Oversized Air-conditioning Systems and Overcooled Buildings in Hot and Humid Climates

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Description

Oversized air-conditioning systems and overcooled buildings in hot and humid climates

Why are air-conditioned buildings in hot and humid climates so cold that one gets reminded of carrying a jacket when going to office? Would raising the set point temperatures in these buildings do the trick? What are the engineering challenges that necessitate a relook at the way air-conditioned buildings in such climates are designed? This talk will review some of the fundamental issues of cooling and dehumidification facing the HVAC designer and the inevitable and inherent design of an oversized system and its undesirable consequences in terms of an overcooled indoor environment. It will provide an understanding of the psychrometric challenges involved in cooling and dehumidification at peak and part loads in hot and humid climates. Possible solutions to creating a more thermally comfortable and healthy indoor environment that can also save energy will be discussed.

Learning Objectives

1. **Describe the psychrometric challenges involved in cooling and dehumidification at peak and part loads in hot and humid climates**
2. **Quantify the energy penalty resulting from summer overcooling**
3. **Quantify occupant discomfort resulting from inappropriate strategies to avert overcooling**
4. **Describe engineering solutions involved in preventing overcooled buildings in such climates and enhancing thermal comfort and IAQ**

1

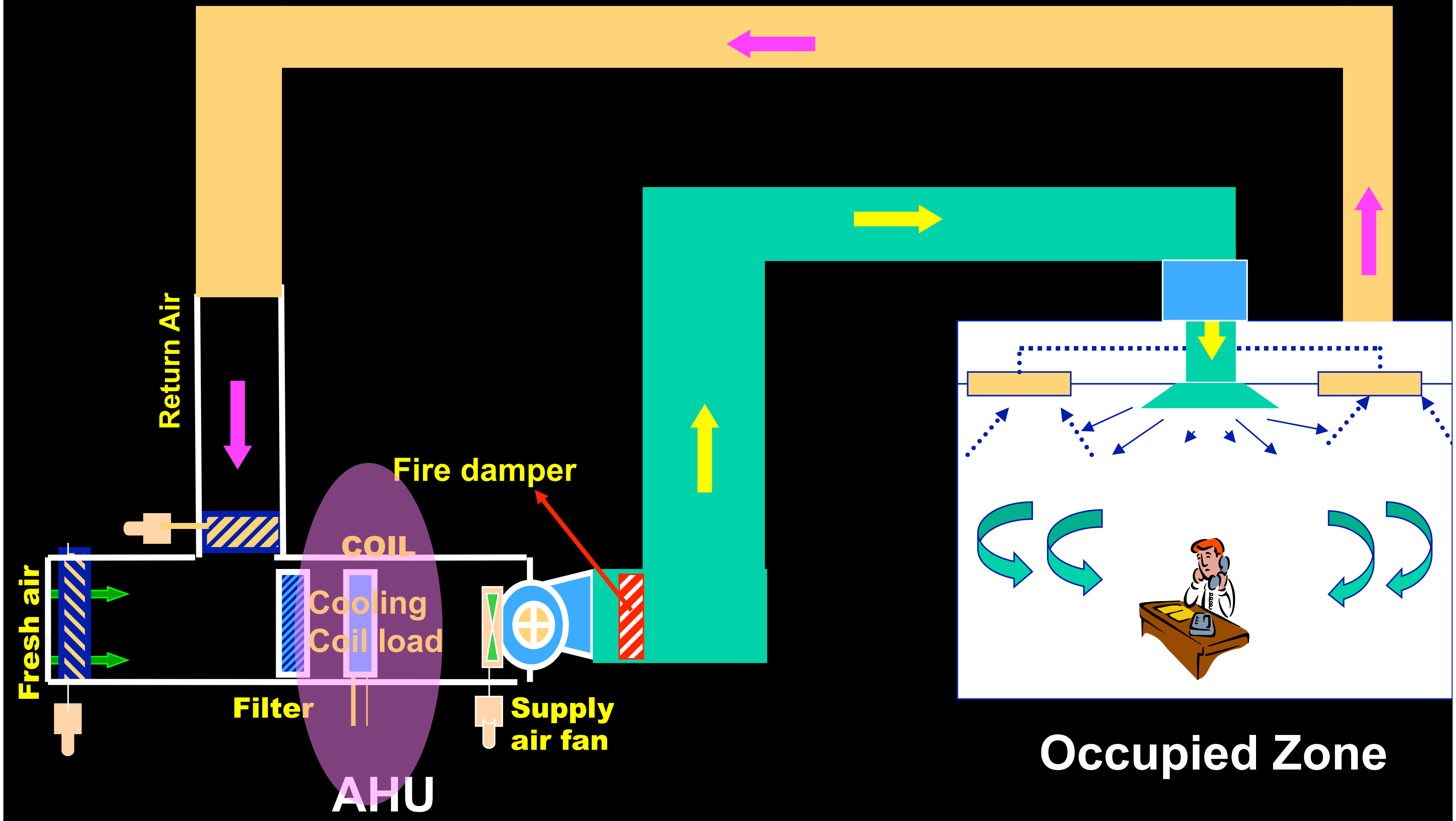
- Why are air-conditioned buildings in hot and humid climates so cold?

2

- Would raising the indoor set point temperature in overcooled buildings be a viable solution?

3

- What are the engineering solutions to the problem of overcooling in buildings in hot and humid climates?



A typical Air-conditioning system

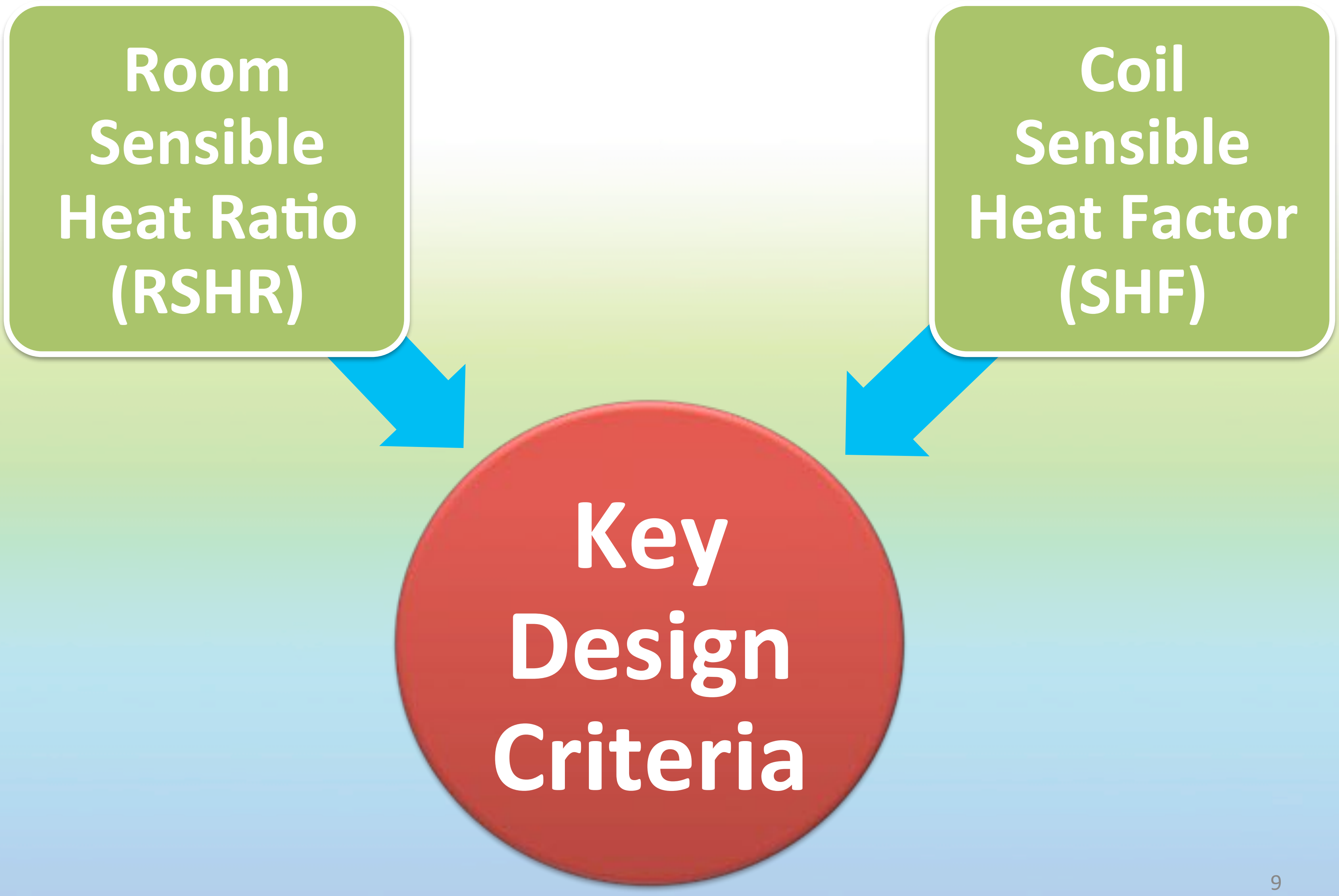
1

Why are air-conditioned buildings in hot and humid climates so cold?

Performance of cooling and dehumidifying coil

Strong bearing on indoor temperature and humidity conditions

Impact on IAQ

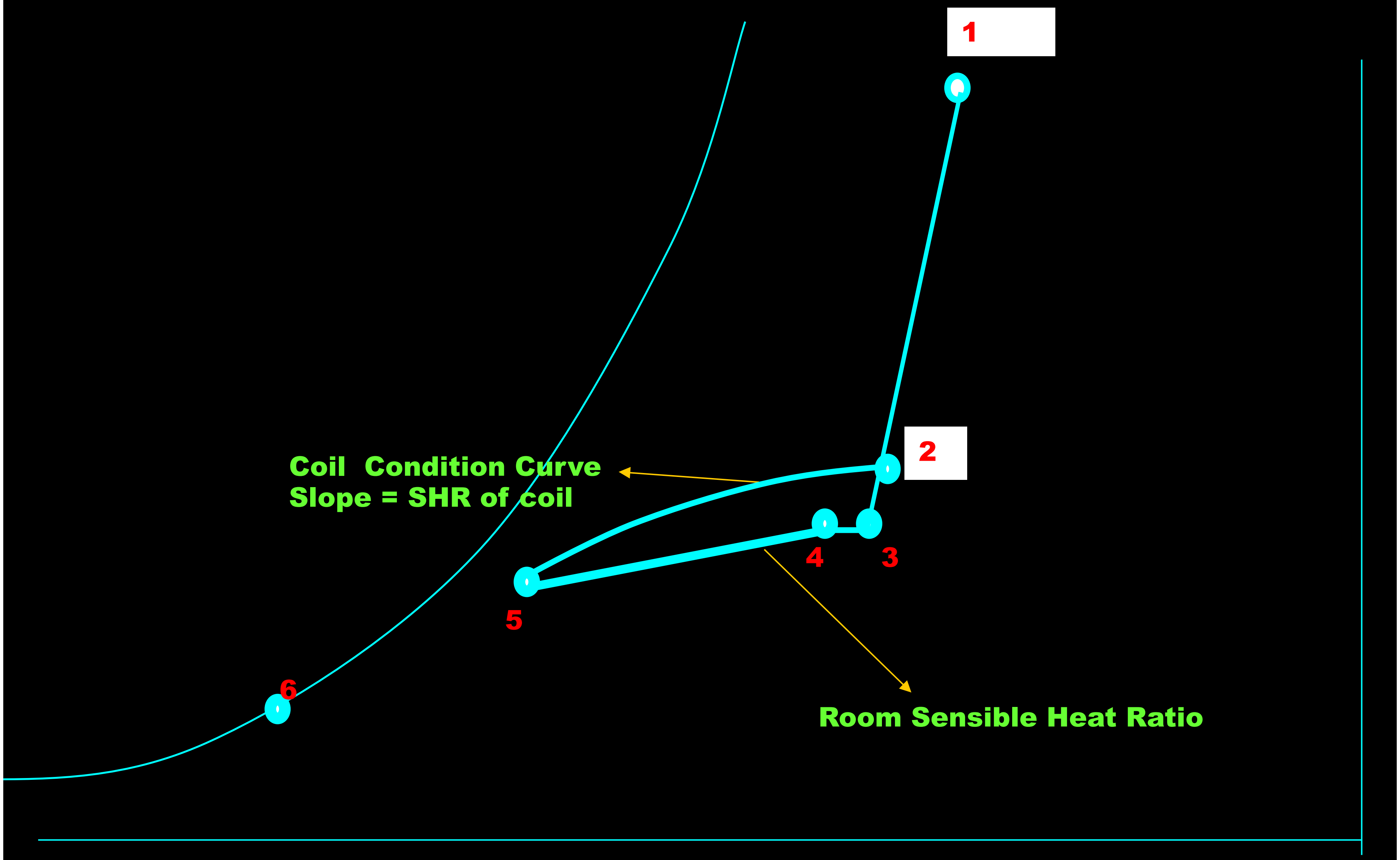

Room Sensible Heat Ratio (RSHR)
Coil Sensible Heat Factor (SHF)
Key Design Criteria

Operation controlled by chilled water modulation

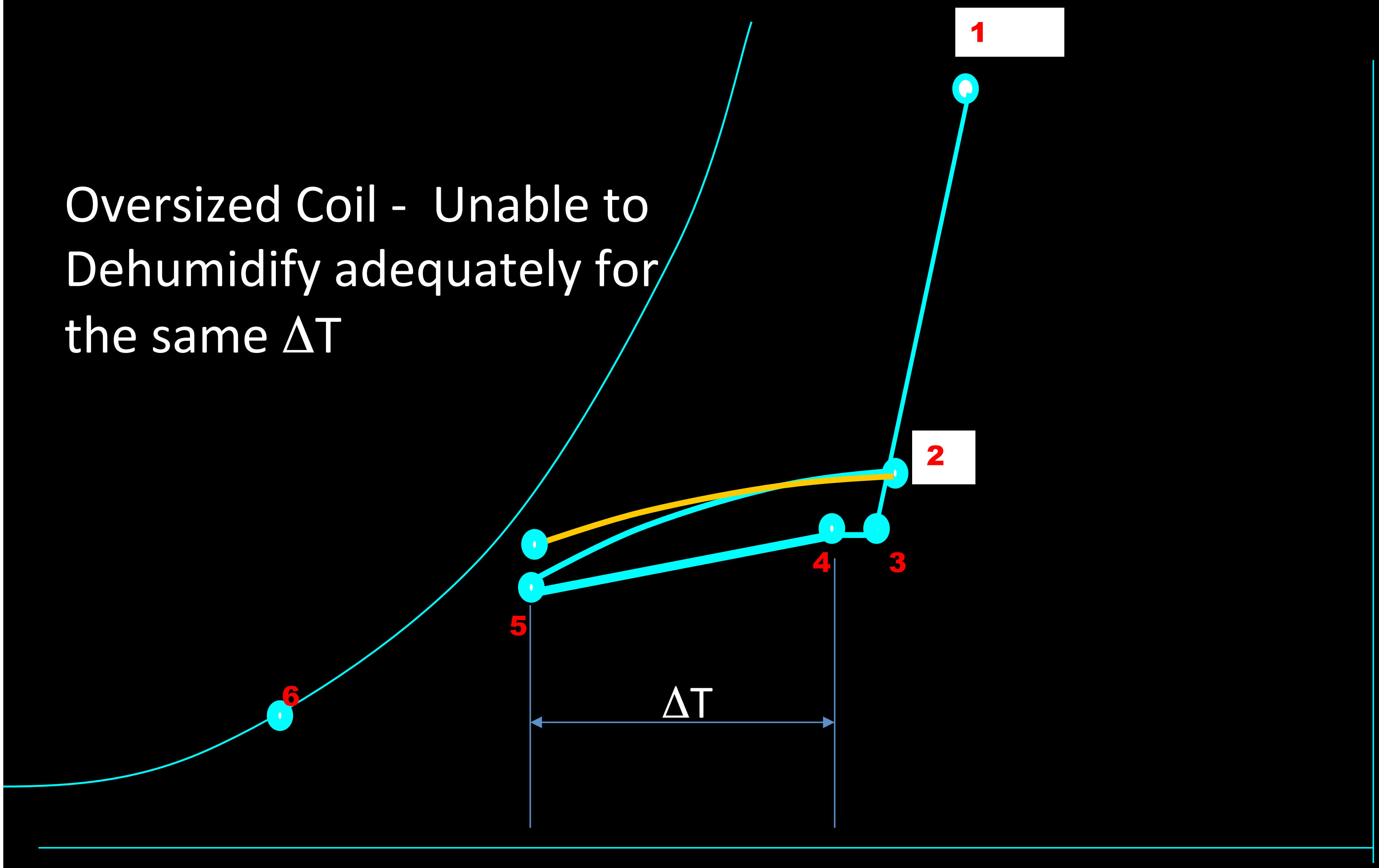
Often leads to problems due to overdesign

Inhibits dehumidifying performance exactly when it is needed to dehumidify more

COIL IS TOO BIG

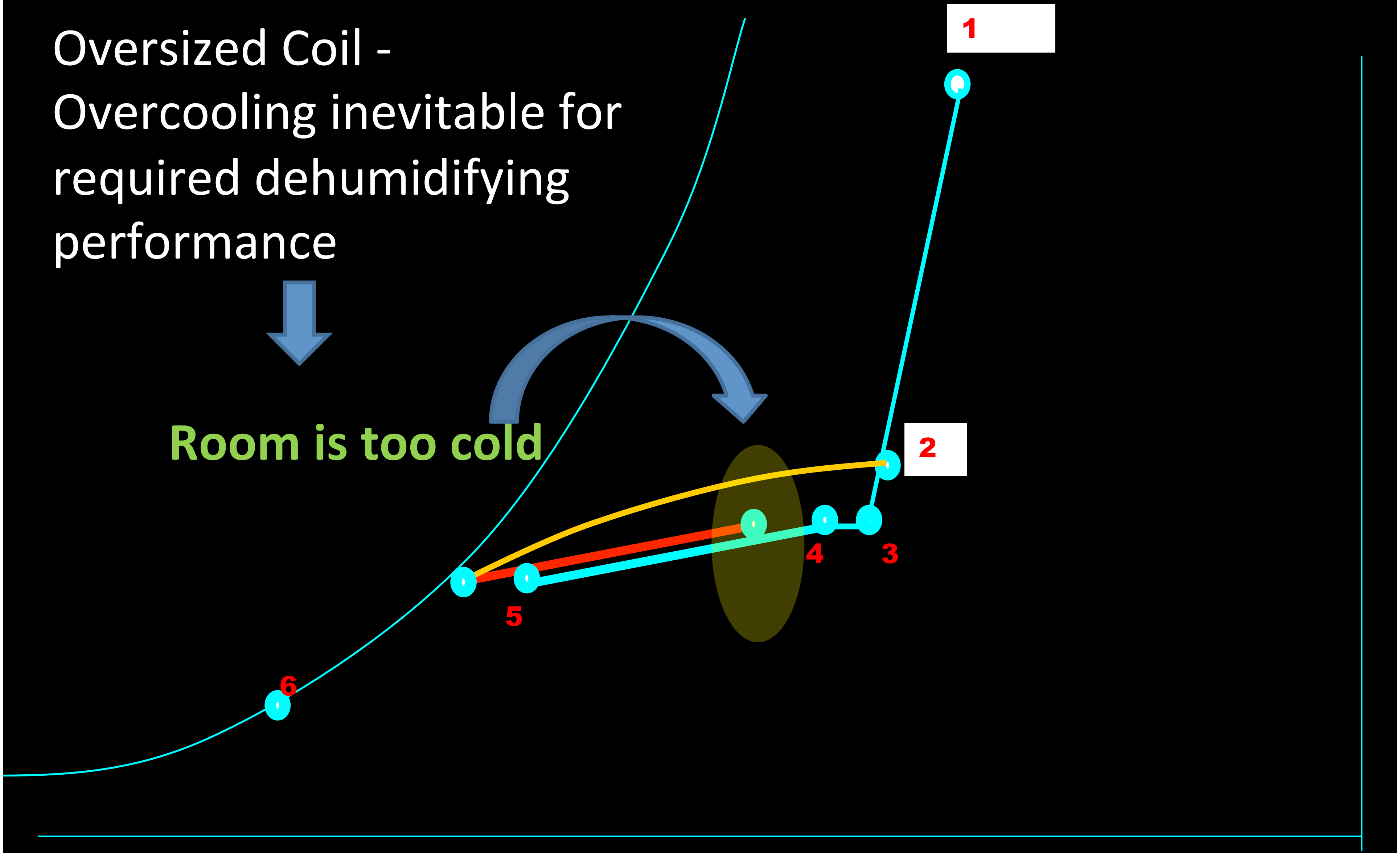

1
2
Coil Condition Curve
Slope = SHR of coil
4
3
5
6
Room Sensible Heat Ratio

Oversized Coil - Unable to Dehumidify adequately for the same ΔT



Oversized Coil - Overcooling inevitable for required dehumidifying performance

Room is too cold



2 Would raising the indoor set point temperature in overcooled buildings be a viable solution?

Air-conditioning – energy penalty in hot and humid climate

- **40-60% of total energy consumption in buildings**

Rooms maintained cold - 21°C to 23°C

Dehumidification challenge

High energy costs

Environmental sustainability

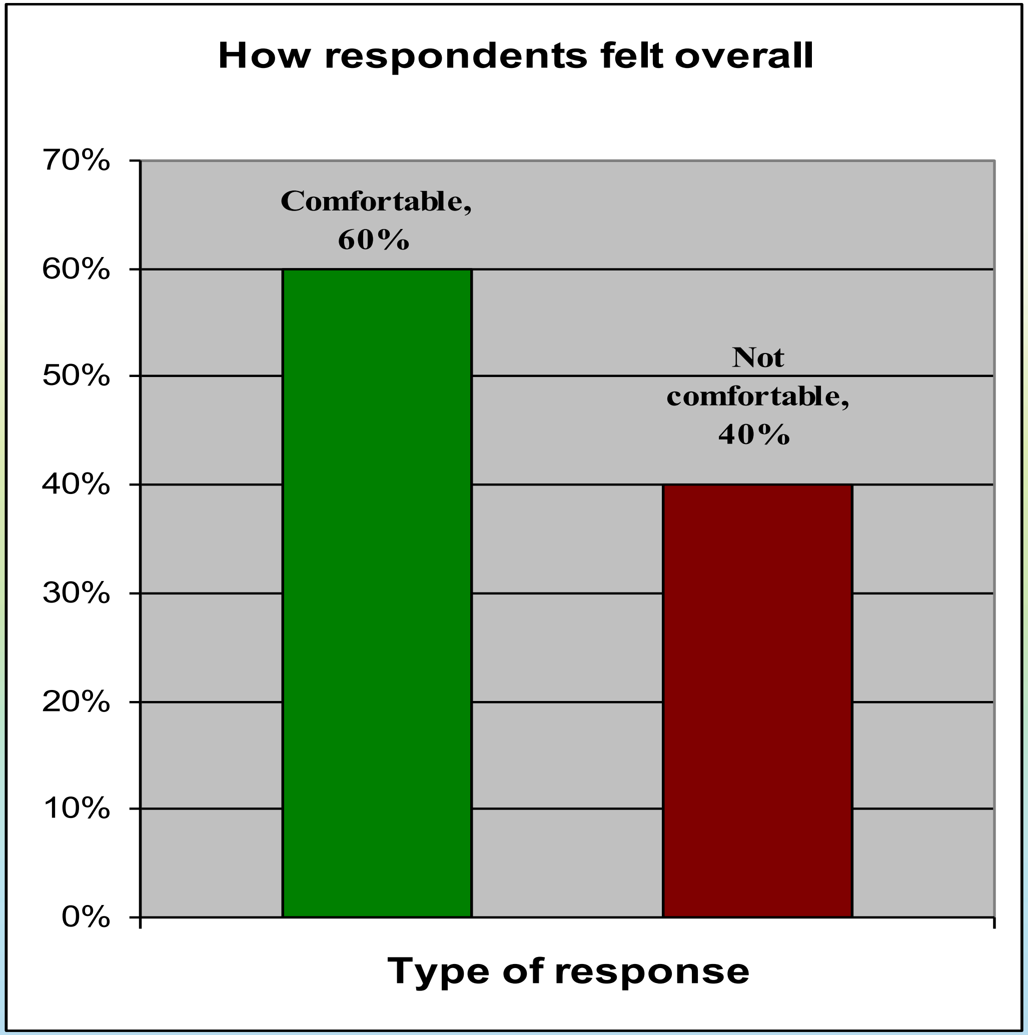
Elevated space temperatures

25°C

METHODS

- **Occupants in 7 offices involved in the study**
- **ASHRAE 7-point scale used for thermal sensation**
- **Total of 146 respondents participated**
- **Measurements of temperature, RH and air velocity**
- **For each office – measurements taken at several spots & two different windows of time 11am – 12nn and 3pm – 4pm**
- **Average PMV and PPD values calculated for each office**

RESULTS



Overall response from all occupants in all offices

Objective measurements of thermal comfort parameters

	11am – 12nn			3 – 4pm		
	Dry Bulb Temp (°C)	Relative Humidity (%RH)	Air velocity (m/s)	Dry Bulb Temp (°C)	Relative Humidity (%RH)	Air velocity (m/s)
A	24	≈50	<0.1	25-25.6	≈50	<0.1
B	24.1	50	<0.1	25.3	≈50	<0.1
C	24.1-24.6	50	<0.1	26.1-27.4	51	<0.1
D	24.4	51	<0.1	24-25	52	<0.1
E	25.7	53-55	<0.1	27	51	<0.1
F	25.2-27	50-54	<0.1	26.3-27.3	51	<0.1
G	22.3-24.5	54-58	<0.1	22.4-24	54-60	<0.1

Comparison between average PPD values calculated from measurements of thermal comfort parameters and actual responses of staff from survey

	Thermal Comfort Questionnaire Survey *(Only 'Comfortable' is taken as acceptable comfort level)*					Measurements of Thermal Comfort Parameters	
	Comfortable (1)	Slightly uncomfortable (2)	Uncomfortable (3)	Very uncomfortable (4)	Percentage of dissatisfied staff (calculated from survey response)	Average PPD (11am-12pm)	Average PPD (3pm-4pm)
A	7	2	0	0	22.2%	20%	10%
B	9	10	1	1	57%	20%	14%
C	7	2	1	0	30%	20%	5%
D	3	3	0	0	50%	20%	14%
E	7	1	0	0	12.5%	7%	6%
F	20	8	1	0	31%	5%	6%
G	31	20	6	1	46.6%	26%	26%
All	84	46	9	2			

Note : Average PPD is calculated from temperature, airflow and relative humidity measurements taken inside the offices during the survey

Large deviation between theoretical predictions (calculated PPD values) and dissatisfaction expressed by respondents thru questionnaire

Poor air movement

➔ reason for thermal discomfort

OBSERVATIONS

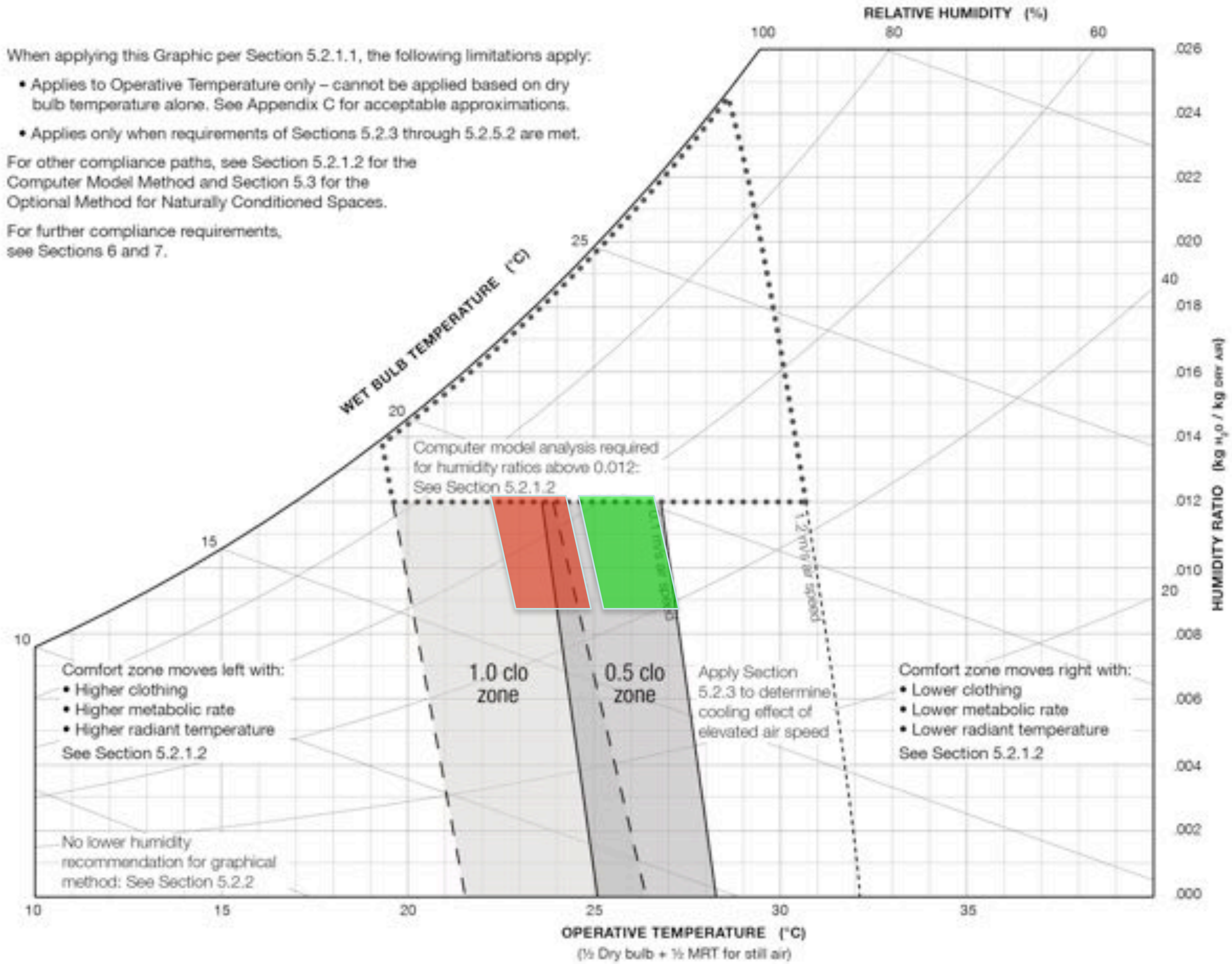
Raising temperature alone does not necessarily Achieve optimal thermal comfort

Humidity needs to be addressed

ASHRAE Standard 55-2013

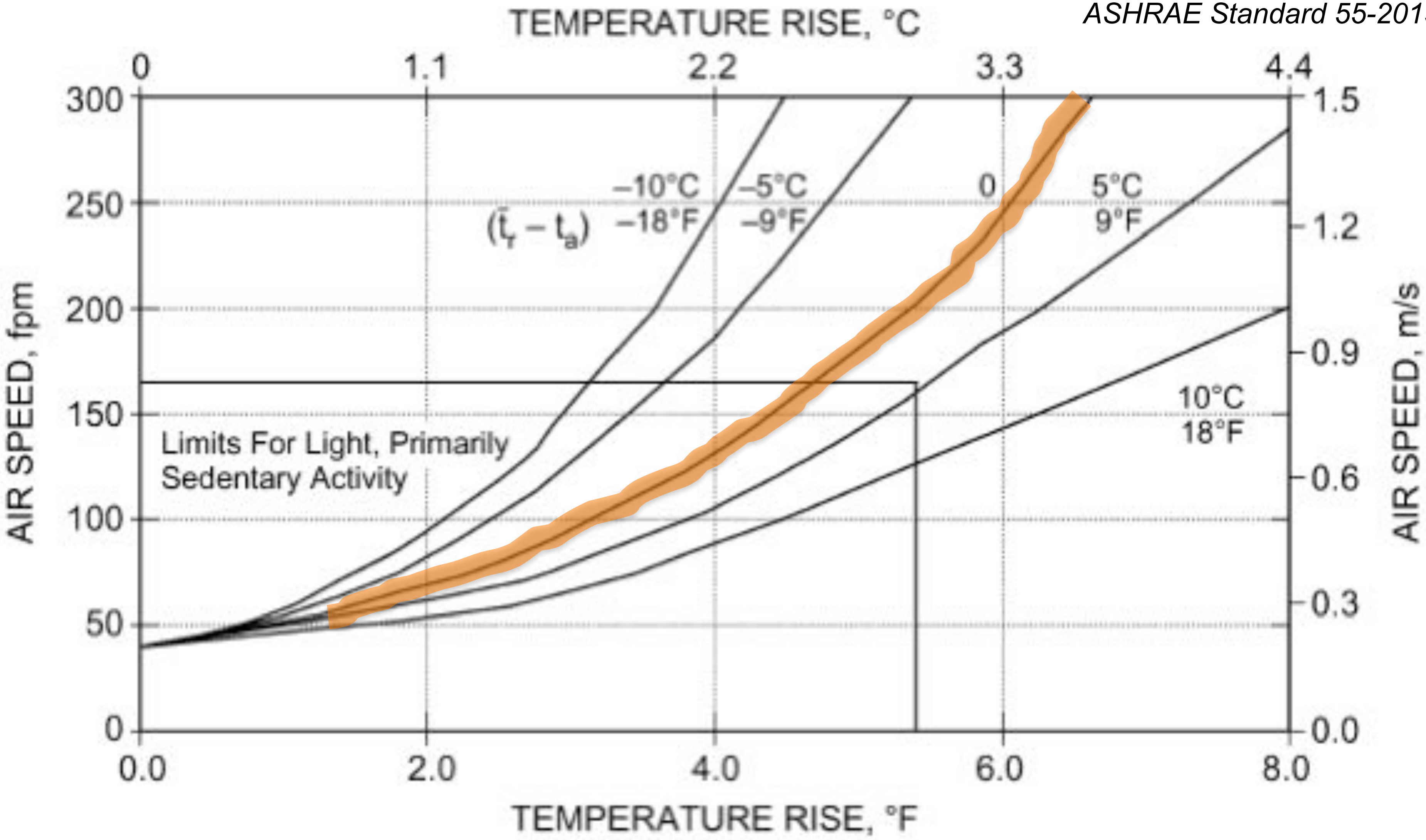
Elevated air speeds at warmer temperatures

Tropically acclimatized subjects do prefer higher air velocities in the range of 0.3-0.9 m/s



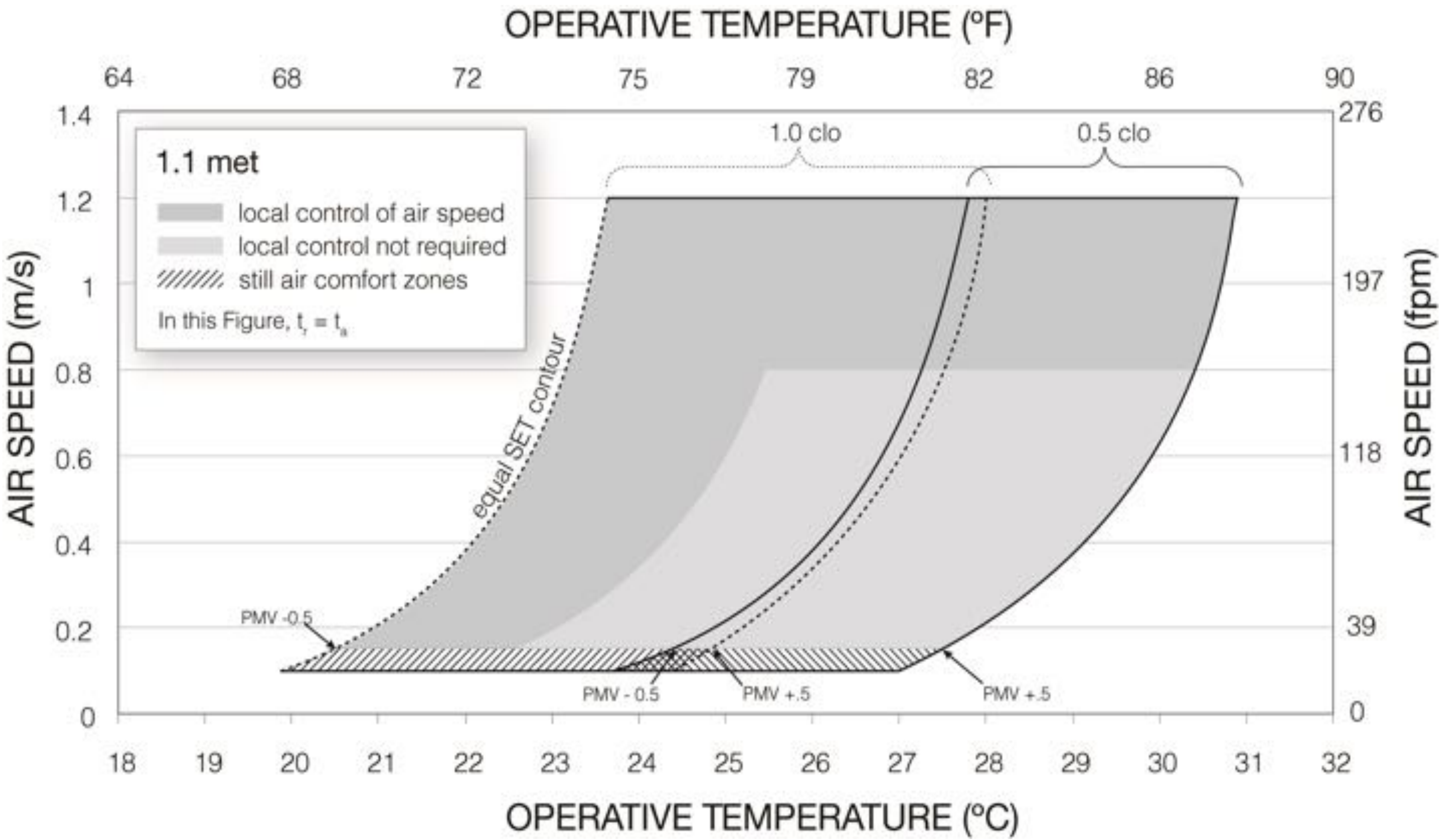
The Graphic Comfort Zone Method: Acceptable range of operative temperature and humidity for spaces that meet the specified criteria (1.1 met; 0.5 and 1.0 clo)

ASHRAE Standard 55-2013



Air speed required to offset increased air and radiant temperature

When the mean radiant temperature is low and the air temperature is high, elevated air speed is less effective at increasing heat loss. Conversely, elevated air speed is more effective at increasing heat loss when the mean radiant temperature is high and the air temperature is low.


OPERATIVE TEMPERATURE (ºF)
64
68
72
75
79
82
86
90
1.4
1.2
1
0.8
0.6
0.4
0.2
0
276
197
118
39
0
AIR SPEED (m/s)
AIR SPEED (fpm)
1.0 clo
0.5 clo
1.1 met
local control of air speed
local control not required
still air comfort zones
In this Figure, $t_r = t_a$
equal SET contour
PMV -0.5
PMV - 0.5
PMV +.5
PMV +.5
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
OPERATIVE TEMPERATURE (ºC)

3

What are the engineering solutions to the problem of overcooling in buildings in hot and humid climates?

Oversized Coil with dynamic change of effective surface area in operation

Cooling coil optimisation in hot and humid climates for IAQ and energy considerations

Oversized Coil with dynamic change of effective surface area in operation

Simulation Approach

Hypothetical building

Actual Maximum Cooling load = 100 kW

Oversized Cooling Coil = 200 kW

RESULTS

No	Parameter	Values
1	Outside Air	32°C DBT & 75% RH
2	Entering coil condition	26°C DBT & 65% RH
3	Return air condition	24.5°C DBT & 60% RH
4	Space condition	24°C DBT & 63% RH
5	Leaving coil condition	13°C DBT & 12.5°C WBT
6	Chilled water supply temperature	6°C

DBT – Dry Bulb Temperature
WBT – Wet Bulb Temperature
RH – Relative Humidity

Oversized Coil – Compared with Dynamically Varying Coil

		Base	Series A1	Series B1	Series C1
Air Side Data	Air Flow (m3/s)	6.45	6.45	6.45	6.45
	Face Velocity	2.52	2.52	2.52	2.52
	Air off DB	13	18.3	18.4	18.8
	Air off WB	12.5	17.2	17.1	17.1
	Capacity(kW)	200	100	100	100
	SHR (%)	52%	62%	61%	58%
Physical Data	Rows	6	6	4	3
	Fin Density	9	9	9	9
Fluid Side Data	Fluid on Temp	6	6	6	6
	Fluid off Temp	12	14.8	13.2	10.8
	Fluid flow rate (l/s)	7.96	2.7	3.33	4.95
	Actual PD	56.5	13.1	17	17.8

Oversized Coil – Compared with Dynamically Varying Coil

		Base	Series A3	Series C3
Air Side Data	Air Flow (m3/s)	6.45	6.45	6.45
	Face Velocity	2.52	2.52	2.52
	Air off DB	13	20.6	21.3
	Air off WB	12.5	19.2	19.1
	Capacity (kW)	200	50	50
	SHR (%)	52%	87%	75%
Physical Data	Rows	6	6	3
	Fin Density	9	9	9
Fluid Side Data	Fluid on Temp	6	6	6
	Fluid off Temp	12	14	11.5
	Fluid flow rate (l/s)	7.96	1.5	2.18
	Actual PD	56.5	10	6.9

Dehumidifying performance – further improvement

Low Face Velocity – High Coolant Velocity (LFV-HCV) method of air-conditioning

Driving force for condensation – Interface temperature

Low heat transfer coefficient on air side

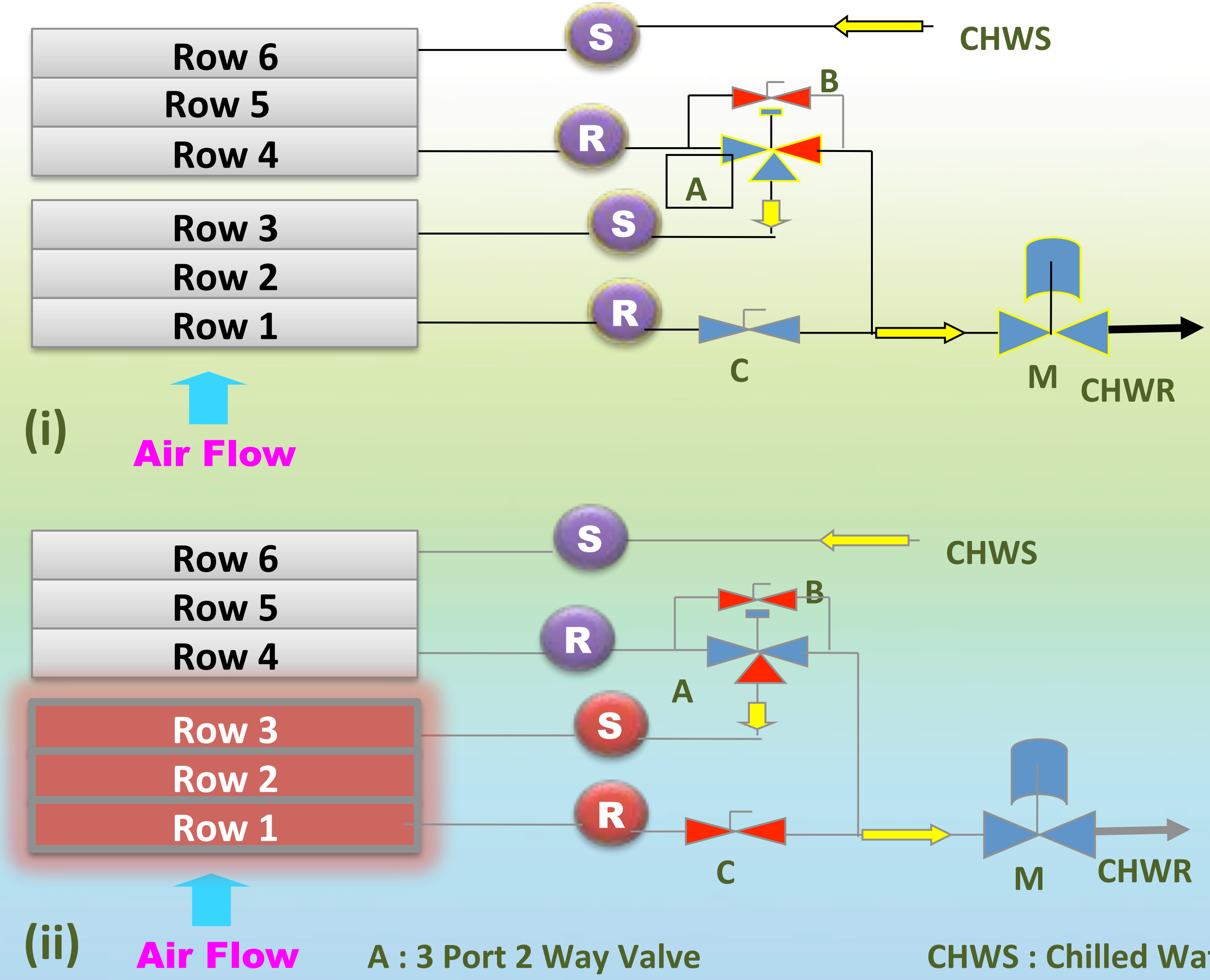
&

High Heat Transfer Coefficient on water side

Low Face Velocity – High Coolant velocity
LFV/HCV

Oversized Coil with dynamic change of effective surface area in operation

LFV-HCV concept – further enhancement



A : 3 Port 2 Way Valve
B : By-Pass Valve
C : On/Off Valve
M : Modulating Valve

CHWS : Chilled Water Supply
CHWR : Chilled Water Return
S : Supply Manifold
R : Return Manifold

Key Findings

- Practical challenge related to the operation of an oversized cooling and dehumidifying coil highlighted
- SHR of the coil used as the basis of measuring dehumidifying performance
- Changing the effective surface area of the coil from 6-rows to 3-rows results in a significant reduction of SHR - particularly in combination with reduced airflows common with VAV systems
- Significant improvement in the dehumidifying performance of the oversized coil during its actual operation stages ➔reduction in the energy consumption of the cooling and dehumidification process

Space temperature difference, cooling coil and fan—Energy and IAQ issues

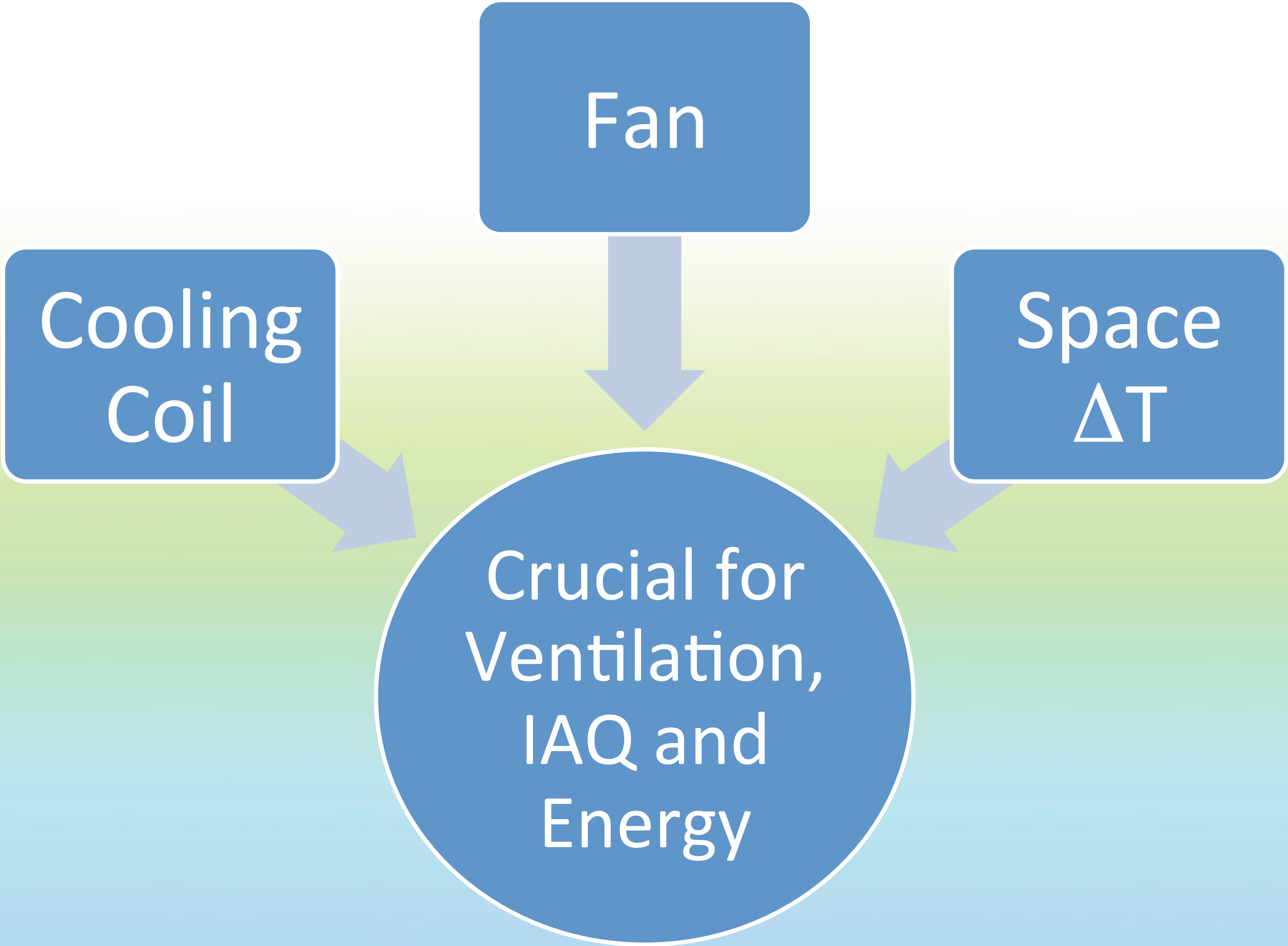
Simulation Approach

Hypothetical building

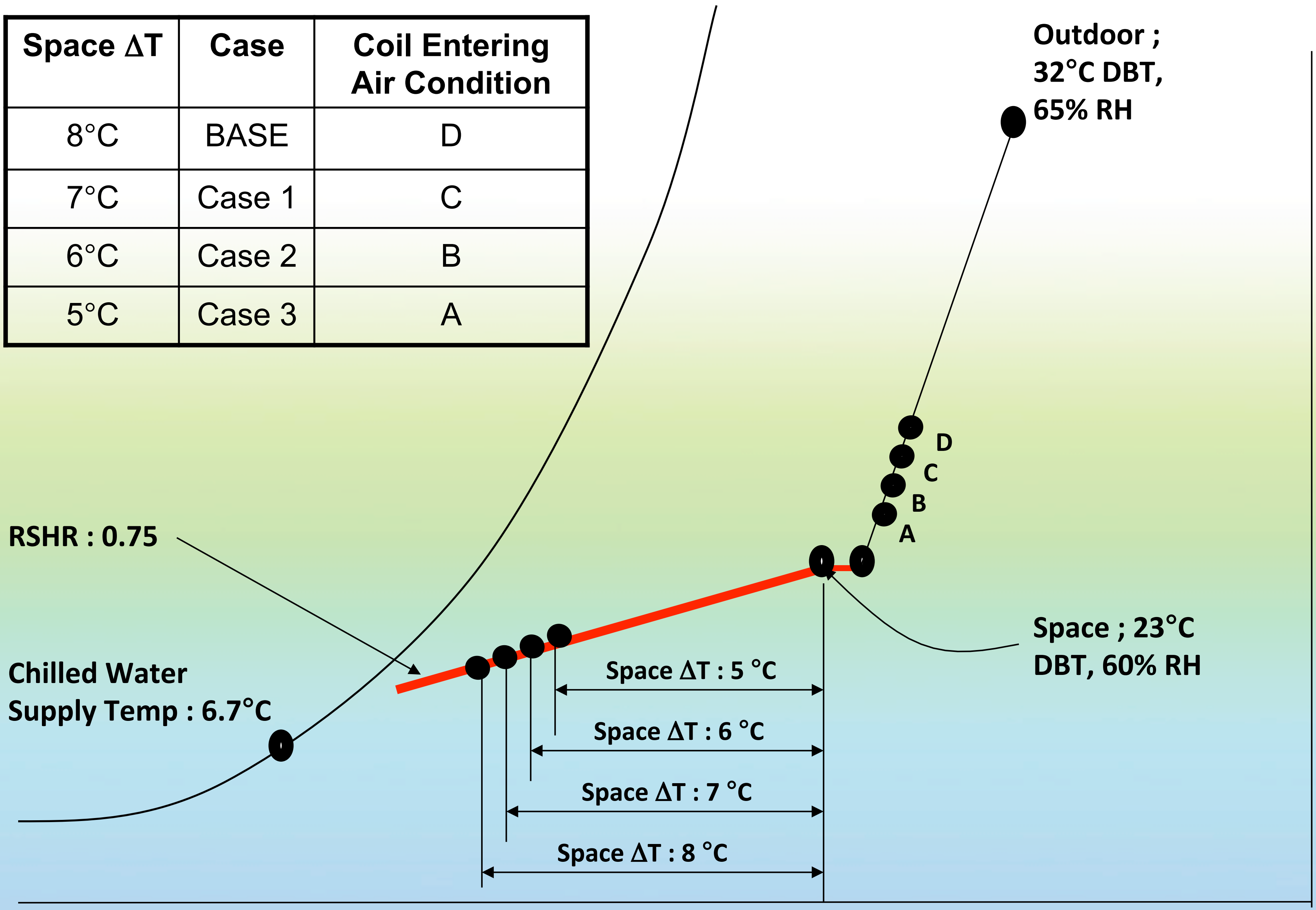
1200 m^2 office space

Space Cooling load : 100 kW

RSHR : 0.75


Fan
Cooling Coil
Space ΔT
Crucial for Ventilation, IAQ and Energy

Space ΔT	Case	Coil Entering Air Condition
8°C	BASE	D
7°C	Case 1	C
6°C	Case 2	B
5°C	Case 3	A

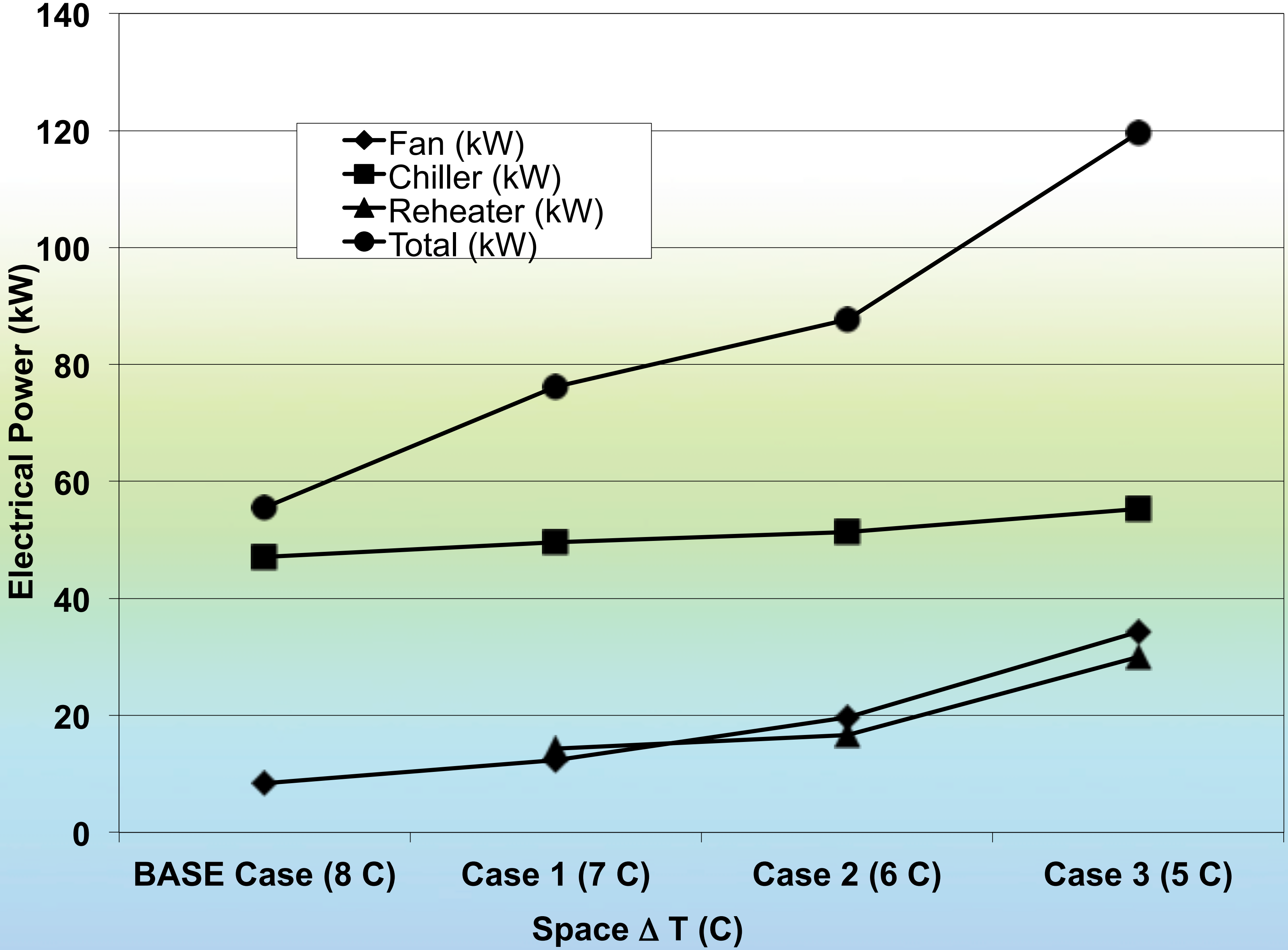


Psychrometric overview of the cases studied

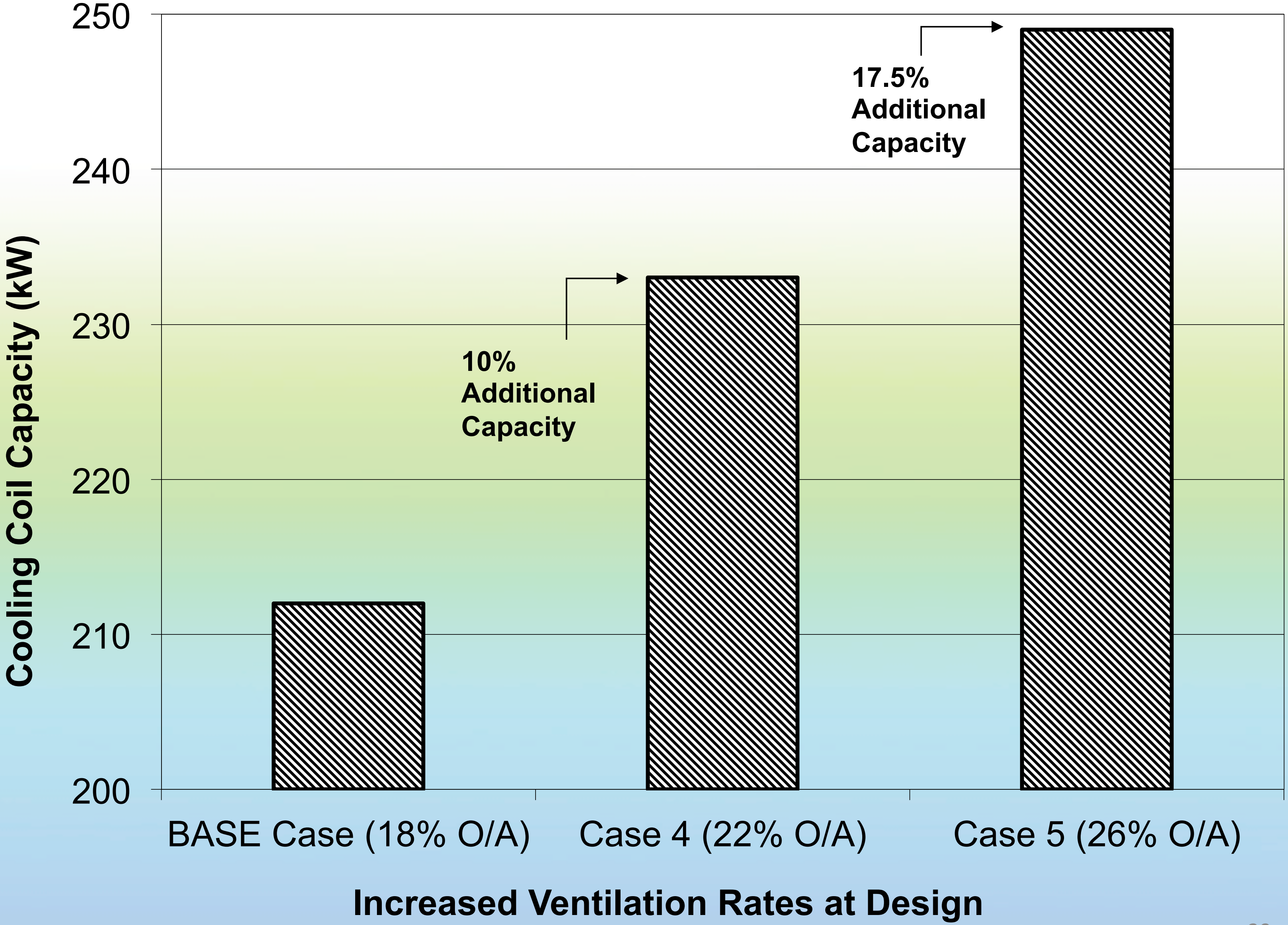
		BASE Case	Case 1	Case 2	Case 3	Increased Ventilation at Peak load	
						Case 4	Case 5
Space total cooling load (kW)		100	100	100	100	100	100
Room Sensible Heat Ratio - RSHR		0.75	0.75	0.75	0.75	0.75	0.75
Space DBT(°C)		23	23	23	23	23	23
Space Relative Humidity (%)		60	60	60	60	60	60
Space temperature difference – Space ΔT (°C)		**8**	**7**	**6**	**5**	**8**	**8**
Entering air DBT (°C)		D 25.3	C 25.1	B 25	A 24.9	25.7	26
Entering air WBT (°C)		19.8	19.6	19.4	19.2	20.3	20.7
Leaving air DBT (°C)		15	15	16	16.5	15	15
Leaving air WBT (°C)		14.1	14.3	14.7	15	14.1	14.1
Air Volume (m^3/s)		**10.3**	**11.74**	**13.7**	**16.5**	**10.3**	**10.3**
Outdoor air Percentage (%)		**18**	**15.8**	**13.5**	**11.2**	**22**	**26**
Face velocity (m/s)		2.48	2.48	2.48	2.48	2.48	2.48
Air pressure drop (Pa)		151	151	129	144	129	129
Cooling coil capacity (kW) (inclusive of overcooling, if any)		**212**	**223**	**231**	**249**	**233**	**249**
Reheat (kW)		--	14.3	16.7	30	--	--
Chilled water supply temperature (°C)		6.7	6.7	6.7	6.7	6.0	6.0
Chilled water return temperature (°C)		13	13.4	13.6	14.1	12.2	12.6
Chilled water flow rate (lps)		8	8	8	8	9	9
Water pressure drop (kPa)		32	17.9	39	17.4	39.5	39.5
Coil Geometry	Tube diameter (mm)	16	16	16	16	16	16
	Tubes high	32	48	32	64	32	32
	Finned height (mm)	**1299**	**1947**	**1299**	**2594**	**1299**	**1299**
	Finned length (mm)	**3200**	**2430**	**4250**	**2560**	**3200**	**3200**
	Fin density (fins/inch)	**11**	**11**	**9**	**6**	**9**	**9**
	Circuiting	FULL	FULL	FULL	FULL	FULL	FULL
	Number of rows	4	4	4	6	4	4

Psychrometric and coil performance parameters for various space ΔTs

Space ΔT	Supply Air DBT	Total Air Flow	Increase in total air flow	Outdoor Air	Fan Power		Cooling Energy (COP=4.5)		Reheat Electrical Power	Total Electrical Power	Additional Energy/ power Required
							Cooling Capacity	Electrical Power			
°C	°C	m^3/s	%	%	hp	kW	kW	kW	kW	kW	%
8	15	10.3	--	18	11.2	**8.4**	212	**47.1**	**--**	**55.5**	--
7	16	11.74	14	15.8	16.5	**12.3**	223	**49.6**	**14.3**	**76.2**	37.3
6	17	13.7	33	13.5	26.36	**19.7**	231	**51.3**	**16.7**	**87.7**	58
5	18	16.5	60	11.2	46	**34.3**	249	**55.3**	**30**	**119.6**	115.5



140
120
100
80
60
40
20
0
Electrical Power (kW)
Fan (kW)
Chiller (kW)
Reheater (kW)
Total (kW)
BASE Case (8 C)
Case 1 (7 C)
Case 2 (6 C)
Case 3 (5 C)
Space ∆ T (C)


250
240
230
220
210
200
Cooling Coil Capacity (kW)
17.5% Additional Capacity
10% Additional Capacity
BASE Case (18% O/A)
Case 4 (22% O/A)
Case 5 (26% O/A)
Increased Ventilation Rates at Design

KEY FINDINGS

- Total power requirements, comprising overcooling, reheating and increased fan power increases significantly as the Space ΔT decreases from 8 to 5°C.
- Total power for a design involving a Space ΔT of 5°C can be as high as a factor of 2.2 of the total power required for a design with a Space ΔT of 8°C.
- Implication of higher supply air flow requirements on the sizes of the ducts.
- For a given space cooling load and a given Space ΔT, increased design ventilation rates to address part-load ventilation requirements can lead to an additional installed cooling capacity of 17.5%.
- This implies a larger than desired effective surface area of the cooling coil which would lead to inefficient dehumidifying performance at part-load operating conditions.
- Separate tracks for ventilation/outdoor air and recirculation air desirable for both energy efficiency and IAQ.

CONCLUSIONS

1

Why are air-conditioned buildings in hot and humid climates so cold?

This is due to oversized design of AHU, particularly an oversized cooling coil. An oversized coil will tend to provide less dehumidification unless "overcooling" is employed. The situation worsens during PART LOADS when considerably more dehumidification is demanded in hot and humid climates.

2

Would raising the indoor set point temperature in overcooled buildings be a viable solution?

Raising the indoor set point temperature in overcooled buildings that is being operated with an oversized cooling coil will not work due to the issues addressed in Q1. The problem is still essentially one of handling dehumidification requirements using a large coil.

3

What are the engineering solutions to the problem of overcooling in buildings in hot and humid climates?

A cooling coil that has the capability of dynamically varying its effective surface area is one possible solution, as this would help to achieve enhanced dehumidification without having to overcool. Coil optimisation varying the water-side parameters and optimisation of Space ΔT and air flow quantities are also options. Another possible solution is to decouple the requirements of VENTILATION from that of COOLING.

Thank You for your Attention

Q & A

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