

ASHRAE Distinguished Lecturer Talk

**ASHRAE India Chapter, New Delhi
18 June 2014**

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

**Professor Chandra Sekhar
PhD, Fellow ASHRAE, Fellow ISIAQ
Department of Building, School of Design and Environment**





ASHRAE is a Registered Provider with The American Institute of Architects Continuing Education Systems. Credit earned on completion of this program will be reported to CES Records for AIA members. Certificates of Completion for non-AIA members are available on request.

This program is registered with the AIA/CES for continuing professional education. As such, it does not include content that may be deemed or construed to be an approval or endorsement by the AIA of any material of construction or any method or manner of handling, using, distributing, or dealing in any material or product. Questions related to specific materials, methods, and services will be addressed at the conclusion of this presentation.



EDUCATION PROVIDER

Prof CHANDRA SEKHAR

is a USGBC Education Provider committed to enhancing the professional development of the building industry and LEED Professionals through high-quality continuing education programs.

As a USGBC Education Provider, we have agreed to abide by USGBC-established operational and educational criteria, and are subject to course reviews and audits for quality assurance.

**GBCI
CMP**

**Oversized Air-Conditioning Systems and
Overcooled Buildings in Hot and Humid Climates**

Approved for **2** GBCI CE Hours for LEED Professionals.

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

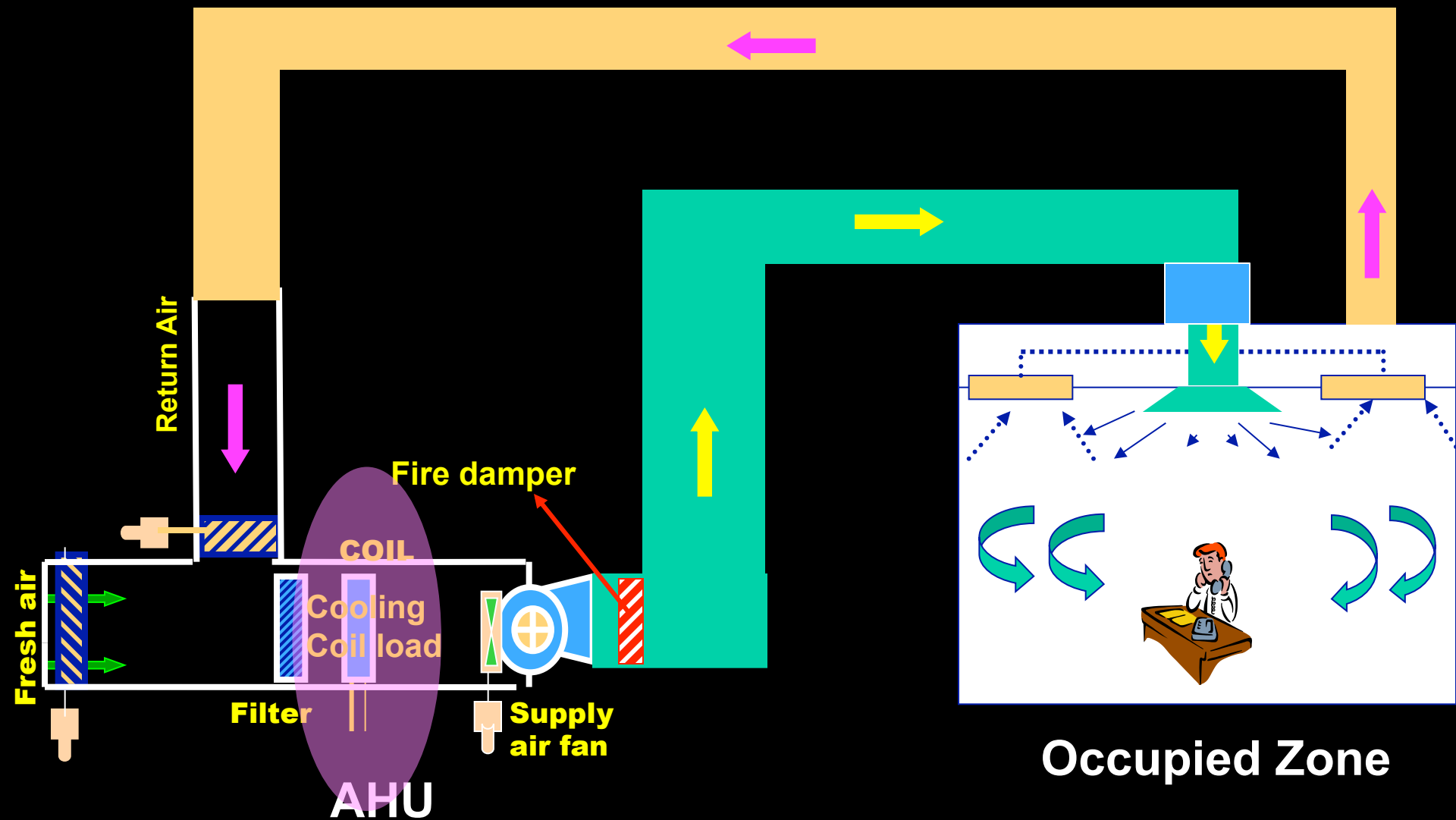
- Why are air-conditioned buildings in hot and humid climates so cold?

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

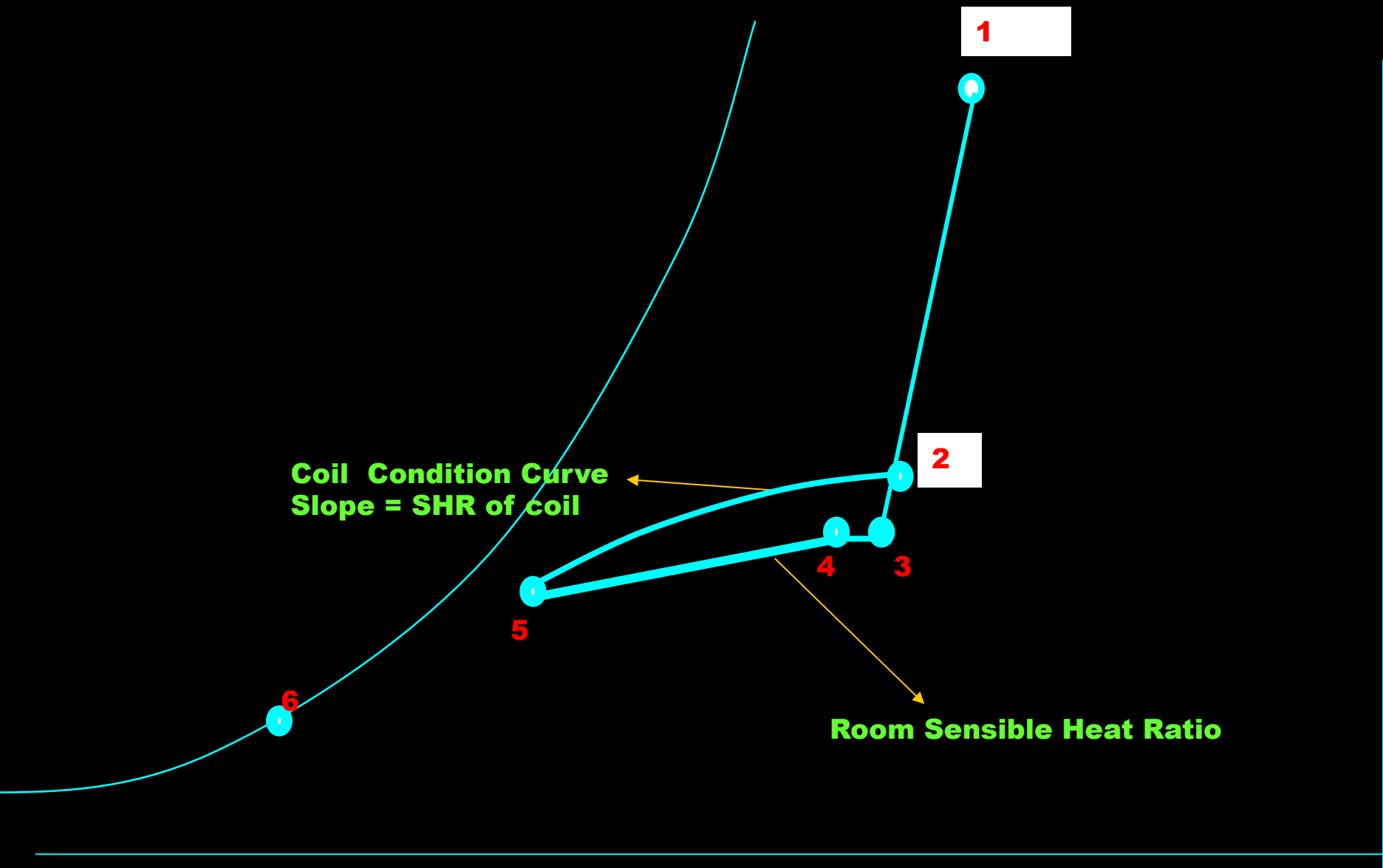


Often leads to problems due to overdesign



Inhibits dehumidifying performance exactly when it is needed to dehumidify more

COIL IS TOO BIG



1

2

3

4

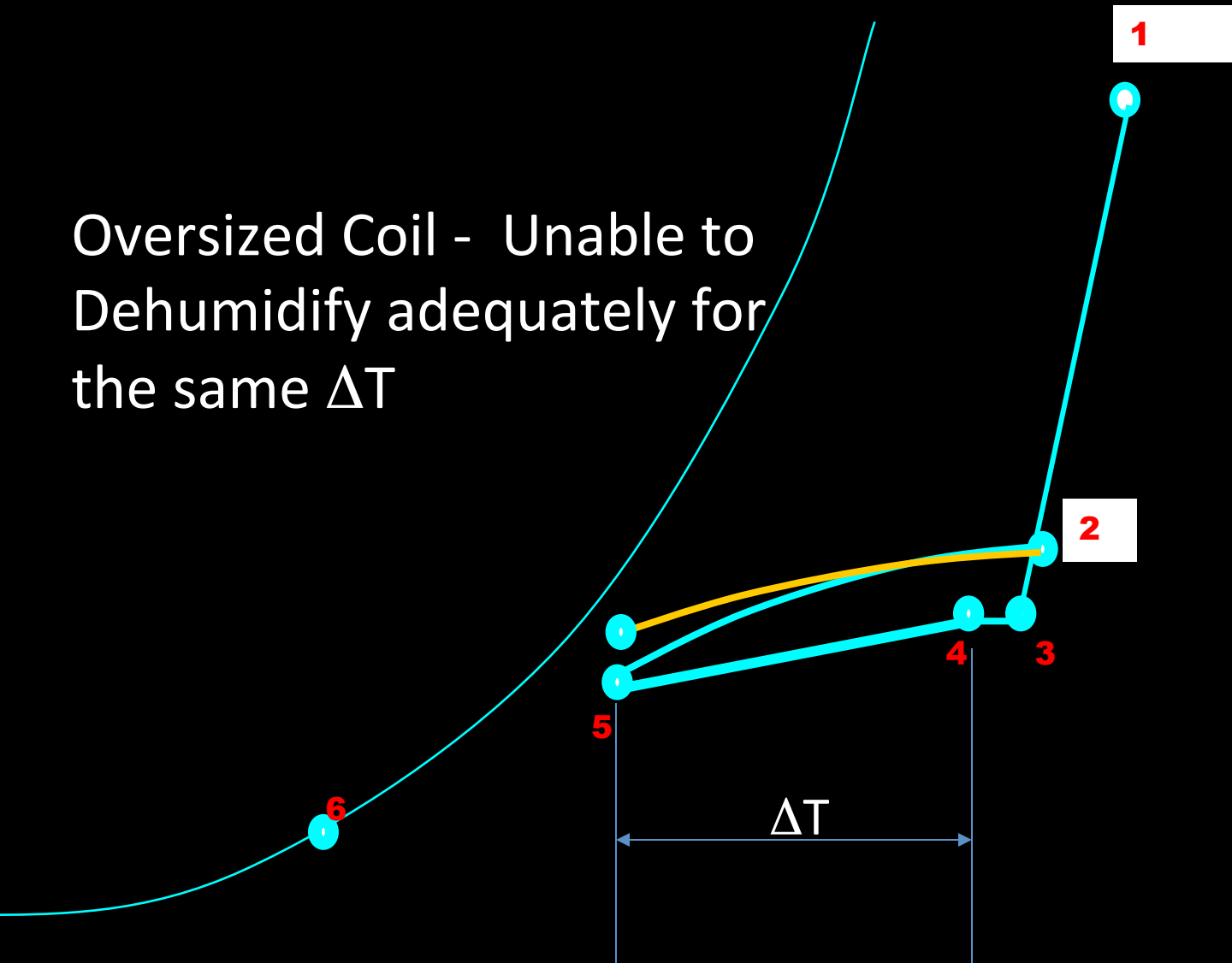
5

6

**Coil Condition Curve
Slope = SHR of coil**

Room Sensible Heat Ratio

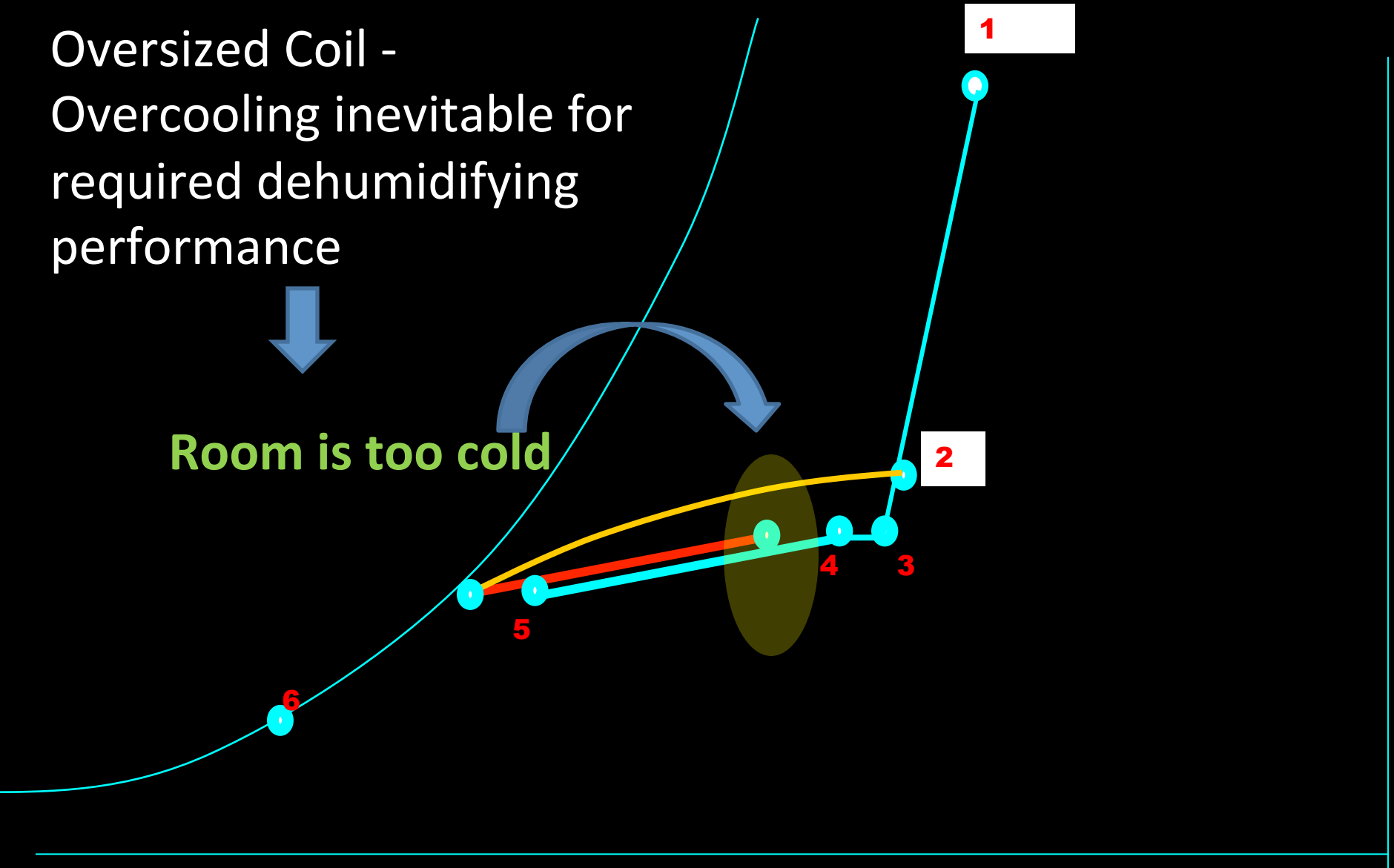
Oversized Coil - Unable to Dehumidify adequately for the same ΔT



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

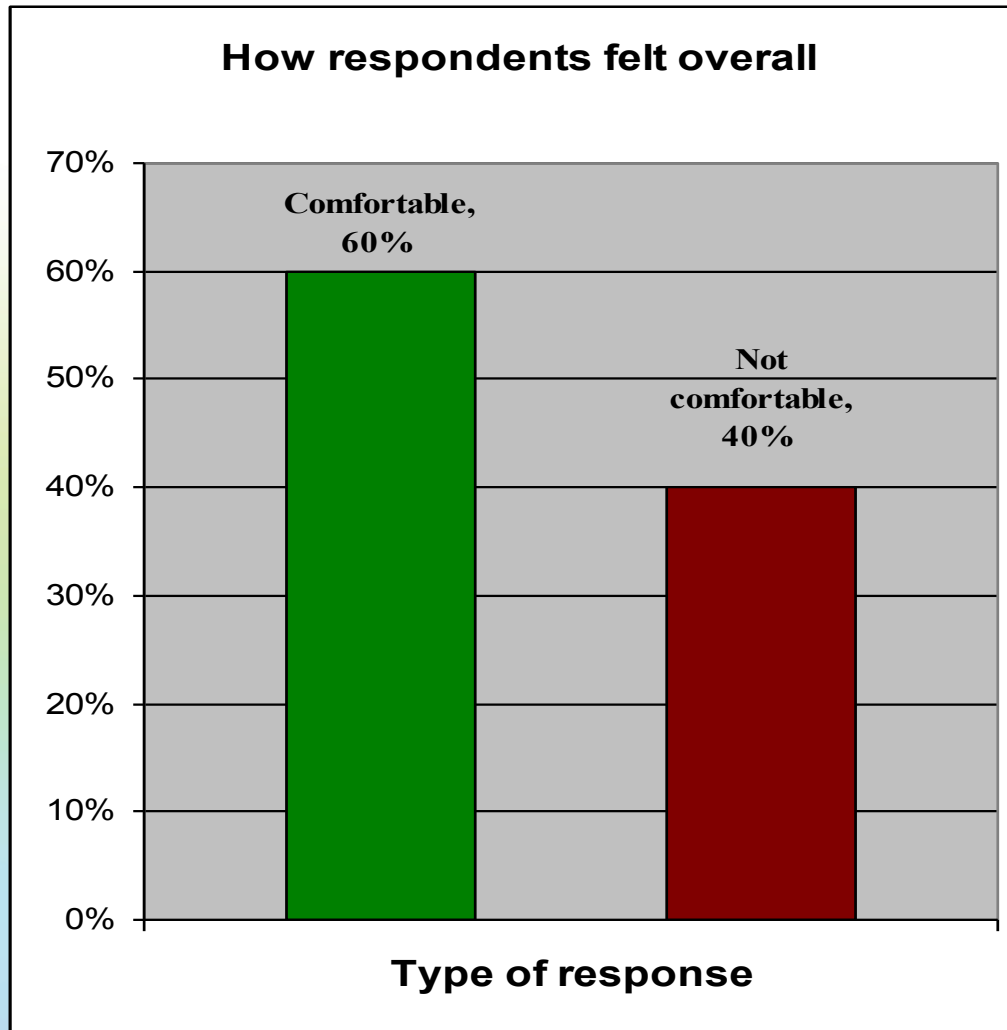
Elevated space temperatures

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 <i>(Only 'Comfortable' is taken as acceptable comfort level)</i>					Measurements of Thermal Comfort Parameters	
	Comfor table (1)	Slightly uncomfor table (2)	Uncomfo rtable (3)	Very uncomfor table (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**

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

Design load = 200kW

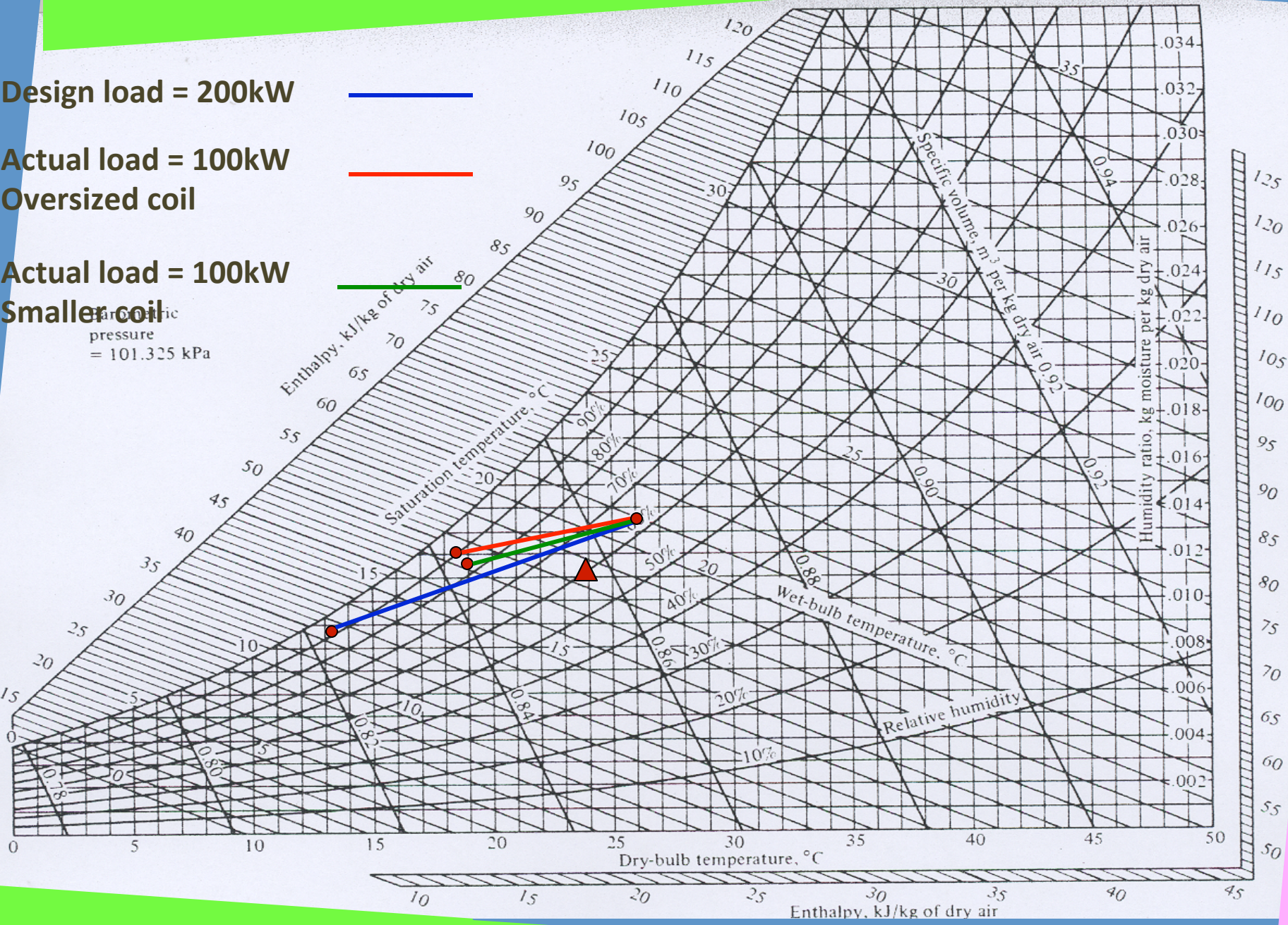
Actual load = 100kW

Oversized coil

Actual load = 100kW

Smaller coil

Static pressure = 101.325 kPa



Design load = 200kW

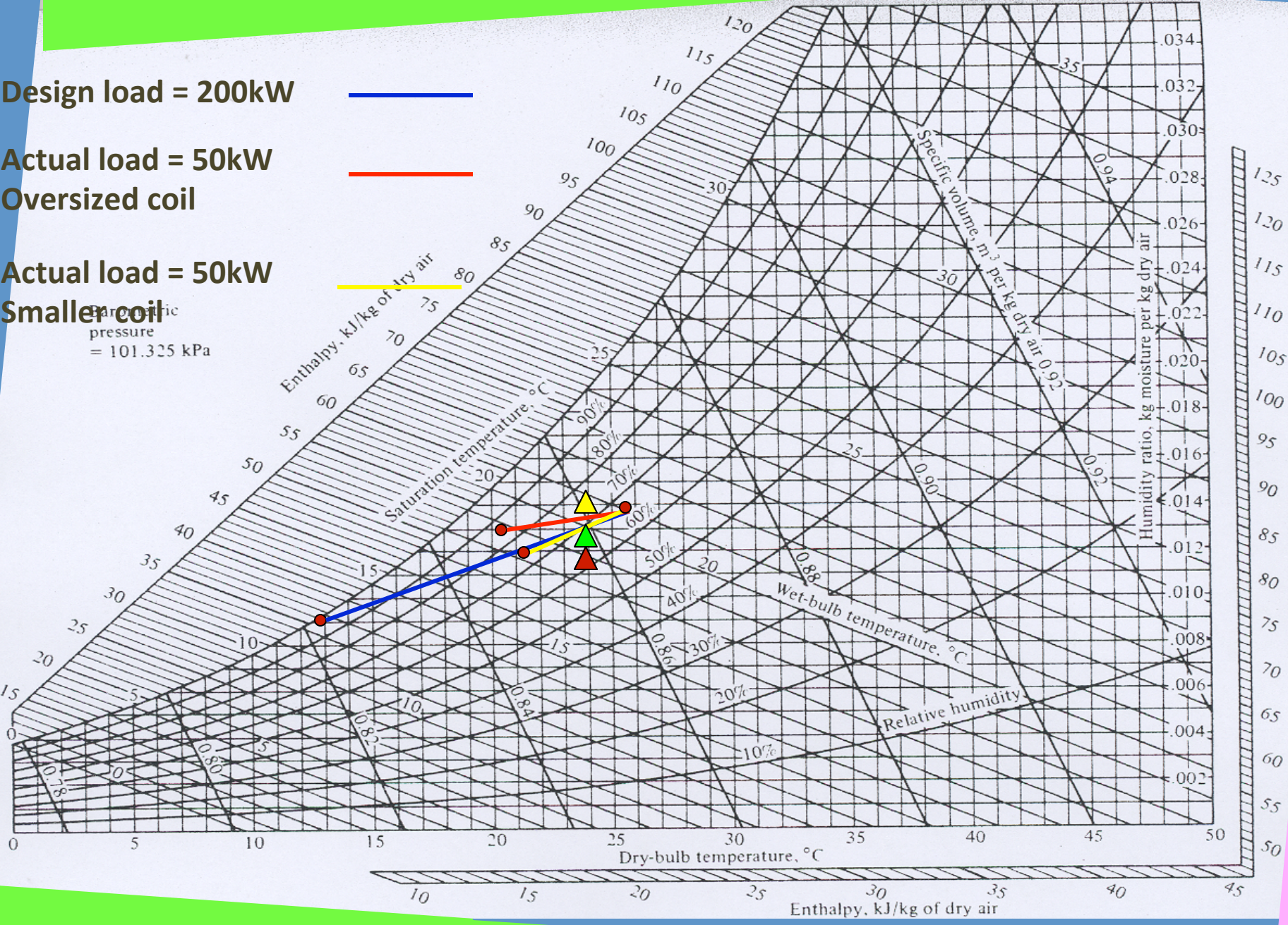
Actual load = 50kW

Oversized coil

Actual load = 50kW

Smaller coil

Barometric pressure
= 101.325 kPa



Blue line

Red line

Yellow line

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

Dehumidifying performance – further improvement

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

**Simultaneous heat and mass transfer
Considerably influenced by air velocity
Thru minimum free flow area of coil**

**Steeper coil condition curve obtained
With Low Face Velocity**

**Local slope of coil condition curve is a
Function of “residence time” – time
For which air has been in contact
With the surface**

**Increased dehumidification per unit
Of sensible cooling**

**Typical low face velocities :
1 – 1.2 m/s**

**Flow resistance significantly reduced
- Fan power reduced**

**Shallower coils
Smaller fin density – wider fin spacing**

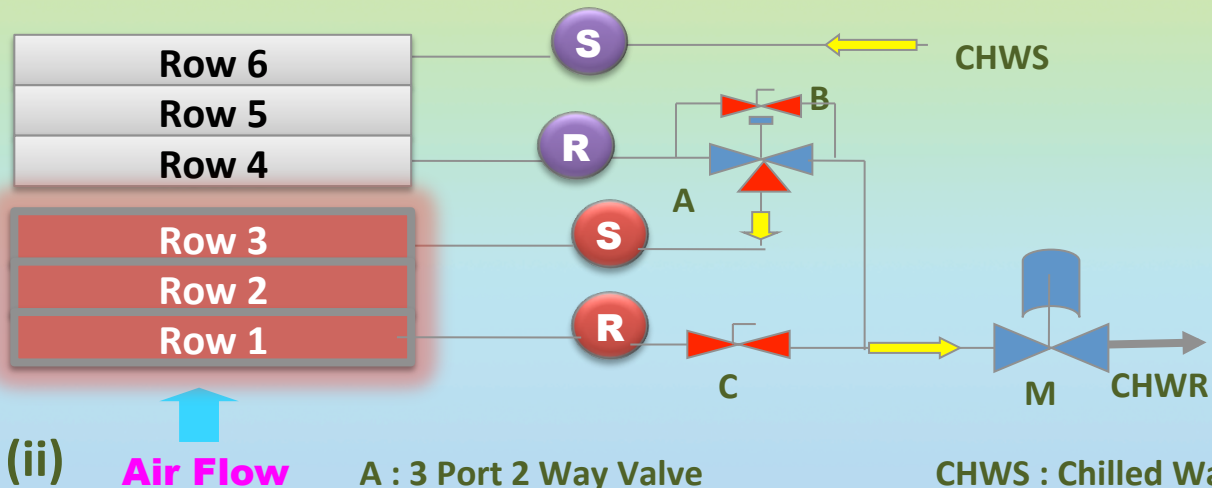
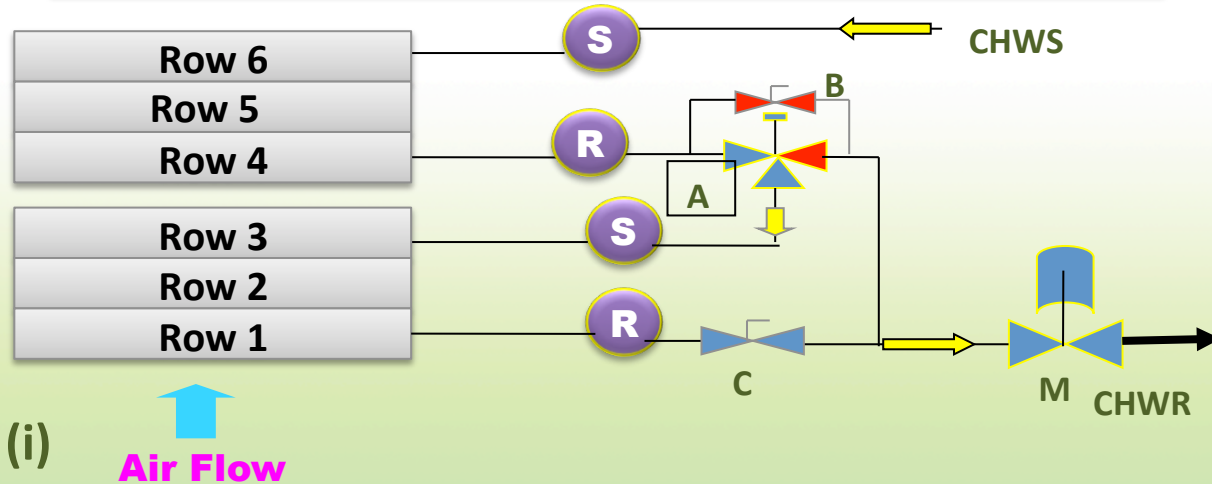
**Driving force for condensation –
Interface temperature**

**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

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



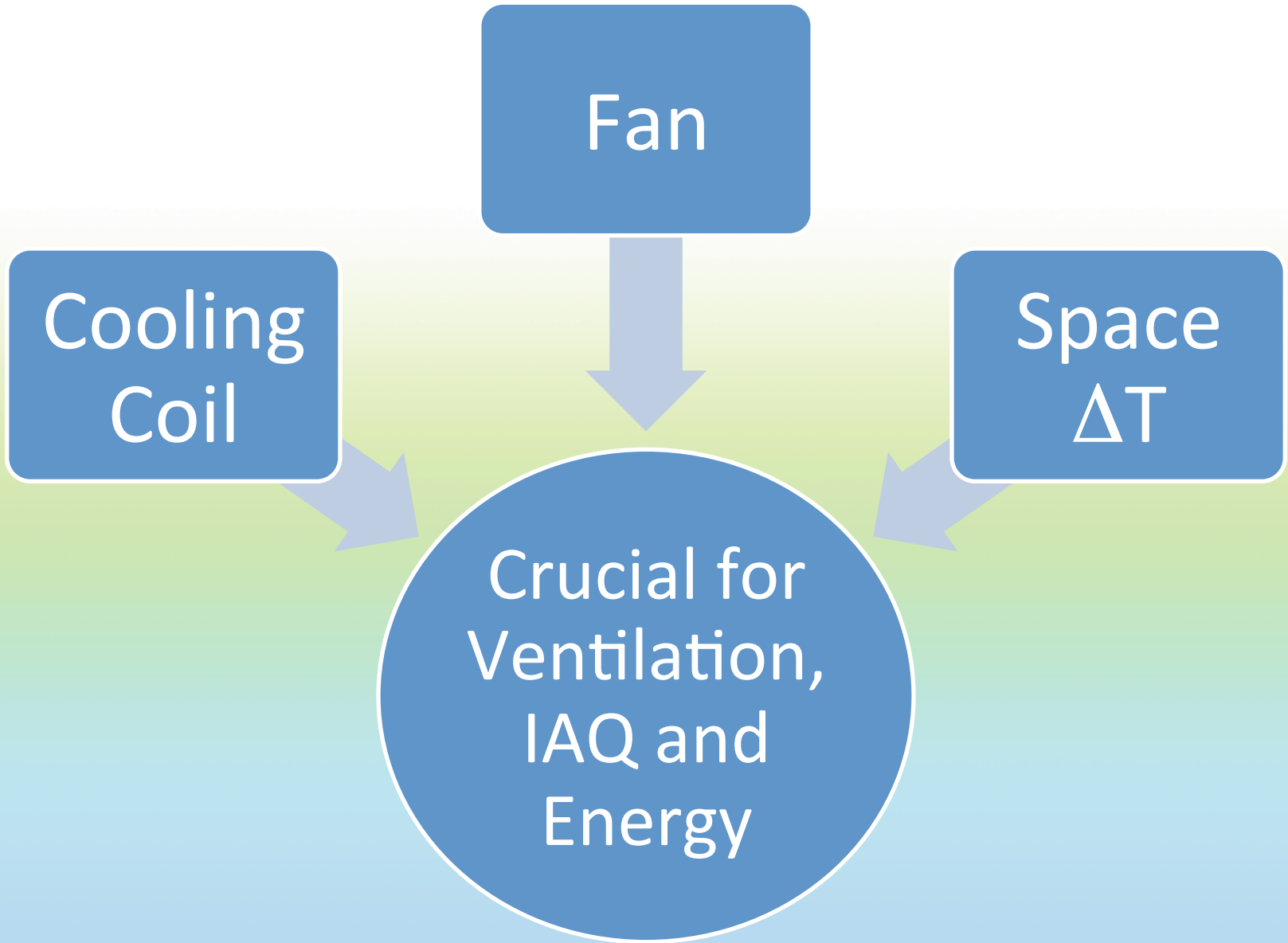
Simulation Approach

Hypothetical building

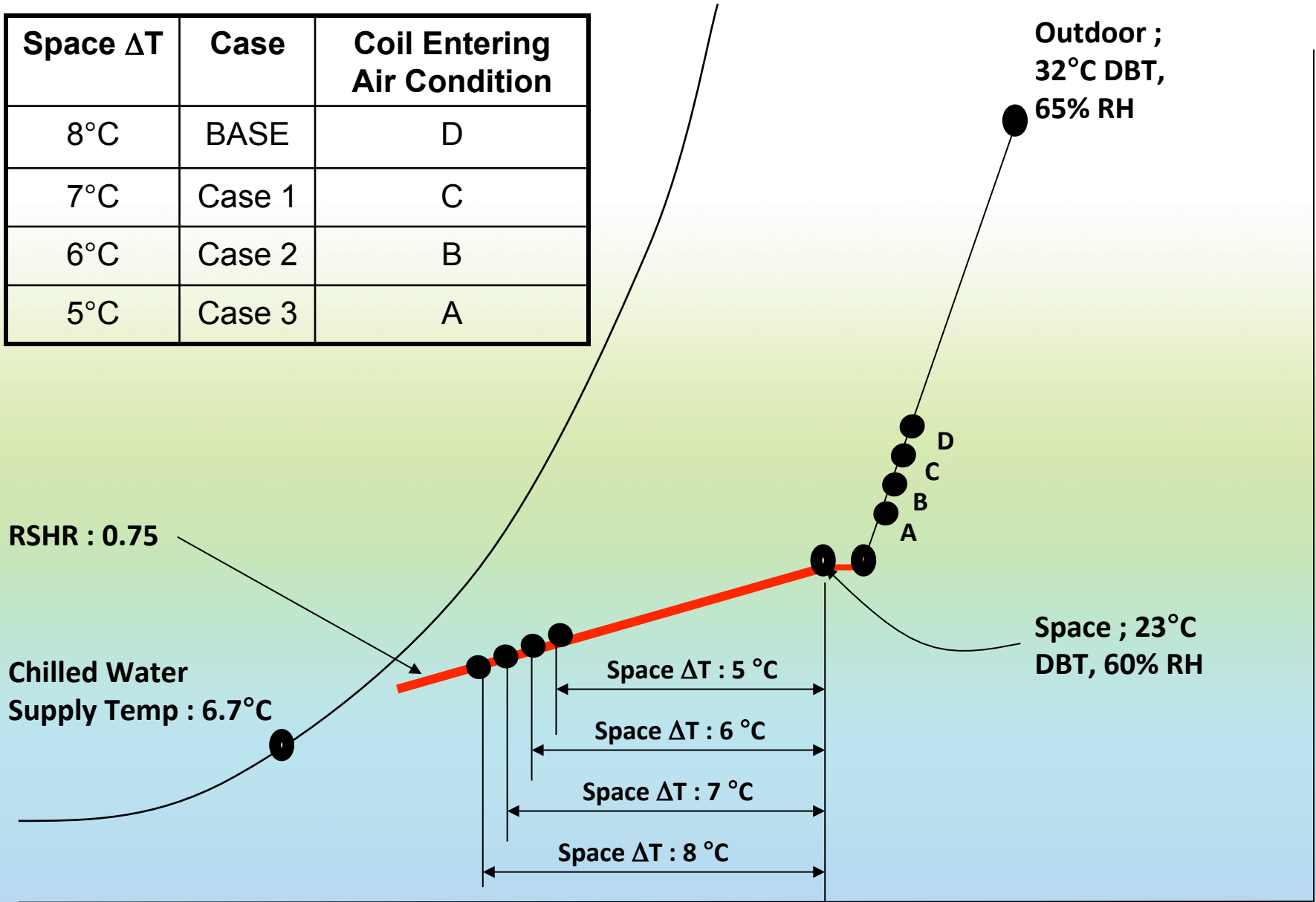
1200 m² office space

Space Cooling load : 100 kW

RSHR : 0.75



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

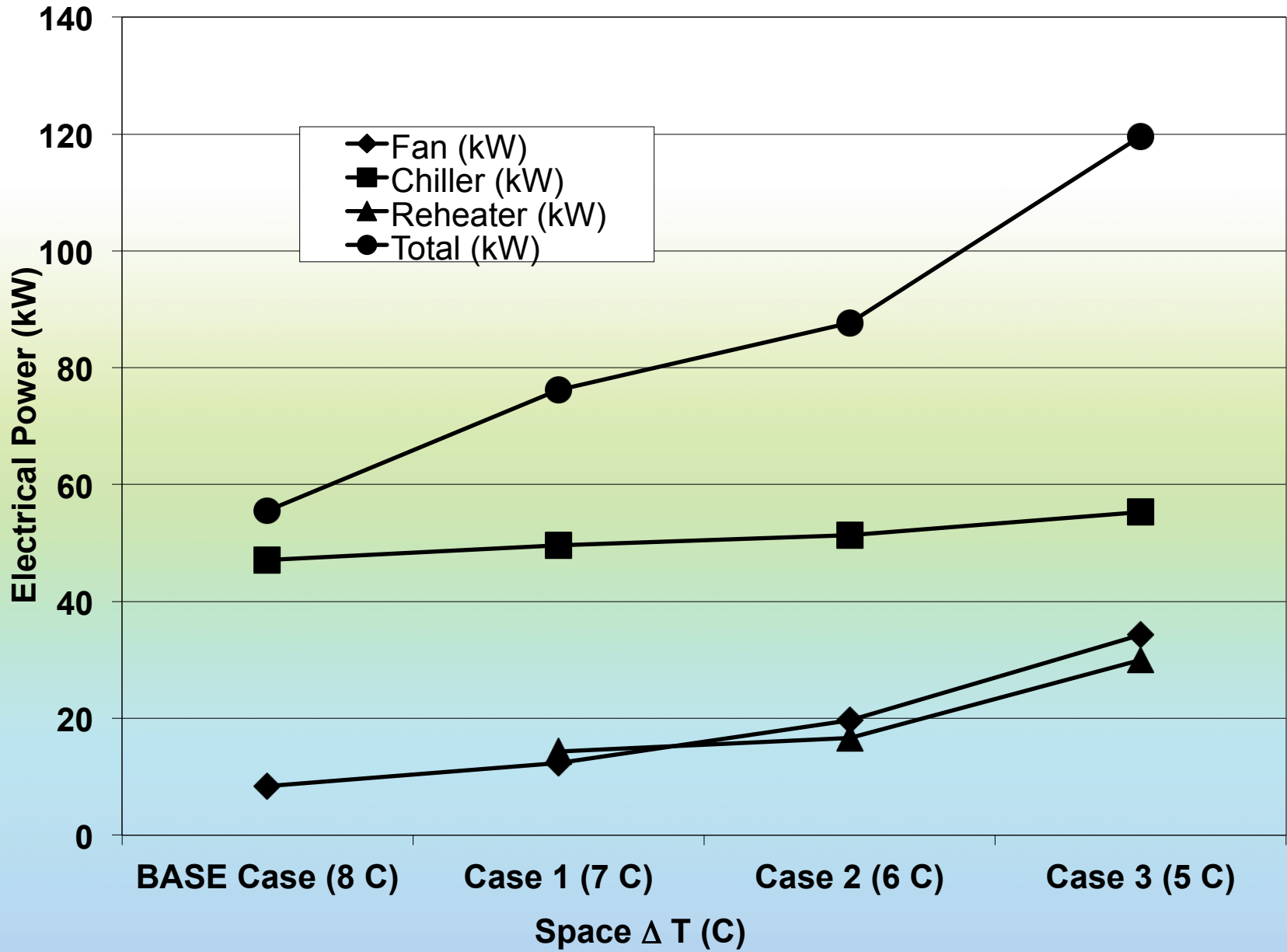


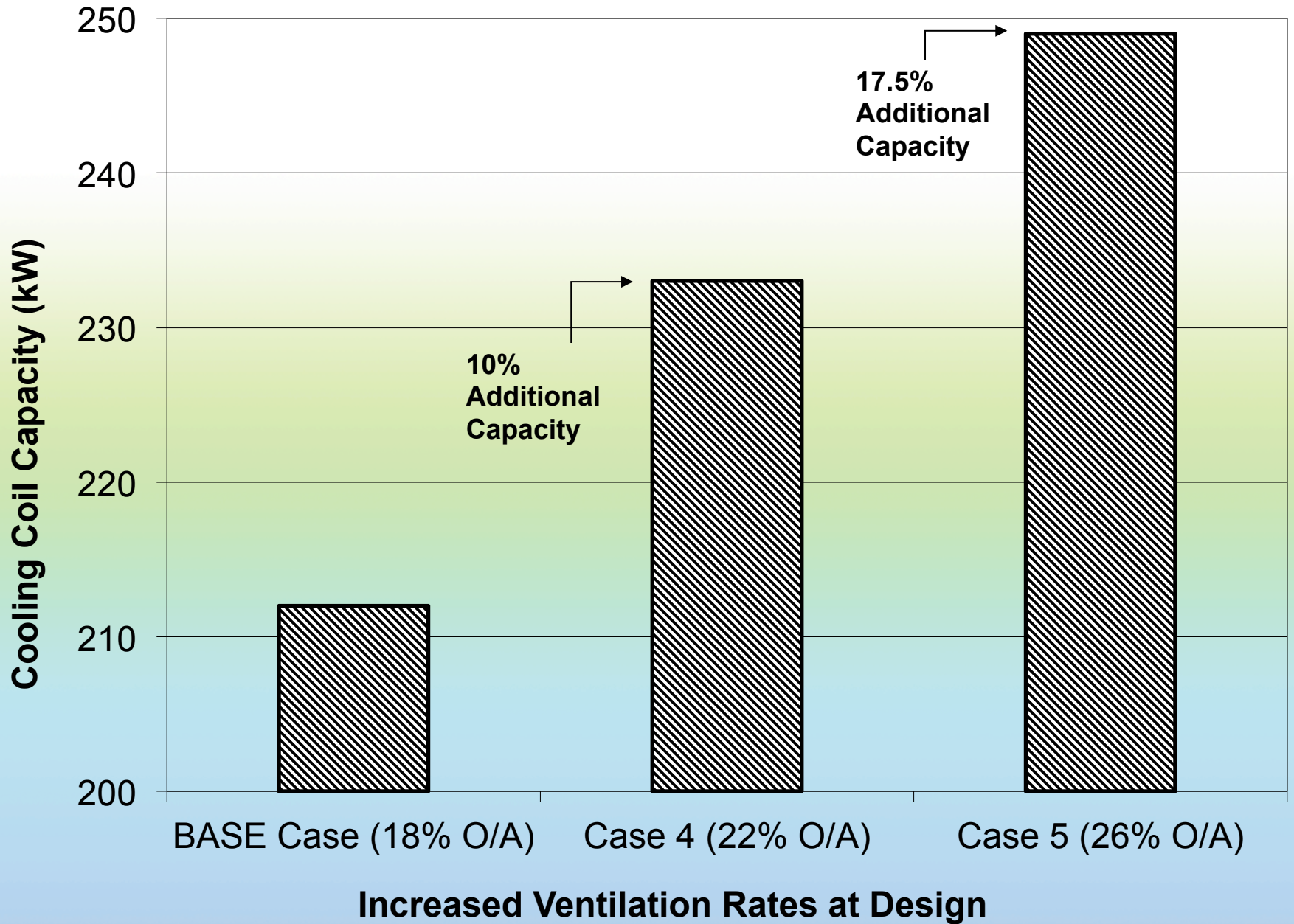
Psychrometric overview of the cases studied

		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 (m³/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 ΔT s

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			
$^{\circ}\text{C}$	$^{\circ}\text{C}$	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





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 Slide

Q & A

