Design and control of mixed-mode cooling and ventilation in low energy residential buildings in India

by

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A Doctoral Thesis

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This is to certify that I am responsible for the work submitted in this thesis, that the original work is my own except as specified in the acknowledgements or footnotes. I confirm that neither the thesis nor the original work contained therein has been submitted to this or any other institution for the award of a degree.

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Abstract

Energy security, climate change and economic growth are matters of critical international importance in an effort to achieve a sustainable future. Energy consumption in buildings contributes to higher greenhouse gas emissions than the industrial or transportation sectors combined. In India, the energy in the residential sector accounts for almost 50% of the total energy consumption. The need for comfortable internal environments, healthy indoor air quality and the consequences of global warming are all contributing factors to the high reliance on mechanical cooling and ventilation systems. In recent years, financial growth and increase in disposable income in India, have accelerated purchases of such mechanical systems. In metropolitan cities of India with extreme climates (hot and dry, warm and humid), the use of these systems increases by 30% every year. This upward trend is likely to continue in response to occupants' higher comfort expectations and the continuous increase of the outside temperature during the summer months due to climate change. This could further impact the climate and the electricity grid. Innovative solutions should establish reliable strategies for cooling purposes by utilizing the use of natural ventilation.

Mixed-mode buildings rely on both mechanical and natural systems to maintain comfortable conditions. Although the performance of mixed-mode buildings has already been studied and there is evidence for its positive impact on the reduction of energy demand, there is still a lack of knowledge on the best methods for controlling mixed-mode buildings. Today, the majority of the available algorithms for the control of mixed-mode systems are very simplistic and at a primitive stage of development. Typically, the control algorithms "make the decision" based on a predefined static set-point temperature, disregarding other important parameters, such as relative humidity, the position of windows and activity of occupants. Control algorithms that would account for a variety of parameters are of paramount importance to achieve energy savings whilst maintaining thermal comfort conditions.

The aim of this research was to investigate the impact on thermal comfort and energy savings of novel and sophisticated control algorithms in mixed-mode residential buildings in India.

Initially, it was important to identify all the control parameters that were important to be included in the control algorithms. Then the control algorithms were designed and presented in flow charts. To analyse the performance of the proposed control algorithms, computer simulations were performed, whilst a validation analysis was conducted to provide evidence

of the validity of the control algorithms. Computer modelling comprised of co-simulations, using Dynamic Thermal Modelling (DTM) (EnergyPlus) and equation-based tools (Dymola using the Modelica language). The coupling of these was achieved using the Functional Mock-up Interface (FMI) for model exchange. The co-simulations enabled to examine the energy saving potential that can be achieved by the proposed control algorithms. In order to evaluate the ventilation performance of the proposed control algorithms, the ventilation rates and ventilation effectiveness of the systems were analysed using Computational Fluid Dynamics (CFD). This allowed the final analysis which included the evaluation of the ventilation performance of the control algorithms by calculating the ventilation effectiveness. To provide evidence of the proposed control algorithms and simulation approach, a validation study was done using data from an experimental chamber in India.

This research has contributed to the existing body of knowledge by providing four main conclusions concerning the design and control of mixed-mode ventilation and cooling systems: i) to deliver comprehensive guidelines on the design and control of mixed-mode buildings, and the ways in which the co-simulations can be implemented to improve the existing control algorithms that can be found in the literature; ii) the use of the co-simulations showed that the developed control algorithms, when dampers/windows and ceiling fans are used, can improve the predicted hours of thermal comfort by up to 1900h compared to the scenarios when the ceiling fans were turned off, while achieving up to 55% energy reduction depending on the city; iii) the CFD simulations predicted that cross ventilation with the maximum opening areas for windows and dampers in combination with the operation of the ceiling fans can dilute the contaminants and/or heat in the building resulting in comfortable internal environments resulting in heat removal effectiveness of 1.65; and iv) the accurate and validated control algorithms that were developed in this research can be used for any study that requires control of mixed-mode buildings regardless of the geometry of the building. The use of co-simulations provides great flexibility since the same control algorithms can be used in any geometry or building location without the need for any modification of the code.

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This PhD was closely linked with a wider international research project, LECaVIR, that involved Loughborough University and CEPT University led by Professor Malcolm Cook. This collaboration has been sealed by joint publications, highlighting the tangible sign of the positive teamwork. The success of this collaboration has set the basis for future research projects between these two universities. The author would like to thank all the researchers that were involved with the LECaVIR project and especially during the collection of the experimental data. This includes Deepta Mishra and Jayamin Patel from CEPT University. Special thanks to Professor Jonathan Wright for his valuable inputs. Finally, this study was financially supported by the Engineering and Physical Sciences Research Council (EPSRC) as part of the activities carried out by the London-Loughborough Centre for Doctoral Training in Energy Demand, a partnership between University College London (UCL) and Loughborough University, for what the author expresses his gratitude.

I am thankful to my parents (Evangelos and Panagiota); sister (Angeliki) and brother in law (Giorgos) for their love, support and sincere belief in me. I would like to thank all my friends for being supportive all these years, and special thanks to my partner Dr Theoni Maria Massara for her endless encouragement especially in the last months of writing. I would like to dedicate this PhD thesis to my two nieces, Panagiota (Nagia) and Anna. I hope this PhD will ignite their intellectual curiosity and will help them chase their dreams and achieve every goal, academic or not, in their lives.

I would like to close with a quote from Albert Einstein:

"***Education is not the learning of facts, but the training of the mind to think***."

and from George Box

"***All models are wrong, some are useful***."

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Nomenclature

AC	Air Conditioning
ACH	Air Change Rate
AFN	Airflow network
AHU	Air Handling Unit
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BAU	Business As Usual
BCVTB	Building Controls Virtual Test Bed
BMS	Building Management System
BS EN	British Standard European Norm
CAV	Constant Air Volume
CFD	Computational Fluid Dynamics
CIBSE	Chartered Institution of Building Services Engineers
CO_2	Carbon Dioxide
CVRMSE	Coefficient of Variation of Root Mean Square Error[%]
DEC	Direct Evaporative Cooler
DTM	Dynamic Thermal Modelling
DX	Direct Expansion
ECBC	Energy Conservation Building Code
EMS	Energy Management System
FCU	Fan Coil Unit
FMI	Functional Mock-up Interface
FMU	Functional Mock-up Unit
GCRF	Global Challenges Research Fund
GHR	Global Horizontal Radiation
HVAC	Heating Ventilation and Air Conditioning
IAQ	Indoor Air Quality
IDC	Indirect Evaporative Cooler
IMAC	Indian Model for Adaptive Comfort
ISHRAE	Indian Society of Heating, Refrigerating and Air-Conditioning Engineers
LECTB	Low Energy Cooling Test Bed
MBE	Mean Biased error [%]
MM	Mixed-Mode
MPC	Model Predictive Control
NV	Natural Ventilation
PI	Proportional Integral
PID	Proportional Integral Derivative
PMV	Predicted Mean Vote
PPD	Predicted Percentage Dissatisfied
PPO	Purpose Provided Openings
RH	Relative Humidity
VAV	Variable Air Volume

VCS	Ventilation and Cooling Strategy
WAC	Water-Cooled Air Conditioning
WBT	Wet-buld Temperature

List of Abbreviations

$q_{analytical}$	Volumetric flow rate	[m³/s]
$\Delta\vartheta$	Temperature difference between the internal and external	[K]
g	Gravitational accelation	[m/s²]
h_a	Height of the opening for natural ventilation	[m];
$\bar{\vartheta}$	Mean air temperature	[°C]
$q_{measurement}$	Measured volume flow rate	[m³/s]
C_i	Measured concentrations at point i	[ppm]
C_s	Steady-state concentration	[ppm]
$\langle C\rangle$	Average concentration of the contaminants at the room	[ppm]
ε_t	Heat removal efficienty	[-]
ε_j	Concentration Removal Efficiency	[-]
s_i	Simulated data	[-]
m_i	Measured data	[-]
N_P	Number of data points	[-]
$T_{air,masterbedroom}$	Average air temperature	[°C]
$V_{masterbedroom}$	Volume of masterbedroom	[m³]
$\dot{m}_{total}$	Total air flow through the openings due to wind and buoyancy forces	[m³/s]
$\dot{m}_b$	Airflow due to buoyancy forces only	[m³/s]
$\dot{m}_w$	Airflow due to wind forces only	[m³/s]
C_d	Discharge coefficient	[-]

$\boldsymbol{A_w}$	Effective area of the windows	[m^2]
$\boldsymbol{V_r}$	Wind speed	[m/s]
$\boldsymbol{C_p}$	Specific heat capacity of air	[kJ/(KgK)]
$\boldsymbol{Q_{gains}}$	Total internal gains	[W]
$\boldsymbol{h_a}$	Vertical distance between centres of the openings	[m]
$\mathbf{T_{comfort}}$	Comfort temperature provided by the standards	[°C]
$\mathbf{T_{om}}$	outdoor mean temperature	[°C]
$\boldsymbol{T_{int,aver}}$	Average internal air temperature	[°C]
$\boldsymbol{T_{i,CSP}}$	Cooling setpoint	[°C]
$\boldsymbol{T_{i,HSP}}$	Heating setpoint	[°C]
$\boldsymbol{T_{out,aver}}$	Average external temperature	[°C]
$\boldsymbol{\bar{t}}$	Average value of internal and external temperature	[°C]
$\mathbf{q_{ic}}$	convective flux from surface i	[J/(kgK)]
$\mathbf{A_i}$	Area of the surface	[m^2]
$\mathbf{Q_{internal}}$	Internal heat gains	[W]
$\mathbf{Q_{heatextraction}}$	Heat extraction rate from the room	[W]
$\mathbf{V_{room}}$	Volume of the room	[m^3]
$\mathbf{C_p}$	Specific Heat of the Air	[J/(kgK)]
$\Delta\boldsymbol{T}$	Temperature Difference	[°C]
$\Delta\boldsymbol{t}$	Sampling time internal	[h]
$\boldsymbol{\rho}$	Density	[kg/m^3]
$\mathbf{q_i}$	conductive heat flux on the surface i	[J/(kgK)]
$\mathbf{q_{ir}}$	radiative heat flux from solar	[J/(kgK)]
$\mathbf{q_{ik}}$	radiative heat flux from surface I to k	[J/(kgK)]
$\mathbf{h_{ik,r}}$	linearized radiative heat coefficient between surface i and k	[W/(m^{2}K)]
$\mathbf{T_i}$	Air temperature of surface i	[K]

$\mathbf{T_k}$	Air temperature of surface k	[K]
$\mathbf{h_c}$	convective heat transfer coefficient	[W/(m2K)]
$\mathbf{T_{room}}$	Room air temperature	[K]
$\mathbf{P_v}$	Vapour Pressure	[mm Hg]
$\mathbf{P_b}$	Barometric Pressure	[1 atm (1,01325*10^5 Pa)]
$\mathbf{R_H}$	Relative Humidity	[%]
Q	air mass flow rate	[kg/s]
$\mathbf{C_t}$	reference condition temperature correction factor	[-]
$\mathbf{C_Q}$	Air mass flow coefficient	[kg/s.m2]
$\mathbf{C_d}$	Discharge coefficeint	[-]
$\mathbf{Q_{sensible}}$	Sensible Cooling load	[kW]
U-Value	Thermal transmittance	[W/(m2K)]
$\mathbf{E_{fan}}$	Energy consumption by the fan	[kWh]
n	Number of hours the fan operated	[h]
$\mathbf{Q_{total}}$	Total gains in a room	[W]
$\mathbf{Q_{external}}$	External heat gains	[W]

Chapter 1: Introduction

1.1. Background

1.1.1. Global trends

Recent studies have shown that the building sector is responsible for approximately 20% of the world's total energy consumption (EIA, 2017). The energy that is used for space heating and cooling in residential buildings is 43% in USA (Levine *et al.*, 2012), about 50% in India (Kapoor *et al.*, 2011) and almost 30% in the UK (DECC, 2016) of the total energy consumption in buildings for each country respectively. Achieving long-term sustainable growth is a problem that requires worldwide attention and international collaborative effort.

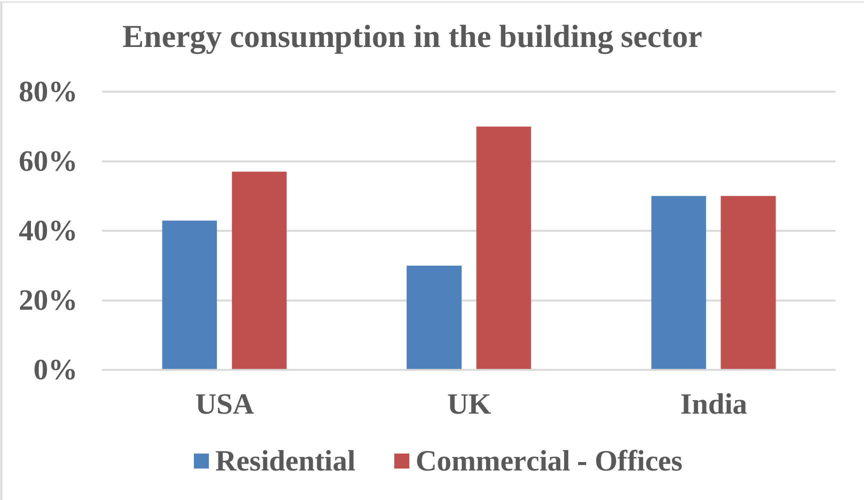


Figure 1-1: Energy consumption in the building sector

1.1.2. Indian trends

In India, the increasing population and urbanization, the decline of air quality and at the same time the higher living standards expectations are all characteristics of the majority of the developed and developing countries, such as in India. Hence, the way in which our society will overcome all these challenges will shape and the future of our planet. The UK Government introduced in 2016 the Global Challenges Research Fund (GCRF) which aims to help the developing countries, such as India, to address the challenges the will face in the future (BIS, 2016). This scheme provides a great opportunity for a collaborative effort between the UK and India to tackle one of the greatest challenges of our society, to reduce energy demand in the building sector.

The sustained economic growth in India in recent years has resulted in a relentless increase in energy demand in several sectors such as buildings, transportation and industry. Residential buildings in India account for almost 22% of the total electricity consumption (Central Electricity Authority, 2013) and the projections indicate that the consumption will continue to increase by 8% annually (Kumar, 2011). Approximately, 31% of the Indian population, 400 million, live in urban areas and this figure is expected to increase to 50% by 2050 (United Nations, 2016). To accommodate the increase of the population, recent studies have shown that the aggregate floor area of buildings in India is expected to increase by 500% with approximately 20 billion m^2 of new building floor area by 2030 (Kumar, 2011).

Due to the constant increase in the disposable income in Indian houses, the consumer purchase power is expected to increase, which will lead to an increase in the use of domestic appliances, especially for cooling purposes. The purchase of Air-Conditioning (AC) units is becoming commonplace in residential buildings rather than a luxury (Rawal and Shukla, 2014). Nowadays, more people tend to rely on mechanical cooling technologies when indoor conditions are not comfortable (Rawal and Shukla, 2014). The electricity demand is expected to increase in the next few decades (USAID, 2014) and since fossil fuel power plants are dominating the electricity production in India, it is estimated that the CO_2 emissions will increase by 860% (Shnapp and Laustsen, 2013). Furthermore, the increase in thermal comfort expectations in combination with the expected growth in the floor area of the residential sector in India could lead to a growth in demand for cooling (Rawal and Shukla, 2014). More specifically, a recent study showed that energy consumption in Indian residential buildings is expected to increase by eight times by 2050 (Rawal and Shukla, 2014). The growth in air-conditioning units is likely to add further pressure on the unstable electricity infrastructure of India (Indraganti *et al.*, 2014), which could lead to increased difficulty in meeting peak demand (Phadke, 2014). Hence, cooling strategies that could maintain acceptable indoor conditions without solely relying on AC units should be developed as a measure to mitigate climate change.

In the cities of India, the cooling electricity consumption dominates the total peak electricity consumption (Phadke, 2014). In previous years, the cooling was mostly achieved by natural ventilation solutions; however, today more and more people are using AC units instead, due to the deeper penetration of the mechanical systems. Room AC units sales in India have grown at an annual rate of 17% in the last 12 years (BSRIA, 2015). However, this has been observed in other countries too; in 2014, the purchase of AC units in the UK increased by 22% compared to 2013 (Antonijevic, 2014). This upward trend would continue to increase in response to occupants' higher comfort expectations and the continuous increase of the outside temperature during the summer months. This could have a significant negative impact on the climate and the electricity grid. Hence, a solution that could minimize the use of the AC units is urgent. Innovative solutions should establish reliable strategies for cooling purposes by using low energy cooling technologies and utilize the use of natural ventilation.

The effective cooling of residential buildings in warm weather conditions, such as the predominant Indian climate, requires urgent attention by designers. Relying only on mechanical cooling technologies will significantly impact on energy consumption, and natural ventilation strategies cannot always maintain thermally comfortable internal environments due to extreme weather conditions. Hence, space cooling of residential buildings should be carefully designed particularly considering that new build properties have at least a 60-year lifespan and should be thus futureproofed (Rawal and Shukla, 2014). It is essential that during the design stage, design teams and engineers will utilize the use of natural ventilation to maintain comfortable internal conditions. For additional cooling supply, low energy cooling technologies should be used. This approach of conditioning spaces by using a combination of natural ventilation, from operable windows, and mechanical systems, such as AC units that provide air distribution and cooling, is known as "*hybrid*" or "*mixed-mode*" ventilation/cooling.

The mixed-mode ventilation strategies have been used in the past mainly as energy savings measures. By using mixed-mode cooling systems, energy consumption could be minimised while providing comfortable internal conditions (Brager and Baker, 2008). The difference between conventional cooling and mixed-mode systems is that the latter is a sophisticated system that can be automatically controlled to switch from natural to mechanical mode in order to minimise the energy consumption without compromising the internal thermal conditions. Natural ventilation may not be suitable for all climates and times of the year and thus low energy mechanical cooling systems can be used during these periods/climates in order to maintain comfortable internal conditions with the minimum energy use. Few examples of the mixed-mode buildings are: *The Natural Resources Defence Council in California* (CBE, 2007)*; The Chicago Centre for Green Technology* (CBE, 2007)*; The Weber Centre at Judson University in Chicago* (Kaiser *et al.*, 2007)*; The Wilkinson Building at the*

University of Sydney, Australia (Rowe, 2003) as well as many examples of mixed-mode residential buildings in the UK (Loveday *et al*., 2016) and in India (Babich *et al.*, 2017).

Although the performance of mixed-mode buildings has been already studied and there is evidence for its positive impact on the reduction of energy demand, there is still lack of knowledge on the methods to control of mixed-mode buildings. Today, the majority of the available algorithms for the control of mixed-mode systems are very simplistic and at a primitive stage of development. Typically, the control algorithms “make the decision” based on a predefined static set-point temperature, disregarding other crucial parameters, such as relative humidity, the position of windows and activity of occupants. The suitability of these algorithms has been examined by others, in office buildings and not in residential spaces. However, previous studies have shown that by 2050, the floor space in residential buildings in India will account for almost 85% of the total floor space (Rawal and Shukla, 2014). Thus, it is crucial to examine the impact on the energy consumption of more flexible control algorithms in residential buildings, and especially in cases where automated control is preferred over manual control. By using automated control of the systems, it is more likely to achieve the predicted energy savings compared to the manual control.

1.2. Aims and Objectives

The research reported here explored the prospects of energy saving by evaluating the potential of natural ventilation and low energy cooling systems, the cautious use of air conditioning units and ceiling fans, digitally controlled by the novel mixed-mode strategies for residential buildings in India.

The **aim** of this research was to investigate the impact on thermal comfort and energy savings of novel and sophisticated control algorithms in mixed-mode residential buildings in India.

The main ***research questions*** that this research attempts to address are:

- Which control strategies should be employed in mixed-mode buildings, in order to meet thermal comfort expectations in India?
- Could sophisticated control algorithms for mixed-mode buildings reduce the energy demand without compromising the indoor air quality of apartment blocks in India?

The above research aim and questions will be addressed by the following ***objectives***:

1. Identify control objectives, parameters and design flexible control algorithms for low energy mixed-mode residential building in Indian residencies. Develop flexible control algorithms for mixed-mode residential buildings in Indian residencies that will utilize the use of natural ventilation and minimize the use of mechanical cooling and implement them into the Modelica language;
2. Validate the co-simulations using data from the experimental chamber in CEPT, India. The validation of the results is important to ensure that the developed control algorithms generate accurate results. Additionally validate the methodology to calculate the ventilation effectiveness using data from the experimental chamber to ensure the that proposed methodology to calculate the ventilation effectiveness generates accurate results. The calculations of the ventilation effectiveness was done using Computational Fluid Dynamic software;
3. Perform simulations (Dynamic Thermal Modelling coupled with Dymola) for an application demonstration case to examine the applicability of the control algorithms under a typical residential building type in India; and
4. Perform simulations (Computational Fluid Dynamic) for the application demonstration case to examine the ventilation effectiveness and thermal comfort conditions for a variety of outdoor conditions.

1.3. Scope and boundaries of the research

The scope of this research was to design control algorithms for mixed-mode residential buildings in India. To design these control algorithms, it was essential to identify the control parameters that were used at the control algorithms. The boundaries that were used in this research were: i) the use of residential buildings in the Indian context, ii) the climatic conditions were specifically for India and iii) the validation of the proposed control algorithms was made against data from an experimental facility;

1.4. Outline

The thesis is divided into eight chapters. A thesis flow diagram (shown in Figure 1.4) provides an overview of the thesis structure.

Chapter 2 presents findings from an extensive literature review that has been conducted, including discussion on mixed-mode buildings, thermal comfort standards used for mixed-mode buildings, and control. Chapter 3 discusses the proposed research design including the research methods that have been employed to address the research questions. It includes the geometries that were used for the validation study and the demonstration case. Chapter 4 presents a detailed explanation of the sophisticated control algorithms that have been developed for the purpose of this research. Chapter 5 highlights the most important outputs from the co-simulations for the demonstration case. For the co-simulations, the geometries described in Chapter 3 and the control algorithms, Chapter 4, were used to perform the co-simulations and estimate the energy savings that can be achieved by implementing the proposed control algorithms. Chapter 6 and 7 presented the results from the analysis regarding the energy saving potentials and the ventilation effectiveness of the proposed control algorithms respectively. Chapter 8 summarized the main conclucions and the limitations of this research and provided ideas for future work.

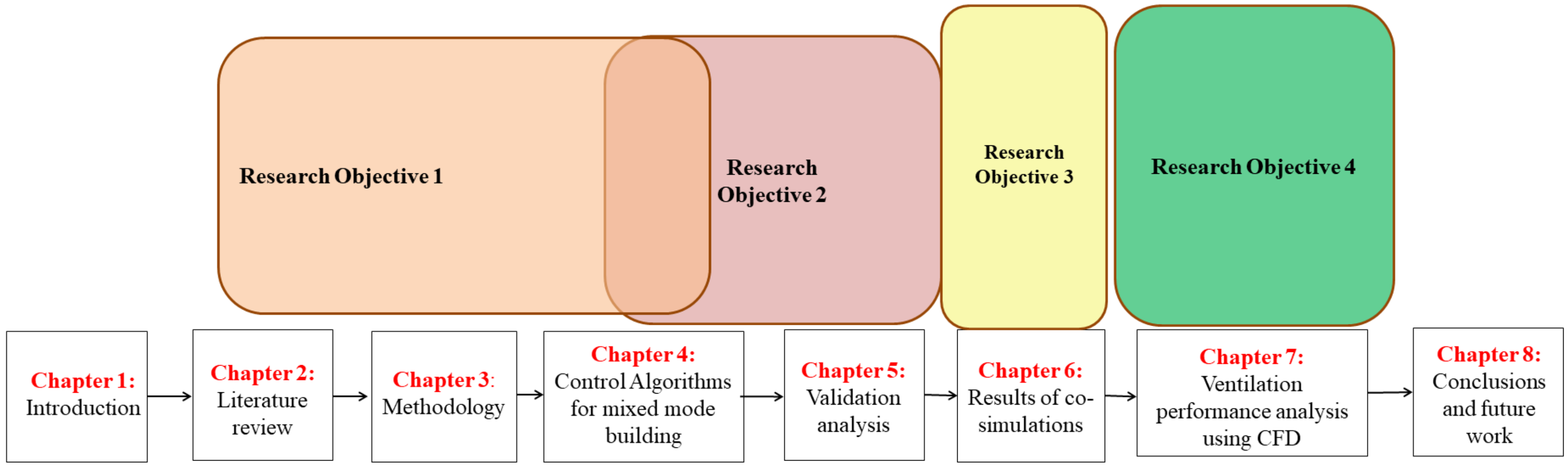


Figure 1-2: Thesis structure navigation flow diagram

Chapter 2: Literature Review

2.1. Introduction

The purpose of this chapter was to review previous relevant work in mixed-mode buildings in order to identify gaps in the literature and establish the research focus. The chapter begins by discussing the classification of mixed-mode systems and their advantages and it addresses the thermal comfort standards that are used for mixed-mode buildings. The importance of control systems in buildings is highlighted in this chapter and previous studies in control algorithms in mixed-mode buildings are presented. Next, the different cooling technologies that have been used in mixed-mode buildings are reviewed. Finally, it provides an overview of the current simulations techniques and software that are used to assess the energy performance, control and internal environment of mixed-mode buildings.

2.1. Mixed-mode ventilation systems

Natural ventilated or free-running buildings are of great interest to designers, architects and building owners, especially due to their potential to consume considerably less energy throughout the life span of a building (Dhaka *et al.*, 2015). Nowadays, energy-efficient and sustainable buildings have become of paramount importance; buildings should be designed in such a way to interact and utilize the outdoor environment to maintain acceptable indoor environment. The extent to which the natural solutions for ventilation, cooling, lighting etc. could be utilized, depends on the weather conditions and on the building design.

Historically, natural ventilation has been used in different forms to ventilate buildings. A building could be naturally ventilated if there is at least one external opening to allow the air to flow in and out. The driving forces for a natural ventilation system are the wind and buoyancy forces. These forces use the space of the building to supply and extract air mainly through the openings which are known as purpose-provided-openings (PPOs) as the paths for the air (Jones *et al.*, 2016). However, natural ventilation through unintentional or adventitious openings could result in either uncomfortable indoor conditions for the occupants or high heat losses for the building (Etheridge, 2012). Wind creates complex and unsteady airflow around buildings in response to either the dominant outdoor conditions or the buildings' characteristics (Santamouris and Asimakopoulos, 1996). Wind velocity, temperature, humidity as well as topographical properties, all form the micro-climate around the building while the building design determines the performance of a natural ventilation system. The difference between the external and internal air temperatures results in different air densities which are the prevailing buoyancy forces. Accordingly, the flows that are created naturally, from both wind or buoyancy-driven forces, are more complex and thus harder to predict compared with those generated by mechanical systems (Heiselberg, 2004).

Therefore, it is important to investigate the effectiveness of passive environmental control systems in residential buildings and identify their potential for application in new buildings or as a retrofitting option. The effectiveness of any passive environmental control system can be demonstrated by analysing the climatic conditions of the region as well as the traditional architecture or the vernacular architecture of the place. For the purpose of the current research project, analysis of the Indian climatic,geographical characteristics and traditional architecture will be implemented.

In the past 50 years, mechanical cooling systems were introduced in buildings to meet the indoor environmental needs of the occupants. These systems have developed to provide constant airflow and maintain indoor conditions with a very narrow temperature range. The shift to mechanical cooling was driven mainly due to an urgent need for more airtight buildings and to reduce the heat losses of a building (Terkildsen, 2013). However, natural ventilation over recent years has attracted great attention as an energy saving measure to mitigate the use of mechanical systems for space cooling. Both natural and mechanical systems have developed with the focus of delivering thermally comfortable internal environments and adequate indoor air quality (IAQ) with the minimum energy consumption, however, in some cases, especially in very warm and humid climates, natural ventilation systems often have been found to be unable to provide acceptable IAQ and thermally comfortable internal environments (Delsante and Vik, 2000; Heiselberg, 2002). The use of mechanical cooling is unavoidable in these cases, hence, a combination of mechanical and natural ventilation strategies should be considered.

2.1.1. Classification of mixed-mode systems

Although there is not a standard mixed-mode design approach for buildings due to their unique design and requirements, there is a classification of the different mixed-mode systems based on how each system integrates the natural ventilation and mechanical cooling. The Center for the Built Environment at the University of California developed a summary report to describe the different mixed-mode ventilation systems that are available (CBE, 2007) as:

- *concurrent mixed-mode* (figure 2-1A) where both systems operate at the same space at the same time. The mechanical system (Heating, Ventilation and Air-Conditioning (HVAC) operate as supplementary ventilation or cooling while the occupants are free to operate the windows based on their personal preference. This strategy is the most known and used mixed-mode ventilation strategy typically for open-plan offices and residential buildings.
- *change-over mixed-mode* (figure 2-1B) where the mechanical systems and natural ventilation operate at the same space but in different periods of time. This system requires an automation control system to determine which mode should be used. The selection is mainly based on a variety of input parameters such as outdoor conditions and occupancy. The automation systems inform the occupants to either open / close the windows.
- *zoned mixed-mode* (figure 2-1C) where the mechanical cooling and natural ventilation operate at the same time but in different spaces of the building. This ventilation strategy is suitable mainly for big office buildings where some offices could use natural ventilation by operating the windows and some other rooms i.e. conference rooms could use HVAC systems for fresh air distribution.

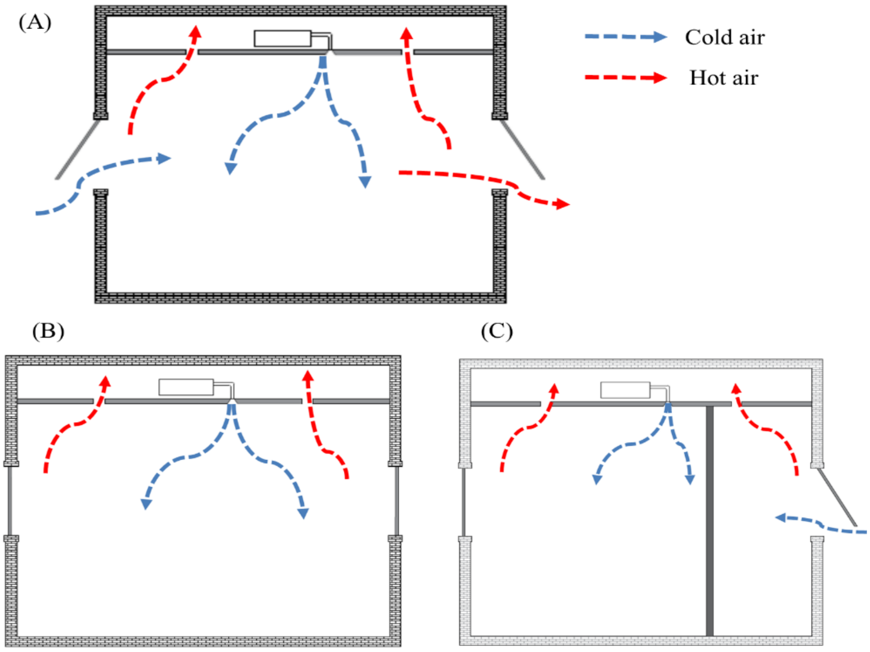


Figure 2-1: Classification of mixed-mode systems. Concurrent mixed-mode system (A); Change-over mixed-mode system (B); and Zoned mixed-mode system (C).

2.1.2. Advantages and disadvantages of mixed-mode buildings

A well-designed mixed-mode building could eliminate the use of mechanical cooling for a considerable period of time, depending on the climate conditions, and lead to energy demand and CO_2 reductions. As an example, a research conducted by Emmerich, (2006) showed that the simulated energy consumption for an office building in Minneapolis, USA decreased by approximately 48% (3500 kWh) over a period of one year when mixed-mode ventilation was used instead of purely mechanical cooling.

In addition, mixed-mode strategies, fundamentally provide flexibility and redundancy in the space conditioning systems of buildings. This could result in potentially smaller HVAC units compared to purely mechanical ventilated buildings for the same floor area. Finally, mixed-mode buildings could provide higher occupant satisfaction, due to their interaction with the system, in addition to energy savings. As the adaptive theory describes, when the occupants have their freedom to adjust the indoor environment by operating the windows in order to meet their thermal preferences, they tend to accept a wider range of internal air temperatures (Brager and de Dear, 1998). Furthermore, previous studies have shown that mixed-mode buildings maintained better IAQ compared to purely mechanical ventilated buildings. A cross-sectional field study between six countries in Europe and in the USA revealed that the air-conditioned buildings have approximately 30-200% higher indices of sick building syndrome[1] symptoms (Seppänen and Fisk, 2002).

Another advantage of mixed-mode building over fully naturally ventilated building is that in periods of the year that the outdoor air quality is very poor, the occupants can rely on the use of mechanical ventilation to supply outdoor air. Using the filters the incoming air can have better quality. On the contrary, fully naturally ventilated building do not provide this flexibility so either the quality of indoor air will become very poor or the occupants will not operate the windows and this might impact the thermal comfort conditions.

Nevertheless, a mixed-mode ventilation strategy could add complexity to the building design and also the occupants of the mixed-mode building should familiarise themselves with the buildings' control. Hence, there is a concern that the lack of occupants' experience with these systems could result to waste of energy, especially for the concurrent mixed-mode strategies (CBE, 2007). The need for humidity control is essential during the natural ventilation mode in mixed-mode buildings as the internal environment is highly affected by the external conditions (CBE, 2007). Hence, there is a need for detailed investigation of advanced controls of mixed-mode buildings in response to outdoor air properties parameters, such as air temperature, relative humidity, wind speed.

2.2. Thermal Comfort and Indoor Air Quality in mixed-mode buildings

People spend more than 85% of their time indoors, either at workplaces or at their homes (Klepeis *et al.*, 2001). Mixed-mode buildings are of great interest to designers and building owners as their energy consumption is less compared to purely mechanical ventilated/cooled buildings (Dhaka *et al.*, 2015). Occupants of mixed-mode buildings though, experience variable internal environment throughout the day (de Dear and Brager, 1998).

Thermal comfort is defined by CIBSE Guide A, (2015) as: "*where there is broad satisfaction with the thermal environment i.e. most people are neither too hot nor too cold*". Two different methodological approaches are used to assess the thermal comfort of space: the human heat balance approach and the adaptive approach (Dhaka *et al.*, 2015). The first approach has been developed by Fanger, (1970) based on experimental work conducted in a

[1] Sick Building Syndrome (SBS) describes a range of symptoms such as headaches, dizziness, nausea, fatigue etc that could be linked to spending time in a building, most often a workplace (NHS, 2017).

controlled environment (climatic chambers). From this research, a comfort index was calculated to capture the percentage of people that will be probably thermal dissatisfied based on the internal conditions. To calculate this comfort index, known as Predicted Mean Vote-Percentage Predicted Dissatisfied (PMV-PPD), six thermal environmental variables are being used: room air temperature; globe temperature; relative humidity; air velocity, clothing insulation and activity.

An experimental study conducted in residential buildings concluded that the indoor conditions at residential spaces are not comparable to those during the experiments for calibration of the PMV and PPD indices (Peeters *et al*., 2009). The internal environment for residential buildings is not in steady-state conditions since both the activity level and clothing insulation could vary in a very small timescale. Hence, they concluded that the majority of the residential buildings should be considered as free-running buildings. Following this, a field study conducted in naturally ventilated Indian residential buildings during summer and monsoon periods revealed that the Fanger's PMV index failed to accurately capture the actual sensation vote (Indraganti, 2010). Specifically, the PMV figure was always higher compared to the actual sensation vote. The occupants voted within the comfort range of -1 to +1 (7-point scale) for outside temperatures of 26.0°C to 32.5°C while the PMV predicted to be thermally uncomfortable for those temperatures. The research concluded that the thermal environment in naturally ventilated spaces changes according to the outdoor conditions and the PMV index could not capture it (Indraganti, 2010). Both studies emphasized the inability of Fanger's model to capture the occupants' interaction with their local environment. This is also highlighted by the fundamental assumption for the adaptive principle "*If a change occurs such as to produce discomfort, people react in ways which tend to restore their comfort*" (Nicol and Humphreys, 2002). The occupants of a building should not be considered as passives receivers of the given thermal conditions but they interact/"control" constantly with the environment via feedback loops until they "reach" their desirable thermal conditions (Brager and de Dear, 1998).

2.2.1. Thermal comfort models in mixed-mode and naturally ventilated buildings

During the last two decades, the adaptive model approach (Brager and de Dear, 1998; Nicol and Humphreys, 2002) has gained great attention worldwide within the research community. Experimental studies have proven that the use of the adaptive approach provides higher accuracy of results (Karyono, 2000; van der Linden *et al.*, 2002; Wong *et al.*, 2002; Fato Francesco Martellotta and Chiancarella, 2004; Feriadi and Wong, 2004) regarding the thermal sensation. Today, several internationally recognised standards, namely:

I. ASHRAE Standard 55 (2013) which is the standard in North America dealing with thermal comfort;
II. the European standard EN 15251 (BS EN15251 CEN, 2007); and
III. CIBSE Guide A (CIBSE Guide A, 2015) which is the standard for the UK have included on their methods for assessing thermal comfort the adaptive approach.

These standards have included the adaptive approach for assessing the thermal comfort of naturally ventilated spaces. However, the current standards use a “black to white” approach when it comes to the classification of the buildings; categorising buildings in those using AC units and those that are naturally ventilated. However, buildings are much more complicated since they could combine the use of AC for a specific period and utilize natural ventilation for some other period. In these standards, there is no distinct guidance on how to examine thermal comfort or to control/operate a mixed-mode building. For instance, the ASHRAE standard 55 suggests the use of the adaptive approach models on the purely naturally ventilated buildings and that the adaptive approach is not applicable to buildings equipped with mechanical cooling/heating units. This description excludes the mixed-mode buildings from the wider scope of the adaptive approach method. However, the flexible nature of a mixed-mode building allows it to operate in passive cooling mode predominantly, and to use additional mechanical cooling for a very small period depending on the regional climate conditions. Nevertheless, based on ASHRAE standard 55, mixed-mode building should be assessed using Fanger’s PMV approach (ASHRAE-Standard-55, 2013). On the contrary, the European standard BS EN15251 (BS EN15251 CEN, 2007) uses the term “free-running” building for the use of the adaptive approach. Hence it includes conditions where the occupants can either open or close the windows and there is no strict clothing guidance. Despite their growing popularity, today remains unclear how to assess the thermal conditions for a mixed-mode building. However, since the energy consumption and the IAQ of a building are closely related to the criteria that are used to control the indoor environment, it remains still unclear what are the actual positive benefits from a mixed-mode building (Olesen, 2007).

Adaptive control algorithms have been developed to assess the thermal conditions for free-running buildings with no mechanical cooling equipment. The algorithms aimed to provide an alternative approach to the static thresholds temperatures. ASHRAE-Standard-55, (2013) contained a graph that correlates the outdoor and indoor comfortable air temperature in naturally ventilated buildings:

$$T_{comf} = 0.31T_{om} + 17.8 \qquad \text{Eq. 2-1}$$

where T_{om} is the monthly outside temperature in °C. The ASHRAE Standard 55 model defines two comfort regions with 80% and 90% acceptability limits and formula 2-1 refers to the central line of the model, as it is shown in figure 2-2.

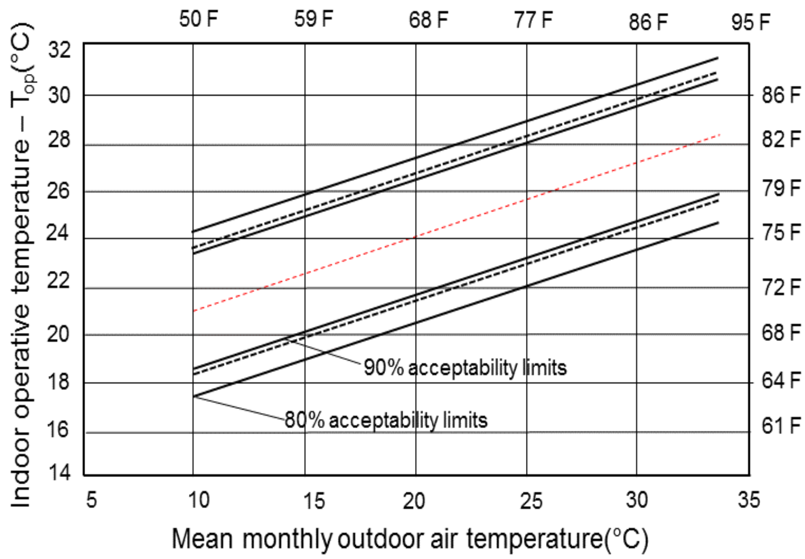


Figure 2-2: Acceptable operative temperature ranges (Excerpt from ASHRAE-Standard-55, (2013))

The BS EN15251 CEN, (2007) recommends a slightly different formula to predict the indoor air comfortable temperature.

$$T_{comf} = 0.33T_{om} + 18.8 \qquad \text{Eq. 2-2}$$

where T_{om} is the running mean outdoor temperature in°C.

The formula was developed by data collected in the EU project Smart Controls and Thermal Comfort (SCATs). When the indoor operative temperature is above 25°C then increased ventilation rates could balance any discomfort conditions (BS EN15251 CEN, 2007). The majority of the data were from free-running naturally ventilated office buildings during periods where no heating was needed but are applicable also for single residential buildings, apartment buildings and offices (BS EN15251 CEN, 2007).

Even though the proposed formulas from ASHRAE 55 and BS EN15251 look similar there are a few major differences. Firstly, the databases for creating these formulas are different since, for equation 2-1, ASHRAE formula, data were used from the world database of field experiments conducted by de Dear, (1998) while for equation 2-2, data from the European SCATs were used and specifically data that were collected in the UK. The ASHRAE formula (Eq. 2-1) is valid for naturally ventilated buildings only (ASHRAE-Standard-55, 2013), while the BS EN15251 could be applied to any free-running building (BS EN15251 CEN, 2007). In addition, for the ASHRAE formula (Eq. 2-1), the monthly mean outdoor air temperature was used while for the BS EN15251 contemporaneous weather data were used. This approach allowed the formula to handle better varying weather conditions.

An experimental study conducted by Singh *et al*., (2011) examined the thermal sensation of occupants at residential buildings in the North-East of India covering three different climate zones. The study was performed throughout the whole year to capture the votes during all seasons. The authors calculated both the actual votes from the occupants and the PMV index. The results showed that the PMV overestimated the indoor thermal conditions during summer months and underestimated it during winter months compared to the actual vote for all the climate zones. An explanation for the discrepancy of the results is that the PMV did not include adaptive opportunities. More specifically, the results highlighted that the occupants adopt different levels of adaptations in different seasons but in the same climate zone. The survey showed that the occupants used different adaptation techniques between winter and summer months. The actual mean vote of the occupants during winter time was great than in the summer. Furthermore, the authors concluded that if dynamic control of the mechanical cooling units was used, then there is potential for great energy savings. However, the authors pointed out, as many within the research community, that by constraining the applicability of the adaptive thermal comfort models at purely naturally ventilated buildings as suggested by the standards, will limit the energy savings potential of the mixed-mode buildings (Olesen, 2007).

In addition, Indraganti *et al*., (2014) conducted a field study for 14 months in 28 offices in two different geographical and climatic regions of Chennai and Hyderabad, India. Most of the office buildings in Chennai were mechanical cooled while the majority of the examined buildings in Hyderabad were in mixed-mode operation. The results from this study revealed that the average indoor globe temperatures for the majority of the buildings in both cities were above 26°C which considers being the upper limit based on National Building Indian Code (Bureau of Indian Standards, 2005). An adaptive comfort model for the naturally ventilated building was created based on the data from this study.

$$T_{comf} = 0.26T_{om} + 21.4 \quad \text{Eq. 2-3}$$

while for the mechanical equipped buildings, the linear relationship between the outdoor and indoor comfortable temperature is given by:

$$T_{comf} = 0.15T_{om} + 22.1 \quad \text{Eq. 2-4}$$

The results for both cities highlighted that the occupants felt thermally comfortable for indoor temperatures well above than the one proposed by the Indian and international standards. Furthermore, the analysis showed that the use of fans offset discomfort conditions for both cities. Hence, the introduction of higher air motion in Indian buildings might offer better thermal performance and energy savings.

The previous studies captured the thermal sensation over a variety of building types but for specific climate regions. Manu *et al*., (2016) examined the thermal sensation for 16 buildings in India for three seasons and the buildings were located in all the available climate zones of India. This research proposes a single adaptive thermal model for the mixed-mode buildings while the previous studies proposed different models for the natural and mechanical operation mode of the building. The adaptive thermal comfort model for mixed-mode buildings is presented below:

$$T_{comf} = 0.28T_{om} + 17.9 \quad \text{Eq. 2-5}$$

Where the outdoor temperature T_{om} is a 30-day running mean air-temperature ranging from 13-38.5°C. Similarly to the ASHRAE thermal comfort model, 80 and 90% acceptability limits are defined in the model as shown in the figure below (figure 2-3).

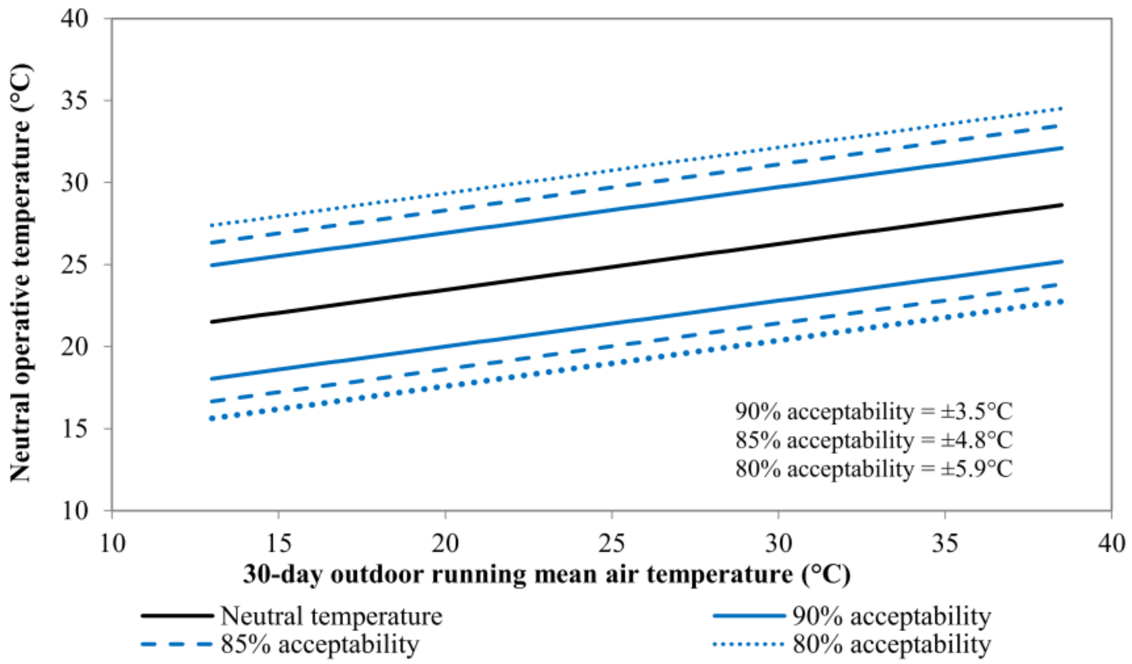


Figure 2-3: Acceptable operative temperature ranges for mixed-mode buildings from IMAC model (Excerpt from Manu et al., (2016))

The proposed comfort temperature by this research for mixed-mode buildings is lower compared to those proposed by ASHRAE Standard 55 and BS EN15251. Furthermore, the above study showed that Indians are more adaptive than the prevailing from ASHRAE Standard 55 and BS EN15251. Moreover, the outcomes from this field study demonstrated that the static Fanger's PMV model over-predicts the thermal sensation on the mechanically cooled buildings.

As the different adaptive comfort models suggest, it is feasible to achieve the same levels of thermal comfort for a space with higher levels of internal air temperature, when the adaptation methodology is used. In order to maximize the potential of the adaptive theory, is important to include it during the design of the control algorithms of a mixed-mode building.

2.3. Control systems of buildings

A successful control strategy for mixed-mode buildings needs to account for the best balance between energy consumption and indoor conditions. As described previously, this balance is affected mostly by the climatic conditions, the building design and the occupants' thermal preferences. Although the key scope is to maintain acceptable levels of IAQ, there is a need to effectively control the mixed-mode systems as they can be subject to either low ventilation rates, that could result to unacceptable IAQ, or high ventilation rates, that could result to unnecessary energy use. It is thus essential to provide an advanced control strategy for the mixed-mode systems to constantly provide adequate IAQ with the minimum energy consumption.

2.3.1. Principles of Control

The main objective of every control system is to maintain the control variable close to a set-point. A control system consists of three elements: a sensor, a controller and a control device (CIBSE Guide H, 2009). The sensor measures a variable and transmits its value to the controller which sends a signal to the control device, which reacts according to the signal to adjust the parameters, as shown in figure 2-4.

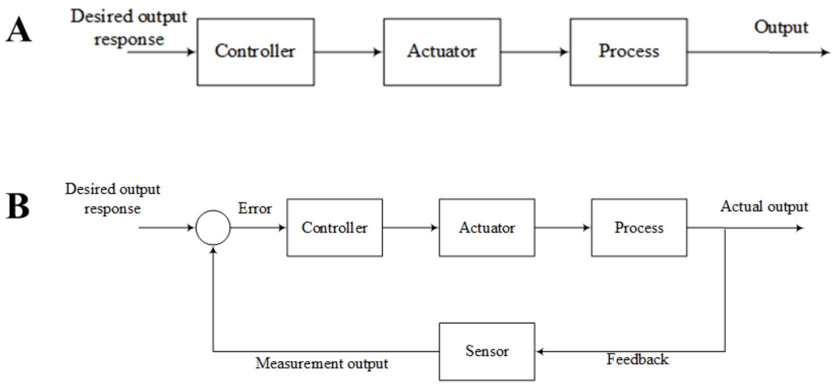


Figure 2-4: Principles of control for an open loop (A) and closed feedback loop (B) (Excerpt from (CIBSE Guide H, 2009)).

To control the variables, such as internal air temperature or ventilation rates, in a building, two control techniques are implemented: *an open-loop control* (figure 2-4A) and a *closed-loop control (*figure 2-4B*)*. In an *open-loop control* the control action is not affected by "process output". A central cooling unit that uses a timer to control the operation of the compressor is a good example of open-loop control. In this case, the cooling is supplied for a given period, regardless of the indoor temperature of the building. Often, this type of controller is used to more simple processes because of its simplicity and low-cost. However, when more accurate control is needed then it is essential to feed the output of the system back to the controller. This configuration is known as *closed-loop control (*figure 2-4B*)*, where the control action is dependent on the process output. In the previous example, a thermostat to measure the indoor temperature would send a signal to the controller to ensure that the indoor air temperature will be always within the set-point values.

In real buildings, it is common to have more than one inputs and outputs for a control loop. When a loop of controls is interconnected to each other they create a control sequence where both the inputs and outputs are continuous variables, such as the temperature. These advanced control loops are actually a set of complex interlocks where the operation of one is conditional on the state of another set (CIBSE Guide H, 2009).

2.3.2. Control modes

The control modes describe the way in which a change in the controlled variable will affect the control system (CIBSE Guide H, 2009). The most common control modes that are used for the buildings are:

- *Two position (on/off) control* typically gives a minimum and maximum output. A common domestic room thermostat is a very good representation of this controller. It is designed to maintain the air temperature within a given temperature range, defined by the set-points temperatures.
- *Proportional control* provides an overall control action which is proportional to an error signal. It is a type of linear feedback control system. In steady-state conditions, a proportional controller produces an offset or error, which increases with the load on the system. Following the previous example with the ON/OFF control, when the thermostat reaches its upper set-point instead of sending a signal to shut down the boiler, it reduces slightly the output of the boiler, in proportion to the error, so that the air temperature decreases gradually until it reaches the target point. This control provides a “smoother” control output compared to the two-position control.
- *Integral control* is a control action that is proportional to a control error integrated over time, however, is not often used by itself, at the control process of a building (CIBSE Guide H, 2009).
- *Proportional plus integral control (PI*) is the most common control system for HVAC units. If it has been set correctly then it could provide with almost stable control with zero error. However, the engineers should carefully design a PI controller. The additional integral action might result in a problem known as wind-up. The integral action accumulates the offset or error over time. Hence the accumulated value tends to

increase as long as the error is positive and decreases when is negative. If the control loop cannot “force” the error to be negative, then the accumulated value will increase constantly until it reaches its maximum value and remains at this stage until a condition change to the point where the system could “force” the offset to be negative. This could result in a huge waste of energy (Sellers, 2007).

- *Proportional plus integral plus derivative control (PID)* is another control loop feedback mechanism. The sum of three actions: a control action of the proportional to the control error P; a control action proportional to the integral of the control error I; and a control action proportional to the first derivate of the control error I forms the *PID controller*. The controller calculates the error over time as the difference between the desired set-point value and the actually measured process variable and corrects this value by using the P-I-D control actions. This is the most suitable controller for controlling slow variables such as temperature. However, it is more difficult to accurately tune since it has three different controllers.

2.3.3. Control and controlled parameters

The control and controlled parameters should be carefully selected based on the strategy employed. Typically, the control variables that are currently being used by the majority of the control strategies in mixed-mode buildings, are the internal air temperature, CO_2 concentration and less frequently the position (opening factor) of the windows (Brager *et al.*, 2007). Moisture sensors are often used at mixed-mode buildings in combination with surface temperatures of radiant cooling systems to prevent condensation rather than as an input to the control strategy (Brager *et al.*, 2007).

Furthermore, several modifiers could be used to ensure thermal comfort and avoid any unfavourable indoor conditions. Modifiers are considered all the variables that could be used as additional inputs to the controllers and can alter the controllers’ outputs. For example, the ventilation rate, the wind speed or rain could be used as a modifier. If the wind speed is very high, then this information could be used to modify the opening for the lower level. Considering all the inputs and modifiers, different control actions could be implemented for both the natural and mechanical systems to ensure comfortable levels of thermal comfort with the minimum energy consumption.

2.4. Control Algorithm Studies for mixed-mode buildings

The main challenge in the design of control systems in mixed-mode buildings is finding the balance between energy use, indoor thermal conditions and IAQ, users' satisfaction and robustness. The development of the control algorithm relies on many technical parameters such as building type, ventilation system, climate conditions, solar shadings, outdoor pollution etc. but also on social parameters such as occupants' dress outfit, thermal expectations etc.

Although users should be freely able to control their own environment, automatic control is preferred in many cases of mixed-mode buildings due to the complexity of the mixed-mode systems. Heiselberg, (2002) mentioned that occupants want to have a system that could maintain comfortable internal environments but allow them to adjust a parameter when the internal conditions are undesirable.

A review of the literature suggested that there is no generic control algorithm for mixed-mode buildings (Brager *et al.*, 2007). Most of the times the algorithms are in the form of flowcharts and a building energy management system (BMS) is used to implement these controls. However, the combination of the sensors and actuators that are used and the control strategies adopted can vary considerably between each case.

Brager *et al*., (2007) summarized several controls algorithms that are used in mixed-mode buildings. A more general control sequence was presented with the intention of capturing all the potential inputs and outputs and the control signals that will be used to effectively control a mixed-mode building. The control algorithm uses the internal air temperature to "decide" whether to operate the windows. The logic diagram includes also a control of the CO_2 concentration in order to adjust the opening of the windows. Furthermore, the wind speed and the existence of rain was taken into account at the control process. A similar study for a mixed-mode school building in the UK used a *changeover (see Figure 2-1)* strategy. Stack ventilation through a tall atrium and wind tower are used to ensure adequate ventilation rates. At this case, the outside temperature has been used to modulate the window openings. At the same time, the wind speed was used as a modifier variable, where the openings might be reduced in increments as the wind speed gets too high in order to avoid discomfort conditions due to high air velocities. However, the control algorithm did not consider the relative humidity, the ventilation rates or the existence of rain parameters at the process of controlling the openings. Hence, it is very likely that during some periods of the year the IAQ will be very low because the windows will be closed due to high wind speeds and not adequate fresh air will be provided.

A school building in the Netherlands used operable windows and mechanical cooling, to provide good IAQ and thermal comfort conditions (van Der Aa, 2002). The control of the ventilation strategy differs from the previously mentioned cases as it is based on measurements of both CO_2 levels and temperature and alter the control strategy between summer and winter. The control system uses different control algorithms for summer and winter. In addition, a similar approach was used to control a mixed-mode building in Norway (Tjelflaat, P., 2002). The school uses a mixed-mode ventilation system to maintain adequate IAQ and thermal comfort conditions in the classrooms. The ventilation is demand-controlled by a CO_2 sensor in each classroom. When the CO_2 levels exceed the threshold values the extract hatch opens. The opening of the hatch is adjusted based on the CO_2 levels. When the ventilation rate is low, a supply fan switches on. In both cases, higher priority was given to the IAQ due to the negative correlation between CO_2 levels and cognitive function of the students (Coley *et al.*, 2004). The studies reported (van Der Aa, 2002; Tjelflaat, P., 2002; Coley *et al.*, 2004) here due to their focus on schools are using IAQ as the main control parameter. In residential cases, it is more challenging to apply control algorithms using the IAQ as the main control parameter especially because there are no standards indicating the levels of indoor contamination in residential buildings.

Ezzeldin and Rees, (2013) presented in their research a control algorithm for a mixed-mode office in arid climates. A different approach was used in this study since the control algorithms that were responsive to the occupancy schedule. A *concurrent (see Figure 2-1)* ventilation strategy was used for the purpose of this research. Natural ventilation had the highest priority and when the outdoor temperature was above the set-point value, mechanical cooling was used to provide a thermally comfortable internal environment. The variation of the opening of the windows was used to modulate the ventilation rate and hence the IAQ. During the non-occupied periods, night cooling was used when the outdoor temperature was below the zone air temperature. Furthermore, the wind speed and the occurrence of rain considered as barriers to the natural ventilation strategies and hence the control of those parameters was crucial for the optimal control of the mixed-mode ventilation strategies. The control algorithm that was used although had as an objective to achieve acceptable levels of IAQ did not directly measure the ventilation rates or CO_2 levels and hence it is possible that during periods of time the ventilation rates to be below the desirable values which result to poor IAQ. Although this study was focused on non-domestic buildings, the main principles of controlling the building can be used in residential buildings as well as the main control parameter was the indoor temperature.

Most of the controls algorithms that can be found in the literature as well as the above mentioned, use CO_2 concentration and internal air temperature as the control parameters of the ventilation strategies. Ideally, different control strategies should be employed for summer and winter conditions, since mixed-mode ventilation is significantly influenced by the micro-climate. However, most of the examples in the literature do not distinguish different control strategies for different periods of the year for both hot and cold climates. Furthermore, in all the cases the air temperature set-points were predefined static values, and hence they failed to capture the adaptive possibilities that the occupants might have. It would be valuable to investigate a varying set-point temperature for different periods of the year in order to maximize the performance of each ventilation strategy, particularly in residential buildings as most literature examples have focused on commercial buildings.

The majority of the control algorithms reported in the literature are focusing on non-domestic buildings. The main limitation to "transfer" them into the residential sector are i) that occupants in non-domestic buildings have less "freedom" to control the environment and ii) the occupancy patterns are easier to predict. For those reasons, the use of static setpoints over a period are more common. However, in the residential sector occupants interact constantly with their environment and occupancy patterns differ almost every day. Hence a dynamic selection of setpoints and more advanced control algorithms are required. The implications of more sophisticated control algorithms in residential buildings should, therefore, be examined further.

2.5. Control of internal environment in mixed-mode buildings

Mixed-mode buildings allow occupants to actively adapt to their environment, for example by operating windows in order to meet their thermal requirements (Nicol and Humphreys, 2002). The adaptation that a mixed-mode building, sometimes, results in uncertainties in the energy performance calculations. Hoes *et al*., (2009) in their research showed that by neglecting occupants' behaviour at free-running buildings during the calculations of the energy and thermal performance of the building might be a big source of error at the final results. Following this, a study by Eguaras-Martínez *et al*., (2014) revealed that the deviation in the energy performance of a building by including and excluding the occupants could reach up to 30%. Another source of uncertainty, while simulating the energy consumption of a building is the lack of guidance on the modelling of occupants' behaviour. The control of mixed-mode buildings requires the use of either manual or automatic controls. Previous studies have highlighted schedules to operate windows mainly based on occupancy patterns of a building (Rajapaksha *et al.*, 2003; Gratia and De Herde, 2004; Koinakis, 2005), on the outside temperatures (Eftekhari and Marjanovic, 2003; Breesch *et al.*, 2005) and also based on the need to improve the IAQ of a building (Gratia and De Herde, 2003; Eicker *et al.*, 2006). These natural ventilation control strategies in conjunction with the use of mechanical cooling must be carefully designed to assure adequate IAQ and comfortable thermal conditions with the minimum energy consumption for cooling purposes.

2.5.1. Window operation and mixed-mode building energy performance

One of the main design characteristics that many have focused on, in the mixed-mode buildings, is the occupant behaviour and especially how the occupants operate the windows. The window operation could have a significant impact on the overall thermal performance of mixed-mode buildings. Commonly, the buildings' occupants are controlling the indoor environment and hence the opening/closing of the windows according to their personal thermal preferences (Brager *et al.*, 2004; Borgeson and Brager, 2008).

The frequency of window opening in mixed-mode buildings is lower compared to the naturally ventilated buildings (Rijal *et al.*, 2009). Study on occupant control at mixed-mode buildings for buildings in Europe and Pakistan revealed that regardless of the geographical region, the likelihood of opening the windows is higher compared to the use of ceiling fans. Additionally, the results indicated that for mixed-mode buildings the occupants rely on the window opening to maintain an acceptable level of indoor conditions when the outside air temperature is below 26.8°C and 24.9°C in Pakistan and Europe respectively. Furthermore, a longitudinal field study in an office building at Loughborough, UK revealed that the opening of a window is affected by the season, the gender of the occupants, the floor level and by the personal thermal preferences (Wei *et al.*, 2013). Also, this study showed that the end of the day window position has an impact on the energy and thermal performance of the building the following day. Hence, it is essential to effectively control the opening of the windows to optimize the performance of the building.

Wang and Chen, (2013) examined the effects of different windows' operation on the energy-saving potential for a mixed-mode office building. The building was using a Constant Air Volume (CAV) cooling system along with natural ventilation. For the purpose of this study a building with a floor area of 600 m^2 and a total window area of 18 m^2 was simulated using EnergyPlus for different climate zones in the USA (very hot & dry, very hot & humid, marine and warm & humid). The results estimated the savings in the electricity consumption could vary from 10 to 71.8%, based on the climate and the window opening. Although this study identified the optimal window operation for every climate zone, the selection of the operation was based on predifed position of the windows (0%, 25%, 50% and 100% opening factors). Hence, future studies should examine a graduate window opening. This could provide a better insight into the actual optimal control of the window opening. Furthermore, since the control of the openings was based on a pre-defined position, there was no feedback loop to adjust the opening length according to the indoor or outdoor conditions. This approach might have resulted in a waste of energy for some days because maybe the external air temperature was not that high. Hence, more sophisticated control of the openings is required to optimally select the opening area of the windows.

A similar study investigated the effect of the window opening on the energy performance of a mixed-mode (Wang and Greenberg, 2015). This study examined the impact of different window openings for medium-size office. EnergyPlus was selected as the simulation tool to run different ventilation scenarios and the results highlighted that the window operation has a significant effect on the overall energy performance and indoor environment of the office building. More specifically, the simulations predicted energy savings potential up to 47% during summertime under a variety of US climate conditions. The energy savings were achieved by utilizing the opening of the windows in conjunction with the ventilation strategies. However, in this study, as also in the study by Wang and Chen, (2013), an open loop control was used to modulate the window opening which has limitations with regards to the control trigger of the window operation. Although both studies simulated similar buildings in similar locations, the results varied. It is likely that the position of the windows, the different mechanical cooling systems in combination with the different building sizes of the buildings resulted in this discrepancy on the results.

Acknowledging the difficulty in predicting accurately the occupants’ behaviour (Ackerly and Brager, 2013), in regards to window operation, a different approach is needed in order to optimize the energy performance of mixed-mode buildings without compromising the IAQ. Many consider the use of automatically controlled windows as the most plausible solution to cope with the unpredicted natural of humans’ behaviour. Most recent studies have focused on implementing a Model Predictive Control (MPC) strategies to automated systems. An MPC is an algorithm that attempts to optimize the behaviour of a system by computing a sequence of future manipulated variable adjustments (Qin and Badgwell, 2003). Tanner and Henze, (2014) developed an innovative approach to optimize the building automatic controlled window opening in mixed-mode buildings by incorporating stochastic models of occupants’ behaviour developed by Page *et al*., (2008). The analysis revealed that the deterministic models for the occupants’ behaviour overpredict the energy-saving potential for mixed-mode buildings. Most of the studies that include MPC controls are focused on non-residential buildings since the occupancy is higher than the residential buildings and hence it is more difficult to satisfy each occupant without wasting energy. Furthermore, as proposed by Spindler and Norford, (2009a, 2009b) in their studies, a more complex approach to estimate the energy performance of a residential building is not feasible, since the occupants most of the times have direct control over the system.

It is therefore essential to develop more sophisticated control algorithms to control the window operation in mixed-mode residential buildings, which will take into account the outside conditions, the internal conditions and the window position in order to “decide” the optimum window position. Furthermore, previous studies have focused on commercial buildings and there are no studies investigating the impact of the window opening in residential mixed-mode buildings. Therefore, this research will shed light on the effective control of windows in mixed-mode residential buildings.

2.5.2. Air motion in mixed-mode buildings to improve thermal comfort

In mixed-mode buildings, occupants control the natural ventilation strategies either by window operation or by the use of fans. Several studies have shown that further energy saving could be achieved by introducing air motion in spaces in combination with adaptive thermal comfort criteria.

Regardless of the recommended formulas for the comfort temperature by the energy standards, research has shown that higher indoor airspeeds could have a positive effect on the thermal sensation of the occupants. For hot and dry climates in North India and Iraq, Nicol, (1974) showed that for moderate airspeeds above 0.25m/s, both the thermal discomfort and skin moisture could be reduced. This has an equivalent effect such as by reducing the air temperature by approximately 4°C. A field study in Pakistan (Nicol *et al.*, 1999) showed that the use of fan ceilings at dry conditions could allow the occupants to be thermally comfortable for indoor air temperatures almost 2°C higher compared to the suggested values by ASHRAE standard 55 for a wide variety for outdoor temperatures.

Similar results were obtained by a field study in Singapore examined the thermal sensation of 24 participants at the tropical weather conditions (Gong *et al.*, 2006). This study revealed that the participants preferred higher airspeeds, compared to values suggested by Nicol, (1974). Specifically, the occupants felt comfortable for airspeeds between 0.3-0.45m/s when the air temperature was 23°C, while for an ambient temperature of 26°C the participants favoured higher airspeed up to 0.9m/s. However, this airspeed range is higher than the proposed by ASHRAE standard 55 (2013). Furthermore, subjects voted for thermal sensation to be between "*neutral"* and "*slightly cool*". In line with the previous studies, an experimental study with 232 participants in Brazil highlighted that there is a relationship between indoor airspeed, thermal sensation and operative temperature (Cândido *et al.*, 2008). Particularly, this research highlighted that for operative temperature between 26.5-27.5°C and airspeed from 0.25-0.5m/s the majority of the participants were thermally comfortable, voting that they would not prefer any change at the indoor conditions. When the operative air temperature was 25.5°C and the airspeed was above 1.5m/s, 80% of the participants were thermally comfortable. All the studies indicated the positive impact of the air motion on the thermal sensation of the occupants in free-running buildings.

Rijal *et al*, (2009) carried out a study by using data from a European study Smart Controls and Thermal Comfort SCATs and from a Pakistan-trans study. Both the projects used mixed-mode buildings. The analysis of the European transverse data showed that the occupants in mixed-mode building preferred to use fans to cool the building instead of using mechanical cooling systems. Similar results observed by the Pakistan transverse study. The outcomes from both studies suggested that occupants prefer higher air movement if they feel slightly warm despite the type of building, mixed-mode or purely naturally ventilated. These results are in line with a previous study by Zhang *et al*., (2007) who found that people prefer to have air motion in their room if they feel warm. Another study focused on a mixed-mode office building in the warm and humid South East Queensland, Australia during the summer months (Healey, 2014). The participants had complete control of the internal environment (opening/closing windows, ceiling fans etc). The results of this experimental study revealed that most of the users chose to use natural ventilation instead of mechanical cooling. The occupants showed a cognitive tolerance to slightly warmer conditions since they preferred having a “connection” with the outdoor conditions. The use of fans helped to mitigate any discomfort conditions due to high internal air temperatures. In general, the authors concluded that the adaptive thermal comfort models described better the actual thermal sensation of the occupants. Another field study was carried out in naturally ventilated and mixed-mode buildings in Jaipur, India (Dhaka *et al.*, 2015). The study indicated that the neutral temperature for all the seasons was 27.2°C, while specifically for winter was 25.6°C and for summer period 29.4°C. Most of the occupants used fans and operable windows to compensate for thermal discomfort conditions. The studies revealed that people from warm climate conditions were more adaptive to climatic variations and generally feel thermally comfortable for higher temperatures compared to previous studies (de Dear and Brager, 1998).

As highlighted from these studies, the air motion could increase the energy saving potential of mixed-mode buildings. Hence it is essential during the design stage of mixed-mode building to incorporate the controlled use of fans to increase the thermal performance of a building with the minimum cooling energy consumption.

2.6. Cooling technologies for mixed-mode buildings

Buildings have cooling loads due to sensible and latent heat gains from inward flow of heat due to higher external air temperature, internal accumulation from electronic equipment & occupants and from the solar radiation (USAID, 2014). All these factors could compromise the thermal internal conditions of a building. However, thermal comfort is seen as one of the ultimate objectives in the design of a building. Most of the times the whole performance of a building is evaluated by the internal conditions that it can achieve. Satisfactory thermal comfort depends on the design and adequate control of the building and its systems (Kalua, 2016). For a mixed-mode building, it is even more important as it seeks to optimally integrate natural ventilation and mechanical cooling systems, to reduce the energy consumption without compromising the indoor air quality.

The control of the mechanical cooling systems used in mixed-mode buildings is an important factor to consider during the early stages of the building design. Mechanical cooling systems that are commonly used in mixed-mode buildings are: the central air conditioning (AC) systems that include Constant Air Volume (CAV) and Variable Air Volume (VAV) systems, air handling units (AHUs), radiant systems etc.; distributed air conditions systems that include Fan Coil Units (FCUs) and Heat Pump units; Local/Direct Expansion (DX) systems- Splits, window ACs, multi-splits, rooftop units; and Ventilation Systems-Evaporative cooling (single, two or three-stage).

2.6.1. Central air conditioning systems: Radiant cooling, Variable Air Volume (VAV) and Constant Air Volume (CAV)

One of the most popular mechanical systems for mixed-mode buildings is the hydronic radiant cooling via chilled slabs, walls or panels. These systems have proven to be favourable due to their energy performance but also due to occupants' satisfaction (Borgeson, 2009).

Ezzeldin *et al*., (2008) evaluate the performance of a mixed-mode office building in arid climates compared to an active VAV system and the EnergyPlus simulations predicted energy savings for the mixed-mode systems up to 60% compared to the conventional design where VAV was used to condition the space. Additionally, Wang and Chen, (2013) compared the cooling performance of a typical mixed-mode office building, utilising natural ventilation and active CAV cooling system, in five different climate zones in the USA against buildings with fully mechanical CAV cooling system. The authors used EnergyPlus for dynamic thermal modelling, predicting that the energy saving potential could vary by 5 to 75% for very hot and humid climate zone to marine climate respectively.

Similarly, Borgeson *et al*., (2009) used EnergyPlus to examine the performance of mixed-mode office buildings using radiant cooling systems. The predicted energy performance of the mixed-mode buildings was compared against purely mechanical ventilated buildings using VAV systems. The results from the simulations revealed that for all the 16 different climate zones examined, radiant cooling could achieve up to 65% energy savings compared to the buildings that used only VAV cooling systems. Furthermore, the analysis showed that in hot and dry climates, natural ventilation alone could not maintain a comfortable internal environment and hence the use of supplement mechanical cooling deemed essential.

2.6.2. Distributed air conditioning systems using fan coil units (FCUs) and heat pump units

Rowe, (2003) performed computer simulations and on-site monitoring at 25 mixed-mode offices in the University of Sydney, Australia to examine their energy consumption and internal air temperature. The offices used natural ventilation with a supplementary reverse cycle heating/cooling system having occupant-controlled fan coil unit in each room. The simulated results indicated an up to 75% reduction of the cooling energy demand relative to relying purely on mechanical means.

Ji *et al*., (2009) examined the energy savings potentials from a changeover mixed-mode office building in south China, using a ground source heat pump to cool the chilled ceilings against a building that relied purely on a ground source heat pump to cover the cooling demand. This resulted in up to 35% energy savings compared to a conventional mechanical cooling system. Ward *et al*., (2012) compared the energy consumption of a building with heat pump against a mixed-mode building. During this research, an office building with no thermal mass simulated, for 8 different climate zones in Australia. Three cases were compared, i) a heat pump was used to condition the indoor air, ii) a heat pump was used in combination with operable windows and ceiling fans with higher set-point temperature and iii) a purely naturally ventilated building. The findings suggested that for moderate climates the energy saving potential for a mixed-mode building could be up to 42% compared with the first case. In tropical climates, a mixed-mode ventilation strategy has a lower impact on the cooling demand, nevertheless, the simulations showed energy savings of 14%.

2.6.3. Local/Direct Expansion (DX) systems or Residential Air conditioners

HVAC systems can essentially be divided into heating only, cooling only, ventilation only and air conditions systems. In building applications, an HVAC system can be characterized by the location of its component, the cooling capacity, the number of zones that will be conditioned. Generally, in a building environment HVAC systems can be classified as central and local/Direct Expansion (DX) systems (figure 2-5).

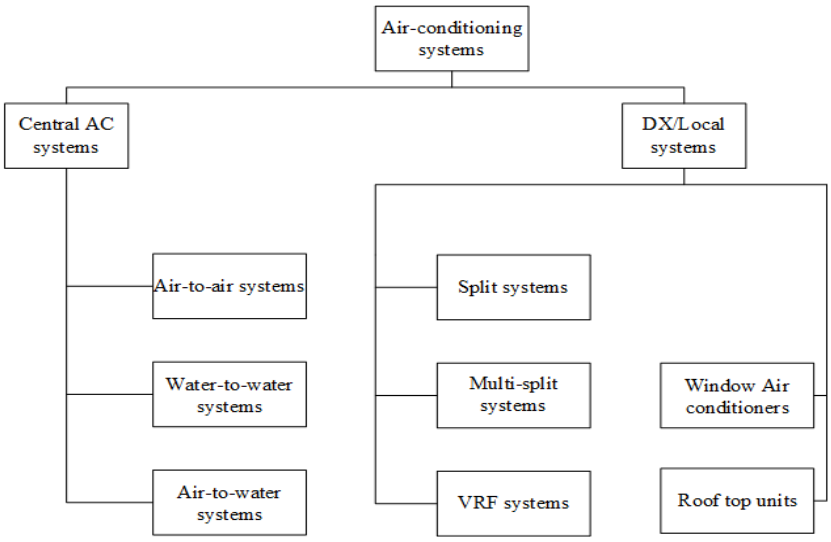


Figure 2-5: Classification of HVAC systems. (Excerpt from (USAID, 2014)*)*

Local systems or DX systems are used to cool or heat the air inside a room or zone through the refrigerant that flows directly from the compressor to the part of the DX system that is placed either inside or adjacent to the zone (USAID, 2014). The difference between the local and the central systems is that the latter cool or heat the fluid (water or air) which then cools or heat the zone. Small DX systems can be used mainly for one zone and the components of the systems are either within the zone or on the boundary of the zone and the external environment. Larger DX systems can be used for more than one zone. Due to their size and simplicity of the design, distribution systems such as ductwork and terminal units, are not necessary since only the refrigerant needs to be carried from the system to the conditioned space (USAID, 2014).

In DX unitary systems, evaporators are in direct contact with the supply air. Thus, the cooling coil which carries the refrigerant is considered also the evaporator. An expansion valve is used to circulate the refrigerant. The DX systems are categorized based on the location of the refrigeration cycle components. If the evaporator, condenser, compressor and expansion valve are packaged together, then the unit is called a *packaged DX system* (figure 2-6). If the components are split, then the system is called *DX split* or *split air-conditioners.* Controls for these systems are provided either on the system itself or through a remote thermostat control.

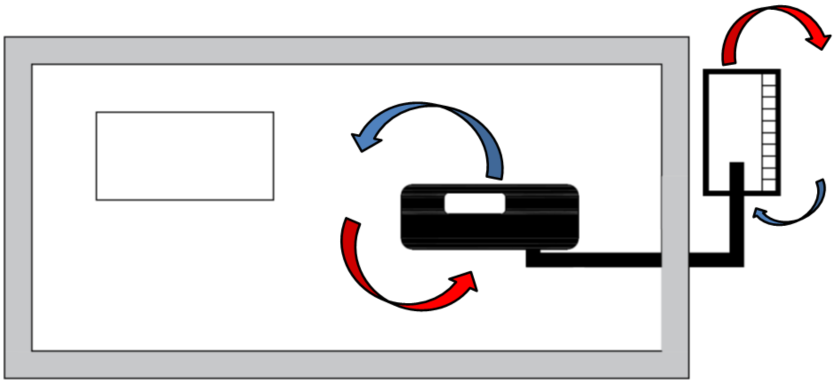


Figure 2-6: Packaged split air conditioning unit.

The term "*Residential Air conditioner*" can be applied in all the systems that are used for cooling a residential building. The main distinctions between the air conditions systems are i) whether they are ducted or not (described above), whereas the ducted AC units are mainly used to serve big buildings with space-specific cooling load requirements, ii) based on the fluid (air or water) that they use as the cooling medium, iii) whether they use vapour compression cycle or an alternative refrigeration cycle (absorption or evaporative cooling) and iv) based on the location of the condenser, whether the condenser is thermally coupled with the outside air, the ground or uses water as the external heat reservoir.

In many countries, the split AC units dominate the room AC market. They are among the most energy-efficient mechanical cooling systems for residential or small to medium commercial buildings. In India the split AC units accounted for almost 75% (Phadke, 2014) , in China and Japan the split AC units dominate the market with 99% and about 100% of the total market share respectively (Phadke, 2014) while in EU 85% (Phadke, 2014) of the residential houses use split AC units for cooling purposes. In India, in the year 2011-12, about 80% of the window units and 50% of the split units were sold to serve cooling demand for the residential sector while the rest of the units were used at small or medium commercial buildings (Antonijevic, 2014). Moreover, the market trends indicate that the share of the residential sector increases faster than the commercial (Chunekar *et al.*, 2011).

Several studies have examined the application of alternative *room air conditioners* units compared to the conventional split AC units. Yik *et al.*, (2001) examined the energy performance of water-cooled air conditioning systems (WAC) and compared the results against conventional AC units, in commercial buildings in Hong Kong. The analysis of the results showed that the annual electricity consumption of the buildings that used AC units was considerably higher compared to the buildings with WAC systems. However, due to the limited amount of freshwater in Hong Kong, the study proposed the implementation of district cooling systems using seawater which will generate better feasibility of the systems but it will increase the capital cost.

A packaged DX system that is used in many residential or office buildings is a *window air conditioner* which consists of a vapour compression refrigeration cycle components, fans and the control system (figure 2-7). Since it is a packaged unit, all the components are placed on the boundary between the zone and external environment, because they carry the external heat exchange unit, which is the condenser, and the interior which is the evaporator inside the package. Due to their size, they do not require any ducts to circulate the refrigerant and hence they are preferred in residential buildings. When a window air conditioner has air damper to control the air changes then it's called *packaged terminal air conditioners*.

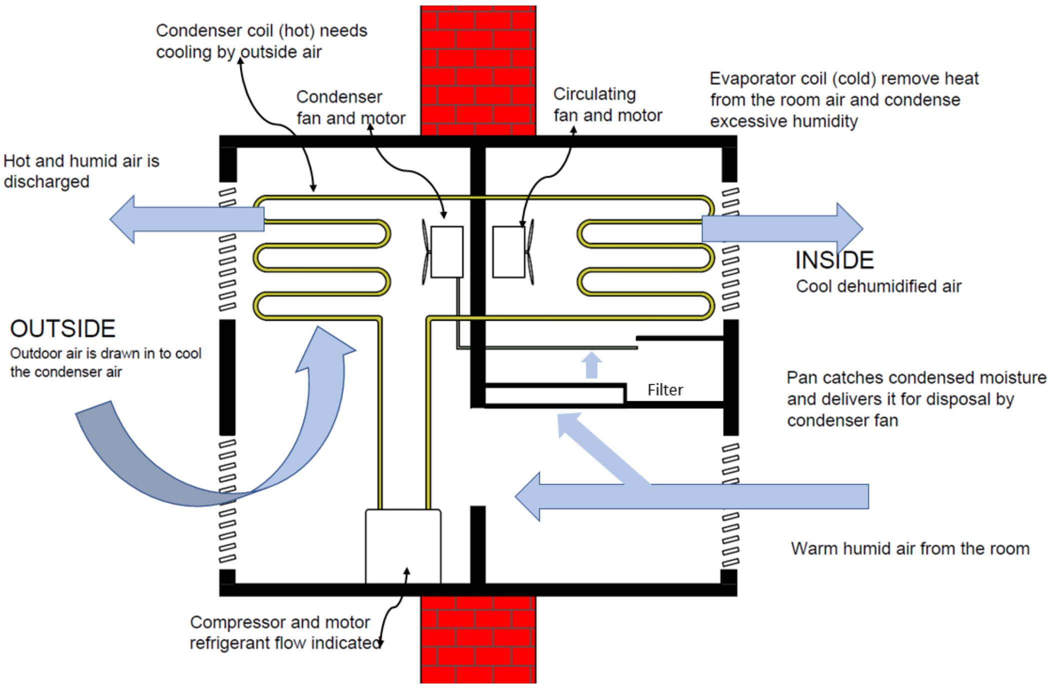


Figure 2-7: Wall-Window air conditioner (Excerpt from USAID, (2014))

Single-duct units are used mainly for room air conditioning purposes. The condensers emit the hot air directly to the exterior environment through a duct. The advantage of these systems is that they can move freely but they should be located close to an opening (window or door) so the duct can expel the hot air out. Ideally, a hole should be made at the envelope of the building in order to reduce the heat losses through the opening. However, since they are low-cost devices, they tend to be used by end-users with intermittent cooling space needs and hence most of the times there is no proper installation of these devices (Shah *et al.*, 2013).

The double-ducted units are very similar to the single-duct units. There are two types of these units. The first type is very similar to the single-ducted unit. The difference is that the condenser has holes which enable the condenser to draw air from the outside. The second type is for permanent use since the unit is placed directly on the wall (Shah *et al.*, 2013).

2.6.4. Evaporative cooling for mixed-mode ventilation systems

Evaporative cooling is an environmentally friendly and energy-efficient method for cooling buildings in hot and dry climates (Heidarinejad *et al.*, 2009). It has been used by people as the first method to improve indoor comfort and reduce the indoor air temperature (Mathews *et al.*, 1994). This system operates using induced processes of heat and mass transfer where the working fluids are either water or air (Camargo *et al.*, 2005). In the previous years, evaporative cooling has been renounced due to the commercial boom of AC units in both residential and commercial buildings. Nowadays, evaporative cooling has regained the attention due to the pending energy shortage in many developing and developed countries and due to the environmental challenges, the whole world is facing. Evaporative cooling has many advantages since it has a very low energy consumption compared to other mechanical cooling system and due to the use of water instead of other environmental harmful refrigerants.

There are two principal methods that are often used for evaporating cooling purposes in buildings: i) *Direct Evaporative Cooling (DEC)* (figure 2-8A) and ii) *Indirect Evaporative Cooling (IEC)* (figure 2-8B). DEC has been used for many years, generally is the simplest system among the two and hence it has extensively been used. The DEC consists of a fan that draws hot outside air, passes it through a porous wetted medium and injects it into the building (Erell, 2007). When no energy is added or extracted from the system the process can be characterized as adiabatic. During this process, the water absorbs the heat from the warm outside air during the evaporation process at the porous medium and sensible heat converts to latent, at constant wet-bulb temperatures (WBT), increasing simultaneously the moisture content of the air and thus the outlet air temperature is lower compared to the inlet air temperature at the DEC. (Santamouris and Asimakopoulos, 1996; Erell, 2007).

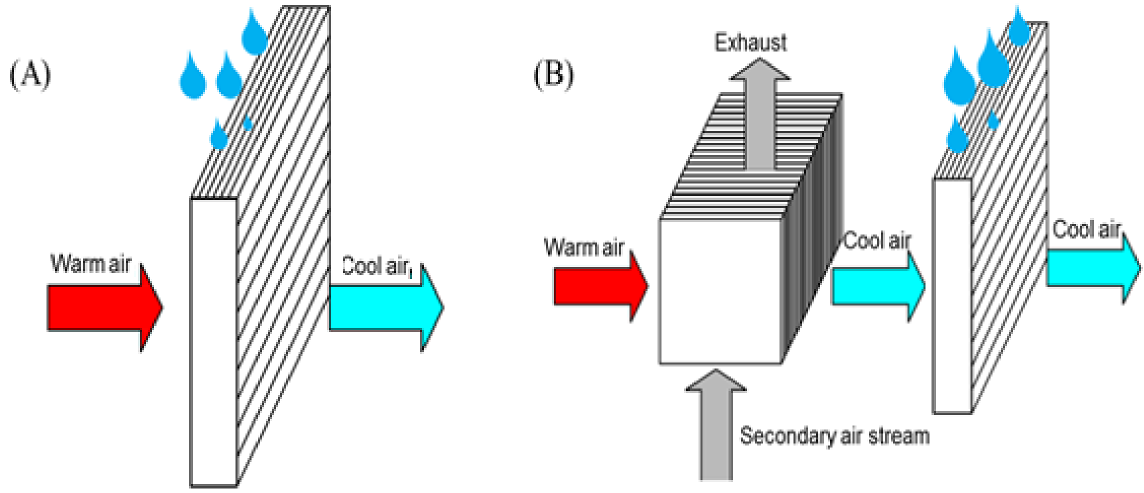


Figure 2-8: Direct Evaporative Cooling (A); Indirect Evaporative Cooling (B)

The efficiency of this process depends on various parameters such as the temperature moisture content of the air, the velocity of ventilated air, the surface temperature of the porous material and the design characteristics of the system (Santamouris and Asimakopoulos, 1996). As far as it is known, DEC systems are the oldest and the most widespread cooling systems for both residential and commercial buildings (Heidarinejad *et al.*, 2009). One of the disadvantages of this technology is that the contact of the air with the water results to vaporised water and this might cause discomfort conditions to the occupants of a building. Furthermore, DEC systems are not suitable for severe climatic conditions or in humid weather conditions and hence alternative evaporative cooling technologies have been developed (Mehere *et al.*, 2014).

IEC can be classified into two different categories based on the heat and mass transfer occurring in the heat exchanger (Zhao *et al.*, 2008):

I. *Dry Surface ICE*: In this type, a secondary fluid (air) is cooled by direct evaporative cooling prior to entering the indirect heat exchanger. Then, the secondary fluid cools the primary air in a conventional air to the air heat exchanger (figure 2-9A).
II. *Wet Surface IEC*: In this type, non-adiabatic evaporation occurs at a wet surface heat exchanger. Two different streams of air are used, the primary air that is cooled in dry passages which are completely separated from the wet passages where the secondary air and water flow. Evaporation occurs in wet passages where the heat is removed from the primary air thought impermeable separating wall and evaporates water into

the secondary air. Heat removal in dry passages and heat and mass transfer in wet passages occur at the same time and is practically inseparable (figure 2-9B).

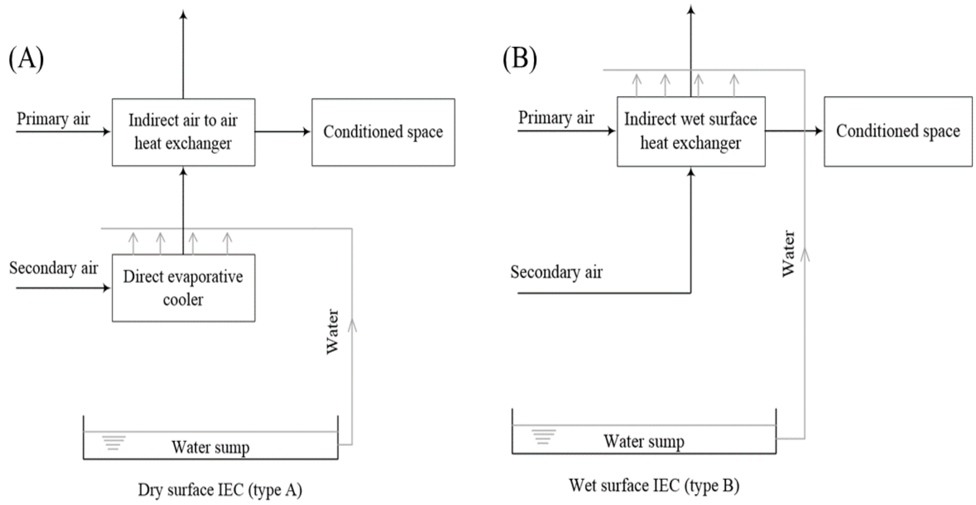


Figure 2-9: Dry surface IEC (A); Wet surface IEC (B).

For both approaches, the vaporised water is taken away by the secondary working medium. Using this configuration, the primary air is cooled without increasing its moisture content of the spaces compared to the DEC which always increasing the moisture content of the air. Based on this distinguished advantage over DEC, IEC has better potential to become an alternative cooling system due to better hygiene air quality which is preferable in all the occupied spaces.

Due to high cooling energy demand in countries with hot climates, such as India, several studies have examined the energy-saving potential of IEC systems. Jain and Hindoliya, (2013) investigated the performance of IEC in three different climatic zones in India, namely composite, hot and dry and moderate. These three different climate conditions were selected as it is more likely for an IEC to be suitable. The monthly average environmental data for Bangalore (moderate), Jodhpur (hot and dry) and Delhi (Composite) were used over a period of May. The results of the analysis predicted that for Jodhpur and Delhi the IEC systems can maintain comfortable internal conditions based on the psychometric chart whilst for moderate climates, Bangalore, the analysis showed that IEC cannot provide the comfortable internal environment. Furthermore, the results showed that the energy-saving potentials were 58% and 56% for Delhi and Jodhpur respectively compared to a conventional AC.

However, IEC technology has a few disadvantages as well. The biggest challenge is the IEC systems are dependent on the conditions of the ambient air. The driving force for both IEC and DEC is the temperature difference between the dry-bulb and wet-bulb temperature, hence for humid or mild climates, this difference is small which results in a limited cooling capacity.

Depending on the climate conditions of a region, it is a common practice to combine both DEC and IEC to increase the cooling capacity of a system (Mazzei and Palombo, 1999). A two-stage indirect/direct evaporative cooling system combines a first indirect stage which is added to a second direct stage (see figure 2-10). This configuration enables the air to cool more compared to a stand-alone DEC. This configuration is preferred in locations where the higher levels of humidity do not allow a stand-alone DEC to sufficiently cool the indoor air. An experimental study in Iran climatic conditions was performed using two air simulators to provide the outdoor conditions for different cities in Iran (Heidarinejad *et al.*, 2009). The results of this experimental studied showed that under various outside conditions, the effectiveness of an IEC system varied from 55-61% while for the two-stage IEC/DEC the effectiveness of the system varied from 108-111%. The energy savings for all the examined cases for the IEC/DEC systems were from 64-68% compared to a conventional mechanical vapour compression system. Other researchers have examined the characteristics of IEC/DEC systems which utilise a dry surface heat exchanger (Scofield & DesChamps, 1984). In the first stage, the ambient air is sprayed with water before it enters the counterflow plate heat exchanger. The added moisture in the secondary fluid results to lower output temperature of the primary medium. If further conditioning of the primary air is needed, then is achieved in a conventional cooling tower. This study showed that this configuration could achieve up to 30% energy savings over a year compared to a conventional mechanical cooling system. El-Dessouky *et al.*, (2004) performed a field study using a dry surface two stages evaporating cooling system in Kuwait. The weather conditions during the summer in Kuwait can reach up to 45°C in dry conditions. The authors examined operation parameters of the system such as water flow rate to the IEC unit or the thickness of the plates in the heat exchanger. The results of this study revealed that the effectiveness of the IEC/DEC system is at a range of 90-105% throughout the year.

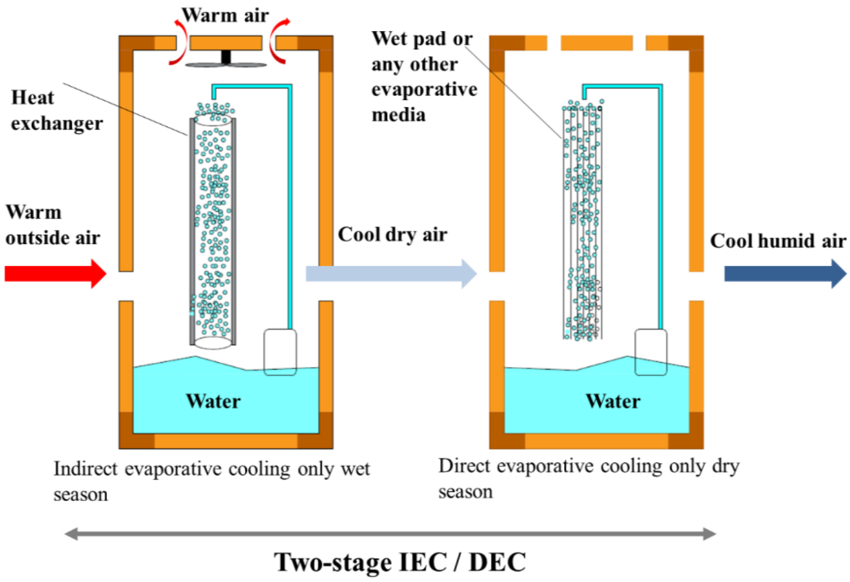


Figure 2-10: Two-stage evaporative cooling

2.7. Modelling of mixed-mode residential buildings

To meet the increasingly stringent building energy performance targets, more attention has been recently given on the operational optimization of buildings (Wetter, 2009). This requires taking into account at the design stage all the system-level interaction between the building elements, the mechanical systems and their controls. For this detail system-level analysis, a multi-physics simulation is required to optimize the operation of the building using coupled thermal, electrical and controls models. However, there is no definitive guide from any standard or professional communities on how to model a mixed-mode building (Gandhi *et al.*, 2015). The methods that have been used in the past from researchers very greatly. In general, three categories of modelling are used by researchers or practitioners i) stand-alone airflow network tools, ii) coupled airflow network models with dynamic thermal modelling tools and iii) Computational Fluid Dynamics (CFD) tools (Gandhi *et al.*, 2015).

2.7.1. The principles of dynamic thermal modelling

A dynamic thermal modelling (DTM) simulation tool solves the energy balance equation for room air and the surface heat transfer to calculate the energy requirements for a space. The energy balance equation for room air is:

$$\sum_{i=1}^{N} q_{i,c}A_i + Q_{internal} - Q_{heat\_extraction} = \frac{\rho V_{room} C_p \Delta T}{\Delta t} \qquad \text{Eq. 2-6}$$

Where $\sum_{i=1}^{N} q_{i,c}A_i$ is the convective heat transfer from enclosure surfaces to the room air, $q_{i,c}$ is the convective flux from the surface *i* [J/(kgK)], N is the number of enclosure surfaces, A_i is the area of the surface *i* [m²], $Q_{internal}$ is the internal heat gains from lighting [W], electrical appliances, occupants, infiltration etc, $Q_{heat\_extraction}$ is the heat extraction rate of the room [W], $\frac{\rho V_{room} C_p \Delta T}{\Delta t}$ is the energy change due to ventilation in the room, where ρ is the density of the air [kg/m³], V_{room} is the volume of the room [m³], C_p is the specific heat of the air [J/(kgK)], ΔT is the temperature difference between the internal and external air [K] and Δt is the sampling time interval which is usually assumed to be 1h (Zhai *et al.*, 2002).

The convective heat flux from a wall is calculated based on the energy balance for each surface. The energy balance equation for a surface, wall or window, is:

$$q_i + q_{ir} = \sum_{i=1}^{N} q_{ik} + q_{i,c} \qquad \text{Eq. 2-7}$$

Where, q_i is the conductive heat flux on the surface *i*, q_{ir} is the radiative heat flux from solar and internal sources and q_{ik} is the radiative heat flux from the surface *i* to *k* (Zhai *et al.*, 2002).

The radiative heat flux is determined by the formula:

$$q_{ik} = h_{ik,r}(T_i - T_k) \qquad \text{Eq. 2-8}$$

Where $h_{ik,r}$ is the linearized radiative heat coefficient between the surfaces *i* and *k* [W/(m²K)]
whilst T_i [K] and T_k [K] are the air temperature of the interior surface *i* and *k* respectively (Zhai *et al.*, 2002).

Finally, the convective heat transfer can be determined by:

$$q_{i,c} = h_c(T_i - T_{room}) \qquad \text{Eq. 2-9}$$

Where h_c is the convective heat transfer coefficient and T_{room} is the room air temperature.

Most of the DTM simulation tools assume a uniform room air temperature and hence the interior surface temperature can be derived by Eq. 2-7. Space cooling or heating is calculated by Eq. 2-6. As described from the above equations, the interior convective heat transfer from enclosures is the connected with the room air temperature and the surface energy balance equations. Hence the accuracy of the calculated energy for the room will be affected by the accuracy of the convective heat transfer coefficient.

2.7.2. The principles of CFD modelling

CFD software applies numerical techniques to solve the Navier-Stokes equation for fluid flow. It solves the conservation equation of mass for the contaminant species and the conservation equation of energy. All the governing conservation equations can be expressed as below:

$$\frac{\partial \Phi}{\partial \mathrm{t}} + (\mathrm{V} \cdot \nabla)\Phi - \Gamma_{\Phi}\nabla^2\Phi = \mathrm{S}_{\Phi} \qquad \text{Eq. 2-10}$$

Where Φ is the V_j for the air velocity component in the j direction, I for mass continuity, T for temperature, C for different gas contaminants, *t* for the time, *V* is the velocity vector, Γ_{Φ} is the diffusion coefficient and S_{Φ} is the source term. Φ could also be used as a turbulence parameter. *C* can stand for water vapour and various gaseous contaminants. For the buoyancy flows, the Boussinesq approximation is usually employed which neglects the effect of pressure changes on density. The buoyancy-driven force is treated as a source term in the momentum equations. However, due to the turbulence nature of the airflow in most cases, a turbulence model should be applied to solve the equations for the flow. As described in equation 2-10, due to the non-linear nature of the Navier-Stokes equation, it is impossible to achieve analytical solutions for room airflow. Therefore, the CFD software applies the finite-volume method to discretise the equations. To achieve converged solutions, a number of iterations are needed (Zhai *et al.*, 2002).

2.7.3. The principles of equation-based modelling

To design and operate energy-efficient buildings, it is important to accurately incorporate the dynamic performance of the system over a wide range of time scales and operation conditions (Wetter, 2009). At the time scale of hours, the system dynamics determine how much energy can be stored or to what extent natural ventilation can be used. At the time scale of minutes or seconds, the system dynamics determine the dynamic behaviour of equipment that might have a negative impact on the energy performance of a building. For instance, at an evaporative cooling coil, if the compressor is switched off then the condensate might evaporate into the supply air which will increase the humidity levels inside the space (Henderson and Rengarajan, 1996). Therefore, to design control algorithms that exploit system's behaviour to increase the system-level efficiency, the simulation tools need to incorporate the changes in efficiency at various steady-state and dynamic operating conditions (Wetter, 2009).

However, most of the DTM or CFD simulation tools do not perform control evaluation for the feedback control of building systems, such as the ventilation systems or HVAC (Zuo *et al*., 2016). The majority of DTM simulation tools, such as EnergyPlus are designed for whole-building energy simulations over a long period of time and often perform annual simulations. To keep the simulations within a computationally affordable range, these tools very often used idealised controllers directly in a component model and there is no local feedback control loop. Hence, these simulation tools cannot be used to design or evaluate a ventilation system with local feedback control. Due to the increasing interest over part-load performance and system controls especially for low energy buildings, energy simulation tools need to account for the non-linear dynamic behaviour of the energy systems and include them at the energy performance analysis (Wetter, 2009). To implement the control algorithms in energy simulations, an equation-based object-oriented modelling language, Modelica, is often used. Modelica is an object-oriented language for system modelling and simulation developed by Fritzson & Engelson (1998). Models are described by differential, algebraic and discrete equations (Wetter, 2009). It uses a standardized interface to represent the objects using icons. The mathematical relationships between the variables are encapsulated and in combination with the graphical interface, make the models reusable.

A modelling suggestion in Modelica is that each component should represent a physical device with physical interface ports (Wetter, 2009). Models in equation-based languages, like Modelica, do not require specifying the sequence of computer assignments needed to simulate a model. Therefore, the user can specify the mathematical formulas, package them in icons and store them in a hierarchical library (Wetter *et al.*, 2016). This improves the management of complex and large systems easier and promotes the reusability of the models. It could also help in the debug process of a model since it allows testing subsystem models every time which in general are easier to handle and identify potential errors (Wetter, 2009). A simulation environment is responsible to analyse the formulas, optimizes them using computer algebra, translate them in executable code and links them with numerical solver (Wetter *et al.*, 2016). The translation from formulas to executable code requires the analysis of all the equations to detect for instance all the algebraic loops and convert the equations to a form that can be solved efficiently using Block Lower Triangularization and Tearing (Cellier and Kofman, 2006).

2.7.4. Coupling approaches

Energy performance simulation tools are not designed to examine the performance of the controls of a cooling system or HVAC unit, to perform detailed daylight analysis or detailed indoor airflow distribution of the air. There are various software to address these tasks. Hence, to extend the capability of the DTM simulation tools they have linked through co-simulations to a variety of software such as CFD, multi-zone airflow network, Dymola (Nouidui *et al.*, 2014).

A co-simulation, often, refers to a technique where individual component models described by differential-algebraic or discrete equations could be simulated by different simulation programs which run simultaneously and exchange data that depend on state-variables during run-time (Nouidui *et al.*, 2014). A co-simulation often consists of:

1. Domain solvers. It should be stated clearly which part of the code calculates with terms in the overall solution scheme.
2. Geometry and grid modeller.
3. Master programme or software environment for co-simulation.

Figure 2-11 describes the coupled simulation approaches that have been used in the past. In figure 2-11A the coupled simulation is defined as simulations where "one simulation tool controls the simulation and calls the other software when necessary". A more generalized view on couple simulations is where each simulation tool has its own geometry modeller and a master program coordinates the simulation process.

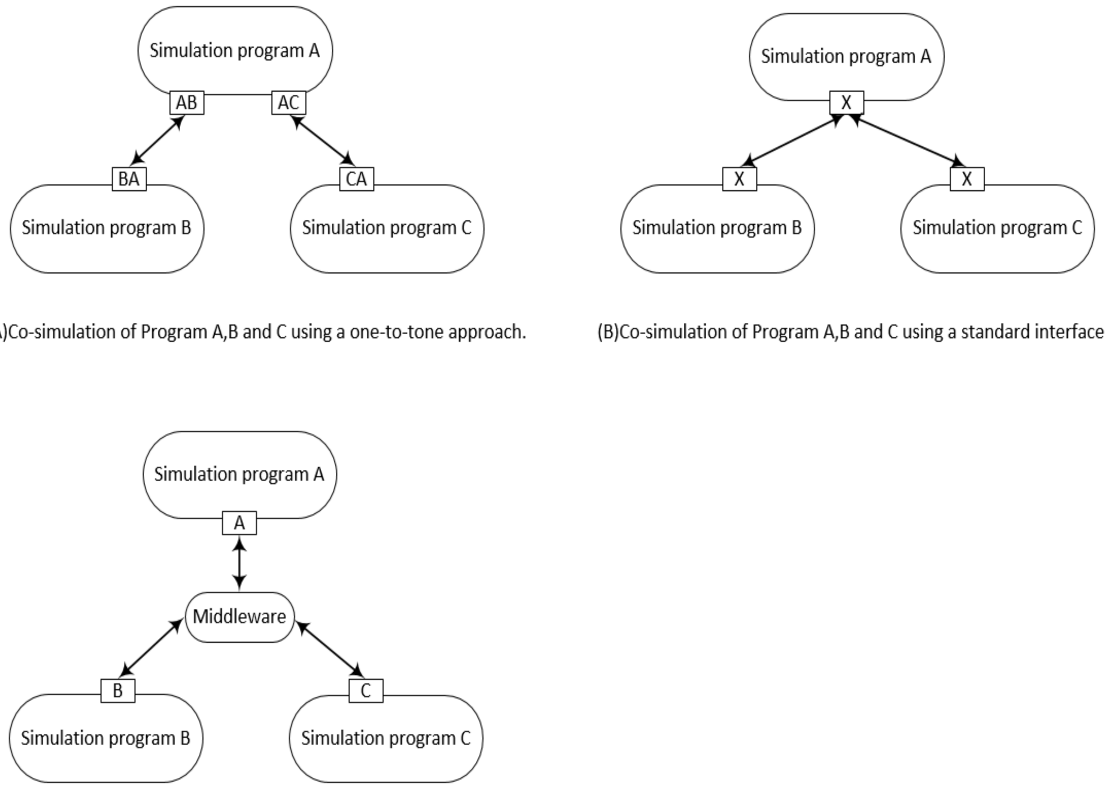
Based on these two different interactions between the two software, a coupled simulation could be characterized as (Djnaedy *et al.*, 2003):

- *Internal coupling*: the two software are specifically edited to work together. Usually, this approach requires code modification at one or both software.
- *External coupling*: No coding is needed. Usually, a master program is used for coupling the software.

In 2010, a standardisation effort initiated focusing on the facilitation of co-simulation, integration of models into different simulation environments and execution of models on embedded systems, see figure 2-11B. The Information Technology for European Advancement (ITAE2) project MODELISAR released the first specification of Functional Mock-up Interface (FMI) for model exchange. FMI standardizes the interface of models that include differential, algebraic and discrete equations (MODELISAR, 2017), is tool-independent and support both model exchange or co-simulation of dynamic models using XML files, C-header files, C-code in source or binary files (Nouidui *et al.*, 2014). A simulation tool which implements the FMI standard is called Functional Mock-up Unit (FMU). The FMI functions are called by a simulator to create one or more instances of the FMU (Nouidui *et al.*, 2014). Usually, the functions that call the FMU run together with other models. An FMU requires either a model exchange or be self-integrating for co-simulation (Nouidui *et al.*, 2014).

A new open-source software environment, the Building Controls Virtual Test Bed (BCVTB), developed by Lawrence Berkeley National Laboratory (Wetter, 2011), uses a graphical interface that allows the coupling of different simulation tools for data exchange. The BCVTB provides a great feasibility, since it gives the option to the users to use the simulation software that are most suited for building performance simulations, control of the systems or simulate airflow in the building and it constantly connects the software together, as it exchanges data with all the simulation tools, see figure 2-11C. A common application of the co-simulation environment is the performance evaluation of integrated building energy and control systems. Common applications in energy systems are the extension of the simulation capabilities of building simulation tools, such as EnergyPlus through the use of more modern languages such as Modelica (Fritzson and Engelson, 1998).

The most important distinction between the BCVTB and the coupling approaches in figure 2-11A is that BCVTB uses a modular middleware to couple the simulations programs whilst the other approaches couple the two simulators directly. The middleware allows the users to couple the energy performance simulators and control interfaces graphically.



(A)Co-simulation of Program A,B and C using a one-to-tone approach.

(B)Co-simulation of Program A,B and C using a standard interface approach.

(C)Co-simulation of Program A,B and C using a middleware approach.

Figure 2-11: Different coupling techniques for co-simulation (Excerpt from Nouidui et al., (2014))

2.7.5. Applications of co-simulation approaches for mixed-mode buildings

Nowadays, most of the dynamic thermal modelling tools such as EnergyPlus are embedded with Airflow Network object mainly to evaluate natural ventilation strategies. These models assume that there is a uniform distribution of the air temperature across the room and they do not take into account the momentum effect (Wang and Chen, 2009). To the contrary, CFD software predicts the airflow distribution for internal or external environments, and that is the reason that over the last years, CFD simulations have been commonly used to simulate natural ventilation strategies in buildings (Chen *et al.*, 2007). Over the last years, the development of computing power in combination with a user-friendly interface in most of the commercially available CFD simulations tools have made CFD tools more popular across the building simulation community (Tong *et al.*, 2012; Tong and Zhang, 2015). However, CFD simulations cannot predict the thermal performance or the energy consumption of a building. Hence, both simulation approaches should be combined to ensure the selected design strategy will result in the minimum energy consumption without compromising the IAQ of the building.

EnergyPlus is a widely used energy simulation tool, with a built-in Airflow Network to simulate buoyancy and forced airflows between different zones in a building. The Airflow Network consists of nodes which are linked to airflow components such as doors or windows. The AirFlow Network solves the mass conservation equation in order to calculate the airflow rate and the pressure at each airflow component (DOE, 2018). The use of built-in AirFlow Networks is suitable only for uniform air distributions (Wang and Chen, 2007). For more complex airflows in a space, the AirFlow Network cannot generate accurate results. Mora, *et al.*, (2003) compared the predicted velocities profiles for single-zone building using both AirFlow Network and CFD simulations. Afterwards, they compared the simulated results against experimental data for the same case. The analysis of the simulated results revealed that even with a coarse grid resolution, CFD can generate more accurate results for the velocity profile compared to AirFlow Network. However, the use of either natural or mechanical ventilation systems in buildings with more than one room usually results in non-uniform air distribution (Zuo *et al.*, 2016). To address this limitation, over the last years many researchers have coupled simulations between dynamic thermal modelling software and CFD. With this approach, the energy performance of the space can be predicted by the DTM whilst CFD can predict the airflow distribution or the stratification of the temperature.

Predictions for natural ventilation could be challenging due to the complex airflow patterns that are created in large scale naturally ventilated spaces. In the past years, researchers have coupled multi-zone models with CFD to improve the design methods and the accuracy of the simulations results (Tan and Glicksman, 2005). The authors coupled the CFD with a multi-zone model and the results showed that the CFD could predict the detailed airflow in the space of interest, while the rest of the zones would be simulated by the multi-zone model with relatively high accuracy. Wang and Chen, (2009) internally coupled a multi-zone network model with CFD, which was used for most of the zones in the building, while the CFD was used where the multi-zone assumptions were not accurate. The authors used three different coupling methods, where the upwind and downwind total pressure and flow rate were exchanged between the two software. Results showed that each coupling technique resulted to different solutions and the approach where the two software exchanged pressure boundaries conditions performed the best compared to the results from a numerical experiment.

The coupling methods described so far, have effectively improved the predicted accuracy of the results; however, the majority of the methods in these studies have increased the complexity of the models. Although researchers are willing to improve the complexity of their models to achieve higher accuracy of the results, it might not be feasible for the designers and engineers to handle the increased complexity of the models (Chen, 2009). A recent technical report pointed out that the computational time in combination with code modification and advanced user-experience has limited the applicability of coupling methods in the built environment (Gandhi *et al.*, 2014).

A recent study from Malkawi *et al.*, (2016) proposed a less computationally intensive approach to couple DTM and CFD to predict the performance of mixed-mode office building in the USA. The authors performed CFD simulations to identify the outdoor threshold temperatures when the natural ventilation alone is adequate for thermal comfort. Energy performance simulations were performed for a whole year to calculate the estimated cooling energy consumption of the building without natural ventilation strategies. The cooling energy consumption during natural ventilation hours was then subtracted from the total estimated energy consumption and the remaining energy consumption, was the energy needed for cooling during the mixed-mode operation. The authors concluded that using this approach, the level of complexity remains low and hence it is more likely for engineers or designers to adopt this methodology during the early-stage design of a project.

A previous study has attempted to couple an equation-based object-oriented with DTM simulation tool to investigate the control sequence of a VAV system (Wetter, 2011). For this research, the Modelica language was used to simulate the HVAC system and its supervisory control and EnergyPlus to simulate the building envelope. The coupling of the two software was implemented in BCVTB. The use of co-simulation allowed to take advantage of Modelica flexible modelling capabilities and its ability to simulate in detail local loop control and supervisory control loops, the dynamics of HVAC, pressure-dependent flow distribution which is not feasible to simulate in a DTM tool. Furthermore, in EnergyPlus the heat transfer, temperature and humidity concentration of the thermal zone were simulated. The two models exchanged data every 1 minute of simulation time.

In addition, BCVTB has been used to couple EnergyPlus and Matlab/Simulink to test the control of a naturally ventilated building (Wetter, 2011). The building envelope including the natural ventilation was simulated in EnergyPlus, whilst Matlab/Simulink was used for the control of the openings. Both software exchange data via the BCVBT every one-minute of simulation time. The internal temperatures from EnergyPlus were imported to the Matlab/Simulink model to estimate the position of the openings. The analysis of the results revealed that the opening and closing of the windows occur very often and this would be unacceptable for the occupants. Hence, more sophisticated control algorithms are needed. Moreover, BCVTB has been used to couple EnergyPlus and Matlab/Simulink to investigate the energy-saving potential for an office building in Lyon, France by using a novel personalized thermal sensation control algorithm which takes into account the actual thermal sensation vote of the occupants (Nouvel and Alessi, 2012). Every five-minutes of simulation time environmental parameters were measured and the control algorithm, which has been developed in Matlab/Simulink, calculated the thermal sensation vote for each occupant. The analysis of the results indicated that the novel thermal comfort control implemented by the co-simulation could result to energy savings up to 57% in the summertime compared to a purely mechanical air-conditioned case and 22% during the winter period.

Nouidui *et al.*, (2014) presented the development and implementation of an FMU for co-simulation import interface in EnergyPlus. The use of the FMU provides the flexibility to DTM simulation tools, like EnergyPlus, to perform co-simulation with a variety of different simulation tools that are equipped with FMUs. An HVAC unit and the controls were simulated in Modelica. The system and control algorithms were exported as an FMU and then imported in the EnergyPlus model, for the modelling of the building envelope. EnergyPlus was the master simulator since it simulated the FMUs, controlled the simulation time and exchanged data between the different simulation tools. Specifically, the authors simulated two different cases, one where an HVAC system was simulated in Modelica, exported in FMU and then coupled with EnergyPlus, and a second case where a shading controller was developed in Modelica and used via FMU to EnergyPlus. Recently, the use of FMUs as a simulation technique to improve the performance of existing dynamic thermal modelling tools is increasing. Angelopoulos *et al.*, (2018) performed co-simulations using FMU to predict the energy savings that could be achieved by using dynamic cooling setpoints in India. The authors used EnergyPlus to develop the thermal model, and Dymola to develop the proposed control algorithms written in the Modelica language. The use of the FMU improved the performance of the control algorithms as it allowed the development of customized control algorithms with dynamic cooling setpoints. The use of the dynamic cooling setpoints resulted in 40% reduction in the cooling demand. Another study by Borkowski. *et al*., (2018) performed co-simulations to improve the performance of the dynamic thermal modelling tools when adaptive building skins are modelled. The developed control strategies were modelled in Dymola and the analysis showed that the developed modelling framework predicted building energy demand with higher accuracy compared to cases where only dynamic thermal modelling tools were used. Less. *et al*.,(2019) used FMU to perform co-simulations to assess the energy-saving potential by implementing smart control algorithms for ventilation systems in the USA. The co-simulations were performed using EnergyPlus and CONTAM. CONTAM was used to enable the more accurate prediction of the contaminants, that were used to develop control algorithms based on the IAQ of the space. The analysis showed energy savings up to 45% when the smart controls were applied. The concept of co-simulations using FMUs has been used also in optimization studies. A recent study by Arendt. *et al*., (2019) presented co-simulations using EnergyPlus and Dymola to develop a framework that can be used for multi-objective optimization studies. This study improved the existing frameworks since the adoption of the FMU provided the flexibility to the controller that can be used by a variety of simulation software. Angelopoulos *et al*.,(2019) used an FMU to develop control algorithms, in Dymola, for evaporative coolers and to improve the very simple control approach used by the majority of the simulations tools. The resulted showed an improvement of about 35% of the performance of the direct evaporative coolers when co-simulations are used. As these studies suggested, the use of co-simulations, can provide flexibility to develop constumized control strategies and hence improve significantly the predictions for energy savings.

As mentioned in the previous subsections, there is a need for sophisticated control algorithms in mixed-mode buildings in order to utilize the use of natural ventilation and minimize the use of mechanical cooling. Recent studies have highlighted the importance of this research to develop sophisticated control algorithms for mixed-mode buildings by coupling EnergyPlus and Dymola. The control algorithms will take advantage of Dymola’s flexible modelling capabilities and its ability to simulate in detail local loop control and supervisory loop control and incorporate them in EnergyPlus.

2.8. Summary

The study of the literature review suggests that with the emergence of climate change, the increasing demand for cooling, and the higher expectations for thermal comfort, there is an urgent need for energy-conscious design and control of the buildings. A variety of control approaches for natural and mixed-mode systems were reviewed in this chapter. It was pointed out the need for more sophisticated controls in mixed-mode buildings to reduce energy consumption without compromising the indoor conditions.

A big part of the literature focused on the non-domestic buildings, mainly because there is a lack of studies focusing on residential buildings. This justifies the importance of the current research work to shed light on the effective control of mixed-mode residential buildings.

This chapter highlighted the importance of using the adaptive theory when assessing comfort conditions for mixed-mode buildings. The different mixed-mode systems and the principles of the control approaches for the mixed-mode buildings were identified as well. A literature review on the performance of mixed-mode buildings showed their potential to deliver energy savings. However, there is a gap in the knowledge on how to control mixed-mode buildings effectively. The literature review presented in this chapter identified the control parameters that should be considered when designing control algorithms for mixed-mode buildings, which enable the design of the proposed control algorithms (Chapter 4) and evaluate their performance (Chapter 6,7). For the validity of the results, it was important to perform a validation analysis (Chapter 5) to provide evidence of the validity of the results. The following chapters address these issues and present the research to fill the gaps in knowledge reported.

Chapter 3: Methodology

3.1. Introduction

Research design is the logical process that needs to be enabled in order to span the gap between the initial questions, aim and objectives, to the final conclusions (Yin, 2016). This approach includes the philosophical background of the research, methodology and the methods that will be employed to answer the research questions (Scotland, 2012; Creswell, 2014). Based on the underlying research philosophy that depicts the adopted research methodology, there are three main approaches to research (Blumberg *et al.*, 2005; Creswell, 2009):

- qualitative research is used to understand the underlying reasons, opinions and motivations. The data are collected via unstructured or semi-structured interviews. The sample size is relatively small.
- quantitative research which is used to quantify the problem by generating numerical data that can be analysed using statistical approaches. Usually, it is used to generalize results from a relatively large population. The collection of the data is more structured compared to the qualitative
- mixed methods which adopt a research strategy that employs more than one type of research method. The methods could be a mix of qualitative and quantitative methods, a mix of quantitative methods or a mix of qualitative methods.

3.2. Overall Research approach

For this research, a quantitative research approach was adopted following the perspective of deductive reasoning (Blumberg *et al.*, 2005). The development of the knowledge followed a "top-down" approach from more general to more specific, from building physics to numerical methods for dynamic thermal modelling simulations and the evaluation of the performance of mixed-mode buildings. The aim and objectives of this research emerged from reviewing the existing literature and identifying the gaps. The produced quantitative data were used to produce quantifiable results and reach scientifically proven conclusions. More specifically, the research employed the action strategy, which is an iterative process that involves (Atweh *et al.*, 1998):

- Plan a change;
- Act;
- Observe the outcomes; and
- Reflect on those.

This process is reflected in figure 3-1. For example, in objective 3, 4 & 5 after the planning and acting of the methodology, the observation of the results will reflect the next step which is either going back to objective 3 to adjust the models or move forward to objective 6.

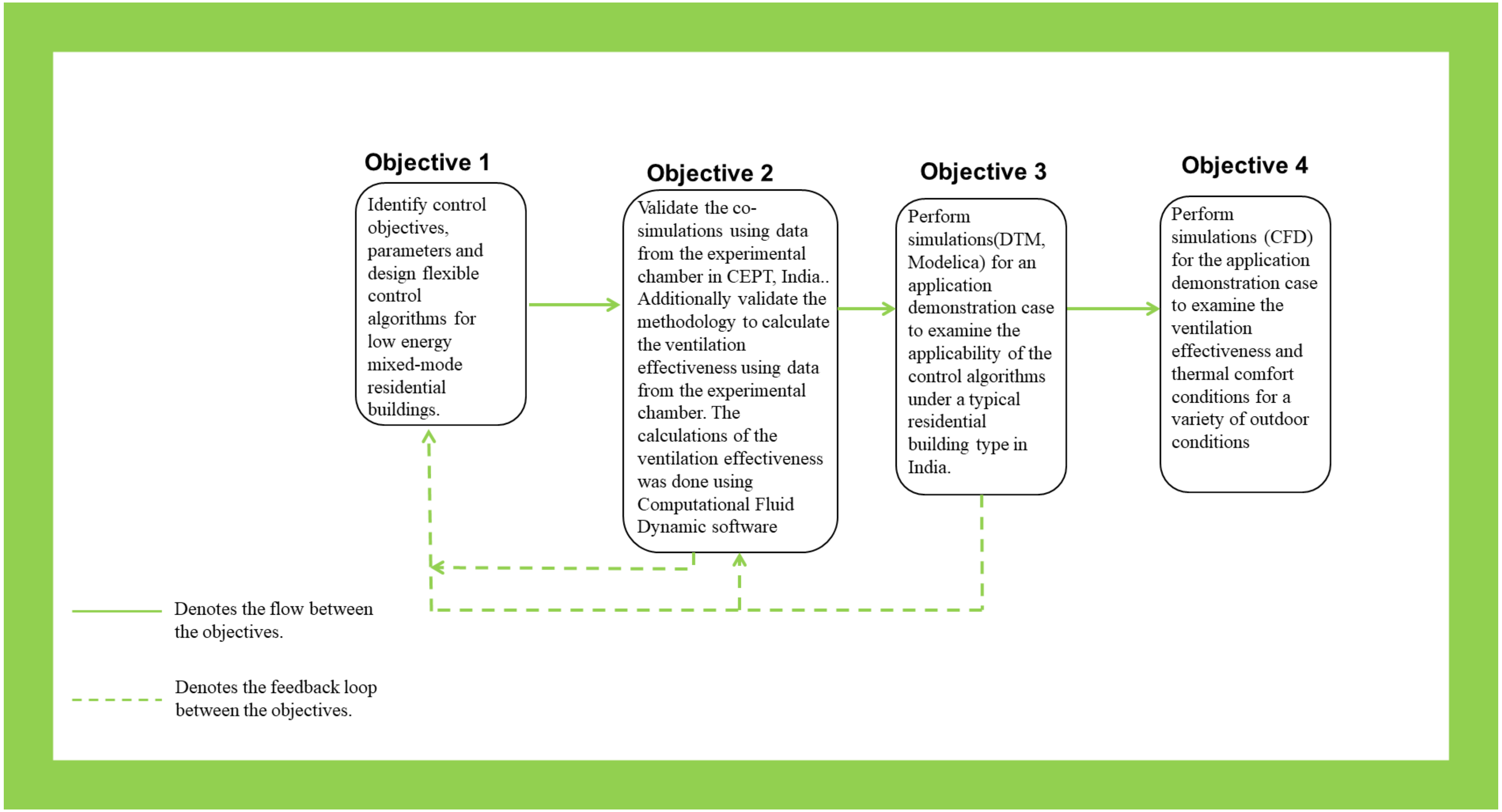


Figure 3-1: Flow chart showing objectives and their connectivity

3.3. Methodology overview

Within the first objective, previous research in mixed-mode buildings and more specifically on the control strategies of mixed-mode buildings, the indoor air quality & thermal conditions and their energy consumption were identified. The detailed literature review highlighted the importance of mixed-mode ventilation in buildings, and its' potential to reduce cooling demand especially in countries with predominately warm climates. A gap in the knowledge regarding the control of mixed-mode buildings and the absence of studies focusing on mixed-mode residential buildings was identified. A number of suitable simulation techniques and software for the performance evaluation of mixed-mode buildings were reviewed.

This research has identified the different control parameters and sensing points in buildings and low energy cooling systems to control a mixed-mode building. Based on the literature and on the models of adaptive theory for thermal comfort the most appropriate control parameters were selected (Objective 2). The design approach for mixed-mode buildings is common to both domestic or non-domestic buildings. The criteria to assess the results might be different. This research is focused on domestic buildings. The developed sophisticated control algorithms determine the most appropriate ventilation and cooling mode based on the prevailing indoor and outdoor conditions and based on the mode (natural ventilation or mixed-mode) the algorithms "decide" the operation of the building/system components for each operation mode. The objective of this task was to develop the flowcharts, (figure 3-1), that used to develop the control algorithms in the Modelica language (Objective 3). These flowcharts can be used as a generalised methodology to control and operate mixed-mode buildings. The control algorithms can be used either from architects and/or building services engineers to design the control algorithms for mixed-mode buildings in hot climates.

The next step was to perform the coupling of the two software, EnergyPlus and Dymola (Objective 4). The coupling of the two software was paramount importance since the literature review identified the limitations of the DTM tools to accommodate advanced control algorithms. Hence the coupling of the two simulation tools was a critical aspect of this research. To predict the performance of the control algorithms and their suitability in mixed-mode residential buildings, the two-aforementioned software were coupled using the "functional mock-up unit" as the common interface. New concepts must be tested in a controlled environment first, and only when their accuracy is proven they can be used to other cases. For this purpose, the validation of the predicted performance was conducted using a purposely built experimental chamber in CEPT, India (Objective 4). These measurements enabled the co-simulations and CFD simulations to be validated for both natural and mixed-mode operation and provided evidence for the validity of the developed flexible control algorithms. The experimental facility is able to simulate a wide range of weather conditions, and is equipped with a wide range of scientific measurement instruments, measurements of air temperature, relative humidity and CO_2 concentration can be recorded.

In order to evaluate the suitability of the developed control algorithms within a complex building scenario that would best represent real-life performance, a case-study apartment building was selected in India and the operation of the controls was computationally assessed. For this task, co-simulations were performed to examine the energy saving potential for a typical apartment block (Objective 5). In aiding further understanding of the airflow throughout the living space, CFD simulations were used (Objective 6). These highly detailed CFD simulations were performed to examine the IAQ, ventilation rates, ventilation effectiveness and thermal comfort conditions of the residences after implementing the developed controlled algorithms. PHOENICS CFD has been selected as the simulation software as it has been widely used by researchers to examine airflow in naturally ventilated spaces (Spentzou *et al.*, 2017). The use of a case-study building is accompanied by limitations. Although the selected geometry represents the majority of new-builts in India, the findings are limited in these specific archetypes. However, the focus of this current PhD was to develop the methodology of how to control mixed-mode buildings and they use of the case-study building was necessary to demonstrate the potential savings. Hence, the relative energy savings from the different scenarios can be used as a guide for different geometries however the absolute values of the energy demand will be, of course, different.

3.4. Control parameters for mixed-mode buildings

To maintain comfortable internal conditions and maximize the use of natural ventilation solutions, it is important to identify all parameters that affect the performance of the mixed-mode systems (objective 2). These parameters could then be controlled to ensure comfortable internal conditions. As identified in section 2.4, the most common parameters that are used to control a mixed-mode or a purely naturally ventilated system are:

- Internal operative or dry-bulb temperature (Ezzeldin and Rees, 2013; Sorgato *et al.*, 2016; Babich *et al.*, 2017);
- Occupancy status (Ezzeldin and Rees, 2013; Sorgato *et al.*, 2016); and
- Wind and rain (Ezzeldin and Rees, 2013)

Relative humidity is a factor linked with occupant comfort and thermal satisfaction; it should be therefore considered during system control in mixed-mode ventilation. However, no studies have detailed RH as a control parameter for natural or mixed-mode ventilation systems. As the analysis of the climate showed (section 6.2), humidity is an important parameter that should be considered in the control algorithms to eliminate discomfort internal conditions.

For this research, the control parameters that are used for the operation of the natural ventilation systems (windows and dampers), the ceiling fans and the mechanical systems are the following:

- Internal dry-bulb and operative air temperature;
- Indoor and outdoor relative and absolute humidity levels;
- Occupancy status, Daytime and nighttime; and
- Rain and wind.

3.4.1. Internal dry-bulb and operative air temperature

Although Ezzeldin and Rees, (2013) used dry-bulb temperature to control the window operation and the mechanical systems in their study, it was highlighted as a limitation of it and as it was proposed for future work to use operative temperature. Sorgato *et al*., (2016) used temperature as a control parameter to operate windows, without specifying whether drybulb or operative temperature was used. In practice, it is more common to use the drybulb temperature to control a system, especially due to most current sensors in the market measuring drybulb temperatures. However, standards refer to operative temperature as an index to assess thermal comfort. Effective control of a system, and hence the maximum energy saving potential is a complex task. When a system is automatically controlled then it is favourable to use the drybulb air temperature to maximize the saving potentials. When occupants have the systems’ control, they respond to both radiant and drybulb components of the temperature and hence it is most likely that they will use the operative temperature to control the system. Accordingly, for this research, both the air drybulb and operative air temperatures were used to control the systems.

3.4.2. Indoor and outdoor relative and absolute humidity levels

The relative humidity is a parameter that has been acknowledged in many research papers as an important control parameter for thermal comfort, however, there are not any reported control algorithms for mixed-mode building that incorporate the humidity as a control parameter. For this research, a *"detailed humidity control"* was developed and used in the control algorithms.

For the "detailed" approach, the actual humidity ratio for indoor and outdoor was calculated for every hour. The following steps were used to calculate the humidity ratio:

- **STEP–I** Barometric pressure:

$$Pb = 101.3 * [\frac{293 - 0.0065 * (El)}{293}]^{5.26} \qquad \text{Eq. 3-1}$$

Where *El* is the elevation of the city [m], Pb [1 atm]

- **STEP-II** Dew point temperature:

$$Td = \frac{[237.3 * C]}{[1 - C]} \qquad \text{Eq. 3-2}$$

Where C is an intermediate quantity:

$$C = \frac{[\ln\left(\frac{RH}{100}\right) + \{\frac{17.27 * T}{237.3 + T}\}]}{17.27} \qquad \text{Eq. 3-3}$$

Where T is dry bulb temperature [K] and RH is the relative humidity for each hour.

- **STEP–III** Vapor pressure:

$$Pv = 0.6108 * e^{(\frac{17.27*Td}{Td+237.3})}$$ Eq. 3-4

Pv is the vapor pressure in [mmHg]

- **STEP–IV** Humidity ratio:

Eq. 3-5

$$PW = 0.6219 * [\frac{Pv}{Pb - Pv}]$$

The window operation is based on the setpoint temperatures determined by adaptive models. Hence, for the upper and lower setpoint temperatures, the absolute humidity was calculated using the relative humidity setpoint values of 30 and 70%. The 30-70% acceptability range is often considered in the literature as an appropriate range to maintain acceptable indoor conditions (Berglund, 1998; Arens *et al.*, 1999; ASHRAE-Standard-55, 2013). In total, four different values of absolute humidity setpoints were calculated. Two values for the lower setpoint temperature and for RH=30% and RH=70% and two values for the higher setpoint temperature and for RH=30% and RH=70%. Afterwards, the minimum and the maximum values of these four values were calculated and were used as the absolute humidity limits to operate windows. Hence, if the absolute humidity of the outdoor air was in between these limits the windows would operate. This check ensures that the entering air is not too humid or too dry.

3.4.3. Occupancy status, Daytime and nighttime control parameters

The occupancy status of spaces has been used as a decision parameter to control the natural and mechanical systems. When occupants are present, control algorithms have been designed to maintain comfortable indoor conditions and minimize the use of mechanical systems whenever possible. When occupants are absent, the control algorithms are designed to provide the minimum ventilation rate, to restrict the opening of windows, and to turn off the mechanical systems.

Two different control strategies, one for daytime and the other for nighttime ventilation were designed. To define the day and nighttime, the global horizontal radiation[2] (GHR) was used. It was assumed that it is nighttime when GHR is zero. Night ventilation relies on the cooler external air during the night to remove the internal heat stored in the thermal mass of the building and to reduce the internal air temperature. It can increase the building structure's capacity to act as a heat sink for the following day, cooling the internal environment's air and delaying peak temperatures. In this way, it creates a time lag, postponing the moment on which the maximum acceptable indoor temperatures are reached, and mechanical ventilation systems become unavoidable if thermal comfort temperatures are to be attained (Givoni, 1994, 1998; Santamouris, 2006; Lissen *et al.*, 2007). Due to safety issues, during nighttime, the window opening was restricted to 20% of the maximum effective area. This restriction limits the potential for night cooling due to natural ventilation but as previous studies on warm and humid climate have shown, the probability to use nighttime natural ventilation in residential building is limited to almost 10% compared to the daytime (Kubota *et al.*, 2009). Although the use of window restriction during nighttime might limit the potential for nighttime cooling, it represents real-life occupant behaviour and expectations in response to safety and noise issues.

3.5. Ventilation and Cooling Strategies and selected cities

To address Objective 2 of this research, it was essential to identify and design the control strategies for mixed-mode buildings. The selection of the mixed-mode ventilation and cooling control strategies was based on the climate applicability; energy saving potential; indoor air quality; and ease of implementation.

The selection of ventilation and cooling strategies (VCS) was done with increased complexity. VCS 1 and 2 use only the windows/dampers and cooling system to maintain thermally comfortable internal conditions. These systems are widely used in any residential apartment in India. VCS 3 incorporates the use of ceiling fans. The elevated speed by the use of ceiling fans has a positive impact on the thermal sensation in hot and humid climates. Lastly, VCS 4 is a strategy that incorporates dehumidifier and mechanical ventilation. This is a scenario that can result in the best thermal conditions for the occupants. For all these

[2] "The GHR consists of the total amount of direct and diffuse solar radiation received from on a horizontal surface" (DOE, 2018), and is assumed to be an indicative of sunrise and sunset hours.

strategies, the analysis focused on the predicted energy savings that can be achieved as well as on the thermal comfort conditions based on the adaptive theory. The four mixed-mode ventilation and cooling strategies that were studied in this research are presented below:

i. **VCS 1:** Natural Ventilation (windows only) and mechanical cooling system;
ii. **VCS 2:** Natural Ventilation (windows and dampers) and mechanical cooling system;
iii. **VCS 3:** Natural Ventilation (windows and dampers), ceiling fan and mechanical cooling system;
iv. **VCS 4:** Natural Ventilation(windows and dampers), ceiling fan, mechanical ventilation, dehumidification and mechanical cooling system;

For each of the ventilation and cooling strategies two different cases with or without nighttime window restriction were examined:

- *No restriction during nighttime:* When cooling was needed in the space, priority was given to the natural ventilation solutions. Natural ventilation was available for both day and nighttime with no restriction on the opening area.
- *Nighttime restriction opening:* This ventilation scenario is similar to the "*No restriction during the nighttime";* however, for this scenario during nighttime due to safety issues and noise, windows are restricted to operate at 20% of their maximum allowable opening area.

To examine the energy saving potential of each of the above control strategies as well as their ability to deliver a comfortable internal environment throughout the year, the base case scenario was simulated:

Base case scenario

- *Fully mechanical mode.* During this, windows and dampers remain closed; when heating or cooling was needed the mechanical system was turned on. This case represents the ideal case for thermal comfort, but it is the worst-case scenario regarding energy consumption.

In order to identify the most appropriate/representative locations for the analysis, it was important to analyse the climatic conditions of India. The Indian Energy Conservation

Building Code (Bureau of Energy Efficiency, 2009) divides India into 5 zones: “*hot and dry*”, “*warm and humid*”, “*composite*”, “*temperate/moderate*” and “*cold*”.

The “*hot and dry*” regions are characterized by high temperatures, reaching 45°C during the day, low levels of humidity, 25-40%, and rainfall less than 500mm/year. In the “*warm and humid*” regions, the average external temperature is about 35°C but due to intense rainfalls, 1200mm/year, the average outdoor humidity is about 70-90%. The “*moderate*” regions are characterized by relatively high average daily temperatures, approximately 30-35°C, but during the night it drops to 17-24°C with relative humidity about 60-85%. In the “*cold*” regions, the average outdoor temperature can reach as low as 4°C during the night. The humidity is low, approximately 10-50%. Finally, the “*composite*” category consists of regions that for at least 6 months do not fall into any other category (Bureau of Energy Efficiency, 2009).

The second step on selecting the most appropriate locations is to analyse the population density across these climatic zones. The Indian population is estimated at approximately 1.33 billion (WorldPopulationReview, 2019) people and the geographical area of 3.29 million km^2 (Census, 2019). Data indicates that almost 93% of the population of India is based in three climatic regions namely “*hot and dry*”, “*warm and humid*” and “*composite*” (Census, 2019), which account for almost 95% of the total land area of India.

Table 3-1 summarizes the 10 most populated Indian cities and their climatic zones. As highlighted in the table, there is no city in the “cold” zone, and specifically, the biggest city in the “cold” zone is Gauhati with a population slightly above 1 million people (Census, 2019). Hence based on the population, on the climatic conditions and on the suggestions of the Indian Energy Conservation Building Code (Bureau of Energy Efficiency, 2009) the cities that were selected for the current research work are: ***Ahmedabad*** (Hot & dry), ***Bangalore*** (Moderate), ***Chennai*** (Warm & humid), ***New Delhi*** (Composite). The selection of cities with the higher population was made so the proposed control algorithms can have a wider applicability and will result in larger scale energy savings. Chennai was selected over Mumbai as Chennai is one of the fastest-growing cities in India and more new buildings are expected to be built there. Hence the positive effect of using the proposed control algorithms is higher.

Table 3-1: Population of the 10 most populated cities in India (Census, 2019)

City	Climatic zone	Population [millions]
Mumbai	Warm and humid	12.44
New Delhi	Composite	11.03
Bangalore	Moderate	8.44
Hyderabad	Composite	6.99
Ahmedabad	Hot and dry	5.57
Chennai	Warm and humid	4.65
Kolkota	Warm and humid	4.50
Surat	Hot and dry	4.46
Pune	Warm and humid	3.12
Jaipur	Composite	3.05

Table 3-2 summarizes the different ventilation and cooling scenarios that were examined for different climatic conditions, using a variety of sophisticated control algorithms that were developed in this research. All scenarios were evaluated for all four selected cities aiming to assess the energy savings and thermal comfort conditions that can be achieved based on the developed control algorithms.

Table 3-2: Summary table with the different scenarios

Available Systems	Windows	Dampers	Ceiling Fan	Mechanical cooling	Dehumidifier	Mechanical Ventilation
Living Room	Yes	Yes	Yes	Yes	Yes	Yes
Master Bedroom	Yes	Yes	Yes	Yes	-	-
Small bedroom	Yes	Yes	Yes	Yes	-	-
AC (Split) unit / Evaporative Cooler						
Base case Strategy 1	Not Available	Not Available	Not Available	Available	Not Available	Not Available
VCS 1	Available	Not Available	Not Available	Available	Not Available	Not Available
VCS 2	Available	Available	Not Available	Available	Not Available	Not Available
VCS 3	Available	Available	Available	Available	Not Available	Not Available
VCS 4	Available	Available	Available	Available	Available	Available

3.6. Simulation software

The aim of this research was to develop and evaluate the performance of the flexible control algorithms for mixed-mode buildings. To address this proposal, computer simulations were essential to develop control algorithms and to evaluate their performance under a variety of different scenarios.

The use of simulation tools provides flexibility in the design process of a building or a control strategy since it can be used to assess their thermal performances relatively easy. To evaluate the thermal performance of a building, DTM software were used, since they can deliver building thermal modelling analysis in very small timestep, even sub-hourly for variables, such as internal temperatures, loads. The main objective of all the DTM tools is to perform whole building energy simulation analysis, and not to focus on the control of the HVAC or natural ventilation systems. DTM tools have limited capabilities to develop customised control algorithms. Hence, the use of additional software that will focus on the design of the control strategies for natural and mechanical systems was essential. To develop the control algorithms, an object-oriented simulation tool was selected, as it offers great flexibility to develop and customize the control algorithms and most importantly it facilitates the functions to perform co-simulations. Finally, to perform the analysis of the airflow internal the building, calculate the ventilation effectiveness and the temperature stratification CFD simulations were performed.

3.6.1. Dynamic Thermal Modelling tool

DTM simulations were performed to assess the energy-saving potential of the developed control algorithms in a variety of scenarios (Objective 4). The DTM simulations evaluated the energy performance of a variety of mixed-mode ventilation scenarios including different cooling technologies under four different climatic conditions.

The software EnergyPlus (DOE, 2018) was selected for this research because it is an open-source, freely available whole building energy simulation tool. EnergyPlus has been commonly used by others to evaluate the performance of natural ventilated and mixed-mode buildings (Ezzeldin and Rees, 2013; Wang and Chen, 2013; Wang and Greenberg, 2015; Sorgato *et al.*, 2016). EnergyPlus allows users to model energy consumption for heating, cooling, ventilation, plug-loads and lighting as well as thermal comfort. The program consists of ‘objects’, each holding information regarding a particular aspect of the building, for instance, objects to define materials properties, and objects to create schedules. The interesting aspect of the software is that it allows the user to perform a particular task using different combinations of objects. For instance, the impact of ventilation on energy consumption can be assessed in various ways; it can be examined by simply entering a single value (e.g. 1 air change per hour) or more advanced options can be used such as the Airflow network that provides the ability to simulate multizone airflows driven by wind. Another very useful feature of EnergyPlus is the Energy Management System (EMS) that allows the user to write a custom computer program in EnergyPlus Runtime Language (Erl) to create a specific output list or to override built-in routines. This feature allows the user to write custom computer programs to create outputs not available by EnergyPlus (e.g. sums of hours exceeding a threshold temperature) or to override routines embedded in the software. Furthermore, the use of EMS is essential to create customized variables for co-simulation purposes. Although EnergyPlus does not provide any user-friendly design interface to create geometries, a building geometry can be developed by using the coordinates of each point or each object (wall, windows, doors etc), which can increase the levels of complexity in the model. Hence, DesignBuilder software (DesignBuilder, 2017) was used for this research in order to create building geometries. The user-friendly design interface and its compatibility with EnergyPlus were the main factors that led to the use of this tool. The geometry created can then be exported as an .idf file, and then imported in EnergyPlus to perform the simulations.

3.6.2. Object-oriented modelling tool

For this research, Dymola was the second simulation tool that was selected to address Objective 3. Dymola is a commercial tool based on the Modelica language (DYMOLA, 2018) and uses different libraries to build more complex models. Such a library is the Buildings library from Lawrence Berkeley National Laboratory (Wetter *et al.*, 2014), which is a free library for modelling building energy and control systems which allow the development of feedback loops. The use of Building’s library provides the option to the user to either develop the whole model into Dymola (building envelope, mechanical systems, control strategies) or develop each model in the simulation tool that is most suitable and link the afterwards in a co-simulation environment.

Dymola provides an interface that the user can create a model with a “drag and drop” approach by using the already existing blocks or the user can create a customized model or block by writing a script. For this research, the Building’s library was used to develop control algorithms for natural ventilation and mechanical systems. To create the proposed control algorithms and to address Objective 3, customized code written in Modelica was used aside with the Building’s library to capture all the different control strategies. Below it is demonstrated an example of customized code to calculate the daily heating/cooling setpoints based on the IMAC thermal comfort model. This code takes as input the hourly outdoor temperature, it calculates the neutral comfort temperature and then it gives as outputs the upper and lower limits based on the 90% acceptability range. All the developed control algorithms for this research were developed in the form of code written in the Modelica language, similar to the one presented in Appendix A: Modelica code for control algorithms.

3.6.3. Coupling technique

The coupling of the two-simulation tools allowed to take advantage of the flexibility that Modelica language provides to develop the control algorithms, combined with the advantages of EnergyPlus to address Objective 5 of this research.

The FMI standard (Nouidui *et al.*, 2014; MODELISAR, 2017) was used to couple the two simulation tools. For co-simulation, the first simulation program exports an FMU through its FMU export interface. This FMU is then imported to the other simulation software, through its FMU import interface. The solver of the second software solves both the imported FMU as well as the models that are been developed in that software.

Both EnergyPlus and Modelica have a function for importing/exporting FMUs and hence it is feasible to couple EnergyPlus and Modelica. The created FMU is independent of the software that will be imported, and hence it can be used in any software that is compatible with FMUs. The data exchange between the two software occurs in discrete communication point in time. The communication timesteps are defined by the user and must be the same in both software, to allow for the co-simulations. The software that the FMU is imported at, is called "*master simulator"* and controls the data exchange between the two software and synchronizes the simulations. For this research, Dymola was the "*master simulator"*.

The implementation of the co-simulation was done by using the EnergyPlus module *ExternalInterface* (DOE, 2018). EnergyPlus has 6 objects that can be used to exchange variable data and overwrite schedules. These objects are:

- *ExternalInterface:FunctionalMockup UnitExport:To:Schedule*
- *ExternalInterface:FunctionalMockup UnitExport:To:Actuator*
- *ExternalInterface:FunctionalMockup UnitExport:To:Variable*

- *ExternalInterface:FunctionalMockup UnitExport:From:Schedule*
- *ExternalInterface:FunctionalMockup UnitExport:From:Actuator*
- *ExternalInterface:FunctionalMockup UnitExport:From:Variable*

These objects can be used either to send information for a variable, schedule to FMU or to receive information from the FMU. The objects that have in their name "*To"* were used to receive the information from the FMU and pass it to EnergyPlus whilst the objects with "*From*" in their name pass the information from EnergyPlus to Modelica through the FMU.
To create the FMU from EnergyPlus, the EnergyPlusToFMU software was used (Nouidui and Wetter, 2018). *EnergyPlusToFMU* is a software package written in Python, which supports model export as an FMU for co-simulations using the FMI standard. To create the FMU, the use of window's command line was necessary. The steps to create the FMU is to run the compiler and then call the python package *EnergyPlusToFMU.py* and also the EnergyPlus data dictionary, *Energy+.idd* as well as the weather file (.epw) and the input data file (.idf) in order to create the FMU. It should be mentioned that the created FMU is a unique file based on the given inputs: i) the EnergyPlus file (.idf) and ii) the weather file for the location. For this research, it was required to perform co-simulations at different locations and hence different weather files were used, thus separate FMU files were created for each location.

The created FMU is a compressed library containing all the functions needed to couple EnergyPlus models with the Modelica. The FMU is then imported to Dymola to run the co-simulations. EnergyPlus and Dymola ran at the same time during the co-simulation and in every timestep, $T_{timestep}$=600 sec, the two-software exchanged information. This process is completely automated, and the user does not have to interact with the software after the co-simulation had started. For the proposed co-simulations, Dymola was the master simulator controlling the communication of the two software and was responsible for exchanging the information. Figure 3-2 shows the variable exchange between the two software. This figure presents the main variables that have been used in all the models.

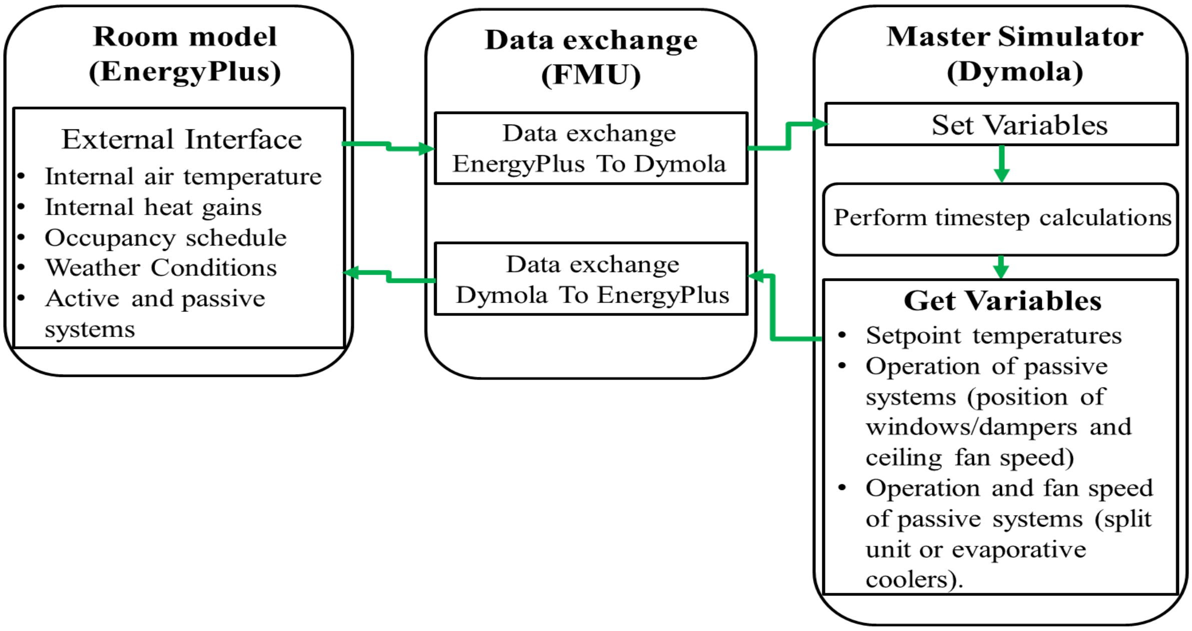


Figure 3-2: Variable exchange between the two simulation tools

This simulation approach provides great flexibility as the user can easily adjust all parameters in each software and run the simulations again without having to adjust the simulation setup up each time.

Figure 3-3 illustrates the interaction between the two simulations tools. The white box in the middle “TwoBedApart_fmu” represents the model from EnergyPlus. This is the FMU containing all the information regarding the building geometry, mechanical systems, internal gains and occupancy schedules. One the left side of the box are the required inputs from Modelica. For this case, the required inputs from Modelica are the position of the opening; the heating and cooling setpoints; the availability schedules for cooling and heating coiling; ceiling fan;mechanical ventilation and dehumidifier. This information is calculated for each simulation timestep (600 sec) and sent to FMU in order to calculate the internal temperature, cooling and heating demand. The yellow blocks on the left side of the “TwoBedApart_fmu” are blocks which the output signal is identical to the sampled input single at sample time, and they are used to ensure that during the exchange of information between the two software the signal will remain the same.

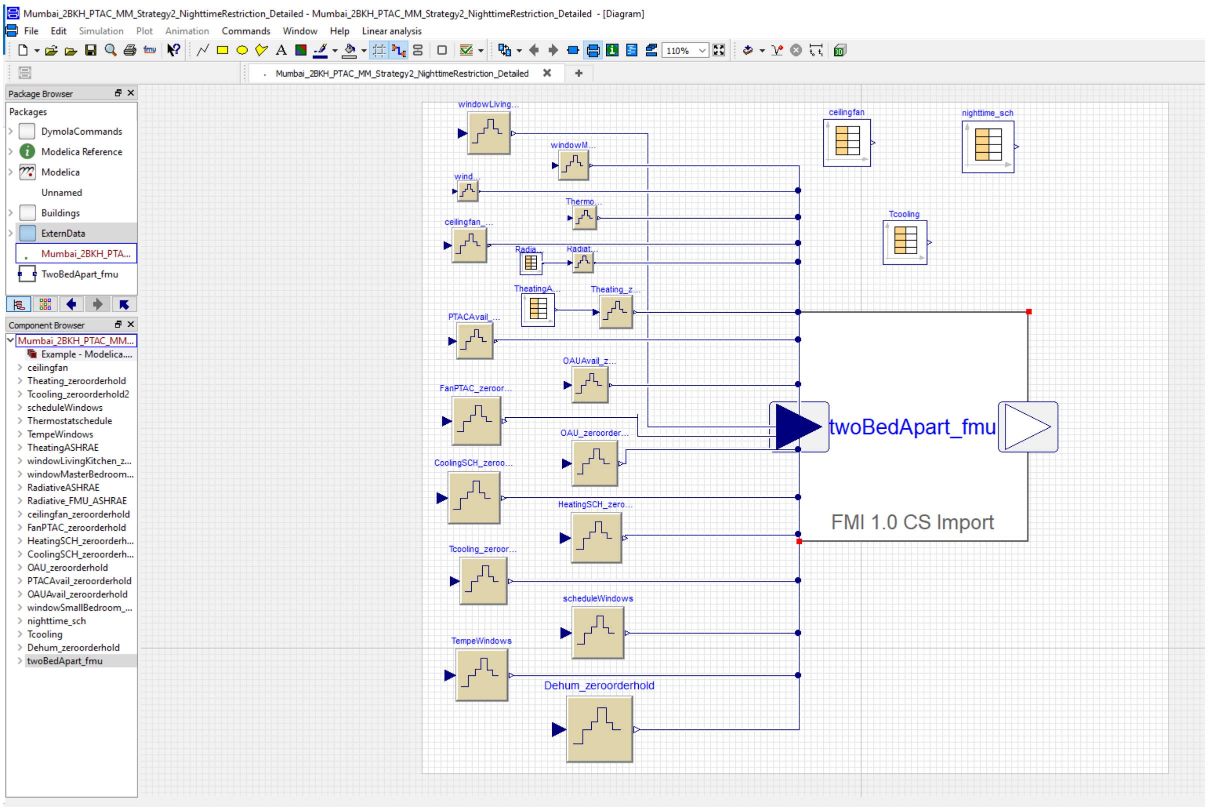


Figure 3-3: Screenshot of the Dymola interface, showing the FMU (white box), inputs from Modelica to FMU (left), and outputs from Modelica to EnergyPlus (right)

3.6.4. CFD Simulation design

For this research, the CFD simulations were developed and run using the PHOENICS CFD software version 2018v1.1-64 bit developed by Concentration, Heat & Momentum Limited (CHAM, 2019). The FLAIR interface was used to carry out steady-state simulations. FLAIR is a variant of PHOENICS which is designed to run airflow and thermal simulations and is mainly used to perform CFD simulations by building services engineers. Additionally, CHAM has developed a coding language INFORM, Fortran-based language, embedded into PHOENICS software which allows the user to create customized code to enhance the design and/or the analysis of the simulated results.

This is a well-established tool for simulating and analysing fluid flow, air velocity and air temperature by quantitively predicting the airflow patterns (Walker, 2006). PHOENICS has been used by researchers to examine the airflow patterns in naturally ventilated spaces (Chiang and Lai, 2009; Angelopoulos *et al.*, 2017; Spentzou *et al.*, 2017).

CFD software solves the conservation equations for mass, momentum and energy flow to predict the airflow patterns into space (CHAM, 2019). Generally, all the flows are turbulent, and hence when simulating a case turbulent models are required to solve the equations.

3.6.4.1. CFD modelling techniques

For this research, the κ-ε RNG turbulence model was used with the Boussinesq approximation for representing the buoyancy effects.

This model considers density variations (Cook, 1998) in the momentum equations and has been widely tested and used as the main turbulence model for steady-state modelling of buoyancy-driven flows (Dascalaki *et al.*, 1999; Visagavel and Srinivasan, 2009). The two differential equations that are used in this model are the κ-ε RNG for the turbulent kinetic energy and dissipation of turbulent energy respectively. The κ-ε RNG has been used in the past when modelling indoor airflows. Chen (1995) compared different turbulence models and concluded that when simulating indoor airflows, κ-ε RNG turbulence models were performed better compared to others. Ji *et al.*, (2007) stated in their research that for an appropriate modelling of indoor air-flows the two-equation eddy viscosity turbulence models (κ-ε RNG) are suitable. In line with this statement, Hussain and Oosthuizen, (2012) documented that when the mean air temperature and the flow rate are examined then the two-equation eddy viscosity turbulence models are suitable.

3.6.4.2. Convergence criteria and mesh sensitivity

When performing CFD simulations it is very important to ensure that the model converged. The three main parameters that were used to determine whether a converged had achieved were:

- source balance
- residual behaviour
- spot value

Source balance: the error in the mass equation should be less than 0.1% of the total mass entering the domain (kg/s) and also the value of the enthalpy residual should be below 1% of the total heat input.

Residual behaviour: Residual values or errors should be reduced by a factor of 100 from their initial sweeps. Ideally, these values should reach a constant value or at least with minor oscillations. In order to have a reasonable convergence these changes in the values should be small (Walker, 2006).

Spot values: spot values at specific locations, which representative of the domain, should reach a relatively constant value (CHAM, 2019).

To monitor these values, all the CFD simulation tools are using a probe where all these parameters are calculated. The location of the monitoring point or the probe location as it specified in PHOENICS, should be carefully placed. Hence it should be away from any sources that could alter the velocity or the temperature of the fluid quickly, such as a heat source (CHAM, 2019).

Another critical aspect of the CFD simulations and the accuracy of the results is the mesh structure. A good quality mesh structure can have a positive impact on the accuracy of results, but it can also increase the computational time significantly. Hence, it is essential to have the optimum number of cells that can generate accurate results in the minimum time. A more detailed mesh is usually applied in the regions of greater interest5 such as openings, heat sources or the domain's boundaries or in general in places within the domain where there are higher chances of a more turbulent flow compared to the rest of the domain (van Hooff *et al.*, 2013).

Accuracy and reliability of simulation studies are of concern and validation studies are imperative. The proposed control algorithms were validated to provide a higher degree of accuracy of outputs, prior to their application to a demonstration case building. For the validation study, a unit space was created in EnergyPlus that replicated a real-life experimental chamber. The application study involved the design of an apartment building with the main focus on a single two-bedroom apartment. For the purpose of these two different geometries, the experimental chamber for the validation study and the two-bedroom apartment.

3.7. Description of the experimental chamber

For rigorous research, new concepts must first be tested in a controlled environment. Only then can the principles be applied to full-scale buildings. For this research, experimental work in the form of measurements was conducted in a full-scale low energy characterization chamber located in CEPT, India. The facility consists of an outer chamber (10m x 8m x 8m) and an inner chamber (5.05m x 3.95m x 3.00m) (figure 3-4), which represent the external and internal environments respectively. The inner chamber represents a typical Indian bedroom. Two air handling units are installed, one for each chamber. In the chamber, only buoyancy-driven cases can be simulated. The outer chamber can maintain a wide range of conditions (15-40±0.1°C air temperature, 20-80±3% relative humidity) at steady state allowing the evaluation of steady-state at the inner chamber too. The inner chamber is equipped with one motorized window (1.2m x 1.2m) and four motorized vents (0.4m x 0.4m). Building materials and properties are presented respectively in table 3-3 & 3-4.

Table 3-3: Construction material and thermal properties

Material	Conductivity [W/(mK)]	Specific Heat [J/(kgK)]	Density [kg/m^3]
Kota Stone	3.02	668	3102
Sand Mortar	0.88	896	2800
Plain Cement Concrete	0.72	840	1860
Cement Putty	0.114	742	1070
Plaster (dense)	0.5	1000	1300
AAC Block	0.35	1100	780
XPS	0.029	1525	37
RCC *(Reinforced Cement Concrete) [2% steel]*	2.50	1000	2400
Cement Mortar	0.720	920	1650
Ceramic Tiles Flooring	0.920	820	1950
Acrylic Paint	0.201	1342	745

Air temperatures and surface temperatures were measured using PT100 RTD probes. The accuracy of the temperature sensors was ±0.15°C based on the manufacturers' data. Twelve internal air temperature sensors were used in a grid configuration, and six surface temperature sensors, one at each wall, roof and floor. The sensors were calibrated to ensure the accuracy of the readings. CO_2 was used for the trace gas measurements. A CO_2 source was placed at the centre of the inner chamber, to represent the internal generated CO_2, and measurements of the CO_2 rate were taken. CO_2 sensors were placed at the low- and high-level vents and at the outer chamber. The accuracy of the CO_2 sensors was provided by the manufacturer and was ±30ppm +3% of the reading.

Table 3-4: Inner experimental chamber envelope properties

Construction	Thickness (mm)	U-value [W/(m^{2}K)]
Floor		
Kotta Stone	25.0	0.46
Sand Mortar	60.0	
RCC	150.0	
XPS	50.0	
Plain Cement Concrete	75.0	
Roof		
Acrilic paint	0.6	0.36
Plaster	20.0	
RCC	150.0	
XPS	70.0	
Cement Mortar	30.0	
Ceramic Tiles	10.0	
Walls		
Acrylic paint	0.6	0.36

Putty Plaster AAC Block XPS AAC Block Plaster Ceramic Tiles	2.0 20.0 100.0 50.0 150.0 20.0 0.6	
Window		
Glass air gap Glass	3.0 6.0 3.0	 2.6

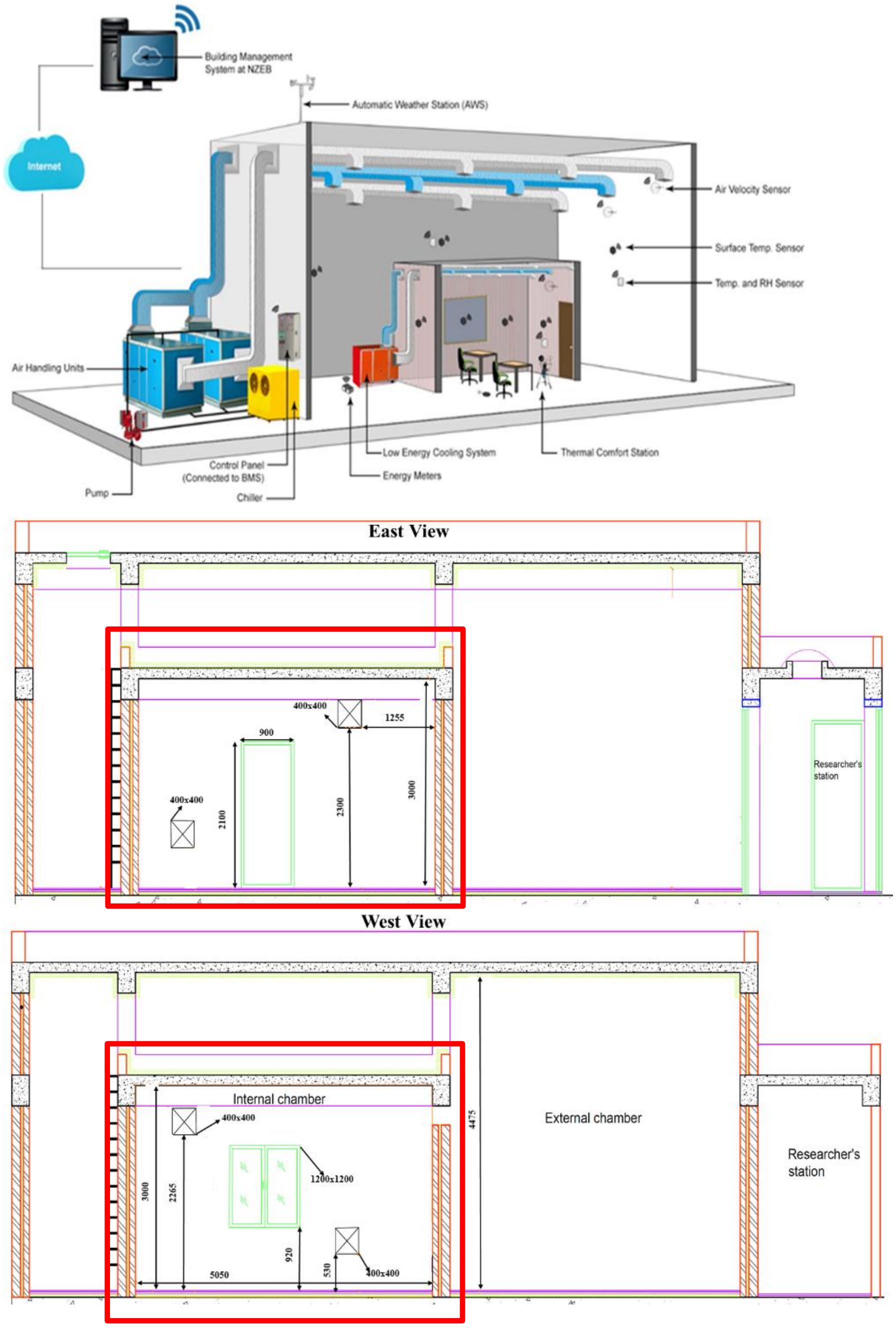


Figure 3-4: Experimental chamber in CEPT, India. The internal chamber is highlighted within the light red square.

3.7.1. Experimental set-up

The validation experiments were conducted in a chamber within a chamber as shown in figure 3-4. The LECTB has two chambers – indoor environment chamber and outdoor climate chamber. Outdoor climate chamber replicates outdoor weather conditions for low energy cooling systems and indoor environment chamber (located inside the outdoor chamber) replicates indoor space conditions. There are two Air Handling Units (AHU) dedicated to condition and maintain the environment condition in the outdoor climate chamber (AHU-1) and indoor environment chamber (AHU-2). To provide internal sensible and latent heat load inside the indoor environment chamber a separate AHU was used (AHU-3).

The natural ventilation window NV 1 and NV 2 and the sliding window W 1 as shown in figure 3-5 were used for the experiments. The control of the openings and the systems were implemented through the Building Management System (BMS) of the chamber. The positions of the air conditioner's indoor, outdoor unit and ceiling fan are shown in figure 3-6. The arrangement for CO_2 cylinder and piping for tracer gas experiment is represented in figure 3-7.

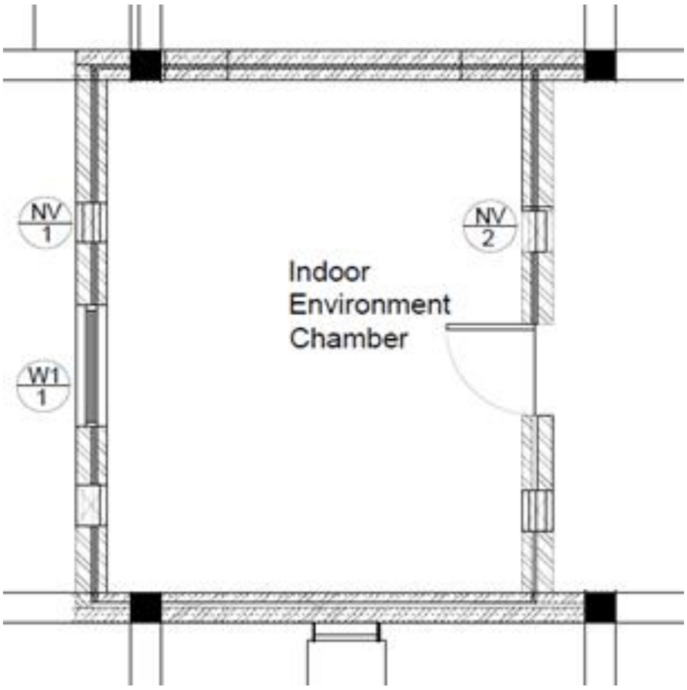


Figure 3-5: Natural Ventilation openings to be used for experiments

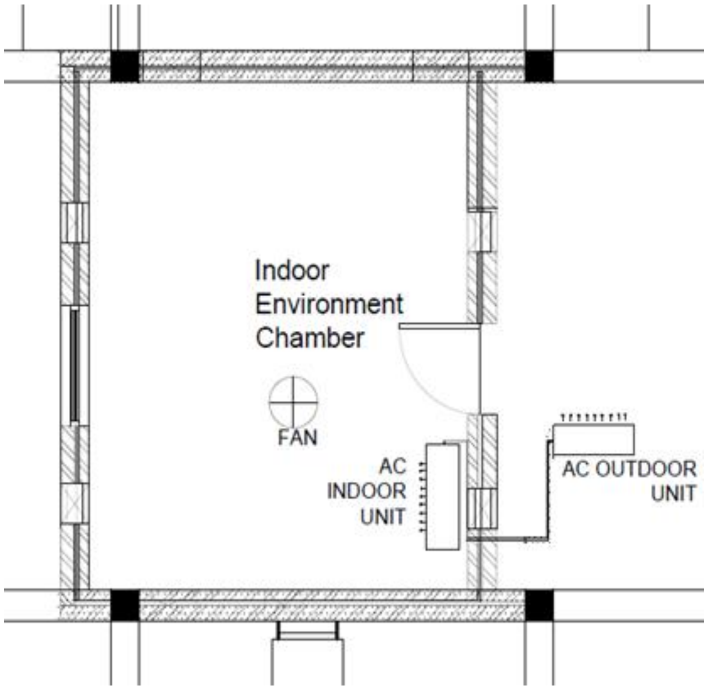


Figure 3-6: Air Conditioner (AC) indoor and outdoor unit location

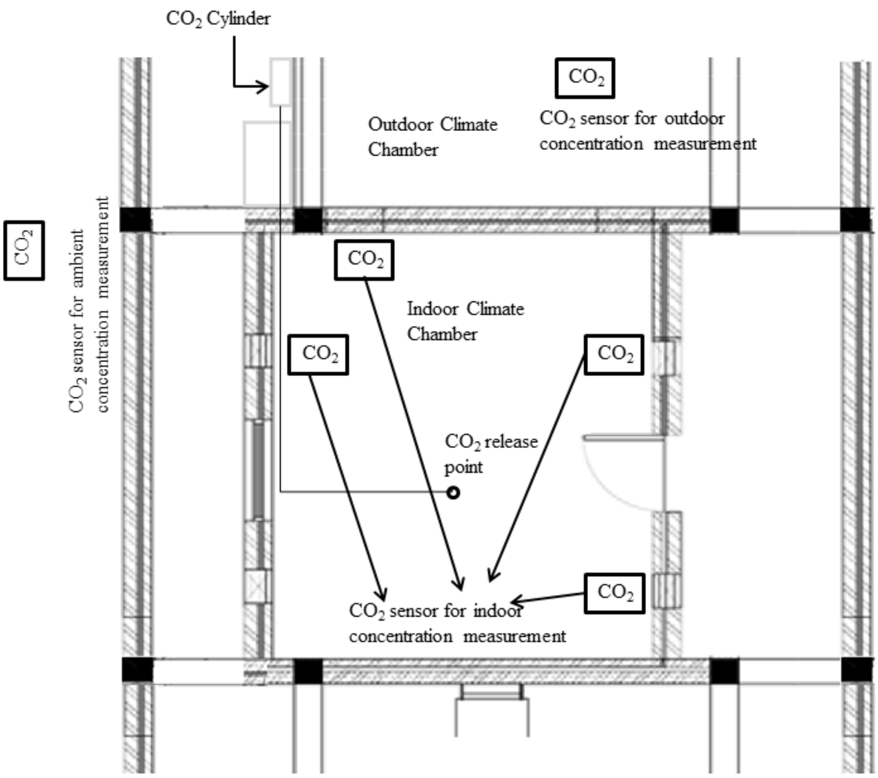


Figure 3-7: CO2 Cylinder, piping and CO2 Sensor locations

Key Measurements:

Measurements of the key parameters were essential to validate the co-simulation and CFD models. The following key parameters were measured in detail:

- Air Exchange Rates for configurations of table 5-2;
- Indoor Temperature at various points in the chamber for configurations of table 5-2;
- Indoor Humidity for configurations of table 5-2.

Location of Sensors and ceiling fan:

- A total of 13 flexible air temperature sensors were used and were arranged using a 3x3x3 meter 3-dimensional grid (figure 3-8). Four air temperature and Relative Humidity (RH) sensors and two CO_2 sensors (one for the outdoor and one for the indoor environment chambers) were installed as shown in figure 5-4. The temperature + RH sensors are shown in figure 3-9. Details for the sensors can be found in Table 3-6;
- Six sensors were used to monitor the indoor environment chamber CO_2 concentration and one each will be used to monitor outdoor climate chamber and ambient CO_2 concentration.

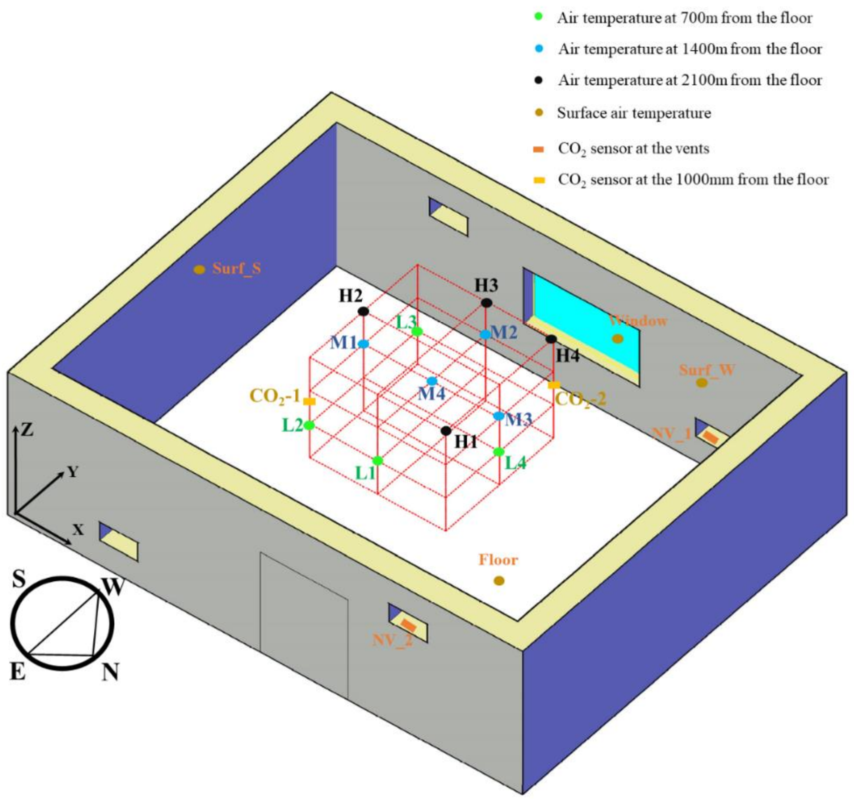


Figure 3-8: Location of sensors in the space of the indoor environment chamber

Table 3-5: Sensor location in the experimental chamber. The coordinates are given in mm.

	Location from South-East Corner of Indoor Environment Chamber		
Sensor Name	**X Coordinate**	**Y Coordinate**	**Z Coordinate**
L1	1860	1260	700
L2	1230	1260	700
L3	1230	2920	700
L4	2690	2090	700
M1	1230	2090	1400
M2	1860	2920	1400
M3	2690	2090	1400
M4	1860	2090	1400
H1	2690	1260	2100
H2	1130	2090	2100
H3	1860	2920	2100
H4	2690	2920	2100
CO_2_1	1230	1260	1000
CO_2_2	2690	2920	1000
NV_1	3670	3950	550
NV_2	3670	0	2500
Window	2480	3950	1650
Surf_W	3150	3950	1530
Surf_S	0	2000	2800
Floor	3670	1000	0

The ceiling fan was placed at X=2635mm, Y=2085mm and Z=2375mm.

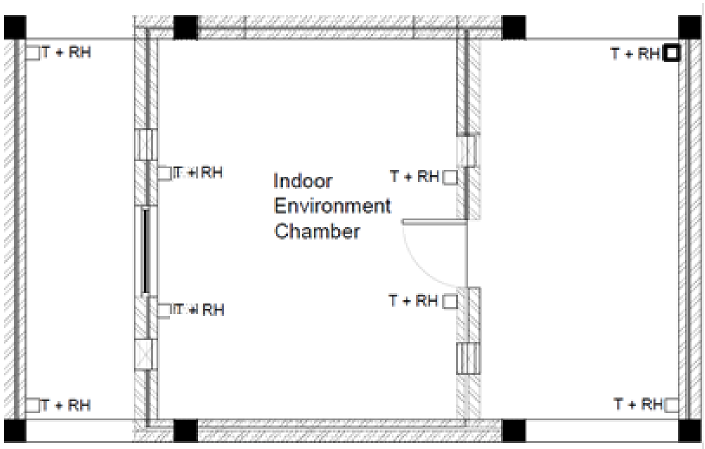


Figure 3-9: Air Temperature + Relative humidity sensors at LECTB

Table 3-6: Sensors used for the experiments and their accuracy

Instrument		Parameter measured	Quantity	Accuracy
Testo 480 Climate Measurement Instrument	**Globe Thermometer**	Globe temperature (°C)	1	±1.5°C At -40 to 375°C
	IAQ Probe	CO_2 (ppm)	1	±(75 ppm + 3 % of mv)
	Turbulence Probe	Air velocity (m/s)	1	±(0.03 m/s + 4 % of mv)
PT100 RTD		Air temperature(°C)	12	± 0.15°C
		Surface Temperature(°C)	7	± 0.15°C
CO_2 Sensor		CO_2 concentration (ppm)	2	±30 PPM + 3% of Reading
IAQ Sensor		CO_2 concentration (ppm)	2	50ppm± 3%

The timeline of each experiment is illustrated in figure 3-10. Prior to each individual experiment performed, the process required to reach steady-state (Stage A), followed by the release of the CO_2 (Stage B), the experimentation period (Stage C) and finally the period required to cool down the chamber in preparation for the next experiment (Stage D). The four stages are as follows:

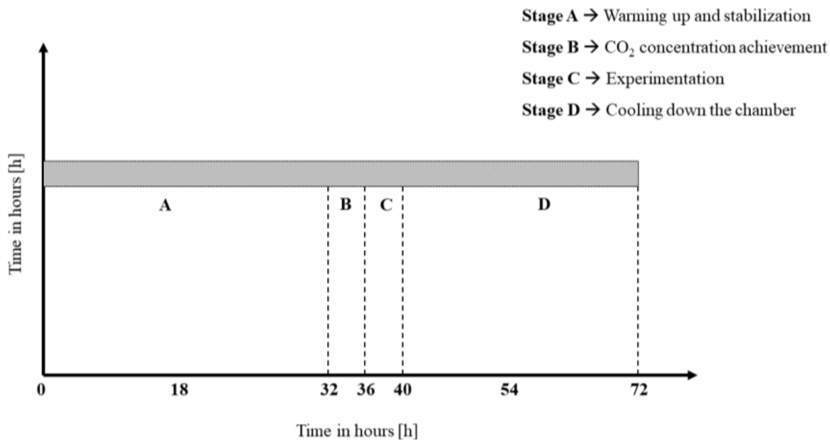


Figure 3-10: Timeline of each experiment

Stage A: Chamber Stabilization

- Outdoor temperature and RH were stabilized as per the specified conditions using AHU-1. The temperature and RH were controlled with an accuracy of ±0.3°C and ±3% respectively. The chamber was stabilized using the temperature and RH values from the co-simulations 1 hour prior to the start of the experiment. For instance, for scenario 1, table 5-2, the experiment was scheduled to run on the 2nd of July at 11 pm. To stabilize the conditions in the chamber the average temperature and RH from 22:00 of the 2nd of July were used. The same process was applied in all the scenarios.
- Indoor temperature and RH were stabilized to desired conditions for the conduction of the experiment using AHU-2 with and accuracy of ±0.3°C and ±3% respectively.
- Fresh air quality was maintained to achieve near equal CO_2 concentration to ambient in the outdoor chamber.

Stage B: Arrangement of cooling systems and heat loads

- Fan, window, and AC were set up as per the configurations in table 5-2.
- Internal loads were maintained through AHU-3 as per table 5-2.
- CO_2 tracer flow will be maintained to achieve 2000ppm inside the indoor chamber.

Stage C: Experimentation

- The indoor conditions (temperature, RH, and CO_2 level) were monitored for four hours. A total of 13 flexible and four fixed temperature points, as well as two CO_2 concentration points, were selected, as shown in figure 3-8.

Stage D: Cooldown of the chamber

- After the completion of the experiments, the AHUs were turned off in order to cool down the internal chamber.

3.8. Description of the demonstration case

To accommodate the rapid increase of the population, it is expected that the floor area in India to increase by 500% with approximately 20 billion m^2 of new building floor area by 2030 (Kumar, 2011). Hence focusing on residential buildings is of great importance as there is a huge potential to reduce energy demand for cooling.

Following the evaluation of the developed control algorithms in a controlled environment, their applicability in real buildings was investigated to predict their energy savings potential (Objective 3). For this purpose, a typical apartment block was used. A study by Rawal & Shukla, (2014) analysed approximately 57 building designs across India and concluded that there are two typical layouts of building and four types of residential buildings. This analysis showed that the most typical apartments in India can be clustered as shown in figure 3-5.

Figure 3-11: Indian residential building plan, general layout for two-bedroom apartments for design case 1 (left from the dashed line) and design case 2 (right from the dashed line). In the red square, the flat that was used for this research is highlighted (Excerpt from Shukla et al., (2014))

Rawal & Shukla, (2014) reported that the majority of the Indian urban residences use reinforced cement concrete for structural stability, whilst the walls are constructed by brick and cement block masonry. Two different types of building envelopes will be considered in this study: one from the Business as Usual (BAU) and one from the Energy Conservation Building Code (ECBC). Based on this study, the envelope properties are presented in table 3-5. The same building envelope characteristics have been used for the DTM simulations and it was assumed that the infiltration rate for all the different construction methods will be 1ACH (Rawal and Shukla, 2014). Figure 3-6 shows the plan view and the dimensions for each room for the 2-bedroom apartment that was simulated.

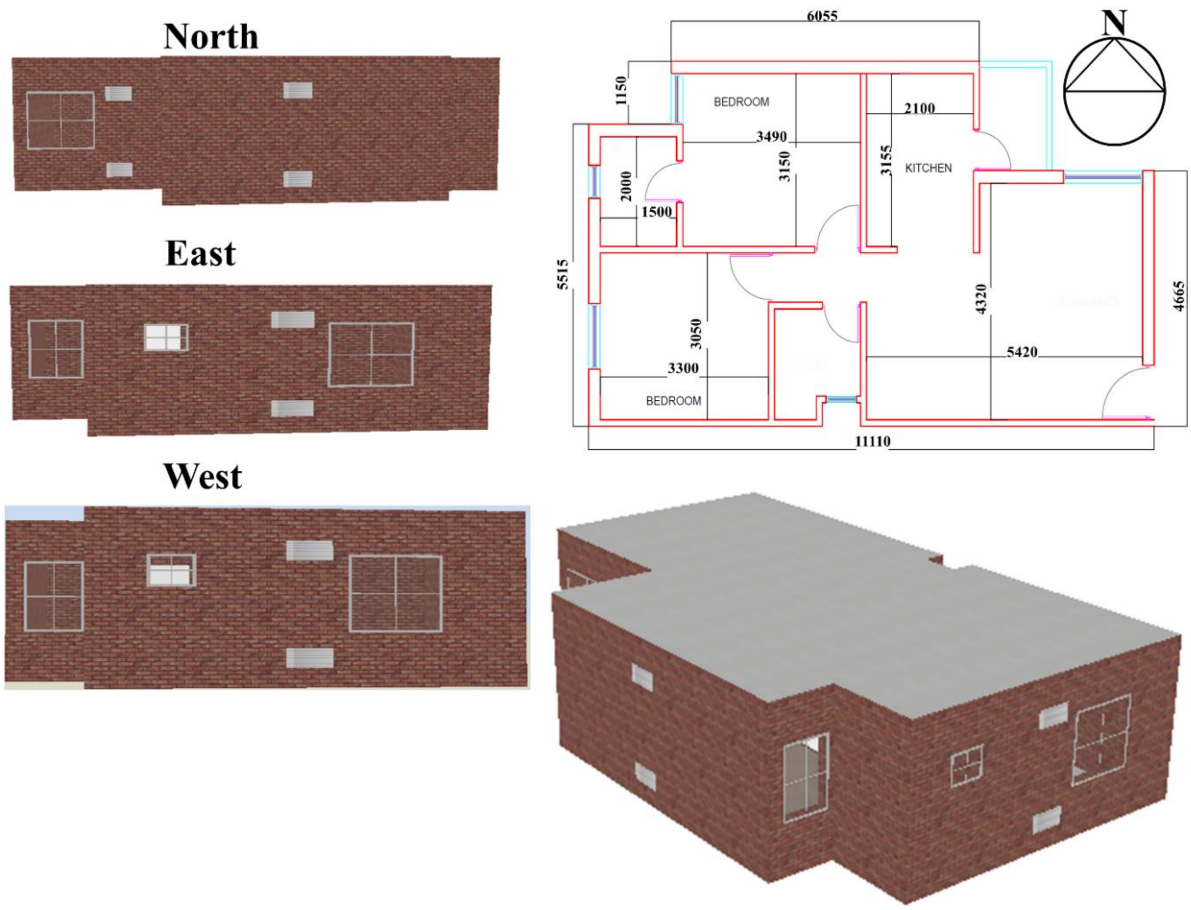


Figure 3-12: Floor plan for a 2-bedroom apartment, Case 1. Dimensions are in mm. The 3D shows the location of the windows/dampers for the 2-bedroom apartment.

Table 3-7: Building envelope properties. (Excerpt from (Shukla et al., 2014))

	BAU Envelope Properties	
Wall	-20mm cement plaster	U-Value:

	-Uninsulated 230mm brick wall -20mm cement plaster	1.722W/m^{2}K
Roof/floor	-10mm cement plaster -150mm concrete roof -12mm Tiles	U-Value: 2.942W/m^{2}K
Window	Single-glazed 6mm	U-Value: 5.8W/m^{2}K

3.8.1. Modelling inputs for the demonstration case

The selected apartment has a floor area of 60m^2 and internal height of 3m. For the purpose of this analysis, the two bedrooms and the living room were considered to be separate thermal zones with separate control over the systems, meaning that each system (passive or active) can operate irrespectively of the others. Although this adds complexity into the model and increases the computational time, it provides higher accuracy of calculations and a better understanding of the airflow between the different zones. For each thermal zone, mechanical systems, ceiling fans, windows and dampers were designed and control via the proposed control algorithms.

As figure 3-5 indicates, the flat that was analysed has all the external walls exposed to the environment except the south walls. Hence for this analysis, the south and partition walls, ceiling and floor were assumed to be adiabatic. For this analysis, the BAU envelope properties, table 3-3, were used. For the cooling systems, both the AC split units and the evaporative coolers were simulated based on the different scenarios. The cooling systems were located in the two bedrooms and the living rooms, similar to the ceiling fans. The values for the internal heat gains, occupancy schedules and a number of occupants were discussed in subsection 3.8.2.

To model the natural ventilation in EnergyPlus, the *AirflowNetwork* model was used as it has been used extensively in the literature to model natural ventilation in buildings (Bre *et al.*, 2016; Sorgato *et al.*, 2016). This model was selected over other models in EnergyPlus because it can simulate the multizone airflows driven by wind or/and by buoyancy forces

(DOE, 2018). Additionally, by using the *AirflowNetwork* model, it was possible to simulate the airflow through cracks in the exterior walls and around the windows/dampers when they were closed. The *AirflowNetwork* model provides zone level control of the windows/dampers as well as modulation of the opening to avoid any large temperature swings (DOE, 2018). For window-blindings, typical internal blindings were used and their control was manual. Hence, they could operate only when the rooms were occupied during daytime.

Due to software limitations, ceiling fans in EnergyPlus were added with the following approach was used. To represent the electricity consumption of the ceiling as well as the internal heat gains in the zone, an object in EnergyPlus called "other equipment". In this object, it was assigned a value for the power of 50W to represent the power of a typical residential ceiling fan used in the Indian context (Babich *et al.*, 2017). To calculate the energy consumption of the ceiling fan, the power was multiplied by the hours of use n:

Eq. 3-6

$$E_{fan} = n * 50$$

The proposed control algorithms were used to control the systems for the different ventilation and cooling scenarios presented in Section section 3.5 – table 3-2.

3.8.2. Internal and external heat gains

A very important parameter to consider while designing natural or mixed-mode ventilation strategies is the total amount of heat gains in the space. The total amount of heat gains, in Watts, for a space is the sum of the internal and external heat gains (Eq. 3-7).

Eq. 3-7

$$Q_{total} = Q_{external} + Q_{internal}$$

External heat gains

External heat gains are the amount of solar radiation absorbed and transmitted by the building envelope, via opaque walls and windows into the internal environment of the building. These heat gains derived through the DTM simulations. To calculate the heat gains from the building envelope, the heat conduction rates from outside to inside (heat gains) and from inside to outside (heat loss) providing the total heat flow to the thermal zone via external solar radiation were calculated. For the windows, the total amount of direct and diffuse solar radiation entering the internal environment through the window and the heat conduction rates from outside to inside (heat gains) and from inside to outside (heat loss) were used.

Internal heat gains

To calculate the total amount of heat gains due to electronic equipment, data from the LECaVIR project were used regarding the electronic equipment usage schedule as well as the typical installed W/m^2 for a 2-bedroom apartment (Cook *et al.*, 2018). The average electronic equipment density was assumed to be 18.7 W/m2, while the assumed usage schedule is presented in figure 3-7 (Cook *et al.*, 2018).

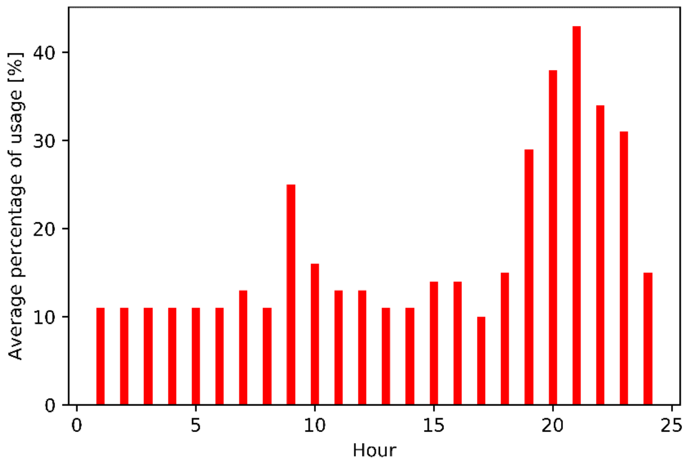


Figure 3-13: Daily average percentage of usage of electronic equipment

The percentages of the graph refer to the whole floor area, for example at 2 a.m. the internal gains from the use of electronic equipment are: 11%*18.7 W/m^2*60 m^2=123.4W.

The total amount of internal heat gains is the sum of heat gains by the occupants and heat gains by any electronic equipment in the room. To calculate the occupants' heat gains, the metabolic rate from CIBSE Guide A, (2015) was adopted whilst the occupancy scheduled was used by LECaVIR project (Cook *et al.*, 2018). The assumed metabolic rate was (CIBSE Guide A, 2015):

- From 08h00 to 18h00: 75W per person
- From 18h00 to 08h00: 41W per person

Table 3-6 summarizes the occupancy schedule and heat gains from the occupants. It is assumed that the flat was occupied by 4 people in total, and the bedrooms can be used during day-time from the occupants for other activities such as studying. That's the reason that it is assumed, especially during weekends, that bedrooms can be occupied even during day-time. The total amount of heat gains was calculated for every timestep (600 sec) to calculate the required airflow rate using equation Eq. 3-7.

Table 3-8:Schedule of occupants per room for 24 hours and internal heat gains with occupants

	Weekdays				**Weekends**			
	Bedroom en-suite	**Bedroom small**	**Living room**	**Maximum total number**	**Bedroom en-suite**	**Bedroom small**	**Living room**	**Maximum total number**
Number of occupants								
08h00 to 18h00	0	0	2	2	2	2	4	4
18h00 to 08h00	2	2	0	4	2	2	0	4
Heat gains [W]								
08h00 to 18h00	0	0	150	-	150	150	300	-
18h00 to 08h00	82	82	0	-	82	82	0	-

Chapter 4. Control algorithms for mixed-mode buildings

4.1. Overview

In this chapter, the control of the passive and active systems for mixed-mode buildings that were identified in the literature review are presented. A detailed analysis of the control of both active and passive systems is presented based on mathematical equations. Following this, the novel control algorithms developed in this research are presented (Objective 1) in this chapter were used in later chapters to evaluate the energy saving potentials for hot climates. Furthermore, a feasibility study of the adaptive comfort model used in this research (IMAC) was presented to examine the applicability of the model under the weather conditions.

4.2. Control of passive and active systems for mixed-mode buildings

As it was identified in the literature review, mixed-mode buildings provide the flexibility to combine both mechanical and natural systems to maintain thermally comfortable internal environments with the minimum energy consumption. To achieve this, it is essential to incorporate sophisticated control algorithms for the windows/dampers and ceiling fans as well for the mechanical systems to ensure minimum energy consumption. This subsection describes how the passive and active systems were modelled in the simulation software.

4.2.1. Control of passive systems: Windows and dampers

The majority of the DTM tools control the operation of windows based on the temperature difference between the indoor and outdoor dry-bulb temperatures. In every time step, EnergyPlus checks whether the internal air temperature is within the setpoint temperature for natural ventilation and whether internal air drybulb temperature is higher than the outdoor temperature. This research, however, proposes a more advanced methodology to operate the windows, the “detailed control”.

When natural ventilation is required, the opening of the windows is modulated according to the indoor and outdoor conditions based on a linear relationship, see figure 4-1 (DOE, 2018). Windows are kept closed when the difference between the internal and external temperature is greater or equal to 20°C and are fully open when the difference is zero (Wang and

Greenberg, 2015). Use of dynamic opening of the windows has not been previously used by many in literature.

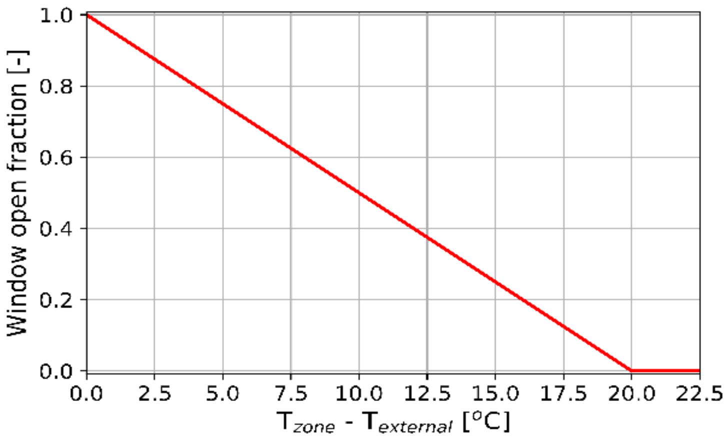


Figure 4-1: Modulation of window opening according to the temperature difference

The current research proposes a new approach to control the window operation, which is in response to the required airflow rate to delivery cooling based on the internal heat gains rather than on the temperature difference between inside and outside. For each timestep, the internal heat gains are calculated using the Eq. 4-1 (CIBSE Guide B, 2005):

$$Q_{gains} = \dot{m}_{total} C_p \left[T_{int,aver} - T_{i,CSP}\right] \qquad \text{Eq. 4-1}$$

Where Q_{gains} is the total heat gains [W]; $\dot{m}_{total}$ is the total airflow through the openings due to wind and buoyancy forces [m^3/s]; C_p is the specific heat capacity of air [kJ/(KgK)]; $T_{int,aver}$ is the average internal air temperature [°C], $T_{i,CSP}$ is the cooling setpoint [°C].

Then, by using the Eq. 4-2, Eq. 4-3 & Eq. 4-4 the required mass flow rate for the given internal and external temperatures is calculated (CIBSE Guide B, 2005):

$$\dot{m}_b = C_d A_w \left[\frac{2\left[T_{int,aver} - T_{out,aver}\right] h_a g}{\bar{t} + 273}\right]^{0.5} \qquad \text{Eq. 4-2}$$

$$\dot{m}_w = 0.05\, A_w V_r \qquad \text{Eq. 4-3}$$

$$\dot{m}_{total} = \sqrt{\dot{m}_s{}^2 + \dot{m}_w{}^2} \qquad \text{Eq. 4-4}$$

Where $\dot{m}_b$ and $\dot{m}_w$ are the airflow due to buoyancy and wind forces respectively [m^3/s]; C_d is the discharge coefficient [-]; $A_w$is the effective area of the windows [m^2]; $T_{out,aver}$ the average external air temperature [°C], h_a is the vertical distance between the centres of the openings [m]; g is the acceleration due to gravity [m/s^2]; $\bar{t}$ is the average value of the internal and external temperature [°C]; V_r is the wind speed [m/s].

Finally, by combining formulae Eq. 4-1 and Eq. 4-4 the effective area of the windows is:

$$A_w = \frac{\dot{m}_{total}}{\sqrt{0.05^2 V_r{}^2 + C_d{}^2 \left[\frac{2[T_{int,aver} - T_{out,aver}]h_a g}{\bar{t} + 273}\right]}} \qquad \text{Eq. 4-5}$$

Hence the modulation of the windows was implemented in such a way to match with the calculated required effective area for the given conditions. Dampers are also used in many applications to improve the performance of natural ventilation solutions. The control logic of the dampers was assumed to be identical to the windows. However, when natural ventilation was needed, dampers were assumed to have the highest priority to operate and when they were fully open then the windows were “allowed” to open.

To simulate the windows/dampers into EnergyPlus the *AirFlowNetwork* (AFN) module was used (DOE, 2018). The AFN provides the ability to simulate single or multizone air-flows. The AFN consists of a number of nodes that are connected by the airflow components through the surface linkages. In EnergyPlus, surface linkages are the heat transfer surfaces that both faces are exposed to the air. The AFN assumes that air flows from one node to another, and hence it cannot predict internal air circulation in a thermal zone (EnergyPlus, 2017). The airflow through a crack in the walls, floors and roof are calculated in the AFN as a function of the pressure difference(EnergyPlus, 2017):

$$Q = Crack_{Factor} * C_T * C_Q * (\Delta P)^n \qquad \text{Eq. 4-6}$$

where,

Q is the air mass flow rate [kg/s]; $Crack_{Factor}$ is the multiplier for a crack; C_T is the reference condition temperature correction factor [dimensionless]; C_Q is the air mass flow coefficient [kg/s at 1 Pa]; ΔP is the pressure difference across crack [Pa]; and n is the airflow exponent [dimensionless]. The values are ranging from 0.5 for fully turbulent flows to 1.0 for fully laminar flows (EnergyPlus, 2017).

When the openings are open, the volume flow rate is calculated by the formula below (EnergyPlus, 2017):

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}} \qquad \text{Eq. 4-7}$$

where,

Q is air mass flow rate [kg/s];C_d is discharge coefficient [dimensionless] depends on Reynolds number and geometry of the windows; ΔP is pressure difference across the openings [Pa],A=surface opening area [m^2]; and ρ is the air density [kg/m^3].

The input variables that are required from EnergyPlus to establish the AFN are: wind pressure coefficient (C_p); airflow coefficient (C_Q); flow exponent (n) for each crack, windows and doors and the discharge coefficient (C_d) for each opening at each opening factor.

Wind pressure coefficient (C_p)

AFN uses the wind pressure coefficient to determine the wind-driven pressure on the external surfaces of the building. The (C_p is depended on a number of factors such as building geometry, façade characteristics, wind speed and direction (Cóstola *et al.*, 2010). Therefore, the C_p is very difficult to calculate so assumptions have to be made. The window pressure coefficient values were determined by the ASHRAE Fundamentals Handbook (ASHRAE, 2009).

Air mass flow coefficient (C_Q) and flow exponent (n) for each crack and windows when they are closed

AFN requires the C_Q for each crack on the internal or external wall, floors and ceilings at 1 Pa pressure difference across the crack. These cracks are very difficult to characterize even with the visual inspection so all the datasets that are available are based on estimations.

In the DesignBuilder database (DesignBuilder, 2019), there are five different crack templates: Very poor, poor, medium, good and excellent according to the leakiness level. Based on values from the literature (Beizaee, 2016; Sorgato *et al.*, 2016; EnergyPlus, 2017) table 4-1 summarizes the assumed air mass flow coefficient and flow exponent for the cracks and windows/doors for a "medium" crack template.

Table 4-1: Crack characteristics for the building's elements

Building element	**Air mass flow coefficient (C_Q) [kg/s.m^2] at 1Pa**	**Flow exponent(n) [-]**
Cracks		
External walls	0.0001	0.70
Internal walls	0.003	0.75
Internal floors	0.0009	0.70
External floors	0.0007	1.0
Roof	0.0001	0.70
Windows / Doors when they are closed		
External windows	0.00014	0.65
External doors	0.0014	0.65
Internal doors	0.02	0.60
External vents	0.008	0.66

Discharge coefficient (C_d)

A very important parameter when calculating the required effective area of windows or dampers is the discharge coefficient, C_d. The C_d is a function of the shape of windows/dampers, the thermodynamic properties of the air (air temperature, Reynolds number), wind speed and direction. Considering the C_d as a constant number throughout the simulation is not accurate. For single-side ventilation, typical values that can be found in the literature are $C_d = 0.25$, while for cross ventilation C_d ranges from 0.6 to 0.9 (CIBSE, 1986; Awbi, 2003; CIBSE A, 2005; CIBSE Guide A, 2015). Heiselberg and Sandberg, (2006) have shown that for windows' areas from 0.5 to 0.6 m^2 the C_d can vary from 0.6-0.8 while for smaller areas C_d can ranges from 0.8-1.0. For the dampers, C_d ranges from 0.4-0.6 depending on the louvres' geometric properties such as the shape and angle of the metallic louvres. In the UK typical rainproof louvres with and angle of 45° have a C_d from 0.3-0.5 (CIBSE, 1986).

To summarize, the assumed values for the C_d for the windows and dampers are presented in table 4-2.

Table 4-2: Discharge coefficient for different window and damper opening factors.

Window-single sided			
Opening factor	**0**	**0.5**	**1.0**
C_d	0	0.25	0.25
Window-cross ventilation			
Opening factor	**0**	**0.5**	**1.0**
C_d	0	0.6	0.6
Damper			
Opening factor	**0**	**0.5**	**1.0**
C_d	0	0.3	0.3

4.2.2. Ceiling fan

The adjustment of the indoor air velocity using ceiling fans is one of the most important behavioural adaption mechanisms for the occupants of a building. Air movement, especially in warm and humid climates, can improve the thermal sensation of occupants (Zhai *et al.*, 2015). ASHRAE-Standard-55, (2013) has incorporated in their documentation, the advantages of the use of a ceiling fan by increasing the acceptable range for operative air temperature by 1.2°C, 1.8°C and 2.2°C for internal air velocities of 0.6m/s, 0.9m/s and 1.2m/s respectively. It is assumed that the ceiling fan can operate in 3 fan speed levels: i) Fan speed level 1 that results in internal air velocity of 0.6m/s; ii) Fan speed level 2 that results in internal air velocity of 0.9m/s and iii) Fan speed level 3 that results in internal air velocity of 1.2m/s.

When ceiling fans operated, the setpoint temperature for natural ventilation was increased based on the air velocity. When the control algorithms "decided" that ceiling fans should operate the setpoint temperature for cooling was increased based on the fan speed. For instance, if the cooling setpoint without the use of ceiling fan was found to be $T_{i,CSP}$=29°C and the control algorithm decided to operate the ceiling fan at Fan speed level 1 then the control algorithm adjusts the cooling setpoint accordingly to include the effect of the air movement so the new setpoint temperature is $T_{i,CSP}$=(29+1.2)°C=30.2°C.

4.2.3. Mechanical systems

The extreme weather conditions during some periods of the year can jeopardise the use of natural ventilation solutions. The use of mechanical systems was deemed essential to maintain acceptable internal conditions throughout the year.

Two different mechanical cooling/heating systems have been used for this study, in conjunction with the standalone humidification, dehumidification and mechanical ventilation systems. The mechanical systems that were studied are: i) Split AC units; ii) Evaporative cooler; iii) Outdoor Air Unit and iv) Dehumidification Unit.

i) Split AC Units

Split AC units are the most common cooling/heating mechanical systems that are used in India, and projections show that sales of split units growing with a rate of 10-15% annually. In EnergyPlus, the ZoneHVAC:Packaged Terminal Air Conditioner object was used to model a typical residential split unit. It consists of an outdoor mixer, direct expansion (DX) cooling coil, a heating coil and a fan that can be placed either between the outdoor mixer and the cooling coil (blow through a fan), see figure 4-2, or it can be placed after the heating coil (draw-through). For this research, an electric heating coil was used at it is more common for residential cases. The cooling coil was assumed to be a single-speed DX coil, while the fan was modelled as a constant volume fan.

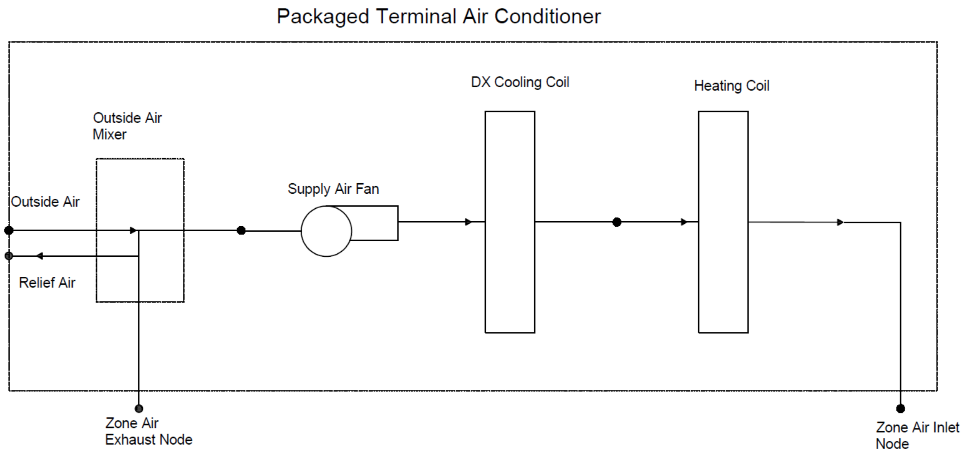


Figure 4-2: Schematic diagram of a packaged terminal air conditioner with a blow through fan

The control of the split unit was done in two stages. Initially, the availability of the unit was controlled based on the cooling and heating setpoint values. If the unit was available, then the second stage of the control was done in each component of the ZoneHVAC:Packaged Terminal Air Conditioner object. In section 4.2.1.3 the control algorithm for the split unit is presented.

The control of the cooling coil was implemented by parameterizing the total cooling capacity of the unit based on the wet-bulb temperature of the air entering the cooling coil and the dry-bulb temperature of the air entering the air-cooled condenser coil, see Eq. 4-8.

$$z = 0.942587793 + 0.009543347 * x + 0.000683770 * x^2 - 0.011042676 * y + 0.000005249 * y^2 - 0.000009720 * xy \quad \text{Eq. 4-8}$$

Additionally, a cooling capacity curve modifier was used with the independent variable being the ratio of the actual airflow rate to the rated airflow rate, see Eq. 4-9 (DOE, 2018).

$$z = 0.8 + 0.2 * x + 0 * x^2 \quad \text{Eq. 4-9}$$

For the heating coil, a simpler control was implement based on the internal air temperature and the heating setpoint. When the internal temperature was below the heating setpoint, the mechanical heating was turned on. The fan was scheduled to be available based on the availability of the ZoneHVAC:Packaged Terminal Air Conditioner unit.

ii) ***Evaporative Cooler***

Evaporative coolers are commonly used in Indian residencies as the main cooling system. The control of the evaporative cooler requires the design of advanced control algorithms. The control algorithms are detailed in Section 4.2.1.44.3.1.

To model the evaporative cooler the following objects were used: i) *AIRLOOPHVAC* object is an air-loop object that is used to model the air distribution from the outside to the zone, either a single or multiple thermal zone; ii) the cooling pad, *EvaporativeCooler:Direct:CelDekPad* or *EvaporativeCooler:Direct:ResearchSpecial* where the heat transfer occurs between the cooling media and the air and iii) the fan, *Fan:OnOff* or *Fan:VariableVolume*, that is used to drive the air so it can pass through the cooling media. The evaporative cooler cannot be controlled to perform partially, it can be either 100% on for the whole timestep or completely off. Hence, a variable volume fan was used to modulate the airflow that passes at the evaporative cooler and to meet the cooling loads in a similar way as in the other HVAC systems. For this research, advanced control algorithms were developed for the variable speed fan (Angelopoulos *et al.*, 2019) can be found in section 4.2.1.4. This control approach uses the predicted zone load for the current timestep and adjusts the fan speed to meet the sensible cooling load, if possible. This method provides great flexibility since the design engineer can specify which the minimum cooling load that will trigger the unit to operate.

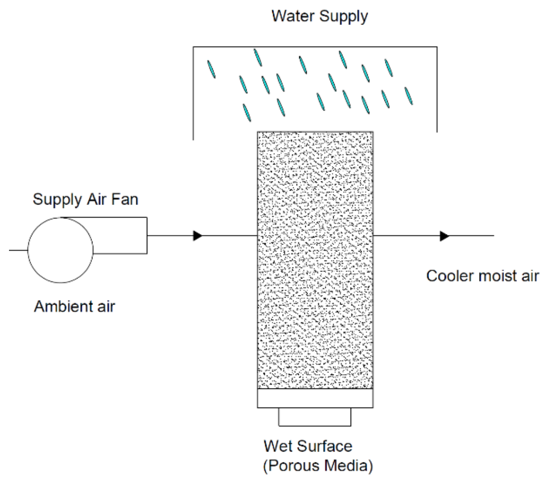


Figure 4-3: Schematic diagram of a direct evaporative cooler

For the evaporative cooler, a direct stage was used with a rigid evaporative pad and water that was recirculated. The air after exiting the supply fan passed through the pad where an almost adiabatic saturation of the air occurred, see figure 4-3. The area of the pad is an essential parameter to achieve a lower temperature. The exiting air of the pad had a lower dry-bulb temperature but also higher humidity. The thermodynamic process that occurs at the water pad, is assumed to be a simultaneous heat and mass transfer. Since it’s an adiabatic process the enthalpy remains constant and hence the wet-bulb temperature of the exiting air is equal to the entering air.

To maintain thermally comfortable conditions when heating was required, a standalone convective electric heater was used. The control of this unit was based on the heating setpoint point. When the internal temperature was below the heating setpoint, the mechanical heating was turned on. As mentioned, the use of an evaporative cooler increases the levels of humidity inside the zone and to eliminate discomfort condition due to high levels of humidity, it is essential to use a dehumidification system.

iii) ***Outdoor Air Unit***

A ZoneHVAC:OutdoorAirUnit object was used in EnergyPlus to provide additional outside air, when needed, to improve the indoor air quality. The purpose of the unit was not to condition the air(heating/cooling) hence it consisted of supply and an exhaust constant volume fan. This unit was controlled by checking whether additional outside air was needed in the space. If yes, the fans were switched on. Effective control of this unit was crucial because bringing more outside than the required could have a negative impact on the internal air temperature and humidity levels.

i) ***Dehumidification Unit***

Despite the use of systems that can increase the humidity levels of the thermal zone (evaporative cooler), the extreme climatic conditions, especially in the humid climatic zones, requires the use of a standalone dehumidification system to maintain comfortable internal environments. A ZoneHVAC:Dehumidifier: DX object in EnergyPlus was used. The control of this unit is similar to the DX cooling coil. A biquadratic performance curve, Eq. 4-10, was used to modulate the water removal based on the dry-bulb temperature and relative humidity of the air entering the dehumidifier.

The system's nodes were connected to the nodes before the air reached the thermal zone to ensure that the air reaching the thermal zone had the minimum possible relative humidity levels.

$$z = -2.724878664080 + 0.100711983591 * x - 0.000990538285 * x^2 0.050053043874 * y - 0.000203629282 * y^2 - 0.000341750531 * xy \qquad \text{Eq. 4-10}$$

4.3. Control algorithms for mixed-mode buildings

The aim of this research was to develop suitable control algorithms for the effective operation of NV and MM cooling strategies in residential apartment buildings in India. This section describes the flexible control algorithms developed under this research project. These control algorithms are focused on the ventilation and cooling scenarios identified in section 3.5 – table 3-2.

4.3.1. List of Flexible Control Strategies

The control strategies for mixed-mode residential buildings are developed as a set of stand-alone yet interconnected control modules. A "***rule-based***" approach was used for all the developed control algorithms. These control modules incorporate building and system characteristics, and operation patterns as control inputs for the envelope and system components. Applicable control modules should be selected and applied based on the installed envelope and system components in buildings. The following control strategies are developed under this project:

- Building Operation Mode – “*Master Control”*: This control module determines a suitable operation mode of the building based on defined setpoints as well as prevalent indoor and outdoor conditions.
- Operation of Natural Ventilation Components: This control module determines the operation of ventilation components, such as windows and dampers, in the building based on ventilation setpoints as well as prevalent indoor and outdoor conditions.
- Operation of Split Air Conditioning Devices: This control module determines the operation of split air conditioning devices based on defined cooling setpoints as well as prevalent indoor conditions.
- Operation of Evaporative Cooling Devices: This control module determines the operation of evaporative cooling devices in the building based on defined cooling setpoints as well as prevalent indoor conditions.

4.3.1.1. “Master” control algorithm

The “*master”* control algorithm, figure 4-4, determines the suitable operation mode of the building based on defined setpoint as well as prevalent indoor and outdoor conditions. Depending on the wind speed and internal air temperature the “master” control algorithm uses the sub-control algorithms such as the nodes D and E to control the building. For the maximum wind speed the upper limit of 18m/s was used to prevent damages to the windows (DOE, 2018).

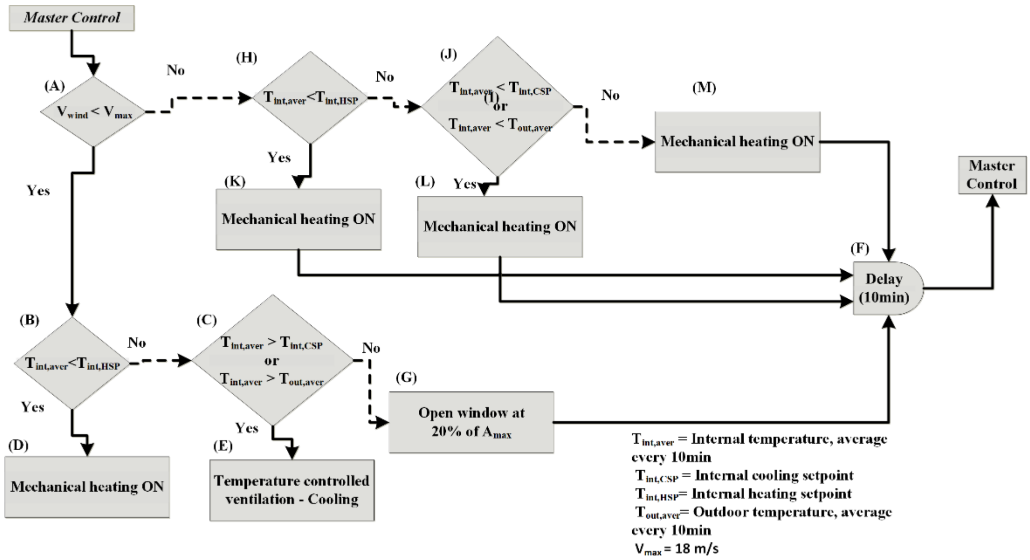


Figure 4-4 Master control algorithm

When none of the blocks B and C are satisfied, the algorithm opens the windows to the minimum required position in order to achieve the minimum ventilation rate of 0.6L/sm^2 as suggested by ASHRAE-ADDENDA, (2006). A delay function of 10 minutes is used to avoid changes in the control strategies such as a change in operation mode from NV to mechanical cooling/heating or opening/closing of windows for a sudden fluctuation in the temperature. There is no occupancy status check at the "master" control algorithm. This check occurs within the sub-control algorithms of blocks D and E.

4.3.1.2. Operation of natural ventilation components

Figure 4-5 to 4-9 show the control algorithms for the cases when nighttime restriction was applied in the windows. The same control algorithms were used for the cases with no window restriction with an exception that block H had been removed, see Appendix B: Control algorithms for no window restriction

Figure 4-5 highlights the control logic for ***VCS 1*** and with window opening restriction during night ventilation. When cooling is required during occupied hours, natural ventilation is preferred over the mechanical cooling and when there is no cooling potential from natural ventilation, mechanical cooling is switched on. If the zone's internal air temperature rises above the cooling setpoint, then mechanical cooling is operating. The natural ventilation is preferred as long as the internal air temperature is higher than the external, the cooling setpoint is higher than the outdoor air temperature to ensure that there is a thermodynamic advantage for additional ventilation. When natural ventilation is required, the opening of the windows will be modulated according to the indoor and outdoor conditions. It is assumed that the difference between the internal and external air temperatures as well the internal air temperature and the internal cooling setpoint should be higher than the selected deadband temperature of 0.5°C in order to avoid any unnecessary use of the mechanical cooling systems or operation of the windows (DesignBuilder, 2017).

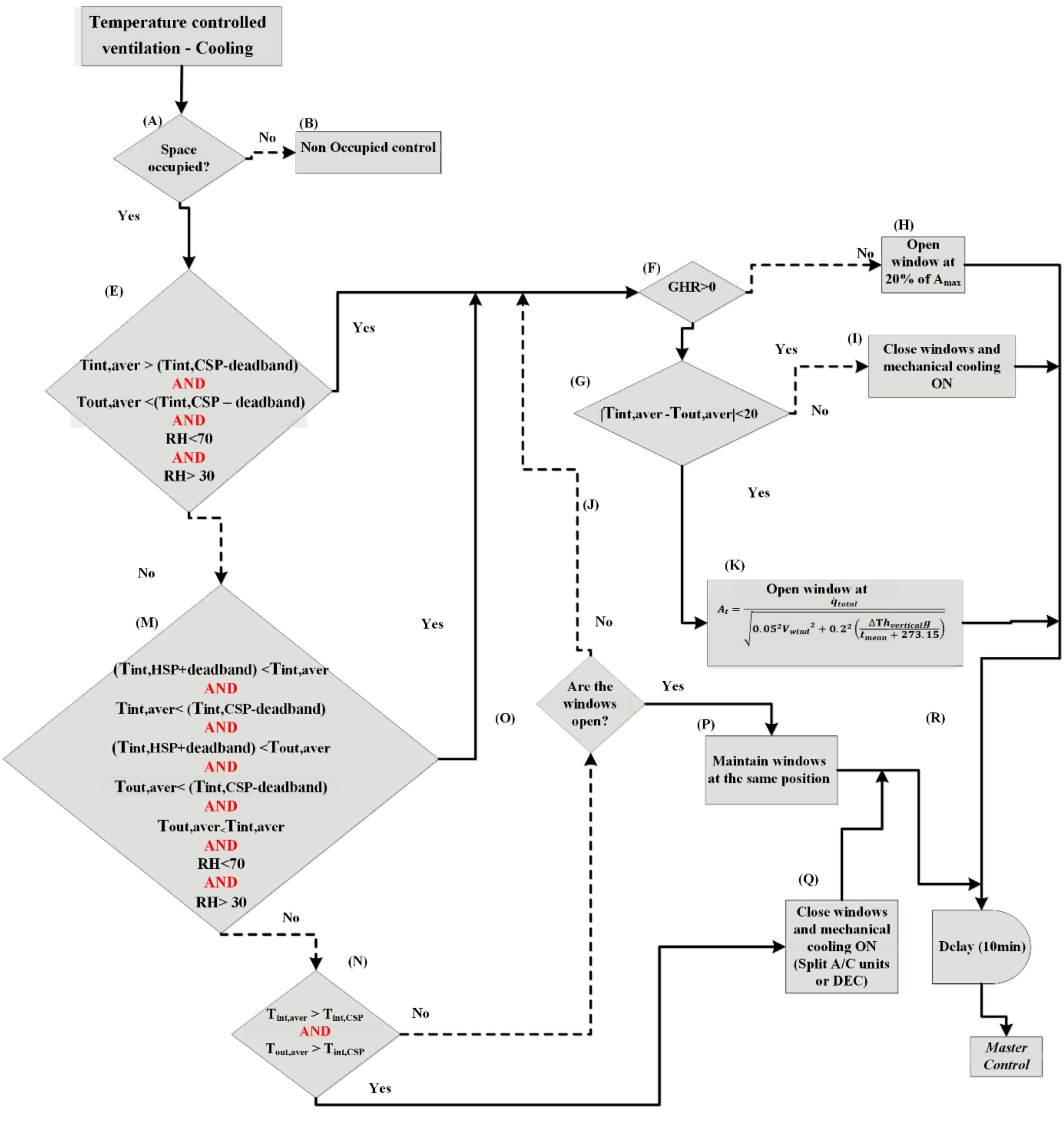
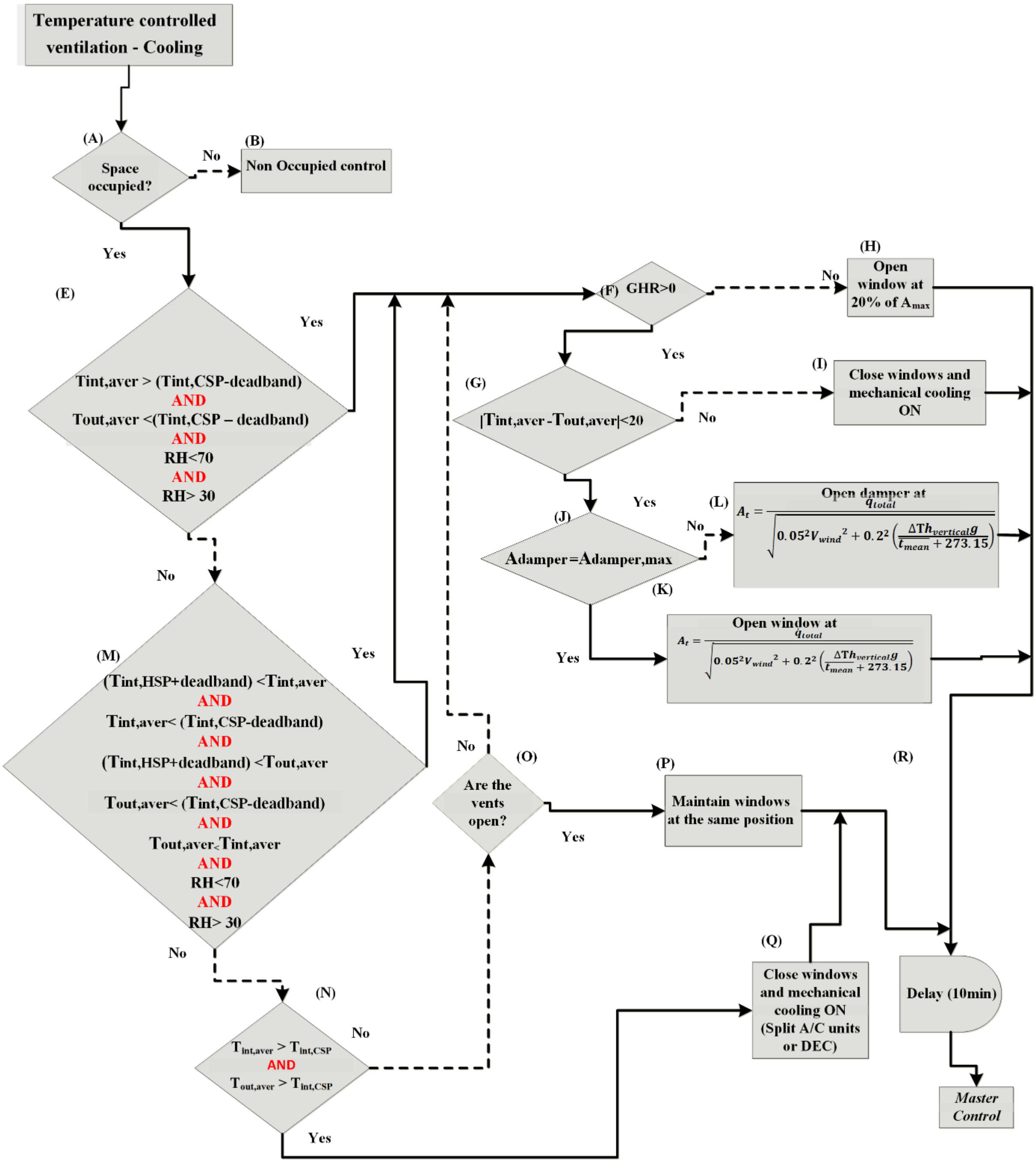


Figure 4-5: Control algorithm for ventilation and cooling scenario (VCS) 1

Figure 4-6 highlights the control logic for ***VCS 2.*** In this case, the control logic of the windows is the same as described above with an additional check on the dampers. When natural ventilation is needed, priority is given to operate the dampers. Only when the dampers were fully opening, windows were opened. The methodology to calculate the opening area of the dampers is identical to the one for windows.



$$A_t = \frac{q_{total}}{\sqrt{0.05^2 V_{wind}{}^2 + 0.2^2 \left(\frac{\Delta Th_{vertical} g}{t_{mean} + 273.15}\right)}}$$

Figure 4-6: Control algorithm for ventilation and cooling scenario (VCS) 2

Figure 4-7 highlights the control algorithm for the operation of the air circulation devices (ceiling fans) included in the control algorithm (***VCS 3***). The ceiling fan is assumed to operate when the internal air operative temperature is above 28°C (Manu *et al.*, 2014). When the ceiling fan is switched on, the setpoint temperature for the operation of the cooling systems is set to be higher than the value proposed by the adaptive theory in order to take into account the positive impact of air movement to occupants' the thermal sensation, as discussed in Section 4.1.2. The control of the ceiling fan is set to ON/OFF.

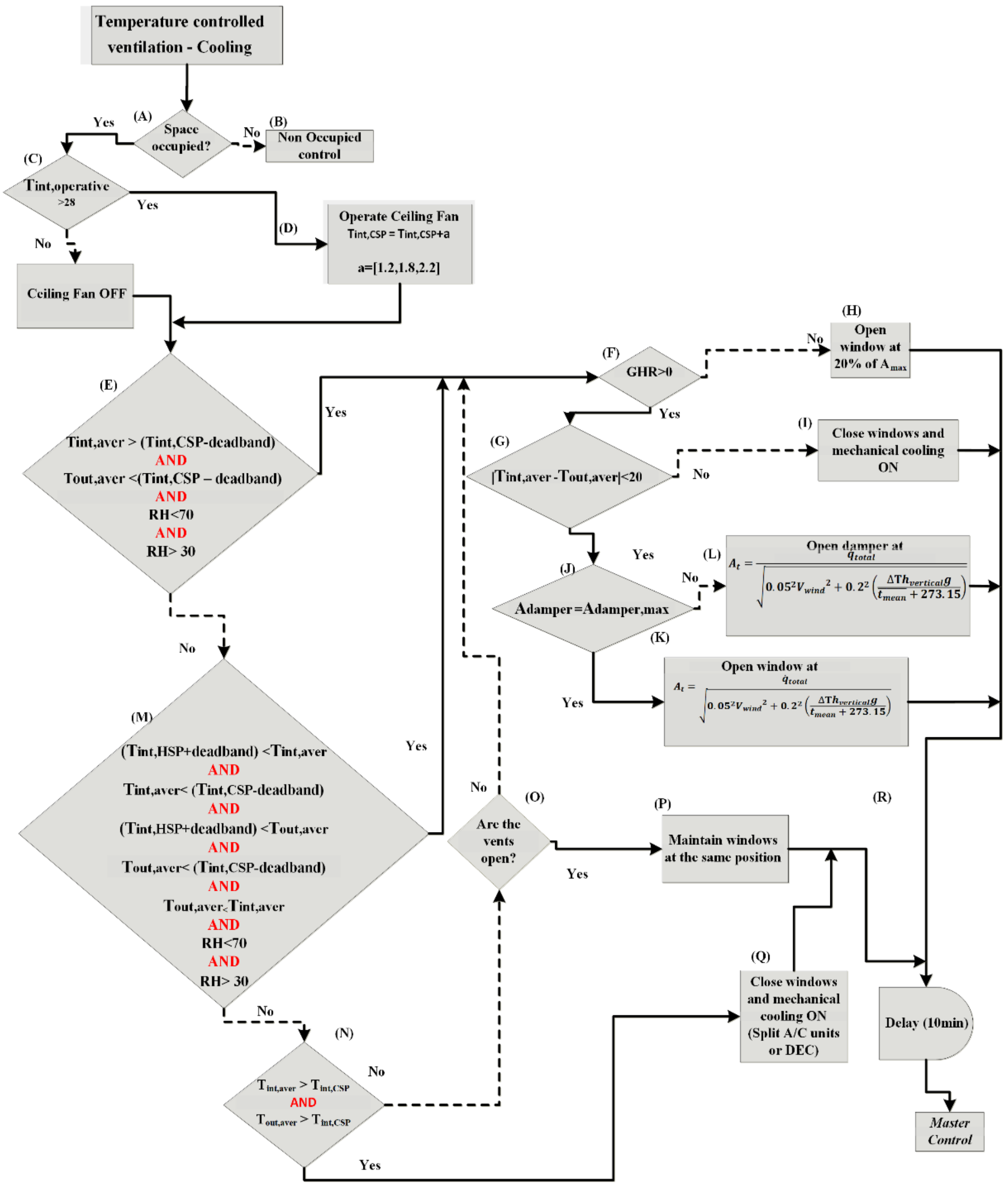


Figure 4-7: Control algorithm for ventilation and cooling scenario (VCS) 3

Finally, figure 4-8 presents the control algorithm for ***VCS 4***. The control approach for this scenario includes an additional check on the levels of internal relative humidity and when above the setpoint value the mechanical dehumidifier is activated. The rest of the control logic remains the same as discussed above.

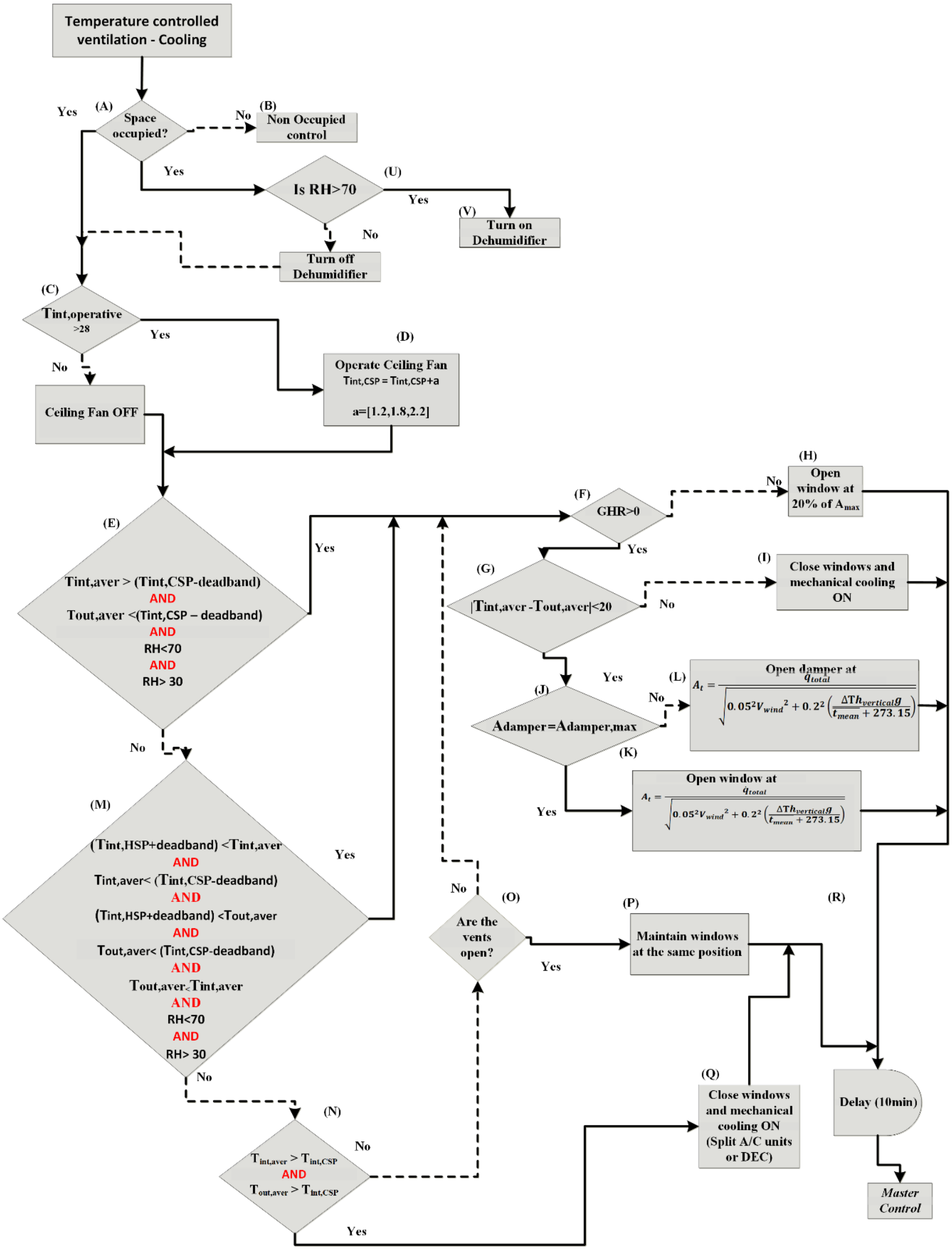


Figure 4-8: Control algorithm for ventilation and cooling scenario (VCS) 4

Figure 4-9 shows the control algorithm for ***unoccupied periods***. A check is made whether the internal air temperature and outdoor air temperature are within the setpoint temperatures. If they are, then the windows operate in order to take advantage of the cooler outdoor air and pre-cool the building for when it is occupied. However, due to safety reasons, the opening of the windows is restricted to 20% of their maximum openable area.

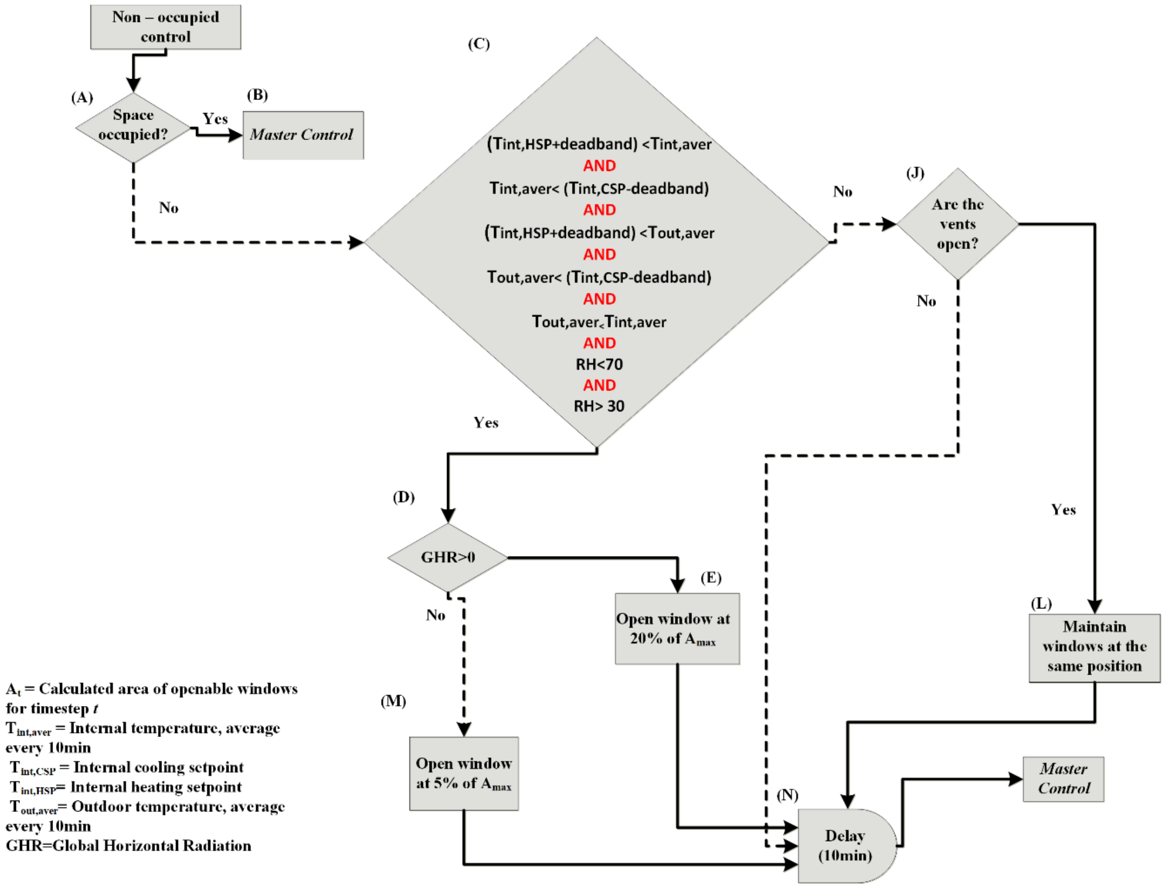


Figure 4-9: Control algorithm for non-occupied periods

It should be mentioned that the “master” control algorithm (figure 4-4), as well as the control algorithm for the unoccupied periods (figure 4-9), remained the same in all the strategies. Also, during unoccupied hours, the ceiling fan did not switch on, since the air velocity inside the room provides better comfort sensation to occupants.

4.3.1.3. Operation of split unit

The control approach of the fixed speed split A/C unit, see figure 4-10, is based on predefined cooling setpoints as well as prevalent indoor and outdoor conditions. Since a fixed speed split unit was used the control approach was based on an ON/OFF approach for both the fan and the cooling coil. The supply fan and the coiling coil were modelled to run simultaneously. At each time-step, the sensible cooling load, $\dot{Q}_{COOLING\ LOAD}$, to meet the cooling setpoint was calculated in Modelica. To eliminate the short cycling of the split unit, a deadband of 0.5°C is used. The algorithm, see figure 4-10, uses as inputs the most recent value of the zone air temperature ($T_{int,aver}$) and the current cooling setpoint(T_{CSP}). The disadvantage of this strategy is that the split unit is operated at full load at periods when there is no need for this amount of cooling. To eliminate the use of the split at full load when there is no need, the control of the cooling coil was implemented by parameterizing the total cooling capacity of the unit based on the wet-bulb temperature of the air entering the cooling coil and the dry-bulb temperature of the air entering the air-cooled condenser coil, see section 4.1.3.

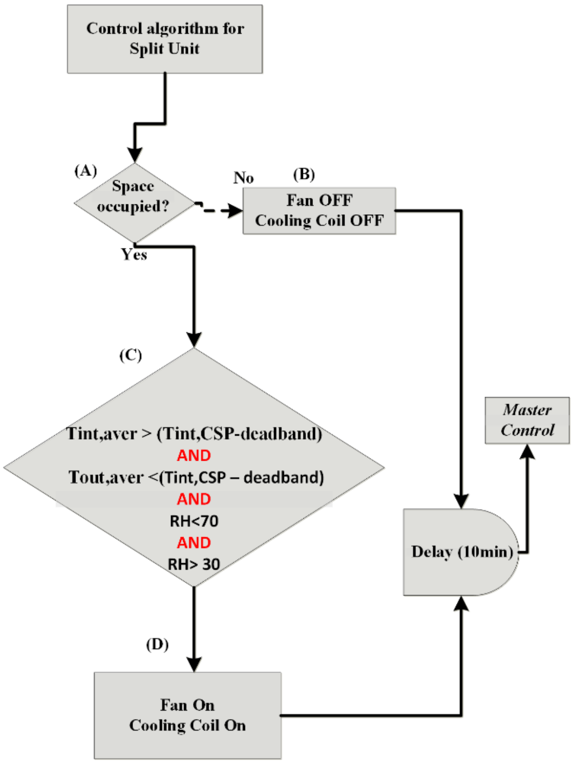


Figure 4-10: Control algorithm for split unit

4.3.1.4. Operation of direct evaporative coolers

The “basic” control algorithm, figure 4-11, uses a similar methodology to the majority of the DTM tools. The control of the DEC systems is based on the On/Off approach of the evaporative cooler and the fan. When there is a need for cooling, the control algorithms turn on the evaporative cooler without any extra control over the fan speed. The fan operates at its maximum airflow. The decision to operate or not the DEC takes place at the beginning of each timestep. When the unit operates, it runs at the full design airflow rate regardless of the required amount for cooling. This control method might not be optimal, but it is similar to how most thermostats operate in real applications. To eliminate the short cycling of the DEC unit, a deadband of 0.5°C is used. The control algorithm, see figure 4-11, uses as inputs the most recent value of the zone air temperature ($T_{int,aver}$) and the current cooling setpoint(T_{CSP}). This approach has a disadvantage of operating the DEC unit at full load at periods when there is no need for this amount of cooling. However, this is the most common control approach found in DTM tools.

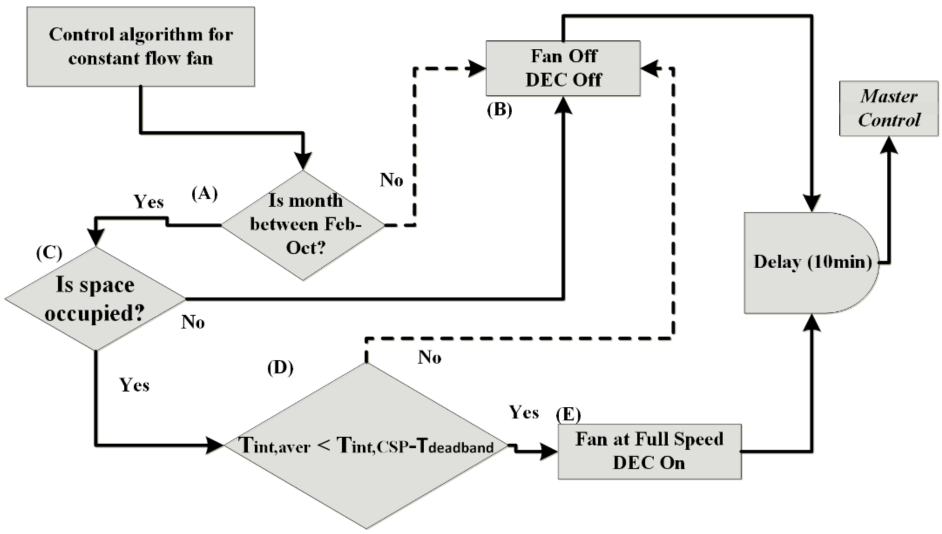


Figure 4-11: Control algorithm for DEC unit and a constant flow fan

The “advanced” control algorithm, see figure 4-12, uses a more detailed approach to control the DEC unit and the fan (Angelopoulos *et al.*, 2019). For the “advanced” control algorithms a variable speed fan, with 3 fan speed levels (33%,66% and 100% of its maximum airflow), was used instead of a constant volume fan that was used in the previous cases. This control method uses a more sophisticated approach to modulate the fan speed based on the required cooling load at each timestep. This value is then compared with the maximum value of the cooling load that the DEC unit can provide. To avoid running the DEC unit at periods when there is no need for cooling, a deadband temperature of 0.5°C value for the temperature is used. This control approach eliminates the cases where the unit is turned on and off constantly. Then the sensible cooling provided by the unit $\dot{Q}_{FULL,OUTPUT}$ is calculated and compared against the cooling load of the zone $\dot{Q}_{COOLING\ LOAD}$ as presented in figure 4-12.

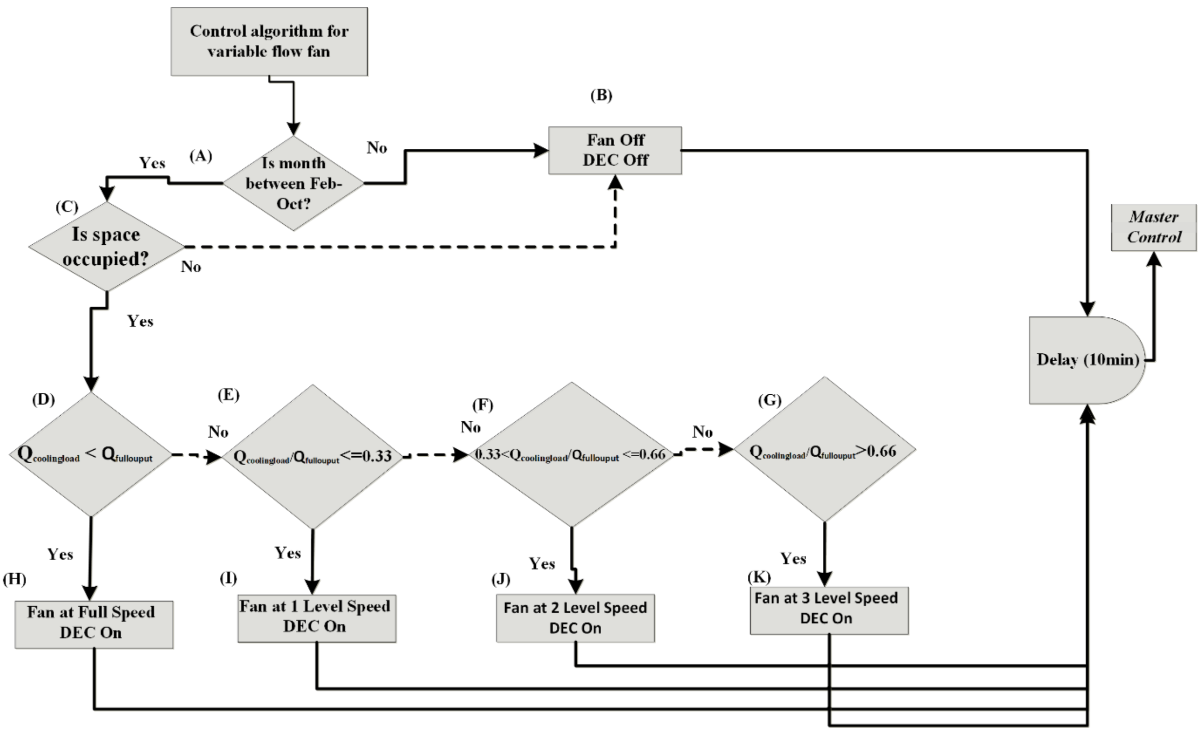


Figure 4-12: Control algorithm for DEC unit with a variable flow fan

4.4. IMAC models' applicability

Adaptive control algorithms have been used for this research to determine the dynamic cooling and heating setpoints. These algorithms aimed to provide an alternative approach to the static thresholds temperatures that traditionally used for fully mechanical conditioned or mixed-mode buildings. The assessment of the thermal comfort was based on the adaptive comfort limits proposed by the IMAC model for 90% acceptability ranges, see Eq.4-11 (Manu *et al.*, 2016).

The IMAC model was developed based on the data from thermal comfort surveys of office buildings across India. Although this work is focused on residential buildings only, the lack of any other comfort model for the Indian climate makes the IMAC model the most appropriate to use. Due to its relevance in the Indian context, it has been recently integrated into the National Building Code of India (Standards, 2016). This comfort model suggests a broader acceptability temperature range compared to the ASHRAE 55 comfort model. In this research the adaptive comfort model for the mixed-mode building proposed by IMAC was used:

$$T_{comf} = 0.28T_{om} + 17.87 \quad \text{Eq. 4-11}$$

Where the outdoor temperature T_{om} is a 30-day running mean air-temperature ranging from 13-38.5°C. Similarly, to most of the thermal comfort models, 80 and 90% acceptability limits are defined in the model as shown in the figure below (figure 4-13) and the temperature bands are i) 90 % band = ±3.5°C, and ii) 80 % band = ±5.9°C.

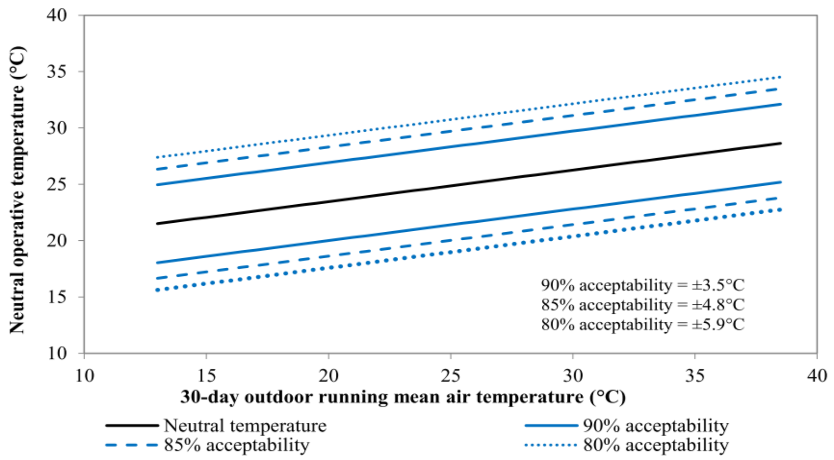


Figure 4-13: Acceptable operative temperature ranges for mixed-mode buildings from IMAC model (Excerpt from Manu et al., (2016)).

The adaptive theory is based on the temperature limits as the only matrix to assess whether operative temperatures are within limits. However, as the literature suggests, the external relative humidity could be an equally important parameter when assessing the thermal comfort conditions of space.

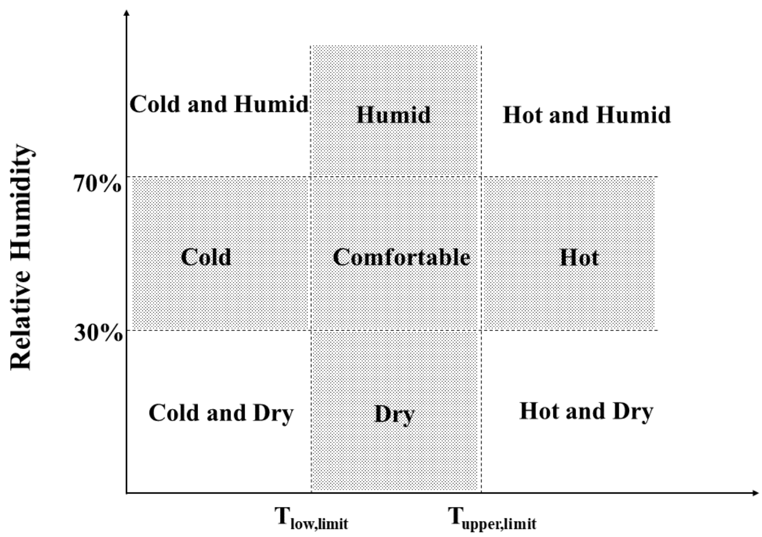


Figure 4-14: Categorization of annual hours based on temperature and humidity thresholds

Hence, based on the lower and upper temperature limits, derived by the IMAC comfort model, as well as on the threshold values of relative humidity (RH), the hours of the year could be categorized into nine types of environmental conditions, see figure 4-14.

T_L = Lower limit of 80% acceptability band, and T_U = Upper limit of 80% acceptability band. The upper and lower limits of temperature will change for each day and each location since they are derived from the 30-day running mean of outdoor dry-bulb temperature. The 80% acceptability band was used to maximize the potential of natural ventilation solutions. The 30-70% acceptability range is often considered in the literature as an appropriate range to maintain acceptable indoor conditions (Berglund, 1998; Arens *et al.*, 1999; ASHRAE-Standard-55, 2013). The environmental conditions specific to each location for defining the operation modes based on temperature and humidity thresholds are shown in table 4-3.

Table 4-3: Conditions for operation modes per the temperature (based on the IMAC models) and RH thresholds

Operation mode	Temperature thresholds	RH thresholds
Natural Ventilation	Within the 80% acceptability band	30% -70%
Heating	< Lower limit of 80% acceptability	30% -70%
Heating and Dehumidification	< Lower limit of 80% acceptability	>70%
Cooling	> Upper limit of 80% acceptability	30% -70%
Cooling and Dehumidification	> Upper limit of 80% acceptability	>70%
Dehumidification	Within the 80% acceptability band	>70%
Humidification	Within the 80% acceptability band	<30%
Heating and Humidification	< Lower limit of 80% acceptability	<30%
Cooling and Humidification	> Upper limit of 80% acceptability	<30%

4.5. Summary

This chapter illustrates the developed control algorithms for mixed-mode residential buildings. These control algorithms were used later on to evaluate the energy saving potentials that can be achieved in hot climates. The control algorithms were presented in the format of flowcharts that were converted into the Modelica language for the simulations. Finally, the applicability study of the IMAC model showed that is feasible to use mixed-mode buildings in the Indian climate during periods of the year using passive systems.

Chapter 5. Validation Analysis

5.1. Overview

In order to ensure rigorous research, new concepts must first be tested in a controlled environment; only then can the principles be applied to full-scale buildings. In this chapter, the results of the initial testing and the validation of the co-simulations are presented. Detailed measurements of air temperatures and CO_2 concentrations have been collected in the environmental chamber and used to validate the experimental set-up (presented in Chapter 4) modelled in EnergyPlus (DOE, 2018) and the proposed control algorithms modelled in Dymola (DYMOLA, 2018). The validation analysis is of paramount importance for this research as it provides solid evidence of the validity of the developed control algorithms and the simulation approach. The use of a controlled environment, however, is a very idealistic case which might lead to over-predicting the performance of the control algorithms since all the parameters are well controlled and it does not represent a real case. However, this controlled approach to validation, for new concepts, is far superior to attempting validation using 'noisy' field data which possesses many inherent uncertainties.

Figure 5-1 shows the steps of the validation analysis and which part of this research were validated.

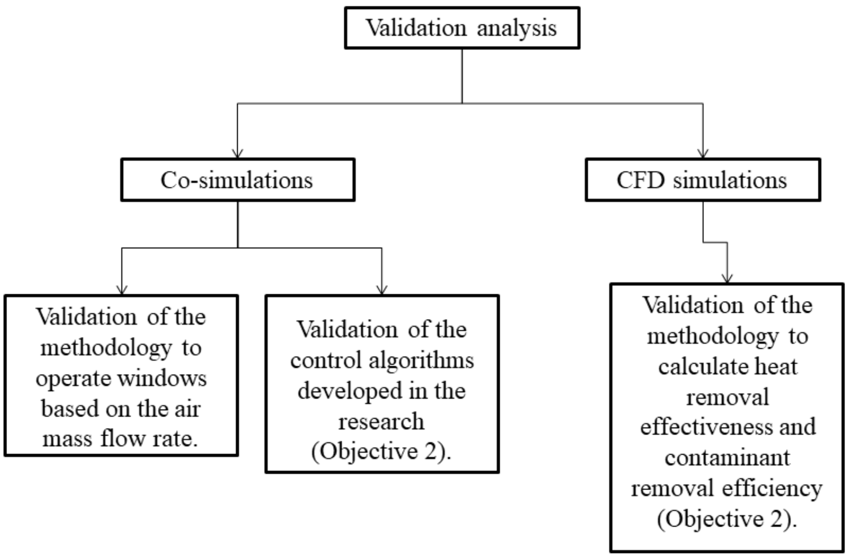


Figure 5-1: Flow chart of validation analysis

5.2. Uncertainty of measurements and analytical models

The use of the CO_2 to calculate the ventilation rate is a method that has been previously used by others. To calculate the ventilation rates by using the CO_2 concentration measurements the following formula was used (Etheridge and Sandberg, 1996; Nikolopoulos *et al.*, 2012) :

$$q_{measurement} = V \frac{\ln\left(\frac{C_S - C_0}{C_s - C_1}\right)}{\Delta T} \qquad \text{Eq. 5-1}$$

where $q_{measurement}$ is the volume flow rate[m^3/s]; V is the volume of the room [m^3]; ΔT=period between C_0 and C_1 measurements [s]; C_0 and C_1concentrations measured at the start and the end of the observation period respectively[ppm]; and C_s the steady-state concentration [ppm].

Furthermore, another important parameter to assess in this validation study was the ventilation effectiveness of the natural and mixed-mode ventilation systems. In the literature, there are different definitions of the ventilation effectiveness of a system. The following are the most commonly used metrics in the literature (Cao *et al.*, 2014):

I. The Contaminant Removal effectiveness (CRE) metric is used to assess the ability of a system to deliver outside air into a space and remove contaminated air from the space. The CRE for a space is given by the Eq 1 below (Mundt *et al.*, 2004):

$$\langle \varepsilon \rangle = \frac{C_{exh} - C_S}{\langle C \rangle - C_S} \qquad \text{Eq. 5-2}$$

where, C_{exh} is the concentration of the contaminants at exhaust vent; C_S is the concentration of the contaminants at the supply vent and $\langle C \rangle$ is the average concentration of the contaminants at the room. Sandberg (Sandberg, 1981) expanded

Eq 1 and calculated the steady-state local contaminant removal efficiency at a point j (Coffey and Hunt, 2007):

$$\varepsilon_j = \frac{C_{exh} - C_S}{C_j - C_S} \qquad \text{Eq. 5-3}$$

where, C_{exh} and C_S is the concentration of the contaminants at the exhaust vent and the supply vent respectively and C_j is the steady-state concentration of the contaminant at the point j.

II. The heat removal ventilation effectiveness (HRE) (Cao *et al.*, 2014) is a similar metric to the CRE but it assesses the effectiveness of the system to remove heat from the space and the mathematical expression is the following (Cao *et al.*, 2014):

$$\varepsilon_t = \frac{T_{exh} - T_S}{T_{oc} - T_S} \qquad \text{Eq. 5-4}$$

where, T_{exh} is the temperature of the exhaust air; T_S is the temperature of the supply air and T_{oc} is the temperature in the occupied zone.

The collection of experimental data is accompanied by uncertainties. These uncertainties introduce an error in the results where many times can alter the significance of the data. To quantify the uncertainty of Eq. 5-1 the following equations were used:

Sums and differences $\Delta Z=[\Delta A^2+\Delta B^2]^{0.5}$ Eq. 5-5

Products and Quotients $\Delta Z/Z=[(\Delta A/A)^2+(\Delta B/B)^2]^{0.5}$ Eq. 5-6

Where ΔZ is the final error and ΔA & ΔB are the errors associated with the values A & B respectively.

Further, table 5-2 summarizes the accuracy of the sensors that were used for this experiment as provided by the manufacturers. To quantify the discrepancies in the results two statistical indices were used to provide more evidence of the validation method:

i. Mean Bias Error (MBE): captures the mean difference between the measured and the simulated data and is a good indicator of the overall bias in the model (Coakley *et al.*, 2014). A limitation of this method is that the positive and negative errors cancel each other when summed which might lead to positive bias compensates for negative bias.

$$MBE\ [\%] = \frac{\sum_{i=1}^{N_P}(m_i - s_i)}{\sum_{i=1}^{N_P}(m_i)} \qquad \text{Eq. 5-7}$$

where,

m_i is the measured data; s_i is the simulated data; and N_P is the number of data points.

ii. Coefficient of Variation of Root Mean Square Error (CVRMSE): captures the error between the measured and simulated data without the cancellation effect described in MBE (Granderson and Price, 2013).

$$CVRMSE\ [\%] = \frac{\sqrt{\frac{\sum_{i=1}^{N_P}(m_i - s_i)^2}{N_P}}}{\bar{m}} \qquad \text{Eq. 5-8}$$

where,

m_i, s_i and N_P are the same as defined in Eq. 5-7 and $\bar{m}$ = average value of the measured data.

The selection of CVRMSE was made because is the same metric that ASHRAE Guideline 14 (ASHRAE, 2014) uses to access the accuracy of the simulated data compared to the measured and also has been used by previous researchers as well (Coakley *et al.*, 2014; Granderson and Price, 2014).

As stated in ASHRAE 2009 (ASHRAE, 2009) there are three methods to assess the accuracy of the simulation models: the empirical validation; the analytical verification. In *empirical validation*, results from building simulation tools are compared to data measured from real buildings or experimental chambers (ASHRAE, 2009). In this method, the measured data are accompanied by errors/uncertainties that might impact the accuracy of the data. Hence it is important when presenting a study with empirical validation to highlight the error/uncertainties of the measured data. In *analytical verification*, the simulation results are compared to the results obtained by solving steady-state equations referred to in literature. The analytical verification is useful to compare new models to well-established ones (Clarke, 2011).

The simulation results are compared to the measured data from the experimental chamber. To provide further evidence of the accuracy of the results, analytical models were used to calculate the ventilation rates for the given input values and the values were compared to the measured ones. The use of two different validation methods provides undeniable evidence of the accuracy of the results and the validity of the developed control algorithms of mixed-mode buildings.

The analytical models for the cross-ventilation were used to calculate the expected ventilation rates for each scenario. The results from the analytical models were compared to the predicted values from the co-simulations in order to provide additional evidence of the validity of the outputs. The formula to calculate the ventilation rates for buoyancy-driven natural ventilation flows through multiple openings is (CIBSE Guide A, 2015):

$$q_{analytical} = C_d A_b \left(\frac{2\Delta\vartheta h_a g}{\bar{\vartheta} + 273.15}\right) \quad \text{Eq. 5-9}$$

where, $q_{analytical}$ is the volume flow rate[m^3/s]; C_d is the discharge coefficient of the openings [-]; A_b is the effective area of the openings for natural ventilation [m^2]; $\Delta\vartheta$ temperature difference between the internal and external temperatures[K]; g gravity force [m/s^2]; h_a height of the opening for natural ventilation [m]; and $\bar{\vartheta}$ is the mean temperature [°C].

5.3. Simulated scenarios for validation analysis

Few studies in the literature have focused on validation of natural ventilation. The majority of the studies used the internal air temperature as the main parameter to assess and validate the performance of the natural ventilation systems (Zhai *et al.*, 2011). However, there is no other study in the literature that validates sophisticated control algorithms for mixed-mode or for natural ventilation buildings. The validation study presented in this research is a unique approach to validate mixed-mode ventilation and cooling strategies and it can be used in the future as a reference point for other researchers to validate their cases.

For the validation study, average hourly data were used to compare the simulated results to the measured data and to the analytical models. Steady-state conditions (averaged hourly data) over dynamic conditions were selected as the main purpose of the validation study was to examine the validity of the developed control algorithms in this research. To avoid this time lag, which might cause discrepancies in the results, steady-state data were used since the focus of the validation study was to examine the validity of the proposed control algorithms and the co-simulation technique rather than the response time of the BMS system to send the signal and/or how fast/slow the windows/dampers operate.

By selecting hourly data, it was feasible to examine a variety of different outdoor and indoor conditions and windows/dampers configurations. Specifically, the controls of the windows/dampers and ceiling fans in the experimental chamber were made via the BMS of the facility. In real-life applications, there is a small-time lag between the time that the control algorithm “decides” to operate the windows/dampers and the time that the BMS requires to send the signal to the windows/dampers. In the co-simulations on the other hand, the signal is transferred instantaneously from the control algorithm to the windows/dampers.

To simulate the steady-state conditions for the experiments and the co-simulations, the following cases were selected, see table 5-4. Co-simulations were performed using the geometry of the experimental chamber and the weather file of Ahmedabad, India, and the control algorithms see figure 3-12 in section 4.3.1.2. The results were analysed and clustered into groups based on the outside air temperature. The outside air temperature was grouped into 1°C interval and for each group, the number of hours the windows and/or dampers operated was counted (table 5-3). This analysis allowed the calculation of the frequency that each scenario occurred, and then the cases with the highest frequencies, highlighted cells in table 5-3, were selected to be simulated into the experimental chamber.

Table 5-1: Detailed analysis of co-simulation results. The highlighted cells (green colour) are those which were selected to be simulated into the experimental chamber. The hatched cells denote the scenarios that were not simulated into the chamber.

Scenarios				**1**	**2**	**3**	**4**		**5**		**6**			**7**	**8**	**9**	**Hours**
Window	Closed	Closed	Closed	80-100% Open	80-100% Open	60-80% Open	60-80% Open	40-60% Open	40-60% Open	40-60% Open	40-60% Open	Closed	Closed	Closed	Closed	Closed	
Damper	Closed	Closed	Closed	80-100% Open	80-100% Open	80-100% Open	80-100% Open	60-80% Open	80-100% Open	60-80% Open	80-100% Open	20-40% Open	40-60% Open	40-60% Open	60-80% Open	60-80% Open	
AC	ON	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	
Ceiling fan	ON	ON	OFF	ON	OFF	ON	OFF	OFF	OFF	ON	ON	ON	OFF	ON	OFF	ON	
Temperature [-) [°C]																	
9-10	2.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2.0
10-11	11.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	11.0
11-12	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
12-13	103.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	103.3
13-14	75.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	75.7
14-15	100.8	0.0	0.0	0.0	0.0	0.0	0.2	0.0	0.7	0.0	0.0	0.0	0.8	0.0	2.2	0.0	104.7
15-16	61.7	0.0	0.0	0.0	0.0	0.0	3.7	0.0	3.2	0.0	0.0	0.0	21.3	0.0	79.7	0.0	169.5
16-17	8.0	0.0	0.0	0.0	0.0	0.0	10.8	0.0	7.5	0.0	0.0	0.3	23.5	0.0	126.3	0.0	176.5
17-18	3.3	0.0	0.0	0.0	1.0	0.0	16.2	0.2	18.5	0.0	0.0	1.7	30.5	0.0	134.0	0.0	205.3
18-19	3.5	0.0	0.0	0.0	6.7	0.0	45.0	0.0	20.5	0.0	0.0	2.3	18.0	0.2	130.5	0.0	226.7
19-20	3.3	0.0	0.0	0.0	12.7	0.0	40.0	0.0	24.5	0.0	0.0	1.0	22.5	2.7	93.2	0.0	199.8
20-21	4.3	0.0	0.0	0.0	29.2	0.0	48.3	0.2	29.2	0.0	0.0	0.0	9.5	5.2	108.3	0.0	234.2
21-22	5.0	0.0	0.0	0.0	42.2	0.0	50.8	0.0	27.5	0.0	0.0	0.0	0.2	18.2	102.8	0.0	246.7
22-23	3.5	0.0	1.5	0.0	73.3	0.0	45.5	0.0	18.7	0.0	0.0	0.0	0.2	25.7	81.5	1.3	251.2

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23-24	9.2	0.0	14.5	0.0	77.2	0.0	37.2	0.0	21.0	0.0	0.0	0.0	0.0	24.2	37.2	26.7	247.0
24-25	21.7	0.0	25.5	0.0	122.2	0.0	53.2	0.0	26.5	0.0	2.0	0.0	0.0	8.5	15.7	42.0	317.2
25-26	39.2	0.0	59.3	0.0	172.7	5.2	136.8	0.0	21.5	1.2	13.2	0.0	0.0	0.7	3.8	63.7	517.2
26-27	78.0	0.0	79.0	0.7	236.5	30.3	161.7	0.0	11.0	1.8	44.3	0.0	0.0	0.0	1.7	38.0	683.0
27-28	136.7	0.0	40.7	15.5	270.2	101.2	104.2	0.0	0.8	0.7	25.7	0.0	0.0	0.0	0.0	3.7	699.2
28-29	200.3	1.7	13.7	101.7	244.0	106.3	30.2	0.0	0.0	0.0	3.2	0.0	0.0	0.0	0.0	0.0	701.0
29-30	207.7	13.2	21.8	226.3	80.5	66.8	0.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	616.7
30-31	313.2	27.5	16.0	222.7	5.5	12.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	597.3
31-32	423.7	4.8	2.0	51.3	0.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	482.0
32-33	374.3	0.5	0.0	6.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	381.2
33-34	294.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	294.8
34-35	239.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	239.3
35-36	221.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	221.7
36-37	193.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	193.8
37-38	149.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	149.0
38-39	131.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	131.8
39-40	105.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	105.8
40-41	83.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	83.3
41-42	64.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	64.7
42-43	29.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	29.2
43-44	2.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2.0
Hours	3704.8	47.7	274.0	624.5	1373.8	322.3	784.0	0.3	231.0	0.00	88.3	5.3	126.5	85.2	916.8	175.3	**8760.0**
	42.3%	0.5%	3.1%	7.1%	15.7%	3.7%	8.9%	0.0%	2.6%	0.0%	1.0%	0.1%	1.4%	1.0%	10.5%	2.0%	

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The next step was to analyse the simulated results and find all the hours of the year that the conditions were met. For instance, in scenario 1 (table 5-3) the simulated results were analysed to find all the hours of the year that the external air temperature was between 29-30°C and the window/dampers openings were 80-100% opened respectively. As explained in section 3.7, the experimental chamber is able to simulate buoyancy only driven natural ventilation/cooling cases only. Due to this, when selecting the values for the selected scenarios from the co-simulations that fulfils all the requirements, the time of the year with the lowest wind speed was selected. For this analysis, a tailor-made Python script was created to enable to sort the results and analyse them. This was an essential part of the analysis, as the co-simulations reported the results in timesteps, as such, in total 52560 results were reported for each variable.

Table 5-2: Experimental scenarios

Scenario	Date	Time	T_{out} [°C]	$T_{internal}$ [°C] 1h prior experiment	Window opening area [m^2]	Damper NV1 opening area [m^2]	Damper NV2 opening area [m^2]	Assumed air velocity [m/s]	Total internal heat load [W]
1	02-Jul	23:00-00:00	29.3	31.3	0.44	0.1707	0.1729	0.6	305
2	04-Aug	08:00-09:00	27.4	27.9	0.501	0.1829	0.1842	0.0	216
3	11-Mar	21:00-22:00	27.3	29.4	0.416	0.1703	0.1726	0.9	324
4	14-May	03:00-04:00	27.3	29.4	0.398	0.1698	0.172	0.0	262
5	05-Dec	00:00-01:00	20.5	25	0.281	0.1501	0.1548	0.0	285
6	16-Aug	02:00-03:00	26.3	29.7	0.315	0.1558	0.1592	1.2	322
7	06-Nov	22:00-23:00	23.8	30.2	0.0	0.0916	0.1016	1.2	322
8	31-Jan	07:00-08:00	18.3	23.2	0.0	0.1325	0.13346	0.0	267
9	07-Aug	06:00-07:00	25.6	30.7	0.0	0.1333	0.1385	0.6	326

5.4. Co-simulation and control algorithms validation analysis

In this subsection, the results of the validation study of the co-simulations are presented. To visually represent the uncertainties during the measurements, error bars were used in the measured data. The analysis showed that the simulated results are in great agreement (100% agreement) with the measured data whenever the error bars were included. The results are considered to be in agreement when the simulated data overlap with the measured data (included the error bars). The key parameter to validate mixed-mode ventilation strategies is the ventilation rate, due to its significant impact on the performance of the natural ventilation solutions and also on energy consumption.

Figure 5-2 presents the comparison between simulated and measured air changes and the results from the analytical models. Including the uncertainties (error bars) there is an excellent agreement (100%) in all the simulated and measured data. In most of the scenarios, air changes per hour were predicted higher for the co-simulations relative to the measured values but still were within the error bars margins. This can be explained by the fact that for the simulated scenarios, the wind was simulated for all the cases. On the contrary, only buoyancy-driven cases were simulated in the chamber hence the experimental data did not account for the wind forces. The effect of the wind resulted in slightly higher predictions of the ventilation rates. The higher impact occurred in scenario 3 & 6 where the highest wind speeds were observed (5.9 and 7.2 m/s respectively). Further, for the scenarios where only the dampers operated (scenario 7,8 & 9), the co-simulations underpredicted the ventilation rates but still the predictions were within the error bars margins of the measured data, possibly due to the effect of the wind speeds on the pressure coefficient at the openings. Due to the very small effective opening area of the dampers, the small wind speed resulted in a turbulent flow which affected the levels of outdoor air that entered the chamber. In the experiments though, the flow was less turbulent since the driving forces were only due to buoyancy and hence higher volumes of outdoor air were measured to enter the space.

The largest difference between the simulated and measured data occurred in scenario 6, where the co-simulations overpredicted the ventilation rate by 12% compared to the experiments but still the predicted values were within the error bars. This difference could be explained by the fact that in scenario 6 the ceiling fan was used at its maximum speed producing internal air velocity of 2.2 m/s. The use of the ceiling fan created a turbulent flow inside the chamber that might have affected the distribution of the CO_2 and hence the calculations of the ventilation rate. On the other hand, the co-simulation assumes a uniform distribution on the air and hence the flow remained laminar, which might be the explanation of this relatively high difference.

A general trend that was observed in Figure 5-2 was the simulated data overpredicted the ACH by approximately 5-11% compared to the simulated data. This can be explained by the dynamic nature of the co-simulations. The experimental design has captured a snapshot of the co-simulations. However, the co-simulations were dynamic, and all the conditions varied over time. Parameters such as external air temperature, heat gains etc varied constantly over time but for the experiments, hourly average values were used. This might be another explanation of the small discrepancies in the results. However, by including the uncertainty of the measurements there is an excellent agreement, 100%, between the results, see figure 5-2, which provides solid evidence of the validity of the developed control algorithms and the simulation approach.

When comparing the analytical models and the simulated data, there is an 80% agreement between the data with an exception in scenario 1 and 9. The analytical models use the temperature difference, the discharge coefficient and the height of the openings as the only parameters to determine the ventilation rates, see Eq. 5-9. However, the airflows through openings and hence the calculation of the ventilation rates is a more complex task. Also, the heat gains, internal or external, are not considered in the analytical models as they use the dry-bulb air temperature and not the operative, which accounts for the radiant component of the temperature. These observations explain the discrepancy and the trends in the results between the simulated and the analytical models.

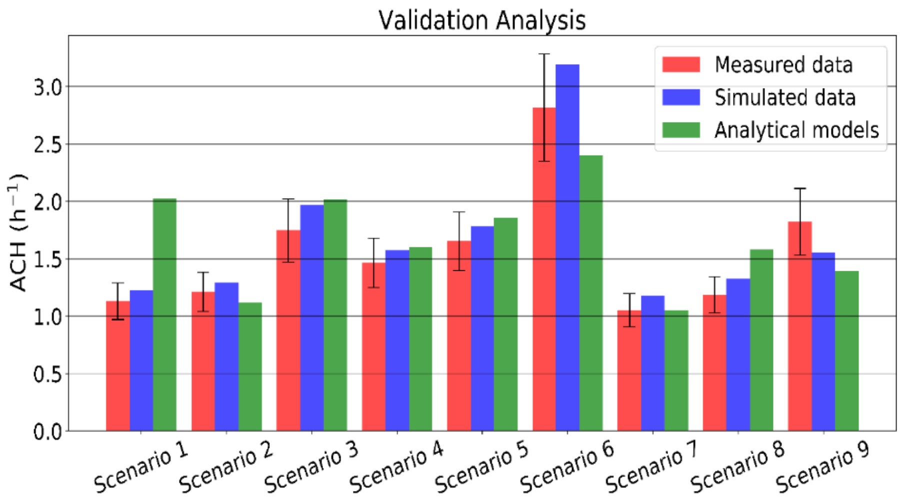


Figure 5-2: Comparison of experimental, simulated and analytical models. Error bars highlight the uncertainties during the experiments.

Using Eq. 5-7 & Eq. 5-8 the analysis showed that MBE=7.25% and CVRMSE=20.64%. Based on ASHRAE Guideline 14, the acceptable limits for hourly calibration of building energy simulation models are 10% and 30% for MBE and CVRMSE respectively. Hence both indices are within the proposed limits.

Table 5-3: Summary of validation analysis data for the co-simulations.

Scenario	**T_{int} [°C]**	**T_{out} [°C]**	**Concentration [ppm]**		**Measurements**			**Analytical modes**		**Co-simulations**	
			Initial	Steady -state	[m^3/s]	ACH [h^{-1}]	error [%]	[m^3/s]	ACH [h^{-1}]	[m^3/s]	ACH [h^{-1}]
1	30.4	29.3	1561	521	0.014	1.131	14.1	0.026	2.023	0.016	1.229
2	27.5	27.4	1640	510	0.015	1.211	14.1	0.014	1.117	0.016	1.291
3	28.3	27.3	1566	435	0.022	1.747	15.7	0.026	2.016	0.025	1.968
4	28.5	27.3	1710	470	0.019	1.465	14.7	0.02	1.603	0.02	1.577
5	23.9	20.5	1629	445	0.021	1.654	15.4	0.024	1.858	0.023	1.784
6	28.4	26.3	1575	404	0.036	2.817	16.5	0.03	2.399	0.04	3.194
7	27.6	23.8	1550	540	0.013	1.053	13.8	0.013	1.055	0.015	1.179
8	22.6	18.3	1882	538	0.015	1.187	13.3	0.02	1.584	0.017	1.328
9	29.0	25.6	1550	430	0.023	1.823	15.6	0.018	1.394	0.02	1.558

5.5. CFD simulations and validation analysis

This subsection presents the validation study for the CFD simulations. To validate the ventilation performance of the control algorithms, it was essential to validate the methodology to calculate the ventilation effectiveness of the natural and mixed-mode ventilation control algorithms. This subsection presents the mesh sensitivity study conducted in order to select the mesh density for the CFD simulations and then presents the validation analysis for the CFD. To calculate the ventilation effectiveness indices (see Eq. 5-2 & 5-3), a customised INFORM code was developed within the CFD software to calculate the ventilation effectiveness. INFORM is an addition to PHOENICS CFD software, which enables the user to define a series of algebraic formulae to calculate customize variables using the PHOENICS Input Language. A part of the code is presented in Appendix: Inform Code for PHOENICS.

5.5.1. Mesh sensitivity analysis

As stated in section 3.6.4.2, a mesh sensitivity analysis is essential to assess the mesh density and the convergence of the simulations. Different mesh densities were analysed to examine which one can provide acceptable levels of accuracy in the predicted results with the minimum computational time compared to the rest of the cases.

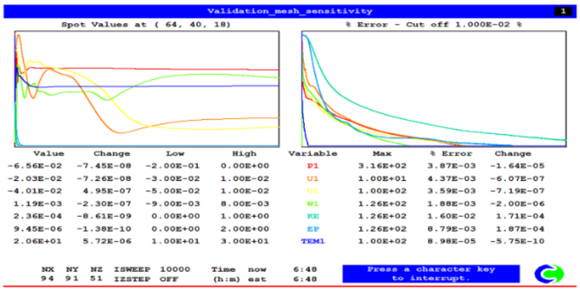
To analyse the results from the mesh sensitivity it was essential to examine whether they had reach convergence. Hence, as stated in section 3.6.4.2, the spot values, the source balance and the residuals' behaviour were analysed to conclude if the examined mesh densities generated stable results. The selected variables for the mesh sensitivity analysis are the air temperature and air velocity at the probe position.

Table 5-4: Summary of different mesh densities and computational time

Mesh number	**Number of cells [X,Y,Z] - Total**	**Computational time [h:m]**
1	[70,77,38] – 204820	2:29
2	[80,68,64] – 348160	4:55
3	[94,91,51] – 436254	6:48
4	[112,109,63] – 769194	12:03
5	[128,120,71] – 1090560	16:47

Based on the results from table 5-7 it can be concluded that all the five simulations have achieved convergence. This was achieved within 10000 iterations, as the residuals of mass and enthalpy equations were below the limit set in the convergence criteria (see section 3.6.4.2), suggesting that the CFD will accurately predict the results. Furthermore, in order to identify and select the most suitable mesh density to be used for the CFD simulations, the key variables were analysed, figure 5-4. The internal air temperature had a maximum variance of around 9% between mesh 1 and 5. Meshes 2, 3 and 4 had very low variance (3%) and hence the least detailed mesh could be used. Similarly, for the air velocity graphs, the maximum variance was calculated to be 8% between mesh 1 and 5, whilst for the rest of the cases, the difference in the predicted results was less than 2%.

Mesh 3



Mesh 5

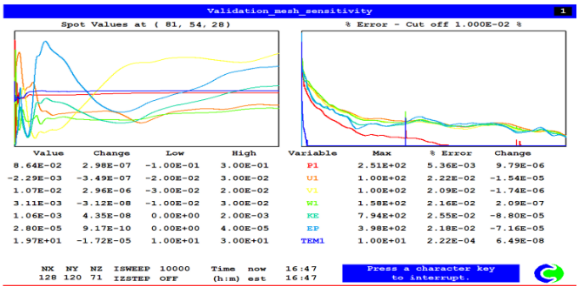


Figure 5-3: Spot values for each mesh density. Top plot represents Mesh 3 which was selected for CFD simulations; bottom plot represents Mesh 5 which did not reach convergence

Table 5-5: Summary of source balance and residual behaviour for the examined mesh densities.

	Source balance (kg/s)			**Residual behaviour**		
	Positive Sum	**Negative Sum**	**error(%)**	**Initial**(10^5)	**Final** (10^5)	**error(%)**
1	0.70120	-0.70120	<0.1	2.041750	-2.0411749	<0.1
2	0.59789	-0.59789	<0.1	1.744517	-1.7444516	<0.1
3	0.60981	-0.60982	<0.1	1.778683	-1.7786930	<0.1
4	0.65763	-0.65578	<0.1	1.919649	-1.9203698	<0.1
5	0.67598	-0.67603	<0.1	2.0467308	-2.0467310	<0.1

The simulation time for each case is presented in table 5-6. The small difference in the results between the mesh 2, 3 and 4 suggests that a further increase in the mesh density will have a minor impact on the final results. The mesh that was selected for the rest of the simulations was **mesh 3** as it needs almost 50% less computational time, compared to mesh 4, to generate as accurate results.

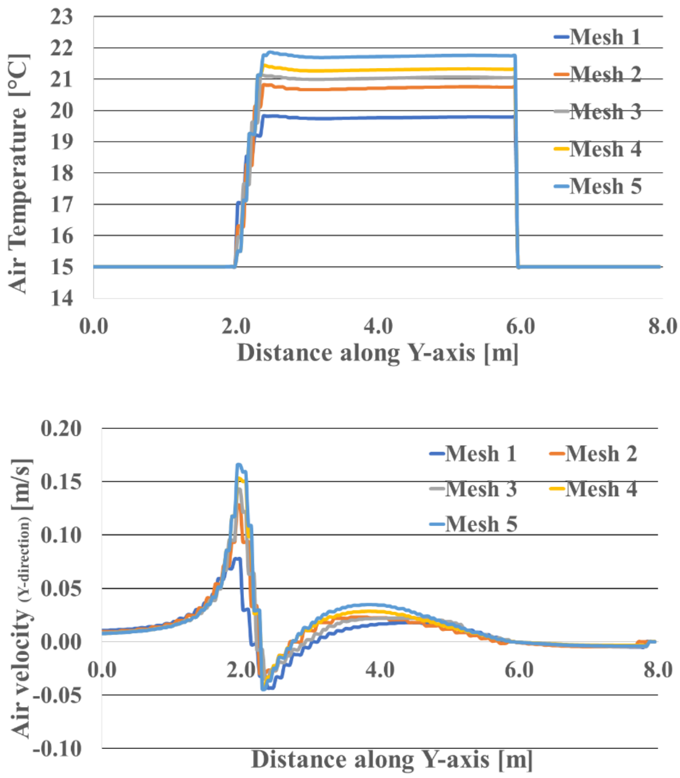


Figure 5-4: Air temperature (top) and velocity parameter(bottom) for mesh sensitivity study.

5.5.2. Validation analysis for the CFD simulations

In this subsection, the results of the validation study of the CFD simulations are presented. To visually represent the uncertainties during the measurements, error bars have been used in the measured data. The key parameters to validate the CFD simulations were the temperature distribution, the CO_2 concentration, the heat removal efficiency and the ventilation effectiveness.

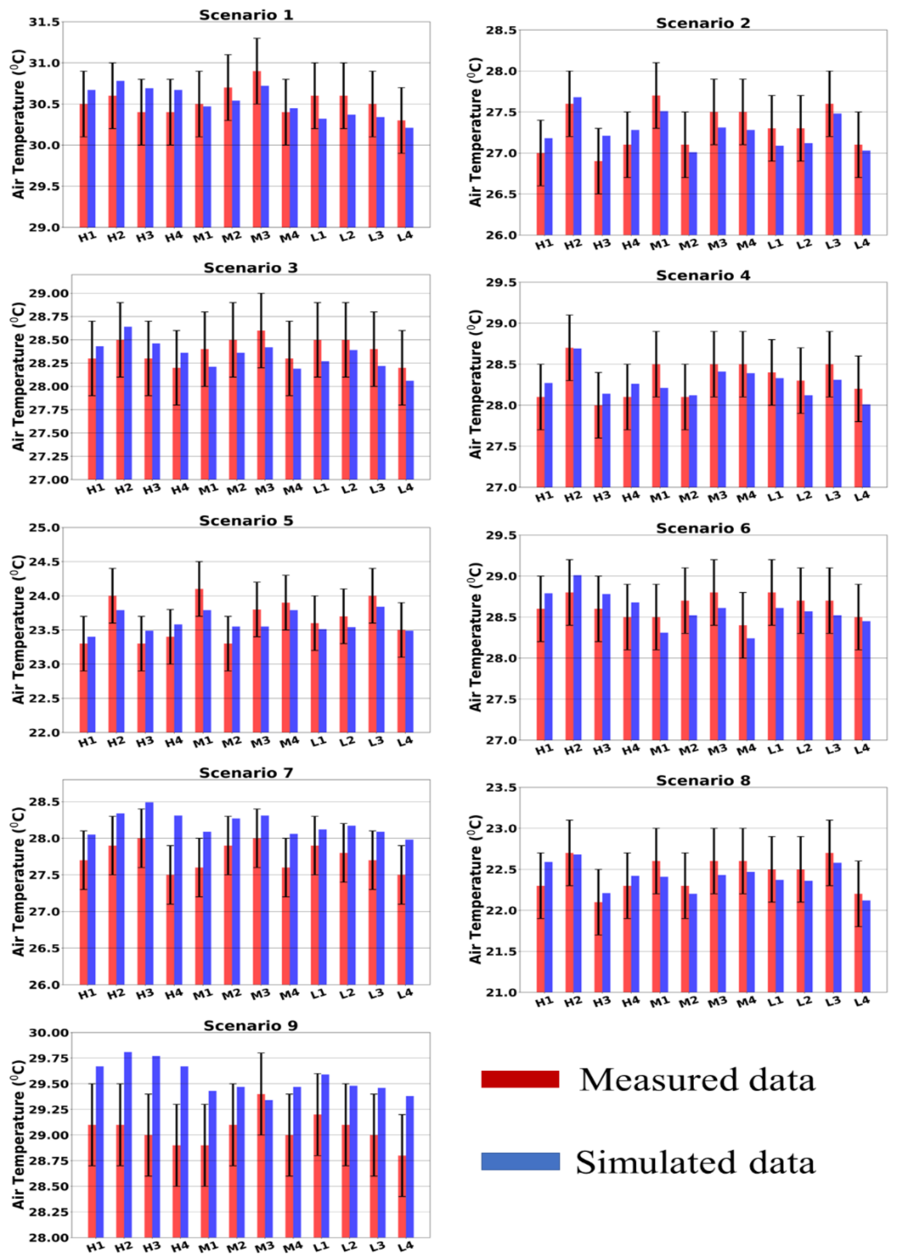


Figure 5-5: Comparison of experimental, simulated data for the internal air temperature for each sensor. Error bars highlight the uncertainties during the experiments.

Figure 5-5 presents the comparison between the simulated and the measured temperatures for the scenarios presented in table 5-4. Taking into the calculated uncertainties for the experimental data there is a 91% agreement between the measured and the simulated data for all the scenarios when the ceiling fan was turned off. For the cases where the ceiling fan was turned on, the simulated results were in agreement with the measured for all the scenarios expect 7 & 9. In both scenario 7 & 9, only the dampers operated and the small effective opening area might have resulted in a more turbulent flow in the chamber compared to what the CFD simulations predicted, resulting in the small discrepancies. The temperatures from the CFD simulations were recorded in the exact same positions as the sensors were placed in the experimental chamber (table 5-1). As expected in buoyancy-driven cases, the temperature was stratified, which is clearly visible in figure 5-6. The use of the ceiling fan resulted in more turbulent flow inside the chamber, mainly due to the swirling of the fan. The discrepancies in some scenarios, when the ceiling fan was on, occurred mainly due to the complexity of modelling a ceiling fan and its effects on the airflow patterns. The fan was model by using a circular object to generate the swirling effect in the air. This object increased the velocity of the internal air, but it did not rotate.

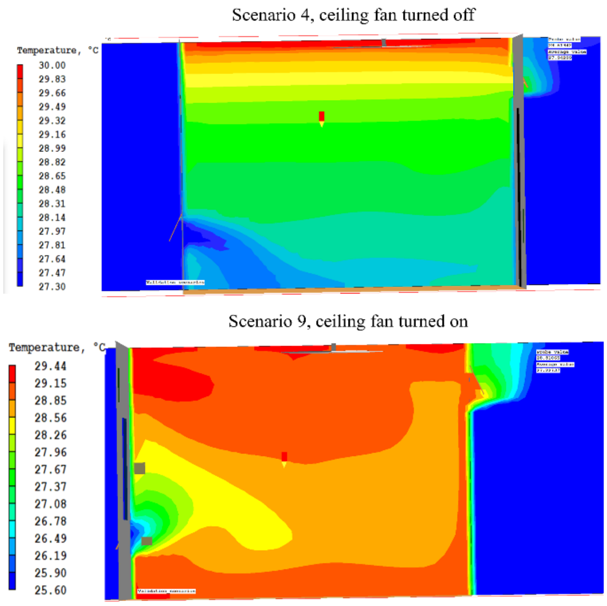


Figure 5-6: Temperature distribution for scenario 4(top) and 9 (bottom).

Also, the objected used in the CFD model was a very simple object and did not have the exact same geometry of the blades as in the experimental chamber. These differences might have caused the small discrepancies in the results.

Despite this observation, the discrepancies of the results in some of the cases when the ceiling fan speed was relatively high were still small. In conclusion, the CFD generated accurately resulted regarding the air temperature distribution in the chamber.

Assessing the accuracy of the temperature distribution was important, as the air temperatures were used to determine the HRE (Eq. 5-4). To calculate the uncertainties for equation 5-4, the equations 5-5 and 5-6 were used. Table 5-6 summarizes the results of the analysis. For a better representation of the results, a bar plot was used (figure 5-7). The error bars represent the uncertainties for the heat removal efficiency calculations. Since the calculation of the heat removal efficiency is based on temperatures, the average value of the temperature measurements was calculated and the error of the average value of the temperature was reported in figure 5-7. In particular, the analysis of the results when the ceiling fan was turned off showed 100% agreement between the measured and simulated data for the heat removal efficiency.

Table 5-6: Summary of validation analysis for the heat removal ventilation effectiveness.

Scenario	**Measurements**						**CFD predictions**			
	T_{oc}	**T_{exh}**	**T_{sup}**	**HRE[-]**	**error[%]**		**T_{oc}**	**T_{exh}**	**T_{sup}**	**HRE[-]**
1	30.53	30.27	29.3	0.784	9.52		30.52	30.32	29.3	0.837
2	27.31	27.30	27.4	1.091	9.10		27.27	27.26	27.4	1.022
3	28.39	28.35	27.3	0.962	10.05		28.33	28.39	27.3	1.054
4	28.33	28.31	27.3	0.985	9.95		28.27	28.18	27.3	0.902
5	23.66	23.54	20.5	0.963	9.93		23.61	23.66	20.5	1.015
6	28.63	28.63	26.3	0.999	10.55		28.59	28.77	26.3	1.080
7	27.76	26.97	23.8	0.801	9.35		28.47	27.03	23.8	0.692
8	22.45	21.87	18.3	0.860	8.57		22.40	21.98	18.3	0.897
9	29.05	28.30	25.6	0.783	10.15		29.68	28.35	25.6	0.674

The calculation of the heat removal efficiency is based on the temperature difference between inside and outside, and since the CFD simulations predicted very accurately the temperature distribution, there is an 80% agreement between the measured and simulated data for the heat removal efficiency as well. For the scenarios with the ceiling fan off, the two data sets were in 100% agreement, whilst when the ceiling fan was on there were some discrepancies in scenarios 7 and 9. The same observation was made in the temperature distribution analysis. Overall, the method of assessing the effectiveness of the natural ventilation based on the heat removal found to be very accurate as the analysis highlighted.

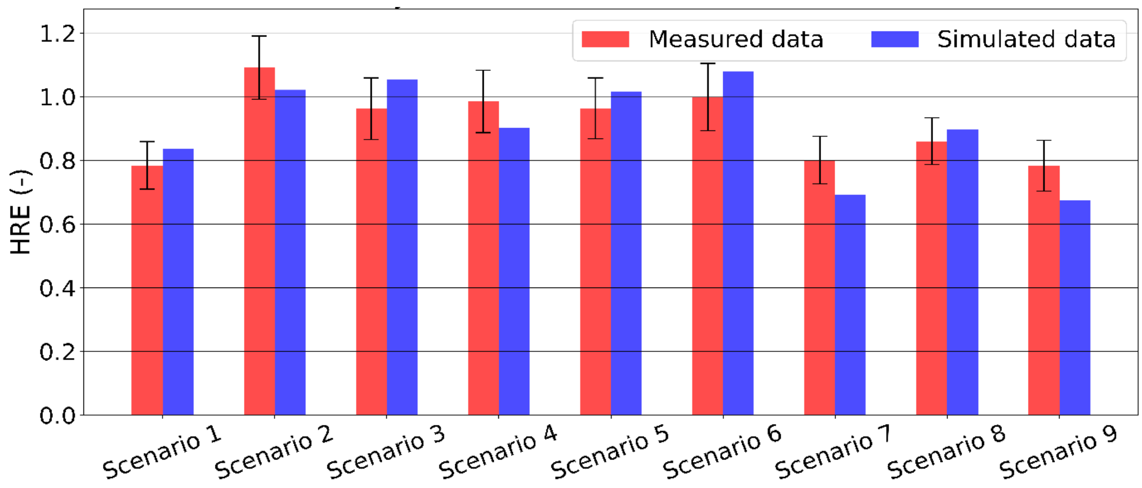


Figure 5-7: Comparison of experimental and CFD simulations for heat removal effectiveness. Error bars highlight the uncertainties during the experiments

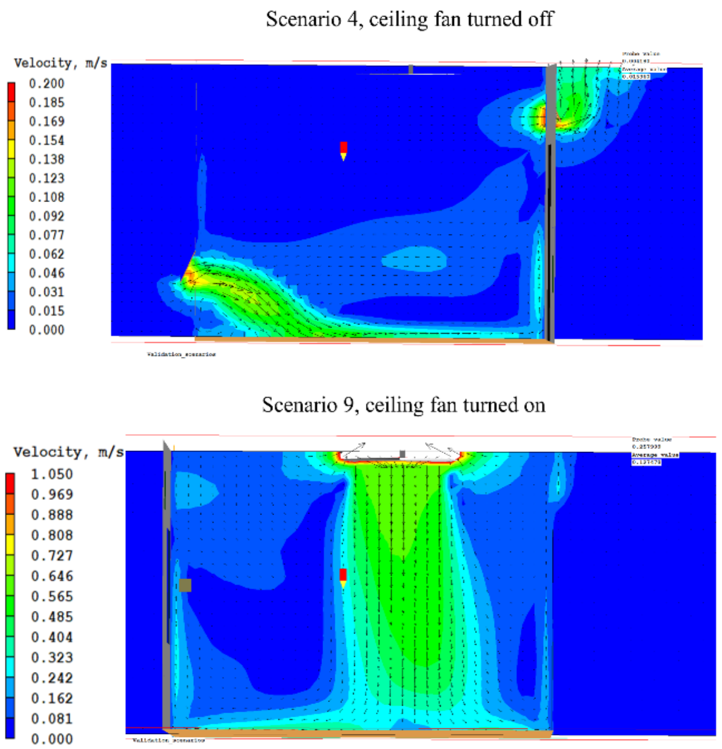


Figure 5-8: Air velocity distribution inside the chamber when the ceiling fan was turned off (top) and on (bottom).

The second method that was used to assess the effectiveness of the natural and mixed-mode ventilation system was the contaminant removal ventilation effectiveness (CRE) method. This method evaluates the ability of the system to remove the contaminants' concentration in the room. Firstly, an analysis of the contaminant concentration was made. Similar to the temperature distribution analysis, the CO_2 concentration from the CFD simulations was measured in the exact same positions as the probes were placed during the experiment. As indicated in figure 5-9, when including the uncertainties during the experiments (error bars), the measured and simulated data overlapped in all the cases expect scenario 7 where the operation of the ceiling fan resulted in high internal air velocities and only the dampers were open.

The CO_2 sensors were placed in the centre of the chamber and underneath the ceiling fan. When the fan operated at the maximum speed, the airflow patterns that were created, were turbulent and hence that might have impacted the readings in the sensors. The difference in the airflow patterns and how they were affected by the operation of the ceiling fan can be identified in figure 5-8. The air velocity inside the chamber when the ceiling fan was turned off was almost negligible as expected due to the small buoyancy forces. However, when the ceiling fan was turned on, the air velocity was higher and specifically under the ceiling fan. The placement of the sensors almost underneath the ceiling fan might have impacted their readings. Furthermore, only two sensors were used to measure the CO_2 concentration in the chamber and the other two were placed in the openings of the two dampers. The higher discrepancies occurred in the two sensors placed in the centre of the chamber whilst for the other two, the measured and simulated results showed 100% agreement.

Analysing the accuracy of the CO_2 measurements was essential since the CO_2 measurements were used to determine ventilation effectiveness. To calculate the uncertainties for Eq. 5-3, the equations 5-5 and 5-6 were used. Since only two sensors were used to measure the CO_2 concentration inside the chamber, the ventilation effectiveness was measured at the centre of the chamber. Table 5-7 summarizes the results of the analysis. For a better representation of the results, a bar plot was presented as well (figure 5-9), showing with error bars the uncertainties for the ventilation effectiveness calculations.

Table 5-7: Summary of validation analysis for the contaminant removal ventilation effectiveness.

Scenario	**Measurements**						**CFD predictions**			
	C_j	**C_{exh}**	**C_{sup}**	**CRE [-]**	**error[%]**		**C_j**	**C_{exh}**	**C_{sup}**	**CRE [-]**
1	352	561	2000	0.873	10.48		379	609	2100	0.866
2	404	480	1801	0.946	10.23		435	526	1881	0.937
3	394	409	1978	0.991	11.06		424	489	2066	0.960
4	387	479	1907	0.939	11.18		416	549	1991	0.916
5	385	464	1881	0.947	10.92		414	564	1964	0.903
6	365	404	2000	0.976	11.85		393	479	2089	0.949

7	349	539	2000	0.885	10.28		394	748	2140	0.797
8	391	574	2000	0.886	9.63		421	673	2089	0.849
9	377	414	1858	0.975	11.17		405	478	1941	0.953

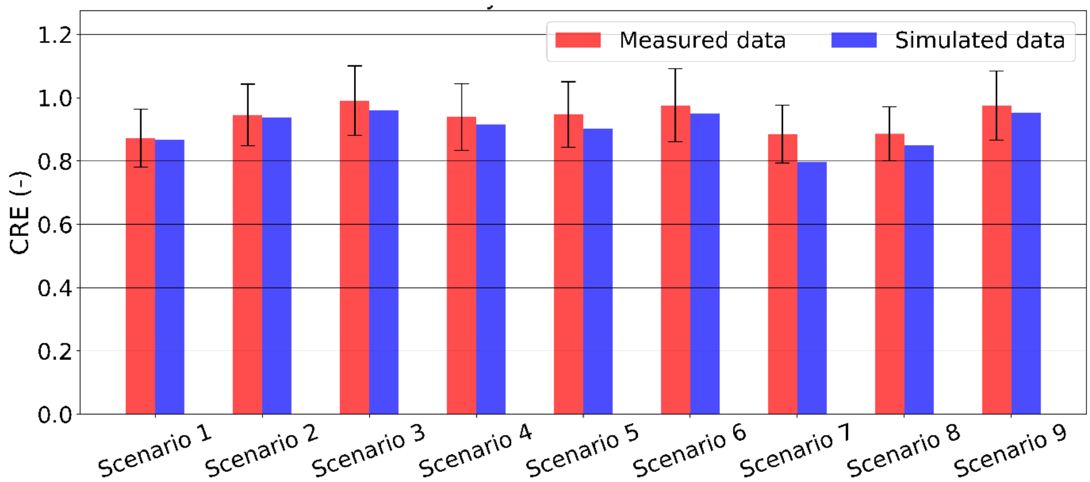


Figure 5-9: Comparison of experimental and CFD simulations for contaminant removal effectiveness. Error bars highlight the uncertainties during the experiments

As the graphs show (figure 5-8 and 5-9), there is a 90% agreement in the simulated and measured data. For the examined scenarios the predictions for the ventilation effectiveness are overlapping with the calculated when the errors bars are included with an exception in scenario 7. This is because the calculation of the ventilation effectiveness is based on the CO_2 concentration measurements and as explained for scenario 7, the CFD models overpredicted the values. Overall it can be said that the CFD can accurately predict the ventilation effectiveness and also the methodology to simulate the CO_2 concentration was accurate, so the methodology to determine the ventilation effectiveness of natural or mixed-mode ventilation systems can be used in the demonstration case.

Using Eq. 5-7 and Eq. 5-8 the analysis showed that MBE=4.36% and CVRMSE=10.71% for the CRE and MBE=5.39% and CVRMSE=12.64% for the HRE. Based on ASHRAE Guideline 14, the acceptable limits for hourly calibration of building energy simulation models are 10% and 30% for MBE and CVRMSE respectively. Hence both indices are within the proposed limits.

5.6. Summary

An overview of the experimental work and the experimental set-up was provided that allowed the validation of the control algorithms and the simulation approach (co-simulations and CFD) developed in this research. Due to the complexity of designing control algorithms for mixed-mode buildings and to test the performance of the developed control algorithms and the co-simulation approach, it was essential to validate the algorithms used, in a well-controlled environment. In the experimental work, an experimental chamber in CEPT university in India was used to collect data to validate both the co-simulations and the CFD models.

The importance of the proposed control algorithms clearly emerged from the gap in the literature regarding the sophisticated control of mixed-mode buildings. The validation of the proposed control algorithms and the co-simulation approach was implemented into two steps. Initially, the predicted values of the ventilation flows were compared against the measured values from the experimental chamber. The co-simulations were able to accurately predict, in all the scenarios, the ventilation flow rates since there was 100% agreement in the results when including the uncertainties in the measurements. The highest discrepancies occurred in Scenario 6 but still the predictions overlapped with the measured data when the uncertainties of the experiments were calculated. Also, the MBE and CVRMSE (7.25% & 20.64% respectively) indices were calculated and the results were within the proposed limits by ASHRAE Guideline 14. The second step of the validation process was to compare the predictions of the co-simulations to the analytical models for single-side natural ventilation. The comparison revealed again an 100% agreement in the results with an exception of Scenarios 1 and 9. The comparison against the measured and the analytical models showed that the co-simulations generated accurate results.

Furthermore, the validation study included the analysis of the ventilation effectiveness and heat removal efficiency as two parameters to assess the performance of the control algorithms. For this analysis, CFD simulations were carried out and the predictions for the temperature and CO_2 distribution were compared against the measured data. The analysis for the CO_2 showed 81% agreement between the simulated and measured data and this observation was further reflected when calculating the ventilation effectiveness and compared against the CFD predictions. The highest discrepancy in the results occurred for scenario 7, where the high internal air velocities due to the ceiling fan and the very small opening areas of the dampers resulted in a turbulent flow inside the chamber which might have an impact on the measurements of the CO_2 concentration. For the temperature distribution comparison, the analysis showed over 90% agreement between the simulated and the measured data with an exception in Scenarios 7 and 9 where the CFD tool overpredicted the internal air temperature.

Overall, this analysis provided evidence that the developed control algorithms and the co-simulations are generating accurate results and hence the performance predictions of the control algorithms for the energy savings and ventilation effectiveness, included in Chapters 6 and 7, are accurate. In addition, the analysis highlighted that the CFD models and the methodology to determine the ventilation effectiveness of the natural ventilation systems are accurate and hence the same methodology can be applied in other geometries (demonstration case) to examine the performance of the natural and mixed-mode ventilation systems. Most importantly, the validation study for both the co-simulations and the ventilation effectiveness presented in this research is a unique approach to validate control algorithms for mixed-mode buildings and evaluate their thermal performance and their ability to delivery comfortable internal environments. It provides evidence for the validity of the control algorithms and the simulation approach that has been used.

Using a controlled environment (chamber) for the validation study however, is accompanied by limitations as well. Controlling all the variables to eliminate uncertainties created a “perfect” condition which does not represent reality. In a real building with occupants, conditions will change constantly and it will create a very dynamic environment. So there is a possibility that in real applications the control algorithms might not behave the same way. However, when testing new concepts it is better to eliminate the uncertainties in order to understand better the behaviour of the control algorithms.

Chapter 6. Results of co-simulations

6.1. Overview

In this chapter, a feasibility study for the use of natural ventilation solutions in all the climate zones in India is firstly presented. The developed control algorithms were applied in a typical 2-bedroom apartment (section 3.8) to examine the impact of the different control algorithms on the operation of the window. Afterwards, the predictions from the co-simulations for the energy saving potential, as well as the predictions for the internal air temperature for the different control algorithms, were assessed. It should be highlighted that this chapter is focused on the demonstration case study, a typical 2-bedroom apartment. The mechanical cooling systems that were used are i) a split AC unit and ii) DEC units, in the master bedroom, small bedroom and living room. This analysis addresses Objective 5 of this research.

6.2. Feasibility for natural ventilation solutions in four Indian climate zones

The weather files developed by the Indian Society of Heating Refrigeration and Air-conditioning (ISHRAE) are used. ISHRAE 2.0, are the latest weather files developed by ISHRAE to improve the accuracy of simulation studies (Huang, 2015). These were developed using data from 1993 to 2007, and are the most recent weather dataset that could be found online for India (Huang, 2015). To assess the feasibility of free-running buildings the IMAC adaptive models were used and the following steps were followed:

- Calculate the daily average outdoor dry-bulb temperature using the arithmetic mean of the minimum and maximum temperatures[3]; and
- Calculate the 30-day running mean outdoor dry-bulb temperature for a day by using the arithmetic mean of the outdoor dry-bulb temperature of the previous 30-days.

[3] The arithmetic mean of the hourly values of the temperature was also calculated and compared to the values from step II. The values from both approaches were almost identical and hence the arithmetic mean of the minimum and maximum values was used for the rest of the analysis.

6.2.1. Indian Model of Adaptive Comfort (IMAC) applicability

IMAC adaptive comfort model is applicable within a certain range of 30-day running mean outdoor dry-bulb temperature values of 13-38.5°C. The calculated 30-day running mean for each day of the year was used to predict the hours per year that each model is applicable for in each of the four selected locations. The discomfort conditions were predicted for the hours of the year with dry-bulb temperature falling outside the temperature limits. Table 6-1 summarizes the number of hours within the applicability range of the IMAC thermal comfort model for the four locations.

Table 6-1: Number of hours within the applicability range of comfort models for each location.

Ahmedabad- IMAC MM	8760
Bangalore- IMAC MM	8760
New Delhi- IMAC MM	8448
Chennai- IMAC MM	8760

For each of the proposed cities, a graph showing the 30-day outdoor running mean and the comfortable range from the adaptive thermal comfort models was plotted to examine whether the climate for each city can provide comfortable conditions.

Hot and Dry-Ahmedabad

Ahmedabad is in the west part of India (23.03°N and 72.58°E) and at 53m above the sea level. The climate is hot and dry with three main seasons: summer, monsoon and winter. Except for the monsoon period, Ahmedabad receives a very small amount of rainfall. The average maximum temperature during the summer months is 43°C while during winter period can reach up to 30°C (IMD, 2017). Figure 6-1 displays the outdoor 300-day running mean temperature throughout the year. It can be seen that with an exception during the months of April to July, the outdoor temperature is within the acceptability ranges proposed by IMAC model suggesting that it is feasible to operate in mixed-mode.

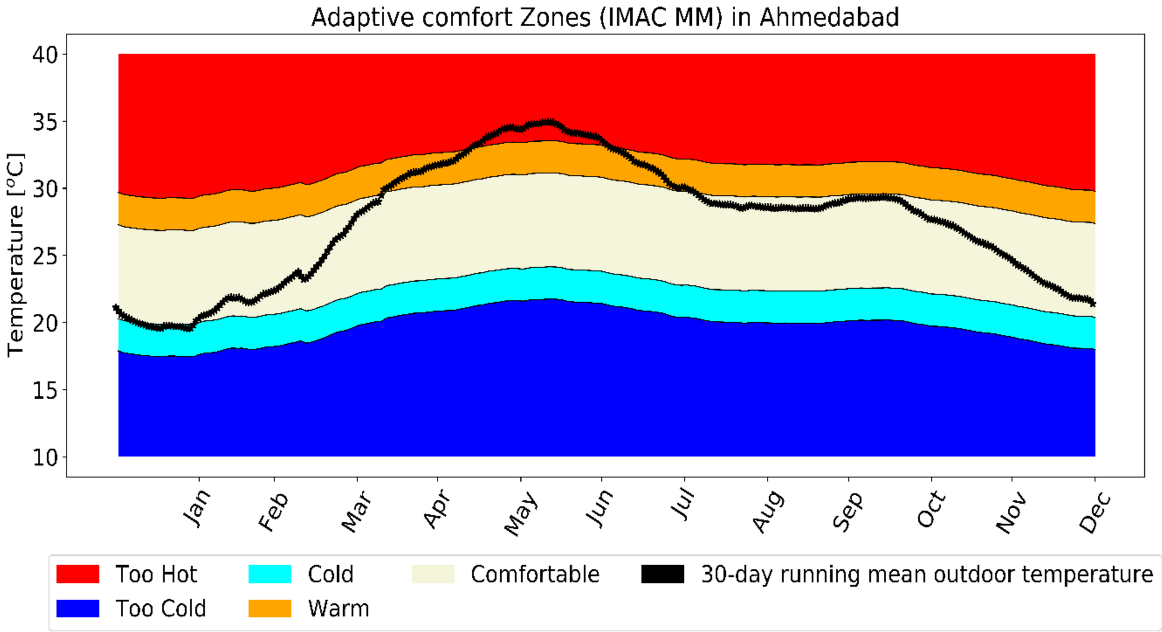


Figure 6-1: Adaptive comfort zones for Ahmedabad; Red and dark blue denote uncomfortable conditions outside the 80% acceptability range; Beige denotes the comfortable zone; and Orange and cyan denote slightly uncomfortable conditions, between 80-90% acceptability.

Warm and Humid-Chennai

Chennai is located on the south-east coast of India (13.1°N and 80.3°E and on average 7m above the sea level). Due to its location, Chennai features a warm and humid climate, with the hottest period of the year to be in late May/early June with maximum temperatures to be around 35-40°C and the coolest month is January with average temperatures around 25°C. The humid period for Chennai is from July until December and the highest rainfall occurs between October to December (IMD, 2017). As suggested by figure 6-2, the external air temperature is within the limits from both thermal comfort models, however, it is important to incorporate mechanical systems especially due to the extremely high levels of humidity.

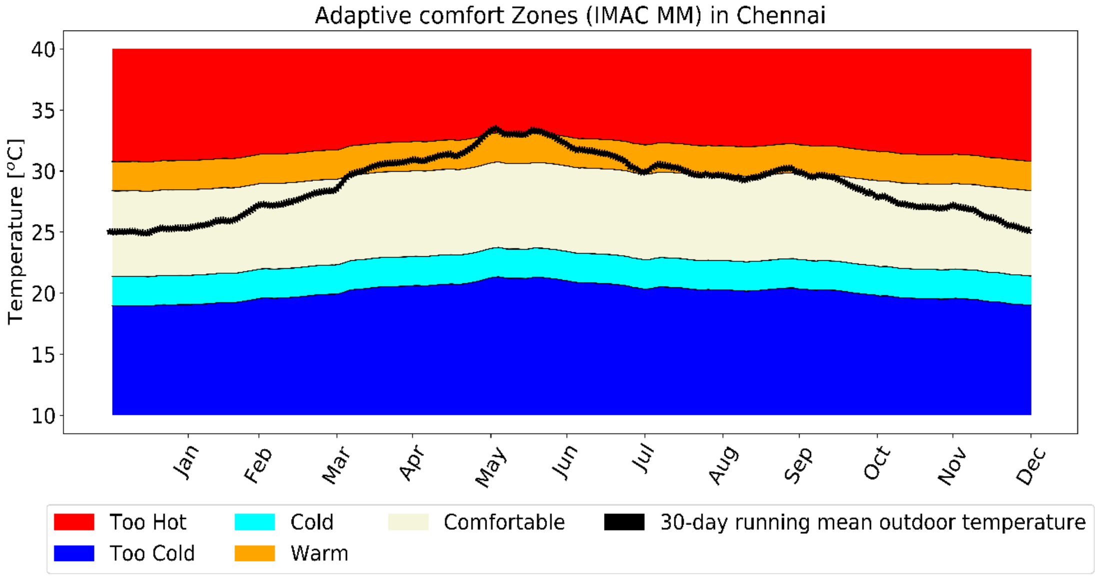


Figure 6-2: Adaptive comfort zones for Chennai; Red and dark blue denote uncomfortable conditions outside the 80% acceptability range; Beige denotes the comfortable zone; and Orange and cyan denote slightly uncomfortable conditions, between 80-90% acceptability

Temperate/Moderate-Bangalore

Bangalore is located on the south of India (12.97°N and 77.56°E) and has an elevation of 900m. Due to its high elevation, Bangalore features a temperate climate with distinct wet and dry seasons. The hottest month is April with a maximum air temperature of 35°C whilst the coolest is January with an average low air temperature of 15°C. Bangalore receives rain due to the north-east and south-west monsoons with the wettest month to be September (IMD, 2017). The temperate climatic conditions of Bangalore are favourable for the use of natural ventilation solutions almost throughout the whole year, as highlighted in figure 6-3.

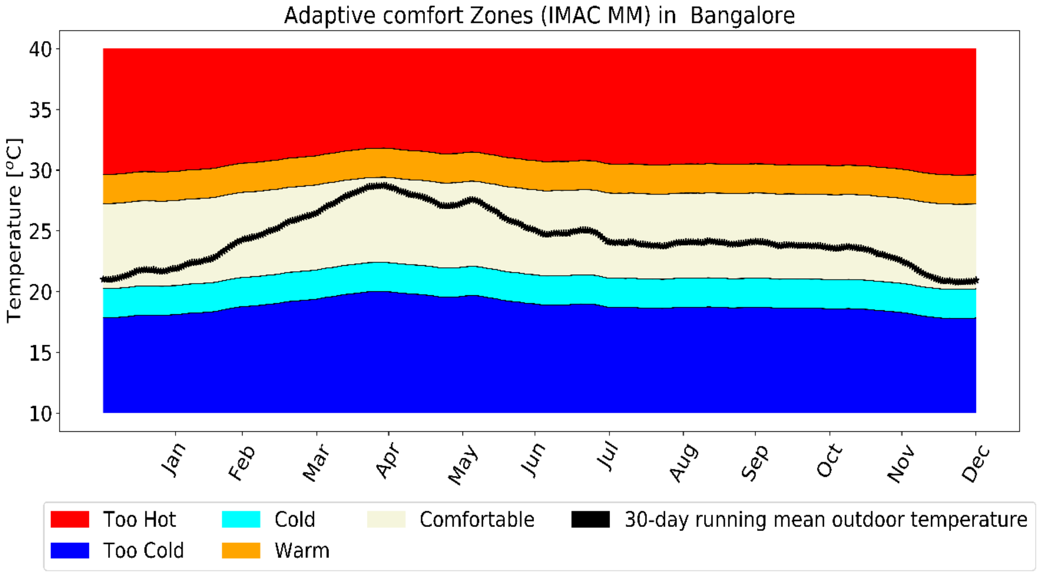


Figure 6-3: Adaptive comfort zones for Bangalore; Red and dark blue denote uncomfortable conditions outside the 80% acceptability range; Beige denotes the comfortable zone; and Orange and cyan denote slightly uncomfortable conditions, between 80-90% acceptability.

Composite-Delhi

Delhi lies in northern India (28.61°N and 77.23°E) and has an elevation of 216m. The average temperature varies from 40°C during summer months to 20°C during winter. The monsoon period occurs in the middle of the summer with the highest levels of precipitation during July and August (IMD, 2017). Based on figure 6-4, the weather conditions are favourable for natural ventilation solutions during the winter months due to lower external temperature.

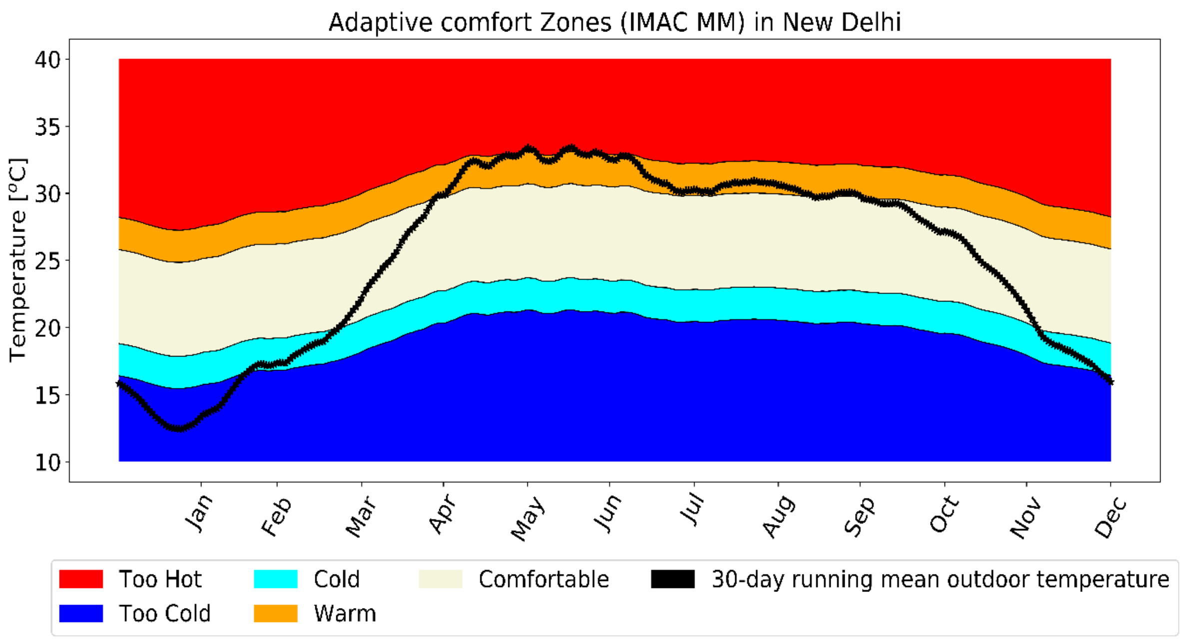


Figure 6-4: Adaptive comfort zones for New Delhi; Red and dark blue denote uncomfortable conditions outside the 80% acceptability range; Beige denotes the comfortable zone; and Orange and cyan denote slightly uncomfortable conditions, between 80-90% acceptability.

The analysis above showed the potential for natural ventilation for all the climatic zones throughout the year using only the 30-day outdoor running mean. However, to assess the feasibility for natural ventilation solutions in all the climatic zones in India in detail, the hourly levels of outdoor relative humidity should be included in the analysis to calculate the potential for each operation mode, see table 3-5. It is important to mention that this analysis is solely based on outdoor conditions and does not include any building design. This means that this analysis highlights the theoretical potential for each operation mode since the building thermal mass was ignored.

Figure 6-5 presents the monthly operation mode distribution for all the studied cities. This analysis justifies the use of natural ventilation solution for each climate zone in India. Moreover, the analysis highlights the need for humidification and dehumidification for both locations to maximize the use of natural ventilation without compromising the thermal comfort of the occupants.

Hence, it is crucial to consider both temperature and humidity in the design of the control algorithms for mixed-mode buildings.

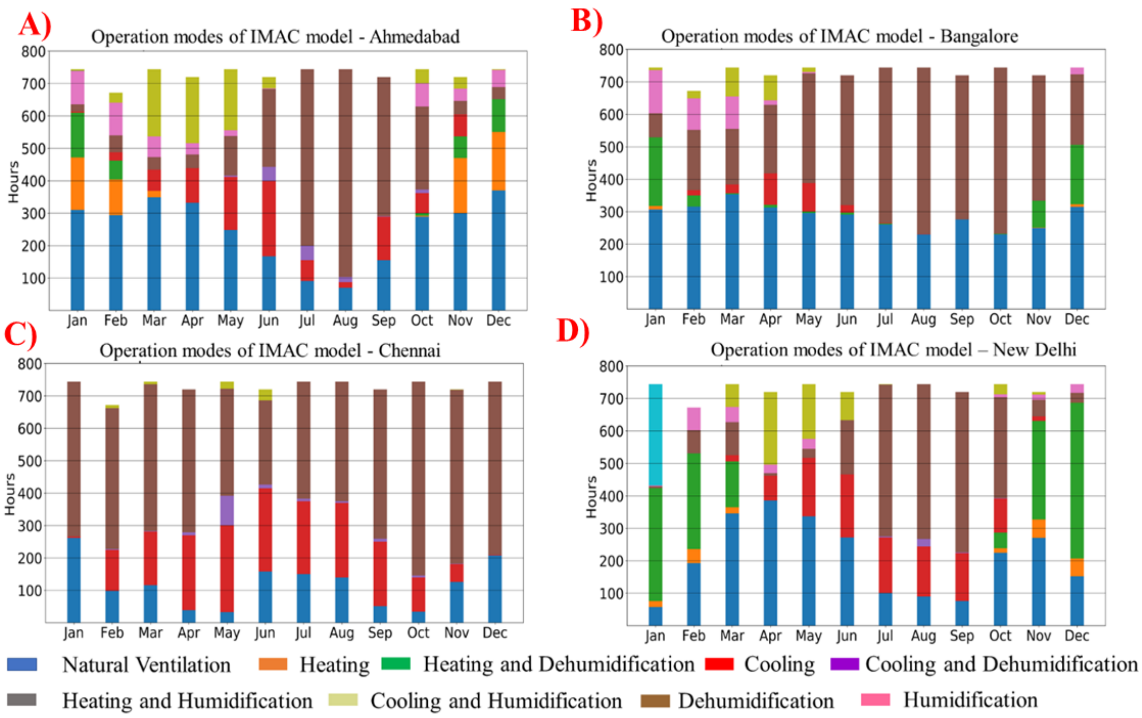


Figure 6-5: Operation mode for all the cities for the IMAC thermal comfort model.

For all locations, dehumidification seems to have an important role, especially during the summer months when it is the monsoon period, followed by cooling mode and natural ventilation. It is important to consider humidity as an equally important parameter to temperature when designing control strategies since as is was depicted by the climatic analysis dehumidification is required throughout the year. In Bangalore, (figure 6-5B) the analysis proposes that from May to October, it is feasible to maintain a comfortable internal environment using natural ventilation and dehumidifier. Overall, for all locations, the analysis highlighted the potential for using natural ventilation solutions throughout the year to minimize the use of mechanical systems. However,

As the analysis suggests, India could accommodate the use of natural ventilation strategies for cooling and ventilation purposes in all the climatic zones during the year. Based on the specific weather conditions of each place, the use of natural ventilation solution could cover almost 50% of the time. However, due to the extreme weather conditions (high air temperatures or high levels of humidity), the uncontrollable use of natural ventilation could have a negative impact on the cooling demand of buildings.

6.3. Two-bedroom apartment - Demonstration case

The analysis of the climate zones in India indicated that it is feasible to accommodate natural ventilation solutions for all the climatic zones without compromising thermal conditions for occupants in certain periods of the year. To assess the impact of the different cooling and ventilation strategies on the internal conditions and their energy-saving potential, the case-study two-bedroom apartment was evaluated at four climates representative India cities. The analysis below highlights the most significant outcomes from the simulations. The two different cooling systems will be analysed separately and at the end, a comparative analysis of the energy-saving potential from both systems will be presented. As mentioned in table 3-2 in section 3.5, the examined mechanical systems were the split unit and the DEC units. For each of the mechanical system, all the scenarios were analysed (VCS 1,2,3,4). It should be mentioned that the co-simulations reported the results in timesteps, every 10 minutes simulation time, so the average value for each hour was calculated.

6.3.1. Temperature comparison among the ventilation and cooling strategies

As discussed in Section 2.4, the internal air temperature is an important parameter for the control algorithms. It is essential, hence, to analyse the results and assess the impact of the different ventilation and cooling strategies on the internal air temperature.

The average zone air temperature is analysed in detail to understand better the impact of the window opening, ceiling fans and mechanical ventilation on the internal environment of the apartment. For this purpose, the weighted average air temperature was calculated for the whole apartment. To control the natural or mechanical systems, the average zone air temperature was used. The volume of each zone was used as the weighted factor. To calculate the weighted average air temperature, the following formula was used:

$$T_{zone,aver} = \frac{T_{air,masterbedroom} * V_{masterbedroom} + T_{air,smallbedroom} * V_{smallbedroom} + T_{air,livingroom} * V_{livingroom}}{V_{masterbedroom} + V_{smallbedroom} + V_{livingroom}}$$

,where $T_{air,masterbedroom}$ and $V_{masterbedroom}$ are the average air temperature [°C] and volume of the masterbedroom [m^3] respectively. Similarly, the rest of the terms are referred to the average air temperature and volume for the smallbedroom and livingroom.

Figure 6-6 & 6-7 illustrate the daily outdoor air temperature, average internal air temperature as well as the heating and cooling setpoints and the operation of ceiling fans and windows for VCS 2 and VCS 3 respectively for one week for Bangalore. The selection of this specific week was made because the outdoor conditions favoured the use of the windows/dampers for a long period, so it was interesting to examine the impact of the different control algorithms on the internal air temperature and window operation. The cooling and heating setpoints are determined by the IMAC adaptive thermal comfort model.

When the nighttime restriction was applied at the windows, the internal air temperature was higher compared to the case where windows could operate without restrictions for all the ventilation and cooling strategies. For instance, in figure 6-6A and for the 7th of September, the window restriction resulted to almost 2°C higher internal air temperature (figure 6-6A dashed black line) compared to the case with no window restriction (figure 6-6B solid black line). The restriction at the windows limited the amount of cool air that entered the apartment and hence limits the potential to cool down the thermal mass of the apartment. Due to the time lag of the building, the stored heat released later in the day which resulted in higher internal air temperatures and higher demand for cooling.

As presented in figure 6-6 & 6-7, the use of ceiling fans increases the cooling setpoint temperature which led to the more frequent operation of windows. By increasing the cooling setpoint, the temperature range to operate in natural ventilation mode was wider compared to the cases where the ceiling fan was turned off. It also reduced the number of hours the internal temperature was outside the setpoint limits especially for the case with no window restriction. It has been shown that effective control of ceiling fans, can improve the overall thermal performance of the building by utilizing the use of natural ventilation and can achieve higher energy-saving potential.

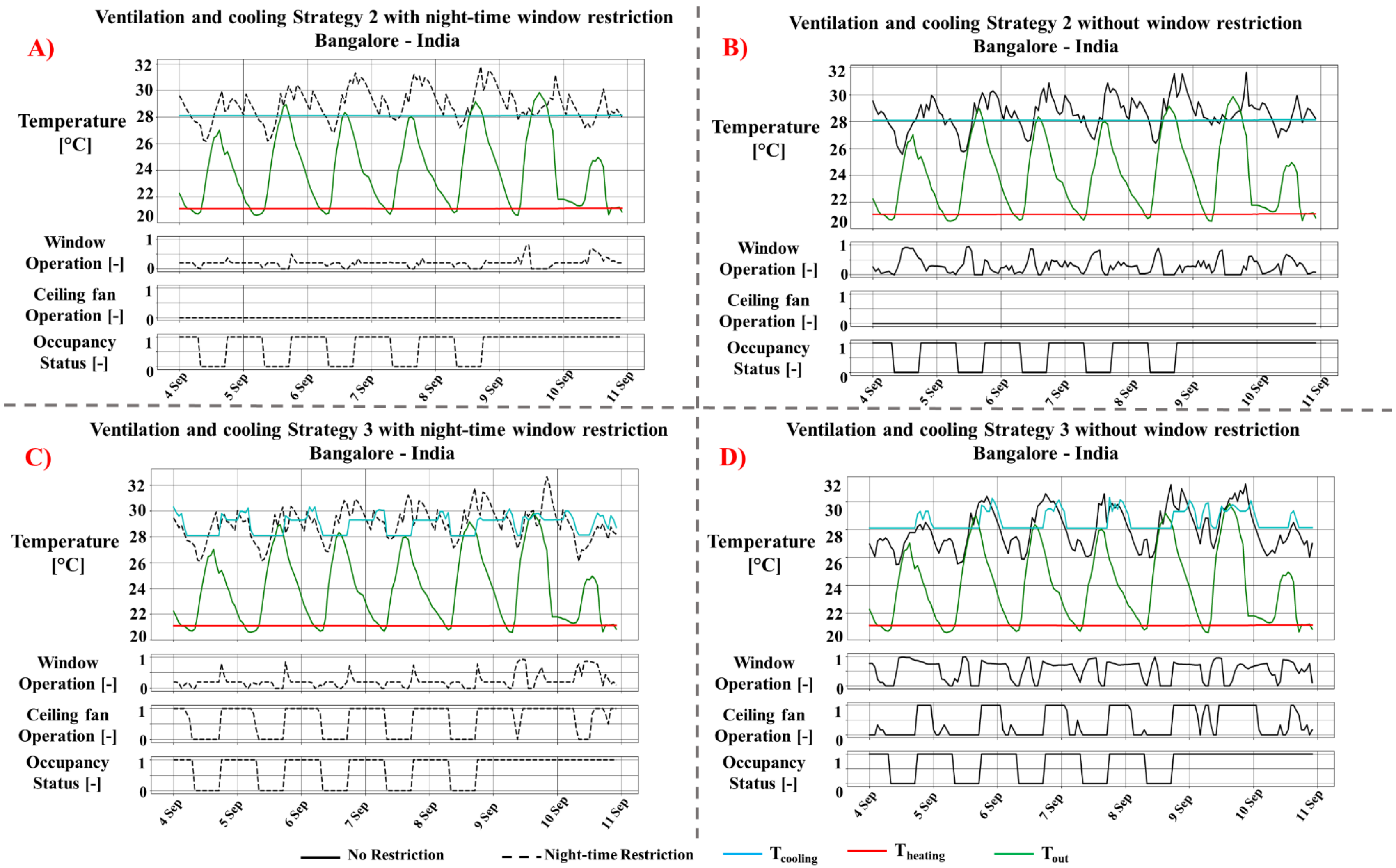


Figure 6-6: Comparisons of daily average air temperature, window and ceiling fan operations for VCS 2 (top) and VCS 3 (bottom) for the city of Bangalore with a split unit as the cooling system.

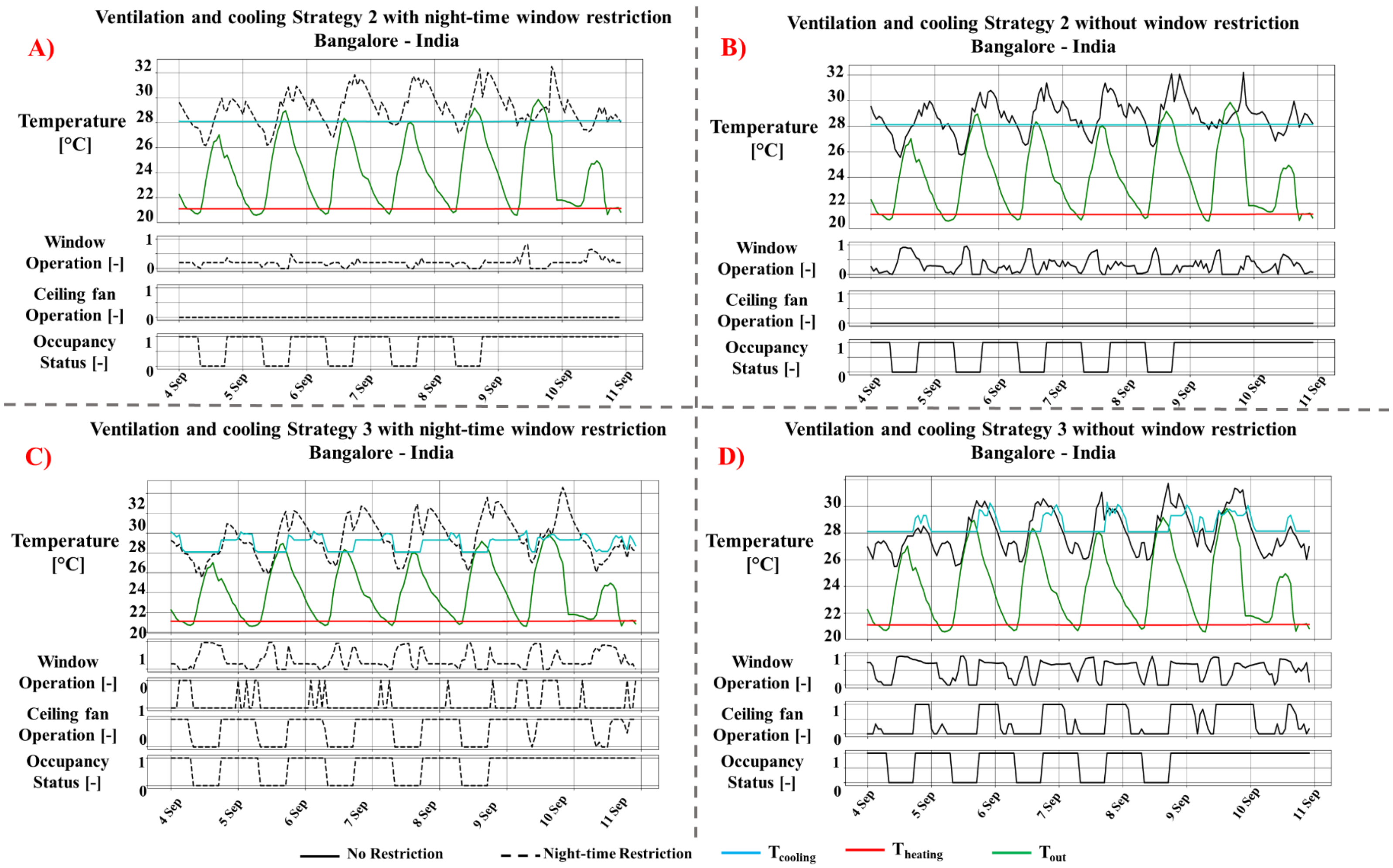


Figure 6-7: Comparisons of daily average air temperature, window and ceiling fan operations for VCS 2 (top) and VCS 3 (bottom) for the city of Bangalore with an evaporative cooler as the cooling system.

As figure 6-6 & 6-7 indicate, the profile of the internal air temperatures is affected by the VCS that has been selected as well as the cooling systems. The inclusion of the ceiling fans resulted in elevated cooling setpoints and hence the hourly air temperature was within the comfort limits for a longer period of times compared to ventilation and cooling VCS 2, where no ceiling fans were used. It should be mentioned that the windows/dampers, ceiling fans and cooling systems were available to operate only when the flat was occupied, which explains the high internal air temperatures, during day-time. This resulted in a time lag when cooling the flat during the nighttime since the internal air temperature was reported to be in some cases much higher than the cooling setpoint. As explained in table 4-4, the apartment was unoccupied during the day time in weekdays. The high outdoor air temperature and the high levels of solar radiation were the main two reasons that led to the very high internal air temperature when then flat was unoccupied. However, that seems the most reasonable approach that the occupants of a flat will control their systems. In both ventilation and cooling scenarios, the use of nighttime window restriction resulted in higher internal air temperature, due to lack of sufficient cooler air flowing in and reducing the temperature of the thermal mass.

To have a better understanding of how each VCS affected the internal air temperature, an analysis was made to calculate the temperature distribution, see figure 6-8 & 6-9. This analysis enables the developers of control algorithms to understand how the different control approaches affect the internal air temperature, which is an index to assess the levels of thermal comfort.

Figure 6-8 & 6-9 illustrate the impact of the different control algorithms on the temperature distribution in the flat when split units and evaporative coolers were used respectively. The black bars show the hours of the year the internal air temperature was between the temperature interval, for example between 24-25°C. The graphs highlight the internal air temperature distribution for each city and cooling system for VCS 2 & 3. The co-simulations predicted that the average internal air temperature was approximately 1°C lower for the cases without any window restriction compared to the nighttime restriction cases, regardless of the city or cooling systems. When the ceiling fans were used, and hence the cooling setpoint was elevated, the average internal air temperature increased as well as expected, approximately 1~1.5°C. In the city of Bangalore, with the most moderate climate among the examined cities, the co-simulations showed that the internal air temperature is distributed around a wider range of temperatures (27-30°C) compared to the most extreme climatic conditions, such as Chennai, where the internal air temperature was distributed around the cooling setpoint temperature (~32°C) since the use of mechanical cooling dominated the operation mode. A similar observation was made for New Delhi and Ahmedabad as well.

In the studied areas with high levels of relative humidity, Chennai and New Delhi, the use of the evaporative cooler resulted in higher internal air temperature compared to the use of split units. The use of the evaporative cooler affected the average annual value of the internal air temperature as well as the temperature distribution. A general observation based on all the subplots in figures 6-8 & 6-9 suggests that the distribution of the internal air temperature is shifted towards the highest temperature ranges for New Delhi and Chennai compared to the cases with split AC units. For Ahmedabad and Bangalore, with drier and more moderate climatic conditions respectively, the use of the evaporative cooler resulted in slightly higher average internal air temperature but not significantly higher as in New Delhi and/or Chennai.

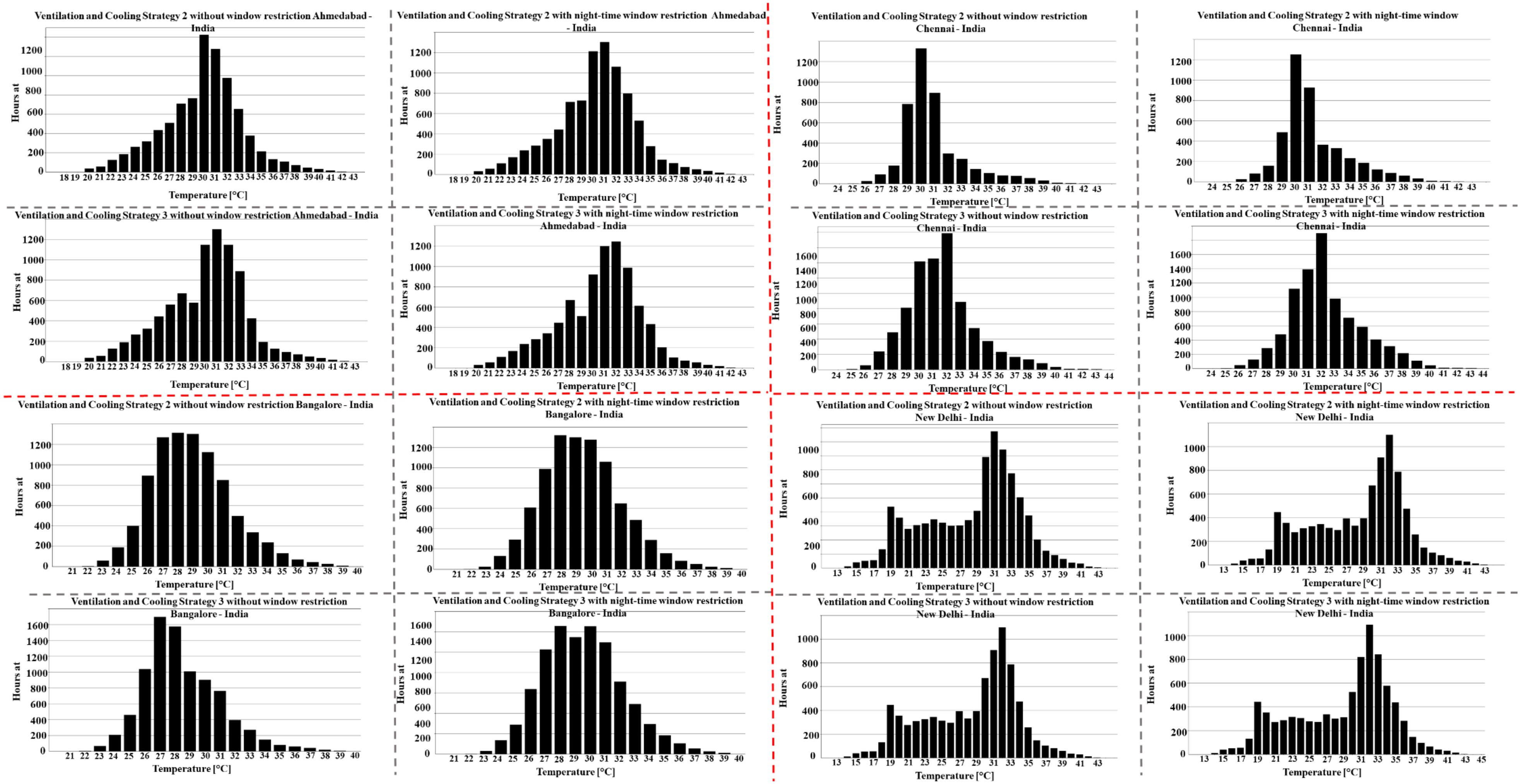


Figure 6-8: Temperature distribution for VCS 2 and 3 for all the examined cities when split units were used as the mechanical cooling systems

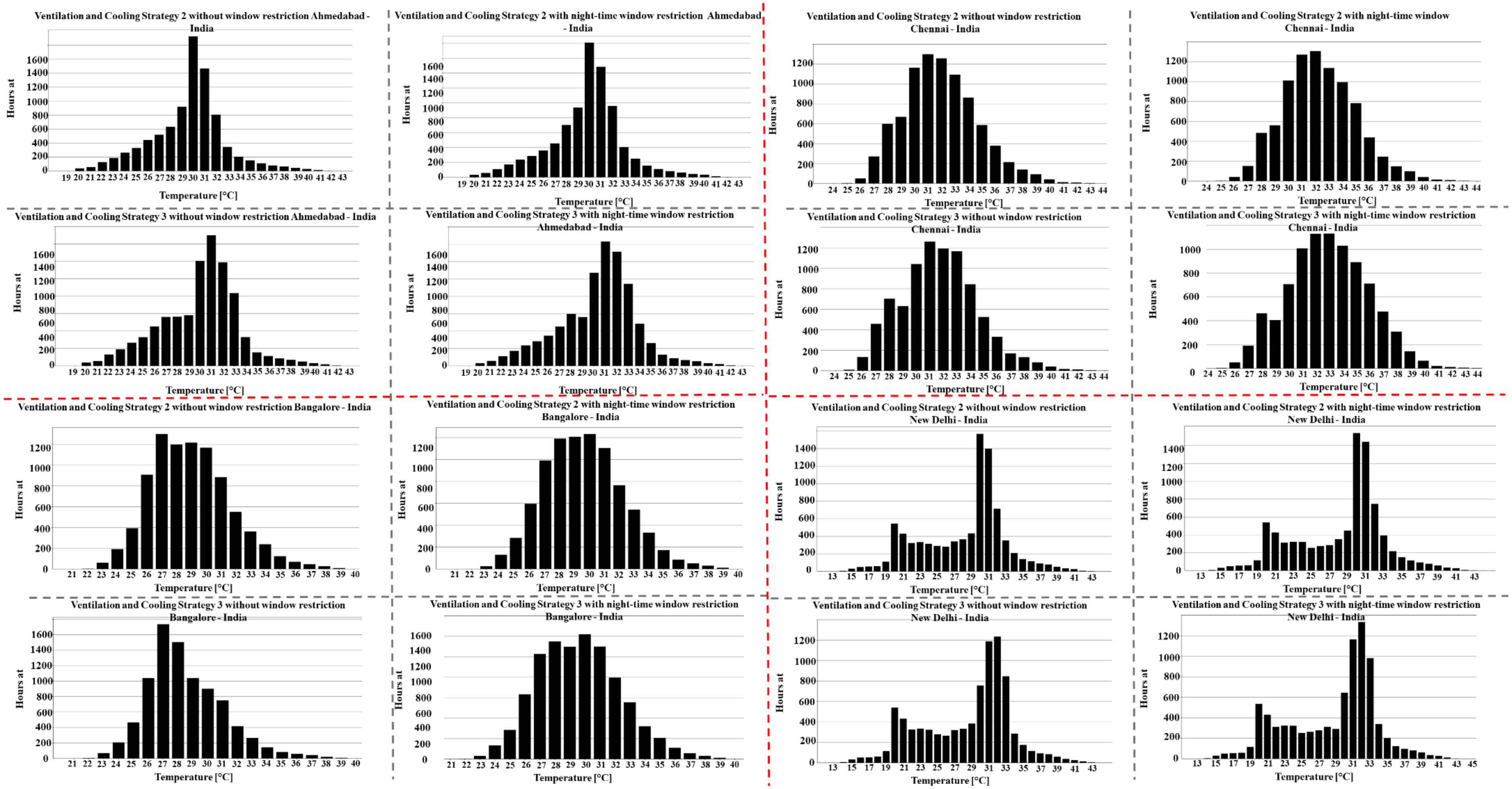


Figure 6-9: Temperature distribution for VCS 2 and 3 for all the examined cities when evaporative coolers were used as the mechanical cooling systems

As the analysis showed, the internal air temperature is affected by a number of parameters. However, to better understand the performance of the different control strategies, other parameters should be analysed to conclude how effective is each VCS. A very important parameter that should be examined, in mixed-mode buildings, is the hours of the year the natural ventilation was selected over the mechanical mode and how many hours the internal operative temperature was within the comfort limits.

6.3.2. Window operation, predicted internal temperature and comfortable hours for the examined cities and for all the ventilation and cooling strategies

The use of different control algorithms has an impact on the control and operation of the windows. Figure 6-10 to 6-13 show the behaviour of the window opening for the VCS 2 and 3 for the whole year. The use of more nighttime natural ventilation has benefits as it can take advantage of the cooler outside air to cool down the thermal mass of the building and possibly to mitigate the risk of uncomfortable internal conditions due to high internal air temperature during the next day. However, the use of nighttime ventilation is not always favourable mainly for safety and noise issues.

As the figures 6-10 to 6-13 show, the operation of the windows/dampers is highly affected by the ventilation and cooling strategies employed, the restriction for the nighttime ventilation and the climatic conditions of each location. The moderate climate of Bangalore favoured the use of natural ventilation for a longer period throughout the year compared to the rest of the locations. The use of sophisticated control algorithms resulted in using natural ventilation during the day time in Bangalore from May to December as figure 6-11 suggests. In places with extreme weather conditions, such as Ahmedabad, the control algorithms operated the openings mainly during the nighttime when the outdoor air temperature was within the limits. In Chennai, the combination of high external air temperature and high levels of outdoor relative humidity resulted in fewer hours of natural ventilation compared to the rest of the cities. However, the sophisticated control algorithms can combine the use of passive and active techniques to maintain comfortable internal environments and achieve energy savings, even for extreme climatic conditions, as the analysis shows in the next section (section 6.3.3).

Overall, the nighttime restriction affected not only the operation of the openings during the night but also during the day. In figures 6-10 to 6-13, graphs A and C, highlight the windows' operational behaviour throughout the year for every hour with 20% of their maximum allowable opening area and without (graphs B and D figures 6-10 to 6-13) restriction at the window's opening. Although the nighttime restriction might limit the advantages of utilising the cooler outside air, it seems a solution that is more possible to be adopted by the occupants of an apartment, especially for those at low floor level apartments especially due to safety reasons. From the graphs, it can be seen a difference in the opening percentage predictions from the co-simulations. The nighttime restriction affected the opening percentage during the daytime as well. When the restriction was applied, the internal air temperature predicted to be higher compared to the case without any restriction and hence the control algorithms operated more frequently the mechanical systems resulting in closed windows regardless the climatic conditions. Thus, the co-simulations calculated lower percentages of windows' opening compared to the cases with no window restriction. The opening area of the windows was calculated according to the required ventilation rates to deliver natural cooling. In cases without any restrictions, the algorithms calculated the opening area to be approximately 50-60% of their maximum allowable opening area during the nighttime compared to the 20% when the restriction was applied. This resulted in lower internal air temperatures, compared to the cases with nighttime restriction, which favoured the use of natural ventilation during the day-time, as preconditioning had occurred. Similar windows operation patterns were predicted for all four climates with the difference of course at the actual percentage of the openings, see figures 6-10 to 6-13.

A general observation from the graphs is that despite the climatic conditions, the use of ceiling fans increased significantly the operation of the windows. The higher setpoint temperatures for natural ventilation led to predictions of higher windows/dampers opening percentages compared to cases without the ceiling fans. The restriction of the openings during the nighttime ventilation affected the operation of the windows during the daytime as well. In Bangalore with the most moderate climate compared to the rest of the cities, the co-simulations predicted a more frequent operation of the windows/dampers throughout the year. On the other hand, in Chennai with high outdoor air temperatures and relative humidity, the co-simulations predicted that the windows/dampers to operate less frequently compared to the rest of the cities.

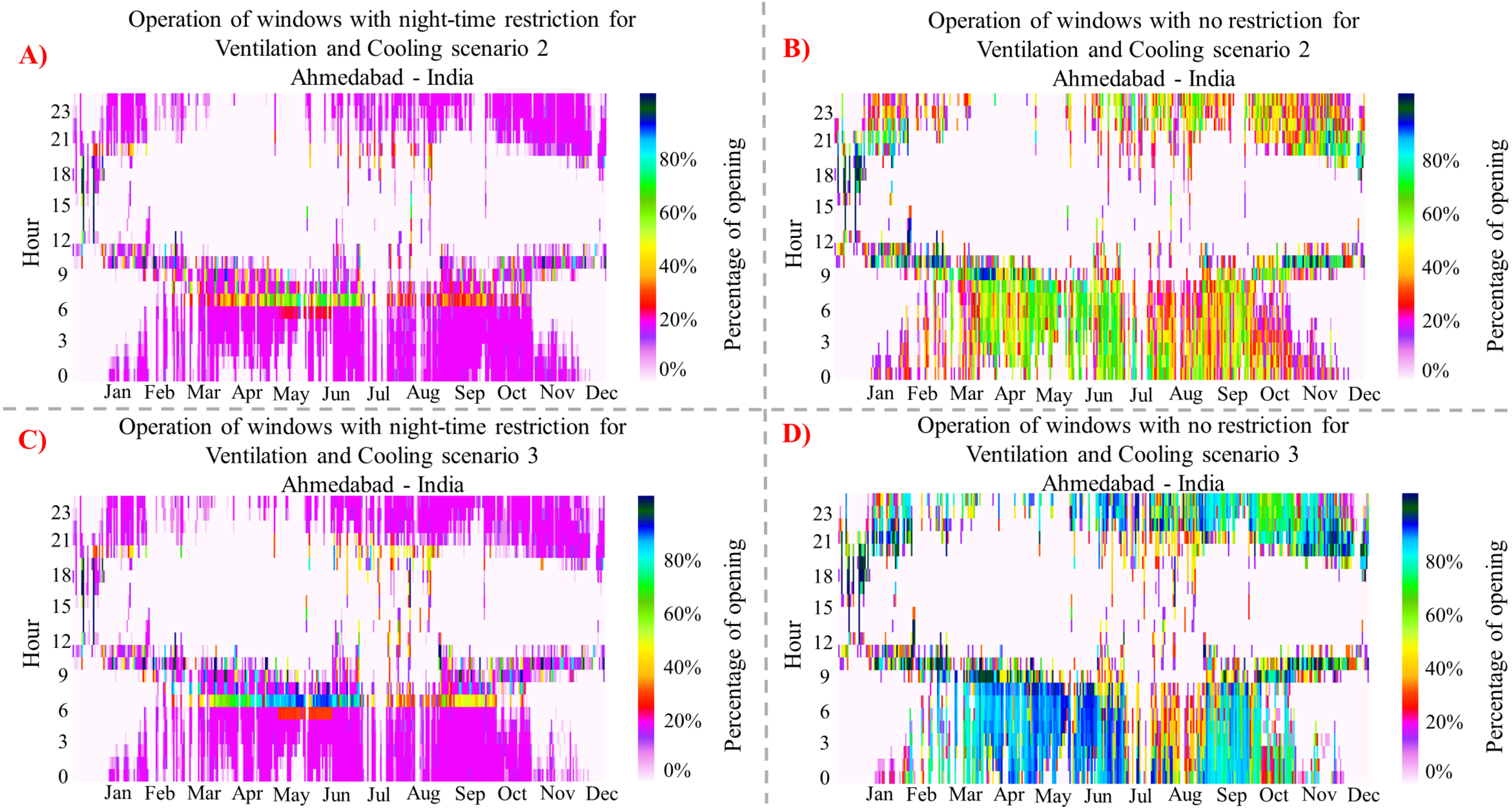


Figure 6-10:Percentage of window opening for different control algorithms with and without nighttime window restriction for ventilation and cooling VCS 2 and VCS 3 for Ahmedabad, India when split A/C units were used as the cooling systems.

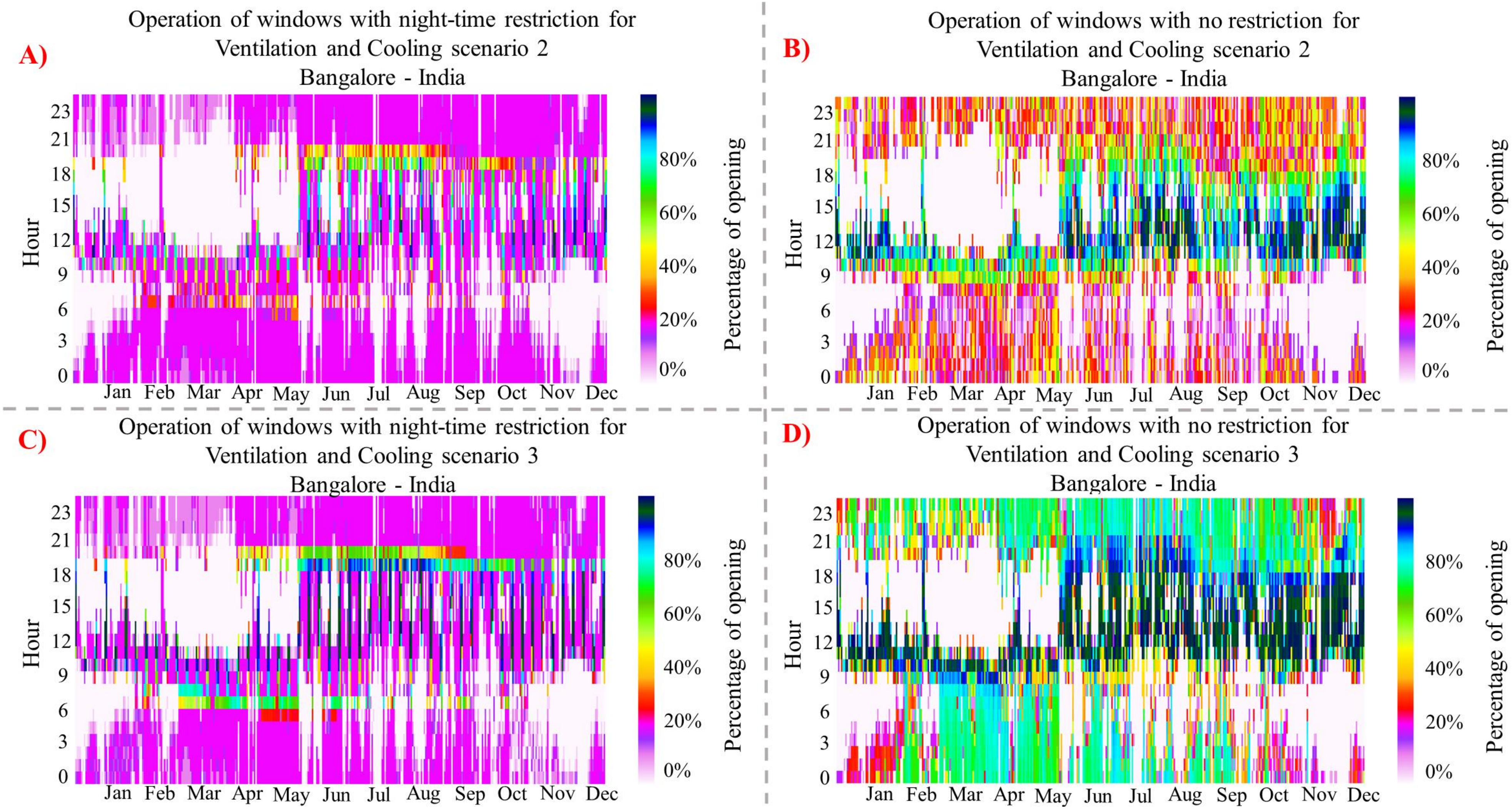


Figure 6-11:Percentage of window opening for different control algorithms with and without nighttime window restriction for ventilation and cooling VCS 2 and VCS 3 for Ahmedabad, India when split A/C units were used as the cooling systems.

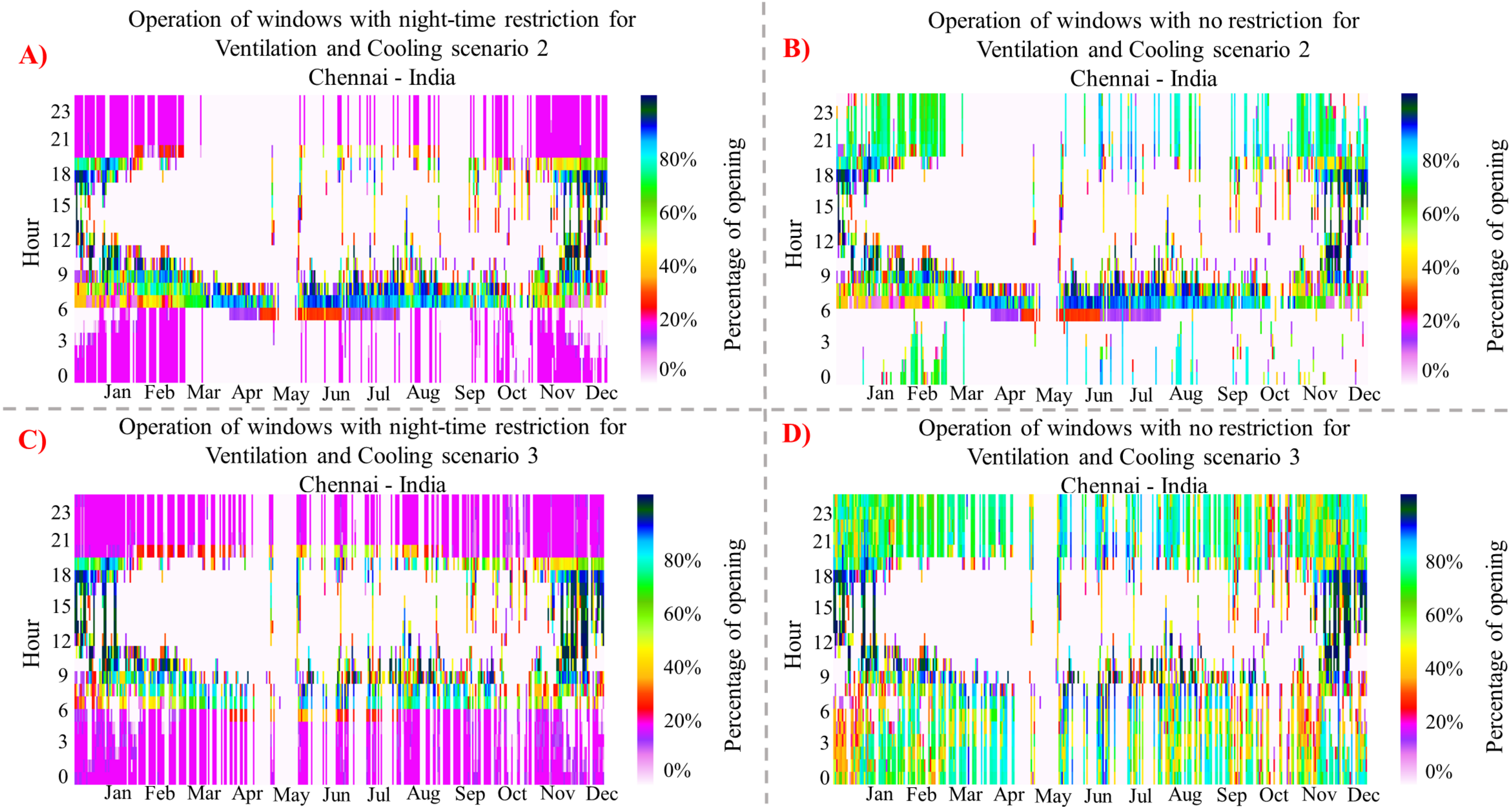


Figure 6-12: Percentage of window opening for different control algorithms with and without nighttime window restriction for VCS 2 and VCS 3 for Chennai, India when split A/C units were used as the cooling systems.

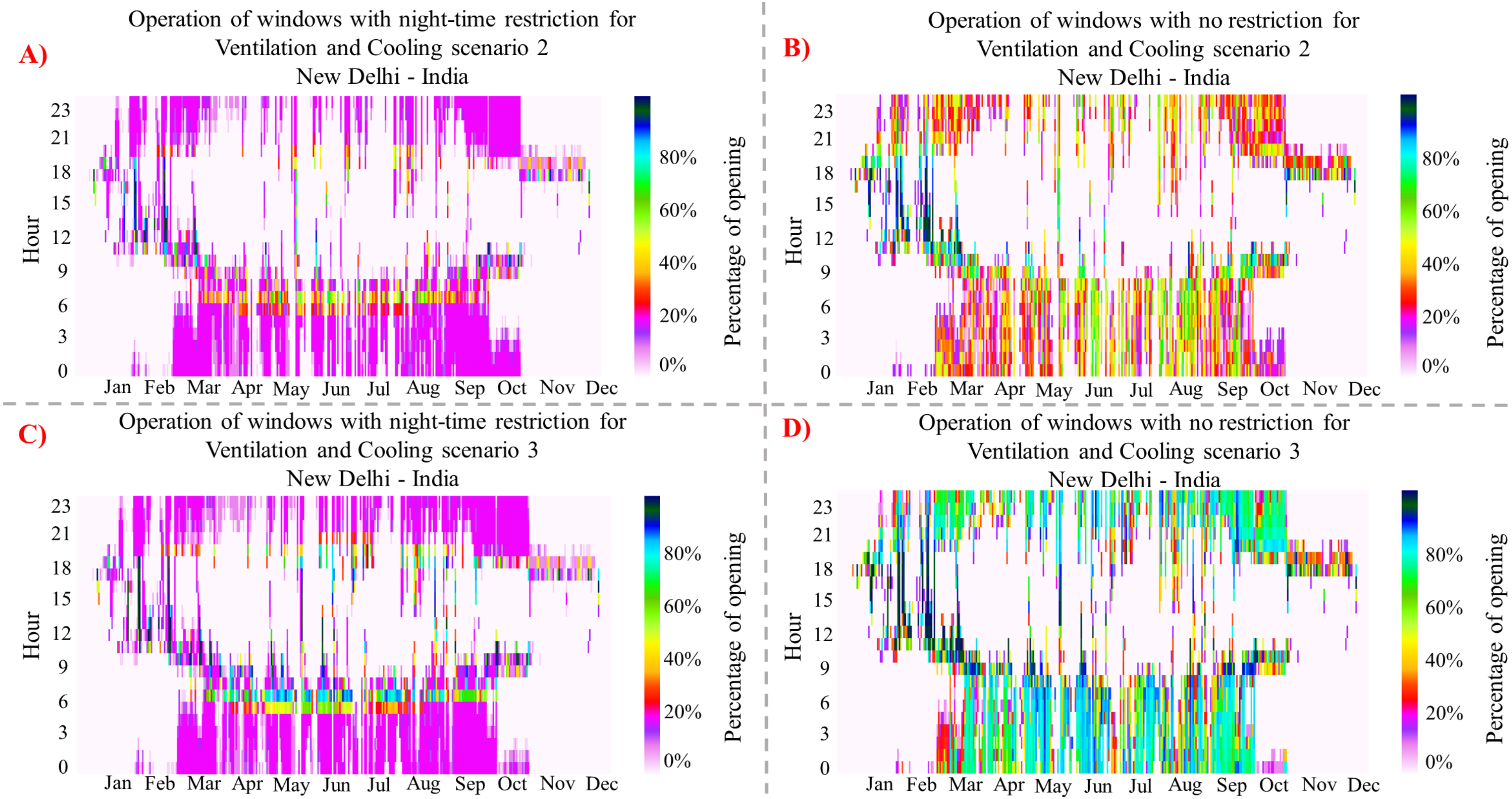


Figure 6-13:Percentage of window opening for different control algorithms with and without nighttime window restriction for VCS 2 and VCS 3 for New Delhi, India when split A/C units were used as the cooling systems.

These graphs (figure 6-10 to 6-13) provided a visual aid on the performance of windows/dumpers for different ventilation strategies and controls. To have a deeper understanding of the performance of the different ventilation and cooling strategies in each city, a detailed analysis of the internal air temperature and hours of thermal comfort was performed. In order to calculate the hours of natural ventilation, the data from the co-simulations were analysed. The hourly data of the operation of the windows were measured to calculate the hours of natural ventilation. The hourly average zone internal air temperature was compared against the hourly comfort temperatures proposed by the IMAC model for 90% acceptability. This analysis enabled to calculate the hours of natural ventilation and the hours of the year the control algorithms maintained comfortable conditions.

Table 6-2 & 6-3 summarize the hours of the year natural ventilation was preferred over the use of split units and evaporative coolers respectively, to provide ventilation/cooling and maintain the internal conditions within the comfort limits as a result of the controls. N*atural ventilation hours* are the hours of the year that the co-simulations predicted exclusively natural ventilation/cooling mode. C*omfort hours* are the total amount of hours that the internal operative temperature was within the comfort limits, provided by IMAC thermal comfort model, during the natural ventilation operational mode. As it has been explained in Section 2.4, the windows/dampers operated only during occupied hours, and hence the percentage refers to the total numbers of hours the windows/dampers operated.

Table 6-2: Total number of hours of natural ventilation and comfort hours for all cities when split A/C units were used as the cooling systems.

City	**Ahmedabad**		**Bangalore**		**Chennai**		**New Delhi**	
Window opening percentage	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)
VCS 1								
Comfort hours [h]	1287	1557	2788	2977	948	1458	986	1247
Natural Ventilation hours [h]	2247	2398	4673	4879	1797	2647	1876	2247
Comfort hours/ (Natural Ventilation hours)%	**57.3%**	**64.9%**	**59.7%**	**61.0%**	**52.8%**	**55.1%**	**52.6%**	**55.5%**
VCS 2								
Comfort hours [h]	1758	2018	3578	3987	1267	1787	1685	1934

Natural Ventilation hours [h]	2378	2680	4757	5087	2058	2700	2230	2487
Comfort hours/ (Natural Ventilation hours)%	**73.9%**	**75.3%**	**75.2%**	**78.4%**	**61.6%**	**66.2%**	**75.6%**	**77.8%**
VCS 3								
Comfort hours [h]	2457	2788	4673	5304	2449	2946	1968	2315
Natural Ventilation hours [h]	2745	3058	5087	5677	3126	3575	2384	2679
Comfort hours/ (Natural Ventilation hours)%	**89.5%**	**91.2%**	**91.9%**	**93.4%**	**78.3%**	**82.4%**	**82.6%**	**86.4%**
VCS 4								
Comfort hours [h]	1977	2249	3973	4079	2143	2317	1578	1789
Natural Ventilation hours [h]	2687	2788	4869	4939	2887	2967	2047	2177
Comfort hours/ (Natural Ventilation hours)%	**73.6%**	**80.7%**	**81.6%**	**82.6%**	**74.2%**	**78.1%**	**77.1%**	**82.2%**

Table 6-3: Summary table of the total number of hours of natural ventilation and comfort hours for all the cities when evaporative coolers were used as the cooling systems.

City	**Ahmedabad**		**Bangalore**		**Chennai**		**New Delhi**	
Window opening percentage	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)	Nighttime restriction (up to 20%)	No-restriction (up to 100%)
VCS 1								
Comfort hours [h]	1205	1457	2610	2786	887	1365	923	1167
Natural Ventilation hours [h]	2202	2350	4580	4781	1761	2594	1838	2202
Comfort hours/ (Natural Ventilation hours)%	**54.7%**	**62.0%**	**57.0%**	**58.3%**	**50.4%**	**52.6%**	**50.2%**	**53.0%**
VCS 2								
Comfort hours [h]	1645	1889	3349	3732	1186	1673	1577	1810
Natural	2330	2626	4662	4985	2017	2646	2185	2437

Ventilation hours [h]								
Comfort hours/ (Natural Ventilation hours)%	**70.6%**	**71.9%**	**71.8%**	**74.9%**	**58.8%**	**63.2%**	**72.2%**	**74.3%**
VCS 3								
Comfort hours [h]	2346	2662	4462	5064	2338	2813	1879	2210
Natural Ventilation hours [h]	2696	3003	4996	5576	3070	3511	2341	2631
Comfort hours/ (Natural Ventilation hours)%	**87.0%**	**88.6%**	**89.3%**	**90.8%**	**76.2%**	**80.1%**	**80.3%**	**84.0%**
VCS 4								
Comfort hours [h]	1822	2073	3662	3759	1975	2135	1454	1649
Natural Ventilation hours [h]	2622	2721	4751	4819	2817	2895	1997	2124
Comfort hours/ (Natural Ventilation hours)%	**69.5%**	**76.2%**	**77.1%**	**78.0%**	**70.1%**	**73.8%**	**72.8%**	**77.6%**

As expected, the city of Bangalore, with moderate climatic conditions, predicted to have the highest number of hours of natural ventilation compared to the rest of the cities with more extreme weather conditions. As expected, VCS 3 favours the use of natural ventilation for a longer period compared to VCS 2. Particularly the use of ceiling fans can result in almost 8-25% more use of natural ventilation compared to VCS 2 for all the examined cities and for both cooling systems (split A/C and DEC units). The higher percentages of thermal comfort were predicted in the city of Bangalore due to the moderate climatic conditions.

The most significant improvement of the predicted hours of natural ventilation was observed for the city of Chennai, with a warm and humid climate compared to the rest of the cities. The introduction of the dampers and ceiling fan, VCS 3, had a positive impact, as it increased the predicted hours of natural ventilation by almost 27% compared to VCS 1 when the split unit was used as the mechanical system. Chennai has a high mean outdoor air temperature throughout the year; hence the use of windows only was predicted unable to maintain a thermally comfortable internal environment. By introducing dampers (VCS 2) and ceiling fans (VCS 3), it was possible to utilize the use of natural ventilation and increase the predicted hours for thermal comfort by 18% and 16% for the split A/C and DEC units respectively. For Bangalore, with a moderate climate, the predictions of thermal comfort hours during the natural ventilation improved by approximately 16% when both dampers and ceiling fans were introduced (VCS 3) compared to VCS 2 when the split A/C units were used whilst for the DEC units the improvement was predicted to be 14%. Due to the moderate outdoor conditions, the predicted hours of natural ventilation were the highest compared to the rest of the cases, even when only the windows were used as passive systems.

The use of the evaporative cooler had an impact on the window operation (table 6-3). The co-simulations predicted that the city of Bangalore will have the most hours of natural ventilation in the year, due to its moderate climatic conditions. In general, the co-simulations predicted fewer hours of natural ventilation for all the cities compared to the cases with split units. A plausible explanation for this observation is that the use of evaporative cooler resulted in higher internal air temperatures compared to the cases with split units. Due to the higher internal air temperatures, the control algorithms decided to operate in the mechanical mode for more hours to maintain thermally comfortable internal environments, hence it limited the use of natural ventilation. In places with high levels of outdoor humidity, such as Chennai, the evaporative coolers did not manage to maintain the internal air temperatures within the comfort limits, and this affected the hours of natural ventilation and also the comfort hours. Although the current research proposed an advanced control approach for the DEC units, the high levels of outdoor humidity limited the availability of the system. However, the advanced control of the DEC units resulted in better performance of the units compared to the conventional approaches that can be found in most of the DTM tools (Angelopoulos *et al.*, 2019).

Overall, in all the climates studied, the nighttime restriction resulted in a small reduction of the predicted hours of natural ventilation ranging from 3-9% depending on the city and the VCS. By restricting the opening during the nighttime, less volume of the cooler outdoor air entered the apartment, resulting in higher internal air temperature compared to the cases without window restriction. Nighttime ventilation can cool down the thermal mass of the building, providing a “cool sink” for the following morning, which results in lower internal air temperature and hence less reliance on mechanical means for cooling. Additionally, the use of stand-alone mechanical ventilation and dehumidifier units slightly decreased the number of hours for natural ventilation as is shown in table 6-2 & 6-3. By removing humidity from the internal air, the internal air temperature increased. This could be an explanation of a slightly higher internal air temperature that resulted in fewer predicted hours of natural ventilation compared with VCS 3. The same observations were made for both cooling systems.

The hours during the natural ventilation mode, that the control algorithms managed to maintain the internal operative temperature within the comfort limits were further predicted. Traditionally PMV-PPD method is typically used for fully conditioned spaces, where there is no natural ventilation. For mixed-mode cases, as presented here, the most appropriate comfort models to use are the adaptive comfort models as the literature suggested (Deuble and de Dear, 2012; Sorgato *et al*., 2016; Babich *et al*., 2017). For this reason, table 6-2 & 6-3 present the number of hours of the year during only natural ventilation that the internal operative temperature is outside the comfort limits proposed by each adaptive model. In the IMAC thermal comfort model, 90% acceptability was considered. The use of windows and dampers as the only passive systems to provide natural ventilation proved inadequate to maintain the internal conditions within the desirable limits. Regardless of the climatic conditions or cooling system, the analysis showed that for approximately 60-75% of the hours the natural ventilation mode was preferred and the internal operative temperature was within the limits. In Bangalore with moderate climatic conditions, the co-simulations predicted the highest percentage of comfort hours for both cooling systems but were still outside the desirable limits. The results showed that the windows/dampers solely cannot provide enough outside air to mitigate the internal heat gains, the high levels of the outdoor air temperature and of relative humidity. In addition, considering the strict comfort limits, imposed by IMAC model, the analysis showed that it is not feasible to rely only on the windows/dampers to achieve comfort conditions. The use of additional systems to improve the internal conditions of the flat, during the natural ventilation mode, was deemed essential for all the climatic conditions. The introduction of the ceiling fans in the bedrooms and living rooms, as discussed in section 4.2.2, resulted in higher setpoint temperatures meaning that the natural ventilation was preferred for a wider range of external temperatures. This justifies the higher percentages of thermal comfort conditions predicted for the VCS 3 over the VCS1 and VCS2. The inclusion of the ceiling fans in the control algorithms increased the number of comfort hours for all the cities and for both cooling systems. The most significant improvement was observed for the moderate climate of Bangalore and the hot and dry climate of Ahmedabad. In the cases without any window restriction, the analysis showed that it is feasible to meet the desirable limits for 90% acceptability for Bangalore and Ahmedabad for both cooling systems. Furthermore, for the other two locations, Chennai and New Delhi, the analysis highlighted the potential of the ceiling fan as an additional system to improve the performance of the mixed-mode ventilation and cooling strategies. The co-simulations predicted that the comfort hours can be up to 85% of the time, which is very close to the very strict acceptability limit.

In general, the cases with the split A/C units resulted in higher percentages of thermal comfort compared to the cases with the DEC units by approximately 7-11% depending on the location. The co-simulations predicted that the split units can maintain internal air temperatures at lower levels compared to the evaporative coolers. This has an impact also when the natural ventilation was selected as the operation mode. When the mechanical-mode was selected, the split units maintained internal air temperatures in lower levels, and when the natural ventilation was used, it was “easier” to maintain the internal air temperature within the comfort limits. This phenomenon occurs because the development of the internal air temperature is a dynamic process, and hence is affected by different parameters. So it is important when designing a mixed-mode ventilation strategy to include both the natural and the mechanical systems in order to better design the most appropriate control algorithm. Similarly, when there was no restriction in the opening area of the windows, the analysis showed higher percentages of thermal comfort compared to the nighttime restriction.

A parameter to analyse in order to assess the performance of mixed-mode ventilation control strategies is their ability to utilize the use of natural ventilation and also to maintain the internal environment within the comfort limits. However, the co-simulations predicted that natural ventilation solely cannot maintain acceptable indoor conditions for the Indian cities, especially during the daytime, and hence for this period mechanical systems should be used. The analysis of the energy-saving potential, that each control strategy can achieve, is paramount importance when selecting the most suitable ventilation strategy for a city.

6.3.3. Comparison of energy-saving potential of the different ventilation and cooling strategies

Energy-saving potential for the different control algorithms, ventilation and cooling scenarios and examined climates are presented here. The co-simulations were performed for the whole year, and the predictions of the total energy demand were compared against the base case, which was the fully mechanical mode case with no window operation.

Figure 6-14 presents the relative energy savings against the base case scenario (red bar). For all VCSs the difference between nighttime restriction and no restriction of the window openings is approximately between 5-10%. As expected, when there is no restriction at the openings the total energy demand is lower since the building thermal mass cools during nighttime. This could be also justified by the predictions of the hours of natural ventilation. The co-simulations predicted more hours of natural ventilation when there was no restriction on the windows' openings and hence fewer hours of mechanical mode compared to the cases with nighttime restriction. However, the difference in the energy savings between these two cases is not significant, mainly because it is assumed that the free area of the window is 50% of the geometric area of the window (slide window) and the effective area of the window is the product of the free area multiplied by the discharged coefficient. To achieve a greater variation in the results based on the window opening area, a different type of window has to be selected which will result in a bigger effective area. As expected, the co-simulations predicted that the highest energy savings can be achieved in Bangalore, since the moderate climatic conditions favour the use of natural ventilation for a longer period compared to the rest of the cities.

The use of ceiling fans (VCS 3 and VCS 4) resulted in the highest percentages of energy savings for all the cities and cooling systems, approximately 40-55% compared to the base case scenario. The ventilation and cooling setpoints due to the air movement proved to be the most efficient measure to reduce the use of mechanical cooling systems. For instance, by including the ceiling fan as part of the control strategies, the co-simulations predicted an additional 20-25% energy savings compared to the cases without the ceiling fan (VCS 2). The positive effects of the use of the ceiling fans are more significant in warm and humid climatic conditions, such as Chennai, since the air movement increases the convective and evaporative heat exchange between the occupant and the environment. The convective heat exchange is driven by the temperature difference whilst the evaporative by the water vapour pressure difference. For the same temperature difference, the air motion has a more positive effect on the humid climates compared to the drier climates. The occupants in warm and humid climates are more likely to use higher fan speeds to maintain the same levels of thermal comfort for the same temperature compared to the occupants in drier places. This can be reflected also in the predictions of the energy savings since the co-simulations calculated that the inclusion of the ceiling fans can result to up 25% additional energy savings for Chennai and New Delhi whilst for Bangalore or Ahmedabad the additional savings were up to 20%.

Additionally, the use of stand-alone mechanical ventilation and dehumidifier units slightly decreased the potential of energy savings as shown in figure 6-14. This can be explained due to the thermodynamic process of a dehumidifier. Basically, the humid air of the room enters the dehumidifier where it’s cooled to its dew point which results in releasing its moisture. The dry air is then heated, due to latent heat which is a natural process that occurs in the condenser. Also, the compressor releases heat which can result in higher air temperature of the exhaust air compared to the inlet temperate. This could be an explanation of a slightly higher internal air temperature that resulted in fewer predicted hours of natural ventilation compared with VCS 3.

The use of split units resulted in higher energy-saving potential compared to the evaporative cooler. This can be explained because the analysis showed that the split units can maintain the internal air temperature in lower levels compared to the DEC units. This favoured the use of natural ventilation for longer periods of time and hence the mechanical systems operated for fewer hours. Furthermore, the evaporative coolers are affected by the levels of relative humidity in the air. Direct evaporative cooling occurs when the water and the air come into direct contact, and the transfer of energy from the air to the water takes place when the air has relative humidity less the 100% (Jain and Hindoliya, 2014). In a DEC system, a fan forces the air through a wet surface for evaporation. The heat and mass transfer between air and water results in a decrease of the air dry-bulb temperature and increase of its humidity levels, and in an ideal case, this process is adiabatic (Watt and Brown, 1997). The minimum temperature that can be reached is determined by thermodynamics and is the wet-bulb temperature of the incoming air. Consequently, this process is more efficient when the levels of the relative humidity of the incoming air to the evaporative cooler are low. In cases with high levels of relative humidity, the efficiency of the evaporative coolers drops and this is reflected also in their ability to maintain the internal air temperature within the comfort limits. This is an explanation of the lower predictions of energy savings in cities with high levels of outdoor relative humidity.

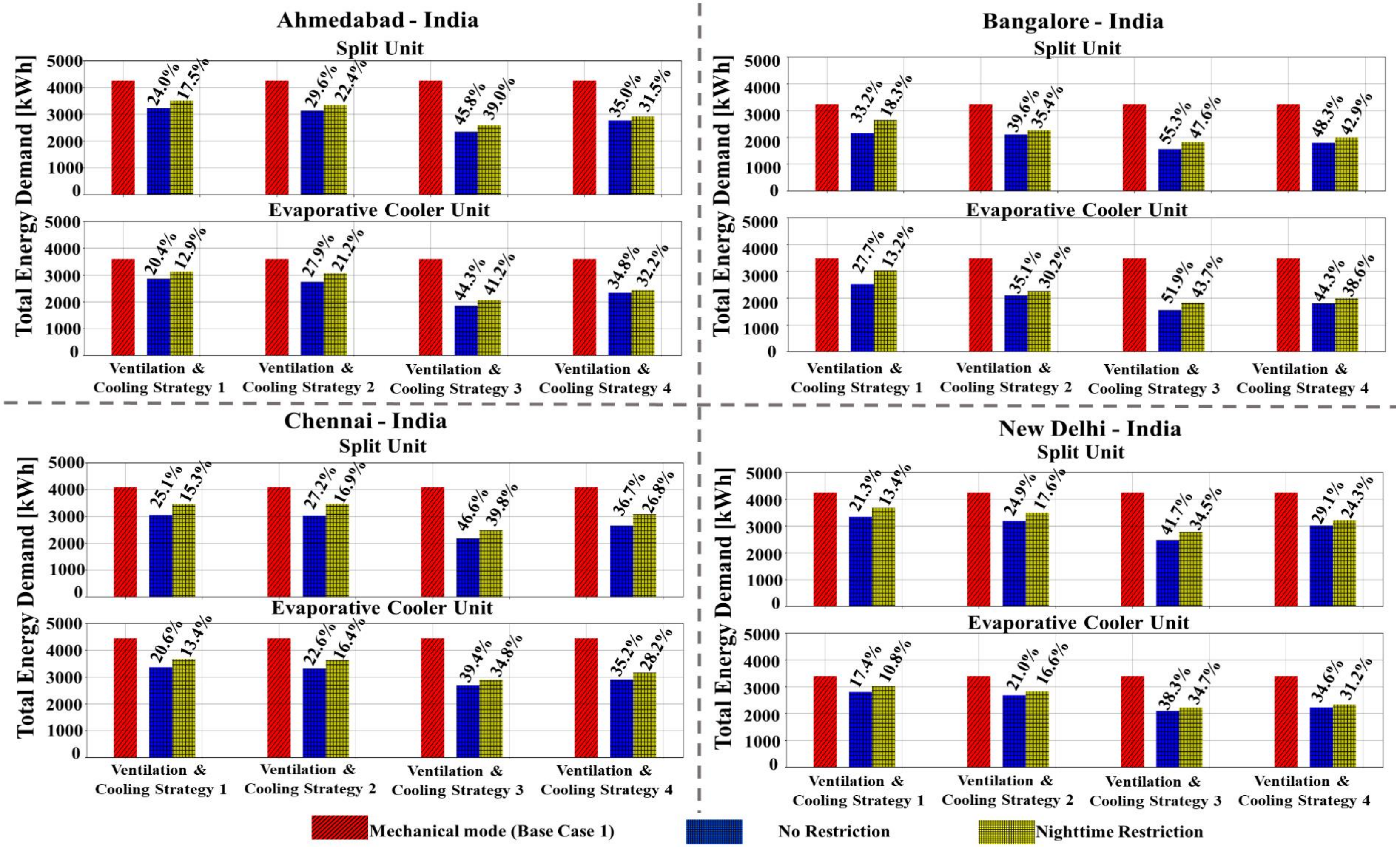


Figure 6-14: Energy-saving potential for the different ventilation and cooling strategi

6.4. Summary

This chapter illustrates the predicted energy savings, hours of thermal comfort that can be achieved for a typical 2-bedroom apartment in different climatic conditions. The results from this study show that it is possible to achieve energy savings for mixed-mode buildings while maintaining thermally comfortable internal conditions. The potential of the examined ventilation and cooling strategies to achieve energy savings has been demonstrated through co-simulations. The performance of the proposed control algorithms for mixed-mode ventilation and cooling strategies was evaluated with respect to the energy savings and the predicted comfort hours.

Using the co-simulations it was predicted that by controlling only the openings, VCS 2, (windows and dampers) up to 25% reduction in the energy consumption was achieved depending on the location of the building when AC split units were used as the cooling systems. The moderate climate of Bangalore favoured more the use of day and nighttime ventilation, whilst more extreme climatic conditions (hot and dry, warm and humid) relied more on the mechanical cooling systems to maintain comfortable internal conditions. The use of dynamic setpoints for cooling, based on the adaptive theory, improved significantly the predictions of the energy savings over the static setpoints, but more importantly, it captures the existing connection between the occupants of mixed-mode buildings and their interaction with the natural or mechanical systems. When DEC units were used, the co-simulations predicted energy savings up to 21% for VCS 2. The use of DEC unit as the cooling systems resulted in slightly lower energy savings since the higher levels of relative humidity restricted the use of the DEC unit and the internal air temperature was higher.

The inclusion of the ceiling fans (VCS 3) as part of the control algorithm reduced the energy predictions for cooling demand by 45%, 55%, 47% and 42% for Ahmedabad, Bangalore, Chennai and New Delhi when the split AC units were used. When the DEC units were used, the co-simulations predicted 44%, 52%, 39% and 38% for Ahmedabad, Bangalore, Chennai and New Delhi compared to the base case scenarios. As previous studies have highlighted, the elevated internal airspeeds have positives impact on the thermal sensation of the occupants, and hence when the ceiling-fan operated, the control algorithms increased the setpoint temperatures accordingly to the airspeed.

The hours of thermal comfort that each VCS maintained were analysed. The inclusion of the ceiling fans in the control algorithms resulted in 1100h more comfort hours, compared to the scenarios when the ceiling fans were turned off, in Ahmedabad, and these findings are in line with previous research (Babich *et al.*, 2017). For the rest of the cities, the inclusion of the ceiling fans resulted in 1900h, 1500h and 1000h more hours of thermal comfort for Bangalore, Chennai and New Delhi respectively. The operation of the ceiling fans had a positive impact, as the internal airspeeds allowed higher cooling temperatures to achieve the same levels of thermal comfort. In Bangalore, with a moderate climate, the inclusion of the fan as part of the control algorithms resulted in 93% of comfort hours for the whole year. When DEC units were used as the cooling systems, the co-simulations predicted fewer hours of thermal comfort compared to the split unit scenarios. The elevated levels of relative humidity in addition to the slightly higher predicted internal air temperatures compared to the split units resulted in approximately 8-13%, 7-15%, 19-24%, 17-22% fewer hours of thermal comfort for Ahmedabad, Bangalore, Chennai and New Delhi respectively. With regards to thermal comfort and energy savings, it is proposed to use the adaptive thermal comfort theory to calculate the cooling setpoints for all the locations.

For all VCSs the difference between nighttime restriction and no restriction of the window openings is approximately between 5-10% when comparing the energy predictions. As expected, when there is no restriction at the openings the total energy demand is lower since the building thermal mass cools during nighttime. This could be also justified by the predictions of the hours of natural ventilation. The co-simulations predicted more hours of natural ventilation when there was no restriction on the windows' openings and hence fewer hours of mechanical mode compared to the cases with nighttime restriction. Similarly, when there was no restriction in the opening area of the windows, the analysis showed higher percentages of thermal comfort compared to the nighttime restriction. In all the VCS, the use of nighttime window restriction resulted in higher internal air temperature, due to lack of sufficient cooler air flowing in and reducing the temperature of the thermal mass. However, that seems the most reasonable approach that the occupants of a flat will control their systems, especially for safety reasons. Hence, it is important to design the control algorithms considering both cases, with and without restriction, to have a better understanding of the performance of the control algorithms.

As the analysis showed, when designing control algorithms for mixed-mode buildings, it is essential to include all the systems (passive and active) as each of them have an impact on the performance of the other and together impact the performance of the control algorithms. It is crucial to understand the performance of the mechanical systems and to control them in such a way that will minimize their influence during the natural ventilation mode when designing the control algorithms of mixed-mode buildings. Based on this research, it is proposed to include the ceiling fans as part of the control algorithms, as it improves significantly the predictions for energy savings and comfortable hours. In cities with extreme weather conditions, such as Ahmedabad, Chennai and New Delhi, the use of the split units seemed the most appropriate as the maintained the internal air temperatures within the comfortable limits for more hours throughout the year compared to the DEC unit cases whilst achieved less energy consumption. For the moderate climate of Bangalore, both the DEC and split units maintained the internal temperatures within the comfort limits for a similar period of time whilst achieving similar energy savings.

Chapter 7. Ventilation performance analysis using CFD of a two-bedroom apartment

7.1. Overview

Analysis of the ventilation performance of the two-bedroom apartment (demonstration case) using CFD simulations was carried out and is presented here. The examined cases were selected based on the co-simulations presented in Chapter 6 and consist of a variety of outdoor conditions, window and dampers opening sizes. Steady-state CFD simulations were performed. A mesh sensitivity study for the geometry used demonstrates the mesh density used for the simulations. Ventilation performance was assessed using the internal air temperature and airspeed, the heat removal (HRE) and contamination removal effectiveness, and CRE respectively.

It was important to perform CFD simulations to examine the impact of the different control algorithms in the thermal comfort conditions. The use of the co-simulations in Chapter 6 could not provide detailed information regarding temperature stratification and ventilation effectiveness. The co-simulation cannot assess these parameters because they can assume an even distribution of temperature in the space. Hence the use of CFD was essential in order to examine the ventilation performance of the proposed control algorithms.

7.2. Mesh sensitivity study

To ensure that the results were independent of the mesh resolution, six different mesh densities were investigated as shown in table 7-1 for the buoyancy-driven cases with 15% increase in cells from the previous mesh size, evenly distributed on the “X” and “Y” axis. The size of the domain was:

$$[X, Y, Z] = [10.1, 14.4, 2.75]\ [m, m, m]$$

As stated in section 3.6.4.2 the spot values, the source balance and the residuals’ behaviour were analysed to examine how the mesh densities generated stable results. This analysis showed the mesh densities that provided accurate results in less computational time. The variables chosen for this mesh sensitivity analysis include the air temperature and air velocity at the probe position. All mesh densities reached the acceptable level of convergence as defined in section 3.6.4.2 with an exception of mesh B-6. The lack of convergence was due to high residual errors. Although the relaxation factors were reduced to 0.1 and the mesh structure was adjusted at the openings, the residual error did not reach the limits to assume convergence for the simulations.

Table 7-1: Summary of different mesh densities and computational time for buoyancy-driven cases

Mesh number	**Number of cells [X,Y,Z] - Total**	**Computational time [h:m]**
B-1	[51,72,37] – 135864	2:58
B-2	[59,83,43] – 206632	4:31
B-3	[67,96,49] – 314262	5:48
B-4	[78,110,56] – 477953	7:42
B-5	[89,126,65] – 726906	13:28
B-6	[103,145,74] – 1105534	17:26

Table 7-2: Summary of source balance and residual behaviour for the examined mesh densities for buoyancy-driven cases.

	Source balance (kg/s)			**Residual behaviour**		
	Positive Sum	**Negative Sum**	**Error (%)**	**Initial**(10^5)	**Final** (10^5)	**error (%)**
1	0.790954	-0.79095	<0.1	2.2893	-2.28865	<0.1
2	0.67442	-0.67442	<0.1	1.956029	-1.95596	<0.1
3	0.687866	-0.68788	<0.1	1.994337	-1.99435	<0.1
4	0.741807	-0.73972	<0.1	2.145667	-2.19917	<0.1
5	0.762505	-0.76919	0.8	2.294884	-2.30769	0.6
6	1.079775	-1.14823	6.0	2.962924	-3.35927	11.8

Further, as indicated in figure 7-1 and table 7-2, the residual error reduction within the computational time for all the mesh densities was acceptable with an exception in mesh B-5 & B-6. The number of iterations for all the mesh densities was 10.000, and based on the data in figure 7-1 and table 7-2 the results for the mesh densities 1 to 4 have achieved convergence as the residuals of mass and enthalpy equations were below the limit set (section 3.6.4.2) in the convergence criteria, suggesting that the CFD will accurately predict the results.

Mesh 3

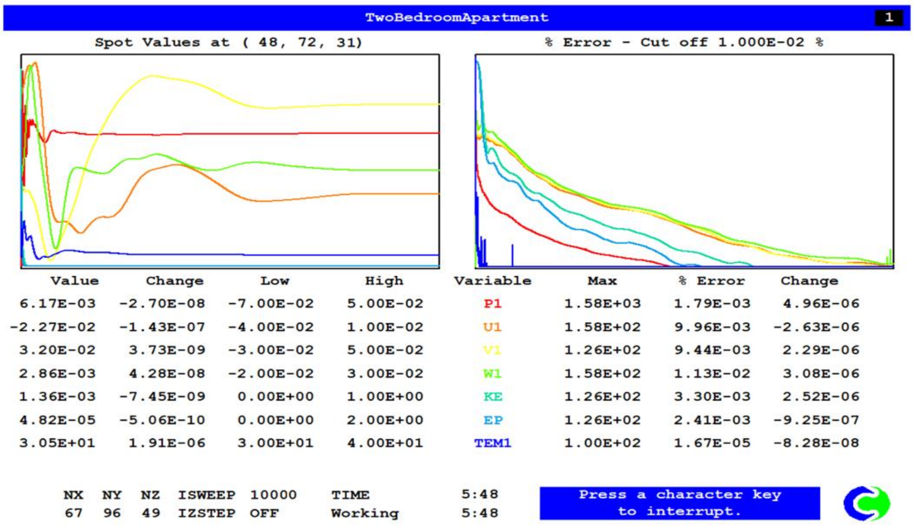


Figure 7-1: Spot values for mesh density B-3 for the buoyancy-driven cases.

As shown in figure 7-2, the internal air temperature had a maximum variance of around 9% predicted between mesh 1 and 4 and a minimum of 1.8% predicted between B-3 and B-4. Further, for the air velocity graphs, the maximum variance was calculated to be 15% between mesh 1 and 4. The simulation time for each case is presented in table 7-1.

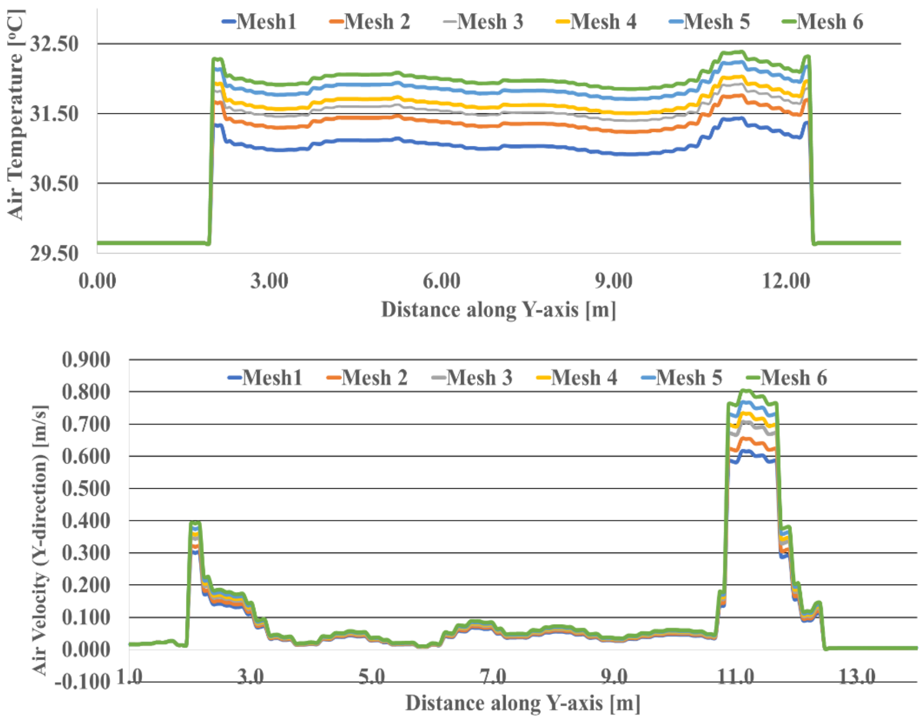


Figure 7-2: Air temperature (top) and velocity parameter(bottom) for mesh sensitivity study for the buoyancy-driven cases.

The small difference in the results between the B-3 and B-4 suggested that a further increase in the mesh density will have a minor impact on the results. The mesh that was selected for the buoyancy-driven simulations was **B-3** as it needs almost 25% less computational time, compared to mesh 4, to generate as accurate results.

The whole domain approach was used, and it was essential to have a large domain to allow the wind profile to develop fully. At the whole domain approach, the internal and external environments are simultaneously modelled, allowing prediction of the airflow around the building, through the ventilation openings, and at the interior of spaces.

Based on previous studies, the domain size was proportional to the building's height (H) and more specifically the X-direction was 9H, the Y-direction was 13H and the Z-direction was 4H (Evola and Popov, 2006). The size of the domain was:

$$[X, Y, Z] = [24.75, 35.75, 11]\ [m, m, m]$$

A similar analysis for the mesh independent was done for the wind-driven cases. To ensure that the results were independent of the mesh resolution, five different mesh densities were investigated as shown in table 7-3 for the wind-driven cases with 15% increase in cells from the previous mesh size in each direction, being evenly distributed on the "X" and "Y" axis.

Similar to the buoyancy cases, the spot values, the source balance and the residuals' behaviour were analysed to conclude if the examined mesh densities generated stable results. A convergence check was made for all the mesh densities, and all the results have reached an acceptable level of convergence (see section 3.6.4.2) with an exception of W-5.

Table 7-3: Summary of different mesh densities and computational time for the wind-driven cases

Mesh number	Number of cells [X,Y,Z] - Total	Computational time [h:m]
W-1	[78,102,72] – 572832	6:02
W-2	[90,117,83] – 871206	8:47
W-3	[103,135,95] – 1324995	11:35
W-4	[119,155,110] – 2015152	21:56

W-5	[136,178,126] – 3064794	26:45

Table 7-4: Summary of source balance and residual behaviour for the examined mesh densities for the wind-driven cases.

	Source balance (kg/s)			**Residual behaviour**		
	Positive Sum	**Negative Sum**	**error(%)**	**Initial**(10^5)	**Final** (10^5)	**error(%)**
1	3.322007	-3.32199	<0.1	9.61506	-9.61233	<0.1
2	2.832564	-2.83256	<0.1	8.215322	-8.21503	<0.1
3	2.889037	-2.8891	<0.1	8.376215	-8.37627	<0.1
4	3.115589	-3.10682	0.28	9.011801	-9.14608	1.49
5	3.202521	-3.74011	16.79	9.638513	-12.5720	30.45

As indicated in figure 7-3 and table 7-4, the residual error reduction within the computational time for all the mesh densities was acceptable with an exception in W-5. Although several attempts were made to improve the accuracy of W-5, all residual errors remained very high. The number of iterations for all the mesh densities was 10.000, and based on the data in figure 7-3 and table 7-4, the results for the mesh densities W-1 to W-4 have achieved convergence as the residuals of mass and enthalpy equations were below the limit set in the convergence criteria, suggesting that the CFD will accurately predict the results.

Mesh 2

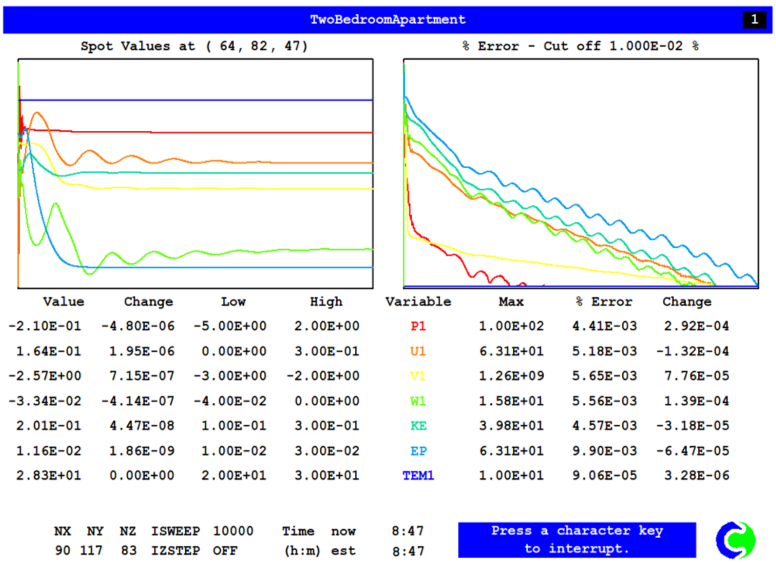


Figure 7-3: Spot values for mesh density W 2 for the wind-driven cases.

For the wind-driven CFD mesh sensitivity study, figure 7-4 shows the variance of the internal air temperatures for the five meshes examined. The maximum variance of around 6.5% was predicted between mesh 1 and 4. Meshes 2 and3 had very low variance (~1.5%). These levels of variance are very small and hence a less detailed mesh can be used. Similarly, for the air temperature graph, the maximum variance was calculated to be 15% between mesh 1 & 4 for the air velocity in the Y-direction. The simulation time for each case is presented in table 7-3.

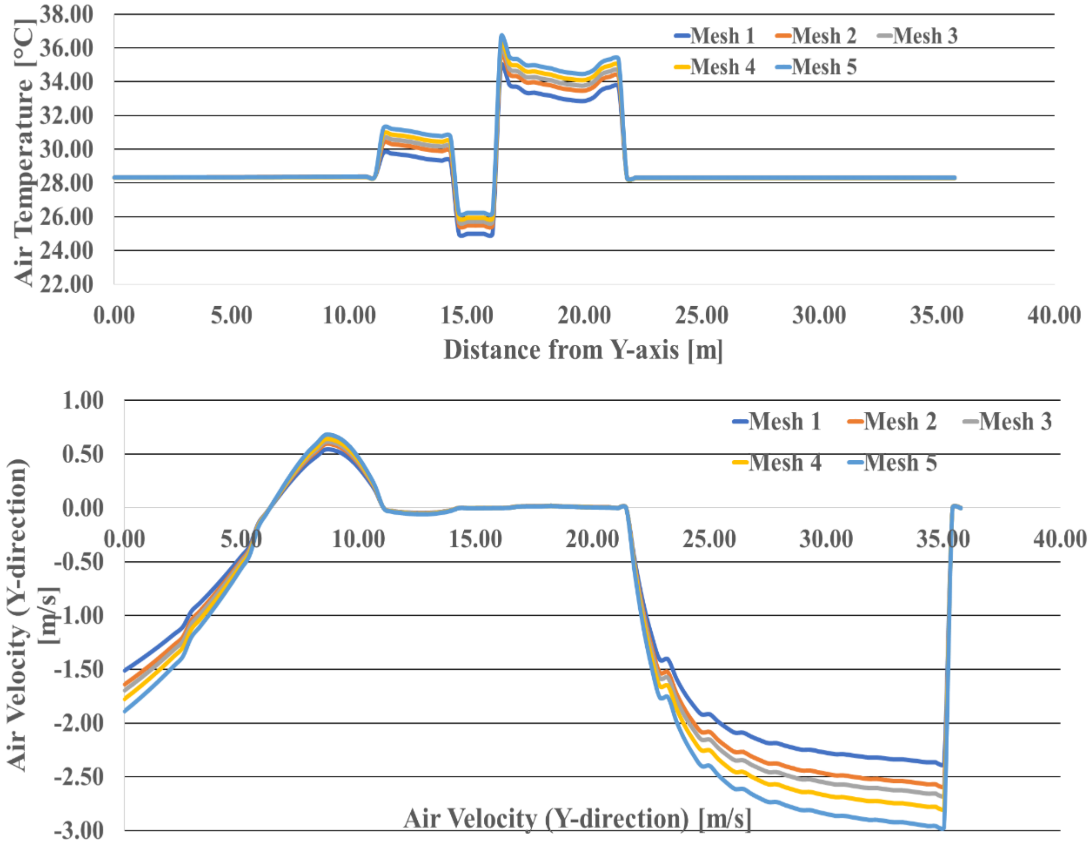


Figure 7-4: Air temperature (top) and velocity parameter(bottom) for mesh sensitivity study for the wind-driven cases.

The small difference in the results between the W- 2 and W-3 suggested that a further increase in the mesh density will have a minor impact on the final results. The mesh that was selected for the rest of the wind-driven simulations was **mesh W-2** that required 25% less computational time, compared to mesh W-3 to generate accurate results.

7.3. Examined scenarios for the CFD analysis

To select the cases for the CFD analysis, the co-simulation results from section 6 were analysed. Specifically, the results from the co-simulations for the city of Ahmedabad were analysed. For this analysis, the timestep data from the co-simulations were used. The results were analysed and clustered into groups based on the outside air temperature. The outside air temperature was grouped into 1°C interval and for each group, the number of hours the windows and/or dampers operated was counted (table 7-5). This analysis allowed the calculation of the frequency that each scenario occurred, and then the cases with the highest frequencies highlighted cells in table 7-5 were selected to be simulated into the CFD. For the wind-driven cases, only 1 outdoor temperature was used. Based on the co-simulations results, the wind speed for that specific temperature was used in varying directions. Furthermore, to examine in detail how the airflow patterns are affected within the apartment, in buoyancy-case 6 & 7 the same boundary conditions were used with an exception of the internal doors. In buoyancy-case 6 the internal doors kept open allowing cross-ventilation flows, whilst in buoyancy-case 7 the internal doors kept closed restricting the flow between each room. This allowed a direct comparison of the results. The same occurred for the wind-cases 1-4 where the internal doors kept open and wind-cases 5-8 where the internal doors were closed. The selected cases for the CFD simulations are presented in table 7-6. The selection of these specific cases was made in order to represent a variety of combinations for natural and mixed-mode ventilation strategies and to capture the cases that the co-simulations predicted to occur more frequently.

Table 7-5: Detailed analysis of co-simulation results. The highlighted cells (green colour) are those which were selected to be simulated into the CFD.

																	Hours
Window	Closed	Closed	Closed	80-100% Open	80-100% Open	60-80% Open	60-80% Open	40-60% Open	40-60% Open	40-60% Open	40-60% Open	Closed	Closed	Closed	Closed	Closed	
Damper	Closed	Closed	Closed	80-100% Open	80-100% Open	80-100% Open	80-100% Open	60-80% Open	80-100% Open	60-80% Open	80-100% Open	20-40% Open	40-60% Open	40-60% Open	60-80% Open	60-80% Open	
AC	ON	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	OFF	
Ceiling fan	ON	ON	OFF	ON	OFF	ON	OFF	OFF	OFF	ON	ON	ON	OFF	ON	OFF	ON	
Temperature [-) [°C]																	
9-10	2.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2.1
10-11	11.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	11.6
11-12	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
12-13	108.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	108.5
13-14	79.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	79.5
14-15	105.9	0.0	0.0	0.0	0.0	0.0	0.2	0.0	0.6	0.0	0.0	0.0	0.9	0.0	0.0	0.0	107.6
15-16	64.8	0.0	0.0	0.0	0.0	0.0	3.9	0.0	3.0	0.0	0.0	0.0	22.4	0.0	0.0	0.0	94.0
16-17	8.4	0.0	0.0	0.0	0.0	0.0	11.4	0.0	7.1	0.0	0.0	0.3	24.7	0.0	0.0	0.0	51.9
17-18	3.5	0.0	0.0	0.0	0.8	0.0	17.0	0.2	17.6	0.0	0.0	1.8	32.0	0.0	0.0	0.0	72.8
18-19	3.7	0.0	0.0	0.0	5.3	0.0	47.3	0.0	19.5	0.0	0.0	2.4	18.9	0.2	2.3	0.0	99.5
19-20	3.5	0.0	0.0	0.0	10.1	0.0	42.0	0.0	23.3	0.0	0.0	1.1	23.6	2.8	83.7	0.0	190.0
20-21	4.5	0.0	0.0	0.0	23.3	0.0	50.7	0.2	27.7	0.0	0.0	0.0	10.0	5.4	112.7	0.0	234.6
21-22	5.3	0.0	0.0	0.0	33.7	0.0	53.4	0.0	26.1	0.0	0.0	0.0	0.2	19.1	140.7	0.0	278.4
22-23	3.7	0.0	1.5	0.0	58.7	0.0	47.8	0.0	17.7	0.0	0.0	0.0	0.2	27.0	137.0	1.4	294.9
23-24	9.6	0.0	14.5	0.0	61.7	0.0	39.0	0.0	20.0	0.0	0.0	0.0	0.0	25.4	97.8	28.0	296.0

24-25	22.8	0.0	25.5	0.0	97.7	0.0	55.8	0.0	25.2	0.0	2.1	0.0	0.0	8.9	103.8	44.1	385.9
25-26	41.1	0.0	59.3	0.0	128.1	5.4	140.7	0.0	20.4	1.2	13.8	0.0	0.0	0.7	98.0	66.9	575.7
26-27	81.9	0.0	79.0	0.7	149.3	31.8	164.7	0.0	10.5	1.9	56.6	0.0	0.0	0.0	85.6	39.9	701.9
27-28	143.5	0.0	40.7	16.3	185.3	161.7	109.4	0.0	0.8	0.7	27.0	0.0	0.0	0.0	39.0	3.9	672.7
28-29	210.3	1.7	13.7	106.8	206.2	106.2	31.7	0.0	0.0	0.0	3.3	0.0	0.0	0.0	16.5	0.0	751.7
29-30	218.1	13.2	21.8	240.7	64.4	70.2	0.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	4.0	0.0	632.7
30-31	328.8	27.5	16.0	233.2	4.4	13.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	1.8	0.0	624.8
31-32	444.9	4.8	2.0	53.9	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	505.7
32-33	393.0	0.5	0.0	6.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	400.2
33-34	309.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	309.6
34-35	251.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	251.3
35-36	232.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	232.8
36-37	203.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	203.5
37-38	156.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	156.5
38-39	138.4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	138.4
39-40	111.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	111.1
40-41	87.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	87.5
41-42	67.9	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	67.9
42-43	30.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	30.6
43-44	2.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2.1
Hours	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	108.5	8760.0

	T_{out} [°C]	Wind speed [m/s]	Wind direction	Window opening percentage [%]	Damper opening percentage [%]	Ceiling fan operation	Assumed air velocity [m/s]	Internal doors position
Buoyancy-driven forces								
Buoyancy Case-1	25.8	-	-	0	80	ON	0.6	open
Buoyancy Case -2	21.3	-	-	0	100	OFF	0	open
Buoyancy Case -3	27.5	-	-	60	80	ON	0.9	open
Buoyancy Case -4	29.6	-	-	100	100	ON	0.6	open
Buoyancy Case -5	30.6	-	-	100	100	ON	1.2	open
Buoyancy Case -6	28.3	-	-	60	100	ON	0.6	open
Buoyancy Case -7	28.3	-	-	60	100	ON	0.6	closed
Wind-driven forces								
Wind Case - 1	28.3	3.5	North	60	100	ON	0.6	open
Wind Case -2	28.3	3.5	South	60	100	ON	0.6	open
Wind Case-3	28.3	3.5	East	60	100	ON	0.6	open
Wind Case -4	28.3	3.5	West	60	100	ON	0.6	open
Wind Case -5	28.3	3.5	North	60	100	ON	0.6	closed
Wind Case -6	28.3	3.5	South	60	100	ON	0.6	closed
Wind Case -7	28.3	3.5	East	60	100	ON	0.6	closed
Wind Case -8	28.3	3.5	West	60	100	ON	0.6	closed

Table 7-6: Simulated cases for CFD simulations for buoyancy and wind-driven forces.

For each examined case, the thermal boundary conditions for the walls were provided by the co-simulations. In the two bedrooms, a heat source boundary condition was applied in the bed to represent the heat gains by the occupants.

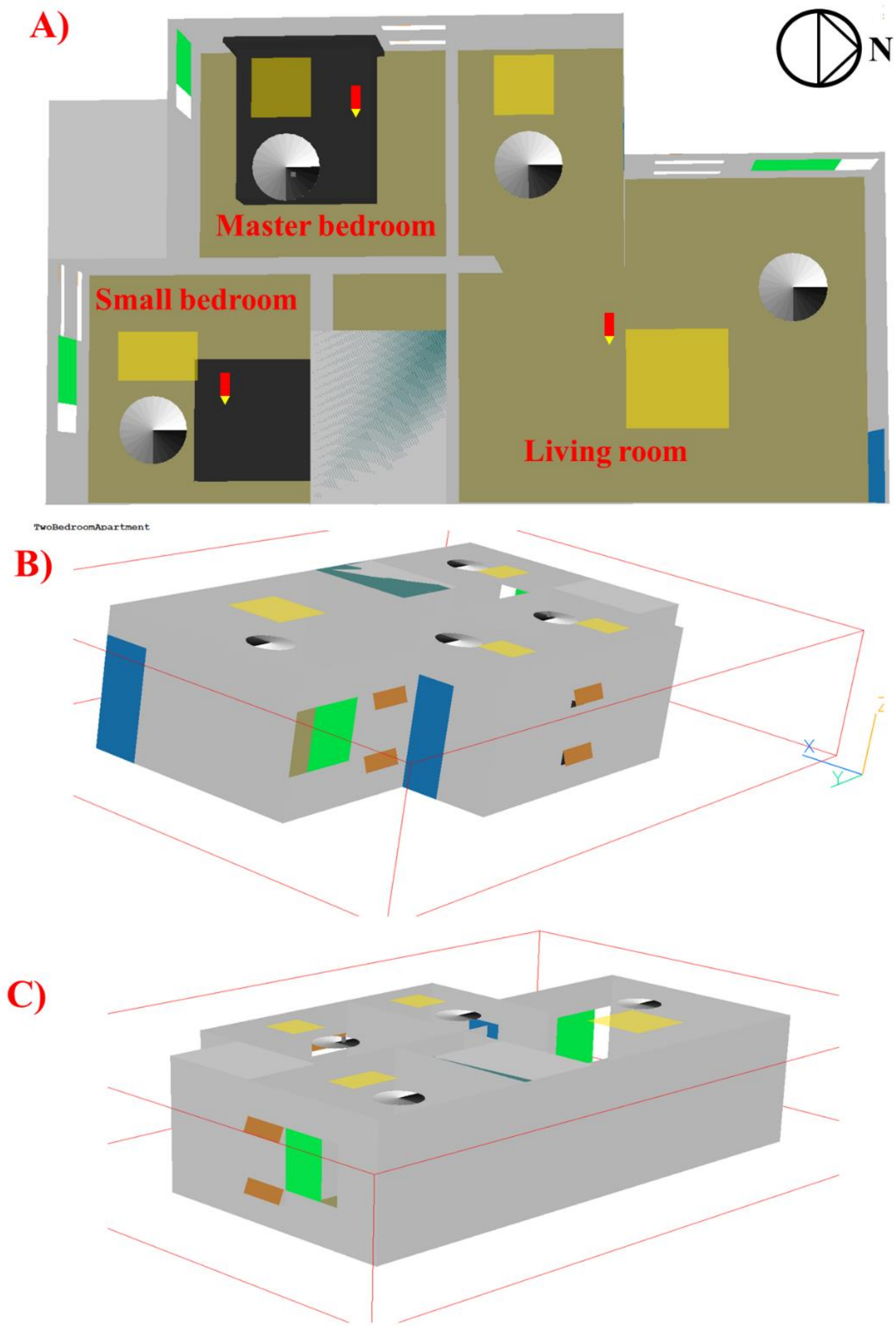


Figure 7-5: CFD model for the two-bedroom apartment.A) The red/yellow arrows show the positions where the values were calculated; plot B & C shows the position of the dampers.

Boundary conditions for the walls, floors and ceilings for each room, were imported from the co-simulations for that specific timestep. Occupants were represented as blockages one per living space. The boundary conditions for the heat generated by the occupants were imposed on the beds. Table 7-7 summarizes the boundary conditions for all the examined cases.

Table 7-7: Heat gains of each of the rooms for the CFD simulations, all in W/m^2.

Space	W/m^2	Buoyancy Case-1	Buoyancy Case-2	Buoyancy Case-3	Buoyancy Case-4	Buoyancy Case-5	Buoyancy Case-6	Buoyancy Case-7	Wind Case-1	Wind Case-2	Wind Case-3	Wind Case-4	Wind Case-5	Wind Case-6	Wind Case-7	Wind Case-8
Master Bedroom	**Walls**	5.2	5.8	4.9	3.9	6.1	5.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5
	Lighting	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3
	Other heat gains	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8
Small Bedroom	**Walls**	5.3	5.9	5.0	4.0	6.2	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
	Lighting	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3
	Other heat gains	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8
Kitchen/ Living Room	**Walls**	5.5	6.2	5.3	4.1	6.5	5.9	5.9	5.9	5.9	5.9	5.9	5.9	5.9	5.9	5.9
	Lighting	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3
	Other heat gains	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9

7.4. Analysis of ventilation performance and flow fields for a typical two-bedroom Indian apartment

Comparison of airflow rates, air temperature and velocity distributions and the ventilation effectiveness were performed for a variety of outdoor conditions, windows/dampers openings and ceiling fan operation, presented in table 7-6 using steady-state buoyancy and wind-driven simulations. Since there are no standards values for Indian houses, the proposed limits by CIBSE (2006) were used for the proposed ventilation rates, see table 7-8. To calculate the ventilation effectiveness indices (see Eq. 5-2 & 5-3), a customised INFORM code was developed within the CFD software to calculate the ventilation effectiveness. A part of the code is presented in Appendix:Inform Code for PHOENICS.

Table 7-8: Environmental conditions used for the ventilation rates

Air change rates (ach^{-1})	Above 1 ach^{-1} for adequate ventilation (CIBSE A, 2005)
	Between 5-10 ach^{-1} (CIBSE Guide B, 2005) and up to 30 ach^{-1} for cooling.

7.4.1. Buoyancy-case 1-2: Operation of dampers and ceiling fans - Buoyancy-driven flow.

Due to the small effective opening area of the dampers, when only dampers operated (Buoyancy cases 1-2) insufficient ventilation rates (1.8~2.2 ach^{-1}) and very low internal air velocities of 0.1~0.22 m/s were predicted in the living spaces.

The heat plumes and associated air movement emanating from the heat sources (walls, ceiling lights and beds) are clearly represented in figure 7-6. The velocity vector plot shows that the air is entering into the room through the low-level dampers, travelling along through the corridor towards the rest of the rooms, where is rising up to the ceiling and exiting the room through the high-level dampers. The distortion caused to the airflow entering the room by the dampers' configuration is noticeable, as the air does not enter the room at a horizontal plane, instead, it is forced to fan out higher into the room in comparison to the sliding windows.

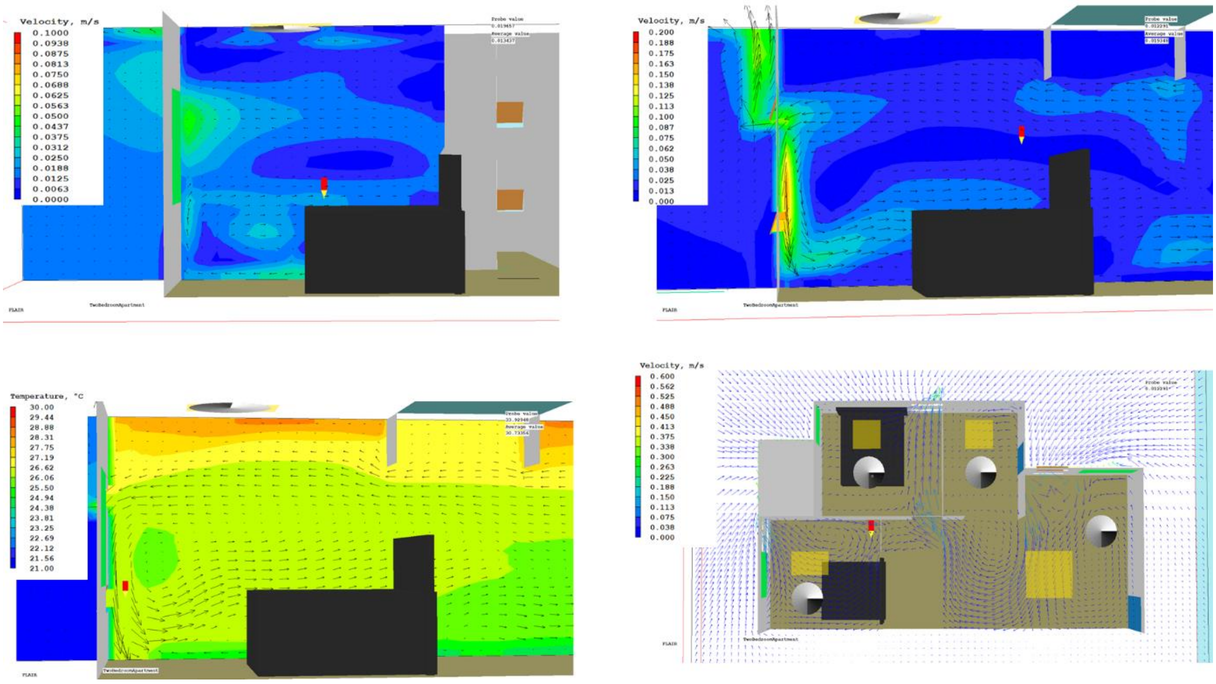


Figure 7-6: Predicted internal airspeeds and temperature distribution for case 4. Top right and left graphs highlight the air velocity fields generated by the ceiling fan in the small bedroom at plane X=6.0m and X=4.5m respectively; bottom left graph shows the temperature distribution for the small bedroom at plane X=4.5m while the bottom right graph shows the airflow fields for the whole apartment.

As conveyed in figure 7-6, the exiting air is forced to flow down under the damper and then back up behind it to leave the external area. Due to that the similar surface temperatures (walls) to the air temperature, the buoyancy forces were small and hence the air velocity of case 4 was very low, as reflected in figure 7-6.

For case 3 (figure 7-7) where the ceiling fan was operating, the airflow patterns and the velocity vector plot (top left graph) were similar to previous studies (Jain *et al.*, 2014; Babich, 2017). For instance, for the two bedrooms, at a low level, very close to the floor and underneath the ceiling, the airspeed was higher compared to the rest of the locations inside the bedrooms. The use of the ceiling fan and most specifically the axial velocity of the ceiling fan determined the internal airspeed in each room. This was predicted to be from 0.3 to 1.3 m/s depending on the room and the position of the probe. The placement of the ceiling fan is essential, as it affects greatly the airflow generated in the living spaces, and it might lead to discomfort in response to high internal air velocities. In case 3, the ceiling fan speed was relatively low and was predicted to be approximately 0.7 m/s, 0.75 m/s, 0.68 m/s for the master bedroom, small bedroom and living room respectively, which are within the acceptable limits for occupants in domestic buildings in India (Indraganti, 2010; Dhaka *et al.*, 2015). The temperature contours plots show a noticeably stratified air temperature distribution for the case where only thedampers operated, and the ceiling fan was turned off (case 4). On average, the temperature difference between the floor and ceiling was around 2°C.

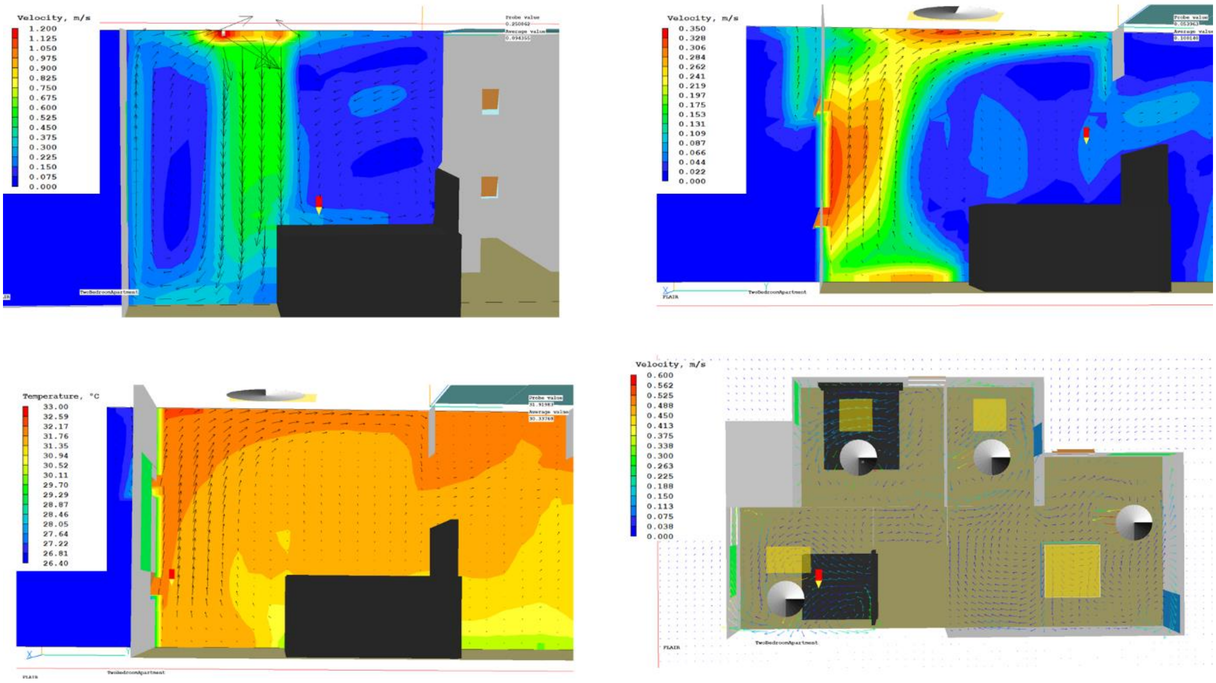


Figure 7-7: Predicted internal airspeeds and temperature distribution for case 3. Top right and left graphs highlight the air velocity fields generated by the ceiling fan in the small bedroom at plane X=6.0m and X=4.5m respectively; bottom left graph shows the temperature distribution for the small bedroom at plane X=4.5m while the bottom right graph shows the airflow fields for the whole apartment.

The relatively small ventilation rates, that were predicted in the cases where only the dampers operated, could not balance the heat generated by the internal sources and this resulted in higher internal air temperatures. The CFD simulations predicted relatively high internal air temperatures for the two bedrooms, mainly due to higher internal heat gains and the small effective area of the dampers. The use of the ceiling fan (case 3) resulted in less distinctive stratified temperature distribution. The swirling effect by the use of the ceiling fan impacted the airflow patterns inside the rooms and the CFD simulations predicted that the internal air was better mixed compared to case 4. The results showed a more uniform distribution of the air temperature in the room, where colder air is well-mixed with the warmer air and hence the average temperature of the room predicted to be slightly higher compared to case 3. The elevated internal airspeed, by the use of the ceiling fan, has positive impacts on the thermal sensation and allows the use of higher cooling setpoints without compromising the thermal comfort sensation for the occupants as previous studies have shown (ASHRAE-Standard-55, 2013; Manu *et al.*, 2016).The ventilation effectiveness of the systems was predicted. As explained in Chapter 5, both the HRE and CRE were calculated for each room separately.

The analysis showed that the distribution of the CO_2 concentration affected by the operation of the fan as well as the configuration of the dampers. The use of the dampers was unable to remove the heat or concentration of the contaminant for both cases. The analysis showed that the HRE for case 3 & 4 was 0.58 & 0.67 respectively, whilst the CRE was calculated to be 0.38 & 0.47 for case 3 & 4 for the master bedroom only. The results for the rest of the rooms can be found in see table 7-9 for both CRE and HRE. Although both cases utilized the used of cross ventilation, the relatively small effective opening area of the dampers was found to be inadequate to remove the heat or the contaminants from the space. This can be explained from the relatively low ventilation rates that were predicted for both cases.

7.4.2. Buoyancy Cases 3 to 7: Combined operation of windows, dampers and ceiling fans - Buoyancy-driven flow.

The combined operation of windows and dampers (buoyancy cases 3-7) resulted in higher ventilation rates compared to buoyancy cases 1 & 2. Specifically, the ventilation rates predicted to be from 4.7-9.6 ach^{-1} and internal air velocities from 0.4-0.8 m/s. Adequate ventilation rates relative to the values presented in table 7-8.

The heat plumes and associated air movement emanating from the heat sources (walls, ceiling lights and beds) are represented in figure 7-8. The velocity vector plot shows that the air is entering into the room through the low-level dampers and the low side of the windows, travelling along the rooms, towards the corridor and the rest of the rooms, rising up to the ceiling then exciting the room through the high-level dampers and the high side of the windows. The distortion caused to the airflow entering the room by the dampers' and windows' configuration is noticeable. For the dampers, as discussed in subsection 7.4.1, the air does not enter the room at a horizontal plane, instead, it is forced to fan out higher into the room in comparison to the sliding windows.

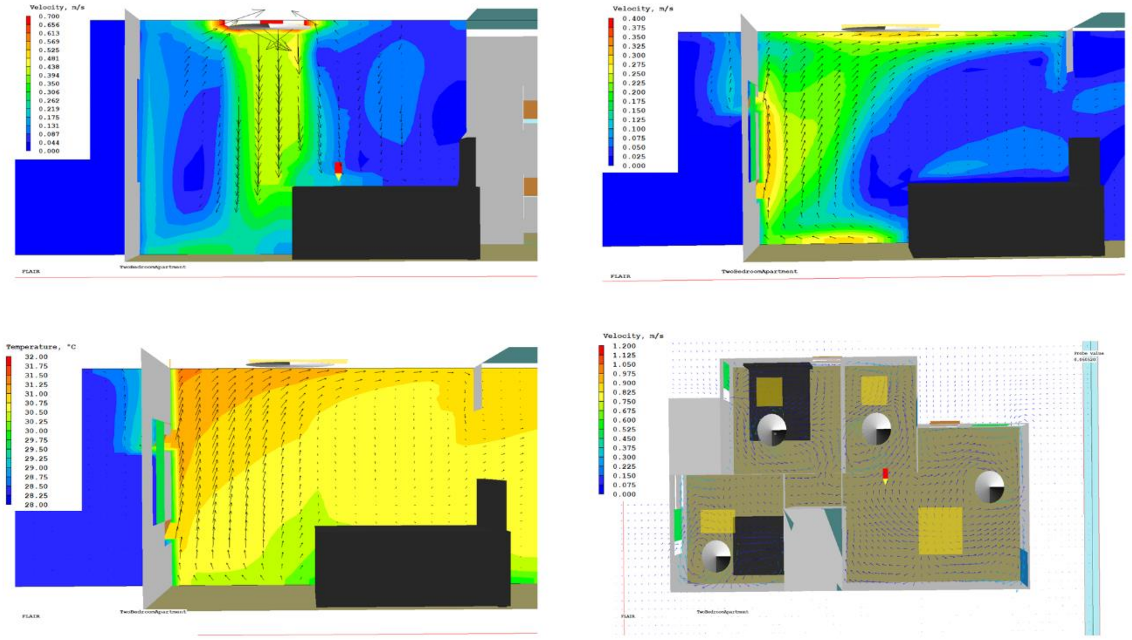


Figure 7-8: Predicted internal airspeeds and temperature distribution for case 6. Top right and left graphs highlight the air velocity fields generated by the ceiling fan in the small bedroom at plane X=6.0m and X=4.5m respectively; bottom left graph shows the temperature distribution for the small bedroom at plane X=4.5m while the bottom right graph shows the airflow fields for the whole apartment.

The horizontal sliding-windows resulted in cold (outside) air to enter through the low side of the window; while the hotter air to exit through the high side of the window. For the buoyancy case 6, where the internal doors kept open, the air circulated inside the apartment resulting in slightly lower internal air temperatures compared to buoyancy case 7. For all buoyancy-driven cases, the driven factor of the airflow is the density and temperature difference between the inside and outside air. Because of the imposed boundary conditions on the surfaces (walls and window), the predicted internal air velocity was relatively low, especially for the cases where the ceiling fans did not operate (figure 7-8). For buoyancy case 6 where the ceiling fan was operating, the airflow patterns and the velocity vector plot (figure 7-9) were similar to previous studies (Jain *et al.*, 2014; Babich, 2017). Similarly, to buoyancy case 6, the use of the ceiling fan and most specifically the axial velocity of the ceiling fan determined the internal airspeed in each room. The internal airspeed was predicted to vary between 0.3 and 1.2 m/s for the different cases. For Case 6 with the same ceiling fan speeds as case 3 but with open windows, about 25% higher internal airspeeds at the same locations were predicted, which were within the range of values predicted by others in domestic buildings in India (Gong *et al.*, 2006; Indraganti, 2010). This was due to the total effective opening area increased by almost 180% between case 3 and 6 due to the operation of the window, which resulted also in higher ventilation rates by approximately 65%. The higher predicted volume of cooler air that entered.the room had a positive impact on the internal air temperature and the temperature stratification as expected.

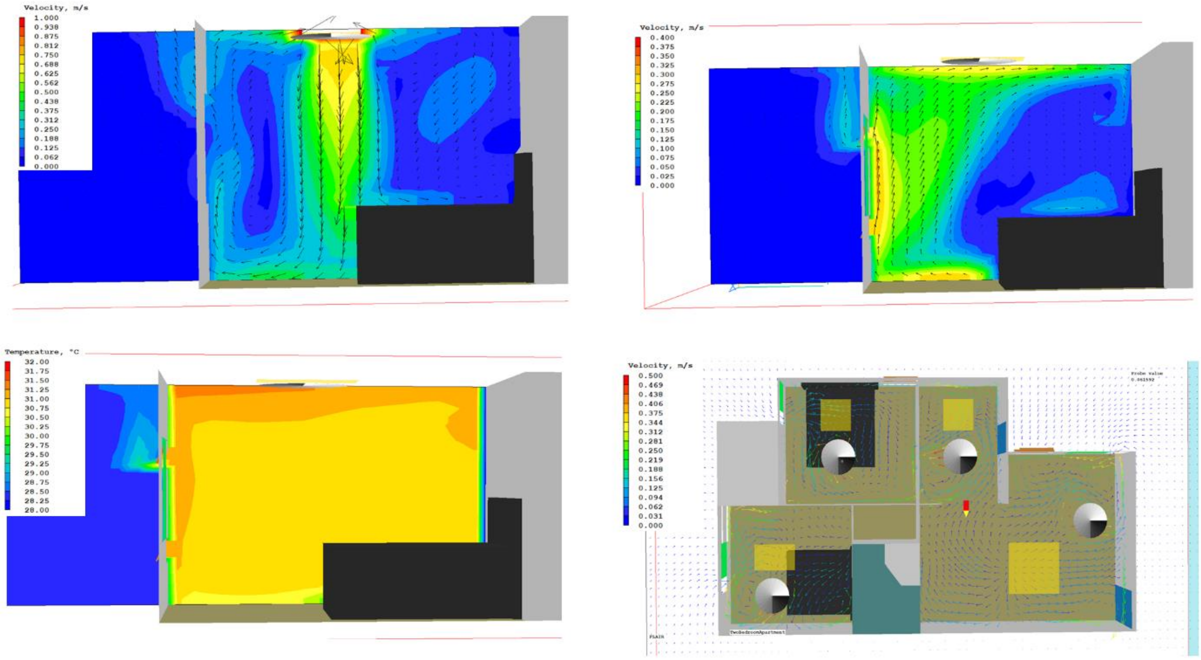


Figure 7-9: Predicted internal airspeeds and temperature distribution for case 3. Top right and left graphs highlight the air velocity fields generated by the ceiling fan in the small bedroom at plane X=6.0m and X=4.5m respectively; bottom left graph shows the temperature distribution for the small bedroom at plane X=4.5m while the bottom right graph shows the airflow fields for the whole apartment.

When the internal doors were kept closed (buoyancy case 7), comparable airflow distribution across all spaces was predicted to case 6 that has the same boundary conditions (figure 7-9). The closure of the internal doors restricted the air to flow freely throughout the apartment resulting in single-sided instead of cross ventilation, which impacted both the temperature stratification and ventilation rates (figure 7-11). The internal airspeed was predicted to be almost 11% higher for buoyancy case 7 compared to case 6 with the highest values to be 0.41m/s for the master bedroom. This can be explained because the single-sided ventilation resulted in more turbulence flow inside the rooms due to the ceiling fan and hence higher internal airspeeds were predicted. As mentioned earlier, closing the internal doors resulted in single-sided ventilation and hence less outside air entered the rooms. With the single-sided ventilation, the opening of each room were both the inlets and outlets which had an impact on the predictions of the ventilation rates as the analysis showed. The temperature contour plots show a stratified air temperature distribution. On average, the temperature difference between the floor and ceiling is around 2.2-2.4°C for buoyancy case 6 while for buoyancy case 7 was found to be 2.5-2.7°C. The relatively higher ventilation rates, that were predicted in case 6, resulted in better dilute of the heat generated by the internal sources and this had a positive impact on the predicted internal air temperatures. The CFD simulations predicted relatively higher internal air temperatures for the two bedrooms compared to the living room, mainly due to higher internal heat gains and the smaller total effective area of the openings. In line with the rest of the cases, the use of the ceiling fan resulted in less distinctive stratified temperature distribution. The swirling effect using the ceiling fan impacted the airflow patterns inside the rooms and the CFD simulations. The results showed a more uniform distribution of the air temperature in the room, where colder air is well-mixed with the warmer air. The operation of the windows resulted in lower internal air temperatures, as the more outside air entered the rooms compared to buoyancy case 1 & 2 with only the dampers.

Overall, the use of ceiling fan contributes to a better and more uniform distribution of fresh air in the spaces relative to the cases without the use of ceiling fan. Regardless the operation of the windows, the use of fan dominated as expected the development of the airflow patterns inside the room and can be used to improve the distribution of the air inside the rooms and mitigate the development of the stratified air.

The HRE and CRE were calculated for each room separately in order to evaluate ventilation effectiveness using the customised code developed for this research (Inform Code for PHOENICS). Higher ventilation rates attributed to the open windows delivered heat and/or CO_2 concentration reduction, within the acceptable levels for thermal comfort. This was evident in the HRE and CRE calculations for all the rooms. The CFD simulations predicted higher ventilation effectiveness indices for all the cases where windows and dampers where used. When the ceiling fans were operating, the analysis revealed slightly higher CRE and HRE values compared to the cases where the ceiling fans were turned off. The CRE and HRE values varied from 0.83-1.21 and 0.78-1.44 respectively depending on the cases and rooms. The highest values of CRE and HRE were predicted to be in the living room, where the volume of the room was bigger compared to the master and small bedroom. The calculations for the CRE and HRE were performed in the occupied zone (1.4m above the floor level), and hence the bigger volume of the living room in combination with higher ventilation rates (see table 7-9) resulted in a better dissolution of the heat and the concentration of the contaminants and hence higher predicted CRE and HRE values. A very interesting observation was made for buoyancy case 7 where the internal doors were kept closed. The closure of the internal doors restricted the air to flow between each room. Hence, instead of having cross-ventilation, the CFD analysis predicted single-sided natural ventilation for each room. The predicted ventilation rates for each room were considerably smaller compared to buoyancy case 6, approximately 10-12%, with the same boundary conditions. The CFD simulations predicted smaller HRE and CRE indices as expected due to the smaller ventilation rates. Hence, when designing natural ventilation solutions, it is essential to include the operation of the internal doors as part of the analysis because it can have an impact on ventilation performance.

Similar to the buoyancy case 1 & 2, the analysis showed that the distribution of the CO_2 concentration affected by the operation of the fan. The use of the ceiling fan resulted in a higher predicted value of the CRE and HRE indices (see table 7-9) due to a better distribution of the internal air temperature and CO_2 concentration as explained above.

7.4.3. Wind Cases 1 to 8: Window, dampers and ceiling fan under wind and buoyancy-driven flows.

This case-study apartment was modelled on its own, excluding the surrounding buildings and apartments Figure 7-10). This is a limitation of the model as it does not account for the effect of the surrounding building on the ventilation performance and on the pressure distribution at the openings.

When single-sided ventilation was in place (all internal doors were closed), wind cases 5 to 8, similar trends were predicted with the buoyancy-driven cases. More specifically, although the predicted ventilation rates were higher compared to the buoyancy-driven cases for the same boundary conditions by approximately 7 to 22%, they were still relatively lower compared to the cases with the internal doors open. As expected, the wind direction dominated the ventilation rates for each room. The rooms with windward facing openings had the highest ventilation rates. In particular, when cross-ventilation was achieved through the open internal doors, CFD simulations predicted higher ventilation rates by approximately 7 to 27% relative to single-side ventilation cases (wind cases 5-8). Ventilation rates for both single-sided and cross ventilation scenarios were in line with values reported by others in similar cases for Indian residences (de Faria *et al.*, 2019).

Under wind-driven forces, the wind cases 1-4 with the cross-ventilation predicted to achieve temperature reductions up to 1.2°C, while the internal airspeed increased from 5-30% compared to the single-sided cases.

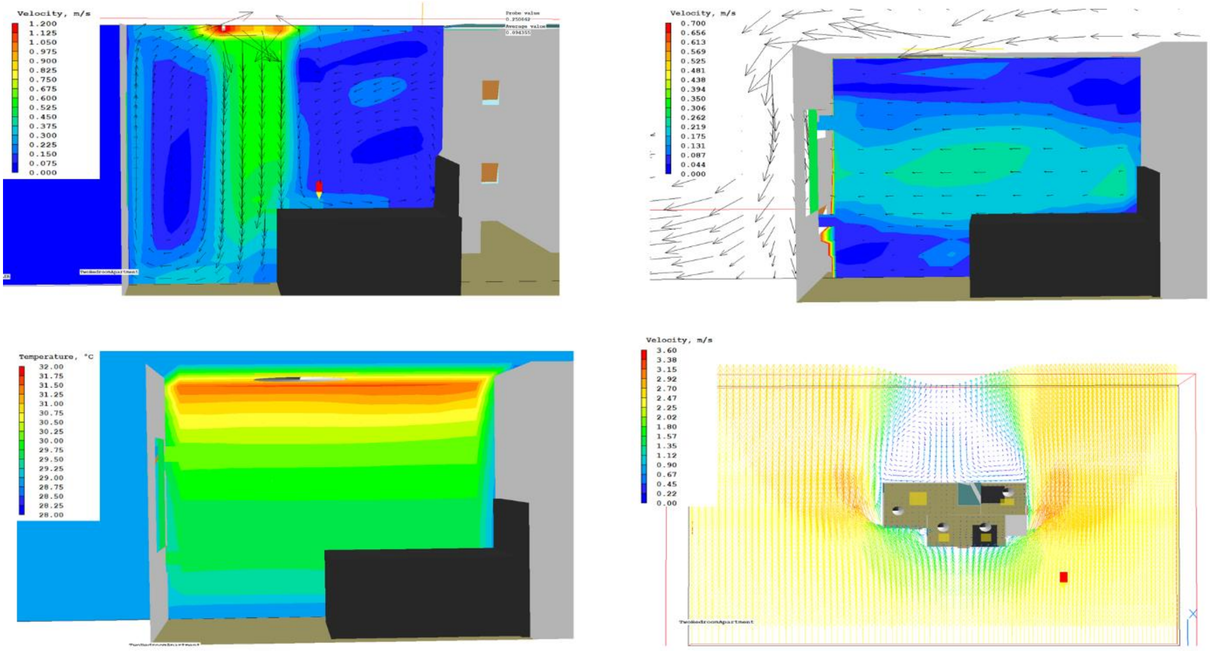


Figure 7-10: Predicted internal airspeeds and temperature distribution for case 8. Top right and left graphs highlight the air velocity fields generated by the ceiling fan in the small bedroom at plane X=16.75m and X=15.5m respectively; bottom left graph shows the temperature distribution for the small bedroom at plane X=15.5m while the bottom right graph shows the airflow fields for the whole domain for west wind direction.

Due to the use of the ceiling fans in the examined scenarios, the predicted values of the internal airspeed was mainly affected by the ceiling fan and less by the external wind speed. As expected, the rooms where the wind direction was normal to the openings had the highest internal airspeed. The wind-driven flows still resulted in the stratification of the air temperature, as shown in the buoyancy cases. The inclusion of the wind effect did not alter significantly the temperature distribution compared to the buoyancy cases, at least qualitatively.

In line with the buoyancy-driven cases, the CFD simulations predicted relatively higher internal air temperatures for the two bedrooms compared to the living room, mainly due to higher internal heat gains and the smaller total effective area of the openings for the wind-driven cases. In line with previous cases, the use of the ceiling fans resulted in less distinctive stratified temperature distribution. The swirling effect by the use of the ceiling fan impacted the airflow patterns inside the rooms and the CFD simulations predicted that the internal air was well mixed resulted in less stratified temperature distribution. The temperature difference between the floor and ceiling was from 1.6-2.4°C. Although the wind-driven forces had an impact on the flow patterns inside the rooms, the use of the ceiling fan dominated the development of the flow patterns.

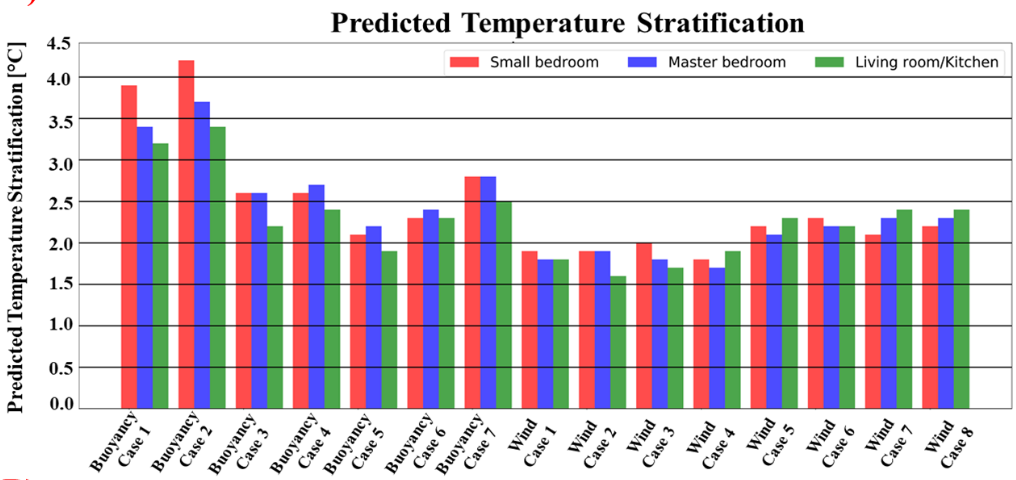
Finally, a very important parameter for the CFD simulations was the ventilation effectiveness of the systems. The wind-driven forces resulted in higher ventilation rates in all the cases and this had a positive impact on dissolving the heat and/or CO_2 concentration. This observation can be reflected in the HRE and CRE calculations for all the rooms. The CFD simulations predicted higher ventilation effectiveness indices for all the wind-driven cases compared to the buoyancy-driven cases. The CRE and HRE values varied from 0.97-1.56 and 0.97-1.61 respectively, depending on the wind direction and rooms. The highest values of CRE and HRE were predicted to be in the living room, where the volume of the room was bigger compared to the master and small bedroom for the South wind direction since higher ventilation rates were predicted for the same wind direction. The calculations for the CRE and HRE were performed in the occupied zone (1.4m above the floor level), and hence the bigger volume of the living room in combination with higher ventilation (see table 7-9) resulted in a better dissolution of the heat and the concentration of the contaminants and hence higher predicted CRE and HRE values. As observed in the buoyancy-driven case, the closure of the internal doors restricted the air to flow between each room resulting in single-sided ventilation which reflected in smaller ventilation rates as well as smaller HRE and CRE indices compared to the cross-ventilation cases.

Similar to the buoyancy-driven cases, the analysis showed that the distribution of the CO_2 concentration affected by the operation of the fan. The use of the ceiling fan resulted in a higher predicted value of the CRE and HRE indices (see table 7-9) due to a more uniform better distribution of the internal air temperature and CO_2 concentration as explained above.

7.5. Comparison of results for all the examined cases

As indicated in figure 7-11, temperature stratification in each room for each case was predicted. The buoyancy cases 1-2 with only the dampers open resulted in higher stratification of the temperature which affects the thermal comfort conditions of the occupants. The low ventilation rates were inadequate to remove the internal heat hence it is essential to incorporate other sources of cooling (mechanical cooling) in cases where the opening areas of the windows/dampers are small. For the rest of the buoyancy cases (buoyancy case 3-7), the CFD simulations predicted temperature stratification up to 2.7°C, which is considered acceptable. The lowest values were predicted for buoyancy case 5 where the windows and dampers were fully open (100% opening) and the predicted ventilation rates were high. In general, for al the wind cases (wind case 1 to 8), the CFD simulations predicted temperature stratification from 1.6 to 2.4°C, which is in line with the high ventilation rates that were predicted for these cases. A general observation was that for the cases where the internal doors were kept closed (buoyancy case 7, wind case 5 to 8) the predicted temperature stratification was higher due to the lower ventilation rates. Hence, it is an important parameter to consider when design natural and mixed-mode ventilation strategies for buildings.

A)



B)

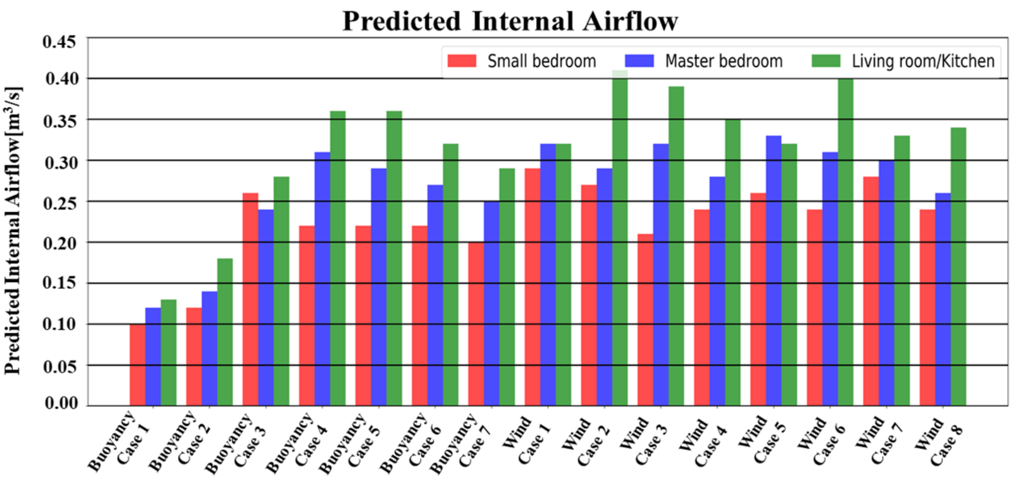


Figure 7-11: Average predicted stratification air temperatures for the three rooms(plot A); Average predicted airflow rates for the three rooms (plot B).

The higher ventilation flow rates had a positive impact on the calculations of the CRE and HRE indices. In general, the cases with the higher ventilation rates found to have higher values of CRE and HRE as well. Although a direct comparison of the results is inappropriate due to the different boundary conditions for each case, a general observation is that the cases with cross ventilation resulted in higher CRE and HRE for both buoyancy and wind-driven flows and the single-sided ventilation cases (buoyancy case 7, wind case 5 to 8) resulted in lower predicted HRE and CRE. Hence it is important when designing a control algorithm for natural and mixed-mode ventilation systems to analyse also the HRE and CRE in order to conclude whether or not the proposed strategy is efficient (table 7-9).

	HRE [-]			**CRE[-]**		
	Living room	**Master Bedroom**	**Small Bedroom**	**Living room**	**Master Bedroom**	**Small Bedroom**
Buoyancy Case-1	0.62	0.58	0.56	0.39	0.38	0.37
Buoyancy Case-2	0.68	0.67	0.61	0.48	0.47	0.46
Buoyancy Case-3	0.82	0.76	0.78	0.89	0.92	0.86
Buoyancy Case-4	1.29	1.27	1.2	1.2	1.13	1.12
Buoyancy Case-5	1.5	1.48	1.36	1.29	1.2	1.22
Buoyancy Case -6	1.06	0.99	0.95	1.04	0.93	0.87
Buoyancy Case-7	0.87	0.85	0.82	0.88	0.82	0.8
Wind Case - 1	1.53	1.47	1.45	1.48	1.47	1.44
Wind Case -2	1.69	1.58	1.51	1.66	1.59	1.57
Wind Case-3	1.53	1.45	1.42	1.51	1.34	1.29
Wind Case -4	1.02	1	0.94	1.1	0.98	0.96
Wind Case -5	1.47	1.46	1.37	1.41	1.33	1.34
Wind Case -6	1.21	1.19	1.05	1.17	1.06	1.04
Wind Case -7	1.23	1.2	1.13	1.18	1.11	1.13
Wind Case -8	1.21	1.12	1.09	1.03	0.98	0.96

Table 7-9: Summary table of the ventilation effectiveness indices for all the rooms and all the cases.

7.6. Summary

The results from the steady-state CFD simulations show that it is possible to implement mixed-mode ventilation strategies in domestic buildings in India and provide occupants' thermal comfort when outdoor temperature is between 28-29°C and ceiling fans are used. To assess the thermal comfort within the apartment, the temperature stratification and ventilation rates were used, while the ventilation effectiveness was used to analyse the performance of the natural and mixed-mode systems. The potential of the examined cases to maintain the thermal conditions of the building within the comfort limits has been demonstrated through steady-state CFD simulations, of a typical 2-bedroom apartment in India. The performance of the natural and mixed-mode ventilation cases was evaluated with respect to the ventilation rates, internal air temperatures and airspeeds and ventilation effectiveness of the systems.

The inclusion of the ceiling fans, as part of the control approach, enhanced significantly the thermal conditions inside the rooms. The use of the ceiling fans resulted in less stratified temperatures and higher internal airspeeds ranging from 0.12-0.52 m/s depending on the case, for the buoyancy-driven forces. Under the wind-driven forces, the CFD simulations predicted internal airspeeds higher than the buoyancy-driven cases by 7-30% depending on the cases. The use of the ceiling fans dominated the development of the airflow patterns inside the rooms; hence the impact of the external wind speed was not as significant as expected. As expected, the same observations were made for buoyancy-driven cases. It is essential hence to incorporate the ceiling fans into the early-stage design of apartments as they can have a significant impact on the internal airspeeds and might result in uncomfortable internal airspeeds.

The inclusion of the windows, dampers and ceiling fans were found to be the most effective way to reduce the temperature stratification in the apartment. The analysis showed that in the buoyancy-cases 3-7 the stratification from floor to ceiling was from 1.9 to 2.8°C depending on the room and the case. For the buoyancy-cases 1-2, it was predicted to be from 3.1 to 4.2°C mainly due to the small opening areas of the dampers that resulted in very low ventilation rates.

In extreme climatic conditions found in countries such as in India, it is important for the natural and mixed-mode strategies, not only to provide outside air but also to reduce the internal air temperature whenever possible to reduce the dependence on mechanical systems.

The CFD analysis demonstrated the potential of natural and mixed-mode strategies to provide thermally comfortable internal conditions.

It was further explored, how the operation of the internal doors can impact the development of the flow inside the apartment. The closure of the internal doors resulted in single-sided natural ventilation instead of cross ventilation. The CFD simulations explored two cases with the internal doors closed and found out that the internal air temperature was increased by approximately 7% for the same room and the same boundary conditions. The airflow restrictions from space to spaces imposed by the closed doors can significantly influence the performance of the ventilation strategy. Hence, it should be considered when sizing windows and dampers since bigger opening areas are required to provide adequate outside air. However, sizing of the openings was outside the scope of this research, and that is the reason for using a very typical layout for a 2-bedroom apartment in India.

Finally, the ventilation effectiveness of the different cases was quantified using the HRE and CRE indices. As expected, the cross-ventilation cases with the bigger opening areas resulted in higher ventilation rates and also in higher ventilation effectiveness indices, whilst the single-side scenarios resulted in the smallest ventilation effectiveness indices. This research provides a unique insight into the ventilation performance, as it analysed the ventilation effectiveness indices (HRE and CRE) under a variety of cases. The cross-ventilation case with 100% windows and dampers opened, predicted to have the highest values for CRE and HRE indices. The very high ventilation rates resulted in the most effective way to dilute the contaminants' concentration and the heat generated, and the CFD simulations predicted CRE=1.25 and HRE=1.42. Similarly, for the wind-driven cases, the CFD simulations predicted that the cross-ventilation with the ceiling fan on resulted in the higher values for CRE and HRE compared to the rest of the scenarios, with CRE=1.62 and HRE=1.65. The lower values were predicted for the single-side buoyancy-driven case due to the very small predicted ventilation rates. The CRE and HRE indices ranged from 0.80-0.88 and 0.81-0.87 respectively for the single-sided ventilation systems

Although the CFD simulations predicted the ventilation performance of the different cases under steady-state conditions, they allowed through predicted temperature and airflow distributions across the living spaces the better understanding of the performance of the ventilation strategies. The use of cross-ventilation in combination with windows, dampers and ceiling fan operation can result in comfortable internal conditions as the analysis indicated. However, due to the nature of the steady-state simulations, few assumptions were made. Many characteristics such as the occupants' behaviour, surrounding buildings etc were neglected due to the lack of information and hence these are the limitations for the CFD results. Also, the number of the examined scenarios was sufficient to provide an understanding of the ventilation performance of the mixed-mode ventilation and cooling strategies, but still, it was relatively small to draw conclusions for all the examined locations and weather conditions

Chapter 8. Conclusions and Future work

8.1. Overview of the research carried out

The research presented in this thesis has developed a better and deeper understanding of the control algorithms for mixed-mode ventilation and cooling systems for Indian residencies. This research focused on Indian residencies and Indian climatic conditions and investigated the energy savings that can be achieved using a variety of control algorithms whilst maintaining thermal comfort conditions.

To accommodate the rapid population growth in India, it is expected that the floor area of the domestic sector will increase by almost 500% in the next 10 years (United Nations, 2016). Following this, the increase of the disposable income in developing countries, such as in India, in combination with the higher expectations for thermal comfort will increase the demand for cooling and reliance on mechanical systems, such as ACs, for cooling and comfort. In view of climate change, retrofitting existing building stock and ensuring low-energy operation of new builts is of significant importance. Accordingly, control algorithms were developed and tested in control environments for mixed-mode ventilation and cooling systems currently used and/or new that can ensure occupant comfort and energy consumption reductions for existing and new domestic buildings. As a result of this research, designers will be able to accurately identify the most suitable control algorithm and predict energy-saving potential by implementing the selected control algorithm. This research provides guidance on how to control a mixed-mode building and evidence of their performance. The results can be used by designers of buildings or policymakers to write or revise national standards.

In this research, data from the experimental chamber in CEPT, India were used to validate the proposed control algorithms. Computer simulations were used to model the proposed control algorithms with two main purposes: the results were analysed to examine the energy-saving potential by comparing different scenarios whilst maintaining thermal comfort internal conditions, and by comparing the predictions for the ventilation rates, ventilation effectiveness indices and air temperature & airspeeds distributions. The former was addressed by using co-simulations, where the developed control algorithms were translated from a flow chart format into the Modelica language while the building geometry was developed in

EnergyPlus. The co-simulations were performed using the FMI standardized method. The latter was achieved by using CFD simulations and involved the accurate definition of the boundary conditions. This chapter is divided into three main sections. The first presents the key findings and conclusions of this research, highlighting the contribution to knowledge. The second section describes the wider significance of this work with regards to the impact on the academia, industry and policy. Finally, the third part summarizes the limitations of this research and suggests ideas and directions that could be followed by future work to address these limitations.

8.2. Principal findings of this research and contribution to knowledge

This research contributed to the existing body of knowledge by providing:

1. Comprehensive guidelines about the design and control of mixed-mode buildings, and how the co-simulations can be implemented to improve the existing control algorithms that can be found in the literature;
2. An accurate and validated methodology for the co-simulations, that can be used for any research study that requires the control of mixed-mode ventilation and cooling systems. The use of co-simulations provides great flexibility as they can be used regardless of the geometry and/or the location of the buildings as long as the geometry is exported as an FMU. Most importantly, the validation method that was used for this research provides a new framework to validate control algorithms for mixed-mode buildings and can be used by future research to validate new control algorithms; and
3. An accurate and validated method to estimate the ventilation effectiveness of the mixed-mode systems. This methodology can be applied in any CFD model, regardless of the geometry, to assess different ventilation systems.

Findings of this research suggesting that the proposed control algorithms and ventilation and cooling scenarios examined in this research achieved energy savings compared to the base case scenarios whilst delivering thermal comfort conditions. The evaluation of thermal comfort conditions was performed according to the Indian Model for Adaptive Comfort (IMAC) while previous work was used to compare the results for energy predictions. This research has shown that the co-simulations are able to improve the performance of the existing models for modelling mixed-mode ventilation and cooling systems.

Firstly, the development of the control algorithms into the Modelica language and of the co-simulations models was essential for this research. The validation analysis showed that the co-simulations and the developed control algorithms can accurately control the mixed-mode systems. For the co-simulations, 100% of the predicted values are within the error bars of the respective measured data, whilst for the CFD simulations, the agreement was 81%. The qualitative comparison of the developed airflow due to the swirling effect was identical to previous studies (Babich, 2017). Furthermore, the current research proposed an innovative approach to control the openings based on the required airflow rate to delivery cooling based on the internal heat gains rather than on the temperature difference between inside and outside. This method was validated and can be used by other researchers in the future to control the opening of windows/dampers.

The next step of this research was to perform the co-simulations using the geometry for a typical 2-bedroom apartment in India. The key findings of this research project are included in the following list:

- Using the co-simulations it was predicted that by controlling only the openings (windows and dampers) up to 25% reduction in the energy consumption was achieved depending on the location of the building when AC split units were used as the cooling systems. The moderate climate of Bangalore favoured more the use of day and nighttime ventilation, whilst more extreme climatic conditions (hot and dry, warm and humid) relied more on the mechanical cooling systems to maintain comfortable internal conditions. The use of dynamic setpoints for cooling, based on the adaptive theory, improved significantly the predictions of the energy savings over the static setpoints, but more importantly, it captures the existing connection between the occupants of mixed-mode buildings and their interaction with the natural or mechanical systems. The use of DEC unit as the cooling systems resulted in slightly lower energy savings since the higher levels of relative humidity restricted the use of the DEC unit and the internal air temperature was higher.
- Using the control algorithms for the DEC unit improved its performance over the conventional control approaches that can be found by the majority of the DTM tools. The use of the cooling load, for the variable speed fan cases, as the control parameter instead of the air temperature resulted in more accurate predictions of the internal air temperature (Angelopoulos *et al.*, 2019).

- The inclusion of the ceiling fan as part of the control algorithm reduced the energy predictions for cooling demand. The energy consumption for space cooling reduced by 45%, 55%, 47% and 42% for Ahmedabad, Bangalore, Chennai and New Delhi for the split units while for DEC the co-simulations predicted 44%, 52%, 39% and 38% respectively compared to the base case scenarios. As previous studies have highlighted, the elevated internal airspeeds have positives impact on the thermal sensation of the occupants, and hence when the ceiling-fan operated, the control algorithms increased the setpoint temperatures accordingly to the airspeed. The ceiling fans were placed in the two bedrooms and living room.
- The evaluation of the comfort hours that each VCS was used to analyze the ability of the control algorithms to maintain acceptable levels of thermal comfort. The inclusion of the ceiling fans in the control algorithms resulted in 1100h more comfort hours, compared to the scenarios when the ceiling fans were turned off, in Ahmedabad, and these findings are in line with previous research (Babich *et al.*, 2017). For the rest of the cities, the inclusion of the ceiling fans resulted in 1900h, 1500h and 1000h more hours of thermal comfort for Bangalore, Chennai and New Delhi respectively. The operation of the ceiling fans had a positive impact as the internal airspeeds allowed higher cooling temperatures to achieve the same levels of thermal comfort. In Bangalore, with a moderate climate, the inclusion of the fan as part of the control algorithms resulted in 93% of comfort hours for the whole year. When DEC units were used as the cooling systems, the co-simulations predicted fewer hours of thermal comfort compared to the split unit scenarios. The elevated levels of relative humidity in addition to the slightly higher predicted internal air temperatures compared to the split units resulted in approximately 8-13%, 7-15%, 19-24%, 17-22% fewer hours of thermal comfort for Ahmedabad, Bangalore, Chennai and New Delhi respectively. The IMAC model was used to assess the thermal comfort conditions using the 90% acceptability range.
- For all VCSs the difference between nighttime restriction and no restriction of the window openings is approximately between 5-10% when comparing the energy predictions. As expected, when there is no restriction at the openings the total energy demand is lower since the building thermal mass cools during nighttime. This could be also justified by the predictions of the hours of natural ventilation. The co-simulations predicted more hours of natural ventilation when there was no restriction

on the windows' openings and hence fewer hours of mechanical mode compared to the cases with nighttime restriction. Similarly, when there was no restriction in the opening area of the windows, the analysis showed higher percentages of thermal comfort compared to the nighttime restriction. In all the VCS, the use of nighttime window restriction resulted in higher internal air temperature, due to lack of sufficient cooler air flowing in and reducing the temperature of the thermal mass. However, that seems the most reasonable approach that the occupants of a flat will control their systems, especially for safety reasons.

A series of steady-state CFD simulations were performed to analyse the ventilation performance of the natural and mixed-mode ventilation strategies for buoyancy and wind-driven flows.

- The inclusion of the windows, dampers and ceiling fans were found to be the most effective way to provide ventilation rates from 4.8 to 9.6 ach^{-1} which is within the proposed limits by the standards (CIBSE A, 2005; CIBSE Guide B, 2005). In extreme climatic conditions found in countries such as in India, it is important for the natural and mixed-mode strategies, to provide outside air both in order to reduce the internal air temperature whenever possible and hence to reduce the dependence on mechanical systems. The CFD analysis demonstrated the potential of natural and mixed-mode strategies to provide thermally comfortable internal conditions. Overall, the inclusion of the ceiling fans, as part of the control approach, enhanced significantly the thermal conditions inside the rooms. Similar to previous studies (Babich, 2017) the CFD simulations showed that the use of the ceiling fan dominated the development of the internal airflows. The use of the ceiling fans resulted in less stratified temperatures and higher internal airspeeds ranging from 0.12 to 0.52 m/s depending on the case, for the buoyancy-driven forces. Under the wind-driven forces, the CFD simulations predicted internal airspeeds higher than the buoyancy-driven cases by 7 to 30% depending on the cases. It was further explored, how the operation of the internal doors can impact the development of the flow inside the apartment. Results from all CFD simulations performed demonstrated the potential for wind-driven ventilation to improve upon the buoyancy-driven ventilation, assisting reductions in indoor temperatures of up to 1°C relative to the buoyancy-driven flows.

- Finally, the ventilation effectiveness of the different cases was quantified using the HRE and CRE indices. As expected, the cross-ventilation cases with the bigger opening areas resulted in higher ventilation rates and also in higher ventilation effectiveness indices, whilst the single-side scenarios resulted in the smallest ventilation effectiveness indices. This research provides a unique insight into the ventilation performance, as it analysed the ventilation effectiveness indices (HRE and CRE) under a variety of cases. For the mixed-mode buoyancy-driven ventilation scenarios, the CRE and HRE indices ranged from 0.80-0.88 and 0.81-0.87 respectively for the single-sided ventilation systems (when internal doors were closed) and from 0.39-1.25 and 0.58-1.54 for the cross ventilation systems (internal doors open). The systems with the highest predicted values for CRE and HRE for the buoyancy-driven flows were the scenario with cross ventilation, with the ceiling fans on, the windows opened at 100%, high and low levels dampers opened at 100%. The very high ventilation rates resulted in the most effective way to dilute the contaminants' concentration and the heat generated, and the CFD simulations predicted CRE=1.25 and HRE=1.42. Similarly, for the wind-driven cases, the CFD simulations predicted that the cross-ventilation with the ceiling fan on resulted in the higher values for CRE and HRE compared to the rest of the scenarios, with CRE=1.62 and HRE=1.65.

8.3. Recommendations based on this research

Based on the outcomes of this research the following recommendations can be made:

- With regards to thermal comfort and energy savings, it is recommended to use the adaptive thermal comfort theory to calculate the cooling setpoints for all the locations. In cities with extreme weather conditions, such as Ahmedabad, Chennai and New Delhi, the use of the split units seemed the most appropriate as the maintained the internal air temperatures within the comfortable limits for more hours throughout the year compared to the DEC unit cases whilst achieved less energy consumption. For the moderate climate of Bangalore, both the DEC and split units maintained the internal temperatures within the comfort limits for a similar period of time whilst achieving similar energy savings.

- Based on this research, it is proposed to include the ceiling fans as part of the control algorithms, as it significantly improves the predictions for energy savings and comfortable hours. In cities with extreme weather conditions, such as Ahmedabad, Chennai and New Delhi, the use of the split units seemed the most appropriate as the maintained the internal air temperatures within the comfortable limits for more hours throughout the year compared to the DEC unit cases whilst achieved less energy consumption. For the moderate climate of Bangalore, both the DEC and split units maintained the internal temperatures within the comfort limits for a similar period of time whilst achieving similar energy savings. Additionally, it is important to incorporate night-time ventilation in the building whenever is possible to take advantage of the cooler outside air during the night and reduce the demand for cooling the next day.
- To achieve higher values of ventilation effectiveness and hence better performance of the ventilation systems this research suggests to use cross ventilation for the passive systems with the additional use of ceiling fans. More specifically this research suggeststhe use of low and high levels dampers together with the ues of windows and ceiling fas to achieve the most effective way to remove contamintants and/or heat from a space.

8.4. Wider significance of this research: impact on academia, industry and policy

The findings of this research have an impact on a wide range of stakeholders ranging from academia, industry and policy.

Firstly, the major impact of this research on academia is clearly the contribution to knowledge regarding the development and publication of the control algorithms for mixed-mode buildings. Secondly, this research has a positive impact on the industry. The developed control algorithms for mixed-mode buildings, in the form of the flowcharts, can be used by architects and/or designers at an early design stage. The use of the flowcharts can be used to decide during the early stage design of the building, by the control engineers, what is the most appropriate control approach to use in the BMS of the building. In addition, the control algorithms can be used by consultancy companies focusing on the control of natural and/or mixed-mode buildings, to quantify the energy savings that can be achieved by using one of the proposed control algorithms. This research has identified what are the most appropriate systems that should be used and how to control them in order to achieve energy savings while maintaining acceptable levels of thermal comfort. The use of open-source models (EnergyPlus and Modelica) to develop the co-simulations is of paramount importance for companies as it is not required to buy any licence for EnergyPlus and in the near future, the open-source compiler for Modelica will be available. Finally, the proposed methodology to calculate the ventilation effectiveness can be used by consultancy companies to assess the ventilation performance of the proposed control algorithms and provide solid evidence to the clients that the control algorithms not only achieve significant energy savings, but they maintain comfortable internal environments. For the proposed methodology to calculate the ventilation effectiveness, there was no need for high-performance computers or any additional computational power so it can be easily adopted by consultancy companies.

Lastly, although this research did not suggest any specific changes in the current national building code of India, it provides the basis for a dialogue with policymakers and suggests few indications that may be considered. Policymakers should promote the proposed control algorithms in the future guidelines, so the practitioners and architects are more likely to use them. The guidebook should include the flowcharts as well as indicative results from the current research, so when an engineer is designing a house at a specific place in India, the energy-saving potentials can be estimated based on the current analysis. This research provides solid evidence of the most appropriate control algorithms for mixed-mode buildings.

8.5. Limitations of this research and future work

This research evaluated the feasibility of the developed control algorithms for the operation of mixed-mode systems by application of these in an experimental facility. This validation study showed that both the developed control algorithms and the co-simulations can generate accurate results. This was a significant contribution in the development of control algorithms for residential mixed-mode building; however, it did not take into account the impact on the uncertainties concerning occupant’s presence and interaction with building systems, as well as the contribution of the building’s thermal mass. Future research could explore the performance of the developed controls when employed in occupied spaces. Additionally, for the validation of this research an experimental chamber was used. Although it minimizes the uncertainties during the measurements, it is an “idealised” case and hence it cannot repsresnt a real case i.e. a real building. Hence future work should focus on validating the control algorithms using data from a real building.

For the occupancy profiles, internal heat gains values from the literature based on the Indian context were used. However, as occupants tend to interact constantly with the control strategies in a room to adjust the environment and meet their preferences, it is essential to for future research to conduct surveys in occupants to examine their preferences for thermal comfort and how they would interact with the systems. This would enable to develop more accurate occupancy and internal heat gains profiles.

The developed control algorithms did not account for automated control of the shadings. Also, the control algorithms based on the rule-based approach (if statements). This approach seemed the most appropriate for this research as it can be adopted by the control engineers and implemented into the BMS of the buildings. Future research should be focused on Model Predictive Control (MPC) methods to include probabilistic models for occupancy, future weather files and hence be able to control the buildings for few hours or even couple of days in advance.

Forthly, this research used evaporative coolers as one of the cooling systems and proposed an advanced control algorithm to improve their performance. However, the analysis did not include the water consumption that is required to run the DEC unit. Hence, future research should focus on accounting the water consumption that the DEC will need and in case of large scale implementation such as several apartments, where this water will come from since there might be a water shortage.

The development of the control algorithms for mixed-mode buildings was based on utilizing the use of natural ventilation based on temperature and humidity. However, the use of natural ventilation can be affected by other parameters such as outdoor levels of pollution, noise etc. The research did not account any of those parameters because there were no available data for the outdoor pollution levels or noise levels. Hence by including those control parameters at this stage would have only added extra uncertainties. Future research should include outdoor pollution and noise levels as the control parameters. Also, future research should explore the use of IAQ filters that can be placed in the windows so the air entering the building would be as clean as possible. In many large cities, especially in developing countries such as India, outdoor pollution and noise levels are very high, so it is essential future research to build on the current control algorithms and improve them even further.

The main focus of the current research project was the development of control algorithms. For the CFD simulations, various assumptions were made to reduce the computational time, especially for the wind-driven scenarios. The surrounding buildings, the street obstacles (trees, cars etc) were not included in the analysis to reduce the computational time. Also, the number of the examined scenarios was sufficient to provide an understanding of the ventilation performance of the mixed-mode ventilation and cooling strategies, but still it was relatively small to draw conclusions for all the examined locations and weather conditions. Future research might focus on developing more scenarios for the CFD simulations to examine whether there is a correlation of the ventilation performance (ventilation rates, ventilation effectiveness) and the wind speeds, outdoor air temperatures, shape of the building, solar gains etc.

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Appendix

A. Modelica code for control algorithms

```
function Temperature_IMAC
  input Real T_hourly2[8760,4];
  output Real TcomfortIMAC_MM[8760,2],TcomfortIMAC_upper_90MM[8760,2],
TcomfortIMAC_lower_90MM[8760,2];
protected
  Real runningwindow[30],test[395],Trun[365],X2[365,24],T_average_temperature2[365,2],
T_max_daily[365,2],T_min_daily[365,2],T_daily_average[365,2];
algorithm
for i in 1:365 loop
    for j in 1:24 loop
       X2[i,j]:=T_hourly2[j + (i -
 1)*24, 3]"Create a 2X2 matrix. In the first collumn are the dates and in the second the hourly temperature";
    end for;
    T_max_daily[i,1]:=i;
    T_max_daily[i,2]:=max(X2[i, 1:24]);
  end for;
for i in 1:365 loop //finds the min for everyday
    for j in 1:24 loop
       X2[i,j]:=T_hourly2[j + (i -
 1)*24, 3] "Create a 2X2 matrix. In the first collumn are the dates and in the second the hourly temperature";
    end for;
     T_min_daily[i,1]:=i;
     T_min_daily[i,2]:=min(X2[i, 1:24]);
end for;
for i in 1:365 loop //finds the average for everyday based on max and min
    for j in 1:24 loop
       X2[i,j]:=T_hourly2[j + (i -
 1)*24, 3] "Create a 2X2 matrix. In the first collumn are the dates and in the second the hourly temperature";
    end for;
    T_average_temperature2[i,1]:=i;
    T_average_temperature2[i,2]:=sum(X2[i, 1:24])/24;
end for;
  for i in 1:365 loop
     T_daily_average[i,1]:=i;
     T_daily_average[i,2]:=(T_max_daily[i, 2] + T_min_daily[i, 2])/2;
```

```
  end for;
for i in 1:365 loop //calculates the running mean
  if i<=30 then
     for j in 1:30 loop
       test[1:365]:=T_daily_average[1:365,2];
       test[366:395]:=T_daily_average[1:30,2];
       runningwindow[j]:=(test[335+j+(i-1)]);
     end for;
       Trun[i]:=sum(runningwindow[1:30])/30;
  else
     for j in 1:30 loop
       runningwindow[j]:=(T_daily_average[j + (i - 31), 2]);
     end for;
      Trun[i]:=sum(runningwindow[1:30])/30;
  end if;
end for;
for k in 1:365 loop //creates a matrix with daily values of comfort temperature for IMAC thermal comfort model
   for l in 1:24 loop
    TcomfortIMAC_MM[l+(k-1)*24,2]:=0.28*Trun[(k - 0)]+17.87;
    TcomfortIMAC_MM[l+(k-1)*24,1]:=l+(k-1)*24;
   end for;
end for;
 for k in 1:365 loop  //creates a matrix with daily values of cooling setpoint based on the IMAC thermal comfort
model
   for l in 1:24 loop
    TcomfortIMAC_upper_90[l+(k-1)*24,2]:=TcomfortIMAC_MM[l+(k-1)*24,2]+3.5;
    TcomfortIMAC_upper_90[l+(k-1)*24,1]:=TcomfortIMAC_MM[l+(k-1)*24,1];
   end for;
 end for;
 for k in 1:365 loop  //creates a matrix with daily values of of heating setpoint based on the IMAC thermal comf
ort model
   for l in 1:24 loop
    TcomfortIMAC_lower_90MM[l+(k-1)*24,2]:=TcomfortIMAC_MM[l+(k-1)*24,2]-3.5;
    TcomfortIMAC_lower_90MM[l+(k-1)*24,1]:=TcomfortIMAC_MM[l+(k-1)*24,1];
   end for;
 end for;
end Temperature_IMAC;
```

B. Inform Code for PHOENICS

The following code files allow for the calculation of the key parameters needs to calculate the ventilation effectiveness, and ventilation flow rate.
In this script two small sized openings were being used, both bi-directional but movement in both directions is still calculated. This is part of the whole code that was applied in all the openings (windows/dampers in both the experimental chamber and the 2-bedroom apartment).

The save14 block in these simulations is used to calculate the average contaminant concentration, and total volume of the breathing zone.
save14begin # start of code block. Inform code is stored in particular save block
depending on the functionality of the code.
(make voltot is 0.0) # First declare the variable and set to zero, total volume
(make conttot is 0.0) # total contaminant concentration
(store1 of voltot at ROOMAV is sum(vol) with imat< 100) # User store1 function to
store the main calculations
object is a user de_ned object with the dimensions of the breathing zone where the
breathing zone variable are calculated
(store1 of conttot at ROOMAV is sum(C1*vol) with imat< 100) # the imat< 100
statement is to stop any calculation occuring in cells occupied by objects
text document after simulation run ends
(print of cont avg is conttot/voltot)
(table in room avg.csv is get(voltot ,conttot/voltot) with head(volume ,cont avg)!sweep) # table function writes
assigned variables to a document e.g.
csv which is stored in the d priv1 folder in the phoenics directory
save14end # end of code block

The save15 block is used in these simulations to calculate the total contaminant concentration entering and leaving the space, and the velocity of air entering and leaving the space.

save15begin # start of code block
(make asump is 0.0)
(make asumm is 0.0)
(make vsump is 0.0)
(make vsumm is 0.0)
(make contsump is 0.0)
(make contsumm is 0.0)
(store1 of asump at WIN1 is sum(anorth) with imat< 100 ! if (v1.gt.0.)) # v1.gt.0
means only calculate for cells where the velocity in the v1 direction (Y direction) is greater
than zero. For X direction change v1 to w1 and for Z direction change v1 to u1.

```
(store1 of vsump at WIN1 is sum(anorth*v1) with imat< 100 ! if (v1.gt.0.)) # anorth
is cell direction facing along the Y direction, aeast is facing along the X direction and
ahigh is facing along the Z direction.
(store1 of contsump at WIN1 is sum(anorth*C1) with imat< 100 ! if (v1.gt.0))
(print of av vel pos is vsump/asump)
(print of tot vel pos is vsump)
(print of cont pos is contsump)
(store1 of asumm at WIN1 is sum(anorth) with imat< 100 ! if (v1.lt.0.))
(store1 of vsumm at WIN1 is sum(anorth*v1) with imat< 100 ! if (v1.lt.0.))
(store1 of contsumm at WIN1 is sum(anorth*C1) with imat< 100 ! if (v1.lt.0))
(print of av vel neg is vsumm/asumm)
(print of tot vel neg is vsumm)
(print of cont neg is contsumm)
 (table in win1.csv is get(vsump, vsumm, vsump/asump, vsumm/asumm, contsump,
 (make asump2 is 0.0)
(make asumm2 is 0.0)
(make vsump2 is 0.0)
 (make contsump2 is 0.0)
(make contsumm2 is 0.0)
(make agesump2 is 0.0)
 (store1 of asump2 at WIN2 is sum(anorth) with imat< 100 ! if (v1.gt.0.))
(store1 of vsump2 at WIN2 is sum(anorth*v1) with imat< 100 ! if (v1.gt.0.))
(store1 of contsump2 at WIN2 is sum(anorth*C1) with imat< 100 ! if (v1.gt.0))
 (print of av vel pos2 is vsump2/asump2)
(print of tot vel pos2 is vsump2)
(print of cont pos2 is contsump2)
 (store1 of asumm2 at WIN2 is sum(anorth) with imat< 100 ! if (v1.lt.0.))
(store1 of vsumm2 at WIN2 is sum(anorth*v1) with imat< 100 ! if (v1.lt.0.))
(store1 of contsumm2 at WIN2 is sum(anorth*C1) with imat< 100 ! if (v1.lt.0))
 (print of av vel neg2 is vsumm2/asumm2)
(print of tot vel neg2 is vsumm2)
(print of cont neg2 is contsumm2)
 (table in win2.csv is get(vsump2, vsumm2, vsump2/asump2, vsumm2/asumm2, contsump2,
contsumm2, asump2, asumm2) with
head(Total Vel p, Total Vel m, avg vel p, avg vel m, cont p OUT, cont m IN, Area OUT, Area IN)!sweep)
save15end # end of code block
```

C. Control algorithms for no window restriction

Ventilation and Cooling Strategy 1

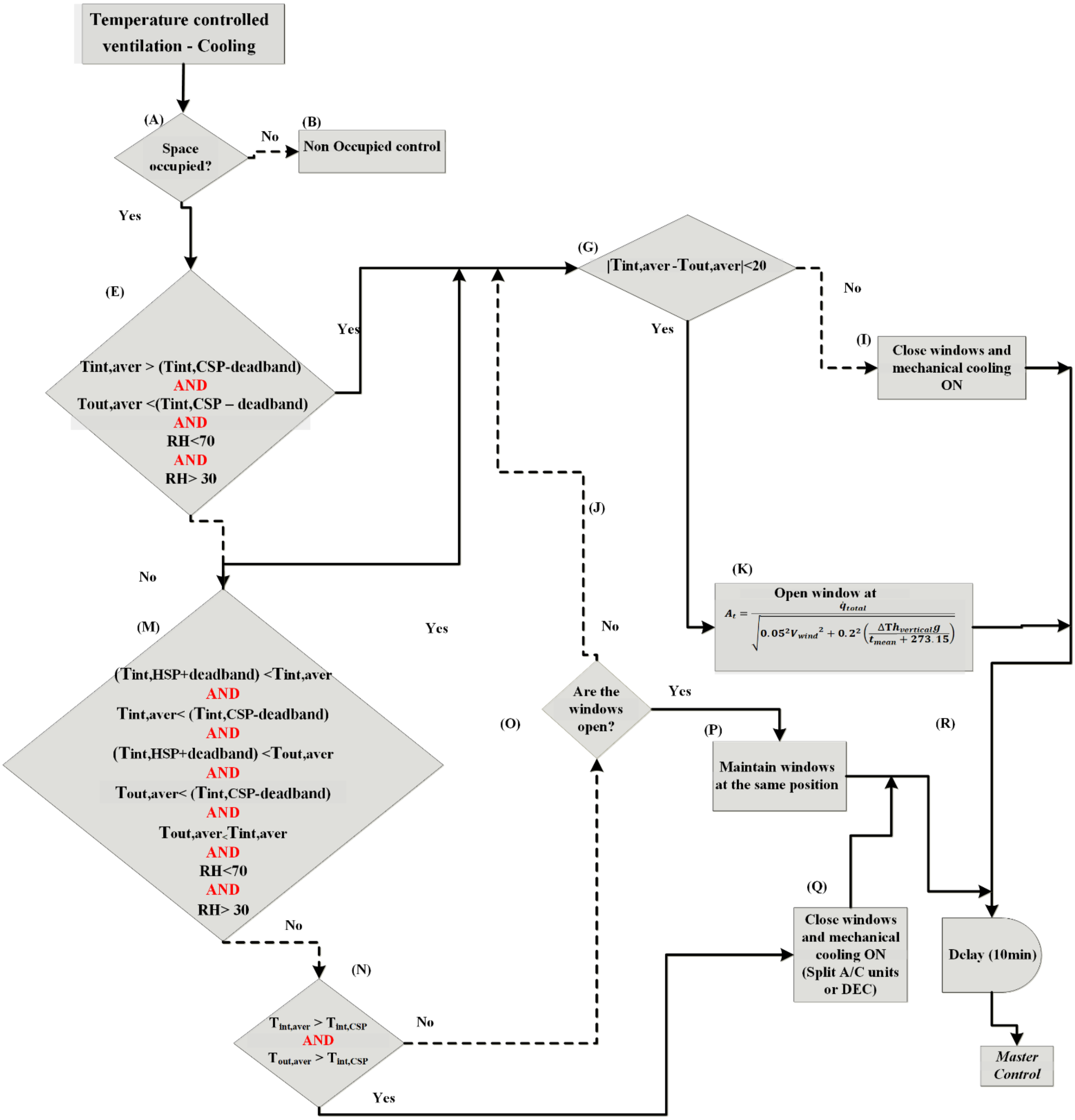

Figure B 1: Control algorithm for ventilation and cooling scenario (VCS) 1

Ventilation and Cooling Strategy 2

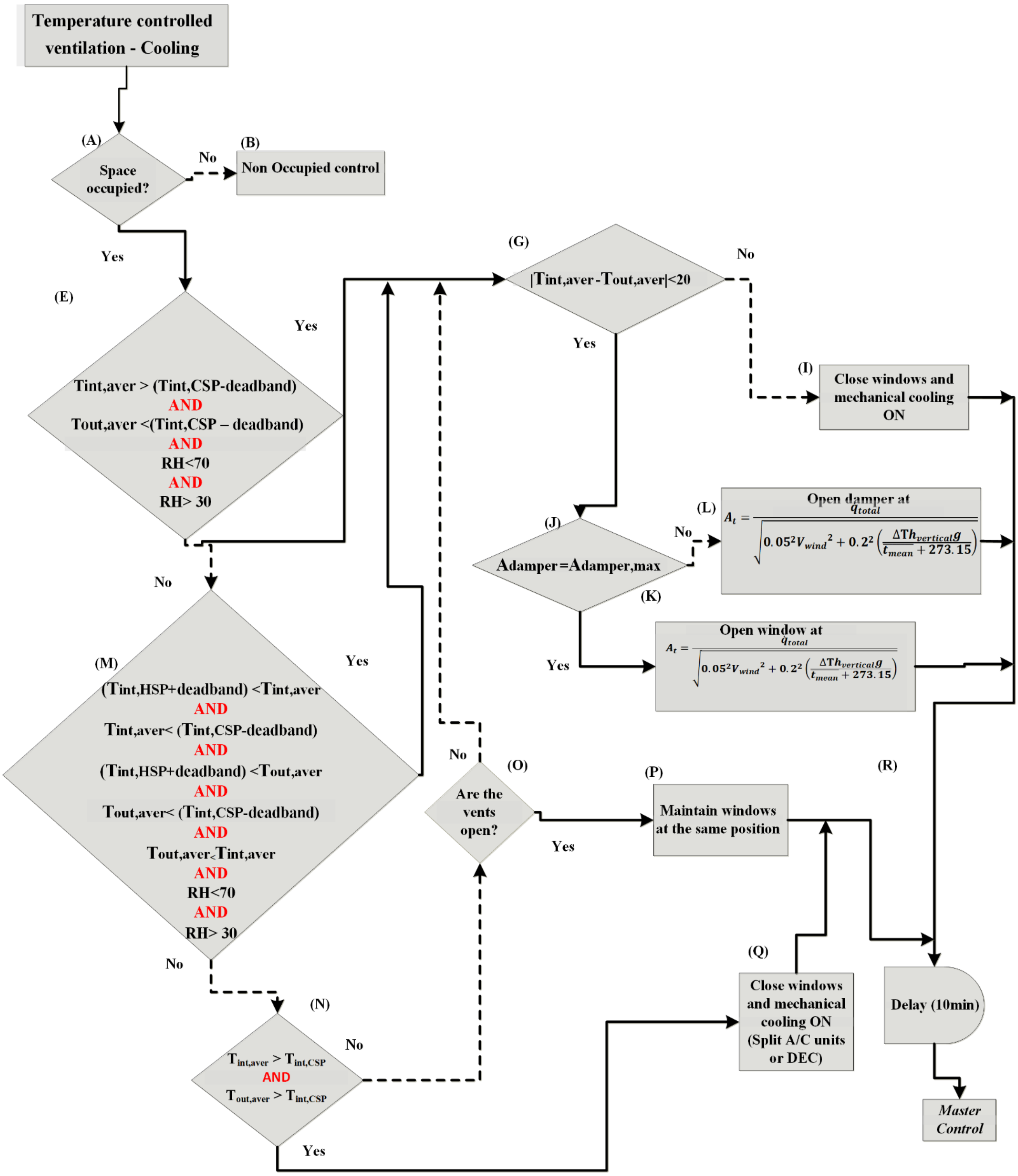

Figure B 2: Control algorithm for ventilation and cooling scenario (VCS) 2

Ventilation and Cooling Strategy 3

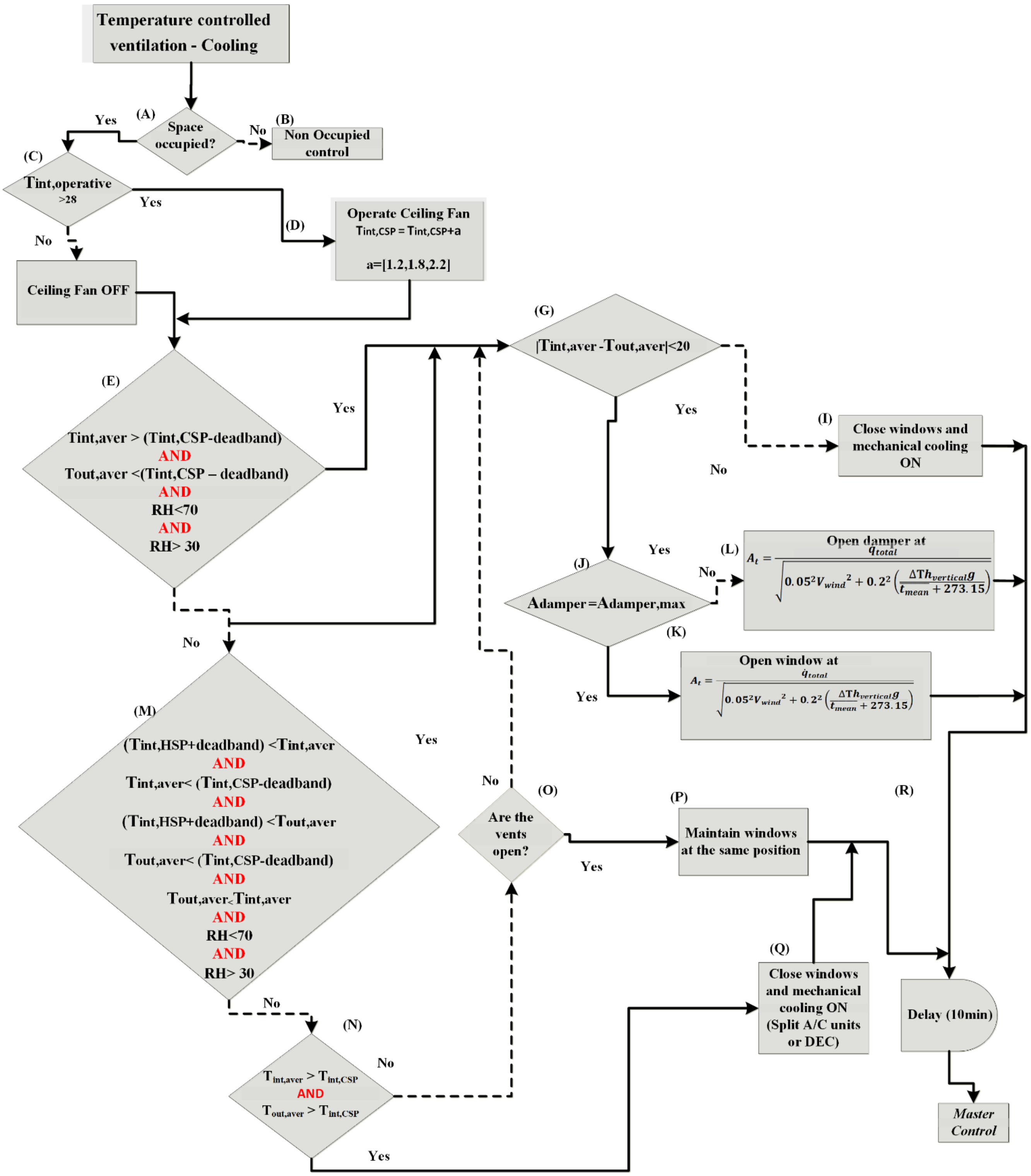

Figure B 3: Control algorithm for ventilation and cooling scenario (VCS) 3

Ventilation and Cooling Strategy 4

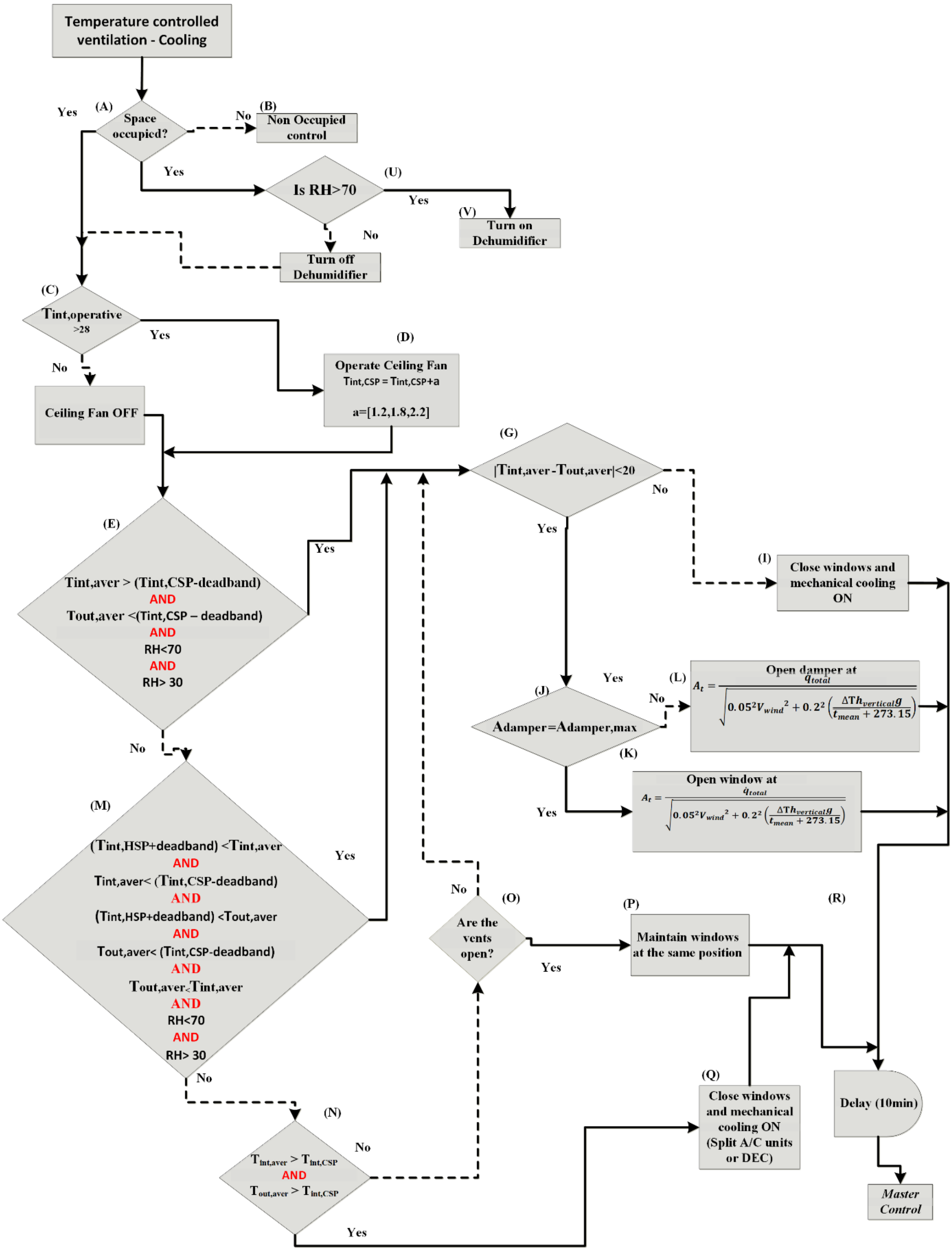

Figure B 4: Control algorithm for ventilation and cooling scenario (VCS)