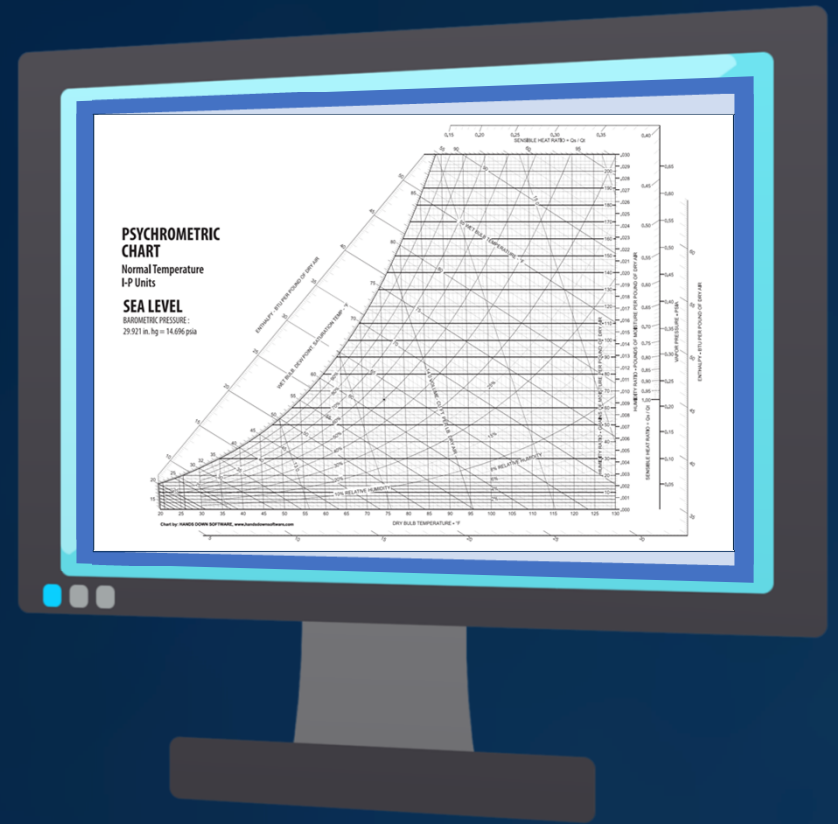


Psychrometric HVAC Equations and Real-World Application



Presented By: Chris Adams, PE



OVERVIEW

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SPEAKER INTRODUCTION

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HVAC RULES

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PSYCHROMETRIC OVERVIEW

4

CRITICAL HVAC EQUATIONS

5

REAL WORLD APPLICATION

Today's Presenter

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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
Lower RH?*

Turn on the **Heat**
It's Relative 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 !!!

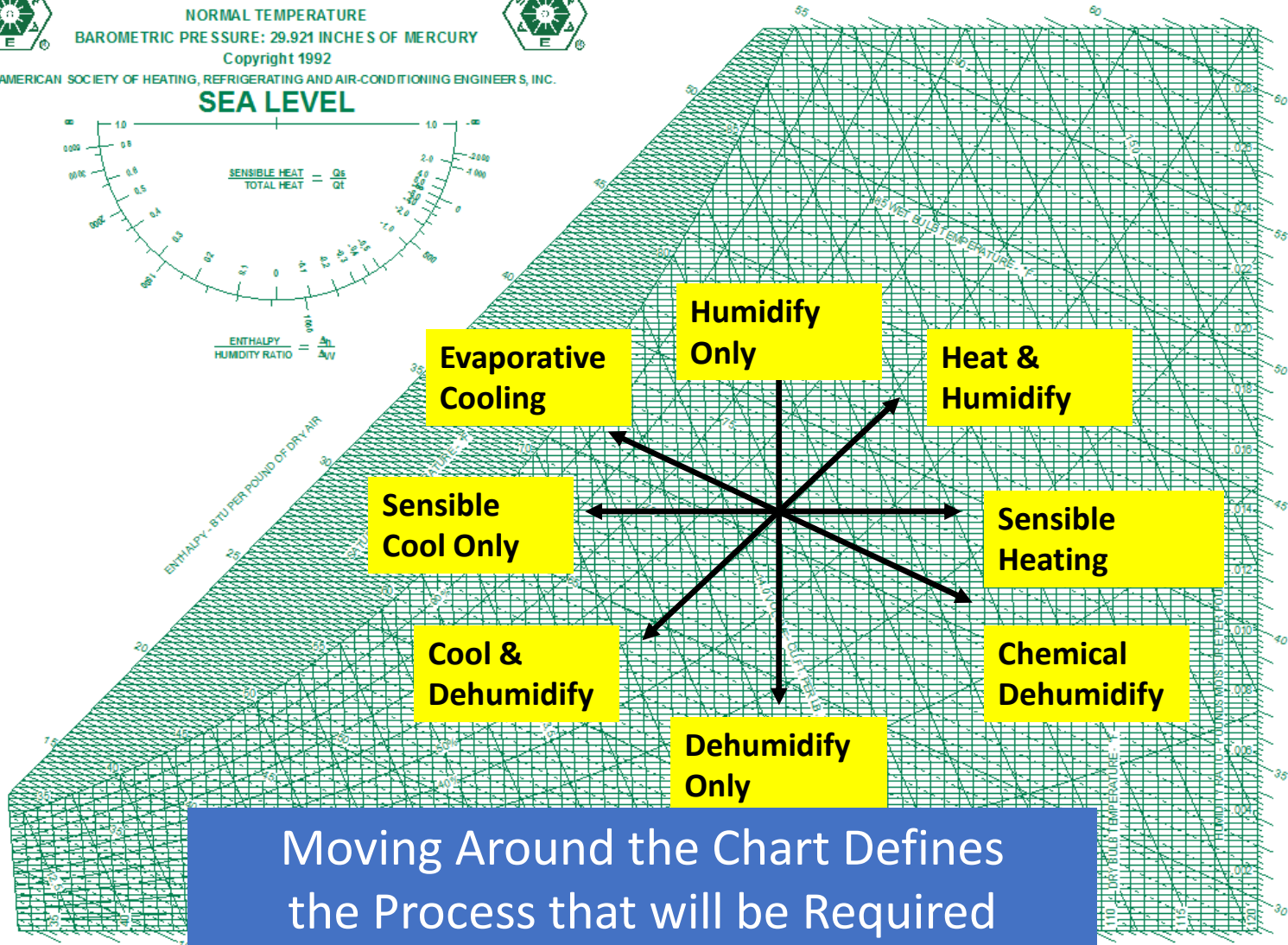
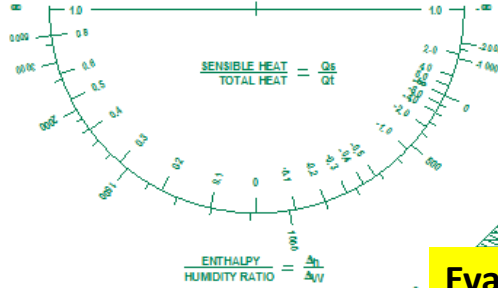


ASHRAE PSYCHROMETRIC CHART NO.1
 NORMAL TEMPERATURE
 BAROMETRIC PRESSURE: 29.921 INCHES OF MERCURY
 Copyright 1992



AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

SEA LEVEL



Moving Around the Chart Defines the Process that will be Required

ENTHALPY - BTU PER POUND OF DRY AIR

Use the RIGHT Conditions!

Design - Given

School

Pool

Climatic Data - ASHRAE 1997 Fundamentals

COOLING USA 768 Elevation, Feet English (IP)
HEATING North Carolina 35.22 North Latitude Metric (SI)
WIND Charlotte 80.93 West Longitude

SUMMER COOLING			Evaporation			Dehumidification			
DB °F	MWB °F	gr/lb	WB °F	MDB °F	gr/lb	DP °F	MDB °F	°F wb	
0.4%	94	74	100	77	88	126	74	82	75.79
1%	91	74	102	76	87	125	73	80	74.85
2%	89	73	101	75	86	117	72	80	74.02

Psychrom...

Altitude	768
Barometric Pressure	29.100
Atmospheric Pressure	14.293
Dry Bulb Temp	72
Wet Bulb Temp	61.301
Relative Humidity	55
Humidity Ratio gr/lb	66.4059
Specific Volume	13.9865
Enthalpy	27.6516
Dew Point Temp	54.972

Psychrom...

Altitude	768
Barometric Pressure	29.100
Atmospheric Pressure	14.293
Dry Bulb Temp	84
Wet Bulb Temp	71.453
Relative Humidity	55
Humidity Ratio gr/lb	99.4079
Specific Volume	14.4088
Enthalpy	35.7601
Dew Point Temp	66.111

Psychrom...

Altitude	768
Barometric Pressure	29.100
Atmospheric Pressure	14.293
Dry Bulb Temp	81.5
Wet Bulb Temp	75.79
Relative Humidity	77.416
Humidity Ratio gr/lb	129.9378
Specific Volume	14.4408
Enthalpy	39.9296
Dew Point Temp	73.755

Hospital OR

Hotel Hall

Psychrom...

Altitude	768
Barometric Pressure	29.100
Atmospheric Pressure	14.293
Dry Bulb Temp	60
Wet Bulb Temp	50.091
Relative Humidity	50
Humidity Ratio gr/lb	39.5538
Specific Volume	13.5879
Enthalpy	20.5488
Dew Point Temp	41.332

Psychrom...

Altitude	768
Barometric Pressure	29.100
Atmospheric Pressure	14.293
Dry Bulb Temp	75
Wet Bulb Temp	65.212
Relative Humidity	60
Humidity Ratio gr/lb	80.3860
Specific Volume	14.1099
Enthalpy	30.5697
Dew Point Temp	60.190

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

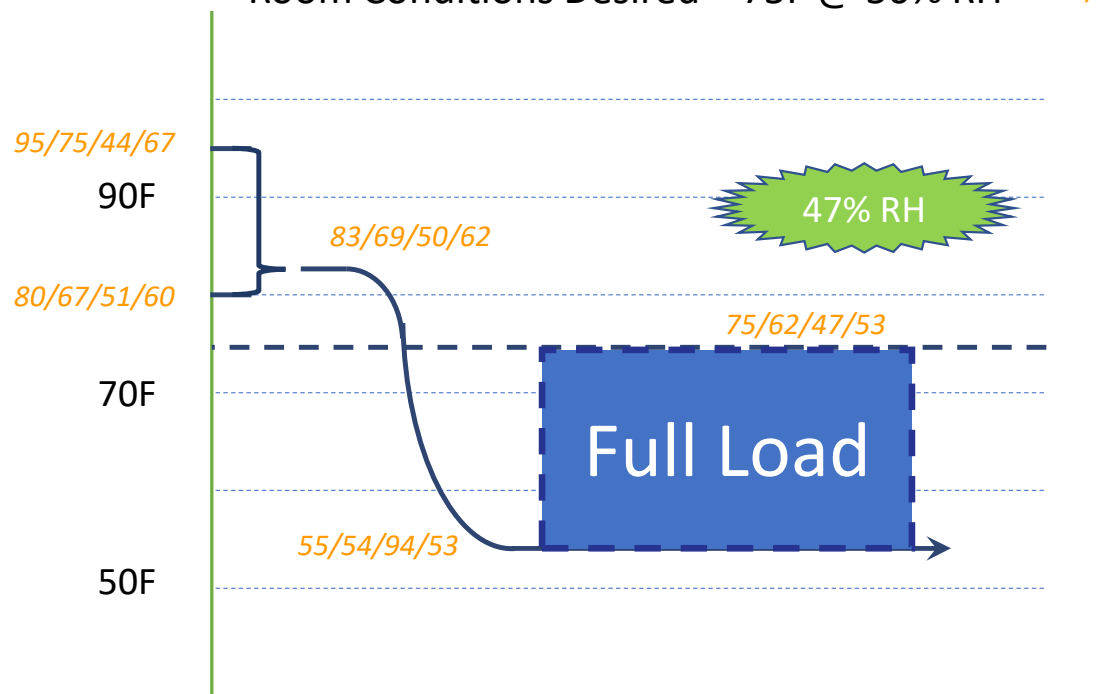
RH Tells the Least Information by Itself



PSYCHROMETRIC OVERVIEW

Psychrometric View – Hot / Semi-Humid Air

Room Conditions Desired – 75F @ 50% RH *75/63/50/55*



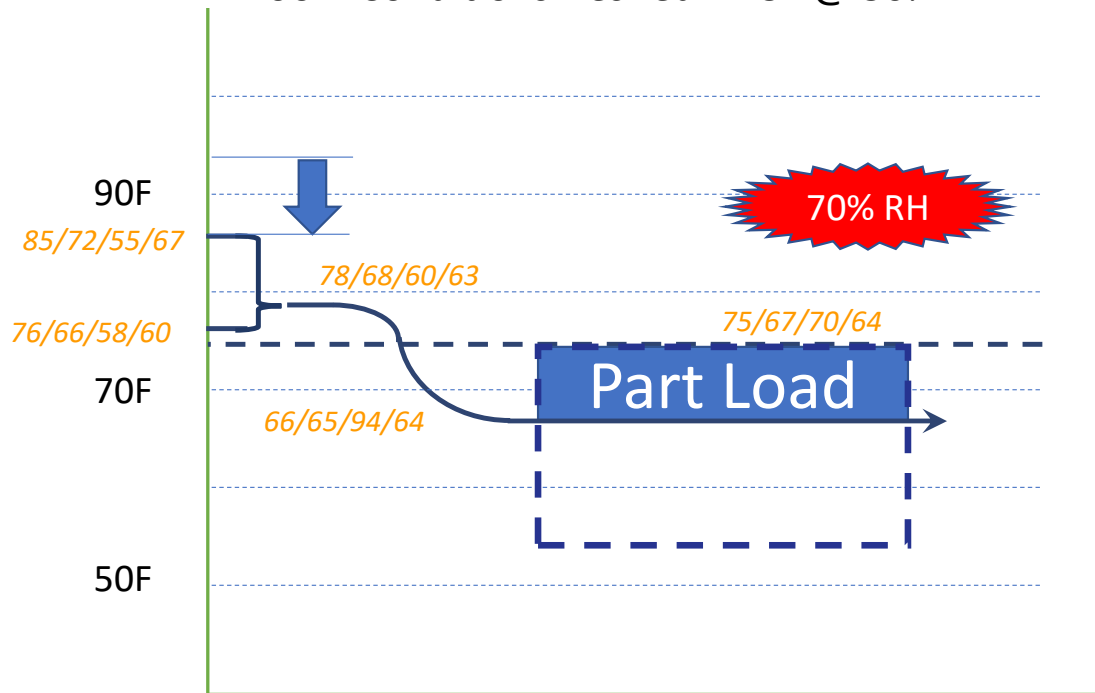
Hot & Dry Day (25% O/A)

- 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

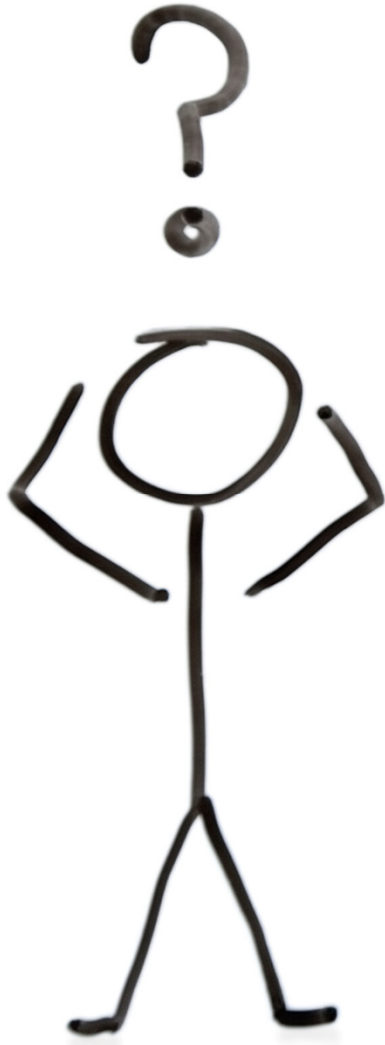
Room Conditions Desired – 75F @ 50% RH *75/63/50/55*



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!

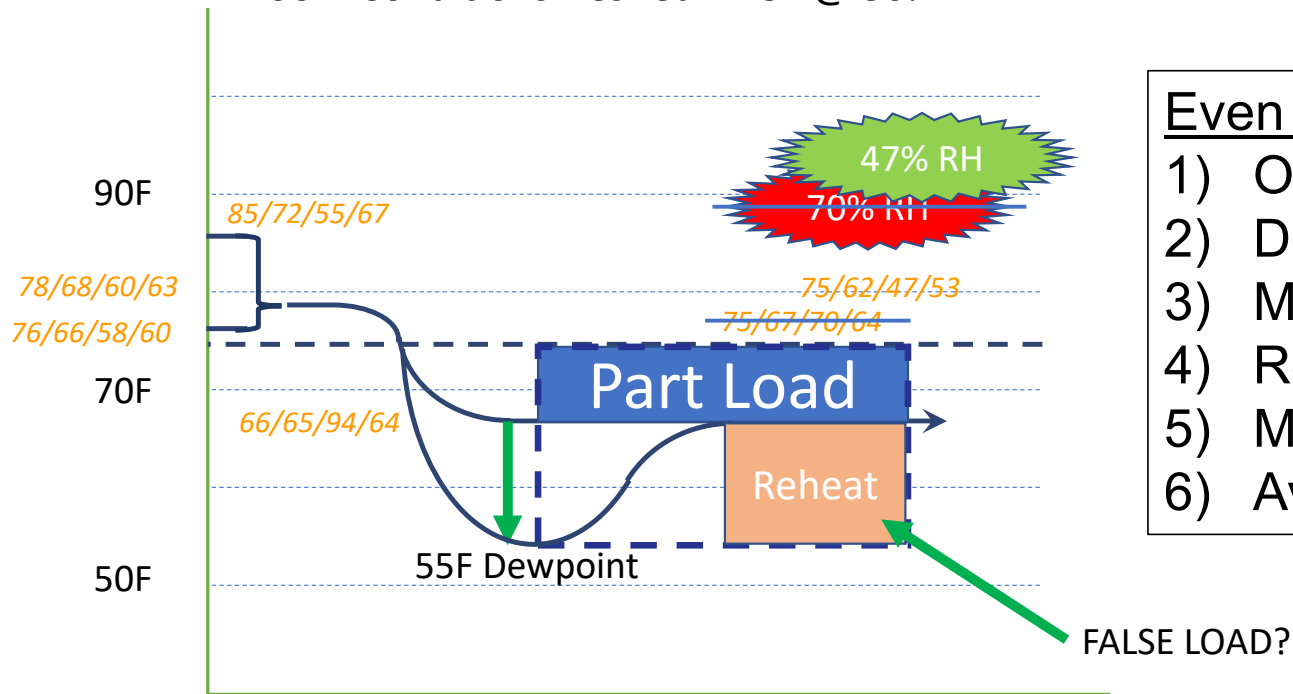
Industry Methods / Techniques

- 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

Room Conditions Desired – 75F @ 50% RH *75/63/50/55*



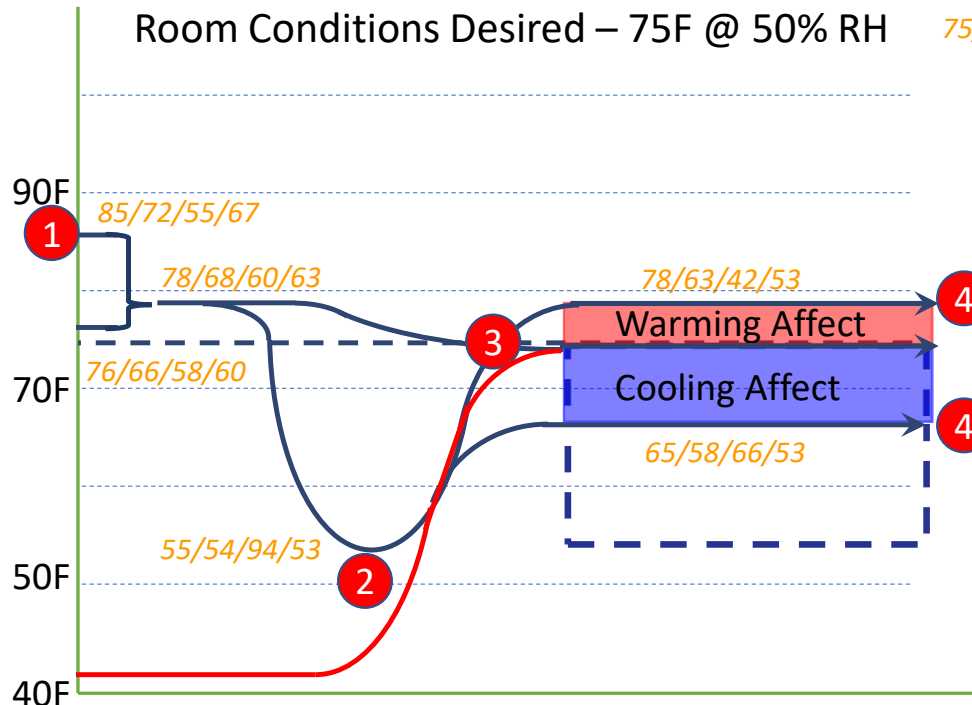
Even Cooler (25% O/A)

- 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?

Room Conditions Desired – 75F @ 50% RH *75/63/50/55*



Ride the Psychrometric Roller Coaster

Cooling
+
Dehum
+
Reheat

Vent +
Heat

Heating
+
Humidify

Required Control Sensors

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

Optional Enhancement

- 1) Room Humidity

(db/wb/rh/dp)

Equations to Know

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

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

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

CRITICAL HVAC EQUATIONS

Equations to Know

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

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

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

Commit These to Memory!

Sensible Heating (Equation 1)

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

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

$$Q = 1.1 * 3,000 * (72 - 42) = 99,000 \text{ BTU}$$

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

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

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

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

Cooling w/ Moisture Removal (Equation 2)

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

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

$$Q = 4.5 * 3,000 (40.1 - 27.7) = 167.4 \text{ MBH or } 13.95 \text{ Tons}$$

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

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

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

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

Water Side Review

Job Information

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

Static Pressure

External: 0.25 in. wg.
 Evaporator: 0.19 in. wg.
 Filters Clean: 0.08 in. wg.
 Dirt Allowance: 0.15 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 ft ² / 4 Rous / 14 FPI	
Evaporator Face Velocity:	205.7 fpm	

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

Economizer: 0.00 in. wg.
 Heating: 0.08 in. wg.
 Cabinet: 0.06 in. wg.
 Total: 0.81 in. wg.

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			Unit Information	
Job Name:	SPASHRAE		Approx. Op./Ship Weights:	1696 / 1696 lbs. (±5%)
Job Number:	Job #11		Supply CFM/ESP:	3000 / 0.25 in. wg.
Site Altitude:	768 ft		Final Filter FV / Qty:	216.00 fpm / 4
Refrigerant	R-410A		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			Heating Section (**)	
	Gross	Net	Primary Heat Type:	Heat Pump - Not operational when the indoor coil entering temperature is less than 43.0 °F
Total Capacity:	196.25	194.58 MBH		
Sensible Capacity:	79.45	77.78 MBH	Auxiliary Heat Type:	Hot Water Heat
Latent Capacity:	116.80 MBH		Heating CFM:	3000
Mixed Air Temp:	82.00 °F DB	76.00 °F WB	Total Capacity:	99.7 MBH
Entering Air Temp:	82.00 °F DB	76.00 °F WB	OA Temp:	42.0 °F DB / 39.4 °F WB
Lv Air Temp (Coil):	56.07 °F DB	55.98 °F WB	RA Temp:	68.0 °F DB / 54.0 °F WB
Lv Air Temp (Unit):	56.59 °F DB	56.19 °F WB	Entering Air Temp:	42.0 °F DB / 39.4 °F WB
Supply Air Fan:	1 x 220 @ 0.57 BHP		Leaving Air Temp:	72.0 °F DB / 53.1 °F WB
SA Fan RPM / Width:	830 / 4.920"		Entering Water:	180.0 °F
Evaporator Coil:	14.6 ft ² / 4 Rous / 14 FPI		Leaving Water:	165.3 °F
Evaporator Face Velocity:	205.7 fpm		GPM / Head:	14 / 3.1 ft
			Water Velocity:	2.48 fpm
			FA / RD / FPI / FV:	5.83 ft ² / 1 / 8 / 514.3

Water Side Review

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

	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

$$Q = 99,000 \text{ BTU}$$

	Starting Point	Ending Point
Water Temp	180	165.3
GPM	14	14

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

$$Q = 500 * 14 * (180 - 165.3) = 102,900 \text{ BTU}$$

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
<i>Design</i>	3400	1020	3400	42.0	78.4	180.0	141.9	7.3	7.0
<i>Actual</i>	3141	1073	3053	77.2	102.2	190.8	122.7	7.1	6.9

Balance Equations (Q_{air} = Q_{water})

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

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

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

$$\begin{aligned} Q(\text{air}) &= 1.1 * \text{cfm} * (\text{Delta T}) \\ &= 1.1 * 3,053 * (102.2 - 77.2) \\ &= 83,958 \text{ BTU} \end{aligned}$$

$$\begin{aligned} Q(\text{water}) &= 500 * \text{gpm} * (\text{Delta T}) \\ &= 500 * 6.9 * (190.8 - 122.7) \\ &= 234,945 \text{ BTU} \end{aligned}$$

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

$Q = 4.5 * \text{cfm} * (\text{Delta } h) \dots \text{Solve for CFM}$

25 Ton Unit = 300,000 BTU * 1.05% = 315,000 BTU

$\text{CFM} = 315,000 \text{ BTU} / (4.5 * (32.1 - 22.0))$
= 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)

$$Q(\text{glycol}) = 472 * \text{gpm} * (\text{Delta T})$$

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

$$\begin{aligned} Q(\text{glycol}) &= 472 * \text{gpm} * (\text{Delta T}) \\ &= 472 * 9 * (160 - 128) \\ &= 135,936 \text{ BTU} \end{aligned}$$

$$\begin{aligned} Q(\text{air}) &= 1.1 * \text{cfm} * (\text{Delta T}) - \text{Solve for CFM} \\ \text{CFM} &= 135,936 / (1.1 * (80 - 40)) \\ &= 3,089 \text{ cfm} \end{aligned}$$

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

The screenshot displays a software interface for a VCCX2 Controller. The main window shows various control panels:

- Mode of Op:** Forced Occupied, Cooling RH Mode.
- Supply Air Status:** Supply Air 71.1°F, Supply SP 70.0°F.
- Binary Input Status:** Proof of Airflow ON, Dirty Filter OFF, Remote Occupied OFF, Remote Cooling OFF, Remote Heating OFF, Remote Dehum OFF, Emergency Shutdown ON.
- Main Relays:** Main Fan ON, Relays #2-#8 Not Used.
- EM1 Expansion Relays:** Relays #9-#13 Not Used.
- 12 Relay Expansion Relays:** Relays #14-#21 Not Used.
- Control Settings:** Mode Cooling SP 72.0°F, Mode Heating SP 68.0°F, Space Temp ALL 0.0°F, Space Humidity 0.0%, Slide Offset 0.0°F, Return Air 0.0°F, Return Humidity 0.0%, Return Enthalpy 0.0 btu/lb.
- Outdoor Air Status:** Outdoor Air 88.0°F, Outdoor RH 46.2%, Outdoor Wetbulb 72.3°F, Outdoor Dewpoint 64.8°F.

A psychrometric chart is overlaid on the interface, showing a process line from 88/72/46/65 (Outdoor Air) to 71/67/81/65 (Supply Air). The chart indicates the system is in cooling and reheat control but is not reaching the target conditions.

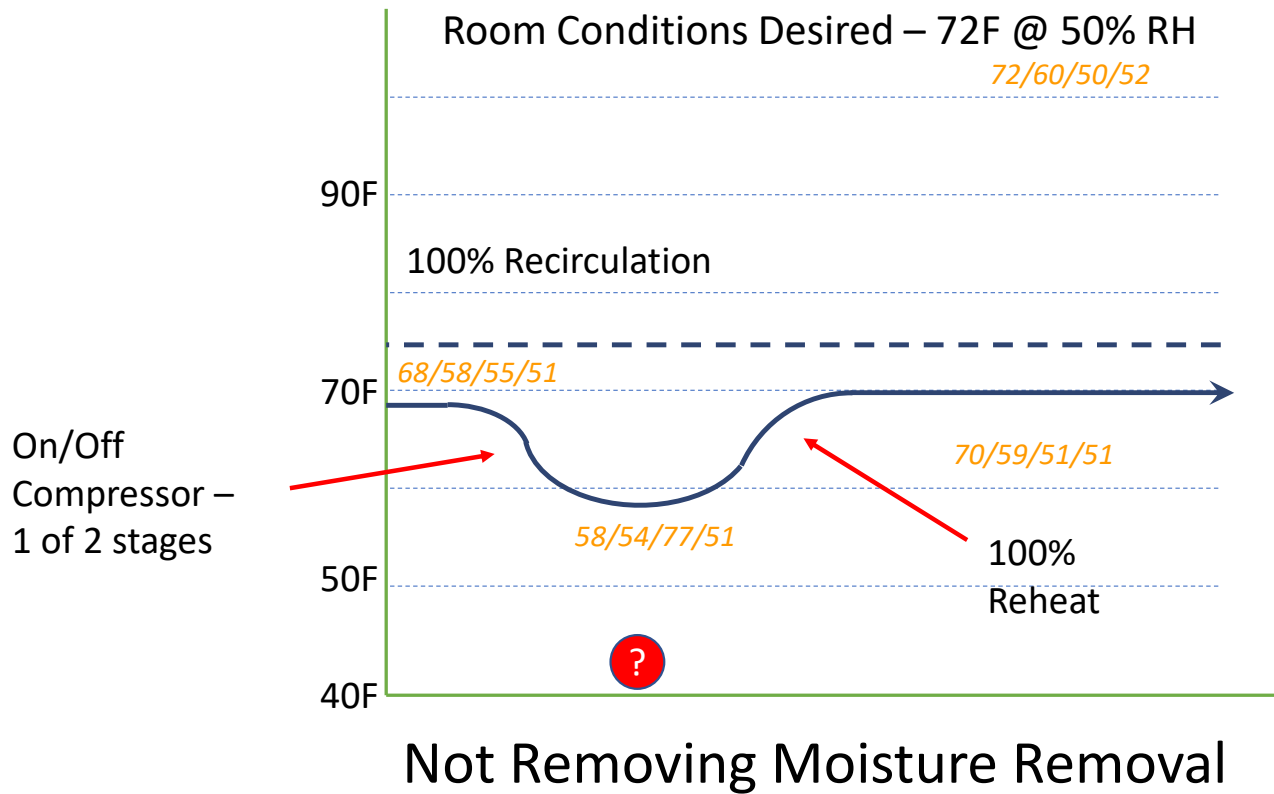
The RSM-D Status window shows the Refrigeration Module status:

	A1	A2	B1	B2	C1	C2	D1	D2
Comp Position	97%	0%	0%	0%	0%	0%	0%	0%
Cond Position	77%	0%	0%	0%	0%	0%	0%	0%
Suction Pr.	115.4 PSI	227.4 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI
Head Pr.	371.6 PSI	223.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI	0.0 PSI
Saturation Temp	40.09°F	77.36°F	0.00°F	0.00°F	0.00°F	0.00°F	0.00°F	0.00°F
Coil Setpoint	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F
Discharge Temp	172.3°F	0.0°F	0.0°F	0.0°F	0.0°F	0.0°F	0.0°F	0.0°F
Comp Relay	ON	OFF	OFF	OFF	OFF	OFF	OFF	OFF
Comp Status	Enabled	OFF	OFF	OFF	OFF	OFF	OFF	OFF

Annotations on the psychrometric chart:

- 88/72/46/65 (Outdoor Air)
- 71/67/81/65 (Supply Air)
- Coil Reading 40F and 77F, but 0% Reheat
- In Cooling + RH Control Not getting there...Low on Charge
- Off the Psychrometric Roller Coaster

RH Control Concern



(db/wb/rh/dp)

Rule of Thumb Example

Church Application - How Many People?

Example Calculation

1,000 People

Rule of Thumb Formula

1) $1,000 * 550 \text{ BTU / Person} = 550,000$ or 48.8 Tons

2) $1,000 * (7.5 \text{ cfm O/A / Person}) / 200 \text{ cfm/ton} = 37.5$ 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

