

CFD-Based Quantification and Optimization of Local Indoor Air Quality of Different Ventilation Systems

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***CFD-Based Quantification and Optimization of Local Indoor Air
Quality of Different Ventilation Systems***

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ABSTRACT

Indoor Air Quality (IAQ) is an important consideration in the design of buildings to promote the well-being of occupants. Previous indoor environmental studies quantified and optimized air quality based on boundary flow rates evaluated at the window, ventilation supply inlet, and other openings, or using room volume-averaged values. However, due to the existence of imperfect mixing conditions, the aforementioned quantities do not represent the entirety of indoor environments. For example, corners, windows, near ventilation inlet regions, occupant zones, and outlet may have varying indoor air qualities – creating distributions. With field experiments being very costly and hard to implement owing to spatial and temporal resolutions and the number of measuring devices needed, this dissertation aims to quantify and optimize local air quality in a room in an urban scenario and in an office, with different ventilation modes using computational fluid dynamics (CFD). First, this study analyzes an idealized room model in an urban environment with cross-ventilation affected by the sheltering effect. With two window opening cases: parallel and perpendicular to the flow direction, this determines the IAQ by placing several scalar (representative contaminant) point sources and creates corresponding contaminant distributions. The study explains the effects of the air introduction in the cross-ventilation model and the source location on the resulting contaminant distributions. Furthermore, the study also quantifies the local ventilation and nature of contaminant transport using ventilation efficiencies namely, Age of Air, Purging Flow Rate, and Net Escape Velocity. Second, this study investigates the IAQ in an office room with a hybrid displacement-personalized ventilation system while having more realistic contaminant sources coming from two occupants through their exhaled carbon dioxide (CO₂). With 25 simulation cases determined by Taguchi design of experiment coupled with “technique for order preference by similarity to ideal solution” (TOPSIS), the study optimizes the IAQ together with occupants’ thermal comfort and ventilation system’s energy requirement, by varying the ventilation system inputs: displacement ventilation (DV) supply flow rate, DV supply temperature, DV recirculation, personalized ventilation (PV) supply flow, and PV supply temperature. This also compares optimization procedures based on occupant-centric IAQ value such as inhaled CO₂ concentration versus room volume-averaged CO₂ concentration. Third, also through the use of Taguchi-TOPSIS procedure and with the same office room, this study explores the effect of changing the ventilation system from hybrid displacement-personalized to hybrid mixing-personalized. This compares the differences between the optimized results of occupant-centric IAQ, as well as thermal comfort and energy requirements, which then eventually determines which hybrid system is better. The summaries of the chapters’ contents of this dissertation are as follows:

Chapter 1 introduces the general motivation of the dissertation and the overarching objectives, as well as the parallel objectives.

Chapter 2 outlines the methodology used in the dissertation, particularly the computational fluid dynamics (CFD) technique and its corresponding turbulence modeling.

Chapter 3 discusses the spatial distributions of several contaminants placed in different locations inside an idealized cross-ventilation model surrounded by urban-like block arrays. The outcomes showed varying distributions for every contaminant, even for contaminant sources placed near each other, which can be attributed to imperfect mixing conditions. Utilizing ventilation efficiencies age of air (AOA), purging flow rate (PFR), and net escape velocity (NEV), local ventilation and contaminant transfer mechanisms were discussed. The results indicate that values of AOA were generally low in the inlet of the cross-ventilation model, while high in the corners. On the other hand, an inverse correlation was seen between AOA and PFR, which can be proven by comparing their equations. As can be discussed by NEV, in addition to flow velocity, turbulence becomes significant as driving force for contaminant transfer following the direction of high to low concentration. This can particularly be observed in regions where the mean flow is weak.

Chapter 4 focuses on the quantification and optimization of IAQ together with thermal comfort and ventilation energy requirements in a two-occupant office room with hybrid displacement-personalized ventilation (DV-PV) system. The results showed significant differences in the metrics used for optimization, particularly the volume-averaged values (volume-averaged temperature and CO₂ concentration) versus occupant-centric values (average skin temperature and inhaled CO₂ concentration), owing to imperfect mixing conditions. Inhaled CO₂ concentration appeared to be heavily dependent on the resulting local flow field. The occupant-centric optimization yielded lower energy requirements. The study also showed significant effects of thermal plumes and parameters of the main ventilation system, making the PV have a setting of 6 L/s to function as intended.

Chapter 5 compares the occupant-centric IAQ, thermal comfort, and ventilation energy requirements of hybrid displacement-personalized ventilation (DV-PV) in Chapter 4 with that of the hybrid mixing-personalized ventilation (MV-PV) system. DV-PV generally produced better results due to its contaminant and thermal stratification. However, for the same settings, IAQ of MV-PV may sometimes be better than its DV-PV counterpart, especially when the supply jet of the main ventilation goes directly to the occupants. Both hybrid systems yielded their best setting when the personalized ventilation for both occupants was turned off, implying that when the system is well-designed, the use of PV might not be necessary to maintain the

inhaled concentration lower than the limit while keeping within the acceptable range of skin temperatures and at low energy requirements. In addition, the Taguchi-TOPSIS method failed in the optimization of MV-PV due to non-linear relationship between input and output parameters.

Chapter 6 finally summarizes the key findings of this study and recommended future works that can be explored and be used to improve the current studies.

KEYWORDS: Computational fluid dynamics, Large eddy simulations, Reynolds-averaged Navier-Stokes, Cross-ventilation, Hybrid ventilation, Displacement ventilation, Mixing ventilation, Personalized ventilation, Indoor air quality, Thermal comfort, Ventilation energy requirements, Ventilation indices, Net escape velocity, Purging flow rate, Age of air.

TABLE OF CONTENTS

ACKNOWLEDGMENT	i
ABSTRACT	iii
LIST OF FIGURES.....	x
LIST OF TABLES.....	xiii
ACRONYMS	xiv
NOMENCLATURE	xv
CHAPTER 1 – BACKGROUND OF THE STUDY	1
1.1 Motivation.....	1
1.2 Objectives of the Study	2
References	3
CHAPTER 2 – GENERAL METHODOLOGY	6
2.1 Computational Fluid Dynamics	6
2.2 Turbulence Model	7
2.2.1 Reynolds-Averaged Navier-Stokes equation (RANS).....	7
2.2.2 Large Eddy Simulation (LES)	9
References	10
CHAPTER 3 – QUANTIFICATION OF LOCAL INDOOR AIR QUALITY IN AN IDEALIZED URBAN CROSS-VENTILATION MODEL	11
3.1 Background.....	11

3.2 Methodology	14
3.2.1 Ventilation Metrics and Efficiencies	14
3.2.2 Simulation settings	18
3.2.3 Validation	20
3.3 Indoor Concentration Fields	22
3.3.1 Instantaneous Scalar Distribution and Velocity Vector Field	22
3.3.2 Mean Scalar Distribution and Velocity Field	24
3.4. Ventilation Indices/Efficiencies Results and Other Summaries	26
3.4.1 Net Escape Velocity	26
3.4.2 Purging Flow Rate	30
3.4.3 Age of Air	31
3.4.4 Relationship of Overall Ventilation Flow Rates and Ventilation Indices	32
3.4.5 Overall Effect of Flow Introduction Owing to the Outdoor Flow Pattern and Window Openings	33
3.4.6 Overall Effect of Scalar Source Location	33
3.5. Conclusion	34
References	36
CHAPTER 4 – QUANTIFICATION AND OPTIMIZATION OF LOCAL AND AVERAGE INDOOR AIR QUALITY AND THERMAL COMFORT IN HYBRID DISPLACEMENT-PERSONALIZED VENTILATION IN AN OFFICE	41
4.1 Background	41

4.2. Methodology	43
4.2.1 Computational domain and boundary settings.....	43
4.2.2 Occupant model and zone definitions	44
4.2.3 Numerical settings, mesh independence, and validation and verification	46
4.2.4 Taguchi DOE and optimization cases	49
4.2.5 Multicriteria optimization: TOPSIS and S/N ratio.....	52
4.3. Result and Discussions	54
4.3.1 Individual measures	54
4.3.2 Multi-objective measures.....	59
4.4. Limitations of PV in Current Study	65
4.5. Conclusion	70
References	71
Appendix	75
 CHAPTER 5 – COMPARISON OF LOCAL INDOOR AIR QUALITY AND THERMAL COMFORT UNDER HYBRID MIXING-PERSONALIZED AND HYBRID DISPLACEMENT-PERSONALIZED VENTILATION IN AN OFFICE.....	 80
5.1 Introduction	80
5.2 Methodology	81
5.2.1 General methodology.....	81
5.3 Result and Discussions	82

5.3.1 Inhaled CO ₂ Concentration	82
5.3.2 Average Skin Temperature	85
5.3.3 Energy Requirements	87
5.3.4 Taguchi-TOPSIS of the 2 Hybrid Systems	90
5.3.5 Optimization of MV-PV	91
5.3.6 Effect of MV settings on the PV	92
5.4 Conclusion	92
References	94
CHAPTER 6 – SUMMARY AND FUTURE WORK	95
6.1. Summary	95
6.2. Future Work	96
6.2.1. For Chapter 3	96
6.2.2. For Chapters 4 and 5	97

LIST OF FIGURES

Figure 1. Diagram of dissertation structure.....	3
Figure 3.1. Control volume (CV) for the NEV computation. The variables CP , CN , CS , CE , and CW represent the concentration values at the numerical grid center. The velocity components uin , $uout$, vin , and $vout$ denote the boundary values of the CV.....	15
Figure 3.2. Schematics of (a) numerical domains in staggered (ST) arrangement, (b) STR openings, and (c) LAT openings (d) Infinite replication of building domain at spanwise and streamwise directions. H ($= 100$ mm) is the building height with opening size equal to 10% of $H2$	19
Figure 3.3. Definitions of passive scalar source locations. (a) STR, and (b) LAT conditions.	20
Figure 3.4. Comparison between indoor velocity contours (upper half figures of (a, b), and (e, f)) and vectors (lower half figures of (a, b) and (e, f)) using CFD (Adachi et al. [32]) and WTE (Ikegaya et al. [23]); current study velocity magnitudes and standard deviations (c, d, g, h) versus WTE (Ikegaya et al. [23]) on the x–y plane at $z/H = 0.5$. (a–d) STR and (e–h) LAT conditions. (a, e) LES (previous), (b, f) WTE, and (c, d, g, h) LES (current) versus WTE comparison of profiles for streamwise and spanwise velocity components. vh is the magnitude of the time averaged horizontal velocity components, and $uref$ is the reference wind speed defined at $z = 4H$. Figures are cited from Adachi et al. [32] and modified for comparison.	21
Figure 3.5. Comparison between current LES indoor velocity magnitudes and standard deviations versus WTE (Ikegaya et al. [23]) for the STR case on the x–z plane at $y/H = 2.0$	22
Figure 3.6. STR and LAT cases instantaneous scalar distribution of $Ci1$ and $Ci4$ at arbitrary time t	23
Figure 3.7. STR and LAT time-averaged scalar distribution of $Ci1$ and $Ci4$	25
Figure 3.8. STR and LAT time-averaged scalar distribution of Cuc	25

Figure 3.9. a) LAT time-averaged scalar distribution of C_{dw} and the fluctuating flow introduction b) from the $y/H = 1.5$ inlet and c) from the $y/H = 2.5$ inlet.....	26
Figure 3.10. STR time-averaged net escape velocity vector field for C_{uc} : (a) Velocity, (b) NEV^* , (c), NEVA, and (d) NEVT.	27
Figure 3.11. Magnified NEVT and NEV^* of (a) C_{uc} and (b) C_{dc}	28
Figure 3.12. NEV^* decomposition computed at the source location of the different scalar distributions for both (a) LAT and (b) STR cases.....	30
Figure 3.13. Purging flow rates of the (a) LAT and (b) STR cases.....	31
Figure 3.14. Age of air of (a) LAT and (b) STR cases.....	32
Figure 4.1. Room layout and schematic diagram of coupled HVAC system.....	44
Figure 4.2. a) Skin segmentation, and b) nasal cavity of computer-simulated person....	46
Figure 4.3. Velocity magnitudes of experimental [15] (bracket) vs. CFD (scatter plot) along line shown under PV flow rate of 13 L/s.....	47
Figure 4.4. Cases and optimization process	52
Figure 4.5. CO_2 concentration values from 25 simulations	55
Figure 4.6. a) Velocity field and b) concentration distribution of Simulation 2.....	56
Figure 4.7. Temperature values from 25 simulations	57
Figure 4.8. DV and PV energy inputs for 25 simulations.....	58
Figure 4.9. Optimization cases values: a) CO_2 concentration; b) temperature; c) energy	62
Figure 4.10. Skin-temperature distributions from optimization. a) Case 1, b) Case 3, and c) Case 4.....	65

Figure 4.11. Concentration distributions of a) optimum Case 3, b) Simulation 4, and c) Simulation 12.....	67
Figure 4.12. a) Velocity field and b) temperature distribution of Simulation 1	68
Figure 4.13. Concentration distribution and velocity vector field of optimum Case 5...	69
Figure 5.1. Room layout and schematic diagram of hybrid MV-PV system	81
Figure 5.2. Inhaled CO ₂ concentration values of the two hybrid systems for the 25 simulations	82
Figure 5.3. Concentration and velocity distributions of Simulation a) 8, b) 14, and c) 18	83
Figure 5.4. Concentration distributions of simulation 1 of a) MV-PV and b) DV-PV system	84
Figure 5.5. Concentration distributions of simulation 15 of a) MV-PV and b) DV-PV system	85
Figure 5.6. Average skin temperature values of the two hybrid systems for 25 the simulations	86
Figure 5.7. Velocity distributions of simulation a) 11 and b) 21 of the MV-PV system ..	87
Figure 5.8. Energy requirements of the two hybrid systems for the 25 simulations	88
Figure 5.9. Outlet temperatures of the two hybrid systems for the 25 simulations	89

LIST OF TABLES

Table 3.1. NEV*/NEVA for each source location.	30
Table 4.1. Summary of computational settings and boundaries	47
Table 4.2. Taguchi design	49
Table 4.3 Neutral Skin Temperatures [37]	50
Table 4.5. Input parameters determined via S/N ratio analysis for each optimization case	61
Table 5.1. Objective function values and corresponding ranks of the separated and combined analyses of the two hybrid systems	90

ACRONYMS

AHU	Air-Handling Unit
AOA	Age of Air
ASHRAE Engineers	American Society of Heating, Refrigerating, and Air-Conditioning
CFD	Computational Fluid Dynamics
CSP	Computer Simulated Person
CV	Control Volume
DOE	Design of Experiment
DV	Displacement Ventilation
GDH	Gradient Diffusion Hypothesis
HVAC	Heating, Ventilation, Air-Conditioning
IAQ	Indoor Air Quality
LAT	Window on Lateral Direction
LES	Large Eddy Simulation
MCDM	Multicriteria Decision Making Methods
MV	Mixing Ventilation
NEP	Net Escape Probability
NEV	Net Escape Velocity
OHU	Outdoor Air-Handling Unit
OZVA	Occupant-Zone Volume-Averaged
PFR	Purging Flow Rate
PPM	Parts per Million
PV	Personalized Ventilation
RA	Reynolds-Averaged
RANS	Reynolds-Averaged Navier-Stokes
RVA	Room Volume-Averaged
SGS	Sub-grid Scale
SIMPLE	Semi-Implicit Method
SST	Shear-Stress Transport
STR	Window in Streamwise Direction
SVE	Scale for Ventilation Efficiencies
S/N	Signal-to-Noise
TOPSIS	Technique for Order Preference by Similarity to Ideal Solution
UDF	User-Defined Function
UFAD	Underfloor Air-Distribution Systems

VF	Visitation Frequency
WTE	Wind Tunnel Experiment

NOMENCLATURE

A_{DV}	Displacement-ventilation inlet area
A_{NC}	Nasal-cavity inlet area
A_{PV}	Personalized-ventilation inlet area
ACH	Air change rate per hour
C	Scalar Concentration
C_{pm}	Perfect mixing concentration
C'	Fluctuating scalar concentration
$\bar{C}$	Mean scalar concentration
D	Molecular diffusivity
D_{eff}	Effective diffusivity
E_{DV}	Displacement-ventilation energy requirement
E_{PV}	Personalized-ventilation energy requirement
FA	Fresh-air-intake flow rate
F_1	Blending Function
f_{res}	Respiratory frequency
H	Building height
h_{DVs}	Displacement-ventilation-supply enthalpy
h_m	Mixture enthalpy
h_o	Outdoor air enthalpy
h_{PVs}	Personalized-ventilation-supply enthalpy
h_{ro}	Room outlet enthalpy
k	Turbulent kinetic energy
L_z	Domain Height
$\dot{m}$	Net Scalar Generation Rate
P	Pressure
P'	Fluctuating pressure
$\bar{P}$	Mean pressure
Q_t	Skin heat flux
R	Recirculation rate
$R_{clothing}$	Clothing thermal resistance
S_ϕ	Scalar Source
t	Time
T_{skin}	Skin temperature

u	Instantaneous velocity
u'	Fluctuating velocity
$\bar{u}$	Mean velocity
u_{BRE}	Nasal-cavity-inlet exhalation velocity profile
u_{BRI}	Nasal-cavity-inlet average inhalation velocity
u_{DV}	Displacement-ventilation inlet velocity
$u_{n,i}$	Velocity perpendicular to the face of each cell
u_{PV}	Personalized-ventilation inlet velocity
u_τ	Friction Velocity
V_{room}	Room Volume
V_t	Tidal volume
$\dot{V}$	Volume flow rate
$\dot{V}_{BR}$	Breathing-air flow rate
$\dot{V}_{DV}$	Displacement-ventilation inlet flow rate
$\dot{V}_{gross}$	Gross Ventilation Rate
$\dot{V}_{inst}$	Instantaneous Ventilation Rate
$\dot{V}_{net}$	Net Ventilation Rate
$\dot{V}_{PV}$	Personalized-ventilation inlet flow rate
δ_{ij}	Kronecker delta
ε	Dissipation rate
ΔA_i	Area of a grid point
ν	Kinematic viscosity
ν_{sgs}	Sub-grid scale kinematic viscosity
ν_t	Turbulent viscosity
ρ	Density
σ_t	Turbulent Schmidt number
σ_{tsgs}	Sub-grid scale turbulent Schmidt number
τ_{ij}	Reynolds stress tensor
ω	Specific dissipation rate

CHAPTER 1 – BACKGROUND OF THE STUDY

1.1 Motivation

Spaces with occupants require good indoor air quality (IAQ) supplied through ventilation in which fresh, clean air is introduced from outdoor to indoor to supply oxygen, displace harmful and unwanted air pollutants [1], and prevent the transmission of airborne diseases such as COVID-19, among others [2], [3]. Several ventilation systems such as natural ventilation (NV), displacement ventilation (DV), mixing ventilation (MV), and hybrid systems coupling NV with DV or PV or by adding a personalized ventilation (PV), have been utilized and studied to provide good IAQ in different indoor environments. These environments include classrooms, residential and office buildings [4], [5], [6], [7].

A typical practice for quantifying IAQ involves using a representative average value of the contaminants present in the room. For example, rooms are maintained at a certain level of CO₂ concentration depending on the number of occupants. American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) recommends room CO₂ concentration of less than 1000 parts per million (ppm) [8]. Furthermore, engineers can also quantify IAQ by relating it to the ventilation flow rate [9] – which represents the volume of fresh air introduced into the space. Estimates from ASHRAE standard states that to maintain acceptable indoor level of CO₂ ppm, a minimum of 15 cubic feet per min per person of fresh air should be introduced [10].

However, the average contaminant concentration and ventilation flow rate may not represent every sections of indoor space owing to imperfect mixing conditions. In other words, inlet and high-velocity regions may have good IAQ as fresh air reaches these regions, and it is easier to remove the contaminant because of high velocities. Unoccupied zones may also maintain good IAQ, as they are regions without contaminant sources such as exhaled air from occupants. In contrast, corners and regions with low-velocity and recirculating flows may have higher concentrations as they may serve as contaminant sinks in which contaminants may find it harder to escape from the area. Occupied zones may also have bad IAQ as these regions have contaminant mainly the occupants. Due to this, generating the spatial distributions of contaminant concentration is essential in designing the ventilation system compared to relying on the average concentration quantities or average ventilation flow rate. Furthermore, in parallel to average concentration and ventilation flow rate, local IAQ can also be explained or quantified through the use of local ventilation efficiencies, such as age of air [11], [12] and net escape velocity [13], [14].

With knowledge of where good local IAQ is needed, i.e. occupant zone [15] or breathing zone (more localized), settings of ventilation systems such as supply flow rate, temperature, and angle can be adjusted. In some cases, if an adjustment on ventilation settings is limited, room layout can also be changed. To some extent, these adjustments have resulted to energy savings [16], [17], [18]. Having said that, IAQ spatial distributions and its corresponding analysis are an important step in understanding the dispersion of indoor contaminants, and then eventually to the determination of optimum ventilation settings.

In addition, generating IAQ spatial distributions is quite difficult to do. Spatial and temporal resolutions, and also number of measuring devices, serve as limitations to fully capture the distributions [19]. This has resulted to the popularization of the computational fluid dynamics (CFD) approach. Having representative measurement points, experimental results both field and wind tunnel (WTE) validated the use of CFD. For example, CFD studies on NV by Shirzadi et al. [20] and Adachi et al. [9] showed good agreement with WTE.

1.2 Objectives of the Study

Against the said background, this dissertation generally aims to use CFD technique to generate contaminant spatial distributions of indoor environments with different ventilation modes to quantify IAQ. In particular and in parallel to the overarching objective:

- 1) The dissertation aims to analyze the IAQ of an idealized cross-ventilation model in an urban environment affected by the sheltering effect. Several passive scalars (representative contaminant) were defined in different parts of the room (inlet and corners), and corresponding contaminant distributions were generated. The study explains the effect of the air introduction in the cross-ventilation model and the contaminant source location on the resulting distributions. Furthermore, the study also quantifies the nature and magnitude of contaminant transport using local ventilation efficiencies: Age of Air, Purging Flow Rate, and Net Escape Velocity.
- 2) The dissertation aims to analyze the IAQ of an office room with hybrid displacement-personalized ventilation system while having a more realistic contaminant sources coming from two occupants through their exhaled carbon dioxide (CO₂). With 25 simulation cases determined by Taguchi design of experiment coupled with “technique for order preference by similarity to ideal solution” (TOPSIS), the study optimizes the IAQ together with the occupant’s thermal comfort and ventilation system’s energy requirement, by varying the ventilation system inputs: displacement ventilation (DV) supply flow rate, DV supply temperature, DV recirculation, personalized ventilation (PV) supply flow, and PV supply

temperature. This also compares optimization procedures based on occupant-centric IAQ value such as inhaled CO₂ concentration versus room volume-averaged CO₂ concentration.

- 3) The dissertation aims to analyze the IAQ of an office room with hybrid mixing-personalized ventilation system, having two occupants as contaminant sources. With the same procedures for occupant-centric optimization of the previous objective, this compares the differences between the optimized results of IAQ, as well as thermal comfort and energy requirements with those of the hybrid displacement-personalized ventilation system. This eventually determines which hybrid system performs better.

Overall, this study may potentially help in the design of indoor environments in providing good air quality. The structure of this dissertation is illustrated in Figure 1.

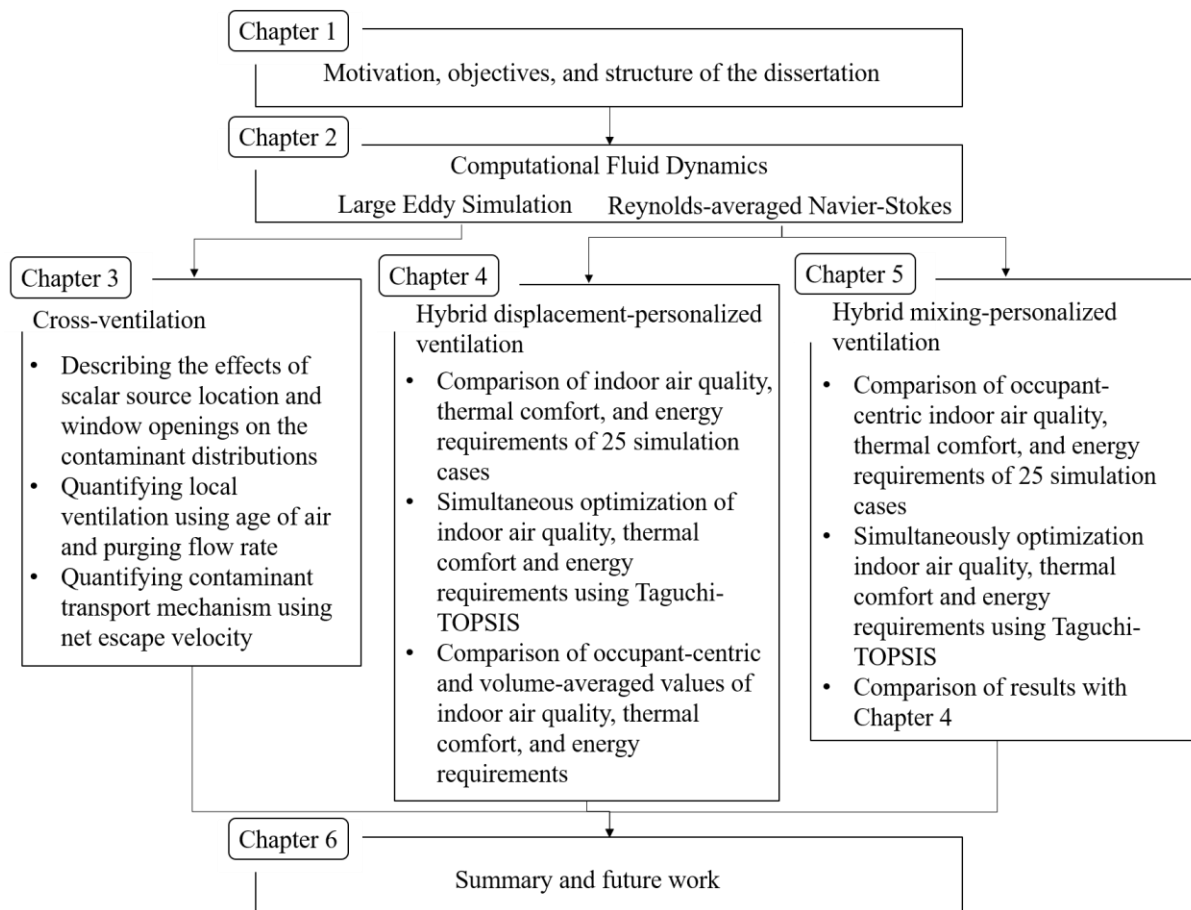


Figure 1. Diagram of dissertation structure.

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CHAPTER 2 – GENERAL METHODOLOGY
2.1 Computational Fluid Dynamics

Due to the limitations of experiments in generating spatial distributions of IAQ, this study utilized computational fluid dynamics (CFD). CFD is a branch of fluid mechanics where equations governing fluid motion, such as continuity, momentum, and energy, are solved numerically to determine fluid flow properties like velocity, temperature, and pressure [1], [2]. The governing equations are PDEs in nature and are generally in the form of a transport equation as shown in Eq. (2.1) for a given fluid property Φ . By discretizing the domain into smaller volumes or grids called cells, the PDEs are approximated to systems of linear equations for each cell.

$$\frac{\partial(\rho\Phi)}{\partial t} + \frac{\partial(\rho\Phi u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\Gamma_\Phi \frac{\partial\Phi}{\partial x_j} \right) + S_\Phi \quad (2.1)$$

where x is the direction, ρ is the fluid density, t is time, u_j is the velocity, Γ_Φ is the diffusivity, and S_Φ is the source/sink. The equation's first to fourth terms represent the unsteady, convection, diffusion, and the source terms of the property, respectively.

This dissertation limits its scope to incompressible flow where the change in density is negligible. Eqs. (2.2) – (2.3) express the continuity and momentum equations.

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (2.2)$$

$$\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} = -\frac{1}{\rho} \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\nu \frac{\partial u_i}{\partial x_j} \right) + f_i \quad (2.3)$$

where P is pressure, ν is the kinematic viscosity, and f_i are the external body forces. Here, Eq. (2.2) follows the principle of the conservation of mass, where mass cannot be created or destroyed but can only be transferred. Eq. (2.3), on the other hand, follows Newton's second law where the rate of change of momentum is equal to the sum of forces acting on a body.

In addition to the continuity and momentum equations, we also utilized the scalar transport equation to generate contaminant distribution for air quality modeling as expressed in Eq. (2.4).

$$\frac{\partial(C)}{\partial t} + \frac{\partial(Cu_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left(D \frac{\partial C}{\partial x_j} \right) + S_C \quad (2.4)$$

where C is the general contaminant concentration and D is the molecular diffusivity.

2.2 Turbulence Model

In fluid flow, velocity and pressure fluctuations exist, characterized by an irregular and chaotic motion that cascades from large to small eddies [3]. This phenomenon is called turbulence. Due to its complexity, Navier-Stokes equation employs models to approximate turbulence. In this dissertation, we utilized two Eddy viscosity models, particularly the Reynolds-Averaged Navier-Stokes equation (RANS) and Large Eddy Simulation (LES).

2.2.1 Reynolds-Averaged Navier-Stokes equation (RANS)

RANS equation are time-averaged equations of fluid motion. Using Reynolds decomposition [4], where velocity, pressure, and other properties are decomposed into their mean and fluctuating components (Eq. (2.5) – (2.7)), the original equations can be changed into Reynolds-averaged (RA) form.

$$u = \bar{u} + u' \quad (2.5)$$

$$P = \bar{P} + P' \quad (2.6)$$

$$C = \bar{C} + C' \quad (2.7)$$

Here, the overbar represents the temporal average of the fluid property while the apostrophe represents the temporal fluctuations from the mean.

By substituting the Reynolds decomposed variables (Eq. (2.5) – (2.7)) into their corresponding transport equations (Eq. (2.2) – (2.4)), and then performing time-averaging, the RA form of the continuity, Navier-Stokes, and scalar transport equations are as follows:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (2.8)$$

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial \bar{u}_i \bar{u}_j}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\nu \frac{\partial \bar{u}_i}{\partial x_j} \right) + f_i - \frac{\partial \overline{u_i' u_j'}}{\partial x_j} \quad (2.9)$$

$$\frac{\partial \bar{C}}{\partial t} + \frac{\partial \bar{C} \bar{u}_j}{\partial x_j} = \frac{\partial}{\partial x_j} \left(D \frac{\partial \bar{C}}{\partial x_j} \right) + S_c - \frac{\partial \overline{C' u_j'}}{\partial x_j} \quad (2.10)$$

Here, the average of the fluctuations, e.g. $\overline{\Phi'}$, vanishes since its average value is equal to zero. However, the average of the product of fluctuations, e.g. $\overline{\Phi' \Phi'}$, remains. In RANS (Eq. (2.9)), Reynolds stress (Eq. (2.11)) appears due to the averaging of the product of the velocity fluctuations. Modeling is required to solve the equation, which will be discussed in the next section. Furthermore, a similar term also appears in the scalar transport equation but as the

product of scalar and velocity. Using gradient diffusion hypothesis (GDH) [5], the average of the product of the fluctuations relates to the mean scalar gradient (Eq. (2.12)).

$$\overline{u_i' u_j'} = \frac{2}{3} k \delta_{ij} - \nu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} - \frac{2}{3} \frac{\partial \bar{u}_k}{\partial x_k} \delta_{ij} \right) \quad (2.11)$$

$$\overline{C' u_j'} = - \frac{\nu_t}{\sigma_t} \frac{\partial \bar{C}}{\partial x_j} \quad (2.12)$$

where, δ_{ij} is the Kronecker delta which is equal to 1 when $i = j$, and equal to zero otherwise, ν_t is the turbulent viscosity, and σ_t is the turbulent Schmidt number.

2.2.1.1 SST k - ω model

Bousinessq hypothesis [6] relates the Reynolds stress term to known quantities, particularly the mean velocity gradients and the turbulent viscosity as shown in Eq. (2.11). This eventually requires additional two transport equations: turbulence kinetic energy (k) and dissipation rate quantities, such as ε (dissipation rate) or ω (specific dissipation rate), depending on the turbulence model used. This dissertation used SST (shear-stress transport) [7] k - ω model as one of the studied involves impinging flow and separation. This model overcomes the limitations of its predecessors: the standard k - ω model (due to free-stream turbulence dependence) and the standard k - ε model (due to problems with adverse pressure gradients and flow separation). Solving these additional transport equations helps solve the RANS equation.

Eq. (2.13) and Eq. (2.14) show the transport equations for turbulence kinetic energy and specific dissipation rate, respectively.

$$\frac{\partial k}{\partial t} + \frac{\partial k \bar{u}_j}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right) + P_k - \beta^* \omega k \quad (2.13)$$

$$\frac{\partial \omega}{\partial t} + \frac{\partial \omega \bar{u}_j}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\left(\nu + \frac{\nu_t}{\sigma_{\omega 1}} \right) \frac{\partial \omega}{\partial x_j} \right) + \alpha S^2 - \beta \omega^2 + 2(1 - F_1) \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i} \quad (2.14)$$

where P_k is the turbulent kinetic energy production term, $\beta^*(=0.09)$ is a model constant. Here, the last term of Eq. (2.14) is a blending function that shifts the dissipation rate from ε to ω using the variable F_1 , which is solved using Eq. (2.15) – (2.18). One of the equations has a variable d , which is the distance of the evaluated cell to the closest wall, indicating the blending function's dependency on the distance from the wall. Particularly, SST k - ω utilizes ω near wall regions while ε far from the wall.

$$F_1 = \tanh(\arg_1^4), \quad (2.15)$$

$$\arg_1 = \min \left(\max \left(\frac{\sqrt{k}}{\beta^* \omega d}, \frac{500\nu}{d^2 \omega} \right), \frac{4\rho\sigma_{\omega 2}k}{CD_{k\omega}d^2} \right) \quad (2.16)$$

$$CD_{k\omega} = \max \left(2\rho\sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i}, 10^{-10} \right) \quad (2.17)$$

$$\Phi = F_1 \Phi_1 + (1 - F_1) \Phi_2 \quad (2.18)$$

where $\alpha_1 (=0.556)$, $\alpha_2 (=0.44)$, $\beta_1 (=0.075)$, $\beta_2 (=0.0828)$, $\sigma_{k1} (=0.85)$, $\sigma_{k2} (=1)$, $\sigma_{\omega 1} (=0.5)$, and $\sigma_{\omega 2} (=0.856)$ are model constants. To get the value of α and β of Eq. (2.15), Eq. (2.18) can be utilized.

2.2.2 Large Eddy Simulation (LES)

Large eddy simulation is an eddy viscosity model in which large-scale turbulent eddies are directly solved while small-scales are modeled to decrease the computational cost [8]. LES uses low-pass filtering for a flow property Φ given by Eq. (2.19). $G(x, \tilde{x}, \Delta)$ is the filter kernel based on representative length scale $\Delta = (dxdydz)^{1/3}$. This filter kernel separates the resolved scale (filtered scale) and the small sub-grid scale (SGS).

$$\langle \Phi(x, t) \rangle = \int G(x - \tilde{x}, \Delta) \Phi(x', t) dx' \quad (2.19)$$

where

$$\Phi(x, t) = \langle \Phi(x, t) \rangle + \tilde{\Phi}(x, t) \quad (2.20)$$

Here, the brackets represent the resolved scale, while tilde represents the SGS or the modeled part of the equation. Applying the filter to the continuity, momentum, and the scalar equations yields to the following:

$$\frac{\partial \langle u_i \rangle}{\partial x_i} = 0 \quad (2.21)$$

$$\frac{\partial \langle u_i \rangle}{\partial t} + \frac{\partial \langle u_i \rangle \langle u_j \rangle}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \langle P \rangle}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\nu \frac{\partial \langle u_i \rangle}{\partial x_j} \right) - \frac{\partial (\langle u_i u_j \rangle - \langle u_i \rangle \langle u_j \rangle)}{\partial x_j} \quad (2.22)$$

$$\frac{\partial \langle C \rangle}{\partial t} + \frac{\partial \langle C \rangle \langle u_j \rangle}{\partial x_j} = \frac{\partial}{\partial x_j} \left(D \frac{\partial \langle C \rangle}{\partial x_j} \right) - \left(\frac{\partial \langle C u_j \rangle}{\partial x_j} - \frac{\partial \langle C \rangle \langle u_j \rangle}{\partial x_j} \right) + S_c \quad (2.23)$$

where the last term of Eq. (2.22) is the unresolved stress tensor or the SGS stress given by (Eq. (2.24)). In our study, we used the standard Smagorinsky model where the stress tensor is assumed to be proportional to the resolved mean velocity strain $\langle S_{ij} \rangle$ and SGS kinematic viscosity ν_{SGS} . Here, $C_s = 0.12$ is the Smagorinsky constant.

$$\langle u_i u_j \rangle - \langle u_i \rangle \langle u_j \rangle = -2\nu_{SGS} \langle S_{ij} \rangle + \frac{1}{3} \tau_{kk} \delta_{ij} \quad (2.24)$$

$$\langle S_{ij} \rangle = \frac{1}{2} \left(\frac{\partial \langle u_i \rangle}{\partial x_j} + \frac{\partial \langle u_j \rangle}{\partial x_i} \right) \quad (2.25)$$

$$\nu_{SGS} = (C_s \Delta)^2 |\langle S_{ij} \rangle| \quad (2.26)$$

$$|\langle S_{ij} \rangle| = \sqrt{2 \langle S_{ij} \rangle \langle S_{ij} \rangle} \quad (2.27)$$

From there, the final form of the momentum equation is shown in Eq. (2.28). For the scalar transport equation, GDH is again used making Eq. (2.29) the final form.

$$\frac{\partial \langle u_i \rangle}{\partial t} + \frac{\partial \langle u_i \rangle \langle u_j \rangle}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \langle P \rangle}{\partial x_i} + \frac{\partial}{\partial x_j} \left((\nu + \nu_{SGS}) \frac{\partial \langle u_i \rangle}{\partial x_j} \right) \quad (2.28)$$

$$\frac{\partial \langle C \rangle}{\partial t} + \frac{\partial \langle C \rangle \langle u_j \rangle}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\left(D + \frac{\nu_{SGS}}{\sigma_{tSGS}} \right) \frac{\partial \langle C \rangle}{\partial x_j} \right) + S_c \quad (2.29)$$

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CHAPTER 3 – QUANTIFICATION OF LOCAL INDOOR AIR QUALITY IN AN IDEALIZED URBAN CROSS-VENTILATION MODEL

3.1 Background

Since ancient times, because of cost effectiveness, different facilities have utilized wind-induced natural cross-ventilation, which does not rely on energy-consuming equipment and only utilizes moving atmospheric air (driven by pressure difference and buoyancy) via windows and other building openings [1], [2]. Questions on its reliability have lingered owing to its unpredictability and lack of control of the amount of ventilation, which depends on the building location, windows, and other opening positions, and even the constant weather variation. Despite this, its availability [3], [4] and application in different establishments, such as office buildings [5] and classrooms [6], [7] have proved its effectiveness.

The complexity of natural cross-ventilation and the demand of facilities for energy efficiency have prompted researchers to conduct experimental and numerical simulations by integrating and modifying factors such as the incident wind speed and direction [8], number and position of window openings [9], [10], geometry of wind catchers [11], [12], [13], and building location [14]. In addition, particularly for urban settings, several obstructions such as tall buildings can potentially alter the approaching wind direction, speed, and turbulence that consequently influence the wind-induced natural ventilation of a target room or space when compared with the ventilation of an isolated one. Researchers have termed this phenomenon as sheltering effect. To understand this, Gough et al. [15] conducted field experiments and observed a 50%–90% decrease in ventilation rate when comparing an isolated 6 m × 6 m × 6 m building with a building sheltered by blocks arranged in a staggered manner. Moreover, Hirose et al. [16] conducted measurements to clarify the relationship between the approaching flow and change in pressure on the leeward and windward sides of a target building surrounded by blocks arranged in a square array to determine the temporal variations in indoor ventilation. In addition to ventilation characteristics, Gough et al. [17] compared approaches in determining the ventilation rate of a building located in a staggered array, one using the tracer gas decay method, and another using an empirical model for pressure measurements. Both methods exhibited limitations in predicting the relationship between the wind speed and direction with the ventilation rate. Accordingly, Hirose et al. [18] studied pressure distribution and canopy flow in relation to varying wind directions in a building within a square array. They observed a discrepancy in the pressure coefficient values between the computed and experimental values, particularly on the side edge of the target block, which may affect the estimation of the ventilation rate using the conventional approach.

In addition to field measurements, several research groups have focused on wind tunnel experiments (WTE) to describe these phenomena. Tominaga and Blocken [19] conducted velocity and concentration field measurements and flow visualization using a gas tracer technique to compare the cross ventilation of a building model with isolated and sheltered (square array) conditions. Their study suggested that the surrounding buildings diminished the turbulent kinetic energy, increased the concentration fluctuation, and reduced the airflow rate to approximately 30% compared with isolated cases. Ikegaya et al. [20] used time-resolved particle image velocimetry (PIV) to visualize the airflow inside a generic block surrounded by two array configurations: staggered and square. Their experiment showed that the outdoor building configuration significantly affected the indoor airflow. They concluded that instantaneous velocity, in addition to mean flow distributions, must be considered in the ventilation design because of the observed unsteadiness. Moreover, Shirzadi et al. [21] used a split fiber probe to measure the instantaneous velocity of a building under sheltered conditions using eight neighboring blocks. They varied the approaching wind angle and observed that it significantly affected the transient nature of indoor ventilation, particularly the instantaneous velocity, penetration length, and vortex structure. Furthermore, Golubic et al. [22] studied the impact of the wind incidence angle and velocity, building spacing, and window configuration on the air change rate of a building sheltered by a square array. The results indicated lower air change rates for sheltered cases and favored less densely spaced buildings in terms of ventilation.

For CFD simulations, Adachi et al. [23] reproduced cross-ventilation phenomena under sheltered conditions conducted in a previous WTE [20]. They employed an urban-like block with square and staggered array configurations, for which turbulent statistics and structures were experimentally studied by Cheng and Castro [24], to clarify the importance of considering the urban-like sheltering effect for indoor ventilation. Subsequently, Hirose et al. [25] extended the research by incorporating thermal comfort aspects to demonstrate the number of inlet measurement positions required to represent an accurate ventilation rate. Furthermore, other studies used LESs using the lattice-Boltzmann method [26] and combined LES and RANS simulation [27] to decrease the computational time of LES for the natural ventilation of a room surrounded by buildings.

Many indoor-only and coupled outdoor–indoor simulations have employed such overall metrics to evaluate ventilation. However, these metrics may fail to explain the relationship between indoor flow structures and their effect on ventilation characteristics in different indoor locations. In other words, the local concentrations may not necessarily reflect the value of the overall boundary ventilation rates owing to the imperfect mixing condition. Therefore, various studies have proposed different ventilation indices that utilize the concentration distribution

field in addition to the velocity field, such as mean age of air (AOA) [28], [29], scale for ventilation efficiencies (SVEs) [30], visitation frequency (VF) [31], [32], purging flow rate (PFR) [33], [34], and net escape velocity (NEV) [35], [36]. The AOA describes the freshness of air depending on the time a parcel of air spends in a room; the shorter the time, the better the local ventilation [28], [29]. Kato and Murakami [37] developed AOA calculations for steady-state and concentration distribution analysis CFD-based simulations, and it was eventually defined as SVE3. In contrast, SVE1 is defined as the average concentration or the residence time of the contaminant at a local point in a room [30]. Accordingly, the VF describes the number of times a certain parcel of air visits a local domain; the lower the value, the better the ventilation [31], [32]. The PFR represents the amount of ventilation in terms of the flow rate required to remove contaminants from a local domain [33], [34]. Finally, Lim et al. [35] clarified the definition of the NEV, which was initially conceptualized by Sandberg. They defined the NEV as an effective velocity scale representing the escaping population of a contaminant at the source location. Notably, when normalized with the convective velocity at a point, the NEV provided insight into whether convection or turbulent diffusion governs contaminant transport, as demonstrated in RANS CFD studies [38], [39], [40].

Previous studies on the cross-ventilation of a room surrounded by urban-like block arrays were limited to the computation of the overall ventilation rate based on boundary volume flow rates, although they considered the turbulent effects on the ventilation rate by employing LESs. Therefore, this study aimed to extend the cross-ventilation study of Adachi et al. [23] for a building surrounded by an urban-like array by incorporating ventilation indices such as the AOA, PFR, and NEV to understand the relationship between overall ventilation rates and local ventilation indices. We attempted to further explain the effect of the introduction of outside flow owing to the surrounding buildings and window openings on the scalar concentration profile and then the ventilation effectiveness in the indoor space. In addition, a previous study applied the concept of NEV only to RANS simulations; therefore, our study explored its use in unsteady simulations using LES. We clarified the variation in the NEV with changes in the velocity and concentration fields. To consider the local ventilation rates despite computationally demanding LES, our simulations simultaneously solved the scalar transport equation of multiple passive scalar contaminants emitted from different source locations to determine its effects on the concentration field that may potentially be used in actual applications, such as the positioning of occupants (contaminant-generating subjects) in a room.

3.2 Methodology

3.2.1 Ventilation Metrics and Efficiencies

The local ventilation rates at different locations in an indoor space do not reflect the boundary net flow rate due to imperfect mixing conditions resulting from the formation of vortices and stagnation points inside the room. Such vortices produce high-velocity points where parcels of air are constantly moving, stagnation points where parcels of air are seemingly not moving, and even circulation points where parcels of air are moving but only circulate in that local domain and cannot escape the room. In addition, concentration gradients drive the transport of contaminants to other parts of the domain, thereby affecting the local ventilation rates. The study of Adachi et al. [23] was limited to the use of boundary-based metrics such as the net, gross, and instantaneous ventilation rates, as defined in their paper. To extend their work by quantifying the variations in local ventilation rates owing to imperfect mixing condition, we employed ventilation indices. This section defines boundary-based volume flow rates (ventilation rates and the equivalent air exchange rate) and ventilation indices (AOA, PFR, and NEV).

3.2.1.1 Air Exchange Rate

One of the most commonly used boundary-based ventilation rates is the air exchange rate (ACH). It is defined as the number of times the indoor air is replaced by the outside air (which is assumed fresh and clean) per unit time, typically on an hourly basis, as given by Eq. (3.1).

$$ACH = \frac{\dot{V}}{V_{room}}, \quad (3.1)$$

where $\dot{V}$ is the volume flow rate of air in m^3/h and V_{room} is the volume of the indoor space in m^3 .

3.2.1.2 Net, Gross, and Instantaneous Ventilation Rates

To evaluate the ACH, the ventilation rate $\dot{V}$ can be split into different values that may or may not consider the reversal of flow at the boundary and also be evaluated instantaneously as used in the study of Adachi et al. [23]: the net ventilation rate $\dot{V}_{net}$, gross ventilation rate $\dot{V}_{gross}$, and instantaneous ventilation rate $\dot{V}_{inst}$, as given by Eqs. (3.2–3.4), respectively:

$$\dot{V}_{net} = \sum_i \overline{u_{n,i}} \Delta A_i, \quad (3.2)$$

$$\dot{V}_{gross} = \sum_i |\overline{u_{n,i}}| \Delta A_i, \quad (3.3)$$

$$\dot{V}_{inst} = \frac{1}{t} \int \sum_i |u_{n,i}| \Delta A_i dt, \quad (3.4)$$

where $u_{n,i}$ is the wind speed perpendicular to the face of each of cell i located at the ventilation building openings (windows) in m/s, ΔA_i is the corresponding area of the grid point in m^2 , t is the time in seconds, and overbar indicates the temporal averaging during t . Here, $\dot{V}_{net}$ is defined by the temporally and spatially averaged wind speed at the window boundary. The reversal of flow is considered a negative value, and in cases in which the total entering flow is the same as the total exiting flow, this will yield a zero value, although air exchange is apparent due to the inhomogeneity of the flow within the opening. To address this problem, we can compute $\dot{V}_{gross}$ using the absolute value of the temporally averaged $u_{n,i}$. Accordingly, $\dot{V}_{inst}$, which considers the instantaneous reverse flow at each time step during the duration t , can also be computed. According to comparisons of the three ventilation rates by Adachi et al. [30], $\dot{V}_{inst}$ becomes significant when the mean velocity is small, whereas the differences in $\dot{V}_{net}$, $\dot{V}_{gross}$, and $\dot{V}_{inst}$ are subtle when the inflow is dominated by the mean velocity.

3.2.1.3 Net Escape Velocity

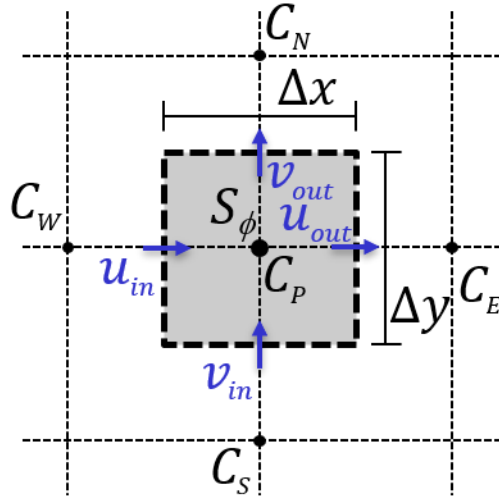


Figure 3.2. Control volume (CV) for the NEV computation. The variables C_P , C_N , C_S , C_E , and C_W represent the concentration values at the numerical grid center. The velocity components u_{in} , u_{out} , v_{in} , and v_{out} denote the boundary values of the CV.

The concept of the NEV, as proposed by Sandberg and derived by Lim et al. [35] for CFD simulations, is a vector quantity and is defined as the effective velocity scale representing the velocity of the escaping population of a contaminant at a source or control volume discretized in Figure 3.2. In their study, the NEV were derived from a RANS-based scalar transport equation (Eq. (2.10)). Two types of NEV were formulated depending on whether it accounts

for the returning contaminants (NEV) or only the outgoing flux (NEV*) of the control volume being considered. The j -th components of NEV and NEV* are defined as follows:

$$\overline{NEV_j} = \frac{1}{\overline{C_p}} \left(\overline{u_{j,out} C_{out}} - D_{eff} \left. \frac{\partial \overline{C}}{\partial x_j} \right|_{out} - \overline{u_{j,in} C_{in}} + D_{eff} \left. \frac{\partial \overline{C}}{\partial x_j} \right|_{in} \right) = NEP \times \overline{NEV_j^*}, \quad (3.5)$$

$$\overline{NEV_j^*} = \frac{1}{\overline{C_p}} \left(\overline{u_{j,out} C_{out}} - D_{eff} \left. \frac{\partial \overline{C}}{\partial x_j} \right|_{out} \right) = NEP \times \overline{NEV_j^*} + (1 - NEP) \overline{NEV_j^*}. \quad (3.6)$$

Here, NEP indicates the net escape probability (Chung et al. [40]), and $\overline{C_p}$ is the temporally averaged scalar concentration in the control volume. When only the outgoing flux is scaled by the concentration $\overline{C_p}$, it defines the velocity scale determined by the total scalar flux, namely, NEV*. Because NEV* is decomposed into two parts, as shown in Eq. (3.6), the NEV is considered the escaping part of NEV*. When normalized with the mean velocity at a point, the NEV provides insights into whether advection or turbulent diffusion governs contaminant transport, as demonstrated in previous studies [38], [39], [40]. The results of the study by Lim et al. [35] show that the NEV* computed in each of the control volumes is slightly different from the velocity field owing to the contribution of the turbulent and molecular diffusion, which is further discussed in subsequent sections.

As this study focused on the application of the NEV to LES, we consider NEV^* based on the spatially filtered equations used in LES. By applying temporal averaging to the filtered equations, the temporally averaged filtered scalar transport equation can be expressed as follows.

$$\frac{\partial \langle \overline{C} \rangle}{\partial t} + \frac{\partial \langle \overline{u_j} \rangle \langle \overline{C} \rangle}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\left(D + \frac{\nu_{sgs}}{\sigma_{tsgs}} \right) \frac{\partial \langle \overline{C} \rangle}{\partial x_j} - \langle \overline{u_j} \rangle' \langle \overline{C} \rangle' \right) + S_\phi \quad (3.7)$$

Here, the brackets indicate the filtering operation used to extract the resolved scale components of the velocity and scalar concentration. The overbar indicates the temporal average and their fluctuation parts from the mean are defined as $\langle u_i \rangle' = \langle u_i \rangle - \overline{\langle u_i \rangle}$ and $\langle C \rangle' = \langle C \rangle - \overline{\langle C \rangle}$. The variable D is the molecular diffusivity ($= 1.5 \times 10^{-5} \text{ m}^2/\text{s}$), ν_{sgs} is the sub-grid scale (SGS) kinematic viscosity, and Sc_{sgs} ($=1$, assuming passive scalar) is the sub-grid scale turbulent Schmidt number.

By following the same procedure used to derive Eq. (3.6), $\overline{NEV_j^*}$ is expressed as follows:

$$\overline{NEV_j^*} = \frac{1}{\langle \overline{C} \rangle} \left(\langle \overline{u_j} \rangle \langle \overline{C} \rangle + \langle \overline{u_j} \rangle' \langle \overline{C} \rangle' \right), \quad (3.8)$$

where the molecular and SGS diffusion terms are neglected because advection and turbulent diffusion dominate scalar fluxes. In studies to determine the indoor velocity distributions, Adachi et al. [23] and Hirose et al. [25] showed that the resolved mean velocity and turbulent

intensity distributions agreed well with the experimental data (we recall this aspect in Section 3.2.3). These results indicated that the contribution of the SGS to the total velocity fields is much smaller owing to the advection and turbulence of the grid-scale velocity determined with very fine grid resolutions. Therefore, we approximated the total scalar fluxes using advective and turbulent parts. In addition, the determination of NEV^* was simplified in the LES calculation based on the mathematical definition of NEV^* by Ikegaya et al.[36]. According to their study, NEV^* can be defined by the macroscopic concentration, or $\overline{\langle C \rangle}$, and scalar flux at each grid without considering the boundary values of the scalar fluxes at a control volume. In contrast, the scalar fluxes should be integrated on the boundary is required to determine the NEP and, accordingly, to estimate the NEV at a source location (i.e., $NEV = NEP \times NEV^*$). Because of the limitation in data storage in the LES, we could not store all the data required for determining the NEP in this study. Therefore, in this paper, we only scrutinize the distribution of NEV^* determined by LESs.

3.2.1.4 Purging Flow Rate

The local purging flow rate (L-PFR, m^3/s) is the net airflow rate for the transfer or dilution of a contaminant from a local domain (Eq. (11)) [34].

$$L - PFR = \frac{\dot{m}}{\overline{\langle C \rangle}}, \quad (3.9)$$

where $\dot{m} = \int_{CV} S_\phi dV$ denotes the net scalar generation of the target control volume in kg/s . To determine the L-PFR from the concentration obtained by LES, we simply calculate the temporal-averaged concentration at the source location.

3.2.1.5 Age of Air

AOA, or SVE3, describes the mean time required for the supply air to reach a local point in a room [30]; it is proportional to the concentration of a point in when a temporally uniform contaminant-generation rate is uniformly distributed in a given space ($\dot{m}$), and it is given by

$$SVE3 = \frac{\overline{\langle C \rangle}}{C_{pm}} = \frac{\overline{\langle C \rangle}}{\dot{m}} \dot{V}, \quad (3.10)$$

where $C_{pm} = \dot{m}/\dot{V}$ is the perfect mixing concentration (kg/m^3) (equal to the average concentration at the boundary).

3.2.2 Simulation settings

3.2.2.1 Computational Domain

We performed isothermal simulations for both the airflow and concentration fields for cross-ventilation conditions sheltered by surrounding buildings with two different opening conditions. As this research to author's knowledge was the first one to employ ventilation indices in a coupled indoor-outdoor simulation considering sheltering effect, and also considering the availability of experimental data for validation, we employed a generic block array arranged in staggered manner, with one building as ventilation model with two facing openings that were consistent with the dimensions of buildings in the WTE of Ikegaya et al. [23], and numerical simulation of Adachi et al. [32] and Hirose et al. [34]. Figure 3.3 shows a schematic of the numerical simulation domain, where three solid blocks were used as obstructions for a ventilated block (Figure 3.3b, 3.2c) with two window openings. Two cases of window positions were simulated: one with window openings on the streamwise faces (STR) and the other with the window openings on the lateral faces (LAT). We set the length of the sides of the blocks as $H = 100$ mm, with the ventilated block having a wall thickness of 2 mm, and the windows positioned at the center of the face of the building with an area of 10% of H^2 . The computational domain was set as $4H \times 4H \times 4H$, where the packing density of the blocks on the ground was 25%. A uniform grid size of $0.01H$ and a time step of 0.0001 s were set for the maximum Courant number to be maintained at a value of less than 1, resulting in a total number of cells of 64,000,000 [50]. The wall unit at the first grid, $z_1^+ = z_1 u_\tau / \nu$, where z_1 is the first grid height and u_τ is the local friction velocity on the solid wall surfaces, was maintained at less than 10.

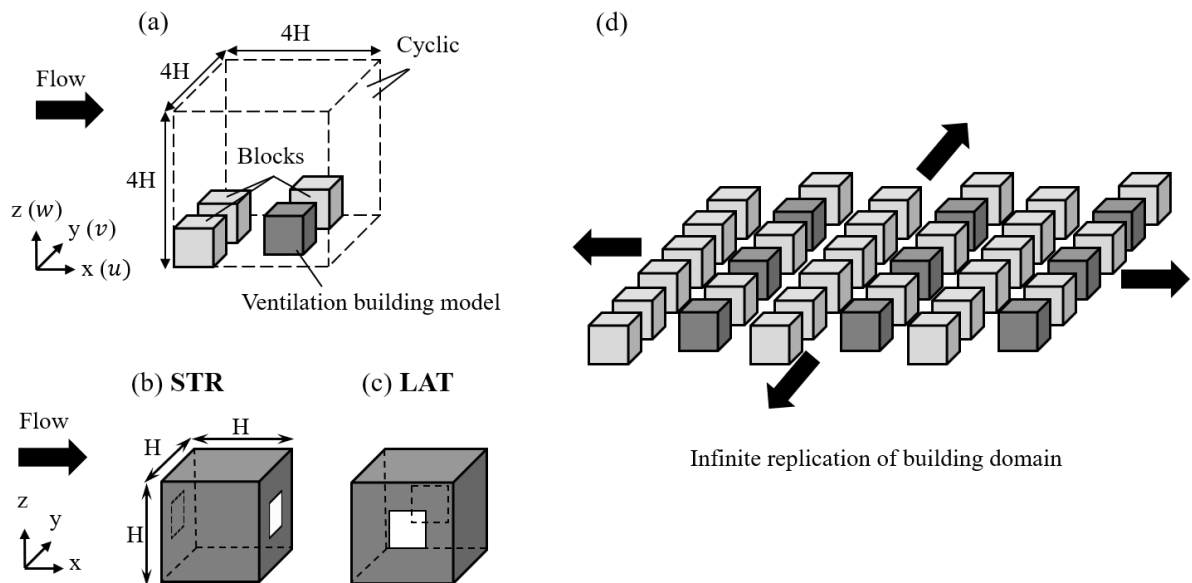


Figure 3.3. Schematics of (a) numerical domains in staggered (ST) arrangement, (b) STR openings, and (c) LAT openings (d) Infinite replication of building domain at spanwise and streamwise directions. H ($= 100$ mm) is the building height with opening size equal to 10% of H^2 .

3.2.2.2 Numerical Setup

OpenFOAM software was used to perform LESs with the finite volume method for incompressible continuity, Navier–Stokes, and scalar transport equations described in Chapter 2 and Eq. (3.7).

To reduce the simulation time, this study used the velocity and pressure fields generated using CFD simulations by Adachi et al. [32] at $t = 60$ s, where the flow fields were fully turbulent and adjusted to the building arrangements as initial conditions. The velocity, pressure, and scalar concentration fields were then calculated. We used cyclic boundary conditions in both the streamwise and spanwise boundaries such that the numerical domain was replicated in the horizontal direction (Figure 3.3d), with a constant streamwise pressure gradient with a magnitude of $-dp/dx = u_\tau^2/L_z$, where $L_z = 4H$ is the domain height and u_τ is the friction velocity. We set the boundary condition for the block and ground surfaces as no-slip and for the top of the domain as free-slip.

Additionally, seven scalar sources were positioned in the x–y plane in the middle of the ventilation building domain (at $z/H = 0.5$) for both the STR and LAT cases (Figure 3.4). These scalar sources were independently computed to investigate the behavior of the scalar dispersion from the following locations and conditions: 1) the inlet of the ventilation domain with intermittent reversal of flow ($C_{i1} - C_{i4}$), 2) possible stagnation points, particularly in the corner of the ventilation domain (C_{uc} and C_{dc} for the corner near the upwind and downwind walls, respectively), and 3) near the downwind section of the ventilation building (C_o for the outlet of the STR case, and C_{dw} for the near downwind wall of the LAT case). In addition, we specified a uniformly generated source within the indoor region, denoted as C_a , for the computation of SVE3. Finally, the cyclic boundary condition was also employed in the spanwise and streamwise directions, whereas a zero gradient was employed for the top, bottom, and surfaces of the domain ($\partial C_i/\partial x_j = 0$).

For numerical schemes, we followed the conditions employed by Adachi et al. [32]. Second-order backward and central schemes were used for temporal and spatial terms, respectively, whereas pressure-implicit split-order method were utilized to couple the velocity and pressure [52].

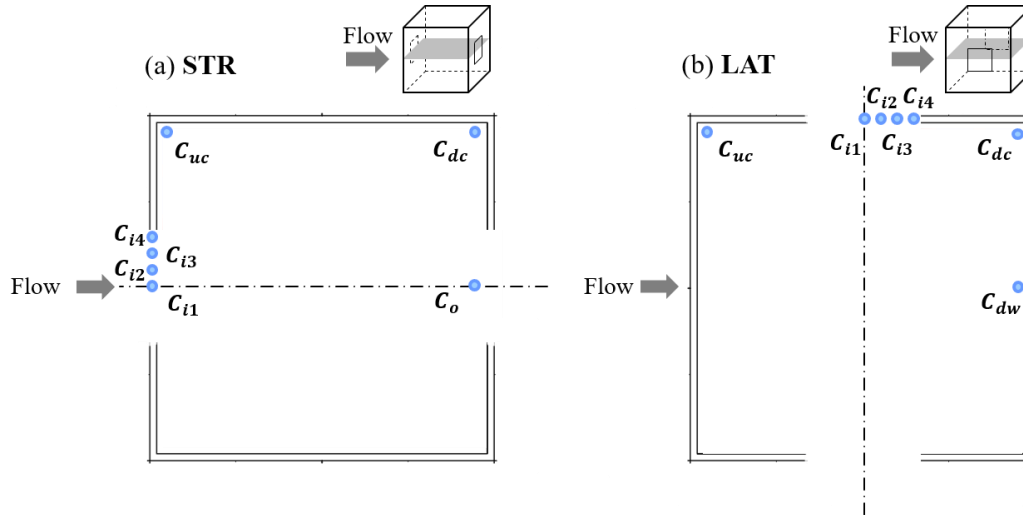


Figure 3.4. Definitions of passive scalar source locations. (a) STR, and (b) LAT conditions.

3.2.3 Validation

A Figure 3.5a, 3.4b, 3.4e, and 3.4f depict the velocity vector and contour plots on the x–y plane at $z/H = 0.5$ for both the STR and LAT cases, which illustrate the comparisons made by Adachi et al. [32] between their CFD simulation and the WTE using PIV by Ikegaya et al.[23]. In addition, as this research used the simulation settings of Adachi et al.[32], we reproduced the mean velocities and root-mean-square (rms) velocities at the x–y plane at $z/H = 0.5$ (Figure 3.5c, 3.4d, 3.4g, and 3.4h) and on the x–z plane at $y/H = 2.0$ (Figure 3.6), and we compared them with the previous WTE results. Generally, the velocity fields and rms values of the CFD produced results that were in good agreement with the WTE, which makes the former a viable method for predicting the indoor velocity field of a room surrounded by an urban-like array. Furthermore, note that the discrepancies, particularly at the inlet of Figure 3.6 where the velocities were generally high, were considered by Adachi et al. [32] to be due to PIV measurement errors.

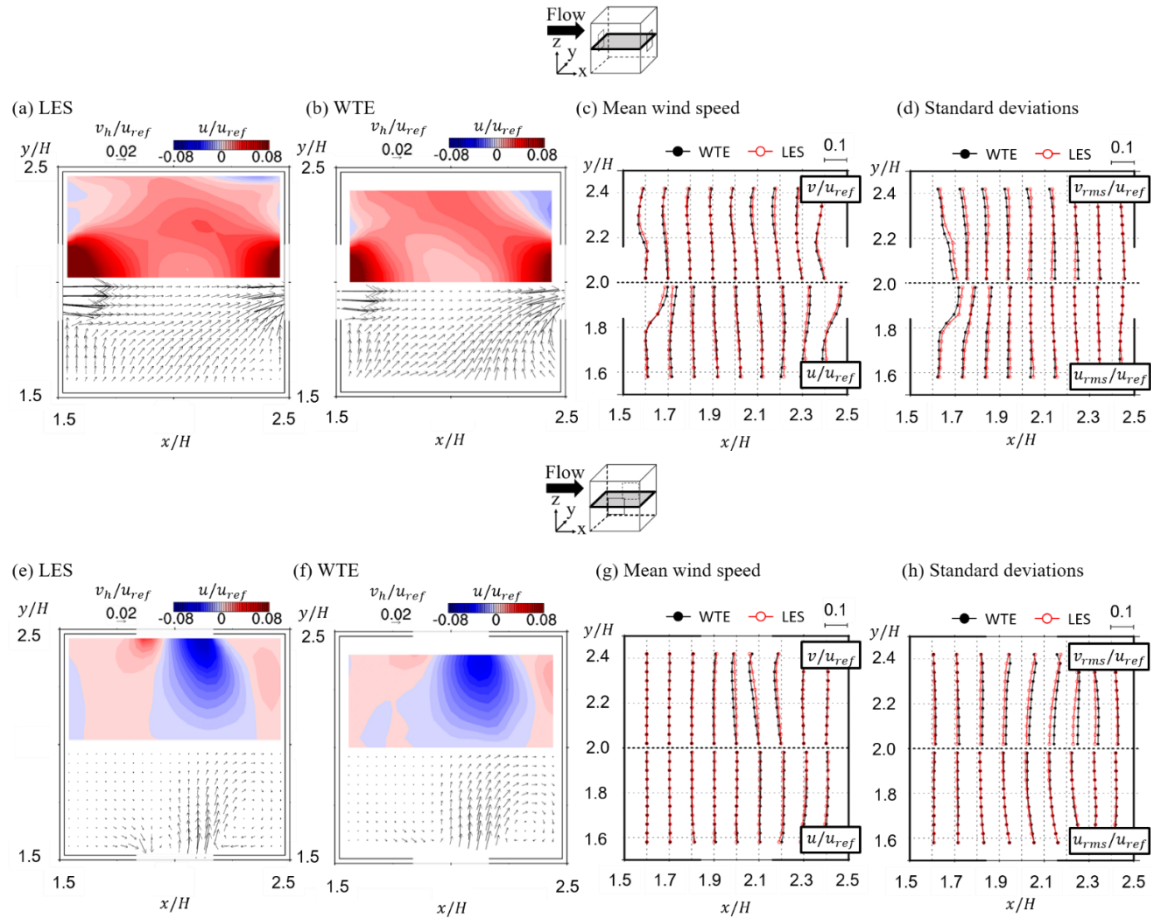


Figure 3.5. Comparison between indoor velocity contours (upper half figures of (a, b), and (e, f)) and vectors (lower half figures of (a, b) and (e, f)) using CFD (Adachi et al. [32]) and WTE (Ikegaya et al. [23]); current study velocity magnitudes and standard deviations (c, d, g, h) versus WTE (Ikegaya et al. [23]) on the x - y plane at $z/H = 0.5$. (a–d) STR and (e–h) LAT conditions. (a, e) LES (previous), (b, f) WTE, and (c, d, g, h) LES (current) versus WTE comparison of profiles for streamwise and spanwise velocity components. v_h is the magnitude of the time averaged horizontal velocity components, and u_{ref} is the reference wind speed defined at $z = 4H$. Figures are cited from Adachi et al. [32] and modified for comparison.

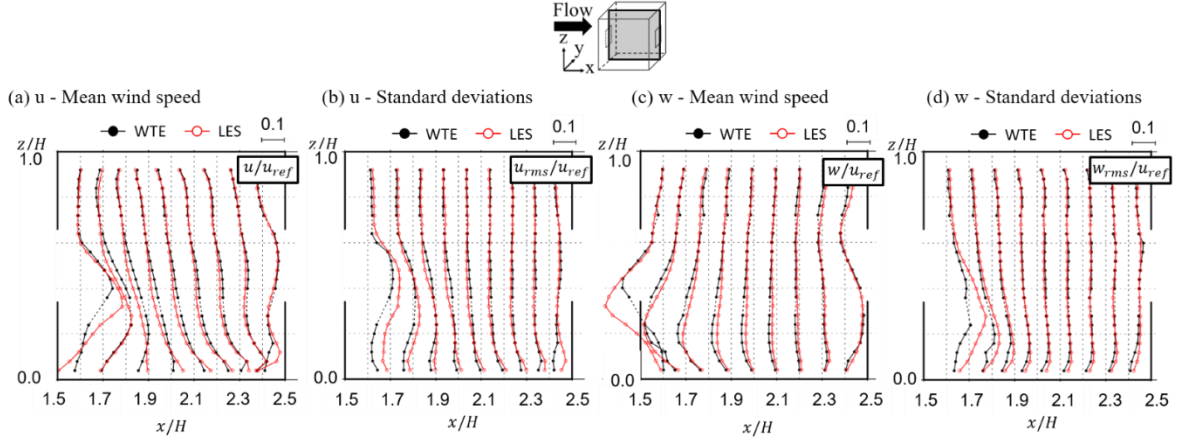


Figure 3.6. Comparison between current LES indoor velocity magnitudes and standard deviations versus WTE (Ikegaya et al. [23]) for the STR case on the x - z plane at $y/H = 2.0$.

3.3 Indoor Concentration Fields

3.3.1 Instantaneous Scalar Distribution and Velocity Vector Field

In this section, we discuss the qualitative characteristics of instantaneous scalar dispersions to investigate the effect of the introduction of instantaneous flow on scalar entrainment in indoor spaces. Figure 3.7 shows the instantaneous scalar distributions of scalars positioned at the inlet of the indoor spaces for both the LAT and STR cases. The concentration distributions were superimposed in the velocity vector field projected on the x - y plane at $z/H = 0.5$ for both the LAT and STR cases, taken at an arbitrary time t of the simulation. The scalar fields were normalized using the perfect mixing concentration C_{pm} , and the velocity fields were normalized using the reference velocity u_{ref} .

Here, only scalar fields of C_{i1} and C_{i4} are presented because C_{i3} yielded almost the same distributions as C_{i4} , whereas C_{i2} yielded scalar distributions between C_{i1} and C_{i3} . For the STR case, the concentrations of scalar C_{i1} (Figure 3.7a) were almost scattered throughout the indoor space, whereas the concentrations of C_{i4} (as well as C_{i2} and C_{i3}) (Figure 3.7b) accumulated on the upper side of the x - y plane ($y/H > 2.0$). This can be explained by the stagnation represented by the short arrows (low-magnitude velocities) that can be observed in the middle of the indoor space, which prevented concentration transfer from the upper to the lower side. This occurred specifically in scalar concentrations C_{i2} to C_{i4} that were located near the upper half of the field ($y/H > 2.0$), in contrast to C_{i1} which was very close to the middle ($y/H = 2.0$). Meanwhile, the scalar concentration C_{i1} (Figure 3.7c) of the LAT case produced scalar-concentration scattering outside the indoor space extending up to $y/H = 3.0$ and inside the indoor space extending down to $y/H = 2.0$, in contrast to the C_{i4} (Figure 3.7d),

where the concentration scattered mostly in the indoor space. First, the behavior of C_{i1} can be attributed to the fluctuating flow experienced by the LAT case, as demonstrated in the study by Adachi et al. [32] and in Figure 3.10b and 3.8c, whereas the behavior of C_{i4} resulted from the boundary layer formed as it was positioned near the window wall and, in effect, decreased the magnitude of flow fluctuations, enabling the concentration to scatter inside. These observations indicate that the scalar source location affects the resulting concentration field although the locations of the source of $C_{i1} - C_{i4}$ are very close to each other.

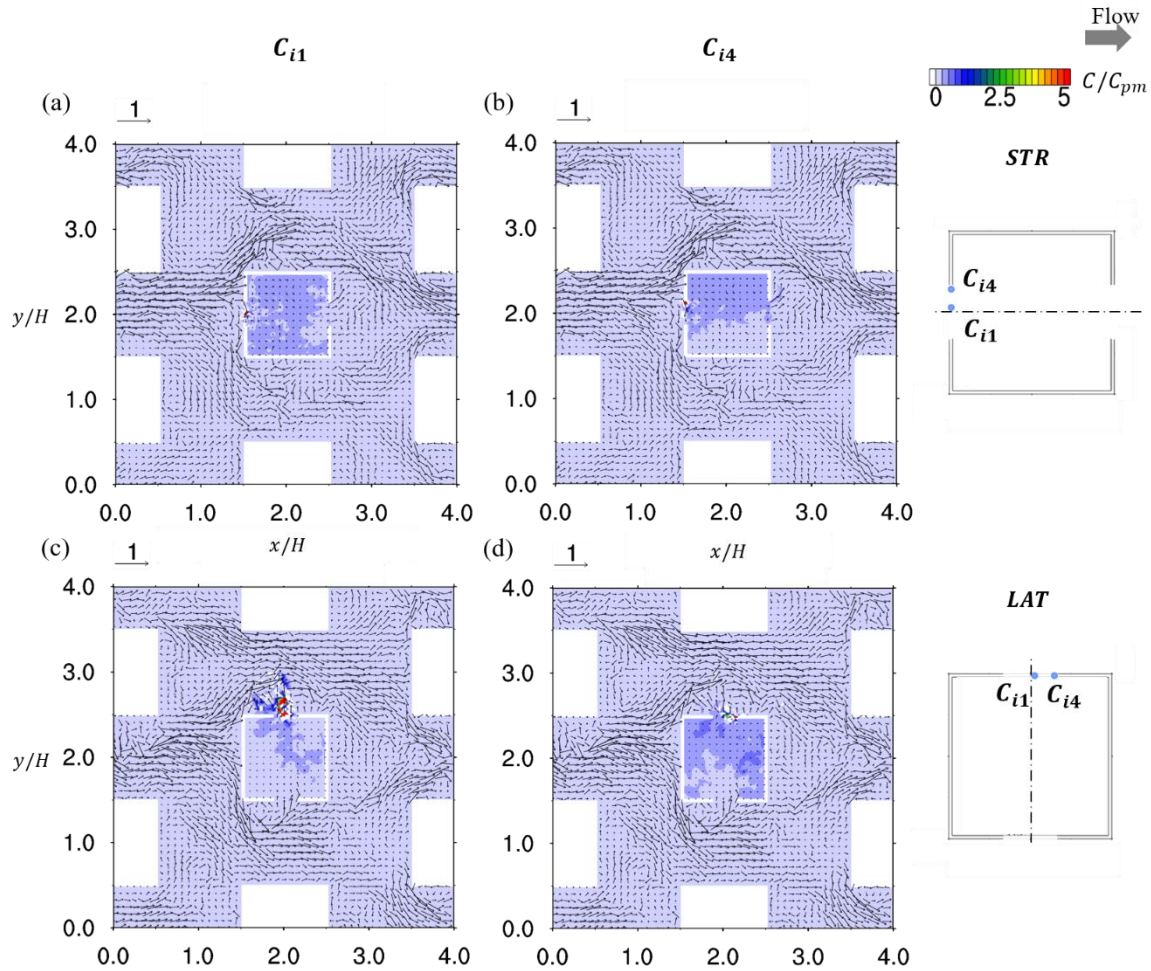


Figure 3.7. STR and LAT cases instantaneous scalar distribution of C_{i1} and C_{i4} at arbitrary time t .

In contrast, scalar sources positioned at the corners or stagnation points, such as the scalar C_{uc} of both STR and LAT cases, had concentration accumulations near their respective sources with slight change for the entire duration. Finally, for the STR case, where the scalar source was positioned near the outlet, the concentration of the scalar C_o immediately left the indoor space after generation; therefore, almost no concentration can be observed inside the indoor space.

3.3.2 Mean Scalar Distribution and Velocity Field

In this section, we discuss the qualitative and quantitative characteristics of temporally averaged scalar dispersion. Figure 3.8–3.8 depicts the mean scalar distributions superimposed with the mean velocity vector field taken on the x – y plane at $z/H = 0.5$ for both the STR and LAT cases. The scalar fields were normalized with the perfect mixing concentration C_{pm} , whereas the velocity fields were normalized using the reference velocity field u_{ref} .

Figure 3.8 shows the time-averaged scalar distributions of C_{i1} and C_{i4} for the STR and LAT cases. In the STR case, scalars C_{i1} and C_{i4} exhibited similar behaviors to their instantaneous counterparts. For C_{i1} (Figure 3.8a), the scalar concentration spread throughout the entire x – y plane, except near the inlet ($x/H = 1.5$). This lower concentration occurred because of the flow introduction being directed towards the bottom or negative z -direction with velocity values of approximately 1.08 to 1.5 m/s as observed in the study by Adachi [32]. In contrast, C_{i4} (Figure 3.8b) was concentrated at $y/H > 2.0$, with some scattering at $y/H < 2.0$ particularly near the outlet side ($x/H = 2.5$). For the LAT case, C_{i1} (Figure 3.8c) had a time-averaged concentration distribution accumulated near the boundary, which can be explained by the fluctuations or frequent reversal of flow, particularly in the middle of the inlet (Figure 3.10b and 8c). Moreover, scalar C_{i4} (Figure 3.8d) was scattered inside the room because its source location was near the side of the window that experienced less frequent reversal of flow owing to the no-slip condition, and its local mean flow field was directed towards the indoor space.

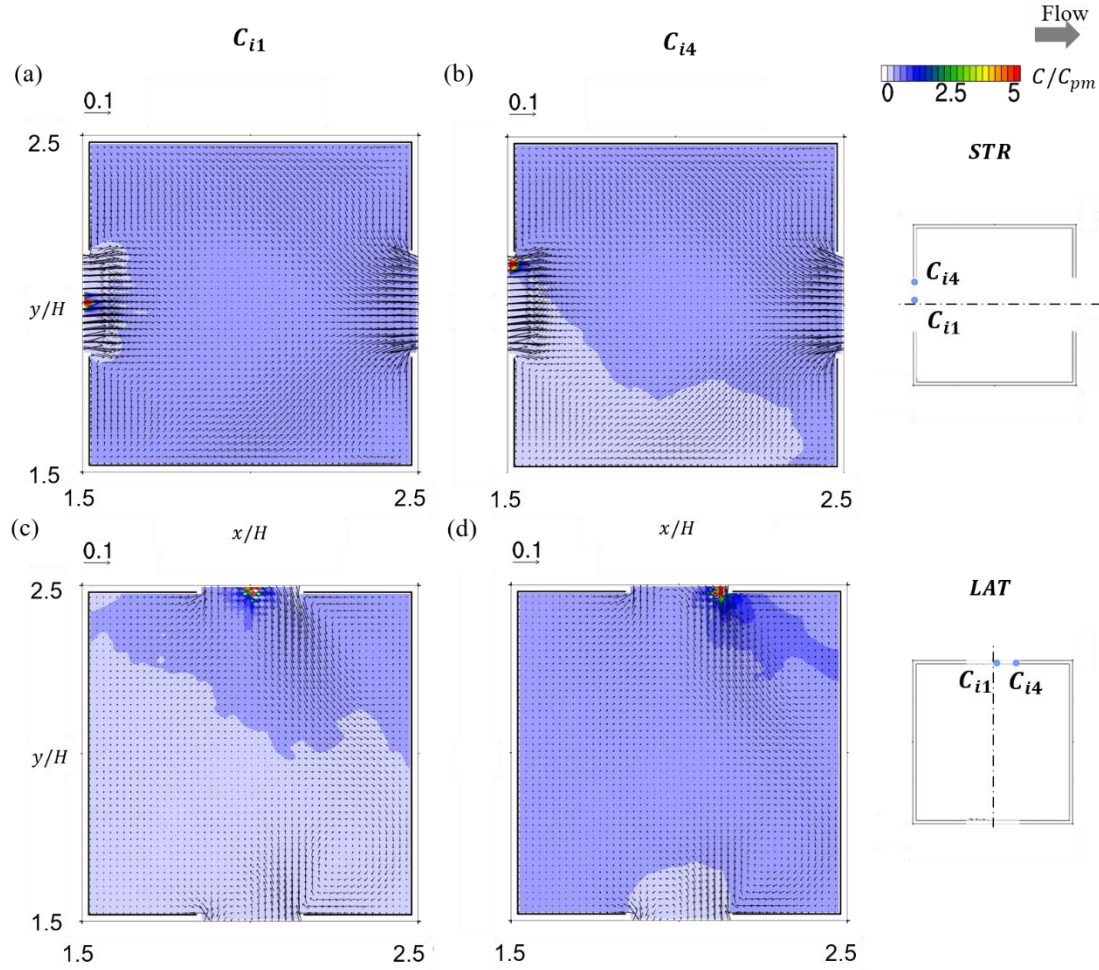


Figure 3.8. STR and LAT time-averaged scalar distribution of C_{i1} and C_{i4} .

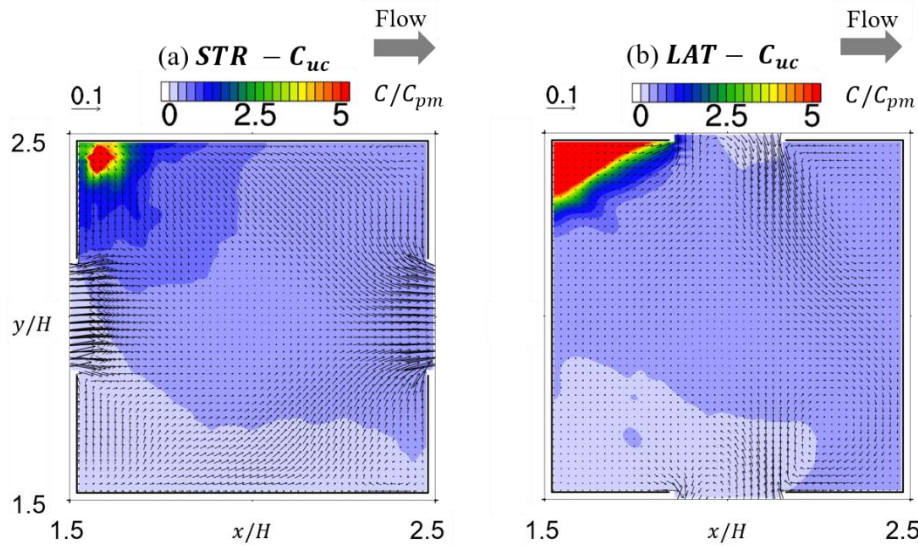


Figure 3.9. STR and LAT time-averaged scalar distribution of C_{uc} .

As with their instantaneous counterparts, the scalars positioned at corners or stagnations points, such as scalar C_{uc} (Figure 3.9), of both the STR and LAT cases had their respective

mean scalar concentrations accumulated near the source location. In contrast, the mean scalar distribution for C_{dw} (Figure 3.10a) remained near its source location and scattered to $y/H = 1.5$ and $y/H = 2.5$ (both near $x/H = 2.5$) owing to the fluctuations in the inlet flow introduction (Figure 3.10b and 9c).

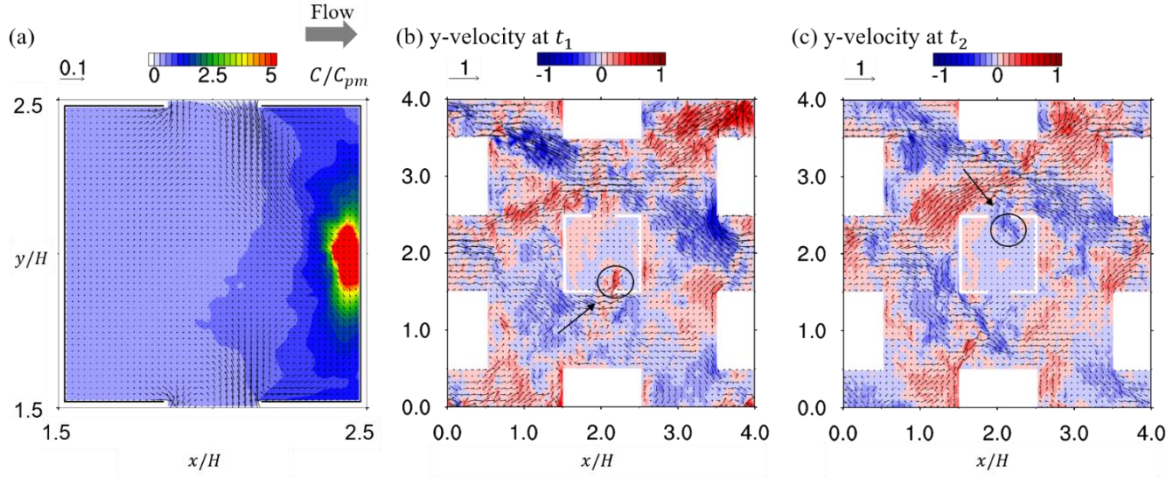


Figure 3.10. a) LAT time-averaged scalar distribution of C_{dw} and the fluctuating flow introduction b) from the $y/H = 1.5$ inlet and c) from the $y/H = 2.5$ inlet.

3.4. Ventilation Indices/Efficiencies Results and Other Summaries

3.4.1 Net Escape Velocity

In this section, we discuss the time-averaged NEV* field superimposed in the concentration field inside the indoor space taken at the x–y plane at $z/H = 0.5$ for both the STR and LAT cases. Here, NEV* was decomposed into two components: the advection part (NEVA) and turbulent diffusion part (NEVT) for further examination.

$$\overline{NEV_j^*}_a = \frac{1}{\langle C \rangle} \left(\overline{\langle u_j \rangle \langle C \rangle} \right) = \overline{\langle u_j \rangle} \quad (3.11)$$

$$\overline{NEV_j^*}_t = \frac{1}{\langle C \rangle} \left(\overline{\langle u_j \rangle' \langle C \rangle'} \right) \quad (3.12)$$

The scalar fields were normalized using the perfect mixing concentration C_{pm} , and the velocity fields were normalized using the reference velocity u_{ref} .

NEVA (Figure 3.11c) appeared to be the velocity vector field (Figure 3.11a), with the flow direction coming from the inlet of the indoor space. This can be explained by referring to Eq. (3.8) and citing the first term on the right-hand side of the equation as the NEVA part (Eq. (3.11)). The concentration component was canceled out, leaving the velocity term, making NEVA equal to the velocity. However, NEVT (Figure 3.11d) exhibited a different vector field,

particularly at the inlet location where the vectors pointed against the direction of the flow. This counterflow demonstrated a path directed upstream of the inlet, where the observed concentration values were lower (even zero) than that in the indoor space, thus following the transfer of concentration from a region of high concentration to low concentration. Moreover, the NEV* in Figure 3.11b illustrates how the velocity field (as well as NEVA) was altered due to NEVT (velocity representation of the concentration gradient). Thus, we can conclude that the NEVT is the “induced velocity field” owing to the concentration gradient, and NEV* is the local ventilation considering both the velocity and concentration gradient.

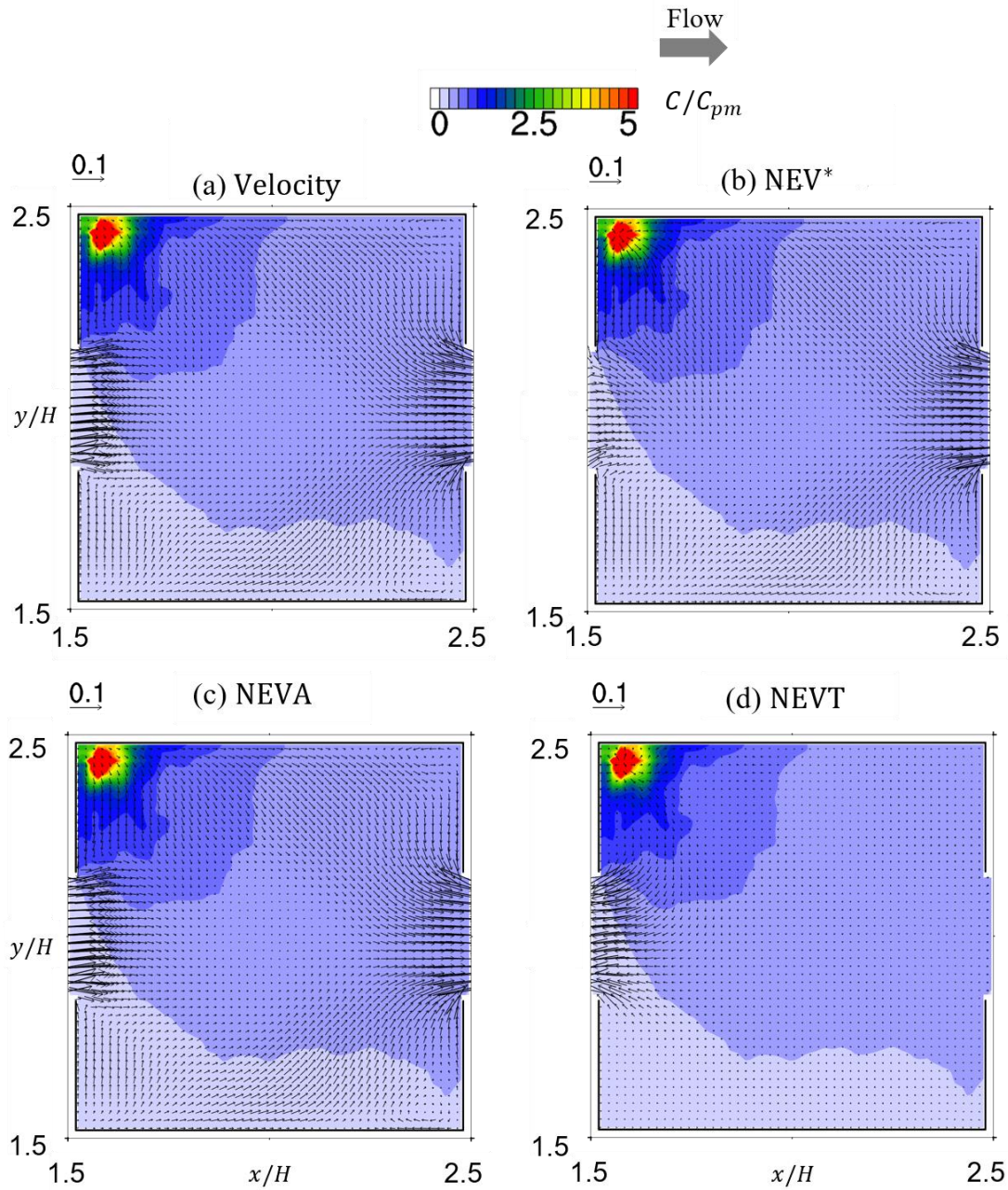


Figure 3.11. STR time-averaged net escape velocity vector field for C_{uc} : (a) Velocity, (b) NEV*, (c), NEVA, and (d) NEVT.

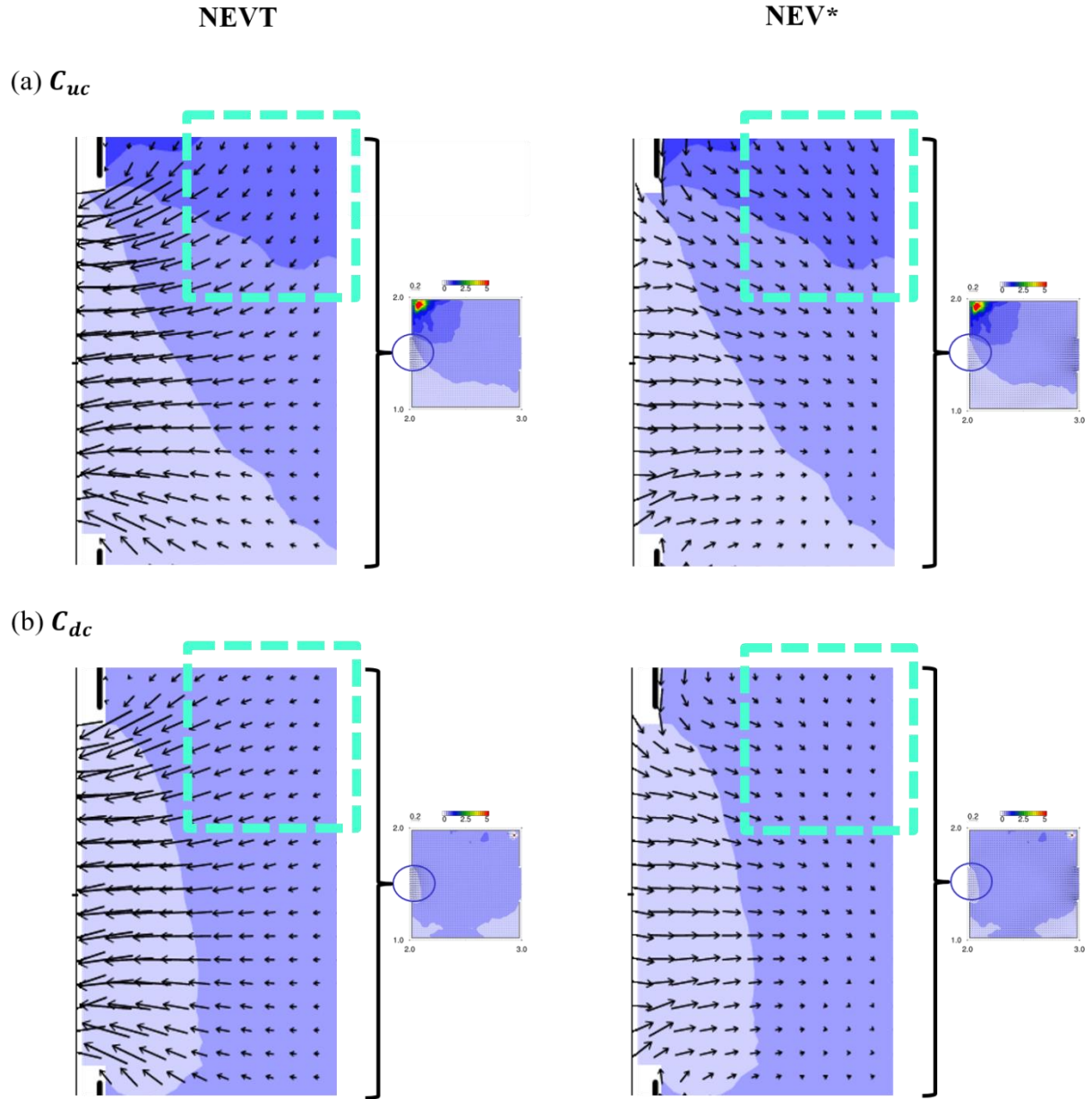


Figure 3.12. Magnified NEVT and NEV* of (a) C_{uc} and (b) C_{dc} .

In addition, Figure 3.12 presents the magnified indoor inlets of scalars C_{uc} and C_{dc} of the STR case. Note that they had the same velocity field (as well as NEVA). The location in which the differences between their respective NEVT and NEV* were noticeable is enclosed in the box with dashed corners. Clearly, the NEVT fields of C_{uc} and C_{dc} exhibited different behaviors, which resulted in differences in the resulting NEV* fields.

Figure 3.13 demonstrates the time-averaged NEV* decomposition of its advection (NEVA) and turbulent diffusion (NEVT) components. The velocity values were normalized using the reference velocity u_{ref} . Generally, the normalized values of the NEV* components of the STR cases yielded higher values than those of the LAT cases because of the higher

velocities experienced in the indoor space in the STR case. The NEV^* components of the STR case (Figure 3.13b) reached up to 0.3 for scalar distributions with a scalar source at the inlet of the indoor space ($C_{i1} - C_{i4}$), whereas the LAT cases (Figure 3.13a) yielded only up to 0.039. For the STR case, NEVA produced a greater value than NEVT except for C_{uc} , which meant that advection generally dominated the resulting scalar distribution in comparison with turbulent diffusion. For the LAT case, where the mean velocities were relatively low, NEVT became comparable with NEVA, which meant that the effect of turbulent diffusion was significant. This finding reinforced the speculation by Adachi et al. [32] that the effect of turbulence on ventilation should be considered when mean velocity field is relatively low. Flow fluctuations producing near-zero time-averaged values and stagnation points at the corner can still have ventilation, where turbulent diffusion serves as the driver for scalar transport. Finally, this proved that the boundary-based ventilation flow rates, which are computed based on the boundary velocity, are limited in predicting the ventilation at different sections of the indoor space because of the non-uniformity of the indoor velocity and the effect of turbulent diffusion.

Also in Figure 12, cases in which NEVA was greater than NEV^* (C_{uc} of STR case) and vice versa (C_{uc} and C_{dc} of the LAT case) can be observed. This $NEVA > NEV^*$ condition means that the advection velocity has a different direction from the induced velocity owing to the concentration gradient, thus decreasing the overall ventilation index NEV^* . On the other hand, $NEVA < NEV^*$ means that the advection velocity has a direction similar to that of the induced velocity due to the concentration gradient, thus increasing the overall ventilation index NEV^* . Thus, the induced velocity owing to the concentration gradient may or may not increase the overall ventilation (NEV^*) depending on its direction with respect to the velocity. In addition, the $NEV^*/NEVA$ values at the location of the different scalars were computed and tabulated (Table 3.1). A value of less than 1 indicates diminishing ventilation (the direction of velocity and induced-velocity are different), whereas a value greater than 1 indicates that ventilation is amplified due to turbulent diffusion.

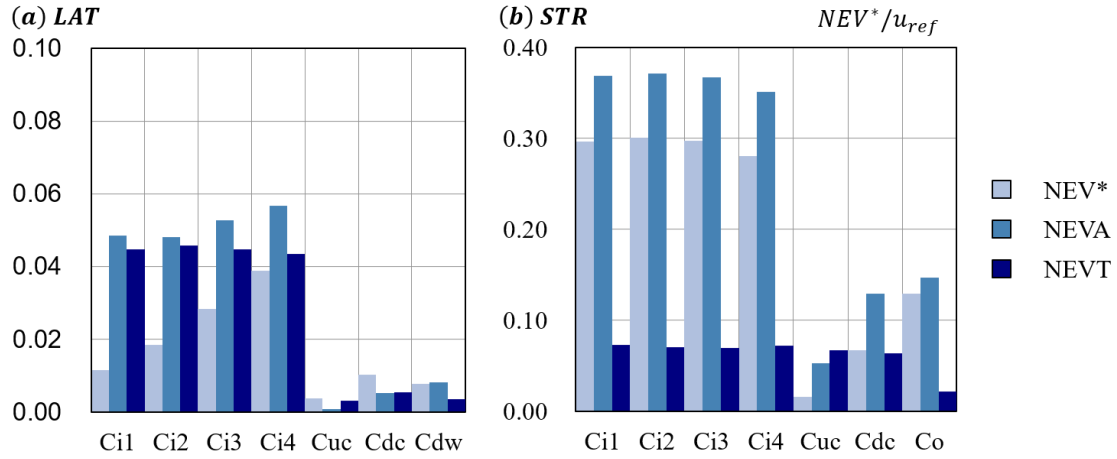


Figure 3.13. NEV* decomposition computed at the source location of the different scalar distributions for both (a) LAT and (b) STR cases.

Table 3.1. NEV*/NEVA for each source location.

LAT		STR	
C_{i1}	0.24	C_{i1}	0.24
C_{i2}	0.39	C_{i2}	0.39
C_{i3}	0.54	C_{i3}	0.54
C_{i4}	0.68	C_{i4}	0.68
C_{uc}	4.24	C_{uc}	4.24
C_{dc}	2.00	C_{dc}	2.00
C_w	0.96	C_w	0.96

3.4.2 Purging Flow Rate

A Figure 3.14 depicts the PFRs at the scalar source location of the different scalar distributions for both the LAT and STR cases. Note that all of the scalars had the same generation rate $\dot{m}$, making the variation in PFR solely dependent on the time-averaged concentration at the given source locations. Almost uniform PFRs were observed at the scalars that had their source near the inlet ($C_{i1} - C_{i4}$) for both cases (STR and LAT). For LAT cases, these values were 0.25–0.26, whereas they were 0.32–0.34 for STR cases. This indicated that the STR sources had a higher rate of removal of concentration (PFR), making the time-averaged concentration in the area small in comparison with the LAT case.

Furthermore, the PFR values at the stagnation points (C_{uc} , C_{dc} , and C_{dw}) of the LAT case were approximately 0.05. For the STR case, scalar C_o (positioned at the outlet of the case) had a relatively similar PFR as its inlet counterparts because its source was positioned where the

flow magnitude was almost the same as that of the inlet. Furthermore, the scalar C_{dc} in the STR case produced the highest PFR.

As Figure 3.13 and Figure 3.14 show, the correlation between PFR and NEV^* were unclear, although the two indices quantitatively increased when the mean velocity was high. Compared with the STR case, the lower correlations between NEV^* and PFR in the LAT condition implied that these indices may differ when the turbulent flow contributes to the determination of the local concentration.

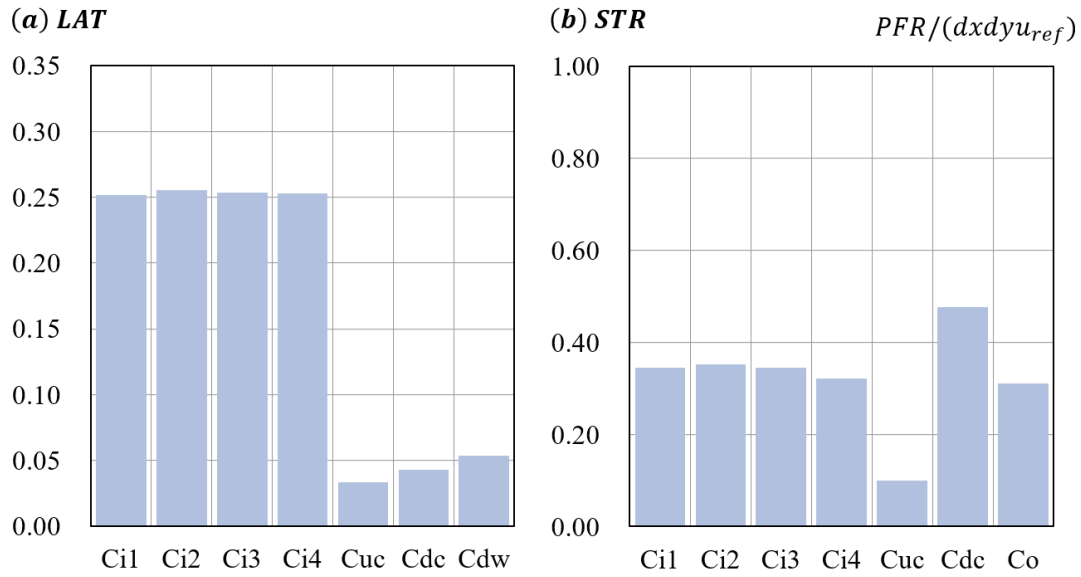


Figure 3.14. Purging flow rates of the (a) LAT and (b) STR cases.

3.4.3 Age of Air

Figure 3.15 presents the AOA distributions for the STR and LAT cases. Note that these two were computed based on uniform generation; therefore, the AOA was dependent only on the boundary flow rate and the resulting concentration profile. Here, the AOA of the STR and LAT cases were incomparable because they had different boundary ventilation flow rates: 5.3×10^{-4} and $1.2 \times 10^{-4} m^3/s$, respectively. Additionally, the computation of the AOA used the gross ventilation rate because the net ventilation rate of the LAT case approaches zero value because of fluctuations in the direction of the flow at the boundary. The LAT case generally had a lower AOA because the perfect mixing concentration C_{pm} was very high owing to the low gross ventilation rate (see Eq. (3.10)).

For the LAT case, the AOA was generally low near the inlet/outlet boundaries and generally high at the upwind wall ($x/H = 1.5$). Although the time-averaged velocities in the inlet/outlet boundaries were low, the instantaneous fluctuations enabled the concentrations to

be transported from the location; thus, they also produced a low AOA. This again demonstrated the significance of the turbulence when the mean flow is weak. However, for the STR case, a very high AOA was also observed near the middle where both the instantaneous and time-averaged velocities were low.

NEV*, as well as its components NEVA and NEVT, were also superimposed on the distribution of AOA. Here, the NEVTs at the inlet were always directed towards the outside of the indoor space, as those were the regions in which the concentration was generally lower. Thus, the NEVTs counteracted the NEVAs that were directed to the indoor space in both cases. The low AOAs at the inlet in both cases were due to the high ventilation rate they experienced—due to both advection and turbulent diffusion for the LAT case, and generally only advection for the STR case, as illustrated by the NEV* vectors in Figure 3.15.

Finally, we observed that the inlet and outlet of the two cases generally had low AOAs. These regions also had a high PFR, as described in Section 3.4.2. An inverse relationship between AOA and PFR is expected from Eqs. (3.9) and (3.10).

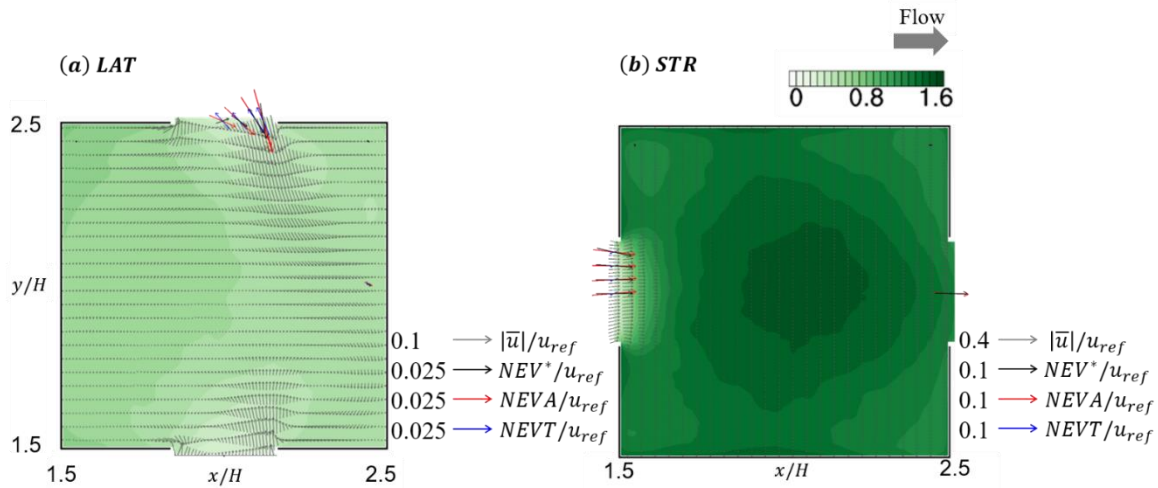


Figure 3.15. Age of air of (a) LAT and (b) STR cases.

3.4.4 Relationship of Overall Ventilation Flow Rates and Ventilation Indices

The work of Adachi et. al [32] utilized the boundary-based ventilation rates (Eq. (3.2) and Eq. (3.3)) that are based on temporally-averaged and then spatially-summed velocities at the window boundary. Dividing these ventilation rates by the total volume of the indoor space will yield to air change rate (Eq. (3.1)) – that is typically used as ventilation representation in building design. These values, being evaluated at the boundary using velocity, may not necessarily be equal to the ventilation at each sections of the indoor space, particularly at mesh level. Scrutinizing the mean velocity field inside the room, high velocity points at the inlets

and then stagnation points at the corners, make these kinds of boundary-based ventilation rate insufficient when we talk about the ventilation at local level – which are beneficial in predicting contaminant transport inside the indoor space and then to the building design. In addition, although they accounted for the turbulent fluctuations in addition to velocity through the computation of instantaneous ventilation rate (Eq. (3.4)), its value may again not reflect the turbulent fluctuations at mesh level. As shown in the different distributions (concentration (Figure 3.7-3.9), NEV^* (Figure 3.11-3.13), and PFR/AOA (Figure 3.15)), various locations in the indoor space yielded different values with that of the inlet values. These distributions happen due to imperfect mixing conditions, therefore, making the boundary-based ventilation rates not representative of the local ventilation.

3.4.5 Overall Effect of Flow Introduction Owing to the Outdoor Flow Pattern and Window Openings

Based on the previous sections, the flow introduction due to the outdoor flow and window openings significantly affected the resulting indoor concentration and eventually the ventilation indices. For example, in the STR case where the flow introduction had higher velocity magnitudes, the resulting NEV^* s (particularly for scalar concentrations located at the inlet of the indoor space, $C_{i1} - C_{i4}$, had their ventilation or scalar concentration transport dominated by advection. One of these scalar concentrations (C_{i1}) also produced a small concentration near the source location, as observed on the target plane (x–y plane at $z/H = 0.5$) because of the mean flow introduction directed towards the negative z-direction (Figure 3.8a), while allowing the concentration to spread throughout the domain. In addition, this downward flow introduction generated stagnation at the middle of the same target plane that resulted to high AOA (Figure 3.15b). In contrast, for the LAT case, where the flow introduction had a lower mean velocity, the resulting NEV^* s (particularly for scalar concentrations located at the inlet of the indoor space, $C_{i1}-C_{i4}$, and at the downwind corner C_{dc}), had their ventilation or scalar concentration transport significantly affected by turbulent diffusion in addition to advection. This has manifested in the resulting mean scalar distribution of C_{dw} (Figure 3.10a), that even though have generally low mean flow field, the scalar was able to spread out due to the fluctuating flow introduction coming from both of the two windows of the LAT case. These show how the resulting flow introduction due to outdoor flow and window openings should be taken into consideration in harnessing natural ventilation effectively.

3.4.6 Overall Effect of Scalar Source Location

The seven scalars (of both STR and LAT cases) studied in this paper resulted to different concentration distributions even having the same respective velocity fields. Generally, the local

mean velocity field and the local flow fluctuations near the scalar source location dictated the concentration distributions. For instance, for both the STR and LAT cases, scalars C_{i1} and C_{i4} that were positioned near each other, yielded different scalar distributions. Scalar C_{i1} being positioned at the middle of the window inlet followed the general flow introduction: being pushed to the bottom of the indoor space and then eventually to the other location for the STR case (Figure 3.8a), while being pushed back and forth due to flow fluctuation near the window therefore spreading only near its location for the LAT case (Figure 3.8c). On the other hand, the resulting mean concentration distributions for C_{i4} (Figure 3.8b and 3.7c) deviated from C_{i1} for both of the cases. As these scalars are positioned near the wall, the no-slip condition lowered the local velocity magnitudes thus affecting the resulting spread of the scalars. Furthermore, for both of the cases, scalars positioned at the corner yielded concentration distributions that are concentrated near their respective source location as the local flow field were stagnant. These show how the scalar source location should be taken into consideration in natural ventilation design. For example, these scalars can represent an occupant that may be infected by an airborne disease, therefore, indoor lay-out can be to prevent cross contamination while relying on natural ventilation.

3.5. Conclusion

In this study, we conducted isothermal LESs to calculate the ventilation of an indoor space sheltered by urban-like block arrays. Previous studies have reported boundary-based ventilation rates computed using the velocity fields. In contrast, this study integrated the concentration field to compute the ventilation indices: 1) age of air (AOA), 2) purging flow rate (PFR), and 3) net escape velocity (NEV*), as imperfect mixing condition exists within the indoor space. We separately solved seven scalar concentration distributions with sources located at the indoor space inlet, corners, and outlet to investigate the effects of the high-velocity points, velocity fluctuations, velocity circulation, and stagnation points on local ventilation. Finally, two opening positions were simulated, streamwise (STR) and lateral (LAT), as these cases produced different velocity fields from the previous study.

Both the instantaneous and mean concentration fields indicated that the scalar source location affects the resulting indoor ventilation in addition to the velocity profile. Although they had the same velocity fields, the concentration fields of all seven scalar concentrations (for both STR and LAT) yielded different results. For example, a significant difference was observed for concentrations located at the inlet boundaries, C_{i1} and C_{i4} in both cases, although their respective source locations were near each other. The nature of the flow near the source locations, whether they were positioned in high-velocity, fluctuating velocity, circulating velocity, or stagnation regions, dictated the individual concentration distribution.

Accordingly, the scalar sources positioned at the inlet of the LAT case, where the mean flow field approached zero, could still disperse inside the indoor space. This is because the instantaneous velocity fluctuation (turbulence) still drives scalar concentration transport even when the average velocity is zero. This finding reinforces the speculation of Adachi et al. [32] that turbulence must be considered when the mean flow is weak.

Moreover, NEV^* , which represents the velocity of the escaping and returning concentration population, was calculated to evaluate the local ventilation in the indoor space. Previous studies have derived such ventilation index using RANS simulations. This study, on the other hand, determined the NEV^* in time-dependent simulations. In addition, we divided NEV^* into two components: 1) advection component ($NEVA$) and 2) turbulent diffusion component ($NEVT$). The $NEVA$ field appeared to be only the velocity field, whereas the $NEVT$ field appeared to be a velocity field in which the vectors were directed from a region of higher concentration to a lower concentration. Thus, we can conclude that $NEVT$ is the induced velocity field due to the concentration gradient, and NEV^* is a ventilation index in terms of the velocity scale accounting for the effects of advection and turbulent diffusion.

In addition to NEV^* , we computed the AOA and PFR. With the same scalar generation rate for each scalar concentration, the PFR represents the necessary flow rate that determines the scalar concentration in a specific local domain (cell). Higher purging flow rates were observed at locations with lower scalar concentrations. The PFR then implies that if there is a higher concentration removal rate, a lower concentration can be observed in a certain local domain, and vice versa. In contrast, AOA is the ratio of the concentration at a specific local domain over the perfect mixing concentration or the average concentration at the boundary of the indoor space. A very high AOA can be observed at locations with high concentrations, and low ages at low concentrations. This implies that as an increasing number of scalars accumulate in a specific local domain, the value of the representative AOA increases, and vice versa. Moreover, an inverse proportion with a multiplier of a boundary-based flowrate is expected between the PFR and AOA, and there are no observable relationships with NEV^* . We speculate that NEV , not NEV^* , is much more related to these two ventilation indices; thus, additional research is necessary.

This study also demonstrated that the introduction of outdoor flow owing to outdoor flow patterns and window openings significantly affects the resulting concentration field and ventilation. This was manifested by the continuous high-velocity flow for the STR case in comparison with the fluctuating flow for the LAT case, where their corresponding ventilation was significantly affected by advection alone, and both advection and turbulent diffusion, respectively.

As urbanization is inevitable due to the increasing population, this study gives a glimpse on how to better utilize the natural ventilation inside building spaces in cities, that the general boundary-based flow and air change rates utilized in the previous studies may fail to provide. Although this utilized generic blocks, this study showed how the ventilation rate typically evaluated in boundaries, does not reflect the entirety of the indoor space. Different levels of ventilation have manifested at different sections of the indoor space due to different flow patterns. For instance, corners of the STR case have very low NEV* and high AOA as these regions are typically the stagnation points where contaminants settle in contrast with the inlet where air continuously flows that drives the contaminant transport. By exploring the indoor flow field response to the varying outdoor flow field through the use of concentration distribution and then the ventilation indices, this study may contribute to the design of indoor lay-out such as the positioning of occupants to acquire enough ventilation, and prevent cross contamination in the case of airborne diseases, thus maximizing the natural ventilation potential. The current study through the different scalar source locations that can represent the occupants in real life scenarios answers the questions: “Will the virus exhaled by this occupant spread throughout the indoor space? will it just settle? or will it be directly exhausted outside?”, while considering the effect of natural ventilation.

Overall, this LES study demonstrated the limitations of boundary-based ventilation rates while emphasizing the general importance of scalar source location, introduction of outdoor flow, and the effects of advection and turbulent diffusion on local ventilation. We have shown that the use of ventilation indices, based on velocity and concentration fields, is necessary as ventilation is dependent on both advection and turbulent diffusion, and not solely on advection, which highlighted the limitation of the previous sheltering effect studies.

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CHAPTER 4 – QUANTIFICATION AND OPTIMIZATION OF LOCAL AND AVERAGE INDOOR AIR QUALITY AND THERMAL COMFORT IN HYBRID DISPLACEMENT-PERSONALIZED VENTILATION IN AN OFFICE

4.1 Background

Most heating, ventilation, and air-conditioning (HVAC) systems installed in facilities such as households, offices, and schools are designed based on well-mixed conditions that are expected to provide an acceptable temperature and ventilation (depending on the system) for the entire space. However, this design results in potential energy-saving losses, as the air in unutilized spaces is conditioned as well. In reality, such a design may fail to provide the necessary thermal comfort and indoor air quality (IAQ) to the occupants, even though sensors that adjust the HVAC system settings detect acceptable ranges, which can be attributed to imperfect mixing conditions. Occupant-centric systems have been developed to address these challenges. These systems are based on the segregation of the indoor space into two zones: the task zone (or the zone occupied by the occupants) and the ambient zone (the unoccupied zone where ventilation and air conditioning are not required) [1]. Depending on the occupants' requirements, the settings of the HVAC system can be adjusted to that of a specific occupied zone. Furthermore, systems can be rendered more occupant centric by focusing on the occupants' average skin-surface temperature and other related metrics, such as thermal comfort and inhaled contaminant values, instead of the IAQ value for the occupied zone.

For decades, standalone personalized-ventilation (PV) systems or PV systems installed with the main ventilation have been adopted to realize occupant-centric systems. The PV system adjusts the temperature and IAQ by supplying fresh and conditioned air to the immediate surroundings of the occupants [2]. For standalone systems, Song et al. [3] summarized PV systems with controls based on indoor CO₂ concentrations, which generally provided lower CO₂ indoor concentrations at lower energy consumption. PV variations such as desk-[4] and ceiling-mounted systems have been reported [5], [6]. Meanwhile, other PV studies have focused on sleeping environments. Ning et al. [7] and Mao et al. [8] investigated the effects of supply-air temperature, flow rate, and vane angle on a bed-based PV system. Meanwhile, Mao et al. [9] and Mao et al. [10] performed the simultaneous optimization of energy and thermal comfort by varying the aforementioned parameters, which yielded 10%–19% energy savings. However, PV systems can cause thermal discomfort as they are typically designed to be positioned near the occupant [11]. Increasing the supply flow rate or decreasing the temperature beyond certain levels might cause discomfort such as skin irritation, thus posing a challenge in managing both thermal comfort and IAQ for standalone PV systems. Ma et al. [12] simultaneously verified breathable air quality and sensation on different parts of the

body by varying PV supply temperatures and velocities. Hence, PV is generally more reliable when coupled with the main ventilation system. Yang et al. [6] discovered that combining PV and mixing ventilation (MV) can provide energy savings of up to 16%, particularly at elevated room temperatures. Moreover, Xu et al. [13], Liu et al. [14], and Katramiz et al. [15] evaluated the transmission of airborne infection by varying the PV flow rate and occupants' position and distance, while being incorporated with MV. Al Assaad et al. [16] demonstrated that integrating PV into a chilled ceiling system affords energy savings of up to 15% while maintaining good IAQ, as indicated by the CO₂ concentration and thermal comfort based on the segmented skin temperature. Finally, other studies focused on the design of PV systems in densely populated areas, such as aircraft cabins and schools [17], [18], [19].

The most commonly evaluated metrics for typical indoor systems are the IAQ, thermal comfort, and energy requirements. For different inputs, the abovementioned outputs may indicate contrasting responses, thus necessitating the use of multicriteria decision making (MCDM) methods. In MCDM methods, the first challenge is to obtain an adequate amount of data through experiments or simulations to characterize the responses to the inputs. In several indoor analyses, the Taguchi design of experiment (DOE) was used in conjunction with the Taguchi signal-to-noise (S/N) ratio function to minimize the number of test cases, such as in the optimization of underfloor air-distribution (UFAD) systems for data centers [20], [21] and in natural ventilation [22]. The Taguchi S/N ratio is sufficient for optimizing a single parameter. However, to perform simultaneous optimization, the Taguchi design and analysis can be paired with the MCDM method, such as the technique for order preference by similarity to ideal solution (TOPSIS). The TOPSIS combines several outputs into one objective function, in which a value closer to 1 implies proximity to the ideal value, considering all the output variables. In the elaborated occupant-centric studies, only Mao et al. [10] used the TOPSIS to simultaneously optimize thermal comfort and energy consumption. Similarly, Qin et al. [23] used this method to simultaneously optimize the IAQ, thermal comfort, and cooling load in a UFAD system. In a recent study, Yang et al. [24] adopted the TOPSIS along with the Taguchi design to optimize four temperature indices simultaneously in UFAD systems for data centers. They successfully identified combinations of inputs outside the Taguchi design that yielded an optimized value, thus proving the viability of the Taguchi–TOPSIS method for indoor environmental analysis.

The studies reviewed above generally discussed the effects of several inputs of occupant-centric systems on thermal comfort, IAQ, and energy separately, or on two of their parameters at a time, instead of simultaneously optimizing all three parameters. In addition, most optimization studies have focused on adjusting the input parameters to optimize PV systems while maintaining the main ventilation parameters, even though the latter significantly affect

the general flow field, temperature, and IAQ. Hence, this study aims to simultaneously optimize thermal comfort, IAQ (in terms of CO₂ concentration), and energy consumption by varying both the main ventilation parameters (in this case, the displacement-ventilation (DV) parameters), particularly the supply temperature, flow rate, and recirculation value, and the PV input parameters, particularly the supply temperature and flow rate, in an office room with two occupants. We perform several steady-state computational fluid dynamics (CFD) simulations by varying the input parameters based on a DOE to obtain several outputs and then perform multicriteria optimization to determine the best combination of DV and PV input parameters that yields optimum, balanced, and acceptable output parameters. To analyze the difference in results and the effects on the optimization, we vary the output parameters used for optimization, which include the segmented skin temperature, average skin temperature, occupant-zone (defined by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers standard (ASHRAE) [25]) volume-averaged temperature, and room volume-averaged temperature, as thermal-comfort parameters; and inhaled CO₂ concentration (units: parts per million, ppm), occupant-zone volume-averaged CO₂ concentration, and room volume-averaged CO₂ concentration, as IAQ parameters. This study may aid in the design of more occupant-centric ventilation systems that can provide acceptable thermal comfort and IAQ to occupants at relatively low energy inputs.

4.2. Methodology

4.2.1 Computational domain and boundary settings

We utilized a two-occupant office room measuring 4.8 m × 3.4 m × 2.6 m, as was used in previous studies [15], [26], [27], [28]. We specifically used the PV model validated by Katramiz et al.[15] measuring 10 cm in diameter and positioned 40 cm away from the occupants' face (Figure 4.16a). In contrast with the previous study [15], we used a simple DV instead of MV as our test case by allowing air to be supplied from a 0.6 m × 0.3 m supply inlet located at the bottom of one side of the room and then exhausted to an exhaust outlet of the same dimensions located at the top of the opposite side of the room. The air-handling units for the DV and PV were separated (Figure 4.16b). For the DV, a certain percentage of indoor air was exhausted outdoors, whereas some were recirculated. Subsequently, the recirculated air was mixed with the outdoor air and conditioned by the DV air-handling unit (AHU) before being supplied indoors. To couple the recirculation in the calculations, we used the user-defined function (UDF) of ANSYS Fluent. The mass-weighted average concentration at the outlet was computed and then multiplied by the recirculation rate or the fraction of outlet air being resupplied to the DV inlet. This value was then added to the product of the fraction of outdoor air intake and the outdoor concentration. The resulting sum served as the concentration at the

DV supply inlet. Meanwhile, the PV system extracted 100% of its air outdoors, which was then conditioned in the PV outdoor air-handling unit (OHU) before being supplied near the occupants. Therefore, the concentration at the PV inlet is the same with the concentration outdoors. We assumed an outdoor temperature of 14.9 °C, based on the average temperature of Hakata, Fukuoka, Japan during the final week of March [29], and an outdoor CO₂ concentration of 420 ppm. Furthermore, we assumed that the room was located inside an apartment building, where the floor, ceiling, and side walls interacted with adjacent rooms with indoor temperatures similar to those of the room of interest. Only the walls located behind the occupants interacted with the outdoor conditions. The thermal resistance of the wall was set to 1.1 m²-K/W[30]. For the internal-heat gain, we specified 11.9 W/m² for the lighting load [31] and 40 W for each monitor. Finally, long-wave radiation was employed by applying a surface-to-surface model to the software.

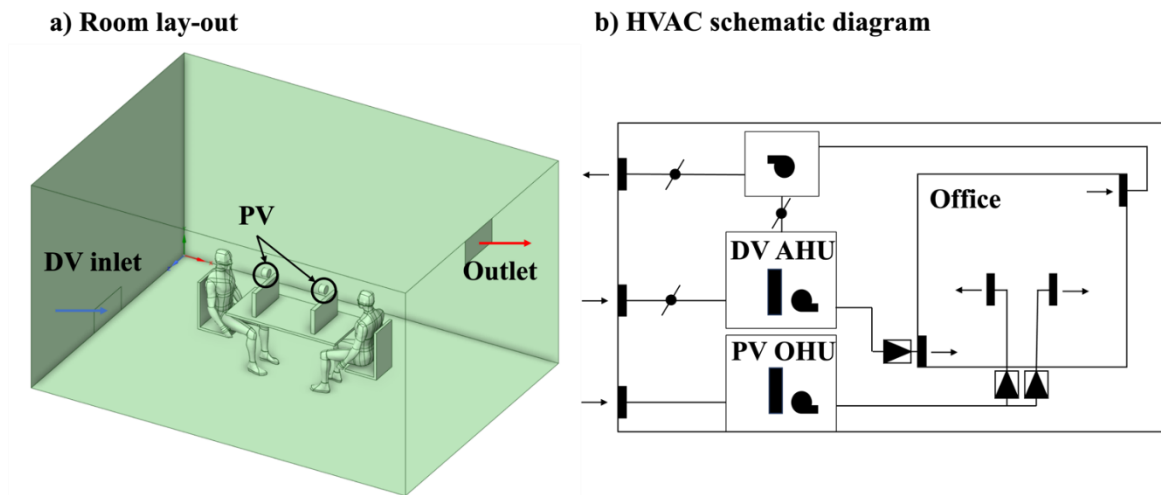


Figure 4.16. Room layout and schematic diagram of coupled HVAC system.

4.2.2 Occupant model and zone definitions

In this study, we used a sitting computer-simulated person (CSP) validated by Yoo and Ito [32], where we included the nasal cavity to accurately analyze the breathing flow (Figure 4.17b). For each simulation, we estimated an equivalent steady-state procedure for a transient breathing airflow rate (Eq. (4.1)) using a tidal volume (V_t) of 0.5 L and a respiration frequency (f_{res}) of 15 min⁻¹ [33]. The estimation was performed in four segments: 1) Steady-state computation of the flow field without breathing; 2) Based on procedure 1 as the initial condition, a 2 s transient exhalation based on the breathing volumetric profile expressed in Eq. (4.1) was performed with an exhaled CO₂ concentration of 40,000 ppm; 3) continuation of the transient simulation for another 2 s without the breathing flow rate while allowing the exhaled CO₂ to disperse based on the flow field; and 4) steady-state inhalation, where the inhalation flow rate was computed based on the averaged value from 2 s using Eq. (4.1) and with a steady

contaminant source based on the concentration field computed in procedure 3. In this process, 40,000 ppm was computed from a CO₂ emission rate of 0.005 L/s (18 L/h) based on previously summarized data [34]. This estimation procedure resulted in a shorter simulation time compared with performing a transient simulation and waiting until the steady state.

$$\dot{V}_{BR} = \frac{\dot{V}_{peak} \sin(2\pi f_{res} t)}{60} = \frac{\pi f_{res} V_t \sin(2\pi f_{res} t)}{60}, \quad (4.1)$$

where $\dot{V}_{BR}$ is the breathing air flow rate in L/s, and t is time in seconds.

As our focus was on occupant-centric metrics, we computed the skin temperature in units of degrees Celsius (Eq. (4.2)) based on the energy balance between a human and an indoor environment using Fanger's model assuming control of an experimental thermal manikin [35], [36]. Additionally, we partitioned the CSP skin into 10 segments (Figure 4.17a) based on a partition performed by Lai et al. [37] and Olsen [38], in which the computed neutral skin temperatures served as the reference temperatures for the subsequent sections. Finally, we disregarded the effect of humidity in our computations.

$$T_{skin} = 36.25 - (0.054 + R_{clothing}) \times (Q_t), \quad (4.2)$$

where 36.25 is the core temperature of human (units: °C), 0.054 is the thermal resistance of the body (units: m²-K/W), $R_{clothing}$ is the thermal resistance of the clothing (units: m²-K/W), and Q_t is the heat flux from the skin surface (units: W/m²). The clothing resistance (units: clo) were set as follows: 1) 0.25 (long-sleeved, dress shirt) for thorax, abdomen, upper arms, and lower arms; 2) 0.28 (thick trousers and brief) for upper legs; 3) 0.24 (thick trousers) for lower legs; and 4) 0.12 (boots with socks) for feet [39]. Fanger's model is based on the one-node concept, which is not a thermoregulation model originally intended for explicitly reproducing the inhomogeneous distribution formed inside or on the surface of the human body. However, when Fanger's model was applied for the skin-surface temperature control of an experimental thermal manikin, non-uniform thermal resistance was formed on the manikin skin surface as a result of reproducing the actual clothing condition. In this study, we aimed to reproduce this situation by employing conditions that provide thermal resistance corresponding to the clothing resistance at each segment.

Finally, as one of the cases to be compared is based on ASHRAE's definition of an occupied zone [25], we defined a volume inside the domain with boundaries 0.3 m away from the wall and beginning from the floor up to a height of 1.8 m.

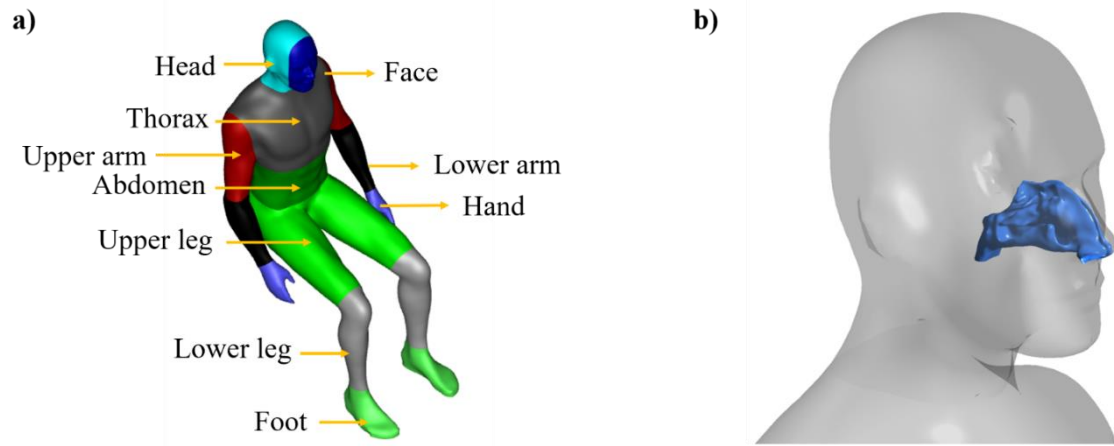


Figure 4.17. a) Skin segmentation, and b) nasal cavity of computer-simulated person.

4.2.3 Numerical settings, mesh independence, and validation and verification

We used the shear-stress transport model (SST $k-\omega$) to solve the steady-state Reynolds-averaged Navier–Stokes (RANS) equations via a semi-implicit method for pressure–velocity coupling in ANSYS Fluent. We utilized the Boussinesq approximation to model the buoyancy effects. To simulate the CO_2 concentration inside the domain, we used a scalar transport equation with the molecular diffusivity and turbulent Schmidt number listed in Table 4.2. Mesh-independent tests were conducted meticulously to minimize the number of cells without compromising the computed results. Four grid designs (0.88, 1.48, 2.41, and 4.90 million cells) were tested based on two values: 1) the human model area-weighted average temperature and 2) the mass-weighted velocity at a plane in the middle of the domain ($z = 1.7$ m). By computing the deviation of the abovementioned values based on values obtained using the number of cells above, we achieved mesh independence when we adopted 2.41 million cells, i.e., when the errors for the two abovementioned values were 0.00% and 0.13%, respectively. Therefore, for the subsequent simulations, we adopted a grid design with 2.41 million cells.

In this study, we performed CFD analysis by considering benchmark tests and guidelines, and the appropriate boundary conditions were set to ensure the desired quality [40], [41], [42], [43]. In particular, the CSP applied to this analysis was the same as the model whose prediction accuracy had been verified in previous studies [32]; furthermore, it was applied to the current analysis under conditions that maintained the grid design and numerical/boundary conditions. Additionally, we performed a comparison with the results of a previous study by Katramiz et al. [15] for the PV setting of 13 L/s (Figure 4.18). The scatter plot represents the CFD data, whereas the brackets represent the experimental data. Deviations were observed at the backs of the occupants, which can be attributed to differences in the main ventilation. The current

study used DV, whereas the previous study used MV. Nevertheless, the velocities measured from the faces of the two occupants showed good agreement.

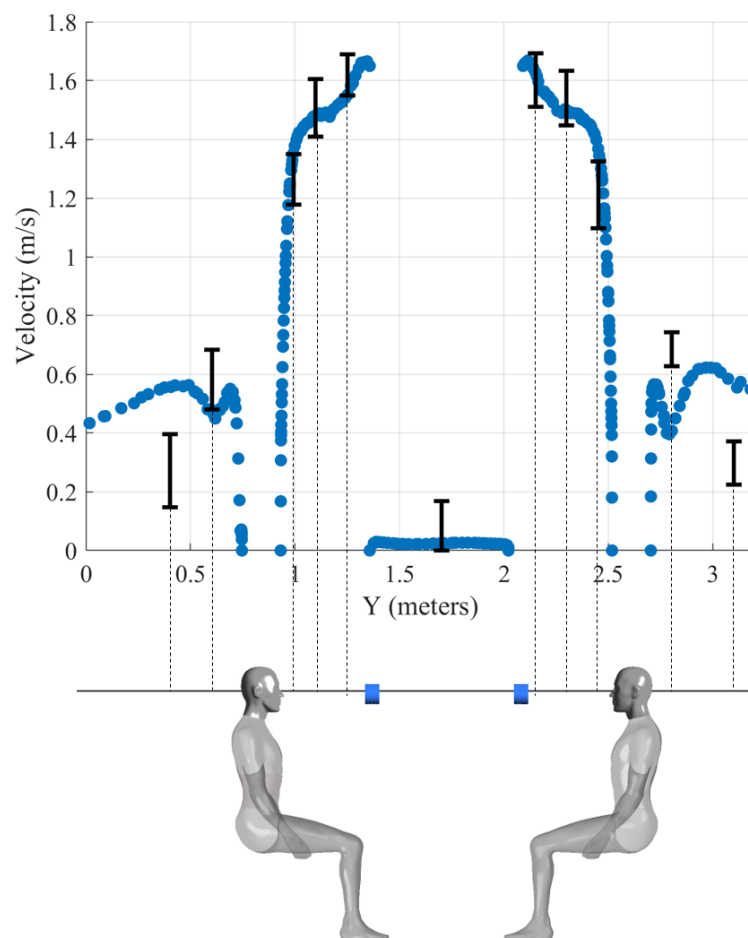


Figure 4.18. Velocity magnitudes of experimental [15] (bracket) vs. CFD (scatter plot) along line shown under PV flow rate of 13 L/s.

Finally, the settings used to solve the flow, temperature, and scalar fields are summarized in Table 4.2.

Table 4.2. Summary of computational settings and boundaries

Computational Settings	
Turbulence Model	SST $k-\omega$ [44]
Pressure–Velocity Coupling	SIMPLE
CO ₂ Molecular Diffusivity	$1.64 \times 10^{-5} \text{ m}^2/\text{s}$
Turbulent Schmidt Number	1.0
Long-Wave Radiation Model	Surface-to-surface model
Flow Field Boundaries	
DV Supply Inlet Velocity	Computed from DV flow rate

	$(u_{DV} = \frac{\dot{V}_{DV}}{A_{DV}(1000)})^*$
PV Supply Inlet Velocity	Computed from PV flow rate
	$(u_{PV} = \frac{\dot{V}_{PV}}{A_{PV}(1000)})^*$
Nasal-Cavity Inlet Velocity (exhalation)	Computed from first 2 s of breathing flow rate ($u_{BRE} = \frac{\dot{V}_{BR}}{A_{NC}(1000)})^*$ [33]
Nasal-Cavity Inlet Velocity (inhalation)	$u_{BRI} = -0.667$ m/s, average inhalation velocity [33]
Turbulent Intensity	10%
Turbulent Length Scale	7% of supply inlet hydraulic diameter
Thermal Boundaries	
Outdoor Temperature	14.9 °C
Wall Resistance	1.1 m ² -K/ W [30]
Human Thermoregulation Model	Fanger's Model [35], [36]
Human Core Temperature	36.25 °C
Human Resistance	0.05 m ² -K/ W
Clothing Resistance (Thorax, Abdomen, Upper Arms, Lower Arms)	0.25 m ² -K/ W [39]
Clothing Resistance (Upper Legs)	0.28 m ² -K/ W
Clothing Resistance (Lower Legs)	0.24 m ² -K/ W
Clothing Resistance (Feet)	0.12 m ² -K/ W
Lighting (Ceiling)	11.9 W/m ² [31]
Monitor	40 W
Emissivity	0.95 (Skin surface), 0.9 (Wall surface)
Scalar Boundaries	
DV Supply Concentration	Computed based on recirculation
PV Supply Concentration	420 ppm
Equivalent Occupant CO ₂ concentration associated with exhalation	40000 ppm [34]
Wall surface	Zero gradient

* A_{DV} , A_{PV} , and A_{NC} denote inlet areas of displacement ventilation, personalized ventilation, and nasal cavity, respectively.

4.2.4 Taguchi DOE and optimization cases

In this study, we simultaneously optimized the thermal comfort, IAQ, and AHU/OHU energy requirements based on PV and DV input variables. Here, the required energy input is calculated from the temperature difference and air flow rate, ignoring fan power and heat loss in the ducts. We used the Taguchi design to minimize the number of simulations. Using equal weights or importance (1/3) for each category mentioned above, we adopted the TOPSIS as the MCDM method. In this procedure, the thermal comfort, IAQ, and AHU/OHU energy requirements were combined into one objective-function value that describes its closeness to the ideal value. Using these objective-function values in each simulation, we performed an S/N ratio analysis of the Taguchi method [45] to determine the input variables that would yield the optimum value. This method was similarly and pioneering used by Yang et al. [46] to optimize data centers, except that they used other methods to determine the weights of variables that they wished to combine.

We varied the typical input variables used in previous studies, namely the supply temperature and flow rate. Specifically, we varied the PV supply flow rate from 0 to only 6 L/s at increments of 1.5 L/s, whereas we varied the PV supply temperature from 21 °C to 25 °C at increments of 1 °C to avoid thermal discomfort as the PV systems were located near the occupants [25]. Furthermore, we varied the DV supply temperature and flow rate from 16 °C to 20 °C (at increments of 1 °C) and from 30 to 110 L/s (at increments of 20 L/s), respectively. Finally, we allowed the outlet flow of DV to be recirculated (R) at 0.20, 0.35, 0.50, 0.65, and 0.80. We used five parameters with five levels each, for which we adapted the L_{25} Taguchi design. Table 4.3 presents the corresponding 25 simulations used for the multicriteria optimization.

Table 4.3. Taguchi design

Simulation	DV Flow Rate (L/s)	DV Supply Temperature (°C)	PV 1 Flow Rate (L/s)	PV 2 Flow Rate (L/s)	PV Supply Temperature (°C)	DV Recirculation Rate
1	30	16	0	0	21	0.2
2	30	17	1.5	1.5	22	0.35
3	30	18	3	3	23	0.5
4	30	19	4.5	4.5	24	0.65
5	30	20	6	6	25	0.8

Chapter 5: Comparison of Local Indoor Air Quality and Thermal Comfort Under Hybrid Mixing-Personalized and Hybrid Displacement-Personalized Ventilation in an Office

6	50	16	1.5	3	24	0.8
7	50	17	3	4.5	25	0.2
8	50	18	4.5	6	21	0.35
9	50	19	6	0	22	0.5
10	50	20	0	1.5	23	0.65
11	70	16	3	6	22	0.65
12	70	17	4.5	0	23	0.8
13	70	18	6	1.5	24	0.2
14	70	19	0	3	25	0.35
15	70	20	1.5	4.5	21	0.5
16	90	16	4.5	1.5	25	0.5
17	90	17	6	3	21	0.65
18	90	18	0	4.5	22	0.8
19	90	19	1.5	6	23	0.2
20	90	20	3	0	24	0.35
21	110	16	6	4.5	23	0.35
22	110	17	0	6	24	0.5
23	110	18	1.5	0	25	0.65
24	110	19	3	1.5	21	0.8
25	110	20	4.5	3	22	0.2

Table 4.4 Neutral Skin Temperatures [37]

Locations	Temperature (° C)
Mean Skin	33.6 ± 0.5
Head	34.3 ± 0.5
Face	32.8 ± 0.8
Thorax	34.8 ± 0.7
Abdomen	35.0 ± 0.9
Upper arm	32.8 ± 1.1
Lower arm	33.3 ± 1.1
Hand	33.5 ± 1.5
Upper leg	32.4 ± 1.1
Lower leg	32.1 ± 1.0
Foot	33.8 ± 1.6

We examined four cases based on the type of output parameter used to represent thermal comfort and IAQ (Figure 4.19). Case 1 focused on the average skin temperature of each

occupant as a thermal-comfort parameter, and the occupant's inhaled CO₂ concentration was regarded as the IAQ parameter. To observe the effect of skin-temperature segmentation, we changed the thermal-comfort parameter from skin temperature to the skin temperature of individual body parts (segmented temperature), which we labeled Case 2. We used the study by Lai et al. [37], which was relatively similar to the study by Olsen [38], as reference for the average neutral skin temperature and segmented neutral skin temperature, which comprised 10 body parts. The reference values are listed in Table 4.4. In Case 3, the occupant-zone volume-averaged (OZVA) temperature was used as the thermal comfort parameter, whereas the OZVA CO₂ concentration was used as the IAQ parameter. In Case 4, the room volume-averaged (RVA) temperature and CO₂ concentration were used as the thermal comfort and IAQ parameters, respectively. For Cases 3 and 4, we used 23 °C as the reference acceptable indoor temperature, which is the intermediate value in the range recommended by the ASHRAE [47]. A limit of 1000 ppm was set for all IAQ parameters [48]. These four cases provided a clear comparison, as we rendered the thermal comfort and IAQ more occupant centric. Finally, the PV and DV energies were computed using Eqs. (4.3) and (4.4), respectively.

$$E_{PV} = \frac{\rho \dot{V}_{PVs} \times (h_{PVs} - h_o)}{1000} \quad (4.3)$$

$$E_{DV} = \frac{\rho \dot{V}_{DVs} \times (h_{DVs} - h_m)}{1000} \quad (4.4)$$

$$h_m = R \times h_{ro} + (1 - R) \times h_o, \quad (4.5)$$

where $\dot{V}_{PVs}$ and $\dot{V}_{DVs}$ are the PV and DV volume flow rates (units: L/s), respectively; ρ is the corresponding air density (units: kg/m³); h_{PVs} and h_{DVs} are the enthalpies of the supply air (units: kJ/kg) from the PV and DV handling units, respectively; h_o is the enthalpy (units: kJ/kg) of the outdoor air; and h_m is the enthalpy of air (units: kJ/kg) entering the AHU of the DV, which is a mixture of outdoor and recirculated air. Here, h_m is computed using Eq. (4.5), where h_{ro} is the room outlet enthalpy (units: kJ/kg). As we did not consider humidity, we computed the enthalpies based on a constant humidity ratio of 0.011 kg of vapor per kilogram of dry air.

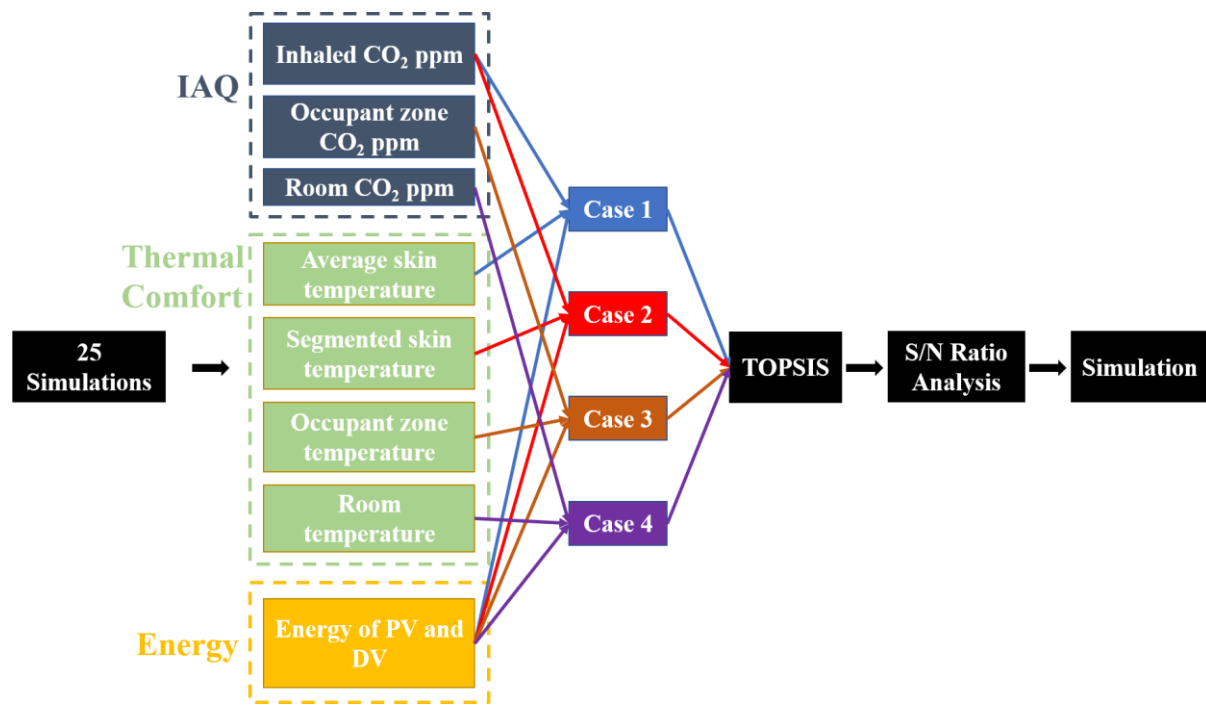


Figure 4.19. Cases and optimization process

4.2.5 Multicriteria optimization: TOPSIS and S/N ratio

After performing 25 steady-state simulations based on the Taguchi DOE and obtaining the corresponding output parameters for the four cases, we post-processed the data such that the ideal values was the lowest or approximately zero. We computed the absolute value of the difference between the average skin temperature (Case 1) and the segmented skin temperature (Case 2) with their corresponding average neutral skin temperatures (Table 4.4). Thus, the best solution was the closest to 0 and the worst solution indicated the highest value. In Cases 3 and 4, their values were subtracted from the reference temperature of 23 °C. For CO₂ concentrations, the best solution was consistently the closest to 0. In terms of energy, the PV consistently indicated positive energy, as OHU underwent heating at all times. However, the DV can undergo cooling or heating depending on the simulation case; therefore, we obtained the absolute value. The PV and DV energies demonstrated better solutions as they approached 0. Subsequently, the post-processed data are summarized as a decision matrix D.

We solved for the TOPSIS value and the objective-function closeness coefficient using Eqs. (4.6)–(4.10). In the TOPSIS, a value of approximately 1 implies an almost ideal solution, whereas a value of approximately 0 implies otherwise [49]. In this process, we first normalized D using Eq. (4.6). Specifically, we obtained the square root of the sum of the squares (y_{ij}) of each post-processed output parameter (x_{ij}). Subsequently, we used these values to normalize their corresponding parameters in D.

After normalizing D , we multiplied each normalized value per parameter by its weight (Eq. (4.7)). We used equal weights of $1/3$ for the three categories: thermal comfort, IAQ, and energy input. If several parameters are available for a category, e.g., the thermal comfort of Case 1 comprises two parameters (i.e., the average skin temperatures of Occupants 1 and 2), then we divide the weight of $1/3$ by 2. Therefore, the average skin temperatures of Occupants 1 and 2 have a weight of $1/6$. Subsequently, we can obtain a weighted normalized matrix.

$$y_{ij} = \frac{x_{ij}}{\sqrt{\sum_{j=1}^n x_{ij}^2}} \quad (4.6)$$

$$w_{ij} = y_{ij} \times W_j \quad (4.7)$$

For each column of the weighted normalized matrix, we determined the best (M^+) and worst (M^-) values for each parameter of the weighted normalized matrix. For post-processing, we set the parameters such that they would indicate the best value if their value was the minimum or closest to zero; therefore, M^+ and M^- were specified as the minimum and maximum values per parameter, respectively.

Subsequently, we compute the distance between each simulation based on the best and worst values using Eqs. (4.8) and (4.9), respectively. For every simulation number, we obtained the square root of the sum of squares of the difference between each of the parameters with their corresponding maximum (and also for the minimum using Eq. (4.9)). Finally, the closeness coefficient or objective function was determined using Eq. (4.10).

$$C_j^+ = \sqrt{\sum_{i=1}^n (w_{ij} - M_i^+)^2} \quad (4.8)$$

$$C_j^- = \sqrt{\sum_{i=1}^n (w_{ij} - M_i^-)^2} \quad (4.9)$$

$$C = \frac{C_j^-}{C_j^+ + C_j^-} \quad (4.10)$$

Once the closeness coefficients of the objective function values were solved for each of the 25 simulations, “the larger the better” S/N ratio was computed using Eq. (4.11). Here, y_i denotes the objective-function value per simulation and n the number of simulations. Subsequently, based on the parameter per level (e.g., parameter 1 is the DV flow rate, and level 1 of that parameter is 30 L/s), we calculated the average S/N ratio. Finally, we determined the optimum input parameters by identifying the levels per parameter that yielded the highest

average S/N value, which were then used as input parameters for new simulations to simultaneously obtain the optimum thermal comfort, IAQ, and energy per case. We provide a summary of the optimization process in Figure 4.19. As an example, we included the resulting table per process for the optimization of Case 4 in Appendix A (Tables 4A1–4A6).

$$S/N = -10 \times \log \left[\frac{1}{n} \left(\sum_{i=1}^n \frac{1}{y_i^2} \right) \right] \quad (4.11)$$

4.3. Result and Discussions

Figure 4.20, Figure 4.22 and Figure 4.23 the thermal comfort, IAQ, and energy results, respectively, including the different measures for each case. We did not include the segmented skin temperatures because this would result in 20 additional column plots per point.

4.3.1 Individual measures

First, the flow rates of the Taguchi table were initially set such that every simulation was estimated to have RVA values lower than 1000 ppm based on the ASHRAE standards. As such, the RVA values were lower 1000 ppm (Figure 4.20). For every simulation, the RVA and OZVA CO₂ concentrations were similar to each other owing to the small volume difference between the two zones. Furthermore, the OZVA CO₂ concentrations were slightly lower than the counterpart RVA values, which is attributable to the volume of CO₂ being pushed into the exhaust system outside the occupant zone. We computed the average CO₂ concentration at the outlet and discovered that all the values were higher than the RVA CO₂ concentration. For example, in simulation 1, the RVA value was 715 ppm, the OZVA value was 681 ppm, and the average CO₂ concentration at the outlet was 803 ppm.

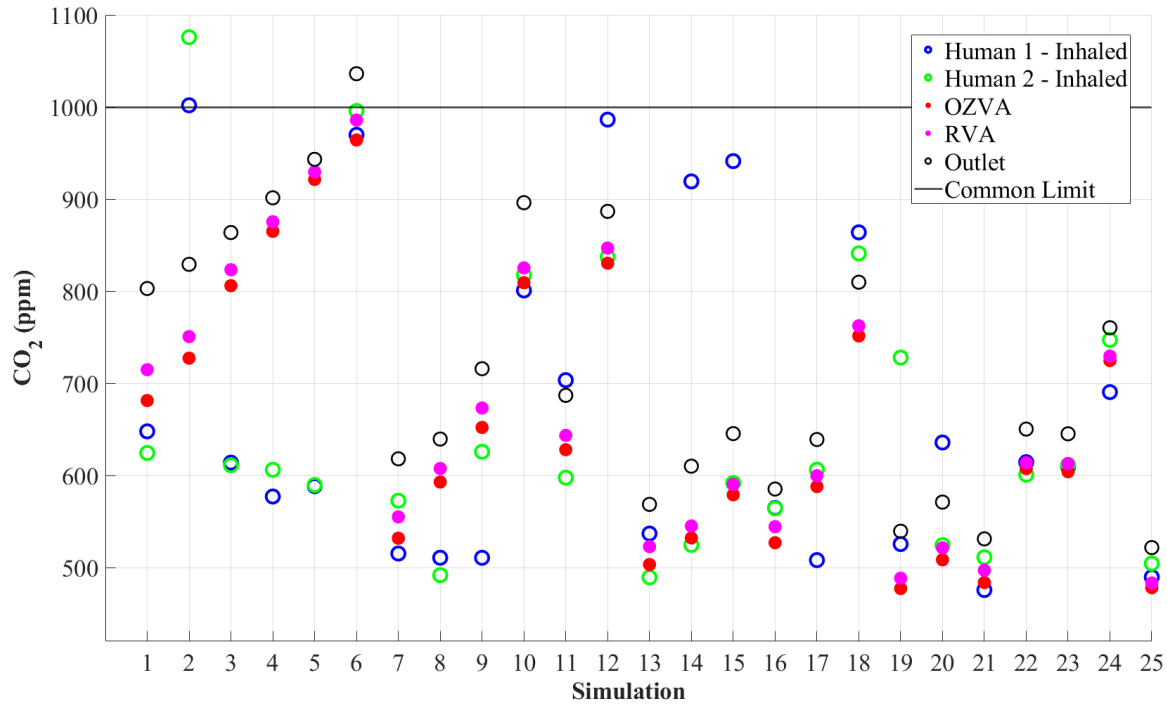


Figure 4.20. CO₂ concentration values from 25 simulations

In general, the two average values depend on the total fresh-flow-rate intake (FA), which can be estimated using Eq. (4.12). Simulation 25, which indicated the highest total outdoor air intake, showed the lowest RVA CO₂ concentration (i.e., 483 ppm). By contrast, Simulation 6, which indicated the lowest total outdoor air intake, showed the highest RVA (i.e., 986 ppm). These two examples show the RVA trend, where a high FA yielded a low RVA value, and vice versa. Furthermore, based on a comparison between Simulation 21 ($FA = 82$ L/s) and Simulation 19 ($FA = 79.5$ L/s), although the FA of the former was greater than that of the latter, the CO₂ concentration of Simulation 19 was lower (i.e., 489 ppm) compared with that of Simulation 21 (i.e., 497 ppm). This is because of the imperfect mixing condition, where Simulation 19 can exhaust more CO₂ with an average CO₂ concentration of 540 ppm, as compared with only 531 ppm afforded by Simulation 21. The same trend was observed for the OZVA CO₂ concentration, where Simulation 19 indicated a lower value (477 ppm) than Simulation 21 (481 ppm). Additionally, between Simulations 19 and 25, even though the RVA value of the first case was greater than that of the second case, the OZVA value of the first case was less than that of the second case. This phenomenon can be attributed to imperfect mixing conditions.

$$FA = \dot{V}_{DV}(1 - R) + \dot{V}_{PVs1} + \dot{V}_{PVs2} \quad (4.12)$$

As a more occupant-centric IAQ metric, the inhaled CO₂ concentration exhibited a behavior that differed significantly from those of the OZVA and RVA values. As an example, Simulation 2 indicated an inhaled CO₂ concentration greater than 1000 ppm, although it did not show the highest OVA and RVA values. Additionally, its RVA value was lower than those

of Simulations 3–6, 10, 12, and 18. This can be attributed to the local flow field and concentration distribution near the occupants (Figure 4.21). Here, the PV jet cannot reach the occupants' nostrils, which consequently creates a vortex near the occupant, thus increasing the local concentration and inhaled CO_2 concentration. The lowest average inhaled CO_2 concentration was observed in Case 21, which can be attributed to the extremely high FA and the high flow rate of the PV supplying fresh air directly to the nostrils of the occupants.

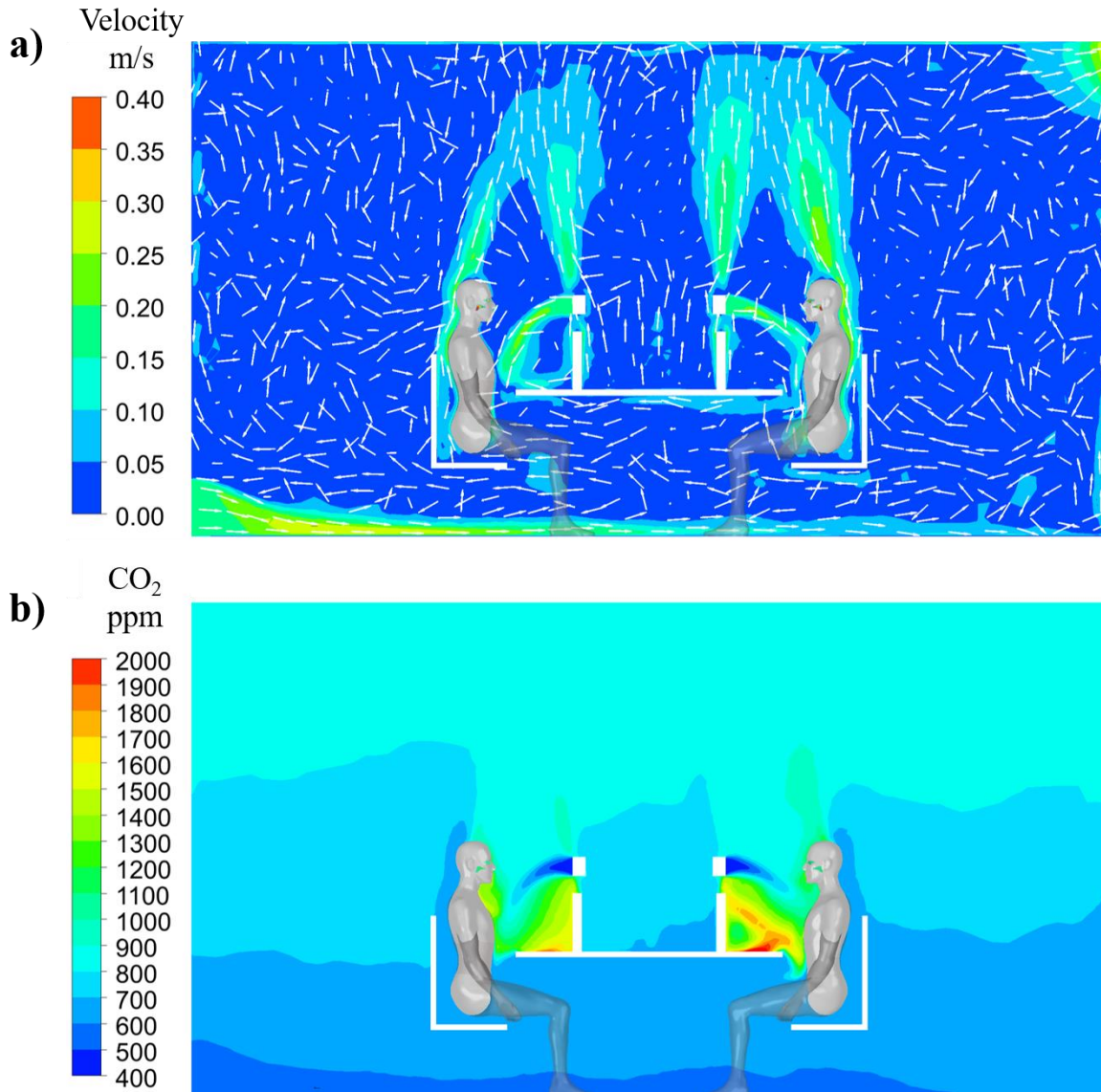


Figure 4.21. a) Velocity field and b) concentration distribution of Simulation 2

In general, the RVA and OZVA values depend on FA . The higher the FA , the lower were the RVA and OZVA values. In addition, we observed several degrees of nonlinear relationships when comparing different IAQ metrics such as RVA to RVA values, RVA to OZVA values, and inhaled CO_2 concentration in terms of both RVA and OZVA values. This can be attributed to the imperfect mixing condition, and the degree of nonlinearity was higher for the more

occupant-centric metric because of its dependence on the local flow field and concentration distribution.

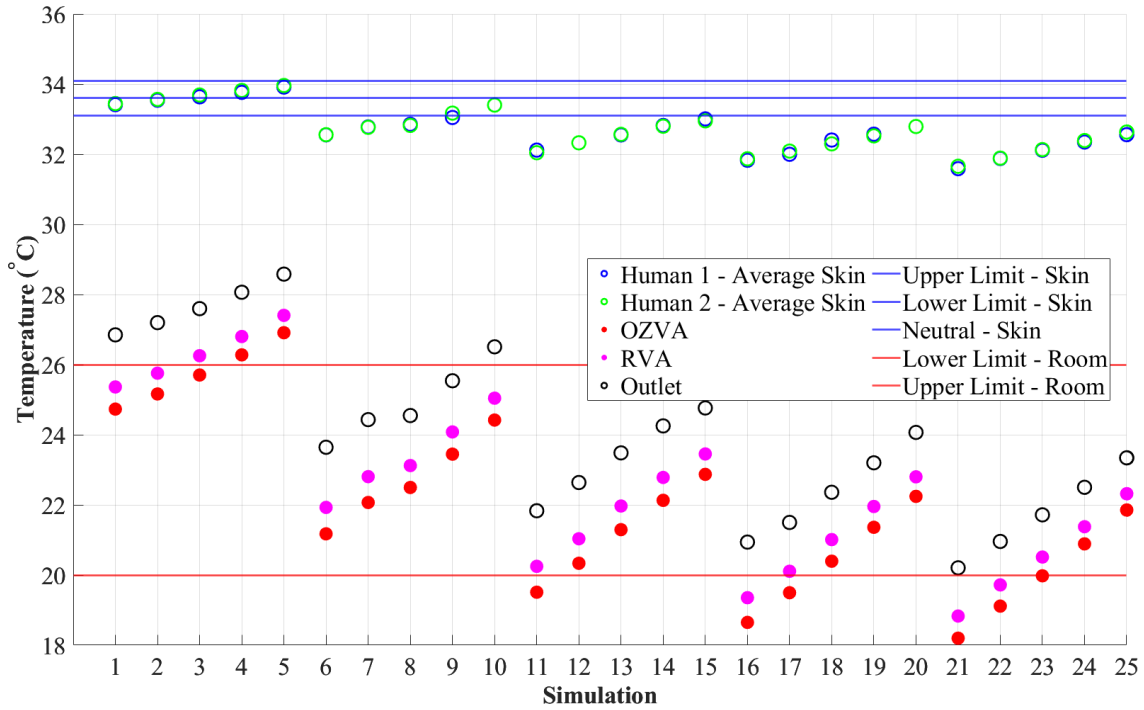


Figure 4.22. Temperature values from 25 simulations

In terms of thermal comfort, the RVA and OZVA temperature ranges were 18.8 °C–27.4 °C and 18.2 °C–26.9 °C (Figure 4.22). All the RVA temperatures showed higher values than their OZVA counterparts. This is attributable to the high-temperature flow being pushed to the exhaust outlet, which was outside the occupant zone, thereby reducing the OZVA temperature. For example, Simulation 1 indicated the following: an RVA temperature of 25.4 °C, an OZVA temperature of 24.7 °C, and a mass-weighted average temperature at the exhaust of 26.8 °C. Additionally, we observed some simulations that indicated the same RVA temperatures but different OZVA temperatures. For example, both Simulations 7 and 20 indicated an RVA temperature of 22.8 °C, whereas their OZVA temperatures were 22.1 °C and 22.2 °C, respectively. This can be attributed to the higher temperature at the exhaust of Simulation 7 with a mass-weighted average temperature of 24.4 °C, compared with 24.1 °C of Simulation 20.

In terms of the area-weighted average skin temperature, Simulations 1–5 and 10 indicated a neutral temperature of 33.6 ± 0.5 °C. This was observed in the RVA and OZVA temperature ranges of 25 °C–27.4 °C and 24.4 °C–26.9 °C, respectively. Additionally, we verified the segmented temperature. First, for each simulation, we obtained the absolute value of the difference between each segment value and the segment neutral temperature, and then obtained

the average value. However, we obtained the lowest average difference of 1 °C in Simulation 2, which indicates the difficulty in simultaneously satisfying all the segmented neutral skin temperatures.

At the same flow rate, e.g., Simulations 1–5 (30 L/s) or 6–10 (50 L/s), we observed an increase in the OZVA and RVA values. This is because, at each of the abovementioned consecutive five simulations, although they had the same flow rates (as defined in Table 4.3), the inlet temperature increased from 16 °C to 20 °C. In addition, we observed that the average skin temperatures of the occupants increased, albeit at a lower rate. This demonstrates the prominent effect of the DV inlet temperature on the average human skin temperature. Furthermore, as we increased the flow rate, we observed an abrupt decrease in the RVA and OZVA temperatures. This was similarly observed for the average human skin temperature, which indicates the prominent effect of the DV flow rate.

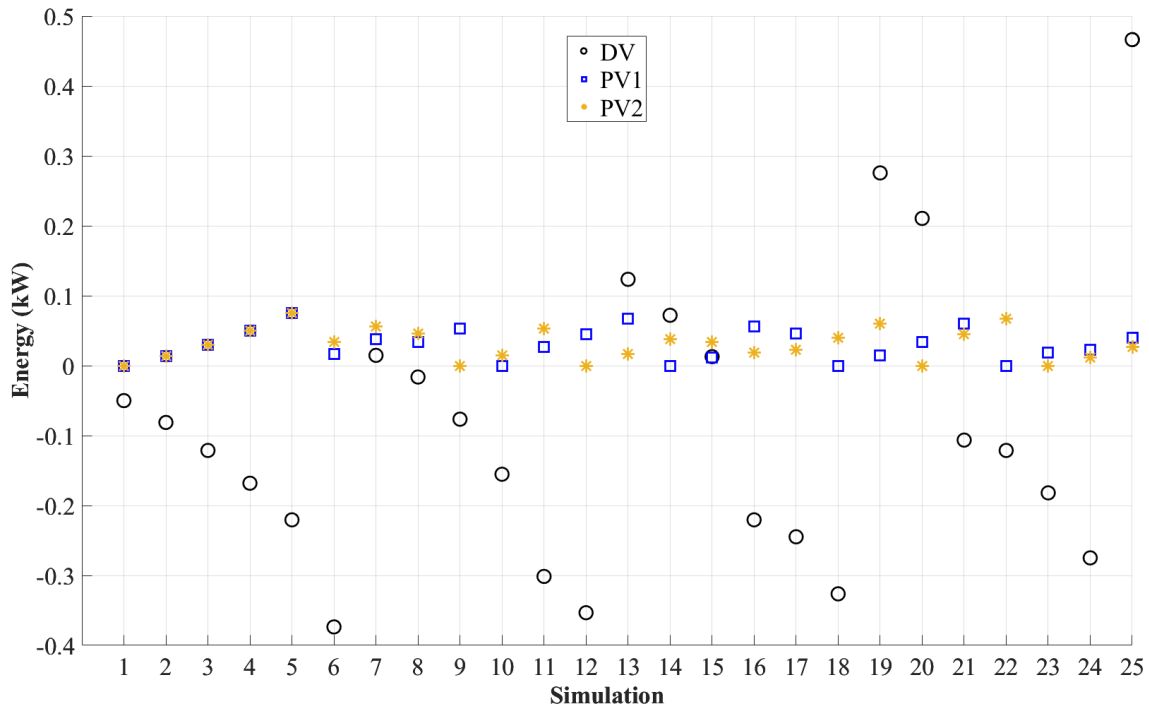


Figure 4.23. DV and PV energy inputs for 25 simulations

In terms of the DV energy requirement, Simulation 15 indicated the lowest energy input of only 0.01 kW for heating (Figure 4.23). Moreover, as the DV underwent recirculation, the AHU functioned as a cooling or heating unit, thereby changing the sign of the energy input. Meanwhile, the lowest average energy requirement for the PV was indicated in Simulation 1, where both PV systems were closed.

We observed that the best simulation settings may differ depending on the output parameters used in the metrics of thermal comfort and IAQ. For example, for the thermal-comfort metric, the ASHRAE standard defined the room temperature to be 20 °C–26 °C. However, for more occupant-centric metrics such as the average skin temperature, this range may still extend to 27.4 °C and 26.9 °C for the RVA and OZVA values, respectively. In addition, to ensure the desired IAQ, having an acceptable RVA or OZVA CO₂ concentration does not guarantee an acceptable inhaled CO₂ concentration owing to imperfect mixing conditions. Thus, using an occupant-centric system may result in an RVA value that deviates slightly from published standards in terms of thermal comfort and IAQ.

4.3.2 Multi-objective measures

4.3.2.1 TOPSIS results

Table 4.4 lists the objective-function value or closeness coefficient for each simulation for the different TOPSIS cases defined in Section 4.2.5. We ranked each case from the highest (highlighted in red) to the lowest. Additionally, we added a new case (Case 5) distinct from the four initial cases as can be seen in Table 4.4, Table 5.5, and Figure 4.24. This case will be discussed particularly in section 4.4.

For Cases 1 and 2, which differed by only the thermal-comfort parameters (i.e., the average skin temperature and segmented skin temperature), both indicated Simulation 1 as the closest to the ideal solution. This can be attributed to the zero-energy input for both PV systems and the relatively low energy input for the DV. In addition, although both cases did not exhibit the lowest inhaled CO₂ concentration, they indicated acceptable values that were below the mean values of the 25 simulations. In terms of thermal comfort, Simulation 1 in TOPSIS Case 1 yielded an average skin temperature within the range of neutral skin temperature of 33.6 ± 0.5 °C (Figure 4.22). Meanwhile, for Case 2 with 20 thermal comfort parameters (10 segmented temperatures per occupant), the average deviation of Simulation 1 from the neutral segmented temperature was 1.1 °C, which is extremely close to the lowest value of 1 °C. Thus, Simulation 1 had a high objective-function value for TOPSIS Case 2.

Meanwhile, for TOPSIS Cases 3 and 4, where the thermal comfort and IAQ parameters were the OZVA and RVA temperatures and CO₂ values, respectively, Simulations 14 and 15 yielded the highest objective-function values. First, these two simulations had relatively low OZVA and RVA CO₂ concentrations compared with the other simulation cases. Furthermore, their RVA and OZVA temperatures were extremely close to 23 °C, which is the ideal temperature set for Cases 3 and 4. Additionally, their energy requirements were lower than

those of the other simulations (Figure 4.23). Thus, both yielded higher objective-function values in Cases 3 and 4.

Table 4.4. Objective-function values of each simulation for different cases using TOPSIS

Simulation	Case 1		Case 2		Case 3		Case 4		Case 5	
	Value	Rank	Value	Rank	Value	Rank	Value	Rank	Value	Rank
1	0.99	1	0.97	1	0.75	9	0.88	6	0.99	1
2	0.89	5	0.79	10	0.60	15	0.77	10	0.96	2
3	0.91	3	0.80	9	0.38	20	0.54	15	0.95	5
4	0.76	10	0.54	18	0.17	24	0.28	21	0.87	10
5	0.53	15	0.28	25	0.06	25	0.10	25	0.73	12
6	0.39	20	0.37	23	0.63	14	0.52	16	0.17	23
7	0.74	11	0.76	12	0.91	5	0.87	7	0.93	7
8	0.80	8	0.82	8	0.93	4	0.92	5	0.94	6
9	0.90	4	0.89	3	0.89	6	0.94	3	0.95	4
10	0.92	2	0.86	6	0.71	11	0.84	8	0.89	9
11	0.28	25	0.43	22	0.36	21	0.27	22	0.20	22
12	0.37	22	0.44	21	0.52	17	0.41	18	0.15	24
13	0.64	12	0.72	13	0.85	7	0.77	9	0.78	11
14	0.84	7	0.89	2	0.98	1	0.96	2	0.91	8
15	0.87	6	0.88	4	0.97	2	0.98	1	0.96	3
16	0.33	23	0.60	17	0.29	22	0.25	23	0.35	18
17	0.38	21	0.61	16	0.41	19	0.34	20	0.34	20
18	0.43	18	0.53	19	0.58	16	0.49	17	0.21	21
19	0.48	16	0.51	20	0.79	8	0.72	11	0.39	17
20	0.78	9	0.82	7	0.95	3	0.92	4	0.66	13
21	0.33	24	0.61	15	0.27	23	0.25	24	0.55	15
22	0.45	17	0.76	11	0.42	18	0.36	19	0.59	14
23	0.61	13	0.88	5	0.66	13	0.59	14	0.53	16
24	0.54	14	0.68	14	0.72	10	0.65	13	0.34	19
25	0.39	19	0.36	24	0.71	12	0.67	12	0.12	25

4.3.2.2 Optimized results

Based on the resulting objective function from each simulation, we performed the Taguchi S/N ratio analysis to determine the input parameters for each optimization case, as presented

in Table 5.5. Figure 4.24 shows the results obtained after performing the simulations using the determined input parameters.

Table 5.5. Input parameters determined via S/N ratio analysis for each optimization case

Case	DV Flow Rate (L/s)	DV Supply Temperature (°C)	PV 1 Flow Rate (L/s)	PV 2 Flow Rate (L/s)	PV Supply Temperature (°C)	DV Recirculation Rate
1	30	19	0	0	21	0.35
2	50	18	0	0	21	0.35
3	50	18	1.5	0	21	0.2
4	50	19	1.5	0	21	0.2
5	30	19	6	4.5	25	0.35

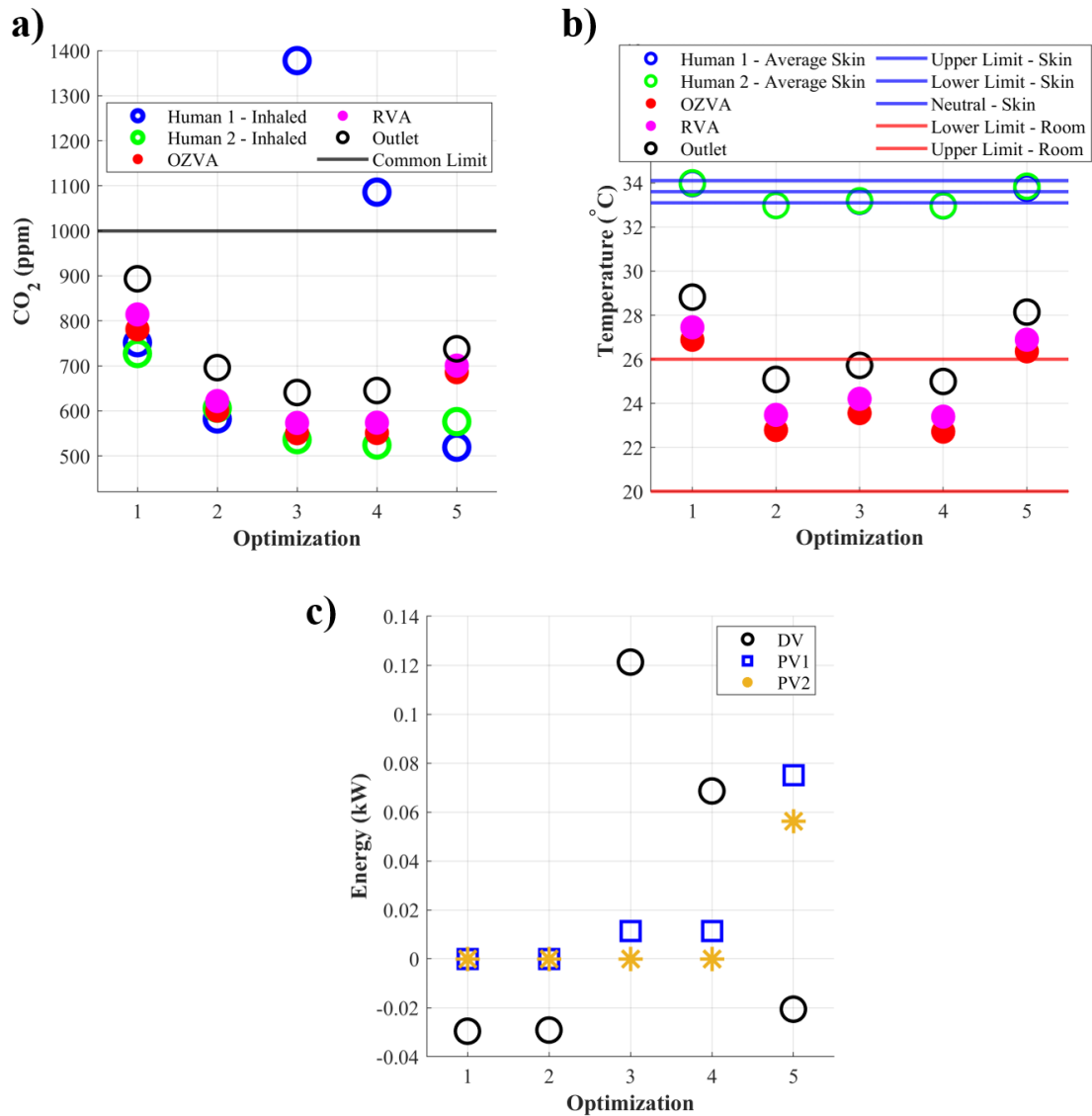


Figure 4.24. Optimization cases values: a) CO₂ concentration; b) temperature; c) energy

The objective of multicriteria optimization is to determine a combination of output parameters that are close to the ideal values of the three metrics and do not necessarily exhibit the best values for each simultaneously. For example, in Case 1, the average skin temperatures for Occupants 1 and 2 is 34 °C. Although not the best result, because the closest is the Simulation 2 of the Taguchi method, the value remains within the range of neutral skin temperature (Figure 4.24b). Meanwhile, the optimization case showed above-average inhaled CO₂ concentrations of 752 and 728 ppm for Occupants 1 and 2, respectively (Figure 4.24a)), which is within the acceptable range. In terms of the energy input, the DV energy input was one of the lowest among the 25 simulations, whereas the PV energy input was the lowest (both 0 kW) because the PV systems were turned off. Comparing the best Taguchi result of Case 1 (Simulation 1) with the optimized case (Figure 4.24c), the former indicated slightly better IAQ

and thermal comfort. However, the optimized case indicated a lower DV energy input by 40% (from -0.05 to -0.03 kW), whereas both had the same PV energy input.

For the optimization of Case 2 (segmented skin temperatures), the average deviation from the neutral segmented temperature was 1.6 °C, which was slightly higher than the Taguchi best for Case 2 (Simulation 1), with a value of 1.1 °C. However, its inhaled CO₂ concentration was lower (i.e., 582 and 606 ppm, for Occupants 1 and 2, respectively) compared with the Taguchi best for Case 2 (648 and 624 ppm for Occupants 1 and 2, respectively). Furthermore, its DV energy input was 40% lower than that of the Taguchi best for Case 2, although both had the same PV energy input.

For the optimization of Case 3, the deviation of the OZVA temperature from the set temperature of 23 °C was only 0.6 °C, which was better than that of the Taguchi best for Case 3 (Simulation 14), i.e., a deviation of 0.8 °C. In terms of the OZVA CO₂ concentration, the optimized case indicated a slightly higher value of 551 ppm compared with that of the Taguchi best (532 ppm). By contrast, the optimized DV energy input increased by approximately 70%, whereas the average PV energy input decreased by 70%.

For the optimization of Case 4, the deviation of the RVA temperature from the set temperature of 23 °C was 0.6 °C, which was slightly higher than that of the Taguchi best for Case 4 (Simulation 15), i.e., a deviation of 0.5 °C. In terms of the RVA CO₂ concentration, the optimized result was lower, with a value of 573 ppm compared with 590 ppm. However, the optimized result of the DV input increased by five times, whereas the average PV input decreased by four times.

In these four optimization cases, we observed that compared with each corresponding best simulation inside the Taguchi table (Table 4.4), some of the categories, i.e., thermal comfort, IAQ, and energy input, may worsen slightly, while another category may improve simultaneously. Owing to this compensation, the closeness coefficient remains higher than that of the best case in the Taguchi table. In particular, even though the closeness coefficients of the best cases in the Taguchi table were between 0.97 and 0.99, the resulting optimum cases yielded higher values, albeit by small percentages.

4.3.2.3 Comparison of Optimized Cases

Cases 3 and 4 have almost identical input parameters. The only difference is the DV supply temperature, which resulted in significant changes to the output parameters. For the IAQ, all parameters remained the same, except for the inhaled CO₂ concentration in Occupant 1. This

can be attributed to the resulting concentration distribution and flow field near the nostrils of Occupant 1, which changed the local flow field near the nostrils of Occupant 1, similar to Figure 4.21, except that the high inhaled CO₂ concentration was only limited to Occupant 1. Additionally, Case 3 indicated higher OZVA and RVA temperatures because its supply temperature was greater than that of Case 4. The skin temperature increased as well, which is the same trend as that described in the previous sections. Furthermore, the DV energy input of Case 3 was lower than that of Case 4.

Generally, as the optimization objectives for Cases 3 and 4 are based on OZVA and RVA metrics, these two cases cannot satisfy the more occupant-centric parameters, particularly the inhaled-breathing-zone CO₂ concentration (Figure 4.24a). Furthermore, they have higher energy requirements, as shown in Figure 4.24c. This can be attributed to the fact that these two optimization cases attempt to optimize a larger indoor space, as compared with optimization of Cases 1 and 2. However, Case 3, which focuses on OZVA values, generally indicated output parameters that were more similar to those of Cases 1 and 2 owing to the smaller space of interest. In particular, the optimized result for Case 3 yielded a skin temperature that was close to the range of the average neutral skin temperature, as shown in Figure 4.24b. This was similarly observed when we compared the skin-temperature distributions of Cases 1, 3, and 4, as shown in Figure 4.25. Here, the temperature distribution of Case 3 was closer to the neutral temperature range of 33.6 ± 0.5 °C of the human skin, as represented by more yellow to orange regions (33.5 °C–34.5 °C) on the arms and torso while green (32.5 °C–33.0 °C) on the head and lower legs, compared with Case 4, which showed more yellow green to yellow regions (33 °C–34 °C) on the arms and torso while blue green to green on the head and lower legs (32.0 °C–33.0 °C). This made the average skin temperature of the latter case to be lower from the neutral range.

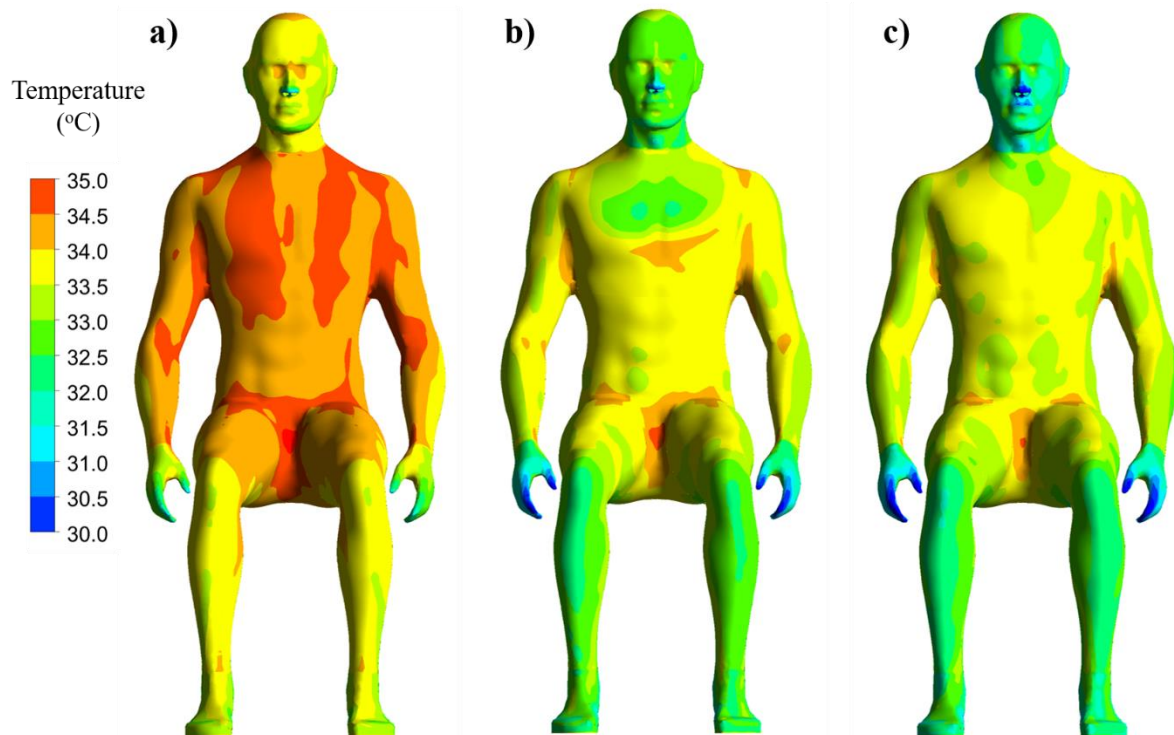


Figure 4.25. Skin-temperature distributions from optimization. a) Case 1, b) Case 3, and c) Case 4

In terms of the more occupant-centric measures, the optimizations of Cases 1 and 2 differed only by their DV input temperature and flow rate (Table 5.5). Furthermore, they both featured closed PVs that were solely dependent on the DV. A higher inhaled CO_2 concentration was indicated in Case 1; however, it remained within the range (Figure 4.24a). In terms of the average deviation from neutral temperatures, the computed value for Case 1 was 1.1 °C, as compared with 1.6 °C for Case 2, although both had similar DV energy inputs. However, we considered the uncertainties that may arise in Case 2 (the segmented case). This is because it comprised 20 thermal-comfort parameters, and the weight of each of the 20 parameters was 1/60. This weight is extremely low and may significantly affect the optimized case, as the 20 parameters for the thermal-comfort category were to be satisfied. Therefore, this should be tested under different scenarios. Finally, the optimization of Case 1 can be regarded as the current best optimization procedure for discussing occupant-centric metrics.

4.4. Limitations of PV in Current Study

Based on our results, particularly for Cases 1 and 2, both the optimized cases and Taguchi bests did not involve PV. In addition, although the optimized results of Cases 3 and 4 involved opening one of their PV systems at 1.5 L/s, it was only performed to decrease the OZVA and RVA CO_2 concentration and was not necessarily intended for PV (Figure 4.26a), which

provides fresh air directly to the occupant's breathing zone. This was manifested by the high inhaled CO₂ concentration in the occupants when the PV was turned on. This general phenomenon can be attributed to the range of PV flow rates used in the current study, which was 6 L/s at the maximum. We set such limits based on the ASHRAE standard to prevent thermal discomfort, particularly for PV systems that are generally near the occupant and operating at elevated air speeds. However, extremely low air speeds caused the PV systems, particularly at low velocities, to not function as intended. For example, in Simulation 2 (Figure 4.21) and optimized Case 3 (Figure 4.26a), the PV jets could not reach the nostrils of the occupants. Meanwhile, although demonstrating a higher flow rate (4.5 L/s), Simulation 12 (Figure 4.26c) showed an upward skew for its Occupant 1's PV jet, which created a suction effect on the exhaled CO₂ concentration to be re-inhaled. This can be attributed to the higher velocity originating from the DV at a flow rate of 70 L/s. By contrast, we demonstrated a PV system with 4.5 L/s that functioned as intended, as shown in Simulation 4 in Figure 4.26b. In addition to the DV flow rate, the effects of the thermal plume could not be disregarded. Figure 4.27 shows the velocity-field and temperature distributions for Simulation 1. The figure shows velocity vectors pointing upward from the two occupants and two monitors, which were the heat sources. These thermal plumes featured induced velocities of up to 0.3 m/s, which affected the PV jets.

Meanwhile, we observed cases in which the PV systems functioned as intended. This occurred when the setting was at its maximum (6 L/s). As indicated in almost all cases with the PV operating at 6 L/s, the inhaled CO₂ was generally low (primarily less than 600 ppm). Thus, in the current PV system, the setting in which the PV functions as intended is restricted, regardless of the settings of the other parameters.

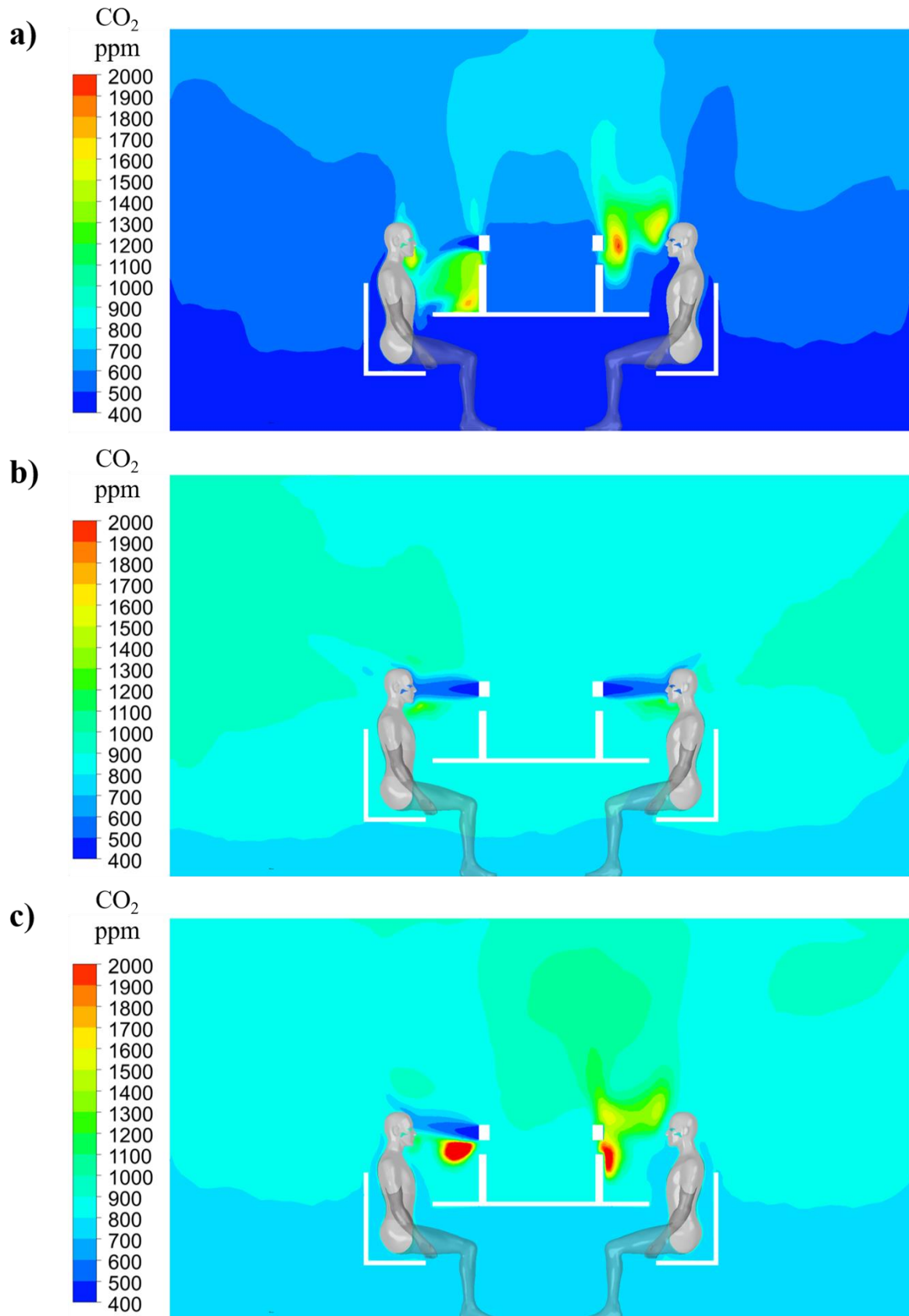


Figure 4.26. Concentration distributions of a) optimum Case 3, b) Simulation 4, and c) Simulation 12

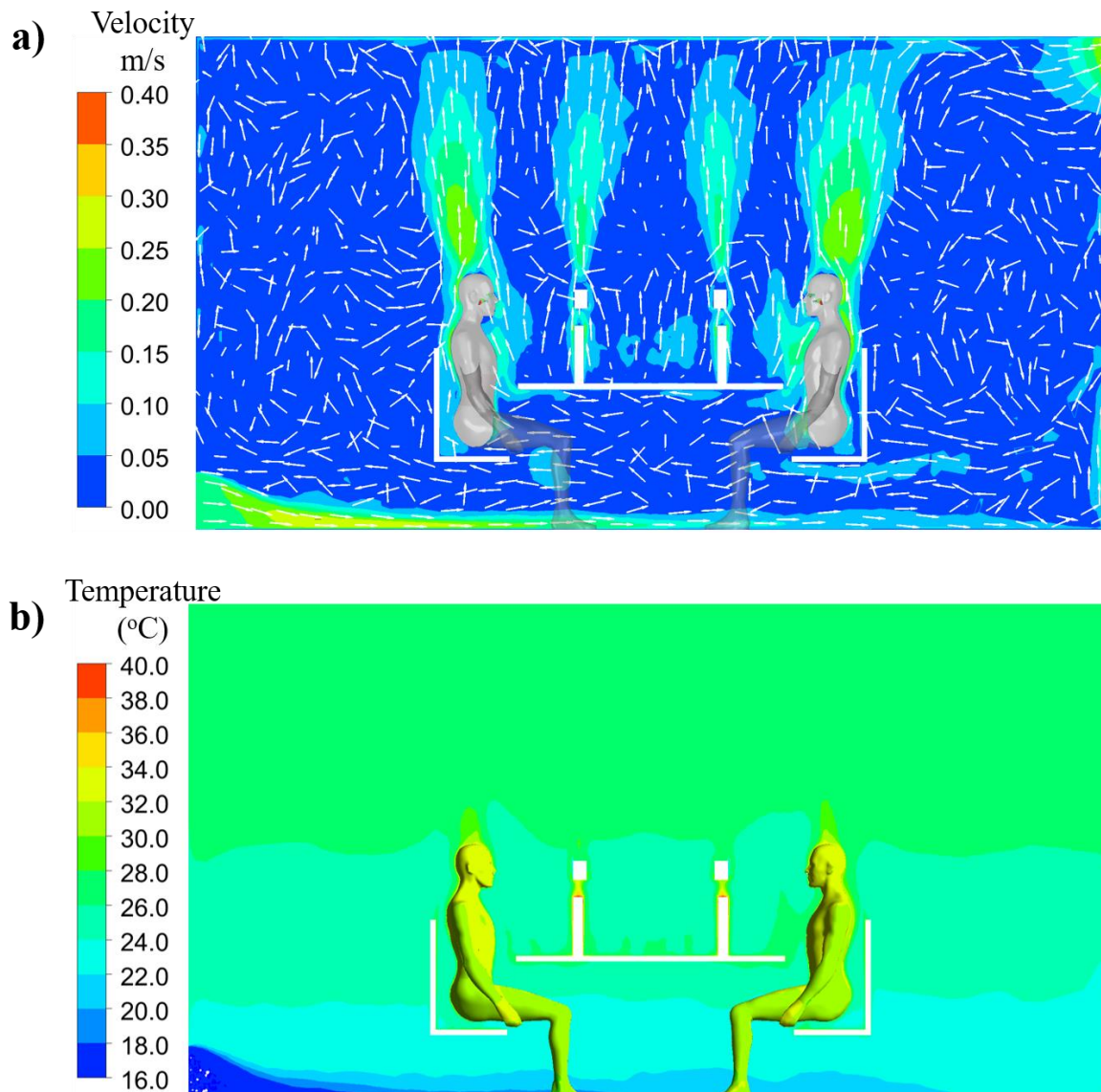


Figure 4.27. a) Velocity field and b) temperature distribution of Simulation 1

We attempted to determine a scenario in which the optimum case would involve the PV systems functioning as intended. Hence, we changed the weights of the categories used in the Case 1 optimization from 1/3 to 1/5 for the IAQ, 2/5 for the thermal comfort, and 2/5 for the energy input. This was justified because we initially set every simulation to less than 1000 ppm. In addition, for the energy input, we assigned more weight to the DV than to the PV, as the DV generally exhibited higher energy-input magnitudes, i.e., almost six to seven times higher. Thus, we assigned weights of 7/8 and 1/8 to the total 2/5 weight of the energy input for the DV and PV, respectively. The input settings are listed in Table 5.5. As shown, the inputs for PV 1 and PV2 were 6 and 4.5 L/s, respectively, which were the two highest settings in our current study. By contrast, the resulting inhaled CO₂ concentration was extremely low, i.e., below 600 ppm (Figure 4.24a). This is primarily caused by the PV jets traversing directly to the human nostrils,

as shown in Figure 4.28. In addition, this optimization yielded average human skin temperatures that deviated from the neutral temperature by 0.2 °C (Figure 4.24b). Furthermore, the DV power input decreased by approximately 60% owing to its larger weight (Figure 4.24c). However, because the two PVs were opened, the PV power input increased significantly.

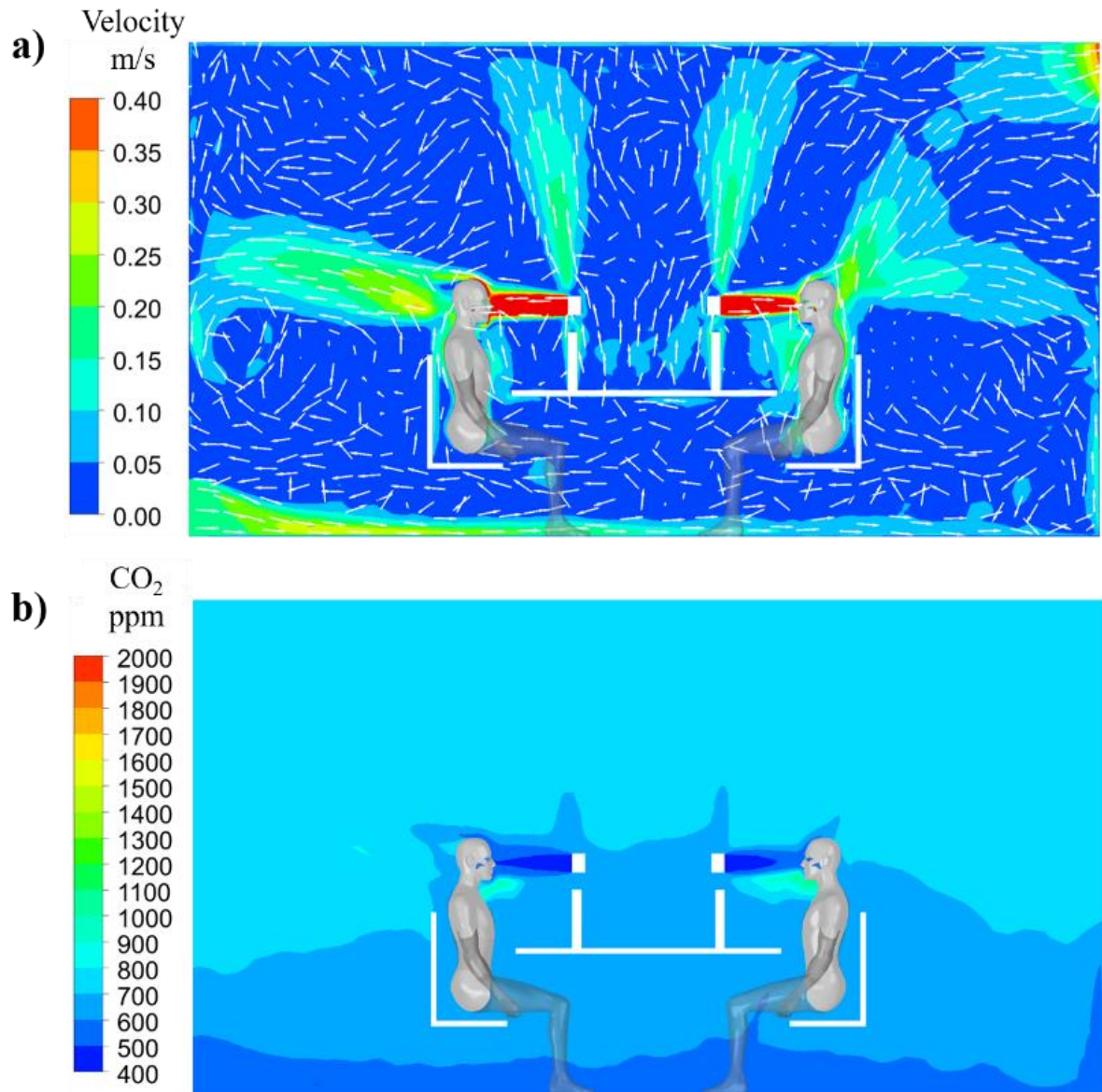


Figure 4.28. Concentration distribution and velocity vector field of optimum Case 5

This section shows that for the current PV system with jets flowing horizontally, the settings of the flow rate are affected significantly by thermal plumes and the properties of flow originating from the DV. Hence, the operating flow rate of the PV is limited; in this case, it does not exceed 6 L/s. Additionally, operating these PV systems at the maximum allowed speed increases energy consumption. Therefore, future studies and optimizations involving the

position of PV systems and flow angles should be conducted while accounting for the thermal plumes and properties of flow originating from the main ventilation.

4.5. Conclusion

This chapter conducted multicriteria optimization for a hybrid DV–PV system. Using the Taguchi DOE, we performed 25 CFD simulations to characterize the responses of thermal comfort, IAQ, and energy input by varying both the DV and PV parameters, i.e., the DV and PV flow rates, the DV and PV supply temperatures, and the DV recirculation. By assigning equal weights, we utilized the TOPSIS to combine thermal comfort, IAQ, and energy input into one objective function, and subsequently performed Taguchi's (S/N) ratio analysis to determine the input settings that would yield the optimum solution. Furthermore, four optimization cases were investigated by varying the thermal comfort and IAQ parameters from volume-averaged to occupant-centric values, such as the skin temperature and inhaled concentration.

Owing to imperfect mixing conditions, nonlinear relationships were observed when we compared the IAQ and thermal comfort between simulations as well as between optimization cases. For example, a low RVA CO₂ concentration did not imply a low OZVA or inhaled CO₂ concentration. As we focused on more occupant-centric metrics, such as the inhaled CO₂ concentration, these nonlinearities between different IAQ metrics intensified, as they were now dependent on the local flow field and concentration distributions. This was similarly observed for the different thermal-comfort parameters. Simulations with skin temperatures within an acceptable range did not indicate values within the published range of acceptable indoor temperature. Nevertheless, we observed a strong linear dependency of the thermal-comfort parameters on the DV supply flow rate and temperature, whereas the IAQ parameters showed that only RVA and OZVA metrics were more dependent on the total fresh air supplied to the room.

The four optimization cases yielded higher objective-function values than their Taguchi counterparts, but by an extremely small percentage, as the best simulations in the Taguchi table exhibited a higher objective-function value of 0.97–0.99. Nonetheless, this confirmed the viability of the Taguchi–TOPSIS method for the multicriteria optimization of indoor environments. Meanwhile, the optimizations of Cases 1 and 2, which use more occupant-centric metrics, yielded lower energy requirements, as they were more focused on local areas compared with Cases 3 and Case 4, which required an extremely large indoor space. Thus, the optimization metrics of Cases 1 and 2 were better as they focused on the metrics experienced by the occupants. However, further analysis should be performed for Case 2, as it involved 20 thermal-comfort parameters to satisfy because it utilized segmented skin temperatures.

Additionally, we observed that the PV systems of the optimized cases were turned off. First, this can be explained by the fact that the 25 simulations were set such that they exhibited an acceptable estimated RVA IAQ. In addition, as the PV flow rate for avoiding thermal discomfort at elevated air speeds was limited, most of the flow rate settings for the PV systems hindered them from functioning as intended, which was to have their jet directly enter the occupants' nostrils such that they inhaled extremely low CO₂ concentrations. This can be attributed to the effect of thermal plumes (or the buoyancy effect) and the DV settings, which altered the direction of jet traversing from the PV systems. Nevertheless, under our PV design with jets flowing horizontally, only simulations with a maximum setting of 6 L/s functioned appropriately in general, regardless of the other input settings. Therefore, future studies and optimizations involving the position of PV systems and flow angles should be conducted while accounting for the thermal plumes and properties of flow originating from the main ventilation.

Overall, this study may potentially help in the design and control of ventilation systems in providing occupant-centric IAQ and thermal comfort at low energy requirements. Lastly, the current study disregarded the effects of humidity in the calculation, as we assumed uniform value inside the office, matching the outdoors, and the only potential sources of increase/decrease are occupants who only perform light work. Therefore, future studies can improve the optimization by considering humidity effects.

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Appendix

Table 4A1. Case 4 Raw Values

Simulation	RVA CO ₂ (ppm)	RVA Temperature (°C)	QDV (kW)	QPV-average (kW)
1	715	25.4	(0.05)	0.00
2	751	25.8	(0.08)	0.01
3	823	26.3	(0.12)	0.03
4	875	26.8	(0.17)	0.05
5	930	27.4	(0.22)	0.08
6	986	21.9	(0.37)	0.03
7	555	22.8	0.01	0.05
8	608	23.1	(0.02)	0.04
9	673	24.1	(0.08)	0.03
10	825	25.0	(0.16)	0.01
11	644	20.2	(0.30)	0.04
12	847	21.0	(0.35)	0.02
13	523	22.0	0.12	0.04
14	545	22.8	0.07	0.02
15	590	23.5	0.01	0.02
16	544	19.4	(0.22)	0.04
17	600	20.1	(0.24)	0.03
18	763	21.0	(0.33)	0.02

Chapter 5: Comparison of Local Indoor Air Quality and Thermal Comfort Under Hybrid Mixing-Personalized and Hybrid Displacement-Personalized Ventilation in an Office

19	488	22.0	0.28	0.04
20	521	22.8	0.21	0.02
21	497	18.8	(0.11)	0.05
22	614	19.7	(0.12)	0.03
23	613	20.5	(0.18)	0.01
24	730	21.4	(0.27)	0.02
25	483	22.3	0.47	0.03

*Red characters in parentheses indicate negative values

Table 4A2. Case 4 Post-Processed Matrix D

Simulation	RVA CO ₂ (ppm)	Distance from 23°C	QDV (kW) absolute value	QPV-average (kW)
1	715	2.4	(0.05)	0.00
2	751	2.8	(0.08)	0.01
3	823	3.3	(0.12)	0.03
4	875	3.8	(0.17)	0.05
5	930	4.4	(0.22)	0.08
6	986	1.1	(0.37)	0.03
7	555	0.2	0.01	0.05
8	608	0.1	(0.02)	0.04
9	673	1.1	(0.08)	0.03
10	825	2.0	(0.16)	0.01
11	644	2.8	(0.30)	0.04
12	847	2.0	(0.35)	0.02
13	523	1.0	0.12	0.04
14	545	0.2	0.07	0.02
15	590	0.5	0.01	0.02
16	544	3.6	(0.22)	0.04
17	600	2.9	(0.24)	0.03
18	763	2.0	(0.33)	0.02
19	488	1.0	0.28	0.04
20	521	0.2	0.21	0.02
21	497	4.2	(0.11)	0.05
22	614	3.3	(0.12)	0.03
23	613	2.5	(0.18)	0.01
24	730	1.6	(0.27)	0.02
25	483	0.7	0.47	0.03

Table 4A3. Case 4 Normalized Matrix D

Simulation	RVA CO ₂	Distance from 23°C	QDV absolute value	QPV-average
1	0.208717321	0.19891832	0.045732	0
2	0.219193944	0.231983683	0.074315	0.077236
3	0.24039276	0.273937077	0.110967	0.175633
4	0.255590778	0.320009541	0.153841	0.29519
5	0.271414756	0.370830763	0.201759	0.44014
6	0.287819615	0.090233134	0.341869	0.147595
7	0.162064342	0.016330642	0.013451	0.275087
8	0.177394032	0.010224014	0.014852	0.233295
9	0.196577617	0.090983204	0.070055	0.154472
10	0.240974905	0.17206313	0.142072	0.043908
11	0.187892748	0.231656211	0.275793	0.231708
12	0.247250719	0.165464874	0.323263	0.131724
13	0.152634168	0.086870027	0.112985	0.245991
14	0.159187638	0.018300522	0.065908	0.110035
15	0.172388252	0.038087715	0.011769	0.133311
16	0.158956086	0.306945195	0.201759	0.22007
17	0.175115537	0.243515899	0.223952	0.199967
18	0.222623385	0.167470953	0.298603	0.115854
19	0.142623066	0.088011547	0.252199	0.219541
20	0.152240802	0.016819745	0.19268	0.098397
21	0.145134101	0.351053672	0.097405	0.307357
22	0.179259767	0.276199072	0.110967	0.196793
23	0.178957023	0.209341006	0.166451	0.055017
24	0.213029949	0.136558989	0.251526	0.099984
25	0.141119452	0.057379912	0.426608	0.19309

Table 4A4. Case 4 Weighted Normalized Matrix D

Simulation	RVA CO ₂	Distance from 23°C	QDV absolute value	QPV-average
1	0.06957244	0.066306107	0.007622	0
2	0.073064648	0.077327894	0.012386	0.012873
3	0.08013092	0.091312359	0.018495	0.029272
4	0.085196926	0.106669847	0.02564	0.049198

5	0.090471585	0.123610254	0.033626	0.073357
6	0.095939872	0.030077711	0.056978	0.024599
7	0.054021447	0.005443547	0.002242	0.045848
8	0.059131344	0.003408005	0.002475	0.038883
9	0.065525872	0.030327735	0.011676	0.025745
10	0.080324968	0.057354377	0.023679	0.007318
11	0.062630916	0.077218737	0.045966	0.038618
12	0.082416906	0.055154958	0.053877	0.021954
13	0.050878056	0.028956676	0.018831	0.040999
14	0.053062546	0.006100174	0.010985	0.018339
15	0.057462751	0.012695905	0.001962	0.022219
16	0.052985362	0.102315065	0.033626	0.036678
17	0.058371846	0.081171966	0.037325	0.033328
18	0.074207795	0.055823651	0.049767	0.019309
19	0.047541022	0.029337182	0.042033	0.03659
20	0.050746934	0.005606582	0.032113	0.016399
21	0.048378034	0.117017891	0.016234	0.051226
22	0.059753256	0.092066357	0.018495	0.032799
23	0.059652341	0.069780335	0.027742	0.00917
24	0.071009983	0.045519663	0.041921	0.016664
25	0.047039817	0.019126637	0.071101	0.032182

*Red and yellow highlights denote the best and worst values per parameter, respectively.

Table 4A5. Case 4 Distance from best (C+) and worst (C-), Closeness Coefficient C, and S/N Ratio Value

Simulation	C+	C-	C	S/N Ratio
1	0.004496	0.01339	0.75	-2.51465
2	0.006416	0.009771	0.60	-4.38439
3	0.009952	0.006004	0.38	-8.48989
4	0.0151	0.003053	0.17	-15.4851
5	0.022719	0.001434	0.06	-24.5268
6	0.006734	0.011325	0.63	-4.0533
7	0.002155	0.021219	0.91	-0.84016
8	0.001658	0.021701	0.93	-0.6396
9	0.001824	0.015425	0.89	-0.97058
10	0.004543	0.011244	0.71	-2.94781
11	0.009119	0.0051	0.36	-8.9056

12	0.007106	0.007808	0.52	-5.62143
13	0.002633	0.014769	0.85	-1.42491
14	0.000461	0.022288	0.98	-0.17792
15	0.000689	0.021178	0.97	-0.27792
16	0.012166	0.005048	0.29	-10.6549
17	0.008537	0.005955	0.41	-7.72452
18	0.006144	0.008444	0.58	-4.74896
19	0.003617	0.013427	0.79	-2.07198
20	0.001197	0.020731	0.95	-0.48742
21	0.015737	0.005806	0.27	-11.3888
22	0.009371	0.006717	0.42	-7.58664
23	0.005313	0.010214	0.66	-3.63774
24	0.004222	0.010785	0.72	-2.86967
25	0.006063	0.015003	0.71	-2.94803

Table 4A6. Case 4 S/N analysis

Level	DV flow rate	DV temperature	PV1 flow rate	PV2 flow rate	PV temperature	Recirculation
1	-11.0802	-7.50345	-3.5952	-2.64636	-2.80527	-1.95995
2	-1.89029	-5.23143	-2.88506	-4.45634	-4.39151	-3.41562
3	-3.28156	-3.78822	-4.31855	-4.67873	-6.10398	-9.32665
4	-5.13756	-4.31506	-7.06982	-6.54819	-5.80748	-7.74016
5	-5.68617	-6.2376	-9.20713	-8.74613	-7.96752	-8.36404

*Bold cells denote the highest value per parameter which served as the combination of the level of inputs needed to achieved optimum objective function value C.

CHAPTER 5 – COMPARISON OF LOCAL INDOOR AIR QUALITY AND THERMAL COMFORT UNDER HYBRID MIXING-PERSONALIZED AND HYBRID DISPLACEMENT-PERSONALIZED VENTILATION IN AN OFFICE

5.1 Introduction

In contrast with displacement ventilation (DV) system, mixing ventilation (MV) allows the supply air to circulate around the indoor environment. It aims to provide near well-mixed or perfect mixing conditions where the contaminant distributions and even the temperature are near homogenous throughout the domain. Although one might say that DV is better than MV, the reality is that most establishments, especially during the pre-COVID 19 era, had the latter type of ventilation installed. Therefore, switching to DV may require high capital cost, making the MV with a personalized ventilation retrofit the way to go.

Several studies were conducted to show the effect of integrating personalized ventilation into MV. Notable parameters varied in those studies are the PV flow rate and orientation between occupants, evaluating the effect on local indoor air quality, while also fixing the main ventilation settings. In particular, Xu et al. [1] investigated the effect of varying PV flow rates on the transmission of virus particles from an infected occupant to a healthy occupant. They categorized two types of exposure: indirect and direct, claiming that PV installed in front of the infected person may increase those types. The infection probability among the test cases ranged from 0.28 to 0.85. With varying occupant orientation (side-by-side, back-to-back, and face-to-back) in addition to PV flow rates, study of Liu et al. [2] showed reduction in infection risk by 65% under back-to-back occupant orientation and a PV flow rate of 9 L/s. Similarly, Katramiz et al. [3] found lower exposure levels when the infected person used lower PV flow rates for side-by-side orientation, while higher PV flow rates for face-to-face orientation. They also explored the cross-contamination for both coughing and breathing events of the occupants. Previous literatures also compared hybrid MV-PV with other hybrid systems. For example, He et al. [4] compared the effects of changing three ventilation strategies MV, DV, and under-floor air distribution (UFAD) system, with or without PV. Their study showed that DV performed best in preventing cross-contamination even without PV. They also observed that infection risk would always decrease when PV is paired with MV, while may sometimes increase the risk when it is paired with DV or UFAD, depending on the PV flow rate. These observations were also present in the study of Li et al. [5] when they compared hybrid DV-PV and MV-PV.

From the mentioned literatures, MV-PV studies mainly focused on the parametric analyses of the indoor air quality by varying the flow rate. This is also the trend in the comparison of hybrid MV-PV with other hybrid systems. In addition, as thermal comfort and ventilation

system's corresponding energy requirements were also equally important besides indoor air quality, they were often neglected in the analysis. Therefore, this section particularly aims to extend the previous studies by conducting simultaneous optimization of indoor air quality (in terms of inhaled CO_2 concentration), thermal comfort (in terms of average skin temperature of the occupants), and energy requirements of hybrid MV-PV system by varying both MV and PV input parameters, particularly supply flow rate and temperature and the recirculation rate of the main ventilation. Furthermore, as the general methodology are the same with Chapter 4, this also aims to compare the results of hybrid MV-PV with the hybrid DV-PV in Chapter 4. Lastly, this study attempts to reinforce the significance of accounting the main ventilation settings in optimizing occupant-centric ventilation system.

5.2 Methodology

5.2.1 General methodology

The general methodology including the computational domain, boundary conditions, numerical settings, thermoregulation model, breathing parameters, and optimization conditions and procedures was the same with Chapter 4. The only difference is the location of the inlet which is now near the ceiling of the computational domain. Shown in Figure 5.29 is the room lay-out and the schematic diagram.

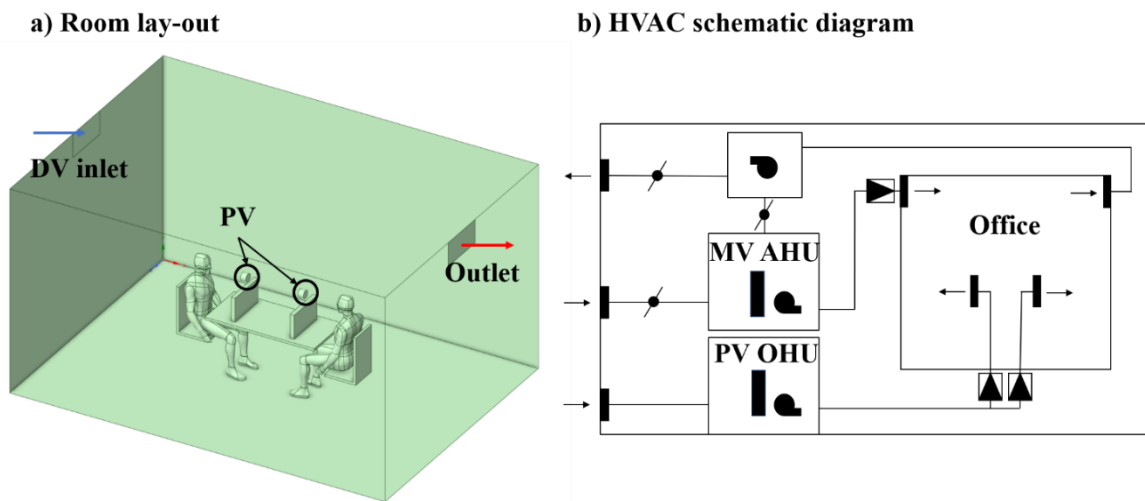


Figure 5.29. Room layout and schematic diagram of hybrid MV-PV system

For the output being optimized, this study focuses on the occupant-centric measures for IAQ and thermal comfort, which are inhaled CO_2 concentration and averaged skin temperature, respectively.

5.3 Result and Discussions

5.3.1 Inhaled CO₂ Concentration

Figure 5.30 shows the inhaled CO₂ concentrations of occupant 1 (left) and 2 (right) of the hybrid MV-PV system. We also included the data of hybrid DV-PV system from the previous chapter for comparison. For the hybrid MV-PV system we can see that there are plenty of cases where at least one of the two occupants exceeded the limit of 1000 ppm. These are simulations 2, 6, 10, 12, 14, and 17 – 20, which is 9/25 of the Taguchi design, in contrast with the DV-PV system with only simulation 2 exceeding the limit. For the MV-PV system, simulation 8 yielded the lowest average inhaled concentration (520 ppm), which can be attributed to both of the PV jets reaching the nostrils of the human (Figure 5.31a). Simulation 14 (Figure 5.31b) on the other hand has the highest record of inhaled CO₂ concentration, particularly experienced by occupant 2. This can be attributed to the PV jet of occupant 2 not reaching the nostrils of the occupant, which generated stagnation region in its breathing zone, thus allowing the accumulation of CO₂ in that region. Furthermore, out of the 25 cases, we can see in simulations 16 – 20 that occupant 1 has the higher inhaled concentration compared to occupant 2. In these 5 simulations, the supply inlet flow rate is 90 L/s. Figure 5.31c shows that the jet from the supply inlet goes directly to the space between the two occupants. This jet creates a vortex at the vicinity near the nostrils of the occupant 1, thus increasing the concentration.

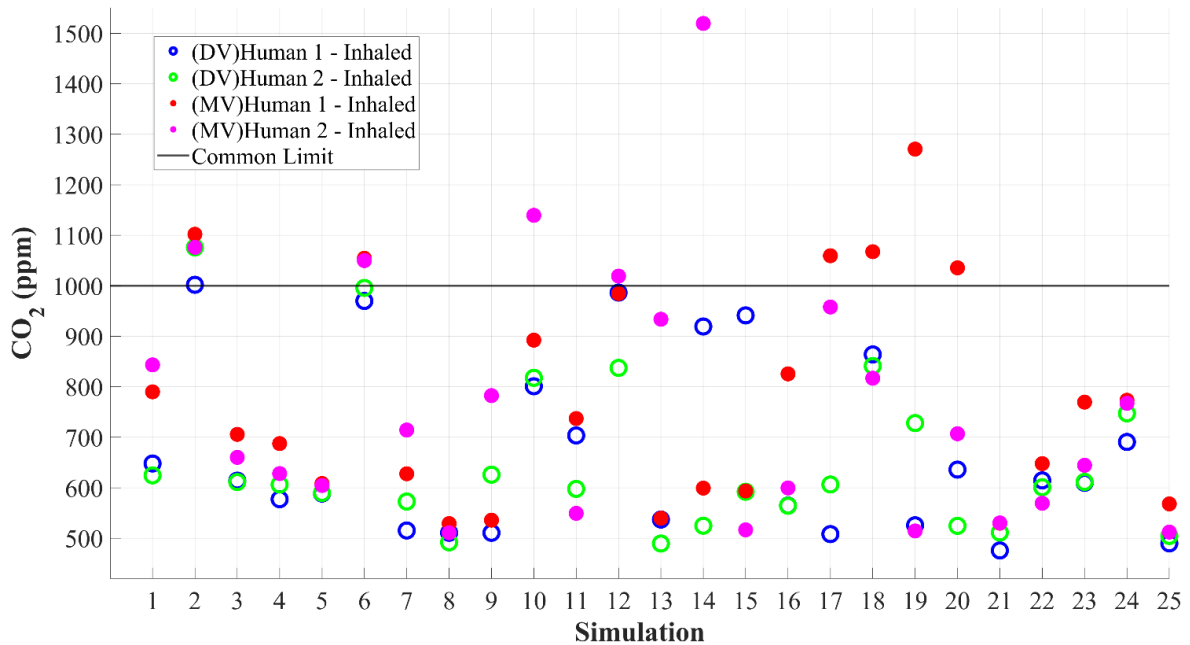


Figure 5.30. Inhaled CO₂ concentration values of the two hybrid systems for the 25 simulations

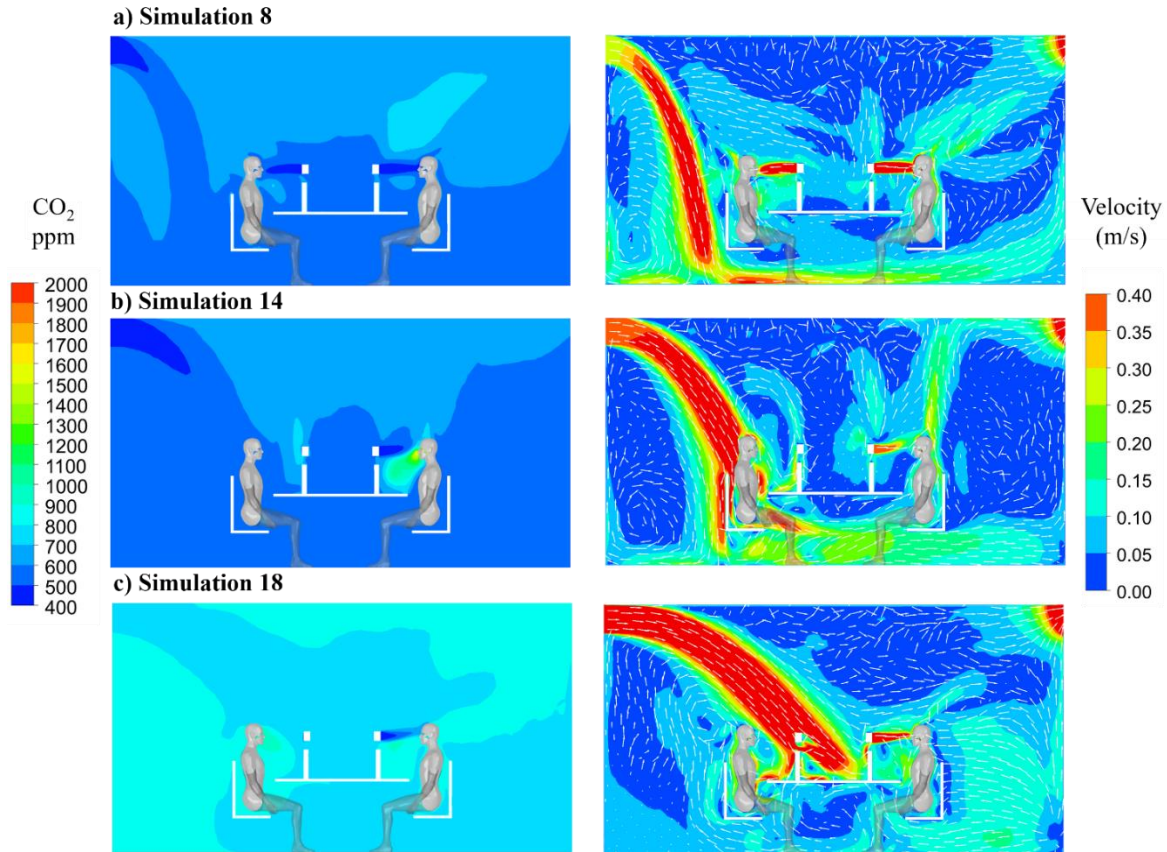


Figure 5.31. Concentration and velocity distributions of Simulation a) 8, b) 14, and c) 18

Looking again at Figure 5.30 the average inhaled concentrations of the two occupants are higher in MV-PV compared to DV-PV, except for 2 simulations (simulation 11 by little amount and simulation 15 by a huge amount). This can be explained by 2 factors. One is the general function of the main ventilation used. The DV system forms concentration stratification from the bottom to the top of domain, having the lowest at the bottom as the supply inlet is positioned at the bottom. Since the position of breathing zone of the occupants are in the middle of domain, they generally fall into a lower CO₂ concentration region. In contrast, MV allows the concentration to be well-mixed with less stratification, therefore, the concentration tends to be higher. As an example, the stratification can be seen in simulation 1 of hybrid DV-PV compared to hybrid MV-PV (Figure 5.32). However, this trend can still be affected by the local flow field near the inhalation zone of the occupants as a result of flow interaction coming from MV and PV. For example, in simulation 15 (Figure 5.33), average inhaled CO₂ concentration is lower in MV-PV (555 versus 767 ppm). This can be attributed to the supply jet of the MV directly goes to occupant 1, while the PV directly provides ventilation to occupant 2 on the right side as shown in Figure 5.33a. In contrast, the DV-PV case has the tendency to form higher CO₂ concentration near the breathing zone of the occupant (Figure 5.33b), increasing the inhaled CO₂ concentration (941 ppm).

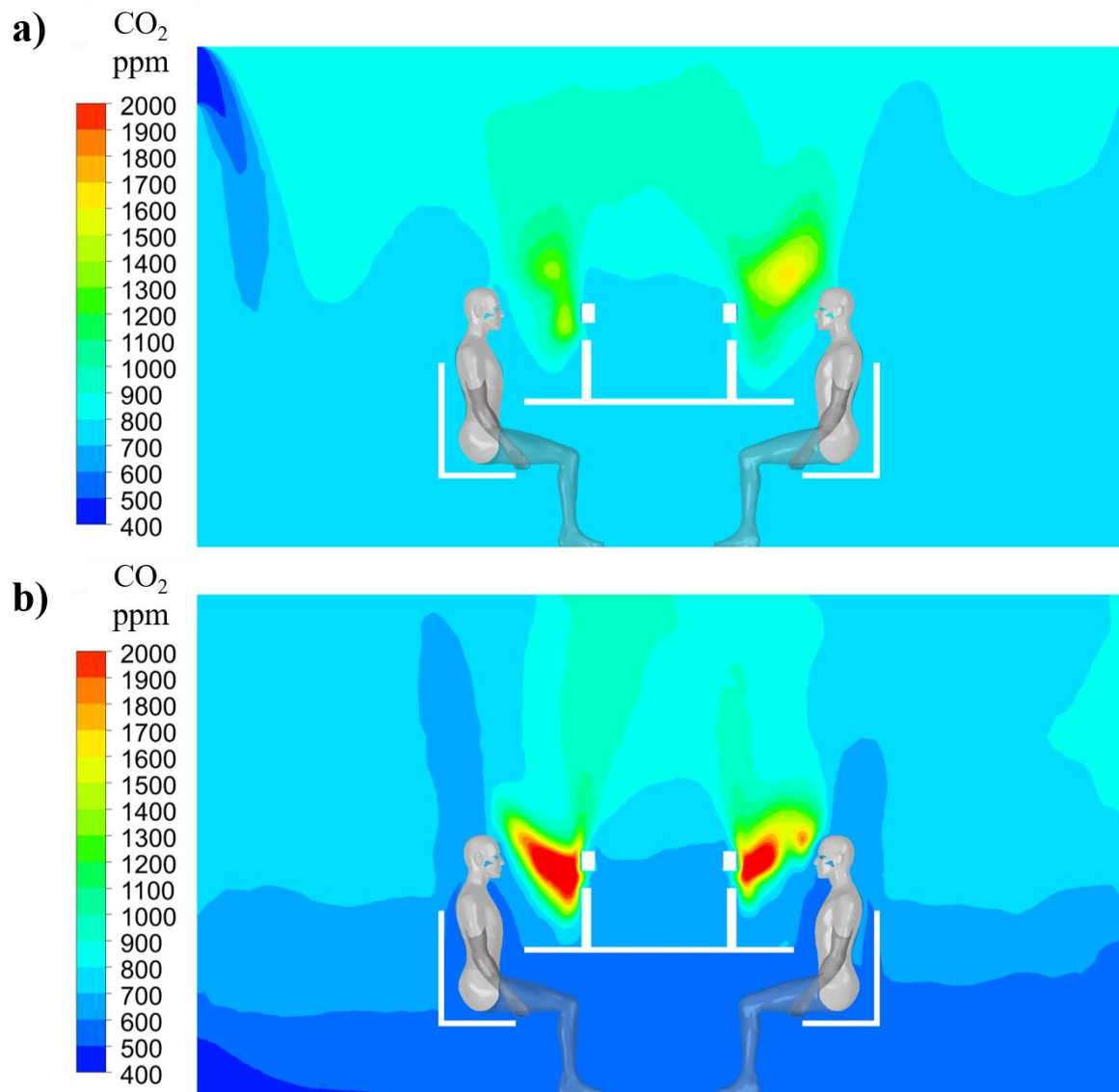


Figure 5.32. Concentration distributions of simulation 1 of a) MV-PV and b) DV-PV system

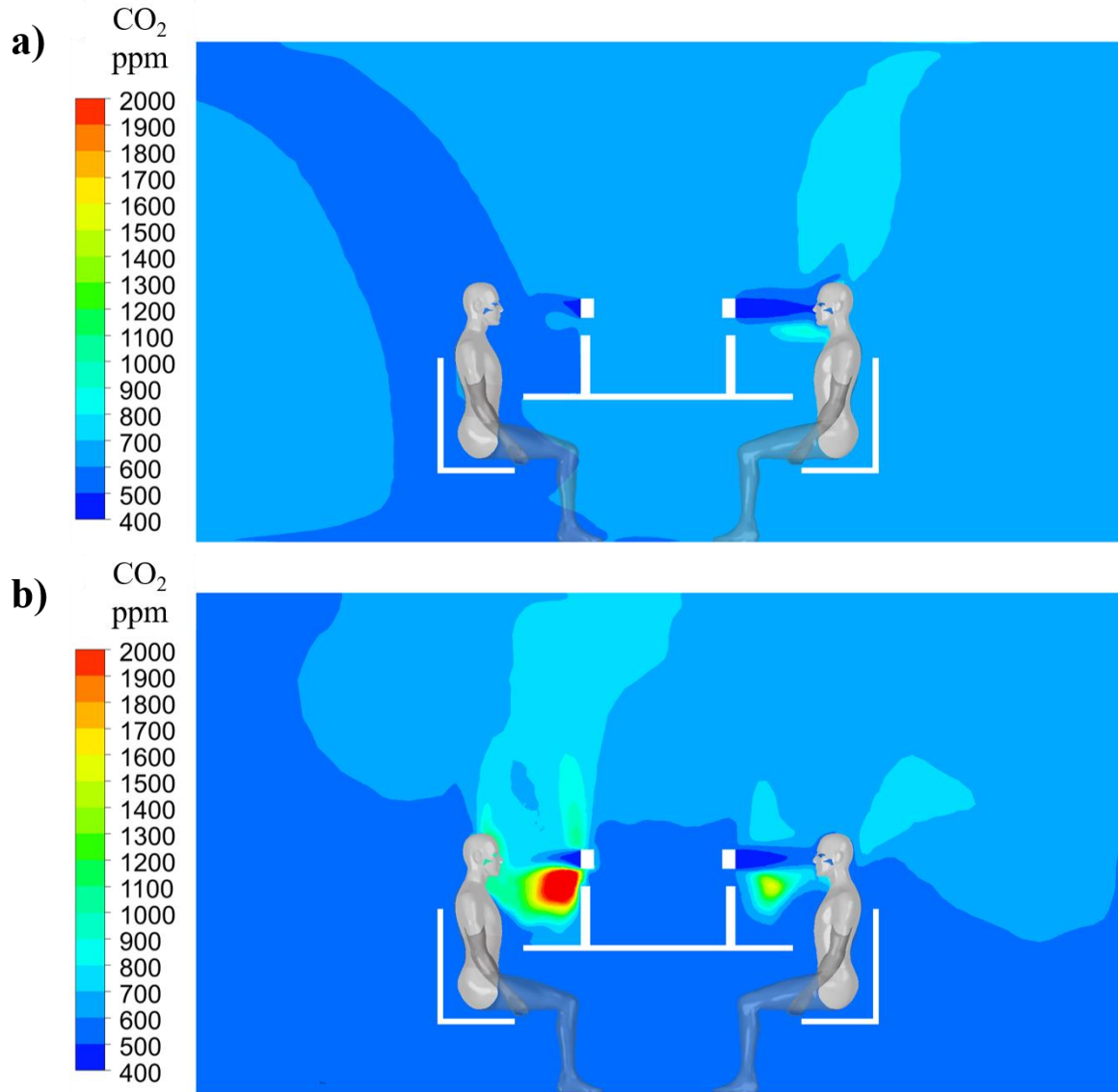


Figure 5.33. Concentration distributions of simulation 15 of a) MV-PV and b) DV-PV system

5.3.2 Average Skin Temperature

Figure 5.34 depicts the average skin temperatures of occupants 1 and 2 for the 25 simulations for both hybrid DV-PV and MV-PV systems. For the MV-PV system, simulations 1 – 5, 9 and 10, fall into the set neutral skin temperature of 33.6 ± 0.5 °C. This is 1 simulation better compared to the DV-PV system wherein simulation 9 is not within the range. In general, the temperature values of the 2 occupants are almost the same in the case DV-PV system which can be attributed to the thermal stratification inherent in DV system. In contrast, differences between the two temperature values were observable in MV-PV system, especially simulations 11 – 25 cases where the mixing ventilation flow rate was 70 L/s and above. This can be explained by the MV supply jet that may directly reach either of the two occupants. For

example, Figure 5.35 shows the velocity distributions of simulation 11 and 21 of the hybrid MV-PV system. In simulation 11 (supply flow rate = 70 L/s), the MV supply jet directly reaches, occupant 1 – making its skin temperature to drop compared to occupant 2. On the other hand, in simulation 21 (supply flow rate = 110 L/s), the MV supply jet directly reaches the occupant 2, making its skin temperature to drop compared to occupant 1. It can also be observed from Figure 5.34 that occupant 1 in the hybrid MV-PV system have smaller skin temperatures except for the last 5 simulations. This can be attributed to the last simulations having a supply flow rate of 110 L/s, which directly reaches the second occupant, thereby decreasing the skin temperature.

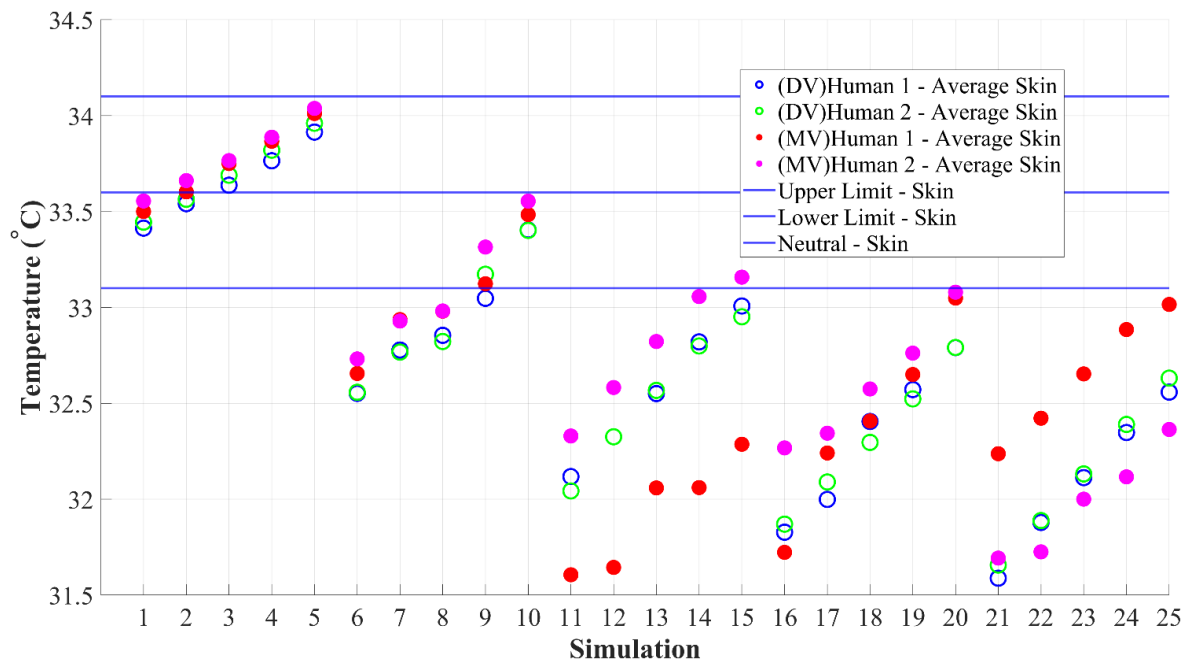


Figure 5.34. Average skin temperature values of the two hybrid systems for 25 the simulations

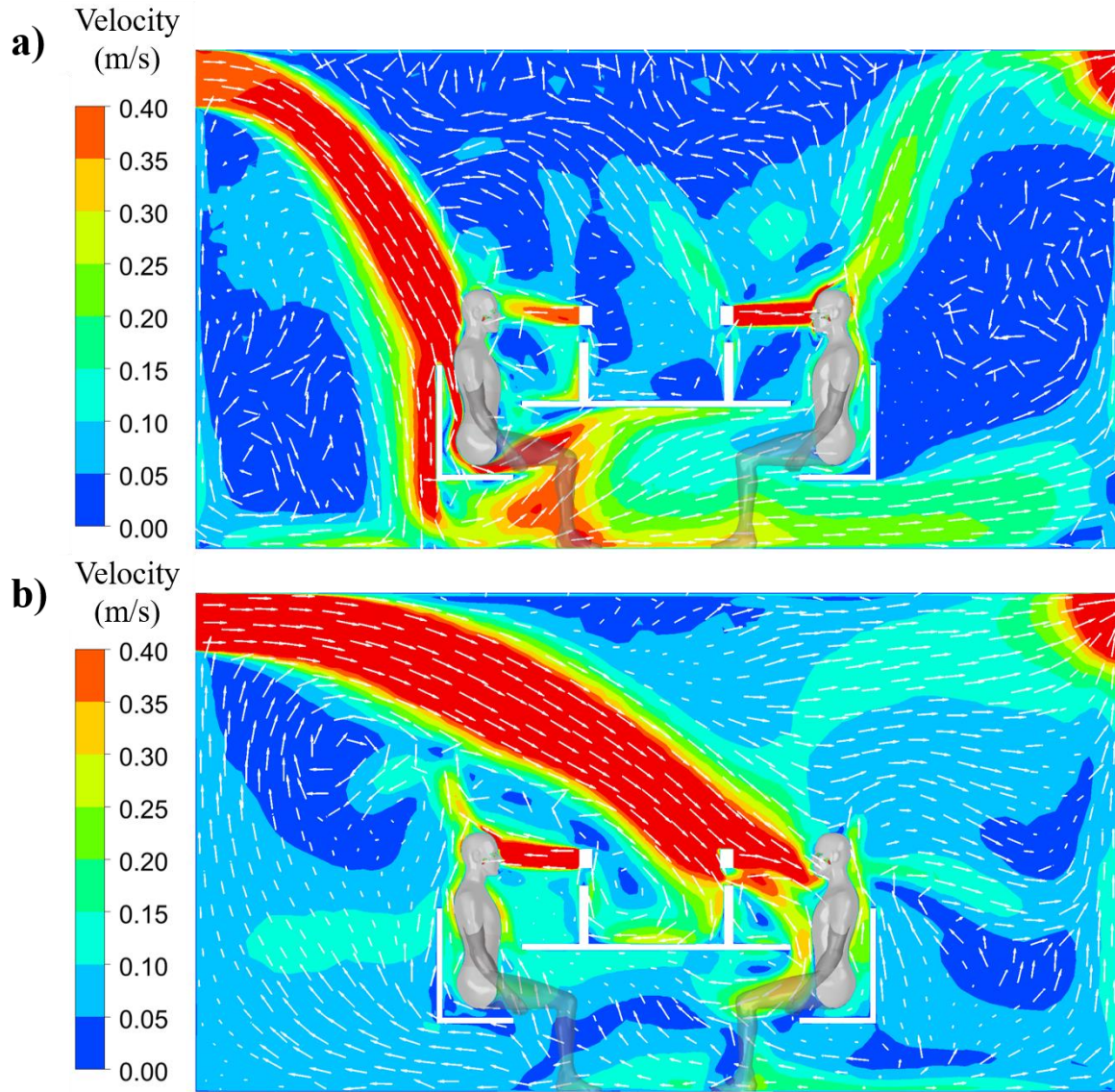


Figure 5.35. Velocity distributions of simulation a) 11 and b) 21 of the MV-PV system

5.3.3 Energy Requirements

Figure 5.36 illustrates the main ventilation requirements for both hybrid systems. It is important to note that PV energy requirements were not included since its boundary conditions and varied parameters are the same for both systems. These conditions made the PV energy requirements of the two hybrid systems the same for each of the 25 simulations. Furthermore, this section compares the energy requirements not in absolute terms (unlike in Chapter 4), but based on which is highest in the set of real numbers for convenience. From the figure, it can be observed that compared to DV, MV energy requirements are higher for simulations 1 – 10 and 16 – 20, while lower for simulations 11 – 15 and 21 – 25. This behavior can be explained by the equation used to compute the energy requirement of the main ventilation system in Chapter 4 (Eq. (4.4)). In the equation, the mass flow rate and the inlet enthalpy are the same for each of

the simulations of the two systems, and only the inlet enthalpy varies, which is a function of the outlet temperature.

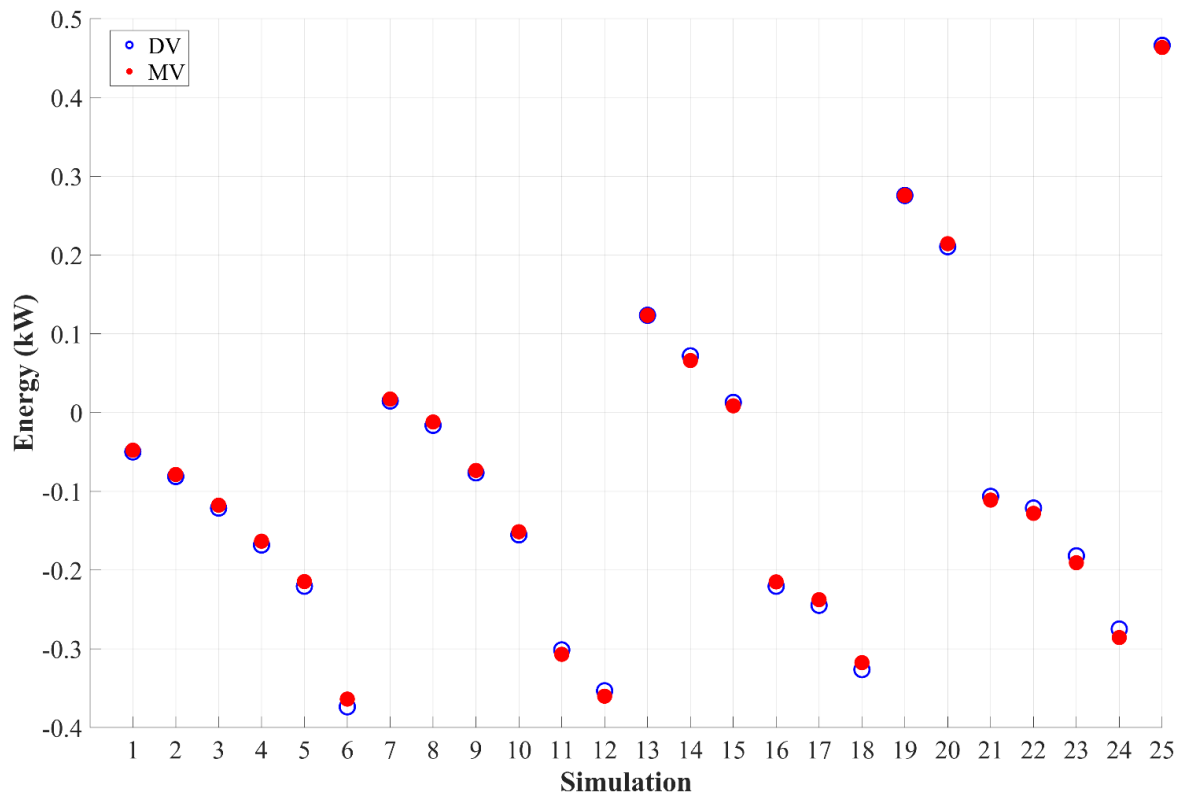


Figure 5.36. Energy requirements of the two hybrid systems for the 25 simulations

As can be observed in Figure 5.37, compared to DV-PV, the MV-PV outlet temperatures are lower for simulations 1 – 10 and 16 – 20, while higher for simulations 11 – 15 and 21 – 25. Therefore, it can be said that the energy requirement becomes low when the outlet temperature is high and vice versa. In terms of what factor contributes to the increase or decrease of the outlet temperature of MV-PV compared to DV-PV, it can be explained by whether the inlet supply jet will directly reach one of the occupants or not. Simulations 11 – 15 with an inlet flow rate of 70 L/s directly reach occupant 1, while simulations 20 – 25 with an inlet flow rate of 110 L/s directly reach occupant 2. This process of reaching the occupant picks up heat from them, who are the main source of heat inside the room, eventually increasing the temperature being exhausted to the outlet.

Talking about the energy in absolute terms, as energy requirement with a positive value means the system needs heating while a negative value means cooling, having an absolute value near zero is the better value. With this, simulation 15 of the MV-PV system yielded the lowest energy requirement of 0.0086 kW of heating, followed by simulation 8 with an energy requirement of higher than 0.012 kW of cooling. Furthermore, the third lowest energy

requirement is seen in DV-PV system in simulation 15 with an energy requirement of 0.013 kW of heating.

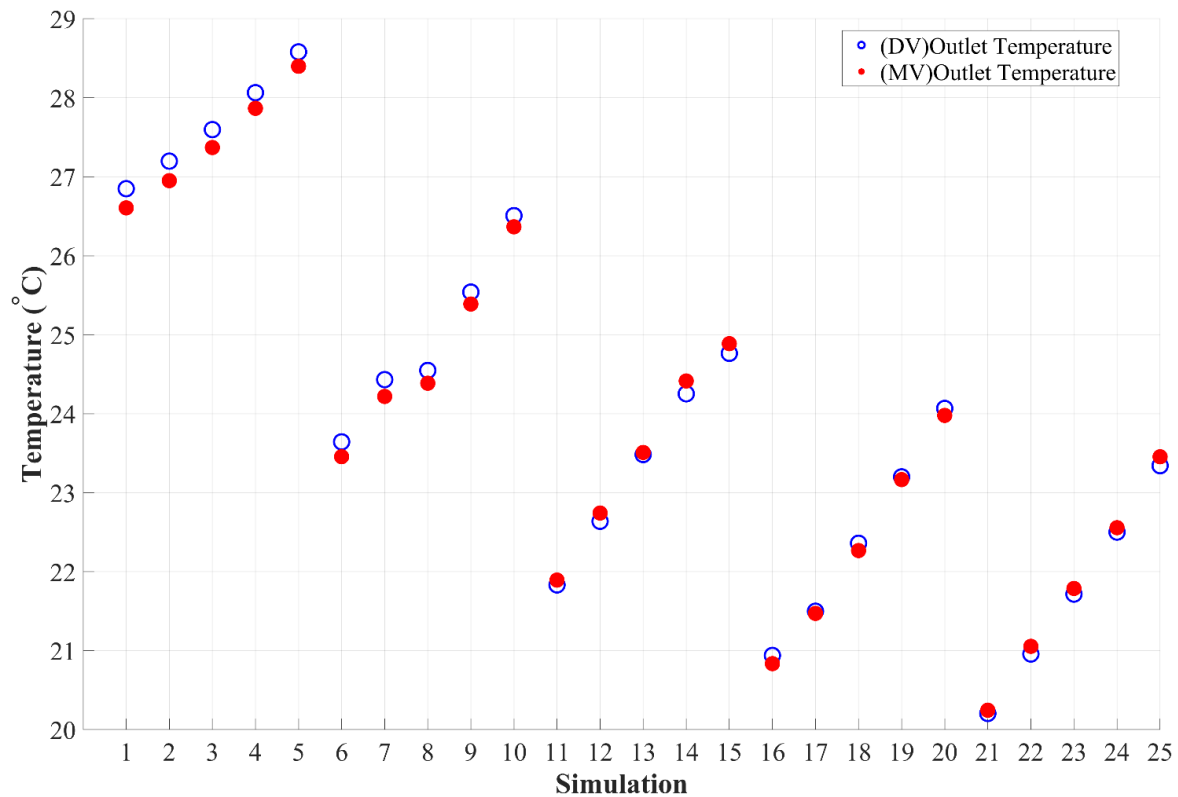


Figure 5.37. Outlet temperatures of the two hybrid systems for the 25 simulations

In terms of contaminant concentration, the DV stratification is generally deemed more beneficial compared to that of MV system, as lower concentrations can be found in the breathing zone of the occupants. In contrast, MV may sometimes be beneficial to one of the occupants especially when its supply jet directly reaches the occupant. However, it still the interaction between the flow field generated by the main ventilation and PV that dictates the local flow field near the occupants and eventually the inhaled concentration as shown in Figure 5.31 and Figure 5.33. In terms of skin temperature, the DV stratification almost maintained similar values for the two occupants, as the main ventilation does not reach directly both of the occupants. In contrast, MV may results in different temperature for the two occupants depending on the direction of the supply jet. In terms of energy requirements, it has been shown that it depends on whether the MV supply jet will reach either of the occupants. Generally, from these comparisons between the two hybrid systems, it has been shown that the main ventilation has a significant effect in the indoor air quality, thermal comfort, and energy requirements, which makes it necessary to include their parameters in the optimization instead of focusing on the PV parameters alone.

5.3.4 Taguchi-TOPSIS of the 2 Hybrid Systems

With equal weights for the IAQ, thermal comfort, and energy requirements, we utilized “technique for order preference by similarity to ideal solution” (TOPSIS) to combine the three output parameters into 1 single objective function, and to rank them based on their proximity to the ideal value (i.e., having very low inhaled CO₂ concentration, within the neutral skin temperature range, and with low energy requirements). Table 5.6 shows the results for the individual TOPSIS analyses of the MV-PV and DV-PV systems. In addition, a combined analysis was done by combining the data sets of the two hybrid systems and performing TOPSIS, thereby having 50 data points. From the individual TOPSIS analyses, both the simulation 1 of the hybrid systems have the highest objective function value in their corresponding Taguchi tables, indicating that the settings of simulation 1 produce results that are near the ideal solution in terms of IAQ, thermal comfort, and energy requirements. It is also important to note that in this simulation, both PVs were turned off. This means that when the main ventilation system is well designed, PV maybe not be needed to achieve low inhaled CO₂ concentration with good thermal comfort and low energy consumption. Regarding the combined TOPSIS analysis, simulation 1 of the DV-PV has the highest objective function value, followed by the simulation 1 of the MV-PV. Comparing their corresponding output parameters (MV-PV versus DV-PV): 1) average inhaled concentration (817 ppm versus 636 ppm), 2) average occupant skin temperature (33.5 °C versus 33.4 °C), and 3) main ventilation energy requirements (0.048 kW versus 0.050 kW), a large discrepancy between the inhaled concentration can be seen, which makes the DV-PV to have a higher TOPSIS value (0.97 versus 0.99). Furthermore, the next two simulations with higher TOPSIS are also from the DV-PV. With a set threshold of 0.90 objective function value, 5 simulations from DV-PV can satisfy while 4 from the MV-PV. In these 9 simulations, higher objective values can be seen from the DV-PV. Based on these results, hybrid DV-PV can be said to have better outputs compared to hybrid MV-PV, albeit by very small percentages.

Table 5.6. Objective function values and corresponding ranks of the separated and combined analyses of the two hybrid systems

Simulation	Individual TOPSIS				Combined TOPSIS			
	MV-PV		PV-DV		MV-PV		DV-PV	
	Value	Rank	Value	Rank	Value	Rank	Value	Rank
1	0.98	1	0.99	1	0.97	2	0.99	1
2	0.92	3	0.89	5	0.90	9	0.91	8
3	0.91	4	0.91	3	0.91	7	0.92	4
4	0.77	10	0.76	10	0.77	20	0.79	18

5	0.56	15	0.53	15	0.57	30	0.58	27
6	0.46	19	0.39	20	0.46	39	0.43	42
7	0.78	9	0.74	11	0.78	19	0.76	21
8	0.85	7	0.80	8	0.85	13	0.82	15
9	0.92	2	0.90	4	0.91	5	0.91	6
10	0.91	5	0.92	2	0.89	10	0.93	3
11	0.32	25	0.28	25	0.34	49	0.36	48
12	0.33	24	0.37	22	0.33	50	0.41	43
13	0.57	14	0.64	12	0.58	29	0.68	22
14	0.68	11	0.84	7	0.66	24	0.85	12
15	0.85	6	0.87	6	0.85	14	0.89	11
16	0.39	23	0.33	23	0.41	44	0.40	45
17	0.39	22	0.38	21	0.40	46	0.45	41
18	0.51	18	0.43	18	0.51	34	0.48	37
19	0.53	16	0.48	16	0.53	33	0.54	32
20	0.81	8	0.78	9	0.80	16	0.80	17
21	0.43	21	0.33	24	0.45	40	0.39	47
22	0.52	17	0.45	17	0.55	31	0.50	35
23	0.66	12	0.61	13	0.67	23	0.64	25
24	0.58	13	0.54	14	0.59	26	0.58	28
25	0.46	20	0.39	19	0.48	36	0.47	38

*red highlight depicts the best simulation in individual TOPSIS analysis, while yellow highlight depicts simulations with objective function value of 0.9 and above in the combined TOPSIS analysis.

5.3.5 Optimization of MV-PV

After computing the objective functions of each of the simulations in the Taguchi table of the MV-PV, Signal-to-Noise ratio analysis was done to determine the input parameters that will yield to the optimum objective function value. The procedure resulted in the following input parameters: 1) MV supply flow rate = 30 L/s, 2) MV supply temperature = 20 °C, 3) PV 1 supply flow rate = 0 L/s, 4) PV 2 supply flow rate = 0 L/s, 5) PV supply temperature = 21 °C, and 6) Recirculation = 0.35. After conducting the simulation, the setting yielded average inhaled concentration of 956 ppm, an average skin temperature of 34.2 °C, and an MV energy requirement of 0.032 kW. Comparing this result with that of the simulation 1 of the Taguchi table (highest TOPSIS value), we see that the average inhaled concentration has increased, although still within the limit of 1000 ppm. In terms of energy, the optimized value has

decreased by almost 40% (from 0.050 kW to 0.032 kW). However, the resulting average skin temperature is now outside the neutral range. Calculating the TOPSIS value, the solution yielded 0.91, which is lower than the highest value (0.98) in the Taguchi table. This indicates the failure of the Signal-to-Noise ratio to yield the optimum value in the MV-PV system. This can be attributed to the non-linear responses of the output to the input, as the analysis is based on the assumption of linear relationship between the parameters. For example, the occupant skin temperature doesn't have a linear increase or decrease with respect to the MV supply flow rate. As discussed in the previous section when the flow rate is 70 L/s, occupant 1 has a higher decrease in average skin temperature compared to occupant 2. In contrast, when the flow rate is 110 L/s, occupant 2 has a higher decrease in temperature. This non-linearity can be illustrated by the behavior of average skin temperature of the MV-PV results in Figure 5.34. In the same figure, the DV-PV has more linear results, therefore, it was able to get optimum results using S/N ratio in the previous chapter.

5.3.6 Effect of MV settings on the PV

In the 25 simulations done, PV has work as intended for the simulations where the settings are 6 L/s. This is also the trend in the previous DV-PV study. The simulations yielded inhaled concentrations of less than 610 ppm for the occupant to whom it is directed. An exception can be observed in simulation 17, where the inhaled concentration of occupant 1 is 1059 ppm even when the setting is 6 L/s. This can be attributed to the main ventilation system that is directed between the occupants, generating a vortex near the breathing zone of the occupant 1 (similar to Figure 5.31c), thus increasing the concentration. In the previous DV-PV study, the jet coming from PV are generally skewed upwards due to the effects of thermal plume and DV supply which are both flowing upwards. In contrast, the current study has more different trends depending again on where the main ventilation supply inlet is directed. For example, in simulation 15 (Figure 5.33a), where flow is directed to occupant 1, the PV effect can no longer be quantified as its jet eventually mixes with the flow coming from the main ventilation. The same goes for simulation 21, as shown in Figure 5.35b

5.4 Conclusion

This chapter conducted multicriteria optimization of IAQ, thermal comfort, ventilation energy requirements for a hybrid MV-PV system. Similar to the previous chapter's general methodology, the Taguchi-TOPSIS method was utilized. In addition, this adopted the occupant-centric output parameters: inhaled CO₂ concentration for IAQ and average skin temperature for thermal comfort. Furthermore, this also compared the results with those of the hybrid DV-PV system from the previous chapter.

For the 25 simulations, MV-PV showed different responses compared to the DV-PV. In terms of IAQ, more simulations had inhaled CO₂ concentrations greater than 1000 ppm, which can primarily be attributed to the well-mixed behavior of MV-PV. In some cases, MV-PV can have better IAQ compared to DV-PV when the supply jet of the main ventilation of the former directly reaches the occupants. However, the concentration stratification of the DV system generally showed better results compared to the MV. In terms of thermal comfort, similar trend to IAQ is observed, which is heavily affected if the supply jet of the main ventilation reaches the occupant. When an occupant is directly experiencing the main ventilation, their average skin temperature drops. This contrasts with DV-PV, where thermal stratification is present, and the temperature of the two occupants are almost always the same. In terms of energy requirements, higher energy in MV-PV is observed depending on the resulting outlet temperature, which is affected by whether the supply jet will directly reach the occupant. Overall, the type of main ventilation heavily affects the output parameters, where concentration and thermal stratification inherent in DV system showed favorable results. This has been supported when we compared the TOPSIS of the corresponding 25 simulations of the two hybrid systems, where the DV systems results have generally have a slightly higher value.

Both simulation 1 (case with both PVs are off) of the two hybrid systems yielded the highest TOPSIS value within their corresponding Taguchi design, with values of 0.99 and 0.98 for DV-PV and MV-PV, respectively. This implies that when the system is well-designed, the use of PV might not be needed to maintain the inhaled concentration lower than 1000 ppm, while having average skin temperatures within the acceptable range, and at low energy requirements. This also shows that MV-PV system, although generally have lower TOPSIS values compared to DV-PV, can still be used as long as correct settings are determined and used.

Applying the S/N ratio to the MV-PV system yielded to lower TOPSIS value (0.91) compared to the best inside the Taguchi design (0.98), indicating the failure of the Taguchi-TOPSIS optimization. This can be attributed to the non-linear response of the output parameters to the input, in this case, heavily influenced by whether the supply jet from the main ventilation will reach either of the two occupants.

Lastly, this study reinforces that the main ventilation system parameters significantly affects the IAQ, thermal comfort, and energy requirements. This study adds to the pool of knowledge in providing good occupant-centric IAQ and thermal comfort at low energy requirements.

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CHAPTER 6 – SUMMARY AND FUTURE WORK

6.1. Summary

This dissertation applied the computational fluid dynamics (CFD) technique, both Reynolds-averaged Navier-Stokes (RANS) and Large Eddy Simulations (LES) eddy viscosity models, to generate contaminant spatial distributions of indoor environments with different ventilation modes to quantify indoor air quality (IAQ).

In chapter 2, the governing equations for fluid motion and also the transport equation for contaminants were presented. These equations are continuity, momentum (Navier-Stokes), and scalar transport equations. As there are unknown quantities to solve the transport equations, two turbulence modeling approaches were introduced: RANS and LES.

In chapter 3, an idealized cross-ventilation model in urban environment affected by sheltering effect was the test case and isothermal LES was used. To understand the effect of window orientation, and the location of the contaminant source on local IAQ, this chapter studied two window opening cases: parallel (STR) and perpendicular (LAT) to the flow, and placed several passive contaminant sources positioned in different parts of the room where flow patterns are expected to be different. These locations were 1) inlet where velocity is expected to be high and there are reversals of flow, 2) corners that are possible stagnation points, and 3) outlet. The outcomes showed generally high mean velocities for the STR case while low mean velocities for the LAT cases. The different scalars yielded different contaminant distributions even for scalars placed near each other, which can be attributed to imperfect mixing conditions. To quantify the scalar transfer or ventilation, this study computed three local ventilation efficiencies: age of air (AOA), purging flow rate (PFR), and net escape velocity (NEV). AOA values were generally low in the inlet of the cross-ventilation model, while high in the corners. On the other hand, an inverse correlation was seen between AOA and PFR, which can be proven by comparing their equations. Furthermore, NEV showed what drives the scalar transfer for different parts of the room. For high-velocity regions, fluid flow drives the contaminant transfer. In contrast, for regions where the mean flow is weak, turbulence becomes significant in contaminant transfer following the direction of high to low concentration.

In chapter 4, an office room with hybrid displacement-personalized ventilation (DV-PV) system was the test case, and the flow field and transport equations were solved using RANS. In contrast to the previous chapter, the study introduced more realistic contaminant sources, which are occupants that produce CO₂ through exhalation. In addition to quantifying IAQ, optimization was done simultaneously with thermal comfort and energy requirements of the

ventilation by varying both DV and PV parameters using Taguchi-TOPSIS (technique for order preference by similarity to ideal solution). Several cases of optimization by varying the IAQ parameters: 1) volume-averaged CO₂ concentration, 2) occupant zone averaged CO₂ concentration, and 3) inhaled CO₂ concentration, and thermal comfort parameters: 1) volume-averaged temperature, 2) occupant zone averaged temperature, 3) average skin temperature, and 4) segmented skin temperature, were explored. The different optimization cases yielded different results which can be attributed to imperfect mixing conditions, where occupant-centric measures of IAQ and thermal comfort are more dependent on the local flow, temperature, and contaminant distributions. Furthermore, the study showed the significant effects of thermal plume and parameters of the main ventilation system, indicating the importance of exploring their effects in personalized ventilation studies. Lastly, occupant-centric optimization yielded lower energy requirements.

In chapter 5, the general methodology was the same as in chapter 4, except that the ventilation system is a hybrid mixing-personalized (MV-PV) ventilation system. In addition, the study focused on the occupant-centric measures: inhaled CO₂ concentration and average skin temperature. Comparing the results with those of chapter 4, the hybrid DV-PV generally produced better results due to its contaminant and thermal stratification. However, for the same settings, IAQ of MV-PV may sometimes be better than its DV-PV counterpart, especially when the supply jet of the main ventilation goes directly to the occupants. Both hybrid systems yielded their best settings when the personalized ventilation for both occupants was turned off. This implies that when the system is well-designed, the use of PV might not be needed to maintain the inhaled concentration lower than the limit while keeping within the acceptable range of skin temperatures and at low energy requirements. Lastly, due to non-linear relationship between input parameter and output parameter, the Taguchi-TOPSIS optimization did not yield ideal results.

6.2. Future Work

6.2.1. For Chapter 3

Chapter 3 focused on simulations of an idealized cross-ventilation model. Therefore, further studies may be conducted by integrating complex and realistic building geometries. Buoyancy effects considering ambient and indoor temperatures, as well as other heat sources such as occupants, may also be integrated in the study. Wind patterns based on realistic conditions are also suggested. Similar to chapters 4 and 5, more realistic contaminant source can be integrated, and simultaneous optimization can be performed.

6.2.2. For Chapters 4 and 5

These two chapters ignored the effect of latent heat transfer, therefore, a mathematical model to integrate sweating for clothed human should be developed. Optimization can also be done with other types of hybrid ventilation systems, such as utilizing impinging jet ventilation system and cross-ventilation system as the main ventilation. Variations in the outdoor climatic conditions can also be explored. As the current study used PV with jets flowing horizontally, angles can be adjusted to account for the effects of thermal plume and main ventilation. Lastly, when a large amount of simulations is conducted, more sophisticated optimization techniques can be utilized.