Original Paper

Indoor and Built Environment

Indoor and Built Environment
2019, Vol. 28(3) 345–361

DOI: 10.1177/1420326X18757835
journals.sagepub.com/home/ibe
SAGE

A parametric study of the combined effects of window property and air vent placement

Qiuhua Duan[1] and Julian (Jialiang) Wang[1,2]

Abstract

The creation of energy-efficient buildings that maintain thermal comfort is a major goal of architectural design. The study on the interrelation of vent location and window properties on indoor thermal conditions and energy performance is still needed. With the rapid development of building window technologies, such combined effects have become more complex. This work using computational fluid dynamics examines airflow patterns, temperature distributions, thermal comfort indices and corresponding heat transfers through exterior windows in an office unit during summer and winter seasons. Our results indicate that, buildings with low-insulated windows, above-the-window air register and under-the-window air register, have obvious advantages in forming comfortable conditions in summer and winter, respectively. Building with highly insulated windows, the central-ceiling placement of air supply vents is capable of merging airflows for the entire room, providing appropriate indoor thermal conditions and significant savings in energy use. The percentage of occupants dissatisfied with the thermal conditions achieved is slightly higher than best performances achieved in other reference models but still within recommended limits. The findings from this research provide an improved understanding of how thermal comfort and energy issues could change in response to different vent locations and types of building windows.

Keywords

Building windows, Building performance, Energy use, Thermal comfort, Air vent placement, Simulation

Accepted 15 January 2018

Introduction

An average individual spends up to 95% of their time indoors; thus, the indoor environment is an important concern in architectural design.[1] Studies have demonstrated strong correlations between indoor environmental conditions and user well-being, productivity and health.[2,3] Air-conditioned office buildings yield improved work productivity and less thermal dissatisfaction by providing a thermally acceptable environment for human comfort.[4] However, a poor distribution of airflow, irregular temperatures leading to indoor contaminant concentration issues, could directly result not only in reduced productivity and economic loss but also in thermal comfort problems (drafts) and 'sick-building' syndrome.[5]

Therefore, a proper HVAC (Heating, Ventilating, Air-conditioning) design, including careful consideration of air vent placement, is a necessity for indoor comfort. An appropriate supply diffuser and return grille can ensure good air circulation in enclosed individual offices, which in turn will result in a more comfortable indoor environment.[6,7] In such design considerations, in addition to features of the HVAC system, engineers must also

[1]Department of Civil and Architectural Engineering and Construction Management, University of Cincinnati, Cincinnati, OH, USA
[2]School of Architecture and Interior Design, University of Cincinnati, Cincinnati, OH, USA

Corresponding author:
Julian (Jialiang) Wang, University of Cincinnati, 765, Baldwin Hall, Cincinnati, 45221 OH, USA.
Email: julian.wang@uc.edu

study building envelope elements such as insulation levels, wall-to-window ratios, surface materials, etc.[8–11]

In particular, exterior windows often comprise a relatively large portion of the building envelope and have relatively lower thermal insulating properties than the opaque components. This means that windows not only provide architectural mechanism to outdoor views and natural light, but they are also one of the most significant building components that could affect overall thermal comfort and heat transfer.[12] Of all building envelope components, a window's inner surface is normally the coldest in winter and hottest in summer. Cold or hot air from the window makes airflow travel across the floor, often causing occupant discomfort. For example, air cools and sinks when warmer room air hits exterior windows during the winter season. The movement of cool air may create cold drafts on the floor, leading to thermal discomfort.

According to the Air Conditioning Contractors of America (ACCA) Manual T,[11] the process for properly locating air vents has four steps: (1) scan the performance data for a size that is capable of delivering the desired airflow to achieve the required air exchange rate (ACH), (2) make sure that the noise level is acceptable, (3) check the flow and (4) check the pressure drop. Although the ACCA Manual T does not recommend particular air vent placement, air supply vents are normally located close to exterior windows, owing to their conventional thermal properties.[12] On the other side, energy use in office buildings is strongly influenced by these exterior window properties. Especially in regard to the air vent placement circumstances, the large contact area and relatively 'high-speed' airflow from vents may accelerate heat transfer between the interior and exterior of the building through windows. We believe, theoretically, traditional vent placement may enhance occupants' thermal comfort conditions, but could undesirably increase heat transfer between the interior and windows, due to high-level thermal convection gained from the airflow. Consequently, the thermal comfort maintained by having the traditional placement of air supply registers close to exterior windows could have a potential to conflict with the energy loss, at least in terms of heat transfer through the windows. Therefore, some literature points out that using high sidewall or ceiling registers without washing the exterior windows could save energy use.[13] However, such vent placement may potentially affect users' thermal comfort due to the high speed and low or high temperatures of conditioned air directly emanating from these vents. This is especially likely in the central zone occupied by users in offices.

This conflict is actually belonging to a popular research question, inspiring a number of studies on how to balance the needs of indoor comfort and energy savings.[3,14] Ideally, the success of a building's design accounts for whether a comfortable indoor environment can be achieved without an increase in energy use.[15] Many researchers have investigated the effect of windows or air vent placement on energy use and comfort.

With regard to the building exterior windows, the window performance is broadly recognised to have a significant impact on human thermal comfort. Arasteh et al.[10] introduced numerous types of windows properties, studying how different windows could affect occupants' satisfaction, health and productivity. Sengupta et al.[16] developed a methodology for quantifying performance of a window regarding human thermal comfort. The results showed that thermal comfort distribution could be significantly affected by the presence of a window. A joint study[17] conducted by changing a window's surface temperature and view factor used an analytical method to study windows and evaluated their effect on indoor comfort. Hassan et al.[18] carried out an investigation on effects of various window combinations on ventilation and thermal comfort in buildings. These researchers considered effects of window location and size ('window-to-wall' ratios) and building orientation relative to prevailing wind speed and direction on the natural ventilation criteria for thermal comfort in a building. The results showed that two adjacent openings for single-side ventilation were not related to window location. Also, two non-adjacent openings (corner and centre-located) were shown to be better than two adjacent openings for single-side ventilation, especially two non-adjacent openings (one far left and one far right).[18]

There have been other studies that attempted to determine how air vents affect air distribution strategies and, in turn, indoor thermal comfort. Chiang et al.[19] analysed diffuser positioning in mechanical ventilation systems and provided guidance for improving the design of radiant cooling ceiling panels. These researchers compared two alternative diffuser positions to the original configuration in order to study how air and temperature distributions could be influenced. Wargocki et al.[20] examined how the air supply rate and condition of the air supply filters could influence environmental perceptions, the intensity of 'sick building' symptoms and operator performance in a call centre. Mijakowski and Sowa[21] proposed humidity-sensitive air inlets in a kindergarten building to improve the performance of passive stack ventilation, but the effect was not sufficient to meet current recommendations of Polish and European standards for indoor environments in newly designed kindergarten buildings. Ghassem et al.[22] found that the height of return air vent from ceiling to floor could influence temperature gradient at the vertical direction, and thermal discomfort could be increased by reducing the height of the return air vent location. Kuo and Chung[23] investigated the effects of air vent location on occupants' thermal comfort in the occupied zone and suggested that the inlet diffuser should be located at the half-upper wall, if a wall-mounted diffuser is necessary. Lam and Chan[24] reported that the outlet vent position could have a great impact on the thermal stratification and annual cooling load in a gymnasium which accordingly would pose an effect on human thermal comfort.

Nevertheless, most studies in the existing literature have separately focused on either a building's exterior windows or air vent placement. The motivation for this research's examination of the correlational impacts of exterior windows and air vent placement on indoor thermal conditions and energy use stemmed from the rapid development of glass materials and fenestration technologies seen in the last decade. Therefore, the key research questions in this research are whether the conventional air vent placement (close to exterior windows) is still necessary to maintain indoor thermal comfort and how energy savings from newly developed and highly insulated windows might offset the possible negative effect on comfort from the central placement of air supply registers (i.e. in the centre of the ceiling) in an office unit.

Even though considerable work has been done on the fundamental issues related to windows such as diffuser position, thermal comfort and energy savings, there is still a lack of research that specifically addresses the role of air vent placement and exterior window properties in indoor thermal environment conditions and the associated energy use. In this research, we performed an extensive computational fluid dynamics (CFD) study of six different combinations of air vent locations and exterior window insulation properties in a typical office unit during the summer and winter seasons. We analysed and compared the indoor thermal conditions and transfer through windows generated by CFD simulations in the 12 design scenarios (six combinations). The conclusions from this work could provide an improved understanding of how thermal comfort and energy issues could change in response to different vent locations and types of building windows. In addition, this research would also shed light on how best to optimize air vent placement and window selection for both energy savings and indoor comfort at the design stage.

Analytical procedure and method

Methodology overview

CFD is a method for developing computer programs that can be used to create simulations and perform studies of natural or engineered fluid dynamics systems.[25] CFD is derived from disciplines of fluid mechanics and heat transfer.[26] Recently, CFD has become an important design tool in engineering, and also an indispensable research tool in architecture.[27] Due to the rapid development of computing capabilities and CFD technology, one can use this technique to visualize and analyse spatial variations in an indoor thermal environment that are attributable to flow and heat transfer from various vent locations and window properties, as well as correlations among these indoor thermal control elements.

A complete CFD analysis consists of three main elements: pre-processor, solver and post-processor. Figure 1 presents the workflow of our research, which also includes the relationships among these three elements in the CFD analysis. First, an office geometry model was built-in Autodesk® CAD based on real geometric information and settings for an office at the University of Cincinnati campus. This information was then imported into Autodesk® CFD 2016. Next, we assigned different materials to every part of the office model (see Table 1). We defined 'Window-low' and 'Window-high' to simulate low and high-insulation windows in the Autodesk® CFD software because these materials were not included in its materials database. In addition, three different air vent placement layouts were assigned to the model to formulate six different combination cases for two different window types. During the simulation and calculation process, we assumed that the airflow inside the office was a stable and incompressible Newtonian fluid that met the Boussinesq approximation. The ceiling, floor and three inner walls were perfectly adiabatic and did not transfer heat. The only way to transfer heat was through the windows to the outside. Thermal radiation was neglected in this study. Based on the governing time-averaged fluid flow and heat transfer calculations, we obtained three types of output: airflow pattern, temperature variation/distribution, predicted mean vote (PMV) and heat transfer through windows. Lastly, we compared these four generated results among the six combinations of windows and air vent layouts in order to understand relationships among window properties and air vent layouts and their impact on heat transfer and indoor thermal comfort.

Room model and boundary conditions

In this study, we simulated a typical office with a dimension of $4.57\ \mathrm{m} \times 3.05\ \mathrm{m} \times 3.05\ \mathrm{m}$ (clear length $\times$ width $\times$ height), following the Cartier coordinate system as shown in Figure 2. The human model, which was the size of an adult man with a seating height of 1.1 m and a shoulder width of 0.5 m, was

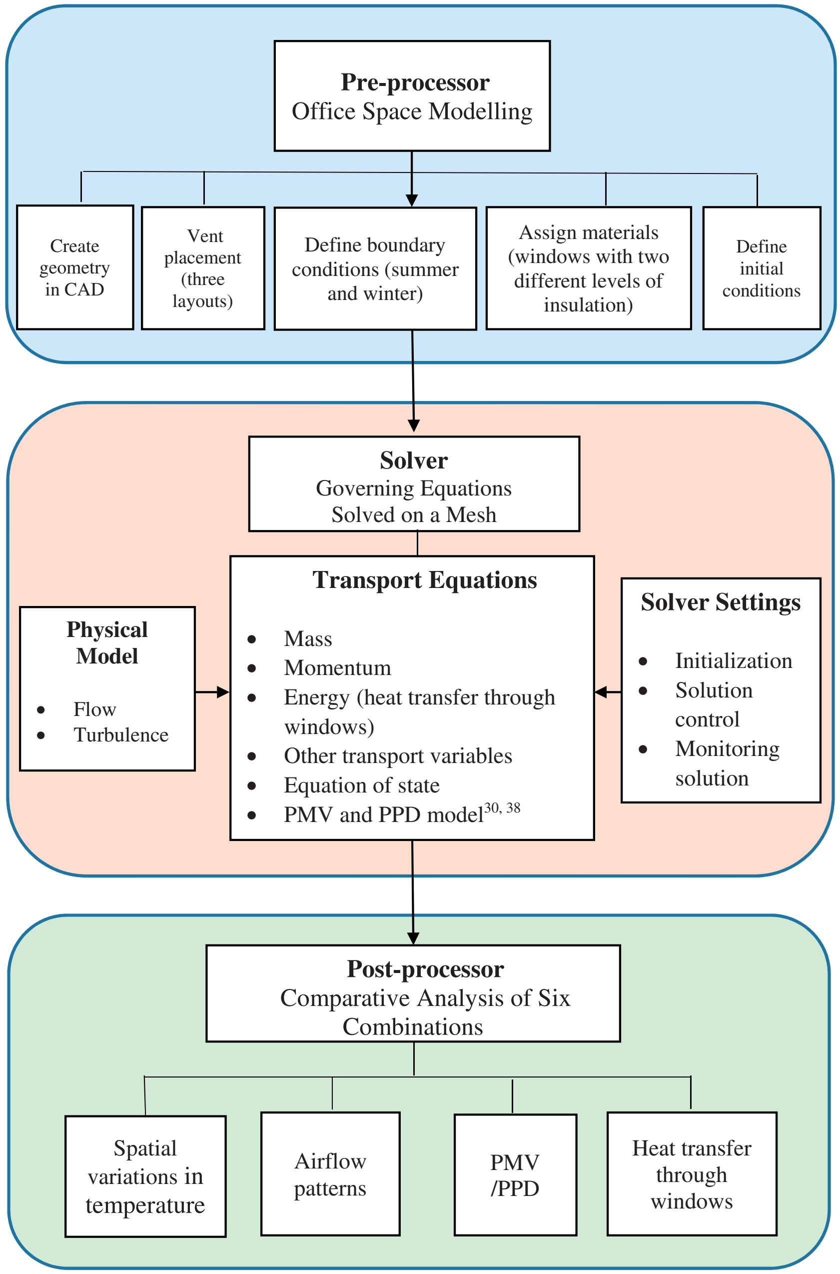

Figure 1. Workflow of this research.

Table 1. Material properties for the private office CFD model.

Parts	Materials	Conductivity (W/m.K)	Density (kg/m^3)	Specific heat (J/kg.K)	Emissivity
Window	Window-low	0.14224	2,700	840	0.8
	Window-high	0.01651	2,700	840	0.01
Human	Human	50	998	4182	0.98
Chair	Wood (soft)	0.12	510	1380	0.8
Desk	Wood (soft)	0.12	510	1380	0.8
Laptop	ABS (polycarbonate)	0.181	1150	1810	0.9
Lamp	ABS (polycarbonate)	0.181	1150	1810	0.9
Room	Air	0.02563	Equation of state	1004	1

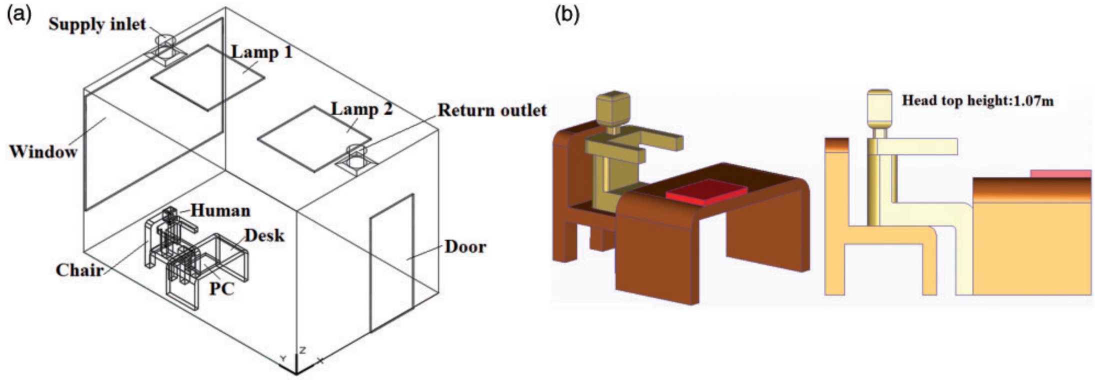

Figure 2. The office model. (a) Details of the office room (the air register placement was different in each of the three representative layouts) and (b) details of the human model, desk, chair and laptop.

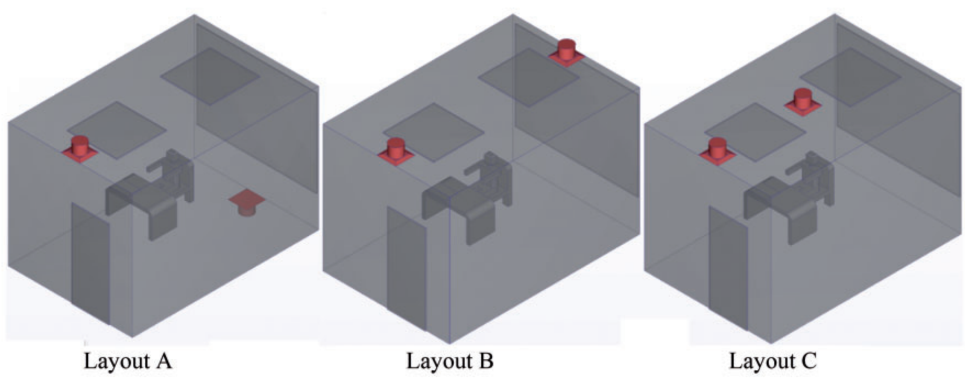

Figure 3. Three typical layouts of air supply registers and return air grilles.

seated near a desk 1.15 m × 0.6 m × 0.6 m (length × width × height), as shown in Figure 3. The door's dimension was 2.18 m × 0.91 m (height × width). The window's dimension was 2.13 m × 2.99 m (height × width). The thickness of both the door and window was 0.0254 m. The radius of the duct was 0.15 m, and the terminal dimension was 0.30 m × 0.30 m. Two lamps (1.22 m × 1.22 m, each) were installed in the ceiling. Regarding the thermal properties, the only thermal transfer boundary was set to the side with the exterior window.

The other surfaces were set as adiabatic materials.

The boundary conditions were set in Autodesk® CFD 2016. In the library of materials, the fluid material was set as a standard default air material with a variable environment. We considered the buoyancy-driven airflow. The k-ε turbulence model was used to solve governing equations. We adopted the coldest and hottest weather in Cincinnati, Ohio, United States for the external boundary conditions for the model. In the summer, the exterior environment temperature was set at 32°C and at −11°C for the winter season. Table 2 shows the relevant thermal boundary conditions.

Table 2. Boundary conditions of the CFD model.

Boundary	Conditions
Supply inlet	Air supply flow rate of 0.047 m^3/s; the air supply temperature was adjusted to keep the office's average air temperature at approximately 25°C
Return inlet	Pressure outlet, gradient zero
Wall	Adiabatic wall
Window	Film coefficient of 5 W/m^2K, reference temperature of 32°C (in summer) and −11°C (in winter)
Human	Generated heat load of 60 W
Laptop	Generated heat load of 60 W
Lamps	Generated heat load of 120 W (60 W each)

Governing equations in the simulation

Fluid flow and heat transfer. The governing equations used in simulation are the conservation of mass, or continuity equation, the conservation of momentum, or Navier-Stokes equations, the conservation of energy, and thermal energy equations. These equations are shown in equations (1) to (4).

The mass conservation equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \bar{V}) = 0 \tag{1}$$

where ρ is the density, $\vec{V}$ is the velocity, $\nabla \cdot \vec{V}$ is the divergence of the velocity.

Navier-Stokes equations

$$\frac{\partial(\rho u)}{\partial t} + \nabla(\rho u \vec{V}) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + \rho f_x \tag{2a}$$

$$\frac{\partial(\rho v)}{\partial t} + \nabla \cdot (\rho v \vec{V}) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \rho f_y \tag{2b}$$

$$\frac{\partial(\rho w)}{\partial t} + \nabla \cdot (\rho w \vec{V}) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho f_z \tag{2c}$$

where u, v, w is the velocity in x, y, z direction, p is the pressure force per unit area, τ_{ij} denotes a stress in the $j-$direction exerted on a plane perpendicular to the $i-$axis; ρf_i denotes the body force on the fluid element acting in the $i-$direction, respectively.

The energy conservation equation

$$\begin{aligned}
&\frac{\partial}{\partial t}\left[\rho(e + \frac{V^2}{2})\right] + \nabla \cdot \left[\rho(e + \frac{V^2}{2}\vec{V})\right] \\
&= \rho \dot{q} + \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) \\
&\quad - \frac{\partial(up)}{\partial x} - \frac{\partial(vp)}{\partial y} - \frac{\partial(wp)}{\partial z} + \frac{\partial(u\tau_{xx})}{\partial x} \\
&\quad + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{zx})}{\partial z} + \frac{\partial(v\tau_{xy})}{\partial x} \\
&\quad + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} + \frac{\partial(w\tau_{xz})}{\partial x} \\
&\quad + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z} + \rho\vec{f} \cdot \vec{V}
\end{aligned} \tag{3}$$

where $(e + \frac{V^2}{2})$ is the total energy; k is the thermal conductivity; T is the local temperature; $\dot{q}$ is the rate of volumetric heat addition per unit mass.

Heat transfer through an exterior window can be calculated by using equation (4)

$$Q = UA(T_{indoor} - T_{outdoor}) \tag{4}$$

where Q is the amount of heat transfer energy from the indoors to the outdoors through a window; u is the overall heat transfer coefficient; A is the window area; T_{indoor} is the indoor temperature; and $T_{outdoor}$ is the outdoor temperature.

Thermal comfort model and settings. Various thermal comfort models have been proposed and developed, such as the Fanger model, the one-node model, the two-node model, the multi-node model, the UC (The University of California) Berkeley comfort model, the local thermal sensation model, the adaptive model, and so forth.[28,29] Each model has its own features and applicability. For example, the Fanger model is based on the body heat balance in the steady-state condition, and the local thermal sensation model considers individual body parts.[29] The Fanger model has been widely used in existing studies and standards and already embedded in the Autodesk CFD software. Based on the assumption of homogenous thermal conditions in this study, we adopted the Fanger model to calculate PMV in this study, as seen in equation (5)

$$\begin{aligned}
PMV = {} & (0.303e^{-0.036M} + 0.028)\{(M - W) \\
& - 3.96 \times 10^{-8} f_{cl}[(t_d + 273)^4 - (t_r + 273)^4] \\
& - f_{cl}h_c(t_{cl} - t_a) - 3.05 \\
& \times [5.73 - 0.007(M - W) - p_a] \\
& - 0.42[(M - W) - 58.15] \\
& - 0.0173M(5.87 - p_a) \\
& - 0.0014M(34 - t_a)\}
\end{aligned} \tag{5}$$

where f_{cl} is the clothing factor $f_{cl} = 1.0 + 0.2I_{cl}$ or $1.05 + 0.1I_{cl}$;

I_{cl} is the clothing insulation [clo];

$$\begin{aligned}
t_{cl} = 35.7 - 0.0275(M - W) - R_{cl}\{ & (M - W) \\
& - 3.05[5.73 - 0.007(M - W) - p_a] \\
& - 0.42[(M - W) - 58.15] \\
& - 0.0173M(5.87 - p_a) - 0.0014M(34 - t_a)\}
\end{aligned}$$

R_{cl} is the thermal insulation from clothing, $R_{cl} = 0.155I_{cl}$;

M is the metabolic rate [W/m^2];

p_a is the vapour pressure of the air [kPa];

t_a is the air temperature [°C];

t_{cl} is the surface temperature of the clothing [°C];

t_r is the mean radiant temperature [°C];

W is the external work;

h_c is the convective heat transfer coefficient, $h_c = 1.2(V)^{1/2}$;

V is the air velocity [m/s].

Thermal comfort depends on the air temperature, humidity, radiant temperature, air velocity, and occupant's metabolic rate and amount of clothing. Each individual would experience these sensations differently based on their physiology and state.[30] Three personal factors – metabolic rate, clothing level and humidity – were set in the CFD model. They are summarized in Table 3.

Six combinations of vent placement and window type

This study investigated an office space with three different configurations of supply inlet and return outlet locations. In Layout A, the air supply register was placed on the floor underneath the exterior window (namely under-the-window), and the air return grille was on the top part of the office, in the centre of the wall near the door. In Layout B, the air supply register was above-the-window; the vent was on the ceiling just above the exterior window and the air return grille was located in the same position as in Layout A. In Layout C, the air supply register was located at the centre of the office ceiling, namely central-ceiling type, and the air return grille was placed in the same position as in Layout A. The locations of the supply and return air vents for these three layouts are shown in Figure 3. We also considered two different window types, those with high and low levels of insulation, which represented traditional single-pane windows with a U-factor of 5.6 W/m^2-K and contemporary triple-pane windows with a U-factor 0.65 W/m^2-K, respectively. There were a total of six combinations of air vent placement and window type and 12 simulation tasks for the summer and winter seasons. Table 4 shows the basic information for these six combinations and 12 simulation tasks.

Table 3. Personal factors in human comfort.

	Summer	Winter
Metabolic rate (W/m^2)	58	58
Clothing (clo)	0.57	1.01
Relative humidity (%)	50	50

Comparative analysis

Analysis of airflow patterns

We used the steady simulation approach in Autodesk® CFD 2016 to observe the overall average room temperature of 25°C. Figures 4 and 5 show the basic vertical airflow patterns under these three different layouts in summer and winter, respectively.

Figures 4 and 5 show airflow patterns for three different layouts in two seasons. From Layout A, we can see that in summer, at the floor level, the airflow emanated uniformly from the supply. The air velocity was almost constant in the room and was generally below 0.14 m/s. The velocity was higher and more uneven in the vicinity of the air supply terminal. This under-the-window type of air register placement did not allow the hot air around the window to fully merge with the cool air coming from the air register. The airflow patterns indicated that in winter the cool air around the window fully and quickly merged with the hot air from the register. There was almost no airflow from the cold side of the window. In contrast, Layout B appeared more effective and complete in regard with the summer airflow mixture. The air velocity around the exterior of the window was, overall, above 0.5 m/s. However, in winter, in the occupant zone, the airflow was clearly unstable and uneven due to incomplete mixing resulting from the above-the-window placement. In particular, with low-insulated windows, the cold air flowed from the window side and was driven to floor level in

Table 4. Simulation tasks for all combinations of vent placement and window type.

Case	Vent placement	Window type (U-factor)	Season
A-SL	Layout A	Low insulation (5.6 W/m^2-K)	Summer
B-SL	Layout B	Low insulation (5.6 W/m^2-K)	Summer
C-SL	Layout C	Low insulation (5.6 W/m^2-K)	Summer
A-SH	Layout A	High insulation (0.65 W/m^2-K)	Summer
B-SH	Layout B	High insulation (0.65 W/m^2-K)	Summer
C-SH	Layout C	High insulation (0.65 W/m^2-K)	Summer
A-WL	Layout A	Low insulation (5.6 W/m^2-K)	Winter
B-WL	Layout B	Low insulation (5.6 W/m^2-K)	Winter
C-WL	Layout C	Low insulation (5.6 W/m^2-K)	Winter
A-WH	Layout A	High insulation (0.65 W/m^2-K)	Winter
B-WH	Layout B	High insulation (0.65 W/m^2-K)	Winter
C-WH	Layout C	High insulation (0.65 W/m^2-K)	Winter

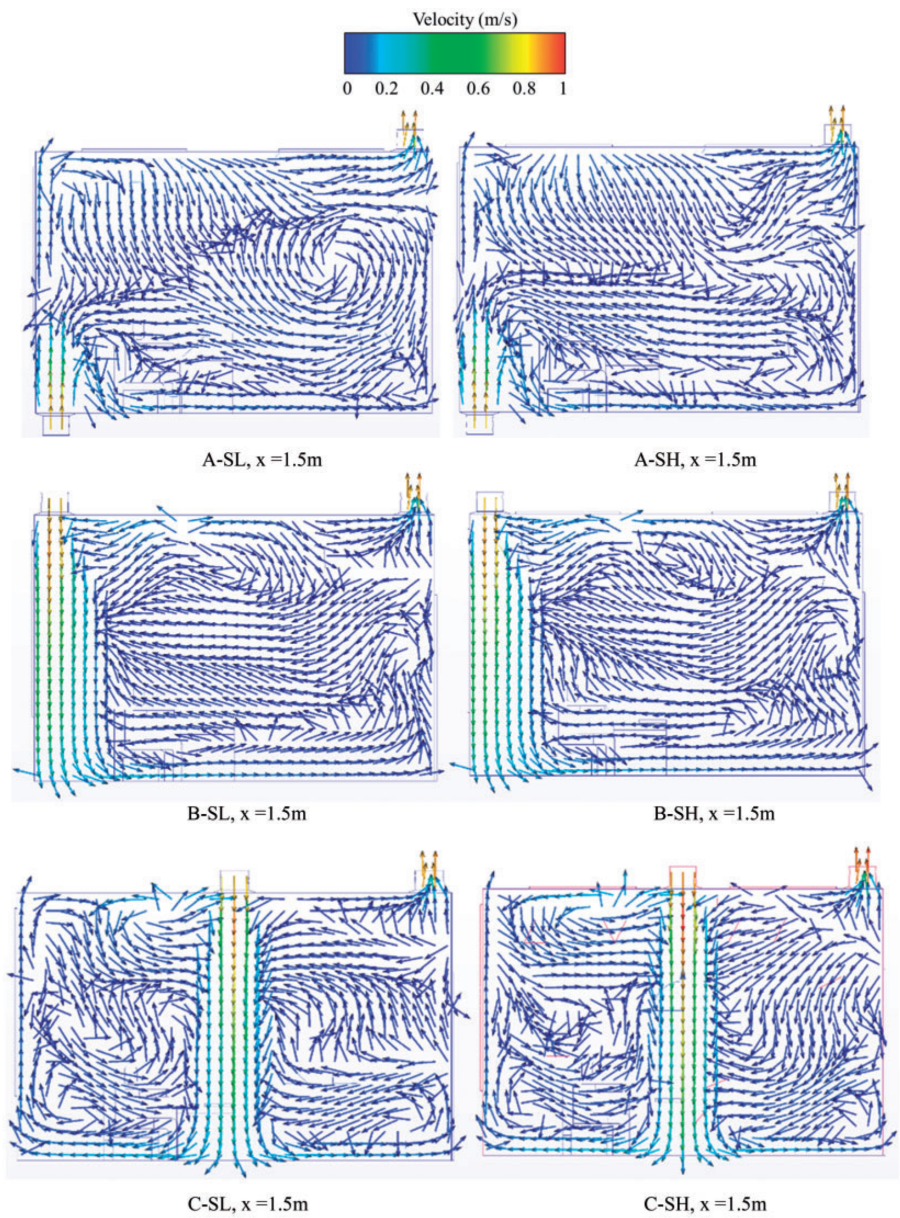

Figure 4. Vertical winter airflow patterns in different layouts.

the user zone. Layout C, the central-ceiling placement, also presented a complete air mixture from the ceiling to the floor, except in the C-WL case. This revealed that some cold airstreams may be moved from the exterior window side of the room to the central user zone. In addition, the air speeds varied from 0.1 m/s to 1 m/s in the user zone, which could lead to some potential discomfort.

To this end, with low-insulated windows, the airflow patterns revealed apparent advantages to using Layout A in winter and Layout B in summer, due to the quicker and more complete airflow mixtures that resulted. This would prevent the adverse effect resulting from hot airflow in summer and cold airflow in winter that would come from the exterior window side of the room. These results also show that Layout C might be effective for the airflow mixture in summer, but apparently not in winter. However, when it came to the three highly insulated window models, the advantages seen in Layout A in winter and Layout B in summer were not distinct from those seen in other layouts. The central-ceiling placement, Layout C, seemed capable of merging airflows for the entire room in cases where the windows were highly insulated.

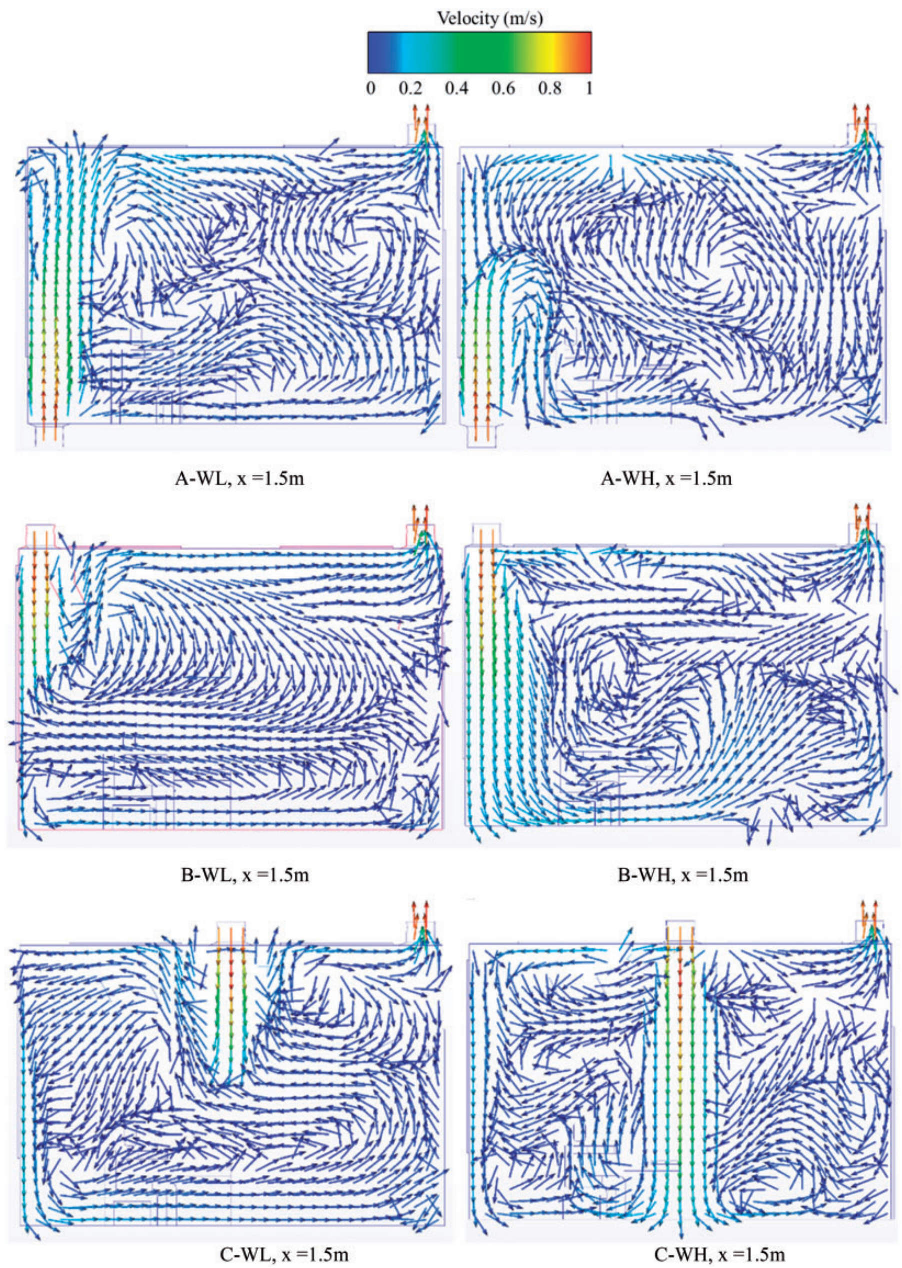

Figure 5. Vertical winter airflow patterns in different layouts.

Analysis of temperature variations

Local thermal discomfort due to vertical air temperature differences has been reported in a number of studies.[30–35] A comprehensive experiments was conducted by Olesen et al.,[35] who reported that 3 K/m should be the limit on the vertical air temperature gradient if thermal discomfort is to be avoided. Another characteristically experimental study was conducted by Wyon and Sandberg.[34,36] They found that the vertical temperature difference should not be more than 4 K/m for there to be an acceptable thermal environment. They also found that when the vertical temperature difference was above 2 K/m, more subjects reported discomfort related to dry eyes.[34]

To avoid local discomfort, ANSI/ASHRAE Standard 55–2013 recommends that the air temperature difference between the 0.1 and 1.7 m height levels should be less than 4 degrees.[37] Figures 6 and 7 present vertical temperature distribution patterns in the vertical central plane of an office in summer and winter, respectively. The average indoor temperature for all cases was approximately 25°C. In order to compare the temperature distributions, we also extracted

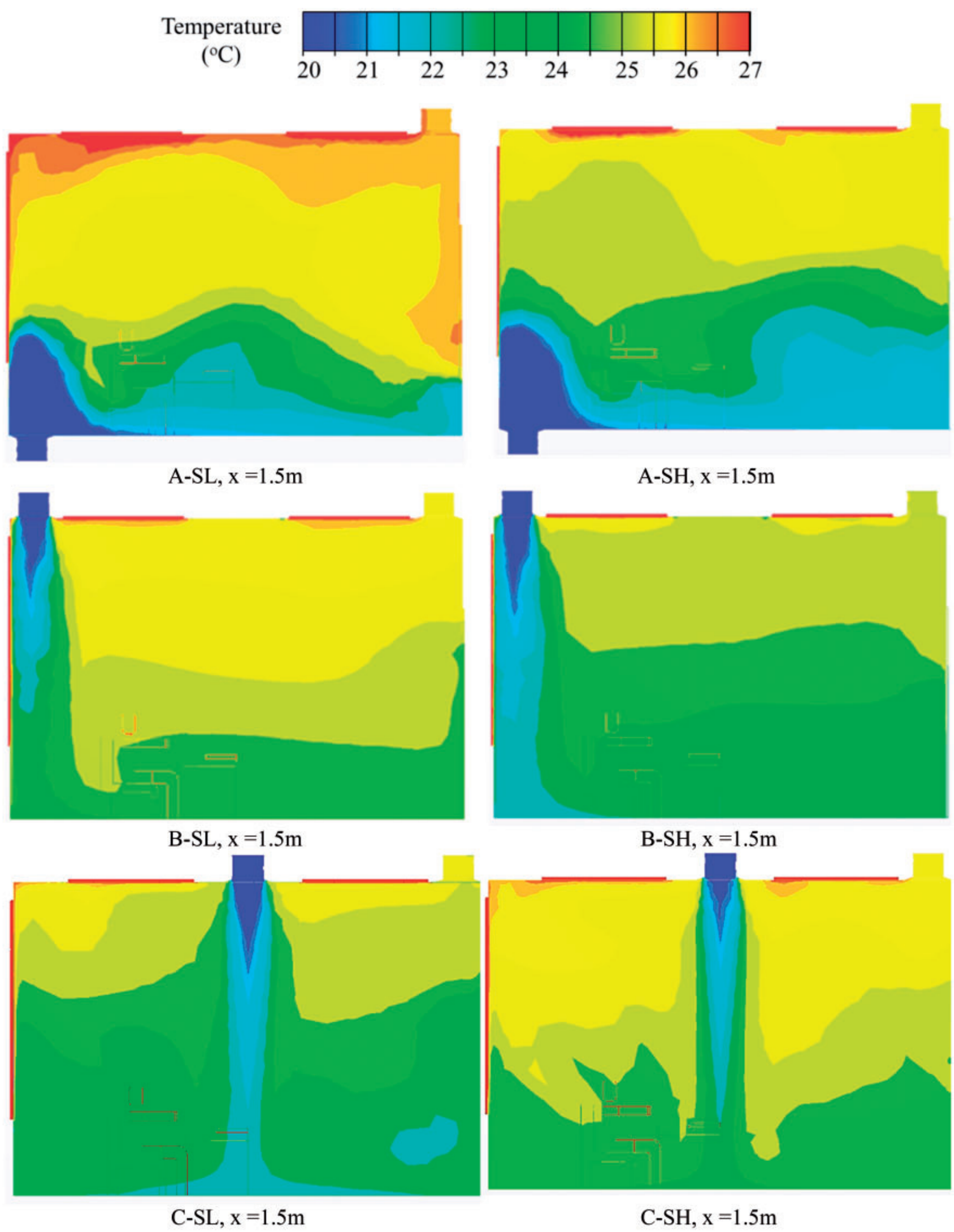

Figure 6. Different cases of vertical summer temperature distributions.

air temperatures at two height levels (1.7 m and 0.1 m) to calculate the vertical temperature difference in the user occupied zone. These results are generalized in Table 5. The acronym 'A-SL' in the table represents Layout A, summer season and low-insulated windows.

In summer (Figure 6), the air temperature distribution showed that Layout B, with the above-the-window type vent placement, provided a uniform vertical temperature gradient (0.7°C) when the window insulation was low. The vertical temperature difference (4.1°C) in Layout A was over the level recommended by ASHRAE 55–2013, so it could have resulted in local thermal discomfort. When it comes to highly insulated windows, the vertical temperature differences in all three cases were reduced, and Layout C presented a similar vertical temperature gradient (0.9°C) to that of Layout B (0.7°C). Layout A's vertical temperature difference was still over the recommended level.

In winter (Figure 7), when the window had low insulation, the air temperature distribution showed that Layout A, with its under-the-window type of vent placement, provided more uniform thermal conditions (a 1.8°C vertical temperature difference) than the other two layouts. Layout C had the highest temperature gradient (4.19°C), and both Layouts B and C generated high vertical temperature differences that were over the

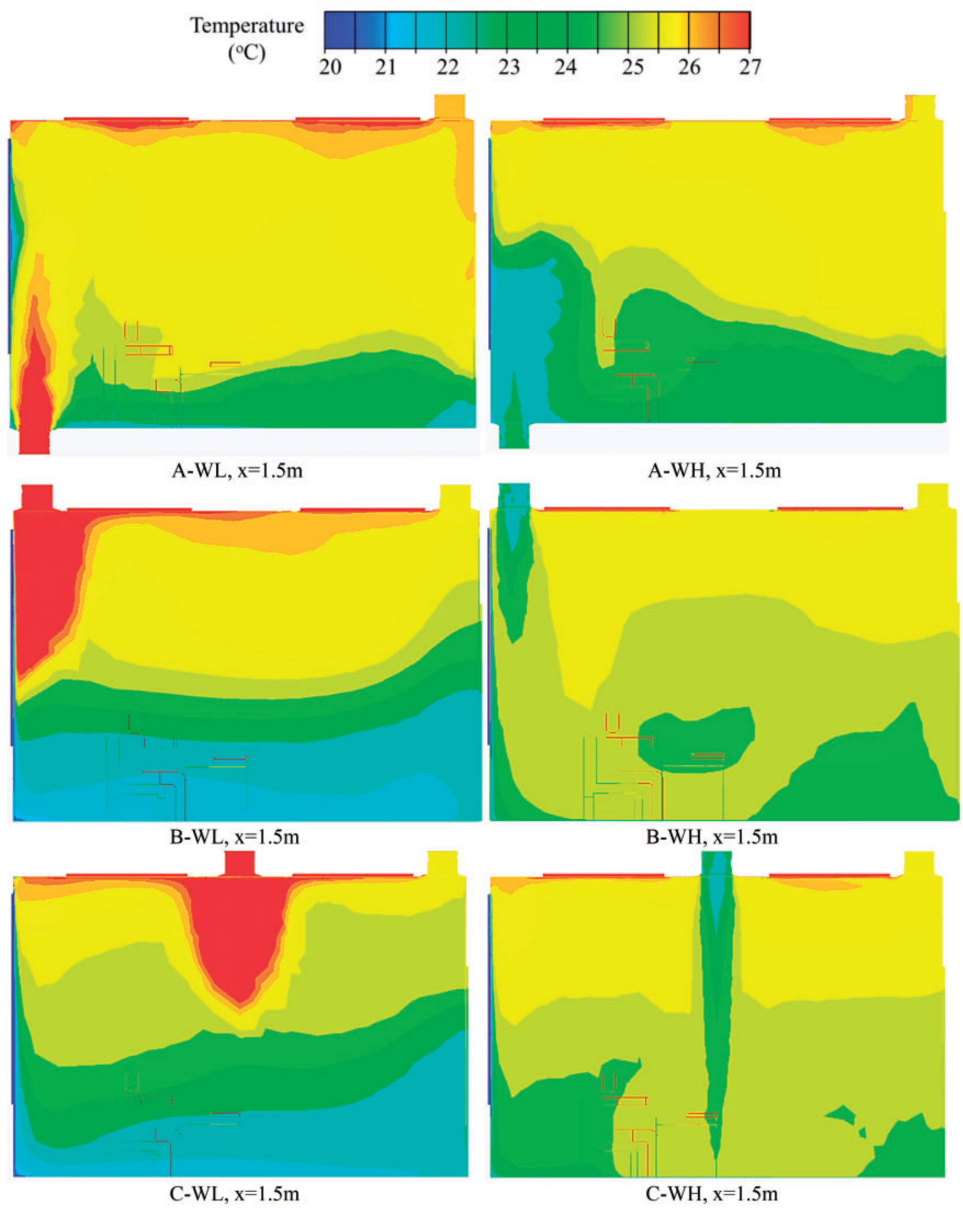

Figure 7. Different cases of vertical winter temperature distributions.

Table 5. Vertical temperature difference and average temperature in the occupant zone.

	A-SL	B-SL	C-SL	A-SH	B-SH	C-SH	A-WL	B-WL	C-WL	A-WH	B-WH	C-WH
Vertical temperature difference 1.7/0.1 m	4.1	0.8	1.3	3.4	0.8	0.9	1.8	4.0	4.2	1.1	2.6	1.6

ASHRAE 55 recommended level. Conversely, comparing the models with highly insulated windows indicated that all three layouts provided more even local thermal conditions. Also, the distinction and advantages of Layout A were attenuated. Layout C also performed well in this situation. These comparisons reveal the consistent effects on vertical temperature differences and airflow patterns that result from combining window properties and air vent layouts. With highly insulated windows, although Layouts A and B performed slightly better in winter and summer, respectively, the central-ceiling type of air vent placement would serve as an acceptable alternative solution in terms of vertical temperature variations.

Analysis of PMV-PPD values

PMV was divided into a seven-point thermal sensation scale, according to the thermal perception of human bodies (+3 hot, +2 warm, +1 slightly warm, 0 neutral, −1 slightly cool, −2 cool, −3 cold).[30] From the simulation scenes, we extracted the PMV indices for two points at height levels of 0.1 m and 1.7 m. We then obtained the average PMV values. To predict the percentile of occupants who would be thermally dissatisfied in a given thermal environment, the predicted percentage of dissatisfied (PPD) index[38] was calculated using equation (6) and thoroughly analysed. The recommended acceptable PPD range from ANSI/ASHRAE Standard 55–2013 is less than 10% of persons dissatisfied with the interior space.[37] Table 6 summarizes these two key thermal comfort indicators obtained from this simulation study.

$$PPD = 100 - 95 \times \ e^{-(0.03353 \cdot PMV^4 + 0.2179 \cdot PMV^2)} \tag{6}$$

The PMV and PPD values in Table 6 demonstrate the advantages of using Layout A in winter and Layout B in summer. They offer the most appropriate PMV indices and the lowest PPD values in cases of low-insulated windows. As the window insulation increased, the differences among the Layouts decreased in terms of percentiles of occupants who were thermally dissatisfied. The PPD values from Layout C were only about 0.5% and 2% higher in summer and winter, respectively, than the best thermal comfort indicators but still within the recommended range 10%.

Analysis of heat transfer through windows

Except for exterior windows, all other model boundaries in this study were set as adiabatic, so that the output of heat transfer by the CFD simulation was only through windows. Using this method, we summarized the heat transfer through the exterior window at eventuate state (about 25°C as the average room temperature) in summer and winter in Tables 7 and 8, respectively. In these Tables, we also documented the window's average inner and outer surface temperatures, which could also reveal the thermal insulation abilities of different windows.

With regard to the impact of window insulation in the summer season, compared to low-insulated windows (Cases A-SL, B-SL, and C-SL), highly insulated windows (Cases A-SH, B-SH, and C-SH) significantly reduced heat gain to a rate of 67.1%, 67.2% and 66.8% (an average of 67%), respectively. This was as expected. Similarly, in the winter season, Cases A-WH, B-WH and C-WH achieved significant heat loss reductions with highly insulated windows, approximately

Table 6. PMV-PPD indicators.

	A-SL	B-SL	C-SL	A-SH	B-SH	C-SH	A-WL	B-WL	C-WL	A-WH	B-WH	C-WH
Average PMV	0.59	0.39	0.67	0.45	0.41	−0.44	−0.07	−0.42	−0.71	−0.15	−0.17	−0.35
Average PPD (%)	12.3	8.2	14.4	9.2	8.5	9.0	5.1	8.7	15.6	5.5	5.6	7.6

PMV: predicted mean vote; PPD: predicted percentage of dissatisfied.

Table 7. Heat gain through exterior windows in summer.

Case	A-SL	B-SL	C-SL	A-SH	B-SH	C-SH
Inner/outer window surface temperature (°C)	27.70/29.68	27.66/29.75	28.07/29.99	25.24/31.24	25.59/31.26	26.13/31.33
Heat gain (W)	73.0	70.7	63.3	24.0	23.2	21.0

Table 8. Heat loss through exterior windows in winter.

Case	A-WL	B-WL	C-WL	A-WH	B-WH	C-WH
Inner/outer window surface temperature (°C)	12.1/1.2	10.79/0.42	10.2/0.04	19.87/−7.46	19.09/−7.56	17.85/−7.70
Heat loss (W)	382.8	380.6	347.1	111.4	108.3	103.6

70.9%, 71.6% and 70% (an average of 70.8%) over Cases A-WL, B-WL and C-WL with their low-insulated windows, respectively. This can be explained via the approximately eightfold increase (i.e. from 5.6 W/m^2-K to 0.65 W/m^2-K) in window insulation. Therefore, an increase in window insulation could greatly improve energy efficiency by an average of 65.4% in summer and 70.8% in winter, based on a change from low-insulation windows.

Conversely, Figure 8 shows a comparison of the different air vent layouts under the same level of window insulation. This reveals that placing air supply vents close to the exterior window (Layouts A and B) resulted in more heat loss/gain compared to the central-ceiling placement (Layout C). With low-insulation windows, the heat transfer difference between Layouts A/B and Layout C was around 13.5% in summer and 10% in winter. With high-insulation windows, the heat transfer difference between Layouts A/B and Layout C was slightly attenuated to 12.3% in summer and 6% in winter. Thus, regardless of the window's insulating ability, air vent placement plays an important role in determining the amount of heat transferred through the window. In other words, this demonstrates that a central-ceiling type of placement can be deemed energy efficient, even under highly insulated exterior window conditions.

Discussion

From a comparative analysis of four aspects – airflow pattern, temperature distribution, thermal comfort index and heat transfer – the above-the-window air supply register in summer and under-the-window air supply register in winter would serve as the best solution for maintaining thermal comfort in both single and triple-pane window conditions, even though such advantages were not significant in cases of highly insulated windows. Meanwhile, a central-ceiling air vent placement might save a certain amount of heat loss/gain through exterior windows. This presents a trade-off between thermal comfort and energy savings. The goal is to achieve the lowest energy use possible while still maintaining an acceptable 10% PPD.

Figures 9 and 10 show the average temperature difference, PPD and the sum of energy use in these two test seasons. The green cube highlights the key recommended ASHRAE levels for vertical temperature and PPD. The points falling within this zone indicate that the situation complied with the recommended comfort levels. Therefore, in these two figures, the closer to the coordinate origin (the red dot in the Figure) the points are located, the more preferable the heat transfer values and thermal comfort levels. Figure 9 illustrates that Cases B-SL, A-WL and B-WL were in the recommended zone, and Layout B was the most appropriate option for both energy use and PPD objectives in cases of low-insulated windows.

Figure 10, on the other hand, demonstrates that almost all of the points fell into the recommended zone. Compared with other layouts, the average PPD in Layout C showed an increase to 8.2%, over the 7.4% of Layout A or 7% of Layout B. The heat transfer amount was reduced from 135.4 W in Layout A and 131.5 W in Layout B to 124.6 W in Layout C, representing an approximately 6% to 9% energy savings. Therefore, in this case study, we identified Layout C as the optimal air vent placement type caused by its relatively large energy savings and minor negative effects on thermal comfort. However, this simplified

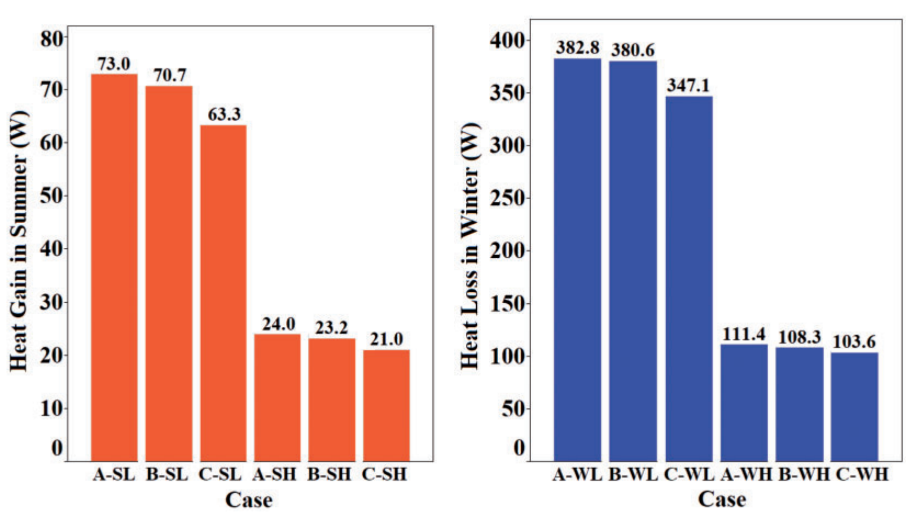

Figure 8. Heat gain/loss through an exterior window.

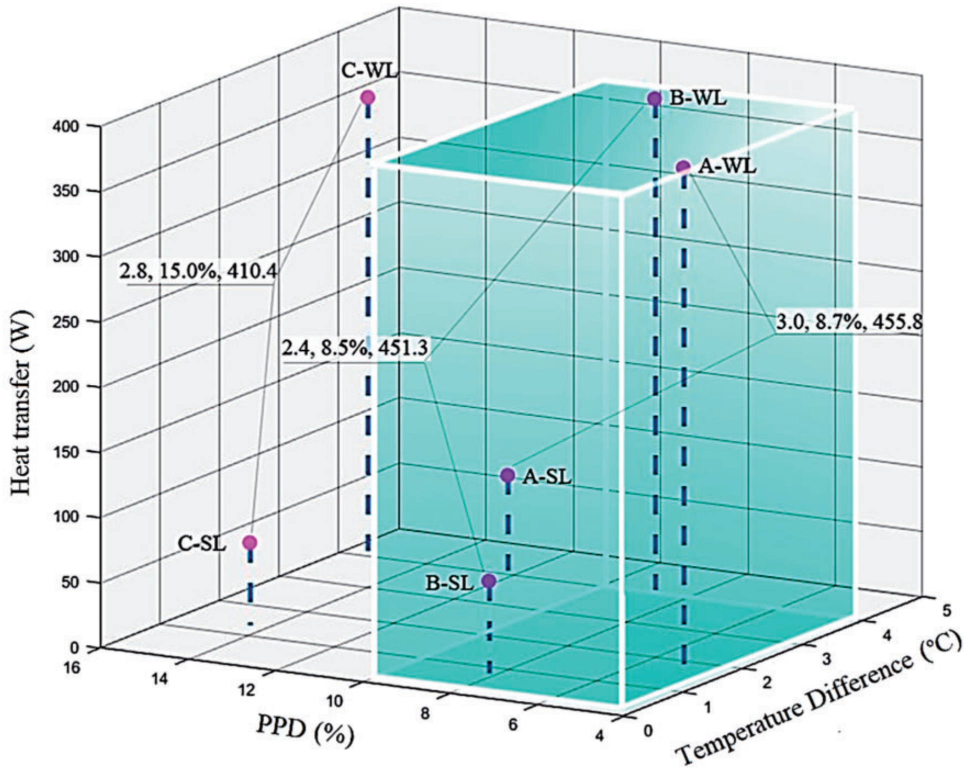

Figure 9. Performances of different air vent placements with low-insulated windows.

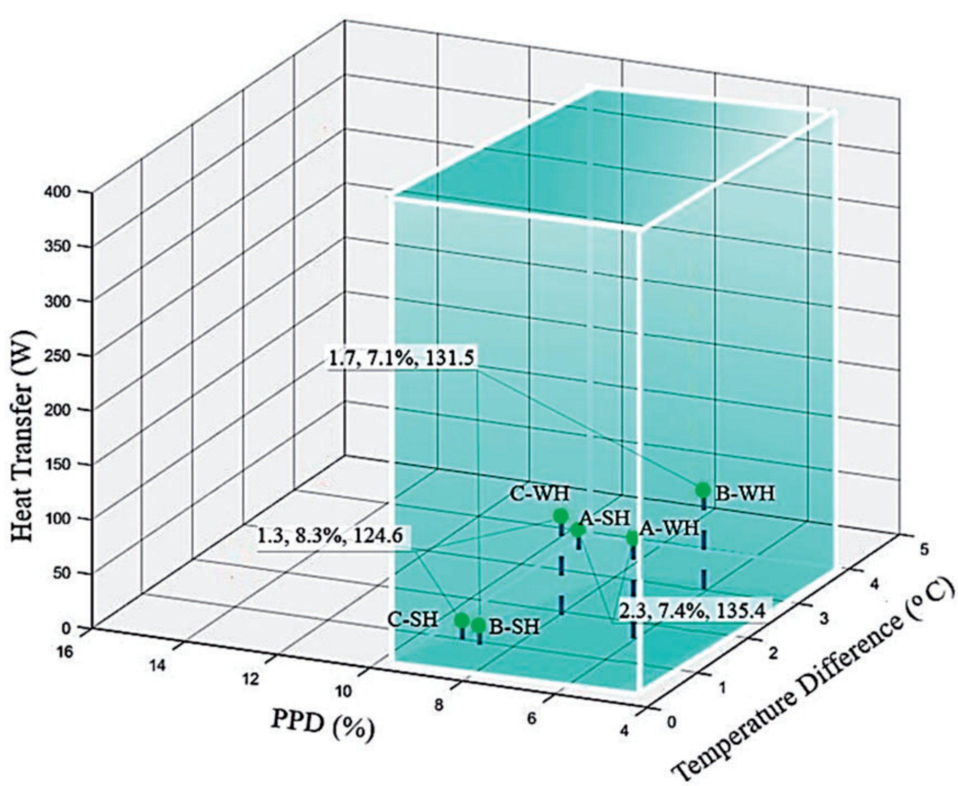

Figure 10. Performances of different air vent placements with highly insulated windows.

trade-off stemmed from the strong energy efficiency of the triple-pane windows and adiabatic boundary wall properties hypothesized in this work. If the energy savings from other types of windows cannot achieve this level, a comparative analysis would be more complex, and further assessment of energy use and indoor comfort would be needed. In addition, many studies have pointed out that user well-being and productivity are significantly related to indoor thermal comfort.[38–41] A PPD change from 6% to 10% might result in a 4.2% increase in employees' maximum loss of performance in their workspaces.[42] Given the considerably high cost of personnel in an office environment, a 1% loss of performance could lead to a significant loss in profit for the organization. Therefore, the extra energy used to enhance indoor comfort could still be regarded as a cost-effective strategy from an investment perspective.

Conclusions

This research was conducted to examine airflow patterns, temperature distributions, PMV/PPD thermal comfort indices and the corresponding heat transfers through exterior windows in a private office during the summer and winter seasons. The objective was to understand the effects of different air vent placements and window insulation on indoor thermal conditions and energy use, as well as to quantitatively assess the trade-offs between indoor comfort and energy use saving with a central-ceiling air vent placement. To this end, extensive CFD simulations and related comparative analyses were conducted on six representative combinations of air vent layouts and window insulation levels. The results indicate that:

- With low-insulated windows (single-pane type), using above-the-window air supply registers in summer and under-the-window registers in winter were shown to have apparent advantages, providing a more complete air mixture, uniform vertical temperature distribution, and acceptable thermal comfort (PMV/PPD index). Even though the heat transfer could be considerably reduced by using a central-ceiling type of air register, the resulting thermal comfort performance was outside of the recommended limits.
- With highly insulated (triple-pane type) windows, the advantages of Layout A in winter and Layout B in summer were not especially distinct. The central-ceiling placement in Layout C seemed to be capable of merging airflows for the entire room, providing appropriate indoor thermal conditions and significant savings in energy use. However, the percentage of occupants dissatisfied with the level of thermal comfort, as compared with the best performances achieved in Layouts A and B, was slightly higher but still within the recommended limits.
- This simulation was limited to the selected window types and office unit features. Considering the variety of window properties, it is possible to find significant complexity in trade-offs between energy savings and thermal comfort. Given the substantial connection between user productivity and indoor thermal comfort, additional investment in improved indoor environments will likely be regarded as acceptable.

The discoveries from this research provide an improved understanding of how thermal comfort and energy issues change in response to different vent locations and types of building windows. This work may also shed light on how best to optimize air vent placement and window design. Moreover, it is becoming increasingly important to improve user satisfaction and associated productivity, especially in office settings. The inter- and intra-relationships between air vent layouts and building window properties present trade-off issues between energy savings by retrofitting and/or HVAC upgrades and financial advantages by improving user satisfaction on indoor environment. At this point, this work presents a quantitatively assessment framework using simulation programs for such trade-off issue.

U-factor is the only window property considered and manipulated in this study, so such parametric study is not able to reflect the effect on energy use and indoor environment by the other window properties, such as solar heat gain coefficient (SHGC) which may play more important roles than conductivity in certain scenarios (for instance south-facing windows in daytime). In the platform of Autodesk CFD, only the 'Solar Window' was able to involve SHGC in the calculation and simulation. However, the key drawback of the current version is that the heat transfer values could not be extracted when it comes to the 'Solar Window' type. Therefore, in this study, regular window construction elements (without the SHGC parameter) were used instead because heat transfer values through windows were readily accessible. This limitation is expected to be resolved in our future work using customized plug-ins or different programs to explore comprehensive parametric controls on window properties including SHGC, emissivity, specific heat, and so forth. In particular, our future study will address the effect of thermal radiation on interior window surfaces with various emissivity levels and specific heat properties of boundary materials.

In addition, this study is at the preliminary stage for a large research scope about the combined effect between building envelopes and indoor system settings. At this stage, only six combinations with three variables (season, air vent location, and window insulation) were involved. Upon this simulation framework, other variables, such as window SHGC, indoor humidity, outdoor temperature, system cubic feet per minute (CFM), can be integrated into the research in the future, and then the optimization techniques can be further investigated and applied to explore the optimal settings upon the needs.

Author's contribution

All authors contributed in the preparation of this manuscript equally.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by The Photothermal Effects of Iron Oxide Nanoparticles on Energy Efficient Windows (a National Science Foundation grant: CMMI-1635089).

References

1. ASHRAE 90.1. *Energy standard for buildings except low-rise residential buildings*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2013.
2. Wagner A, Gossauer E, Moosmann C, Gropp T and Leonhart R. Thermal comfort and workplace occupant satisfaction – results of field studies in German low energy office buildings. *Energy Build* 2007; 39: 758–769.
3. Azizpour F, Moghimi S, Salleh E, Mat S, Lim CH and Sopian K. Thermal comfort assessment of large-scale hospitals in tropical climates: a case study of University Kebangsaan Malaysia Medical Centre (UKMMC). *Energy Build* 2013; 64: 317–322.
4. Kosonen R and Tan F. Assessment of productivity loss in air-conditioned buildings using PMV index. *Energy Build* 2004; 36: 987–993.
5. Yuan X, Chen Q and Glicksman LR. A critical review of displacement ventilation. *ASHRAE Trans* 1998; 104: 78–90.
6. Yu Z, Zhang WL and Fang TY. Impact of building orientation and window-wall ratio on the office building energy consumption. *Appl Mech Mater* 2013; 409–410: 606–611.
7. Moon JW, Chang JD and Kim S. Determining adaptability performance of artificial neural network-based thermal control logics for envelope conditions in residential buildings. *Energies* 2013; 6: 3548–3570.
8. Vanhoutteghem L, Skarning GCJ, Hviid CA and Svendsen S. Impact of façade window design on energy, daylighting and thermal comfort in nearly zero-energy houses. *Energy Build* 2015; 102: 149–156.
9. Ahn B-L, Kim J-H, Jang C-Y, Leigh S-B and Jeong H. Window retrofit strategy for energy saving in existing residences with different thermal characteristics and window sizes. *Build Serv Eng Res Technol*.2016; 37: 18–32.
10. Arasteh D, Carmody J, Lee ES, Selkowitz S and Willmert T. *Window systems for high performance buildings*. New York, NY: Norton & Company, 2004.
11. Hank R. *Manual T: air distribution basics for residential and small commercial buildings*. New York, NY: Air Conditioning Contractors of America, 2015.
12. Duan Q, Wang J and Zhao H. Airflow pattern and thermal comfort in winter by different combinations of air distribution strategies and window types in an office unit. In: *The passive low energy architecture conference* (PLEA), Edinburgh, UK: NCEUB Publisher, 2017, pp. 1061–1068.
13. Cho J, Lim T and Kim BS. Measurements and predictions of the air distribution systems in high compute density (Internet) data centers. *Energy Build* 2009; 41: 1107–1115.
14. Nicol JF and Humphreys MA. Adaptive thermal comfort and sustainable thermal standards for buildings. *Energy Build* 2002; 34: 563–572.
15. Bessoudo M, Tzempelikos A, Athienitis AK and Zmeureanu R. Indoor thermal environmental conditions near glazed facades with shading devices – part I: experiments and building thermal model. *Build Environ* 2010; 45: 2506–2516.
16. Sengupta J, Chapman KS and Keshavarz A. Window performance for human thermal comfort. *ASHRAE Trans* 2005; 111: 254–275.
17. Huizenga C, Zhang H, Mattelaer P, Yu T, Arens EA and Lyons P. *Window performance for human thermal comfort*. Berkeley, USA: Center for the Built Environment, University of California, 2006.
18. Hassan MA, Guirguis NM, Shaalan MR and El-Shazly KM. Investigation of effects of window combinations on ventilation characteristics for thermal comfort in buildings. *Desalination* 2007; 209(1–3 SPEC. ISS.): 251–260.
19. Chiang W-H, Wang C-Y and Huang J-S. Evaluation of cooling ceiling and mechanical ventilation systems on thermal comfort using CFD study in an office for subtropical region. *Build Environ* 2012; 48: 113–127.
20. Wargocki P, Wyon DP and Fanger PO. The performance and subjective responses of call-center operators with new and used supply air filters at two outdoor air supply rates. *Indoor Air* 2004; 14: 7–16.
21. Mijakowski M and Sowa J. An attempt to improve indoor environment by installing humidity-sensitive air inlets in a naturally ventilated kindergarten building. *Build Environ* 2017; 111: 180–191.
22. Heidarinejad G, Fathollahzadeh MH and Pasdarshahri H. Effects of return air vent height on energy consumption, thermal comfort conditions and indoor air quality in an under floor air distribution system. *Energy Build* 2015; 97: 155–161.

23. Kuo J-Y and Chung K-C. The effect of diffuser's location on thermal comfort analysis with different air distribution strategies. *J Build Phys* 1999; 22: 208–229.
24. Lam JC and Chan AL. CFD analysis and energy simulation of a gymnasium. *Build Environ* 2001; 36: 351–358.
25. Sharma A. *Introduction to CFD: development, application, and analysis*. Chichester, UK: John Wiley & Sons, Ltd., 2017, pp.23–39.
26. Tu J, Yeoh GH and Liu C. *Computational fluid dynamics: a practical approach*. Oxford: Butterworth-Heinemann, 2012.
27. Blazek J. *Computational fluid dynamics: principles and applications*. Oxford: Butterworth-Heinemann, 2015.
28. Cheng Y, Niu J and Gao N. Thermal comfort models: a review and numerical investigation. *Build Environ* 2012; 47:13–22.
29. Djongyang N, Tchinda R and Njomo D. Thermal comfort: a review paper. *Renewable Sustainable Energy Rev* 2010; 14: 26–40.
30. Olesen BW and Brager GS. A better way to predict comfort. *ASHRAE J* 2004; 8: 20–26.
31. Wu X, Fang L, Olesen BW, Zhao J and Wang F. Effect of supply air temperature on air distribution in a room with radiant heating and mechanical ventilation. *ASHRAE Trans* 2017; 121: 206–213.
32. Gil-Lopez T, Galvez-Huerta MA, O'donohoe PG, Castejon-Navas J and Dieguez-Elizondo PM. Analysis of the influence of the return position in the vertical temperature gradient in displacement ventilation systems for large halls. *Energy Build* 2017; 140: 371–379.
33. Melikov AK and Nielsen JB. Local thermal discomfort due to draft and vertical temperature difference in rooms with displacement ventilation. *ASHRAE Trans* 1989; 95: 1–8.
34. Wvon DP and Sandberg M. Discomfort due to vertical thermal gradients. *Indoor Air* 1996; 6: 48–54.
35. Olesen BW, Scholer M and Fanger PO. Discomfort caused by vertical air temperature differences. *Indoor Climate* 1979; 36: 561–578.
36. Yu WJ. Local discomfort caused by draft perception in a space served by displacement ventilation system in the tropics. *Indoor Built Environ* 2006; 15: 225–233.
37. ANSI-ASHRAE Standard 55-2013. *Thermal environmental conditions for human occupancy*. Atlanda, USA: American Society of Heating, Refrigerating and Air-conditioning Engineers, 2013.
38. ISO7730:2005. *Ergonomics of the thermal environment - Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria*. Geneva, Switzerland: International Organization for Standardization, 2005.
39. Clements-Croome DJ. Work performance, productivity and indoor air. *J Work Environ Health Suppl* 2008; 4: 69–78.
40. Roelofsen P. The impact of office environments on employee performance: the design of the workplace as a strategy for productivity enhancement. *J Facilities Manage* 2002; 1: 247–264.
41. Wyon DP and Wargocki P. How indoor environment affects performance. *ASHRAE J* 2013; 3: 46–52.
42. Vimalanathan K and Babu TR. The effect of indoor office environment on the work performance, health and well-being of office workers. *J Environ Health Sci Eng* 2014; 12: 113.