Title
Adaptable cooling coil performance during part loads in the tropics—A computational evaluation

Authors
Sekhar, Chandra
Anand, Prashant
Schiavon, Stefano
et al.

Publication Date
2018

DOI
10.1016/j.enbuild.2017.10.086

Title
Adaptable cooling coil performance during part loads in the tropics - A computational evaluation

Permalink

Authors
Sekhar, Chandra
Anand, Prashant
Schiavon, Stefano
Tham, Kwok Wai
Cheong, David
M. Saber, Esmail

Publication Date
2017-11-02

DOI
10.1016/j.enbuild.2017.10.086

Adaptable cooling coil performance during part loads in the tropics - A computational evaluation

Chandra Sekhar[1], Prashant Anand[*1], Stefano Schiavon[2], Kwok Wai Tham[1], David Cheong[1] and Esmail M. Saber[1,3],

[1] Department of Building, School of Design and Environment, National University of Singapore, Singapore

[2] Center for the Built Environment, University of California Berkeley, Berkeley, CA, USA

[3] Department of Civil and Structural Engineering, University of Sheffield, Sheffield, UK

*Corresponding author: Prashant Anand (a0135589@u.nus.edu)

Abstract

Air conditioning and mechanical ventilation systems may be oversized in commercial buildings in the Tropics. Oversized cooling coils may lead to reduced dehumidifying performance, indoor air quality and thermal comfort and increased energy consumption. In this paper, an adaptable cooling coil design is assessed with a general-purpose coil selection software tool, in which the number of active rows changes as a function of the load. For a 100% oversized coil, it is shown that the adaptable cooling coil is able to provide small but relevant improved humidity control down to 25% of the design load. This was obtained without affecting energy performance in typical variable air volume design and control. For specific applications, where variable air volume systems mainly control space humidity, there are also energy savings. The adaptable cooling coil could be seen as providing additional flexibility in the operation of HVAC systems, particularly in the tropics.

Keywords: *Oversized cooling coils, Part load operational issues, Adaptable cooling coil, Dehumidification performance, Energy Saving*

Nomenclature

ACC	*Adaptable Cooling Coil*
AHU	*Air Handling Unit*
CAV	*Constant Air Volume*
DCV	*Demand Control Ventilation*
HVAC	*Heating Ventilating and Air-Conditioning*
IAQ	*Indoor Air Quality*
RR	*Reduced Rows*
LVF	*Low Face Velocity*
HCV	*High Coolant Velocity*

COP	*Coefficient of Performance*
DBT	*Dry-Bulb Temperature*
SEER	*Seasonal Energy Efficiency Ratio*
SHR	*Sensible Heat Ratio*
SHR_{cc}	*Sensible Heat Ratio of Cooling Coil*
SHR_{cz}	*Sensible Heat Ratio of Cooling Zone*
VAV	*Variable Air Volume*
L_{100}	*Actual Peak Load*
L_{75}	*75% of* L_{100}
L_{50}	*50% of* L_{100}
L_{25}	*25% of* L_{100}
FEG	*Fan Efficiency Grade*

1. Introduction

1.1 Issues with oversized coil design

1.1.1 Humidity issues

HVAC systems in buildings are sized based on peak load conditions to provide comfortable indoor space for occupants under relatively extreme conditions. ASHRAE 90.1 ([1]) recommends oversizing the equipment capacities by 15% for cooling and 25% for heating. Nevertheless, in actual scenarios the building HVAC systems could be oversized by designers/contractors due to poor thumb rule calculations or to avoid customer complaints [2]. Studies show that 89% of designers in Hong-Kong [3] and 51% of designers in California use manufacturing software and 17% use rules of thumb (e.g., 28–32.5 m^2 per ton for general office) and previous experiences to design HVAC systems [4]. Another study from northern California concluded that more than 40% of the commercial building HVAC systems are oversized with a degree more than 25% and about 10% are oversized with a degree more than 50% [5].

These design considerations significantly reduce the ability of HVAC system to control humidity at part load, which could cause poor indoor air quality (IAQ) and likely consequence of mold growth and bad odor. This could further lead to adverse health and comfort related issues of occupants. Indoor air quality (IAQ) issues due to poor dehumidifying performance of an oversized coil is a serious concern for the Tropics where a high latent cooling demand prevails for most of the time [6]. The amount of cooling provided to a space is usually controlled based on dry-bulb temperature measured with a thermostat in the space. Nevertheless, the amount of dehumidification provided by the cooling system may not be enough to satisfy the latent load of the space without having to resort to overcooling. In smaller direct expansion air conditioning units, such as the window unit or the split system unit, the fan operates continuously and the compressor cycles on and off depending on the thermostat set points (sensible control). Therefore, in part-load conditions, the continuous fan operation with a shut compressor elevates the evaporator temperature. This scenario is created when the compressor is cycled to the shut-off stage until the thermostat triggers its operation as a result of the indoor temperature rising above the set point. This causes an increased evaporation of moisture content with an increased temperature difference between coil surface and entering air wet-bulb temperature. This further leads to a higher humidity at zone level [7]. The source of this moisture content could be internal latent loads of the building and highly humid ventilation/infiltration air. The mixed air consisting of room air and ventilation air needs to go through the coil as part of the air-conditioning process and would encounter a coil with elevated evaporator temperature that would result in less dehumidification. Earlier studies related to on-site part-load scenarios have shown that the moisture removal capacity of a cooling coil decreases with a decrease in space-load [8]. This happens even in a case when the supply air fan operates uninterruptedly [9]. Therefore, varying the capacity of the cooling coil during part-load periods may not be sufficient to counter the moisture content in an indoor space.

1.1.2 Ventilation issues

In the VAV system, the amount of supply air varies from peak load to part load while off coil air temperature may remain constant if supply air temperature reset is not used. However, due to ventilation demand of space, the supply air volume cannot be reduced below a minimum requirement set by various standards (ASHRAE Standard 62.1, [10]; Singapore Standard 553, [11]). It is a common practice in office buildings in Singapore to size the air handling unit supply air fan to provide 8 to 10 $L/s\cdot m^2$ of air to the conditioned space at peak load. However, outdoor air only constitutes 10 to 20% of this supply air volume and it is a general practice to not reduce supply air volume below 50% of peak design during part load operation. This limit can be overcome with outdoor air damper control and modern VFD fan controls.

1.1.3 Energy issues

Other consequences of over sizing are increased capital cost of the equipment and, with time, additional maintenance and energy use costs [12]. A study had predicted 0.2% reduction in Sensible Energy Efficiency Ratio (SEER) value for every 1% over-sizing of an air conditioning unit. That is equal to an estimated energy saving of 10% for correcting an average over sizing of 50% [13]. However, correcting equipment-sizing problem with existing system is not a common practice and in most cases, replacing the whole system has been seen to overcome these issues [5]. VAV systems provide better humidity control at part load as compared to Constant Air Volume (CAV) systems as most of the air passes through the cooling coil. However, in existing commercial buildings in Singapore and elsewhere in the tropics, automatic outdoor air dampers are not commonly installed, and therefore, the VAV system is forced to operate like a CAV system at part load, which results in high indoor humidity. Therefore, overcooling may be necessary to bring the humidity level within an acceptable range, which requires additional energy to operate the system. VAV strategy is commonly used in large office spaces due to its superior efficiency, while CAV design has been chosen in some applications because of its simplicity and lower initial cost.

Strategies aimed to boost energy-efficiency (e.g., window and wall insulation, external shading, etc.) may lead to reduced sensible load in a space without affecting the latent load. This may cause more challenging humidity control issues [14].

1.2 Importance of the tube-fin interface temperature for dehumidification

The term sensible heat ratio (*SHR*) is defined as the ratio between sensible and total cooling load or capacity. The SHR_{cc} (*SHR* of the cooling coil) and SHR_{cz} (*SHR* of the conditioned zone) are required to be compatible for an adequate humidity control under full load and part load scenarios. However, at part load conditions, when SHR_{cz} decreases, the SHR_{cc} increases. This mismatch has been evident for residential HVAC systems [15], [16] and [17]. Although this mismatch has not been calculated and reported for commercial buildings, this heat and mass transfer principle will be applicable for all HVAC systems. Coils with higher *SHR* could reduce temperature too quickly for adequate moisture removal to occur, while coils with low *SHR* may dry out the air too much and indoor air may become too dry [18]. A simulation study by [6], found that operating a coil at half its design capacity (from 200 to 100 kW) increases the SHR_{cc} by 10%. This 10% increase in *SHR* can increase the indoor space humidity.

1.2.1 Low Face Velocity/High Coolant Velocity

An oversized coil can perform better in dehumidification at part loads with Low Face airflow Velocity (LFV) and High Coolant Velocity (HCV). When air approaches the cooling coil at a lower velocity, the ratio of latent heat exchange between moist air and chilled water across the coil increases, and hence *SHRcc* decreases. This *LFV* results in better moisture control due to a smaller external heat transfer coefficient (h_o). Additionally, *HCV* improves the internal heat transfer coefficient (h_i). The *LFV* along with *HCV* provides a coil-surface temperature as close as possible to the chilled water temperature. These phenomena can be seen through one-dimensional heat transfer analysis. The sensible heat transfer rate between air and water, Q, can be expressed as:

$$\mathrm{Q} = h_0 \cdot A \cdot (T_a - T_s) \tag{1}$$

And

$$\mathrm{Q} = h_i \cdot A \cdot (T_s - T_w) \tag{2}$$

From Eq. (1) and Eq. (2)

$$(T_a - T_s) = \frac{h_i}{h_0}(T_s - T_w) => \frac{T_s - T_w}{T_a - T_w} = \frac{1}{1+\left(\frac{h_i}{h_0}\right)} \tag{3}$$

Where,

T_s - *Coil interface temperature (°C)*

h_o - *Heat transfer coefficients on the air side (W/m^{2}K)*

h_i - *Heat transfer coefficients on the chilled water side (W/m^{2}K)*

T_a - *Air temperature (°C)*

T_w - *Chilled water temperature (°C)*

The coil-surface temperature decreases as $\frac{h_i}{h_0}$ increases (See Eq. (3)). The difference between the dew-point temperature of the air and the surface temperature of the coil causes condensation. Hence, a low h_o and a high h_i can lower the coil-surface temperature which in turn provides a larger condensation for a given dew point temperature. A lower h_o can be achieved through *LFV* and a higher h_i can be obtained through *HCV* [19].

1.3 Adaptable coils

This study assesses an adaptable cooling coil in which the number of active cooling rows varies as a function of the cooling load. This strategy is based on the work of ([20] and [6]). They found that changing the effective surface area of the coil could improve the *SHRcc* of cooling coil at part load [6]. They concluded that with reduction of coil rows from 6 to 3, there would be a significant decrease in *SHRcc*. This study aims to further develop the proposed adaptable cooling coil concept by assessing its interaction with the VAV system design and control. This simulation has been performed for two oversized cooling coils (200 kW and 150 kW) for two scenarios (Fixed/varying outdoor air percentage) and two possible cases (Fixed/Varying space absolute humidity). Simulation results have been assessed based on the ability to maintain indoor humidity levels and energy consumption.

2. Methodology

A three-step method is employed to assess the performance of the adaptable cooling coil. In the first step, two oversized coils have been designed. The design condition was arbitrarily assumed to be 100 kW. One coil was designed to be 100% oversized (200 kW), the other to 50% oversizing (150 kW). The extent of the oversize is an assumption based on anecdotal evidence where 50% oversizing is assumed to be quite typical of conventional designs in Singapore and 100% is an example of an extreme oversizing, both of which are not uncommon in existing buildings. High oversizing could result in existing buildings when energy conserving operational strategies are employed. Oversized space cooling loads are 126 kW and 95 kW for the oversized cooling coil of capacity 200 kW and 150 kW respectively, whereas, the “actual peak” cooling capacity is 100 kW (for an actual space load of 63 kW). The method by which actual space loads are obtained is described in Section 2.1 of this paper.

In the second step, a hydraulic system and sensors are proposed to be integrated with the oversized coils. This integration enables the control of active coil row numbers. Four active row configurations are assessed: 6 (base case), 4, 3 and 2. In the third step, performance assessment of adaptable cooling coil design is performed. For each coil size (150 kW and 200 kW), two VAV controls strategies (dry-bulb control and absolute humidity control), two VAV configurations (with and without motorized outdoor damper) and four possible active rows (6, 4, 3 and 2) were simulated; 32 cases in total. Each case is simulated for four different part-load operations of oversized coil (100%, 75%, 50% and 25% of actual room load). So, a total of 128 (32x4) simulations were performed.

2.1 Design of oversized coils (Base Designs)

The oversized cooling coils (200 kW and 150 kW) are designed using a free access cooling coil selection software (SPC2000 version 8.2, by S & P Coil Products Limited, [21]). SPC2000 is a coil rating and selection program. This software has been selected as it was used previously in similar investigations to explore the performance of coils for different operational scenarios and provided good results [22]. It calculates the optimum specifications of a heat exchanger based on the design input data. The input/output data could include airside data, fluid side data and physical data related to the geometry of the heat exchanger. The configuration of this coil and its operational condition for Singapore climate were selected based on simulation data as done by [6]. The main operational design inputs for SPC are on-coil conditions and airflow rate.

On-coil condition: the design indoor condition of space is considered as 24 °C and 63% RH. A slightly more stringent conditions than design day conditions outdoor parameters (32 °C and 75%) has been used in this study. Based on these constrains, on-coil condition of air has been obtained using psychrometric chart (“ASHRAE Psychrometric Chart No.1, [23] ”) which is 25°C / 67% RH. It has been assumed that at full load, on-coil air is a mixture of hot and humid outdoor air (10%) and conditioned re-circulated indoor air (90%).

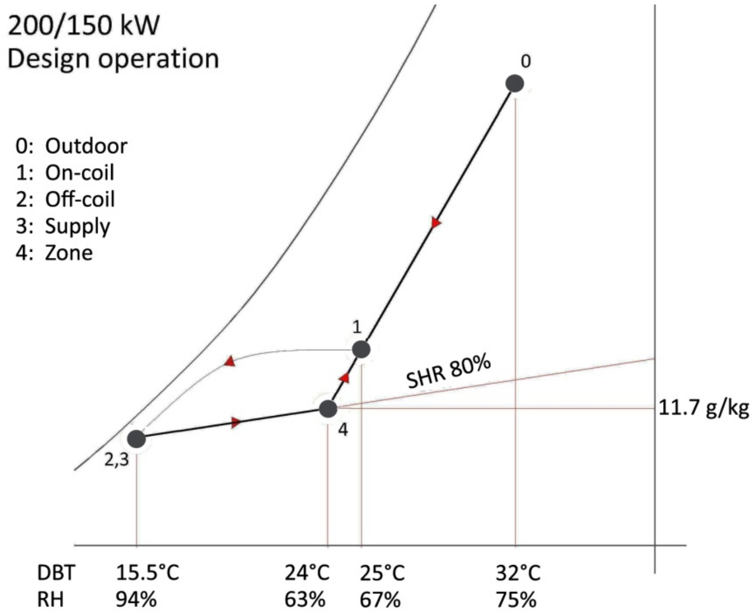


Figure 1 On/off-coil and space conditions for oversized designed coils

Airflow rate: While deciding airflow rate (m^3/s) based on typical cooling load estimation and conventional operation of HVAC system, it has been assumed that the *SHRcz* of oversized space load is up to 80% and difference between zone and off-coil temperature (ΔT) is below 12 °C. Based on these assumptions, numerous simulations have been performed to obtain an adequate design. The same has been verified using a psychrometric chart (See Figure 1) and Eq. (4) and (5).

$$Sensible\ Heat\ Load = 1.213 \cdot Airflow\ rate\ \cdot \Delta T \tag{4}$$

$$Sensible\ Heat\ Load = \left(\frac{Room\ Load \cdot SHRcz}{100}\right) \tag{5}$$

(Where ΔT is the temperature difference of air between off-coil and space)

In all the simulated scenarios, chilled water supply temperature or on-coil water temperature was fixed at 6 °C which is a typical value. The full circuiting (number of circuits is equal to the number of tubes in the height of the coil) has been chosen as circuiting scheme. This also determines the fin height. This implies that every tube in the first row of the coil that is most downstream in the airflow direction is supplied with chilled water at the supply temperature of 6 °C. This is to ensure that a true counter flow arrangement of chilled water flow and airflow is achieved, which would lead to a uniform log-mean temperature difference in the heat and mass transfer process occurring in the cooling coil.

Detailed information regarding other types of circuits e.g. half circuiting can be found in coil design handbooks and guidelines (ASHRAE, [24]). The sizes for fin height (*H*) and fin length (*L*) determine the face area of the coil, which is inversely correlated with the face velocity (V_f). The average air velocity passing through the coil is defined as the face velocity, which can be calculated as airflow rate (*Q*) divided by the face area of the coil (H·L).

Table 1 Configuration of the simulated cooling coil including geometrical specification and typical operational condition of coil in the tropics

Coil parameter	Value		Unit
	200 kW	150 kW	
On-coil air temperature	25	25	°C
On-coil air relative humidity	67	67	%
On-coil water temperature	6	6	°C
Fin material type	Aluminium 0.15 rippled		
Tube diameter	16	16	mm
Tubes high (no of tubes)	44	36	No.
Finned height	1785	1461	mm
Finned length	2500	2250	mm
Fin density	9	9	FPI (fins per inch)
Inlet/Outlet connection	76.2	76.2	mm
Circulate type	F - Full circuiting		

Table 2 Design parameters for 200 kW and 150 kW oversized coils

Design parameters	Values	
Coil Duty, (kW)	150	200
Rows	6	6
Airflow rate, m^3/s	8.0	11.1
Outdoor air %	10	10
Face velocity, m/s	2.4	2.5
SHR_{CC}, %	59	59
Air pressure drop, Pa	187.4	194.2
Fluid on temperature, °C	6	6
Fluid off temperature, °C	14.7	15.2
Fluid flow rate, L/s	3.8	5.2
Actual pressure drop (Pa)	8000	10900
Air on DB, °C	25	25
Air on RH, %	67	67
Air off DB, °C	**16.1**	**16.3**
Air off WB, °C	15.5	15.7
Air off RH, %	94	94
Supply Air temp, °C	**16.2**	**16.2**
Supply Air humidity, %	93	93

SHR_{CZ}, %	80	80
Space absolute humidity, g/kg	11.7	11.7
Space Load (kW) (Latent + Sensible)	94.7	131.4
ΔT	7.8	7.8
Zone DB, °C	24	24
Zone RH, %	63	63

A typical face velocity threshold of 2.5 m/s has been used [24]. In addition, fin material type of aluminum with ripple profile and density of 9 fins per inch (fpi) has been selected in these simulations, which is a common fin material and density supplied in Singapore.

The details of selected coil geometry and information on airside and waterside of the coil are provided in Table 1 and other design parameters are provided in Table 2. The schematic of this full circuiting cooling coil including its design parameters are illustrated in (Figure 2a).

2.2 Design and operative strategies for Adaptable Cooling Coil

Two operational strategies are considered: a) control of coil rows and b) control of outdoor air supply.

2.2.1 Control of coil rows

A reduction in rows of coil may improve the *SHRcc* of cooling coil at part load. This improvement in coil *SHRcc* will further result in improvement of humidity at the zone level. The hydraulic system connection of chilled water flow to cooling coil is required to be redesigned in order to activate each part of the coil separately on demand. The schematics of hydraulic system design for the proposed adaptable cooling coil configuration are shown in Figure 2b. Three-way control valves and modulating valves would be installed at different parts of the hydraulic system to modify chilled water flow loop based on the cooling load. The on/off coil dry-bulb and wet-bulb temperatures could be measured and used to determine which water loops are to be opened or closed. Dry-bulb and wet-bulb temperatures could provide information regarding the real time *SHRcz* of the space and the necessary loops of water could then be deactivated to maintain the *SHRcz* in the acceptable range. A hydraulic system can control the number of rows in operation. A detailed control strategy of hydraulic systems for the practical application is discussed in Section 4 of this paper.

2.2.2 Control of outdoor air supply

Some of the existing VAV systems in Singapore do not employ controllable/motorized/variable outdoor air damper that results in proportional reduction of outdoor air quantity with reducing fan speed. This is quite common in other places, such as the United States. This is the major reason that forces a VAV system to function like a CAV to maintain the minimum outdoor air requirement at part loads. In this study, both the cases of "without" and "with" outdoor air motorized damper (cases A and B) are assessed to evaluate the applicability of adaptable cooling coil in existing and new buildings. To maintain the absolute humidity level of the space at lowest possible airflow rate, the amount of outdoor air supply to AHU space is decreased from 0.52 m^3/s to 0.08 m^3/s. The minimum ventilation requirement for an office space in Singapore is 0.6 L/sm^2 [11]. To maintain the balance between outdoor air and re-circulated air with minimum airflow rate, an automated return air exhaust needs to be introduced.

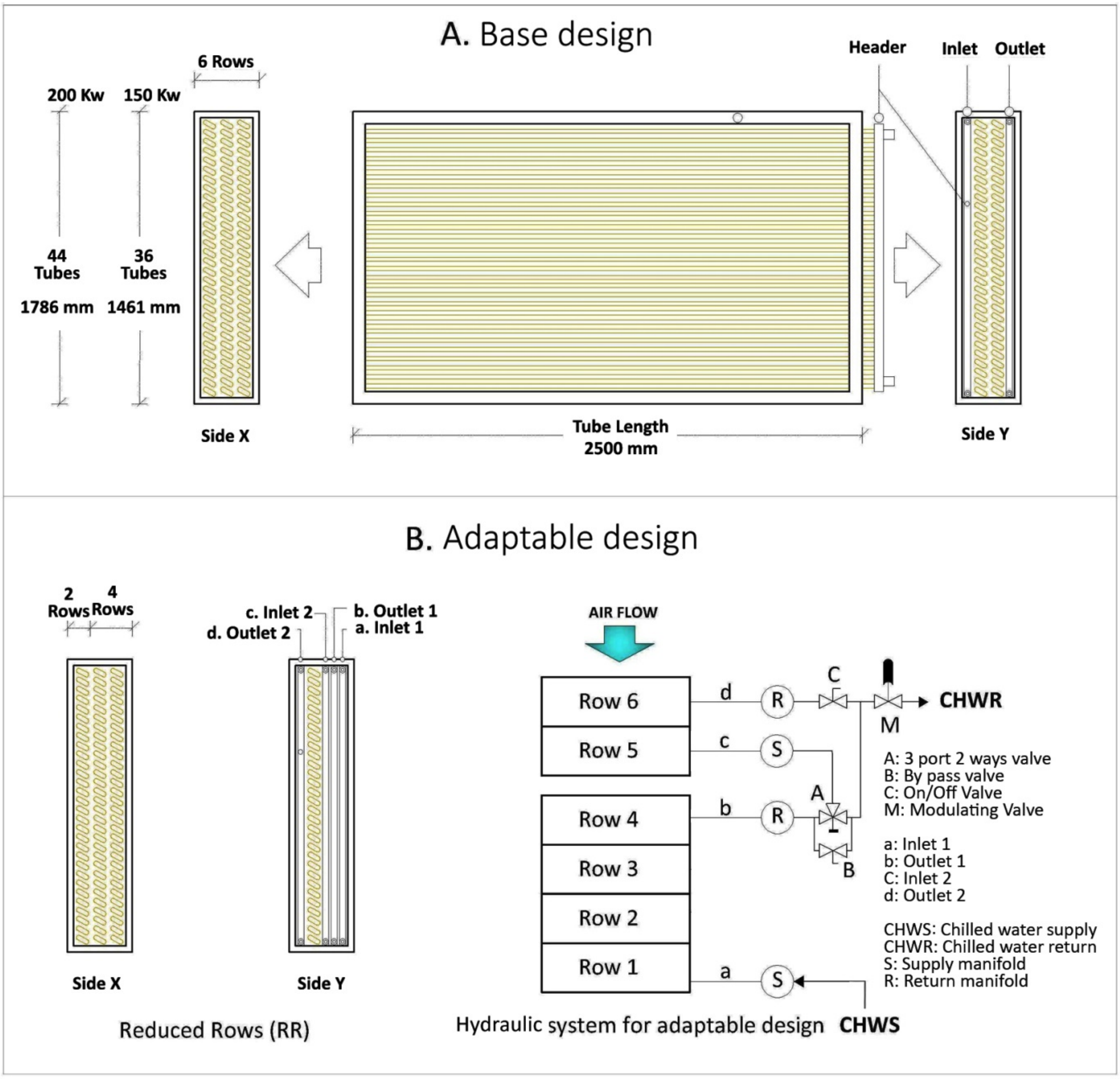


Figure 2 Configuration of a typical full circuiting cooling coil a) base design and b) adaptable design

3.3 Performance assessment of Adaptable Coil

The performance of the adaptable cooling coil has been assessed through simulation and psychrometric analysis for four possible part loads, i.e. 100%, 75%, 50% and 25%. 100% space load is actual peak load = 63 kW (L_{100}), 75% space load is = 47.4 kW (L_{75}), 50% space load is = 31.6 kW (L_{50}) and 25% space load is = 15.8 kW (L_{25}). The design space load was obtained using a psychrometric analysis as per Eq. (6). These part loads for space were obtained as a percentage of design coil duty using Eq. (7).

$$RL_{Oversized} = 1.184 \cdot V_{oversized} \cdot (\Delta h) \tag{6}$$

$$RL_{Part\ loads} = \frac{RL_{Oversized}}{CD_{oversized}} \cdot \%\, RL_{actual\ peak} \tag{7}$$

Where

$RL_{Oversized}$	: Space load for oversized coil
$V_{oversized}$	: Design airflow rate for oversized coil
Δh	: Difference in enthalpy of on-coil and off-coil air (obtained from psychrometric chart)
$RL_{Part\ loads}$	: Space load at part loads
$CD_{oversized}$	: Duty of oversized coil
$\%\ RL_{actual\ peak}$	: Space load as a percentage of actual peak

The off-coil conditions and energy performance analysis for these part loads are done using the results of the SPC simulations. These simulations are performed for base coil design (6 rows in operation) as well as for adaptable coil design (4, 3, and 2 rows in operation). To obtain the on-coil conditions, which are one of the main inputs for SPC simulation, *SHRcz* of part-loads has been assumed to be 75, 70, 65 and 60 for L_{100}, L_{75}, L_{50} and L_{25} respectively. These assumptions are based on a hypothesis that with decrease in space loads, predominantly the sensible loads, the *SHRcz* also decreases. Several simulations have been performed and the outcome of these simulations along with the psychrometric chart is used to obtain optimum space and on-coil conditions for various cases. These simulations have been performed for two oversized cooling coils (200 kW and 150 kW) for two cases and two possible scenarios for each case:

Scenario A (VAV_v - Variable outdoor air): the amount of outdoor air is reduced proportionally to the speed of the fan, therefore the minimum outdoor air may not be provided at low load conditions and this may cause IAQ problems. This is the operation of a VAV system that has a fixed position outdoor air damper.

Scenario B (VAV_f - Fixed outdoor air): The amount of outdoor air provided to the occupant is constant. This is the operation of a state of the art VAV system with variable outdoor air damper. In this case, the occupant always receives the design outdoor airflow rate.

Both Scenarios A and B have been simulated for two different cases. So, a total of four scenarios have been simulated for each oversized cooling coil. The detailed steps for these simulations are discussed below.

CASE 1: Fixed dry-bulb air temperature

In this case, dry-bulb temperature of space is fixed at 24 °C (like design space condition) and absolute humidity is allowed to vary. This type of control is the most commonly used control in commercial and residential buildings. In this case, the airflow rate (m^3/s) was obtained for various part loads as a proportion of oversized design load using Eq. (8).

CASE 1A: Fixed dry-bulb air temperature and variable outdoor air flowrate (VAV_v)

This case represents an existing building without any advanced HVAC control for the outdoor air damper. This case only compares the dehumidification capacity of base and various adaptable coil configurations. Hence, adaptable coil here is the coil that gives off-coil condition closest to design condition (i.e. absolute humidity close to 11.7 g/kg). Secondly, the same airflow rate is used for the analysis of remaining coil configuration of the loads individually. Finally, optimum on-coil, off-coil and space conditions have been obtained using repetitive SPC simulations for various part loads. These simulations have been performed along with psychrometric analysis where fixed *SHRcz* line is moved upward over the psychrometric chart and ΔT was fixed (See Figure 3a).

$$V_{pl} = \frac{V_{DO} \cdot PL}{DL_o} \tag{8}$$

Where,

V_{PL} *: Airflow rate at part load*

V_{DO} *: Design airflow rate (Oversized)*

DL_O *: Design Load (Oversized)*

PL *: Part Load*

CASE 1B: Fixed dry-bulb air temperature and fixed outdoor air flowrate (VAV$_f$)

This case represents a new building with a state of the art VAV system employing outdoor air damper control. The steps of simulation and analysis are similar to CASE 1A.

CASE 2: Fixed space absolute humidity

In this case, absolute humidity (moisture content) of the space is fixed to be 11.7 g/kg (design space condition) and dry-bulb temperature (DBT) is allowed to vary. The variation in DBT is such that the space condition and off-coil condition lie on the same assumed *SHRcz* line on the corresponding psychrometric plots for any load. Fixed absolute humidity is representative of spaces where there is a need to control humidity like in some museums, part of libraries, operating theaters, etc. These types of building represent a relatively small fraction of the built environment.

CASE 2A: Fixed space absolute humidity and variable outdoor air flowrate (VAV$_v$)

In this case firstly, a minimum airflow rate (m^3/s) was obtained for the adaptable coil for various part loads. This has been done through Eq. 4. Here, adaptable coil is one, which gives an off-coil condition closest to design condition (i.e. DBT close to 24 °C). Secondly, the same airflow rate is used for the analysis of all coil configurations (6, 4, 3, 2 rows) at each load. Finally, optimum on-coil, off-coil and space conditions have been obtained using repetitive SPC simulations for various part loads. These simulations have been performed along with psychrometric analysis where fixed *SHRcz* line is moved laterally on the psychrometric chart and ΔT was fixed (See Figure 3b). In this case, base case and various adaptable coil configurations are compared for energy as well as dehumidification performances.

Energy saving through adaptable coil operation

Three types of energy consumption are considered for energy saving calculation a) O_e - Overcooling energy, b) F_e - Fan Energy (to regulate airflow) and c) P_e - Pump energy (to regulate chilled water supply). All energy uses are calculated in kWh. The energy savings are calculated using the adaptable cooling coil best configuration as reference point. This can be calculated using Eq. (9).

$$Total\ Additional\ Energy = Oe + Additional\ Fe + Additional\ Pe \tag{9}$$

Where, Additional $F_e = F_{ce} - F_{ace}$ (9a)

and Additional $P_e = P_{ce} - P_{ace}$ (9b) where, F_{ce} & P_{ce} are the fan and pump energy of that configuration for which this additional energy is calculated and F_{ace} & P_{ace} are the fan and pump energy of adaptable cooling coil.

To obtain the energy saving by adaptable coil configurations, additional set of simulations is performed to obtain coil duty where off-coil temperature is similar to the off-coil temperature obtained for adaptable coil. Further, the difference between adaptable coil duty and the duty of the remaining coils is considered as an overcooling energy saving potential, which could be applied additionally to obtain desired space temperature. These savings in terms of percentages can be obtained through Eq. (10).

$$\%\ Potential\ overcooling\ energy\ saving = \frac{(PL_V - PL_f)}{L_{100F}} \cdot 100 \tag{10}$$

Where L_{100F} = Adaptable Coil duty at part load

PL_v = Coil duty at part load with varying DBT

PL_F = Coil duty at part load with fixed DBT (DBT like Adaptable Coil)

Similarly, the airflow rate (V_f) and actual pressure drop (dP) obtained from simulation is used to calculate the Fan energy (Fe) using Eq. (11). Further, to calculate pump energy (Pe), fluid flow rate (q) in (l/s) obtained from simulation has been used with Eq. (12).

$$F_e(kW) = (dP \cdot V_f)/\mu \tag{11}$$

$$P_e\,(kW) = \frac{q \cdot \rho \cdot h \cdot g}{3.6 \cdot 10^6} / \mu \tag{12}$$

Where,

μ : *Fan efficiency/Pump efficiency (0.85 assumed in this study is within the range of the Fan Efficiency Grade (FEG) given in draft standard AMCA 205 (Movement, 2012)*

ρ : *Fluid density (here fluid is water so density taken is 1gm/cm^3)*

h : *Differential head of pump (2 m is assumed in this case) and*

g : *gravity (9.81 m/s^2)*

CASE 2B: Fixed space absolute humidity and fixed outdoor air flow (VAV$_f$)

In this case, the absolute humidity is constant and the outdoor air flow rate is fixed. The steps of simulation are identical to CASE 2A.

Overall, 16 scenarios of base coil design and 48 scenarios of adaptable design scenarios have been simulated for various part-loads using the coil selection software for 200 kW and 150 kw coil individually. So, a total of 128 simulations have been performed. The summary of simulated scenario can be seen in Table 3.

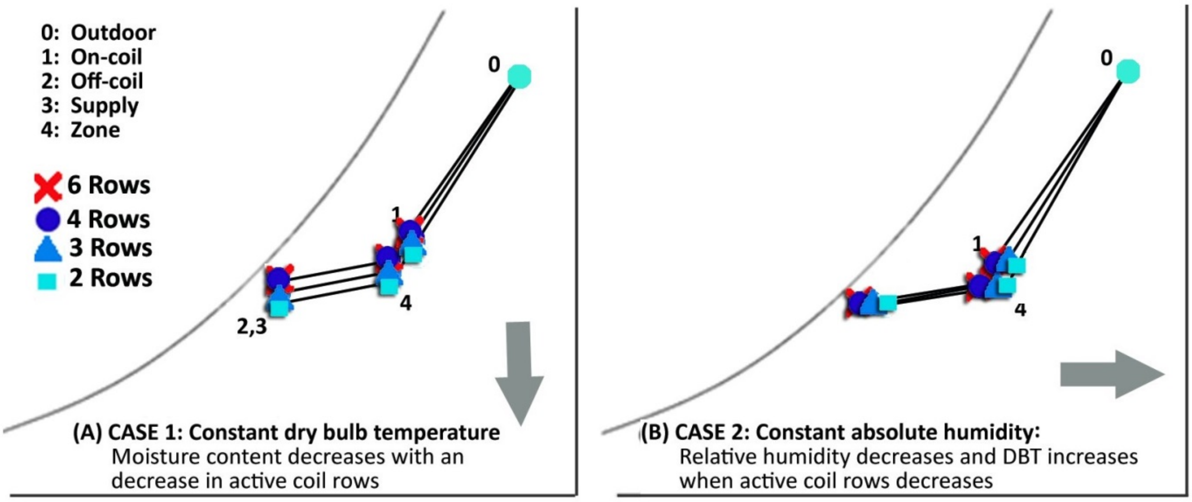


Figure 3 Typical psychrometric analysis for Case 1 & Case 2.

Table 3 Summary of simulated scenarios

Sl. No (A)	Control	Design /Cases	Operation Scenario (Absolute Humidity)	Oversized Coils (B)				Active Rows (Maximum 4 cases) (°C)
				150kW design		200 kW design		
				% of actual peak	Airflow rate (m^3/s)	% of actual peak	Airflow rate (m^3/s)	
	VAV	**Base**	**NA**	**150%**	**8.00**	**200%**	**11.1**	**6**
1	VAV + FOA	RR / 2	Fixed	100%	5.20	100%	5.20	6, 4, 3, 2
2	VAV + FOA	RR / 2	Fixed	75%	3.60	75%	3.60	6, 4, 3, 2
3	VAV + FOA	RR / 2	Fixed	50%	2.00	50%	2.00	6, 4, 3, 2
4	VAV + FOA	RR / 2	Fixed	25%	0.80	25%	0.80	6, 4, 3, 2
5	VAV + FOA	RR / 1	Varying	100%	5.33	100%	5.33	6, 4, 3, 2
6	VAV + FOA	RR / 1	Varying	75%	4.00	75%	4.00	6, 4, 3, 2
7	VAV + FOA	RR / 1	Varying	50%	2.67	50%	2.67	6, 4, 3, 2
8	VAV + FOA	RR / 1	Varying	25%	1.33	25%	1.33	6, 4, 3, 2
9	VAV + VOA	RR / 2	Fixed	100%	4.80	100%	4.80	6, 4, 3, 2
10	VAV + VOA	RR / 2	Fixed	75%	3.20	75%	3.20	6, 4, 3, 2
11	VAV + VOA	RR / 2	Fixed	50%	1.56	50%	1.56	6, 4, 3, 2
12	VAV + VOA	RR / 2	Fixed	25%	0.62	25%	0.62	6, 4, 3, 2
13	VAV + VOA	RR / 1	Varying	100%	5.33	100%	5.33	6, 4, 3, 2
14	VAV + VOA	RR / 1	Varying	75%	4.00	75%	4.00	6, 4, 3, 2

15	VAV + VOA	RR / 1	Varying	50%	2.67	50%	2.67	6, 4, 3, 2
16	VAV + VOA	RR / 1	Varying	25%	1.33	25%	1.33	6, 4, 3, 2
Total Simulation = A x B x C = 16 x 2 x 4								**128**

(RR- reduced rows adaptable design, FOA- Fixed outdoor air, VOA – Variable outdoor air)

3. Results

Out of 128 Simulation cases, 116 were successfully applicable to practical operation scenarios. However, in the remaining 12 cases, the coil was either unable to perform or the coil adaptation for that configuration was not required. This implies that, in some cases, 3 rows configuration was sufficient to maintain space conditions and so the configuration of 2 rows adaptation was not required. These results are also categorized in four groups based on four operational scenarios for 150 kW and 200 kW individually. The outcome of each case is discussed for dehumidification performance and energy saving potential in the subsequent sections.

3.1 CASE 1 – fixed space dry-bulb temperature

3.1.1 CASE 1A Fixed dry-bulb temperature and variable outdoor air flowrate (VAV_v)

Performance at all four loads has been studied for a system in which the supply air varies from peak load to part-loads, while the percentage of outdoor air out of supply air remains constant. The calculated airflow rate and corresponding chilled water flow rate is presented in Table 4.

Table 4 CASE 1A: Airflow rate and fluid flow rate

	L_{100}		**L_{75}**		**L_{50}**		**L_{25}**	
Coil Configuration	Base	Adaptable	Base	Adaptable	Base	Adaptable	Base	Adaptable
Rows	6	3	6	2	6	2	6	2
Coil $_{150}$ F (l/s)	2.00	3.30	1.70	3.70	1.30	3.80	0.30	1.90
Coil $_{200}$ F (l/s)	2.10	3.50	1.82	4.20	0.90	3.47	0.30	1.97
V (m^3/s)	5.3		4.0		2.6		1.3	
Coil $_{150}$ V%	67		50		33		17	
Coil $_{200}$ V%	50		37		25		13	

V - Airflow rate, V% – Percentage of actual design, F – Fluid (chilled water) flow rate, Coil $_{150}$: 150 kW, Coil $_{200}$: 200 kW.

Adaptable coil performance: Room conditions

In the base configuration (6-rows) of VAV operation at L_{100} and part-loads, the off-coil moisture content increases through a decrease in airflow rate. Although the zone level temperature is maintained, the zone level absolute humidity slightly increases (max increase is 1.5 g/kg (from 12.75 to 14.3 g/kg) that correspond to a 10% RH increase (from 67% to 77%). We can conclude that in the extreme conditions selected in this simulation involving a VAV system without motorized damper, the humidity increases but not excessively. In all the cases, the adaptable coil slightly reduces the absolute humidity (maximum reduction is 1.9 g/kg – from 13.8 to 11.9 g/kg). This corresponds to a 10% RH reduction (from 74 to 64%). The minimum variation with desired room condition is seen for L_{100} and the maximum variation is seen for L_{25}. However, from Figure 4a, it can be seen that with increase in fluid flow rate and reduction in active coil rows, the zone level humidity improves. For L_{100}, reduction of active coil rows up to 3 has been found to be adequate to maintain room conditions and for L_{75}, L_{50} and L_{25}, whereas the adaptable coil configuration is the operation of 2 coil rows. The adaptable coils for L_{100}, L_{75} and L_{50} can maintain zonal absolute humidity as close as design (11.7 g/kg). However, for L_{25}, although adaptable coil is able to provide better humidity control as compared to other configurations, the zonal absolute humidity is found to be slightly higher (13.1 g/kg). It is to be noted that the dehumidification capacity of adaptable coil configuration decreases with decrease in zone load and airflow rate. This result is clearly seen with psychrometric analysis as presented in Appendix A (Figure 9).

Adaptable coil performance: Energy Savings

The calculated amounts of pump energy by Eq. (12) shows insignificant additional energy required for operating the coil during adaptation as compared to base operation. For the various cases, the difference in energy use between cases is always less than 0.3% of overall energy required. It can, therefore, be concluded that the VAV system without outdoor damper has slightly increased absolute humidity when working in part-load conditions and that the adaptable coils can slightly reduce the absolute humidity in all load conditions without an energy penalty.

3.2.2 CASE 1B Fixed dry-bulb temperature and fixed outdoor air flowrate (VAV_f)

This case investigates the performance of coils with fixed dry-bulb temperature. However, in this case, a fixed amount of outdoor air 0.8 m^3/s and 1.1 m^3/s (10% of design loads for $Coil_{150}$ and $Coil_{200}$ respectively) is considered for all part-loads. This is the case of a properly designed VAV system.

This case works for all part-loads as there is flexibility to operate at higher absolute humidity levels. Calculated airflow rate and corresponding simulated fluid flow rate for different part-loads are presented in Table 5.

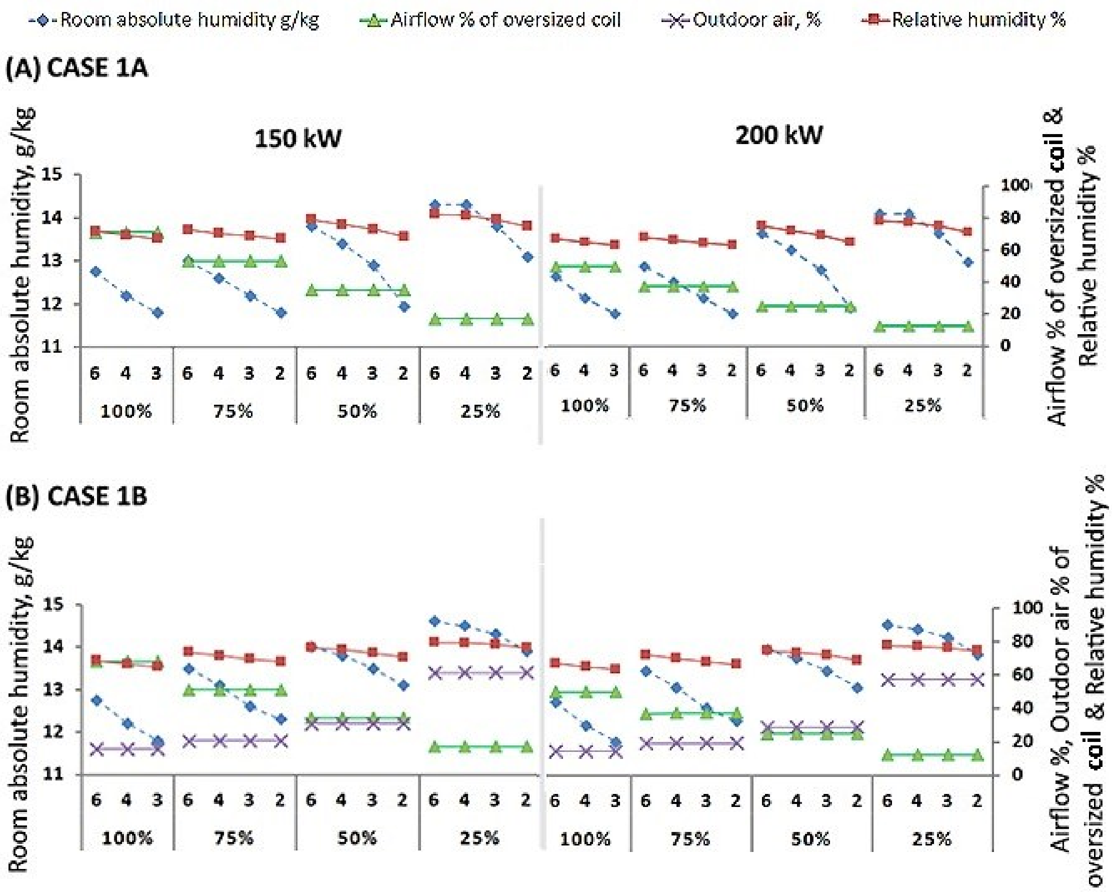


Figure 4 CASE 1A and 1B Results: Comparison of dehumidification performance of coil configurations at various part-loads

Table 5 CASE 1B: Airflow rate and fluid flow rate

	L_{100}		L_{75}		L_{50}		L_{25}	
Coil Configuration	Base	Adaptable	Base	Adaptable	Base	Adaptable	Base	Adaptable
Rows	6	3	6	2	6	2	6	2
Coil $_{150}$ F (l/s)	2.2	3.8	1.75	3.8	1.5	3.8	0.7	3.1
Coil $_{200}$ F (l/s)	2.2	3.7	1.9	4	1.7	5.0	1.2	2.9
V (m^3/s)	5.30		4.00		2.67		1.30	
Coil $_{150}$ V%	67		50		33		17	
Coil $_{200}$ V%	50		37		25		13	

V - Airflow rate, V% – Percentage of actual design, F – Fluid (chilled water) flow rate, Coil $_{150}$: 150kW, Coil $_{200}$: 200kW.

Adaptable coil performance: Room conditions

From Figure 4b, it can be easily seen that the off-coil humidity slightly increases with a decrease in airflow rate. The zone level absolute humidity increases by 1.9 g/kg (from 12.75 to 14.6 g/kg) and 11 RH% (from 67% up to 78%). As seen in earlier cases, the minimum variation with desired room condition is seen for L_{100} and maximum is seen for L_{25}. With an increase in fluid flow rate and reduction in active coil rows, the zone level absolute humidity slightly improves. The maximum reduction is 1.2 g/kg (from 13.5 to 12.3 g/kg), which corresponds to a maximum RH reduction of 5.5% (from 72 to 66.5%). The improvement is less than in the case with fix outdoor damper. Similar to Case 1A, for L_{100}, reduction of active coil rows up to 3 has been found to be adequate to maintain room conditions and for L_{75}, L_{50} and L_{25}, the adaptable coil configuration is the operation of 2 rows coil. In this case, the adaptable coil for L_{100} can maintain zonal absolute humidity as close as design (11.7 g/kg). The dehumidification capacity of adaptable coil configuration decreases with decrease in zone load and airflow rate. These results are presented in the form of psychrometric analysis as shown in (Appendix A, Figure 10).

Adaptable coil performance: Energy Savings

Like Case 1A, the energy saving potential is insignificant here. Overall, it can be concluded that for a state of the art VAV system with outdoor air damper control, there is a slight increase in absolute humidity when working in part-load and that the adaptable coil only marginally reduces the absolute humidity, in particular in part-load conditions. There is no energy benefit in using the adaptable coil.

3.2 CASE 2 - Fixed room condition analysis

3.2.1 CASE 2A Fixed space absolute humidity and variable outdoor air flowrate (VAV_v)

To maintain the zone level absolute humidity of 11.7 (g/kg) as per design load, the airflow rate is calculated using Eq. (4). Similarly, the airflow rate as a percentage of actual design is calculated using Eq. (13) for various part-loads. These calculated airflow rates and corresponding fluid (chilled water) flow rates for various part-loads are presented in Table 6.

Table 6 CASE 2A: Airflow rate and fluid flow rate

	L_{100}		L_{75}		L_{50}		L_{25}	
Coil Configuration	Base	Adaptable	Base	Adaptable	Base	Adaptable	Base	Adaptable
Rows	6	3	6	2	6	2	6	2
$Coil_{150}$ F (l/s)	2.3	3.8	1.8	3.8	1.4	3.0	0.4	2.4
$Coil_{200}$ F (l/s)	2.5	3.5	1.8	3.5	1.3	3.3	0.4	2.2
V (m^3/s)	5.2		3.6		2		0.8	
$Coil_{150}$ V%	65		45		25		10	
$Coil_{200}$ V%	47		33		18		7	

V - Airflow rate, V% – Percentage of actual design, F – Fluid (chilled water) flow rate, $Coil_{150}$: 150kW, $Coil_{200}$: 200kW.

Adaptable coil performance: Room conditions

Calculated airflow rate is used for SPC simulation to find the adaptable coil configuration with varying rows number (6, 4, 3 and 2). An adaptable coil configuration is identified as one, which provides the zone condition at part-loads closest to the design condition of full load. The outcomes of simulation for $Coil_{150}$ and $Coil_{200}$ are presented in Figure 5. Using the adaptable cooling coil slightly reduces the relative humidity (max reduction is 5%, from 68 to 63%) and the temperature is increased (max increase is 1.4 °C from 22.7 to 24 °C). In the base case of VAV operation when 6 rows were active, the off-coil air condition gets cooler through reduction of airflow. The least variation with desired space condition is seen for L_{100} and maximum variation is seen for L_{25}. These results can be seen in the psychrometric analysis (Appendix A, Figure 11).

For L_{100}, reduction of active coil rows up to 3 rows has been found to be adequate to maintain room conditions for both $Coil_{150}$ and $Coil_{200}$. So, there is no need to explore 2 row coil operation here. However, for L_{75}, L_{50} and L_{25}, the adaptable coil is a configuration of 2 rows. It is to be noted that the required amount of fluid flow rate varies as per the variation in airflow rate for both the coils.

Adaptable coil performance: Energy Savings

The energy saving potential by adaptable coil can be seen from Figure 5. Between 1.5-11.8% and 7-18% of energy saving has been calculated for $Coil_{150}$ and $Coil_{200}$. The performance of these adaptation strategies (reduction is coil rows) in terms of energy saving is seen higher for the higher oversized coil ($Coil_{200}$). However, the energy saving potential by adaptable coil configuration decreases with decrease in zone load (highest energy saving at L_{100} and lowest at L_{25}). For $Coil_{150}$ and $Coil_{200}$, highest energy saving possible are 12 and 18% and the lowest are 1.5 and 2%.

Overall, we can conclude that in this specific application, where absolute humidity has to stay constant (e.g., conservation of art pieces), the adaptable coil is able to slightly reduce the relative humidity and increase the dry-bulb temperature, leading to up to 18% energy savings.

3.2.2 CASE 2B Fixed space absolute humidity and fixed outdoor air flow (VAV_f)

This case investigates the performance of coils with fixed space absolute humidity and variable outdoor air damper control that leads to fixed outdoor airflow rate at part-load conditions. The variable OA damper control allows for a fixed amount of outdoor air 0.8 m^3/s and 1.1 m^3/s (10% of design loads for $Coil_{150}$ and $Coil_{200}$ respectively) for all part-load operation. So, the percentage of outdoor air to supply air changes with the ventilation requirement for spaces. The results of this case are presented in Figure 6.

The first noticeable outcome is that the coils studied need outdoor air percentage (ratio of outdoor air to airflow rate) more than 100% to maintain the absolute humidity at L_{25}, which is practically impossible. This is due to fixed and higher outdoor air supply at L_{25}. In the case of L_{25}, the required airflow rate is 0.62 m^3/s and much lower than the fixed outdoor air supply amount of 0.80 m^3/s. So, these coils are unable to maintain absolute humidity for L_{25} even in case of 100% outdoor air for ventilation. These are the cases when excess overcooling could be seen in the real operation of HVAC systems with an increased airflow rate.

In this case, airflow rate and fluid flow rates are obtained like Case-1A for various part-loads and presented in Table 7. Like Case-1A, it can be seen from (Appendix A, Figure 12) that the off-coil air condition gets cooler through reduction of airflow. From Figure 6, a minimum variation is noticed between desired zonal DBT and absolute humidity for various part-loads.

However, there is a slight decrease in zonal temperature up to 0.9 °C and an increase in RH% up to 5% is noticed for L_{100} and L_{75}.

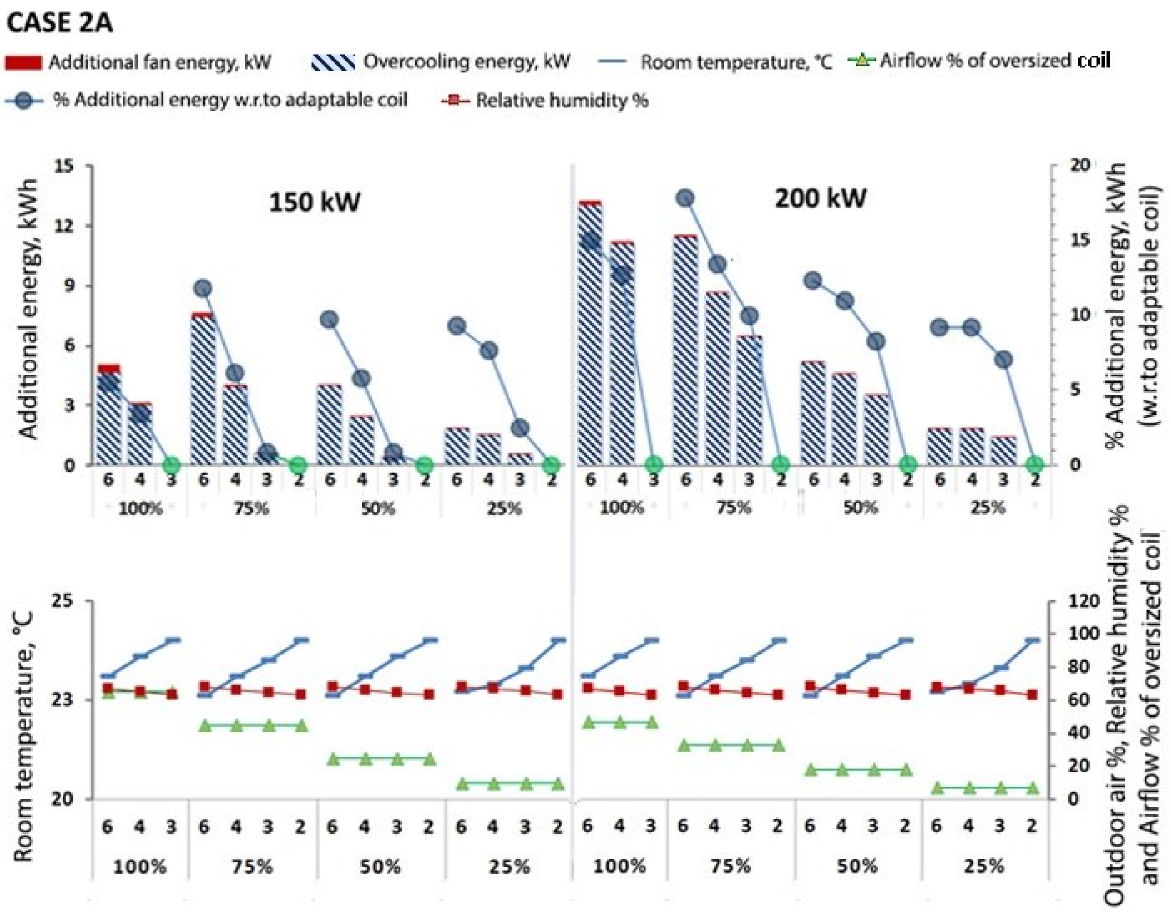


Figure 5 CASE 2A Results: Comparison of energy and dehumidification performance of coil configurations at various part-loads

Table 7 CASE 2B: Airflow rate and fluid flow rate

	L_{100}		L_{75}		L_{50}		L_{25}
Coil Configuration	Base	Adaptable	Base	Adaptable	Base	Adaptable	*Unable to operate*
Rows	6	3	6	3	6	3	
Coil $_{150}$ F (l/s)	2.5	3.8	2.0	3.1	1.7	2.4	
Coil $_{200}$ F (l/s)	2.3	3.4	2.0	3.2	1.8	2.4	
V (m^3/s)	4.8		3.2		1.56		
Coil $_{150}$ V%	60		40		20		
Coil $_{200}$ V%	43		29		14		

V - Airflow rate, V% – Percentage of actual design, F – Fluid (chilled water) flow rate, Coil $_{150}$: 150kW, Coil $_{200}$: 200kW.

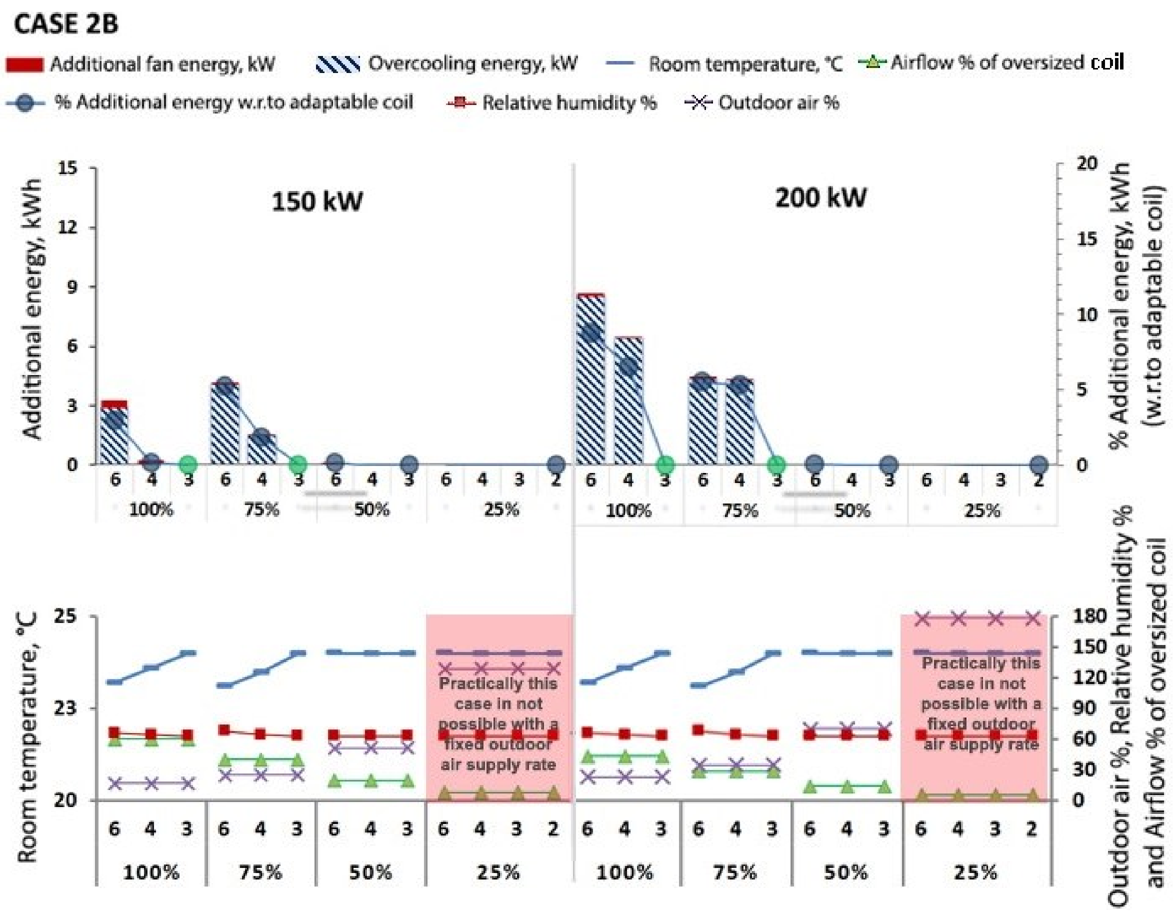


Figure 6 CASE 2B Results: Comparison of energy and dehumidification performance of coil configurations at various part-loads

Adaptable coil performance: Room conditions

From Figure 6, it can be easily seen that with increase in fluid flow rate and reduction in active coil rows, the zone level temperature and relative humidity slightly improves. For L_{100} and L_{75}, reduction of active coil rows up to 3 has been found to be adequate to maintain room conditions for both the coils and so there is no need for 2-row coil operation here. However, it has been seen that at L_{50}, adaptation does not give any better zone condition than actual base design of 6-Row coil operation. This occurs as the off-coil air condition reaches the saturation line for all cases (6, 4 and 3 rows in operation). This can be easily seen from the psychrometric analysis as presented in (Appendix A, Figure 12).

Adaptable coil performance: Energy Savings

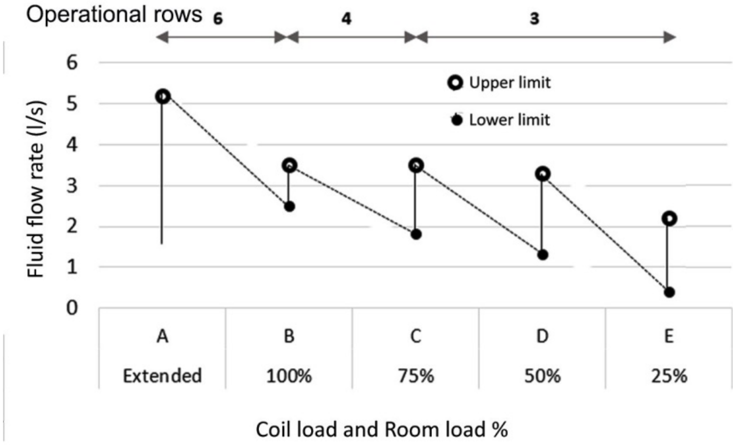
The potential of energy saving using adaptation for $Coil_{150}$ and $Coil_{200}$ is seen to be between 2-5.2% and 5.5-8.8% for L_{100} and L_{75}. Like Case-1, the energy saving potential by adaptable coil configuration decreases with decrease in zone load. Highest energy saving potential through adaptable coil is seen in the case of L_{100} and lowest is seen for L_{75} for both $Coil_{150}$ and $Coil_{200}$.

Adaptation is found insignificant for L_{50} for both cases of enhancing the room condition as well as for saving energy. So, it can be concluded that with fixed amount of outdoor air supply, the percentage of outdoor to supply air increases with decrease in space load. With this increase in ratio of outdoor to supply air, the performance of adaptation decreases and vice versa in case of airflow rate.

4. Adaptable coil control strategy for the practical application

The control strategy of an adaptable coil would be based on the continuous measurement of ΔT_{chw} (Temperature difference between on-coil chilled water and off-coil chilled water) and Q_{chw} [Fluid (water) flow rate ($\frac{l}{s}$) inside the coil]. From Eq. (13), Q_{coil} (Coil load) would be measured. Further, the measured Q_{coil} and Figure 7 (sample graph) would be used together to identify the required adaptation in the coil configuration for the various part loads. For illustration, the graph in Figure 7 is prepared from the simulation outcomes of 200 kW coil and Case 2A. In the case of the 200-kW coil which is 100% oversized, if the actual or measured Q_{coil} is in the range of A (100% of coil load) and B (50% of coil load), there is no adaptation required and the coil will function normally with 6-rows in operation. However, at coil load B, which also corresponds with the actual room load in this study, a change in coil configuration is preferred. A key criterion in the A-B stage of a 6-row coil operation would be the association of the coil load with both Q_{chw} and ΔT_{chw} and it is important to ensure that Q_{chw} does not go below a stipulated (or pre-determined) minimum flow rate. Between B and C, adaptation is made from 6-rows to 4-rows and this is accompanied by an increase of Q_{chw} at the start of the 4-row coil operation, which is desirable from the perspective of enhancing dehumidification. During this operation of B-C, the adapted coil functions as a 4-row coil with Q_{chw} progressively being modulated or reduced with decreasing coil load. Again, the criterion adopted for further adaptation would be further reduction of coil loads associated with reducing Q_{chw} and ΔT_{chw}. A change-over to a 3-row coil would be made if Q_{chw} falls below minimum stipulated flow rate during the B-C stage. The adapted coil functions as a 3-row coil during the C-D stage of operation. Following similar logic, a further adaptation is made from 3-rows to 2-rows at the change-over point of D. The scenario described above is for scenarios corresponding to decreasing loads. In the reverse scenario of increasing loads, the adaptation would move from 2-rows to 3-rows to 4-rows to 6-rows. At every change-over point, a control band of ± 5% of Q_{coil} is to be considered to prevent cycling and determine a clear "change-over" or adaptation from one coil configuration to another. A similar control strategy can be adopted for 150 kW (50% oversized) coil and other operational scenarios. This control strategy of coil provides adaptation at various parts loads and also gives the flexibility of harnessing the full potential of the coil performance, including that of future expansion in terms of room/space loads.

$$Q_{coil} = Q_{chw} \times \Delta T_{chw} \tag{13}$$



	Coil load (kW)	Room load (kW)
A	200	Extended
B	100	100%
C	75	75%
D	50	50%
E	25	25%

Figure 7 Adaptable coil control strategy - Desirable coil configurations at extended and various part loads for Case 2A

5. Limitations of the study

The cooling coil operational scenarios have been simulated under steady state scenarios where input data on air and water sides of coil do not change over time. However, in real conditions, chilled water supply temperature or on-coil air condition may change or fluctuate over time. The humidity buffer effect of building material was not considered. Another limitation of this study is to consider a certain value for operational conditions of the cooling coil investigated. These assumptions have been made based on typical design conditions of cooling coils for the tropical climate of Singapore but there was not much measured data, therefore several assumptions (intensity of the oversize, SHR and how they change in part load, etc.) were based on our professional judgment. We did not explore how the active row control would look like and how it should be implemented.

6. Discussion and conclusions

The outcomes of simulations revealed that in part-load conditions, there is a small increase in absolute humidity (max 1.9 g/kg – from 12.7 to 14.6 g/kg, 11% RH – from 67 to 78%) when we have a heavily oversized cooling coil (50 and 100% oversize and room load conditions are at 25%). This is not a large increase but if the starting conditions were already in the high humidity range, then the value could be a concern. One possible approach would be to design for lower humidity targets to start from and a better-sized coil. Another approach is the one proposed in this paper. Using the adaptable coil would slightly reduce the absolute humidity (max reduction 0.9 g/kg) but its effectiveness is reduced at lower part-load conditions where is most needed. For illustration, L_{50} (lower part-load condition) is represented over a Psychrometric chart for different cases (see Figure 8). It can be easily seen that the adaptation is effective for all cases except for the case of fixed absolute humidity with fixed outdoor airflow. This happens as the performance of adaptation decreases with an increase in ratio of outdoor to supply air. In typical applications (temperature control) there is no energy benefit of using the adaptable coil but in special applications (absolute humidity control) there is a substantial energy benefit (up to 18%). All the conditions are summarized below (see Table 8).

Table 8 Adaptable Coil Impact Matrix:

		CASE 1A				CASE 1B				CASE 2A				CASE 2B			
		INDOOR - Fixed DBT, Varying Absolute Humidity; OUTDOOR AIR % - Fixed				INDOOR - Fixed DBT, Varying Absolute Humidity; OUTDOOR AIR % - Variable with damper control				INDOOR - Fixed Absolute Humidity, Varying DBT; OUTDOOR AIR % - Fixed				INDOOR - Fixed Absolute Humidity, Varying DBT; OUTDOOR AIR % - Variable with damper control			
Load (%)		100	75	50	25	100	75	50	25	100	75	50	25	100	75	50	25
Adaptability (Active coil rows)		3	2	2	2	3	2	2	2	3	2	2	2	3	3	3	NA
Energy saving (%)	OS (50%)	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*5.5*	*12*	*9.7*	*9.3*	*3*	*5.3*	*0*	*NA*
	OS (100%)	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*Ins*	*15*	*18*	*12.3*	*9.2*	*8.8*	*5.5*	*0*	*NA*
Absolute humidity reduction (g/kg) & RH% Reduction	OS (50%)	*1*	*1.2*	*1.9*	*1.2*	*1*	*1.2*	*0.9*	*0.7*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*
		4%	*5%*	*10%*	*7%*	*3.5%*	*5.5%*	*5.5%*	*3.3%*	*4%*	*5%*	*5%*	*4.8%*	*3%*	*5%*	*0%*	*NA*
	OS (100%)	*1*	*1.2*	*0.9*	*0.7*	*1*	*1.2*	*0.9*	*0.7*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*	*NA*
		4%	*5%*	*10%*	*7%*	*3.5%*	*5.5%*	*5.5%*	*3.3%*	*4%*	*5%*	*5%*	*4.8%*	*3%*	*5%*	*0*	*NA*

**Ins – Insignificant, NA – Not Applicable, OS – Oversized*

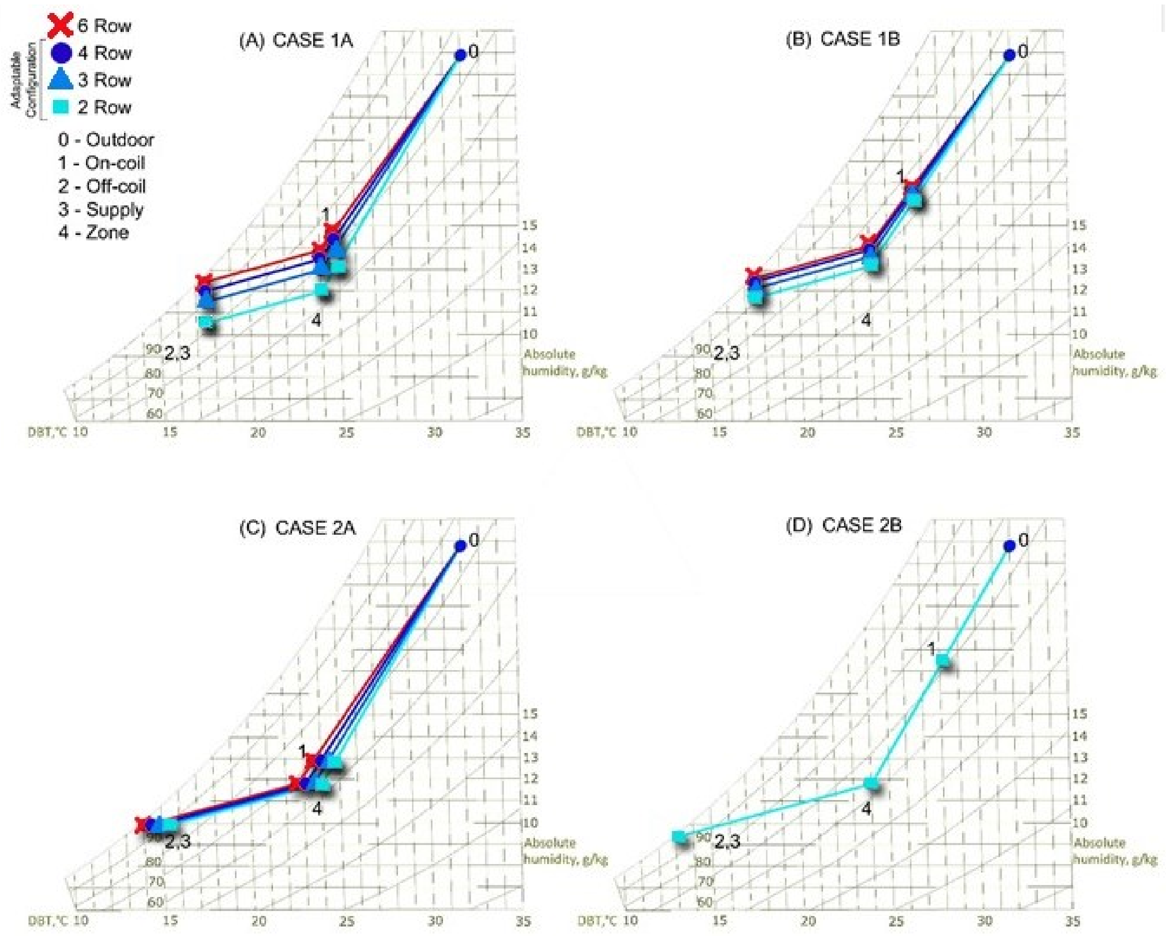


Figure 8 **Comparison between various cases for L_{50} using psychrometric analysis**

It is apparent from this study that the reduction of active coil surface area could be an attractive option for the designer and operator to deal with an oversized coil. Although an evaluation of the cost-effectiveness is not presented in this paper and was beyond the scope of this study, it is envisaged that the capital cost premium would not be high. It is also further envisaged that manufacturer adoption could be quite reasonable, as the strategy does not have significant drawbacks.

7. References

[1] ASHRAE 90.1, Energy Standard for Buildings Except Low-Rise Residential Buildings, Atlanta, G (2013).

[2] G.C. Hourahan, How to Properly Size Unitary Equipment, ASHRAE J. 46.2. (2004) 15–18.

[3] W.K.C. Philip C.H. Yu, Sizing of air-conditioning plant for commercial buildings in Hong Kong, Appl. Energy. 66 (2000) 91–103.

[4] H.H. P. Jacobs, State-of-the-Art Review Whole Building, Building Envelope and HVAC Component and System Simulation and Design Tools, Air-Conditioning Refrig. Technol. Institute, Arlington, VA, USA. (2002).

[5] P.B. Don R. Felts, The State of Affairs-Packaged Cooling Equipment in California, (2000). http://cgec.ucdavis.edu/ACEEE/2000/PDFS/PANEL03/233.pdf (accessed June 15, 2016).

[6] S.C. Sekhar, L.T. Tan, Optimization of cooling coil performance during operation stages for improved humidity control, Energy Build. 41 (2009) 229–233. doi:10.1016/j.enbuild.2008.09.005.

[7] G. Amrane, K., Hourahan, G.C. and Potts, Latent performance of unitary equipment, ASHRAE J. 45 (2003) 28–31.

[8] M. Khattar, M., Ramanan, N. and Swami, Fan cycling effects on air conditioner moisture removal performance in warm, humid climates, Int. Symp. Moisture Humidity, Washingt. DC, April. (1995).

[9] Henderson, H.I., The impact of part-load air-conditioner operation on dehumidification performance: Validating a latent capacity degradation model, Proc. IAQ Energy, pp.115-122. (1998).

[10] ASHRAE, Standard 62.1-2016 Ventilation for Acceptable Indoor Air Quality, American Society of Heating, 2016.

[11] BCA, SS 553: Code of practice for air-conditioning and mechanical ventilation in buildings, Singapore, SPRING, Singapore, 2016.

[12] E. Djunaedy, K. van den Wymelenberg, B. Acker, H. Thimmana, Oversizing of HVAC system: Signatures and penalties, Energy Build. 43 (2011) 468–475. doi:10.1016/j.enbuild.2010.10.011.

[13] H. Mclain, D. Goldenberg, Benefits of Replacing Residential Central Air Conditioning Systems:

Benefits of Replacing Residential Central Air Conditioning Systems, Knoxville, TN, USA, Oak Ridge Natl. Lab. (1985). http://www.osti.gov/scitech/biblio/5866949 (accessed June 15, 2016).

[14] Z. Li, W. Chen, S. Deng, Z. Lin, The characteristics of space cooling load and indoor humidity control for residences in the subtropics, Build. Environ. 41 (2006) 1137–1147. doi:10.1016/j.buildenv.2005.05.016.

[15] D. J. Worthington, Air conditioning apparatus having variable sensible heat ratio, US4984433 A, 1991. http://www.google.com/patents/US4984433#backward-citations.

[16] X. Xu, L. Xia, M. Chan, S. Deng, Inherent correlation between the total output cooling capacity and equipment sensible heat ratio of a direct expansion air conditioning system under variable-speed operation (XXG SMD SHR DX AC unit), Appl. Therm. Eng. 30 (2010) 1601–1607. doi:10.1016/j.applthermaleng.2010.03.016.

[17] Q. Qi, S. Deng, Multivariable control of indoor air temperature and humidity in a direct expansion (DX) air conditioning (A/C) system, Build. Environ. 44 (2009) 1659–1667. doi:10.1016/j.buildenv.2008.11.001.

[18] P. Stephen, Kavanaugh, HVAC Simplified, American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), n.d.

[19] K.J. Chua, S.K. Chou, J.C. Ho, A model to study the effects of different control strategies on space humidity during part-load conditions, Build. Environ. 43 (2008) 2074–2089. doi:10.1016/j.buildenv.2007.12.005.

[20] A. Shaw, R.E. Luxton, Russell Estcourt Luxton and Luminis Pty. Ltd., Air conditioner and method of dehumidifier control, 1989.

[21] S.C.P. Limited, Coil selection and rating software, (2015). www.spcoils.com.

[22] C. Sekhar, U. and Maheswaran, Cooling coil optimisation in hot and humid climates for IAQ and energy considerations, in: 10th Int. Conf. Heal. Build., Brisbane, Australia, 2012. http://scholarbank.nus.edu.sg/handle/10635/114049.

[23] ASHRAE Psychrometric Chart No.1, ASHRAE. (2016). http://www.handsdownsoftware.com/ashrae-chart.pdf (accessed June 16, 2016).

[24] ASHRAE, Air-Cooling and Dehumidifying Coils, in: Ashrae Handbook-HVAC Syst. Equip., ASHRAE, Atlanta, GA, USA, 2012.

Appendix A

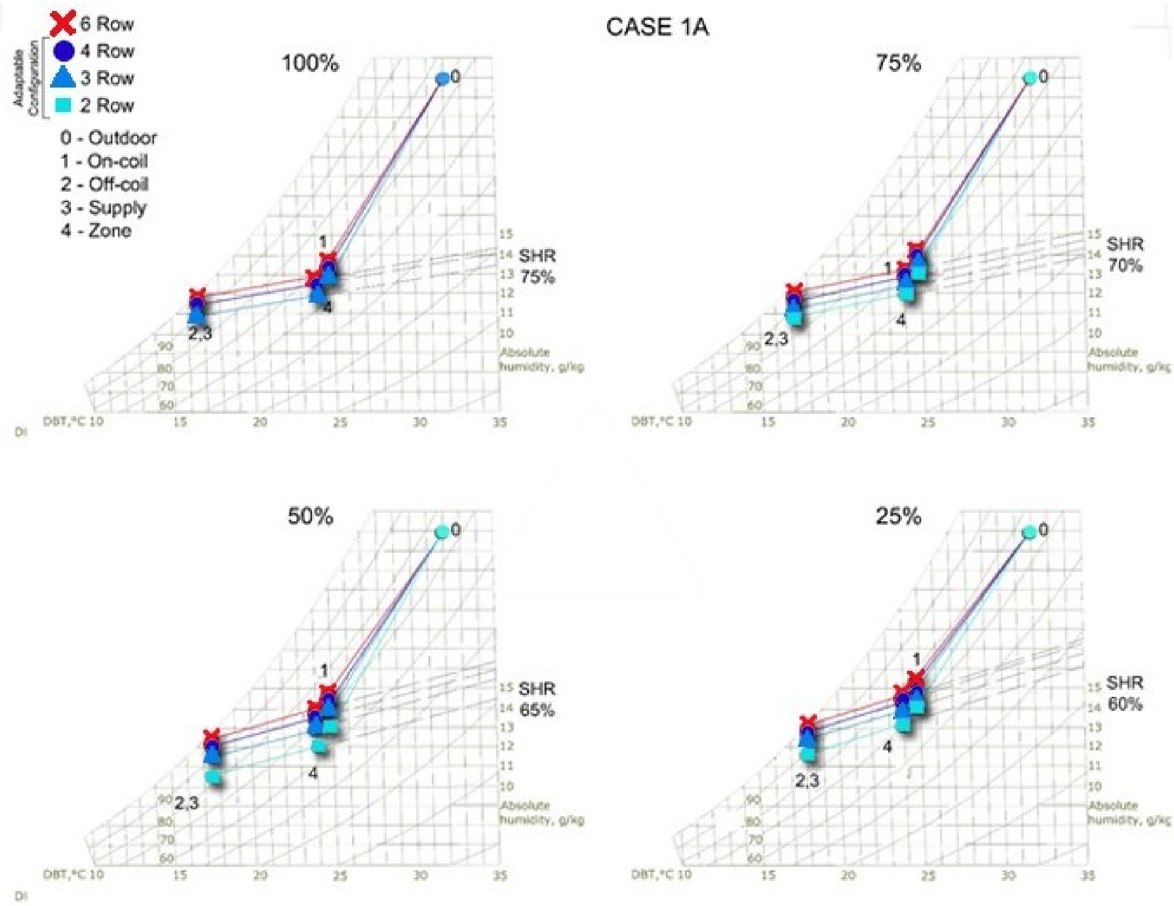


Figure 9 Psychrometric analysis for CASE 1A

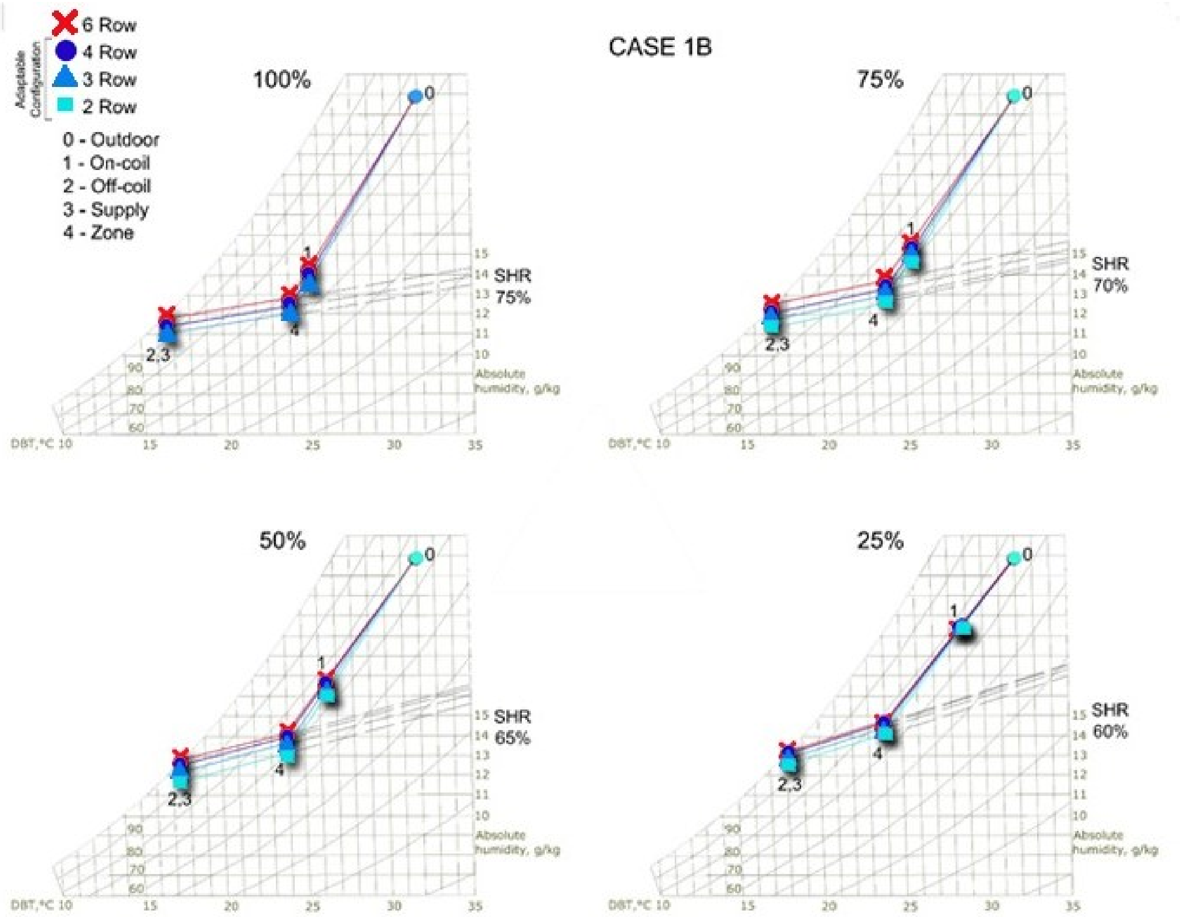


Figure 10 Psychrometric analysis for CASE 1B

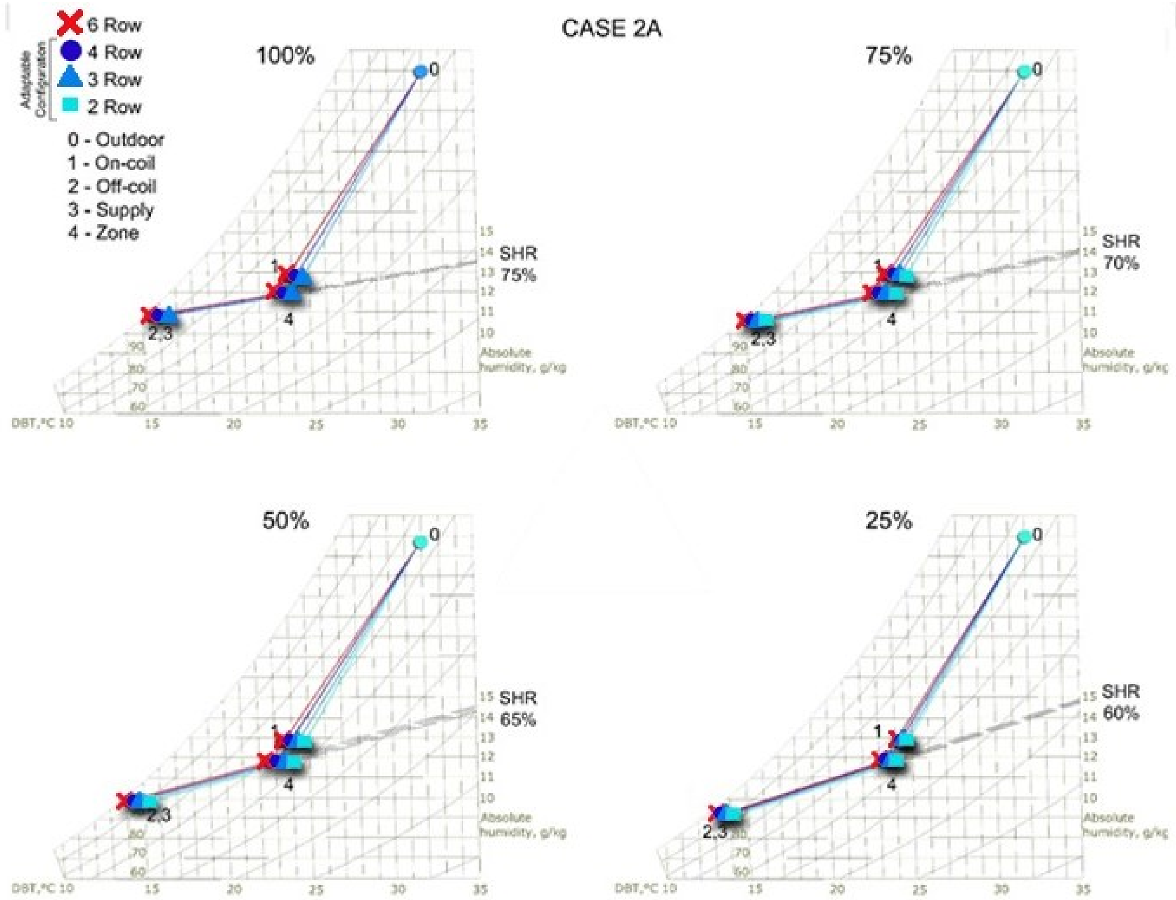


Figure 11 Psychrometric analysis for CASE 2A

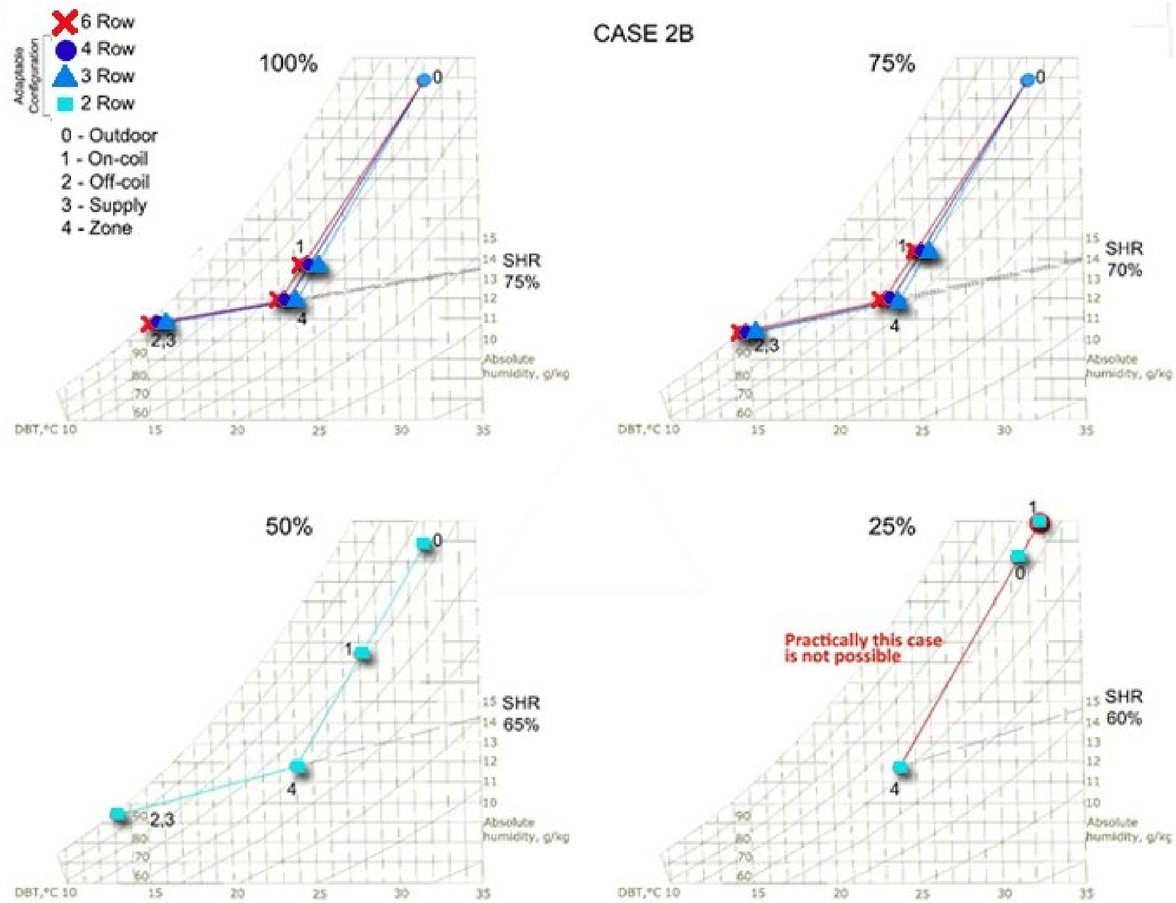


Figure 12 Psychrometric analysis for CASE 2B