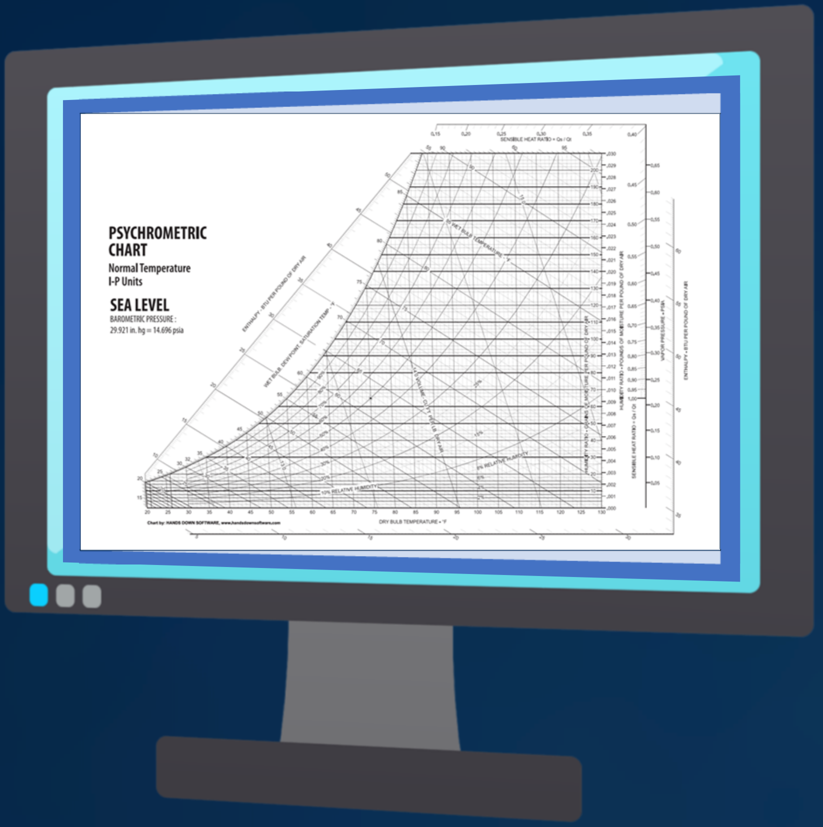
Psychrometric HVAC Equations and Real-World Application



Presented By: Chris Adams, PE

OVERVIEW

Today's Presenter

Chris Adams, P.E.

VP of Engineering
Mechanical Engineer

AAON Training

Education

Bachelor of Science, Mechanical Engineering
NC State University
Master of Business Administration
University of North Carolina Charlotte
Registered Professional Engineer
North Carolina – 037820 (License Number)
Member of ASHRAE – Southern Piedmont Chapter & Regional Vice Chair, CTTC Region IV

Biography

Chris Adams, PE is currently the VP of Engineering for Insight Partners, a Manufacturing Representative Firm that represents Aaon, Marley Cooling Towers, Armstrong Pumps, Samsung VRF, Quantech Chillers, and numerous other air and water treatment lines. Chris' area of expertise is the proper conditioning, treatment, and control for optimal indoor air quality using air or water systems as it relates to Energy Efficiency, Mold/Mildew, Chloramines, and Demand Control Ventilation. Prior experience includes President and Owner of Adams Companies, a Rep Firm covering the Carolinas for air side products. Additional Experience includes a Sales Engineer at General Electric servicing Coal, Natural Gas, and Nuclear Power Plants. During his employment with General Electric, Chris completed the Six Sigma Training Program achieving the highest level of quality control as a Master Black Belt. Chris' additional leadership activities include several Holding / Leasing Companies and board positions within ASHRAE in Region IV, Charlotte, NC, and Greenville, SC as well as a current member and Jack Stickley Fellow for the Lake Norman Lions Club. Hobbies include Scuba Diving as a certified Master Diver and Flying Single Engine Aircraft as a Private Pilot.

HVAC Rule #1

All Manufacturers make GREAT Equipment

When Applied Properly!

Blaming equipment is admitting it is smarter than you!

HVAC Rule #2

What is Easiest Way to <u>Lower RH?</u>

Turn on the Heat

It's *<u>Relative</u> to Temperature*

HVAC Rule #3

It is ALWAYS Controls Fault! ☺

- **With the proper review of the trend data in the controller, you can ALWAYS identify the problem**
- **Most of the time it will be something simple**
- **Controls may or may not be able to fix the issue**

Moisture Tracking

Sources of Moisture Loads (Know what you Have & When)

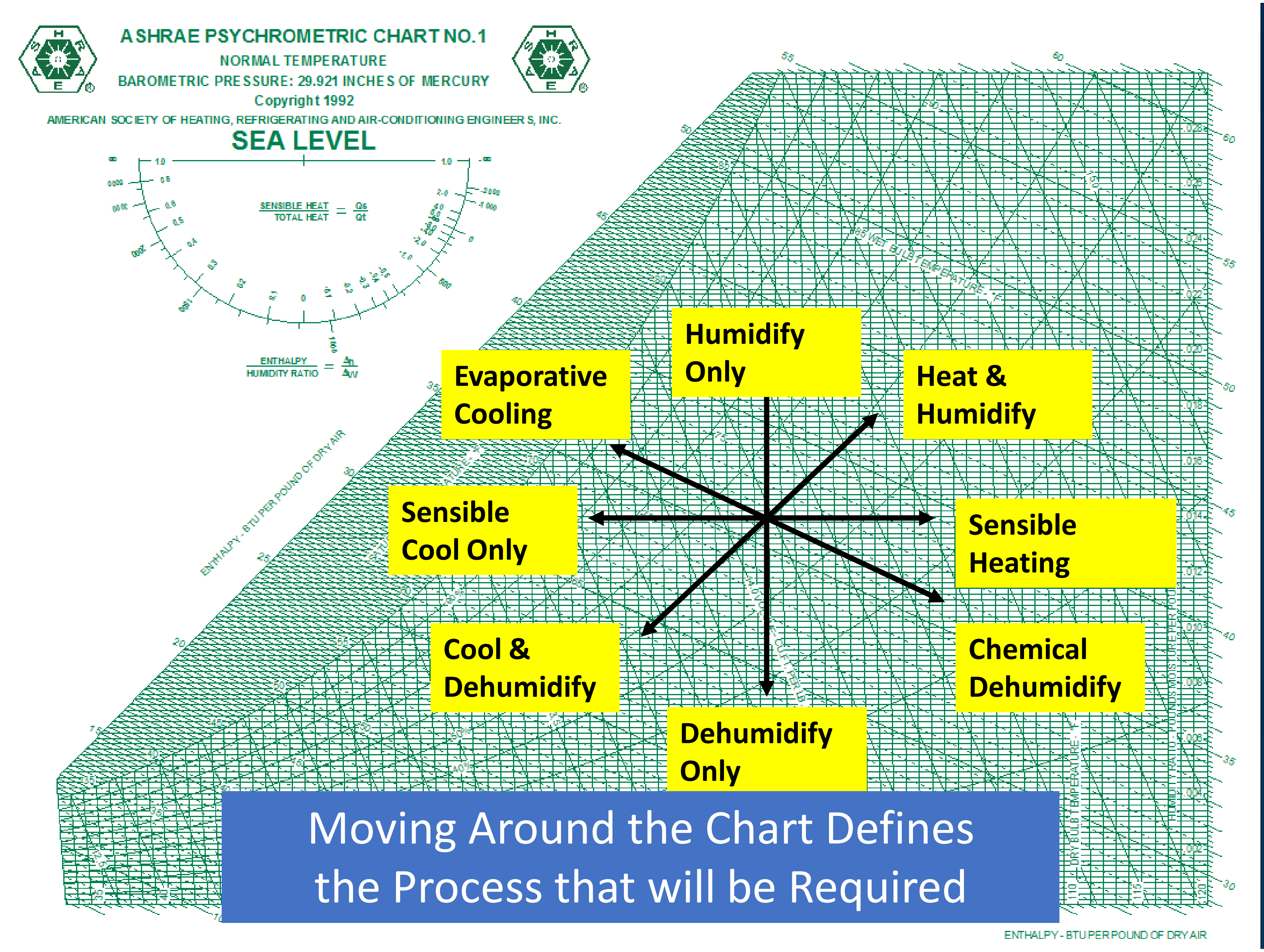
- Permeation: Moisture passing through structure
- Infiltration: Moisture entering space through cracks (Can't Stop)
- Perspiration: People – Respiration
- Condensation: Products – Change in moisture content
- Evaporation: Moisture evaporated from inside
- Make-Up Air: External Source
- Humidifier: Need Moisture Added

Controlling Content Defines Solution
Changes by Season

The “7” Properties of Air

1. **Dry bulb**
2. **Wet bulb**
3. **Dewpoint**
4. **Relative Humidity**
5. **Humidity Ratio**
6. **Volume**
7. **Enthalpy**

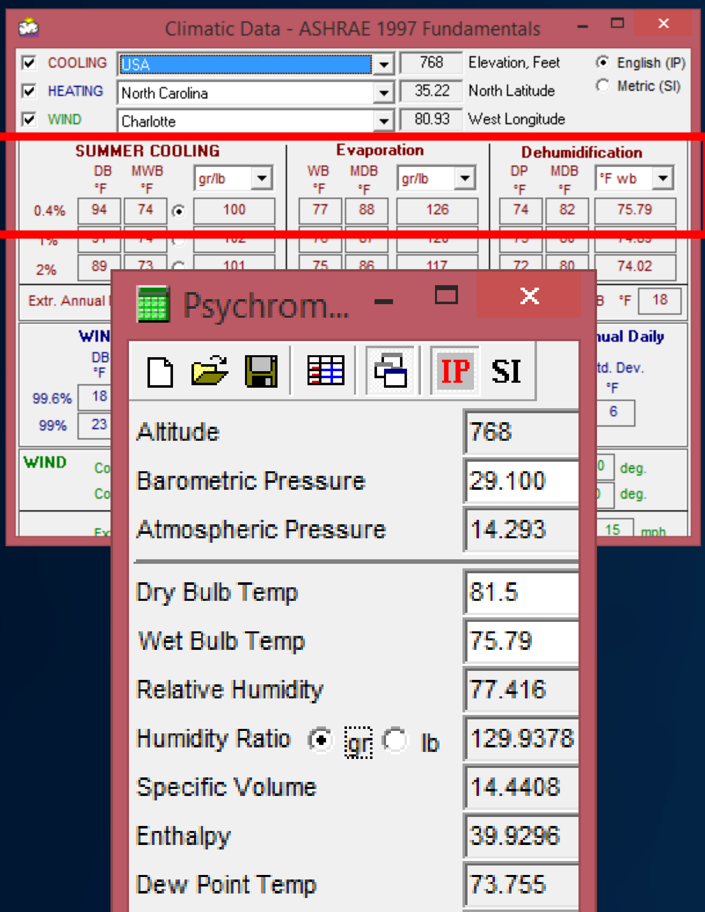
Must Know 2 !!!



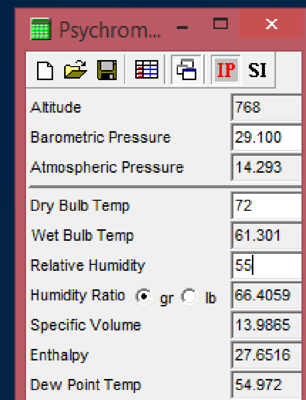
Moving Around the Chart Defines the Process that will be Required

Use the RIGHT Conditions!

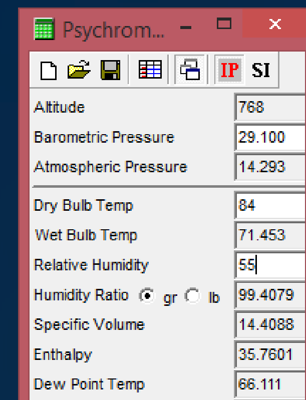
Design - Given



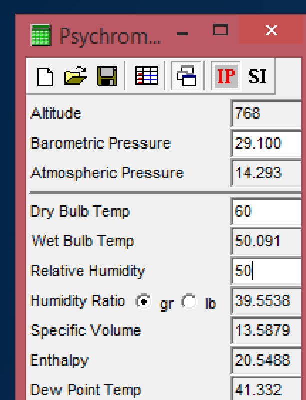
School



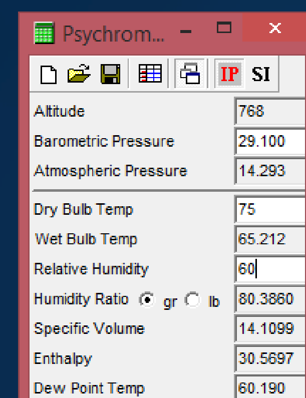
Pool



Hospital OR



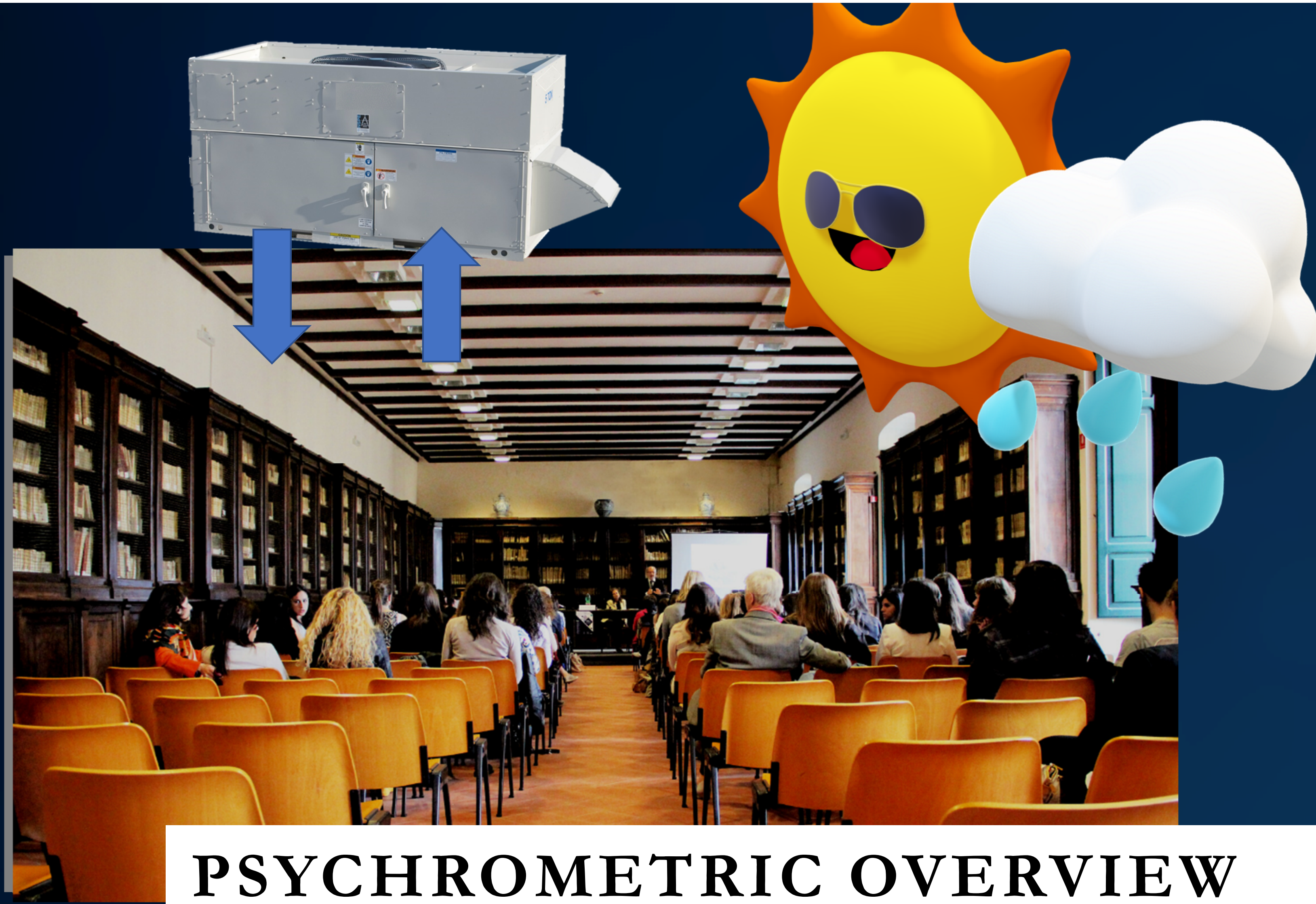
Hotel Hall



Solve the Psychrometric Problem

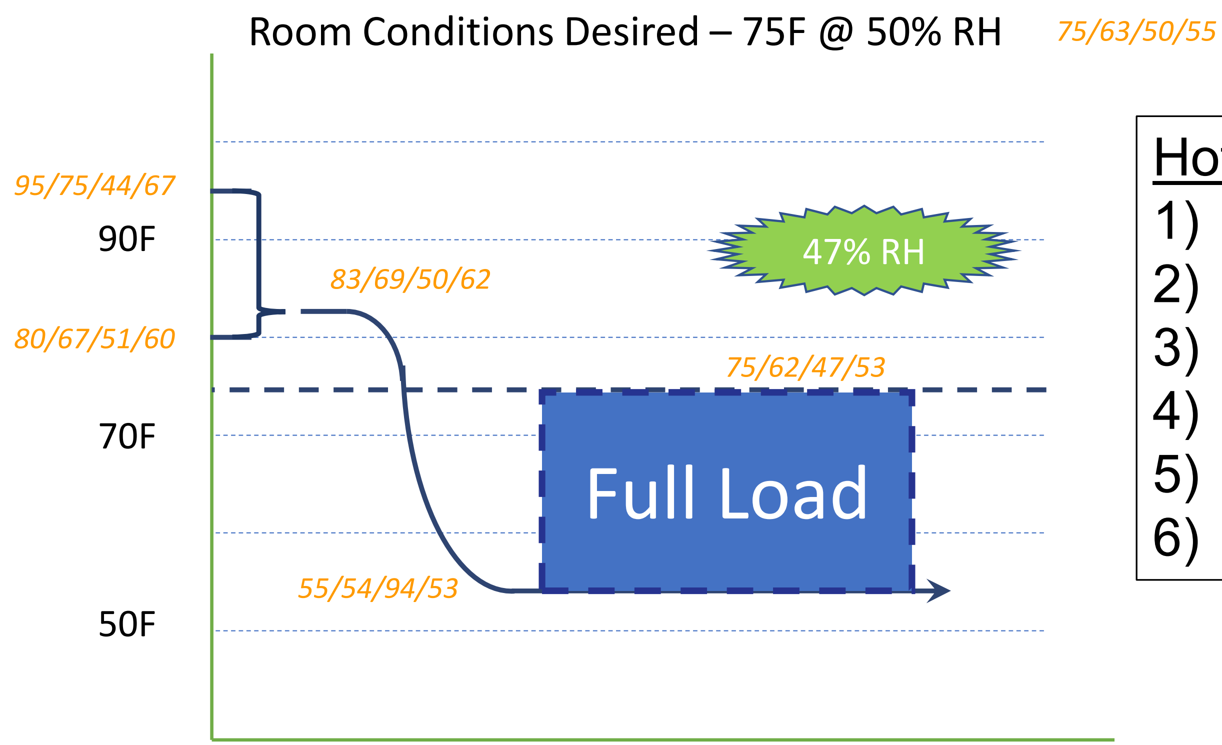
	Dry Bulb	Wet Bulb	Enthalpy	Dewpoint	RH	Grains
Outdoor Air	82	75.8	39.9	73.8	77.4	130
Pool	84	71.4	35.9	66.1	55.0	100
Hotel Hall	75	65.2	30.7	60.2	60.0	81
School	72	61.3	27.7	55.0	55.0	66.9
Hospital	60	50.1	20.6	41.3	50.0	39.8

RH Tells the Least Information by Itself



PSYCHROMETRIC OVERVIEW

Psychrometric View – Hot / Semi-Humid Air

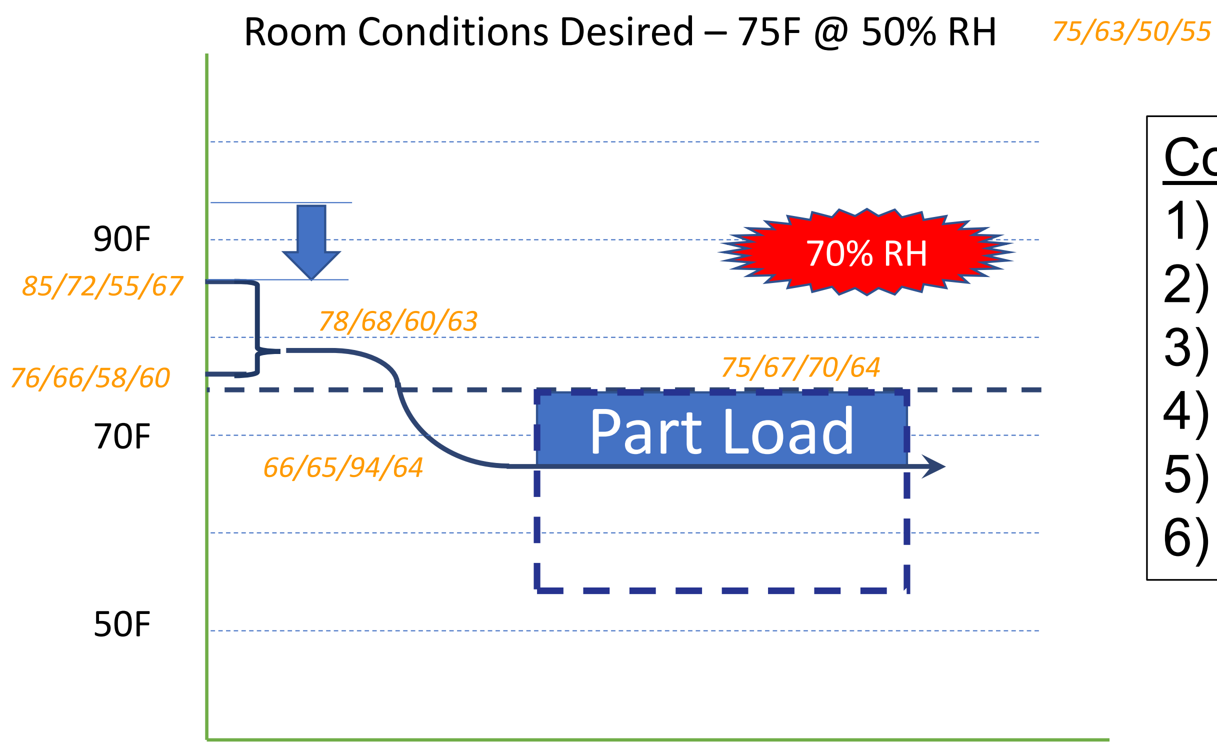


<u>Hot & Dry Day (25% O/A)</u>

1) Outdoor Air = 95db/75wb
2) Dewpoint = 67F
3) Mode = Cooling
4) Return Air = 80db/67wb
5) Mixed Air = 83db/69wb
6) Sized Correctly, ~55F, 94%RH

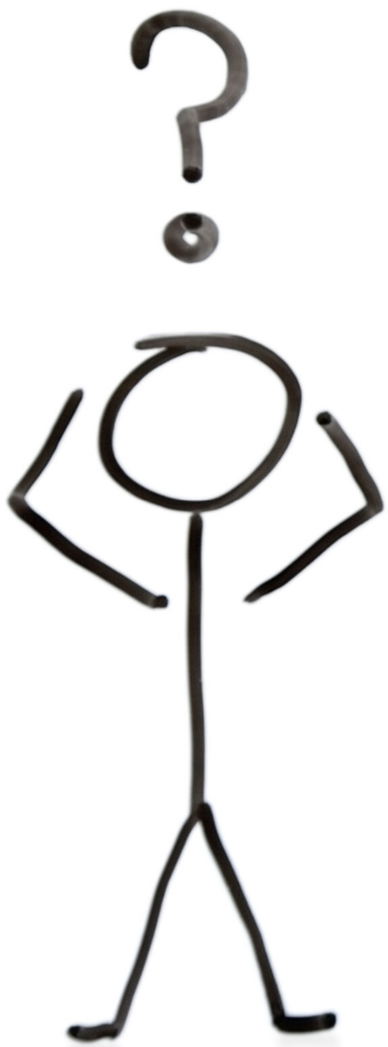
(db/wb/rh/dp)

Psychrometric View – Warm / Semi-Humid Air



Cooler (25% O/A)
1) Outdoor Air = 85db/72wb
2) Dewpoint = 67F
3) Mode = Cooling
4) Return Air = 76db/66wb
5) Mixed Air = 78db/68wb
6) Average LAT, ~66F, 94%

(db/wb/rh/dp)



What Can we DO?

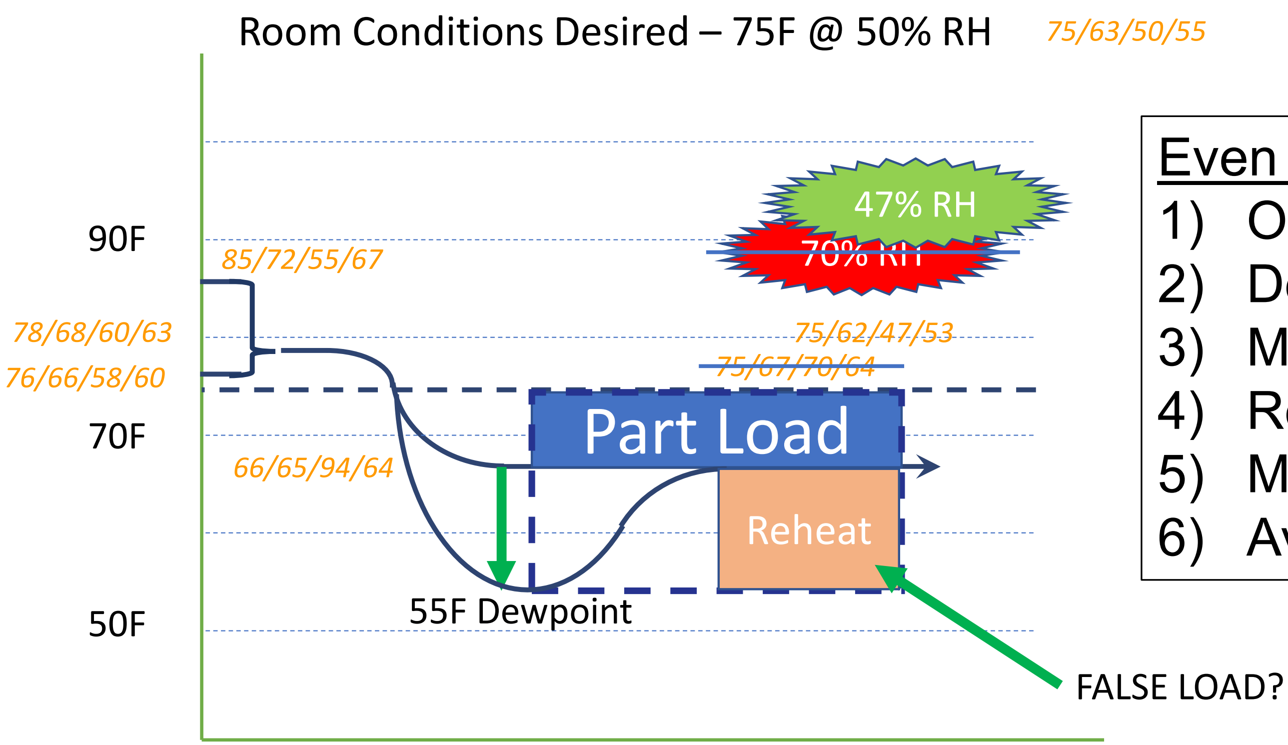
Still MUST Achieve Dewpoint!

Industry Methods / Techniques

- Sensible Load, aka *Hot Gas Reheat*
- Desiccant
- Energy Recovery
- Slow Airflow Down (VAV/SZVAV)
- Wrap Around Coils

It Depends!!!

Hot Gas Reheat View – Must Achieve Dewpoint

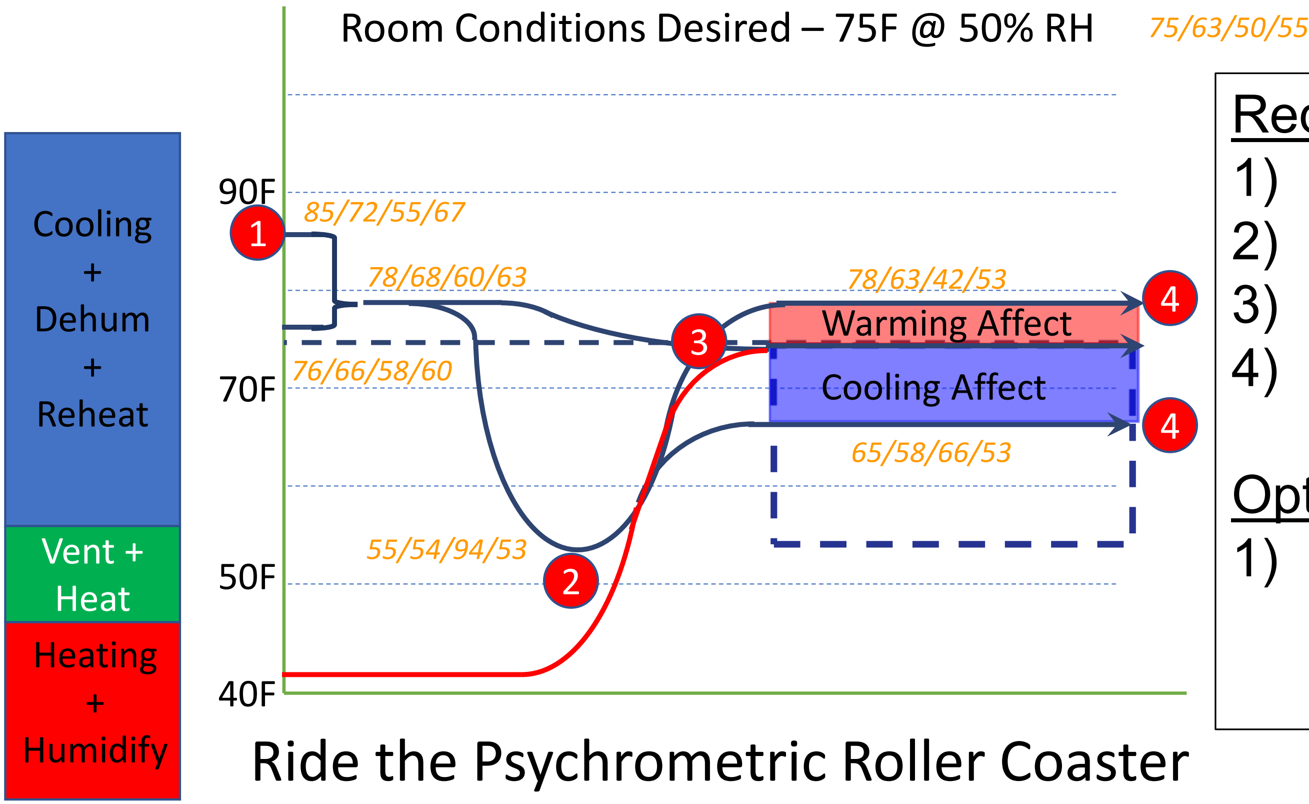


Even Cooler (25% O/A)

1) Outdoor Air = 85db/72wb
2) Dewpoint = 67F
3) Mode = Cooling
4) Return Air = 76db/66wb
5) Mixed Air = 78db/68wb
6) Average LAT, ~66F, 94%

(db/wb/rh/dp)

How is it Controlled?



<u>Required Control Sensors</u>

1) Outdoor Temp & RH
2) Coil Temp (Suction Pressure)
3) Supply Air Temp
4) Room Temp

<u>Optional Enhancement</u>

1) Room Humidity

(db/wb/rh/dp)

Equations to Know

$$Q = 1.1 * cfm * (\Delta T)$$

$$Q = 4.5 * cfm * (\Delta h)$$

$$Q = 500 * gpm * (\Delta T)$$

CRITICAL HVAC EQUATIONS

Equations to Know

$$Q = 1.1 * cfm * (Delta\ T)$$

$$Q = 4.5 * cfm * (Delta\ h)$$

$$Q = 500 * gpm * (Delta\ T)$$

Commit These to Memory!

Sensible Heating (Equation 1)

Sensible Heat (3,000 cfm from 42F, 80% RH to 72F)

Q = 1.1 * cfm * (Delta T)

Q = 1.1 * 3,000 * (72 – 42) = 99,000 BTU

Psychrometric Data

	Starting Point	Ending Point
Dry Bulb (F)	42	72
Wet Bulb (F)	39.4	53.1
Relative Humidity (%)	80	27.1
Grains	32.4	32.4
Enthalpy (h)	15.1	22.3
Dewpoint (F)	36.3	36.3

Moisture Stays the Same (Grains & Dewpoint)
Easiest Solution to Lower RH is Add Heat

Equations to Know

Q = 1.1 * cfm * (Delta T)

Q = 4.5 * cfm * (Delta h)

Q = 500 * gpm * (Delta T)

Cooling w/ Moisture Removal (Equation 2)

3,000 cfm, 100% O/A from 82F db, 76F wb to 72F @ 55% RH)

Q = 4.5 * cfm * (Delta h)

Q = 4.5 * 3,000 (40.1 – 27.7) = 167.4 MBH or 13.95 Tons

Psychrometric Data

	Starting Point	**Ending Point**
Dry Bulb (F)	82	72
Wet Bulb (F)	76	61.3
Relative Humidity (%)	76.5	55
Grains	130.5	66.4
Enthalpy (h)	40.1	27.7
Dewpoint (F)	73.9	55.0

Quick Math, ~ 14 tons, or 214 cfm/Ton

Don't Forget Fan Heat & Room Load

Equations to Know

$$Q = 1.1 * cfm * (\text{Delta } T)$$

$$Q = 4.5 * cfm * (\text{Delta } h)$$

$$Q = 500 * gpm * (\text{Delta } T)$$

Water Side Review

Job Information

Job Name:	*SPASHRAE*
Job Number:	*Job #11*
Site Altitude:	*768 ft*
Refrigerant	*R-410A*

Unit Information

Approx. Op./Ship Weights:	*1696 / 1696 lbs. (±5%)*
Supply CFM/ESP:	*3000 / 0.25 in. wg.*
Final Filter FV / Qty:	*216.00 fpm / 4*
Outside CFM:	*3000*
Ambient Temperature:	*82 °F DB / 76 °F WB*
Return Temperature:	*75 °F DB / 62 °F WB*

Static Pressure

External:	*0.25 in. wg.*	Economizer:	*0.00 in. wg.*
Evaporator:	*0.19 in. wg.*	Heating:	*0.08 in. wg.*
Filters Clean:	*0.08 in. wg.*	Cabinet:	*0.06 in. wg.*
Dirt Allowance	*0.15 in. wg.*	Total:	*0.81 in. wg.*

Cooling Section

	Gross	Net
Total Capacity:	*196.25*	*194.58 MBH*
Sensible Capacity:	*79.45*	*77.78 MBH*
Latent Capacity:	*116.80 MBH*	
Mixed Air Temp:	*82.00 °F DB*	*76.00 °F WB*
Entering Air Temp:	*82.00 °F DB*	*76.00 °F WB*
Lv Air Temp (Coil):	*56.07 °F DB*	*55.98 °F WB*
Lv Air Temp (Unit)	*56.59 °F DB*	*56.19 °F WB*
Supply Air Fan:	*1 x 220 @ 0.57 BHP*	
SA Fan RPM / Width:	*830 / 4.920"*	
Evaporator Coil:	*14.6 ft² / 4 Rows / 14 FPI*	
Evaporator Face Velocity:	*205.7 fpm*	

Heating Section(**)

Primary Heat Type:	*Heat Pump - Not operational when the indoor coil entering temperature is less than 43.0 °F*
Auxiliary Heat Type:	*Hot Water Heat*
Heating CFM:	*3000*
Total Capacity:	*99.7 MBH*
OA Temp:	*42.0 °F DB / 39.4 °F WB*
RA Temp:	*68.0 °F DB / 54.0 °F WB*
Entering Air Temp:	*42.0 °F DB / 39.4 °F WB*
Leaving Air Temp:	*72.0 °F DB / 53.1 °F WB*
Entering Water:	*180.0 °F*
Leaving Water:	*165.3 °F*
GPM / Head:	*14 / 3.1 ft*
Water Velocity:	*2.43 fps*
FA / RD / FPI / FV:	*5.83 ft² / 1 / 8 / 514.3*

Water Side Review

Job Information

Job Name:	*SPASHRAE*
Job Number:	*Job #11*
Site Altitude:	*768 ft*
Refrigerant	*R-410A*

Unit Information

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Heating CFM:	*3000*
Total Capacity:	*99.7 MBH*
OA Temp:	*42.0 °F DB / 39.4 °F WB*
RA Temp:	*68.0 °F DB / 54.0 °F WB*
Entering Air Temp:	*42.0 °F DB / 39.4 °F WB*
Leaving Air Temp:	*72.0 °F DB / 53.1 °F WB*
Entering Water:	*180.0 °F*
Leaving Water:	*165.3 °F*
GPM / Head:	*14 / 3.1 ft*
Water Velocity:	*2.4[illegible] fps*
FA / RD / FPI / FV:	*5.83 ft² / 1 / 8 / 514.3*

Water Side Review

Q = 500 * gpm * (Delta T)

	Starting Point	Ending Point
Dry Bulb (F)	42	72
Wet Bulb (F)	39.4	53.1
Relative Humidity (%)	80	27.1
Grains	32.4	32.4
Enthalpy (h)	15.1	22.3
Dewpoint (F)	36.3	36.3

Q = 99,000 BTU

	Starting Point	Ending Point
Water Temp	180	165.3
GPM	14	14

Q = 500 * gpm * (Delta T)

Q = 500 * 14 * (180 – 165.3) = 102,900 BTU

Rating Page Shows 99,700 BTU…All very close

Church Case Study

Problem

Rooms at the end of the loop either HOT or COLD depending on the Season. Some hotter, some colder.

REAL WORLD APPLICATIONS

Church Example

Final TAB Report – Rebalance

	Airside Information					Waterside Information			
	Clg Max CFM	Clg Min CFM	Heating CFM	EAT °F	LAT °F	EWT °F	LWT °F	Coil ΔP'	GPM
Design	3400	1020	3400	42.0	78.4	180.0	141.9	7.3	7.0
Actual	3141	1073	3053	77.2	102.2	190.8	122.7	7.1	6.9

Balance Equations (Qair = Qwater)

Q(air) = 1.1 * cfm * (Delta T)

Q(water) = 500 * gpm * (Delta T)

Church Example

	Airside Information					Waterside Information			
	Clg Max CFM	Clg Min CFM	Heating CFM	EAT °F	LAT °F	EWT °F	LWT °F	Coil ΔP'	GPM
Design	3400	1020	3400	42.0	78.4	180.0	141.9	7.3	7.0
Actual	3141	1073	3053	77.2	102.2	190.8	122.7	7.1	6.9

Q(air) = 1.1 * cfm * (Delta T)
= 1.1 * 3,053 * (102.2 – 77.2)
= 83,958 BTU

Q(water) = 500 * gpm * (Delta T)
= 500 * 6.9 * (190.8 – 122.7)
= 234,945 BTU

Not Very Close: 84K vs. 235K. WHY?

Church Example

Not Very Close: 86K vs. 235K… WHY?

Caveats

1) No Circuit Setter. Flow was set using memory stops on the associated Control Valve and was calculated using Coil Delta P.
2) No Balancing Dampers installed. Grill adjusted closed to reduce airflow; they are not noisy.

Uh Oh! What COULD this Mean?

Church Example

Potential Issues

Coil Fouled?

Correct Water Control Valve?

Others?

Assessment

- SAT is easy to measure and should be accurate.
- Caveat states they couldn't get GPM measurement.
- Reverse Calculate Flow…Estimate 2.5 gpm.
- Lack of flow under true design day would lack heat.

Start Looking at Coil/Water Loops Real Flow.

School Example

Problem

1) Unable to control the Space RH. It would go above 60% at 72F inside on a hot/humid day.
2) Condensate staying in the drain pan coming down through the unit intermittently.

School Example (3 Similar Sites)

Details Known Prior to Site Inspection

- Designed Airflow: 5,215 cfm
- Nominal 25 Ton Unit with Energy Recovery
- Space Temp in Control
- Space RH out of Control
- Trap Not Flowing – 3.5” Trap
- Static Pressure across Damper Assembly reported as 0.54” wc with a design of 0.07” (~8X Over)
- Fan Static Reported as 3.07” wc. Design Static was 2.19” wc. (0.88” wc over Design)

School Example (3 Similar Sites)

Site Inspection

- Fan Speed 1,584 rpm vs. Design at 1,174 rpm
- Anemometer Airflow Measurement gave estimate of 6,715 – 7505 cfm. (Limited Instrument Accuracy)
- Measured 3.3” wc across fan w/ Semi-Dirty Filters
- Ambient Conditions: 78F db, 71F Dewpoint (~10am)
- Mixed Air Temp: 75F @ 69% RH
- Leaving Air Temp: 54F @ 94% RH

Cooling w/ Moisture Removal (Equation 2)

	Outdoor	Mixed Air	Leaving Air
Dry Bulb (F)	78	75	54
Wet Bulb (F)	73.0	67.7	53.0
Relative Humidity (%)	79	69	94%
Grains	114.7	90.2	58.5
Enthalpy (h)	36.7	32.1	22.0
Dewpoint (F)	71.0	64.2	52.3

Q = 4.5 * cfm * (Delta h) … Solve for CFM
25 Ton Unit = 300,000 BTU * 1.05% = 315,000 BTU
CFM = 315,000 BTU / (4.5 * (32.1 – 22.0))
= 6,931 cfm

Anemometer: 6,715 to 7,505 cfm. Confirms unit must be over CFM.
DOAS Unit Suggestion: Balance at +0% / -10%. 10% of 5,215 = 521 cfm @ 150/ton = 3.5 Tons.

School Example

Summary

- Unit was ~33% Over on CFM
- Designed SAT at 5,215 cfm Equipment would Supply 72F at 50% RH.
- At Actual CFM Design Day, unit would Supply 72F at 62% RH or Higher.
- With Static Pressure at 3.3", as filters loaded, they exceeded 3.5" total & drain trap would no longer drain. CONFIRMED…New Filters allowed unit to drain fine, as time passed, problem would come back. Lowering CFM to proper design, static drops and the issue goes away completely.

College Example

Problem

Site is cold in the wintertime and has days where at 72F the RH will not stay below 60% RH. Site was dealing with the issue for 4 years prior to calling in support.

College Example

Site Inspection During the Winter

Design: 2,000 cfm, 10F, Hot Water Coil, 80F LAT, EWT 180F, 30% Glycol

Actual Readings: EAT 40F, LAT = 80F, GPM = 9, EWT 160F, LWT 128F

College Example

Balance Equations (Q = Q)

Q(glycol) = 472 * gpm * (Delta T)

Q(air) = 1.1 * cfm * (Delta T)

Q(glycol) = 472 * gpm * (Delta T)
= 472 * 9 * (160 - 128)
= 135,936 BTU

Q(air) = 1.1 * cfm * (Delta T) – Solve for CFM
CFM = 135,936 / (1.1 * (80 - 40))
= 3,089 cfm

Drop to Design CFM of 2,000, LAT = 91F, Problem Solved in both Summer and Winter

College Example

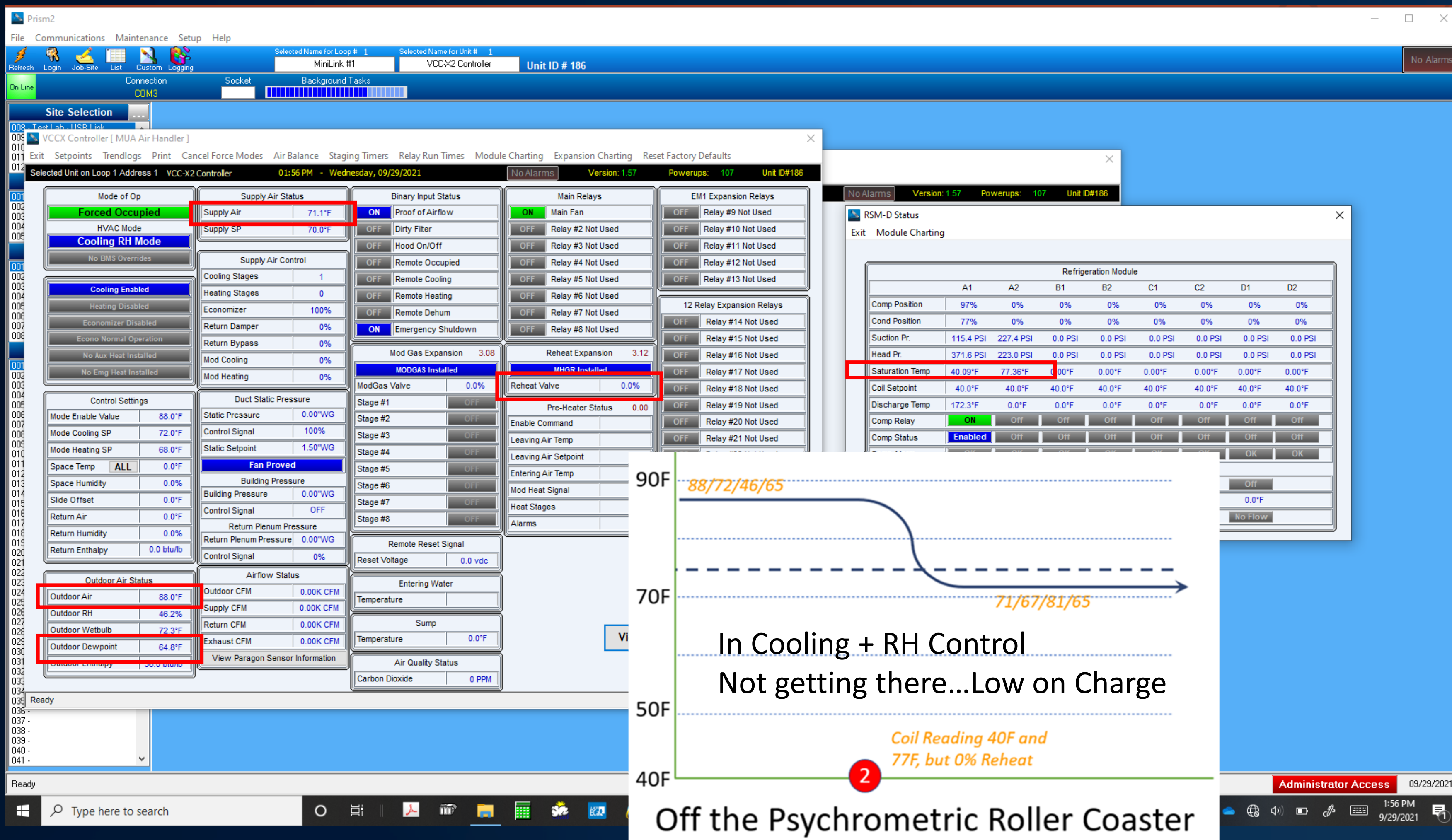
Solution

- Slow the Fan down. 54% too high on CFM
- Review Water Temp. Why 160F vs. 180F?

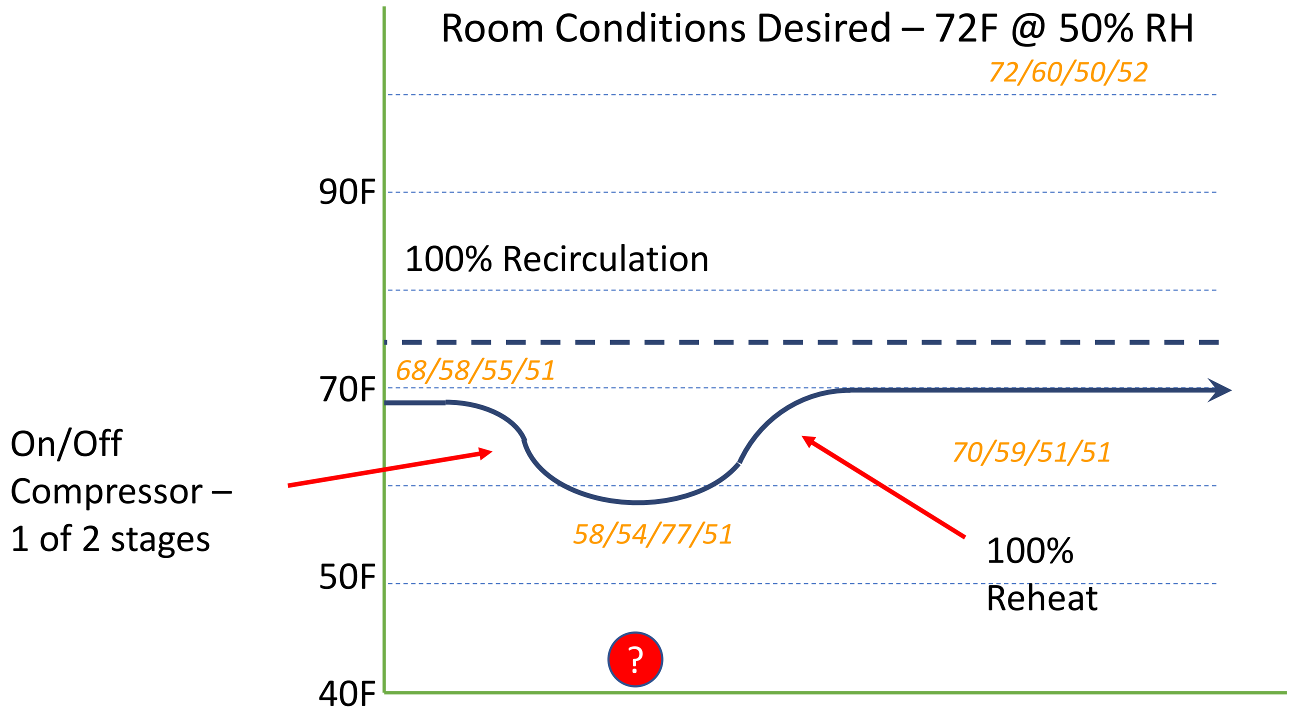
All the gauges were there, they just assumed airflow was correct and blamed equipment.

Customer Suffered with the issue for 4 years before calling for support…Typical Statement, Unit Never Worked since it was installed!

Nursing Home – Controls Review



RH Control Concern



Not Removing Moisture Removal

(db/wb/rh/dp)

Rule of Thumb Example

Church Application - How Many People?

Example Calculation
1,000 People

Rule of Thumb Formula
1) 1,000 * 550 BTU / Person = 550,000 or 48.8 Tons
2) 1,000 * (7.5 cfm O/A / Person) / 200 cfm/ton = 37.5 Tons

Ballpark Tons = 86.3 Tons

With understanding the background, it becomes Trivial to Solve any HVAC Situation with Psychometrics… All it takes is time to do the Math

You can work your way through Solving ANY Room Conditions

Knowing the Theory Makes Solutions Easy

QUESTIONS?

PSYCHROMETRIC CHART

Normal Temperature
I-P Units

SEA LEVEL

BAROMETRIC PRESSURE :
29.921 in. hg = 14.696 psia