Heating Moist Air

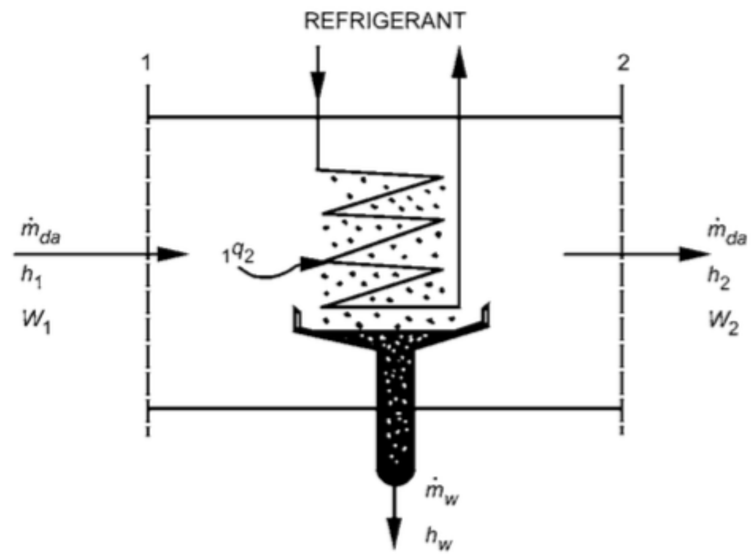


Fig. 3 Schematic of Device for Cooling Moist Air

Generic equations for both heating and cooling processes:

$$Q_{1\rightarrow 2} = \dot{m}_{da}(h_2 - h_1) \qquad Q_{total} = \dot{m}_{da}(h_{exit} - h_{inlet})$$

$Q_{1\rightarrow 2}$ = total rate of heat transfer from state 1 to state 2 (W or BTU/hr or ton)
$\dot{m}_{da}$ = mass flow rate of dry air (kg_{da}/s or lb_{da}/hr)
$h_{exit,2}$ = enthalpy at the exit (J/kg_{da} or BTU/lb_{da})
$h_{inlet,1}$ = enthalpy at the inlet (J/kg_{da} or BTU/lb_{da})

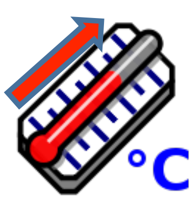
Sensible heat transfer equation

$$Q_{sensible} = \dot{m}_{da} C_p (T_{exit} - T_{inlet}) = \rho_{da} \dot{V}_{da} C_p (T_{exit} - T_{inlet})$$

$Q_{sensible}$ = rate of sensible heat transfer (W or BTU/hr or ton)
C_p = specific heat of air (J/kgK or BTU/lb°F)
ρ_{da} = dry air density (kg/m^3 or lb/ft^3)
T_{inlet} = inlet temperature (K or °F)
T_{exit} = exit temperature (K or °F)

For heating: $Q_{sensible} > 0$

For cooling: $Q_{sensible} < 0$



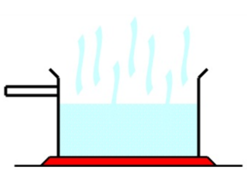
Latent heat transfer equation

$$Q_{latent} = \dot{m}_{da} h_{fg} (W_{exit} - W_{inlet}) = \rho_{da} \dot{V}_{da} h_{fg} (W_{exit} - W_{inlet})$$

Q_{latent} = rate of latent heat transfer (W or BTU/hr or ton)
m_w = mass flow rate of water vapor (kg_w/s or lb_w/hr)
h_{fg} = enthalpy, or latent heat, of vaporization (J/kg or BTU/lb)
* h_{fg} = 2260 kJ/kg or 970 BTU/lb for water
W_{inlet} = inlet humidity ratio (kg_w/kg_{da} or lb_w/lb_{da})
W_{exit} = exit humidity ratio (kg_w/kg_{da} or lb_w/lb_{da})

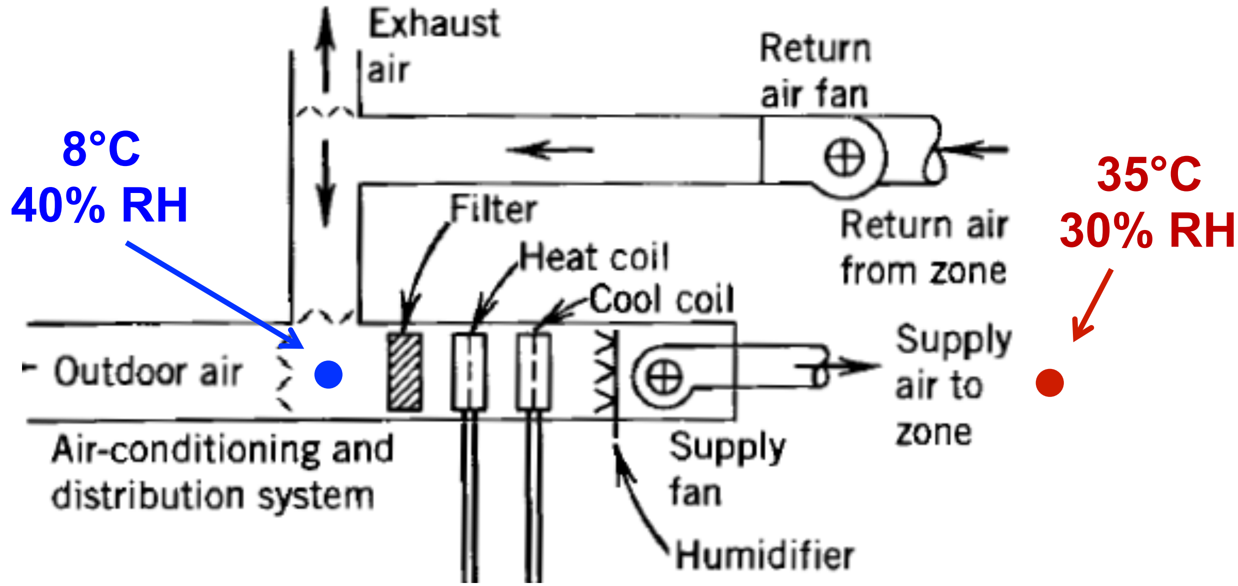
For humidification: $Q_{latent} > 0$

For dehumidification: $Q_{latent} < 0$



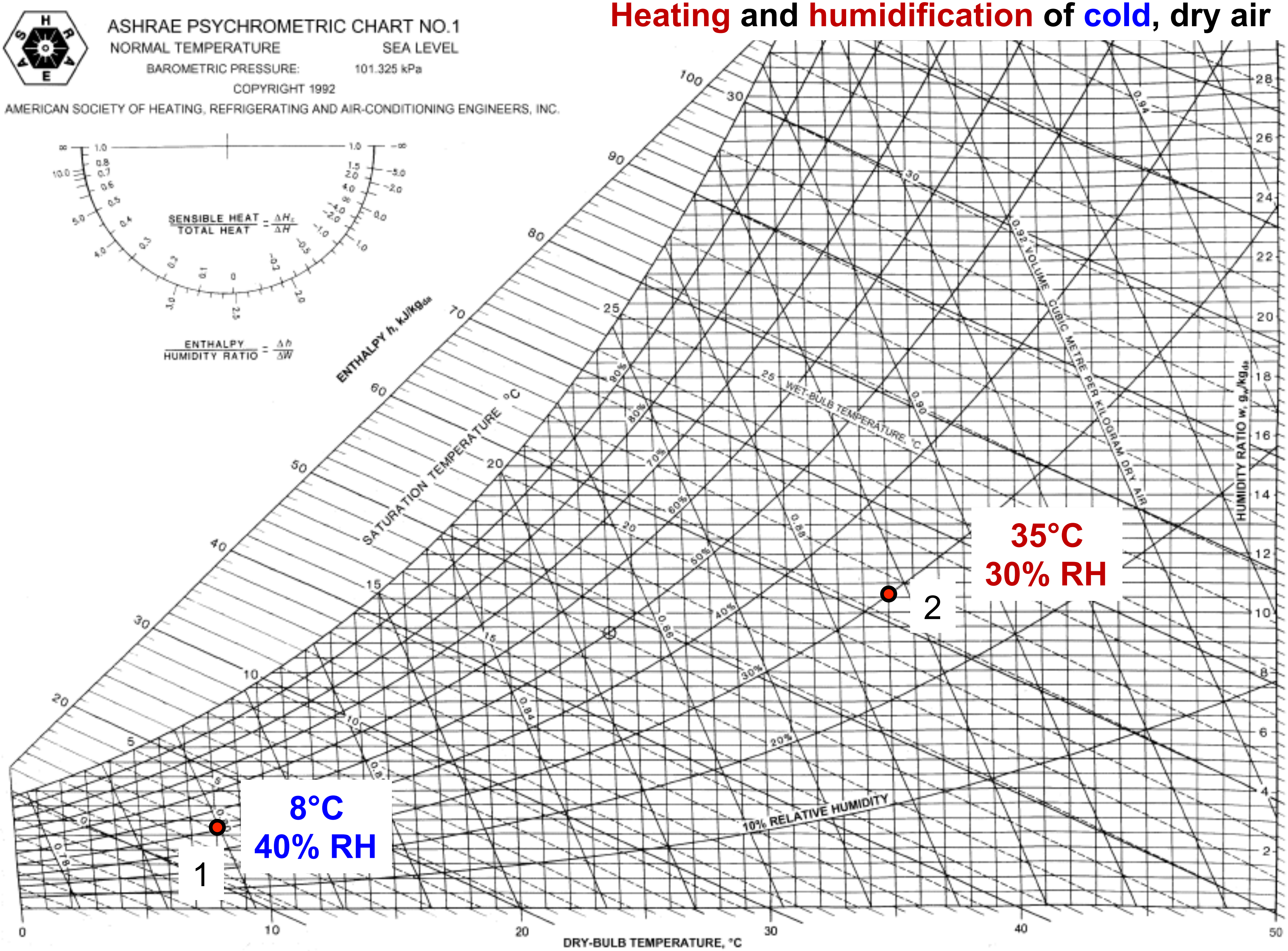
Heating and humidification of cold, dry air

- **Example:** Heating and humidification of air
  - Process: Adding moisture and heat (sensible + latent heating)

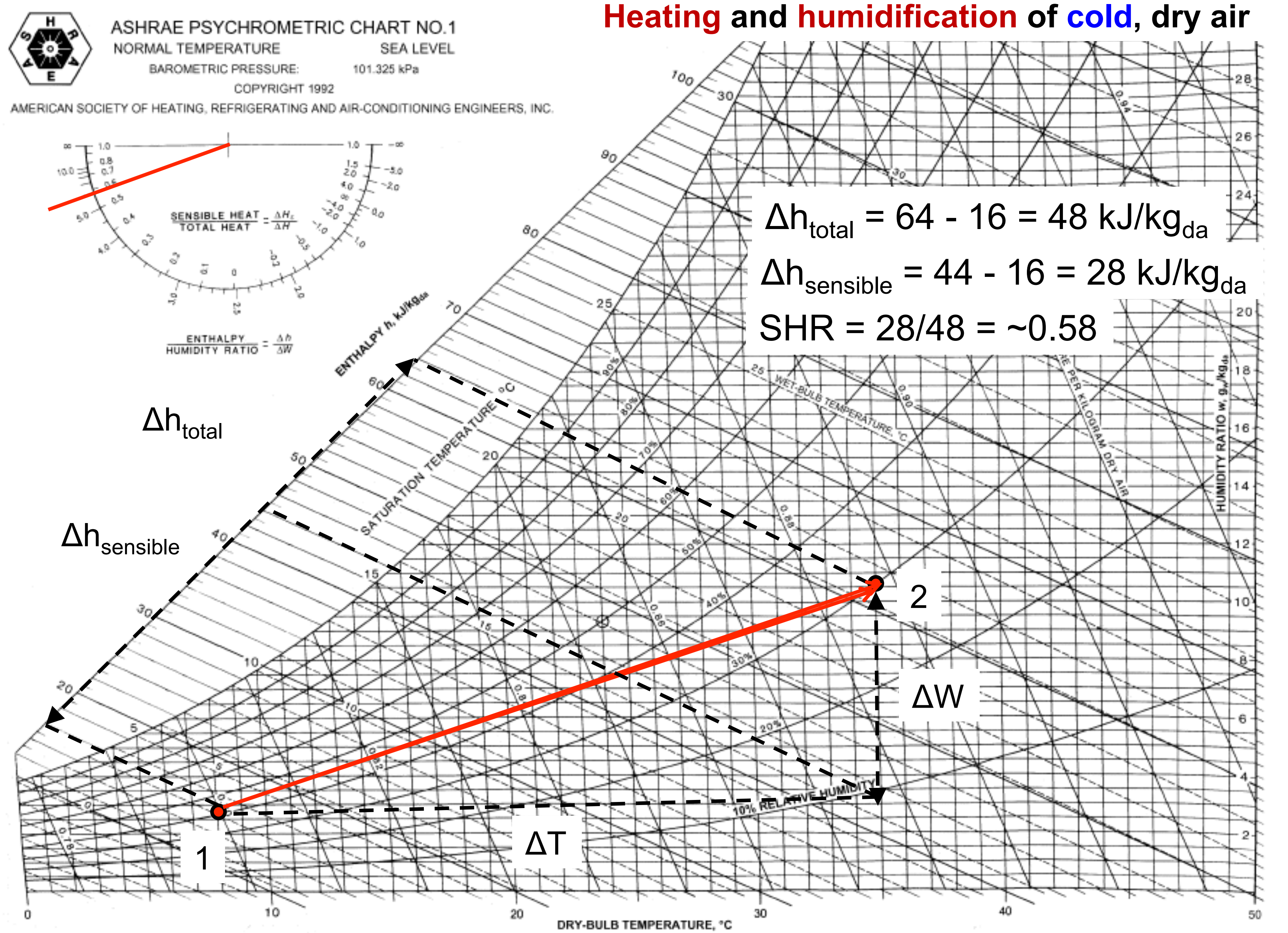


Q1: What is the enthalpy change required?
Q2: What is the total rate of heat transfer if the airflow rate is 10 m^3/s?
Q3: What is the split between sensible and latent transfer?

Heating and humidification of cold, dry air



Heating and humidification of cold, dry air



Δh_{total} = 64 - 16 = 48 kJ/kg$_{da}$

$\Delta h_{sensible}$ = 44 - 16 = 28 kJ/kg$_{da}$

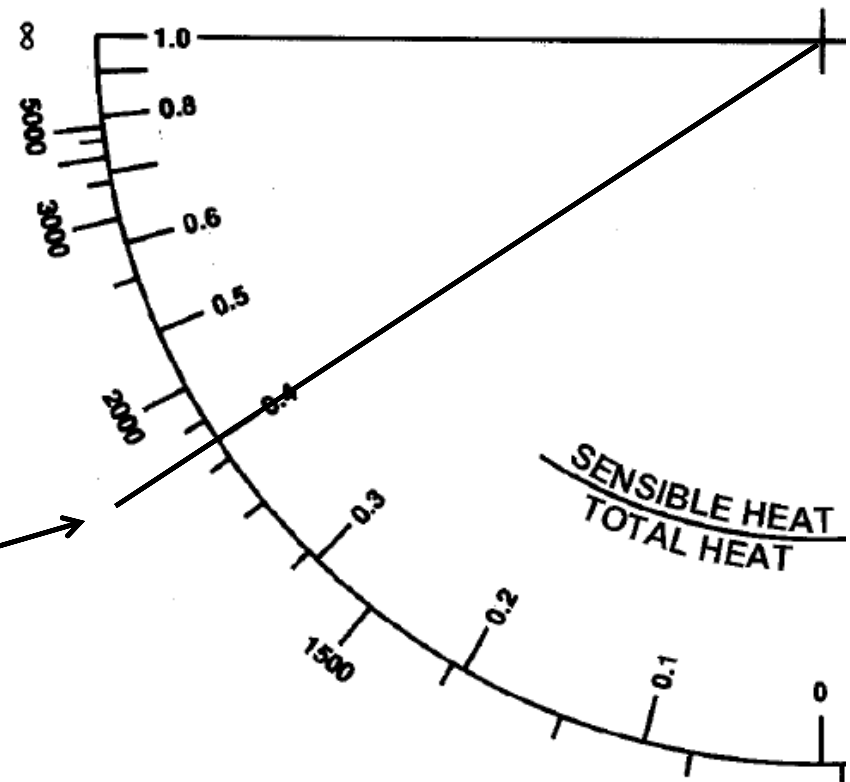
SHR = 28/48 = ~0.58

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

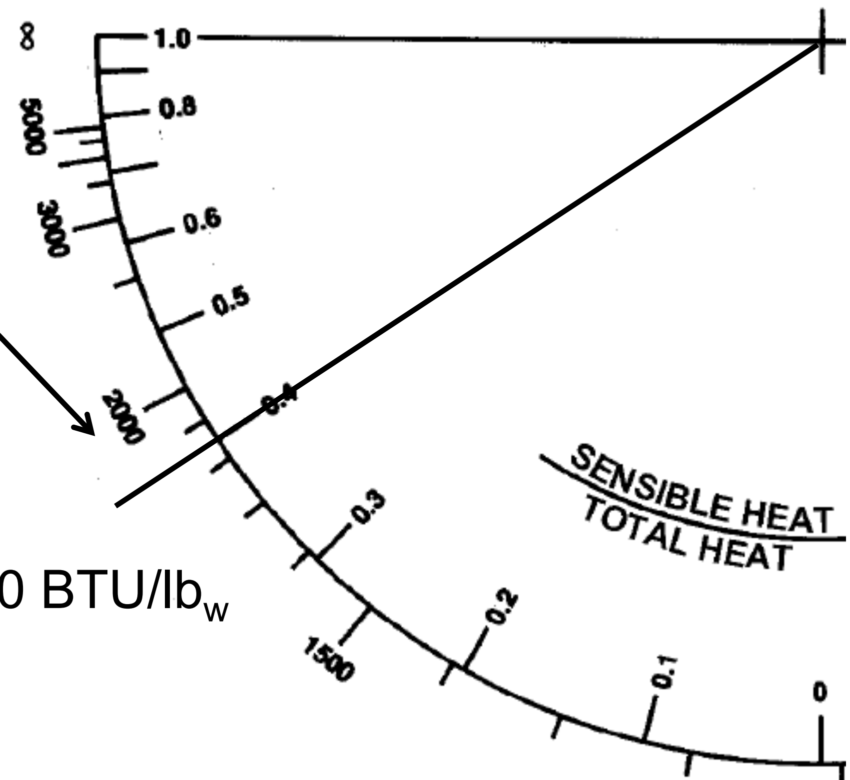


Here is a process
with an SHR ≈ 0.4

Enthalpy protractor ($\Delta h/\Delta W$)

- The other side of the enthalpy protractor tells us:
  - What is the enthalpy change relative to the change in humidity ratio

$$\frac{\Delta h_{total}}{\Delta W}$$

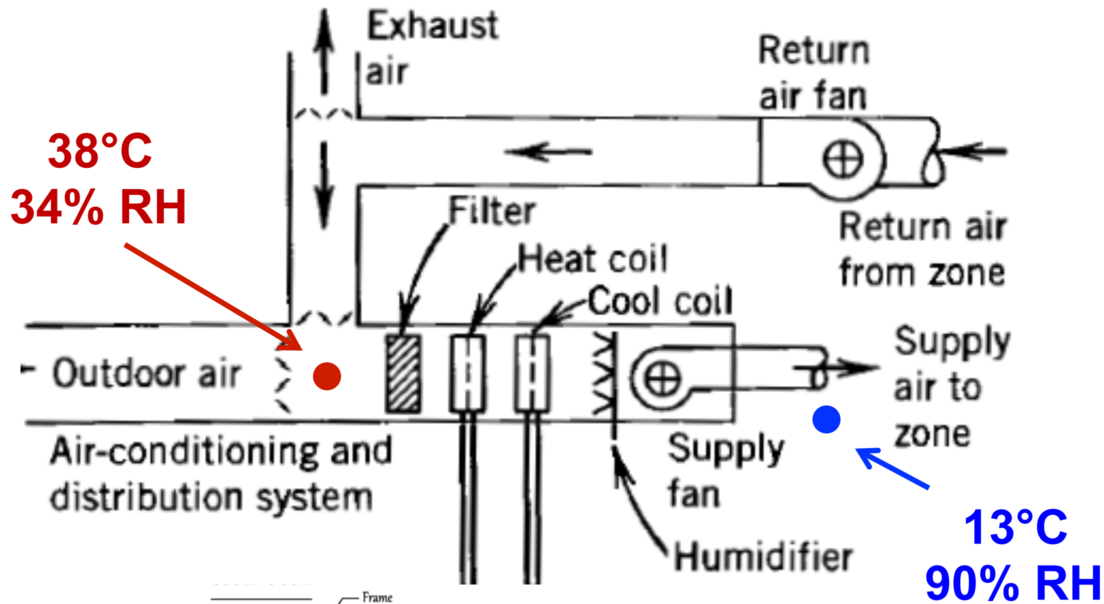


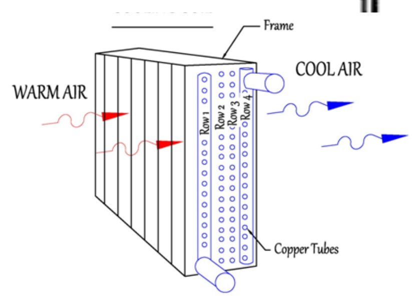
Here is a process with $\Delta h/\Delta W \approx 1850$ BTU/lb_w

SI units: kJ/kg_w

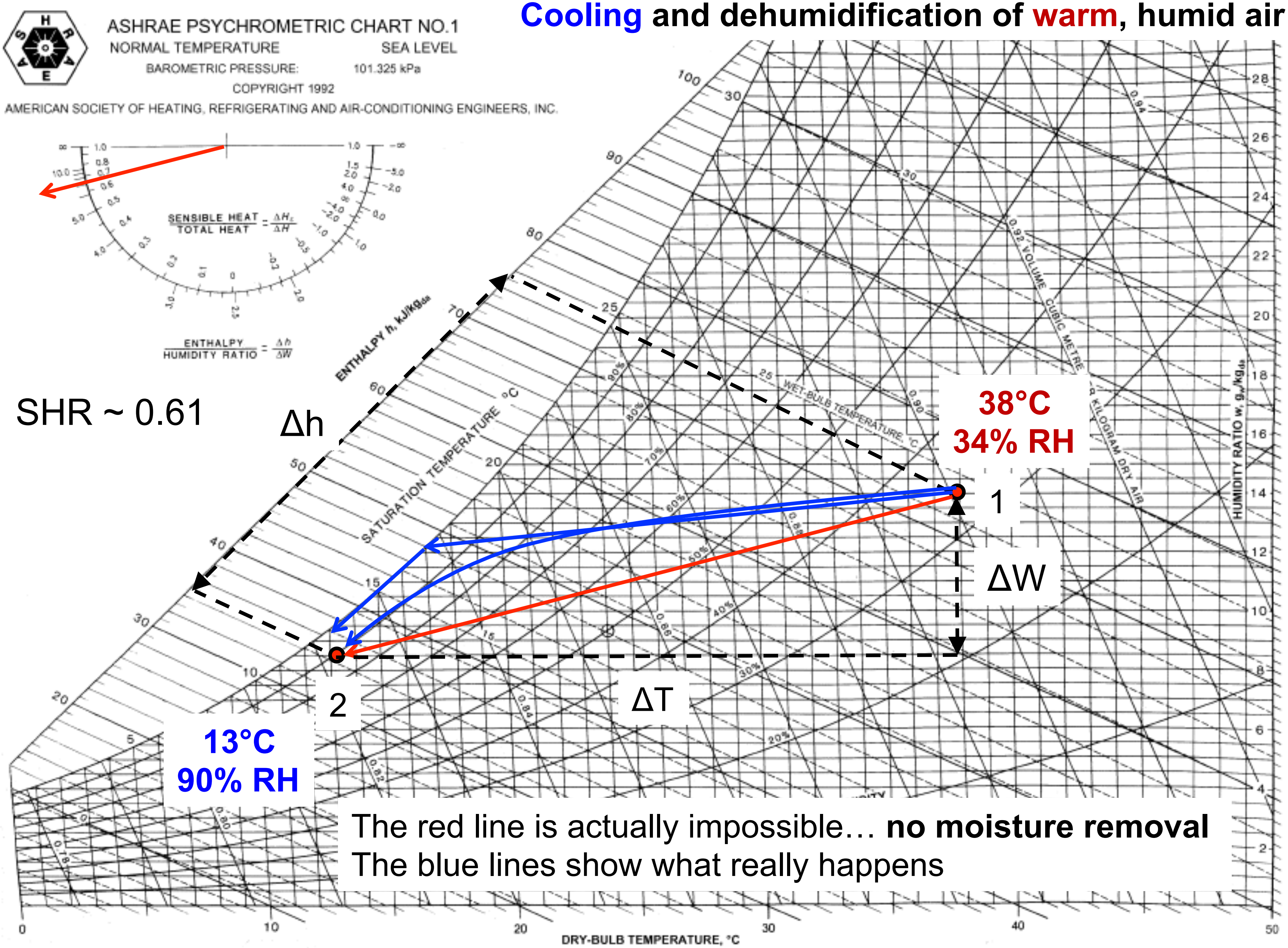
Cooling and dehumidification of **warm**, humid air

- **Example:** Air flowing over a cooling coil
- Removing both moisture and heat
  - Sensible + latent cooling





Cooling and dehumidification of warm, humid air



SHR ~ 0.61

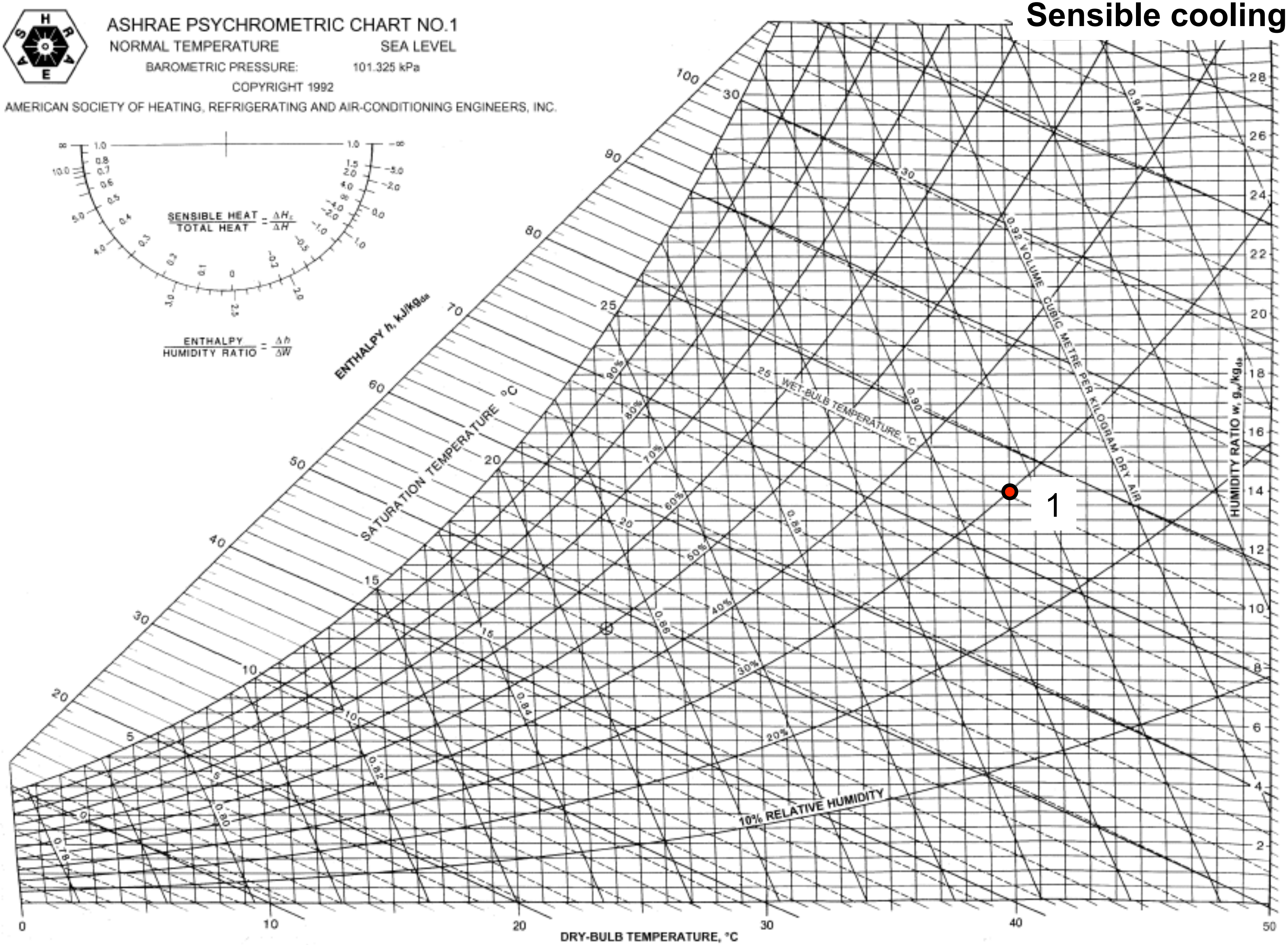
The red line is actually impossible… **no moisture removal**

The blue lines show what really happens

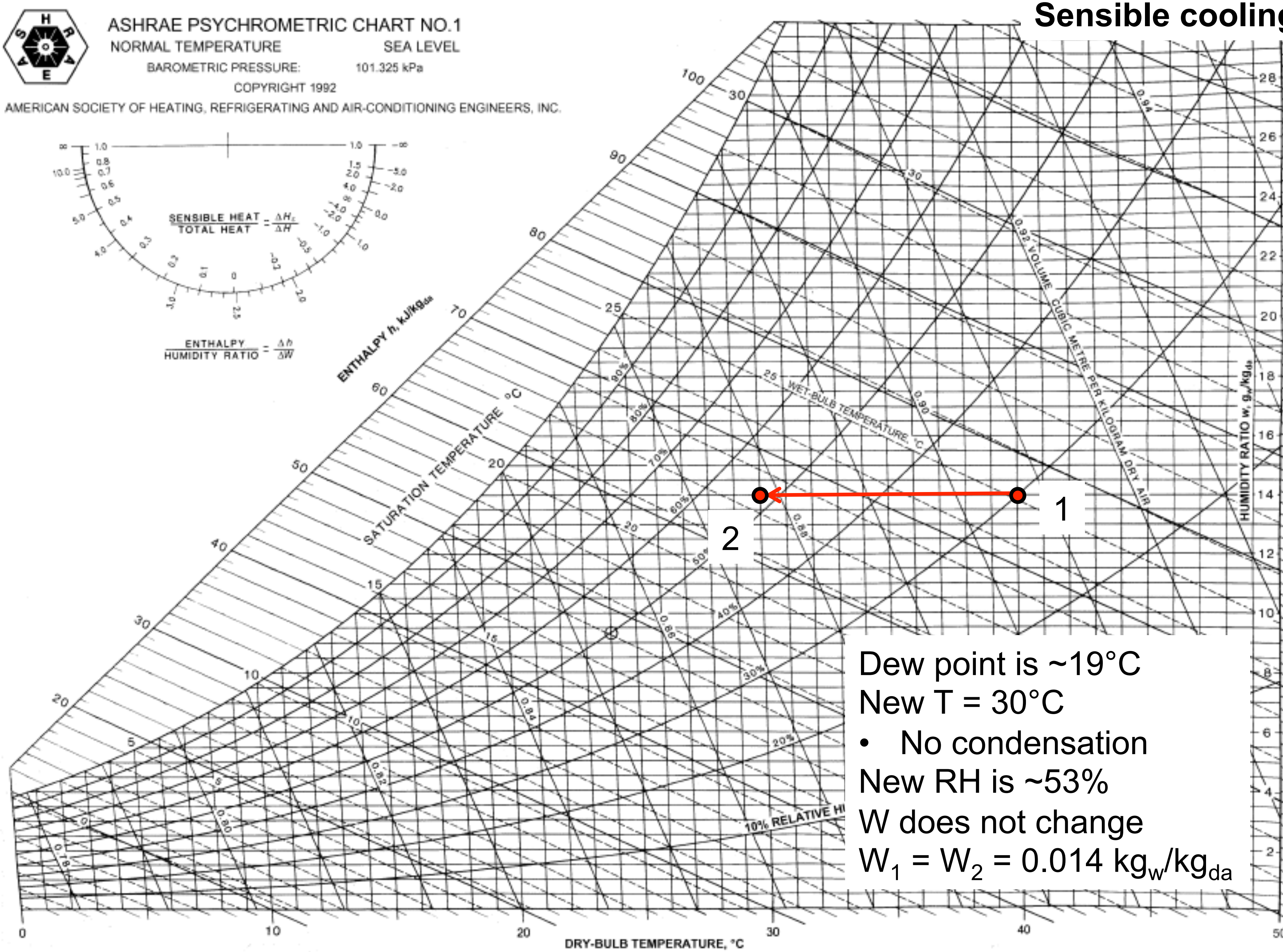
Example: *Sensible* cooling

- Moist air is cooled from 40°C and 30% RH to 30°C without condensation
  - What is the RH at W at the process end point?

Sensible cooling



Sensible cooling



Dew point is ~19°C

New T = 30°C

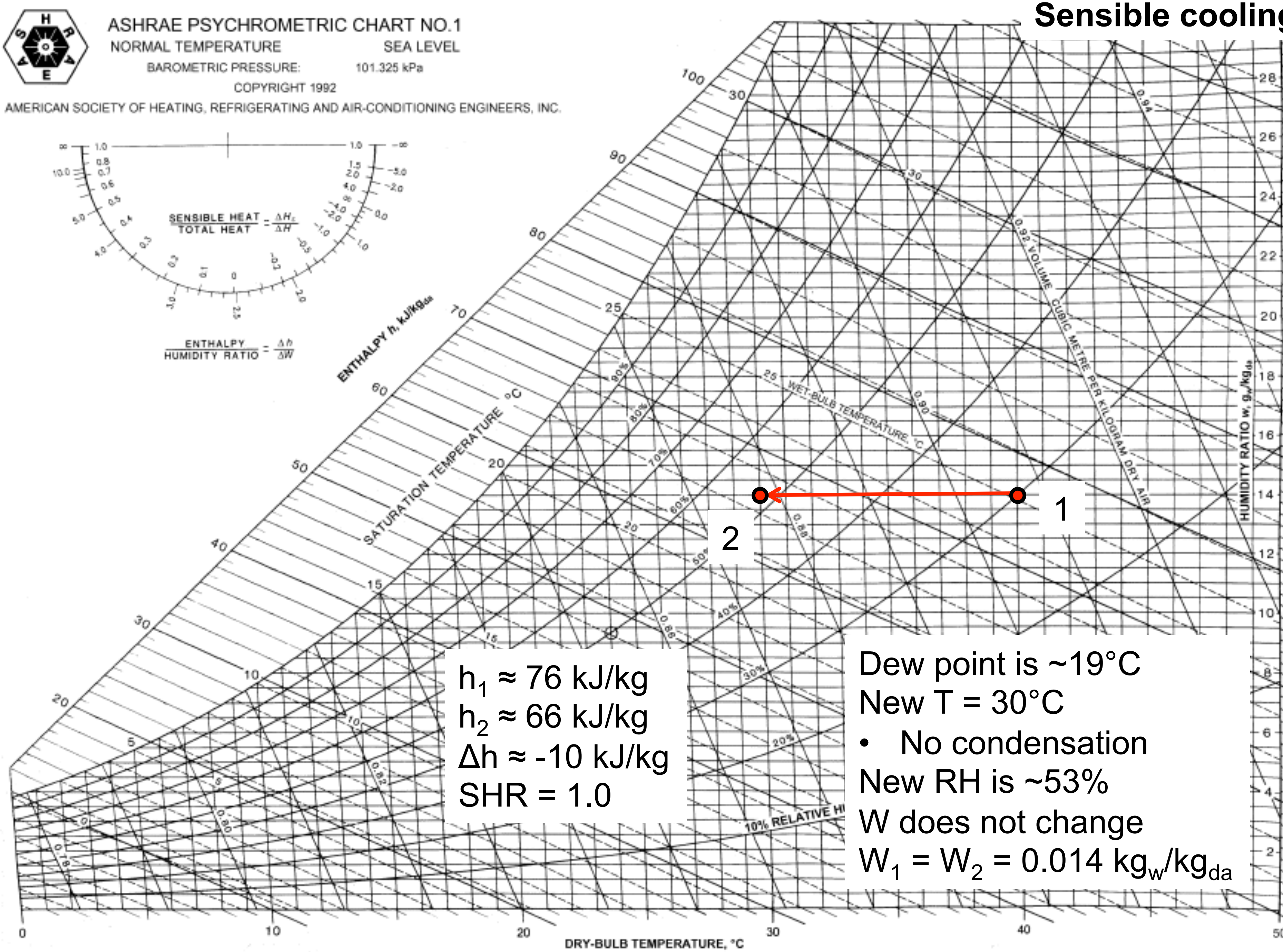
- No condensation

New RH is ~53%

W does not change

$W_1 = W_2 = 0.014\ kg_w/kg_{da}$

Sensible cooling



$h_1 \approx 76$ kJ/kg
$h_2 \approx 66$ kJ/kg
$\Delta h \approx -10$ kJ/kg
SHR = 1.0

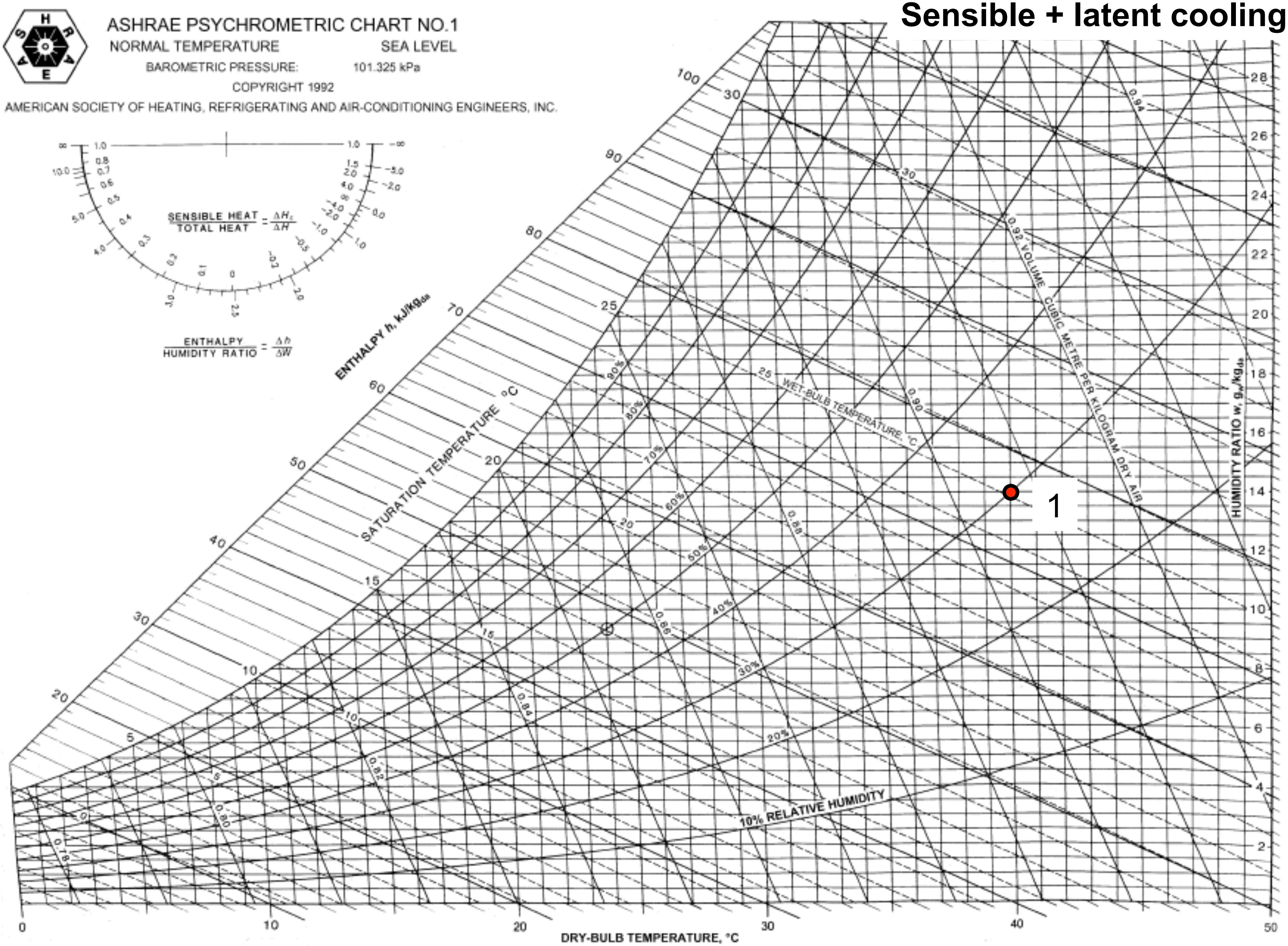
Dew point is ~19°C
New T = 30°C
- No condensation

New RH is ~53%
W does not change
$W_1 = W_2 = 0.014\ kg_w/kg_{da}$

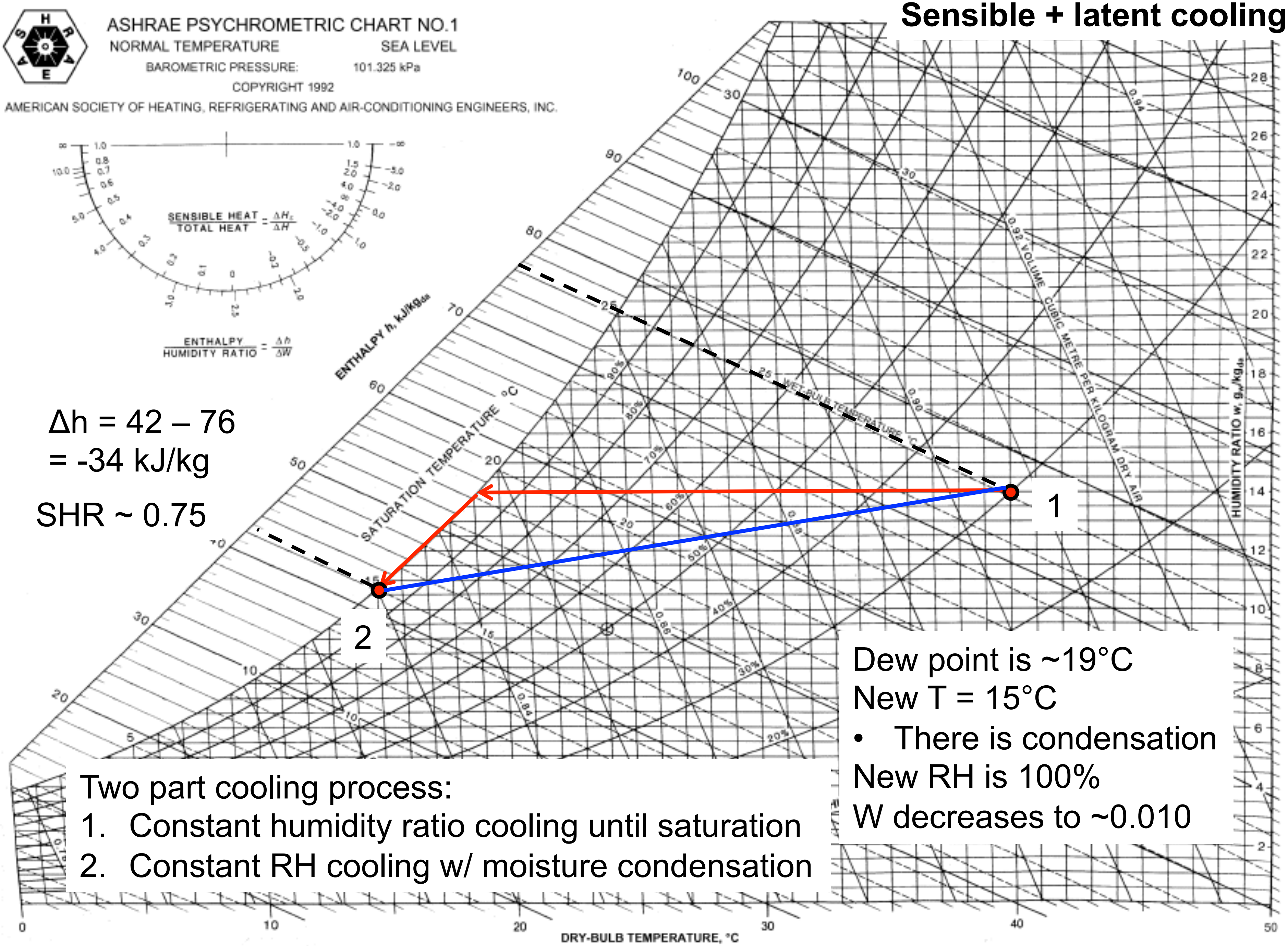
Example: ***Sensible*** **+** ***latent*** **cooling**

- Moist air is cooled from 40°C and 30% RH to 15°C
  - Q1: Does the water vapor condense?
  - Q2: What is RH at W at the process end point?

Sensible + latent cooling



Sensible + latent cooling



$\Delta h = 42 - 76$
$= -34$ kJ/kg

SHR ~ 0.75

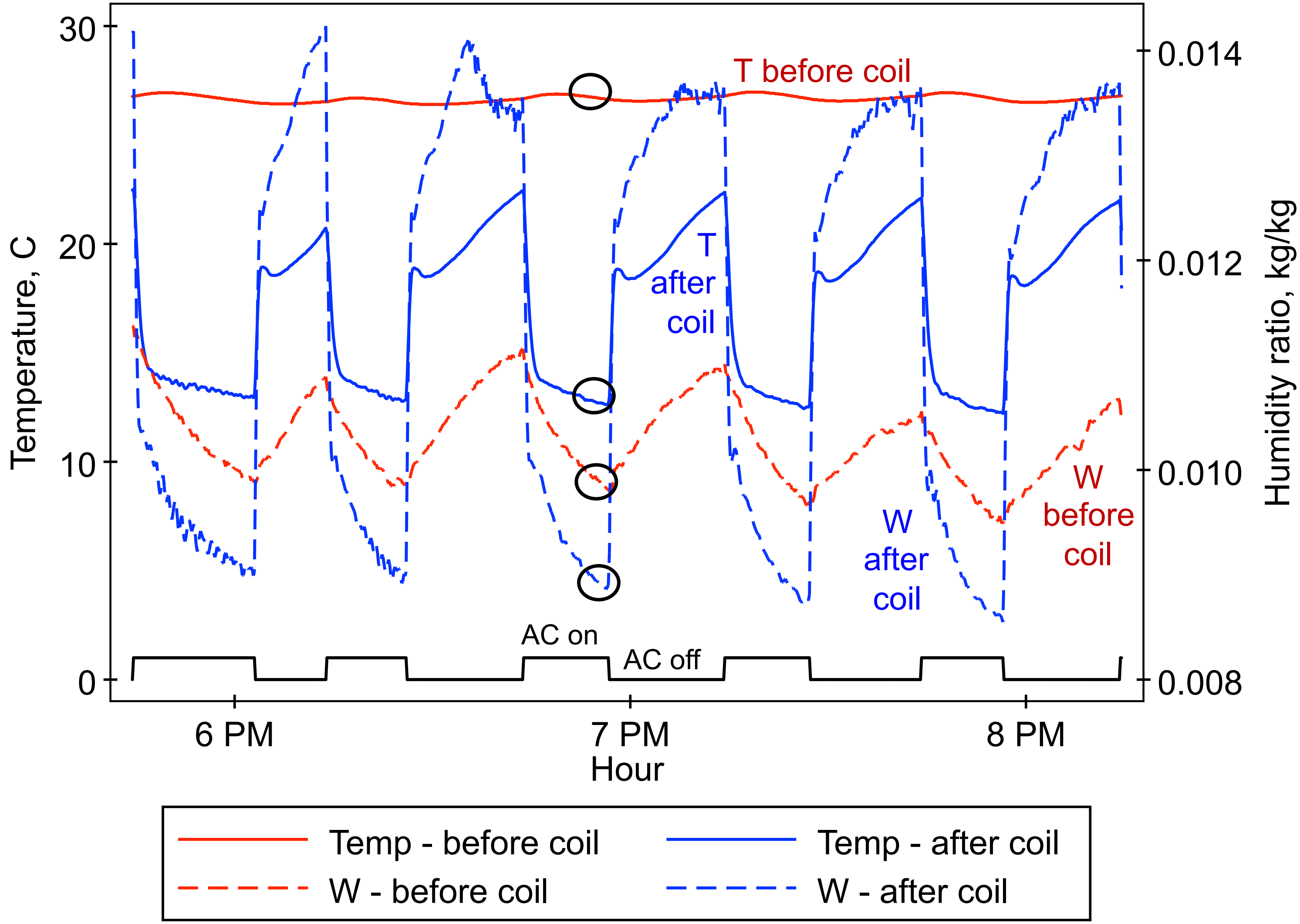
Dew point is ~19°C
New T = 15°C
- There is condensation

New RH is 100%
W decreases to ~0.010

Two part cooling process:
1. Constant humidity ratio cooling until saturation
2. Constant RH cooling w/ moisture condensation

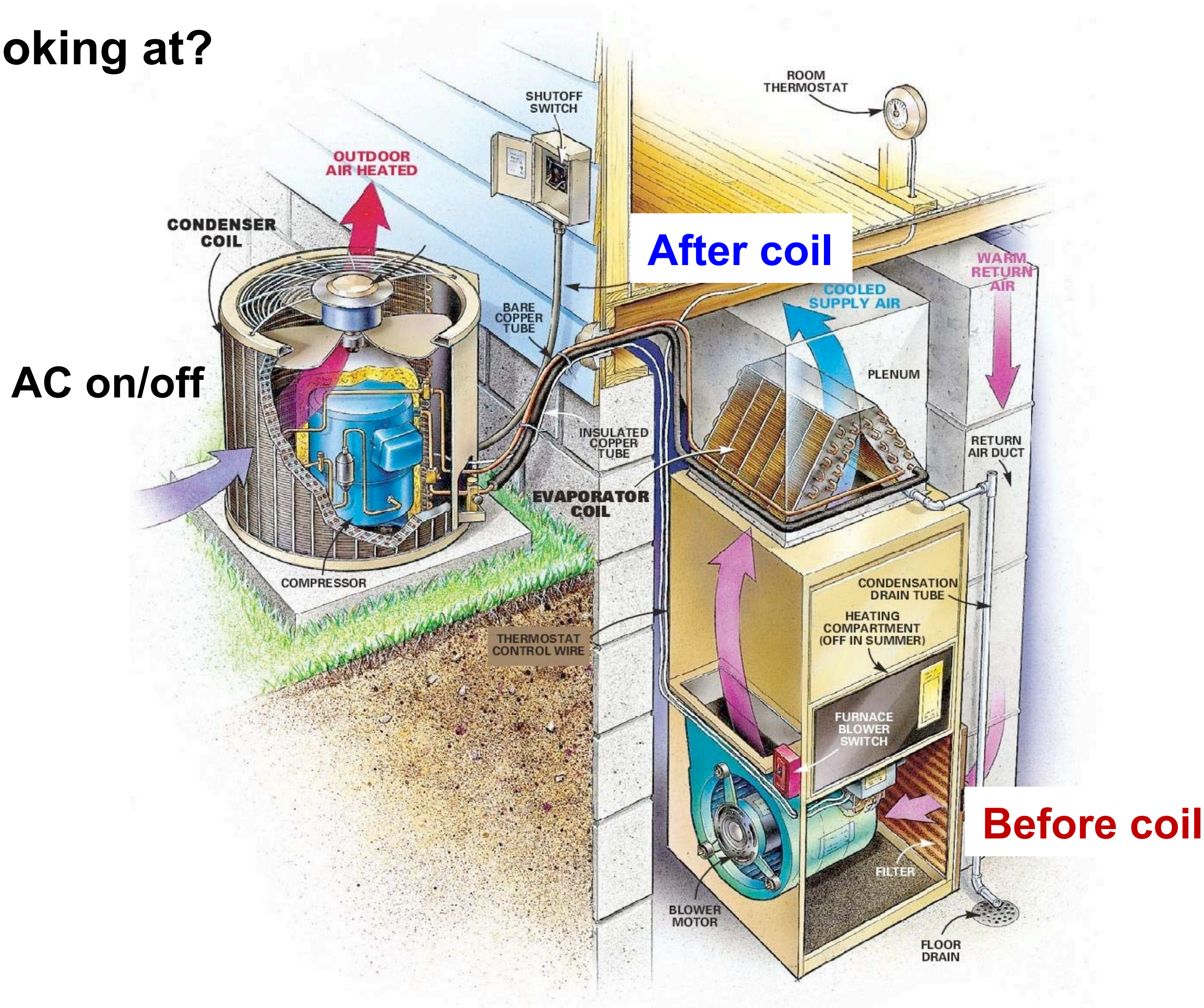
Real data: ASHRAE RP-1299
Energy implications of filters

Temperature and humidity ratio differences across AC coils in homes

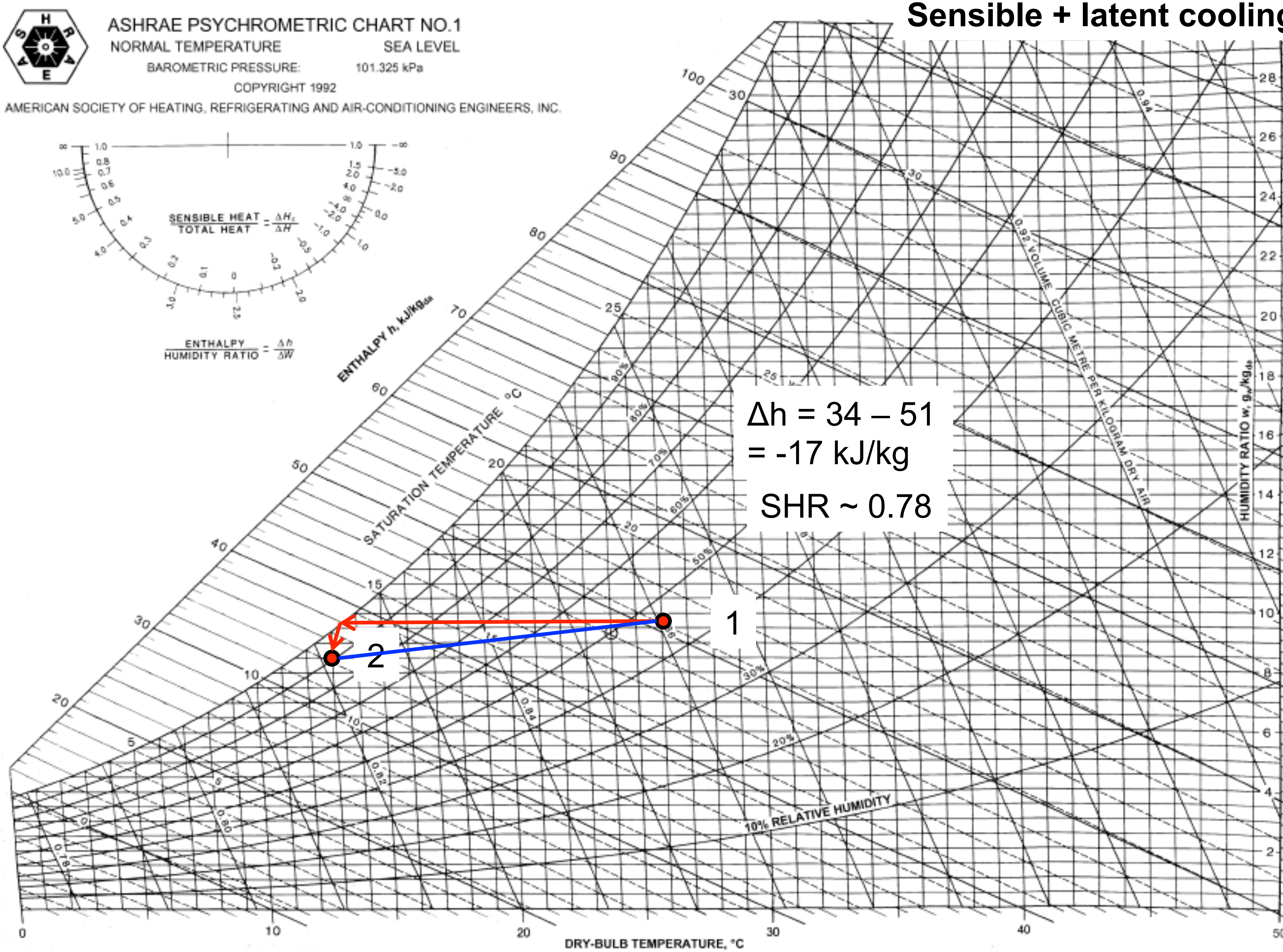


Real data: ASHRAE RP-1299
Energy implications of filters

What are we looking at?

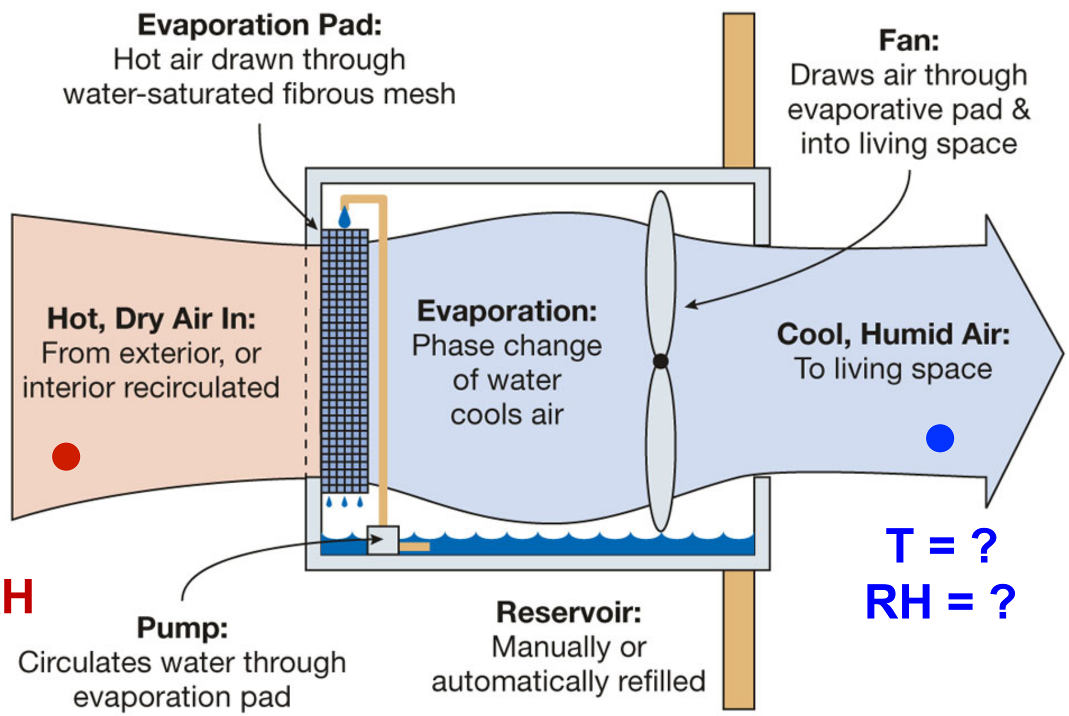


Sensible + latent cooling



Evaporative cooling example

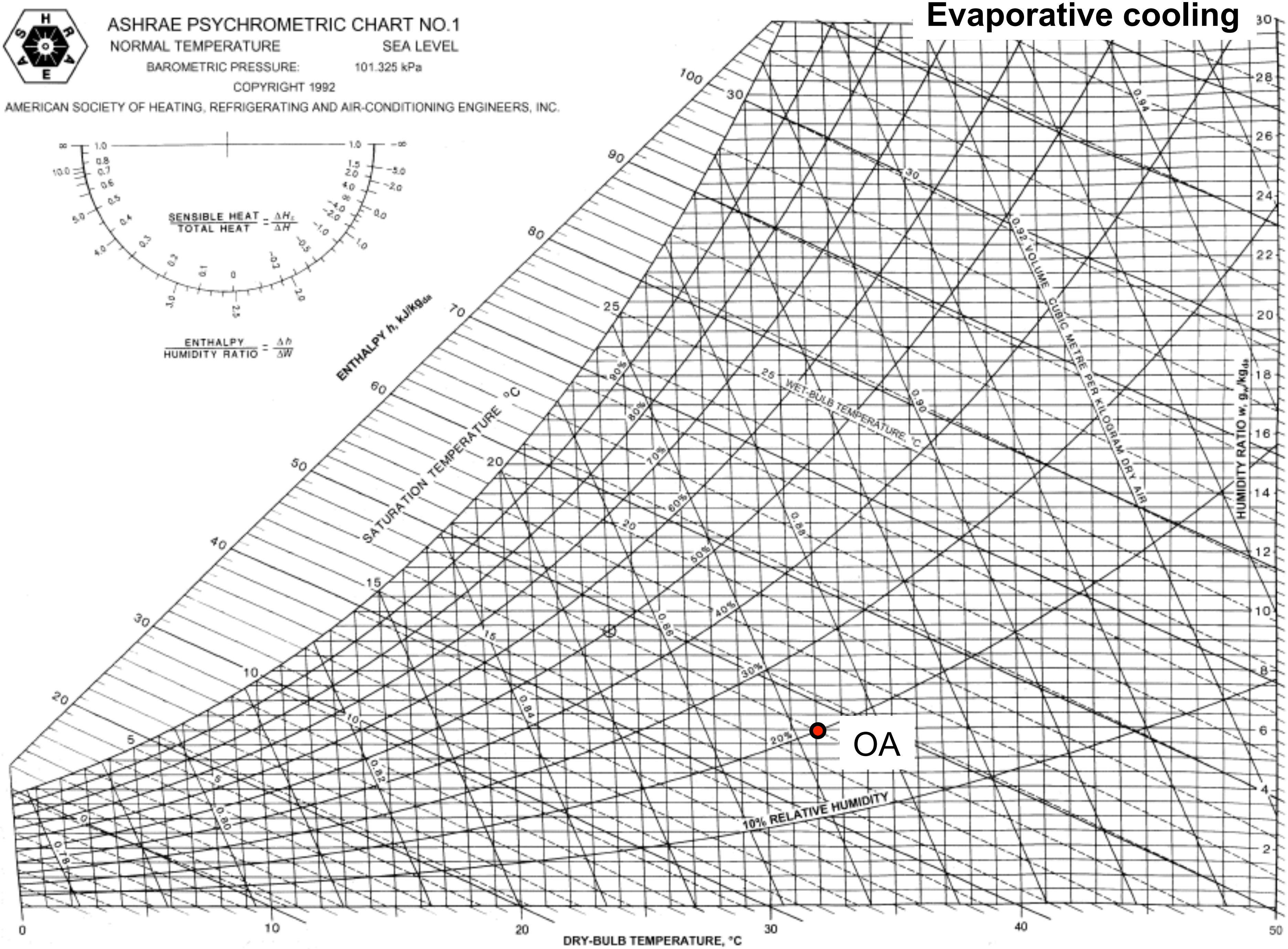
- Hot, dry outdoor air is cooled with an evaporative cooler, or “swamp cooler”
  - **Q1: What is the T, RH, and W of the supply air?**
  - **Q2: Why would we choose this system?**



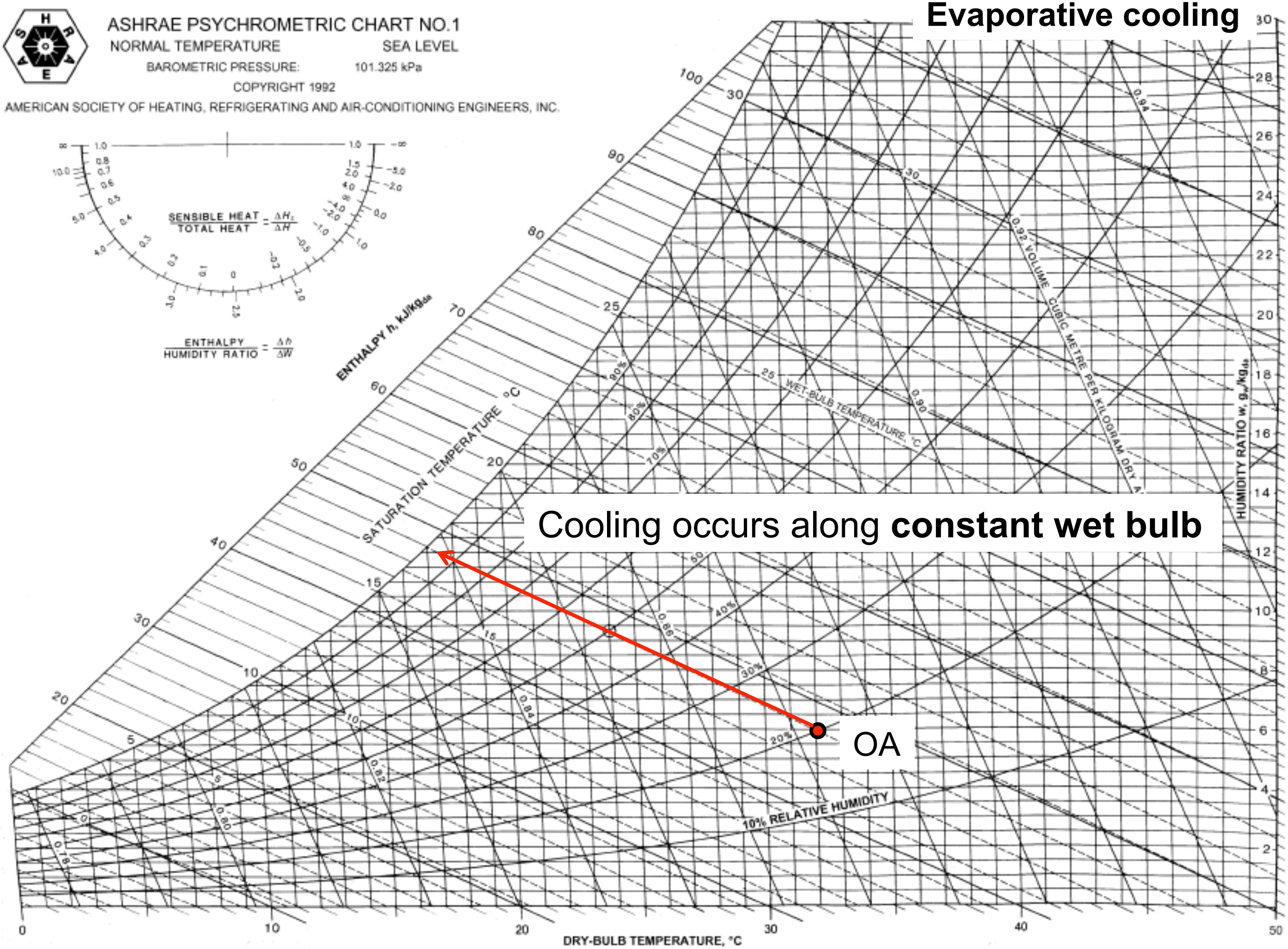
32°C
20% RH

T = ?
RH = ?

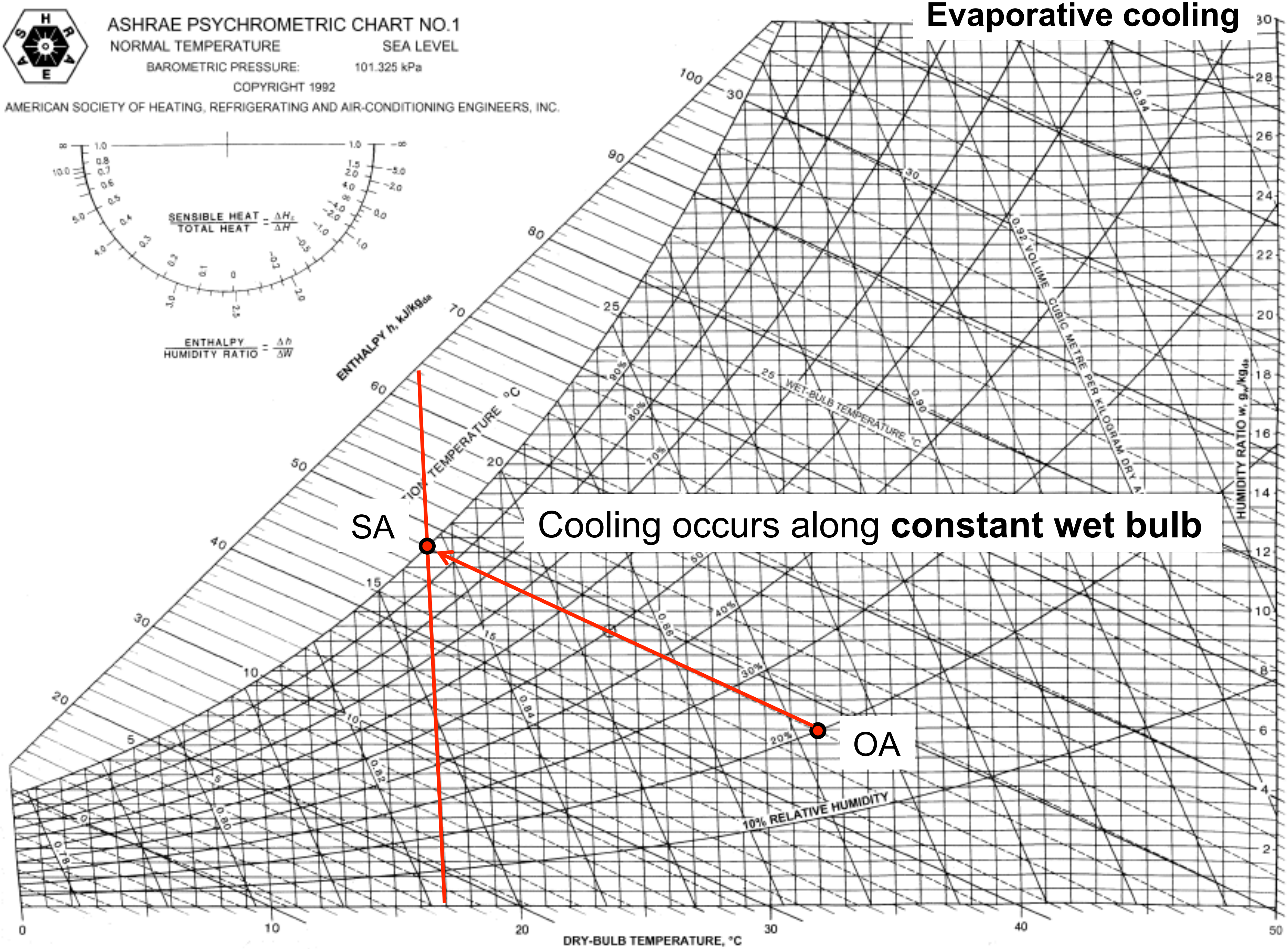
Evaporative cooling



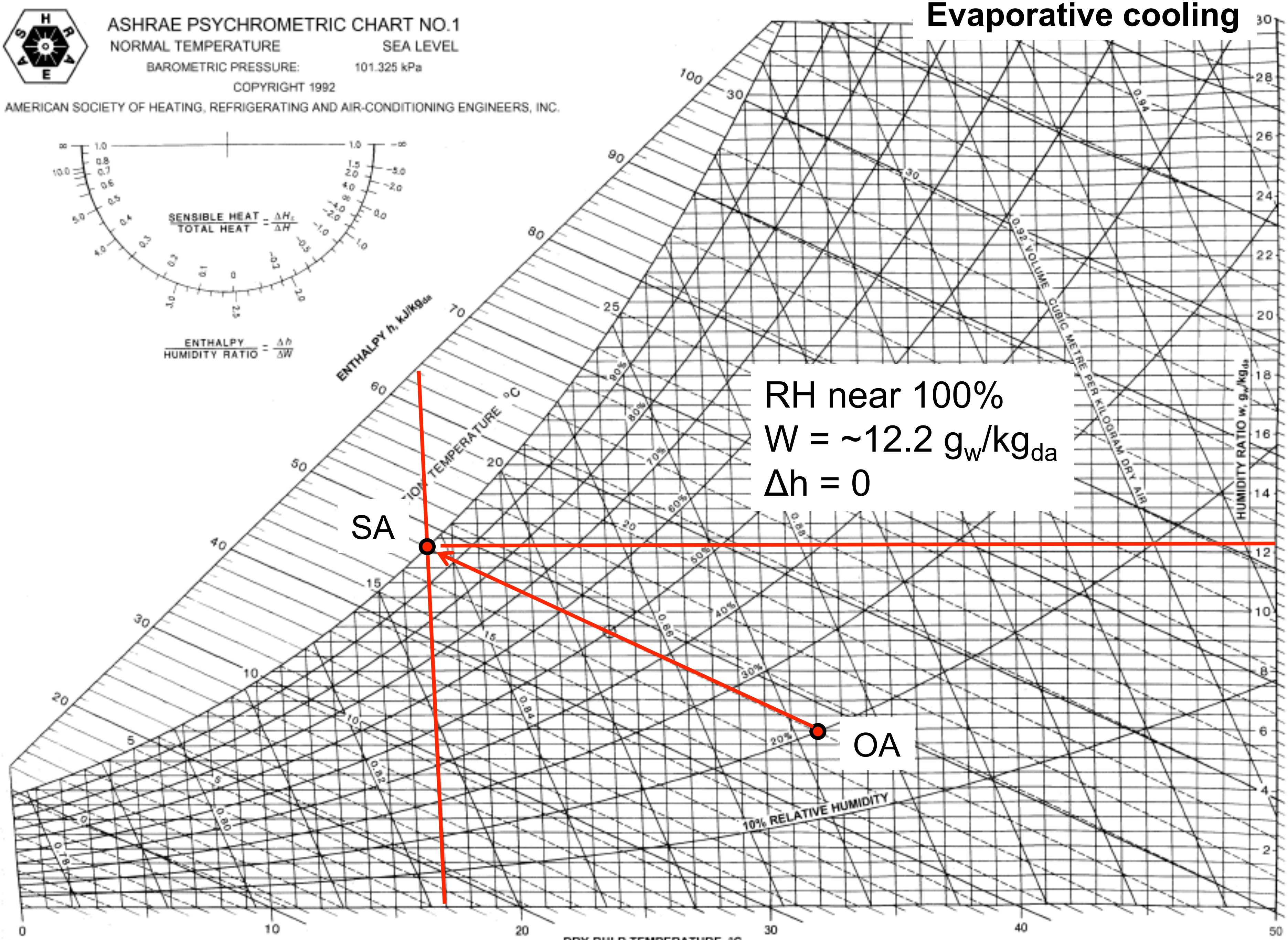
Evaporative cooling



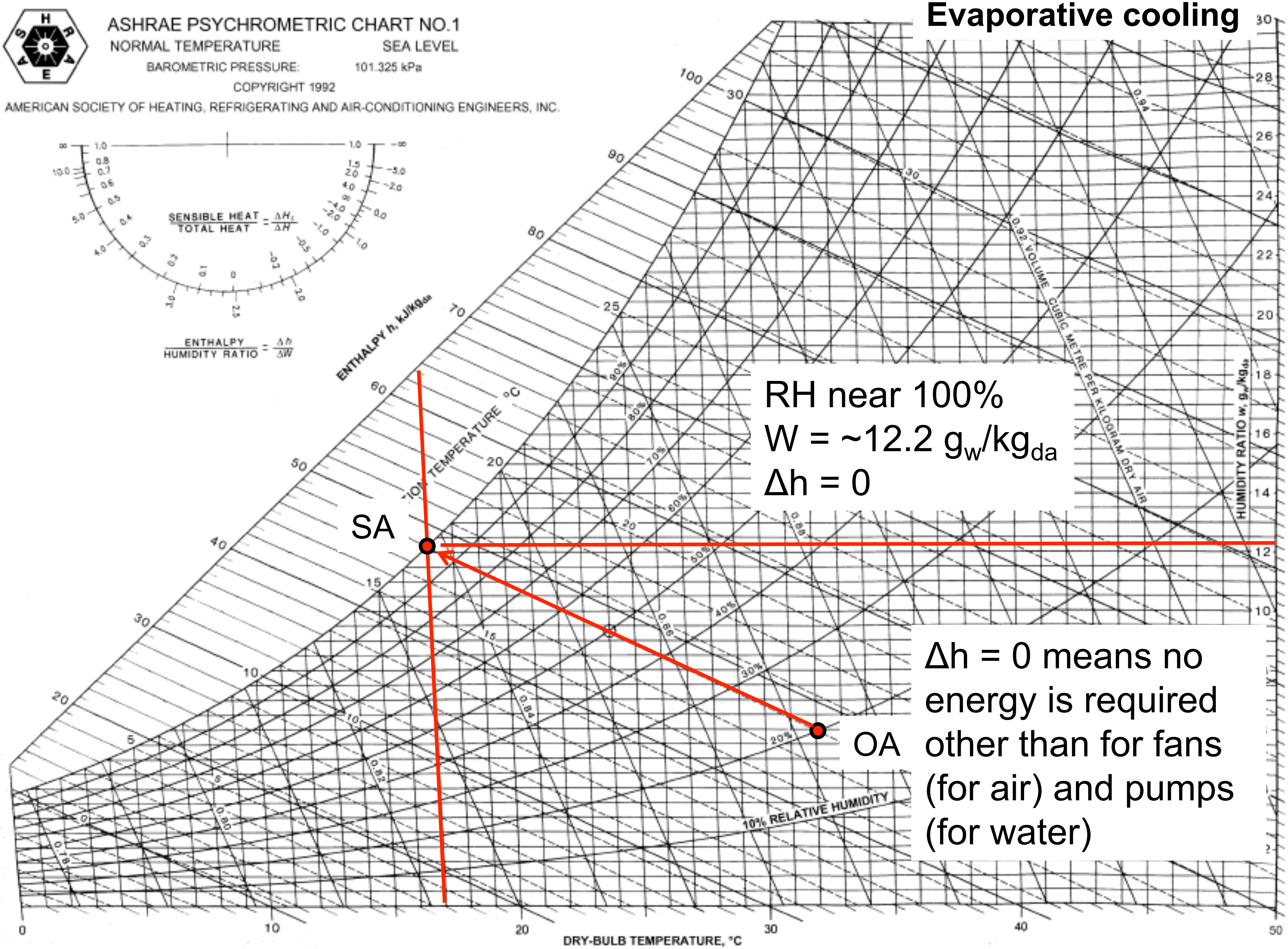
Evaporative cooling



Evaporative cooling



Evaporative cooling

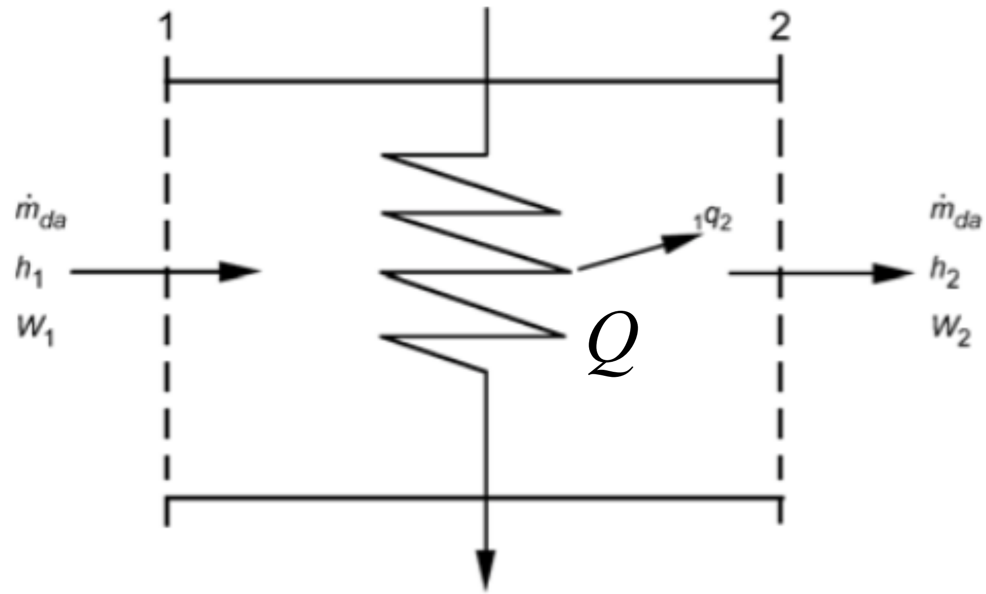


PSYCHROMETRIC PROCESSES

Using energy and mass balance equations

Energy/mass balances for psychrometric processes

Heating air



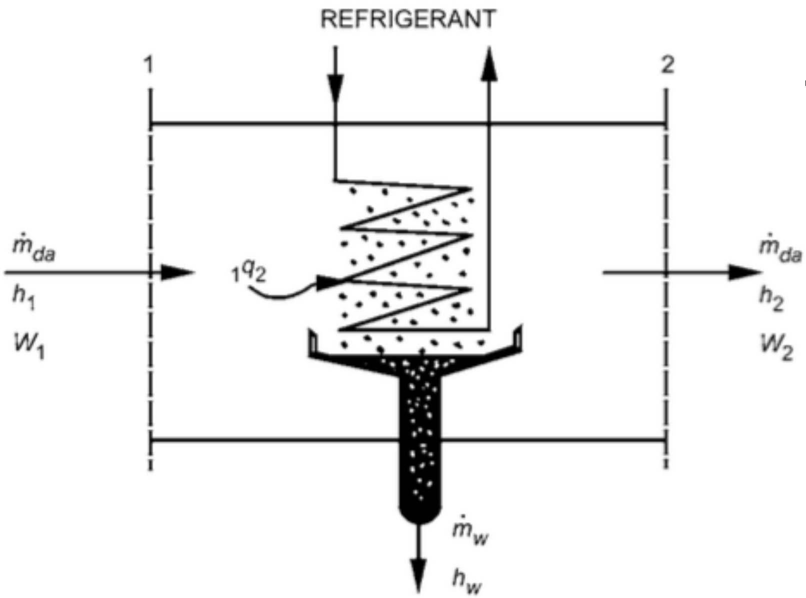
Energy balance: $\dot{m}_{da,1}h_1 + Q_{1\to 2} = \dot{m}_{da,2}h_2$

Mass balance on air: $\dot{m}_{da,1} = \dot{m}_{da,2} = \dot{m}_{da}$

Mass balance on water vapor: $\dot{m}_{da,1}W_1 = \dot{m}_{da,2}W_2$

Therefore: $Q_{1\to 2} = \dot{m}_{da}(h_2 - h_1)$

Energy/mass balances for psychrometric processes



Cooling and dehumidifying

*Note that $h_w = h_g$ for steam/vapor and $h_w = h_f$ for water

Energy balance: $\dot{m}_{da,1}h_1 + Q_{1\to 2} = \dot{m}_{da,2}h_2 + \dot{m}_w h_{w,2}$

Mass balance on air: $\dot{m}_{da,1} = \dot{m}_{da,2} = \dot{m}_{da}$

Mass balance on water vapor: $\dot{m}_{da,1}W_1 = \dot{m}_{da,2}W_2 + \dot{m}_w$

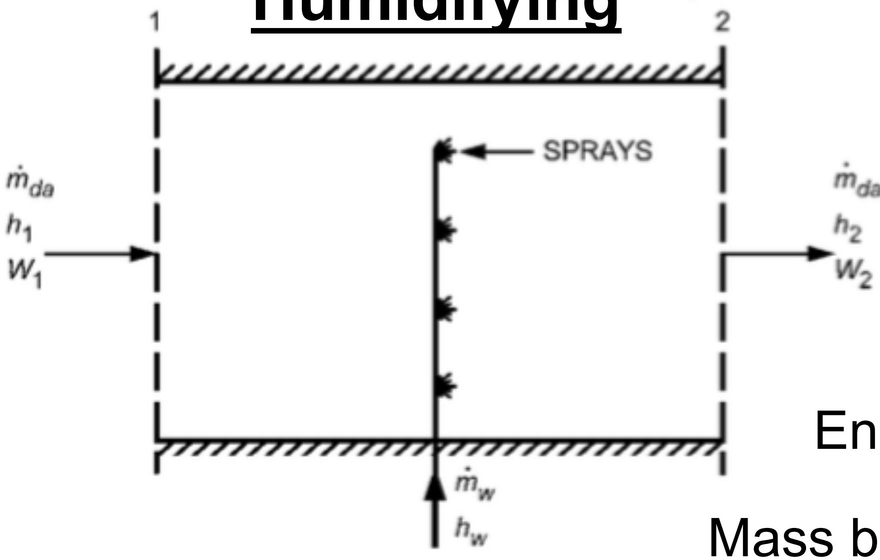
Therefore: $\dot{m}_w = \dot{m}_{da}(W_1 - W_2)$

And: $Q_{1\to 2} = \dot{m}_{da}[(h_2 - h_1) - (W_1 - W_2)h_{w,2}]$

(Q is negative for cooling)

Energy/mass balances for psychrometric processes

Humidifying



Energy balance: $\dot{m}_{da,1}h_1 + \dot{m}_w h_w = \dot{m}_{da,2}h_2$

Mass balance on air: $\dot{m}_{da,1} = \dot{m}_{da,2} = \dot{m}_{da}$

Mass balance on water vapor: $\dot{m}_{da,1}W_1 + \dot{m}_w = \dot{m}_{da,2}W_2$

Therefore: $\dot{m}_w = \dot{m}_{da}(W_2 - W_1)$

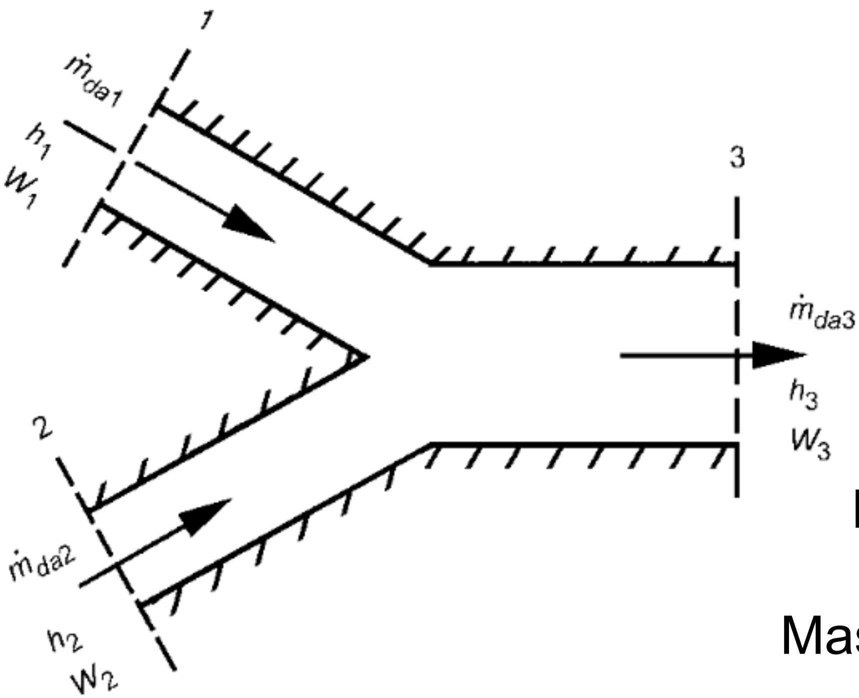
And: $\dot{m}_w h_w = \dot{m}_{da}(h_2 - h_1)$

And: $\dfrac{h_2 - h_1}{W_2 - W_1} = \dfrac{\Delta h}{\Delta W} = h_w$

*Note that h_w = h_g for steam/vapor and h_w = h_f for water

Energy/mass balances for psychrometric processes

- **Mixing**: Often in HVAC systems we mix airstreams adiabatically
  - **Adiabatically** = Without the addition or extraction of heat
  - e.g. outdoor air mixed with a portion of return/recirculated air



Energy: $\dot{m}_{da,1}h_1 + \dot{m}_{da,2}h_2 = \dot{m}_{da,3}h_3$

Mass (air): $\dot{m}_{da,1} + \dot{m}_{da,2} = \dot{m}_{da,3}$

Mass (water): $\dot{m}_{da,1}W_1 + \dot{m}_{da,2}W_2 = \dot{m}_{da,3}W_3$

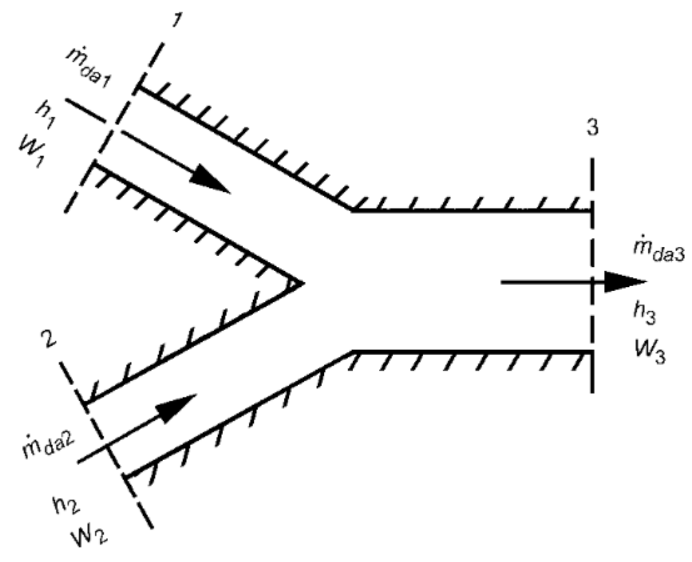
Energy/mass balances for psychrometric processes

- **Mixing:** For most parameters, the outlet conditions end up being the weighted averages of the input conditions based on their mass flow rates
  - Dry bulb temperature
  - Humidity ratio
  - Enthalpy
  - (not RH!)

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

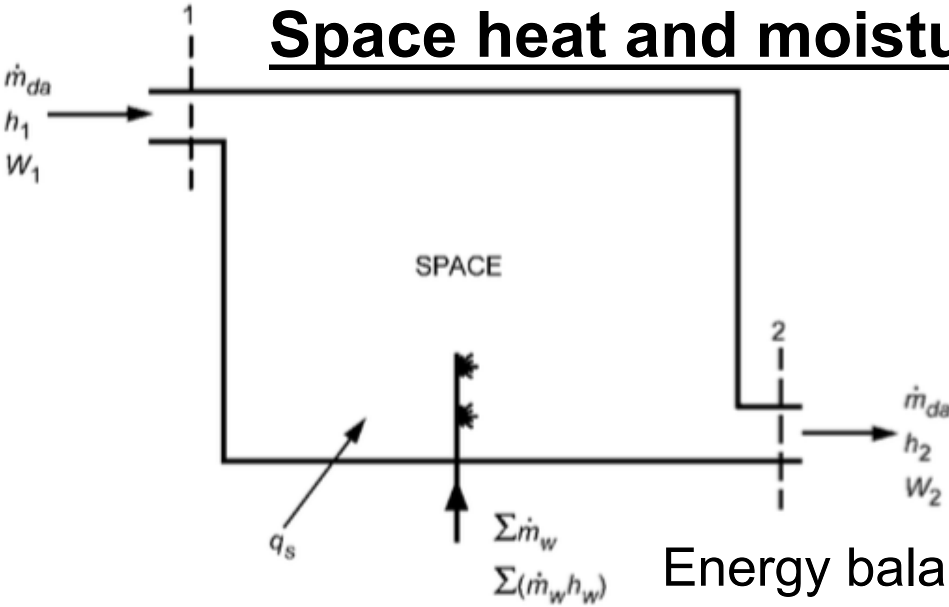
$$W_3 = \frac{\dot{m}_{da1} W_1 + \dot{m}_{da2} W_2}{\dot{m}_{da1} + \dot{m}_{da2}}$$

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



Energy/mass balances for psychrometric processes

Space heat and moisture gains



Energy balance: $\dot{m}_{da} h_1 + Q_{gains} + \sum \dot{m}_w h_w = \dot{m}_{da} h_2$

Mass balance on water vapor: $\dot{m}_{da} W_1 + \sum \dot{m}_w = \dot{m}_{da} W_2$

Therefore: $\sum \dot{m}_w = \dot{m}_{da}(W_2 - W_1)$

Therefore: $\sum \dot{m}_w h_w + Q_{gains} = \dot{m}_{da}(h_2 - h_1)$

And: $$\frac{\Delta h}{\Delta W} = \frac{\sum \dot{m}_w h_w + Q_{gains}}{\sum \dot{m}_w}$$

*Note that $h_w = h_g$ for steam/vapor and $h_w = h_f$ for water

Next time

- Psychrometric processes: example problems