ASHRAE

DL Presentation

Applied Psychrometric Processes related to Elements of Advanced HVAC System Design

Douglas F Zentz

Emeritus Professor

Ferris State University

This ASHRAE Distinguished Lecturer is brought to you by the ASHRAE Society Chapter Technology Transfer Committee

Lecturer presentations and/or opinions do not necessarily reflect the policies or position of ASHRAE or the chapter.

Please!

- Silence Phones
- Distinguished Lecturer Evaluation Forms are very important. Please complete at the end of the presentation and return to the CTTC or Program Chair.

More information on the DL program available at:

www.ashrae.org/distinguishedlecturers

LEADERSHIP WANTED!

www.ashrae.org/volunteer

BECOME A FUTURE LEADER IN ASHRAE – WRITE THE NEXT CHAPTER IN YOUR CAREER

ASHRAE Members who are active at their chapter and society become leaders and bring information and technology back to their job.

YOU ARE NEEDED FOR:

- Society Technical Committees
- Society Standard Committees
- Young Engineers in ASHRAE
- Chapter Membership Promotion
- Chapter Research Promotion
- Chapter Student Activities
- Chapter Technology Transfer

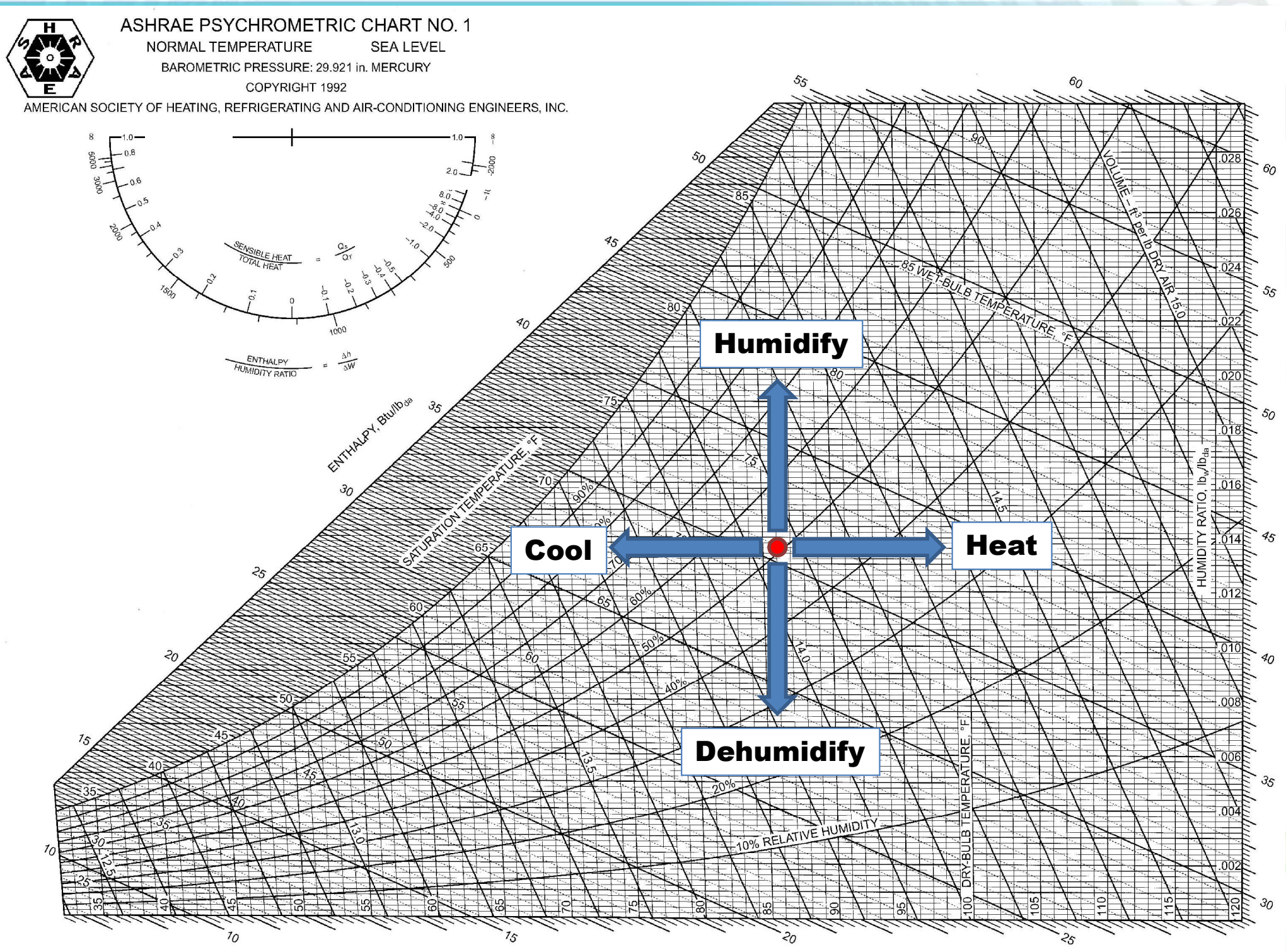
Find your Place in ASHRAE and volunteer

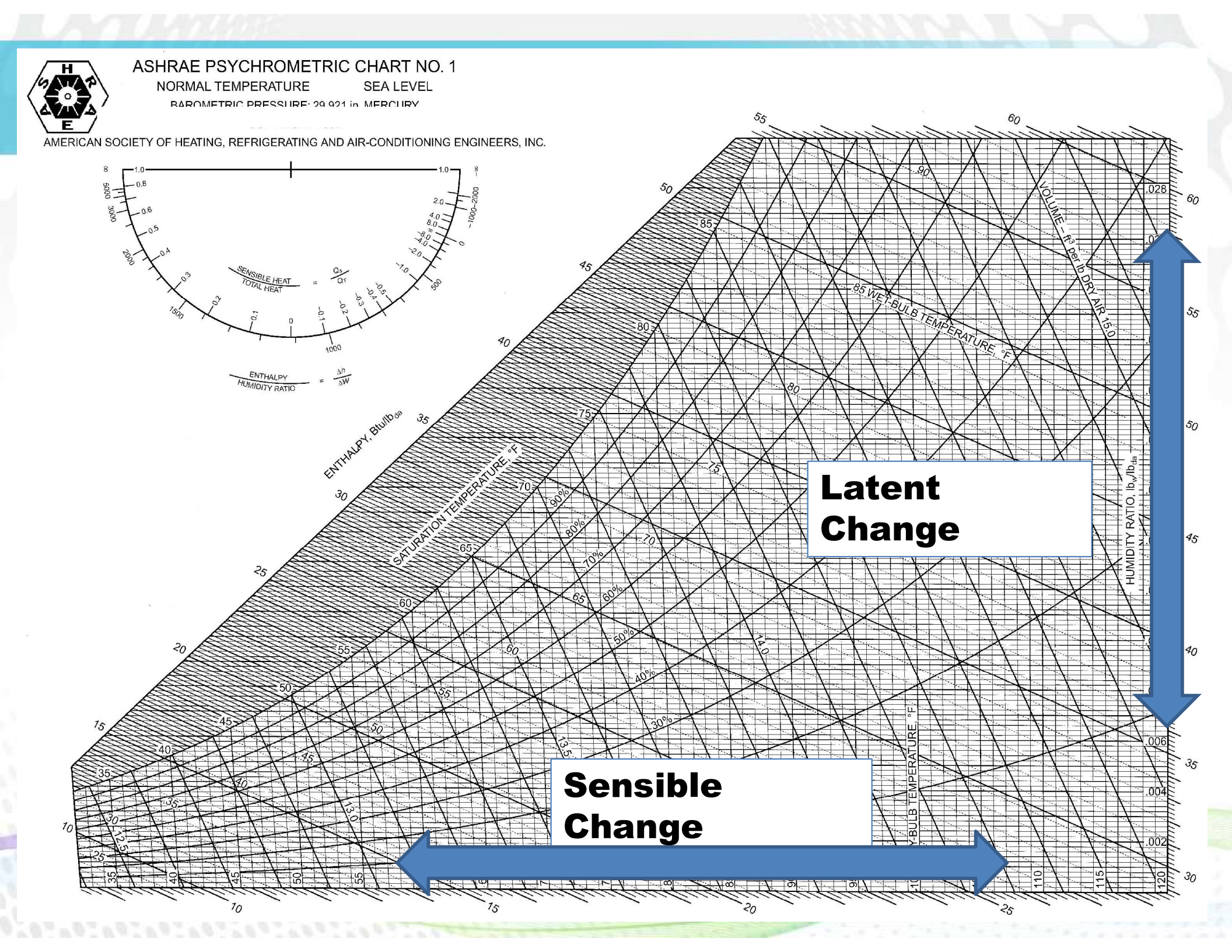
DL Presentation

- Commercialism – follow ASHRAE Policy
- This presentation contains my views
- Try to expand your mind
- Based on Experiences

Overview

- Analyzing the Psychrometric Processes of a Space at the ASHRAE HQ Building
- Utilizing DOAS to decouple the Latent Load and apply this effectively
- Integration of Heat and Energy Recovery
- High Performance HVAC and Psychrometric Processes
- Tips for High Performance AHU Systems


ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. MERCURY
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = Qs/Qt
ENTHALPY / HUMIDITY RATIO = Δh/ΔW
ENTHALPY, Btu/lb da
SATURATION TEMPERATURE, °F
HUMIDITY RATIO, lb w/lb da
DRY-BULB TEMPERATURE, °F
VOLUME – ft³ per lb DRY AIR
85 WET-BULB TEMPERATURE, °F
10% RELATIVE HUMIDITY
Humidify
Cool
Heat
Dehumidify

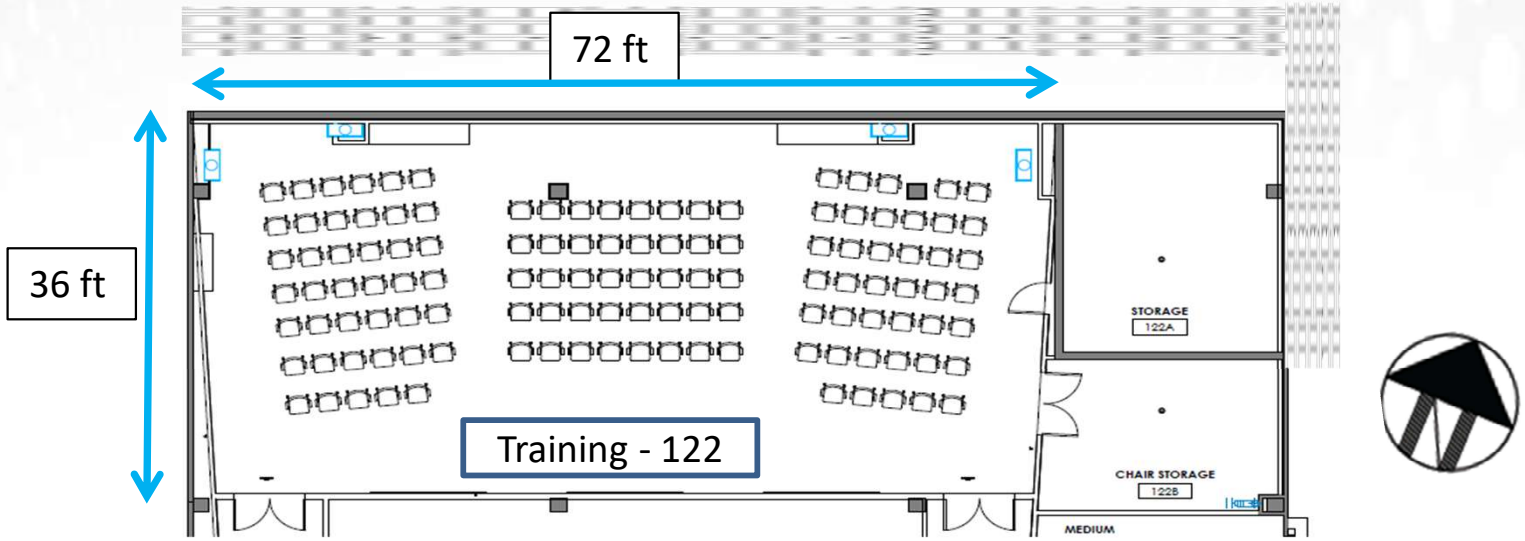

ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. MERCURY
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = Qs/Qt
ENTHALPY / HUMIDITY RATIO = Δh/ΔW
ENTHALPY, Btu/lb da
SATURATION TEMPERATURE, °F
HUMIDITY RATIO, lbw/lbda
VOLUME – ft³ per lb DRY AIR
85 WET-BULB TEMPERATURE, °F
Latent Change
Sensible Change

Movement on Psych Chart

- **Combination of direction more common**

- **Typical cooling process:**
  - Decreases the air temperature (satisfy thermostat) and removes moisture (condensate)
  - Movement on chart: left and down

ASHRAE HQ Training Room



Area

$A_{wall\text{-}SW}$ = 468 ft²

$A_{wall\text{-}NW}$ = 360 ft²

Thermal Transfer

U_{wall} = 0.055 Btuh/(ft²·°F)

$U_{avg\ bsmt}$ = 0.049 Btuh/(ft²·°F)

F_p = 0.53 Btuh/(ft·°F)

Wall Conductive Loads

Table 8.3B Cooling Load Temperature Differences for Calculating Cooling Load from Sunlit Walls—36° North Latitude, July (*Concluded*)

Wall Facing	Wall No. 15												Solar Time, h											
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
N	19	18	16	14	12	11	9	7	6	6	6	6	7	8	10	11	13	15	17	19	20	21	21	20
NE	21	19	17	15	13	11	9	8	7	9	11	14	18	21	23	24	25	26	27	27	27	26	25	23
E	24	22	19	17	15	12	10	9	8	10	13	18	22	27	30	32	33	34	34	33	32	31	29	27
SE	24	22	19	17	14	12	10	8	8	8	10	13	17	21	25	28	30	31	32	32	31	30	28	26
S	23	20	18	16	14	12	10	8	7	5	5	5	7	9	12	15	19	22	25	27	27	27	26	25
SW	33	30	27	24	21	18	15	12	10	9	8	7	7	8	10	13	17	22	27	32	35	37	37	36
W	38	35	31	28	24	21	17	14	12	10	9	8	8	9	10	12	16	21	26	32	38	41	42	41
NW	31	29	26	23	20	17	14	12	10	8	7	7	7	8	9	11	14	17	21	26	30	33	34	33

$$q_{sw,wall} = q_{n,wall} = U \times A \times CLTD_{wall}$$

Use N wall for NW since there is no sun on NW wall

Hour 20: $q_{sw,wall} = 0.055 \times 468 \times 32 = 824$ Btuh

Hour 20: $q_{nw,wall} = 0.055 \times 360 \times 26 = 515$ Btuh

1339 Btuh

Internal Heat Gains

- **People – 65 People/1000 ft^2 x 2,592 ft^2 = 169 People**
  - **Q_S = 169 P x 245 Btuh/P = 41,405 Btuh**
  - **Q_L = 169 P x 155 Btuh/P = 26,195 Btuh**
- **Equipment – 1 Laptop & 3 Projectors**
  - **Q_S = [40 W + (3 x 308 W)] x 3.412 Btuh/W = 3,289 Btuh**
- **Lights – Light 1: 26 W (12) Light 2: 30.4 W (11)**
  - **Q_S = [(26 x 12) + (30.4 x 11)] W x 3.412 Btuh/W**
  - **Q_S = 2,206 Btuh**

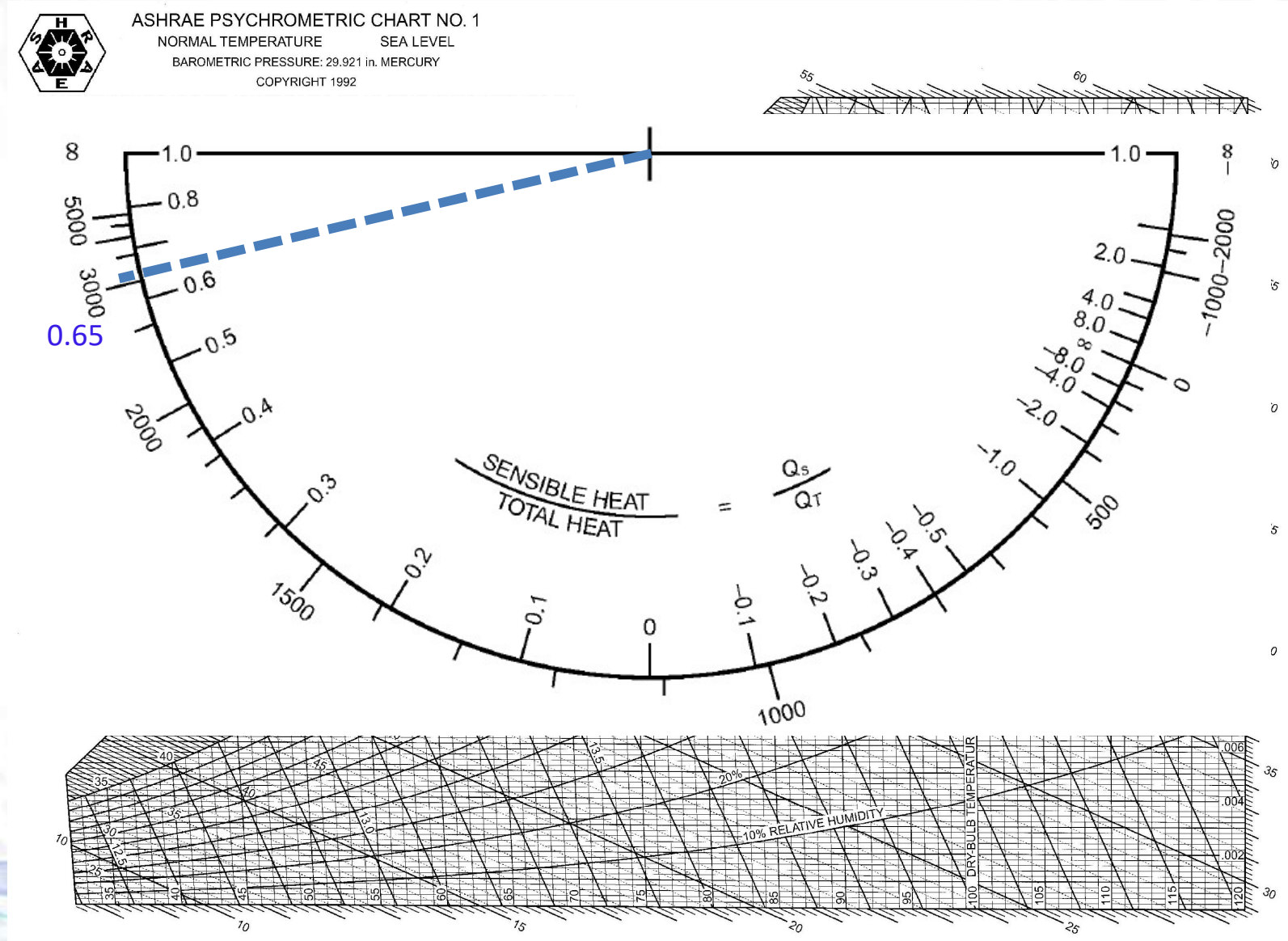
Training Room Total Loads

- **Sensible Load**
  - Q_S = 41,405 (people) + 1,339 (wall) + 3,289 (equipment) + 2,206 (lights)
  - Q_S = 48,239 Btuh
- **Latent Load**
  - Q_L = 26,195 Btuh
- **Total Load**
  - Q_T = 48,239 Btuh + 26,195 Btuh = 74,434 Btuh

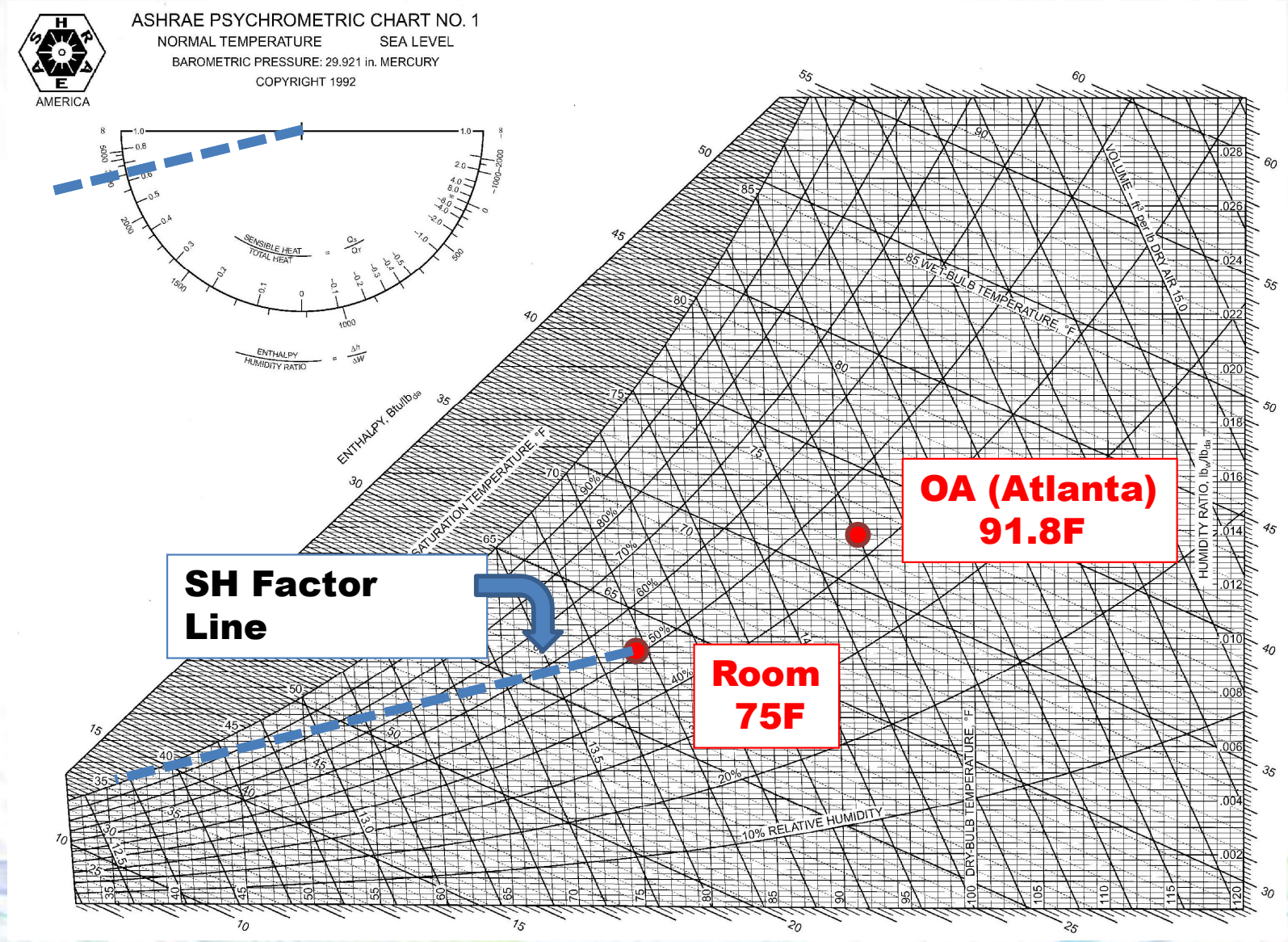
Typical Space SHR

- **Space SHR = Space Sensible/Space Total**
- **Space SHR = 48,239 Btuh/74,434 Btuh**
- **Space SHR = 0.65 or 65%**

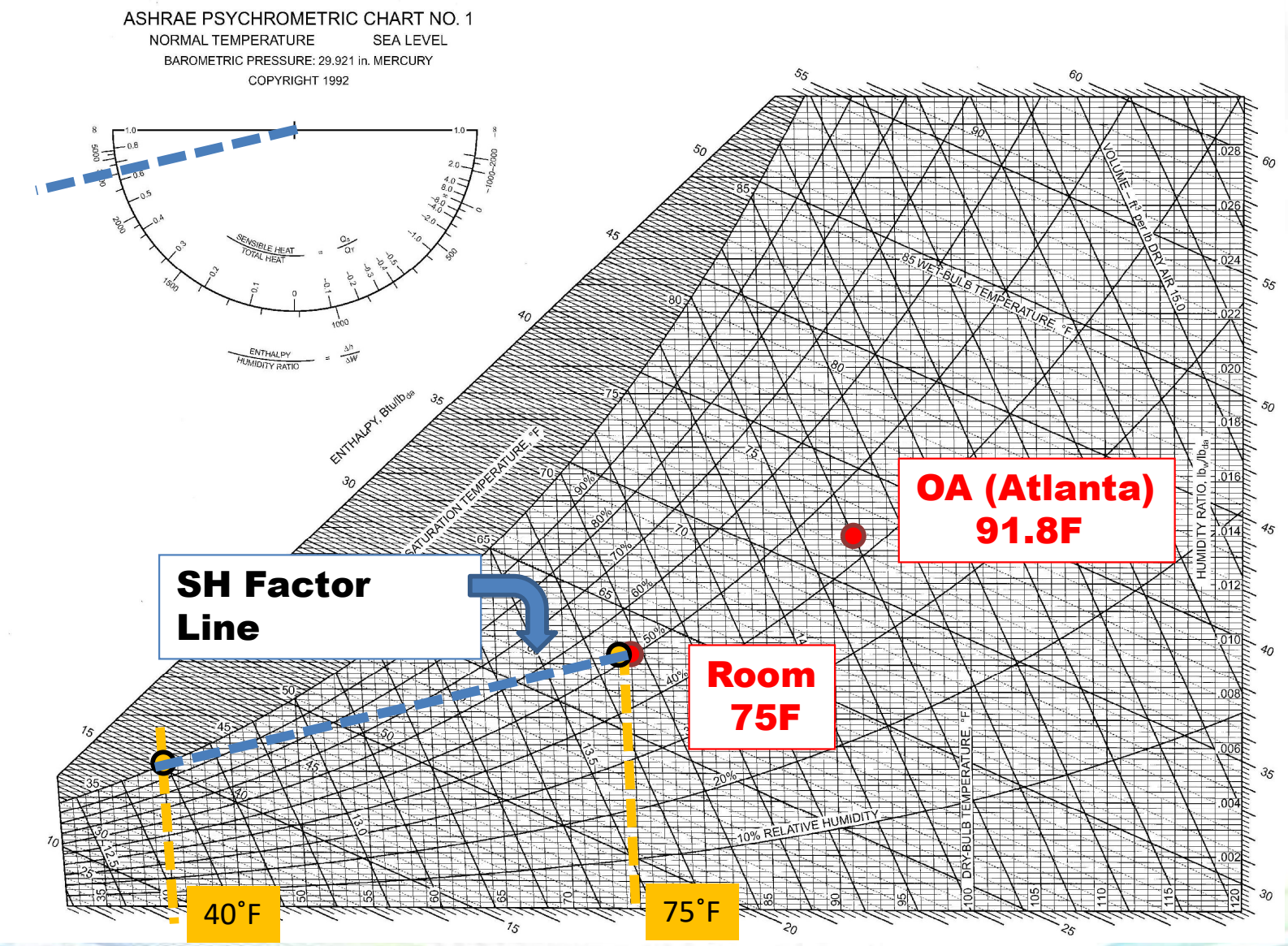
Sensible Heat Ratio Process



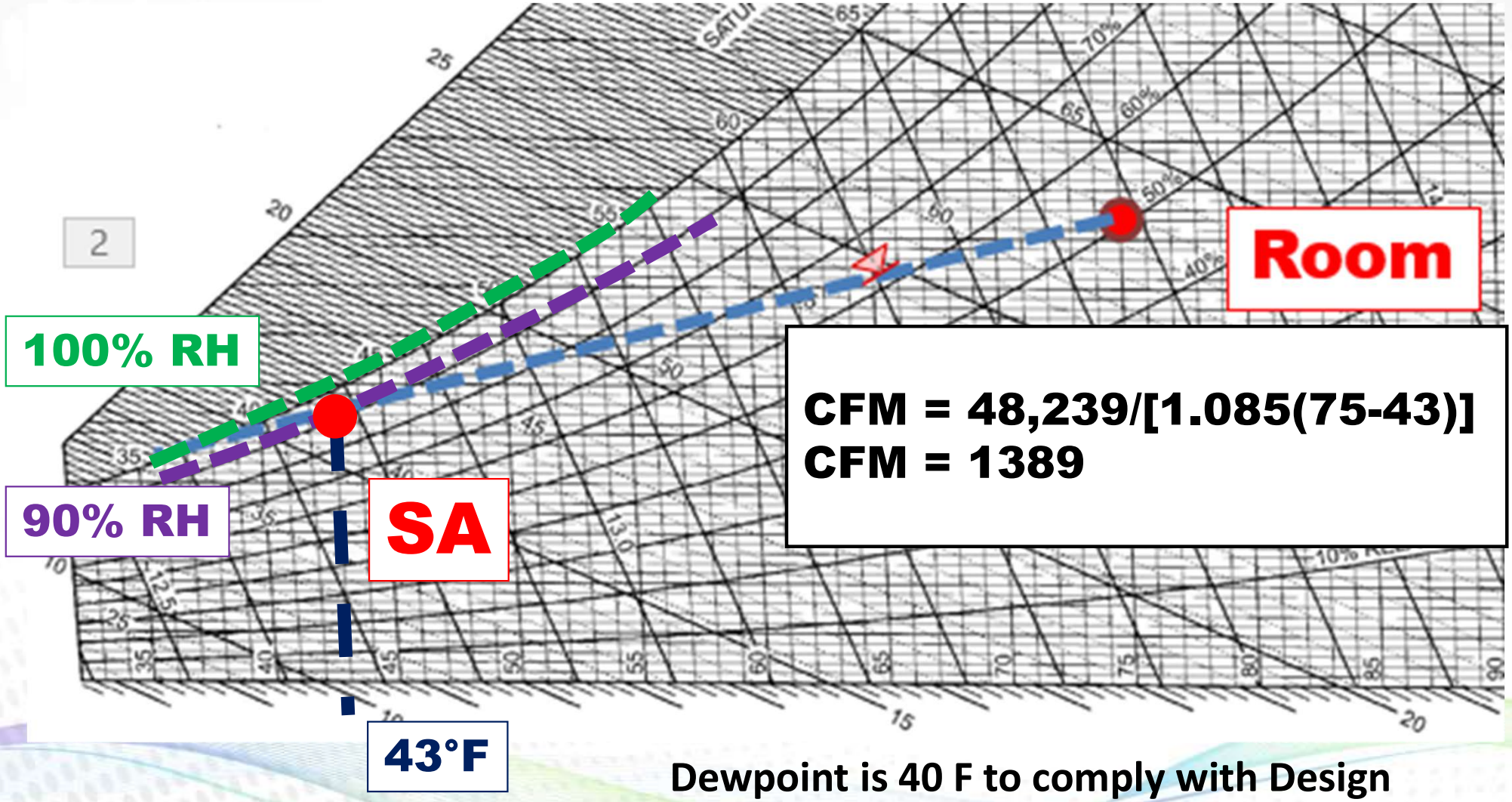
65% SHR Psychrometric Layout



Typical SHR Line to Determine Leaving Air

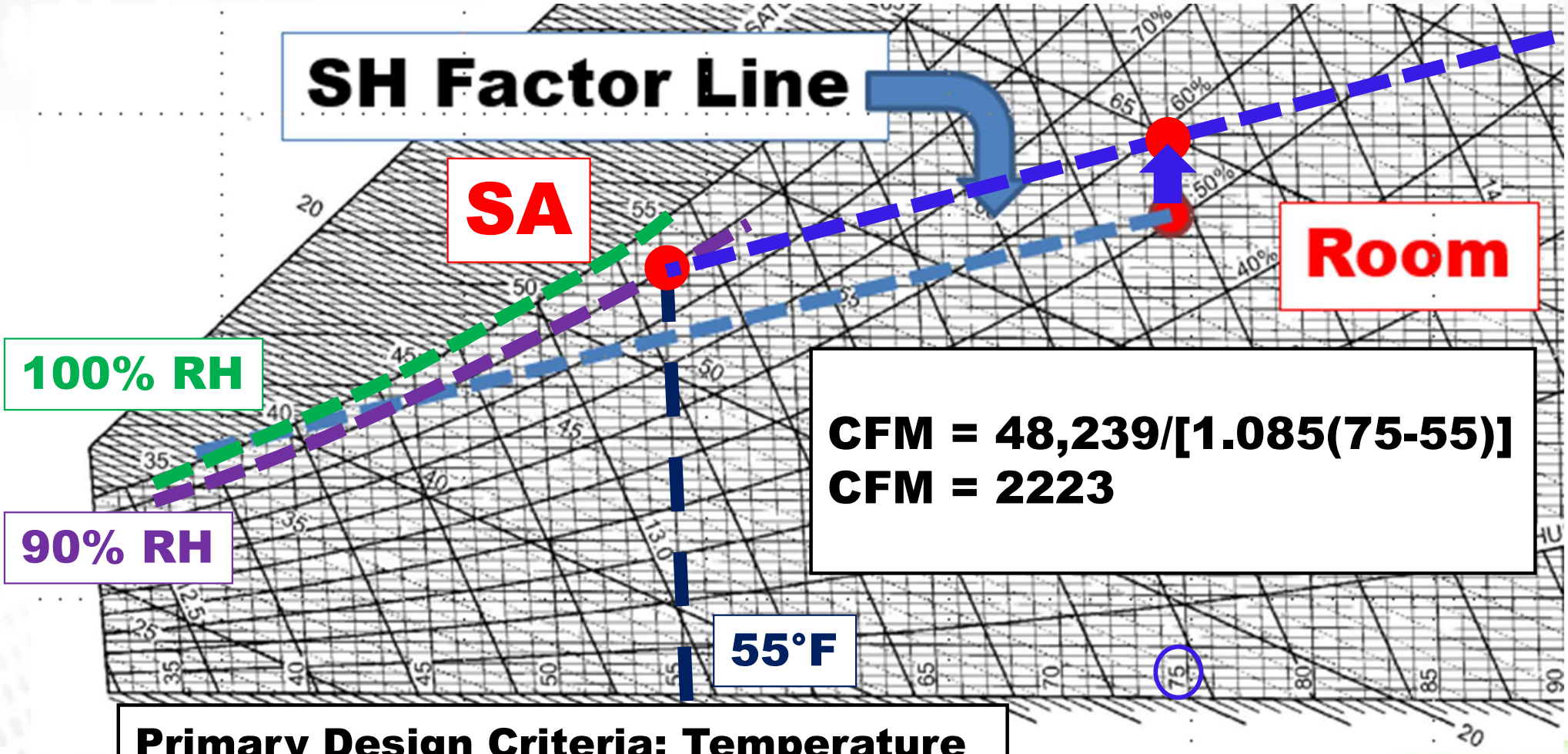


Airflow from Typical Approach

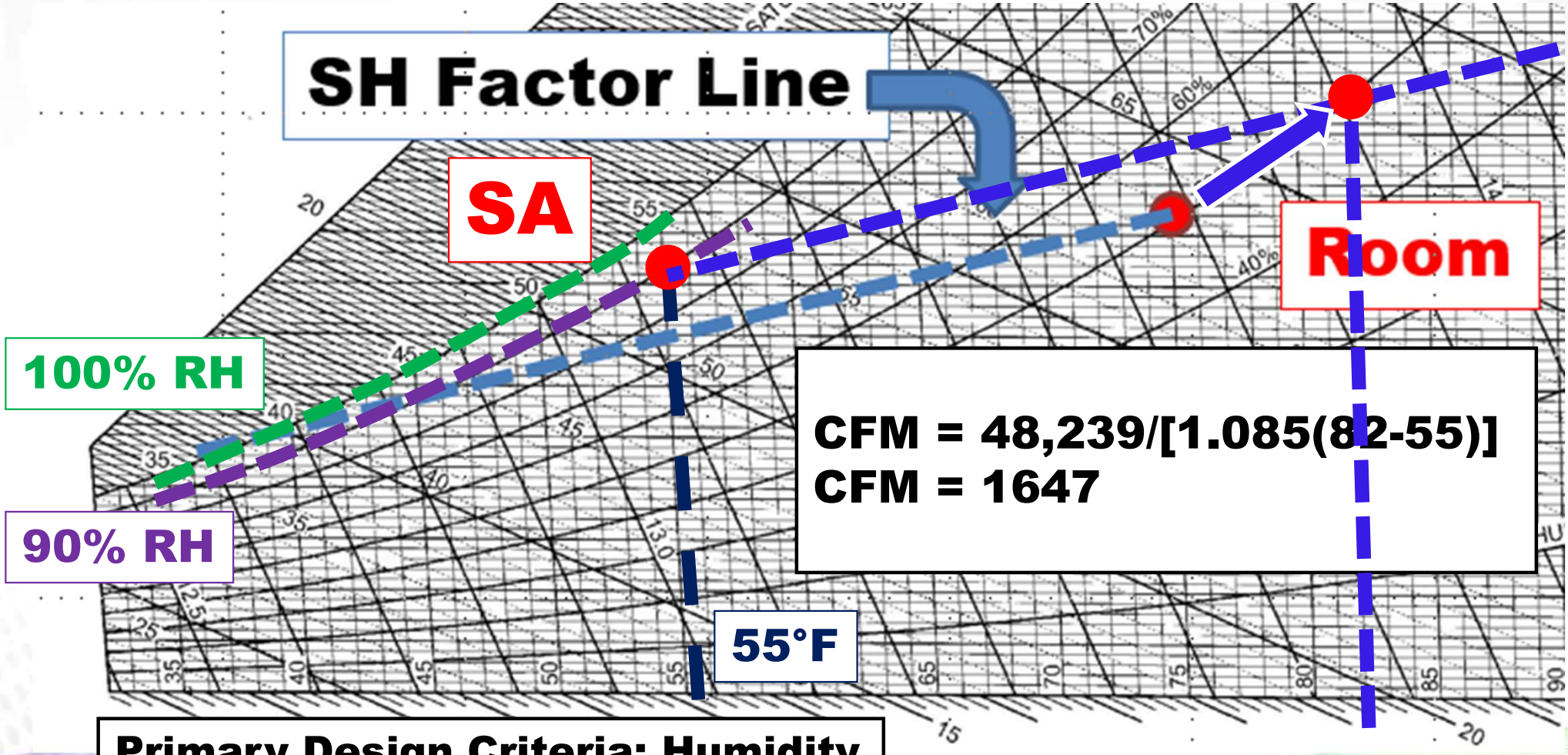


Dewpoint is 40 F to comply with Design

Using 55 F Leaving Air



Keeping 50% RH Maximum



Primary Design Criteria: Humidity
Temperature = 82°F > 75°F

Ventilation Requirements

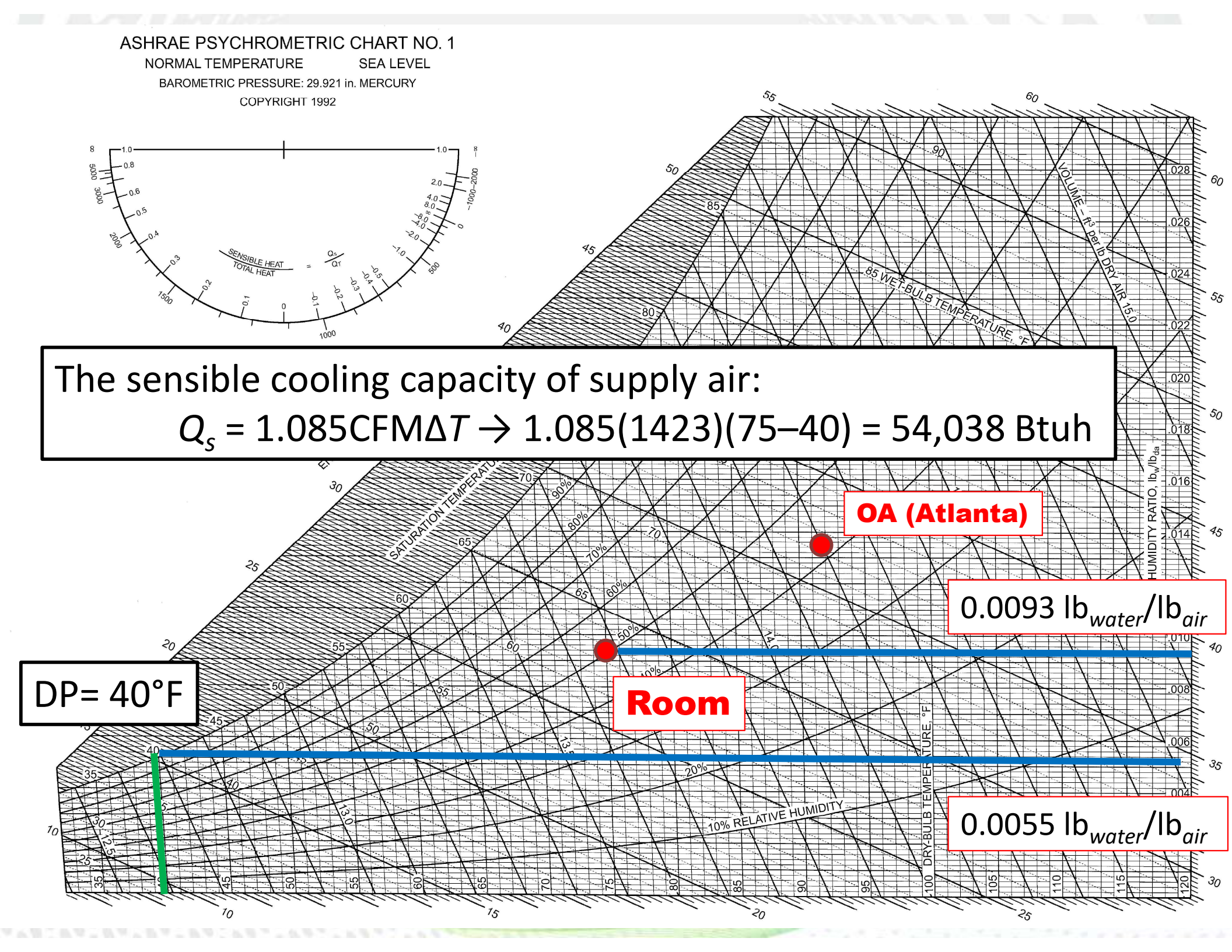
- **Ventilation Rate Equation**
  - $V = (R_P \times P_Z) + (R_A \times A_Z)$
- **Per Standard 62.1**
  - $R_P = 7.5$ CFM/person & $R_A = 0.06$ CFM/ft^2
- **Space Information**
  - $P_Z = 169$ people & $A_Z = 2{,}592$ ft^2
- **Space Ventilation Need**
  - $V = (7.5 \times 169) + (0.06 \times 2{,}569) = 1{,}423$ CFM

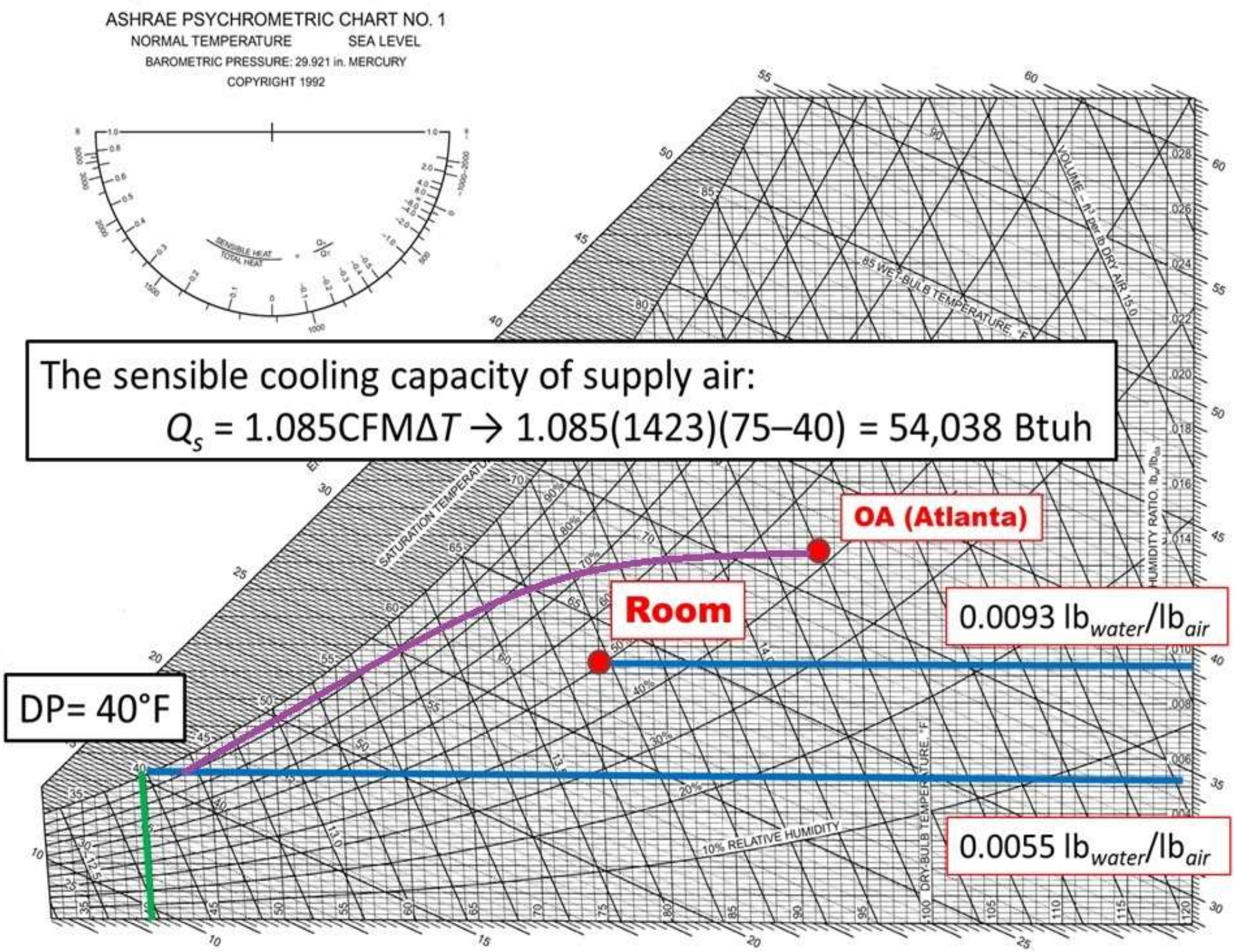
Typical HVAC Thinking

- **Due to SHR being 65%**
  - **Standard 55F Design results in Space Design 59% RH**
  - **OR**
  - **Increased Space Design Set-Point of 82 F**
  - **OR**
  - **Something Different!**
- **Percent Outdoor Air?**
  - **First Calculation: 1,423 CFM/1,389 CFM = 102%**
  - **Second Calculation: 1,423 CFM/2,223 CFM = 64%**
  - **Third Calculation: 1,423 CFM/1,647 CFM = 86%**

DOAS Design

- **Utilize the Standard 62.1 Ventilation Airflow**
  - Calculated 1,423 CFM per Ventilation Rate Equation
- **Use this airflow to "Decouple the Latent Load"**
  - Space Latent Load = 26,195 Btuh
- **Latent Airflow Equation**
  - Q_L = 4,840 x CFM x ΔW
  - ΔW = Q_L /(4,840 x CFM)
  - ΔW = 26,195 Btuh/(4,840 x 1,423 CFM) = 0.0038
- **Required Supply Absolute Humidity**
  - Leaving W = Space W – ΔW = 0.0093 – 0.0038 = 0.0055


ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. MERCURY
COPYRIGHT 1992
SENSIBLE HEAT / TOTAL HEAT = Qs/Qt
VOLUME – ft³ per lb DRY AIR
85 WET-BULB TEMPERATURE, °F
SATURATION TEMPERATURE, °F
HUMIDITY RATIO, lbw/lbda
DRY-BULB TEMPERATURE, °F
10% RELATIVE HUMIDITY
The sensible cooling capacity of supply air:
$Q_s = 1.085CFM\Delta T \rightarrow 1.085(1423)(75–40) = 54{,}038$ Btuh
OA (Atlanta)
0.0093 lb$_{water}$/lb$_{air}$
Room
0.0055 lb$_{water}$/lb$_{air}$
DP= 40°F


ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. MERCURY
COPYRIGHT 1992
SENSIBLE HEAT / TOTAL HEAT
VOLUME – ft³ Per lb DRY AIR
85 WET-BULB TEMPERATURE, °F
SATURATION TEMPERATURE, °F
HUMIDITY RATIO, lb_w/lb_da
DRY-BULB TEMPERATURE, °F
10% RELATIVE HUMIDITY
The sensible cooling capacity of supply air:
$Q_s = 1.085CFM\Delta T \rightarrow 1.085(1423)(75–40) = 54{,}038$ Btuh
OA (Atlanta)
Room
0.0093 lb$_{water}$/lb$_{air}$
DP= 40°F
0.0055 lb$_{water}$/lb$_{air}$

Coil Load

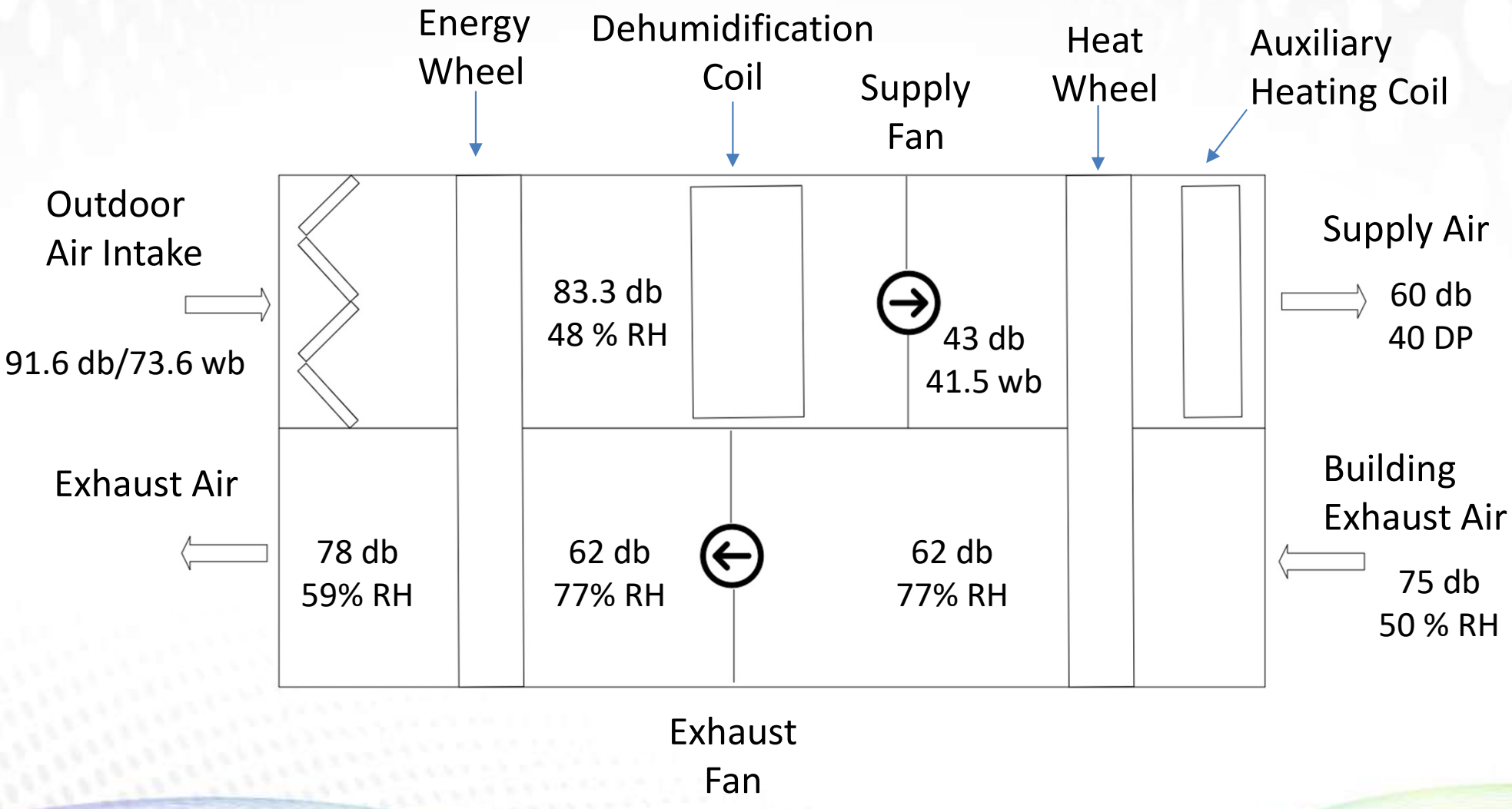
- **Entering Air Conditions**
  - **91.6 F DB / 73.6 F WB / 37.6 Btu/lbs / 45.4% RH**
- **Leaving Air Conditions**
  - **43.0 DB / 41.5 F WB / 16.0 Btu/lbs / 90% RH**
- **Coil Load**
  - **Q_T = 4.5 x CFM x ΔH**
  - **Q_T = 4.5 x 1,423 x (37.6 – 16.0)**
  - **Q_T = 138,316 Btuh**

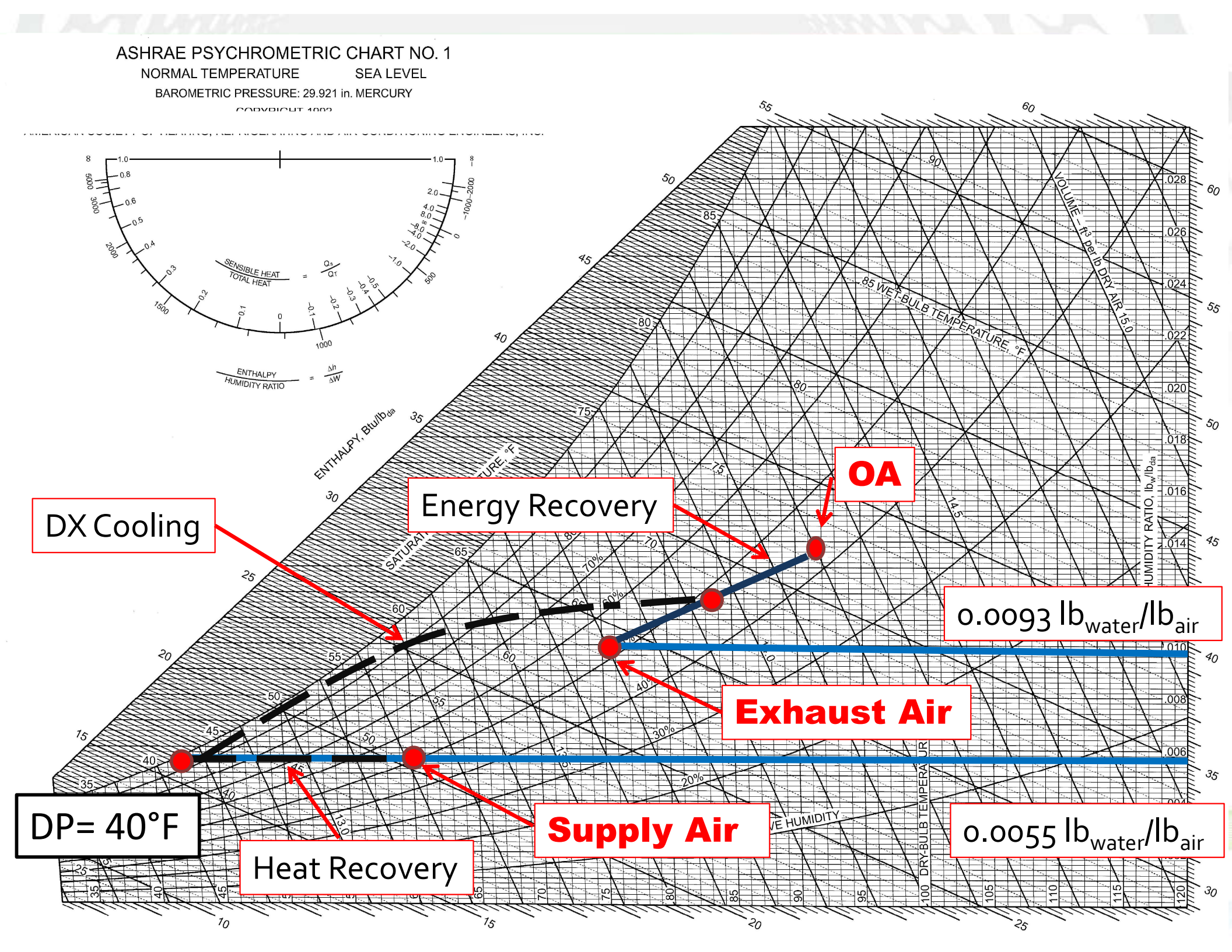
Heat and Energy Wheels

- **Need for an Energy Recovery Wheel to Precondition the Entering Outdoor Airflow**
  - **Per Standard 90.1: assume 50% efficiency via using building exhaust airflow**

- **Good Idea for a Heat Wheel to use Exhaust Airflow to Reheat Leaving Air from the Dehumidification Coil as first-stage of Reheat**
  - **Assume 40% efficiency with sensible energy**

DOAS with Dual Recovery Wheels

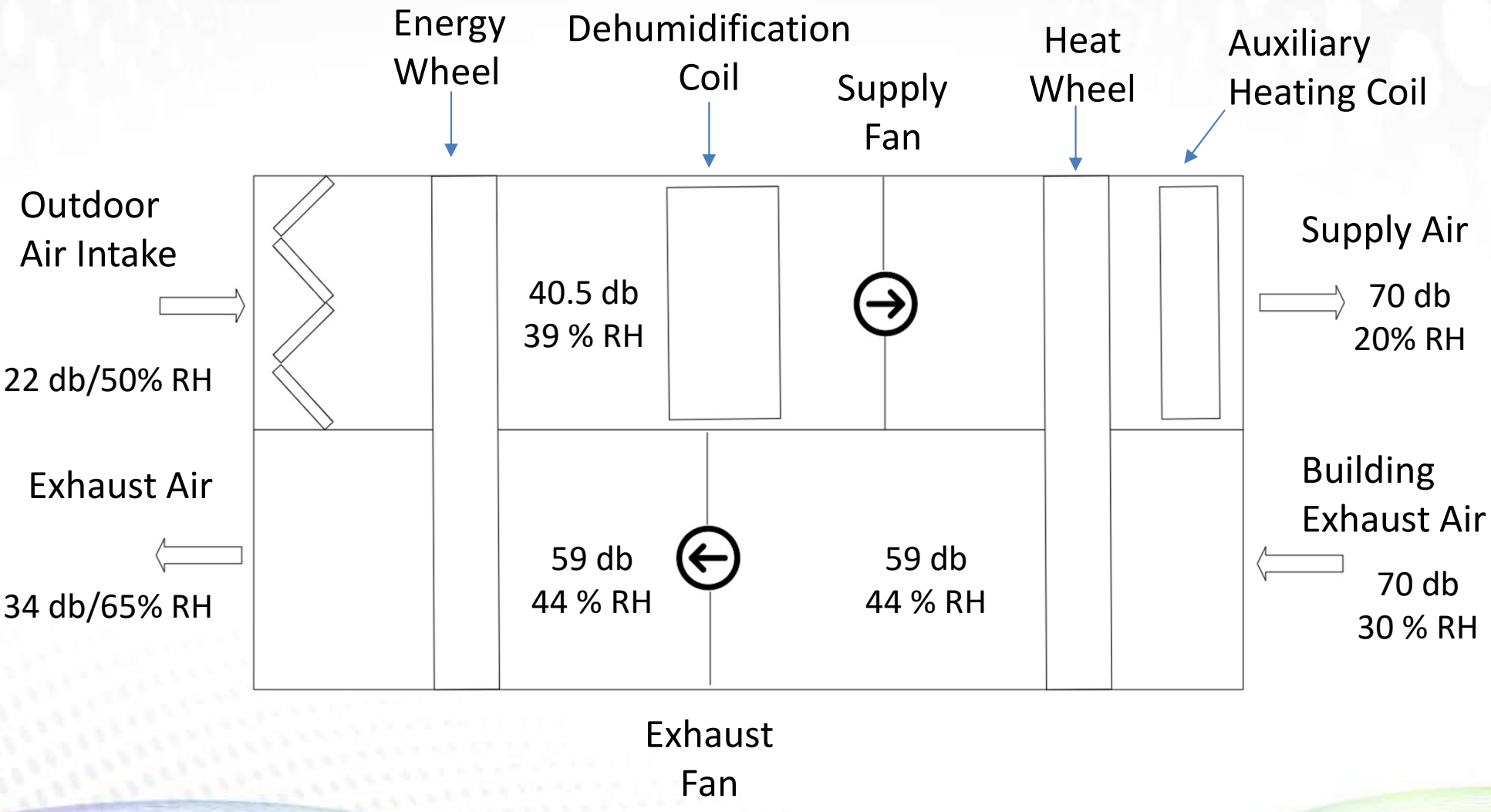


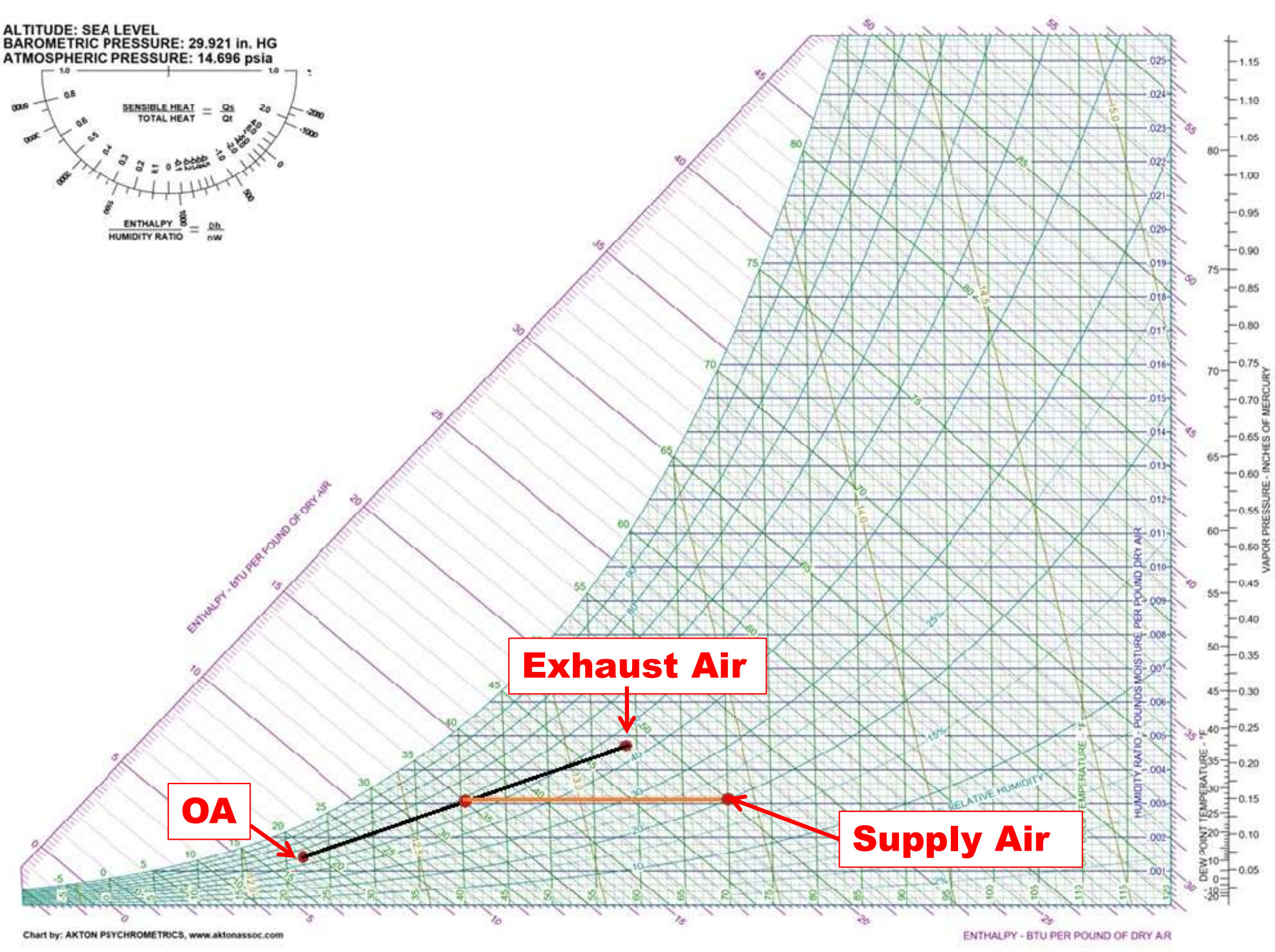

ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. MERCURY
SENSIBLE HEAT / TOTAL HEAT = Qs/Qt
ENTHALPY / HUMIDITY RATIO = Δh/ΔW
ENTHALPY, Btu/lbda
HUMIDITY RATIO, lbw/lbda
DRY-BULB TEMPERATURE, °F
85 WET-BULB TEMPERATURE, °F
VOLUME - ft3 per lb DRY AIR
OA
Energy Recovery
DX Cooling
0.0093 lbwater/lbair
Exhaust Air
DP= 40°F
Supply Air
Heat Recovery
0.0055 lbwater/lbair

Coil Load

- **Entering Air Conditions**
  - **83.3 F DB / 48% RH/ 32.6 Btu/lbs / 48% RH**
- **Leaving Air Conditions**
  - **43.0 DB / 41.5 F WB / 16.0 Btu/lbs / 90% RH**
- **Coil Load**
  - **Q_T = 4.5 x CFM x ΔH**
  - **Q_T = 4.5 x 1,423 x (32.6 – 16.0)**
  - **Q_T = 106,298 Btuh**
- **Savings = 138,316 – 106,298 = 32,018 Btuh**

DOAS in Winter Mode

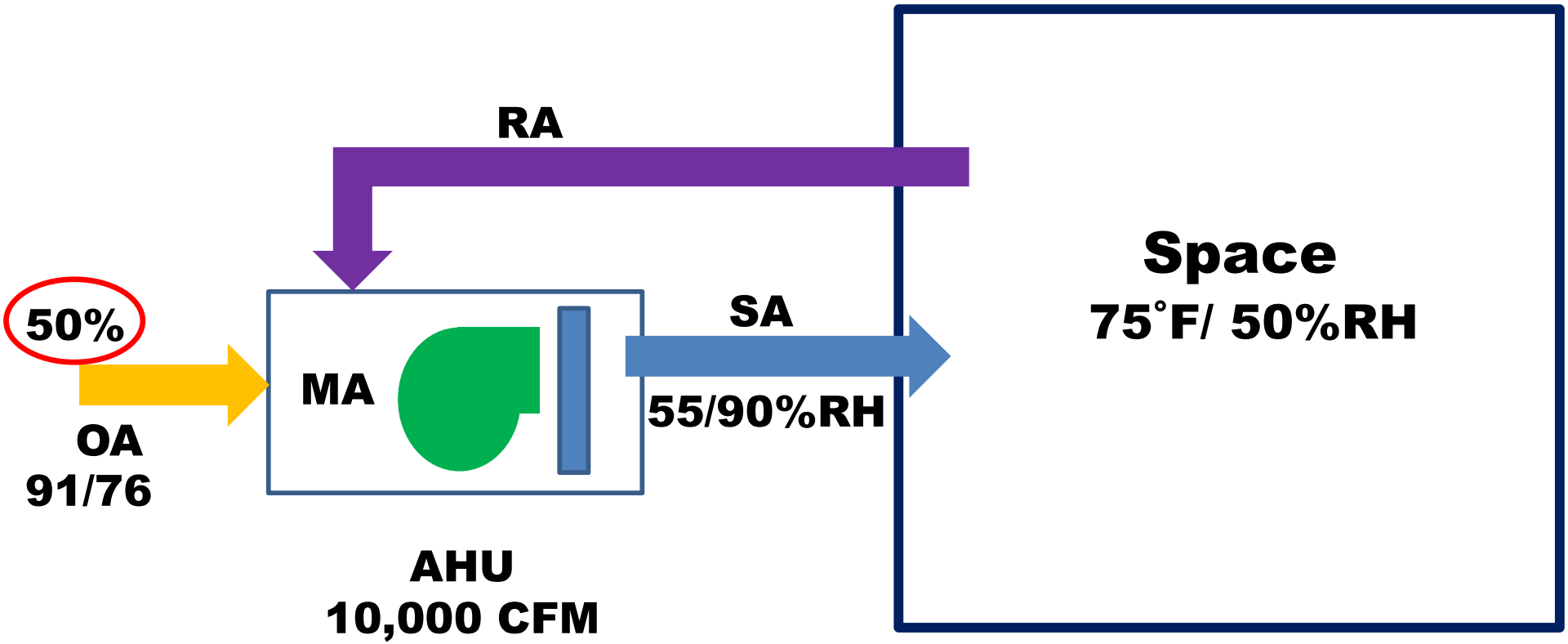



ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Dh / nW
Exhaust Air
OA
Supply Air
ENTHALPY - BTU PER POUND OF DRY AIR
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DEW POINT TEMPERATURE - °F
VAPOR PRESSURE - INCHES OF MERCURY
RELATIVE HUMIDITY
Chart by: AKTON PSYCHROMETRICS, www.aktonassoc.com

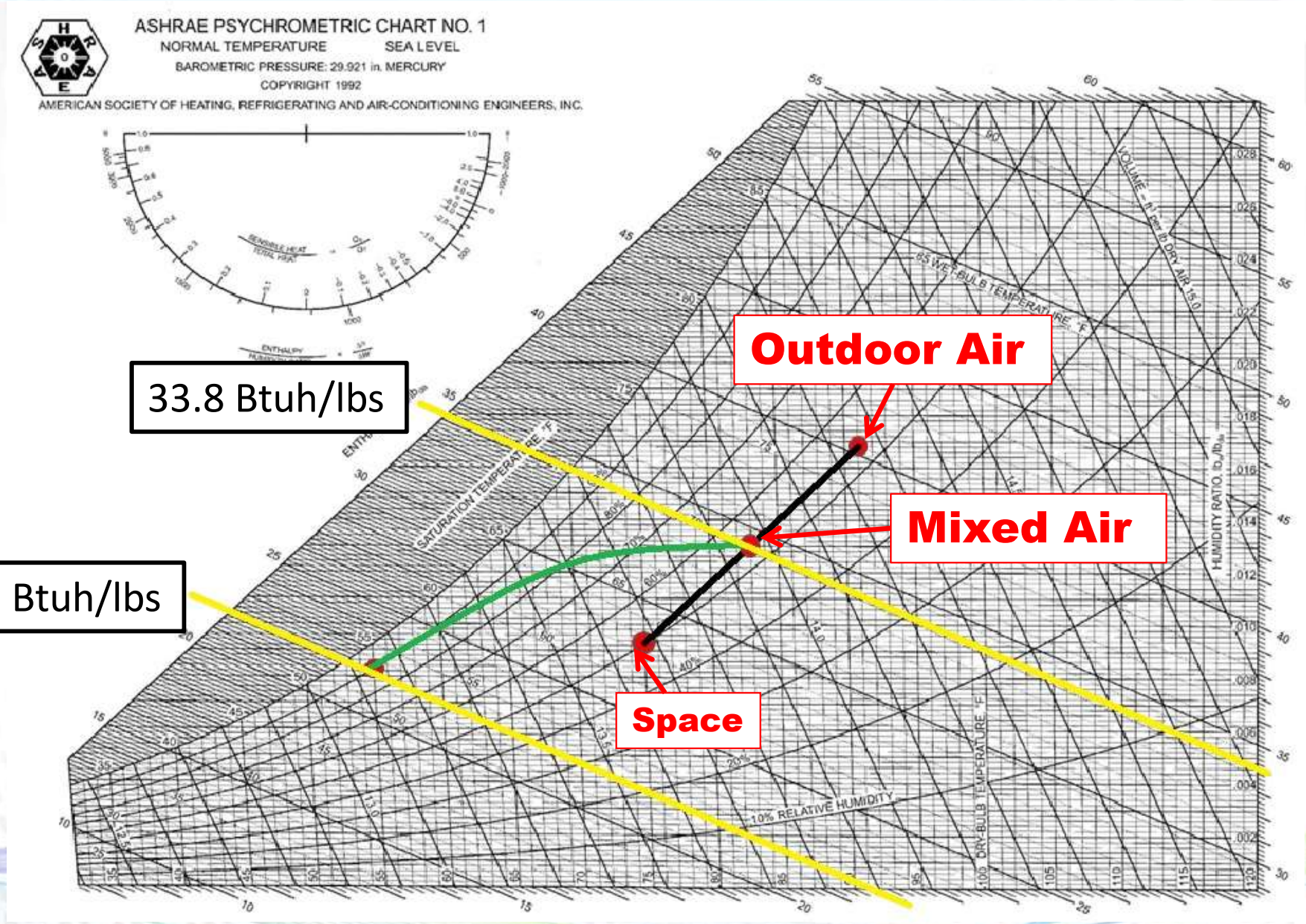
Real Elements of HVAC

- System Heat Gain
  - Return Path Heat Gain
    - Internal energy from Lights, Equipment and People due to convection
    - Elements within the Return Air Path due to conduction and negative pressure in the return air ductwork
- AHU Fans/Motors
  - Calculated BHP power going to the airflow due to motor inefficiency

Example 1



Normal Psychrometric Layout



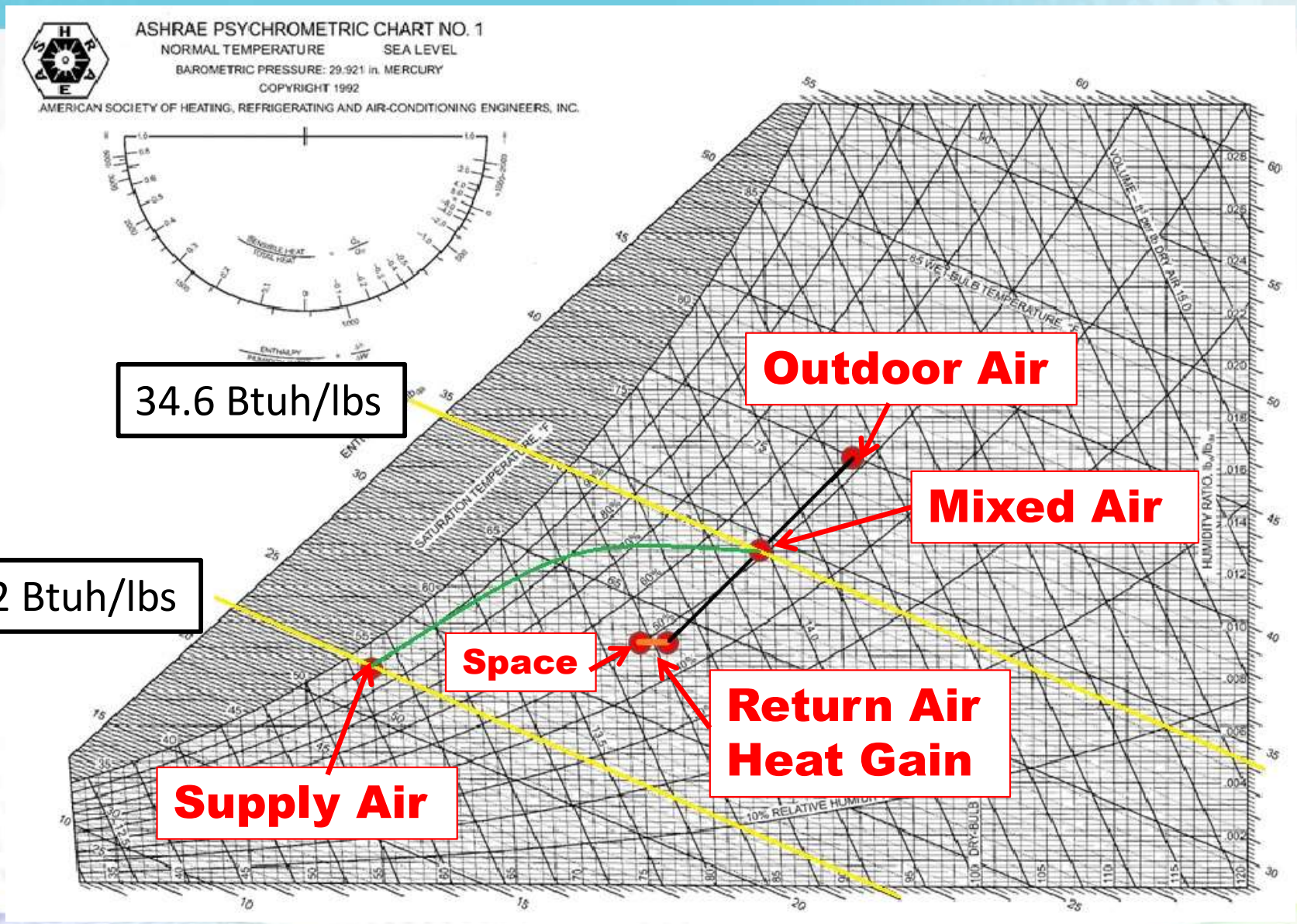
Example 1 Base Coil Load

- **Entering Air Conditions**
  - **83.3 F DB / 69.0 F WB/ 33.8 Btu/lbs / 52.5% RH**
- **Leaving Air Conditions**
  - **55.0 DB / 53.0 F WB / 22.2 Btu/lbs / 90% RH**
- **Coil Load**
  - **$Q_T = 4.5 \times CFM \times \Delta H$**
  - **$Q_T = 4.5 \times 10{,}000 \times (33.8 - 22.2)$**
  - **$Q_T = 522{,}000$ Btuh**

Example 1 Heat Gain to Return Air

- **Element 1 is due to ductwork leakage, there is a 10% introduction of plenum air that is 10 F warmer than the space conditions**
  - **T_{mix} = (0.90 x 75) + (0.10 x 85) = 67.5 + 8.5 = 76 F**
- **Element 2 is from conductive heat transfer from the plenum to the return airflow, and from an analysis of this load it is determined that 1.0 F increase in airflow temperature will occur**
- **Total Heat Gain to the Return Air results in 2.0 F increase**

Psychrometric Layout with Heat Gain



Example 1 Coil Load with RA Heat Gain

- **Entering Air Conditions**
  - **84.0 F DB / 69.8 F WB/ 34.5 Btu/lbs / 52.0% RH**
- **Leaving Air Conditions**
  - **55.0 DB / 53.0 F WB / 22.2 Btu/lbs / 90% RH**
- **Coil Load**
  - **Q_T = 4.5 x CFM x ΔH**
  - **Q_T = 4.5 x 10,000 x (34.5 – 22.2)**
  - **Q_T = 553,500 Btuh**
  - **Increased Load = 553,500 – 522,000 = 31,500 Btuh**

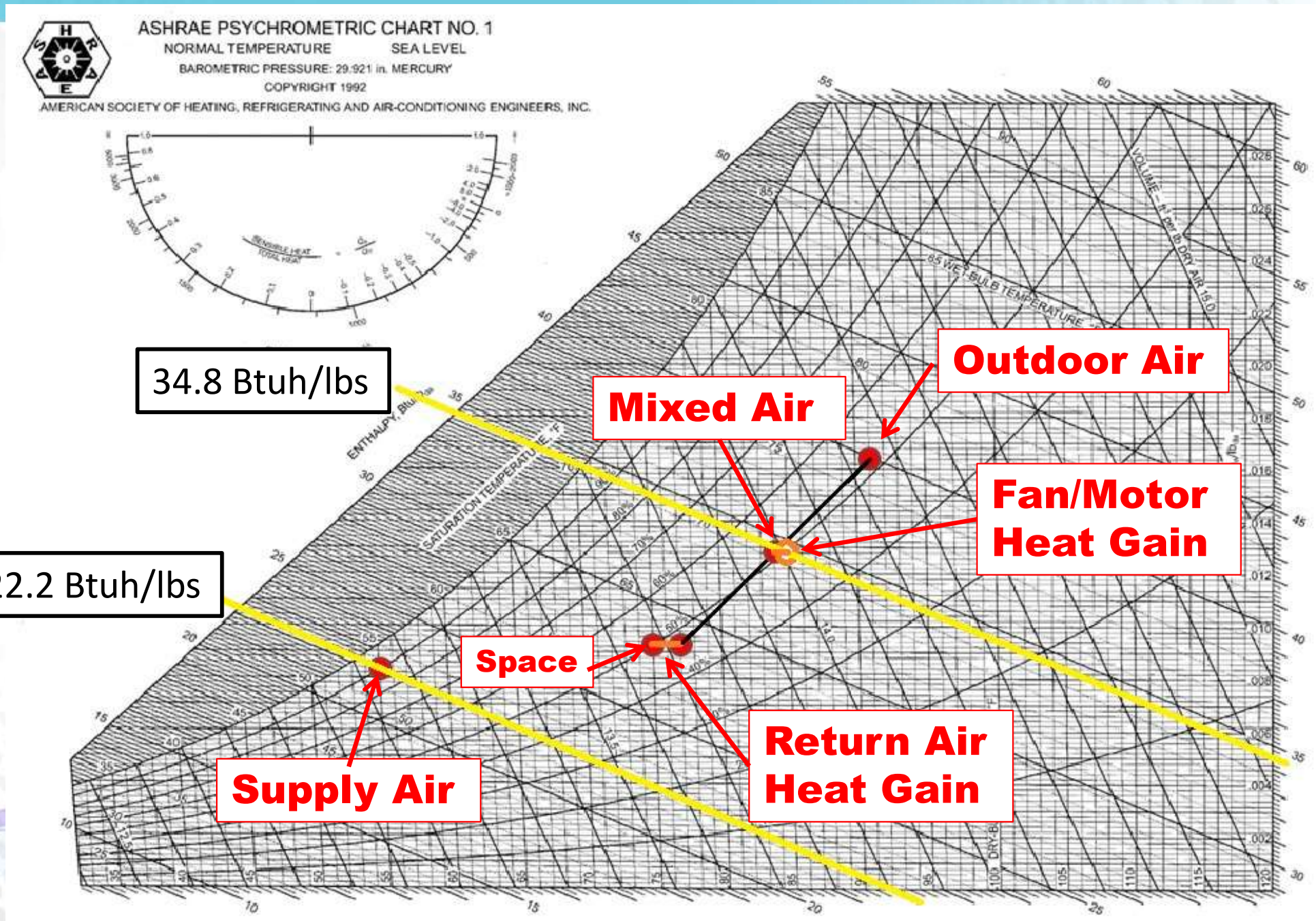
Example 2 Fan/Motor Heat Gain

- **Equation for Fan BHP**
  - BHP = (airflow x static pressure)/(6356 x fan efficiency)
  - Assumptions
    - Fan total static pressure is 4” Static Pressure
    - Fan Efficiency is 60%
  - BHP = (10,000 x 4)/6356 x 0.6) = 10.5
- **Heat Generation (assume 90% efficient motor)**
  - Inefficiency Heat is 10% of 10.5 BHP
  - Btuh = 1.5 BHP x 0.745 kW/1 BHP x 3,412 Btuh/1 kW
  - Btuh = 3,813 Btuh

Example 2 Fan/Motor Heat Air Temp Rise

- **AHU with 90% efficient Motor**
  - **Energy into the airstream is 3,813 Btuh**
- **Equation for Sensible Energy**
  - **ΔTemp = Q/(1.085 x CFM)**
  - **ΔTemp = 3,813 Btuh/1.085 x 10,000)**
  - **ΔTemp = 0.35 F degrees**

Fan/Motor Heat Psychrometric Layout



Example 2 Coil Load with ALL Heat Gain

- **Entering Air Conditions**
  - 84.3 F DB / 70.2 F WB/ 34.8 Btu/lbs / 52.2% RH
- **Leaving Air Conditions**
  - 55.0 DB / 53.0 F WB / 22.2 Btu/lbs / 90% RH
- **Coil Load**
  - $Q_T = 4.5 \times CFM \times \Delta H$
  - $Q_T = 4.5 \times 10{,}000 \times (34.8 - 22.2)$
  - $Q_T = 567{,}000$ Btuh
  - Increased Load = 567,000 – 522,000 = 45,000 Btuh

Tips on AHU Design/Selection

- **Slow Down the Air Velocity inside the AHU**
  - **Remember that Package HVAC Equipment runs at 500-600 fpm and usually only perform to a delta of 6-7 Btu/lbs (enthalpy)**
  - **Air pressure drop drives fan BHP, the lower you can drop the TSP, the Fan BHP will follow. #1 thing to do is increase your AHU cross-sectional area (bigger unit).**
  - **Give your AHU units access space for maintenance, clean units perform better!**
    - **Put Access points between elements**
    - **Instead of an 8-row coil, think 2 4-row coils**
  - **Heat and Energy Wheels require face velocities in the range of 300 – 350 fpm to work well**

Tips on AHU Design/Selection

- **When using coils for Energy Transfer, Remember:**
  - **Approach Temperature Delta Drives Energy Exchange – to increase this, increase the delta between the leaving warmer fluid and the entering cooler fluid!**
  - **Heat Exchangers with more surface area between the fluids become more efficient!**
- **Think about System and Equipment Loads!**
  - **Sometimes they can help your system operation**
- **Rethink System Design/Selection**
  - **Space loads could become coil loads**
  - **DOAS creates many benefits**

Thank You

Douglas F Zentz

Emeritus Professor

Ferris State University