

Precision Ventilation in Open-Plan Offices

A Study of Variable Jet Interaction Between Active Chilled Beams

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**PRECISION VENTILATION IN OPEN-PLAN
OFFICES: A STUDY OF
VARIABLE JET INTERACTION
BETWEEN ACTIVE CHILLED BEAMS**

**BY
HAIDER LATIF**

DISSERTATION SUBMITTED 2023



AALBORG UNIVERSITY
DENMARK

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by

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Dissertation submitted

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ABSTRACT

Acceptable thermal comfort for all office employees is considered a key function of HVAC systems installed in office buildings. Offices with an open-plan layout often have a higher occupant density than conventional office designs and there are areas in such shared workspaces that may include occupants with different energy releases, defined as different metabolic rates. The differences in the metabolic rates of occupants depends upon their different activities and individual body characteristics. It is a widely known fact that thermal comfort preferences may vary due to individual characteristics like gender, age and health condition. Over the years, conventional ventilation systems e.g., mixing ventilation systems, were designed to provide uniform thermal comfort conditions in office spaces. Active Chilled Beams (ACBs) are a type of mixing ventilation system that maintains thermal uniformity, typically in offices, by mixing cold fresh air from the ceiling into warm room air. The thermal uniformity achieved through these systems is insufficient to satisfy all office occupants with different metabolic rates and requires a change in ventilation strategy to improve individual thermal comfort.

The objective of this study was to bring about advancements in the existing mixing ventilation system through the development of a novel ventilation strategy by raising air temperatures and varying air velocities to achieve individual thermal comfort for office occupants. This is achieved by using ACBs with a JetCone feature to adjust the supply of airflow. Variable air velocity zones were established by regulating JetCones in ACBs at different positions to adjust airflow strength and colliding jets according to the occupant's metabolic requirements. This newly introduced precision ventilation strategy was examined in different case studies involving 1, 2 and 4 ACBs for different office configurations. The simulations and experiments were carried out under adiabatic wall conditions with supply air temperature approximately 3.5K lower than room air and heat load of approximately 25 W/m². The total supply of airflow to the room was kept constant and distributed to different zones in the same office space to establish different micro-climate zones. To satisfy occupants with different metabolic rates, different JetCone opening positions were used to establish high, medium and low air velocity zones with air velocities of 0.65 m/s, 0.45 m/s and 0.15 m/s, respectively.

In the occupied zone, an acceptable thermal comfort level ($-0.5 < \text{PMV} < 0.5$) for occupants with higher metabolic rates was maintained by increased local air velocities. For all the cases, the vertical and horizontal air temperature differences (of less than 2°C) in a room showed that precision ventilation is still a mixing ventilation system, even with variable air velocity zones. Occupants with 1.2 met were exposed to low air velocities (less than 0.15 m/s) and had draught rates (DR) of less than 20%, while simultaneously maintaining higher air velocities in the zones with occupants with higher metabolic rates. By using precision ventilation during the cooling period,

the annual energy savings from heating and cooling were 15% and this was achieved by raising the cooling setpoint temperatures from 23°C to 25°C.

DANSK RESUME

Acceptabel termisk komfort for alle personer med kontorarbejde betragtes som en nøglefunktion af HVAC-systemer installeret i kontorbygninger. Kontorer med en åben planløsning har ofte højere persontæthed end konventionelt kontordesign, og der er områder i sådanne fælles arbejdspladser, som kan omfatte personer med forskellige stofskiftehastighed (metabolisk hastighed). Forskellene i personernes stofskiftehastighed afhænger af deres forskellige aktiviteter og individuelle kropsegenskaber. Det er et almindeligt kendt faktum, at præferencer for termisk komfort kan variere på grund af individuelle egenskaber som køn, alder og helbredstilstand. I årenes løb er konventionelle ventilationssystemer, f.eks. opblandingsventilationssystemer, designet til at give ensartede termiske komfortforhold i kontorlokaler. Active Chilled Beams (ACB'er) er en type opblandingsventilationssystem, der bibeholder termisk ensartethed typisk på kontorer ved at blande frisk kold luft fra loftet til varm rumluft. Den termiske ensartethed, der opnås gennem disse systemer, er utilstrækkelig til at tilfredsstille alle personer med kontorarbejde med forskellige stofskiftehastighed og kræver en ændring i ventilationsstrategien for at forbedre den individuelle termiske komfort.

Formålet med denne undersøgelse er at forbedre det eksisterende opblendeventilationssystem gennem udvikling af en ny ventilationsstrategi der hæve lufttemperaturerne og variere lufthastighederne for at opnå individuel termisk komfort for personer med kontorarbejde. Dette opnås ved at bruge ACB'er med skråtstillede dyser til at justere indblæsningsluftstrømmen. Variable lufthastighedszoner blev etableret ved at regulere skråtstillede dyserne i ACB'er på forskellige positioner for at justere kastelængder og luftkollision (colliding jets) i henhold til personernes stofskiftehastighed. Denne nyligt introducerede præcisionsventilationsstrategi blev undersøgt gennem forskellige casestudier, der involverede 1, 2 og 4 ACB'er til forskellige konfigurationer. Simuleringerne og eksperimenterne blev udført under adiabatisk vægforhold med tilluft temperatur ca. 3,5K lavere end rumluft og varmebelastninger på ca. 25 W/m². Den samlede indblæsningsluftstrøm til rummet blev holdt konstant og fordelt til forskellige zoner i samme kontorlokale for at etablere forskellige mikroklimazoner. Forskellige skråtstillede dyser-åbningspositioner blev brugt til at etablere høj, medium og lav lufthastighedszoner med lufthastigheder på henholdsvis 0,65 m/s, 0,45 m/s og 0,15 m/s for at tilfredsstille personer med kontorarbejde med forskellige stofskiftehastighed (metaboliske hastigheder).

Acceptabelt termisk komfortniveau ($-0,5 < PMV < 0,5$) i opholdszonen for personer med kontorarbejde med højere stofskiftehastighed (metaboliske hastigheder) blev opretholdt af øgede lokale lufthastigheder. De lodrette og vandrette lufttemperaturforskelle i rummet på mindre end 2°C for alle tilfældene viste, at præcisionsventilation stadig er et opblendeventilationssystem på trods af zoner med variabel lufthastighed. Personerne med 1,2 met (metaboliske hastigheder) blev udsat for lave lufthastigheder (mindre end 0,15m/s) og havde træk variationer i

opholdszonen i lokaler med stillesiddende aktivitet der er mindre end 20 pct., mens de samtidig bibeholdt de højere lufthastigheder i zonerne med højere stofskiftehastigheder. De årlige energibesparelser ved opvarmning og afkøling blev fundet op til 15 % ved at hæve rumtemperaturerne fra 23°C til 25°C.

PREFACE

Thesis Title:	Precision Ventilation in Open-Plan Offices: A Study of Variable Jet Interaction between Active Chilled Beams
Ph.D. Student:	Haider Latif
Supervisors:	Professor Alireza Afshari, Aalborg University
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This thesis is submitted to the Ph.D. school of engineering as partial fulfilment of the Ph.D. degree. The current study was conducted at the Department of the Built Environment at Aalborg University from September 2019 to January 2023. The experiments related to this study were conducted at the Department of the Built Environment from January 2020 to September 2022. The simulation study was partly conducted at the Center of Integrated Building Systems, School of Mechanical and Aerospace Engineering of Oklahoma State University, Stillwater, USA. The author deeply appreciates the financial support provided by Lindab A/S and Aalborg University. This dissertation is article based and the published scientific articles listed below comprise the main body of this thesis. Co-author statements are available for the Ph.D. school and assessment committee. The current dissertation is available for limited and closed circulation as copyrights may not be ensured.

The main body of this thesis consists of the following papers:

- [1] H. Latif, S. Rahnama, G. Hultmark, K. Rupnik, and A. Afshari, “Design Strategies for Decreasing Cooling Demand and Increasing Individual Thermal Comfort in Open-Plan Offices: A Review,” in *Indoor Air 2020*, Online, p. ABS-0351.
- [2] H. Latif, G. Hultmark, S. Rahnama, K. Rupnik, and A. Afshari, “Active Chilled Beams for Producing Symmetric Air Discharge Patterns Using Jet-Cone System –An Experimental Study,” in *Health. Build. 2021-America*, Online.

- [3] H. Latif, S. Rahn timer, A. Maccarini, G. Hultmark, K. Rupnik, and A. Afshari, "Impact of Thermal Heat Loads on Air Distribution Patterns in Open-Plan Offices with Active Chilled Beams (ACBs)," in *Indoor Air 2022*, Kuopio, Finland.
- [4] H. Latif, G. Hultmark, S. Rahn timer, A. Maccarini, and A. Afshari, "Performance Evaluation of Active Chilled Beam Systems for Office Buildings – A Literature Review," *Sustainable Energy Technologies and Assessments*, vol. 52, p. 101999, 2022.
- [5] H. Latif, S. Rahn timer, A. Maccarini, G. Hultmark, P. V. Nielsen, and A. Afshari, "Precision Ventilation in an Open-Plan Office: A New Application of Active Chilled Beam (ACB) with a JetCone Feature," *Sustainability*, vol. 14, no. 7, p. 4242, 2022.
- [6] H. Latif, S. Rahn timer, A. Maccarini, C. R. Bradshaw, G. Hultmark, P. V. Nielsen, and A. Afshari, "Precision Ventilation for an Open-Plan Office: A Study of Variable Jet Interaction between Two Active Chilled Beams," *Sustainability*, vol. 14, no. 18, p. 11466, 2022.
- [7] H. Latif, A. Maccarini, G. Hultmark, P. V. Nielsen, S. Rahn timer, and A. Afshari, "The Establishment of Design Criteria for Precision Ventilation in Open-plan Offices," Submitted in Building and Environment (December 2022).

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Haider Latif
Aalborg University, January 16, 2023

*To my beloved father, Shahid Latif,
passionate about building science, who motivated me and backed me throughout my
PhD journey*

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Nomenclature

The nomenclature list includes the abbreviations and symbols used in the current Ph.D. thesis.

AC	Air Conditioning
ACBs	Active Chilled Beams
AHU	Air Handling Unit
ATDs	Air Terminal Devices
CAV	Constant Air Volume
CFD	Computational Fluid Dynamics
DR	Draught Rate [%]
DV	Displacement Ventilation
FCU	Fan Coil Unit
HVAC	Heating Ventilation and Air Conditioning
IAQ	Indoor Air Quality
IPV	Intermittent Personalized Ventilation
IR	Induction Ratio
MRT	Mean Radiant Temperature [°C]
MV	Mixing Ventilation
PCBs	Passive Chilled Beams
PD	Percentage Dissatisfied [%]
PIV	Particle Image Velocimetry
PMV	Predicted Mean Vote
PPD	Predicted Percentage Dissatisfied [%]
PV	Personalized Ventilation
RH	Relative Humidity [%]
RMP	Round Movable Panel
SBS	Sick Building Syndrome
UFAD	Underfloor Air Distribution
VAV	Variable Air Volume
VDG	Vertical Desk Grille
VTD	Vertical Temperature Difference [°C]
a	Supply area [m ²]
a_o	Supply air opening area [m ²]
Ar	Archimedes number [-]
C	Convective heat transfer [W/m ²]
C_{res}	Convective heat loss from respiration [W/m ²]
C_p	Specific heat capacity of air [J/kg-K]
E	Evaporative heat loss [W/m ²]
E_{res}	Evaporative heat loss from respiration [W/m ²]
F_{W-h}	Angle factor between thermal mannequin and room configuration [-]
f_{conv}	Convective heat transfer area ratio of thermal mannequin [-]
f_{rad}	Radiative heat transfer area ratio of thermal mannequin [-]

g	Gravity [m/s^2]
Gr	Grashof Number [-]
h_c	Convective heat transfer coefficient [$\text{W/m}^2.\text{K}$]
H	Heat generation inside the space [W]
I	Momentum of airflow to the space [kg.m/s]
K	Convective heat loss [W/m^2]
L	Characteristic linear dimension [m]
M	Metabolic rate [met]
Q	Total sensible heat transfer [W/m^2]
V	Ventilation Rate [m^3/hr]
Q_1	Amount of secondary air induced per second [l/s]
Q_2	Amount of primary air induced per second [l/s]
R	Radiative Heat Loss [W/m^2]
Re	Reynolds Number [-]
S	Rate of body heat storage [W/m^2]
T_a	Surrounding air temperature [$^{\circ}\text{C}$]
T_s	Surface temperature of thermal mannequin [$^{\circ}\text{C}$]
T_r	Mean radiant temperature of thermal mannequin [$^{\circ}\text{C}$]
Tu	Local turbulence intensity [%]
T_i	Indoor air temperature [K] [$^{\circ}\text{C}$]
T_o	Outdoor air temperature [K] [$^{\circ}\text{C}$]
t_a	Local air temperature [$^{\circ}\text{C}$]
u	Supply air velocity [m/s]
u_o	Air velocity at the opening area [m/s]
u_x	Air velocity at a certain distance away from the opening [m/s]
v_a	Local mean air velocity [m/s]
ν	Kinematic viscosity [m^2/s]
W	Mechanical power [W/m^2]
x	Distance away from the opening [m]
x_o	Distance close to the opening [m]
x_s	Penetration Depth [m]
ε_h	Emissivity of thermal mannequin surface [-]
ε_w	Emissivity of the interior surface configuration [-]
β	Coefficient of thermal expansion [K^{-1}]
ΔT	Temperature difference of the human body and surroundings [$^{\circ}\text{C}$]
$\Delta t_{a,v}$	Vertical air temperature difference between head and feet [$^{\circ}\text{C}$]
t_{sk}	Human body skin temperature [$^{\circ}\text{C}$]
t_{am}	Ambient air temperature [$^{\circ}\text{C}$]
ρ	Density [kg/m^3]
σ	Stefan-Boltzmann constant [$\text{W/m}^2.\text{K}^4$]

PART I

INTRODUCTION

CHAPTER 1. INTRODUCTION

1.1. BACKGROUND AND MOTIVATION

Offices in Scandinavia are often designed with open-plan office layouts, which help in promoting communication and knowledge sharing among employees [1]. Office occupants working in such shared spaces may be individually influenced by the thermal environment, which depends upon personal and environmental factors. For the past several decades, HVAC systems in buildings have contributed to improving the level of thermal comfort for office occupants. These HVAC systems consist of different types depending on their function, operation and position in buildings and are designed to provide a satisfactory indoor climate by controlling environmental factors i.e., air velocity, air temperature, humidity and radiant temperature [2]. Energy savings in buildings is another aspect that needs to be taken into account while designing any HVAC system and is considered an important part of European energy policy [3]. Therefore, components of HVAC systems are designed to meet the building's ventilation needs in an energy efficient manner. In addition to this, further modifications in existing HVAC systems are required to increase its applications for achieving individual thermal comfort for occupants in open-plan offices.

Most of the HVAC systems used around the world are designed to establish a uniform thermal environment. Mixing ventilation (MV) is one such type used widely in buildings to establish thermal uniformity through adequate air mixing (between supply and room air) by providing fresh air from the ceiling [4]. However, this uniform thermal environment in any shared office space may involve occupants of different ages, gender, health, clothing, or metabolic rates that can result in individual thermal discomfort [5]. Personalized Ventilation (PV) systems (also known as task ambient or personalized conditioning systems) [6] were introduced to satisfy individuals by establishing micro-climate zones in the same office space. Multiple Air Terminal Devices (ATDs) like desk-chair fans, heating chair etc., were used on a desk-chair arrangement to control air velocity and temperature locally. These PV systems were encouraged in the beginning due to their potential energy savings up to 40%, by increasing the room setpoint temperature and reductions in airflow [7]. However, these systems had a setback due to the direct blow of air from ATD inlets onto the occupants, which can cause discomfort. Furthermore, the large number of ATDs around occupants does not fit well with respect to building aesthetics and the complexity involved in synthesizing all these ATDs together is challenging for HVAC designers.

In the research community, there is a growing consensus on the need for developing modifications to existing or conventional building ventilation systems to fulfill the need for individual thermal comfort. Studies have shown that workers' productivity is enhanced if buildings are provided with ventilation systems that have individual

thermal comfort controls [8]–[10]. In Scandinavia, Active Chilled Beams (ACBs) are a successful HVAC product that have been on the market for the past 20 years. This is due to the potential of saving energy and reducing cost as ACBs use less building space. These ACBs are a type of MV system that provides fresh air from the ceiling to the occupied zone. Like other MV systems, ACBs have been providing thermal uniformity in offices by directing constant airflow from the ceiling to the occupants [11]. The direct blow of air to the occupants is avoided due to the jet attachment to the ceiling and walls i.e., Coanda effect. ACBs with a JetCone feature can regulate airflow patterns by adjustment regulators. Multiple ACBs are required to meet the cooling demand of open-plan offices and the jet collisions caused by these multiple ACBs can produce high air velocities down towards the floor. The variable air velocity zones can be established in areas where occupants with different metabolic rates are located by directing the high-speed colliding jets towards or away from them. Unlike PV systems, advancements in applications of existing ACB systems not only eliminates the need for multiple ATDs in the occupied region but also can provide individual thermal comfort without disturbing the office configuration. Significant energy savings can be achieved with this system by raising the room setpoint temperature and increasing the air velocities (in the specific zones) to maintain the individual thermal comfort level.

1.2. OBJECTIVES

The objective of this work is to utilize ACBs with JetCones to achieve individual thermal comfort at a constant room temperature. This newly named precision ventilation system will establish high and low air velocity zones in open-plan offices by directing variable airflow by JetCone adjustments. The high-speed colliding jets from ACBs (adjusted by JetCones) can be directed towards areas where occupants have high metabolic rates, while occupants with lower metabolic rates can be exposed to shorter air throws. Energy saving potential is evaluated by increasing the room setpoint temperature to certain degrees while maintaining thermal comfort for all occupants (even with different metabolic rates). Figure 1 shows the conceptual diagram of the new proposed system with different micro-climate zones in the same shared office space through high-velocity colliding jets.

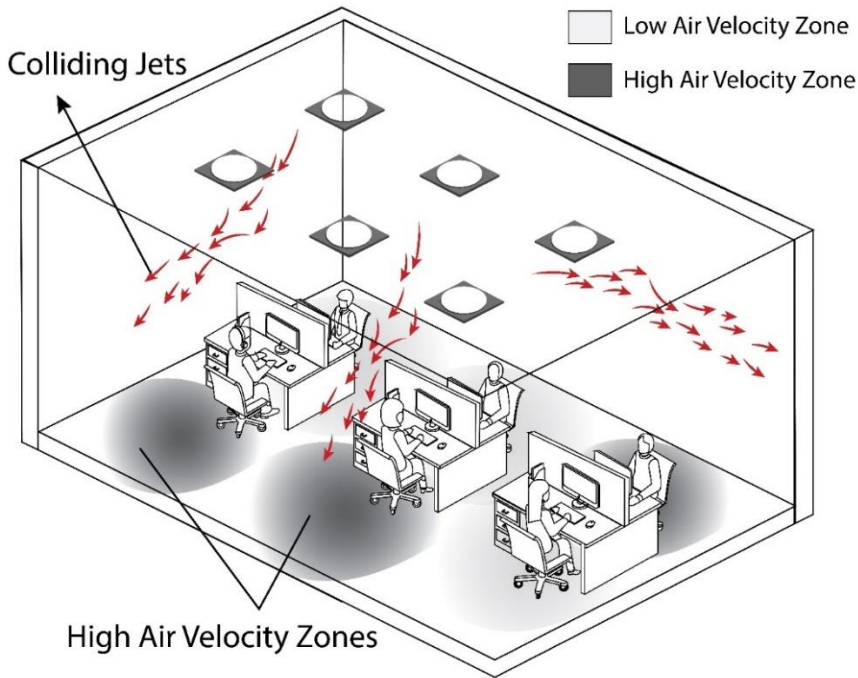


Figure 1: Concept of precision ventilation in an open-plan office

The work was mainly performed using Computational Fluid Dynamics (CFD) simulations validated through laboratory experiments.

1.3. RESEARCH QUESTIONS

The following research questions are addressed in this thesis:

1. Can ACBs create micro-climate zones by directing airflow along the ceiling in the occupied zone without involvement of multiple ATDs in the breathing zone?
2. Through precision ventilation, can individual thermal comfort be achieved for all office occupants with different metabolic rates simultaneously at a constant primary airflow and supply air temperature?
3. Is there any effect of room size, number of ACBs and occupants on the application of precision ventilation?
4. What will be the effect of variable air velocity zones on vertical and horizontal temperature uniformity in the room?
5. What is the maximum room temperature limit at which occupants will start feeling discomfort with precision ventilation?

6. How much annual energy savings will be achieved with an approximate increase in room temperature of 2°C?

1.4. LIMITATIONS

1. In this thesis, the precision ventilation technique is used to provide satisfactory individual thermal comfort for occupants with different metabolic rates, and the aspect of Indoor Air Quality (IAQ) is not considered.
2. The precision ventilation applications are presented through simulations and laboratory studies. Implementation in real office buildings is not addressed.
3. The performance of precision ventilation is evaluated using constant cooling power per square meter in the room.
4. The configuration of the air terminal devices is limited to three cases i.e., 1, 2 and 4 ACBs.

1.5. OUTLINE OF THE THESIS

This thesis is divided into three parts with a total of twelve chapters.

Part I includes three chapters that present the literature and motivation behind this research, results of the main topics of the research project, research gap and concept of this research project.

- **Chapter 1** presents the background, motivation, research questions, objective and limitations of the PhD project.
- **Chapter 2 and 3** details the background of thermal comfort and HVAC technologies used to maintain both overall and local thermal comfort in offices. The need to achieving individual thermal comfort through precision ventilation is also described in these chapters.

Part II involves **Chapters 4-10**, which includes the collection of papers written and published during three years of PhD.

- **Chapter 4**
Design Strategies for Decreasing Cooling Demand and Increasing Individual Thermal Comfort in Open-plan Offices – A Review
- **Chapter 5**
Active Chilled Beams for Producing Symmetric Air Discharge Patterns Using Jet-Cone System – An Experimental Study
- **Chapter 6**
Impact of Thermal Heat Loads on Air Distribution Patterns in Open-Plan Offices with Active Chilled Beams (ACBs)

- **Chapter 7**
Performance Evaluation of Active Chilled Beam Systems for Office Buildings – A Literature Review
- **Chapter 8**
Precision Ventilation in an Open-Plan Office: A New Application of Active Chilled Beam (ACB) with a JetCone Feature
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- **Chapter 10**
The Establishment of Design Criteria for Precision Ventilation in Open-Plan Offices

Part III includes the conclusions and discussion sections of this thesis **with Chapter 11** bringing the results together from all research papers and answering the research questions raised in this PhD project. **Chapter 12** includes the scientific output of this thesis in the form of concluding remarks and suggests future research.

CHAPTER 2. THERMAL COMFORT IN OFFICES

In this digital age, offices around the world still play a significant role in promoting onsite work culture and teamwork. The literature has shown that the physical environment surrounding occupants has a significant impact on occupant's behavior and work productivity [12]–[14]. In this regard, organizations design offices installed with HVAC systems to mechanically maintain a comfortable and a healthy environment among the employees. Hence, a sustainable office design with an efficient HVAC system is an organizational demand to keep its employees satisfied.

2.1. OPEN-PLAN OFFICE LAYOUTS

The concept of the open-plan office was first proposed in Europe by German furniture manufacturers to promote knowledge sharing, save office space and to improve environmental conditions [15]. Open-plan offices involve the absence of partitions and walls between work desks or a workstation whose outer boundaries do not reach the ceiling [16], [17]. This classification of offices based on physical layout is shown in figure 2.



Figure 2: Typical open-plan office layout used in this study

Unlike private office rooms, open-plan offices contain more employees per area, leading to an enhanced communicative environment and pattern of working in groups [18]. These office layouts can vary with respect to their overall space size and use. These layouts are flexible to configuration changes, leading to a reduction in unusable area and can be more cost economical by up to 20% compared to traditional offices [19]. The design flexibility of open-plan office layouts is encouraged due to equal work opportunities for all employees (regardless of seniority) to share the same office space [20]. In the past, open-plan office layouts gained a lot of popularity leading to improvement in work performance and job satisfaction for the employees [21]. A study based on surveys showed that employees job satisfaction and privacy can be improved by implementing different open-plan office layouts [22]. Some studies identified that the physical environment has a significant impact on job satisfaction and refurbishments to open-plan offices can enhance employee satisfaction [23], [24]. But still, the nature of the layout of these offices can also have a negative impact on employees' satisfaction [25]. Based on employee surveys and field studies [26]–[28], these office layouts are discouraged because of the lack of privacy, increased distraction, uniform thermal environment for occupants with different age groups, and acoustics problems. One qualitative study [29] used manager ratings to observe the impact of open-plan offices on employee's performance. Results showed that less privacy leading to low work concentration affects work performance in such setups. An experimental study showed that occupants' privacy remains a concern for open-plan office occupants due to their layout and narrow working space [30]. Some studies indicated noise control as a potential problem for occupants working in open-plan offices [26], [31], [32]. However, these problems do not offset the advantages of the layout of open-plan offices. Despite these disadvantages, open-plan offices are found aesthetically appealing and adaptive and are used widely in Europe and USA. Compared to traditional office designs, controlling environmental parameters (like air velocity, air temperature etc..) seems challenging with such layouts, as occupant density is high and it is difficult to achieve individual thermal comfort due to difference of age, gender, health and activity level of employees. In addition to this, there are also transitional areas in offices which may involve non-sedentary activities where a change of environmental conditions is required to maintain suitable thermal comfort conditions for employees [33]–[35].

2.2. THERMAL COMFORT

As described above, office layouts play an important role in occupants' productivity [36]. Office buildings also require mechanical HVAC systems to fulfill different thermal comfort needs of the occupants. Thermal comfort is defined as *the state of mind, that expresses satisfaction with the thermal environment* [37]–[39]. Another definition specifies the state in which there is no tendency to correct thermal conditions of an environment based on occupant behavior [40].

Despite the known fact that people spend up to 90% of their time indoors [41], [42], it has been reported in the literature that many people still remain unsatisfied with the indoor thermal environment [43], [44]. The thermal comfort in an office environment affects occupants' productivity and work efficiency. Any discomfort with the thermal environment will not only have a negative effect on the ability to perform office tasks but will also affect the well-being of the occupants [45]. An experimental study [46] showed that thermal discomfort negatively affects the well-being and performance of office occupants. A comparative analysis of thermal comfort for two separate offices having two different outdoor climatic conditions showed that the difference in outdoor climate has a substantial impact on the thermal sensation of the occupants [47]. Another study [48] emphasized the need for including qualitative factors that can influence the thermal comfort of office occupants. Further studies also support the fact that improvements in thermal comfort conditions in offices can raise work productivity to a significant extent [43], [49], [50]. These thermal conditions are controlled by six factors classified as personal and environmental factors, as shown in figure 3.

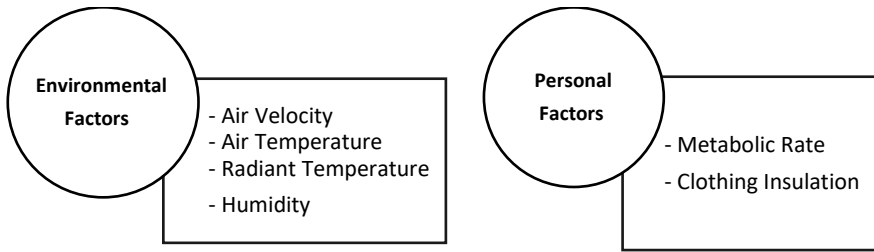


Figure 3: Factors affecting thermal comfort (adapted from [51])

The thermoregulatory system in the human body works on a balance between the rate of metabolic heat produced and losses due to exchange with the environment [52]. This heat balance is established by the six variables mentioned above in order to maintain the occupant's thermal comfort. The heat is exchanged with the surrounding environment by means of convection, radiation and evaporation [53], detailed in the heat balance equation:

$$M - W = C + K + E + R + C_{res} + S + E_{res} \quad (1)$$

here W and M are the mechanical power and metabolic rate, respectively; K , C , E and R are conductive, convective, evaporative and radiative heat losses from skin respectively, whereas E_{res} and C_{res} are evaporative and convective heat loss from respiration respectively [54]. S is the heat storage rate of the human body. For stationary conditions like in offices, W and S are kept zero [55] and conductive heat loss (K) is also neglected due to negligible differences between chair surface and the body. Hence, the total heat loss from the human body is equal to the total metabolic heat production that is mainly from the skin and respiration. In addition to this,

convective and evaporative heat losses from respiration have a very small effect on the surrounding environment [56], [57], leading up to 75% of body heat loss to the surrounding environment by convection and radiation [58]. This heat loss (through convection and radiation) is sensible heat loss (Q) given as:

$$Q = C + R \quad (2)$$

$$Q = h_c (T_s - T_a) f_{conv} + \varepsilon_h \varepsilon_w \sigma (T_s^4 - T_r^4) F_{w-h} f_{rad} \quad (3)$$

In equation 2, C and R are convective and radiative heat transfer (loss) measured in W/m^2 , respectively. Equation 3 involves T_s , T_a and T_r as surface temperature of a thermal mannequin, surrounding air temperature and mean radiant temperature (mrt) of a mannequin, respectively. Whereas f_{conv} shows convective heat transfer area ratio of a thermal mannequin and F_{w-h} as an angle factor between a thermal mannequin and room configuration [59]. According to Fanger [60], [61], the natural convective heat transfer coefficient can be calculated using equation 4.

$$h_c = 2.38(t_{sk} - t_{am})^{0.25} \quad (4)$$

Where t_{sk} is the human body skin temperature; and t_{am} is the ambient air temperature [61]. During forced convection, the convective heat transfer coefficient (h_c) depends on air velocity and can be evaluated by the regression equations 5 and 6 given as:

$$h_c = 4.008 + 6.592 V_a^{1.715} \quad (5)$$

$$h_c = 2.874 + 7.427 V_a^{1.345} \quad (6)$$

Where V_a is the air velocity. Equations 5 and 6 are for the h_c for the whole body (nude seated position downward flow) in forced convection at 20°C and 26°C air temperatures, respectively. For 0.3 m/s and higher air velocities, the regression equations 5 and 6 can be used regardless of the surrounding air temperature [59]. Whereas for lower air velocities, natural convection plays a significant role. The temperature difference created between skin and surrounding air causes the buoyant forces to contribute to mixed or natural convection around the body [61]. These buoyant forces for mixed can be evaluated by equation 7.

$$\frac{Gr}{Re^2} = g\beta\Delta TL/\nu^2 \quad (7)$$

Here L is characteristic liner dimension, β is the coefficient of thermal expansion, ν is kinematic viscosity, ΔT is the temperature difference of the human body and surroundings. The ratio of Grashof (Gr) and Reynolds (Re) numbers is close to unity shows that there are strong buoyant forces contributing to the mixed flow near the body. Hence the heat transfer takes place from the exposed skin and through the clothing [62].

2.2.1. PERSONAL FACTORS

Clothing acts against body heat gains or losses by providing insulation (with different properties) [63]. In studies, the personal factors i.e., metabolic rates and clothing insulation are used according to the Standards recommended in ASHRAE 55 and ISO 7730, under steady state conditions [38], [39], [64]. Clothing insulation is often expressed in clothing factor (clo) units and varies from 0.5 clo in summer to 0.7 clo in winter [65]. In our study, clothing insulation is kept constant and the other personal factor i.e., metabolic rate, is varied to develop the need for creating variable air velocity fields.

The metabolic rate depends on the human activity and personal characteristics of any individual [66]. A higher metabolic rate results in a higher core body temperature and therefore skin temperature needs to be decreased to maintain body heat balance and so thermal comfort. The ISO 7730 Standard and different studies suggest 1.2 met (65 W/m^2) as the reference metabolic rate for a seated office occupant [67]–[69] and 1.6 met for occupants that are involved in light activities while standing [70]. The ASHRAE 55 Standard also restricts sedentary office activities with metabolic rates ranging from 1.0 met to 1.3 met and it encourages the raising of air velocities in warm conditions [39]. However, in real office environments, the metabolic rate of occupants may vary due to individual differences in working behavior, like the current trend in working at standing desks, which increases metabolic rates up to 1.7 met [71]–[73]. This metabolic rate, other than activity level, is dependent on body fat and muscle mass, body size and health, hormonal and genetic factors, and gender and age [74], [75]. Another study classified metabolic rates in buildings into individual characteristics (depending on age, mass, gender etc.) and activity level (office, workout, etc.) [76]. Studies have indicated that gender differences can result in different metabolic rates, with females having up to 30% lower metabolic rates than males [77]. Human bodies that are the same size may contain different amounts of fat and this can affect how hot or cold one would feel in comparison to others. A body with a greater amount of fat will feel warmer. Similarly, younger people can feel less cold than older people because the underlying fat layer beneath the skin (that conserves heat) begins to thin with age [78]. When walking, the body has a metabolic rate of 3.0 met and takes up to twenty minutes to reach normal heat balance at a resting position i.e., 1.0 met [79]. In addition to this, ethnicity can also play a role in having different metabolic rates with Chinese people showing metabolic rates up to 15% less than the western population [80]. Among the six factors mentioned in figure 3, the probability of inaccuracy remains greatest in knowing the real metabolic rate of the human body [76]. Furthermore, different studies maintained thermal comfort for occupants in offices by raising air velocities for the areas in an office where occupant's activities or metabolic rates can be higher [33], [34], [81].

In this thesis, we used a combination of metabolic rates ranging from a sedentary level of 1.2 met to an elevated level of 1.6 met within an office environment. As the large

amount of heat produced by the body through different metabolic processes is balanced by heat loss to the surrounding environment [63], a balance of environmental factors are required in accelerating this process. Environmental factors are controllable and HVAC designers utilize their efforts to satisfy occupants by providing acceptable indoor climate solutions.

2.2.2. ENVIRONMENTAL FACTORS

The literature has highlighted the role of air temperature in maintaining the feeling of thermal comfort and affecting occupants' performance. Li et al. [46] raised the temperature of the room up to 33°C and found that even with the feeling of being thermally neutral, the performance of the occupants declined. In a study, the influence of temperature on thermal comfort for a wide range of temperatures showed that temperatures between 22-26°C were found to be appropriate for work performance while temperatures above this range negatively influences performance [82]. Another study suggested 26 °C was the most comfortable temperature for employees working in offices with installed ceiling fans [83]. In an experimental study, Niemela et al. [84] also showed that employee performance is reduced at room temperatures above 25°C. Some studies even suggest that an increase in temperature not only negatively affects performance but can also cause sick building syndrome (SBS), headaches and fatigue [85], [86]. Therefore, optimum temperature plays a significant role in maintaining thermal comfort and improved work performance [87], [88].

In addition to air temperature, radiant temperature is another aspect that can influence thermal comfort. Mean Radiant Temperature (MRT) is defined as *the uniform temperature of an imaginary enclosure in which the radiant heat transfer from the human body is equal to the radiant heat transfer in the actual non-uniform enclosure* [67], [89]. The MRT is considered to be a complex input parameter when evaluating thermal comfort, therefore MRT is kept equal to ambient air temperature in several thermal comfort studies [90]–[93]. Humidity control is another crucial environmental parameter that is required for maintaining the surrounding environment within a comfortable temperature range [54], [89]. Variations in relative humidity (RH) between 30-85% can vary room temperature by 0.61°C [95]. Another study reported that a 10% decrease in RH results in a 0.3°C increase in room temperature [96]. A comparative study [97] conducted for fixed temperatures i.e., 25°C and 28°C in a climate chamber varied RH from 20% to 90% to observe changes in human thermal comfort. Results showed that a RH above 70% was considered high and may cause discomfort for occupants. Some studies have also reported that humidity has little effect on thermal comfort when other environmental factors are within the comfort range [95], [98], [99]. However, a higher RH becomes more relevant when there is an increase in room air temperature and human metabolic rate [100]–[102].

2.2.3. THERMAL COMFORT EVALUATION CRITERIA

According to ISO 7730 and ASHRAE 55 Standards [38], [39], [64], Predicted Mean Vote (PMV) and Predicted Percentage Dissatisfied (PPD) indices predicts the overall thermal comfort of occupants by using both personal and environmental factors as input parameters. Based on the heat balance of the human body, the PMV index predicts the mean value of votes of a group of occupants on a seven-point thermal sensation scale, as shown in figure 4 [39], [64].

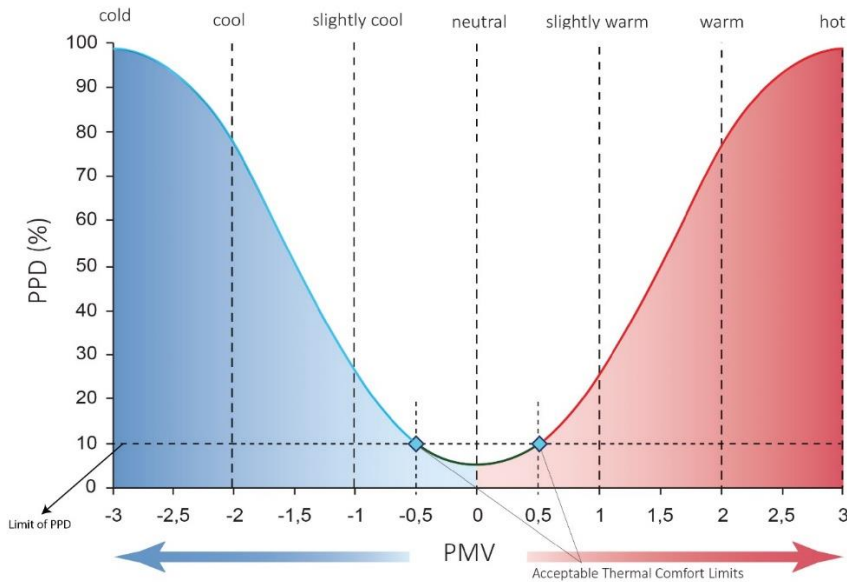


Figure 4: PMV-PPD thermal comfort indices (adapted from [38], [39], [64])

As highlighted in figure 4, the acceptable thermal comfort limit for an occupant feeling slightly warm, slightly cool or neutral is between -0.5 to $+0.5$ and PPD is less than 10% [38]. The scale on the horizontal axis in figure 4 shows 0 as the feeling of being thermally neutral, ± 1 is considered slightly warm or cool and ± 3 is considered to be hot or cold. On the Y-axis, PPD represents the percentage of thermally dissatisfied people.

In addition to this, ISO 7730 Standard [64] also focuses on local thermal comfort caused by radiant asymmetry, vertical temperature differences and draught. Radiant asymmetry causes discomfort for occupants through cold walls/windows or warm ceilings. Different studies have suggested radiant asymmetry to be a cause of thermal discomfort [104]–[106]. Even high radiant temperatures can cause local temperature differences in the different body segments [107]. One study suggested using double-

glazing to minimize thermal discomfort due to radiant asymmetry [108], while another study used diffused ceiling ventilation to minimize the radiant asymmetry caused by a warm ceiling [109]. ISO 7730 Standard for thermal asymmetry states that occupants will feel discomfort with a radiant asymmetry greater than 14°C for chilled ceilings and greater than 5°C for warm ceilings [67]. An experimental study found that radiant asymmetry neither has any influence on operative temperatures nor on occupant's behavior to the surrounding environment to a significant extent [65], but some recent studies have contradicted this claim [110]–[112]. In this thesis, experiments and simulations are conducted under adiabatic wall conditions and discomfort caused by radiant asymmetry is not considered.

The high values of vertical temperature difference (VTD) between ankles and head ($\Delta t_{a,v}$) leads to thermal discomfort and is avoided [113]. The vertical temperature difference is evaluated using regression analysis, as shown in equation 8.

$$PD = 100 / 1 + \exp(5.76 - 0.856 \cdot \Delta t_{a,v}) \quad (8)$$

where $\Delta t_{a,v} < 8^\circ\text{C}$

Equation 8 shows that Percentage dissatisfied (PD) is the function of vertical temperature difference (VTD). Studies have recommended that $\Delta t_{a,v}$ greater than 3°C can cause discomfort for the occupants involved with sedentary activities [109], [114]. Another factor that may cause thermal discomfort is draught which is defined as unwanted local cooling of the human body [115] and is usually caused by high air velocity or air movement in the occupied zone. According to the ISO 7730 Standard [64], [67], the draught model is used to evaluate draught risk that involves group of people dissatisfied with draught. The Draught Rate (DR) is calculated as:

$$DR = (34 - t_a) (v_a - 0.05) 0.62 (0.37 \cdot v_a \cdot Tu + 3.14) \quad (9)$$

The Draught Rate (%) is calculated using the above equation where DR is the function of local mean air velocity (v_a), the local air temperature (t_a) and local turbulence intensity (Tu). This model is applied for a temperature band between 20°C to 26°C, a mean air velocity range up to 0.5 m/s and turbulence intensity between 10% and 60% [64]. Appendix A shows the DR calculations from the actual experimental data for occupants with 1.2 met.

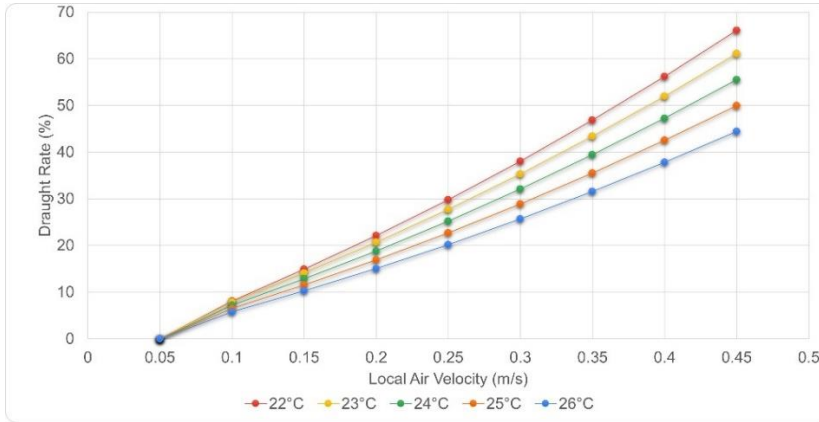


Figure 5: DR for different local air velocities and local air temperatures (illustrated using equation 9)

Figure 5 shows that DR decreases with an increase in temperature at the same local air velocities (keeping T_u constant i.e., 40%). Figure 5 is illustrated using equation 9, which shows that air velocities of up to 0.25 m/s with a temperature range up to 26°C keeps the DR within an acceptable limit i.e., 20%. Further increases in temperature and air velocities leads to $DR > 20\%$ and is considered unacceptable.

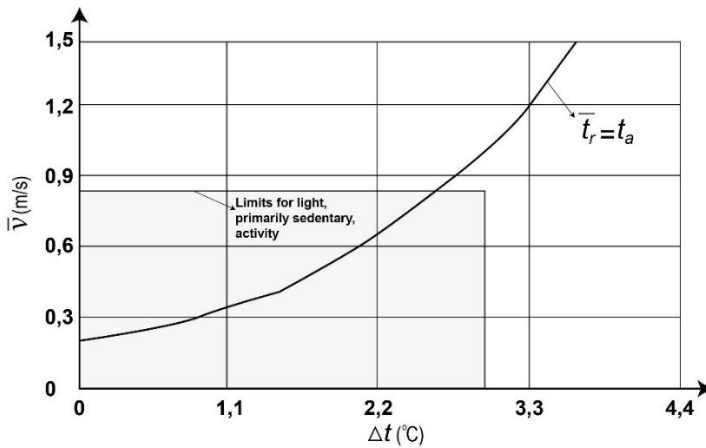


Figure 6: Air velocities required for a temperature rise above 26°C (adapted from [64], [67])

Figure 6 shows air velocity limits for a wide-ranging temperature according to EN ISO 7730 Standard [64]. These higher air velocities in Figure 6 are recommended typically for sedentary activities (with a metabolic rate 1.2 met) and summer clothing

(0.5 clo). In figure 6, the Δt shows a temperature rise above 26°C and the curve is valid for a condition when air temperature (t_a) is equal to mean radiant temperature (t_r). In the case of light or primarily sedentary activity, the curve in figure 6 indicates the limit of $\Delta t < 3^\circ\text{C}$ and $v < 0.82$ m/s.

Thermal Comfort with Elevated Air Velocities

Air velocity is one of the most crucial controllable parameters that influences the convective heat transfer between the human body and the surrounding environment [116]. The air velocities are kept high for industrial applications while for office or residential applications, the air velocities are set according to occupant's changing needs. These needs vary with respect to change in clothing, metabolism or activity and surrounding temperature affecting personal thermal comfort.

For the past two decades, the DR model has been applied to avoid local thermal discomfort by lowering air velocities and air movement according to set standards [117]. These strict air velocity criteria prevented designers being able to maintain thermal comfort and reduce energy use simultaneously at warm room temperatures and for occupants with higher metabolic rates. Among building designers, the energy consumption perspective has changed and has resulted in an increase in the temperature setpoint to certain degrees [118]. This has left users to increase the room air velocities to compensate for warm room temperatures. This shift of negative discomfort caused by draught to the positive benefits of achieving acceptable thermal comfort by raising air velocities in warm indoor climates has been achieved in various studies as shown in table 1.

Table 1: Studies with elevated air temperatures and air velocities

References	Air velocity limit	Room temperature limit
[119]	1 m/s	29°C
[120]	1 m/s	28°C
[121]	1.4 m/s	29°C
[122]–[124]	1 – 1.5 m/s	Up to 31°C
[125]	1.6 m/s	31°C

In addition to table 1, some experimental studies have shown that occupants exposed to room temperatures up to 32°C were also made thermally satisfied by an increase in air velocities [126]–[128]. Some experimental studies even reported thermal comfort satisfaction up to 90-100%, even after increased air velocities for set-point room temperatures that were 26°C or higher [34], [129]. These elevated air velocities are affective towards heat loss to the environment and to offset the warmth created by higher air temperatures. The increase in air movements with a raise in set-point temperature also seems a promising solution for cutting energy use by 30% [130], [131]. In addition to this, the limitations faced by HVAC designers and the scientific

community to follow the set criteria for controlling room air velocities are also stretched to 1.2 m/s by ASHRAE 55 and EN 15251 revised Standards and related studies [132]–[135]. Even ISO 7730 Standards recommend increased air velocities (shown in Figure 6) with an increase in room air and radiant temperatures [38], [136]. In addition, ISO 7730 Standards also recommend increased air velocities for different activity levels (higher metabolic rates) to keep PMV-PPD values within acceptable range. Therefore, minimizing the cooling effect in the room and elevating air velocities can result in significant energy reduction while keeping the occupants thermally satisfied.

From the above literature, it is clear that an increase in air velocities for higher air temperatures is encouraged to save energy and provide thermal comfort simultaneously. The benefits gained by increasing air velocities depends on an increase in temperature in order to save energy, clothing habits, and higher metabolic rates. There are very few studies that have addressed higher air velocities for higher metabolic rates during cooling conditions. For temperatures below 26°C, the DR model limits itself to mainly sedentary activities (metabolic rates of 1.2 met). As the metabolic rate depends not only on a singular factor, elevating air velocities for specific zones at a normal temperature is required to make occupants feel thermally satisfied. In this thesis, we assumed metabolic rates up to 1.6 met (in the specific zones) to establish high air velocity zones in a shared office space to thermally satisfy all occupants. The acceptable PMV range (between -0.5 to +0.5) is achieved by raising air velocities for regions in the office space with occupants that have a higher metabolic rate.

CHAPTER 3. BUILDING HVAC SYSTEMS

These days, buildings are designed for the occupant well-being and to meet their thermal comfort needs [137]. These fundamental needs are met by the control of the indoor climate in buildings with proper ventilation through natural, mechanical, or hybrid (mixed) means. HVAC systems in these buildings are further classified, based on different technologies and design strategies adopted to improve indoor climate, which are described below.

3.1. NATURAL VENTILATION SYSTEMS

The concept of natural ventilation dates back centuries when buildings were designed with large openings in the envelope in order to generate pressure created by the action of wind and buoyancy [138], [139]. The openings in buildings with natural ventilation systems are purposely designed to have natural ventilation across the envelope. These openings can be air vents, windows, doors and chimneys that allow fresh air to move through buildings. The difference in temperature or humidity results in differences in the pressure of wind, causing the recirculation of fresh air throughout the building [140]. The air in the building is raised by the buoyancy effect, whereas momentum is created by inlet and outlet openings. In buildings, natural ventilation can be classified into three types depending on the patterns of the supply and outlet paths as shown in figure 7.

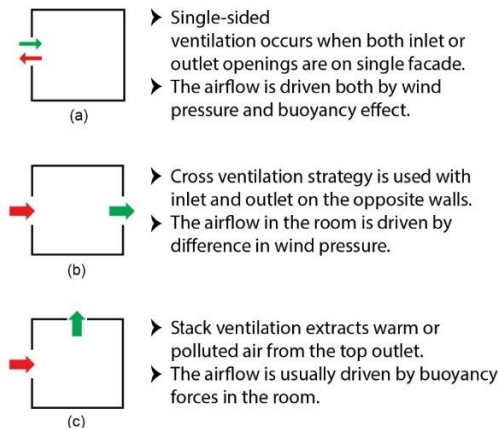


Figure 7: Classification of Natural Ventilation (a) Single-Sided Ventilation (b) Cross Ventilation (c) Stack Ventilation (adapted from [141])

These classifications are applied depending on the location and design of the building in having effective natural ventilation. Some studies have shown that cross-ventilation is considered more effective than stack or single-sided ventilation systems in providing maximum thermal comfort to occupants [141]–[143]. All three types are driven by free forces i.e., wind and buoyancy. These forces are generated by solar or internal heat gains. The natural ventilation in any building can be affected by the size of the openings in facades, wind conditions and internal temperature variations. The size of openings plays a significant role in designing a natural ventilation system to keep the room temperature moderate. A review study has suggested that outlet opening size is the most significant parameter that influences indoor climate conditions during natural ventilation [144]. In addition to this, the positioning and size of the openings (with respect to one other) can also enhance or reduce air velocities in a naturally ventilated room [142], [145]. Different inlet configurations along with their positioning help to keep the indoor environmental conditions comfortable for the building occupants [146], [147]. Whereas different strategies are required to manage internal and avoid external heat gains to make natural ventilation more effective. These heat gains are the source of discomfort and increase the dependency on mechanical ventilation systems. The external heat absorption from the environment is avoided through passive techniques. These passive techniques include providing shading elements and bringing changes to building orientation and architectural plans [148], [149]. In addition to this, natural ventilation can be affected by infiltration, which is caused by an opening not purposely provided and may result in distortion in intended natural airflow patterns [150].

Overall, from the literature, it can be concluded that natural ventilation systems are effective in providing fresh air and significant cost savings. These cost savings are due to their minimal (or none) reliance on electro-mechanical ventilation components for a sustainable indoor climate [151]. However, these naturally ventilated systems are more difficult and complex than designing mechanical ventilation systems, as they may have architectural consequences leading to high infrastructure costs. Furthermore, in non-domestic buildings like offices, natural ventilation is not widely common due to the need for higher ventilation rates to compensate for higher internal heat gains. The ventilation rate (V) required to remove heat from the occupied zone is shown in equation 10.

$$V = \frac{H}{C_p} \cdot \rho \cdot (T_i - T_o) \quad (10)$$

Here,

V = ventilation rate, m³/hr.

H = heat generation inside the space, W

c_p = specific heat capacity of air, J/kg-K

ρ = density of air, kg/m³

T_i = indoor air temperature, K

T_o = outdoor air temperature, K

Hence, the ventilation rate in buildings needs to be sufficiently high to balance the heat generation inside the space [152]. Therefore, building designers must design natural ventilation based on ventilation needs linked with the space. Furthermore, natural ventilation systems are not allowable in all climates due to lack of heat recovery and it becomes necessary to have some form of passive or active systems to fulfil ventilation needs [153].

3.2. HYBRID (MIXED MODE) VENTILATION SYSTEM

The increase in the impact of climate change on the environment has resulted in the development of hybrid ventilation strategies to provide sustainable indoor climate solutions at the expense of minimum energy use [154], [155]. These hybrid ventilation techniques involve the integration of passive or natural ventilation with mechanical ventilation systems. The driving forces in these systems are natural and mechanical, but supplementary mechanical assistance is required in hybrid ventilation in case of extreme conditions. Furthermore, hybrid ventilation systems are personalized according to the climate or season [156].

The hybrid system includes natural ventilation supply openings, fans (to assist local ventilation), heaters and air-conditioning (AC) units. These components work according to the type of requirement (according to seasons) or ventilation needs. Studies have shown that these systems provide satisfactory levels of thermal comfort with significant energy savings [157]–[159]. These ventilation systems take advantage of both natural and mechanical ventilation systems, resulting in a significant decrease in operational costs [160]. The IAQ and better thermal comfort requirements are fulfilled in hybrid systems when compared to natural ventilation due to supplement cooling or heating. The mechanical systems involved in hybrid systems can be of different types like constant air volume (CAV) and variable air volume (VAV), as described below.

3.3. MECHANICAL VENTILATION SYSTEMS

In the past, natural and hybrid ventilation remained a preferred choice for residential buildings due to reduced investment costs. While in commercial buildings, the need for mechanical ventilation is enhanced due to the larger need for maintaining IAQ and thermal comfort for occupants [161]. Since the latter half of the 20th century,

manufacturers and researchers have worked on building HVAC systems that can operate by providing a healthy and comfortable indoor climate at the expense of reduced energy and costs. In mechanical ventilation systems, different mechanical units operate to fulfil the ventilation needs in any closed space without any natural means [162]. Unlike natural or hybrid ventilation systems, mechanical ventilation systems guarantee proper air exchange rates and heat recovery in an energy-efficient manner [163]. According to the IEA, the growth in global demand of electricity for space cooling in buildings is expected to triple by 2050, if tangible steps are not taken to address excessive energy usage (see Figure 8) [164]. Hence, more energy efficient and sustainable HVAC technologies are required to overcome this growing demand.

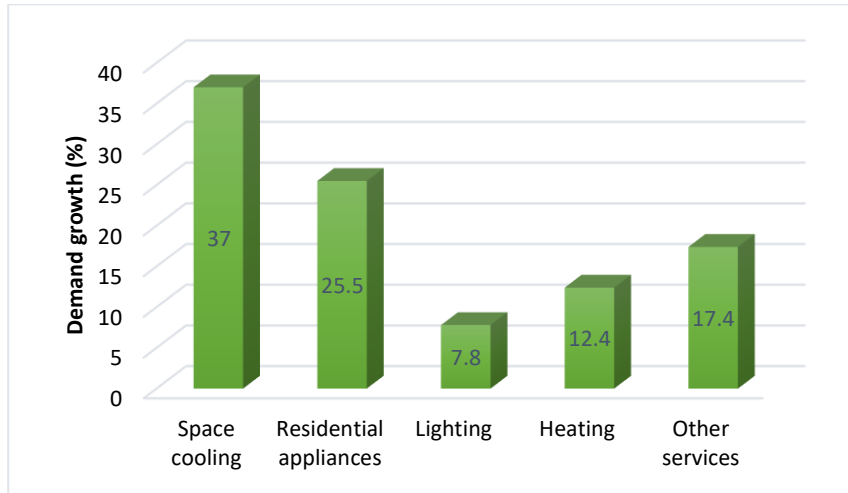


Figure 8: Share of electricity demand growth till 2050 (adapted from [164])

Mechanical HVAC systems in buildings are categorized according to their function, placement and working mechanisms. These adoptions in types are usually recommended by HVAC designers based on building design, cost and energy savings [165], [166]. De-centralized HVAC systems are designed to provide separate air-conditioning units to each zone of a building. Whereas centralized HVAC systems are mostly used in buildings due to their advantage in saving more energy (compared to de-centralized) because of their built-in design ability to cool multiple zones together using a single unit [165]. The air in a centralized system is cooled in one unit and distributed to different zones through the duct network. The fluid transport energy throughout the building facade can be air, water or both. These systems are further classified based on the type of fluid [167]. All-air systems involve only air as a medium for the transport of thermal heat. CAV and VAV systems are typical examples of such systems that provide constant or variable airflows from the Air Handling Unit (AHU). In CAV, the supply air temperature is varied to fulfil the thermal load requirement of the space. While VAV normally involves a constant

supply of air temperatures. A study has shown that a VAV system is more energy efficient and fulfilling in balancing airflows in comparison to CAV systems [168]. Designed for instantaneous load conditions, VAV systems save energy due to reduced fan power and less cooling (or heating) of the primary air [169], [170]. On the other hand, all-water systems involve water as the transfer media using cooling or heating coils. Passive Chilled Beams (PCBs) [171] are examples of such systems that include such coils where natural convection is used to circulate room air. Air-water systems include both air and water as heat transfer media to fulfil cooling or heating demand in buildings. These air-water systems have the potential to save energy and provide better room air circulation through induction units. Active Chilled Beams (ACBs) or Induction units use both air and water to fulfil both sensible cooling and heating demand of the space. The supply air patterns into the space have a substantial impact on occupants' thermal comfort, IAQ and system performance [172]. On the basis of delivering the supply air in the room for providing satisfactory thermal comfort and a healthy environment, HVAC systems are also categorized as displacement and mixing ventilation, as described in detail below.

3.4. DISPLACEMENT VENTILATION

Displacement Ventilation¹ (DV) involves supplying fresh air from or close to the floor surface at low velocity and temperature (lower than the room) [173]. The exhaust outlet is on the ceiling, which allows warm air to be discharged from the ceiling. The temperature in the occupied zone of the room is maintained by natural convection with buoyant forces driving the warm air (from the heat sources) towards the ceiling [174]. The momentum of the air is mainly controlled by thermal plumes instead of supply inlets. In displacement ventilation, high air velocities in the occupied zone would result in high draughts and low air velocities would lead to a higher temperature gradient causing more thermal discomfort compared to mixing ventilation systems [169], [175], [176].

Furthermore, the ceiling height for DV system should be above three meters, therefore its applications are mostly observed where high ventilation is required, like in industry, conference rooms and airports, etc. [177]. The literature has shown that DV systems provide better IAQ than mixing ventilation systems due to the removal of contaminated air by buoyancy from the breathing zone to the ceiling [178]–[180]. In addition to this, the low supply of air velocity and air temperature makes displacement ventilation energy efficient. However, due to design complexities and construction costs of displacement ventilation, mixing ventilation system is mostly used in the commercial sector.

¹ This thesis deals with ventilation techniques used by mixing ventilation, therefore mixing ventilation is more described in detail.

3.5. MIXING VENTILATION

Mixing ventilation (MV) systems provide fresh air into the room through the ceiling or wall outside the occupied region i.e., 1.8 m above the floor [181]. The location of supply inlets outside the occupied zone allows air jets to get completely mixed (if designed properly) with the room air. This adequate air mixing helps to maintain room air temperature uniformity in the room. The uniform temperature and air velocity distribution allows building occupants to feel more comfortable and minimize unwanted draughts. As the buoyant forces of any heat source is increased with the increase of height from the floor [163], [182], cold fresh air gets mixed with warmer room air to reach a comfortable temperature in the occupied zone. The excessive heat is taken out from the occupied zone through induction and out of the room through extract outlets in the room.

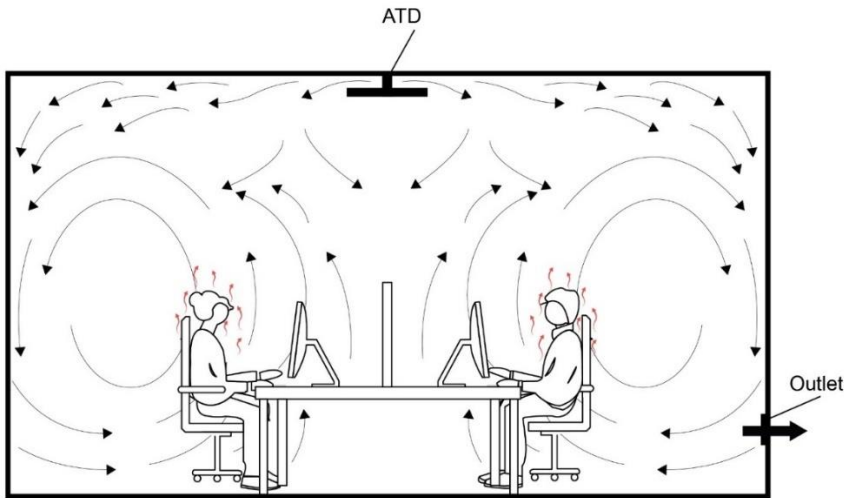


Figure 9: Illustration of mixing ventilation (adapted from [183])

Figure 9 shows the concept of mixing ventilation where airflow from the ceiling ATD flows down along the walls to the occupied zone. The airflow in the room gets totally mixed after mixing with warm room air and maintains thermal uniformity.

3.5.1. SUPPLY INLETS AND ROOM AIR DISTRIBUTION IN MIXING VENTILATION

Mixing ventilation (MV) involve supply air devices mostly mounted inside or on the walls or ceilings. Most of the supply devices include ceiling-mounted diffusers, which often provide both heating and cooling. A study has shown that the selection of supply

devices can affect air distribution patterns and air change effectiveness inside the space [184]. However, the locations of outlets or exhausts in mixing ventilation plays a minimal part in room air distribution patterns [185]. Lee et al. [186] experimentally proved that supply air devices on the ceiling, in the form of diffusers, are more efficient in room air distribution than wall supply devices. Furthermore, furniture configuration can also impact the airflow distribution inside the room [187]. The momentum in the room air with a mixing ventilation system is generated by supply air jets and buoyant forces from the heat sources. Whereas the supply air jets from the supply opening generates the maximum momentum in the room for recirculation, as shown in equation 11.

$$I = \rho a u^2 \quad (11)$$

Here,

I = momentum of airflow to the space, kg.m/s

ρ = density of the air, kg/m³

a = supply area, m²

u = supply air velocity, m/s

3.5.2. ISOTHERMAL WALL JETS IN MIXING VENTILATION

The supply air jets from the ATD openings either flow as free jets or wall jets, depending on the type of ATD. The free jets flow out from the wall while wall jets flow parallel along the wall. The initial momentum in both jets is given by supply air velocities. According to the definition, free jets flow into large spaces infinitely, while wall jets get stuck along the walls (by low pressure difference with the wall) and flow into open spaces [188]. The characteristics of these jets are influenced by the boundary layers and are often studied as independent of space surroundings. However, wall jets are not influenced by downstream conditions. In the case of smaller room dimensions, recirculating room air has a substantial impact on wall jets. This recirculation causes entrainment from the room surroundings to cause turbulent mixing on both sides of the wall jets. The lower pressure (P_1) is created on the space above the jet along the surface, causing the induced jets to push along the surface with higher pressure (P_2), as shown in figure 10. This deflection caused by lower pressure above the supply is known as Coanda effect [189]. The Coanda effect is also achieved if the supply opening is fixed at the same level as the ceiling surface and $P_2 > P_1$, for example in the case of ACBs.

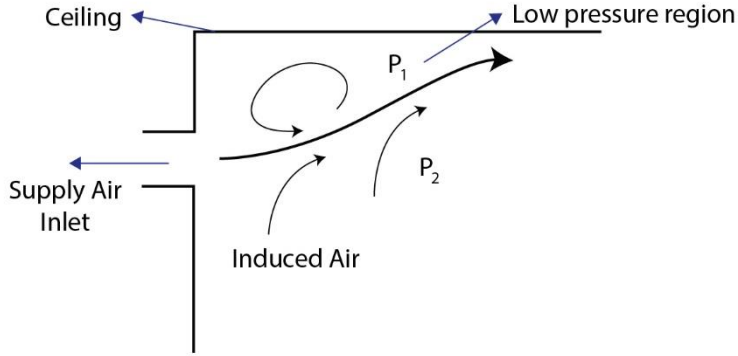


Figure 10: Deflection of free-jet due to the Coanda effect (adapted from [189])

The openings of some of the ATDs have adjustable vanes to regulate the air throw, which may cause a decrease in the momentum. The wall jet has a constant air velocity in the center along the length and initially the momentum is conserved. Equation 12 shows that air velocity has an inverse relationship with distance as air velocity decreases with an increase in distance due to wall friction [190].

$$\frac{u_x}{u_o} = \frac{K_a}{\sqrt{2}} \frac{\sqrt{a_o}}{x + x_o} \quad (12)$$

Here,

u_x = air velocity away from the opening, m/s

u_o = air velocity at the opening, m/s

a_o = supply air opening area, m²

K_a = constant in case of high turbulent flows (varies with different ATDs)

x = distance away from the opening, m

x_o = distance close to the opening, m

3.5.3. NON-ISOTHERMAL WALL JETS IN MIXING VENTILATION

For non-isothermal jets, buoyancy plays a factor in affecting the supply jets and increases the room air circulation [191]. The amount of recirculation can be expressed by Archimedes principle as:

$$Ar = \frac{g\beta l \Delta T}{u_0^2} \quad (13)$$

Where β represents the coefficient of thermal expansion, ΔT is the temperature gradient between supply and return air, and 'g' is gravity. The characteristic length from the diffuser or slot is represented as l . Ar is the Archimedes number, which is defined by the amount of air recirculation in the room. The greater the amount of room recirculation, resulting from high-temperature differences, may affect supply jets and can lead to smaller throw lengths [190], [192]. Throw length is the horizontal distance that an air jet travels at a specific air velocity. Hence the airflow at a higher Ar would result in a decrease in throw length, resulting in penetration depth before supply air jets reach the wall. Both momentum and thermal energy in jets are conserved allowing gravity to move the horizontal cool supply jets downwards. Whereas in three-dimensional spaces, the Coanda effect [189] will make the air jet lift towards the ceiling by pressure forces resulting in supply jets moving to a certain distance from the inlet opening.

Penetration depth [193] is defined as *the point of intersection between the ceiling surface to a line through maximum air velocity in the measured velocity profiles*. Some define it as the distance up to the area where the entrained flow meets the supply jet near the ceiling. The penetration depth (x_s) for three-dimensional space can be related as

$$\frac{x_s}{\sqrt{a_o}} = 0.19K_{sa}K_aAr^{-0.5} - \frac{x}{\sqrt{a_o}} \quad (14)$$

In equation 14, x_s is the distance from the diffuser opening to the point where supply air jet gets separated from the ceiling and deflects down towards the occupied zone. K_a and K_{sa} are constants are treated independent to Ar while considering measuring conditions of Reynold numbers less than 2000. K_a is dependent on ATDs while K_{sa} is dependent on external parameters like heat sources, their location, room dimensions etc. Equation 14 shows that penetration depth is proportional to $1/Ar^{0.5}$ and K_{sa} and K_a are also influencing parameters. Therefore, it can be concluded that different ATDs will have different penetration depths and a proper design procedure is required for different ATD selection for any given space.

The application of mixing ventilation depends upon the climate needs and can be used for cooling, heating or for ventilation purposes. Its applications are applied in hospital wards, offices and industry. If dimensioned properly, mixing ventilation can provide significant energy savings and enhanced thermal uniformity. Despite being the most widely used ventilation system around the world, many requirements in the design of supply ATDs are required to improve thermal comfort locally. These conventional ventilation systems, including MV and DV systems do not accommodate the individual thermal comfort requirements of building occupants [113], [194].

Chilled beams are considered one of the most efficient types of mixing ventilation system, as described below.

3.6. CHILLED BEAMS

Chilled beams are widely used HVAC systems that use mixing ventilation techniques to establish a satisfactory indoor environment for large open spaces [195]. These chilled beams are used to either heat or cool large spaces by either natural or forced convection. Chilled beams work by providing sensible cooling to the spaces through modular beams with flowing chilled water and are mounted to ceilings in a room. These technologies are used as alternatives to conventional VAV systems and are classified into passive and active types based on their operating principle [196]. In PCBs involving mainly convective cooling, ventilation is delivered by a separate AHU system. Whereas in ACBs, ventilated air goes to the beam by a central AHU unit through a duct system.

3.6.1. PASSIVE CHILLED BEAMS

Passive Chilled Beams (PCBs) work on the principle of natural convection, whereby chilled water is circulated in a fin-and-tube heat exchanger [197]. These all-water systems consist of a heat exchanger housed in a casing and are suspended from the ceiling. The supply water in a PCB system is chilled between 14°C to 16°C. The warm air from the room moves upwards to replace the chilled air surrounding the PCBs. The cool denser air falls to the occupied space to provide convective cooling without the need of a fan. The advantages of PCBs include energy savings due to less fan power and low installation costs owing to less ductwork [198]. Kim et al. [171] conducted a performance evaluation between PCBs and conventional all-air systems to show that up to 12% of the total energy can be saved compared to conventional chiller systems. Another study showed that up to 24% of energy (can vary with respect to climate zone) can be saved compared to VAV systems [199]. The cooling capacity of PCBs varies between 60-70 W/m² [170]. Due to their inability to meet high sensible cooling demands and work in heating modes, PCBs are not often used for large office buildings [202]. However, PCB systems can be integrated with DV systems to meet high sensible cooling demand and provide satisfactory IAQ [200]. Shan et al. [201] made parametric analyses after combining PCB and DV systems, which raised the cooling output from 33% to 53% of the total cooling load. Another research showed that a combined DV-PCB system can reduce temperature gradients to a significant extent to avoid thermal discomfort [200]. In terms of performance and overall energy efficiency, ACBs outclass PCBs with their high cooling capacities and induction features as described below.

3.6.2. ACTIVE CHILLED BEAMS

For the past two decades, ACBs are one of the most promising HVAC products used globally including Scandinavia, Australia, and USA [202]. After being invented in 1998, ACBs (also known as induction diffusers) gained success worldwide due to their energy-saving potential and reduction in building construction costs. Unlike PCBs, the cooling load from the spaces is removed partly by water as the working media inside the heat exchanger and partly by supplying air from a fan. The specific heat of water is higher than air and these air-water systems i.e., ACBs, require a less volume of air than conventional all-air systems. The cooling capacities of ACBs are almost twice that of PCBs [203]. ACBs are terminal units that combine hydronic cooling systems with a central ventilation system. These hydronic systems are either two-pipe or four-pipe systems, depending upon their arrangements [204]. Literature has shown that ACBs can provide significant energy savings compared to conventional HVAC systems like VAV [195], [196]. Figure 11 shows a schematic and actual image of the ACB unit.

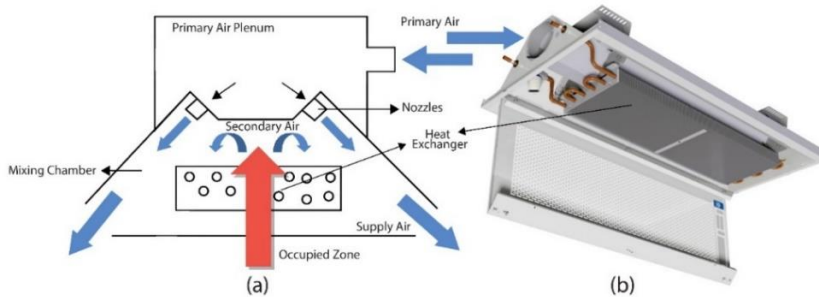


Figure 11: (a) ACB layout (b) Actual ACB unit (adapted from [170])

The chilled water at temperatures between 14-18°C is passed through the heat exchanger enclosed inside the ACB casing. This chilled water temperature is set just above the dew point temperature to avoid condensation that can be formed on the tubes of the ACB heat exchanger [170]. During the heating operation, the coil water temperature is raised to 30-45°C. The primary air from the AHU is transported to the primary air plenum where a pressure between 30-120Pa is maintained [203]. This pressurized primary air is passed through a set of nozzles and gets mixed with the induced (or secondary) air from the room. This warm room air enters the ACB by induction, after which it gets mixed with fresh (primary) air in the mixing chamber. Induction in ACB is a significant feature that improves the energy efficiency of the system. For ACBs, both internal and external induction takes place as a result of pressure differences [205]. Internal induction (within the ACB unit) takes place through a pressure difference between the warm room (secondary) air and the cooling coil. The high positive pressure created inside the primary air plenum ejects high

speed air jets from the ACB nozzles. The high velocity jets increase the dynamic pressure and build up a negative pressure inside the secondary (or mixing) chamber. Therefore, a pressure difference is established between the mixing chamber and room air, which forces warm room air to be sucked into the ACB. This warm air gets cooled down after passing through the cooling coil. This internal induction phenomenon makes the warm room air circulate in the room to the mixing chamber. In addition to this, the pressure difference between low static supply air and warm room air makes the room air move towards the supply air. This phenomenon is called external induction and it facilitates the Coanda effect. This supply air is delivered to the occupied zone using the Coanda effect, which helps with the distribution of air horizontally along the ceiling. This Coanda effect can be enhanced by a high pressure drop and decreasing the temperature difference between the supply and room air [206], [207]. A higher induction ratio (IR) means that primary air is used more effectively and the ACBs will have long life cycle costs. The efficiency of the ACB is measured by the IR, which is expressed as

$$IR = \frac{Q_1}{Q_2} \quad (15)$$

In equation 15, Q_1 is the amount of secondary air induced per second and Q_2 is the amount of primary air induced per second.

The ACB shown in the Figure 11(b) is a two-way ACB that has two inlets for the supply air. ACBs come in different shapes and designs. These modifications in designs are made to improve their efficiency and make their installation easy. Most studies on ACBs are made on 2-way ACBs, as highlighted in a literature review [170]. Experimental research has shown that 4-way ACB results in better air mixing than 1-way ACB, while 1-way ACB showed better efficiency in improving local air quality and is mostly used in patient rooms or single office spaces [207]. Nozzles with different shapes are located at the side of the cooling coil in the ACB unit. In the literature, various design recommendations are detailed to allow uniform airflow from the ACB outlet. A study revealed how a uniform supply airflow from the nozzles could be achieved by either minimizing its size or placing the board in the center of the primary plenum [208]. Studies have also highlighted that changes in the radius and length of ACB nozzles can have a significant impact on the IR [209]–[211]. Nozzles with smaller diameters and optimal lengths between 60 mm to 80 mm increases the IR by 30% [212]. The shape of the nozzles is another factor which influences the function of the nozzles. Another experimental research has shown that cross-shaped nozzle designs facilitate induction more than rectangular or circular shaped nozzles with a constant speed [211].

ACB with JetCones

JetCones were first introduced in 2008 in ACBs to provide airflow adjustments and deal with un-uniformity of supply airflow (presented in Chapter 5) [213]. These JetCones provide adjustments to static pressure and airflow distribution through linear regulation by adjustment regulators. Figure 12 shows the location of the JetCones in the ACB unit with an adjustment regulator. These regulators can be moved up and down to change the position of the JetCones.

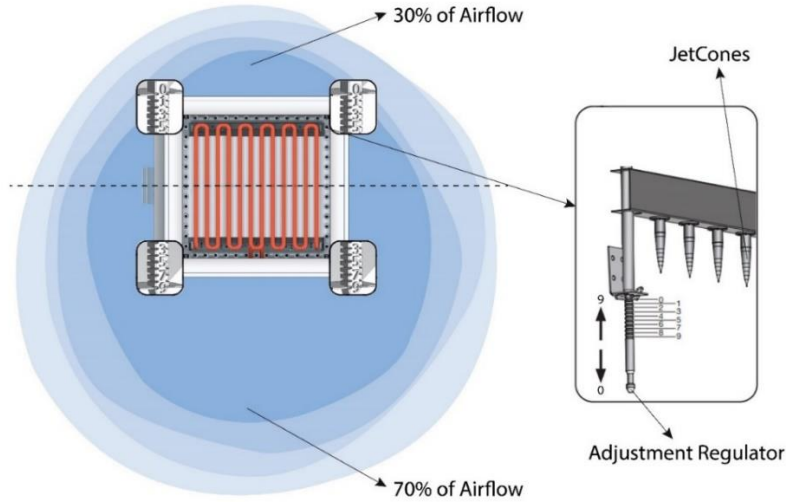


Figure 12: ACBs with a JetCone feature showing asymmetrical airflow distribution (adapted from [214], [215])

This built-in JetCone feature in ACBs provides flexibility in changing supply airflow profiles and can be applied as an alternative to conventional nozzles. The adjustment regulators are embossed with numbers from 0 to 9 to regulate the distribution profile of the supply airflow. The 9 position on the adjustment regulator means that the Jetcones are fully open and maximum airflow is discharged from the ACB opening. Therefore, for a specific fixed primary airflow (q_a) and static pressure (p_{stat}), JetCone positions can be adjusted to allow airflow according to the requirement. Figure 12 shows ACB with different JetCone positions to have uneven airflow distribution from the ACB opening. These positions are set accordingly to distribute airflow inside the ACBs with respect to specific static pressure. The throw lengths can be adjusted by changing regulator positions. The regulator position at 0 allows 30% shorter throw lengths, whereas maximum regulator positions i.e., 9, can increase throw lengths up to 70% (as shown in figure 12).

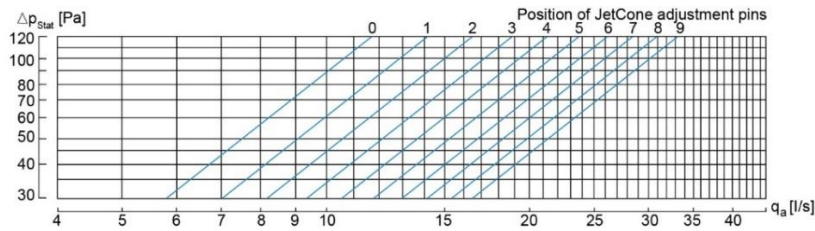


Figure 13: Airflow rate (q_a) and Static pressure (Δp_{stat}) with respect to position of the adjustment pins of ACB unit used in this study (adapted from [214])

For a fixed position of adjustment pin (regulator) and static pressure, an airflow within a specific range can come out of the JetCones. For example, for a given ACB with all JetCones at position 5, there is airflow between 12 l/s (30 Pa) to 24 l/s (120 Pa), as shown in figure 13. Asymmetric airflow distribution can also be achieved by having different adjustment regulator settings. In case of different adjustment regulator settings, the mean value of adjustment regulator positions i.e., “Position of JetCone adjustment pins” should be used in the figure 13, to find the range of airflow rate.

Integration of ACBs with Other Conventional HVAC Systems

Mechanical ventilation systems are integrated with each other to bring improvements in the system’s performance and expand applications of that technology. Table 2 shows the integration of ACBs with different HVAC technologies and control strategies.

Table 2: Integration of ACBs with conventional HVAC systems

Ref.	Integrated system with ACBs	Advantages
[216]	Radiant and convective by chilled ceiling	Improvements in IAQ, local thermal sensation and acceptability.
[217]	Personalized Ventilation system	Energy savings up to 16%, enhanced thermal comfort and IAQ.
[218]	Demand Control Ventilation system	Energy use up to 42% less than conventional VAVs and improved local thermal comfort.
[219]	Direct ground-coupled cooling system	Integration strategy with ACBs lead to an increase in the cooling capacity of ACBs and reduction in the peak load by 29%.

In addition to the advantages in table 2, other energy savings by ACBs include a reduction in fan power due to direct supply of sensible cooling to the breathing zones, an increase of chilled water temperatures from 4-7°C up to 18°C and also due to an induction feature, described in Chapter 7 [170].

Factors Affecting Room Air Distribution by ACBs

The supply air distribution from any ATD plays a significant role in room air distribution patterns and occupants' thermal comfort [220]–[222]. Wu et al. [206] conducted experiments to show that the attachment of ACB supply jets along the ceiling can be controlled by the pressure drop of the ACB unit and temperature difference between the supply and room air. A higher pressure drop results in improved air attachment to the ceiling (Coanda effect), whereas a higher temperature difference between supply and room air results in a negative impact on the Coanda effect [223]. In another laboratory study, Cao et al. [224] observed the turbulent transition of ACB supply jets along the ceiling using a Particle Image Velocimetry (PIV) technique. Results showed that at a short distance from the ACBs openings, supply jets turn completely turbulent due to the increase of volume flow rate in outer supply jets (by large vortex mixing or external induction). Further studies (shown in Table 3) pointed out more factors that may affect ACB's performance and lead to discomfort for occupants.

Table 3: Factors affecting ACB performance

Ref.	Factors	Description
[225]	Room size	- Unlike displacement ventilation systems, it is not recommended to use an ACB system for rooms above 3.5 m in height.
[207]	Colliding jets	- The spaces with multiple ACBs involve colliding jets that falls towards the floor at maximum speeds and can be a source of unwanted local cooling i.e., draught.
[226]	Heat sources	<ul style="list-style-type: none"> - Heat sources in offices usually include people, electrical equipment like computers, lights etc. The heat sources generate plumes that may affect room air distribution patterns (Chapter 6). - Heat source unsymmetrical distribution is one the most critical factors causing discomfort and affecting ACB performance. Asymmetrical heat sources affects air temperature and air velocity distribution, PMV and can increase draught risk [227].
[228], [229]	ACB arrangements	<ul style="list-style-type: none"> - Some experimental results recommend avoiding installation of ACBs parallel to windows to avoid discomfort near the ankle region. - 4-Way ACBs are recommended for establishing better uniformity in the room than 1-Way ACB designs.

Furthermore, condensation and humidity control seems the major issue in ACB systems [230]. Moisture detectors are placed near the ACB cooling coils which shut down the ACB water valve, which results in an increase in temperature in the heat exchanger [231]. Self-regulating ACBs eliminates the need for control components and maintains the balance between room air temperatures and cooling coil water temperatures [232]. Therefore, uniform room temperature (within desired temperature limits) is maintained by self-regulating ACBs by keeping the airflow rate of chilled beams constant. Despite the factors above that can lead to discomfort, ACBs have proven successful in maintaining better thermal uniformity compared to conventional VAV, CAV, FCU and UFAD systems [11], [233]. Most of the research conducted on ACBs is done to analyse the effects of cooling and ventilation [170]. For heating purposes, radiant heating panels or heaters are mostly used in combination with ACBs.

Conventional ACBs are integrated with PV systems to fulfil the requirement of individual thermal comfort [217]. These PV systems are designed to establish micro-climate zones by using different ATDs in the breathing zone, which will now be explained in the next section.

3.7. PERSONALIZED VENTILATION SYSTEMS

As described above, the main aim of conventional HVAC systems is to achieve uniform thermal conditions in a conditioned space. In Scandinavia in the late 20th century, the concept of PV system was introduced with the aim of providing satisfactory thermal comfort and IAQ locally [7]. The aim was achieved through the supply of fresh air to the breathing zone by different ATDs. In PV systems, office occupants are provided with personal control over airflow and positioning of the ATDs to maintain the desired micro-climate around them and room air temperature is set according to the recommended ASHRAE 55 or ISO 7730 Standards [38], [39]. Compared to MV and DV systems, PV systems provide better-perceived air quality and help in minimizing symptoms of Sick Building Syndrome (SBS) when compared to MV systems [234]. In addition to this, PV systems are also termed as personalized conditioning systems that are only used for maintaining individual thermal comfort according to an occupant's preference [7]. This is achieved by individually regulating airflow rates, airflow direction, or air temperatures by different ATDs [235], [236]. In PV designs, additional air distribution ATDs are added in the breathing zone to achieve localized thermal comfort by supplying more cooling through excess air. Some studies (highlighted in Chapter 4) have even used more than five ATDs for a single office workstation to provide local thermal comfort. For example, Kaczmarczyk et al. [234] and Sekhar et al. [237] conducted experimental studies using PV systems that used more than five ATDs around a single workstation to make improvements in the local thermal comfort. Different types of ATDs involved in PV systems coupled with conventional HVAC systems are shown in table 4.

Table 4: Integration of PV with conventional HVAC systems

Ref.	PV System	Integrated System	Purpose
[238]	Round Movable Panel (RMP) and Vertical Desk Grille (VDG)	Mixing and Displacement Ventilation	The proposed system involved a combination of two different types of ATDs i.e., RMP and VDG and were used in conjunction with either MV or DV to mainly improve IAQ. Greater uniform cooling was achieved using RMP as compared to VDG.
[239]	Variable Speed Fan Accompanied with a Cooling Coil	Displacement Ventilation	Overall thermal comfort and energy savings were accessed for DV coupled with PV in a case study. Higher room air temperatures up to 26°C were found to be acceptable for maintaining thermal comfort with energy savings up to 27% less than a DV system alone.
[240]	Personalized Sinusoidal Ventilation	Mixing Ventilation	IAQ and thermal comfort were evaluated in the PV-MV system at different wind frequencies and airflow rates. Optimized frequency of 0.94 Hz and an airflow rate of 7.5 L/s led to simultaneous improvements in IAQ and thermal comfort.
[241]	Intermittent Personalized Ventilation (IPV)	Mixing Ventilation	Experimental and CFD studies involved IPV coupled with MV at different operating frequencies to provide satisfactory IAQ in the breathing zone.
[242]	Ceiling Mounted PV and Individually Controlled Desk Fans	Mixing Ventilation	Energy saving potential between different PV setups integrated with MV systems was evaluated at elevated room air temperatures.

[243]	PV Nozzles	Mixing Ventilation	PV performance related to IAQ was simultaneously evaluated with a seated and walking occupant. A distance of 0.85 m was recommended between the occupants along with a larger PV nozzle size to maintain air quality.
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As shown in table 4, most of the studies on PV systems are conducted either with MV or DV ventilation systems working in the background [244]. Among them, MV systems are mostly used as an integrated system (with PV system) due to their common use in offices.

3.7.1. ENERGY SAVINGS AND SETBACK OF PV

As discussed in the section 2.2.3, energy savings can be achieved through an increase in the cooling setpoint to a higher temperature or by a reduction in airflow rate [242]. The cooling energy is saved by raising the room setpoint temperature and reducing the airflow rates in the background, while PV ATDs maintain the local thermal comfort by blowing air with a desirable air velocity and temperature. In PV systems, energy savings up to 60% are achieved through the increase of room cooling setpoint temperatures up to 30°C [7]. Zhang et al. [245] used palm and foot warmers, head ventilation and hand cooling devices to enhance work productivity by creating non-uniform thermal environments. Energy savings up to 40% were achieved by increasing the ambient temperature band to 30°C. Sekhar et al. [237] and Schiavon et al. [246] conducted an energy analysis using PV systems by raising the cooling setpoint from 24°C to 26°C to achieve 30% and 51% energy savings by reducing cooling, respectively. El-Fil et al. [247] used a ceiling mounted ventilation system along with fans mounted on a chair at different heights to develop a micro-climate to improve thermal comfort and IAQ. Compared to MV systems that were operating to provide uniform thermal conditions, there was a 17% reduction in energy use. With similar desk-chair PV configurations [248], [249], improvements in local thermal comfort were observed by raising personalized fan air velocities to 1.87 m/s with an overall room temperature setpoint raised to 26°C. In another study, an integrated MV-PV setup with an increase in room temperature to 26°C resulted in energy savings up to 27% [239]. Hence, the ventilation and thermal comfort needs at each workstation were achieved by mounting additional ATDs to provide optimal airspeeds around the human body. Energy savings are achieved in PV systems by raising room setpoint temperatures and decreasing airflow rates of the integrated system.

Despite the energy savings described in the literature above, PV systems involve several ATDs in the breathing zone near the occupant to establish a micro-climate according to the occupant's preferences. In a review study [250], improvements in the

designs of PV ATDs are recommended to enhance the system's performance. As PV systems require extra duct work and separate ATDs to reach the breathing zone [251], [252], a setback of PV systems indicated in another review was that a large number of ATDs were needed to be installed in the breathing zone, which is not favourable with an office layout [7]. Furthermore, the complexity of controlling several ATDs and the direct blow of air from each ATD on the occupant may result in discomfort. Table 4 also showed that PV systems are coupled with MV systems in many studies. Therefore, advancements in the design of ATDs in MV systems and the expansion of its applications are required to cover the functions of the PV system without the need for having multiple ATDs in the occupied zone.

3.8. PRECISION VENTILATION SYSTEM

In the above literature, it is evident that ventilation systems in buildings requires greater advancements in order to develop more efficient ATDs for improving individual thermal comfort in offices. Furthermore, controlling individual thermal comfort through cooling is desirable and can be achieved by varying critical parameters like air velocity. As explained above, mixing ventilation systems are the most commonly used ventilation systems and ACB technologies are a typical example of such systems, which are popular due to their ability to provide better thermal uniformity and energy savings. The issue of individual thermal comfort is being addressed in the literature through introducing PV techniques with various ATDs in the breathing zone (Chapter 4). Due to the drawback of the direct blow of supply air onto occupants and the use of several ATDs, a modified form of ventilation strategy was needed which could establish micro-climate zones without bringing changes to the office layout.

In open-plan offices, ventilation is generally based on the supply of cooled air and the formation of a uniform thermal indoor environment. A uniform thermal indoor environment implies that not all occupants will be satisfied with the air temperature in the room. As previously described, ACBs with JetCones can change airflow rates and direction by varying the position of the JetCones (Chapter 5). The primary airflow and supply air temperature are kept constant during the operation, while airflow is directed through the JetCones (instead of conventional nozzles). This new strategy could be utilized to build different air velocity zones in the same open-plan office space. The introduction of this novel ventilation strategy based on ACBs with JetCones can enhance the thermal comfort level of unsatisfied occupants by controlling the airflow from the ceiling. High air velocities can be provided for occupants with higher metabolic rates to improve their thermal comfort level. Unlike PV systems or integrated ACB systems (described in sections 3.6.2 and 3.7), the new precision ventilation technique can be applied to establish micro-climate zones in the same office space by providing a change in air velocities around occupants (without the need for multiple ATDs in the breathing zone) [215]. These different micro-

climate zones will be set to meet individual preferences, leading to greater satisfaction, higher efficiency and increased productivity.

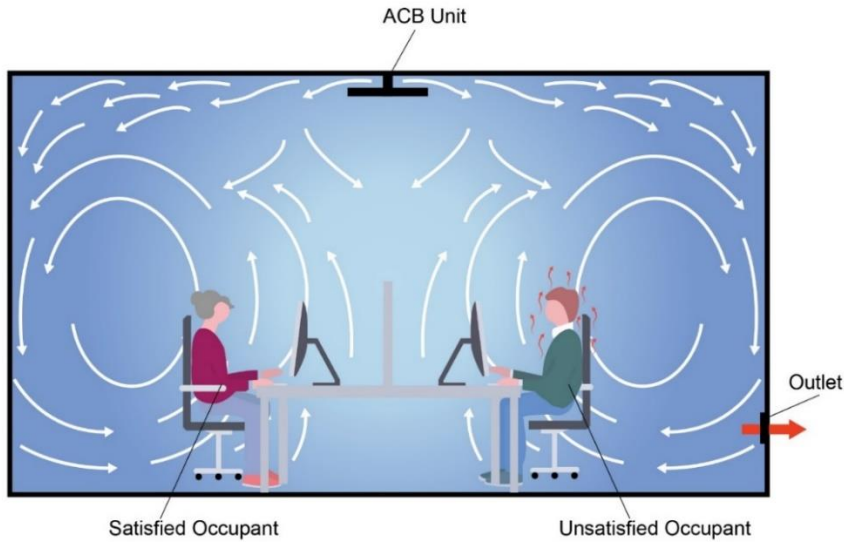


Figure 14: MV with a single ACB

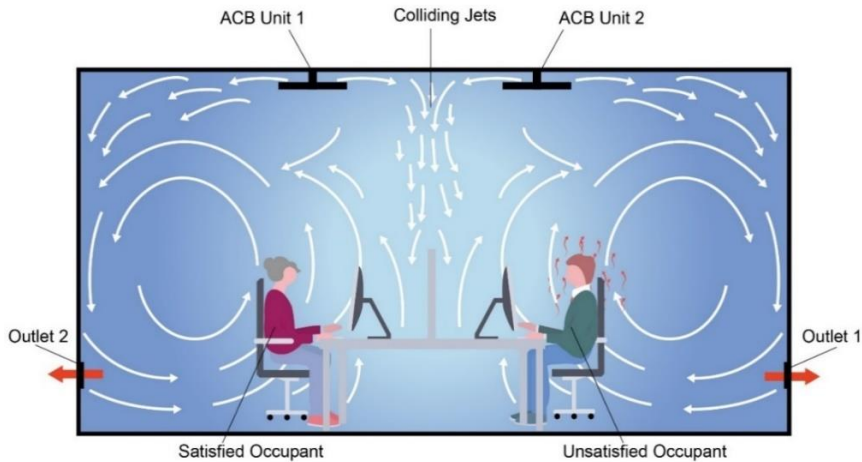


Figure 15: MV with a dual ACB system

Figures 14 and 15 shows a conventional mixing ventilation system with single and two ACBs, respectively. The occupants are exposed to a uniform thermal environment with airflow uniformly distributed in the room. The unsatisfied occupant feels warm due to different metabolic rates or clothing and there is then a need for a change in

environmental conditions surrounding them. Figure 15 shows the same office configuration with mixing ventilation with two ACBs. Here, colliding jets descend in the center of the occupied zone due to symmetric airflow distribution from both ACBs.

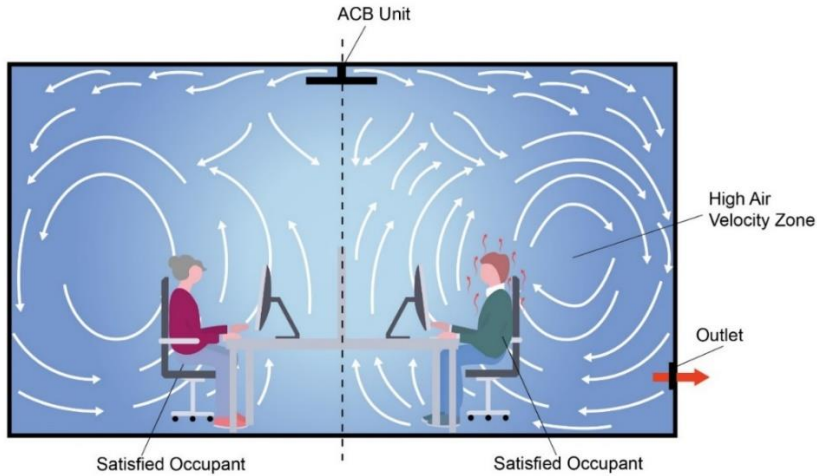


Figure 16: Precision Ventilation with a single ACB

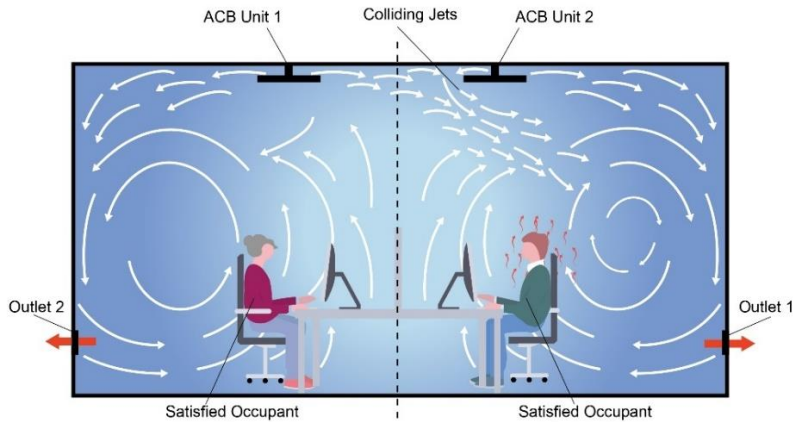


Figure 17: Precision Ventilation with a dual ACB system

Figures 16 and 17 shows modifications in JetCone positions in ACBs leading to a change in airflow distribution. In a room with one ACB, airflow can be pushed towards one part of the room and create a high air velocity zone for the occupant that is feeling warm (see Chapter 8). In a room with two ACBs, the colliding air jets can be moved from the center towards the area of the unsatisfied occupant (see Chapter

9). These high and low velocity zones are divided based on the air velocity requirement of occupants according to their metabolic rate [253]. In precision ventilation systems, the occupants with high metabolic rates can be subjected to higher air velocities while occupants with low metabolic rates are subject to low air velocities. This technique would avoid a direct blow of air to the occupant's body and may prevent any discomfort. This application of ACBs for achieving individual thermal comfort can be used in large open-plan offices with large occupant density, see figure 18.

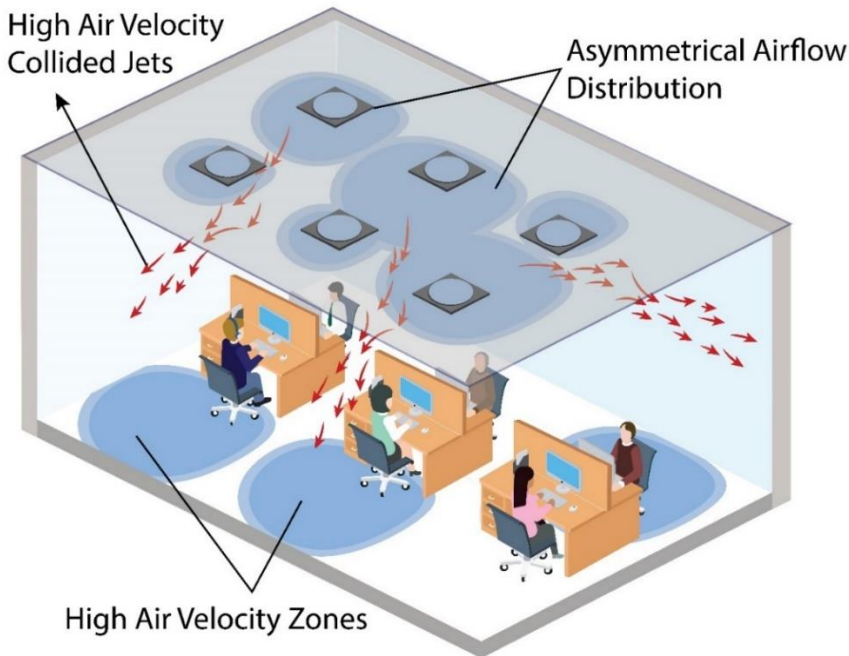


Figure 18: Precision Ventilation with multiple ACBs in an open-plan office layout

Figure 18 shows the depiction of precision ventilation for large scale open-plan offices where asymmetrical distribution of airflow at the ceiling leads to different air velocities in the occupied zone with respect to an occupant's preferences (see Chapter 10). No extra ATDs in the breathing zones are required, which could disturb office aesthetics and does not involve the direct blow of supply air to the human body. Even colliding jets are pushed towards the sitting area of the specific occupant (not directly on the occupant) to avoid discomfort.

In personalized ventilation systems, significant energy savings are achieved by raising the cooling setpoint temperatures and reducing the airflow rates (described in sections 2.2.3 and 3.7). In precision ventilation, the primary airflow from the system stays fixed and variable air velocity zones are established with a fixed supply of airflow

into the room. Therefore, temperature limits are also required to be within a specific range to simultaneously fulfil the ventilation needs of all occupants. Hence, precision ventilation systems can combine the advantages of both the MV and PV systems to achieve individual thermal comfort at the expense of reduced energy use. Energy savings are achieved with an increase in cooling setpoint temperature from 23°C to 25°C. The applications of precision ventilation are tested using CFD and experimental techniques (Chapters 8, 9 and 10) for different cases, as detailed in the next part.

PART II

RESEARCH PAPERS

Introduction to Research Papers

In this part, the collection of papers that are a part of this Ph.D. project are reprinted. The Ph.D. student is the first author in all these research papers and the work is published with Professor Alireza Afshari, Professor Peter V. Nielsen, Associate Professor Göran Hultmark and Associate Professor Samira Rahnama, as senior authors. Thanks to co-authors Dr. Klemen Rupnik from Lindab A/S and Dr. Craig Bradshaw from Oklahoma State University for our collaboration during the past three years.

The first three papers are conference papers published at International conferences. Paper I is a literature review detailing the ATDs that are used to achieve personalized thermal comfort. The review highlights the need for using a single and a more controllable ATD to simultaneously achieve individual thermal comfort for more than one office occupant. Paper II introduces a new feature in ACBs in the form of JetCones and presents an experimental solution to use more reliable JetCones instead of conventional nozzles to achieve symmetrical airflow distribution. Paper III uses the same ACB technology to observe the impact of thermal heat loads on air distribution patterns in an office space. CFD simulations and experiments through smoke tests are carried out, with and without thermal heat loads in an enclosed space in order to observe their impact on colliding jets, room air velocity and room air distribution patterns.

Paper IV involves a detailed literature review on ACBs by investigating their components and their functions, control and design strategies, energy savings and testing techniques and use of their application under office conditions. Paper V presents the concept of precision ventilation for the first time to achieve two different air velocity zones by a single ACB mounted in the ceiling. The applications of ACBs with a JetCone feature are tested in a small-scale laboratory setup for different cases. Numerical simulations are conducted to validate the laboratory results. Paper VI investigates precision ventilation for two ACBs, which involve colliding jets to establish variable air velocity zones. The PMV-PPD indices are used to evaluate the overall thermal comfort, whereas local thermal comfort evaluations were made for this advanced mixing ventilation system. This is also the first paper which established a collaboration with Dr. Craig Bradshaw from Centre of Integrated Building Systems (CIBS), School of Mechanical and Aerospace Engineering, Oklahoma State University (OKSU). A vertical temperature difference between head and ankle of less than 1.5°C is achieved in both papers. Draught rates for papers V and VI (shown in Appendix A) showed acceptance for occupants with metabolic rates of 1.2 met. Therefore, air velocities for specific zones with higher metabolic rate occupants i.e., 1.4 met and 1.6 met were raised to make occupants feel thermally satisfied. Paper VII details the application of precision ventilation for large open-plan offices and highlights the limitations of using this new ventilation strategy. The change in the precision ventilation strategy to establish a single micro-climate zone without

influencing air velocities in the other zones is also discussed in this paper. Annual energy savings up to 15% are reached by raising the cooling setpoint temperature from 23°C to 25°C for a constant primary airflow condition.

The common aim of each of these papers was to highlight the need for improvements in existing ventilation strategies and provide a novel solution to enhance individual thermal comfort in offices.

CHAPTER 4.

PAPER I

Design Strategies for Decreasing Cooling Demand and Increasing Individual Thermal Comfort in Open-Plan Offices: A Review

Haider Latif, Samira Rahn timer, Göran Hultmark, Klemen Rupnik and Alireza Afshari

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Proceedings of the 16th Conference of the International Society of Indoor Air Quality and Climate, Indoor Air 2020

DESIGN STRATEGIES FOR DECREASING COOLING DEMAND AND INCREASING INDIVIDUAL THERMAL COMFORT IN OPEN-PLAN OFFICES – A REVIEW

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SUMMARY

This study presents a state of the art review of ventilation systems that provide occupants with desired individual thermal comfort while minimizing cooling demand in offices. Conventionally, air terminal devices (ATDs) used in building ventilation systems operate to distribute air uniformly to all workstations, but these systems can not satisfy individual thermal comfort preferences. Using task ambient ventilation systems (TASs) with a variety of ATDs is an energy efficient solution which creates separate microclimate zone for each individual in office buildings. However, they cause draft problems due to high air movement. Whereas the proposed ventilation system will set up different operating criteria for ATDs to make them work in different ways through the control of supply air velocity and direction. This review paper discusses the developments in functionality, design and performance of the ventilation systems till date in providing local indoor climate while identifying the future course in design strategies.

KEYWORDS

Air terminal devices, individual thermal comfort, indoor climate, task ambient systems

1 INTRODUCTION

With the increase in urbanized density of buildings, the concept of thermal comfort has also now found a new dimension that calls for more sophisticated technologies to satisfy building occupants with the minimum energy use (Huang and Niu, 2016). According to International Energy Agency (IEA), buildings stand for 40% of total energy demand in the US and Europe and HVAC systems account for a significant proportion of this consumption (Cao et al. 2016). In workplaces, conducive and healthy environment is crucial, where perception about the indoor environments differ between individuals, as people spend up to 90% of their time indoors (Höppe, 2002). It is a widely accepted fact that a comfortable indoor environment of a building is only as a result of proper design and controls of the HVAC systems installed (Graham et al. 2008).

Design strategies adopted to provide the uniform indoor thermal environment is the source of thermal discomfort particularly for occupants sensitive to temperature deviations (Yao et al. 2007; Wang et al. 2007). Task ambient systems (TASs) are used for the past two decades to fulfil individual comfort requirements leading to a reduction in energy use by using various types of multiple ATDs and design configurations (Vesely and Zeiler, 2014). However, the

use of multiple ATDs for TASs in maintaining Indoor Air Quality (IAQ) has reported abnormal air moments i.e. draft in field tests, which have led to more unsatisfied building occupants (Melikov et al. 2002). This paper highlights the developments to date about the task ambient systems, its energy use and its impact on office occupants after proposing new design strategies to have better user control.

2 BUILDING MECHANICAL VENTILATION SYSTEMS

The main purpose of the buildings is to provide a habitable environment to its occupants through advanced ventilation systems and design techniques. For the past three decades, these ventilation systems are mainly classified on the positioning of the primary HVAC equipment and type of transport media used to transfer heat in buildings. These conventional ventilation systems with design modifications can be further classified according to the level of thermal comfort i.e. uniform or individual they provide to the office occupants.

2.1 Classification

Conventionally, HVAC systems are classified as centralized and decentralized systems according to the functions they provide to the whole building (Bhatia, 2011). Centralized HVAC systems are designed to provide conditioning to buildings as a whole single-zone, whereas decentralized or local systems provide separate conditioning to specific zones of the building. Local systems are proven to fulfill the majority of the design parameters (Lin et al. 1992) and consume less energy in comparison with the centralized systems due to their variable flow rate conditions and duct pressure controls (Liu et al. 2019). These systems also work in a flexible manner for their placement on numerous locations of any building, but ATDs used by them mostly provide uniform air distributions to the given space. Hence, both the systems alone fail to fulfill occupant's diverse comfort demands and are unable to give significant energy savings in office buildings (Kalaimani et al. 2018).

Another classification of mechanical ventilation system is based on the working fluid which uses air for thermal distribution in buildings. All air systems transport processed air from Air Handling Unit (AHU) to the conditioned area after extracting the required amount of latent and sensible heat from that space. Variable Air Volume (VAV) systems among them are usually designed to supply variable airflow (as per load requirements) at a constant air temperature. These VAV systems consume less energy due to low fan power as compared to Constant Air Volume (CAV) systems which have constant airflow at variable temperature (Okochi et al. 2016). Majority of such systems used in buildings lack personal comfort controls unless provided with multiple ATDs. The conventional ventilation systems described above are integrated with office furniture to provide individual comfort improvements and energy savings by means of different ATDs design configurations (Bauman et al. 2017).

Thermal comfort is that condition of the mind that expresses satisfaction with the thermal environment or can be stated as a neutral point by the absence of any feeling of discomfort (Djongyang et al. 2010). The widely used Predicted Mean Vote and Predicted Percentage Dissatisfied (PMV & PPD) calculations by (Fanger, 1970) for the analysis of thermal indoor climate uses six primary factors defining the thermal comfort. As discussed above, a uniform thermal environment is the source of thermal discomfort for many individuals and it is, therefore, desirable to control the ventilation individually by predicting the changes in the user's comfort. Task ambient systems (TASs) are used for the past two decades provide a comfortable environment locally with respect to the individual's needs (Melikov et al. 2004). These systems are designed differently from the conventional ventilation systems due to their varying functionalities and design techniques. TASs are mostly all air systems which are

mainly designed on the need for office occupants that show dissatisfaction with room air quality and uniform thermal comfort (Zhang et al. 2007). ATDs are a key component in TASs during the design process in distributing air to end-users and also determines the air characteristics in different micro-climate zones.

3 ROLE OF AIR TERMINAL DEVICES FOR INDIVIDUAL THERMAL COMFORT

The concept of thermal comfort has found a new dimension that calls for more updated technologies as the trend shifts from total ventilation to personalized ventilation. Conventionally for an office setup, ATDs operate in a manner to distribute air uniformly according to the standards prescribed by ASHRAE standard 55 (De Dear et al. 2002) to multiple workstations and entails the risk of unsatisfied users with room air temperature. Whereas TASs provide individual thermal comfort by using a variety of ATDs that increase energy savings and create microclimate zones for each work station in offices when designed properly. These energy savings can reach up to 40% by raising the indoor air temperature range to 18-30°C (Hoyt et al. 2015; Veselý and Zeiler, 2014).

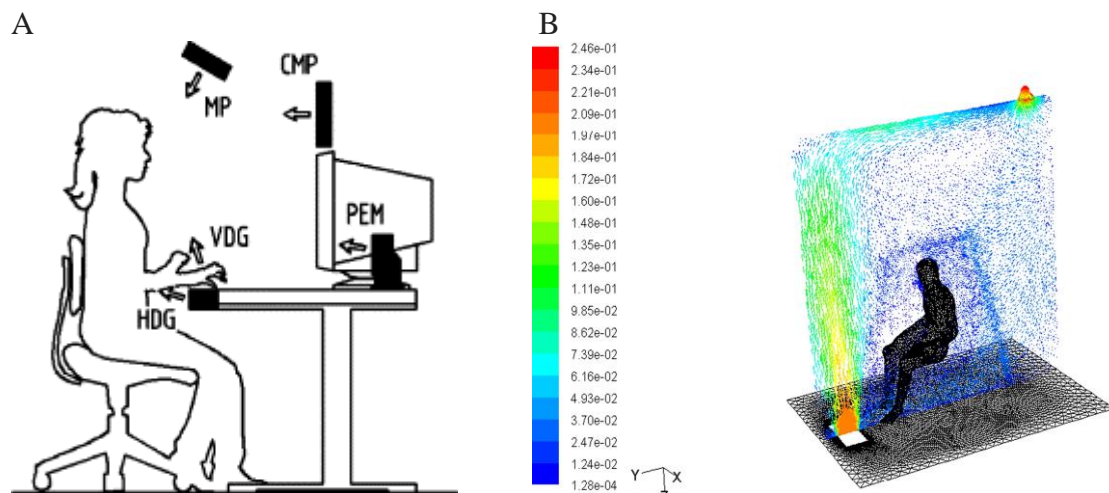


Figure 1. a) TAS with multiple ATDs (Melikov et al. 2002), b) Velocity contours around manikin (Salmanzadeh et al. 2012).

Table 1 highlights the way ATDs are used by TASs and their annual energy savings. Various subjective studies have been carried out to know the perception of IAQ using a number of ATDs. Wei Sun examined the thermal performance of ATDs on 26 body segments of the manikin at different turbulent intensities and discussed the potential draft risks at the facial region and whole body (Sun et al. 2007). Al Assaad et al. (2017) coupled Personalized Ventilation (PV) and Mixing Ventilation (MV) system to assess the velocity and temperature around occupant through computational fluid dynamics (CFD) and used optimal operating conditions to get overall comfort of 0.95 using a thermal manikin. A detailed analysis of the primary air conditioning system in conjunction with a secondary personalized ventilation system for an office set up was made to evaluate ventilation effectiveness in the breathing zones. A subjective analysis made after observing trends of thermal comfort and IAQ showed that the ratings and acceptability improve as airflow rate from PV devices is increased at the same indoor temperature (Sekhar et al. 2005). Various studies on the use of TASs for improving IAQ shows that ATD designs have a great impact on creating suitable and desirable micro-climate around the entire body (Yin and L., 2018; Foda, E. and Sirén, 2013; Zhang et al. 2010).

Table 1. Energy savings and details of ATDs used in TASs.

References	Number of ATDs (per person)	Type of Air Terminal Devices (ATDs)	Energy Savings Compared with Convectional Mixing Ventilation
(Cermak et al. 2006; Sun et al. 2012)	More than 3	CPV (Ceiling Personalised Ventilation) + Chair fans	14.87%
(Makhoul et al. 2013)	More than 2	Ceiling mounted + Desk fans	17%
(El-Fil et al. 2016)	More than 2	Ceiling ventilation + chair mounted fans	13%
(Makhoul et al. 2013)	1 per person	Ceiling based PV Ventilator (co-axial nozzles) integrated with a 4-sided slot diffuser	34%
(Kaczmarczyk et al. 2006)	More than 5	Horizontal Desk Grille + Vertical desk Grille (HDG + VDG), Moveable Panel (MP), Round Movable Panel (RMP), Headset, RMP+HDG	Not Specified
(Zhang et al. 2010)	At least 4	Heated Keyboard/Palm Warmer + Foot Warmer + Head Ventilation Device + Hand Cooling Device	Up to 44%
(Sekhar et al. 2005)	6 independent ATDs for each work- station.	Ceiling Supply with perforated panels	Up to 30%

But, despite all these design techniques, these TASs have failed in satisfying the thermal comfort and air quality needs due to several air streams flowing out of the multiple ATDs simultaneously for a single office individual. Therefore, they are not widely spread in office buildings. In cooling modes, more uniform cool air distribution is required over body areas (Forejt et al. 2004) and this distribution is controlled by several ATDs operating simultaneously at the fixed operating scenarios. However, modifications in control strategies for ATDs of these task ambient systems are required to minimize the risk of the draft. Literature study suggests that most of the VAV systems are integrated with several other ATDs to develop micro-climate zones and are unable to provide fast adjustments to air patterns. This is why in PV applications the irregular air velocity is one of the key factors of draft risks (Liu et al. 2018). A more controlled ATD is required instead of multiple ATDs to provide a more variable airflow pattern in accordance with the desired thermal comfort. Hence, single ATD with better control over regulating airspeed and direction can be a better solution to improve air characteristics for office occupants and can even save more energy.

4 CONCLUSION

Conventional mechanical ventilation systems used for the past many decades did not provide user satisfaction in a uniform thermal environment. Task ambient systems are designed with the potential to provide energy savings up to 40% with its flexible design solutions for the

individual thermal comfort. These energy savings are mostly by decreasing the cooling set point and airflow rate. However, despite such energy-saving potential, most of the authors concluded that these systems are not used in real conditions for the office setups. As large number of air streams flowing out of multiple ATDs used in these systems cause high air movements in the breathing zones, which ultimately leads to the draft discomfort for the occupants. Using multiple ATDs for a single workstation in offices requires more controls to adjust the air movements. This causes an increased risk of the draft. However, the control of airspeed and direction using single ATD can provide better air quality and more energy savings.

5 ACKNOWLEDGEMENT

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CHAPTER 5.

PAPER II

Active Chilled Beams for Producing Symmetric Air Discharge Patterns Using Jet-Cone System - An Experimental Study

Haider Latif, Göran Hultmark, Samira Rahnama, Klemen Rupnik and Alireza Afshari

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ACTIVE CHILLED BEAMS FOR PRODUCING SYMMETRIC AIR DISCHARGE PATTERNS USING JET-CONE SYSTEM – AN EXPERIMENTAL STUDY

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KEYWORDS: Active Chilled Beam (ACB), Asymmetric flow, Jet-cone system

INTRODUCTION: Literature studies show that the efficiency of Active Chilled Beam (ACB) systems can be improved by proper modifications in the geometry of its terminal unit [1]. A typical ACB system comprises of a heat exchanger, primary air plenum, mixing chamber and set of nozzles, as shown in figure 1. Nozzles with different shapes and sizes, in the terminal part of ACB unit discharge air into the occupied zone [2]. A previous study shows that asymmetric flow with large-scale circulation patterns were produced by ACB units and can be a cause of draught discomfort for the building occupants [3]. The uneven air jet patterns have been observed from the nozzles inside the ACB discharge unit. The velocity differences between the nozzles, which are close to the ACB inlet and the ones, which are away from it cause these uneven air jets. Different design modifications have been applied to produce air velocities of equal magnitude from all nozzles. Uniform velocities were achieved either by using small size nozzles (9mm diameter), or by adding a board in the middle of the air plenum to have uniform discharge from the terminal unit [4]. Jet cone system [5], can provide uniformity and controls over supply airflow coming out of the terminal units into the room, compared to conventional nozzle designs applied for ACBs. The jet-cone system can change the magnitude of airflow rates and the air direction through different pin adjustments. This can reduce the risk of draught by providing uniform air distribution patterns in the occupied zones.

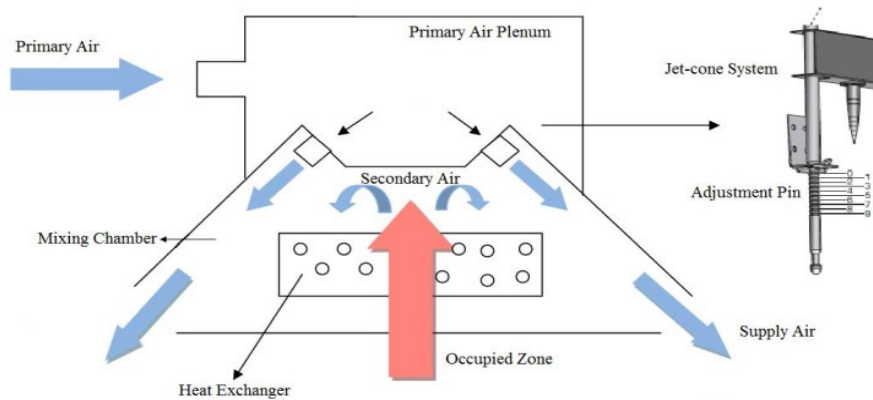


Figure 1 ACB with Jet-cone system

METHODS: The ACB with a jet-cone system was installed for an experimental study to observe the velocity differences between all jet-cones. The testing of this 60*60cm ACB was carried out inside a testing chamber. The jet-cone system consisted of four adjustment pins to control the airflow rates while maintaining required dynamic pressure. Adjustment pins had the scale to

indicate the magnitude of the supply airflow that comes out of the ACB opening in the radial direction. The supply airflow rate was at the minimum level in pin position 0 (up to 30%) and at the maximum level in pin position 9. All the adjustment pins were positioned at 5 to test symmetric airflow symmetry. The velocities were measured by Ball Shaped anemometer to evaluate the air distribution uniformity in the vicinity of the ACB jet-cones. The supply conditions were as follows.

Table 1 ACB Properties

Parameter	Value
Product	Plexus
Function	Cooling 2-pipe system
Product Length	600mm
Primary Airflow rate	20 l/s (constant)
Static nozzle pressure loss	60 Pa

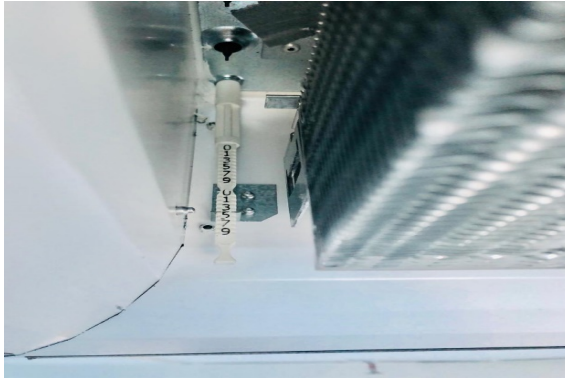


Figure 2 ACB inside view

RESULTS: Measured velocities at 16 different positions inside ACB show an average discharge velocity of 1.85m/s from the jet-cones. Figure 3 shows that the velocity difference is less than 0.05 m/s between all jet-cones, which is comparatively less than the previous ACB designs.

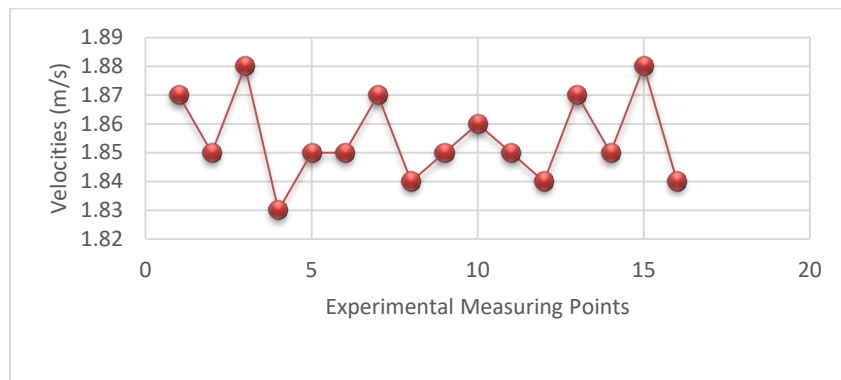


Figure 3 Supply Velocity measurements

CONCLUSIONS: From the initial experimental results, the conclusion is drawn that the jet-cones have negligible velocity differences on all sides of the ACB unit. Placing the board in the middle of the plenum or decreasing the nozzle size also helps in minimizing the velocity differences from the nozzles. While, built-in jet-cone system in ACBs even provides flexible approach in controlling airflow patterns by the change in air velocity and direction. These ACBs are studied further to apply its applications for the thermal comfort in office buildings.

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CHAPTER 6.

PAPER III

Impact of Thermal Heat Loads on Air Distribution Patterns in Open-Plan Offices with Active Chilled Beams (ACBs)

Haider Latif, Samira Rahnama, Alessandro Maccarini, Göran Hultmark,
Klemen Rupnik and Alireza Afshari

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IMPACT OF THERMAL HEAT LOADS ON AIR DISTRIBUTION PATTERNS IN OPEN-PLAN OFFICES WITH ACTIVE CHILLED BEAMS (ACBS)

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SUMMARY

The amount of heat load in open-plan offices has a significant impact on room air distribution, which in return can also influence occupants' productivity. The aim of this study is to investigate the impact of different thermal heat loads on air velocities, colliding effects, and room air circulation patterns. The full-scale laboratory experiments were conducted in a 4.2 m * 4 m * 2.8 m (L × W × H) thermal isolated room. The heat sources of 300 W and 600 W were placed in the occupied region to observe changes in air velocity at 1.1m fixed heights. The average air velocities were increased up to 30% by doubling up the heat loads. The smoke tests and CFD results through vertical and horizontal profiles showed that colliding effect is minimized with an increase of heat loads, but large air circulation patterns (vortexes) are formed in locations of the colliding of jets, near the heat sources and the floor.

KEYWORDS

Active Chilled Beams; air circulation patterns, colliding effect; heat sources; thermal heat loads.

1. INTRODUCTION

Over the years, Active Chilled Beams (ACBs) systems are used for office applications due to their ability to achieve better thermal uniformity compared to conventional HVAC systems (Rhee et al., 2015). This uniformity is maintained mainly due to Coanda effect, induction effect, and different nozzle designs to keep symmetrical air distribution in workspaces (Latif et al., 2022). However, air distribution in rooms with ACBs can be still be affected because of the complex interaction of airflow from the beams outlet with the convection flow produced by heat sources. Furthermore, the flexible changes in the layout of workplaces can further lead to problems related to unwanted air circulation patterns for the occupants. This uneven thermal heat load distribution in an office environment may lead to draught problems. Koskela et al. (2010) made experimentally investigation of air distribution in an office environment with asymmetric workstation layout using ACBs. Results indicated the downfall of colliding inlet jets and the asymmetric layout of ACBs and heat sources as a significant cause of large-scale circulation in offices. The heat load in offices mainly comes from the persons, lighting, and office equipment. The increase or decrease of heat loads in offices is varied mainly due to the increase or decrease of office occupants. Hence heat load and airflow distribution in spaces have a direct impact on occupants' thermal comfort.

Koskela et al. (2012) investigated airflow patterns and thermal comfort in an office environment with ACBs and found out that thermal plumes from heat sources have a notable effect on the velocity distribution in the occupied zone. Melikov et al. (2007b) made an

experimental study using ACBs to show that the heat loads and the airflow rate have a substantial impact on the air distribution in rooms. Results showed that the non-uniformity of the thermal environment increases when the heat load increases. Melikov et al. (2007a) further studied the human response to thermal environment in rooms with ACBs and observed the percent of subjects dissatisfied due to their thermal comfort conditions increased with the increase of the heat loads. Hence literature clearly shows that occupant's thermal comfort is indirectly influenced by the increase of thermal heat loads in the room.

In literature, a lot of work has been done on room air distribution with thermal heat loads. This study is part of the contribution for a major study on office rooms with multiple ACBs and high thermal heat loads. Room air velocities and air circulation patterns are studied by keeping the ACB operating conditions constant. The colliding of jets from multiple ACBs has a significant impact on room air distribution patterns (Dai et al., 2021). Smoke tests and CFD simulations are conducted with multiple ACBs in a room to observe the influence of increasing thermal heat loads on the colliding effect and overall room air circulation.

2. METHODOLOGY

The ACB units used in experiments were tested and rated according to DS/EN 15116 European standard (Company, 2008). Two 0.6*0.6m ACBs were installed in the middle of the test room. The full-scale laboratory experiments were conducted in a 4.2 m * 4 m * 2.8 m (L × W × H) thermal isolated room. Table 1 shows the properties and operating conditions of ACBs.

Table 1 ACB-system details

Units	Values
ACB Dimensions (L*W*H)	0.6m*0.6m*0.2m
ACB Unit	1
Functions	Cooling, Heating & Ventilation
Operating System	Cooling 2-pipe system
Distribution profile	Radial
Primary airflow rate	20 l/s (fixed)
Supply Air temperature	18°C

The vertical velocity measurements at different points near and away from cylindrical dummies were measured by moveable poles attached with Dantec hot sphere anemometers with uncertainty of $\pm 5\%$ of reading at steady conditions. The air velocities were measured at 12 different points at a fixed 1.1m height from the floor, see figure 1.

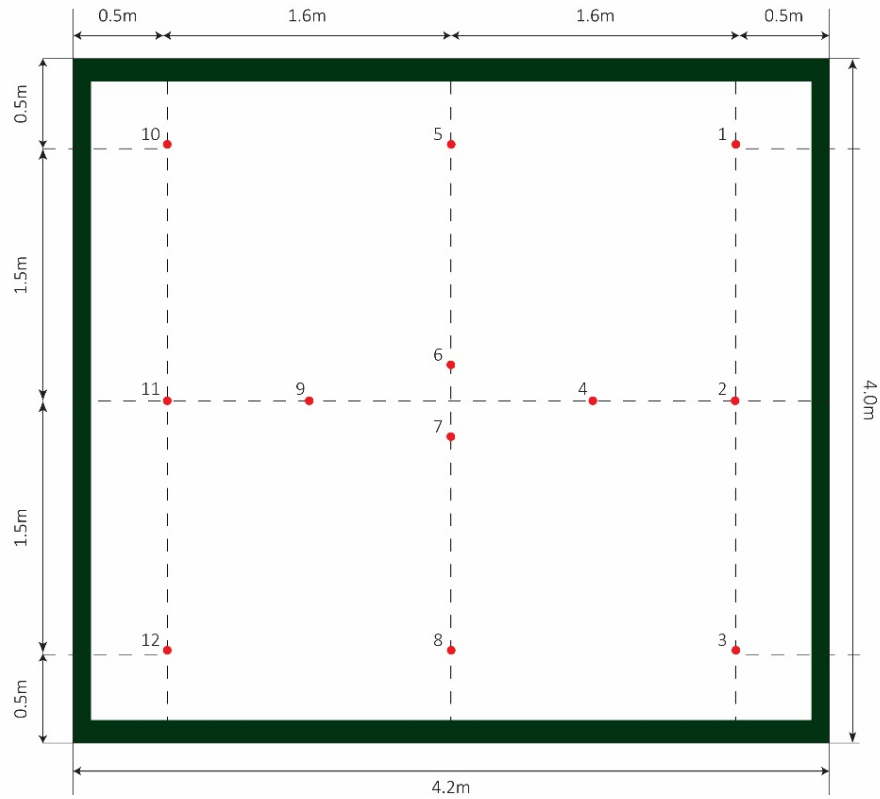
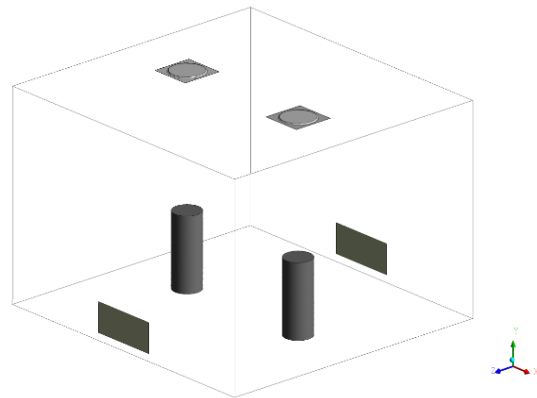


Figure 1 Velocity measuring points

The experiments were conducted for a room with 300 watts positioned in the center, and 600 watts located on all four corners, see figures 2 and 3. The heat loads of 300 and 600 watts were divided into each half of the room and the air velocities near the wall region were ignored. The dummies were positioned facing each other with a distance of 1.5 m (equal to two desks), as shown below.



a)

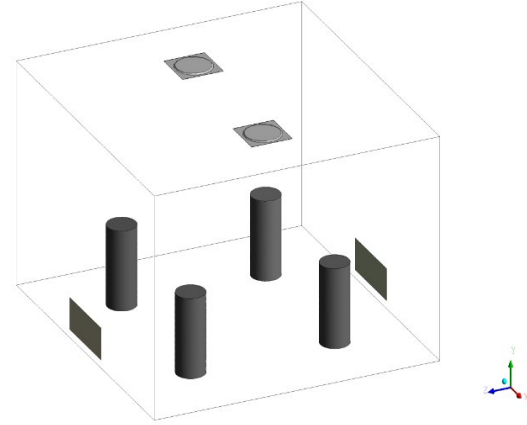


b)

Figure 2 (a) Experimental setup with two dummies (300 Watts) (b) Geometrical model with two ACBs and two dummies



a)



b)

Figure 3 (a) Experimental setup with two dummies (600 Watts) (b) Geometrical model with two ACBs and two dummies

The geometry of ACB units was made circular to reproduce radial distribution pattern and was reduced to only the bottom part for simplicity. The SolidWorks geometric model was imported to ANSYS FLUENT 17.1 version for the CFD simulations. Structured meshes were generated in the entire computational domain to minimize false diffusion. Fine local meshing was done near the critical areas like ACB supply inlets and all heat sources. The pressure-based solver was used for the simulations under steady-state conditions. The RNG $k-\epsilon$ turbulence model was selected for the simulations due to its better accuracy than other RANS models for indoor airflow simulations (Tong et al., 2019), and less computational cost than LES models (Blocken, 2018; Karimi, 2018). Boundary conditions for the velocity inlets of ACBs in both cases were kept constant. The total inlet flow coming out of the inlet opening was 60 l/s, distributed and the convective heat fluxes were applied on the cylindrical dummies. The SIMPLE numerical algorithm was used with the criteria for convergence set $10e-6$.

3. RESULTS & DISCUSSION

The velocities measured at the points (see figure 1) showed a significant increase in air velocities with the increase of thermal heat loads from 300 W to 600 W. The air velocities were seen raised up to 30% with the increase in thermal heat loads.

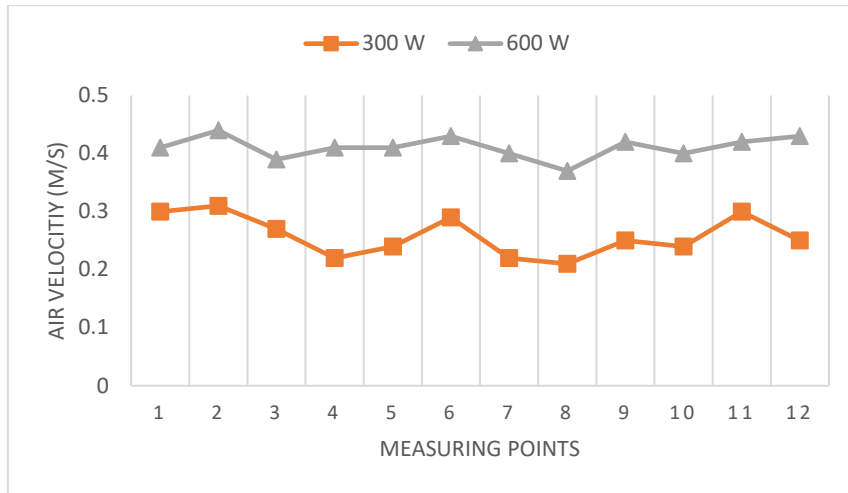


Figure 4 Experimental measuring points

Figure 4 shows that with the increase of thermal loads, air velocities are increased especially on the points near the heat sources. With total heat load of 600 watts, air velocities in the room are increased up to 0.45 m/s.

3.1 Colliding Effect

When the air jets from the two parallel ACBs collide together, they push the jets downward perpendicular to the ceiling surface. Experimental and simulation results for the two cases show that colliding effect starts to diminish with the increase of thermal heat loads. The thermal plumes from the heat sources counter the colliding effect and cause more air circulation in the room as shown in the figures below.

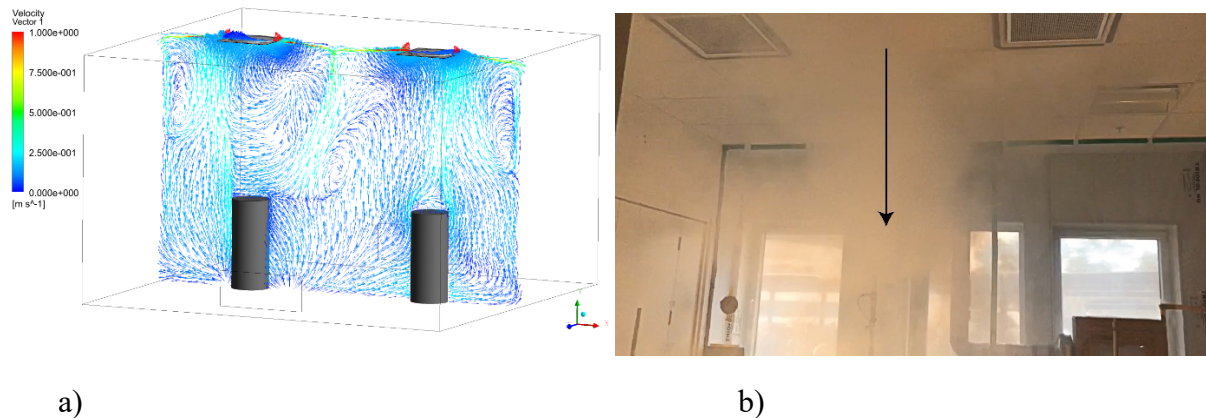


Figure 5 (a) Vertical Vector plane in the middle of the room showing colliding effect (b) Smoke test use to see colliding effect between two ACBs

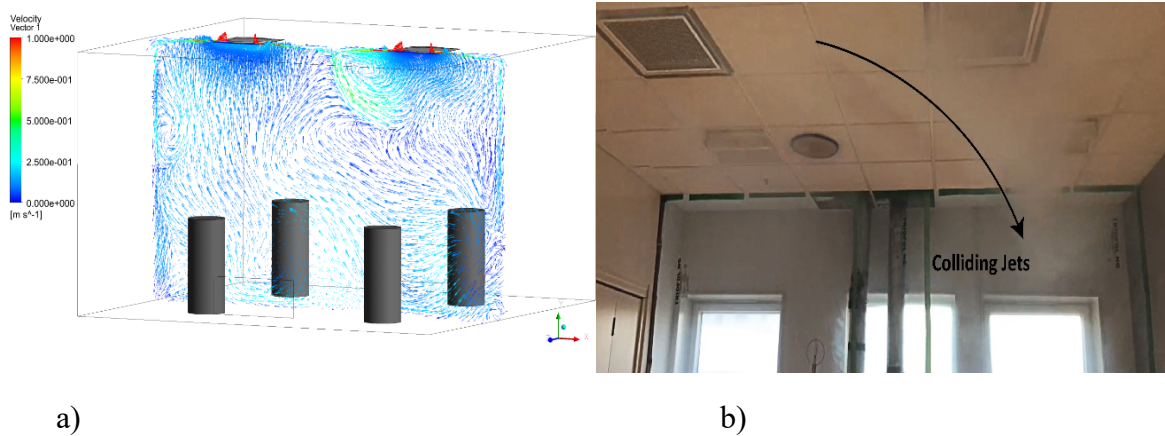


Figure 6 (a) Vertical Vector plane in the middle of the room showing colliding effect (b) Smoke test use to see colliding effect between two ACBs

Figures 5 and 6 show that colliding of jets resulting in the centre of the room is affected by thermal heat loads. This may be because the thermal plumes from heat sources produce a counter effect and are mixed with the downward stream of colliding jets which generates more air circulation.

3.2 Room Air Circulation Patterns

Literature has shown that vortexes formed from the large air circulation in the occupied zone can be a source of thermal discomfort for the building occupants. Experimental and simulation results showed that areas with increased air circulation (vortexes) were found in the occupied zone under the colliding of jets and near heat sources. More vortexes were seen in the center of the room where the colliding effect is minimized by counter thermal heat plumes. With the increase of thermal heat loads, large vortexes are formed as observed in figure 7(b). However, small vortexes were observed close to the room walls and floor.

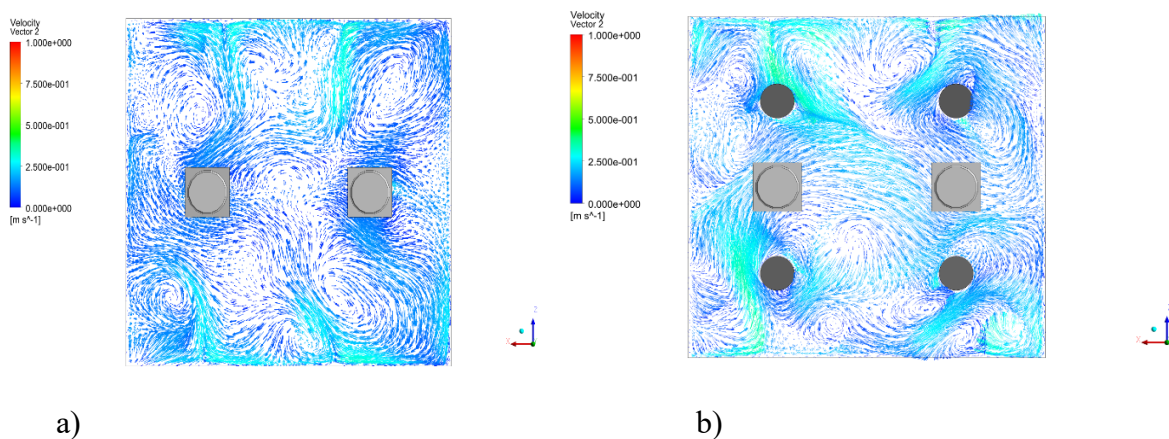


Figure 7 Horizontal planes 1.1m above the floor (a) 300W (b) 600W

Figure 7 shows that more air circulation with large air vortexes and high air velocities are formed near the heat sources and in the center of the room. However, the intensity of the rotating airflow patterns 1.1m above the floor seems less in room with less heat loads.

4. CONCLUSION

Experimental and simulation results showed that the room air distribution patterns are significantly influenced by the thermal heat loads. It was obvious that an increase in thermal heat loads will increase room air velocities. This increase in velocities was seen near the heat sources and colliding of the jets. CFD vertical vector profiles and smoke tests showed that the colliding jets coming from the ceiling were seen diminishing with the increase of heat loads from 300 W to 600 W. These diminishing colliding jets were observed turning to large air vortexes after getting mixed with thermal plumes. These large vortexes formed due to counter-interaction between the colliding jets and thermal plumes were observed near the heat sources and in the center of the room. However, small vortexes were formed near the floor and walls.

5. LIMITATIONS

This study is associated for open-plan office setups, which cannot be considered representative of many offices in Scandinavia. Furthermore, factors like distance of ACB from the walls, change of office configuration, and the location of the exhaust are ignored in the present study, which may also affect the air distribution patterns.

ACKNOWLEDGMENT

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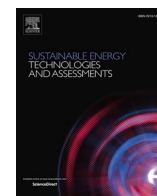
CHAPTER 7.

PAPER IV

Performance Evaluation of Active Chilled Beam Systems for Office Buildings - A Literature Review

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Review article

Performance evaluation of active chilled beam systems for office buildings – A literature review

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ABSTRACT

The applications of Active Chilled Beam (ACB) technology have increased over the two decades and received a lot of attention in research publications due to its ability to provide a suitable indoor climate for building occupants with minimum energy use. In the literature, no comprehensive review on ACB systems has been reported. This study provides a review of the ACB technology by investigating its functions, component and control strategies, design and testing techniques, and its applications for office buildings. Literature shows that ACB systems fulfil sensible cooling demands for office setups and can provide a space-friendly solution by making effective use of ceiling space in buildings. The cooling demands can be met with high induction rates, proper coil circuitry design, and optimum chilled water supply temperature. The high induction rates with more even air-jet patterns inside the terminal unit of ACBs can be achieved by using nozzles of different sizes, cross-sections, and lengths. Studies show that a reduction in energy use up to 30% can be achieved by ACBs as compared to conventional HVAC systems.

Introduction

The rapid growth of world energy use has meant that a critical situation has developed due to the depletion of energy resources and negative environmental impacts [1,2]. The building sector accounts for a significant portion of the world's total electricity usage. The energy demand in residential and commercial buildings has gradually increased, with figures reaching up to 40% in developed countries [3]. According to the International Energy Agency (IEA), the world's total electricity use in buildings is expected to increase by an average of 1.5%/year by 2040 [4]. The new energy-efficient policies for buildings need to be adopted to combat these challenges. In the European Union, continual effort is being made to set out strict building standards and strategies to meet the EU's long-term 2050 goals [5]. Despite these efforts, cooling demand is increasing due to global climate change, affordability of air-conditioning, and increased living standards [6]. A significant fraction of this energy use in buildings comes from Heating, Ventilation and Air Conditioning (HVAC) systems [3,7], as people spend up to 90% of their time indoors [8]. The Global Carbon Capture and Storage (CCS) Institute published projections about the increasing need for cooling equipment in Europe [9]. Hence, sustainable space-cooling

technologies are desired. The IEA's future projections regarding the growth of global electricity demand for space cooling show similar stats, as shown in Fig. 1 [10].

History of chilled beams

Technological change has played a critical role in the green transition and sorting out global issues. These challenges have led many researchers to develop different energy-efficient and cutting-edge technologies. Over the past 50 years, many developments have taken place in manufacturing advanced HVAC products to establish high-quality indoor climate conditions on commercial scales [11,12]. A variety of these HVAC products employed in building applications include diffusers, fan coil units, ventilators, chilled beams, and diffused ceilings, etc. These products have been modified over time to provide suitable Indoor Air Quality (IAQ) and thermal comfort for building occupants [13]. Based on the working principle, chilled beams are categorised into two main types: passive and active. Passive Chilled Beams (PCBs) [14] were first developed in Norway in 1975. These PCBs are operated with a cooling coil in an enclosure. They are suspended from the room ceiling, as in Fig. 2. Chilled water is circulated through the coil, and cooling

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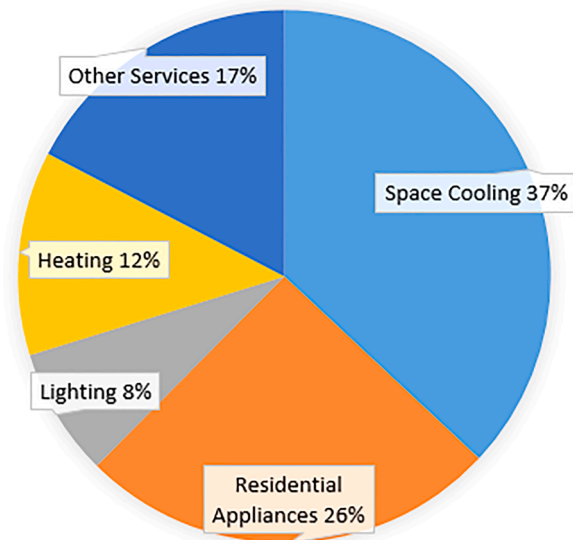


Fig. 1. Share of global electricity demand growth until 2050 [10].

happens in the occupied space through the natural convection of air [15]. Due to low sensible cooling capacity and inability to provide heating and fresh air, PCBs are less applicable for office buildings [16,17]. Therefore, more attention is given to Active Chilled Beams (ACBs) in literature.

ACBs were introduced in 1998 and were suspended from the ceiling to make use of the Coandă effect for horizontal air distribution along the ceiling [16,18]. The ductwork transports fresh air from outside to the pressure box called as primary air plenum. The air inside this plenum is called primary air. The primary plenum is usually set at a pressure that is 30 Pa to 120 Pa above room pressure [19]. These chilled beam designs proved to be more effective than PCBs. 4-Way ACBs were first introduced in 2004 to produce multi-directional or circular air distribution patterns [20]. Both ACBs and PCBs are designed to provide sensible cooling for office buildings. Thus, a Dedicated Outdoor Air System (DOAS) is often required with ACBs to achieve dehumidification. Table 1 shows that the sensible cooling capacities of ACBs are approximately twice that of PCBs [21].

Today, ACB systems are considered to be one of the significant

Table 1

Chilled Beams' Sensible Cooling Capacities [19].

Chilled Beam types	Sensible Cooling Capacities* W/m ²
Passive Chilled Beams	60–70
Active Chilled Beams	130–160

*Taking air speed 0.25 m/s into account in the occupied zone, passive chilled beam and ACB will have maximum cooling capacities of 50 W/m² and 100 W/m², respectively.

energy-saving systems in the countries which use them around the world. Being listed as one of the most promising HVAC-related technologies by the American Council for Energy-Efficient Economy (ACEEE) in 2009 [22], use of ACBs is considered a standard practice in Europe, the USA and Australia as an alternative to conventional Fan Coil and VAV systems.

Working principle of active chilled beams

ACBs (also known as induction diffusers) consist of a heat exchanger to pass chilled or hot water depending on their function. Chilled water temperature, usually between 14 °C and 18 °C, is maintained above the dew point to avoid condensation on the cooling coil [19,23]. ACB systems can be used for heating. Despite their multiple functions, ACB studies relating to the primary heating mode are limited in the literature. In heating mode, water between 30 °C and 45 °C is cycled through the coils. The flow rate of the water determines the room temperature [24]. Processed air from the Air Handling Unit (AHU) is forced into the set of nozzles as primary air. The purpose of the nozzles is to provide high speed primary air and consequently create high dynamic pressure and low static pressure to facilitate induction i.e., pressure differences between mixing chamber and room. The ductwork takes this outside air to the primary air plenum. The primary air plenum is usually set at a pressure that is 30 Pa to 120 Pa above room pressure [19]. The primary air is mixed with the induced air (or secondary air) in the mixing chamber and supplied into the occupied zone as a mixture called supply air [25]. ACBs are mostly fixed in the ceiling to make use of the Coandă effect for proper air distribution. This Coandă effect can be increased with a higher pressure drop or if there is a smaller temperature difference between the discharged air and the air in the room [17]. The warm room air (secondary air) goes up and is induced to the mixing chamber. This secondary air cools (is chilled) down after passing through the

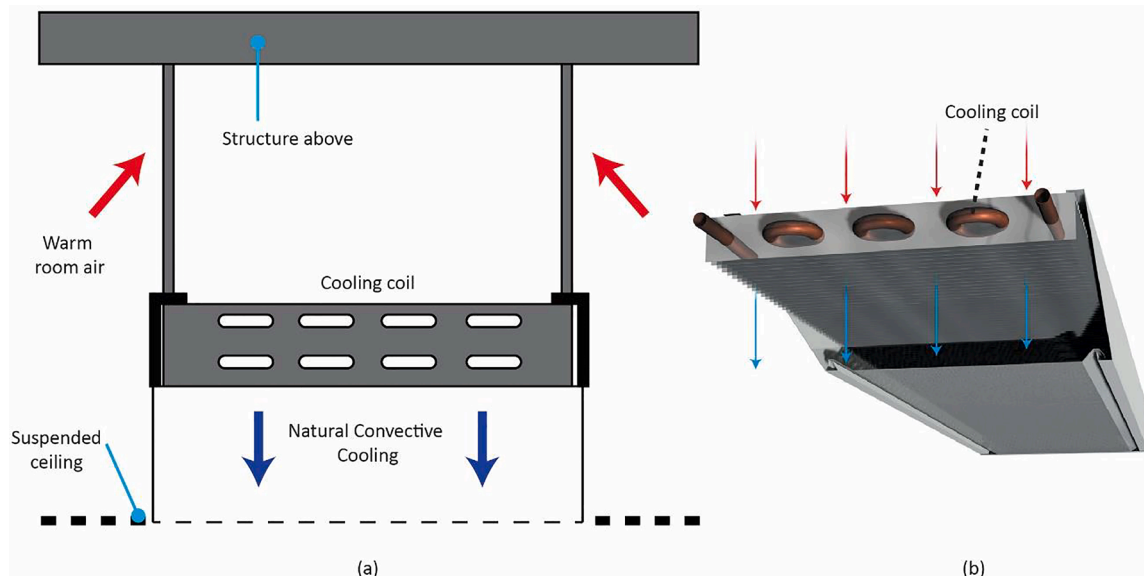


Fig. 2. (a) Passive Chilled Beam working principle (b) Actual passive chilled beam [15].

cooling coils. The heat in the room is taken away by the chilled water inside the cooling coils. The heat transfer between the secondary air and the cooling coil is made forced convection – the operating principle as shown in Fig. 3.

ACBs usually require less ceiling space (up to 0.3 m) and ductwork, classifying it into distinct architecture for office setups [16].

Indoor climate by ACBs

ACBs are a complete indoor climate solution for cooling, heating and ventilation for commercial buildings, hospitals, laboratories, and schools. ACBs can provide satisfactory indoor climate including thermal comfort and Indoor Air Quality (IAQ) [26]. The thermal comfort is provided by the tempered supply air, and the cooling and heating of the air in the room by heating and cooling coil respectively. The Indoor Air Quality (IAQ) is provided first by filtration of the primary air. The room air quality can also be improved by filtration of the recirculated room air before it reaches the cooling coil [27].

Comparison with different HVAC systems

Conventionally, HVAC systems are classified as centralized and decentralized systems according to the functions they provide to the whole building [28]. Centralized HVAC systems are designed to provide air conditioning to the whole building, whereas decentralized or local systems provide separate air conditioning to specific zones of the building. Table 2 shows a detailed comparative overview of centralized HVAC systems for the office buildings.

Most of the studies indicate that ACB systems used in buildings for cooling are more efficient than conventional VAVs in terms of energy use [30]. This is because primary air mostly fulfils the ventilation needs, while chilled water treats the most sensible cooling loads [21]. The comparative analysis between ACBs and conventional VAV systems shows that systems' performance is mainly dependent on the specific building and climate of the locality. The American Council for an Energy-Efficiency Economy reports that energy savings of about 30% can be estimated when comparing ACB systems with traditional VAV systems [22]. However, some still argue that these conventional VAV systems are more energy-efficient than ACBs [31].

In terms of costs, the smaller ductwork and reduced space requirements required in ACB installations lead to the smaller building

volume as compared to conventional VAV systems, and consequently to reduce installation and construction costs [21]. Although the initial product cost of ACB was found more than conventional diffusers by some manufacturers [32], this additional cost can be offset by long-term energy savings and low life-cycle maintenance costs. However, the product cost of the ACB units in comparison to different HVAC system was dependent more on manufacturer and market penetration. The low maintenance cost of ACBs is due to no fans or fewer electric connections required to run the system as compared to traditional VAV systems [16]. Different scientific research carried out over the past 20 years to design ACB systems for office setups is explained through the literature below.

Review objective

With the growing market for ACB systems, a lot of research and analytical studies have been carried out to find out about the performance of ACBs under real office conditions [25]. These studies show that a significant reduction in energy use can be achieved through ACBs due to their high chilled water temperature, optimal nozzle and outlet designs, and their prominent induction feature. Despite its advantages, there is a lack of compiled framework on ACBs in literature for the HVAC research community and field engineers to identify the gaps between research and practice. The objective of this paper is to have a detailed study of the characteristics of ACB systems, the design techniques associated with ACBs and research methods adopted, with a review on the applied experience of their usage for office setups.

Methodology

The terms signifying chilled beams and their impact on thermal comfort and building energy performance were searched through several scientific databases and search engines, including Science Direct, Web of International Patents, Taylor and Francis Online, and Google Scholar. A list of keywords was developed by methodically entering each keyword and combinations of keywords into the search engines, including: ACBs, induction diffusers, air–water systems, ventilation, heating, cooling in combination with thermal comfort and energy performance.

The articles were reviewed, and the compilation process was denoted by a list of relevant articles, patents and reports; and this list was double-checked for any sign of duplication. The screening process for the

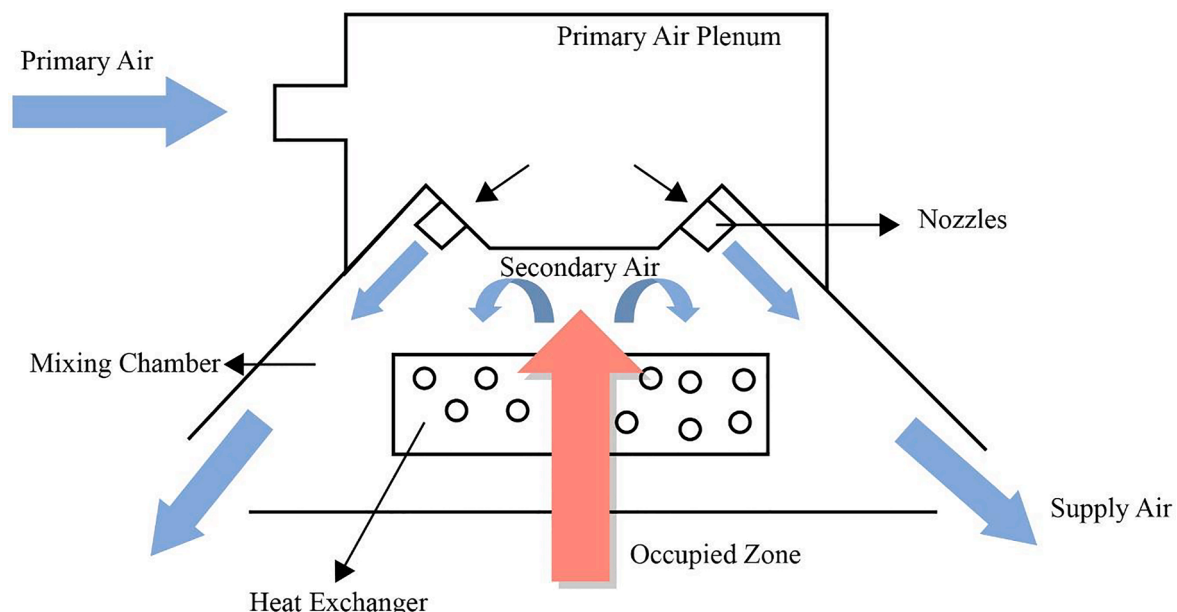


Fig. 3. Working principle of an Active Chilled Beam [23].

Table 2
Comparisons of different HVAC Systems.

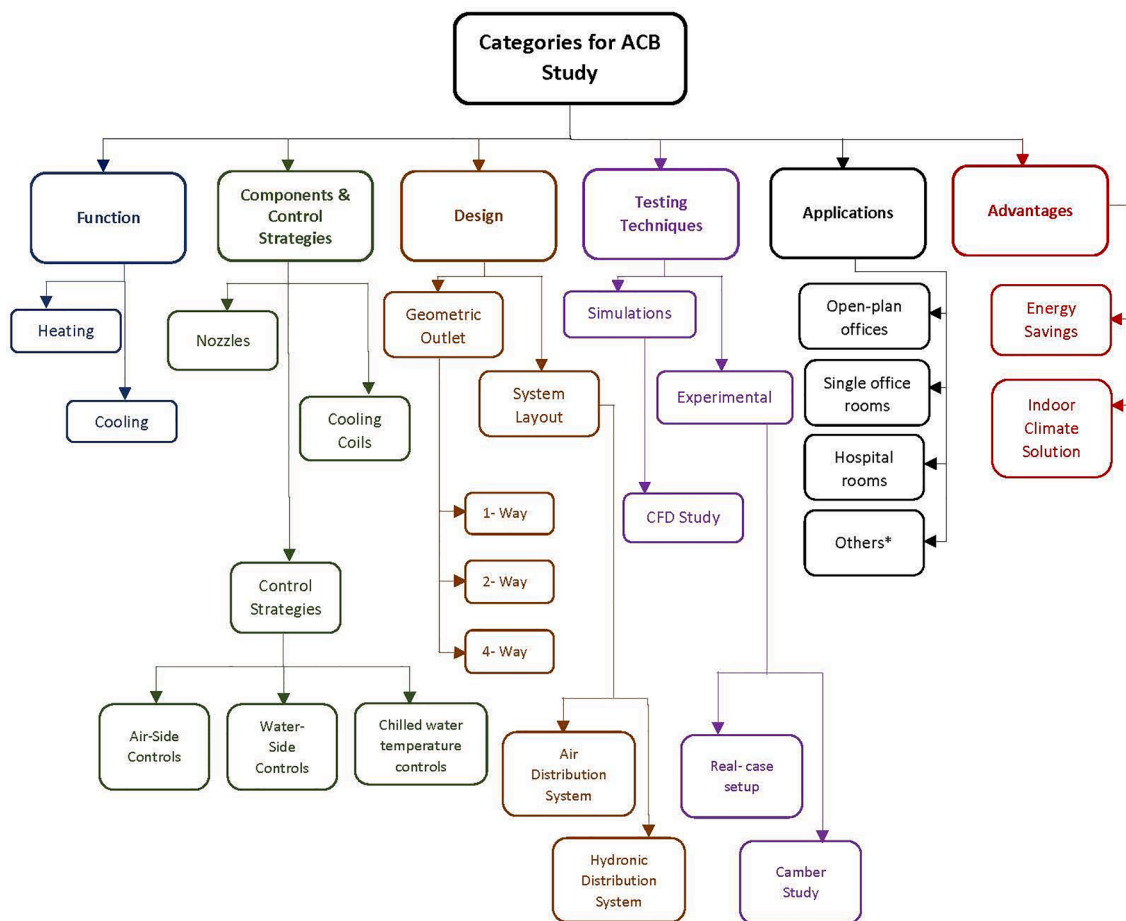
Types	Function	Classifications	Characteristics
All-Air systems* [29]	The thermal heat transfer medium through the building duct system is air.	<ul style="list-style-type: none"> - Constant Air Volume (CAV) system - Variable Air Volume (VAV) system 	CAV <ul style="list-style-type: none"> - Provides constant airflow at variable temperature - Designed for peak load conditions. - CAV systems are less expensive (in terms of initial costs) and simpler to design as compared to VAV systems.
Central HVAC System	Air-water Systems [28]	<ul style="list-style-type: none"> - Fan Coil Unit (FCU) - Active Chilled Beams or Induction Units 	Fan Coil Unit <ul style="list-style-type: none"> - Circulation of the room is done by a fan.
	All-water systems [28]	<ul style="list-style-type: none"> - Fan Coil Units - Passive Chilled Beams 	Fan Coil Unit <ul style="list-style-type: none"> - Circulation of the room is done by a fan
			VAV* <ul style="list-style-type: none"> - Provides variable airflow at a constant air temperature - Designed for instantaneous load conditions. - VAV provide energy savings due to less fan power and less heating or cooling of the primary air. Active Chilled Beams <ul style="list-style-type: none"> - The circulation of the room air is done by induction. Passive Chilled Beams <ul style="list-style-type: none"> - The circulation of the room air is done by natural convection.

*VAV systems studied in this paper focus on cooling room air with the chilled air. The heating system can be of radiators, floor heating, ceiling panels.

shortlisting initially constituted a list of 216 peer-reviewed articles, 38 conferences and 26 patents and reports on ACBs and their performance on commercial scales. These were published mainly between 1998 and 2020. Based on the unique design types and several technologies that are essential in improving efficiency and performance of ACBs, these articles and reports were further categorised. A final list of 150 works of literature was attained after a complete and thorough process. These articles were classified into six categories: ACB functions, components and control strategies, design, testing techniques, applications, and

advantages. These categories were divided to sub-categories to further investigate and give a detailed overview of each main category, as shown in Fig. 4.

Systematically, the review outcomes were then summarised. Subsequently, the summary was checked to determine whether there was detailed information that needed to be included in the review paper.



*Include commercial buildings, laboratories and schools etc.

Fig. 4. Categories for making ACB review.

Categories of research investigation for active chilled beams

Based on literature studies, various design strategies for ACB systems have been formulated to give forward-looking guidance to HVAC engineers, consultants, and researchers with regard to choosing suitable design criteria for the office buildings. An overview of ACB performance is given in detail below by exploring each fundamental component of the system and its design and functions.

Geometric shape and design

The shape and design of ACB systems have a crucial role in the HVAC industry. It is sometimes challenging for engineers to establish a suitable work environment for building occupants, as ACB systems cannot meet high sensible cooling requirements for large spaces [33]. Some spaces are considered suitable for chilled beams, but for other spaces, this technology may not be appropriate. Studies show that open-plan offices with high heat loads can have abnormal air circulation patterns due to asymmetrically placed chilled beams, and these may cause a draught risk [34,35].

The geometry of ACBs is one area where researchers are improving efficiency through design modifications. An ACB unit comprises a heat exchanger, primary air plenum, mixing chamber and nozzles, as shown in Fig. 3. Studies have shown that the arrangement of ACBs installed in open-plan offices may result in a draught sensation due to the asymmetrical layout of terminal units concerning room geometry and heat sources [36]. One type of ACB design applied to a given space may not fulfil high load requirements for other spaces. Upadhyay et al. [37] evaluated the performance of ACB systems for an actual office setup under steady-state conditions and concluded that the design of ACBs requires further research and investigation in a controlled environment. The geometry of ACB systems is mainly designed multiway according to the office space requirements. Based on supply direction, ACBs can be divided into 1-Way, 2-Way and 4-Way (or 360° air distribution) to have the airflow distribution patterns for the required room geometry, see Figs. 5 and 6. Cehlin et al. [26] carried out a comparative analysis by experimentally measuring air change effectiveness (ACE) and air exchange effectiveness (AEE) of 4-Way and 1-Way ACBs in an office setup. Results indicated that 4-Way ACB design provides efficient mixing with the room air and results in a higher mean age of air uniformity throughout the room than with the 1-Way ACB setup. At the same time, local air-change effectiveness (ACE_p) of the unidirectional flow 1-Way ACB design shows improvement in the local air quality and local thermal condition for the single office setups. These 1-Way ACBs are often used for customised applications like patient rooms in hospitals, where there is a need to maintain local IAQ and thermal comfort [38].

Most of the climate chamber studies for ACB systems have been carried out using 2-Way ACBs as terminal units, see Table 3. The 2-Way

ACB units have been used in experimental and simulation studies to provide design guidelines and explore turbulent velocity behaviours in the occupied zones [39]. Cai et al. declared the terminal part of ACB systems as an essential component for assessing the overall system's performance and highlighted heat exchangers issues experimentally using a 2-Way discharge ACB terminal unit [40]. Designing existing building systems, including ACBs, often has several accuracy issues and lacks realism. Fred et al. highlighted differences in existing energy models for 2-Way ACBs (versus the actual 2-Way ACB designs) and, based on this, formulated one guided approach for the modelling community to use to evaluate ACB systems [41].

ACB system layout

An ACB system comprises two parts: the air distribution system and the water distribution system, as described below.

Air distribution system

The air distribution system consists of ACBs, air handling unit (AHU), and ductwork [33]. The AHU mainly has filters, chilled water coil, hot water coil, and fans. This system design often has a heat recovery feature, including a heat exchanger between the exhaust air and the outside air, and humidification/dehumidification components. This air distribution setup provides the primary air to the ACBs and return air to the exhaust through the duct network, as shown in Fig. 7. Some ventilation systems, referred to as Dedicated Outside Air System (DOAS), use 100% outside air (with no recirculation) and dehumidify it before it enters the building space [32]. The system with air recirculation can save energy compared to using DOAS without recirculation [42].

Hydronic distribution system

The hydronic distribution system of ACBs consists of three major parts: generation, distribution, and consumer loads. This water distribution is typically arranged in a two-pipe or four-pipe configuration [43], as shown in Fig. 8. A two-pipe configuration includes one supply pipe and one return pipe. This means that building spaces can only be either heated or cooled at any given time. On the other hand, a four pipe configuration enables simultaneous heating and cooling in the building, as it consists of two supply pipes and two return pipes [44].

Maccarini et al. [45] developed a novel ACB water distribution system that can simultaneously heat and cool building spaces using a two-pipe configuration. The work was supported by simulations in Dymola and later implementation in an office building [46]. Traditionally, two-pipe systems circulate supply water at temperatures of about 30–45 °C and 14–18 °C respectively for heating and cooling operations. The innovative two-pipe system operates with a supply water temperature of

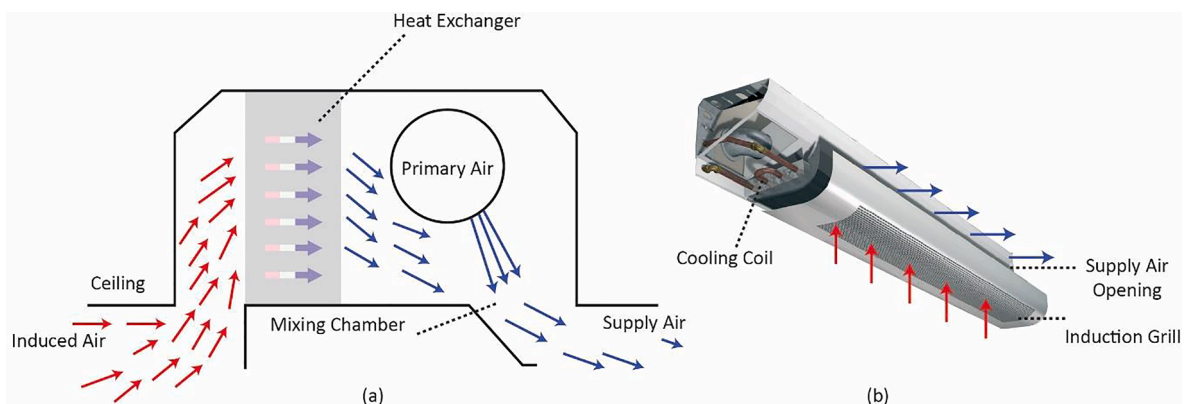


Fig. 5. (a) Working principle of 1-Way ACB (b) Actual 1-Way ACB Unit with one supply-air opening [26].

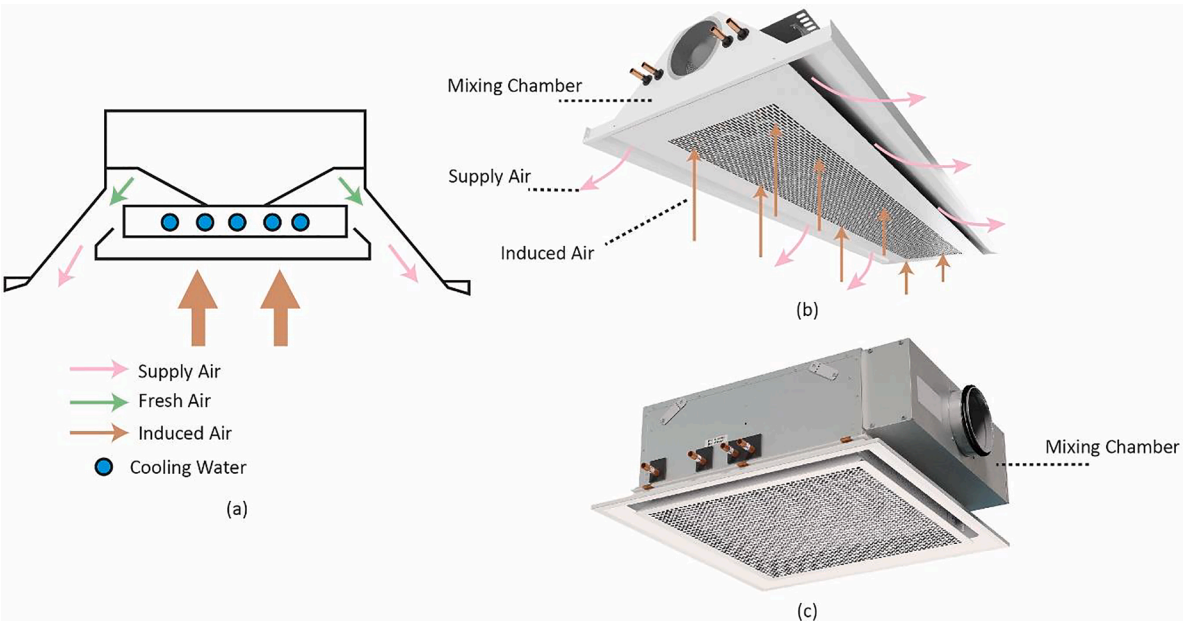


Fig. 6. (a) 2/4-Way ACB principle (b) Actual 2-Way ACB with two supply-air openings (c) Actual 4-Way ACB with four supply-air openings [26].

Table 3
ACB studies according to its design and applications.

Ref.	Design and Geometry					Layout		Operating mode**	
	1-Way	2-Way	4-Way	9 mm round/cylindrical-shaped	Other nozzle shape	Open-plan office	Other setups*	Heating	Cooling
[24,37]		✓				✓		✓	
[17,34]		✓		✓		✓			✓
[40,58–60,77,78]		✓		✓			✓		✓
[68,79]		✓				✓			✓
[80]		✓			✓	✓			✓
[81,82]		✓				✓		✓	✓
[26]	✓		✓			✓			✓
[36,83–85]		✓					✓		✓
[86]		✓					✓		✓
[64]		✓		✓	✓		✓		✓

*other setups include independent ACB studies or single rooms with no activity.
**operating mode in most of the studies is either heating or cooling. Only one case for simultaneously heating and cooling through ACBs is reported in literature.

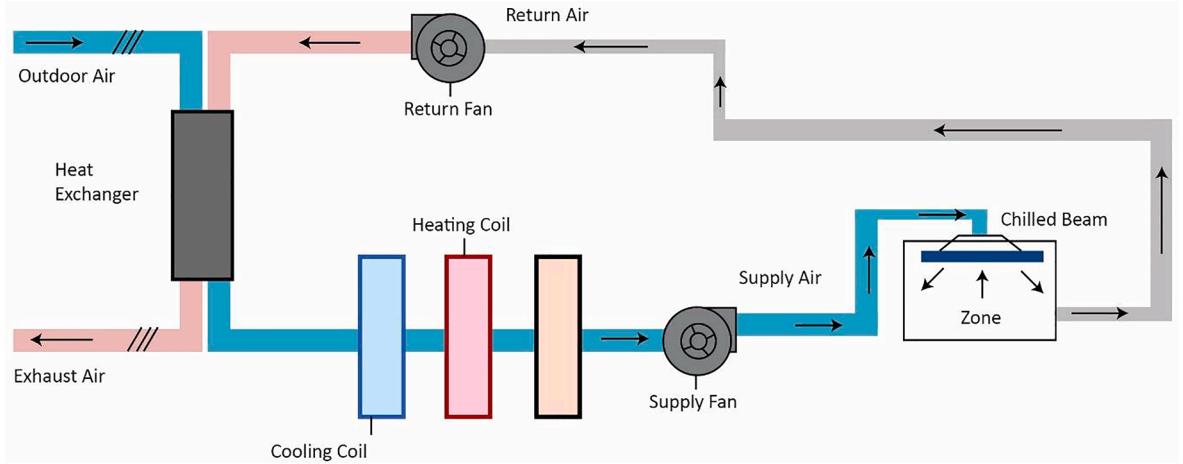


Fig. 7. Air distribution layout of ACBs consisting of ACBs, air handling unit (AHU) and duct work [42].

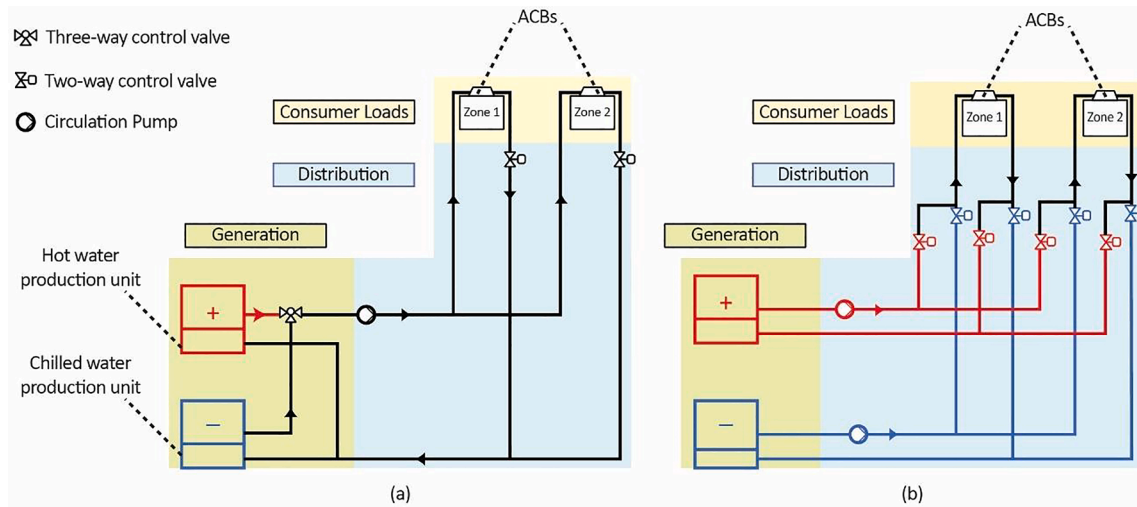


Fig. 8. (a) Two-pipe configuration with one supply pipe and one return pipe (b) Four pipe configuration with two supply pipes and two return pipes [43].

about 22 °C all year round. Results from simulations show that the innovative two-pipe system can save approximately 12–18% of total annual primary energy compared to traditional four-pipe systems, mainly due to useful heat transfer from warm to cold zones and higher potential to provide free cooling [47].

Active Chilled Beam controls

An ACB system consists of water and air distribution systems, as described in the section above. These systems require control strategies to maintain zone temperatures and control chilled water temperature to avoid condensation and maximise the cooling output of the beams [23,48].

Air-side and water-side controls

In ACB systems, the primary airflow rate, water flow rate and zone temperature are control variables for developing any control strategy to minimise energy use and improve thermal comfort [49]. The primary air temperature and chilled water temperature are kept constant, as they are not control variables. The desired room air temperatures are usually

achieved by varying the water flow rate in the water distribution system. Water-side control is a closed loop which controls the room temperature by varying the water flow rate while primary air temperature and airflow rate remain constant. The purpose of air-side controls [40] in ACBs is to meet zone ventilation requirements and minimise draughts by properly maintaining air distribution patterns through variable primary airflow rate. Demand-controlled ventilation (DCV) is also applied, where the primary airflow rate is varied in response to zone occupancy to optimise the system's energy use [50]. Along with the ACB unit, an additional air supply diffuser can be introduced upon increase in the room occupancy. In such cases, water flow rate through the beam is modulated in accordance with the temperature demand of the zone.

Chilled water temperature controls

Since most of the ACBs are designed to be sensible cooling devices, different control strategies are developed to adjust the temperature of the beam [51]. This may eliminate the need for a separate DOAS for dehumidification. Chilled water temperature controls are essential to avoid condensation. In order to avoid condensation, the temperature of the brine is raised, and this results in decreasing of the cooling capacity

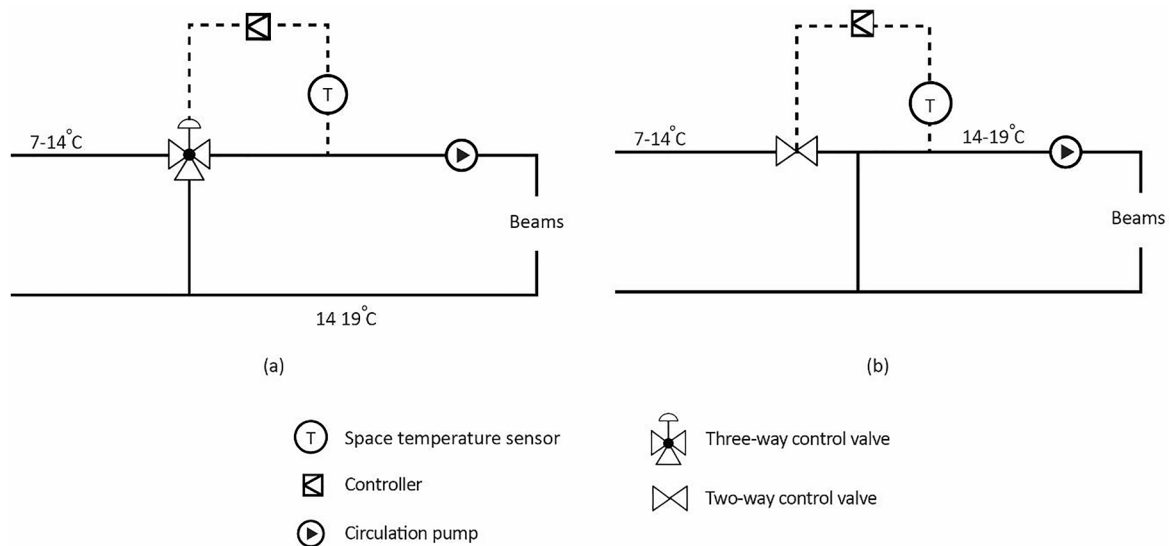


Fig. 9. (a) Open-loop beam chilled-water system supply temperature maintained by mixing primary and secondary chilled water (three-way valve and bypass) (b) Open-loop beam chilled-water system supply water temperature maintained by mixing primary and secondary chilled water (two-way valve and bypass) [23].

of ACBs. Fig. 9a shows the use of a three-way mixing valve to mix primary and return water to maintain the beam chilled-water supply temperature. Fig. 9b shows a two-way modulating valve and bypass are used to mix primary and return water to maintain the beam chilled-water supply temperature.

Three-way or two-way open-loop modulating valves and a bypass are used in beam chilled water systems to maintain supply water temperature by mixing primary and returned water. While, closed-loop beam control systems, as illustrated in Fig. 10, maintain supply temperature by a two- or three-way control valve installed on the suction side of the pump. These control valves maintain the chilled water temperature by regulating primary water flow in the heat exchanger.

Model Predictive Control (MPC) can be applied to optimise system efficiency in terms of energy use or indoor thermal comfort while satisfying system constraints. Using a model of the system, the performance of the system is optimised over a finite future horizon. For instance, in [52], experimental results indicated a 20% electricity saving with MPC control, while an acceptable Predictive Mean Vote (PMV) was fulfilled, i.e. $-0.5 < PMV < 0.5$. In addition to above-mentioned control methods, there is an opportunity for self-regulation [53], i.e. elimination of individual room control systems in ACBs with high temperature cooling. The self-regulating ACBs provide an effective way to maintain balance between chilled water temperatures of ACBs and indoor air temperature without the need for any control components [54]. The flow rate of chilled water in a self-regulating active chilled beam remains constant, as the cooling capacity is instead determined by the room temperature and by the centrally controlled chilled water temperature [55]. Hence, these self-regulating ACB systems keep the indoor air temperature satisfactorily uniform by supplying the building with enough cooling and keep the building within the desirable indoor air temperature limits [46].

Nozzles

Proper modifications in the geometry of the terminal unit of any mechanical ventilation system can improve its efficiency [56]. In an ACB system, the primary air is discharged through multiple sets of nozzles

present on the sides of the heat exchanger. These ACBs come with different nozzle shapes and in different sizes. Bingjie et al. [57] observed uneven air-jet patterns from the nozzles inside the ACB discharge unit. The velocity differences between the nozzles which are close to the ACB inlet and the ones which are away from it cause these uneven air jets. Different design modifications have been applied to produce air velocities of equal magnitude from all nozzles. The uniform air velocities were achieved either by using small nozzles (9 mm diameter) or by adding a board in the middle of the air plenum to produce uniform discharge from the terminal unit. Guan et al. [58] made a regression analysis to show that Induction Ratio (IR) has a negative correlation with the nozzle radius ranging from 1 mm to 10 mm. At the same time, the distance between nozzles has a positive correlation with IR. The radius of the nozzle has a more substantial effect on IR than the distance between nozzles. Another study shows that the smaller ACB nozzle sizes provide efficient heat transfer, which leads to higher IRs for the fixed primary airflow rates [59]. This shows that nozzles with small diameters (up to 9 mm) lessen the primary air resulting in improved IR. Further studies show that choosing the optimal nozzle length between 60 mm and 80 mm in ACBs could increase IR by up to 30% [60]. Ruponen et al. [61] conducted an experimental study by using different-sized circular nozzles with different outlet geometries to indicate that the area ratio of the supply nozzle is another significant parameter that influences the IR. Results indicated that the outlet geometry of the ACB has a negligible effect on the IR. Hence, nozzle sizes and spacing between them are considered key factors by researchers and manufactures when evaluating the performance and IR of ACBs.

Nozzles of ACBs also come in different shapes, as shown in Fig. 11. Commercially, different nozzle designs for ACBs have been tested to improve the IR. The rectangular and elliptic-shaped nozzle geometries have positively affected jet's mixing due to counter-rotating stream-wise vortices and improvement regarding self-induction, respectively [62,63]. Experimental investigations into the performance of ACBs using different nozzle designs have shown that cross-shaped nozzle designs achieve a higher IR than all other nozzle geometries working on constant flow conditions [64].

In addition to this, the shape of these cross-shaped nozzles is further

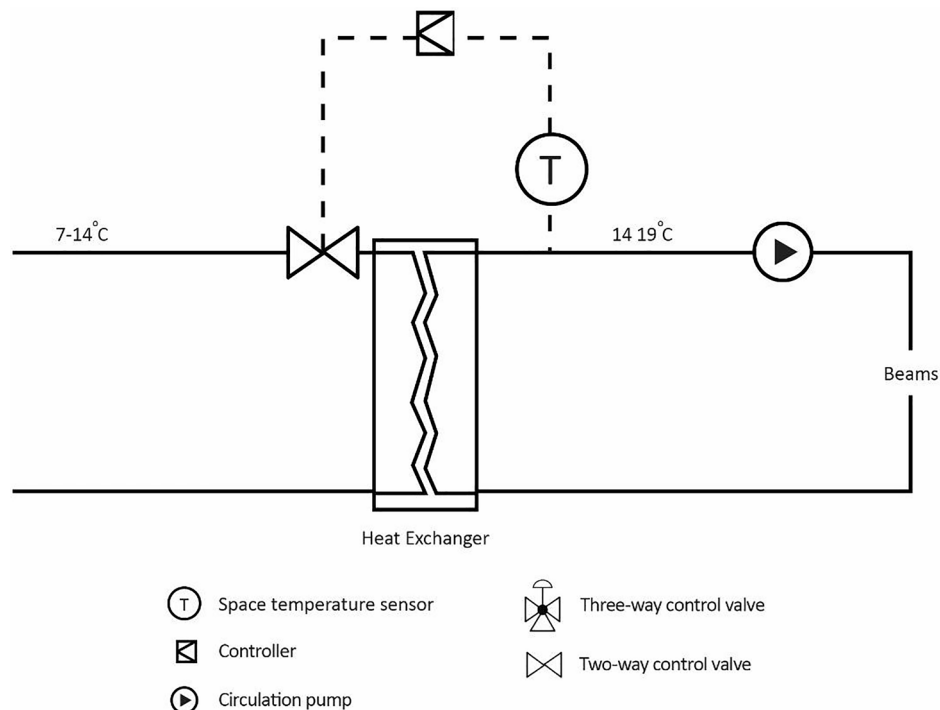


Fig. 10. Closed-loop beam chilled-water system supply temperature maintained by regulating primary water flow in heat exchanger [23].

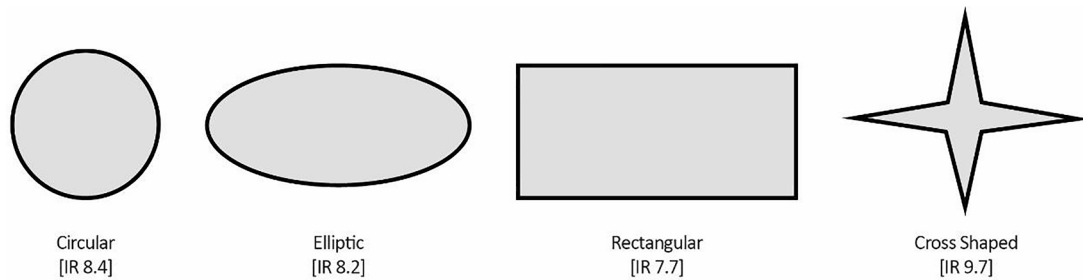


Fig. 11. Nozzle designs for ACBs [64].

modified to lobbed-shaped (by over-lapping cross-shaped nozzles) to enhance mixing and momentum [65]. Jet cones in ACBs can provide better control over supply airflow coming out of the terminal units into the room compared to conventional nozzle designs. This jet-cone feature in ACBs can change the magnitude of airflow rates and the air direction through different pin adjustments [66]. These changes in nozzle designs have a significant impact on the amount of induced air, which is further discussed below.

Induction effect

The induction effect is considered a prominent feature of ACB systems. Literature shows that IR is a key parameter for measuring the efficiency of ACBs units. It is defined as the amount of secondary air entrained per second divided by the amount of primary air supplied per second [67]. Different measuring methods [68] are applied in experimental studies to determine the IR of ACBs under different operating conditions [33]. These methods involve measuring the induced flow through capacity method, air temperature method and air velocity method as explained below.

- Capacity method: This method involves applying an energy equation over the cooling coil. The induced flow is determined by measuring the flow and the temperature rise of the chilled water and the temperature drop of the induced air.

- Temperature method: The induced flow in this method is determined by measuring the temperature of the mixed supply air, the primary air, and the induced air downstream of the coil.
- Air velocity (venturi) method: This method involves the induced airflow rate as a function of the air velocity measured at one point in the throat of the venturi.

The induced flow determined from the above methods is divided by primary airflow of ACB to determine the IRs. The venturi method is considered a reliable method for calculating IR when compared with the results obtained from the European Standard prEN 15116 [69]. However, IRs involving both capacity and temperature suffer difficulties measuring the temperature of the induced air downstream the coil. Experimental results, as shown in Fig. 12, show that the modified capacity method [68] used for primary flow rates (q_p) between $15 \leq q_p \leq 35$ (l/s) gives more consistent and accurate IRs. This method involves combining the expressions of both the capacity method and the temperature method to measure IR without any problem [70].

In literature, different correlations are developed between IR and various ACB operating parameters. Studies show that the primary airflow rate has a weak correlation with Induction Ratio (IR) [71,72], while an increase in plenum pressure gives a slight decrease in IR [73]. The self-regulating effect [54] has a weak influence on the IR of the chilled beam, but large temperature differences can lower the IR to a certain extent [53]. The induction effect for turbulence jet models, especially used for ACBs, is often presented by applying some empirical

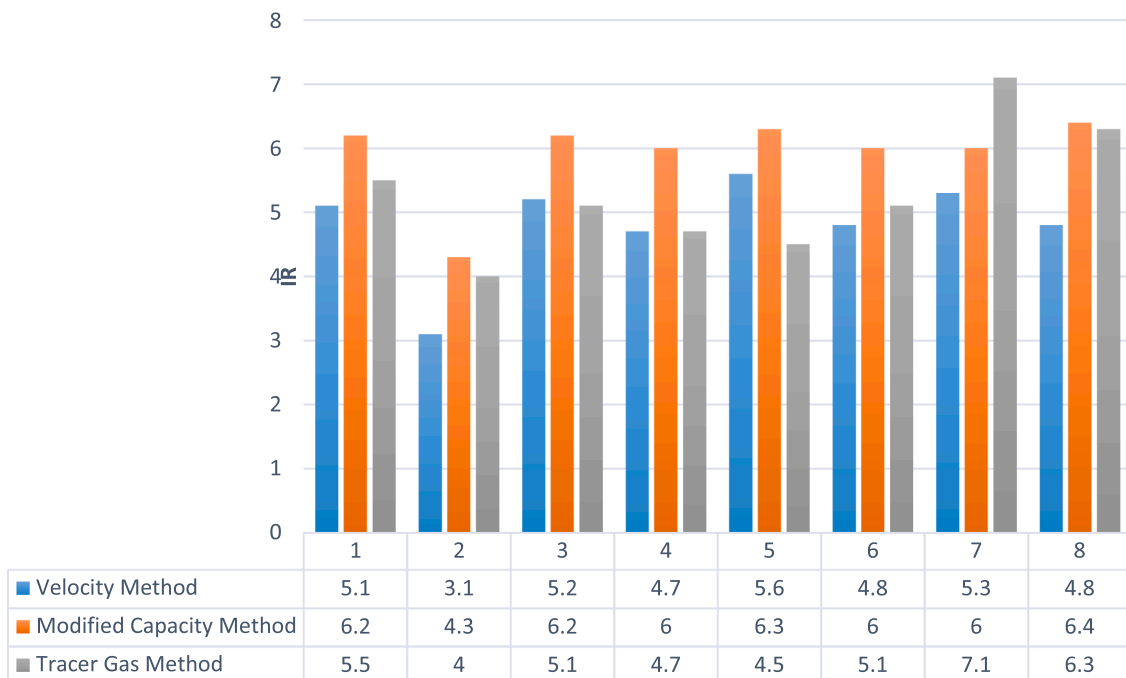


Fig. 12. Experimental results from different methods used for measuring Induction Ratios [68].

hypotheses and modelling techniques. This is due to complications of jet interaction and merging across the cooling coils in the terminal unit described below [74–76].

Table 3 shows that ACB can be used for heating, but thermal stratification must be considered. This is due to the risk of discomfort caused by radiant temperature asymmetry in the room [82]. There are ways to solve this problem for an instant through zone heating or modifications in the heat exchanger design [37].

Cooling coils and filters

Induced air is cooled by the coil or the copper tubing, bonded to aluminium fins, and supplied with chilled water to wick away heat. Higher chilled water temperatures (14 °C to 18 °C) are used in ACB cooling coils compared to conventional air-conditioning systems (which use chilled water temperatures ranging from 4 °C to 7 °C). After mixing in the room, the warm room air ascend upwards towards the cooling coil for the heat exchange [16]. Filters can be placed between the cooling coil and the induction grill to clean up the incoming warm (secondary) air and discharge it back to the room as supply air.

Filters

Indoor air consisting of gases and particles is continuously circulated within the room and moves back towards the cooling coil for the heat exchange [87]. One of the key issues of an ACB is the unclean induced air passing through the cooling coil [88]. This is one of the main setbacks of ACB systems for hospital applications. Fibrous filters placed prior to the cooling coil, as shown in Fig. 13, have been used in air cleaning for the past many years [89]. Studies suggest that these filters reduce the secondary airflows between 6 and 15% due to their significant effect on the pressure drop [90], reducing the ACB efficiency between 22 and 38% [27]. ACBs with filters are observed to be more suitable for hospital applications (than office buildings), where even the slightest risk regarding IAQ in accordance with ASHRAE Standard 170 is not acceptable [91]. Most of the ACB studies in the literature have been conducted without considering the filter effect in the discharge unit.

Cooling capacity

The performance of the cooling coils is mostly represented in terms of the cooling capacity of the cooling system. The cooling capacity of an ACB can be defined as the ability of its cooling system to remove heat from the specific space. The total cooling capacity of these air–water systems is a combination of the air-side cooling capacity and the water-side cooling capacity, as shown in equation (1) [59].

$$P_{\text{total}} = P_{\text{air-side}} + P_{\text{water-side}} \quad (1)$$

Air-side cooling capacity is given as

$$P_{\text{air-side}} = m_{p,a} \cdot C_{p,a} \cdot (t_{r,a} - t_{p,a}) \quad (2)$$

Here,

$m_{p,a}$: mass flow rate of primary air; $C_{p,a}$: specific heat of the air

$t_{r,a}$: induced air temperature; $t_{p,a}$: primary air temperature

Assuming no condensation in the cooling coil, the cooling capacity of the cooling coil of an ACB system can be found as:

$$P_{\text{water-side}} = m_w \cdot C_{p,w} \cdot (t_{w,\text{out}} - t_{w,\text{in}}) \quad (\text{under steady-state condition}) \quad (3)$$

Here,

m_w : mass flow rate of cooling media; $C_{p,w}$: specific heat of liquid media

$t_{w,\text{out}}$: existing water temperature; $t_{w,\text{in}}$: inlet water temperature

It is important to note that there is a risk that the air will condense on the fins of the cooling coil [21]. This has a substantial influence on the functioning of the cooling coils [92].

Cooling coil simulation models

Most of the research work on cooling coils has been incorporated into the modelling techniques due to their function, complex geometry and positioning in the ACB unit. Different types of model are implemented for the cooling coil studies to investigate their function and limitations in a detailed way. A comprehensive review of these models is provided in Table 4.

Hybrid models [96–101] are more reported for the cooling coil in chilled beams due to all the models' advantages. Static models, [93–95] despite the error of around $\pm 10\%$, are reported insufficient in most

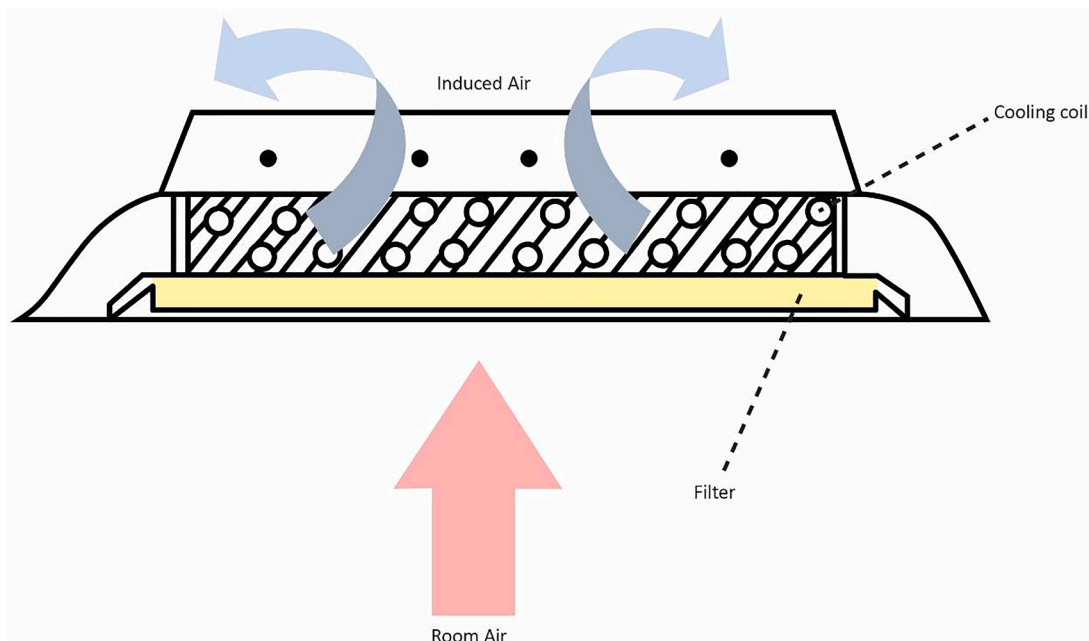


Fig. 13. ACB Cooling Coil with filter [89].

Table 4
Cooling Coil models for ACB applications.

Ref.	Model Type	Function & Applications	Limitations
[93,94]	Theoretical Cooling Coil Model	<ul style="list-style-type: none"> - The models require dimensions of the fin and the tube thickness, diameter, and spacing as inputs in order to calculate the heat transfer coefficients. Applied for ASHRAE simulation toolkits for annual energy calculations of buildings. 	<ul style="list-style-type: none"> - Insufficient robustness - Models are expressed based on physical & conceptual hypotheses without practical validation.
[95]	Empirical Coil Model	<ul style="list-style-type: none"> - The coil model works regardless of the type of fluid in the tubes with a set of constants. - This approach has the capacity to simulate the energy performance of other HVAC components. 	<ul style="list-style-type: none"> - Different sets of constants are required for different fluids.
[96–100]	Hybrid Cooling Coil Models	<ul style="list-style-type: none"> - Model is developed with the air saturation specific heat information. The advantages include simplicity, accuracy, and consistency with the methodology for analysing sensible heat exchangers. - Less computational requirements. - These models have applications in operational optimisation, performance assessment, fault detection and diagnosis in liquid desiccant dehumidification systems. 	<ul style="list-style-type: none"> - Results from these models do not have the sufficient cooling capacity with constant fluid flow rates & exit temperatures in the cooling coil compared with the dynamic models.
[70,101]	Dynamic hybrid models	<ul style="list-style-type: none"> - These models cover mechanical and thermal aspects of air jets & cooling coils with better accuracy and robustness. 	<ul style="list-style-type: none"> - High complexities, as this approach is new to ACB applications. - More dependent on experimental estimations, which may involve high uncertainties. - Compromise is made in capturing exact underlying physics and suitability for engineering applications.
[102–104]	Dynamic Cooling Coil Models	<ul style="list-style-type: none"> - The dynamic behaviour of the heat exchanger in these models is characterised by time constant, delay time and gain in these models. - Easily modified for different types of heat exchanger. - Good for the prediction of transient performance of the fluids flowing through the heat exchanger tubes. 	<ul style="list-style-type: none"> - These comprehensive dynamic models are developed for computer simulations, which are too complex to be applied practically. - Involve extensive numerical and experimental studies from literature for the validations.

Table 4 (continued)

Ref.	Model Type	Function & Applications	Limitations
		<ul style="list-style-type: none"> - Predicts the transient behaviour subject to arbitrary inlet temperatures in shell and tube heat exchangers with parallel or counter current flow. 	

cases compared to dynamic models with less than $\pm 5\%$ [102–104]. Dynamic models for cooling coils in ACBs take advantage of both the physical and empirical modelling approaches and can accurately predict performance in a wide operating range with real-time operations of heat exchangers. However, additional mass balance equations are applied in the case of wet cooling coils in dynamic models. While dynamic modelling for dry cooling coils involves energy equations to achieve reasonable simulation results [105].

Other factors affecting ACBs' performance

In the ACB studies, humidity control, unsymmetrical heat source distribution, collision of jets from multiple ACBs and large air circulation patterns in the open-plan office spaces also affect ACBs' performance, as described below.

Building humidity control

The chilled water temperature in ACBs must be at a certain temperature above room dew point temperature to avoid condensation. The lower the chilled water temperature the higher the power output of the system. This means that lowering the room dew point temperature makes the ACB system work more efficiently [39]. Building humidity control is required for the proper functioning of air–water systems especially in humid climates [23]. For example, opening a window in a humid climate raises the dew point temperature of the air in the room. To avoid condensation in this room, it is common to use moisture detectors or sensors which detect condensation [33,56], and accordingly close the water valve in ACB resulting in a higher temperature in the cooling coil.

Heat sources

Ventilation produces air currents across a room in directions initially determined mainly by the geometry of the ACB. On the other hand, heating, and cooling cause convection currents, affecting the vertical distribution of heat and outdoor air. In open-plan offices, convective flows coming from the heat sources significantly impact airflow patterns produced by the air terminal devices. These airflow patterns inside the room control the thermal comfort in the occupied zone. Heat sources, especially in open-plan offices with different layouts, cause convection currents, which are equal in strength to those produced due to ventilation [84,106,107]. Another study shows that the unsymmetrical distribution of these heat sources is considered the main reason for draught in the rooms with ACBs [26].

Air circulation patterns and colliding jets

The office spaces with high occupancy density (or heat sources) are designed with multiple chilled beams units, which work together to produce favourable indoor climates for the building occupants. Studies show that the collision of inlet jets from these multiple beams is another source of draught discomfort [34,108]. The multiple nozzles in ACB units produce several jets that travel in parallel downstream directions in the same plane and spread along the ceiling as confluent jets [109].

The momentum is conserved better by confluent jets than any other type of jets [110,111]. These jets stay attached to the ceiling until they get deflected by a room wall [112]. Experimental studies show that large air circulations are observed in rooms installed with ACBs because of asymmetric layout of their terminal units [113].

Outdoor conditions

Studies show that outdoor conditions also significantly impact the airflow patterns of rooms and occupant's productivity [107]. Cool windows and outer wall surfaces can create downward flows, and warm surfaces upward flows, which affect the thermal comfort in the room [114]. One study shows that the most critical zone in which people often suffer draught is located close to a wall and floor [83]. However, the risk of a draught is reduced if ACB systems are arranged efficaciously to avoid high induction.

Testing techniques for the ACB study

Most of the studies on ACBs are related to office setups, as shown in Table 2. Different experimental and simulation methods are applied to study the airflow patterns and performance of the ACBs for the different operating parameters. The validated results are used further to improve the ACBs designs and minimise office occupants' draught discomfort.

Simulation tests

In recent years, CFD has been successfully applied to indoor airflow analysis and produced results by using different airflow models [115]. CFD has been a valuable tool for visualising such 3D flows and effective air distribution patterns inside rooms with ACBs [116–118]. Applications of CFD in building ventilation were introduced by P.V. Nielsen, and now, it has become an integral part of scientific research into complex air distribution and ventilation systems [119–121].

Table 5 shows that LES models have the potential to provide more accurate and reliable results than other RANS models in ACB studies but at the expense of higher computational cost [133]. The standard K-epsilon model [124] is reported to be widely used due to its robustness

Table 5
CFD Turbulence Models and their use in ACB studies.

References	Models	Characteristics
[34,36]	k- ω Shear Stress Transport (SST) [122]	<ul style="list-style-type: none"> - Study on airflow near the walls and ceilings, induction plates, and airflow with strong buoyancy. - Not suitable for the ACB studies involving mixed convection
[57–59,123]	Standard k-epsilon model [124]	<ul style="list-style-type: none"> - Robust and reasonable accurate for indoor climate ACB applications. - Low computational cost - Not applicable to near wall region
[125,126]	RNG k-epsilon model (Surface-to-surface) [127]	<ul style="list-style-type: none"> - Enables to use lower Reynolds numbers than standard model. - Better results on mixed convection and impinging jets in ACB study
[128–130]	Realisable k- ϵ model [131]	<ul style="list-style-type: none"> - More accurate than standard model in predicting round jets and complex secondary flows in studies associated with ACB thermal comfort applications - More computational power required than standard model
[128,131]	Large Eddy Simulation (LES) Model [132]	<ul style="list-style-type: none"> - More accurate than all RANS models to investigate external and internal airflows of an ACB - Better for predicting jet width and the jet bending caused by the induction in ACB study - Requires high computational time and cost

and applicability for a wide range of flows in ACB studies. This model is found to be suitable for observing the internal flow of ACB models with less computational time. The comparative analysis from the literature in Table 5 shows that the k- ω -based turbulence models [134] offer better performance than k- ϵ models regarding free shear flows and airflows near the walls. The k- ω SST turbulence model [122] provides detailed characteristics of flow patterns and combines advantages from the k-epsilon and k- ω models [135]. However, the implementation of the required turbulence model is also dependent on the computational facility; therefore, inconsistency is observed in the literature for choosing any specific model for the given ACB application. Other 2D and 3D modelling techniques (using different tools) are also used to predict the fluid-dynamical and thermal performance of complex ACBs [72,136]. However, the simulation software available today has multiple issues when it comes to modelling ACBs, specifically with respect to their accuracy and ease of designing for real-time building applications [41].

Field tests

Generally, the cooling capacities of ACBs are tested and rated by European Standard EN/DS 15115 approved by European Committee for Standardization (CEN) [137]. CEN specifies methods for measuring the cooling capacities of chilled beams with forced airflow. The purpose of the standard is to give comparable and repeatable product data that can be further used for conducting laboratory studies. Most of the experimental studies on ACBs are done in climate chambers or mock-up office setups, see Table 3. In ACB studies, the simulated data obtained through simulations is validated with the field data. Literature shows that during ACB field tests, air change rates, IRs, and IAQ are measured using the tracer-gas method [26,68,82]. Particle image velocimetry (PIV) is another optical measurement technique that validates chilled beam models through field tests [79,135]. The velocity field of an entire testing region with the airflow can be measured through PIV. The flow region is then subdivided into small areas of investigation to make a detailed airflow analysis. As ACB systems involve complex internal geometry, Laser Doppler velocimetry (LDV) is used to observe the internal airflow inside ACBs [77]. In comparison to traditional flow measurement techniques, this optical technique makes a very high spatial resolution possible.

Fig. 14 shows the velocity vector profile of an office setup equipped with ACB by using CFD and PIV techniques. PIV technique is commonly applied for observing jet visualisation during ACB experiments, while CFD tools are applied for the prediction of air distributions in different ventilated spaces.

Energy use and energy savings

Different design strategies are proposed in the literature to minimise the energy use in buildings through modifying conventional HVAC technologies [138–140]. Most of the research on chilled beams is focused on techniques to enhance the energy efficiency and performance of these space friendly ACB terminal units. ACB systems are considered one of buildings' reliable technical solutions due to their energy-saving potential [141]. Below are the reasons why ACB has helped to save energy in buildings.

- One of the main reasons ACBs provide energy savings is because they supply sensible cooling directly to the occupied zones, which ultimately reduces the ventilation fan power consumed to deliver cooling to spaces [16].
- Chilled water flow at higher temperatures (14 °C to 18 °C) in ACBs compared to conventional chillers (4 °C to 7 °C) makes chilled beams up to 20% more efficient (in cooling) than conventional air-conditioning systems [21].

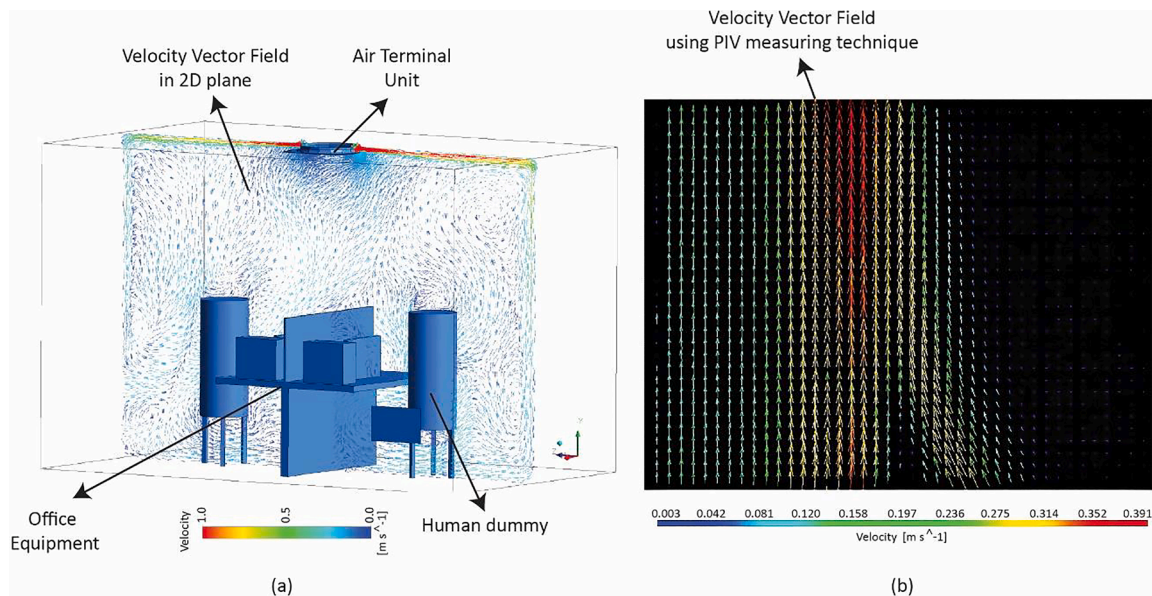


Fig. 14. (a) Velocity vector field in a 3D office room (b) Velocity vector field visualisation through PIV technique [135].

- Water has a higher specific heat than air and can transport a greater amount of energy through the building. This makes ACBs more energy-efficient than all-air systems.
- Built-in precision quality of Dedicated Outdoor Air System (DOAS) helps ACB systems (when coupled together) to provide less airflow to the individual zones in buildings [42]. This reduces the quantity of outdoor air (OA) to be conditioned while meeting Standard 62–2001 ventilation requirements and saves energy [142].
- The induction phenomenon in ACBs eliminates the need to reheat the cooled air, which saves a considerable amount of money and energy [56].
- Energy is saved by using an innovative 2-pipe system for simultaneous heating and cooling through ACB systems. However, a balance point temperature is required to save this annual primary energy [143].
- Chilled beam zone pump modules for controlling the pump speed (according to the requirement) can save energy or increase capacity [144].
- The energy performance of an ACB can be improved by increasing the air-to-water cooling capacity ratio of the system. This air and water-side cooling capacity can be increased by increasing the primary airflow rate and water inlet temperature [59].
- A self-regulating ACB requires less energy for heating and cooling than a system with individually controlled active chilled beams through the control of the chilled water supply temperature aiming at an exhaust air temperature set-point of 24 °C [145].

Thermal comfort is one of the main factors that influence occupants' productivity in office buildings [146]. ACB systems have overall proven successful in providing an acceptable thermal uniformity, even with less air flow rate, than other conventional air distribution systems [71,147]. Along with this, ACBs are integrated with other ventilation systems to further improve local thermal comfort and have more energy savings [148]. Sekhar et al. integrated personalised ventilation (PV) and local fan-induced active chilled beam (PV-ACB) air conditioning systems to achieve 16% energy savings (compared with a conventional VAV system) at 100% peak cooling load [149]. Demand-controlled ventilation (DCV) systems with ACBs, operating at optimal controls, use 7–8% less total primary energy compared to constant air volume (CAV) systems [50]. Kosonen et al. compared chilled beams and radiant panel systems for mock-up office setups and figured out that both the systems are

equally effective in regard to mean radiant temperature and radiant temperature asymmetry at different internal loads in both heating and cooling modes [150]. Wu et al. developed a model-based multi-objective optimisation strategy by combining energy models for the whole ACB system and a PPD model for the building zones [49]. This model controlled primary airflow rate, water flow rate, and zone temperatures to develop a trade-off between energy use and thermal comfort for ACBs. Different heat loads were compared with optimised cases under steady-state experimental conditions. The proposed model provided energy savings of 39.32% with an improved thermal comfort environment. Rahimi et al. used a combination of ACBs and air-cleaning technologies to improve the indoor climate in offices but reported a 38% decrease in efficiency of the chilled beam in exchanging heat [27].

Discussion and future research

As shown in this paper, ACB systems have lots of advantages over conventional HVAC systems in terms of better thermal comfort and energy savings. However, several topics still remain to be discussed in order to promote a healthy dialogue between the HVAC research community and field engineers for long-term future research. Some of these points are discussed as follows:

- Despite the valuable studies on ACB air distribution patterns, most of the studies have been carried out in climate chambers or with simulation tools, which lack realism. Much work needs to be done to evaluate ACB performance and energy savings under real conditions, and this can be done through field tests and by receiving feedback from end-users.
- Most of the authors highlighted the conventional disadvantage of condensation forming on the coils of chilled beams. But no practical solution is addressed in detail. Some recommended having humidity controls within the occupied space to prevent condensation, and others proposed using drainage pans, which does not seem practical and would further increase the cost of the units. ACBs are located inside ceilings, where a separate pump for each beam would be required to drain them. Condensation prevention strategies should be considered as a part of designing ACB systems, which can function well both for sensible and latent cooling modes.
- No known experimental study was found on ACB heating capacities for the office mock-ups. Minimal literature is found to address

thermal comfort by ACB devices in the heating mode. In most of the experiments, the room temperature is raised by using heaters or radiant heating panels. Practically a separate heating system is used in offices equipped with ACBs, which likely will increase the overall cost of building a HVAC system.

- ACBs are not that energy efficient when it comes to dealing with high sensible cooling requirements in large office spaces. Their commercial applications are restricted mainly to offices, school labs and rooms with low ceilings. However, different design strategies are required for ACBs in case of latent load requirements for large office spaces.
- The efficiency of other sustainable energy sources in relation to ACB systems should be investigated. Geothermal heat pumps, solar panels and phase change materials (PCM) seem like favourable technologies to further reduce the primary energy use of these systems.

Conclusion

Over the years, ACBs have become an alternative to conventional HVAC systems in offices due to their advantages in terms of energy and space savings, economy, and thermal comfort. These systems work more efficiently in the cooling modes than heating due to the significant risk of discomfort because of radiant temperature asymmetry. In the cooling mode, operating conditions of the system and humidity levels are controlled to avoid condensation. Literature shows that the suitable geometrical design along with optimal operating conditions for ACBs play a key role in establishing desired thermal comfort for offices while minimizing energy use. Most of the ACB studies are carried out in climate chambers installed with 2-Way ACB designs, where PIV visualisation techniques are applied in experiments to observe 3D airflow patterns and refine CFD models. In addition, studies show that cross-shaped nozzles give high induction rates than conventional round-shaped nozzles, whereas the size of the ACB nozzles has a more significant impact on their efficiency than the nozzle shape. In terms of modelling, hybrid and dynamic cooling coil thermal models are preferred in applied research for ACB cooling coils. This is due to their ability to cover wide thermal aspects close to real conditions and because they have better robustness than static cooling coil models. The use of the filters for cleaning the induced air, placed between the induction grill and cooling coil, is not encouraged in ACB office applications to have low-pressure drop and high induction. Regarding the control of ACB systems, the water-side and air-side controls are used to achieve room temperature control by regulating the water flow rate and variable-volume primary airflow, respectively. However, self-regulation in ACB systems even effectively controls room temperature without the need for any control device or changing water flow rates. Literature shows that energy savings of between 25 and 30% can be achieved from ACBs compared to traditional VAV systems. These energy savings are mainly due to increased chilled water temperature, reduction in reheating cooled air and fan energy, and integration with the other HVAC systems.

CRediT authorship contribution statement

Haider Latif: Conceptualization, Data Curation, Formal analysis, Investigation, Writing - original draft. **Goran Hultmark:** Supervision, Visualisation, validation. **Samira Rahnama:** Supervision, Writing - review & editing. **Alessandro Maccarini:** Supervision, Writing - review & editing. **Alireza Afshari:** Supervision, Funding acquisition, Resources.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Chapter 8.

PAPER V

Precision Ventilation in an Open-Plan Office: A New Application of Active Chilled Beam (ACB) with a JetCone Feature

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Article

Precision Ventilation in an Open-Plan Office: A New Application of Active Chilled Beam (ACB) with a JetCone Feature

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Abstract: Mixing ventilation systems effectively improves thermal comfort in open-spaces due to adequate turbulent mixing of the cold stream with ambient air. This study introduces the concept of precision ventilation for achieving local thermal comfort in a mixing ventilation system. This precision ventilation system provides asymmetrical airflows from an active chilled beam (ACB) to each of the office occupants. These ACBs provide air velocities with different magnitudes and directions. To achieve different magnitudes and directions, JetCones are used to vary the airflow in different parts of the ACB. The performance of the precision ventilation system was analyzed using full-scale laboratory experiments and computational fluid dynamic (CFD) simulations. The full-scale laboratory experiments were conducted in a 4.2 m × 3 m × 2.8 m (L × W × H) thermal isolated room with an open-plan dual desk-chair setup. The jet-cones in the ACB unit were adjusted to throw the required amount of flow to the occupants. The occupants had different metabolic rates of 1.2, 1.4, and 1.6 in a warm office space. The room set point temperatures varied between 23 and 26 °C. The experimental and CFD results show that occupants facing symmetrical airflow distribution and with a constant 1.2 metabolic rate had a similar PMV index. The occupants with 1.2, 1.4, and 1.6 metabolic rate were exposed to asymmetrical airflows, i.e., 30%, 58%, and 70% of the total airflow. Occupants with higher metabolic rates were kept thermally neutral, in the −0.5 to +0.5 PMV range, by increasing the air velocity and room temperature to 0.4 m/s and 25 °C, respectively.

Keywords: active chilled beams; asymmetrical airflows; JetCones; metabolic rates; precision ventilation

1. Introduction

The increasing demand for HVAC in buildings has led researchers towards technological advancements in order to provide desirable thermal comfort for building occupants. Currently, people spend 80–90% of their time indoors with an increasing reliance on air conditioning and mechanical ventilation (ACMV) systems to have a comfortable indoor environment [1]. Similarly, people spend on average 40 h per week in their offices [2]. This dependency has led many researchers to bring greater advancements to existing ACMV systems, which can provide desirable indoor climate solutions for office buildings.

Mixing ventilation [3] is one of the most used ventilation systems for providing space cooling. It involves moving cold air from ceiling level to the floor, driven by momentum and density difference. The falling jets are mixed with ambient air and this establishes a uniform temperature level with low temperature stratification. Active chilled beams (ACBs) are a typical example of mixing ventilation systems that are fixed in the ceiling to make use of the Coandă effect to distribute room air. According to the definition, a Coandă effect is the tendency to entrain fluid from the surroundings so that a region of lower static pressure develops to have the free jet attached to the ceiling [4,5]. This effect is one of the

prominent feature of the ACBs. Rhee et al. concluded that ACBs are considered successful in providing acceptable thermal uniformity compared to other conventional air distribution systems, e.g., displacement ventilation [6].

Thermal comfort is defined as the state of mind that expresses satisfaction with the thermal environment [7]. The literature shows that thermal comfort preferences may vary among office occupants due to a change in metabolic rate, gender, age, and health [8]. Therefore, building occupants might perceive thermal comfort differently, even when they are exposed to the same thermal environment [9]. Individual thermal comfort is an issue that needs to be addressed, which involves personalized control of the micro-climate around the individual according to their needs. Figure 1 shows six factors that affect the thermal comfort of any individual [10].

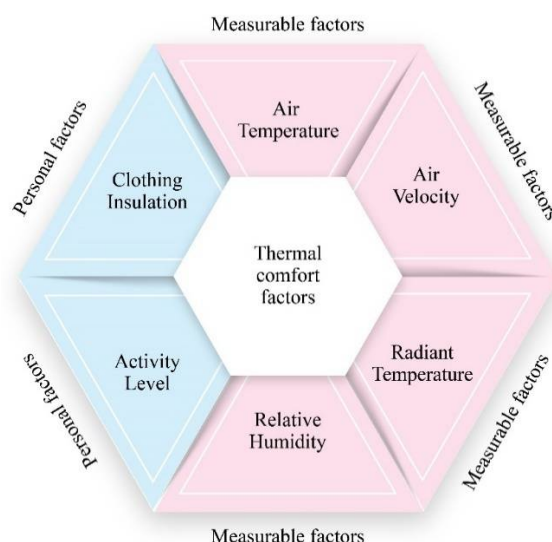


Figure 1. Thermal Comfort Parameters.

The human body has a constant internal temperature and the thermoregulatory system of the body tries to maintain this, despite a wide range of environmental factors [11]. Any imbalance due to inappropriate thermal conditions may adversely affect work productivity in a working environment [12,13]. In addition, the inability to provide individual thermal comfort is considered one of the drawbacks of conventional ACMV systems [14]. An alternative to mixing ventilation system, the personalized ventilation (PV) systems (also known as task ambient systems) [15], have been studied for the past two decades to meet individual thermal comfort requirements in offices. These systems make use of multiple air terminal devices (ATDs) on office workstations to build micro-climate zones. Despite its advantages, PV applications are limited in the HVAC market due to use of multiple ATDs, which appropriately does not fit building aesthetics. Secondly, direct airflow from ATDs to the human body may cause risk of draft or irritation [16]. PV systems coupled with mixing ventilation systems have been studied in order to improve individual thermal comfort [17], but efficiency of these hybrid systems is dependent on the location of multiple ATDs (used in PV systems) to promote mixing [18].

By directing airflow from an ACB outlet that moves along the ceiling towards different zones of a single space, this study presents a mixing ventilation system in a different way. In the past, ACBs used to maintain uniformity of air distribution in rooms have been studied [19,20], but these systems have not been used for individual thermal comfort applications. The aim of this study is to present efficient air mixing with optimal air speeds around occupants to fulfill individual thermal comfort requirements, according to different metabolic rates of the occupants. To enable targeted cooling by varying the air velocity's magnitude and direction through adjustment pins, see Figure 2. This precision ventilation system uses ACB units with a JetCone feature [21].

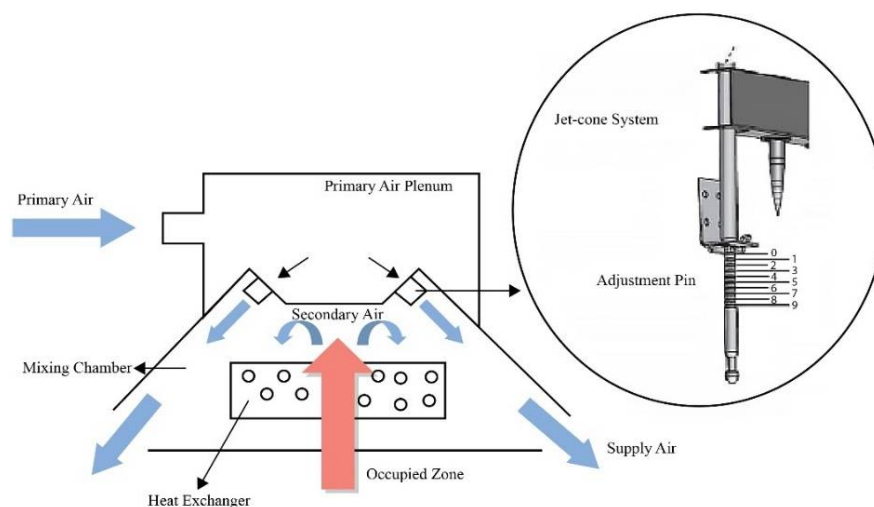


Figure 2. ACB Unit with JetCone system.

Zhang et al. suggested that the room temperatures are often raised for indoor climate studies to save energy [22]. The thermal comfort for these elevated room temperatures can be maintained by increasing the room air velocities. Therefore, this experimental simulation study is conducted for a warm office environment by creating high and low velocity zones around occupants to achieve individual thermal comfort.

2. Experimental Setup

2.1. ACB with JetCones

In the study, the ACB unit with angled nozzles was used, as shown in Figure 3. The plexus was tested and rated according to Danish Standards Foundation DS/EN 15116 [23]. The JetCone feature in the plexus provided control over the primary airflow in the nozzles and pressure in the air inlet. In addition, the nozzles are directed to create a 360° airflow pattern. Four adjustment regulators at each corner gives the ability to adjust the airflow from the ACB to the different parts of the room. These adjustment regulators provide 10 different positions each, which gives 40 different settings to control air diffusion, supply air volume, and pressure.

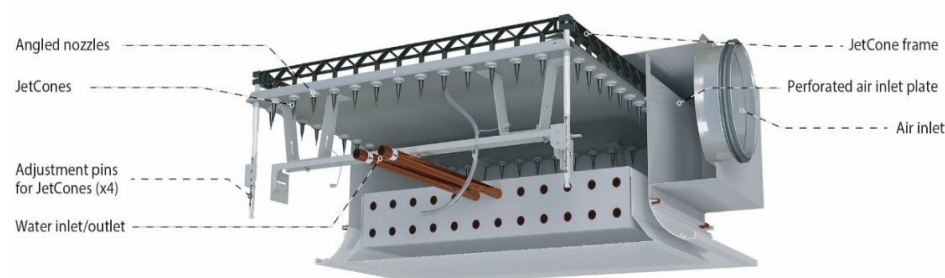


Figure 3. Schematic drawing of an ACB unit with JetCones [24].

The asymmetrical airflow patterns were achieved by adjusting regulator pins from positions 0 to 9. Position 0 of the adjustment regulator allowed minimum airflow coming out of the ACB outlet, whereas pin position 9 allowed maximum airflow to pass through the specific side of the beam outlet. Table 1 shows the complete features of the plexus used in this study.

Table 1. ACB-system details.

Units	Values
ACB Dimensions (L × W × H)	0.6 m × 0.6 m × 0.2 m
ACB Unit	1
Functions	Cooling, Heating & Ventilation
Operating System	Cooling 2-pipe system
Distribution profile	Radial
Capacity	769 W
Primary airflow rate	20 L/s (fixed)
Supply Air temperature	21 °C

The plexus has a standard Ø12 mm water pipe and Ø125 mm horizontal air duct which can be used for cooling, heating, and ventilation applications. This study was limited only to individual comfort through cooling.

2.2. ACB Airflow Distribution

The discharge velocities at 16 different positions of the ACB unit openings were measured to analyze the initial data and airflow distribution. The airflow on each side of the plexus opening was different due to radial discharge and different JetCone settings. The 16 equally spaced velocities at the beam outlet were measured to observe the airflow patterns with three different JetCone settings. Initially, three JetCone settings were measured under non-isothermal conditions (see Figure 4). These include:

- 1- All four-pins at position 5 to have uniform air distribution.
- 2- Two-pins at position 5 and two at position 0 to have more primary flow pushed towards one half of the plexus.
- 3- Two-pins at position 9 and two at position 0 to have maximum discharge on one side and minimal discharge from the remaining beam half.

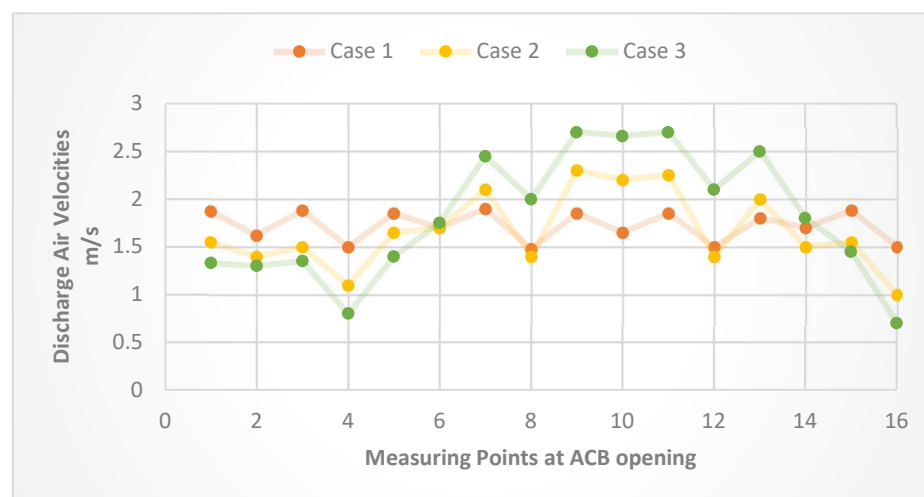
**Figure 4.** Discharge Velocities measured at ACB opening with different ACB pin-settings.

Figure 4 shows that the discharge velocities in case 1 are uniform due to the same pin settings, whereas case 2 and case 3 have maximum discharge velocities in-between points 6 and 14, as this is where JetCones are partially or fully opened. Figure 5 shows the airflow distribution of all three cases. In case 1, the discharge velocities show symmetrical airflow distribution from the JetCones. This means 50% of the total airflow is discharged from each half of the plexus. In case 2, the discharge velocities measured had 58% and 42% of airflow division for the pin positions at 5 and 0. In case 3, the airflow from the sides of the beam outlet with pin position 9 had greater airflow than the sides on position 0. The magnitude of velocities for the third case shows that JetCones opened at position 9 discharge (maximum)

70% of the airflow, while the ones with the pin position at 0 only allow 30% of the airflow from the beam outlet. However, the amount of primary airflow is kept constant.

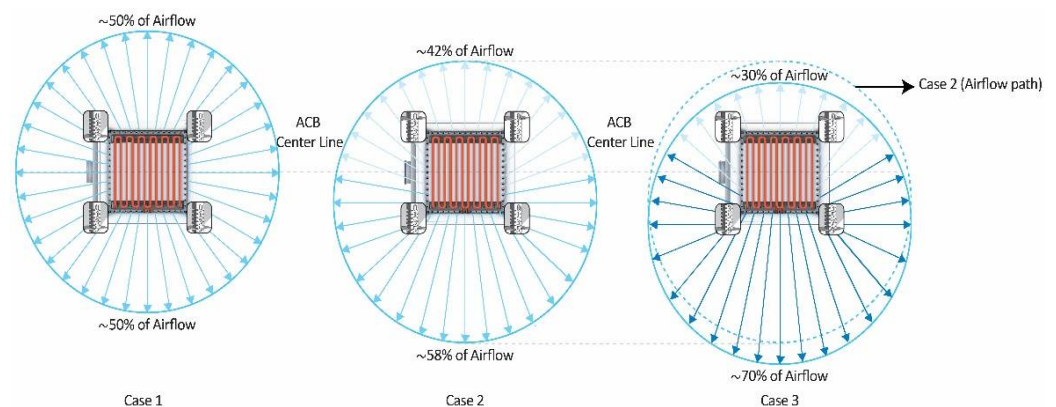


Figure 5. Airflow distribution (Cases 1, 2, and 3).

Table 2 shows the flow division, at a constant primary airflow rate of 20 l/s, based on nozzle pressures set according to the pin positions. The full-scale experimental study for targeted air distribution is applied for the three cases. The measured discharge air velocities were used as inlet velocities for CFD boundary conditions.

Table 2. Pin position properties.

Cases	Adjustment Pin Settings	Static Nozzle Pressure Loss Δp_{stat} (Pa) (Manufacturer's Data)	Airflow Division (Measured) (%)
Case 1	5 + 5 + 5 + 5	75	50/50
Case 2	0 + 0 + 5 + 5	300 (Estimated)	58/42
Case 3	0 + 0 + 9 + 9	80	70/30

2.3. Test Room

The full-scale laboratory experiments were conducted in a room with dimensions 4.2 m \times 3 m \times 2.8 m (L \times W \times H) at Aalborg University Copenhagen. The experiments were carried out with cooling power of ACB balanced with heat sources in the room. The room walls, ceiling, and floor were kept adiabatic without the influence of any solar sources. The open-plan office layout of the test room with two workstations was chosen due to its common use for conducting experiments on ACBs [25]. The ACB was installed in the middle of the ceiling, i.e., 1.8 m away from the two sides (north and south direction), as shown in Figure 6. Table 3 shows the test room parameters set during the experiments.

Table 3. Test room details.

Parameters	Values
Office area	12 m ²
Occupant density	6.0 m ² /person (2 persons)
Office Equipment	2 workstations (2 dummies, Two computers, lights)
Set Room Temperatures	23–26 °C
Occupants (Sensible heat)	65 W/m ² , 80 W/m ² and 95 W/m ² per dummy
Total Average Zone Load	510 Watts

The position of the two dummies (each 1.4 m tall) were 1.85 m apart so that their thermal flux had a negligible effect on each other. These dummies are separated by open-plan 40 cm \times 75 cm desks (L \times W) under the plexus as shown in Figure 6. The human heat source was replicated by placing electric bulbs inside the dummies. The metabolic

rate of each dummy was varied by changing the number and capacity of the electric bulbs installed inside. Other heat sources were computers and lamps (see Appendix A).



Figure 6. Office setup.

The air velocity measurements in the test room were carried out by locating sensors along the length of the room. For all three cases, the measuring points were set in the occupied space to observe airflow distribution inside the room. The air temperature and velocity distributions were measured on points such that heat flux from the dummy did not affect the measurements. The measurements were taken during summer (May–October). The room temperature varied between 23 °C and 26 °C. The measurements were taken at 20 different points at 0.1 and 1.1 m above the floor located at two different zones (see Figure 7). Zone 2 towards the north wall was exposed to high air velocities for cases 2 and 3. The measuring points in both zones were equally distributed such that the airflow distribution in each case could be evaluated.

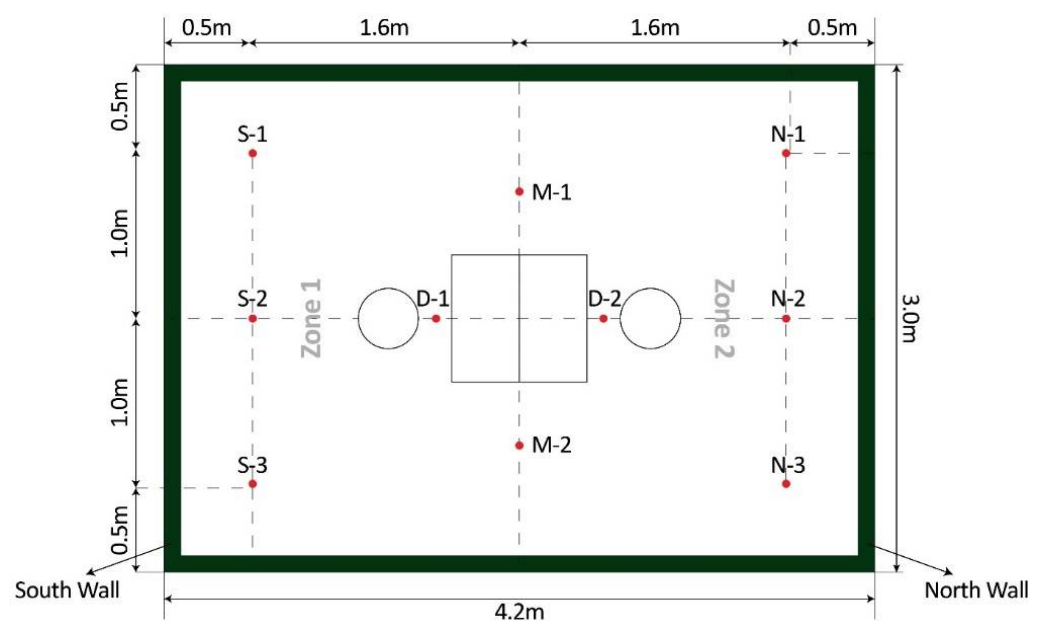


Figure 7. Plan view of test room with the measuring points.

N and S points in Figure 7 were 0.5 m from the north and south walls, respectively. M1 and M2 points in the middle of the room were taken to trace the velocity magnitude. The velocity and temperature data at D1 and D2 points (located in zone 1 and 2, respectively) were used for thermal comfort measurements of the human dummy at different metabolic rates.

2.4. Thermal Comfort Criteria

In the current study, predicted mean vote (PMV) and predicted percentage dissatisfied (PPD) indices are used for evaluation of thermal comfort in an enclosed environment [26]. The expressions of Fanger's PMV and PPD models are given by the Equations (1) and (2).

$$\begin{aligned} \text{PMV} = & (0.303e^{-0.036M} + 0.028) \{ (M - W) - 3.05 \times 10^{-3} [5733 - 6.99(M - W) - p_a] - \\ & 0.42 [(M - W) - 58.15] - 1.7 \times 10^{-5} M (5867 - p_a) - 0.0014 M (34 - T_a) - 3.96 \times \\ & 10^{-8} f_{cl} [(T_{cl} + 273)^4 - (T_r + 273)^4] - f_{cl} h_c (T_{cl} - T_a) \}, \end{aligned} \quad (1)$$

$$\text{PPD} = 100 - 95 \exp [- (0.03353 \text{PMV}^4 + 0.2179 \text{PMV}^2)] \quad (2)$$

PMV was calculated based on mean values of the local air temperature (T_a), mean radiant temperature (T_r), and local air velocity (V_r). As it is generally very close to air temperature in most cases, the mean radiant temperature was assumed equal to air temperature [27]. Equation (1) shows M as the metabolic rate which varies according to activity level, whereas the effective mechanical power (W) is assumed zero. The clothing surface area (f_{cl}) and the convective heat transfer coefficient (h_c) were calculated iteratively. The clothing value is given as 0.6 clo (for summers) due to changing clothing habits of office occupants' w.r.t seasons [28]. The activity level in offices may differ between individuals. Some stay simply seated (relaxed) to getting involved in sedentary or light office activities. The metabolic rates were assumed to be 1.2 met (65 W m^{-2}) to 1.6 (95 W m^{-2}) met per occupant to have different activity levels (heat release) [29]. The heat balance was established by creating high air velocities zones at high room temperatures to maintain individual thermal comfort for occupants with increased metabolic rates. In the present study, draught rate (DR) is not considered due to the need for higher air velocities for the occupants with high metabolic rates at room air temperatures up to 26°C . High air velocities are created around the dummies to maintain body heat balance (with respect to the activity level) and PMV values within the acceptable thermal comfort range. Vertical velocity and temperature measurements near the human dummy were measured by two moveable poles attached with Dantec hot sphere anemometers at steady conditions (see Figure 8). The testing area for PMV measurements is set at the vicinity of the dummy as a vertical line in zones one and two. These parameters were measured at three heights equal to head level (1.1 m), abdomen (0.6 m), and ankle (0.1 m) from the floor of a seated person [30]. The dummies (located in two different zones) with variable metabolic rates exposed to different air velocities (ranging from 0.1 m/s to 0.4 m/s) were used for PMV-PPD measurements. The acceptable thermal comfort range between $-0.5 < \text{PMV} < 0.5$ was considered appropriate [26]. The room air velocity was measured using Dantec hot sphere anemometers with an absolute accuracy of $\pm 2\%$ of the reading (between the range of 0 and 1 m/s) [31]. During the measurements, the average velocity data were collected with a sample time of 180 s according to the requirement for the measurement of the indoor air velocity in EN 13182:2002 [32].

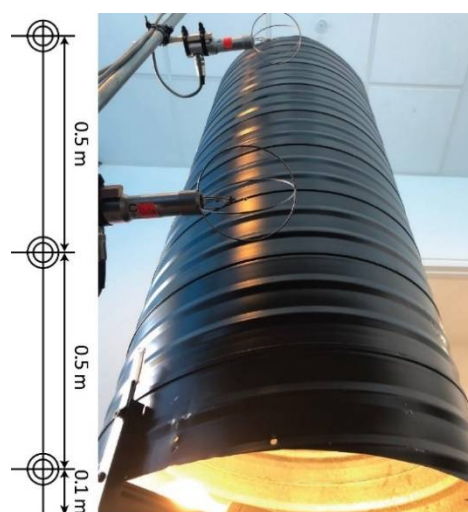


Figure 8. Vertical measuring points with hot sphere anemometers along the dummy.

3. Simulation Model

The geometry of the ACB unit was made circular and simplified to reproduce radial distribution pattern (see Figure 9a). The inlet faces were filleted with 0.03 m radius to have discharge flow parallel to the ceiling surface. The total area of the inlet opening was 0.04 m². The opening of the ACB was divided into 16 equal divisions to have variable velocity spread along the ceiling with respect to JetCone settings. The airflow from each quarter of the round ACB outlet was controlled by one adjustment pin (Figure 9b). The geometry of the ACB was reduced to only the bottom part for simplicity and the magnitude of airflow and temperature of the inlet surfaces were set as a resultant of both primary and secondary airflows.

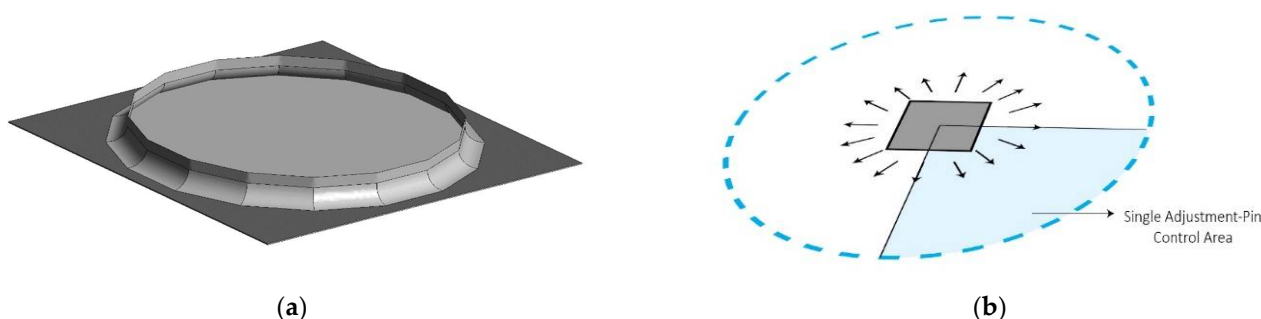


Figure 9. (a) ACB terminal unit geometry (b) Radial flow distribution with pin controls.

The geometry of the room with the actual dimensions was made on the SolidWorks software. The geometry consisted of two fully equipped workstations positioned centrally, along with a radial geometric ACB unit fitted at ceiling level, as shown in Figure 10. The outlet was positioned on one of the walls, next to the workstation towards the north side. The heat loads used in the measurements comprised computers with a human dummy (Appendix A). The cylindrical dummy was applied with the thermal heat flux according to different metabolic rates.

The SolidWorks geometric model was imported to ANSYS FLUENT 17.1 version for the CFD simulations. Tetrahedral meshes were generated in the entire computational domain with fine local meshing done near the critical areas such as ACB supply inlets and all heat sources (Appendix A). The pressure-based solver was used for the simulations under steady state conditions. The RNG k- ϵ turbulence model was selected for the simulations due to better accuracy than other RANS models for indoor airflow simulations [33–35] and less computational cost than LES models [36,37]. Boundary conditions for the velocity inlets of all three cases were used based on the experimental data collected from preliminary

ACB measurements (see Figure 4). The total inlet flow coming out of the inlet opening was 60 L/s (sum of primary and secondary airflows), distributed according to the pin positions (see Figure 5). The convective heat fluxes (see Table 3) were applied to describe the boundary condition of the human body according to metabolic rates of three cases. Other heat sources were also given thermal heat fluxes (Appendix A). The SIMPLE numerical algorithm was selected for coupling pressure and momentum equations and the criteria for convergence was set 10^{-6} [38].

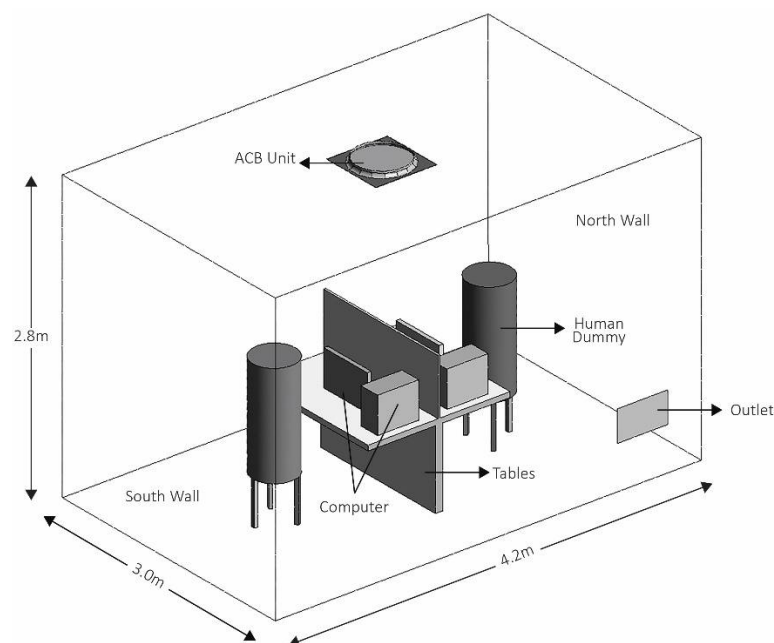


Figure 10. Test room model.

4. Results and Discussion

4.1. Velocity Distribution

Velocity measurements were taken after attaining the steady state conditions inside the room. Measurements were taken at the locations shown in Figure 7 and two different heights (0.1 m and 1.1 m above the floor) to observe velocity distribution. Table 4 shows the average and maximum air velocities for all three cases.

Table 4. Maximum/Average Air Velocity distribution measured in the room.

Maximum/Average Air Velocity	Case 1	Case 2		Case 3	
	Average Velocity	Low Velocity End	High Velocity End	Low Velocity End	High Velocity End
0.1 m	0.14	0.15/0.12	0.28/0.23	0.11/0.08	0.4/0.35
1.1 m	0.18	0.14/0.11	0.25/0.21	0.1/0.09	0.34/0.32

There were no high and low velocity zones for case 1, so only average room air velocities were considered at the measuring points. Case 2 and 3 shows considerable high air velocities towards the north wall of the room due to greater directed airflow. Velocities at 1.1 m above the floor region were found to be slightly higher than at the 0.1 m region. Maximum velocities up to 0.3 and 0.4 m/s were achieved in cases 2 and 3, respectively, whereas uniformity in air distribution was observed in case 1, with an average velocity of 0.16 m/s across the room.

Horizontal velocity profiles in Figures 11–13 show that the velocities from the north and south walls were found to decrease towards the middle of the room. The air velocities at the middle of the room, at points M1 and M2 were seen <0.1 m/s due to minimal directed

flow towards these points. CFD showed similar results of the experiments with a variation of +10% (Appendix A). Figure 14 shows vertical air velocity and temperature points (D1 and D2) for the three cases in the occupied zone.

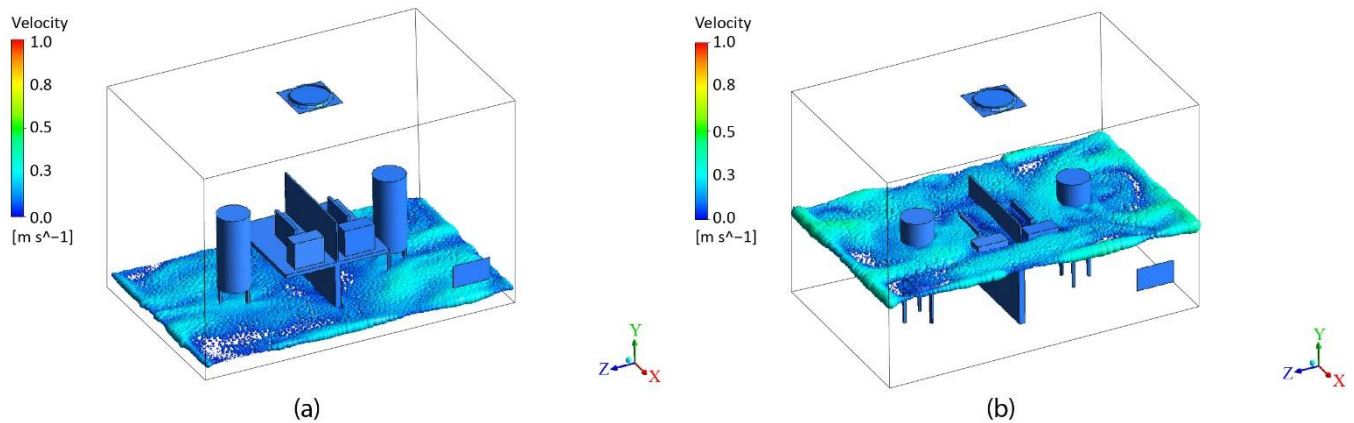


Figure 11. Case 1 Air Velocity Distribution (a) 0.1 m from the floor (b) 1.1 m from the floor.

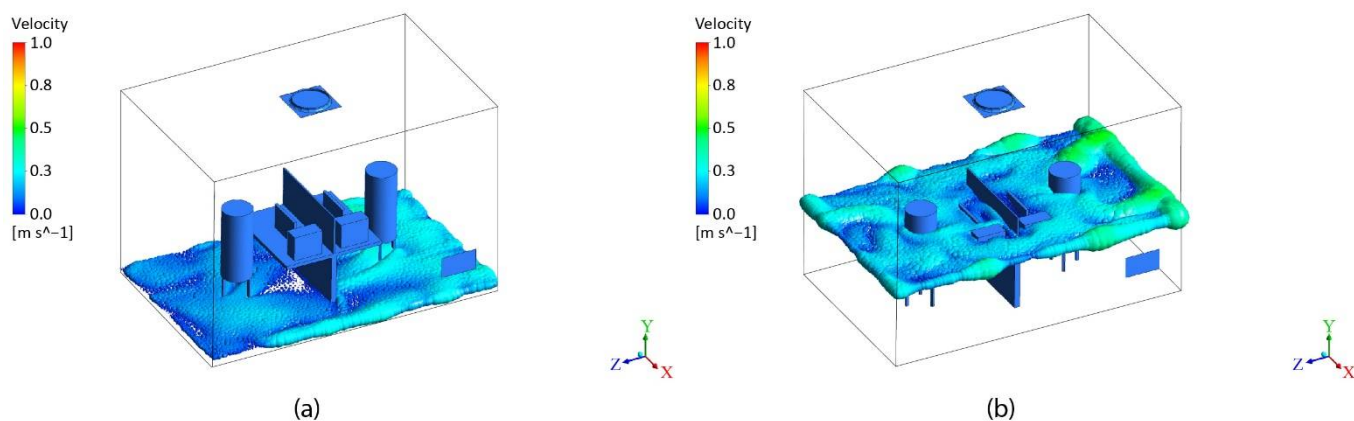


Figure 12. Case 2 Air Velocity Distribution (a) 0.1 m from the floor (b) 1.1 m from the floor.

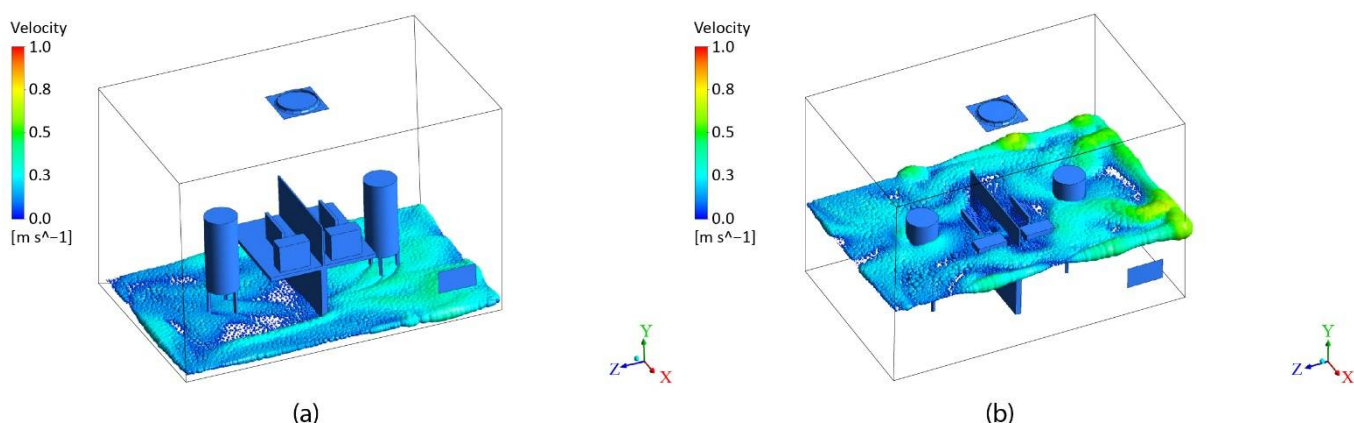


Figure 13. Case 3 Air Velocity Distribution (a) 0.1 m from the floor (b) 1.1 m from the floor.

During the measurements, the local air velocities near the head were found slightly higher than the ankle region. Figure 14 shows that the vertical temperature difference in the occupied zone between the height of the ankle and head was found maximum up to 1 °C. This temperature difference (less than 3 °C) is considered acceptable according to the ISO 7730 standard [26]. The horizontal temperature difference (for all the cases)

between high and low velocity zones was also found within $\pm 1^\circ\text{C}$ range. However, the air temperature stratification was 0.5°C in the center of the test room from the height of 1.0 to 2.8 m from the floor. The rise of temperature in the room was not seen to influence the velocity distribution around occupants.

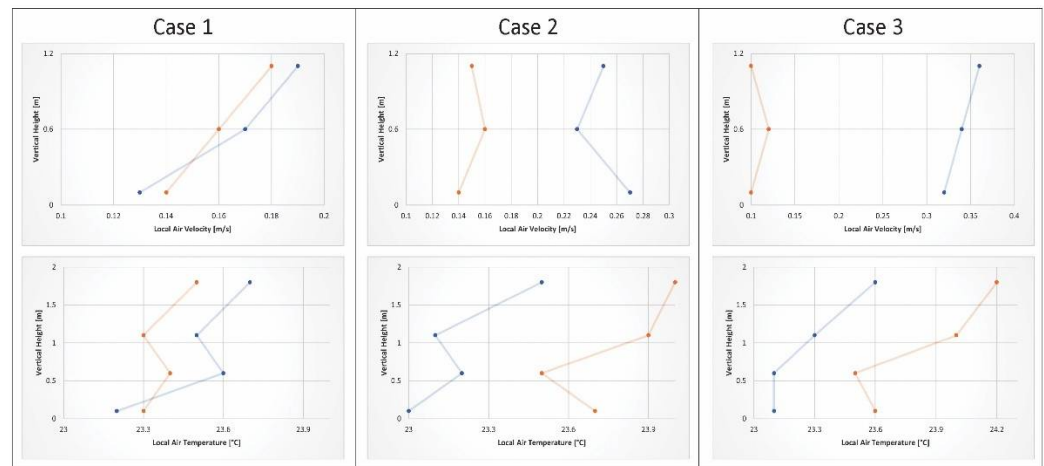


Figure 14. Vertical Local Air Velocity and Temperature distribution in the Occupied Zone.

4.2. Thermal Comfort Calculations

Thermal comfort calculations are made based on measurements in the room. The increase in local air velocity near thermal dummies results in higher convective heat transfer [39]. This change in this environmental variable is required to preserve the heat balance between human body and the surrounding environment. The metabolic rates are kept at these combinations, 1.2 met, 1.4 met, and 1.6 met for the three cases at constant Relative Humidity (RH). Case 1 included both the dummies with 1.2 metabolic rates. The metabolic rates for only dummy 2 (zone 2) in cases 2 and 3 were increased to 1.4 and 1.6, respectively. The thermal comfort for each occupant is evaluated in the mock-up office room using PMV and PPD indices (see Tables 5–7). The PMV and PPD values of two persons with a height of a seated person are determined by taking mean values of air temperature, radiant temperature, and air velocity at locations D1 and D2 (in zones 1 and 2) (see Figure 7). The mean values were taken at heights 0.1 m, 0.6 m, and 1.1 m.

Table 5. Case 1 Measured temperature, RH and Local Air velocities, and calculated PMV & PPD.

Input Data (Dummy 1 and Dummy 2)			PMV		PPD (%)
Ta = Tr (°C)	RH (%)	Vr (m/s)	Dummy 1 (met. 1.2) in Zone 1	Dummy 2 (met. 1.2) in Zone 2	
23	60	0.14	−0.61		12.7
24			−0.29		6.7
25			0.04		5
26			0.36		7.7

Table 6. Case 2 Measured temperature, RH and Local Air velocities, and calculated PMV & PPD.

Ta = Tr (°C)	Dummy 1: Input Data		PMV	PPD (%)	Dummy 2: Input Data		PMV	PPD (%)
	RH (%)	Vr (m/s)	Dummy 1 at Low Velocity Zone (met. 1.2)		RH (%)	Vr (m/s)	Dummy 2 at High Velocity Zone (met. 1.4)	
23	60	0.12	−0.54	11.2	60	0.25	−0.23	6.1
24			−0.23	6.1			0.05	5
25			0.09	5.2			0.32	7.1
26			0.41	8.4			0.6	12.5

Table 7. Case 3 Measured temperature, RH and Local Air velocities, and calculated PMV & PPD.

Ta = Tr (°C)	Dummy 1: Input Data		PMV	PPD (%)	Dummy 2: Input Data		PMV	PPD (%)
	RH (%)	Vr (m/s)	Dummy 1 at Low Velocity Zone (met. 1.2)		RH (%)	Vr (m/s)	Dummy 2 at High Velocity Zone (met. 1.6)	
23	60	0.10	−0.47	9.7	60	0.38	−0.01	5.0
24			−0.17	5.6			0.26	6.4
25			0.14	5.4			0.51	10.5
26			0.46	9.4			0.77	17.4

Table 5 shows PMV and PPD for case one where there is uniform distribution of airflow from the ACB with all JetCones at position 5. In order to comply with ASHRAE 55, the recommended thermal limit on PMV scale is between -0.5 and 0.5 [26]. The range of PMV for both Dummy 1 and Dummy 2 range from -0.61 to 0.36 with set point temperatures from 23 – 26 °C. The metabolic rate was also kept constant, i.e., 1.2 for both the occupants, representing same activity and health condition. The occupants feel slightly cool at 23 °C and almost thermally neutral and satisfied at 24 – 26 °C.

Table 6 shows PMV and PPD for case 2 where there is medium level flow (see Figure 5) directed towards Dummy 2 with met. 1.4 (high heat release) in the zone 2. On the other side, Dummy 1 with met. 1.2 in zone 1 is influenced by the remaining 42% of the flow. Results shows acceptable individual thermal comfort is reached for both the occupants due to targeted velocity distribution. Dummy 1 and Dummy 2 have PMV ranges -0.54 to 0.41 and -0.23 to 0.6 , respectively. Hence, the occupant with elevated metabolic rate gets the high air velocity to maintain the feeling of neutral sensation.

Table 7 shows PMV and PPD for normal and increased metabolic rate of occupants to validate the applications of ACB for individual thermal comfort. Results indicate occupants with 1.6 met maintained thermal sensation within the comfortable range of -0.5 and $+0.5$ [26,40]. However, Dummy 2 with a metabolic rate of 1.6 had PMV index exceeding $+0.5$ at air temperature of 26 °C. Hence, thermal comfort level was reached to a maximum temperature range of 25 °C with elevated air velocities. Further increases in air temperature leads to the feeling of being slightly warm for the occupant.

The variable supply velocities from a single ATD, i.e., ACB directed towards each dummy, resulted in variations in local air velocity around dummies, with percentage of dissatisfied (PPD) less than 15%, even with increased metabolic rates. Figure 15 shows that suitable velocities could act as a catalyst to maintain the thermal comfort for occupants with high metabolic rates to a certain temperature range. Linearity between increased metabolic rates and raised local air velocities is observed in Figure 15a–d. However, change of thermal sensation is seen shifting from left -0.5 range to $+0.5$ if there is no further increase of local air velocities. Therefore, an acceptable thermal comfort range, i.e., -0.5 to $+0.5$, can be maintained if we keep increasing air velocities with the increase of temperature. Zhai et al. [41] used multiple fans to systematically study air movement and comfort in warm-humid office environments. The authors included air velocity, temperature, and RH as controlled parameters and concluded that acceptable thermal comfort can be extended to 3 °C by raising air velocities up to 1.8 m/s and 80% RH %. Figure 14 shows that acceptable thermal comfort limit i.e., -0.5 and $+0.5$ for the occupants, can be extended up to 25 °C with air velocities up to 0.38 m/s. In this study, the ACB unit directs the fixed airflow towards one person (high metabolic rate) at a temperature range of 23 – 26 °C, whereas the temperatures above 25 °C require even higher airflow directed towards the occupant to maintain thermal comfort range. This may disturb the airflow division in ACB and may cause discomfort for the other occupant in the low velocity zone.

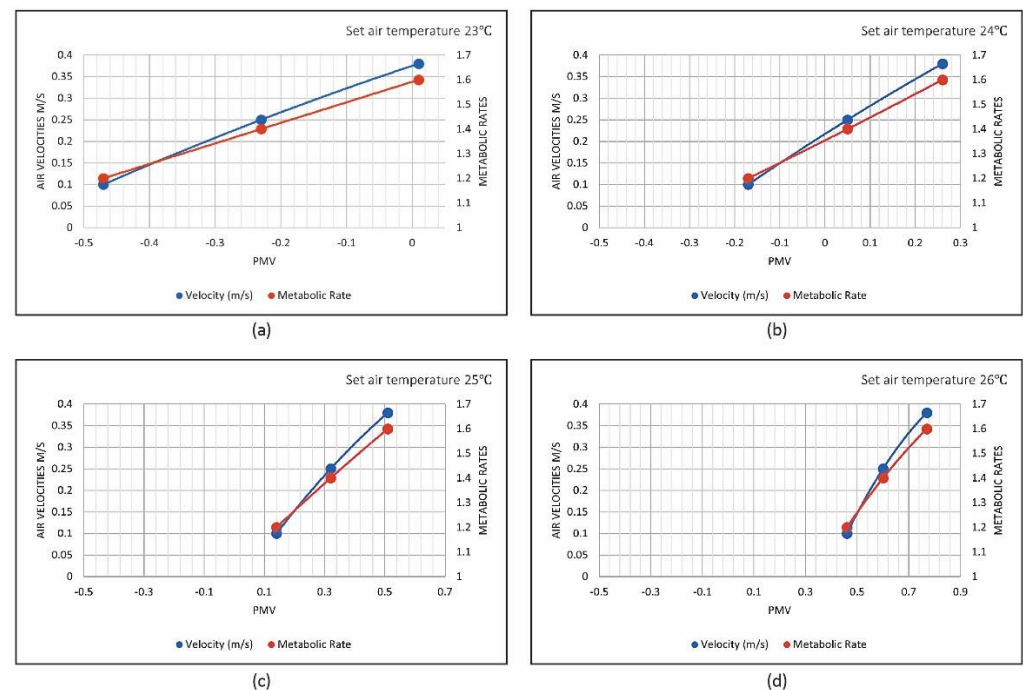


Figure 15. Impact of temperature on thermal comfort for occupants with different metabolic rates. (a–d): different occupants.

5. Conclusions

This study introduced mixing ventilation in a new way to fulfil individual thermal comfort needs for office occupants with a new precision ventilation system. This precision ventilation system takes the advantages of the mixing ventilation and personalized ventilation to provide individual thermal comfort through efficient air mixing by ACBs. Based on the experimental measurements and CFD simulations of the room with ACB, it can be concluded that ACBs with a JetCone feature can be used for individual thermal comfort applications. High air velocities for the office occupants with elevated metabolic rates and higher room temperatures can maintain acceptable thermal comfort levels. Variations in airflow rates by JetCones lead to significant changes in room air distribution patterns. Two airflow divisions in a room with a single ACB unit led to two different micro-climate velocity zones for two office occupants. The fully opened JetCones produced a longer throw to have higher air velocities up to 0.38 m/s around the occupant, while the decrease in throw through the adjustment pins produced velocities as low as 0.1 m/s around dummies.

The need to compensate different metabolic rates developed the need to establish high and low velocity zones around occupants. Results show that variations in air velocities for a single office zone, for two different occupants (varied metabolic rates), was done with precision by ACB with a JetCone feature. This application of ACB through different JetCones settings provides an innovative way of building micro-climate zones in a single office space through mixing and at the same time eliminates the need of multiple ATDs around the occupant.

6. Limitations and Further Work

This study is done in a small office setup with dual table-chair configuration. In further work, precision ventilation in larger scale offices with multiple ACBs where colliding jets may affect the room air distribution patterns will be studied. The colliding of jets can be used by directing the air to occupants in large scale open-plan offices in order to achieve individual thermal comfort.

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Appendix A

Mesh independence test was carried out at line $y_1 = -0.8 \cdot y_2 = -1.2$ in the middle of the room with different mesh densities. The variations after 1.7 million elements were not found significant and this mesh number was considered suitable for the calculation to avoid computational cost (see Figure A1).

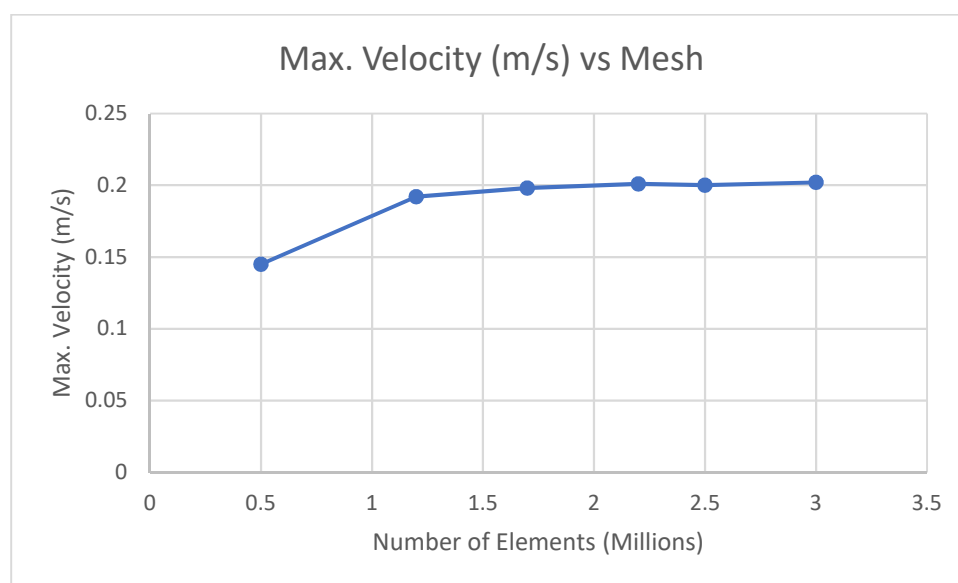


Figure A1. Mesh Independence Test.

The mesh metrics on the horizontal axis in Figure A2 show element quality, where 1 is a perfectly shaped tetrahedral element. Figure A2 shows that most of the elements had a metric range between 0.75 and 0.9. This means that the element quality of the obtained mesh structure is acceptable for carrying out stable numerical computation.

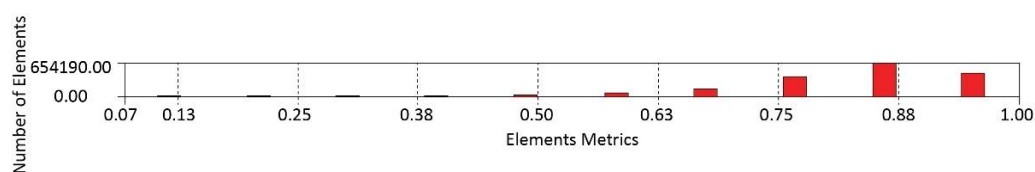


Figure A2. Mesh Element Quality.

The boundary condition in Table A1 were selected with respect to the actual experimental setup used for the measurements. The simulations used the second-order upwind scheme for all the variables (Table A2).

Table A1. Boundary conditions for the simulated model.

Zone	Boundary Type
Inlets	Velocity-inlet
Outlets	Pressure-outlet
Dummy 1	Wall
Dummy 2	Wall
Tables	Wall
Equipment	Wall
Walls	Wall

Table A2. Solution Methods for the simulated model.

Variable	Solution Methods
Scheme	SIMPLE
Gradient	Least Squares Cell Based
Pressure	Second Order
Momentum	Second Order Upwind
Turbulent Kinetic Energy	Second Order Upwind
Turbulent Dissipation Rate	Second Order Upwind
Energy	Second Order Upwind

The air velocities at 12 different points were measured along the length of the room, as shown in Figure 7. These measuring points were taken 1.1 m above the floor. The air velocities at the same fixed positions (as in Figure 7) were measured using probe in CFD post-processing. This experimental and simulation evaluation at these air velocity points was made for case 1 (uniform airflow distribution from ACBs). The measured air velocity points showed a 10% variation from the CFD results as shown in Figure A3.

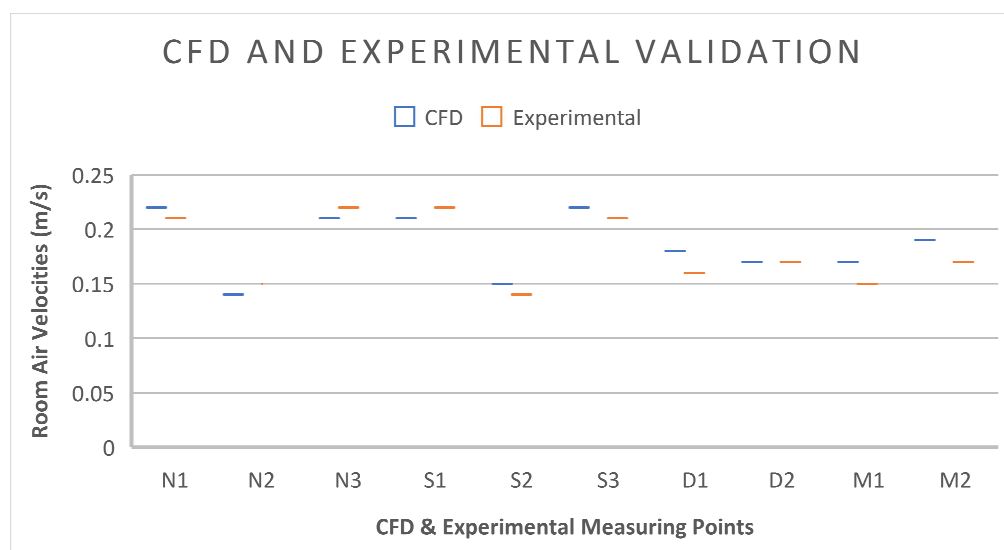


Figure A3. CFD and Experimental validation.

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CHAPTER 9.

PAPER VI

Precision Ventilation for an Open-Plan Office: A Study of Variable Jet Interaction between Two Active Chilled Beams

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Göran Hultmark, Peter V. Nielsen and Alireza Afshari

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Article

Precision Ventilation for an Open-Plan Office: A Study of Variable Jet Interaction between Two Active Chilled Beams

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Abstract: Precision ventilation is developed to achieve thermal comfort for occupants in an office by creating micro-climate zones. The present study aims to achieve individual thermal comfort for occupants with different metabolic rates by using higher airspeeds for enhancing heat transfer. The variable jet interaction between two ACBs with JetCone adjustments cause higher velocity jets to reach different regions of the occupied zone. The colliding jets from the center of a thermal isolated room were moved towards different zones in an office configuration with a constant room temperature of 23 °C. This study was conducted for five different cases in a room divided into four zones according to occupants' metabolic rates. The experimental and CFD results show that occupants facing symmetrical airflow distribution and with a constant 1.2 metabolic rate (Case 1) had a similar predicted mean vote (PMV) index. The zones with higher-metabolic-rate occupants, i.e., 1.4 met and 1.6 met in cases 2 and 3 were exposed to air velocities up to 0.4 and 0.5 m/s, respectively. In case 4, the air velocity in the single zone with 1.6 met occupants was raised to 0.6 m/s by targeted airflow distribution achieved by adjusting JetCones. These occupants with higher metabolic rates were kept thermally neutral, in the −0.5 to +0.5 PMV range, by pushing the high velocity colliding jets from the center towards them. In case 5, the results showed that precision ventilation can maintain the individual thermal comfort of up to three different zones (in the same office space) by exposing the occupants with metabolic rates of 1.2, 1.4, and 1.6 met to airspeeds of 0.15, 0.45, and 0.55 m/s, respectively.

Keywords: active chilled beams; individual thermal comfort; JetCones; metabolic rates; precision ventilation



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1. Introduction

Modern offices are designed with different layouts to promote communication and knowledge-sharing among employees. Open-plan office configurations are the most used office layouts in Scandinavia [1]. These open-plan offices involve the absence of partitions to accommodate many employees by reducing individual space [2]. Studies have shown that employee's satisfaction with comfort and productivity can be enhanced by providing a comfortable environment along with a sustainable open-plan office design [3,4]. It is a widely known fact that people spend up to 90% of their time indoors [5] and the need for individual thermal comfort is necessary to enhance their productivity [6]. This can improve the occupant's ability to perform manual tasks. Thermal comfort is defined as the condition of mind that provides satisfaction with the thermal environment [7,8]. Fanger's theory about thermal comfort showed that thermal comfort is dependent on metabolic rate, clothing insulation, and environmental conditions [9,10]. Hence, for a large-scale office setup, managing individual thermal comfort for the occupants of different ages,

gender, and health seems challenging with conventional HVAC systems such as mixing ventilation [11,12].

Mixing ventilation systems are the most-used ventilation systems due to their ability to provide thermal uniformity [13]. In these systems, the cool air is supplied through the ceiling and the airflow in the room is continuously driven by the inertia of the supply air to maintain temperature symmetry. Active chilled beams (ACBs) are a type of mixing ventilation system. Studies have shown that ACBs are successful in maintaining thermal uniformity due to adequate air mixing [14]. Latif et al. [15] concluded that most of the studies on ACBs are carried out for large-scale open-plan office configurations. These office types have greater occupant density and may require multiple ACBs to meet a sensible cooling demand [16]. A typical ACB generally consists of a primary air plenum, mixing chamber, nozzles, and a heat exchanger. Processed air from the Air Handling Unit (AHU) is forced into the set of nozzles as primary air [17]. The purpose of the nozzles is to provide high-speed primary air and consequently create high dynamic pressure and low static pressure to facilitate induction, i.e., pressure differences between the mixing chamber and room. ACBs with JetCones can provide adjustable airflow patterns by controlling adjustment regulators [18,19]. These adjustment pins (see Figure 1) can be moved from 0 to 9 to vary the magnitude and direction of the airflow coming out of the ACB.

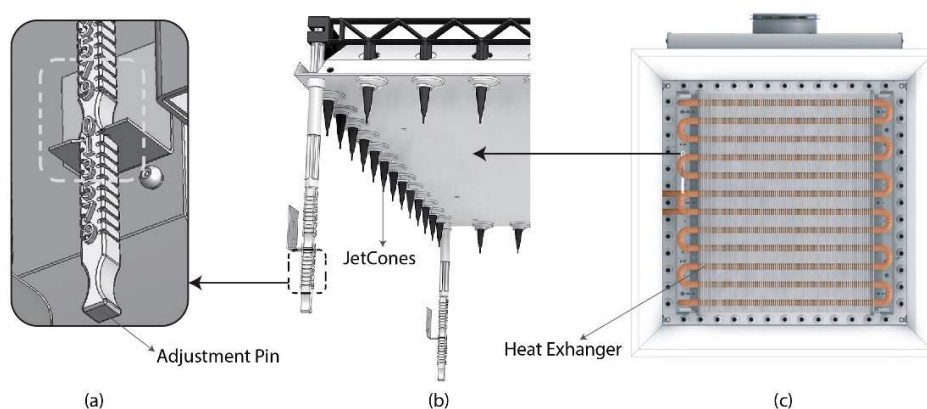


Figure 1. (a) Adjustment pin regulator, (b) JetCones system, and (c) ACB unit without an induction grill [15].

Personalized ventilation systems (PV) (also known as task ambient systems) have been used over the decades to create a micro-climate thermal environment around office occupants [20–22]. These systems include multiple air terminal devices (ATDs) (around an individual or an office desk) which operate simultaneously to blow different air jets directly on the occupants [23,24]. The combination of mixing ventilation and personalized ventilation systems are also applied in the office rooms to build micro-climate zones with different ATDs [25,26]. However, these systems involve multiple ATDs and a direct blow of air on the occupants, which is not pleasing for the office aesthetics. Unlike PV systems, the supply jets from the ACBs inlets spread horizontally across the ceiling due to the Coandă effect [27]. These horizontal jets move along the ceiling and collide with incoming supply jets from other ACBs to result in a colliding effect [28]. Studies have shown that the downfall of these colliding inlet jets creates maximum local airspeeds [29]. In this study, the collision of supply air jets with each other and with the walls is used to create different velocity zones in the same office space. These high- and low-velocity zones are established based on the metabolic requirements of occupants in the same shared office space.

Precision ventilation [30] involves advantages of both mixing ventilation and personalized ventilation to achieve individual thermal comfort. Precision ventilation is achieved by using ACBs (with JetCones) which can direct airflow in different directions along the ceiling. This use of ACBs eliminates the need for multiple air terminal devices around a single workstation in order to achieve individual thermal comfort [23]. In our previous study [30],

precision ventilation achieved individual thermal comfort for a dual desk-chair setup by directing more airflow towards occupants with higher air velocity needs (metabolic rates). This paper expands the previous work by including an experimental and a CFD study in order to achieve precision ventilation by using the variable jet interaction between two ACBs as part of larger open plan offices, i.e., four-desk-chair setup. This variable jet interaction or the movement of colliding jets is achieved by adjusting the JetCones of two ACBs to establish different air velocity zones in an office space.

2. Methodology

2.1. Experimental and Simulation Setup

Full-scale laboratory experiments were conducted in a room with dimensions $4.2 \times 4 \times 2.8$ m (length (L) \times width (W) \times height (H)) at the Aalborg University, Copenhagen. The experiments were carried out during summers (between the months of May–September). The two ACBs used in the study were connected to a duct network, which delivered 15 L/s of primary air to each unit. The supply air temperature from the ACBs was set at 20 °C and a temperature difference (Δt) of 3.5 K was maintained between supply and room air temperatures. The constant airflow of 50 L/s was supplied from each ACB, such that 100 L/s of total airflow (w) was maintained in the room.

$$\text{Heat load (q)} = wC_p\Delta t \quad (1)$$

According to the Equation (1), 22 W/m² of heat load was maintained in the room measuring 16.8 m², which is considered acceptable in Scandinavia. The heat balance in the room was maintained by the cooling power of the ACBs. Table 1 shows the features of the ACBs used in this study.

Table 1. ACB specifications.

Units	Values
ACB dimensions (L \times W \times H)	0.6 \times 0.6 \times 0.2 m
ACB units	2
Functions	Cooling, heating, and ventilation
Operating system	Cooling 2-pipe system
Distribution profile	Radial
Capacity	769 W each

The test room was built with an open-plan office configuration, consisting of four workstations and two ACBs installed in the ceiling, each 2 m apart. Each workstation consisted of a desktop, computer, and a lamp as a heat source as shown in Figure 2a. The room was divided into four zones to establish different velocity (or individual thermal comfort) zones. The positions of the four dummies (1.4 m, the height of a seated person), were 1.85 m apart so that their thermal flux had a negligible effect on each other [31]. The dummies were separated by open-plan 40 \times 75 cm desks (L \times W) under the plexus. The real office atmospheric environment was established for replicating human heat release by placing electric bulbs inside the dummies. The metabolic rate of each dummy was varied by changing the number and capacity of the electric bulbs installed inside. The heat fluxes of 65, 80, and 95 W/m² were applied for 1.2, 1.4, and 1.6 met, respectively. Dantec hot sphere anemometers with an absolute accuracy of $\pm 2\%$ of the reading (between the range of 0–1 m/s) were used to measure the air velocity in the room [32].

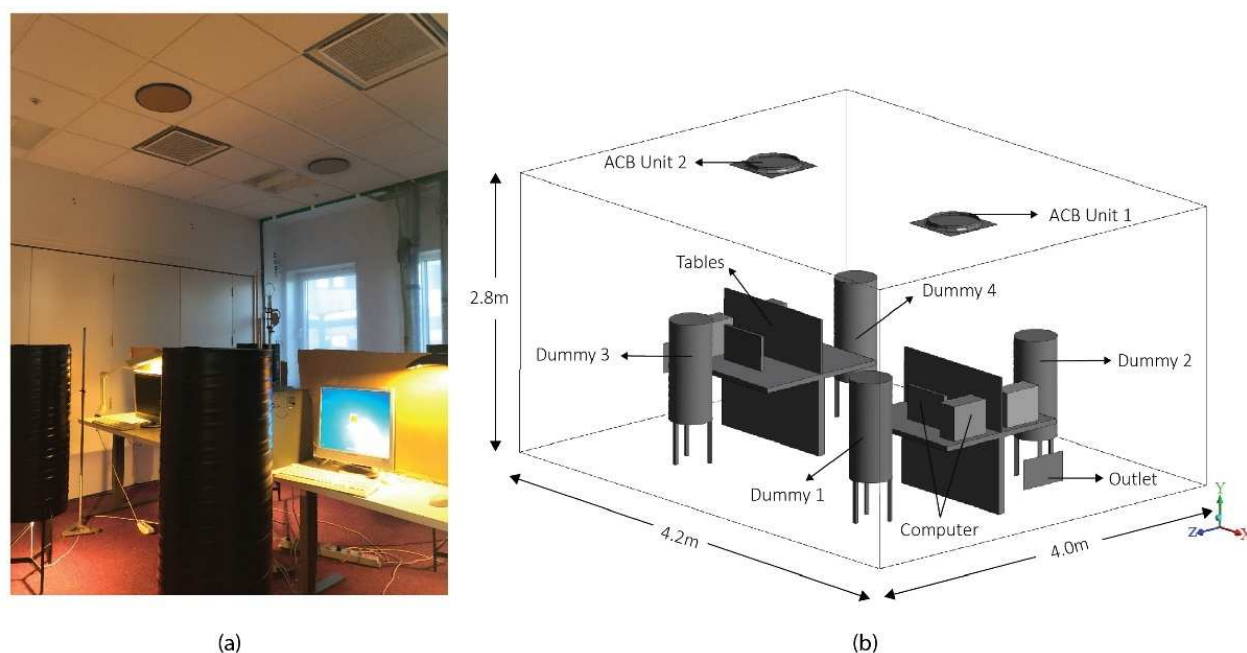


Figure 2. (a) Experimental office setup; (b) office geometry.

Figure 2b shows the geometry of the actual experimental setup made in Solidworks 2020 version. The geometry was imported to ANSYS 17.1 to conduct simulations for different cases [33]. The grid independence test was carried out and tetrahedral meshes were generated in the entire computational domain with a 3.67 million element size (see Appendix A). The RNG k- ϵ turbulence model along with a similar setup (solver, boundary conditions, and numerical algorithm) was applied according to previous studies on precision ventilation [30,34–36]. In simulations, the geometry of the ACBs was made circular in order to have radial flow and 50 L/s airflow (sum of primary and secondary airflows) produced from each ACB unit. The airflow distribution in the ACBs was altered according to the different cases described below.

2.2. ACB Airflow Distribution

In this study, ACBs consisted of four adjustment regulators to be able to adjust JetCones on each side of the unit. Four dummies (D1, D2, D3, and D4) were used as a heat source with heat flux adjusted with respect to changing metabolic rates in each case. Two ACBs in five different cases were investigated to achieve precision ventilation for an open-plan office configuration.

Table 2 shows the cases in which JetCones are positioned according to different metabolic rates. In each case, the adjustment pins of each ACB were regulated relative to the JetCone positions of the other ACB unit. The following cases were studied in this paper under non-isothermal conditions:

1. In case 1, the metabolic rates in all zones were kept at 1.2 met. The adjustment pins of both the ACBs were kept at the same position (position 5) to achieve a uniform thermal environment for all four velocity zones.
2. In case 2, the metabolic rates of the two dummies in zones 1 and 2 were raised to 1.4 met. The pin positions were adjusted to push more airflow towards zones 1 and 2.
3. In case 3, the metabolic rates of the two dummies in zones 1 and 2 were raised to 1.6 met. The pin positions were adjusted to push maximum airflow towards zones 1 and 2.
4. In case 4, the metabolic rate of the dummy in zone 1 was raised to 1.6 met. The pin positions were adjusted to push maximum airflow towards zone 1 to achieve individual thermal comfort.

5. In case 5, the metabolic rates of the two dummies in zones 1 and 4 were raised to 1.6 and 1.4 met, respectively. The pin positions were adjusted to push maximum airflow towards zone 1 and medium level airflow to zone 4.

Table 2. Different cases of precision ventilation with respect to metabolic rates and JetCone positions.

Cases	Metabolic Rates (Met)				Adjustment Regulator Positions	
	D1	D2	D3	D4	ACB 1 (adj. 1 + adj. 2 ...)	ACB 2 (adj. 1 + adj. 2 ...)
Case 1	1.2	1.2	1.2	1.2	5 + 5 + 5 + 5	5 + 5 + 5 + 5
Case 2	1.4	1.4	1.2	1.2	7 + 7 + 0 + 0	7 + 7 + 0 + 0
Case 3	1.6	1.6	1.2	1.2	9 + 9 + 0 + 0	9 + 9 + 0 + 0
Case 4	1.6	1.2	1.2	1.2	0 + 9 + 0 + 0	0 + 9 + 0 + 0
Case 5	1.6	1.2	1.2	1.4	0 + 9 + 0 + 7	0 + 9 + 0 + 7

2.3. Measuring Points and Thermal Comfort Criteria

The horizontal air velocity and temperature measuring points were taken as shown in Figure 3. The vertical velocity measuring points were taken at the heights of the ankle (0.1 m), abdomen (0.6 m), and face (1.1 m) of a seated person (see Figure 3a). Temperature measuring points were taken at six different heights from the floor to ceiling to observe air temperature symmetry (see Figure 3a). The horizontal measuring points were taken at 16 different positions in the room for thermal comfort measurements as shown in Figure 3b.

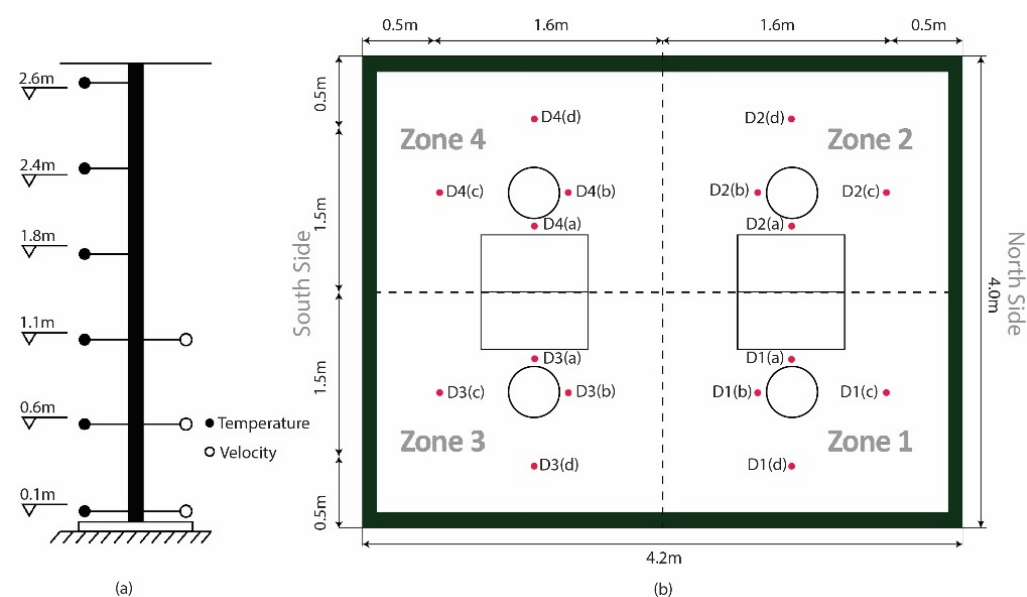


Figure 3. Velocity and temperature measuring points: (a) vertical and (b) horizontal.

Predicted mean vote (PMV) and predicted percentage dissatisfied (PPD) indices were used to calculate thermal comfort for human dummies in an enclosed environment. In the PMV-PPD model, measuring points were taken along the length of the room to calculate thermal comfort, i.e., PMV-PPD indices for each zone. Figure 4 shows the PMV-PPD curve used to represent thermal comfort according to Fanger's method. The horizontal PMV axis represents the seven-point thermal sensation scale (i.e., cold to hot). The vertical PPD axis shows the percentage of people predicted to experience local discomfort. According to ISO 7730 and ASHRAE 55 standards, the acceptable thermal comfort limit lies between -0.5 and 0.5 [37–39], as highlighted in Figure 4.

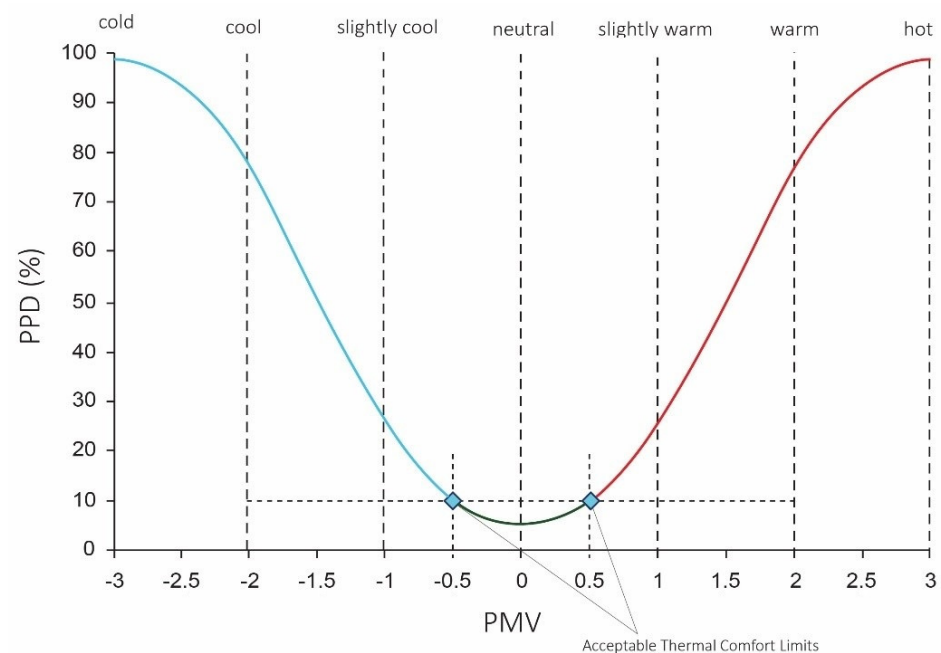


Figure 4. PMV-PPD measurement graph with thermal comfort limits [9].

The Equations (1) and (2) indicate the variables contributing to the calculation of PMV-PPD indices using Fanger's method. These variables were calculated based on mean measured values of the local air temperature (T_a), mean radiant temperature (T_r), and local air velocity (V_r). The mean radiant temperature air temperature was assumed equal to local air temperature [40] and relative humidity (RH) was measured at 60%. The clothing factor was assumed to be 0.6 clo during the summer [41]. The metabolic rates (M) in the equation were varied depending on the type of case. The metabolic rates for 1.2, 1.4, and 1.6 met were kept at 65, 80, and 95 W/m² per dummy, respectively. Whereas the rate of mechanical work (W) was kept at zero. The clothing surface area (f_{cl}) and the convective heat transfer coefficient (h_c) were calculated iteratively.

$$PMV = (0.303e^{-0.036M} + 0.028) \{ (M - W) - 3.05 \times 10^{-3} [5733 - 6.99(M - W) - p_a] - 0.42[(M - W) - 58.15] - 1.7 \times 10^{-5} M (5867 - p_a) - 0.0014 M (34 - T_a) - 3.96 \times 10^{-8} f_{cl} [(T_{cl} + 273)^4 - (T_r + 273)^4] - f_{cl} h_c (T_{cl} - T_a) \}, \quad (2)$$

$$PPD = 100 - 95 \exp [-(0.03353PMV^4 + 0.2179PMV^2)] \quad (3)$$

The room was divided into four zones and PMV-PPD values were measured at 16 different positions (see Figure 3b) for each case. Four different points were taken in each zone to measure air velocity and air temperatures for calculating PMV-PPD thermal comfort indices under different scenarios.

3. Results and Discussion

This study was conducted in five different cases of precision ventilation in an open-plan office through the jet interaction of two ACBs. The experimental results were verified through data validity with CFD. The experimental air velocities measured at 16 different points in the room were compared with the fixed velocity points taken in the ZX velocity plane in a three-dimensional room. The air velocity points (of both experiments and CFD) were considered for case 1 (as shown in Figure 3) at a fixed height of 1.1 m from the floor. Validation showed a less than 10% deviation from the simulation results as shown in Figure 5. The maximum deviation was shown at points D2d and D3d, but the results were still considered acceptable to perform thermal calculations using PMV-PPD equations [42].

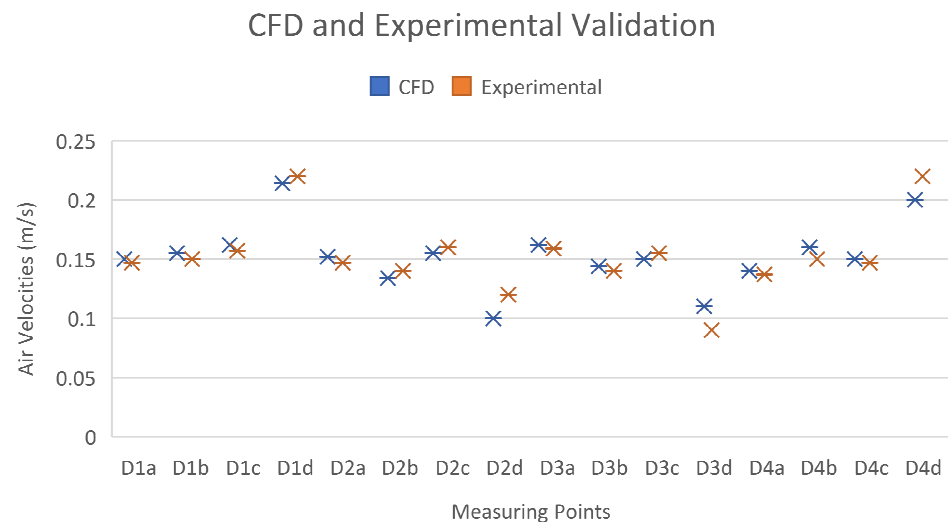


Figure 5. CFD and experimental velocity graph for validation.

3.1. Air Velocity Distribution

Optimal airflow is the requirement of maintaining the heat balance between the human body and the surrounding environment [43]. Occupants with higher metabolic rates require higher air velocities to accelerate the heat transfer [44]. Previous studies have shown that a local air velocity range up to 1.2 m/s is considered acceptable by raising room temperatures up to 28 °C during experiments [43,45]. In the previous study on precision ventilation [30], single ACBs established high velocity zones by pushing more airflow towards the targeted zones. The airflow along the ceiling and walls reached the occupied zone for creating micro-climate zones in the dual desk-chair setup. This study took advantage of high velocity colliding air jets to create high velocity micro-climate zones for an open-plan office setup. The jet interaction between two ACBs of all five cases is shown in the figures below (Figure 6).

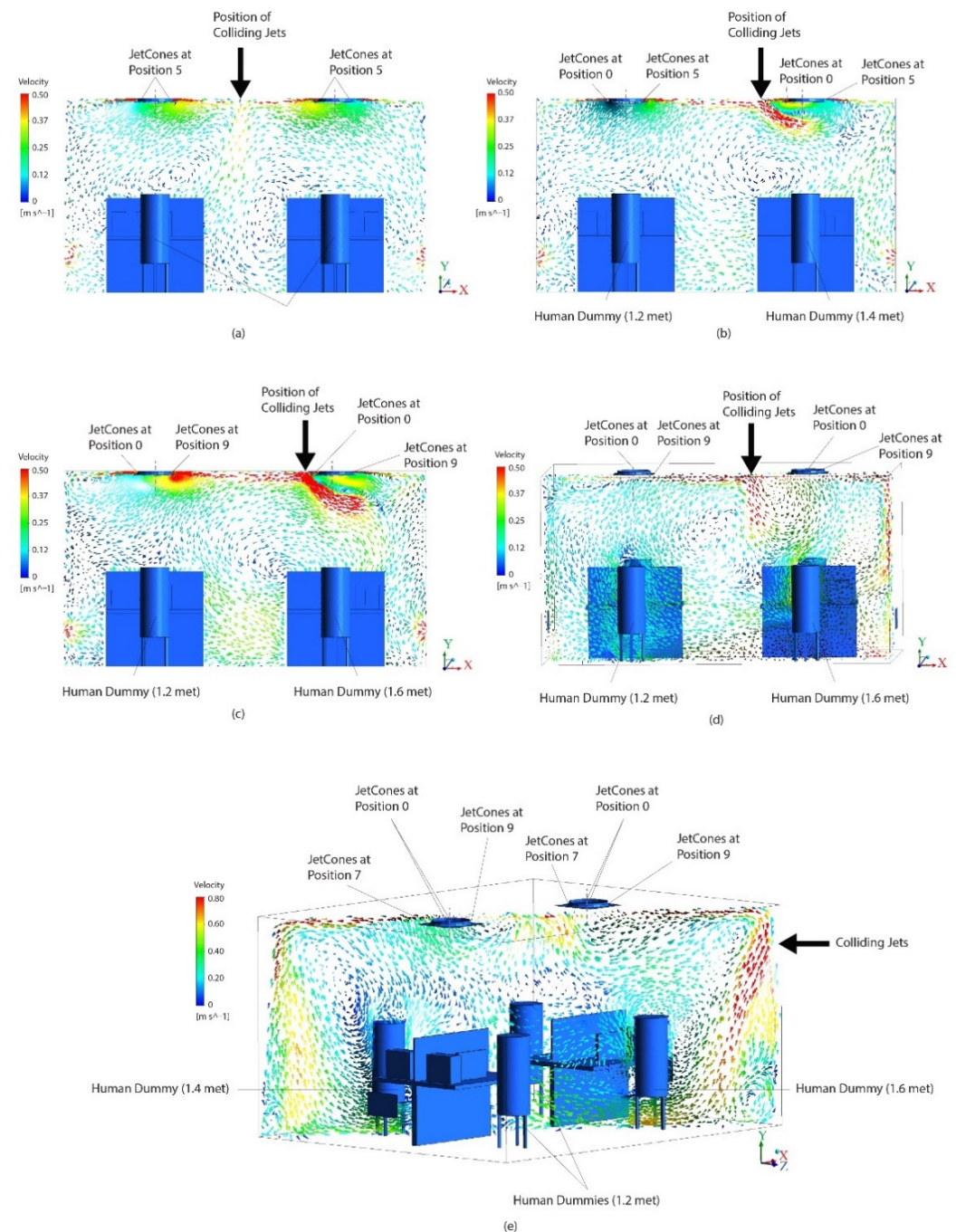


Figure 6. Airflow distribution from the ACBs: (a) case 1, (b) case 2, (c) case 3, (d) case 4, and (e) case 5.

Figure 6 shows the movement of colliding air jets from the two ACBs in an office room. The air velocities up to 0.8 m/s were reached following the collision of supply jets near the ceiling. Figure 6a–c shows that high speed colliding jets are moved (from the center) to the right side of the room by changing the JetCone positions from 0 to 9. In Figure 6b,c it was observed that higher air velocities are seen around dummies with 1.4 and 1.6 met due to the change in position of JetCones from 5 to 9, respectively. The single zone with metabolic rate of 1.6 met in case 4 was dealt with by directing higher air velocity jets towards one corner of the room with JetCone adjustments. Figure 6d shows the 2D velocity plane 1 m away from the center of the room to show the shift in colliding jets from the center to the high metabolic zone, i.e., zone 1. Figure 6e shows the 3D illustration of case 5, where two dummies (positioned diagonally) were supplied with two different velocity streams from the two ACBs. The JetCones were positioned (as shown in Figure 6e) to supply maximum

air velocities (up to 0.8 m/s) to the dummy with 1.6 met and up to 0.6 m/s air velocities to the 1.4 met dummy; whereas the airflow to human dummies with 1.2 met was kept unchanged. To observe the precision of airflow distribution in the occupied part of each zone, the air velocity vectors at heights of 0.1 and 1.1 m from the floor were taken for each case. The JetCones in ACBs were positioned (as in Table 2) to create micro-climate zones with respect to occupants' metabolic rates as shown in Figures 7–11.

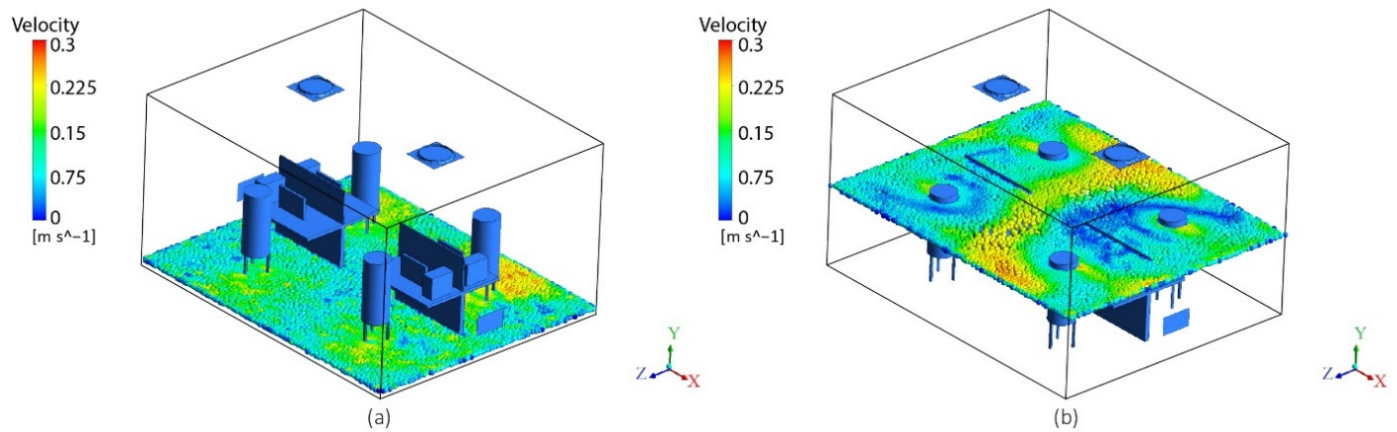


Figure 7. Case 1 air velocity distribution: (a) 0.1 m from the floor and (b) 1.1 m from the floor.

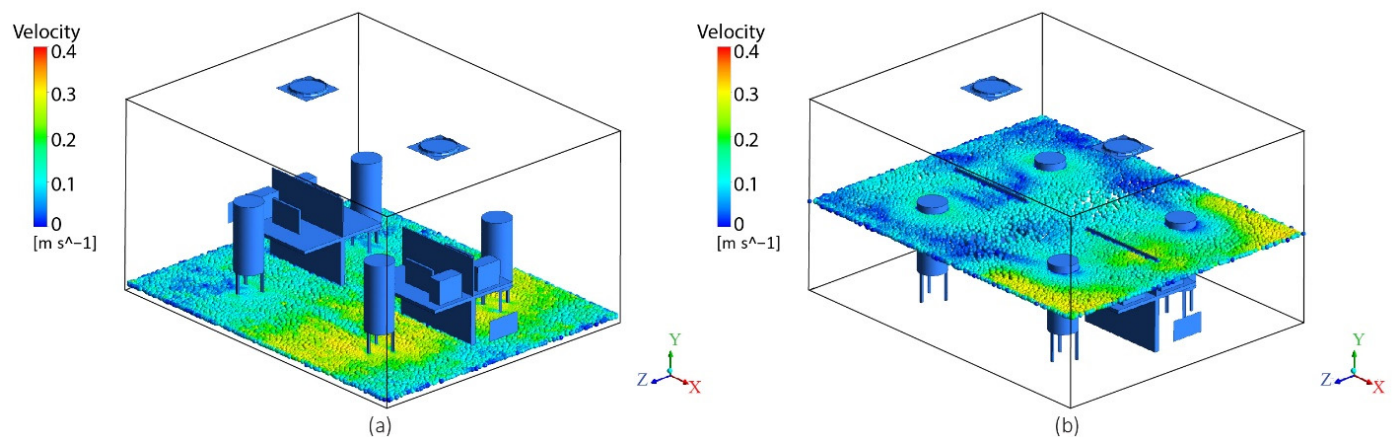


Figure 8. Case 2 air velocity distribution: (a) 0.1 m from the floor and (b) 1.1 m from the floor.

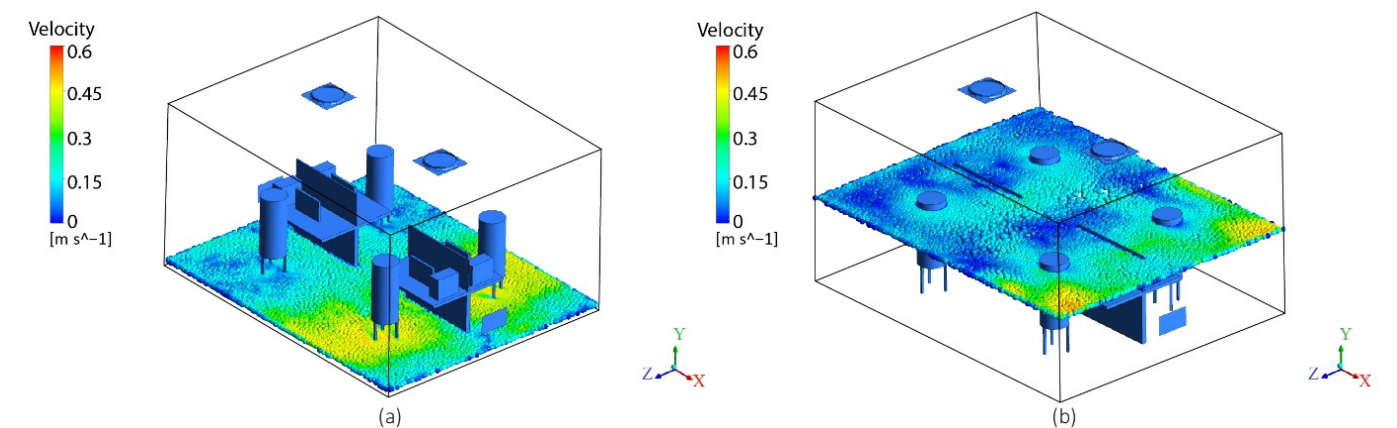


Figure 9. Case 3 air velocity distribution: (a) 0.1 m from the floor and (b) 1.1 m from the floor.

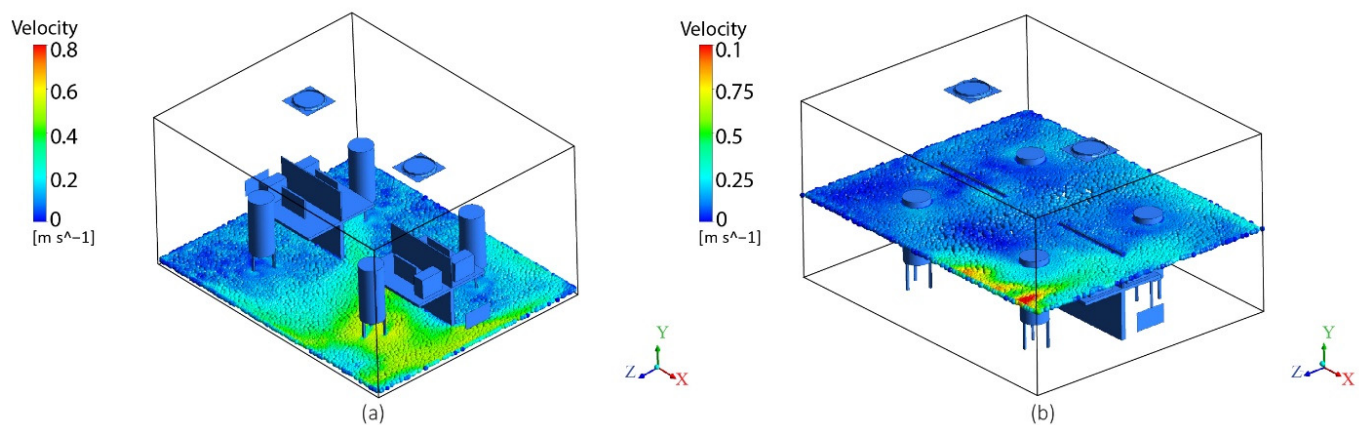


Figure 10. Case 4 air velocity distribution: (a) 0.1 m from the floor and (b) 1.1 m from the floor.

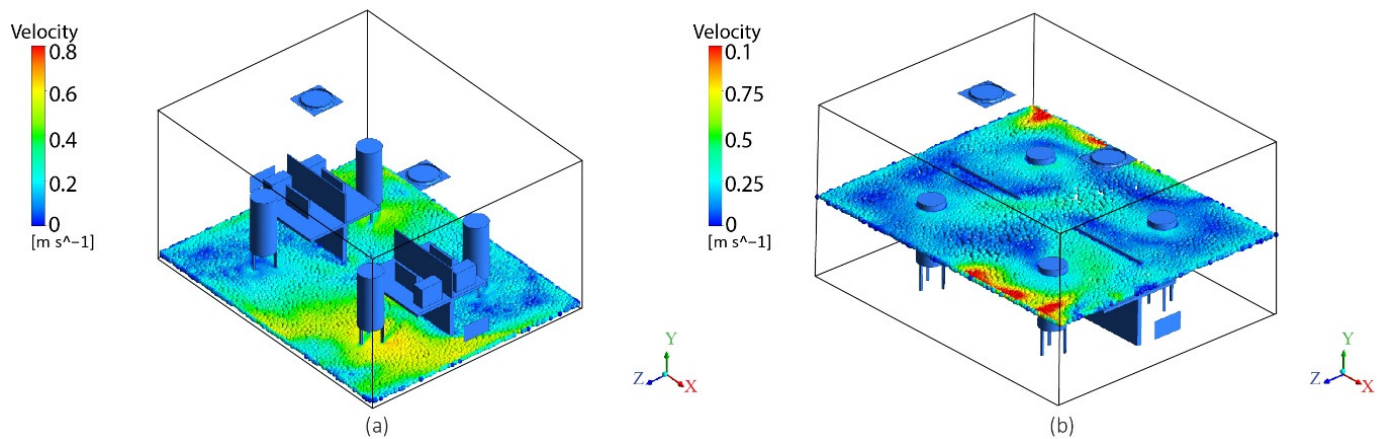


Figure 11. Case 5 air velocity distribution: (a) 0.1 m from the floor and (b) 1.1 m from the floor.

Figures 7–11 show that different velocity zones can be established by moving the colliding jets from the center towards or away from the occupancy zones. This movement of jets was used to create high- and low-velocity zones according to the metabolic rates set for the human dummy. Figure 7 shows the basic case where all dummies had the same metabolic rates, i.e., 1.2, and all four zones were supplied with uniform air velocities. The average air velocities for all four zones were kept at 0.15 m/s, due to the low metabolic rates. The pin positions of both the ACBs in this case were kept at position 5 in order to keep high momentum created by the collision of air jets at the center of the room (away from the four seating zones). In Figures 8 and 9, the high-velocity colliding jets are directed from the center to (zones 1 and 2) the dummies with metabolic rates of 1.4 and 1.6 met, respectively. The average air velocities in these high-velocity zones (zones 1 and 2) for cases 2 and 3 were raised to 0.35 and 0.45 m/s, respectively. This maintained the internal body heat balance and individual thermal comfort. Case 4 (Figure 10) shows velocity changes in only one zone (zone 1) for achieving individual thermal comfort of the dummies with metabolic rates (higher heat release) of 1.6 met. The JetCones of the two ACBs were positioned (see Table 2) to increase the average air velocities of zone 1 to 0.6 m/s. However, the velocities of zone 4 in the same case were also seen to increase at (points D4a and D4b) 0.1 m from the floor (see Figure 10a). Figure 11 shows a case with a combination of metabolic rates (1.2, 1.4, and 1.6 met) in a single office space. The pin positions were adjusted to have air velocities of 0.15, 0.45, and 0.55 m/s in the specific zones with human dummies with metabolic rates of 1.2, 1.4, and 1.6 met, respectively. The results of this case show that precision ventilation can be achieved by creating low- and high-velocity zones in different combinations by targeting the airflow according to the metabolic requirements.

3.2. Air-Temperature Distribution

Vertical air temperature difference plays an important role when considering local discomfort. The problem with local discomfort through the vertical temperature difference is mainly observed in displacement ventilation systems where the airflow supply is near the floor [46]. In mixing ventilation, the local discomfort caused by the vertical temperature difference is low due to adequate turbulent mixing between supply and room air. Figure 12 shows the vertical air-temperature distribution for all five cases measured at 16 different points (see Figure 3). The vertical temperature gradient in the occupied zone (up to a height of 1.8 m from the floor) in all cases was found to be comparatively less than the temperature gradient between the floor and ceiling heights.

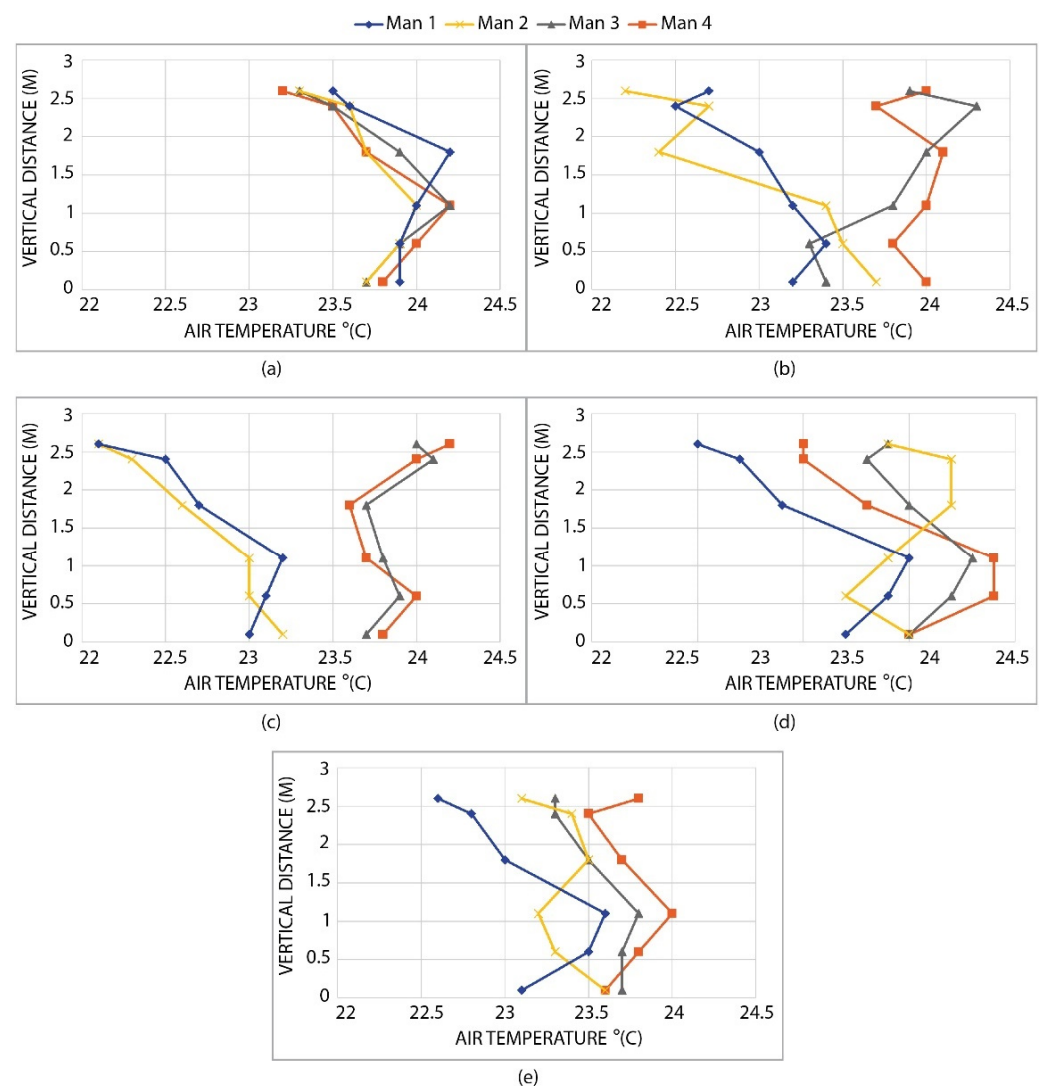


Figure 12. Measured vertical air temperature distribution: (a) case 1, (b) case 2, (c) case 3, (d) case 4, and (e) case 5.

In case 1, the temperature difference for all points was less than 1 °C due to a uniform thermal environment (see Figure 12a). In cases 2 and 3, the airflow is directed more towards zones 1 and 2. Hence, the colliding jets (with higher momentum and lower temperature) were moved towards zones 1 and 2 from the ceiling. The temperature near the ceiling (the height of the seated persons) above zones 1 and 2 was found to be 0.5 °C less than in zones 3 and 4. Hence, a temperature stratification up to 1.5 °C was observed in high-velocity zones 1 and 2, compared to 1 °C for zones 3 and 4 (Figure 12b,c). In Figure 12d,e, the results of cases 4 and 5 show that the maximum vertical temperature gradient of 1.5 °C was reached, which

is within an acceptable thermal limit [47]. The literature shows that greater airflow results in lower vertical temperature difference in the room [48]. However, in precision ventilation, greater airflow in any zone is supplied to the zone where there is more heat release and the vertical temperature difference is seen to be slightly increased above that specific zone. In addition to this, no substantial increase in horizontal temperature difference was observed in any case. The horizontal air temperature difference (from the north to the south side of the room) in all cases was found to be within the ± 1 °C range.

One of the main characteristics of mixing ventilation, to maintain thermal uniformity, was fulfilled through these results. This thermal uniformity was maintained by providing cooling power to balance the heat released from the heat sources in the room. Figure 13 shows the temperature contours of the two selected cases (1 and 3). For all cases, temperature contours were identical to the experimental results with an average room air temperature of 23 °C. Figure 13b shows that thermal uniformity is maintained even when supply air jets at 20 °C are pushed towards a specific side of the room. The temperature difference between the supply air and room air was maintained at around 3.5 K. Room temperatures can be raised in cooling modes to save energy while maintaining thermal comfort by utilizing higher air velocities. This reduction in energy use by increasing room temperatures can lead to an improvement in the overall energy efficiency of the system, hence providing optimal and flexible energy management [49,50].

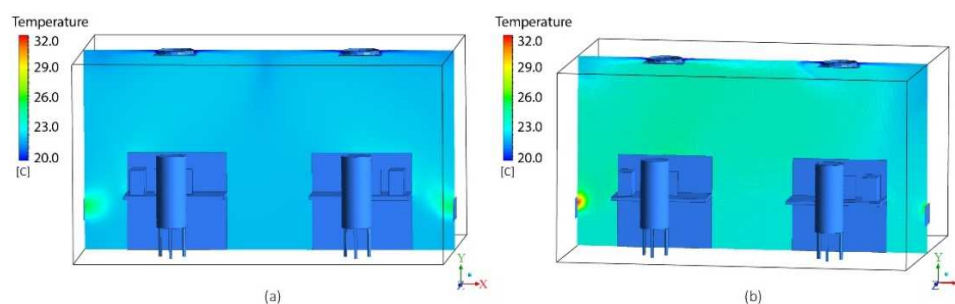


Figure 13. Simulated temperature contour across the central plane of the room: (a) case 1 and (b) case 3.

3.3. PMV-PPD Calculations

PMV and PPD thermal comfort calculations were calculated based on experimental and simulation data from all five cases. The PMV and PPD values at the height of a seated person are determined by taking mean values of air temperature, radiant temperature, and air velocity at 16 different positions (see Figure 3b). The mean values were taken at heights of 0.1, 0.6, and 1.1 m, respectively. The PMV and PPD were calculated from the expressions (2) and (3). The PPD is the function of PMV and was plotted in Figure 14. The PMV and PPD points were plotted after this using measured and assumed variables, as shown in the graphs below.

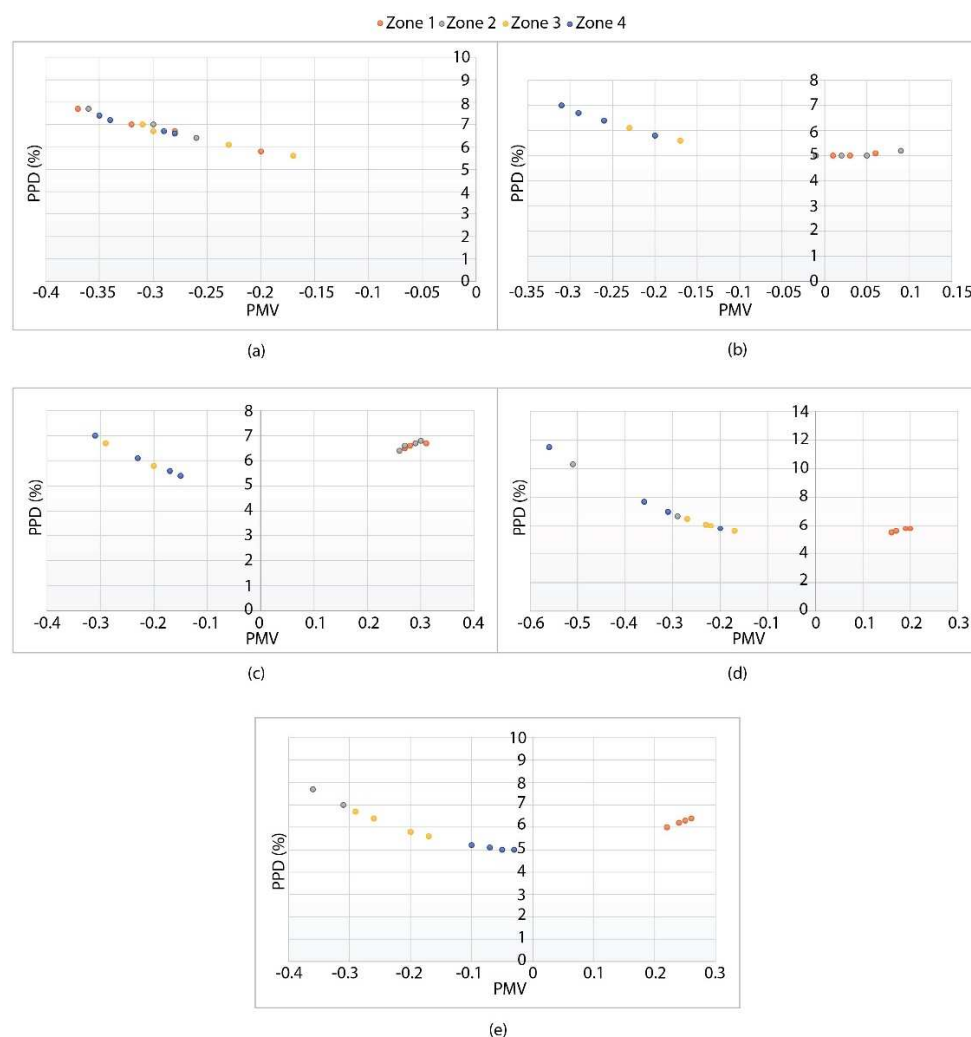


Figure 14. PMV and PPD: (a) case 1, (b) case 2, (c) case 3, (d) case 4, and (e) case 5.

In case 1, Figure 14a shows that ACBs airflow distribution was kept symmetrical and PMV values are found within the acceptable thermal range of -0.5 to $+0.5$ [18]. The PPD values are also less than 10%. Figure 14b,c show PMV-PPD values for cases 2 and 3, where thermal comfort range is maintained by providing higher air velocities to zones 1 and 2. At the same time, PMV-PPD values were maintained for zones 3 and 4 (dummies with a metabolic rate of 1.2 met) by directing less airflow to these zones. Case 4 shows the individual thermal comfort for zone 1 and PMV values are seen to be between 0.2 and 0.3. In case 4, the PMV-PPD values for zone 4 also seem to get slightly affected from the high-velocity zone, i.e., zone 1 (see Figure 14d). Furthermore, the PMV values are increased up to 0.6% and PPD up to 11.5%, but overall PPD values show that more than 80% of occupants were satisfied. Case 5 included a combination of metabolic rates and ACB pin adjustments, which were used to keep the PMV values within the -0.5 to $+0.5$ range, and PPD less than 10% (Figure 14e). Hence, precision ventilation can effectively be applied to create multiple micro-climate zones for real office occupants with different metabolic rates.

4. Conclusions

The occupants with higher metabolic rates working in offices require higher air velocities to balance body heat. In this regard, a precision ventilation technique is used to maintain the individual thermal comfort according to ISO 7730 and ASHRAE 55 standards by creating high and low velocity zones. The collision of supply jets from the two ACBs resulted in maximum air velocities up to 0.8 m/s. These high-velocity colliding jets es-

established micro-climate zones in an open-plan office through ACB's JetCone adjustments. These JetCones adjustments moved these high velocity colliding jets to the targeted zones. The following conclusions were drawn from the experimental and CFD study:

- The change in momentum caused by ACB JetCone adjustments resulted in up to three variable air velocity zones in the same office space. The human dummies with low metabolic rates (1.2 met) could be exposed to air velocities as low as 0.1 m/s. Whereas dummies with metabolic rates of 1.4 and 1.6 met were able to become exposed to 0.45 and 0.55 m/s air velocities, respectively.
- The acceptable PMV range for thermal comfort i.e., -0.5 – 0.5 , could be maintained for the occupants with high metabolic rates using airflow adjustments. This also led to an overall PPD of less than 10% for all five cases.
- Controlling the microclimate solely for a single velocity zone, i.e., case 4 with two ACBs in a large office space showed PMV and PPD values slightly above acceptable limits.
- The local thermal discomfort through the vertical temperature difference was not found to be a problem while implementing precision ventilation. Both vertical and horizontal temperature distribution in all cases was maintained with a temperature difference in the occupied zone of less than $1.5\text{ }^{\circ}\text{C}$.

5. Limitations and Further Work

This study highlighted the significance of jet interaction and colliding air jets from two ACBs for precision ventilation applications. Hence, colliding jets require more in-depth study as psychometric parameters such as the temperature of the room and plumes from the heat sources may affect the distribution.

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Nomenclature

Parameter	Description	Unit
ACB	Active chilled beams	-
AHU	Air handling unit	-
ASHRAE	American society of heating, refrigerating and air-conditioning engineers	-
ATD	Air terminal device	-
CFD	Computational fluid dynamics	-
HVAC	Heating ventilation and air conditioning	-
ISO	International organization for standardization	-
PMV	Predicted mean vote	-
PPD	Predicted percentage dissatisfied	-
RNG	Renormalization group	-
w	Total airflow in the room	L/s
C _p	Specific heat constant for air	J/(kg °C)
Δt	Temperature difference	K
q	Total heat load	W
f _{cl}	Clothing surface area factor	-
h _c	Convective heat transfer coefficient	W/(m ² .K)
t _a	Air temperature	°C
v _{ar}	Relative air velocity	m/s
t _{cl}	Clothing surface temperature	°C
p _a	Partial water vapor pressure	Pa

Appendix A

A grid independence test was carried out with different mesh densities at line $y_1 = -0.5$ $y_2 = -1.1$ in the middle of the two desks. The variations after 3.67 million elements were not observed and this grid size was considered suitable due to the insignificant influence of higher grid size on the solutions and to avoid computational cost, see Figure A1.

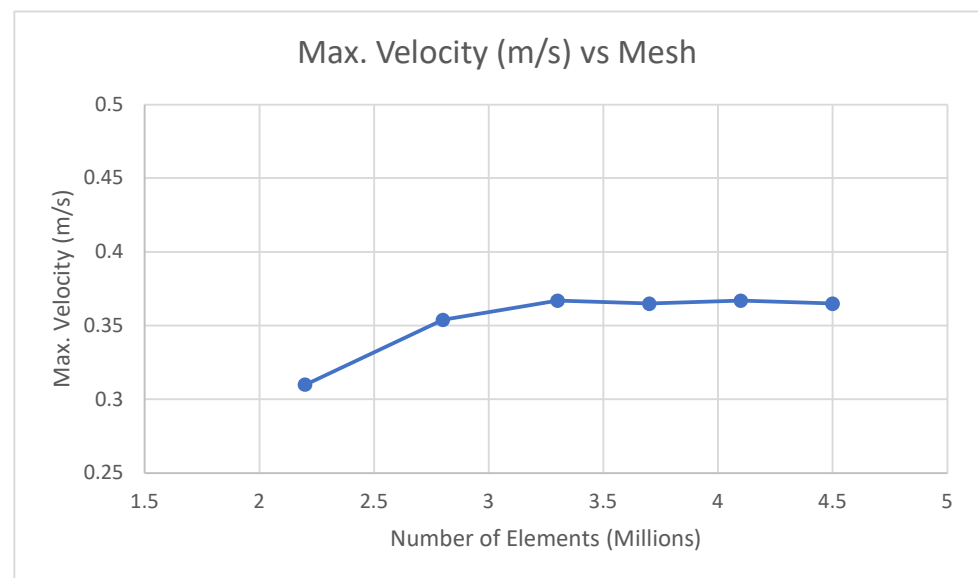


Figure A1. Grid independence test.

The mesh metrics on the horizontal axis in Figure A2a,b show mesh skewness and element quality, respectively. For 3D geometry, skewness with a value of 0 indicates an equilateral cell and value 1 of element quality is a perfectly shaped tetrahedral element. Figure A2 shows that most of the elements had an element quality between 0.75 and 0.95 and a maximum average skewness of 0.1. This means that the skewness and quality of the obtained mesh structure is acceptable for stable numerical computation.

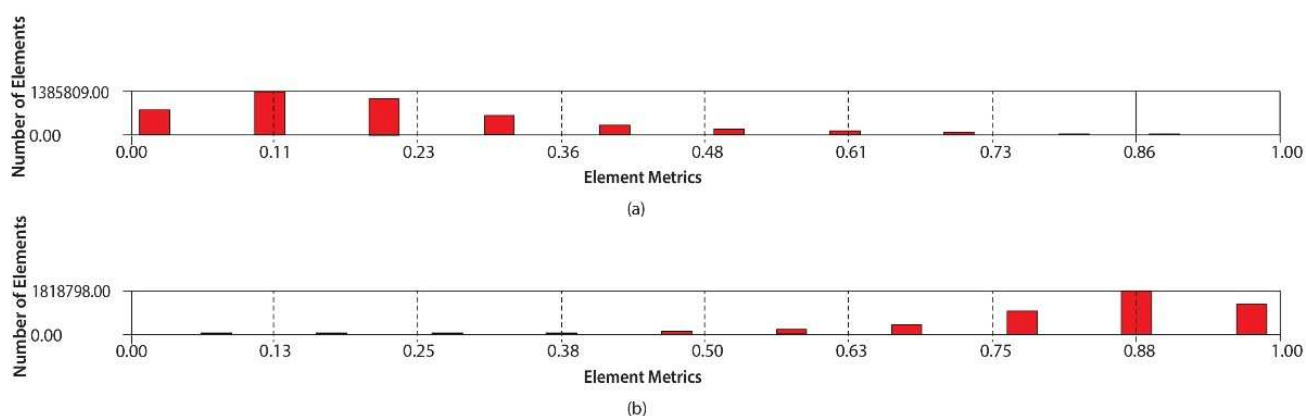


Figure A2. (a) Mesh skewness and (b) element quality.

The boundary conditions were used with respect to the actual setup made for the experiments. The SIMPLE (semi-implicit method for pressure linked equations) algorithmic scheme was applied to implicitly solve pressure and momentum equations. The convergence domain was set to $10e^{-6}$ to have a completely converged solution.

Table A1. Boundary conditions for the simulated model.

Zones	Boundary Type
ACB Inlets	Velocity-inlet
Human Dummies	Wall
Tables and Equipment	Wall
Outlets	Pressure-outlet
Room Walls (Adiabatic)	Wall

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Chapter 10.

PAPER VII

The Establishment of Design Criteria for Precision Ventilation in Open-Plan Offices

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The Establishment of Design Criteria for Precision Ventilation in Open-Plan Offices

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Abstract

Precision ventilation is used in open-plan offices to establish micro-climate zones according to the metabolic requirements of its occupants. The aim of this study is to determine design criteria and highlight the opportunities and limitations for precision ventilation in large-scale offices. The simulations and experiments were conducted for room temperatures between 23-25°C and with a fixed airflow from each ACB set at 50 l/s at steady-state conditions. A temperature difference of 3.5K was maintained between supply and room air in all cases with and without occupants. Results showed that precision ventilation can simultaneously establish low-level (<0.15m/s), medium-level (<0.45m/s), and high-level (<0.65m/s) air velocity zones in the same shared office space such that occupants with different metabolic rates had PMV values maintained within the acceptable limit i.e., -0.5 to +0.5. The establishment of a single air velocity zone in an open-plan office layout without influencing air velocities of other zones was achieved by lowering the airflow rates of two ACBs by 35%. The vertical and horizontal temperature uniformity was maintained with this precision ventilation system with draught rates of less than 20% for occupants with 1.2 met. Comparative analysis using precision ventilation with and without occupants showed that targeted air velocity distribution in the room can be negatively influenced by the absence of any heat sources in any zone. An annual energy saving of up to 15% was achieved by raising the cooling setpoint temperature from 23°C to 25°C.

Keywords: Active Chilled Beams; energy savings; metabolic rates; precision ventilation; open-plan offices

1. Introduction

Thermal comfort in offices plays a crucial role in improving productivity and well-being of the occupants [1, 2]. In open-plan office layouts, employees work in shared spaces and occupant density is high and an employee may require individual control over environmental factors like air velocity and air temperature [3-5]. Thermal comfort depends upon both personal and environmental factors. The personal factors include metabolic rates and clothing insulations and the environment around them is managed by environmental or controllable factors i.e., air speed, air temperature, radiant temperature and humidity. It is widely known that thermal comfort for everyone varies and depends

on an occupant's health, gender, age, or type of activities etc. [6]. Variation in each of the environmental factors will have an impact on personal factors and can lead to the feeling of comfort or discomfort for the occupant [7]. According to ISO 7730 Standard [8], the metabolic rates for seated office occupants is 1.2 met. Studies have shown that an occupant's metabolic rate, other than the level of ongoing activity also depends upon personal characteristics such as body and muscle mass, diet and health, age and gender etc. [9-11]. This metabolic rate may vary individually, and each individual may need a different thermal comfort level under uniform thermal comfort conditions.

Therefore, the need for maintaining individual thermal comfort is not possible with conventional mixing ventilation (MV) systems. A mixing ventilation system such as Active Chilled Beams (ACBs) maintains thermal uniformity by mixing cold supply air (from the ceiling) with warm room air [12]. The ACBs system uses the Coandă effect to distribute air uniformly in the room and is a widely used ventilation system due to their potential energy savings [13]. Unlike MV systems, personalized ventilation (PV) systems have been used for two decades to achieve micro-climates in offices around workstations. The literature shows that these systems even provide significant energy savings by raising the setpoint temperatures and decreasing the airflow of the integrated systems (like displacement or mixing ventilation) in the background [14]. In addition to this, air velocities up to 1.7m/s were raised by PV systems in some studies after raising the setpoint temperature to 29°C [15, 16]. Personalized ventilation systems require multiple Air Terminal Devices (ATDs) along with dimensional changes of these ATDs to build micro-climates around office workstations [17]. Modifications and design changes in offices layout are not encouraged by building architects and are therefore do not often have an application in offices [14, 18].

The individual need for maintaining thermal comfort in open-plan offices can also be fulfilled through a precision ventilation system. Precision ventilation [19] establishes micro-climates around individual occupants to fulfill their thermal needs. High and low air velocity zones are established by moving high or low velocity speed supply jets to the specific zone in a same shared office space. JetCones in ACBs are used for these systems to control the amount of airflow coming out of the ACBs. The total airflow out of the ACB is divided according to the zones with respect to relevant metabolic rates. Precision ventilation with only one ACB involves pushing more airflow towards the zone with a higher metabolic rate (see Figure 1a). While precision ventilation with two ACBs is achieved by moving the high-speed colliding jets towards or away to establish high or low velocity zones, respectively (Figure 1b).

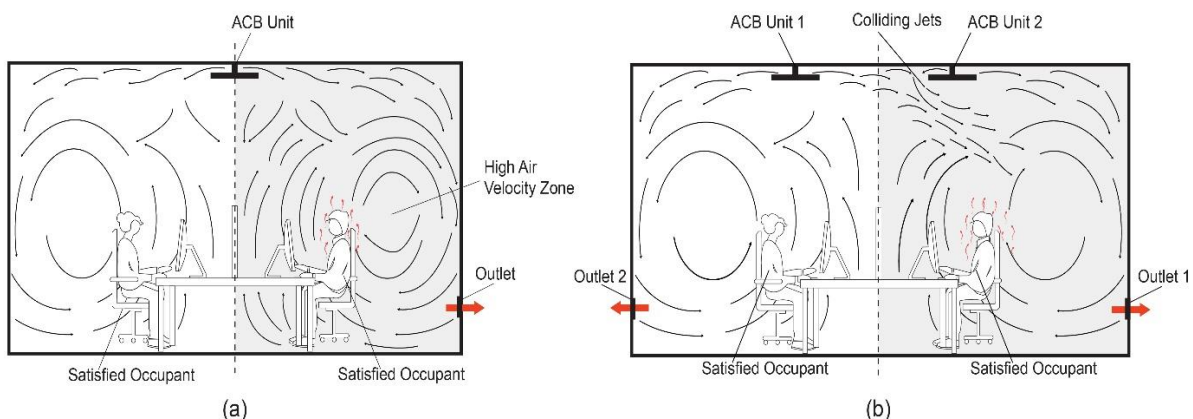


Figure 1 Precision Ventilation Concept (a) 1-ACB (b) 2-ACBs [19, 20]

Multiple ACBs are required to fulfill the high sensible cooling demands in open-plan offices. The concept of precision ventilation holds true for large open-plan office setups with a greater number of ACBs. In this study, the airflow distribution of four ACBs are examined with and without occupants in

the room in four different cases. Without occupants, airflow distribution in the room was uneven in the occupied zone. With occupants in the room, mixed convective airflow along with buoyant forces resulted in targeted airflow distribution and individual thermal comfort was maintained according to relevant levels of metabolic rates. The control of single air velocity zones for multiple ACBs also resulted in some high air velocities in other zones, as seen in a previous study [20]. This study involved a change in design strategy and expands on the application of precision ventilation in a large open-plan office at room temperatures between 23-25°C. The literature shows that significant energy savings can be achieved by raising the cooling setpoint temperature to certain degrees [21-24]. Unlike PV systems, the airflow in a precision ventilation system is kept the same and energy savings can be achieved by raising the cooling setpoint by 2°C. The limitations of the application of precision ventilation for large spaces are also highlighted for future work.

2. Methodology

Experimental Setup

ACBs with JetCones are used in these experiments to control airflow by adjustment regulators as shown in figure 2. The JetCones of each ACB was controlled by four adjustment regulators and maximum airflow is directed when JetCones are set at position 9. The JetCones at position 0 are still able to produce 30% of the airflow. These high speeds airflows are directed towards the areas in the room where metabolic rates of the occupants are high.

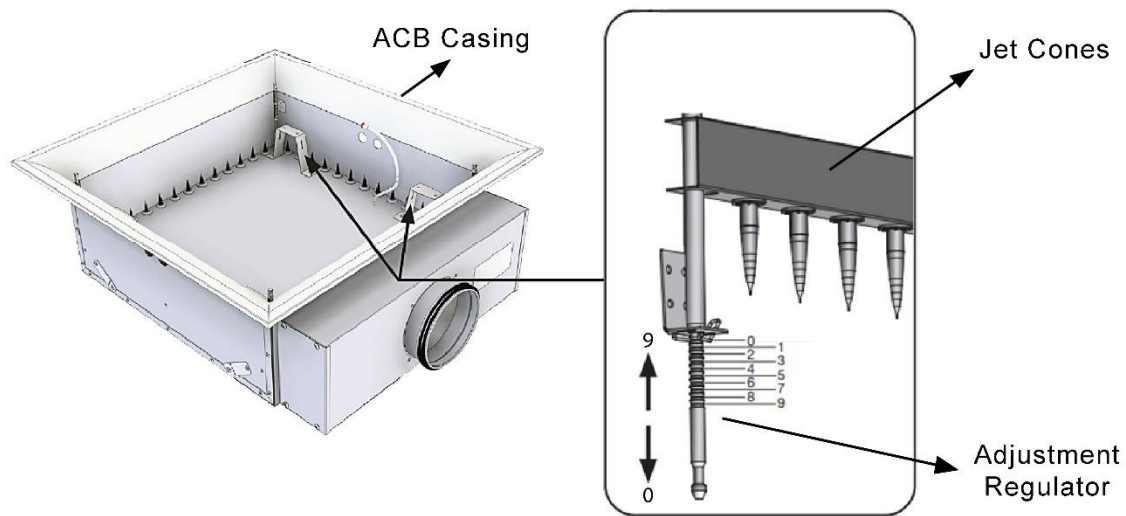


Figure 2 ACB with a JetCone feature [25]

In precision ventilation, the total airflow from the system to the room is kept constant and variable air velocity zones are established by adjusting JetCones, see Table 1. Four ACBs were used in this study in four individual cases to establish air velocity zones according to the metabolic need of the occupant. In case 1, uniform airflow distribution was carried out by adjusting JetCones to position 5. Each ACB produced a supply airflow of 50 l/s and case 1 was kept as a reference case. In case 2, the airflow was directed towards one half of the room for occupants with a metabolic rate of 1.4 met by setting Jetcones to position 7. In case 3, a single high air velocity zone was established for occupants with a metabolic rate of 1.6 met where half of the JetCones of two ACBs were opened to position 9 (maximum airflow). At the same time, the other JetCones were set to position 3 to maintain low level air velocities in the other three zones. In case 4, three different air velocity zones and JetCones were

adjusted so that the total airflow in the entire enclosed space remained at 200 l/s, as shown in Table 1.

Table 1 ACB settings for different case studies

Cases	Number of ACBs	JetCone Positions	Airflow Rate (l/s)
1	ACB 1 + ACB 2 + ACB 3 + ACB 4	All positions set to 5	50 + 50 + 50 + 50
2	ACB 1 + ACB 2 + ACB 3 + ACB 4	(7-7-0-0) + (7-7-0-0) + (7-7-0-0) + (7-7-0-0)	50 + 50 + 50 + 50
3	ACB 1 + ACB 2 + ACB 3 + ACB 4	(0-9-9-0) + (3-3-3-3) + (3-3-3-3) + (3-9-9-3)	60 + 35 + 35 + 70
4	ACB 1 + ACB 2 + ACB 3 + ACB 4	(9-9-0-0) + (3-3-3-3) + (7-7-0-0) + (3-3-7-7)	60 + 35 + 40 + 65

The experimental setup, which is shown in figure 3, involved desk-chair arrangements with human dummies, computers and lamps as heat sources. Electric bulbs were placed inside the dummies as a power source and the number of electric bulbs were changed according to heat release (metabolic rate) from each dummy. The ACBs are installed inside the ceiling so that airflow from each ACB is directed downwards to the occupied zone through walls and from the ceiling, as shown in figure 3. The overall experimental uncertainties were found approximately 4.5% (see Appendix A). Table 2 lists the details of the test room used in the experiments.

Table 2 Test room details

Type	Details
Room Size	38m ²
Heat Load	24w/m ²
Walls	Adiabatic
Temperature Range	23-25°C
Total Airflow	200l/s (50 l/s each)

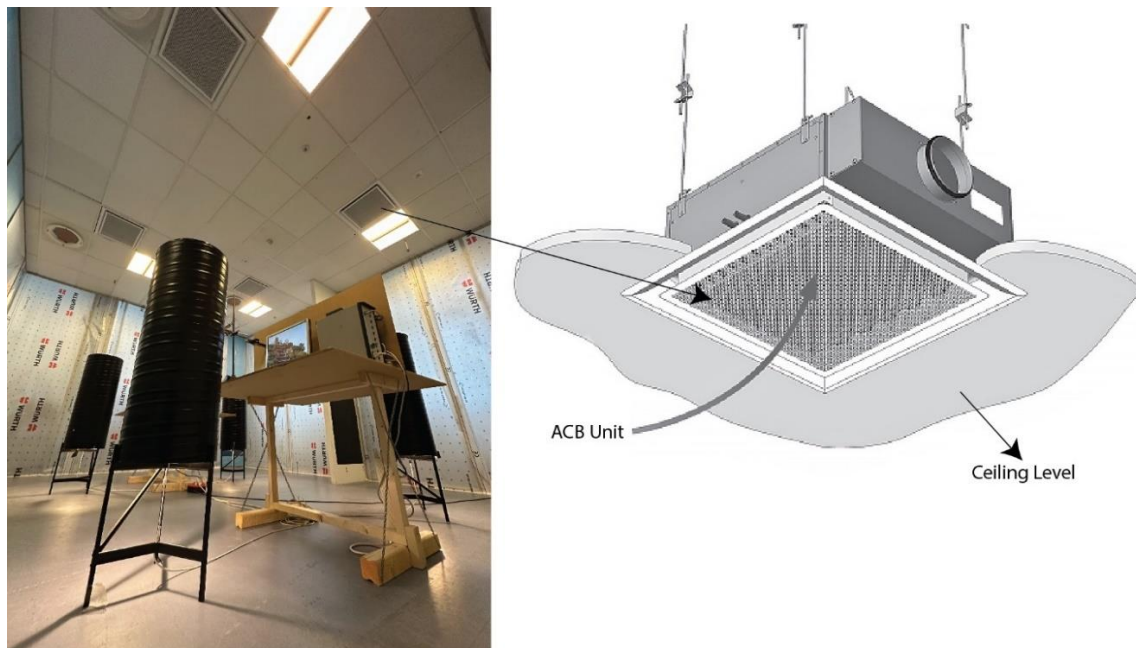


Figure 3 Actual Experimental Setup

Simulation Model

The geometry of the test room was made in Solidworks 2020 with the exact dimensions as the actual setup, as shown in Figure 4. The geometry of the ACB was circular shaped and was divided into 16 divisions so that radial flow emerged from the ACB. The experimentally measured air velocities at 16 positions on ACB inlets were used as inlet air velocities on a radial ACB geometry, as shown in Figure 4(b). Each quarter (four divisions) as highlighted in figure 4(b), controlled one of the four pin positions and the amount of supply airflow from the ACB inlet. Furthermore, to avoid a complex internal ACB geometry of the heat exchanger, internal induction was not considered and the resultant of both the primary air and induced air in the experiments was taken as supply air. The total supply airflow from each ACB was 50 l/s and supply air temperatures were set around 21°C to have ΔT around 3.5K (between supply and room air). For a fixed amount of primary airflow, room temperatures from 23°C to 25°C were recommended for precision ventilation, as shown in the previous study [19]. The simulations were carried out in ANSYS version 17.1, with details of the computational fluid dynamics (CFD) setup shown in Table 3.

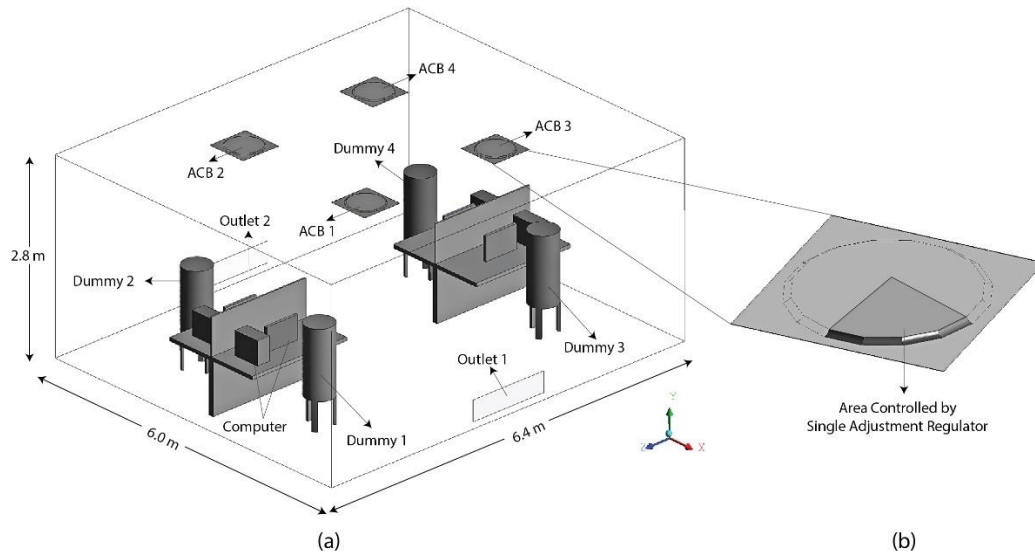


Figure 4 (a) Simulation Geometry (b) ACB Geometry

Table 3 CFD-model details

Type	Specifications
Mesh Shape	Tetrahedral
Mesh Size	4.6 million
Turbulence Model	RNG k- ϵ
Supply Airflow	50 l/s each ACB
Scheme	SIMPLE Numerical algorithm
Convergence Limit	10^{-6}

For indoor climate simulations, RNG k- ϵ is considered more accurate than other RANS models and has less computational time than LES models [26]. The mesh independence test was carried as shown in Appendix B.

Thermal Comfort Evaluation

According to ISO 7730 Standard [27], the overall thermal comfort levels of occupants is evaluated using PMV-PPD indices, which involve a 7-point thermal sensation scale. The scale on the horizontal

axis indicates 0 as the sensation of feeling neutral, positive values as the sensation of feeling warm and negative values as the sensation of feeling cold. The acceptable values for thermal comfort are from -0.5 to +0.5 and a PPD of less than 10% [28]. The relative humidity was assumed at 60% with the mean radiant temperature taken to be equal to room air temperature [29, 30].

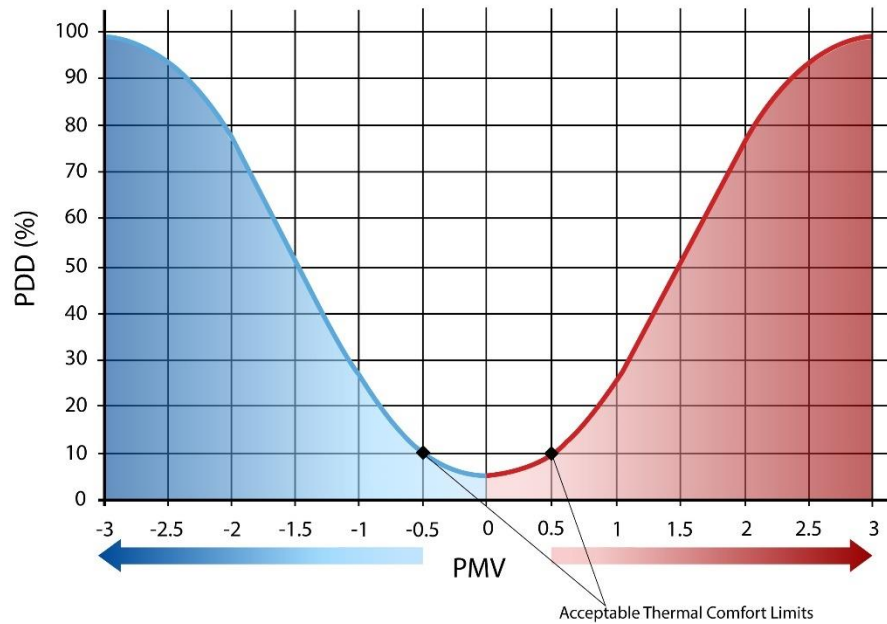


Figure 5 PMV-PPD Indices for Overall Thermal Comfort Evaluation

For the measurement of air velocity and temperature, the horizontal and vertical measuring points were taken at 16 different positions in the room, as shown in figure 6. Figure 6(a) shows the vertical measuring point from the ceiling to the floor. Air velocities that were 0.2 m below the ceiling height were also measured to observe the magnitude of the colliding jets controlled by the ACB JetCones. The room (shown in figure 6b) was divided into four zones according to the office configuration set for establishing variable air velocity zones.

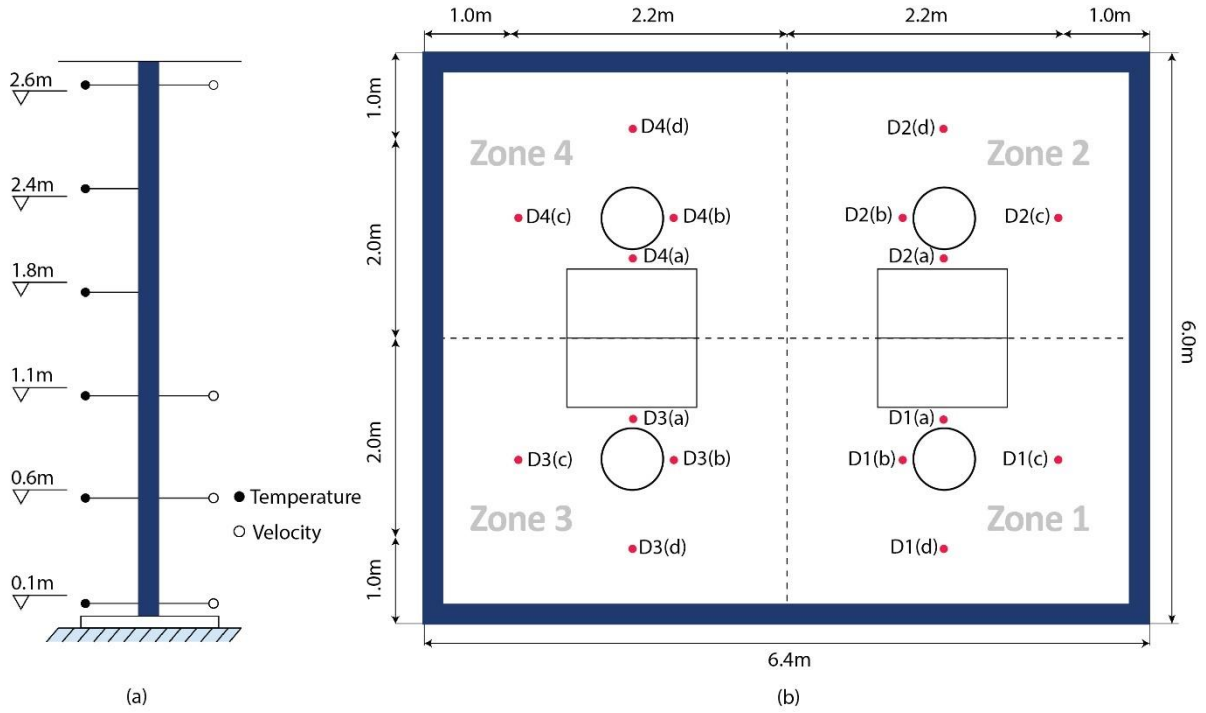


Figure 6 Air Velocity and Temperature Measuring Points (a) Vertical (b) Horizontal

In addition to this, the local thermal comfort level is also considered, which involves the measurement of draught rates, the vertical temperature difference between head and ankles, radiant asymmetry and cold or warm floors. Draught is defined as discomfort caused by unwanted local cooling [27]. Draught model, as shown in equation 1:

$$DR = (34 - t_a)(v_a - 0.05)0.62 (0.37.v_a.Tu + 3.14) \dots (1)$$

The Draught Rate (DR) percentage (%) is calculated using the above expression, where DR is the function of local mean air velocity (v_a), the local air temperature (t_a) and local turbulence intensity (Tu). This model is applied for a temperature band between 20°C to 26°C, a mean air velocity range up to 0.5m/s and a turbulence intensity between 10% and 60%. The draught model is mainly used for sedentary activities involving metabolic rates of 1.2 met. In this paper, local air velocities are raised to balance the large amount of heat release for occupants with metabolic rates of 1.4 and 1.6 met, respectively. To observe vertical temperature difference (VTD), temperature measurements were taken at heights 0.1 m and 1.1 m from the floor, shown in figure 6(a). The temperature measurements at different heights from the floor to the ceiling level were taken to know the vertical temperature stratification in the room. As this study was made for adiabatic wall conditions, discomfort caused by cold/warm floors or surfaces is not considered.

Energy Calculations

The annual energy savings were calculated for the rise in cooling setpoint temperature from 23°C to 25°C by using an energy calculation program, Be18. A single-story building was selected, and energy calculations were made for both heating and cooling demands. Table 4 shows the input parameters for energy calculations.

Table 4 Input Parameters for Energy Calculations

Type	Details
Building External Dimensions	51.6 m × 12.6 m

Conditioned Floor Area	650.2 m ²
Room Height	2.8 m
Ventilation Rate	1.68 l/s m ²
Solar Radiation (G-value)	0.62
U Values for Windows	Around 0.7 W/m ² K
U Values for Walls and Roof	0.08 – 0.25 W/m ² K

3. Results and Discussion

The validation between CFD and the experimental results was carried out at 16 measuring points (shown in figure 6(b)) located 1.1m above the floor. The experimentally measured air velocities were found to be slightly exaggerated compared to CFD velocities and there was up to a 15% difference from each other.

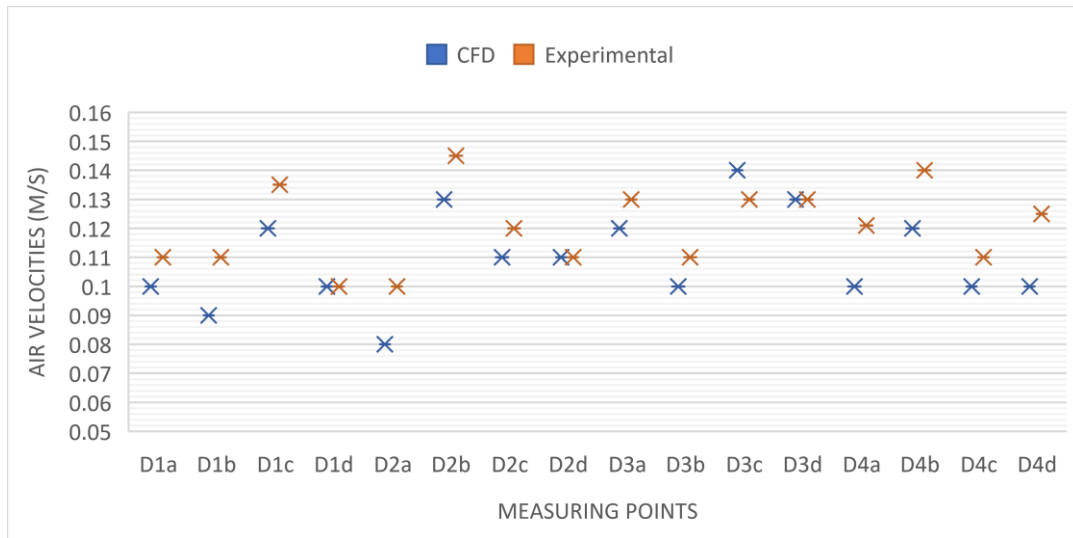


Figure 7 CFD and Experimental Validation

Air Velocity Distribution Without Occupants

The air velocity fields without occupants in the room were initially examined to observe the impact of colliding jets on room air circulation patterns. The success of precision ventilation is dependent on the movement of colliding jets that establish variable air velocity zones in a single office space. Figures 8-10 shows the airflow distribution from four ACBs for the three cases in an empty room with ΔT maintained at 3.5K by applying temperature through the walls. The airflow distribution near the ceiling for all four cases with occupants is shown in Appendix C.

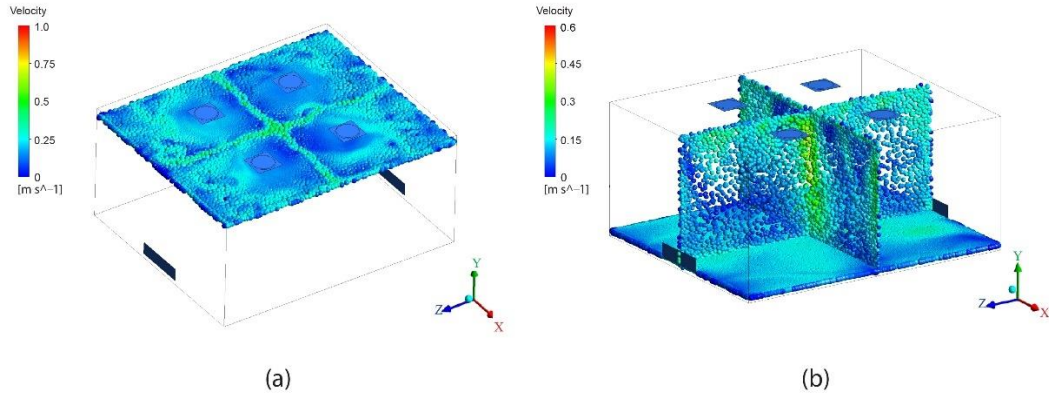


Figure 8 Case 1 (a) 0.2 m from the ceiling (b) 0.1 m from the floor including colliding jets

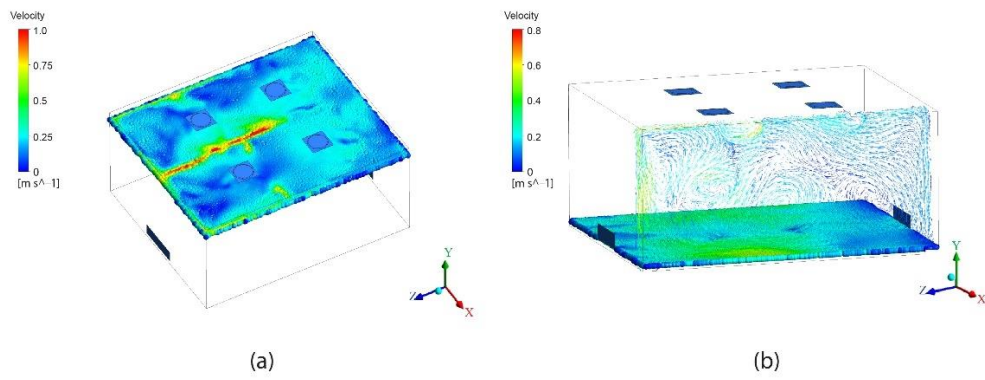


Figure 9 Case 2 (a) 0.2 m from the ceiling (b) 0.1 m from the floor including colliding jets

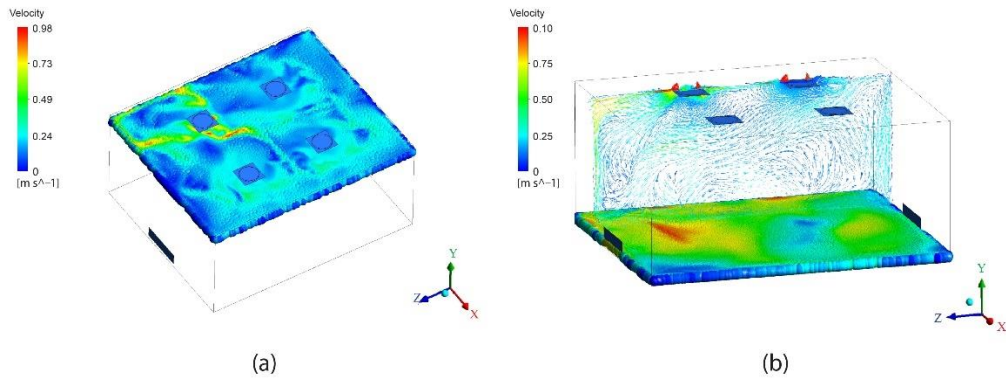


Figure 10 Case 3 (a) 0.2 m from the ceiling (b) 0.1 m from the floor including colliding jets

Figures 8-10 shows the air velocity profiles at 0.2 m from the ceiling and 0.1 m from the floor with air velocity vertical vector planes showing the direction of colliding jets for each case. The results of the first three cases with ACBs JetCones positions set according to Table 1. Figure 8(a) represents case 1, where all colliding jets of ACBs are directed in the center of the room and maximum air velocity is in the middle of four ACBs with air velocities at 0.45 m/s (Case 1). As airflow from all ACBs comes down equally from all sides through the ceiling and walls, therefore air velocity distribution along 0.1 m (above the floor) is uniform. In figures, 9 and 10, the effect of high-speed colliding jets is observed in

the horizontal plane below the ceiling but the airflow distribution at the floor level seems imprecise. The lack of buoyant forces from the heat sources in the room (that are usually mixed with the cold fresh air) and the absence of obstacles in the form of furniture makes air velocity distribution in the occupied zone uneven. Therefore, targeted zones are not achieved in an empty room due to a lack of mixing and absence of occupants.

Colliding Jets with Occupants

The supply air with a temperature that is 3.5°C lower than room air comes out from the ACBs, flows along the ceiling due to the Coandă effect, collides with the walls or other jets and deflects down towards the floor. These colliding air jets gain momentum to succeed in reaching the floor with high air velocities. Precision ventilation uses these high-speed colliding jets to establish variable air velocity zones in offices that do not have any partitions between workstations i.e., open-plan offices. Figure 11 shows the four cases of ACBs where colliding jets are moved in different areas of the room by JetCone adjustments. Figure 11(a) shows colliding jets in the center of the room with air velocities up to 0.4 m/s. In case 2 (figure 11b), the colliding jets are pushed towards ZX plane at air velocities up to 0.6 m/s to meet the air velocity needs of the occupants with a high metabolic rate (1.4 met). In cases 3 and 4, the colliding jets with air velocities of 0.8m/s and 0.6 m/s are pushed towards zones with 1.6 met and 1.4 met, respectively. The position of these jets was kept this way so that the air does not fall directly on the occupant in order to avoid discomfort.

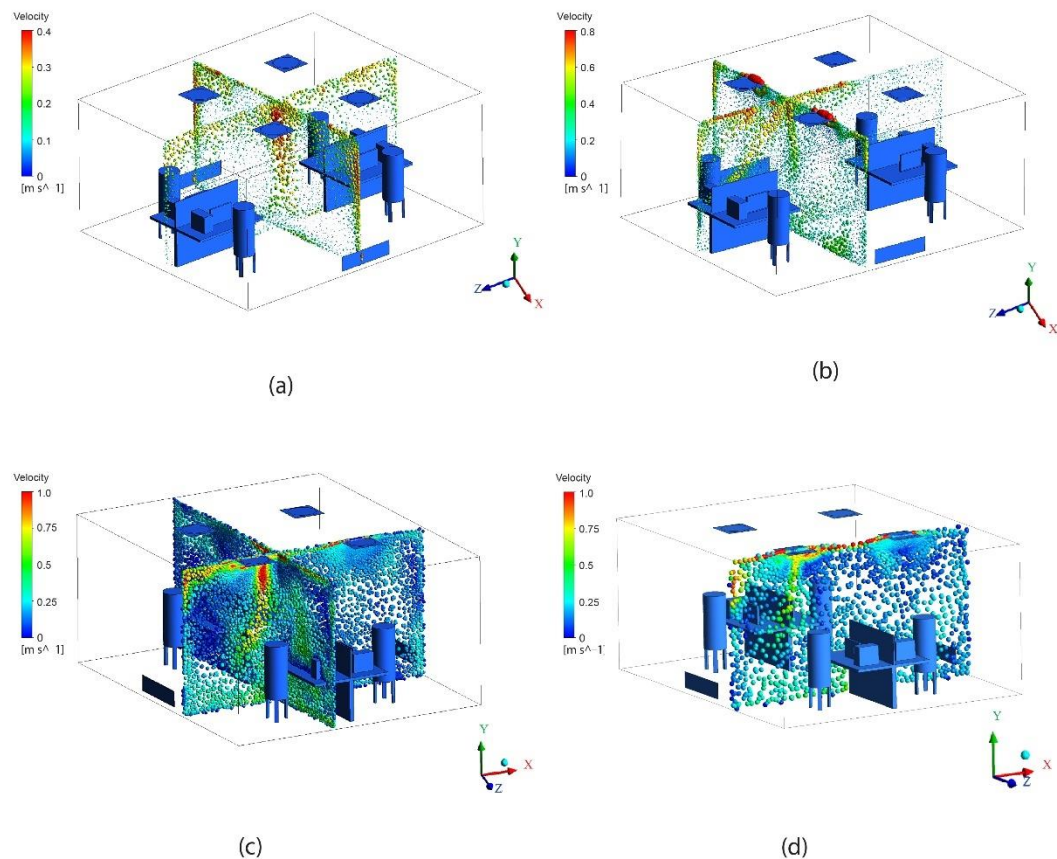


Figure 11 Colliding Jets (a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

In comparison to the previous study with two ACBs [20], the magnitude of the colliding jets was enhanced due to an increase in the number of ACBs and therefore, JetCone positions were adjusted

differently to have suitable air velocities for the occupants with a high metabolic rate and to avoid draught for the occupants with a lower metabolic rate (1.2 met).

Air Velocity Distribution with Occupants

Air velocities in the occupied zone were measured at 0.1 m and 1.1 m from the floor. Figures 12-15 show the air velocity profiles at these heights with the establishment of four air velocity zones. In figure 12, all four air velocity zones had the same magnitude of air velocities due to uniform air velocities in all four zones. Case 2 (shown in figure 13) showed dual velocity zones, as 58% of the total room airflow is pushed towards one side of the room to satisfy occupants with a metabolic rate of 1.4 met. In case 3, a single air velocity zone is established by changing the JetCone positions of ACBs 1 and 4 so that JetCones opened at maximum position (position 9) are facing the targeted high velocity zone (Figure 14). To avoid irregular air velocity distribution in other zones, the air volume of the other two ACBs (ACB 2 and ACB 3) was decreased to 35%. This resulted in air velocities in the other three zones of less than 0.15 m/s. In case 4, the JetCones were positioned as detailed in Table 1 to simultaneously create low, medium and high air velocity zones according to the occupant's metabolic rates (Figure 15).

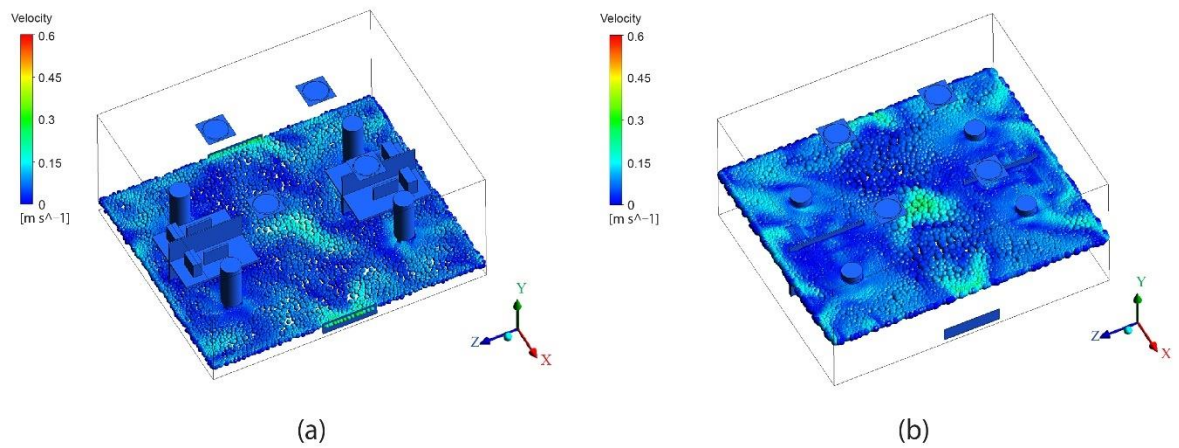


Figure 12 Case 1 (a) 0.1 m from the floor (b) 1.1 m from the floor

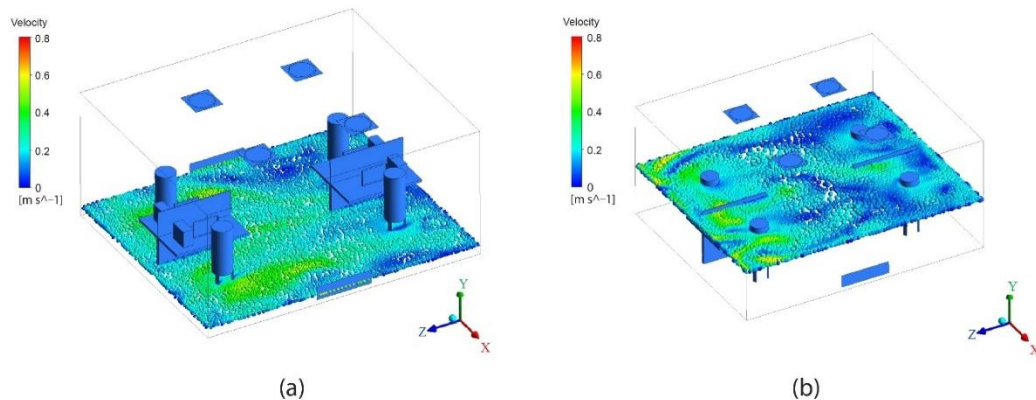


Figure 13 Case 2 (a) 0.1 m from the floor (b) 1.1 m from the floor

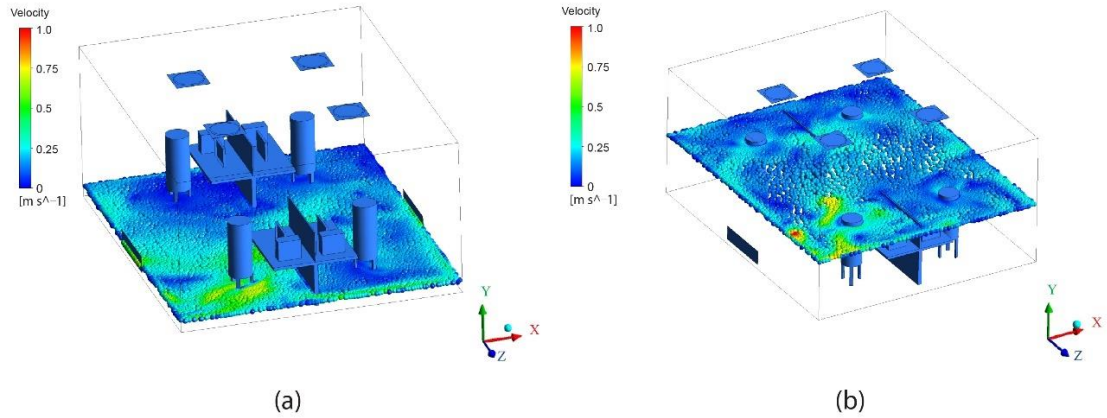


Figure 14 Case 3 (a) 0.1 m from the floor (b) 1.1 m from the floor

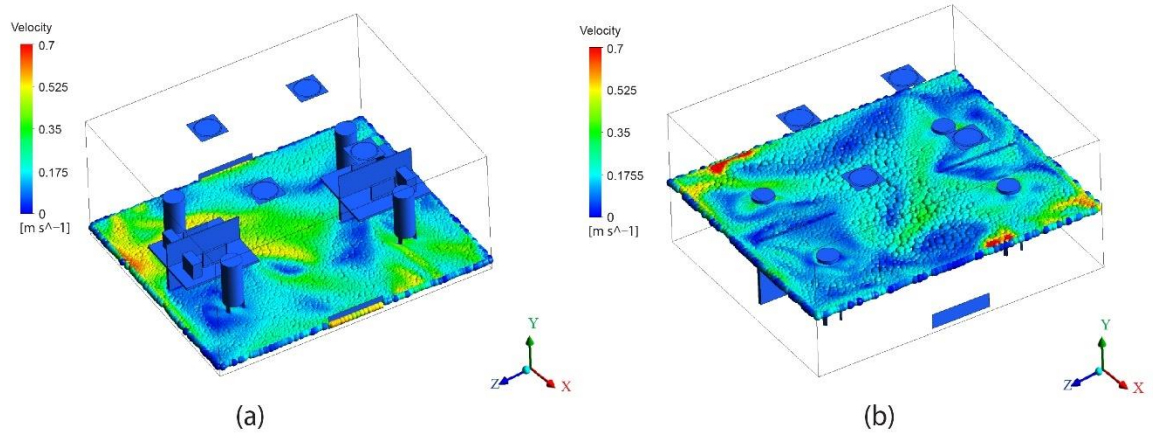


Figure 15 Case 4 (a) 0.1 m from the floor (b) 1.1 m from the floor

Table 5 shows the range of air velocities set for different zones named as low, medium and high air velocity zones. The low air velocity zones were established for normal metabolic rate occupants (1.2 met), while medium and high air velocities zones were established for occupants with metabolic rates ranging from 1.4 met and 1.6 met, respectively.

Table 5 Velocity Ranges for Establishing Variable Air Velocity Zones

Velocity Zones	Air Velocity Range (m/s)
Low	0.07 – 0.15
Medium	0.30 – 0.45
High	0.45 – 0.65

PMV-PPD Calculations

The PMV and PPD values were calculated at fixed points for all four zones, as shown in figure 6(b). These PMV-PPD values for minimum and maximum room temperatures i.e., 23°C and 25°C are shown in figure 16. The mean values of air temperature and air velocities were taken to get PMV-PPD votes for the four cases. In case 1, due to uniform air velocities, PMV values reached the -0.5 limit for the lower air temperature of 23°C but were still within acceptable limits. Figure 16(b) shows that with a

2°C increase in temperature, the feeling of thermal comfort is moved towards slightly warm. In case 2, the higher air velocities for zones 1 and zone 2 for occupants with higher metabolic rates results in a PMV-PPD under the acceptable range. Whereas in case 3, due to a change in the airflow distribution strategy (compared with two ACBs), it resulted in a PMV-PPD index within -0.5 to +0.5. Case 4 involved low, medium and high-level air flow distribution for a single office space, which resulted in all occupants in four zones remaining within acceptable thermal comfort limits. The PMV values at 23°C were towards slightly cool and at 25°C the occupants felt slightly warm.

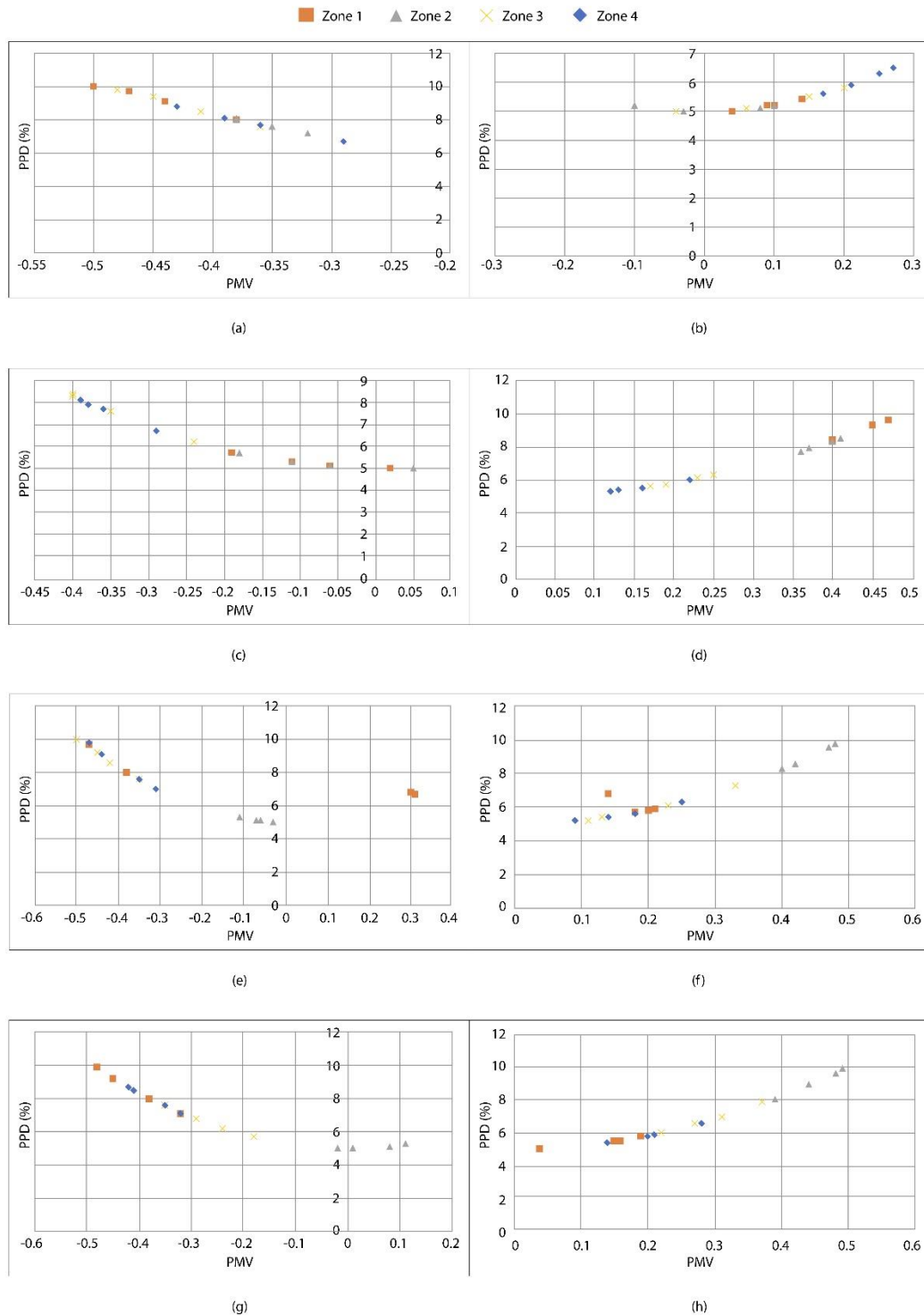


Figure 16 PMV PPD Indices (a) Case 1 at 23°C (b) Case 1 at 25°C (c) Case 2 at 23°C (d) Case 2 at 25°C (e) Case 3 at 23°C (f) Case 3 at 25°C (g) Case 4 at 23°C (h) Case 4 at 25°C

Draught

The Draught Rate for zones with sedentary activity (normal rate of 1.2 met) was calculated for all scenarios, as shown in Table 1. The local air velocity and temperature points were measured at 0.1 m (ankle level), 0.6 m (neck level) and 1.1 m (head level of a seated person). The air velocities at 0.1 m and 1.1 m from the floor were more observed, so the DR for these positions is displayed here. Based on ISO 7730 Standard [27], a DR of less than 10% comes under Category A and a DR of less than 20% comes under Category B. The calculations were made for temperatures between 23-25°C for all four cases and the horizontal measuring positions were taken at points D1(a) and D1(d) (Zone 1); D2(a) and D2(d) (Zone 2); D3(a) and D3(d) (Zone 3); D4(a) and D4(d) (Zone 4). The zones with occupants with a higher metabolic rate (as shown in Table 1) were not considered due to the need for establishing high air velocity zones to compensate for extra heat release from a higher metabolism. Figure 17 shows the DR at 23°C, which have greater DR values than at 25°C, as shown in figure 18. Most DR values for all cases were in category B, with the DR at ankle level slightly higher than head level in most of the cases.

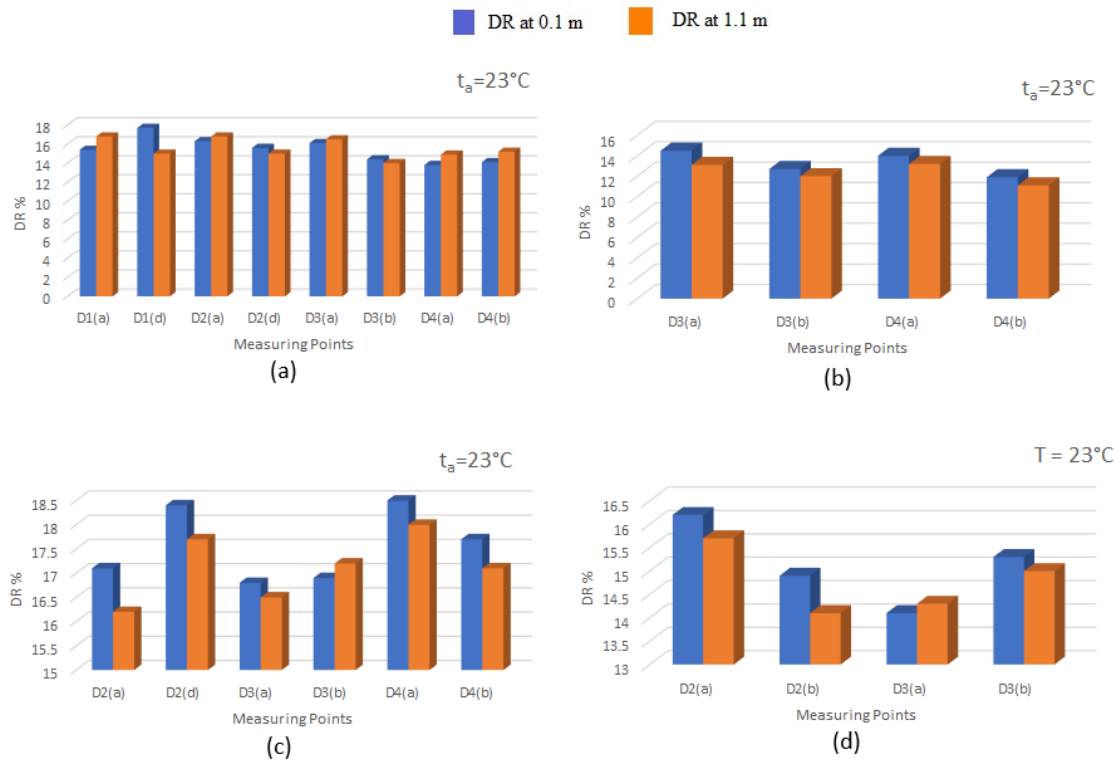


Figure 17 DR at 23°C (a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

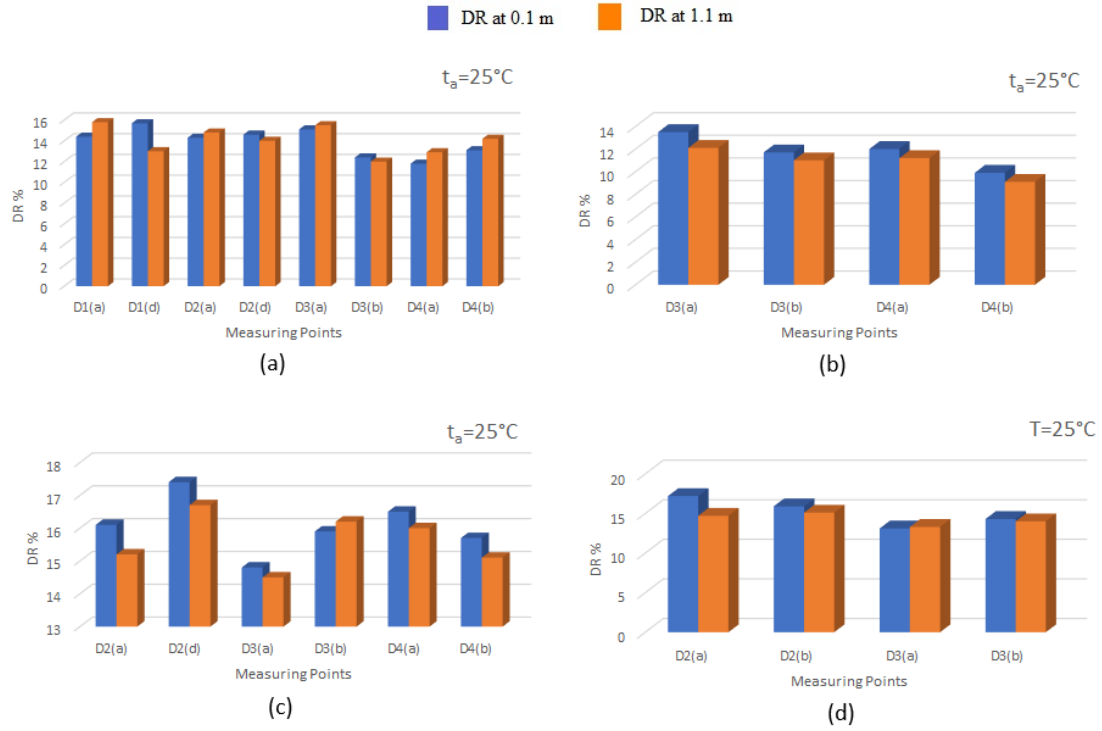


Figure 18 DR at 25°C (a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

Vertical Temperature Difference and Room Stratification

For all four cases, the vertical temperature difference between the head and ankles was found by measuring local air temperature at a height of 0.1 m and 1.1 m near the dummies. For all cases, the vertical temperature difference was measured with a temperature difference reaching up to 1.5°C for the room temperatures of 25°C, as shown in Figure 19. This temperature difference was found to be acceptable according to ISO 7730 Standards for local thermal comfort [27].

Like a MV system, a precision ventilation system maintained both horizontal and vertical temperature uniformities, even with variable air velocity zones. The vertical temperature difference was also found up to 2°C between floor level (0.1 m from the floor) and the ceiling level (0.2 m from the ceiling). The temperature difference (between floor and ceiling) was seen more over the zones with greater heat release from the occupants in the form of higher metabolic rates. In some cases, despite variable air velocity zones having air velocity differences of 0.5 m/s, the measured horizontal temperature difference of all four zones was also found to be less than 1°C.

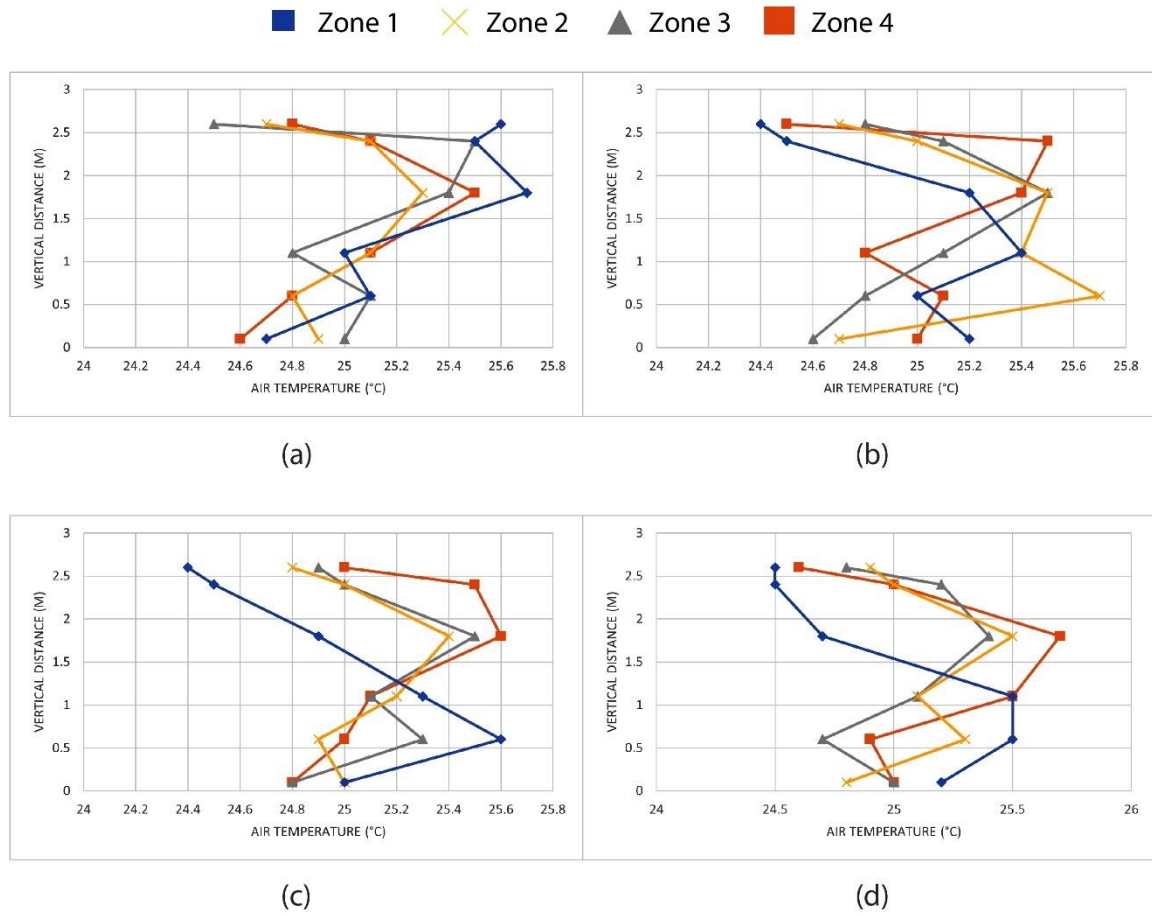


Figure 19 Temperature Measuring Points at 25°C (a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

Energy Savings

Table 6 shows the energy demand of both heating and cooling at 23°C and 25°C. The cooling set point was raised by 2°C while changing the heating temperature setpoints from 21°C to 23°C to make precision ventilation work entire year.

Table 6 Annual energy demands for heat and cooling operations

Function	Heating setpoint 21°C and Cooling setpoint 23°C (kwh/m ²)	Heating setpoint 21°C and Cooling setpoint 25°C (kwh/m ²)	Heating setpoint 23°C and Cooling setpoint 25°C (kwh/m ²)
Heating	29.7	29.7	37.2
Cooling	9.6	3.8	3.8
Total	39.3	33.5	41.0

The results show that the overall annual energy savings when precision ventilation is used only during the cooling season were 15% with the raise of cooling setpoint temperature by 2°C. When the precision ventilation is used entire year (with cooling setpoint 25°C and heating setpoint 23°C) there is increase of energy use by 4%. The energy savings by cooling up to 60% were achieved by raising the setpoint temperature from 23°C to 25°C. Hence for a fixed amount of airflow, precision ventilation

can save significant amount of energy with an increase of room temperatures while keeping all occupants (with different metabolic rates) in the working space satisfied.

4. Limitations

Precision ventilation applications without occupants in the room show that despite an asymmetrical airflow distribution from the ACBs (see Appendix C.), the target air distribution is not achieved without occupants or heat sources in the room. The occupants, as a form of heat source, generates thermal plumes that rise upwards due to buoyant forces. In MV, these buoyant forces play a significant role in mixing room air when cold fresh air is coming from the ceiling [31]. The absence of these heat sources and obstacles (in the form of office furniture) did not let precision ventilation work and form variable air velocity zones. As precision ventilation in large offices establishes multiple variable air velocity zones in the same shared space, this means that absence of any occupants in these office layouts can also influence variable air velocity distribution in the occupied zones.

This study is conducted for a room air temperature between 23°C to 25°C due to the temperature range set in a previous study with one ACB [19]. The precision ventilation applications can be expanded to an even higher air temperature i.e., above 25°C by increasing the amount of primary airflow from the ACBs. In addition, the cooling power per meter square in the room can be increased to evaluate the applications of precision ventilation for higher heat loads.

5. Conclusions

Precision ventilation in large scale open-plan offices involved multiple ACBs to establish low, medium and high air velocity zones by regulating JetCones. The JetCones were positioned to direct high-speed colliding jets with air velocities of 0.8 m/s to the occupants with a metabolic rate of 1.6 met. In contrast, the positions of the JetCones were adjusted to decrease the magnitude of colliding jets to 0.6 m/s for occupants with a metabolic rate of 1.4 met. The PMV-PPD values were maintained within acceptable limits of -0.5 to +0.5. The occupants with 1.2 met were exposed to jets with low air velocities with a DR less than 20% and VTD between head and ankles less than 1.5°C. Both vertical and horizontal temperature uniformity was maintained to a significant extent with vertical temperature stratification less than 2°C. Air velocity distribution profiles in occupied zones in precision ventilation with and without occupants showed a more effective targeted air velocity distribution in cases with occupants in the rooms. An annual energy savings up to 15% were reached by an increase of cooling setpoint temperature by 2°C.

6. Acknowledgements

The authors express their appreciation to Lindab A/S for research support and funding.

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Appendix A: Uncertainties in measurements

The air velocity measurements carried out can be influenced by both systematic and random errors. The main sources that may cause these types of errors in the experimental setups are from the measuring instrument, followed by the correction of the measuring instrument and handling of the measured values. These uncertainties caused by these sources were evaluated using statistical analysis.

A-1 Measuring Instrument and Calibration

The uncertainties caused by the measuring instrument is often removed by calibrating the instrument. The Dantec anemometers used in these experiments had a measuring range of 0.05 m/s to 5.0 m/s with a precision of $\pm 2\%$ in the range 0.05 to 1 m/s and a precision of $\pm 5\%$ in the range 1.0 to 5.0 m/s, as stated by the manufacturers. The same instruments and datalogger were used in all three experimental setups and were calibrated at the indoor climate lab at Aalborg University by following the procedure as explained here [32]. Figure A-1 shows the results for a single calibrated anemometer. The voltage signal from the anemometer was converted to the corresponding air velocity, as shown in figure A1, based on a calibration file.

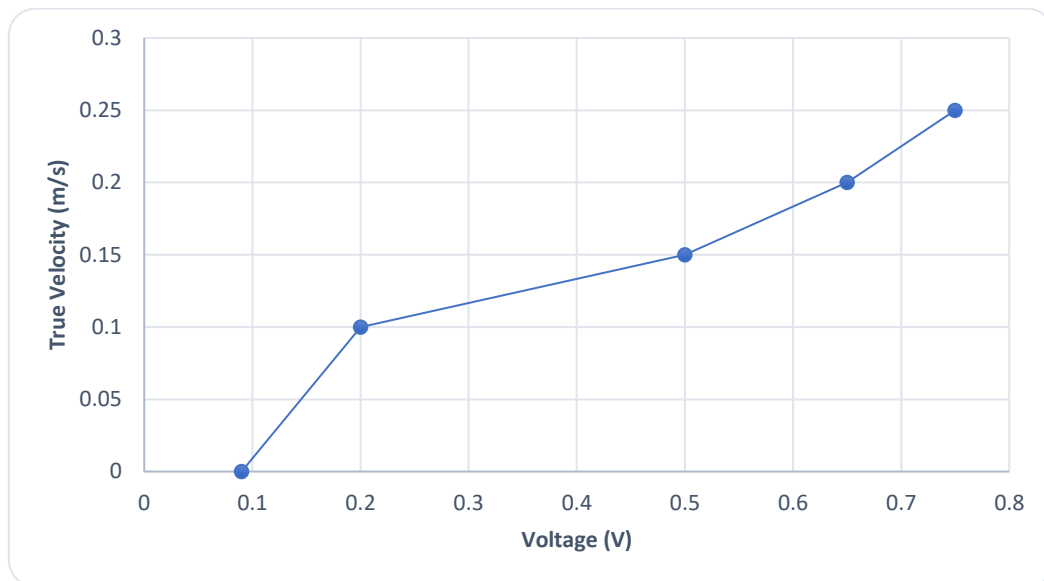


Figure A-1 Example of a calibration curve for a single anemometer

The calibration file for each anemometer was loaded into the program with an instrumental uncertainty of 2%. This range was found to be acceptable for the measurements carried out in this study.

A-2 Handling

In the laboratory setups for all three experiments, the air velocities were measured at fixed heights using bulb shaped hot sphere anemometers. The uncertainty of handling was found after repeated air velocity measurements at 8 points (near the heat sources) at a time interval of 180s in the testing room. An uncertainty of 4% was found after taking average values of conducted measurements.

A-3 Overall Uncertainty

From this data, the uncertainties were mainly from the measuring instrument, procedural handling and random error given by the Mean Root Square method:

$$m_m = \sqrt{m_1^2 + m_2^2 + m_3^2}$$

Here, m_1 is the uncertainty by the instrument of 2%, m_2 is the uncertainty due to experimental procedure, which was calculated to be 3% and m_3 due to reading error was ignored as all measurements were recorded and observed digitally. Therefore, the overall uncertainty m_m was approximately 4.5%.

Appendix B: Mesh Independence Test

A mesh independence test was carried out along Y-direction at line $y_1 = -0.8$ $y_2 = -1.2$ in the middle of the room. The variations after 4.6 million elements were not found to be significant (see Figure A-2) and this mesh number was considered suitable for the calculation to avoid computational cost.

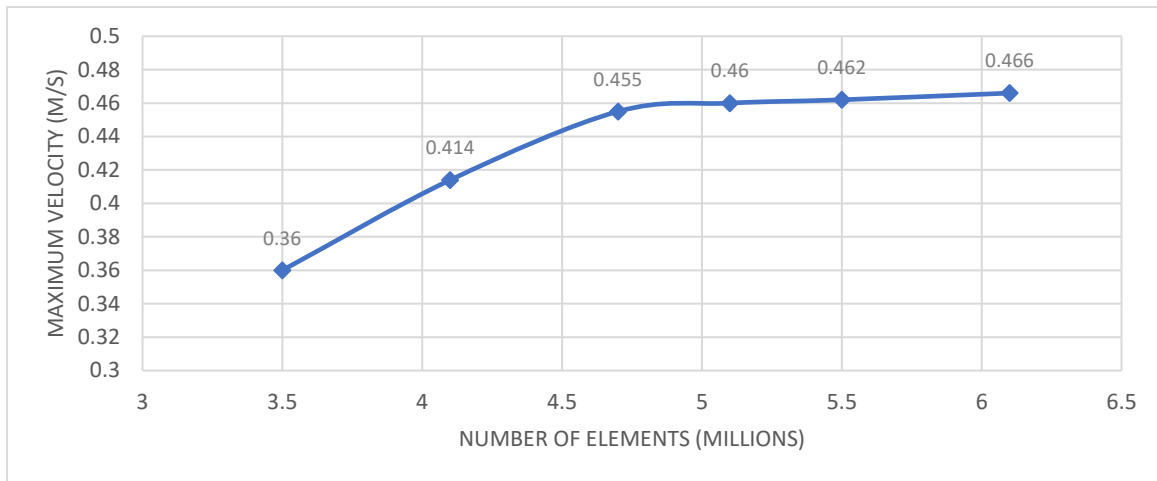


Figure A-2 Mesh independence test

Appendix C: Airflow Along Ceiling

Figure A-3 shows the airflow distribution from four ACBs for four cases with occupants in the room. The airflow distribution (without occupants in the room) at the ceiling level was identical to airflow distribution at the ceiling level with occupants in the room. However, the targeted airflow distribution is not precisely achieved without occupants in the room (see Figures 8-10).

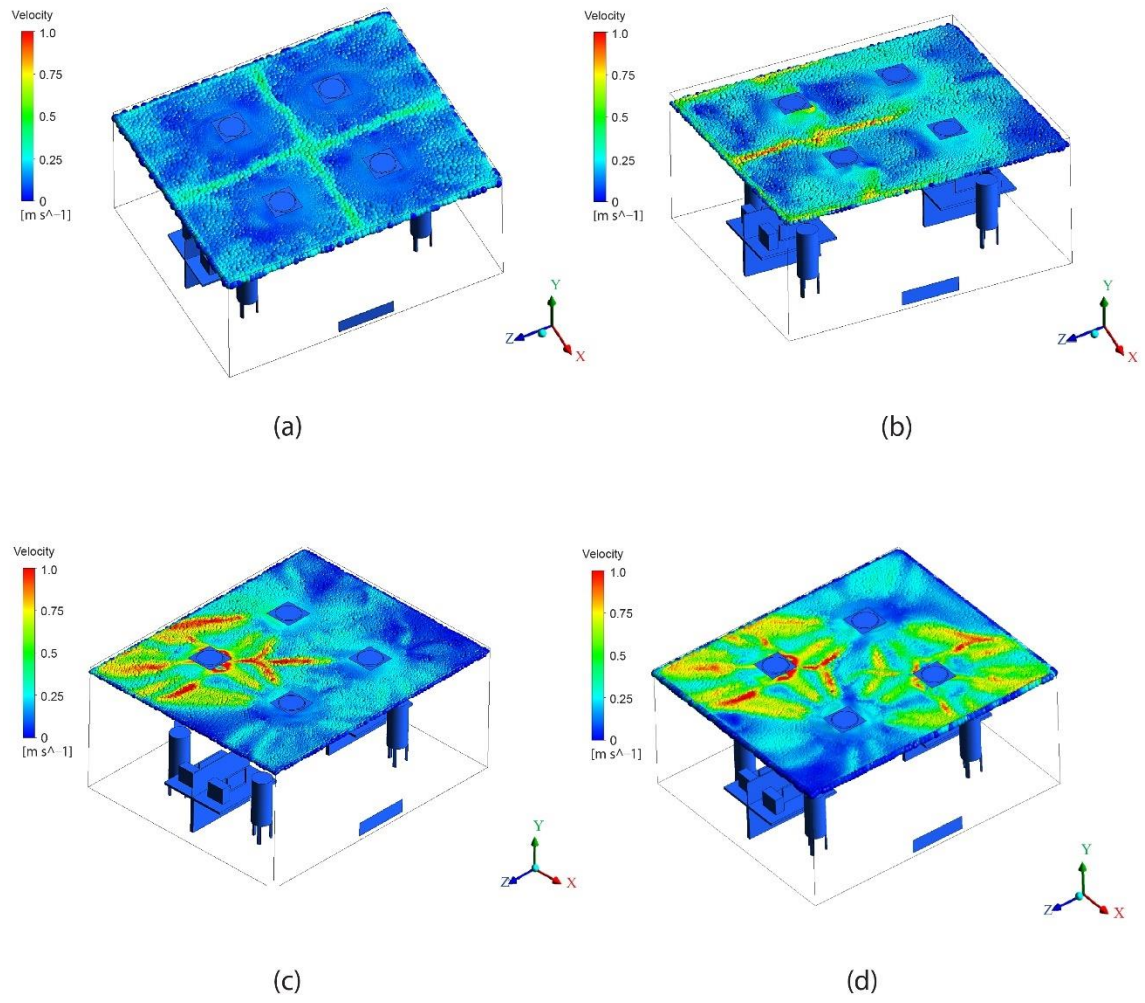


Figure A-3 Air velocity distribution near ceiling (a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

PART III

DISCUSSION AND CONCLUSIONS

CHAPTER 11. DISCUSSION

11.1. FACTORS INFLUENCING PRECISION VENTILATION

The performance of precision ventilation depends upon how well-targeted airflow distribution is implemented while considering certain parameters influencing its performance. These factors include supply airflow and induction, strength of airflow and colliding jets, temperature gradient (between supply and room air) and humidity control and presence of occupants.

Induction in the room and internal induction in ACBs are a complex phenomenon to simultaneously simulate in CFD. Therefore, the simplification of ACB geometry was made and internal induction (within the ACB) was not considered. The simplified ACB geometry included the resultant of both primary and secondary airflows from laboratory tests as supply air in simulations. In addition, it should be mentioned that the supply air (with same properties both in simulations and experiments) plays a more significant role in room air distribution patterns than induced air [254].

Furthermore, precision ventilation uses variable throw lengths and colliding jets to establish micro-climate zones. The supply of airflow out from the ACB moving along the ceiling surface (using Coanda effect) is affected by variations in the pressure drop of the ACB unit and the difference of temperature between the room and supply air [206]. In a study [206], a temperature difference of 4.7°C showed an apparent detachment of supply air jets from the ceiling at a certain distance compared to a temperature difference of 2°C, which showed stronger jet attachment to the ceiling. In addition, colliding jets are also affected by the temperature gradient (between the supply and room air) and thermal plumes (see figures 5 and 6, Paper III). The large temperature differences result in a stronger depth of colliding jets towards the occupied zones. Therefore, for a fixed amount of primary airflow in the current study, the airflow was adjusted through JetCones and a temperature difference of 3°C to 3.5°C was kept constant in all cases to enable throw lengths and colliding jets to be stable in establishing variable air velocity zones. A variation in RH also influences other environmental factors, like air temperature leading to an impact on the thermal comfort of occupants [95], [96]. Whereas some studies contradict this claim and report that RH has a minimal effect on thermal comfort if other environmental factors are kept the same [95], [98]. A RH above 70% can negatively affect the occupant's thermal comfort [97], therefore RH was kept at 60% in simulations.

The current research was conducted at steady state conditions with stationary heat sources and adiabatic wall conditions without doors and windows. [255], [256] shows that the presence of a walking occupant and a room with doors and windows can significantly impact room air distribution patterns, which can also have an impact on precision ventilation performance. In mixing ventilation, thermal uniformity is

established in a room by the significant contribution of buoyant forces from the human body, which gets mixed with cold fresh air from the ceiling [61]. Furthermore, heat sources in a room with ACBs also has a significant impact on room air distribution patterns and may increase draught risk [257]. However, in the present study, even with the presence of high air velocity zones created for high heat sources (high metabolic rates) in the same room, the DR for occupants with normal metabolic rates in other zones was within acceptable limits. In addition, precision ventilation with occupants in a room achieved stabilized and precise conditions compared to a situation without occupants in a large room (see Paper VII). The cases without occupants showed that micro-climatic zones cannot be established precisely in the occupied zone due to the absence of office furniture and heat sources. This means that the absence of any occupants in an open-plan office can influence the formation of different air velocity zones.

11.2. PRECISION VENTILATION PERFORMANCE

The performance of a precision ventilation system can be evaluated by how well individual thermal comfort is maintained for office occupants with different metabolic rates. Precision ventilation involves ACBs that are popular in providing a satisfying thermal comfort environment for occupants through their uniform airflow distribution from the ceiling in a room. ACBs use the Coanda effect to provide fresh air to the occupied zones and maintains thermal uniformity in a room by allowing cold fresh air to become mixed with warm room air. Considering the performance and yielding satisfactory thermal comfort in enclosed places, ACBs outclass conventional HVAC systems like FCUs [231], [233]. In this research, precision ventilation in open-plan offices has been examined by using ACBs with JetCones and metabolic rates from 1.2 met to 1.6 met were used (both in simulations and experiments) to create the demand for having different local air velocities. [258] used occupants in offices involved in different activities, which resulted in an average metabolic rate of 1.4 met during working hours and showed that this can increase up to 1.7 met when the occupant leaves the office chair.

The ACBs come in different shapes and sizes (detailed in Paper IV) and a recent experimental study [259] suggested using $0.6\text{ m} \times 0.6\text{ m}$ 4-ways ACB to have better local thermal comfort than $1.2\text{ m} \times 0.6\text{ m}$ 4-ways ACB. In addition, [260] shows that symmetrical heat source distribution in an office room with ACBs leads to a better thermal comfort performance of ACBs than asymmetrical heat source distribution. Therefore, in this research on precision ventilation, a 4-way ACB with a JetCone feature was used, and variable air velocity zones were established by placing heat sources in the room symmetrically to yield better thermal comfort performance.

The cooling capacity of 15 W/m^2 is considered a minimum according to EN 15116 Standard [261] and should not exceed 80 W/m^2 [203]. Furthermore, while using ACBs in full-scale room experiments, 50 W/m^2 and 80 W/m^2 in [262], and 45 W/m^2 to 92

W/m^2 in [259] showed that local thermal discomfort increases with the increase of heat gains. Therefore, a heat load of approximately 25 W/m^2 was applied to all three office rooms (Papers V-VII) to test precision ventilation applications.

To achieve acceptable individual thermal comfort and IAQ, a PV system provides a solution to control environmental conditions locally around occupants (see section 3.7, Chapter 3). In this thesis, one of the applications of PV systems to provide individual thermal comfort is achieved by having ACBs with JetCones mounted on the ceiling. Even some PV systems involve ceiling-mounted ATDs, such as background mixing ventilation systems, coupled with localized systems like round moveable panels, desk fans, etc. to achieve local thermal comfort [247]–[249]. These multiple ATDs in the occupied zone are one of the setbacks of the PV system, which does not make these systems fit well with office aesthetics. Whereas in [256], ceiling-mounted PV devices were only used to overcome the disadvantages of PV but did not prove effective when compared to conventional PV systems. Precision ventilation involves ACBs with JetCones to establish micro-climatic zones without the need for any ATDs in the occupied zone and to avoid the direct blow of supply air on the occupant.

The high sensible cooling demands in offices are met by the use of multiple ACBs [195]. This thesis involves the investigation of achieving precision ventilation in offices starting from one ACB unit to multiple ACB units (Papers V-VII). There are very few studies conducted on colliding jets and often these high-velocity jets are regarded negatively as a source of draught [227]. The application of precision ventilation with multiple ACBs makes use of high-velocity colliding jets to establish different air velocity zones, regardless of the need for having physical walls or partitions (Papers VI-VII). The magnitude of the colliding jets was observed to increase with the number of ACBs and this magnitude was controlled by the JetCone adjustments. The air velocities of the colliding jets up to 0.5 m/s for two ACBs and 0.8 m/s with four ACBs were observed in some cases, while smoke tests conducted to observe colliding jets (see Figures 5b and 6b, Paper III) also showed denser smoke in the areas of collision.

Precision ventilation with one ACB was carried out for room temperatures between $23\text{--}26^\circ\text{C}$ to evaluate the maximum room temperature limit while maintaining individual thermal comfort for everyone. In this case [215], at 23°C , PMV values were exceeded -0.5 even at air velocities 0.15 m/s . While at 26°C higher air velocities, the PMV values exceeded the $+0.5$ limit. Therefore, in the case of multiple ACBs, local air velocities were further decreased (see Table 5, Paper VII) for temperatures up to 23°C to allow occupants at lower metabolic rates to feel satisfied within a range of $-0.5 < \text{PMV} < +0.5$. The ISO EN 7730 Standard [64], [67] gives guidance to increase air velocities for higher metabolic rates. For a RH of 50%, an operative temperature of 24°C and clothing insulation of 0.5 clo , PMV values within the -0.5 to $+0.5$ range are achieved with air velocities raised up to 0.5 m/s for occupants with 1.4 met and 1 m/s

for 1.6 met, respectively [64]. In the studies [121]–[124], [263], air velocities were increased to a range of 1 m/s to 1.5 m/s for making the thermal environment comfortable for occupants at higher temperatures up to 31°C. Whereas precision ventilation works with fixed primary airflow from a fan and uses JetCones to distribute this airflow into the room. For a fixed amount of airflow in the room, occupants with higher metabolic rates require increased air velocities while keeping occupants with normal metabolic rates satisfied with low air velocities. At higher air temperatures, occupants with a high metabolic rate require even higher air velocities to offset the extra warmth created by higher air temperatures. Therefore, for distribution of 50 l/s in a room with two occupants (Paper V), a temperature limit of 25°C was set to keep all occupants satisfied within the -0.5 to +0.5 PMV range. The air velocities approximately ranging from 0.1 m/s – 0.65 m/s were supplied to these occupants with metabolic rates from 1.2 met to 1.6 met. For temperatures above 25°C, precision ventilation can still work with an increase of airflow from ACBs to the room to satisfy occupants with higher metabolic rates. Therefore, for a fixed amount of airflow, the precision ventilation strategy can only work for a certain temperature range. There is a need to investigate higher air temperatures and higher airflows to provide knowledge about the performance of precision ventilation. Furthermore, a vertical temperature difference (between head and ankle) of more than 3°C can be a source of local discomfort according to ISO 7730 and ASHRAE 55 Standards [64], [39], [67], but in some studies associated with displacement ventilation, temperature difference of 4-5°C was also accepted [264]–[267]. The vertical temperature difference using precision ventilation measured between head and ankles was less than 1.5°C. Despite the formation of variable air velocity zones, the horizontal temperature difference between different air velocity zones was also found less than 1°C.

In PV systems, significant energy savings are achieved by increasing the setpoint to a higher degree or by a reduction in airflow rate [242]. Energy savings by cooling, ranging from 17% to 51% were reached by raising the setpoint temperature from lower room temperatures i.e., 23-24°C to 26°C in different studies involved with PV [237], [239], [246]. For a fixed 50 l/s from each ACB in a room, the cooling setpoint from 23°C to 25°C was raised to implement precision ventilation applications at higher room air temperatures to save energy. A reference single-story office building was selected in Be18 software to calculate annual energy savings at different heating and cooling temperature setpoints (see Table 6, Paper VII). An annual energy savings of 15% were achieved for both cooling and heating because of increasing the cooling set point temperature by 2°C. Energy savings by cooling were achieved at a rate up to 60%. As precision ventilation is a modified form of a mixing ventilation system, during winter the same system can work as a conventional mixing ventilation system with all JetCones at the same level to carry out heating operations.

CHAPTER 12. CONCLUSIONS

12.1. CONCLUDING REMARKS

Precision ventilation can act as a new potential ventilation strategy to make the office environment more comfortable and productive for all office occupants. JetCones instead of conventional nozzles brings a novelty in the application of ACBs, which simultaneously achieved low, medium and high-level air velocity zones in a shared office space. For a room temperature range between 23-25°C, the maximum air velocity limits for 1.2 met, 1.4 met and 1.6 met were set at 0.15 m/s, 0.45 m/s and 0.65 m/s, respectively. The variation in local air velocities with respect to occupants' metabolic rate and room air temperature significantly improved their thermal comfort. The PMV range of $-0.5 < \text{PMV} < +0.5$ was maintained along with a PPD of less than 10%. Even the occupants with normal metabolic rates, seated together with occupants with higher metabolic rates that were seated in higher air velocity zones remained unaffected and were satisfied with a DR of less than 20%. The precision of the air velocities was maintained by the movement of colliding jets (regulated by JetCones). As a mixing ventilation system, temperature uniformity was maintained in the space while keeping a vertical temperature difference between the ankle and head at less than 1.5°C. Despite a horizontal air velocity difference of 0.5 m/s between different zones, the horizontal temperature difference never exceeded 1°C. The annual energy savings up to 15% were achieved by raising the cooling setpoint by 2°C.

The results achieved show remarkable success in the field of achieving personalized thermal comfort by utilizing existing technological advancements, which have not currently been reported elsewhere. Indeed, precision ventilation in open-plan offices is a more advanced form of mixing ventilation strategy to achieve individual thermal comfort for everyone.

12.2. FUTURE RESEARCH

The precision ventilation system was tested at different office scales through laboratory and numerical simulations. Additional research is needed to develop algorithms for multiple ACBs to function with different JetCone settings that are controlled automatically. ACBs with motors regulated by a control system needs to be developed to ensure that any change in the position of JetCones in one ACB will not adversely affect the thermal comfort of other occupants in the same room.

Further studies to assess the formation of variable air velocity zones in precision ventilation at higher heat gains in the offices needs to be conducted. The optimization of the configuration of air terminal devices with respect to the number of people and room area must also be carried out. Finally, as mentioned in limitations, future

research needs to be carried to implement the application of precision ventilation in real office buildings.

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Appendix A. Draught Calculations

Draught is a crucial aspect when considering local thermal comfort calculations. The Draught Rate (DR) for the zones with sedentary activity (normal 1.2 met) was calculated for all scenarios. According to ISO 7730 Standard [64], [67], the DR becomes lower for metabolic rates above 1.2 met. The DR of less than 10% is classified as Category A and a DR of less than 20% is classified as Category B [259]. The local air velocity and temperature points were measured as shown in Papers V and VI at 0.1 m (ankle level), 0.6 m (neck level), 1.1 m (head level of a seated person) and 1.8 m (head level of a standing person). Air velocities at 0.1 m and 1.1 m from the floor were more apparent, so the DR for these positions is only displayed here.

DRs with One ACB

Figures A1 to A3 show the DR calculations made for room temperatures 23-26°C for all three cases with a single ACB unit. The horizontal measuring positions were taken at points S2 and D1 (Zone 1), and D1 and N2 (Zone 2). These points were taken for both zones highlighted in Figure 7 of Paper V. Whereas the DR for higher air velocity zones (in cases 2 and 3) was not considered due to the need to supply higher air velocities for occupants with higher metabolic rates.

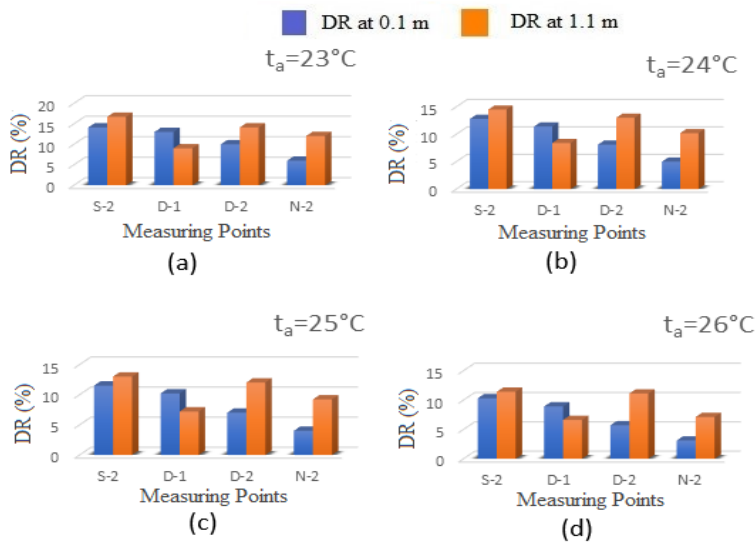


Figure A-1: Case 1 with local air temperatures (a) 23°C (b) 24°C (c) 25°C (d) 26°

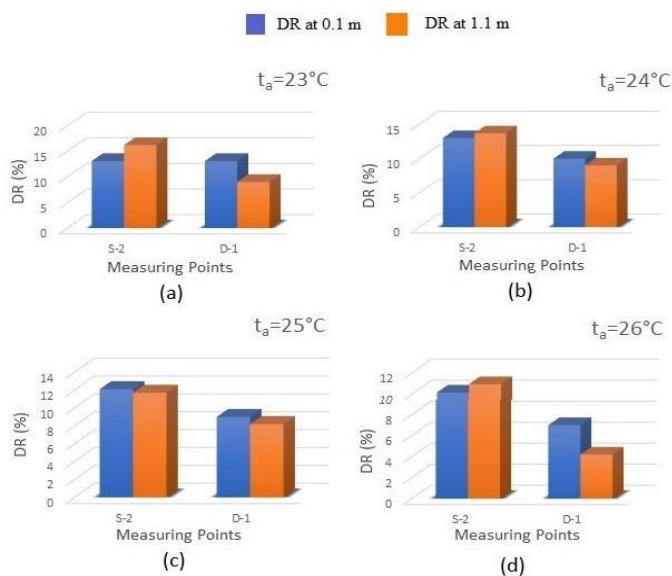


Figure A-2: Case 2 with local air temperatures (a) 23°C (b) 24°C (c) 25°C (d) 26°C

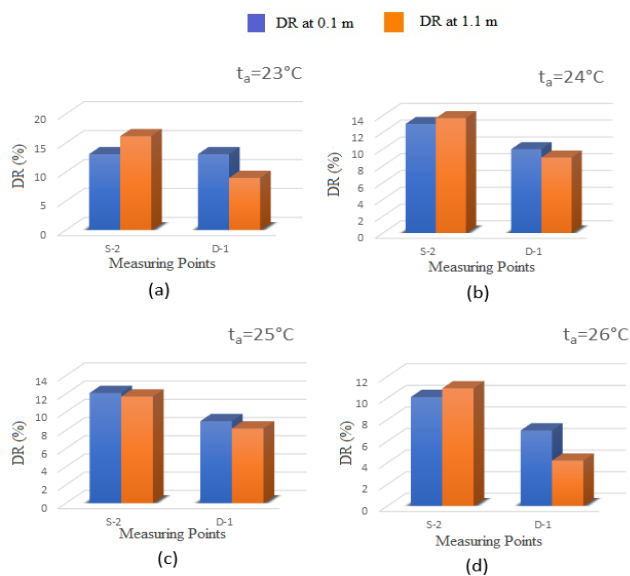


Figure A-3: Case 3 with local air temperatures (a) 23°C (b) 24°C (c) 25°C (d) 26°C

DRs with Two ACBs

Figures A4 to A8 show the DR calculations made for a room temperature of 23°C for all five cases with two ACBs. The horizontal measuring positions were taken at points D1(a) and D1(d) (Zone 1); D2(a) and D2(d) (Zone 2); D3(a) and D3(d) (Zone 3); D4(a) and D4(d) (Zone 4). These points were taken at all four zones highlighted in figure 3 of Paper VI. The zones with occupants with higher metabolic rates (as shown in table 2 of paper VI) were not considered due to the need for establishing high air velocity zones to compensate for extra heat release from a higher metabolism.

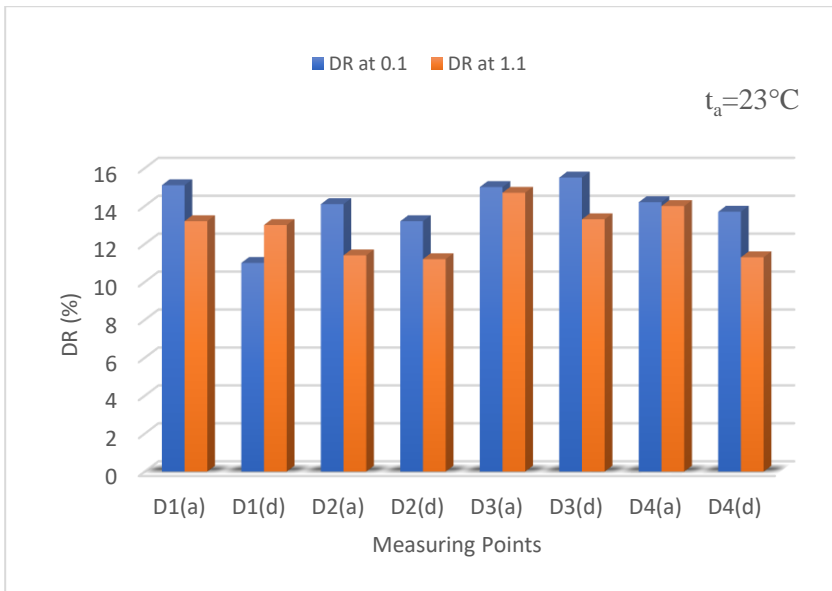


Figure A-4: Case 1 with local air temperature 23°C

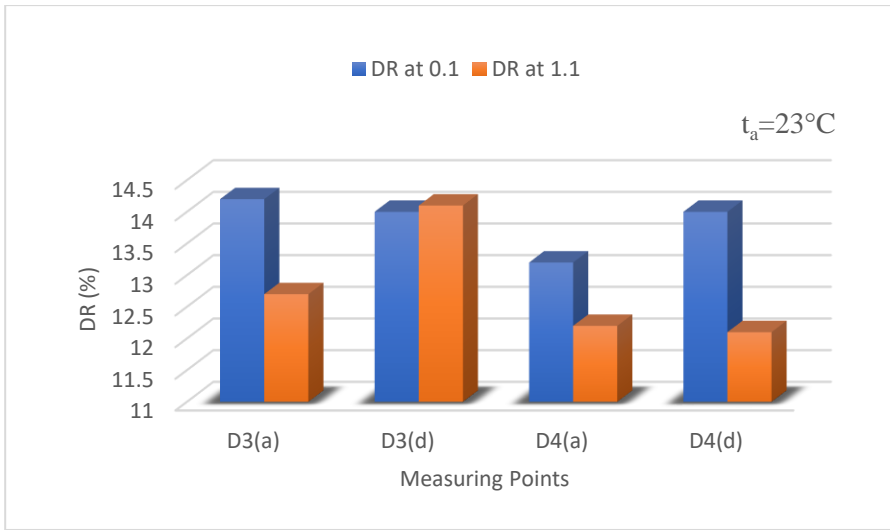


Figure A-5: Case 2 with local air temperature 23°C

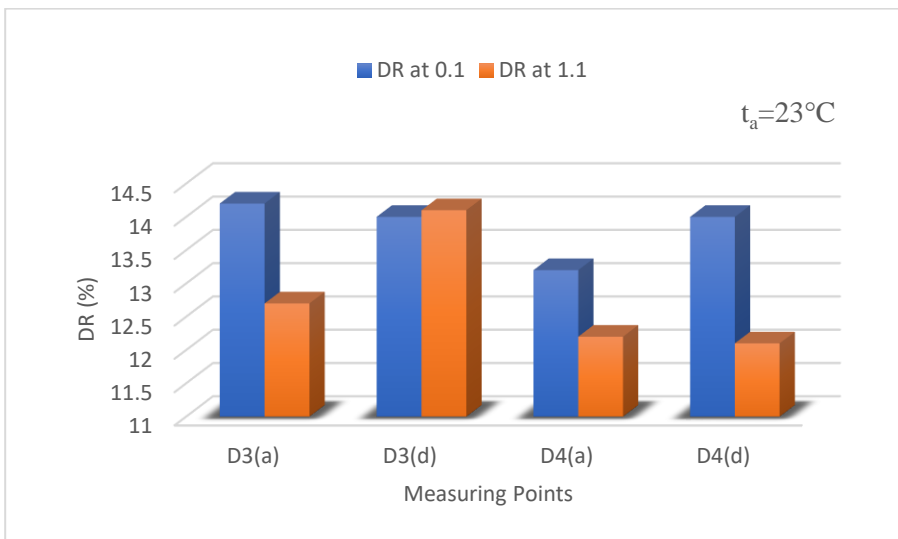


Figure A-6: Case 3 with local air temperature 23°C

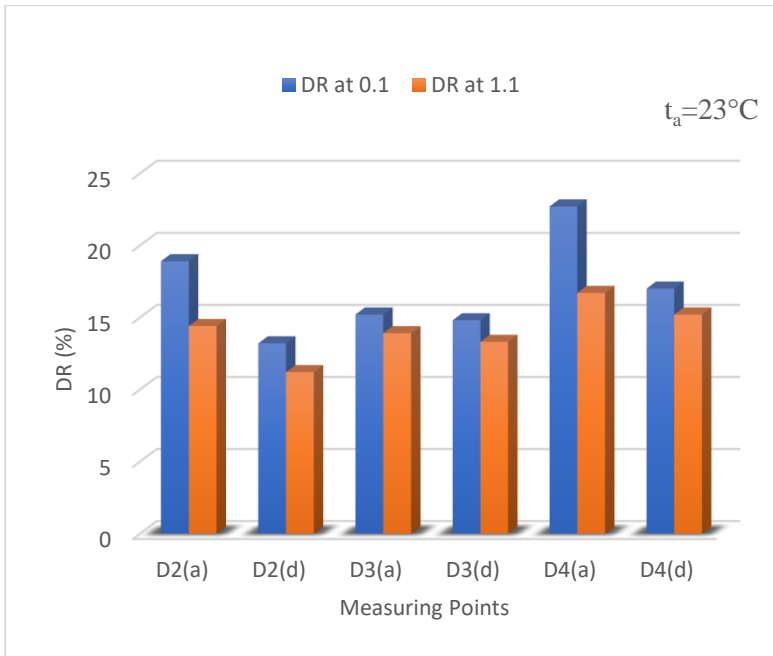


Figure A-7: Case 4 with local air temperature 23°C

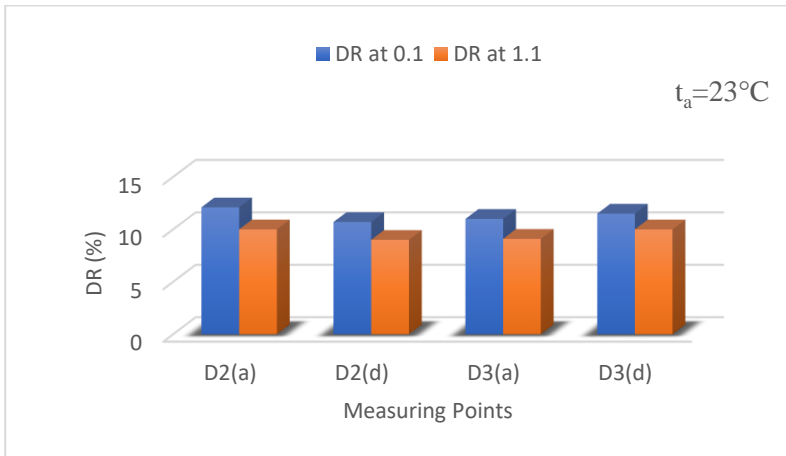


Figure A-8: Case 5 with local air temperature 23°C

Most of the DR values were found within the limits of category B. The DR at ankle level was found to be slightly more than at head level. The DR was within acceptable limits despite the presence of higher air velocity zones in the same shared space. The DR in one of the cases (Figure A-7) with two ACBs was found to be higher than 20%,

which was reduced by the change in precision ventilation strategy with four ACBs in Paper VII.

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