A modelling method for large-scale open spaces orientated toward coordinated control of multiple air-terminal units

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Abstract

The temperature distribution is always assumed to be homogeneous in a traditional single-input-single-output (SISO) air conditioning control strategy. However, the airflow inside is more complicated and unpredictable. This study proposes a zonal temperature control strategy with a thermal coupling effect integrated for air-conditioned large-scale open spaces. The target space was split into several subzones based on the minimum controllable air terminal units in the proposed method, and each zone can be controlled to its own set-point while considering the thermal coupling effect from its adjacent zones. A numerical method resorting to computational fluid dynamics was presented to obtain the heat transfer coefficients (HTCs) under different air supply scenarios. The relationship between heat transfer coefficient and zonal temperature difference was linearized. Thus, currently available zonal models in popular software can be used to simulate the dynamic response of temperatures in large-scale indoor open spaces. Case studies showed that the introduction of HTCs across the adjacent zones was capable of enhancing the precision of temperature control of large-scale open spaces. It could satisfy the temperature requirements of different zones, improve thermal comfort and at least 11% of energy saving can be achieved by comparing with the conventional control strategy.

Keywords

large-scale open space; zonal temperature control; CFD; thermal coupling; heat transfer coefficient; TRNSYS

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1 Introduction

In a humid subtropical area, to condition a large-scale open (indoor) space, such as shopping malls, theatres and sports centers, has a large energy footprint. Compared with a small space, such as a personal office, the unique feature of the large-scale open space is that a good mixture of air inside is difficult to achieve. The air movement inside the space is influenced by supply air diffusers, partially or randomly distributed occupants (Gao et al. 2021a; Wang et al. 2021; Wang et al. 2022), heat transfer through building enclosures, and so on. It results in an inhomogeneous temperature distribution both in horizontal and vertical direction (Huang et al. 2007; Lu et al. 2019; Lan et al. 2022). The traditional control strategies are incapable of dealing with this problem properly, merely at the cost of massive energy use trying to maintain overall indoor thermal comfort. Thus, it becomes a major concern to select appropriate simulation tools to fast predict the airflow field as well as indoor thermal environment for the large spaces, and realize a superior control strategy that adapts to the dynamic features of the space. The indoor dynamic temperature distributions and airflows are required to realize a better control for indoor thermal conditions. There are two established methods to predict indoor airflow as well as contaminant distribution: microscopic and macroscopic models. In specific, microscopic models uses computational fluid dynamics (CFD) to iterate the Navier-Stokes equations to obtain the values of all relevant parameters with a high degree of resolution and accuracy, but it is time-consuming and not suitable for the dynamic
The equation can be the momentum conservation equation from Navier-Stokes control strategy designs. To reduce the computing time, temperature information and accuracy for simulations and the room's division into several air zones gives enough laws in each control volume (Stewart and Ren 2006). Thus, a single room into several subzones (control volumes or flow paths) between those rooms (and with the outside air). The macroscopic models mainly include multizone and zonal models. Multizone models require the user to identify and describe all the rooms of interest and the links (e.g., flow paths) between those rooms (and with the outside air). It neglects the air movement inside the room, and users prefer to apply this model to predict annual energy consumption for the whole building. The zonal model was first proposed in 1970 (Lebrun 1971), which divides a single room into several subzones (control volumes or cells), it is supposed to be well-mixed in each control volume, both the mass and energy satisfy the conservation laws in each control volume (Stewart and Ren 2006). Thus, the room’s division into several air zones gives enough temperature information and accuracy for simulations and control strategy designs. To reduce the computing time, the momentum conservation equation from Navier-Stokes equation can be degraded to simplified mass balance equation with a simple power law model. The energy balance equation is also taken from the Navier-Stokes energy conservation equation but ignoring the effect of conduction terms and viscous dissipation terms (Megri and Yu 2015). By solving the above differential equations, the air mass flow rates and air temperatures inside a given room can be determined. The computing time could be reduced dramatically in this way, presenting a much simpler and easy-to-use dynamic model. The development and application of zonal modeling methods have been comprehensively reviewed in literature (Megri et al. 2005; Megri and Haghighat 2007; Yu et al. 2019; Lu et al. 2020). In general, the accuracy and efficiency of zonal model can be balanced well, and more applications should have imposed attention particularly in dynamic control of large spaces.

In the single-zone control mode, the space temperature is usually seemed to be identical (Kintner-Meyer 2005). While in the multi-control mode, each zone has its own representative temperatures and control configurations. The single-zone control mode is relatively simple, and currently available software, such as TRNSYS and EnergyPlus, provides tools and simulation models to deal with the temperature dynamics and can be used to investigate HVAC control in a transient manner. However, as the whole large space is considered as one single node (such as in a building resistance-capacitance model), those tools and simulation models cannot precisely predict the detailed indoor temperature distribution. Thus, the uneven distribution of temperature issues cannot be flexibly handled solely with the single-zone control mode. The indoor space may suffer from over-cooling in some area while under-cooling in other regions simply due to the uneven distributed occupants inside (Zhou et al. 2015). Thermal comfort cannot be guaranteed, and it may also lead to energy waste. Similar to zonal model, the target space is split into a number of subzones based on the

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**List of symbols**

- \(a_{ij}\) heat transfer from \(j^{th}\) zone to \(i^{th}\) zone (W/K)
- \(d_{ij}\) heat transfer from \(i^{th}\) zone to \(j^{th}\) zone (W/K)
- \(\epsilon_{th}\) heat transfer through the interface of a certain controlled volume \(P\) (W/K)
- \(c_p\) thermal capacitance of the air inside the zone (W/K)
- \(b\) disturbance on the heat transfer from internal and external (W)
- \(c_s\) specific heat capacity of air (J/(kg·°C))
- \(C_p\) specific heat capacity (J/(kg·K))
- \(D\) diffusion conductance (kg/(m²·s))
- \(HTC\) heat transfer coefficient (W/K)
- \(i, j\) superscript and subscript refer to \(i^{th}\) or \(j^{th}\) zone
- \(h, k, l\) cell index
- \(l_v, l_{v+1}\) the first cell index adjacent to the virtual wall
- \(k_{eff}\) effective thermal conductivity (W/(m·K))
- \(K_{hi}\) heat transfer coefficient (W/K)
- \(N\) cell numbers
- \(ny\) total number of grids in \(Y\) direction
- \(nz\) the total number of cells in \(Z\) direction
- \(Q_e\) heat exchange, wall gain across the virtual wall (kJ/h)
- \(Q_{coup}\) thermal coupling (W)
- \(Q_{int}\) internal heat gains (W)
- \(Q_{ext}\) heat gains from external conditions (W)
- \(t\) time (s)
- \(T_{cell,i}\) the \(i^{th}\) cell temperature (K)
- \(T_i\) the \(i^{th}\) average zonal temperature (K)
- \(T_j\) the \(j^{th}\) average zonal temperature (K)
- \(T_s\) supply air temperature (°C)
- \(T_z\) room air temperature (°C)
- \(u_{i,j,k}\) \(X\)-positive direction velocity (m/s)
- \(V_{cell,i}\) the \(i^{th}\) cell volume (m³)
- \(V_z\) room volume (m³)
- \(v_r\) supply airflow rate (m³/s)
- \(\Delta y\) cell distance in the \(Y\) direction (m)
- \(\Delta z\) cell distance in the \(Z\) direction (m)
- \(\rho, \rho_a\) air density (kg/m³)
- \(\lambda\) thermal conductivity of air (W/(m·K))
- \(v_t\) turbulent dynamic viscosity (m²/s)
- \(\sigma_{fr}\) turbulent Prandtl number
- \(\delta x\) thickness of mesh grid (m)
minimum controllable air terminal units in the multi-control mode, each subzone can be controlled independently to deal with uneven distribution of temperature profile, the temperature dynamics of each individual zone can be predicted by zonal model, thus it is necessary for control development.

However, no matter zonal division or controllable subzone division, these zones are connected by interfaces through which airflow and heat flux circulate. For instance, when the multi-zone control mode is considered, thermal coupling effect or heat transfer between adjacent subzones needs to be treated because there are no clear boundary division or partition which separate the entire zone. Here thermal coupling effect between adjacent subzones is also termed as heat transfer between adjacent zones (Kintner-Meyer 2005), the inter-volume interactions (Negrao 1995), the air exchange rate between two neighboring zones (Wang and Jin 1998), the convective heat transfer between the interior partition surface and room core zone (Awbi and Hatton 1999; Zhang et al. 2018b), the heat transfer between air layers (Gao et al. 2007; Gao et al. 2010), the airflow and heat flux circulate between the inter-connected interfaces (Boukhris et al. 2009), the heat transfer between virtual wall (Woradetchumroen et al. 2016), the heat transfer on the layered interface/cross-section/intersection area (Wang et al. 2019; Yang et al. 2020; Hu et al. 2022), the dummy wall (Eydner et al. 2019), and the air coupling between the sub-zones (Laghmich et al. 2022). In the above references, the essence of the physical problem is identical but treated with different terms with respect to different indoor flow mechanisms, which will be illustrated in the following discussion.

Therefore, the issues of multi-control require identifying the thermal coupling effect or heat transfer between subzones in a large indoor space. To explain the regime of air movement in detail, one must understand the driven force behind it, actually the process of heat transfer across the virtual zonal boundaries is relatively complex. The convective heat transfer between zones in an indoor space, either natural connection, driven by the buoyancy force generated by a temperature difference or density difference between the zones, or forced convection, create a strong turbulent airflow by intentionally mechanical ventilations, or a combination thereof. Forced convection, commonly used in mixing ventilation systems equipped with slot, grille, square ceiling or nozzle diffusers. To obtain the heat transfer between adjacent zones, the critical issue is to acquire the key parameter: heat transfer coefficient (HTC). The theory and experiments to determine the inter-zone convective heat transfer coefficient between the zones in natural convection were explored by Barakat (1987). The inter-zone natural convective heat transfer coefficient was theoretically and experimentally summarized, and it is governed by Nusselt (\(Nu\), Grashof (\(Gr\), and Prandtl (\(Pr\)) dimensionless numbers. Togari et al. (1993) introduced this empirical coefficient to obtain the heat migration on the stratified surface and predict the indoor thermal environment. In energy consumption software DOE-2, the coefficient was set as 14.8 \(W/(m^2\cdot°C)\) (Landsberg et al. 1986) and 10 \(W/(m^2\cdot°C)\) in an atrium space separately (Chow 1996). The convective heat transfer for a thermal convention between the surface of the interior partition and the core zone of the room was firstly proposed by Awbi and Hatton (1999). It can be described as a function of the temperature difference and hydraulic diameter. The heat transfer factor between air layers in the vertical direction was calculated with CFD code, and it was related to the laminar and turbulent diffusions with values ranging from 2.0 \(W/(m^2\cdotK)\) to 5.0 \(W/(m^2\cdotK)\) in natural convection and displacement ventilation (Gao et al. 2006; Gao et al. 2010). However, the heat transfer coefficients in the above studies were empirically or experimentally obtained under the condition of the natural convection. Few studies involve forced convection airflows, especially in mechanical ventilation.

In a recent study, Yang et al. (2020) investigated the heat transfer coefficient to predict the cooling load in a stratified air-conditioning system for a large industrial building. The numerical solution for calculating the heat transfer coefficient was presented and it is highly dependent on the turbulent viscosity, the results were inline with the previous studies (Gao et al. 2006; Gao et al. 2010). Wang et al. (2019) also revealed that the heat transfer coefficient (500 \(W/(m^2\cdot°C)\)) can be reached in the vicinity of the nozzle diffuser, where a strong local turbulent intensity occurs (Hu et al. 2022). Thus, the heat transfer coefficient fluctuates between several \(W/(m^2\cdot°C)\) (natural convection) and hundreds of \(W/(m^2\cdot°C)\) (forced convection), depending on the convection regimes (impinging flows, wall jets, free jets or buoyancy-driven flows).

To this end, the zonal model and heat transfer between adjacent zones are introduced. Next, the control of thermal indoor environment can be realized in a dynamic manner. In terms of control, a state-space model was presented based on theory of computational fluid dynamics (CFD) and zonal model to control temperatures in a displacement ventilation (Peng 1996; Yao et al. 2013). However, the method developed is for predicting the zonal temperature in displacement ventilation, and how to combine and interact with a control system is not addressed in detail. In addition, statistical models such as autoregressive with the exogenous (ARX) model, autoregressive moving average model with exogenous input (ARMAX) model have been well developed (Ferkl and Široký 2010; Morosan et al. 2010). Yao et al. (2013) developed a three-zone state-space room model to
investigate the air temperature as well as humidity controls based on experiment results. The virtual wall was proposed to describe the energy interaction between adjacent zones (Woradechjumroen et al. 2014, 2016; Woradechjumroen and Li 2015). However, the identification of the thermal coupling effect through the heat exchange channel was not addressed theoretically. An identification method based on ANN model was proposed to study multi-zone buildings while taking into account thermal interactions effects between neighboring zones (Huang et al. 2015). An average temperature around an occupant, numerically obtained by CFD method, was feedback to a local ON/OFF controller to decrease energy consumption (Alhashme and Ashgriz 2016). Zhang et al. (2018a, 2019) proposed a dynamic temperature control based on multi-node model for both stratum ventilation and displacement ventilation. Cao and Ren (2018) and Cao (2019) compared the online ventilation control strategies with different simulation methods such as CFD, zonal/multizone model, fast fluid dynamics (FFD) method, low-dimensional linear ventilation models (LLVM) and so on. With the developed zonal model, the co-simulation is capable of investigating the thermal environment and energy performance simultaneously. Currently available software, such as TRNSYS and EnergyPlus, provides tools and simulation models to deal with the temperature dynamics and can be used to investigate HVAC control in a transient manner. For example, a new simplified zonal model, in which the airflow transfer between adjacent zones was modeled by mass balance equation due to buoyancy forces, was proposed and implemented in the dynamic simulation software ALMABuild (Campana et al. 2019). A physical-data model based on the multiple linear regression analysis for indoor temperature control design was proposed to realize model-based control (Li et al. 2021). Chen and Li (2021) coupled a multi-zone model with TRNSYS to simulate a radiant floor system. A co-simulation platform with TRNSYS and CONTAM based on zonal demand control was developed in a large scale indoor space to alleviate the aforementioned over-cooling and over-heating phenomenon (Zhou et al. 2022). The DOMA/TRNSYS coupled model was used to predict room temperature distribution in a single-zone building over an entire day (Megri et al. 2022). This paper, therefore, proposed a modified zonal temperature control method with a thermal coupling effect integrated for air-conditioned large-scale open spaces, which attempts to achieve a coordinate control of multiple air-terminal units. The effectiveness of thermal coupling across each subzone is defined as heat transfer coefficients (HTCs). It is modeled and calculated numerically with the aid of CFD tools. It differs from the previous studies that focus on calculating the heat transfer coefficient in displacement ventilation, stratified ventilation or stratum ventilation, most driven by natural thermal convection or heat conduction, of which the turbulent viscosity usually is small and the obtained heat transfer coefficient is between 1 W/(m²·°C) and 50 W/(m²·°C). Moreover, the heat transfer coefficient is calculated experimentally or numerically in steady-state and other dynamic cases were not considered since the change in cooling load inevitably leads to the adjustment of the supply airflows. Therefore, in this paper, forced convection driven by mechanical ventilation with large turbulence viscosity is our primary concern, we try to establish a relationship between heat transfer coefficient and zonal temperature difference since zonal temperature is easily acquired in real practices, or at least determine an approximate range of the heat transfer coefficient which covers all air supply scenarios. Finally, the corresponding zonal model for large scale indoor space can be easily used for simulating indoor dynamic temperature responses with current software such as TRNSYS, with the objective of minimizing energy consumption of the HVAC system, while ensuring thermal comfort at the investigated air-conditioned spaces.

The rest of the paper is structured as follows: Section 2 introduces the principle of the heat coupling effect, numerical solution deduction, and control strategies. Section 3 presents a case study to calculate heat transfer coefficients. The application of the proposed model and its control performance is discussed in Section 4. Finally, Section 5 illustrates the conclusion, limitations and future plans of this study.

2 Methodology

2.1 Room model

It is difficult to get an accurate mathematical room model due to its various external factors such as solar radiation, occupant movement, equipment operation and so on. However, the room model usually can be simplified as the following equation based on the law of energy conservation.

\[
\dot{c}_r \rho_v V_r \frac{dT_r}{dt} = \dot{c}_r \rho_v V_r (T_r - T_i) + Q_{ext} + Q_{int}
\]

where \(\dot{c}_r\) is the room specific heat capacity (J/(kg·°C)); \(\rho_v\) is the air density (kg/m³); \(V_r\) is the room volume (m³); \(V_i\) is the supply airflow rate (m³/s); \(T_i\) is the supply air temperature (°C); \(T_r\) is the room air temperature (°C); \(Q_{ext}\) is the heat gains from external weather condition through walls, roofs, floors, etc. (W); \(Q_{int}\) indicates the internal heat gains (W), such as those from lighting, office equipment, occupants, etc. The Taylor series expansion and Laplace transform are carried out near the working point to obtain the transfer function of the output as the room temperature and the
input as the supply air volume, supply air temperature and heat load disturbance, thus the room temperature control can be realized. However, the above model is based on certain assumptions, such as the airflow in the room is neglected, the temperature in the room is considered to be homogeneous and so on. In fact, in large space such as railway station, shopping mall or hotel concourse, it is obviously unreasonable to assume that the temperature in the entire internal space is uniformly distributed. Therefore, the model is not suitable for conducting temperature control for large space.

2.2 Decomposing a large-open space into multiple zones

In order to realize temperature control for large indoor space, this study divides the whole space into a number of controllable subzones using virtual walls, as shown in Figure 1. The definition of zone should follow the configuration of the controllable terminal units of air-conditioning systems. For example, when VAV boxes are the smallest controllable units in a VAV system, the zone can be defined as a subspace that is served by one independent variable-air-volume (VAV) box (Zhou et al. 2014). Equation (1) can be rewritten as Eq. (2) with an additional term: heat coupling part $Q_{ij}^{coup}$.

$$c_s \rho_s V_s^i \frac{dT_s^i}{dt} = c_s \rho_s v_s^i (T_s^i - T_s^j) + Q_{ext}^{ij} + Q_{int}^{ij} + Q_{coup}^{ij} \quad (2)$$

where $Q_{coup}^{ij}$ describes the thermal coupling for the current subzone with its adjacent zones (W). The heat coupling between zones is the essential result from heat and mass exchange between zones, such as through natural and forced convection, diffusion, dispersion, etc. Here the superscript $i$ represents the current $i^{th}$ zone, $j$ refers to its adjacent subzone.

In order to use the current room model for each zone, the virtual walls are considered adiabatic walls. Each zone has a different temperature due to different thermal environment, and thus heat coupling part between adjacent zones can be calculated by the following equation.

$$Q_{coup}^{ij} = K_{ij} (T_s^i - T_s^j) \quad (3)$$

where $i$ and $j$ refer to the $i^{th}$ and $j^{th}$ zone, respectively; $T_s^i$ and $T_s^j$ are the representative temperature for the $i^{th}$ and $j^{th}$ zone, respectively; $K_{ij}$ is the heat transfer coefficient (HTC) between the $i^{th}$ and $j^{th}$ zone (W/K); $Q_{coup}^{ij}$ is the net heat exchange between the $j^{th}$ zone to the $i^{th}$ zone (W). It should be noted that when the temperature of the whole space is assumed to be homogenous (as in conventional room temperature modeling methods), the heat exchange should be neglected ($Q_{coup}^{ij} = 0$); when the representative temperatures of these zones are different, heat coupling occurs between adjacent zones. If these heat coupling parts between adjacent zones can be described using the heat exchange channel. Correspondingly, the subzones can be considered as typical rooms, and therefore can be simulated using available simulation software, such as TRNSYS, to realize independent zonal temperature controls for large indoor spaces. The advantages of the proposed method are described as follows: each zone can be controlled independently to meet different temperature requirements (track its own temperature set-point) for different subzones while taking the thermal coupling part into account; it is able to enhance thermal comfort and achieve energy conservation to avoid energy waste; it is able to realize energy supply depending on demand for different subzones to avoid sub-cooling or over-cooling.

2.3 Heat coupling between adjacent zones and its calculation

In order to model the heat coupling part, a numerical method, resorting to the tool of computational fluid dynamics (CFD), is proposed to identify the coefficient $K_{ij}$. The detailed indoor temperature profile can be easily obtained as well. Besides, the CFD simulation can be the benchmark for zonal temperature controls.

To use the CFD tool, conventionally, the fluid domain needs to be divided into a large number of controlled volumes...
(grid number) (e.g., several million). For each controlled volume, discrete partial differential equations should be established using a finite volume method. Usually, the SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) method is adopted to obtain numerical solutions.

According to the CFD theory, the convective and diffusive term of the energy equation can be discretized by an upwind difference and central difference scheme, respectively. The detailed methodology for discrete solutions can be found in CFD textbooks (Hoffmann and Chiang 2000). The final discrete equation under the fully implicit time integral method for energy has the following general form:

\[
a_P T_P = \sum a_{ab} T_{ab} + b
\]

(4)

where the coefficient \(a_{ab}\) describes the effects of both convection and diffusion on the heat transfer through the interface of a certain controlled volume \(P\). The term \(b\) describes the disturbance in the heat transfer from internal and external factors, such as occupants, radiation, and diffusers. The coefficient \(a_{ab}\) is related to the coefficient \(K_{ij}\) in Eq. (3) and will be used to estimate the coefficient \(K_{ij}\).

To show how to calculate the coefficient \(K_{ij}\), a simplified temperature zonal model is adopted in this study (Peng 1996). Consider two adjacent zones in this case, titled the \(i^{th}\) zone and the \(j^{th}\) zone, separated by a virtual wall, as shown in Figure 2. In this case, the following equations can be used to express the heat transfer between the \(i^{th}\) zone and \(j^{th}\) zone:

\[
a_{ij} = \sum_{k=1}^{ny} \sum_{l=1}^{nz} \sum_{h=1}^{nx} C_p \left( \Delta y \Delta z \right)_{i,j,h,k} \left( D_{i,j,h,k} + \rho \text{MAX} \left[ -u_{i,j,h,k}, 0 \right] \right)
\]

(5)

\[
a_{ji} = \sum_{k=1}^{ny} \sum_{l=1}^{nz} \sum_{h=1}^{nx} C_p \left( \Delta y \Delta z \right)_{i,j,h,k} \left( D_{i,j,h,k} + \rho \text{MAX} \left[ u_{i,j,h,k}, 0 \right] \right)
\]

(6)

where \(a_{ij}\) is the heat transfer from the \(j^{th}\) zone to the \(i^{th}\) zone and \(a_{ji}\) is the heat transfer from the \(i^{th}\) zone to the \(j^{th}\) zone and vice versa (W/K). \(C_p\) is the air specific heat capacity (J/(kg·K)); \(D\) is the diffusion conductance (kg/(m²·s)); the subscripts \(l, h, k\) represent the cell index; the superscripts \(ny, nz\) are the number of grids of \(Y\) and \(Z\) directions, respectively; \(u\) is the velocity along a certain direction (m/s), in this case, the velocity is in \(X\) direction, where negative means in \(X\)-negative direction. In real applications, the heat transfer between the adjacent zones is not equal due to the difference between the airflow rates supplied to these subzone zones. The disequilibrium of the supply airflow rate leads to the temperature difference eventually. However, even though the airflow rate from both zones is the same, there is still some discrepancy between \(a_{ij}\) and \(a_{ji}\) due to the chaos of turbulence. Besides, the grid meshing, the simulation truncation, and the discretization error may also lead to this discrepancy.

Equation (5) can be expanded by two separate parts:

\[
a_{ij} = \sum_{k=1}^{ny} \sum_{l=1}^{nz} \sum_{h=1}^{nx} C_p \left( \Delta y \Delta z \right)_{i,j,h,k} \frac{k_{eff}}{\partial x}
\]

\[
+ \sum_{k=1}^{ny} \sum_{l=1}^{nz} \sum_{h=1}^{nx} C_p \left( \Delta y \Delta z \right)_{i,j,h,k} \rho \text{MAX} \left[ -u_{i,j,h,k}, 0 \right]
\]

(7)

\(k_{eff}\) is the effective thermal conductivity (W/(m·K)), which can be expressed by the following equation:

\[
k_{eff} = \lambda + \frac{\rho C_p V}{\sigma_R}
\]

(8)

Equation (7) shows that the heat flux through the virtual wall is composed of two parts: (1) the heat flux due to the thermal diffusivity and the turbulent diffusivity of air, and (2) the heat transfer caused by the air mass flows. \(u_{i,j,h,k}\) is the velocity in \(X\)-positive direction (m/s). \(\text{MAX} \left[ -u_{i,j,h,k}, 0 \right]\) means the air mass flows from the \(j^{th}\) zone to the \(i^{th}\) zone. The velocity is negative in this case. Thus, \(\text{MAX} \left[ -u_{i,j,h,k}, 0 \right] = u_{i,j,h,k}\) If the air mass flows from the \(i^{th}\) zone to the \(j^{th}\) zone, the velocity is positive. Thus, \(\text{MAX} \left[ -u_{i,j,h,k}, 0 \right] = 0\). \(\nu\) is the turbulent or eddy viscosity (m²/s). \(\sigma_R\) is the turbulent Prandtl number. Therefore, unlike other simplified thermal models, the “degraded” momentum equations are either only air mass flows considered (Wang and Jin 1998; Wang and Chen 2007; Shan et al. 2020). For instance, the power law model (PLM) is based on the differential pressure driving force in the simplified Bernoulli equation (Dols and Polidoro 2020; Lu et al. 2020); or neglect the air mass flows between adjacent zones (mass transfer). Only conduction and convection effects are considered; it is suitable for natural ventilation driven by buoyancy force (Gao et al. 2007; Gao et al. 2010; Wang et al. 2019; Yang et al. 2020; Hu et al. 2022). It should be noted that the heat transfer coefficient can also be obtained by experimental study and can be expressed by empirical formulas (Awbi and Hatton 1999; Zhang et al. 2019). Equation (7) is retrieved directly from the momentum term in Navier-Stokes equations. Thus, the prediction of indoor
temperature distributions with this model is more accurate and realistic, particularly applicable to intensive turbulence scenarios such as round ceiling diffusers, nozzle diffusers and so on (Gao et al. 2021c). The coupling heat exchange across the virtual wall \( K_{ij} \) is the difference between \( a_{ij} \) and \( a_{ji} \), i.e.:

\[
K_{ij} = a_{ij} - a_{ji}
\]

Given the representative temperature difference between two adjacent zones, an approximate net heat transfer \( Q_{coup} \) via the virtual wall is calculated by Eq. (3). It is noted that Eq. (3) and Eq. (10) have the same meaning but with different units since it can be recognized by TRNSYS in Eq. (10), the unit for \( Q_{coup} \) is \( kJ/h \).

\[
Q_{coup} = K_{ij} \times |T_i - T_j| \times 3.6
\]

where the representative temperature of the \( i^{th} \) (and \( j^{th} \)) zone is

\[
T_i = \frac{\sum_{j=1}^{N} (V_{cell,i}T_{cell,j})}{\sum_{i=1}^{N} V_{cell,i}}
\]

The representative temperature of a zone can be defined as the average temperature of the working region, which should be temperature measured at a place lower than 1.7 m (usually the breathing level for seated occupant is around 1.1 m), higher than 0.1 m above the floor and at least 0.1 m away from the walls, see working zone in Figure 2.

Equations (5) and (6) can be compiled in C++ language and interpreted into FLUENT as User Defined Functions (UDFs) to calculate \( K_{ij} \) since the parameters in the above equations are pre-defined in FLUENT as macros. In order to find the heat transfer coefficient \( K_{ij} \), intensive simulations under different load distributions as well as airflow rates should be conducted. To this end, typical load distributions should be identified and simulated.

### 2.4 Control strategies

To realize zonal temperature control independently, Phase I and Phase II control was proposed and interested reader may find more information in references (Zhou et al. 2014; Zhou et al. 2017a). The control strategy can be illustrated as following: it consists of two control stages namely Phase I and Phase II control, and Phase I control can be regarded as a regular independent zonal temperature control. For example, the temperature in each subzone is undertaken by its corresponding air terminal unit, the air terminal unit is responsible for its own subzone and it does not interfere with other terminal units. However, in real applications, with the absence of physical partitions between subzones in large indoor spaces, if there is a sharp increase of occupants in one of the subzones, the corresponding air terminal unit cannot control the temperature to its set-point value, in this particular circumstance, an extra airflow rate is required from its immediate air terminal units to remove the internal heat gains, this is Phase II control. It should be noted that Phase II control is realized by adding a heat exchange channel in TRNSYS, thus, the influence of thermal coupling effect can be integrated into the control system to investigate the zonal temperature control in large-scale indoor spaces without physical partitions, this part will be addressed in detail in Section 4.

### 3 Case studies

#### 3.1 Boundary condition setup

A room was selected as the target space, with a dimension of 6.87 m × 2.8 m × 6.72 m. In this room, 4 square ceiling diffusers were used to serve the room with a supply air temperature of 17 °C. The maximum airflow rate was designed to be 6000 m³/h, and the outlet vent was set to the side wall with a dimension of 0.8 m × 0.6 m, 0.2 m above the floor. In addition, 6 heaters were used as heat sources in the room, each rated at 0.9 kW.

A CFD model was built for this room to estimate the heat transfer coefficient, two subzones were divided intentionally by a virtual wall in the middle of the room, as shown in Figure 3. There were four supply up-ceiling air inlets inside the room, and one air return outlet in the bottom of the south wall. There were six heaters (three in each subzone) arranged in two rows to represent the heat source generated by the occupants. Inlet 2 and inlet 4 served left subzone, meanwhile controlled by an air terminal unit (VAV box). The right subzone was served by inlet 1 and inlet 3, which controlled by another VAV box. The airflow rate for both zones can be adjustable independently using respective VAV boxes.

The boundary condition setup is listed in Table 1. The momentum method was adopted to simulate the square ceiling diffuser (Chen and Moser 1991; Srebric and Chen 2002), and the discharge angle was set to 45° inclined with the roof. The inlet velocity was defined by UDFs, and a source was added to the momentum to describe the real square ceiling diffusers with an effective area ratio 30% (the UDF program for defining square ceiling diffuser is presented in Appendix A, which is available in the Electronic Supplementary Material (ESM) of the online version of this paper). The Boussinesq hypothesis was adopted in the simulation. In order to calculate the heat flux and airflow mass flows pass through the virtual wall, the boundary condition of the virtual wall was defined to be an “interior” in CFD simulation. To obtain convincible and comparable results, the wall boundary
Table 1 Setup of the boundary condition

| Ventilation scheme | Mixed ventilation |
|-------------------|-------------------|
| Room size         | 6.87 m × 6.72 m × 2.8 m |
| Supply airflow rate | 6000 m³/h |
| Supply air temperature | 17 °C |
| Controlled volumes       | Hexahedral-structured, 1.5 million |
| Supply air inlet         | 4 square up-ceiling diffusers, 0.4 m × 0.4 m |
| Internal heat gains      | 6 electric heaters, 0.6 m × 0.6 m × 0.16 m, 0.9 kW for each |
| Return outlet            | Outflow, 0.8 m × 0.6 m |
| Turbulence model         | Realizable k-ε model |
| Radiation model          | Discrete ordinates (DO) model |
| Numerical discrete methods | Upwind second-order difference scheme; SIMPLE algorithm |
| Enclosure (wall, floor, ceiling) | Temperature: 25 °C |
| Convergence criteria     | Continuity, momentum, turbulent kinetic: 10⁻⁴, energy: 10⁻⁶ |

conditions, the selected turbulence model, and the discrete numerical schemes were kept constant. The mesh generated is hexahedral-structured with a number of 1.5 million, partial meshing scheme on the boundaries is displayed in Figure 3. The grid-independent tests were performed with: 1.5 million, 2.2 million and 2.8 million grid cells, by comparison the average temperature and velocity in the working zone (X = 0.1 to 6.77 m, Y = 0.1 to 1.7 m, Z = 0.1 to 6.62 m), the discrepancy for these three types of cell numbers was no more than 5%, thus we selected 1.5 million grid cells for further calculation to save computational time and resources.

3.2 CFD model validation

To evaluate the CFD model, the airflow and temperature field generated by the CFD model was compared with the flow field measurements from the real room. One of the diffusers was selected to conduct the site measurement for comparison when the airflow rate for each diffuser was 890 kg/h. Figure 4 shows the comparison of the measured and the predicted average Y-direction velocity. Four vertical directions were selected with each being 0.5 m away from the edge of the diffuser. The velocities were recorded at 7 different heights (from 1.6 m to 2.8 m with an interval of 0.2 m) in the vertical direction. Because the duct elbow above the diffuser made the jet asymmetric in these four outlet directions and thus the average velocity was used. The temperature was measured and compared with simulation results as shown in the figure, the temperature difference fluctuated at ±0.5 °C. From the comparison shown in Figure 4, the maximum standard deviation (SD) for velocity and temperature were 0.3 and 0.45 respectively. It can be observed a fair agreement of the airflow velocities between

Fig. 3 HVAC configuration of the case room and meshing diagram in typical boundaries

Fig. 4 Comparison of measured and predicted average Y-direction velocity and temperature at four different locations (0.5 m far away from the four edges of the diffuser)
predicted and measured results, which shows that the CFD model is reliable and can be used to generate data for the thermal coupling analysis.

Using the validated CFD model, the velocity field and temperature field were observed and analyzed at different operational conditions. It is noted that the purpose of the CFD simulation is used to calculate the coefficient $K_{ij}$ under different supply airflow conditions. Figure 5 gives the velocity (bottom) and temperature field (top) in fully-open and half-open conditions. The airflow rate was set to be 1500 m$^3$/h for each diffuser in fully-open condition (inlet 1, 2, 3, 4 fully opened). While in half-open condition, only inlet 1 and inlet 3 were fully-opened, whereas inlet 2 and inlet 4 were closed. The discharge velocity from the diffuser was 3.68 m/s. The simulated average temperature for fully-open and half-open conditions were 20.6 °C and 24.6 °C, respectively. The average velocity for fully-open and half-open conditions were 0.27 m/s and 0.16 m/s, respectively. In fully-open condition, the air in the room was totally mixed, while in half-open condition, both the mass air and heat have a strong movement from supplied area to the non-supplied area. See the velocity and temperature distribution vector in the Y-cut plane in Figure 5. It differs from displacement ventilation where the momentum force is driven by the buoyancy effect. In mixing ventilation the supply diffuser is mounted on the up-ceiling level. The air sinks and settles down to the floor when it reaches the vertical side walls. It was observed that there was a collision of air when two air jets met in the middle and combined into one jet, and the continuous momentum force drove the air to move downward.

To calculate the heat flux due to the turbulent diffusivity, it is imperative to know the turbulent viscosity $\nu_t$ as described in Eq. (8). It greatly impacts the calculation of the heat transfer coefficient since the thermal conductivity of turbulent air mainly depends on turbulent viscosity. Figure 6 shows the calculated turbulent viscosity on the virtual wall, in fully-open and half-open conditions. It varies when the supply airflow changes, therefore, for control applications, this value is difficult to predict since the simulation result are numerically obtained in steady state. The calculated average thermal conductivity is 23.9 W/(m·K) and 13.2 W/(m·K) in fully-open and half-open conditions.

### 3.3 Heat transfer coefficient

To study the heat transfer between adjacent zones, the temperature difference between these two zones were
The temperature difference occurs when the airflow rate or load changes in each zone, so in order to artificially create the temperature difference, 50 combination cases of airflow and load were carried out. For example, the right zone (inlet 1 and inlet 3 are fully opened) was designed to have a fixed load and therefore fixed supply airflow rate; while the supply airflow rate inside the left zone (inlet 2 and inlet 4) was changed by a certain percentage (10%) of its original design value. For control purposes, the model should have a simple structure and be suitable for a wide operational range. Thus, in total, there were 50 cases between different airflow rates and loads which covers most of its operation conditions. The representative temperature of both zones was calculated for each case. The generated representative temperatures were used to calculate the heat transfer coefficient $K_{ij}$ in Eq. (7)–(11).

The simulation results were summarized in Figure 7, which shows the relationship between the heat transfer coefficient and the zonal representative temperature difference. The zonal representative temperature difference, in this case, ranged from 0.0 °C to 0.77 °C, while the maximum temperature difference we observed in a real large-scale space with an area of 1500 m$^2$ was 2.5 °C (Zhou et al. 2015). The calculated heat transfer coefficient ranged from 0 to 6000 W/K (0–312 W/(m$^2$·°C)). It can be observed that the heat transfer coefficient was not constant, the coefficients calculated seemed not in line with our expectations. This is probably due to the disordered turbulent viscosity, which is a key factor that impacts the heat transfer coefficient (see Eq. (8)). When the turbulence viscosity $\nu$ fluctuates, the heat transfer coefficient calculated will oscillate. If the temperature difference is 0, which means there is no heat transfer across the adjacent boundaries, the potential heat transfer occurs between the adjacent zones when the temperature difference exists. Thus, the coefficient has a linear relationship with the temperature difference, here $K_{ij} = 5.508$ kW/K (the upper and lower limits are also given as 10 kW/K and 2.5 kW/K, respectively). The heat exchange increases as the temperature difference increases. To this end, the fitted linear equation can be used in the study presented in the following section.

3.4 Model validation with CFD simulation

The relationship between temperature differences and heat transfer coefficients is obtained by CFD simulation. Then the fitted linear model is integrated into the control platform, the control generates the output temperature for each subzone. At last, we compare the output temperature with CFD simulations at some cases with fixed airflow and loads. The fitted linear model is validated and compared with CFD simulation results, see Table 2. Usually, the TRNSYS software cannot be used to model temperature control for large spaces without physical partitions. For example, In Case II, the TRNSYS simulated temperatures for Zone 1 and Zone 2 are 21.00 °C and 21.99 °C, respectively, nearly 1 °C temperature difference. In fact, the temperatures for Zone 1 and Zone 2 are 21.2 °C and 21.39 °C, respectively. The temperature difference is very close since there is no physical partition in the middle and the airflow rate for both zones is nearly the same (3,000 m$^3$/h and 2,700 m$^3$/h). However, when the HTC is added to the TRNSYS model, the temperatures are improved to 21.42 °C and 21.46 °C, respectively. It is very close to the real situations (21.20 °C and 21.39 °C). In Case IV, the load for Zone 2 is reduced to half and the airflow rate is reduced by half accordingly. It is a very normal case in real applications. The simulated temperature difference with TRNSYS is reached to 6 °C.
without the HTC integrated. The temperatures for Zone 1 and Zone 2 are 20.94 °C and 21.13 °C, respectively, after the HTC integrated. Although there is still a gap with the actual CFD simulated temperature (21.33 °C and 21.69 °C), the introduction of HTC can narrow down the temperature discrepancy between the two sub zones. Case I and Case III also show the room temperature is more uniform and closer to real situations compared with CFD results in the proposed modelling method. Hence the developed model can be used to model temperature control in large space with current software TRNSYS without the aid of CFD software.

### 4 Potential application of the proposed model

#### 4.1 Control platform description

This section investigates the impact of the thermal coupling on the large-scale space temperature control and explains how to use the proposed model to realize zonal temperature control. The space with two zones was still considered, and the structure of the space temperature control was shown in Figure 8. Zone 1 and Zone 2 were controlled separately to maintain the room temperature set-point (this is a typical temperature control infrastructure for large scale indoor open spaces).

To obtain the coordinated zonal temperature control, the TRNSYS control model was constructed, as shown in Figure 9, the building model (Type 56) was built in TRNSYS, which included two subzones. The dimension, the boundary conditions as well as the heat gains setup were consistent with the CFD simulations. It should be aware that two subzone (Zone 1 and Zone 2) shared an internal wall in the building model, there was a heat exchange channel that can be defined on this wall. Thus, the heat transfer acquired in previous section can be added to simulate the dynamic response of temperatures. The supply airflow rate was controlled by two variable speed fans (Type 662), each rated airflow rate was 0.42 m³/s with supply air temperature 17 °C. A continuous typical summer week started from 4440 h to 4608 h with a time interval of 30 min was selected to run the control simulation. The zonal temperatures were controlled by two PID controllers (Type 23), the PID parameters \( P = 0.5 \) and \( I = 0.04 \) were tuned to regulate the temperature response with favorable overshoot and setting times (Tashtoush et al. 2005; Wemhoff et al. 2012). It is necessary to point out again, in TRNSYS, thermal coupling effect in two adjacent subzones was defined as wall gain (kJ/h) (Eq. (10)) on the shared internal wall in building model. Here the wall gain means the coupled heat transfer from/to the jointly owned internal wall surface. Thus, the fitted linear equation for heat exchange in Section 3.3 can be fed into the model to investigate the impact of thermal coupling on the zonal temperature controls. In order to achieve coordinate zonal temperature control, the two phase control strategies were compiled in an external program Matlab (Type 155) and fed into TRNSYS model to control the supply airflow rates (the code of coordinated zonal temperature control strategy is presented in Appendix B in the ESM of the online version of this paper).

#### 4.2 Control simulation analysis

To investigate how the control strategy performed, firstly, an uneven distribution of load was predefined in the two subzones in a typical summer week, as shown in Figure 10. It is a pre-assigned load profile which covers all of the cases:

![Fig. 8 An independent control diagram with two VAV control boxes](image-url)

### Table 2 Comparison between the proposed modelling method and CFD

| Boundary conditions (airflow & load) | CFD Simulation (°C) | TRNSYS without HTC (°C) | TRNSYS with linear fitted HTC (°C) |
|-------------------------------------|---------------------|-------------------------|-----------------------------------|
| Zone 1 | Zone 2 | Zone 1 | Zone 2 | Zone 1 | Zone 2 | Zone 1 | Zone 2 |
| Case I | 3000 m³/h 2.7 kW | 2100 m³/h 2.7 kW | 21.36 | 21.86 | 21.00 | 22.69 | 21.67 | 21.74 |
| Case II | 3000 m³/h 2.7 kW | 2400 m³/h 2.7 kW | 21.20 | 21.39 | 21.00 | 21.99 | 21.42 | 21.46 |
| Case III | 3000 m³/h 2.7 kW | 2700 m³/h 2.43 kW | 21.31 | 21.34 | 20.60 | 21.44 | 20.98 | 21.02 |
| Case IV | 3000 m³/h 2.7 kW | 1500 m³/h 1.35 kW | 21.33 | 21.69 | 19.01 | 24.92 | 20.94 | 21.13 |
(1) Normal case: Zone 1 and Zone 2 have the same or similar load profile in day 1, day 5 and day 7. (2) Special cases are also taken into account when there is only one subzone occupied while the other subzone is empty, such as day 2 and day 3. (3) Extreme case: the heat source from Zone 2 is transferred to Zone 1 or vice versa, see day 4 and day 6. The purpose for setting this scenario is to show how the Phase II control strategy started to intervene.

4.3 Control performance analysis

Phase I control can easily cope with both normal and special cases, as mentioned before. The dynamic response of temperature and supply airflow rate for two subzones are shown in the upper part of Figure 11 and Figure 12. Without heat exchange integrated, the temperatures in Zone 1 and Zone 2 can be tracked to its own set-point except Zone 1 from 4512 h to 4536 h (day 4) and Zone 2 from 4560 h to 4584 h (day 6). For example, in Zone 1, the particular reason for this circumstance is the airflow rate for its corresponding VAV box reaches its rated maximum value: 0.42 m³/s, see Figure 12. Thus, the temperature in Zone 1 increases to 27 °C, which deviates greatly from its temperature set-point 25 °C. Meanwhile, the PID controller is out of control even though the load in day 5 returned to its normal distribution. The controller needs more time (17 hours) to tune-up to track its set-point. This affects the control performance of day 5 (temperature oscillates between...
18 °C and 32 °C from 4536 h to 4557 h). The failure control phenomenon also exists in Zone 2 in extreme day 6. Thus, regular PID control cannot deal with the sharp shift of load variations as a result of controller failure and temperature oscillation.

However, the output of temperature and supply airflow rate suffers a significant improvement after heat coupling effect integrated. For example, in day 4, the maximum internal heat gain in Zone 1 (5.4 kW) has been imposed, while its temperature dynamic response fluctuates a little bit and instantaneously reverted to its set point in the following days, see lower part of Figure 11. The reason can be explained that the heat exchange channel was activated then started to interfere temperature in Zone 1. In this case, the coupled heat gain in the joint internal wall was transferred from Zone 1 to Zone 2. Thus, the delivered supply airflow rates for Zone 1 and Zone 2 varied accordingly: the air supply rate for Zone 1 remained its rated maximum, while the supply airflow rate for Zone 2 increased (an additional airflow rate approximately 0.1 m³/s) to offset the coupled heat gains, see bottom part of Figure 12. From the perspective of energy consumption, 2% more energy consumed by variable speed fans compared with simple independent zonal temperature controls. For Zone 2 in day 6, the temperature was controlled well with an extra airflow rate from Zone 1 intervention to Zone 2 as well. The temperature control for
the following days when the load decreased to its normal range was improved only with small fluctuations. The PMV is calculated by a multiple regression model (Zhang et al. 2017) when the room air temperature and air velocity are determined. Here the average air velocity is extracted by CFD simulations. The PMV for Zone 1 in day 4 is improved from 0.43 to −0.29, but the PMV is fluctuated between −2.8 and 2.2 from 4536 h to 4557 h, which means the thermal comfort was deteriorated without HTCs integrated. Thermal comfort in Zone 2 cannot be guaranteed in day 6 and day 7 as well, see Table 2. To sum up, the accuracy of temperature control of large-scale open spaces was improved by the introduction of HTCs. Therefore, the developed orientated toward coordinated control method for large-scale open spaces of multiple air-terminal units can be achieved.

Meanwhile, the traditional control was also realized in TRNSYS for comparison. The building model is combined Zone 1 and Zone 2 as one zone. A PID controller is connected to the building to control two supply fans. In this conventional control strategy, the indoor temperature will be tracked to its set-point by increasing or decreasing the supply airflow rates. While in the proposed zonal temperature control strategy, each zone can be controlled independently since different zones have different temperature demands. For example, Zone 1 and Zone 2 can be set to 25 °C and 24 °C, respectively, to save energy. The energy consumption of the variable speed fans was 95.71 kWh for the traditional control while 85.33 kWh for the proposed control. Nearly 11% of energy conservation was achieved with the proposed control strategy as shown in Table 3. In a real application, the load is always unevenly distributed, with instances when some subzones might be non-occupied, and in such cases, more energy savings can be obtained with proposed zonal temperature control.

The introduction of heat transfer coefficient indeed could improve the coordinate control performance. Based on the fitted linear equation in Section 3.3, and feedback to the control platform and simulated for comparison. It was found that the temperature for Zone 1 was cooled down from 27 °C to 25.4 °C and 25.1 °C respectively. Apparently, when the heat transfer coefficient increases, the temperature of Zone 1 is much closer to its set-point 25 °C (more airflow rates from Zone 2 transfer to Zone 1), and the temperature response for the following normal days improved. However, due to inferiority of anti-interference ability, the PID controller cannot track its temperature smoothly when the constant $K_i$ is larger than 10 kW/K. A disturbance compensation control or other superior control methods can be adopted to improve the system response performance. The optimal $K_i$ ranges from 3.5 kW/K to 8.5 kW/K for this case, thus the regular PID controller is able to respond to the proposed zonal temperature control. The corresponding temperature difference is from 0.4 °C to 0.7 °C from Figure 7. Two PID controllers are used to realize the zonal temperature control. The PID controller adjusts its parameters to realize a robust and accurate zonal temperature control. In real control applications, the heat coupling effect can be neglected when the temperature difference is less than 0.4 °C, which can be dealt with a simple independent zonal temperature control. It is intended to avoid the interference of heat coupling to the control system and reduce fan energy consumption. The heat interaction between adjacent zones starts to intervene in control system to improve thermal comfort when the temperature difference is larger than 0.4 °C.

## 5 Conclusion, limitations and future work

The proposed strategy presents a novel way to deal with control of large-scale spaces with available technologies, such as the current mainstream software Fluent and TRNSYS. This paper proposes a coordinated zonal temperature control with thermal coupling effect integrated for large-scale open spaces. The proposed method divides a large-scale space into a number of zones and takes account of the thermal coupling between adjacent zones. It differs from the previous studies that emphasize on natural convection flows; in this paper, however, forced convection airflow driven by a mechanical ventilation with strong turbulence is our major concern. The heat transfer coefficients are numerically solved under different air supply scenarios. The relationship between heat transfer coefficient and zonal temperature difference is linearized. The corresponding zonal model for large scale indoor space can be easily used for simulating indoor dynamic temperature responses. Thus, currently available zonal models in popular software can be used to simulate the temperature response of large-scale indoor open spaces when the space temperature is not homogenous. Resorting to CFD simulation,
a method that can estimate the heat transfer coefficient between two adjacent zones has been developed and validated using case studies. An application of this proposed method for control application has also been investigated. The results show that:

- The proposed coordinated temperature control method can handle most of the load changes and the temperature control more practically while the thermal coupling between adjacent subzones were taken into consideration in large-scale spaces.
- Simple independent zonal temperature control consumed slightly less energy as compared to the proposed method. However, at the cost of thermal comfort improved.
- The proposed method could satisfy the temperature requirements of different zones for large-scale spaces, and at least 11% of energy saving can be achieved by comparing with the conventional control strategy.

The process of heat transfer across the virtual zonal boundaries is relatively complex. It is influenced by many factors, such as the flow regime (weak or strong turbulent flow), the geometry of the simulated model, the position of the virtual wall, the supply air velocities from the diffusers, and so on. Due to fluctuations, it is very difficult to get precise values, particularly in strong turbulent flows. However, when the heat exchange between adjacent zones is integrated into the control platform, thermal comfort is improved, which provides a novel way to solve the temperature control, particularly in large-scale open spaces. For control applications, due to its characteristic of inertia and time delay from the temperature sensor, feedback control response of proposed method is much slow. This can be improved by using the occupancy profile detection technique as an alternative way to realize a fast temperature response control. In future work, the proposed strategy will be tested for multiple zones (4 subzones or more) and experiments will be carried out to evaluate the performance of the zonal temperature control when the heat coupling effect is integrated.

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References

Alhashme M, Ashgriz N (2016). A virtual thermostat for local temperature control. Energy and Buildings, 126: 323–339.

Awbi HB, Hatton A (1999). Natural convection from heated room surfaces. Energy and Buildings, 30: 233–244.

Barakat SA (1987). Inter-zone convective heat transfer in buildings: A review. Journal of Solar Energy Engineering, 109: 71–78.

Boukhris Y, Gharbi L, Ghrab-Morcos N (2009). Modeling coupled heat transfer and air flow in a partitioned building with a zonal model: Application to the winter thermal comfort. Building Simulation, 2: 67–74.

Campana JP, Schuss M, Mahdavi A, et al. (2019). A Simplified zonal model for the evaluation of the spatial distribution of air temperature in indoor environments. In: Proceedings of the 16th International IBPSA Building Simulation Conference, Rome, Italy.

Cao S, Ren C (2018). Ventilation control strategy using low-dimensional linear ventilation models and artificial neural network. Building and Environment, 144: 316–333.

Cao S (2019). Challenges of using CFD simulation for the design and online control of ventilation systems. Indoor and Built Environment, 28: 3–6.

Chen Q, Moser A (1991). Simulation of a multiple-nozzle diffuser. In: Proceedings of the 12th AIVC Conference, Ottawa, Canada.

Chen Q, Li N (2021). Fast simulation and high-fidelity reduced-order model of the multi-zone radiant floor system for efficient application to model predictive control. Energy and Buildings, 248: 111210.

Chow WK (1996). Assessment of thermal environment in an atrium with air-conditioning. Journal of Environmental Systems, 25: 409–420.

Dols WS, Polidoro BJ (2020). CONTAM User Guide and Program Documentation Version 3.4. NIST Technical Note 1887. Gaithersburg, MD, USA: National Institute of Standards and Technology.

Eydner M, Toufek B, Henzler T, et al. (2019). Investigation of a multizone building with HVAC system using a coupled thermal and airflow model. E3S Web of Conferences, 111: 04040.

Ferkl L, Široký J (2010). Ceiling radiant cooling: Comparison of ARMAX and subspace identification modelling methods. Building and Environment, 45: 205–212.

Gao J, Zhao J, Li X, et al. (2006). A zonal model for large enclosures with combined stratification cooling and natural ventilation: part 1—model generation and its procedure. Journal of Solar Energy Engineering, 128: 367–375.

Gao J, Gao F, Zhao J, et al. (2007). Calculation of natural ventilation in large enclosures. Indoor and Built Environment, 16: 292–301.

Gao J, Zhang X, Zhao J, et al. (2010). A heat transfer parameter at air interfaces in the BLOCK model for building thermal environment. International Journal of Thermal Sciences, 49: 463–470.

Gao R, Li H, Li A, et al. (2021a). Guardrail-based air supply terminal for improving ventilation effectiveness and saving energy in the waiting zone. Energy Exploration & Exploitation, 39: 1394–1414.

Gao R, Zhang H, Li A, et al. (2021b). Research on optimization and design methods for air distribution system based on target values. Building Simulation, 14: 721–735.
Gao R, Liu M, Zheng Q, et al. (2021c). High-efficiency diffuser based on a normalized evaluation index of jet length and resistance. *Building and Environment*, 195: 107737.

Hoffmann KA, Chiang ST (2000). Computational Fluid Dynamics (Vol IV). Wichita, USA: Engineering Education System.

Hu H, Wang H, Zou Z, et al. (2022). Investigation of inter-zonal heat transfer in large space buildings based on similarity: Comparison of two stratified air-conditioning systems. *Energy and Buildings*, 254: 111602.

Huang C, Zou Z, Li M, et al. (2007). Measurements of indoor thermal environment and energy analysis in a large space building in typical seasons. *Building and Environment*, 42: 1869–1877.

Huang H, Chen L, Hu E (2015). A neural network-based multi-zone modelling approach for predictive control system design in commercial buildings. *Energy and Buildings*, 97: 86–97.

Kintner-Meyer M (2005). Opportunities of wireless sensors and controls for building operation. *Energy Engineering*, 102: 27–48.

Laghmich N, Romani Z, Lapisa R, et al. (2022). Numerical analysis of horizontal temperature distribution in large buildings by thermo-aerulous zonal approach. *Building Simulation*, 15: 99–115.

Lan B, Yu Z, Huang G (2022). Study on the impacts of occupant distribution on the thermal environment of tall and large public spaces. *Building and Environment*, 218: 109134.

Landsberg DR, Misuriello H, Moreno S (1986). Design strategies for energy-efficient atrium spaces. In: Proceedings of ASHRAE Annual Meeting, Technical Paper 2966, Portland, OR, USA.

Lebrun J (1971). Exigences physiologiques et modalites physiques de la climatisation par source statique concentrée. Available at https://bibliotheque.insa-lyon.fr/recherche/viewnotice/id_sigb/26801/id_int_bib/1. (in French)

Li X, Han Z, Zhao T, et al. (2021). Online model for indoor temperature control based on building thermal process of air conditioning system. *Journal of Building Engineering*, 39: 102270.

Lu Y, Xiang Y, Chen G, et al. (2019). Summer dynamic thermal environment for isolated atrium in the severe cold region: on-site measurement and numerical simulation. *Applied Thermal Engineering*, 160: 114108.

Lu Y, Dong J, Liu J (2020). Zonal modelling for thermal and energy performance of large space buildings: a review. *Renewable and Sustainable Energy Reviews*, 133: 110241.

Megri AC, Snyder M, Musy M (2005). Building zonal thermal and airflow modelling—A review. *International Journal of Ventilation*, 4: 177–188.

Megri AC, Haghighat F (2007). Zonal modeling for simulating indoor environment of buildings: review, recent developments, and applications. *HVAC&R Research*, 13: 887–905.

Megri AC, Yu Y (2015). New calibrated zonal model (POMA+) for temperature and airflow predictions. *Building and Environment*, 94: 109–121.

Megri A, Yu Y, Miao R, et al. (2022). A new dynamic zonal model with air-diffuser (DOMA)—Application to thermal comfort prediction. *Indoor and Built Environment*, 31: 1738–1757.

Moroşan PD, Bourdais R, Dumur D, et al. (2010). Building temperature regulation using a distributed model predictive control. *Energy and Buildings*, 42: 1445–1452.

Negrao CO (1995). Conflation of computational fluid dynamics and building thermal simulation. Citeseer. PhD Thesis, University of Strathclyde, UK.

Peng X (1996). Modeling of indoor thermal conditions for comfort control in buildings. PhD Thesis, Delft University of Technology, the Netherlands.

Shan X, Luo N, Sun K, et al. (2020). Coupling CFD and building energy modelling to optimize the operation of a large open office space for occupant comfort. *Sustainable Cities and Society*, 60: 102257.

Srebri J, Chen Q (2002). Simplified numerical models for complex air supply diffusers. *HVAC&R Research*, 8: 277–294.

Stewart J, Ren Z (2006). COwZ—A subzonal indoor airflow, temperature and contaminant dispersion model. *Building and Environment*, 41: 1631–1648.

Tashtoush B, Molhim M, Al-Rousan M (2005). Dynamic model of an HVAC system for control analysis. *Energy*, 30: 1729–1745.

Togari S, Araiz Y, Miura K (1993). A simplified model for predicting vertical temperature distribution in a large space. *ASHRAE Transactions*, 99(1): 84–99.

Wang S, Jin X (1998). CO 2-based occupancy detection for on-line outdoor air flow control. *Indoor and Built Environment*, 7: 165–181.

Wang L, Chen Q (2007). Validation of a coupled multizone-CFD program for building airflow and contaminant transport simulations. *HVAC&R Research*, 13: 267–281.

Wang H, Zhou P, Guo C, et al. (2019). On the calculation of heat migration in thermally stratified environment of large space building with sidewall nozzle air-supply. *Building and Environment*, 147: 221–230.

Wang J, Huang J, Feng Z, et al. (2021). Occupant-density-detection based energy efficient ventilation system: Prevention of infection transmission. *Energy and Buildings*, 240: 110883.

Wang J, Huang J, Fu Q, et al. (2022). Metabolism-based ventilation monitoring and control method for COVID-19 risk mitigation in gymnasiums and alike places. *Sustainable Cities and Society*, 80: 103719.

Wemhoff AP (2012). Calibration of HVAC equipment PID coefficients for energy conservation. *Energy and Buildings*, 45: 60–66.

Woradechjumroen D, Li H, Yu Y (2014). Development and performance evaluation of a steady-state virtual sensor for predicting wall surface temperature in light commercial buildings with an open space. *International Journal of Building, Urban, Interior and Landscape Technology*, 3: 5–20.

Woradechjumroen D, Li H (2015). Simplified instantaneous building load model for coordination control of multiple rooftop units. In: Proceedings of the 6th TSME International Conference on Mechanical Engineering, Petchburi, Thailand.

Woradechjumroen D, Yu Y, Li H (2016). Virtual partition surface temperature sensor based on linear parametric model. *Applied Energy*, 162: 1323–1335.

Yang X, Wang H, Su C, et al. (2020). Heat transfer between occupied and unoccupied zone in large space building with floor-level side wall air-supply system. *Building Simulation*, 13: 1221–1233.
Yao Y, Yang K, Huang M, et al. (2013). A state-space model for dynamic response of indoor air temperature and humidity. *Building and Environment*, 64: 26–37.

Yu Y, Megri AC, Jiang S (2019). A review of the development of airflow models used in building load calculation and energy simulation. *Building Simulation*, 12: 347–363.

Zhang S, Cheng Y, Fang Z, et al. (2017). Optimization of room air temperature in stratum-ventilated rooms for both thermal comfort and energy saving. *Applied Energy*, 204: 420–431.

Zhang S, Cheng Y, Fang Z, et al. (2018a). Dynamic control of room air temperature for stratum ventilation based on heat removal efficiency: Method and experimental validations. *Building and Environment*, 140: 107–118.

Zhang S, Cheng Y, Huan C, et al. (2018b). Heat removal efficiency based multi-node model for both stratum ventilation and displacement ventilation. *Building and Environment*, 143: 24–35.

Zhang S, Lin Z, Zhou P, et al. (2019). Fully mixed air model based cooling load estimation method for both stratum ventilation and displacement ventilation. *Energy and Buildings*, 199: 247–263.

Zhou P, Huang G, Li Z (2014). Demand-based temperature control of large-scale rooms aided by wireless sensor network: energy saving potential analysis. *Energy and Buildings*, 68: 532–540.

Zhou P, Huang G, Zhang L, et al. (2015). Wireless sensor network based monitoring system for a large-scale indoor space: data process and supply air allocation optimization. *Energy and Buildings*, 103: 365–374.

Zhou P, Wang J, Huang G (2017a). A coordinated VAV control with integration of heat transfer coefficients for improving energy efficiency and thermal comfort. *Energy Procedia*, 143: 271–276.

Zhou P, Wang J, Huang G (2017b). An evaluation of heat transfer coefficient in an independent zonal temperature controls with CFD. *Energy Procedia*, 105: 2260–2266.

Zhou P, Wang S, Jin Z, et al. (2022). Data reconstruction of wireless sensor network and zonal demand control in a large-scale indoor space considering thermal coupling. *Buildings*, 12: 15.