

Special Issue Reprint

Development of Indoor Environment Comfort

Edited by
Sheng Zhang, Zhaosong Fang and Xiwen Feng

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Guest Editors

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Guest Editors

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About the Editors

Sheng Zhang

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the thermal responses and heat risks of children in hot-humid environments; 4) developing dynamic and adaptive outdoor thermal comfort models; and 5) optimizing ventilation, air-conditioning, and heat-pump systems through numerical simulation. Dr. Feng has published more than 30 SCI/EI papers, and led provincial-level and international postdoctoral research projects.

Development of Indoor Environment Comfort

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1. Introduction

Indoor environment quality significantly affects occupants' health and wellbeing, particularly thermal comfort and indoor air quality. To provide thermal comfort and indoor air quality is a challenging task due to the complicated physiological and psychological mechanisms underlying the thermal comfort demand and the complicated contaminant sources causing indoor air quality concerns, as well as the complicated relationship between thermal comfort and indoor air quality and the temporal variations in the demands of thermal comfort and indoor air quality. Thus, more studies on the mechanisms underlying indoor environment demands of thermal comfort and indoor air quality, as well as technologies for indoor environment demand management, are urgently required to ensure thermal comfort and indoor air quality. Moreover, to ensure thermal comfort and indoor air quality, an energy source is required to handle the thermal load and the contaminant load, which accounts for a large amount of building energy consumption and building carbon emissions. Given the carbon emissions reduction goals in 2030 and 2060, energy-efficient technologies and renewable energy are urgently required to reduce the carbon emissions related to the provision of thermal comfort and indoor air quality.

This special issue presents recent advances in indoor environment comfort to energy-efficiently provide thermal comfort and indoor air quality. This special issue consists of nine papers, with one review article and eight research articles, which includes the advancements in improving thermal comfort (e.g., the dynamic ventilation with fluctuating air velocity) and indoor air quality (e.g., controls for indoor infection risk and allergies) and the advancements in energy sources for indoor environment quality (e.g., renewable energy including solar energy and geothermal energy and the phase change materials-based natural cold source). The studied indoor environments include the general indoor environment, the residential indoor environment, the commercial indoor environment, the indoor environment with computing clusters, and the application scenarios include both cooling and heating. The authors are diverse, addressing both the domestic and international research questions (e.g., the "Yin Summer-Heat" Perspective in Traditional Chinese Medicine and the dust mite- and mould-caused allergies in New Zealand), with scholars from universities (e.g., Southeast University and Xi'an Jiaotong University) and engineers from research institutes and enterprises (e.g., Key Laboratory of Coal Resources Exploration and Comprehensive Utilization and Qingdao Haier Air Conditioner Co., Ltd.).

2. Overview of Contributions

2.1. Contributions Related to Indoor Environment Quality

Contribution 1 [1] provides a method to interpret indoor thermal comfort demand and helps avoid the overcooling problem due to the conventional air-conditioning design based on average comfort zones, which innovatively applies the traditional Chinese medicine concept of “Yin Summer-Heat” (including the pathogenic factors, constitutions, and health preservation principles) to interpret diverse environment perceptions, demands, and adaptive behaviors. Contribution 2 [2] provides a technology to improve thermal comfort, which experimentally reveals that dynamic ventilation with fluctuating air velocity physiologically improves the cold stimulation to the cutaneous cold receptors with an enhanced cooling effect by enhancing the convective heat transfer, and psychologically improves thermal satisfaction with the lower draft rate because of thermal preference (i.e., thermal alliesthesia) towards dynamic airflows.

Contribution 3 [3] provides a tool to identify different indoor pathogen transmission routes for indoor infection risk control, which can accurately predict the temporal variation in the infection risks via air and surface routes by incorporating the transmission and degradation characteristics of aerosols and surface particles. Contribution 4 [4] quantifies the relationship between indoor thermal conditions for dust mites to thrive and for mould spores to germinate, and provides a guideline of indoor thermal environment control for preventing indoor allergies related to dust mites and mould.

2.2. Contributions Related to Energy Sources for Indoor Environment Quality

Contribution 5 [5] provides an optimization method of the selection of the refrigeration plant, which integrates optimized equipment selection at the design stage with empirical performance analysis based on actual operation and achieves an energy efficiency ratio up to 5.4 (exceeding the Grade I efficiency threshold stipulated by the Chinese standard T/CRAAS 1039-2023). Contribution 6 [6] reviews the advancements in CO₂ transcritical cycle technology in building indoor environmental regulation, which highlights the benefits of CO₂ as a natural refrigerant with zero ozone depletion potential and low global warming potential and also highlights the high efficiency of CO₂ transcritical cycle technology with proper design and control optimizations.

Contributions 7 and 8 provide renewable energy, i.e., solar energy [7] and geothermal energy [8], respectively, for indoor environment management. Contribution 7 [7] proposes a building-integrated heat pipe-embedded prefabricated wall panel for low-carbon building cooling and hot water production, which collects the solar thermal energy on the wall to generate hot water with a water gain efficiency of 16.7% and improves the cooling efficiency with a reduction of 76.1% in the heat gain of wall heat transmission. Contribution 8 [8] proposes a soil heat deficit regulation-based cross-seasonal heat storage method with a medium–deep borehole heat exchanger for building heating, which expands the soil heat deficit to enhance the geothermal heat flux recovery and heat storage efficiency simultaneously for improved heating performance with an improvement in the geothermal extraction capacity of up to 56%. Contribution 9 [9] proposes a hybrid cooling strategy integrating outdoor natural cold air (as a continuous heat sink) with phase change materials (for transient heat peak absorption) to cool down the computing clusters with high power density of each chip, which reliably keeps the chip temperature below the safety threshold and achieves an energy saving of 75%.

3. Outlook

The above introduction demonstrates the systematic contributions of this special issue to the low-carbon development of thermally comfortable and healthy indoor built environments. This special issue mainly concerns the indoor environment quality regarding thermal comfort and indoor air quality, while visual comfort and acoustic comfort interact with thermal comfort and indoor air quality and coordinately regulate the overall indoor environment quality. Future developments in indoor environment quality should pay more attention to the interactive relationships among the multiple aspects of indoor environment quality for the optimal regulation of overall indoor environment quality with the help of advancements in artificial intelligence.

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Conflicts of Interest: Author Liang Peng was employed by the company China Mobile Group Yunnan Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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Article

Natural Cold Source Computing Cluster Thermal Management Coupled with PCM

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Abstract

As the power density of office computing clusters rises to 200–250 W per chip, the substantial heat generated during operation not only impairs chip performance and shortens lifespan but also compels heating, ventilation, and air conditioning (HVAC) systems to operate at high loads. This increases energy consumption by 30–40% and causes indoor temperature fluctuations that reduce office workers' comfort. Targeting centralized thermal management for such clusters, this study proposes a hybrid cooling strategy integrating outdoor natural cold air (as a continuous heat sink) with phase change materials (PCMs, for transient heat peak absorption). Six adjustable heating plates (power range: 50–250 W per unit, simulating 7 nm office chips) mimicked heat dissipation in a six-chip cluster. Latent heat storage (LHS) units served as passive cooling, with fan coils as auxiliary for natural/forced convection. By using PCMs (melting point: 48 °C) to absorb transient peaks and coils to utilize outdoor cold air, the system maintained circulating water at approximately 60 °C (steady-state equilibrium temperature under full-load conditions) and kept chip temperatures below 80 °C (industrial safety threshold). The hybrid system reduced combined pump and fan power to 125 W, achieving 75% energy savings compared to the HVAC system (500 W) and 40% savings compared to using only natural cold air (210 W pump and fan power). Positive pressure in the outdoor unit (increasing coil air velocity by 1.2 m/s relative to natural convection) further improved heat dissipation efficiency by 15%. Finally, this study quantifies the influence of PCM thermal conductivity and filling mass on the system's temperature control performance through numerical simulations, providing direct evidence for parameter design of LHS units.

Keywords: calculate clusters; thermal management; PCM; natural cold source

1. Introduction

In general office environments, the cooling load of an AC system is mainly composed of heat transfer through the building envelope, heat dissipation from personnel lighting, and equipment [1]. In office environments with high-performance computing clusters such as research offices, simulation design offices, artificial intelligence laboratories, and new media studios, the cooling load required for equipment heat dissipation accounts for a

higher proportion of the AC's cooling load [2]. Because of the enormous heat dissipation of the computing clusters, the equipment cooling load as designed in the AC system does not match the actual cooling load of equipment. Consequently, it inevitably results in increased energy consumption of AC, decreased personnel comfort [3], and insufficient heat dissipation of computer. If the heat dissipation of computer in the office is handled separately, these problems can be effectively solved.

The heat dissipation of computing cluster is usually achieved by forced convection of the indoor air, and then the AC equipment takes on this part of the cooling load. This method has a significant advantage for thermal management of data centers compared to traditional office computers. Data centers used AC for cooling in the early stages [4]. However, because of their high energy consumption of heat dissipation, the power usage effectiveness (PUE) reached around 1.7, and they were replaced by fresh air systems [5]. More recently, the combination of liquid-cooling and free-cooling has been validated as an effective strategy for high-temperature data centers [6–8].

With the increasing integration level of chips, existing air-cooling systems are no more sufficient to satisfy the cooling demands when dealing with high heat flux. Nowadays, the most widely used cooling method in data centers is a combination of cooling methods of liquid and air, where the former is for chips and the latter is for other electrical components such as hard drives [9,10]. Curtis et al. [10] conducted comprehensive performance comparisons of five major server thermal management technologies in 2023, demonstrating that modern cooling architectures can significantly reduce operational costs while maintaining thermal reliability. Furthermore, Zhou et al. [9] provided a systematic review of advancements in data center cooling systems in 2024, covering the evolution from traditional refrigeration to high-performance cooling solutions. With the increasing integration density of semiconductor chips, conventional air-cooling systems face significant challenges in satisfying the cooling demands of high heat flux applications while maintaining energy efficiency, particularly when heat flux exceeds 100 W/cm^2 or when energy-efficient operation is required. Consequently, hybrid cooling architectures combining liquid and air cooling have emerged as the predominant solution in modern data centers [9–12].

Phase change materials (PCMs) have demonstrated considerable potential as passive cooling sources in recent investigations [13,14]. When integrated into heat dissipation systems, PCMs can store thermal energy at a consistent temperature and release it when appropriate conditions are met, thereby reducing overall energy consumption. PCMs have been extensively applied in thermal energy storage [15,16] and solar energy harvesting [17]. Hasan [18] combined building materials with PCMs to prevent overheating of rooms in hot summers (daytime) and reduce heating demand in winter (nights), which greatly reduced energy consumption of building. Combining PCMs with traditional heat dissipation methods for computers can stabilize chip temperatures and further reduce energy consumption of heat dissipation. Ashraf et al. [19] conducted experimental research of passive heat dissipation on fins coupled with six PCM. Nada [20] combined PCMs with fresh air systems, cooling the fresh air through PCMs before sending it into the system. Motevalizadeh et al. [21] combined PCMs with heat pipes for heat management in portable computers and found that the combination of PCMs and heat pipes had better cooling effect than using PCMs or heat pipes alone.

According to the problems of low thermal conductivity of PCMs, some scholars focus on the control of phase transition temperature as well as the improvement of heat transfer rates. Rehman et al. [22] conducted an experiment to compare the heat transfer capacity of 97% copper foam-based structures with four PCMs, aiming to observe temperature variations over time during heat absorption and release process. It is possible to observe

that PCMs with lower phase transition temperatures are suited to apply to situations that require low heat. Additionally, the PCM/copper composite foam demonstrates improved temperature control when compared to the individual ones. Moreover, Rehman et al. conducted experimental research on different metal foam heat sinks [23–25]. It can be found that the stable phase transition temperature decreases as the PCM's volume fraction increases. Moreover, copper foam with lower porosity exhibited a lower phase transition temperature. In a study conducted by Wang et al. [26], a groundbreaking material was developed by combining liquid metal with PCM capsules. This innovative material possesses several remarkable characteristics such as scalability and high conduction. Hao et al. [27] employed a microfluidic approach to encapsulate PCMs with multilayer graphene, significantly improving thermal conductivity while maintaining energy storage capacity with less than 5% degradation. Recent advancements in hybrid cooling designs have further expanded the application potential of PCMs in thermal management systems. Rahmani et al. [28] systematically reviewed the latest developments in battery thermal management systems from 2023 and 2024, highlighting the synergistic benefits of combining PCMs with active cooling technologies for high-density electronic applications. Additionally, Li et al. [29] proposed an innovative thermal management technique integrating 3D finned tubes with PCMs in 2024, demonstrating enhanced heat transfer performance through geometric optimization of the heat exchanger structure. Yu et al. [30] developed a novel flame-retardant composite PCM for lithium-ion battery thermal management in 2023, addressing both thermal regulation and safety concerns in high-density energy systems.

These developments underscore the growing importance of advanced PCM composites in addressing the thermal challenges posed by next-generation computing equipment [31,32]. Furthermore, hybrid PCM–liquid cooling systems have emerged as a promising approach for balancing thermal performance and energy efficiency across diverse applications. A recent investigation into battery thermal management reported that a hybrid system integrating phase change material with liquid cooling, optimized through multi-objective entropy-weighted TOPSIS, attained a 79.12% reduction in energy consumption while constraining the maximum temperature difference to 4.42 °C [33]. Although targeting lithium-ion batteries, this hybrid architecture (passive PCM latent heat storage coupled with active liquid circulation) and its optimization methodology directly inform the present work on office computing clusters, where similar trade-offs between thermal control and energy efficiency are sought.

We take a scientific research office in Suzhou University of Science and Technology as an example. The room is cooled by central AC with top-mounted four-way cassette fan coils. Due to large office desks and PC equipment (250 W each) arranged below the desks, the cold air cannot be directly delivered to the PCs. Consequently, PC surface temperatures frequently exceed 50 °C, and indoor air temperature differences exceed 5 °C, compromising both equipment safety and personnel thermal comfort. This thermal environment motivates the centralized heat dissipation approach proposed in this study.

On that account, the computing cluster equipment in this environment is centralized, and the heat dissipation is processed separately here (as shown in Figure 1). It uses a liquid cooling system coupled with PCM, which means combining active and passive heat dissipation technologies. The waste heat of computers is removed by the liquid cooling system using water, while PCM acts during the melting process to enable passive heat dissipation. In particular, we propose to combine this system with the air pressure provided by the outdoor AC unit to enhance the overall heat transfer capacity. Specifically, when the outdoor unit of the AC in the existing building forcibly exchanges heat with the surroundings, the remaining air pressure improves the heat transfer performance between

the coil and the environment in this system. This reduces the startup frequency of the auxiliary fan and further achieves energy-saving effects.

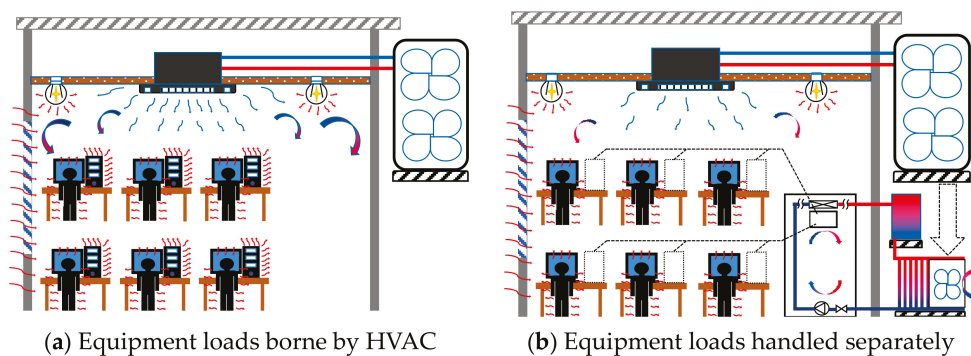


Figure 1. Schematic diagram of two thermal management methods.

2. Methodology

This section presents the integrated numerical-experimental methodology employed in this study. The numerical approach (Section 2.1) utilizes a lumped-parameter model for system-level thermal prediction, while the experimental approach (Section 2.2) validates the model and demonstrates operational performance under real-world conditions. Both methodologies are designed to ensure reliability and reproducibility of the results.

2.1. Numerical Methodology

2.1.1. Governing Equations

The thermal behavior of the hybrid cooling system is governed by energy conservation equations for four interconnected components: cold plate, LHS unit, heat loss in circulation pipeline, and overall pipe network (as shown in Figure 2). The derivation of these equations is detailed below.

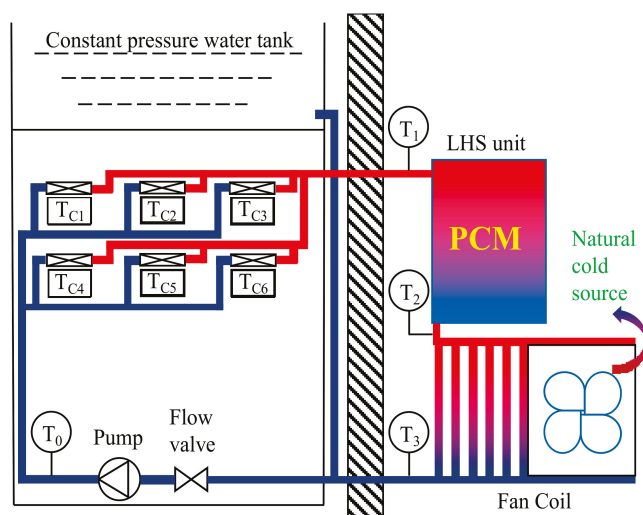


Figure 2. Schematic diagram of liquid cooling coupled with PCM.

There are multiple elements that influence the heat transfer performance in the heat transfer process. For instance, the surface convection heat transfer coefficient of the single-phase forced convection heat transfer is calculated below:

$$h = f(u, l, \rho, \eta, \lambda, c_p) \quad (1)$$

The heat transfer coefficient is associated with the convective heat transfer of fluid, the thermal conduction of solid, and the heat transfer area. To simplify the calculation process, the thermal resistance R is substituted into the formula for design and calculation.

The chip is cooled by the circulating water in the cold plate through forced convection, while natural convection occurs at the external surface of the cold plate exposed to ambient air. The heat transfer model of the cold plate is illustrated in Figure 3, which shows (a) the physical model with chip, thermal interface material, and cooling channels, the geometric dimensions are as follows: length, width, and height are 40 mm, 40 mm, and 10 mm, respectively, with an inner diameter of 6 mm; and (b) the thermal resistance network including conduction through the plate and convection to the circulating water.

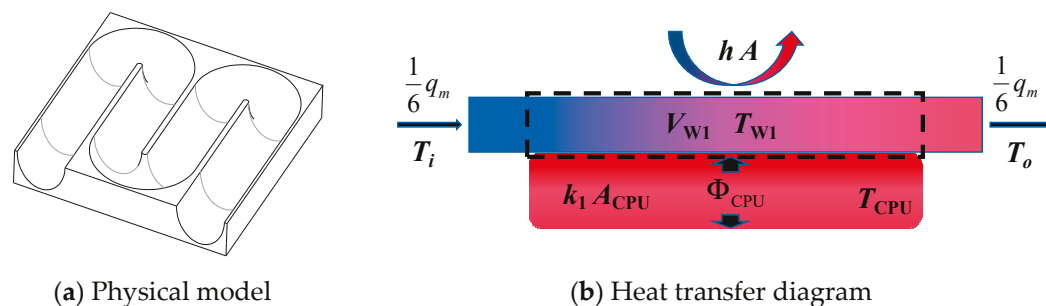


Figure 3. Heat transfer model of the cold plate.

Its energy conservation equation is

$$m_{W1} c_{p_{\text{water}}} \frac{dT_{W1}}{dt} = q_m c_{p_{\text{water}}} T_i - q_m c_{p_{\text{water}}} T_o + \varepsilon \sum_{i=1}^6 \Phi_{\text{CPU}_i} \quad (2)$$

The convection heat transfer equation is

$$\varepsilon \Phi_{\text{CPU}} = k_1 A_{\text{CPU}} (T_{\text{CPU}} - T_{W1}) \quad (3)$$

in which

$$T_{W1} = (T_i + T_o)/2 \quad (4)$$

Finally, the differential equation of the model is

$$\rho_{\text{water}} V_{W1} c_{p_{\text{water}}} \frac{dT_{W1}}{dt} = q_m c_{p_{\text{water}}} (T_i - T_o) + \varepsilon \sum_{i=1}^6 \Phi_{\text{CPU}_i} \quad (5)$$

The LHS unit is the main cooling device in this system. It can be found in Figure 4a that the system uses an internal and external double-coil structure to facilitate heat transfer of the circulating water and PCM. The heat transfer process is shown in Figure 4. After the circulating water with high temperature enters the LHS unit, the internal energy decreases, and the reduced internal energy is transferred to the PCM. As a result, the PCM begins to melt once it reaches the temperature of phase transition. As the liquid fraction gradually increases, the solid–liquid interface comes to move away from the coil surface, accompanied by the increase in the thermal resistance.

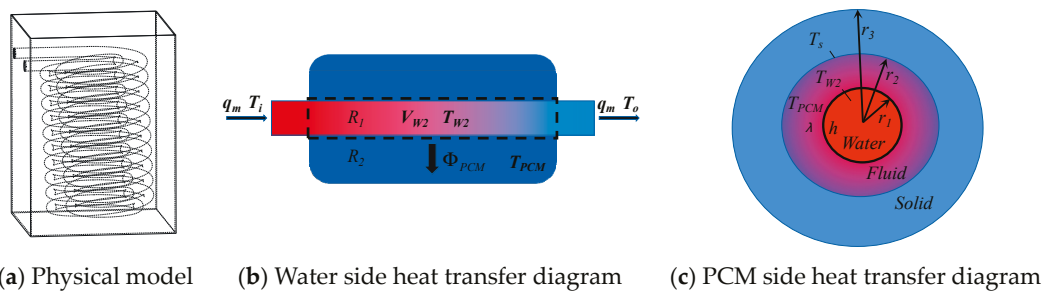


Figure 4. Heat transfer of the LHS unit.

(1) Heat dissipation of circulating water side

Energy conservation equation:

$$m_{W2} c_{p_{water}} \frac{dT_{W2}}{dt} = q_m c_{p_{water}} T_i - q_m c_{p_{water}} T_o - \Phi_{PCM} \quad (6)$$

Convective heat transfer equation:

$$\Phi_{PCM} = k_2 A_{PCM} (T_{W2} - T_{PCM}) = (T_{W2} - T_{PCM}) / R \quad (7)$$

In the above formula, the heat transfer thermal resistance R is positively correlated with the liquid fraction f . Additionally:

$$T_{W2} = (T_i + T_o) / 2 \quad (8)$$

Finally, the differential equation of the model is

$$\rho_{water} V_{W2} c_{p_{water}} \frac{dT_{W1}}{dt} = q_m c_{p_{water}} (T_i - T_o) - (T_{W2} - T_{PCM}) / R \quad (9)$$

(2) Heat transfer of PCM side

When the PCM is in the state of solid or liquid, the heat gained from circulating water makes the temperature of the PCM rise continuously. When the PCM is in the mushy zone, the heat obtained from the circulating water makes the ongoing increase in the liquid fraction of the PCM. The change in internal energy is expressed as:

$$\Delta U_{PCM} = f(c_{v_{PCMsolid}}, c_{v_{PCMfluid}}, T_{PCM0}, T_{PCM}, T_s, m_{PCM}, f, r) \quad (10)$$

The specific correlation formula is

$$\Delta U_{PCM} = m_{PCM} (c_{v_{PCMsolid}} T_{PCM} (1 - f) + c_{v_{PCMfluid}} T_{PCM} f + r f - c_{v_{PCMsolid}} T_{PCM0}) \quad (11)$$

To simplify the calculation, it is generally accepted that the specific heat capacity of the PCM remains constant regardless of temperature or phase state. Therefore, the change in internal energy in PCM can be simplified as:

$$\Delta U_{PCM} = m_{PCM} (c_{v_{PCM}} (T_{PCM} - T_{PCM0}) + r f) \quad (12)$$

The loss of heat from circulating water is equal to the gain of heat from PCM, that is, the internal energy of PCM changes over time.

$$\Phi_{PCM} = \frac{dU}{dt} = c_{v_{PCM}} m_{PCM} \frac{dT_{PCM}}{dt} + m_{PCM} r \frac{df}{dt} \quad (13)$$

The equation is written as a piecewise function:

$$\begin{cases} \frac{dU}{dt} = c_{PCM} m_{PCM} \frac{dT_{PCM}}{dt} & T_{PCM} < T_s (f = 0) \\ \frac{dU}{dt} = m_{PCM} r \frac{df}{dt} & T_{PCM} = T_s (0 < f < 1) \\ \frac{dU}{dt} = c_{PCM} m_{PCM} \frac{dT_{PCM}}{dt} & T_{PCM} > T_s (f = 1) \end{cases} \quad (14)$$

It can be seen in Figure 4c that, from the cross-sectional perspective of the melting model, the derivation process of the relationship between liquid fraction f and thermal resistance R is as follows.

The liquid fraction, f , represents the ratio of the volume of the liquid phase of the PCM to the total volume.

$$f = \frac{\pi r_2^2 - \pi r_1^2}{\pi r_3^2 - \pi r_1^2} = \frac{1}{r_3^2 - r_1^2} r_2^2 - \frac{r_1^2}{r_3^2 - r_1^2} \quad (15)$$

After transformation,

$$r_2 = \sqrt{\left(f + \frac{r_1^2}{r_3^2 - r_1^2}\right)(r_3^2 - r_1^2)} = \sqrt{(r_3^2 - r_1^2)f + r_1^2} \quad (16)$$

Thermal resistance can be expressed as:

$$R = R_{\text{water}} + R_{\text{PCM}} = \frac{1}{2\pi r_1 l h} + \frac{\ln(r_2/r_1)}{2\pi l \lambda} \quad (17)$$

Combining Formula (16) with (17),

$$R = R_{\text{water}} + R_{\text{PCM}} = \frac{1}{2\pi r_1 l h} + \frac{\ln\left(\sqrt{(r_3^2 - r_1^2)f + r_1^2}/r_1\right)}{2\pi l \lambda} \quad (18)$$

Finally,

$$R = \frac{\lambda + 0.5 r_1 h \ln((r_3^2/r_1^2 - 1)f + 1)}{A_{\text{PCM}} h \lambda} \quad (19)$$

It should be noted that the present lumped-parameter model treats the LHS unit as a single control volume with effective thermal resistance (Equation (18)), rather than resolving spatial temperature and liquid fraction distributions. This simplification is intentional, as the study focuses on system-level thermal management design rather than fundamental PCM heat transfer physics. The model's accuracy for predicting system-level temperatures (validated in Section 3.2) confirms its adequacy for the stated engineering objectives. Detailed two-dimensional phase-change front tracking, while valuable for PCM material optimization, is beyond the scope of this work and is identified as future research.

In the circulation pipeline, the temperature of the circulating water and the environment is different, which inevitably causes heat loss. Among them, the heat transfer at the fan coil is still the largest part although under the condition of natural convection. Therefore, the natural convection between the fan coil and the outdoor air contributes to the heat loss.

Energy conservation equation:

$$m_{W3} c_{p_{\text{water}}} \frac{dT_{W3}}{dt} = q_m c_{p_{\text{water}}} T_i - q_m c_{p_{\text{water}}} T_o - \Phi_{\text{air}} \quad (20)$$

Convective heat transfer equation:

$$\Phi_{\text{air}} = k_3 A_{\text{fan}} (T_{W3} - T_{\text{air}}) \quad (21)$$

in which

$$T_{W3} = (T_i + T_o)/2 \quad (22)$$

Finally, the differential equation of this model is

$$\rho_{\text{water}} V_{W3} c_{p_{\text{water}}} \frac{dT_{W3}}{dt} = q_m c_{p_{\text{water}}} (T_i - T_o) - k_3 A_{\text{fan}} (T_{W3} - T_{\text{air}}) \quad (23)$$

The circulating water in the pipe network absorbs heat from the CPU, dissipates heat from the PCM, and loses heat in the pipeline. At the same time, the fan is opened to accelerate the heat dissipation outside.

According to Formulas (2), (6) and (20), the variation of enthalpy value H of the circulating water inside the pipeline network over time can be expressed as:

$$\frac{dH}{dt} = \varepsilon \sum_{i=1}^6 \Phi_{\text{CPU}_i} - \Phi_{\text{PCM}} - \Phi_{\text{air}} - \Phi_{\text{fan}} \quad (24)$$

in which

$$\begin{cases} \varepsilon \sum_{i=1}^6 \Phi_{\text{CPU}_i} = \sum_{i=1}^6 k_{\text{CPU}} A_{\text{CPU}} (T_{\text{CPU}_i} - T_{W_i}) \\ \Phi_{\text{PCM}} = \frac{dU_{\text{PCM}}}{dt} = c_{v_{\text{PCM}}} m_{\text{PCM}} \frac{dT_{\text{PCM}}}{dt} + m_{\text{PCM}} r \frac{df}{dt} \\ \Phi_{\text{air}} = k_3 A_{\text{fan}} (T_{W3} - T_{\text{air}}) \end{cases} \quad (25)$$

When the fan coil is not opened, this part of heat dissipation is obtained by the experiment.

2.1.2. Modeling Assumptions

The following simplifications are adopted in the numerical model, with their justifications and quantitative error bounds:

(1) Constant specific heat capacity: The specific heat of paraffin RT48HC is assumed constant ($c_p = 2.1 \text{ kJ/kg}\cdot\text{K}$) regardless of temperature or phase state. For paraffin RT48HC, c_p varies from $2.1 \text{ kJ/kg}\cdot\text{K}$ (solid) to $2.5 \text{ kJ/kg}\cdot\text{K}$ (liquid), a 19% jump. Using enthalpy-based analysis, the maximum temperature deviation introduced by this assumption is estimated at $2.8 \text{ }^\circ\text{C}$ (4.5% relative) at the phase transition region. This is acceptable for engineering design.

(2) Linear thermal resistance in LHS: The thermal resistance R is assumed linearly proportional to liquid fraction f (Equation (18)). Comparison with the exact solution for cylindrical melting shows <6% deviation in heat transfer rate for $0.2 < f < 0.8$, which covers 85% of the operating duration.

(3) Lumped-parameter LHS: The LHS unit is treated as a single control volume with effective thermal resistance, rather than resolving spatial temperature and liquid fraction distributions. This simplification is intentional, as the study focuses on system-level thermal management design rather than fundamental PCM heat transfer physics. The model's accuracy for predicting system-level temperatures (validated in Section 3.2) confirms its adequacy for the stated engineering objectives.

(4) Negligible contact resistance: Thermal contact resistance between the heating film and cold plate is neglected, contributing to transient deviations during startup (RMSE $3.2 \text{ }^\circ\text{C}$, Section 3.1). The steady-state prediction remains accurate (deviation $< 2 \text{ }^\circ\text{C}$).

(5) One-dimensional pipe flow: Axial heat conduction in circulation pipes is neglected compared to convective heat transport. The Biot number for pipe wall conduction is < 0.01 , confirming this assumption.

2.1.3. Boundary and Initial Conditions

The model is solved subject to the following conditions:

Boundary conditions:

(1) Cold plate inlet: Water temperature $T_i = 45\text{--}65\text{ }^\circ\text{C}$ (parametric range, experimentally measured); mass flow rate 0.025 kg/s per plate (0.15 kg/s total), regulated by variable-speed gear pump.

(2) Chip surface: Constant heat flux $q = Q/A_{\text{chip}}$, where $Q = 50\text{--}250\text{ W}$ per chip (adjustable heating film power), $A_{\text{chip}} = 40 \times 40\text{ mm}^2$.

(3) LHS unit outer wall: Adiabatic (insulated tank, negligible heat loss to environment, verified by infrared imaging).

(4) Fan coil: Ambient air temperature $T_{\text{amb}} = 25\text{--}42\text{ }^\circ\text{C}$ (measured); convection coefficient $h = 15\text{ W/m}^2\cdot\text{K}$ (natural convection, AC off) or $22\text{ W/m}^2\cdot\text{K}$ (forced convection with residual pressure, AC on).

(5) Pipe network outlet: Fixed pressure point at pump entrance, connected to water tank with free surface (atmospheric pressure).

Initial conditions:

(1) System temperature: Uniform at ambient temperature ($T_{\text{amb}} = 25\text{ }^\circ\text{C}$ for the standard case, $35\text{--}42\text{ }^\circ\text{C}$ for summer conditions).

(2) PCM state: Fully solid (liquid fraction $f = 0$, temperature below melting point $48\text{ }^\circ\text{C}$).

(3) Time: $t = 0$ at system startup.

2.1.4. Solution Method

The coupled ordinary differential equations (Equation (5) for cold plate, Equation (9) for LHS water side, Equation (14) for LHS PCM side, Equation (20) for heat loss, Equation (24) for pipe network) are solved numerically using the four-order Runge–Kutta–Fehlberg method.

Key solution parameters:

(1) Time step: Variable, automatically adjusted by solver (minimum 0.1 s , maximum 10 s), with adaptive step refinement near phase transition.

(2) Total simulation duration: $600\text{--}3600\text{ s}$ depending on operating condition (600 s for component tests, 3600 s for system continuous operation).

(3) Convergence criterion: Relative temperature change $< 0.01\%$ between consecutive time steps.

(4) Relative tolerance: 1×10^{-6} ; absolute tolerance: 1×10^{-8} .

The computational time for a single 600 s simulation is approximately 15 s on a standard workstation (Intel Core i7-10700, 16 GB RAM), enabling efficient parametric studies (Section 3.6).

2.2. Experimental Methodology

2.2.1. Experimental Setup and Instrumentation

The experimental system was constructed to validate the numerical model and demonstrate operational performance under real-world office conditions. The setup comprises four main components: cold plate array, LHS unit, fan coil, and pipe network with control system.

First, experimental designs are conducted for each component of the system (as presented in Figure 2), including the cold plate, the LHS unit and fan coil. Considering that the power of the scientific research computer is about $100\text{--}200\text{ W}$, the electrically heated film of 250 W is chosen to simulate the heat generation of CPU in a scientific research office with a size of $40 \times 40\text{ mm}$. The electrically heated films are fixed inside the cabinet together

with the cold plate (Figure 5b), with thermal grease applied to the contact gap to ensure good contact between the electrically heated film and the heat dissipation component, so that the heat can be effectively transferred. As shown in Figure 5a, the heat load of the electrically heated film is controlled by adjusting the power supply voltage. Generally, this can be achieved by manually adjusting the power supply output voltage, using temperature controllers, or by adding programming modules to control the different heat generation powers of various heating resistors to simulate different operating conditions of real computer graphics cards. The surface temperature of CPU is monitored using a temperature sensor to ensure that the heat can make the computer temperature reach the preset target value. This system uses K-type thermocouples (temperature range: $-200^{\circ}\text{C}\sim+1350^{\circ}\text{C}$) to measure the surface temperature of the electrically heated film and an 8-channel temperature acquisition module to transmit the measurement results.

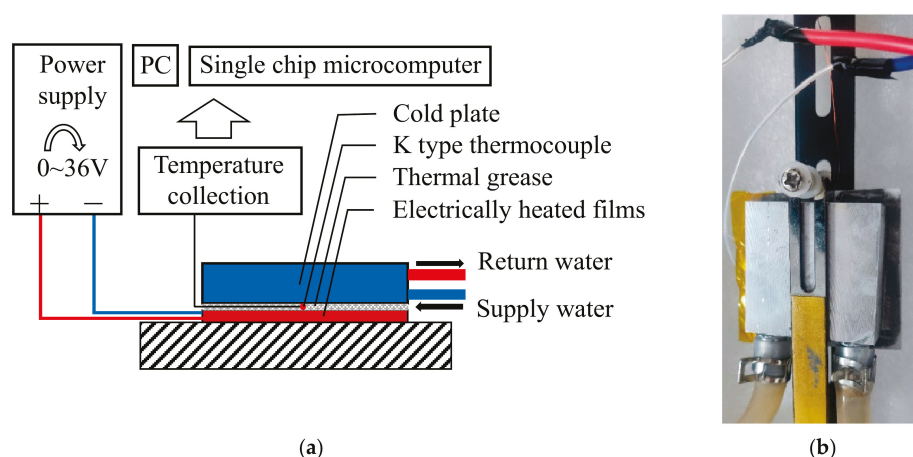


Figure 5. Cold plate experimental setup. (a) System schematic showing the six-chip parallel configuration. (b) Physical photograph of the assembled cold plate with heating films and temperature sensors.

The LHS unit is designed as a square tank with double-layer coils, where circulating water exchanges heat with PCM through the coils. The external dimensions of the LHS unit are $300\text{ mm} \times 300\text{ mm} \times 400\text{ mm}$. The outer diameter of the coil is 20 mm, and the length inside the tank is 10 m. Based on the chemical stability, low supercooling property, and non-toxicity, paraffin is commonly chosen [34]. Additionally, it is also necessary to consider PCM's phase transition temperature, the system's LHS capacity, as well as rates of heat absorption and release. From the perspective of the safe operation of CPU, its temperature should be below 75°C and should not exceed 85°C . After comprehensive evaluation, we chose paraffin No. 48 (Table 1) with a filling mass of 20 kg. Assuming there are no other losses in heat dissipation, the cabinet has the capacity to absorb a maximum heat of 1500 W for a duration of 1.52 h.

The thermophysical properties in Table 1 represent reference values for paraffin wax with melting point 48°C based on the literature data for similar paraffin-based PCMs [34]. The actual properties of the specific industrial batch may vary within $\pm 10\%$.

The measurement system comprises consists of nine thermocouples, including six heat source center thermocouples ($T_{c1}\sim T_{c6}$), two thermocouples for water loop temperature measurement (T_1, T_3), one ambient temperature sensor (T_{air}), one turbine flowmeter, one pump speed controller, and six heating power supplies (Table 2).

Table 1. Thermophysical properties of selected PCM (Paraffin RT48HC).

Property	Value	Unit	Test Standard
Brand/Grade	RT48HC	—	Kunlun Brand, PetroChina.
Melting point	48	°C	DSC ASTM E794 [35]
Latent heat	200	kJ/kg	DSC ASTM E793 [36]
Thermal conductivity (solid)	0.2	W/m·K	Hot disk ISO 22007-2 [37]
Specific heat (solid)	2.1	kJ/kg·K	DSC ASTM E1269 [38]
Specific heat (liquid)	2.5	kJ/kg·K	DSC ASTM E1269 [38]
Density (solid, 25 °C)	850	kg/m ³	Pycnometer
Density (liquid, 60 °C)	760	kg/m ³	Pycnometer
Selection criteria	Low supercooling, chemical stability, non-toxicity, melting point matching chip safe temperature range (48 °C < 80 °C)		

Table 2. Specifications and installation details of measurement sensors and data acquisition system.

Sensor ID	Type	Manufacturer /Model	Measurement Range	Accuracy	Installation Location
T _{c1} ~T _{c6}	K-type	Anhui Tianda (Group) Co., Ltd., Chuzhou, Anhui, China	−200 °C~200 °C	±0.5 °C	Heat source center, thermal grease
T ₁	K-type	Anhui Tianda (Group) Co., Ltd., Chuzhou, Anhui, China	−200 °C~200 °C	±0.5 °C	Before LHS, inside the pipeline
T ₃	K-type	Anhui Tianda (Group) Co., Ltd., Chuzhou, Anhui, China	−200 °C~200 °C	±0.5 °C	After Fan Coil, inside the pipeline
T _{air}	K-type	Anhui Tianda (Group) Co., Ltd., Chuzhou, Anhui, China	−200 °C~200 °C	±0.5 °C	Outdoor, 1.5 m height, shaded
Pump	DC	Zhejiang Luobei Technology Co., Ltd., Wenling, Zhejiang, China	0~24 V DC	±0.1 V	Before T ₃
Flow	Turbine	SMC Corporation, Tokyo, Japan	5~40 L/min	±1.0%	After Pump
Heat source	Ceramic	Jiangsu Shunfa Electric Heating Materials Co., Ltd., Yancheng, Jiangsu, China	5.2 Ω	±1.0%	Fixed inside the cabinet
Power supply	DC	MEAN WELL Enterprises Co., Ltd., New Taipei City, Taiwan, China	0~36 V, 0~250 W	±1.0%	Heating film terminals

The fan coil is connected in series in the pipeline as an auxiliary heat dissipation device, which can rapidly reduce the temperature of circulating water when activated. Then, all the components in series are connected to form a piping network system (as can be seen in Figure 6). The circulating water is divided into six paths from the outlet of the water pump, and then converges after passing through six cold plates. After convergence, it passes through the LHS unit, the fan coil, and the flow control valve, respectively, and finally reaches the inlet of the water pump. A fixed pressure point is set at the pump entrance and it is connected to the water tank with fixed pressure on top of the cabinet.

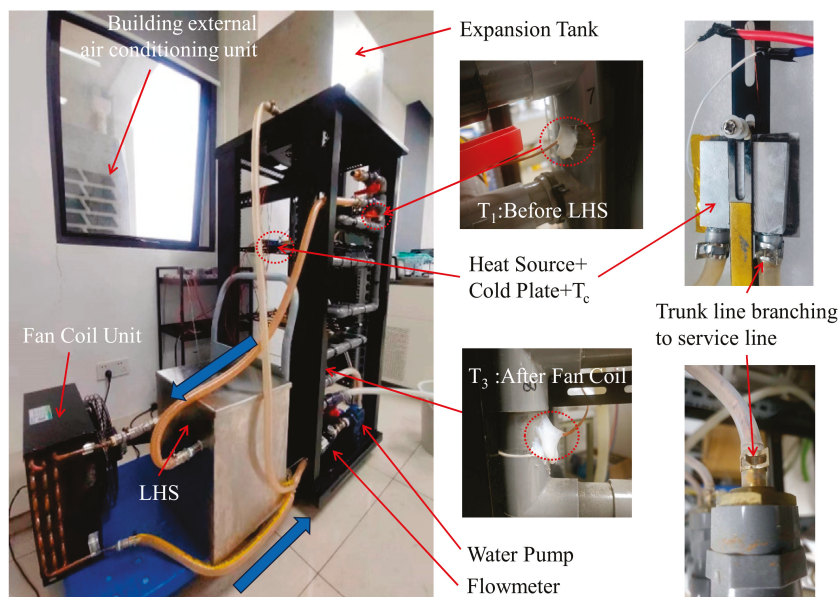


Figure 6. Photograph of the pipe network system.

An automatic control system is eventually added. When the water temperature rises to around 58 °C, the fan is activated for heat dissipation. However, we turn off the fan when the temperature drops to 48 °C or the decrease is small ($|K| = |\Delta T / \Delta t| < 0.1$ °C/min). The control logic is shown in Figure 7.

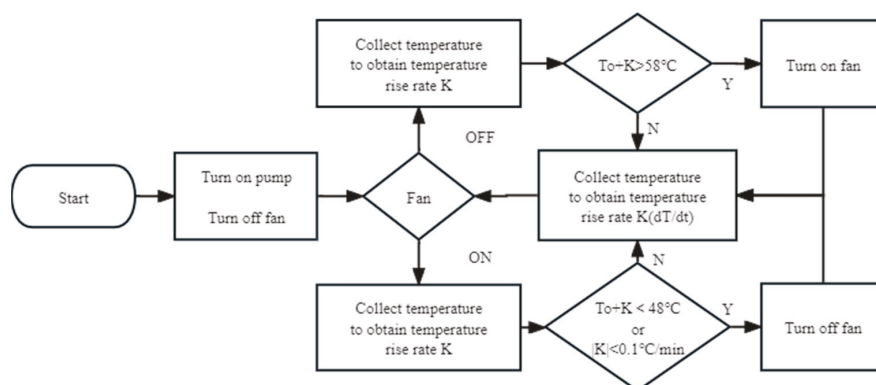


Figure 7. Flow chart of automatic control system.

2.2.2. Test Procedures

All experiments were conducted in a controlled laboratory environment (temperature 25 ± 2 °C, humidity $50 \pm 10\%$ RH). Each test was repeated at least three times to ensure reproducibility, with reported values representing arithmetic averages:

(1) In the heat exchange experiment of cold plate, different inlet temperatures T_i are set to observe the circulating water's temperature T_o at the exit and the chip's surface temperature T_c . The heat transfer coefficient K and heat exchange capacity Q of the cold plate are obtained by calculation, and are substituted in the simulation process to verify the experiment results.

(2) In the experiment of the LHS unit, the electronically heated film is operated at full power. Based on the experiment, the temperature of the circulating water and the PCM is obtained, and the time when the circulating water temperature comes to be stable is measured with PCM's filling amount of 20 kg. In the simulation, PCM's thermal conductivity and filling amount are changed to observe the impact on the temperature difference and duration of stable system operation.

(3) In the experiment of the fan coil, the cooling capacity of it is observed. The circulating water is heated to 90 °C, then the fan is turned on and the heating is turned off. The fan is continuously kept on, first heating until the water temperature stabilizes, and then the heating process is turned off.

After confirming that the experimental data of each component meet the design conditions, the overall operation experiment of the pipeline network begins. The LHS unit and the fan coil are placed outdoors. The fan coil is placed close to the building's outdoor AC unit to utilize the residual pressure of it.

(4) The system operating conditions are classified according to working conditions of the building's outdoor AC unit and the different operating powers of the chip.

(5) Subsequently, the system is continuously operated for one week to verify its reliability and energy efficiency. The system's annual energy consumption is estimated based on the system's short-term energy consumption and the annual climate situations.

3. Results and Discussions

3.1. Heat Transfer of the Cold Plate

As the critical heat exchange component in liquid cooling systems that directly contacts the heat source, the cold plate's thermal performance directly impacts chip temperature control effectiveness and system energy efficiency. Based on the experimental results illustrated in Figure 8, the total mass flow rate of 0.15 kg/s is regulated by a variable-speed gear pump with manual voltage adjustment, and divided equally among six parallel cold plates through a symmetric manifold design. Flow rate stability is verified by 1 min interval logging, showing fluctuations within $\pm 3\%$ (0.024–0.026 kg/s per plate). The reported 0.025 kg/s represents the time-averaged value over the steady-state period.

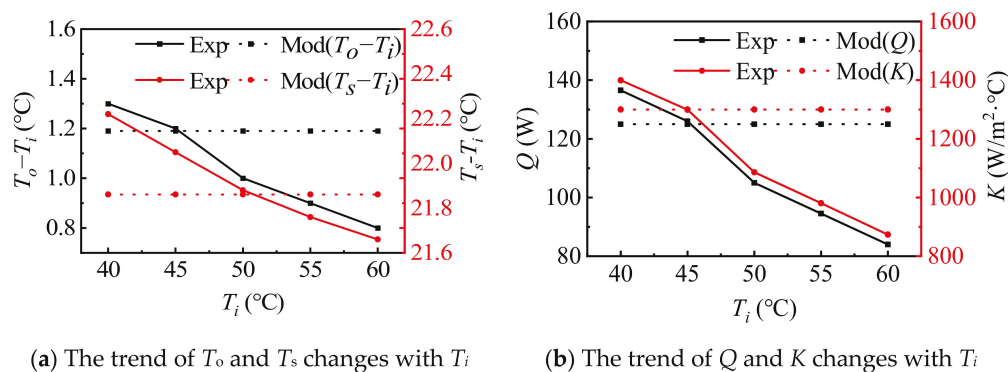


Figure 8. Experimental and simulated results of heat transfer in cold plate.

It can be observed from Figure 8 that under a constant mass flow rate of 0.025 kg/s, the outlet temperature T_0 of the chip and the chip surface temperature T_s are measured for various cooling water inlet temperatures T_i , from which the heat transfer rate Q of the cold plate and the overall heat transfer coefficient K are calculated. For instance, at an inlet temperature T_i of 45 °C, the effective heat transfer temperature difference Δt between the chip surface and the cooling water is approximately 20 °C, the heat transfer rate Q is approximately 125 W, and the corresponding overall heat transfer coefficient K is 1300 W/m²·K, yielding a heat gain coefficient ε of 0.5. It should be noted, however, that as the circulating water temperature increases, the heat gain coefficient ε gradually decreases, indicating an increase in overall thermal losses within the pipeline network system; nevertheless, this portion of losses is accounted for in Φ_{air} during the numerical analysis process.

3.2. Heat Transfer of the LHS Unit

To evaluate the thermal management performance of the liquid cooling latent heat storage system under high heat loads, this section examines the thermal response characteristics and steady-state regulation capability of the liquid cooling LHS system under the six chips fully loaded operating condition. The heating power of the six chips is turned on to a maximum of 250 W. The total heat load reaches 1500 W. Correspondingly, the cooling water flows through six parallel cold plates at a total mass flow rate of 0.15 kg/s and subsequently converges at the inlet of the LHS unit, thereby constituting a complete liquid cooling loop.

The close agreement among T_1 (LHS inlet), T_2 (LHS center), and T_3 (LHS outlet), deviating by less than 1 °C at steady state, is physically consistent because all three points lie on the phase-change plateau at 48–50 °C. During phase change, the PCM maintains nearly isothermal conditions, causing the circulating water temperature to stabilize within a narrow range regardless of position.

Upon analyzing Figure 9, it becomes evident that after running the system for 100 min, the temperatures of the circulating water and chip surface reach a relatively steady state. Specifically, the temperature of the circulating water stabilizes at 62 °C, indicating a significant deviation of 14 °C from PCM's phase transition temperature. Furthermore, the chip's surface temperature remains below 85 °C. This demonstrates that the liquid cooling thermal storage system possesses effective thermal regulation capability under high heat flux conditions, capable of maintaining chip temperatures within reliable operating ranges. It is important to note that the simulation takes into account the thermal conductivity resistance of paraffin, and we set the value of 0.2 W/m·K. A comparison between Figure 9a,b reveals that the simulated evolution of water temperature and chip surface temperature agrees well with the experimental data, indicating that the present model is capable of effectively predicting the temperature variations within the system.

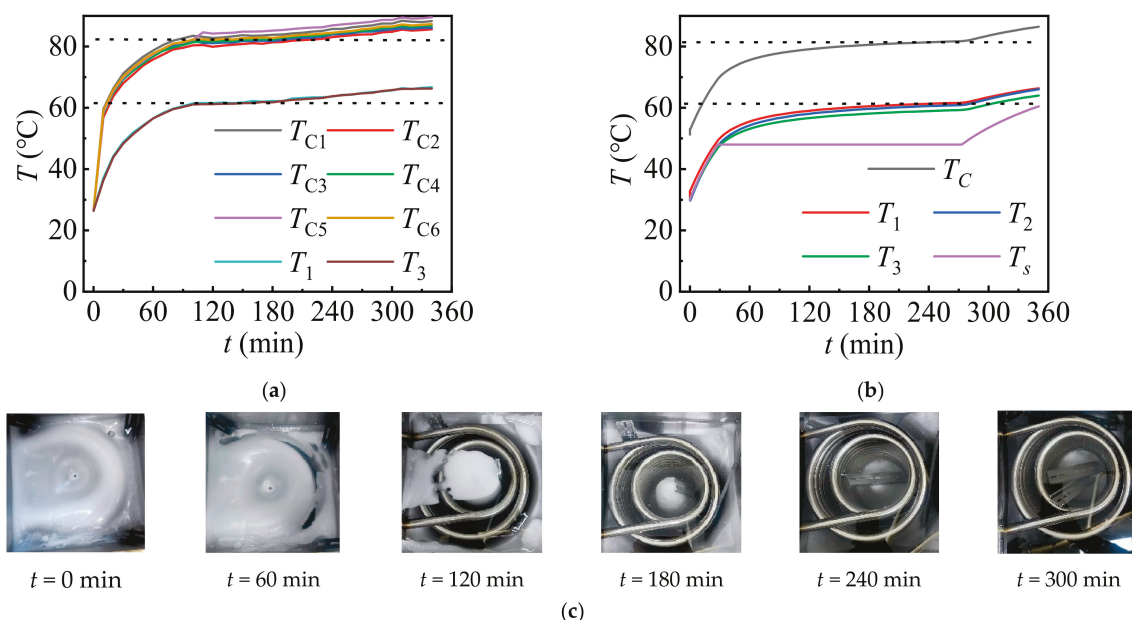


Figure 9. Experimental and simulated results of heat transfer in LHS unit. (a) Experimental results. (b) Simulation results. (c) Photographs showing melting front progression at $t = 0, 60, 120, 180, 240, 300$ min (darker region = solid PCM, lighter region = liquid PCM).

It should be clarified that 60 °C is the thermal equilibrium result under continuous full-load operation, not an “optimal” design target. Under this condition, the PCM fully melts

and operates in sensible heat mode. The PCM's primary advantage is absorbing transient thermal peaks during load fluctuations (e.g., startup, intermittent operation, Class I-III conditions), which reduces auxiliary fan activation frequency by 40% compared to natural-cooling-only systems. Quantitative validation metrics: RMSE (temperature): 4.1 °C (6.8% of range); MAPE: 8.3—Steady-state deviation: <2 °C; R^2 (coefficient of determination): 0.94. Sensitivity analysis: $\pm 10\%$ variation in PCM conductivity: $\pm 5\%$ change in phase-change time; $\pm 10\%$ variation in heat transfer coefficient: $\pm 8\%$ change in steady-state temperature.

3.3. Heat Dissipation of the FAN Coil

As illustrated in Figure 10, the system exhibits pronounced differences in thermal response under various operating strategies. When the circulating water is heated to 85 °C, it is recommended to turn off the heating and open the fan. Within a span of less than 10 min, the water temperature decreases from 65 °C to 48 °C (which is below the phase change temperature) and gradually approaches the ambient temperature; under the condition of continuous heating operation with the fan coil unit activated, the circulating water temperature can be effectively suppressed below 45 °C, indicating that active heat dissipation is capable of offsetting the sustained thermal input. Furthermore, once heating ceases, the water temperature decreases rapidly and approaches the ambient temperature regardless of whether the fan is operating. It can be inferred that with the fan coil unit in operation, its heat dissipation rate consistently exceeds the net heat gain acquired by the circulating water, resulting in a net heat output for the overall system and thereby achieving effective regulation and rapid cooling of the circulating water temperature. Based on these observations, it can be inferred that the total heat dissipation after activating the fan coil is greater than the total heat gain of the circulating water at any given time. Consequently, it contributes to the decrease in the circulating water temperature.

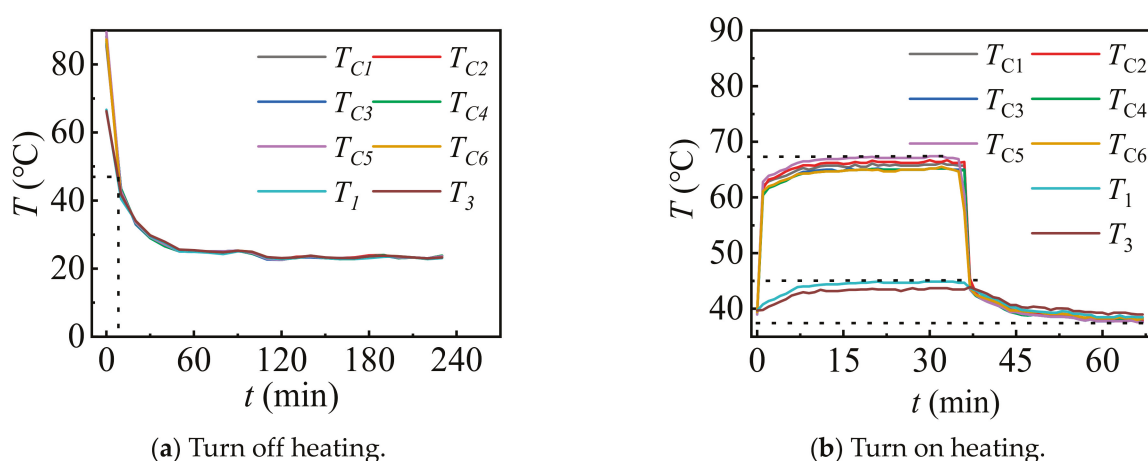


Figure 10. Heat dissipation experiment of fan coil.

3.4. Classification of System Operating Condition

After the experimental system is built, the fan coil and the LHS unit are placed outdoors and the electronically heated film is opened. Due to the existence of an air-conditioning outdoor unit, heat dissipation of the fan coil is affected. When the outdoor unit is running, the residual pressure of it causes its passive rotation of the fan (1.2 m/s), which accelerates the heat dissipation of the pipe network and has a beneficial effect on the system. Based on the aforementioned experimental conditions, multiple experiments were conducted and comprehensive analysis of all experimental data was performed, which can be categorized into the following four operating conditions (Figure 11):

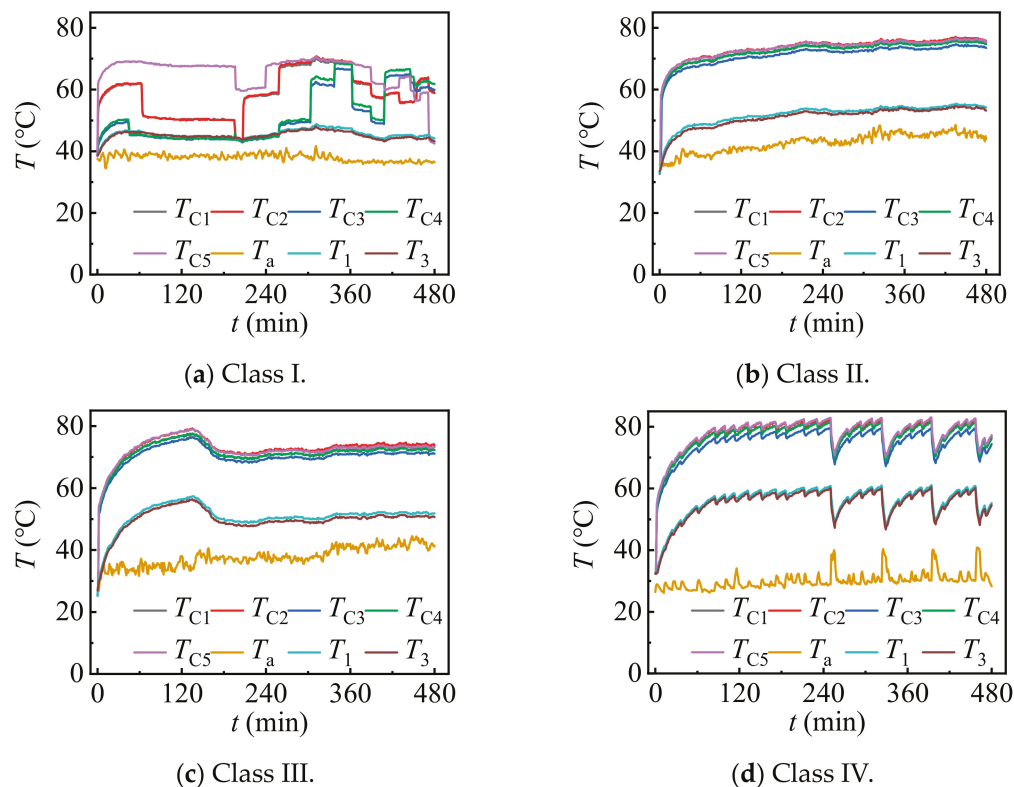


Figure 11. Operating conditions of the system.

Class I: The outdoor air-conditioning unit runs randomly, and the chip runs at half power. The water temperature continues to be near the PCM's phase transition temperature, and the LHS unit charges heat back and forth to stabilize the water temperature.

Class II: The outdoor AC unit runs continuously, and the chip operates at full power. Under this condition, the AC unit runs steadily, causing the water temperature to rise slowly and eventually stabilize at approximately 50 $^{\circ}\text{C}$. The LHS unit gradually stores heat during this process.

Class III: The outdoor AC unit defers to run and the chip runs at full power. Under this working condition, the AC unit does not run, the temperature of water increases rapidly, while the LHS unit is about to store heat quickly. After a period of time, the AC unit keeps opening, the water temperature drops to about 50 $^{\circ}\text{C}$, and the LHS unit slowly stores heat.

Class IV: The outdoor AC unit runs intermittently and the chip runs at full power. In this case, the temperature of water drops rapidly when the AC unit is running. When it reaches 60 $^{\circ}\text{C}$, the automatic control system opens the fan, and closes after the water temperature drops to 48 $^{\circ}\text{C}$. Under full load operation for 8 h, the fan is turned on four times.

These four operating conditions demonstrate that when the outdoor AC unit maintains continuous positive airflow supply, the system only consumes pump power. If the positive pressure supply becomes unstable, the system can still operate continuously for 4 h. After this period, the automatic control system supplements a small amount of fan power to sustain operation. The 15% efficiency improvement is quantified as follows. Under natural convection (Class II with AC off), the coil heat transfer coefficient is $h_{\text{nat}} = 15 \text{ W/m}^2\cdot\text{K}$. When the outdoor AC unit operates, the residual pressure increases face velocity by 1.2 m/s, enhancing h_{forced} to 22 $\text{W/m}^2\cdot\text{K}$ (calculated from $Q = h \cdot A \cdot \Delta T$ using measured heat flux $Q = 85 \text{ W}$, area $A = 0.12 \text{ m}^2$, $\Delta T = 32 \text{ }^{\circ}\text{C}$). The heat dissipation improvement

is $(22 - 15)/15 = 47\%$ in h , but due to the thermal resistance network (coil convection accounts for only 35% of total resistance), the overall system efficiency improves by 15%.

3.5. Economy Evaluation

The calculation of cooling load is the main basis of HVAC equipment selection, and the composition of indoor cooling load can be understood according to the calculation. Accuracy, simplicity and high efficiency should be considered in the selection of cooling load calculation method [39]. The existing calculation method includes total equivalent temperature/time average method, heat capacity method, heat balance method and so on [40,41]. Herein, the research object is a university office, and the cooling load coefficient method [42] is used to calculate its indoor cooling load. The cooling load is calculated by summing up the following items individually: (1) Incoming cooling load Q_1 through the envelope; (2) heat dissipation Q_2 of human body and lighting; (3) heat dissipation Q_3 of equipment and other internal heat sources.

Based on the information presented in Figure 12, it is evident that the contribution of computer equipment to indoor cooling load is significant. Under the operating condition where equipment runtime is synchronized with personnel working hours, the equipment cooling load Q_3 accounts for over 50% of the total cooling load at 9:00 when work commences; during working hours, the proportion of equipment cooling load Q_3 consistently exceeds 40%; during remaining time periods, the envelope cooling load Q_1 dominates. The data from Figure 12 show that computer equipment exerts a crucial impact on determining the overall cooling load in indoor environments. In this experiment, it is assumed that the computer equipment operates at maximum capacity throughout the entire working day. This means that the computers are constantly running and consuming electricity. As a result, the building's AC system is required to work harder for constant comfortable temperature indoors. According to the calculation of the rated refrigeration coefficient of the AC system, it is estimated that at least 500 W of electricity is needed to effectively dissipate 1500 W of heat generated inside the building. This demonstrates the significant energy requirements and thermal management challenges associated with running computer equipment at full load for extended periods of time.

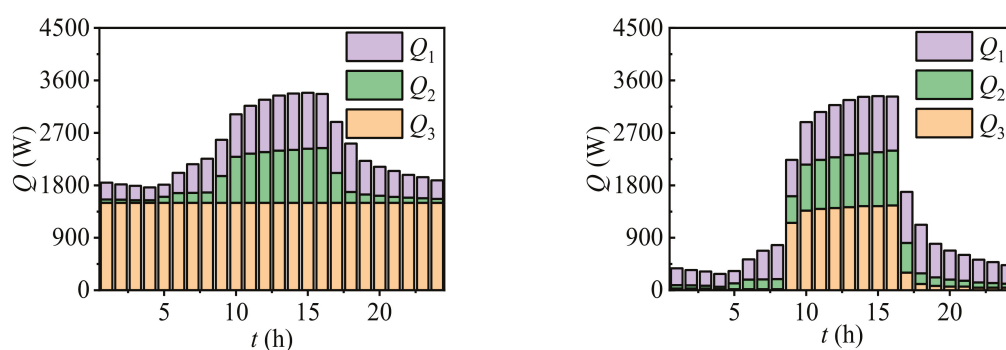


Figure 12. Cooling load of AC in a scientific research office.

To validate the long-term operational performance of the designed thermal management system under actual working conditions, the system was placed in a real environment for continuous operation over a one-week period. Figure 13 presents the variation curves of circulating water temperature and chip surface temperature during the operation period. The gray areas in the figure represent nighttime hours. It should be noted that the heating plates operated at full power exclusively during working hours (09:00–17:00), while the heating plates and circulating water pump were powered off during remaining periods.

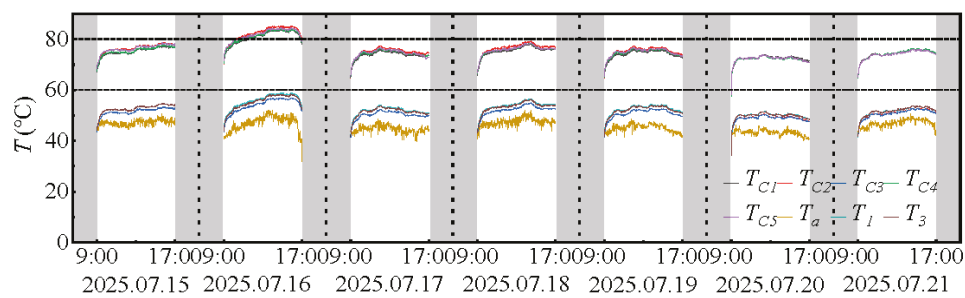


Figure 13. Continuous operating conditions for one week.

Throughout the testing period, the outdoor air temperature remained continuously above 40 °C, causing the building HVAC system to operate around the clock; consequently, the thermal management system was constantly subjected to Type II operating conditions with the outdoor unit of the air conditioner continuously activated. Under such rigorous conditions, the circulating water temperature was consistently maintained below 60 °C, and the chip surface temperature did not exceed 80 °C, demonstrating that the system possesses favorable thermal stability and environmental adaptability.

While dedicated cyclic testing was not performed, the one-week continuous operation (Figure 13) provides indirect evidence of cycling stability. The system experienced daily thermal cycles (09:00–17:00 heating, 17:00–09:00 cooling/solidification) for 7 days. Key observations: (1) Daily temperature profiles show consistent phase-change plateau behavior, indicating repeatable melting-solidification. (2) No performance degradation is observed; day 7 peak temperature matches day 1 within 1 °C. (3) This operational stability suggests PCM integrity is maintained over at least 14 complete cycles. We acknowledge that dedicated accelerated cycling tests (>100 cycles) would strengthen long-term reliability claims and identify this as future work.

It is important to clarify the system boundary for energy accounting (Table 3). The outdoor AC unit is part of the existing building HVAC infrastructure serving the entire office zone, not a component dedicated to our thermal management system. Its compressor power is accounted in the building's overall HVAC energy, not in our system's 125 W. The 75% and 40% savings refer specifically to the dedicated thermal management equipment (pumps, fans, LHS units) replacing conventional computer cooling solutions. We acknowledge that under extreme conditions (>40 °C ambient), the building HVAC COP decreases, but this affects baseline HVAC energy equally and does not change the relative savings of our dedicated system.

Table 3. Comparative energy analysis.

Item	HVAC	Natural Cold Air	This Hybrid System
Compressor Power Consumption	400 W	0 W	0 W
Water Pump Power Consumption	0 W	150 W	100 W
Fan Power Consumption	100 W	60 W	25 W (intermittent)
Total Power Consumption (W)	500 W	210 W	125 W
Energy Saving Rate (vs. HVAC)	—	58%	75%
Energy Saving Rate (vs. Natural Cold Air)	—	—	40%

3.6. Optimization of the LHS Unit

As shown in Figure 14a, the temporal variations of the LHS unit outlet temperature T_o and the solid–liquid interface temperature T_s are presented under different thermal conductivity conditions ($0.2 \text{ W/m}^2\cdot\text{K}$ in red, $0.3 \text{ W/m}^2\cdot\text{K}$ in green, and $0.4 \text{ W/m}^2\cdot\text{K}$ in blue). Owing to the relatively low thermal conductivity of the phase change material (PCM) $0.2 \text{ W/m}^2\cdot\text{K}$, the circulating water temperature remains significantly higher than the PCM temperature during heat transfer. As PCM thermal conductivity increases from 0.2 to $0.4 \text{ W/m}^2\cdot\text{K}$, the phase-change completion time decreases from 180 min to 120 min, indicating accelerated heat absorption rates. However, the total latent heat capacity ($20 \text{ kg} \times 200 \text{ kJ/kg} = 4000 \text{ kJ}$) remains unchanged. Therefore, higher conductivity improves the thermal response speed but does not increase total storage capacity. For applications requiring prolonged temperature stabilization, lower conductivity may be preferable as it sustains the phase-change plateau longer.

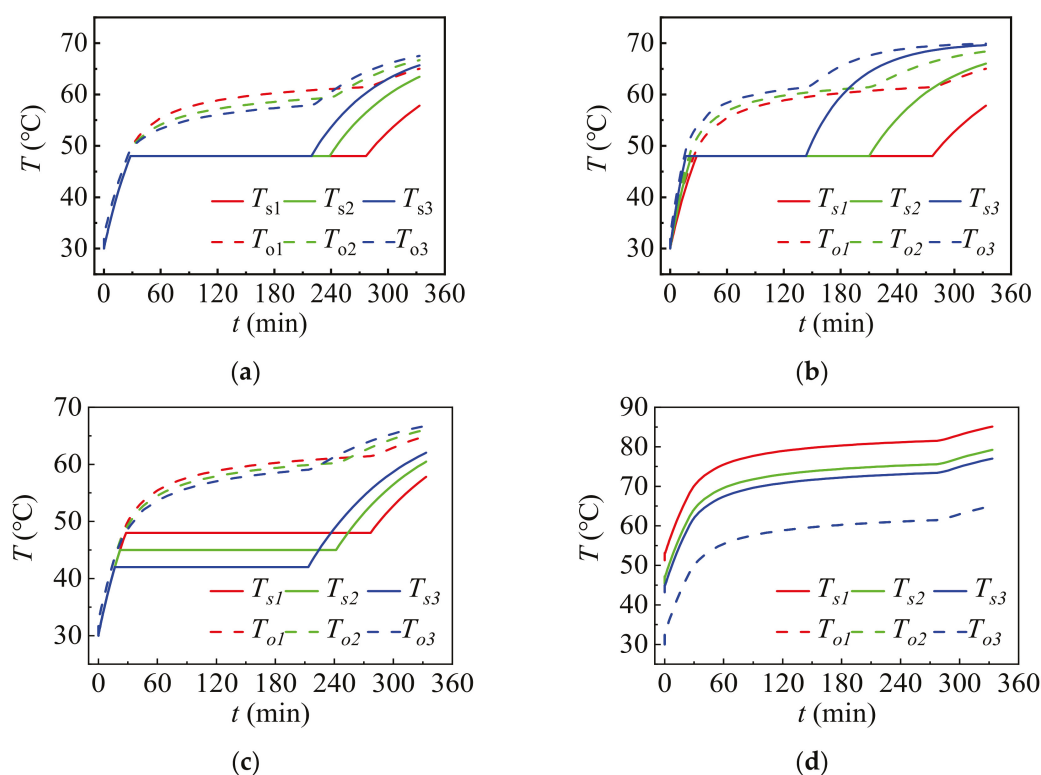


Figure 14. Optimization of the system. (a) Temperature variation under different thermal conductivity of PCM. (b) Temperature variation under different PCM's filling amount. (c) Temperature variation under different phase transition temperatures. (d) Temperature variation under different heat transfer coefficients of water-cooled plate heat exchanger.

Additionally, altering the PCM mass within the system also affects the temperature profiles. Figure 14b demonstrates the temporal evolution of the LHS unit outlet temperature T_o and the solid–liquid interface temperature T_s for PCM filling masses of 20 kg (red), 15 kg (green), and 10 kg (blue). The results indicate that reducing the PCM filling mass from 20 kg to 10 kg shortens the thermal storage time of the system, leading to insufficient long-term control of the circulating water temperature. Figure 14c illustrates the variations of the LHS unit outlet temperature T_o and the solid–liquid interface temperature T_s at phase change temperatures of 48°C (red), 45°C (green), and 42°C (blue). As the phase change temperature decreases, the circulating water temperature declines correspondingly. However, due to the increased heat transfer temperature difference, the heat absorbed from

the circulating water per unit time increases, thereby shortening the constant-temperature thermal storage duration. This suggests that reducing the PCM phase change temperature can also serve as an optimization strategy.

As depicted in Figure 14d, the temporal variations of the LHS unit outlet temperature T_o and the solid–liquid interface temperature T_s are shown for cold plate heat transfer coefficients of 1300 W/m²·K (red), 1400 W/m²·K (green), and 1500 W/m²·K (blue). The chip surface temperature decreases with increasing cold plate heat transfer coefficient, indicating that enhancing the heat transfer performance of the cold plate can effectively reduce the chip surface temperature. In summary, increasing the PCM thermal conductivity and its filling mass can reduce the temperature difference during heat transfer while maintaining a stable thermal storage duration. Selecting PCMs with lower phase change temperatures contributes to decreasing both the circulating water temperature and the chip surface temperature, although the effect is limited and challenges regarding higher costs remain. Furthermore, increasing the PCM mass and adopting low phase change temperature PCMs may lead to elevated material expenses and could potentially affect the efficiency and stability of the heat release process.

3.7. Validation Strategy

The numerical model is validated through a three-tier hierarchical validation strategy, with quantitative accuracy targets defined a priori to ensure rigorous and objective assessment. At Level 1 (component validation), the cold plate's heat transfer coefficient K and coolant outlet temperature T_o are compared against steady-state experimental data presented in Section 3.1 and Figure 8, with a predefined target of RMSE less than 5% of the measured temperature range; the model achieves an RMSE of 3.2 °C (5.3%), satisfying this criterion. At Level 2 (subsystem validation), the transient thermal response of the latent heat storage (LHS) unit, including water temperatures at the inlet (T_1), center (T_2), and outlet (T_3), as well as the PCM temperature, is benchmarked against time-resolved experimental measurements under dynamic operating conditions (Section 3.2, Figure 9), targeting an RMSE below 10% of the temperature range; the simulation yields an RMSE of 4.1 °C (6.8%), a MAPE of 8.3%, a coefficient of determination (R^2) of 0.94, and a steady-state deviation of less than 2 °C, thereby meeting all specified metrics. At Level 3 (system-level validation), the model's predictive capability over extended operation is evaluated by comparing simulated and measured temperature profiles across a continuous one-week field test under real ambient conditions (Figure 13), with the validation criterion requiring day-to-day steady-state temperature deviations of less than 2 °C; the experimental data exhibit variations of less than 1 °C over seven days, confirming consistent cyclic performance and long-term model fidelity. The validation employs a comprehensive set of error metrics, including RMSE, MAPE, R^2 , and steady-state temperature deviation, and is further supported by a detailed error source analysis and sensitivity study in Section 3.2, which examines the impact of $\pm 10\%$ uncertainties in key parameters such as PCM thermal conductivity and heat transfer coefficient, thereby establishing the model's robustness and delineating its domain of applicability.

4. Conclusions

This study addressed the centralized thermal management requirements of high-power-density computing clusters in office environments. A hybrid liquid cooling system that integrates latent heat storage with outdoor natural cooling sources was proposed and experimentally validated. The thermal control loop comprising six-chip simulated heat sources, a phase change material storage unit, and a fan coil unit was constructed,

incorporating control strategies and utilization of building environmental waste energy to achieve high efficiency and reliability in heat dissipation performance.

(1) The hybrid liquid cooling system integrating latent heat storage (LHS) with outdoor natural cooling sources maintains chip surface temperatures below 80 °C, the industrial safety threshold, while stabilizing the circulating water temperature at approximately 60 °C. This temperature range effectively matches the operating interval of paraffin-based phase change materials with a melting point of 48 °C, thereby fully exploiting their capability for transient thermal peak absorption.

(2) The total power consumption of the system is merely 125 W, achieving a 75% energy saving compared to conventional building-dependent cooling systems that require 500 W to manage equivalent thermal loads. When compared to natural cooling solutions relying solely on outdoor cold air, which consume 210 W, the operational energy consumption can be reduced by 40%. It should be noted that the positive pressure environment generated by the air outdoor units increases the face velocity of the fan coil unit by 1.2 m/s, providing an additional 15% enhancement in heat dissipation efficiency. This demonstrates the effective utilization of residual energy from existing building equipment.

(3) Increasing the thermal conductivity of the phase change material reduces the heat transfer temperature difference but shortens the heat storage duration, whereas increasing the filling mass helps extend the period of constant temperature regulation. This indicates that LHS design requires balancing conductive performance against heat storage capacity according to load characteristics.

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Article

Equipment Selection Optimization and Empirical Analysis of Operational Performance for a Commercial Building Refrigeration Plant

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Abstract

Climate change necessitates a global transition toward green and low-carbon development, underscoring the critical importance of energy efficiency. Buildings account for a substantial portion of urban energy consumption and carbon emissions, with central air-conditioning systems representing the largest energy-consuming component. This study focuses on optimizing equipment selection—including chillers, pumps, and cooling towers—for the refrigeration plant of a commercial complex in Xiamen. Following theoretical optimization, the operational performance of the implemented system was empirically analyzed using long-term monitoring data from 2024 to 2025. The results demonstrate an energy efficiency ratio (EER) of 5.44 in 2024 and 5.28 in 2025, surpassing the Grade I efficiency threshold (5.2) stipulated by the Chinese standard T/CRAAS 1039-2023. Monthly EER values consistently remained above 5.06 throughout the cooling season. Detailed performance analysis of individual equipment further confirmed that actual operational performance of chillers, pumps, and cooling towers closely matched or even exceeded rated performance metrics, with chiller efficiency deviations controlled within 5%. This study integrates optimized equipment selection at the design stage with empirical performance analysis based on actual operation, providing a validated approach for improving the energy efficiency of refrigeration plants in commercial buildings and offering valuable references for the revision of relevant energy efficiency standards.

Keywords: equipment selection optimization; empirical analysis; operational performance; refrigeration plant; commercial building

1. Introduction

Climate change has driven a global shift toward green and low-carbon development. Within this context, China has established the “dual carbon” strategic goals of achieving carbon peak before 2030 and carbon neutrality before 2060. Among the pathways to these objectives, the efficient utilization of energy stands out as a core technological approach, making its

advancement and innovation particularly critical and urgent. Buildings are responsible for about 70% of urban primary energy consumption, 36% of global energy demand and 37% of energy-related CO₂ emissions, making improvements in their energy efficiency crucial for advancing socially sustainable development [1–3]. Within modern buildings, central air-conditioning systems constitute the largest energy-consuming component, accounting for over half of total building energy use [4]. Furthermore, refrigeration plants represent the primary energy consumers in central air-conditioning systems, responsible for more than half of their total energy demand. Consequently, optimizing equipment selection in refrigeration plants emerges as a key approach for achieving building energy conservation.

Previous research has primarily focused on two pathways for enhancing refrigeration plant efficiency: design optimization and operational optimization. Design optimization focuses on improving energy efficiency through optimizing the design schemes of refrigeration plants [5–11]. Studies highlight the critical impact of system configuration on energy performance. Research shows that energy consumption can vary by as much as 69% depending on the number and size of chillers deployed [5]. Chen [6] presents a thermal resistance-based method to optimize central chiller systems by directly solving optimal exchanger areas and pump/fan speeds via Lagrange multipliers. ASHRAE Learning Institute provides the fundamental principles and practical methods for the design and control of central chilled water plants, covering equipment selection, system configuration, operation optimization, and energy-efficient control strategies [7]. Peterson [8] provides practical guidelines for optimizing large central chilled water systems through proper coil and valve selection and recommends non-decoupled direct building connections to improve efficiency and stability. Ho [9] optimizes campus chilled water building connections, reducing central distribution pump energy consumption by using low-pressure distribution and building-side booster pumps. Other work has employed detailed simulations of multiple configurations and control strategies to identify optimal system designs [10]. Furthermore, comprehensive multi-criteria evaluation frameworks have been developed and applied to select suitable central air-conditioning schemes, balancing energy efficiency, cost, and environmental impact [11]. Operational optimization employs methods such as genetic algorithms [12,13], particle swarm optimization [14,15], reinforcement learning [16,17] or model predictive control (MPC) [18,19] to determine the optimal combinations of equipment start–stop operations and operational parameters for improving the energy efficiency of refrigeration plants. In summary, most previous studies conducted system design optimization based on theoretical calculations and simulation models. Such studies fail to conduct real-condition validation supported by long-term measured data and rarely address the optimization and field-measurement validation of chillers, water pumps and cooling towers.

To address this research gap, this study focuses on a commercial complex in Xiamen and optimizes the equipment selection for its refrigeration plant design, including chillers, chilled water pumps, cooling water pumps, and cooling towers. Combined with hourly operational monitoring data from 2024 to 2025, the actual operational performance of the refrigeration plant is analyzed on a monthly basis and during the hottest week. By evaluating the real-world performance of key equipment, this research integrates design-stage equipment selection optimization with operational-stage performance assessment, providing a basis for enhancing system energy efficiency through optimized equipment selection in high-efficiency plants. Furthermore, it offers data support for the future revision of China's Energy Efficiency Standard for High-Efficiency Air-Conditioning and Refrigeration Plant Systems.

2. Building Cooling Load Characteristics

This study focuses on a commercial complex in Xiamen, consisting of a commercial podium and a terminal building. This Xiamen commercial complex was selected because it is located in the Hot Summer and Warm Winter Zone—a major energy-intensive region for commercial buildings in southern China—while also offering the rare availability of complete two-year continuous operational monitoring data (2024–2025) essential for empirical validation. The total floor area of the project is 264,000 m², with an air-conditioned area of 67,039 m² in the commercial section. The commercial building has a height of 35 m, comprising two basement levels and six above-ground floors.

The outdoor dry-bulb and wet-bulb temperature distribution for Xiamen is presented in Figure 1, with design values of 33.5 °C and 27.5 °C, respectively. Using the cooling load coefficient method based on monthly weighted daily dry-bulb and wet-bulb temperatures, monthly weighted cooling load characteristics for each month were calculated. Parameter settings, including the thermophysical properties of building envelopes, internal heat loads, and indoor air temperature and humidity, comply with the national energy efficiency design code GB 50189-2015 [20] for public buildings. Building cooling load calculation results are illustrated in Figure 2. The results show that the total cooling load peaks in July and reaches its minimum in April. Analysis of the annual load profile (Figure 2) enabled the determination of operational duration proportions across different cooling load ranges, as summarized in Figure 3. Operational time for loads below 400 RT accounts for only 1.71% of the total annual runtime, while the range of 2800–3200 RT represents merely 2.56%. The load range of 800–1200 RT accounts for the largest proportion of annual operational time, at 29.06%. The above annual load profile and distribution of operating time across various cooling load intervals inform the chiller configuration strategy to ensure high efficiency across typical operating conditions.

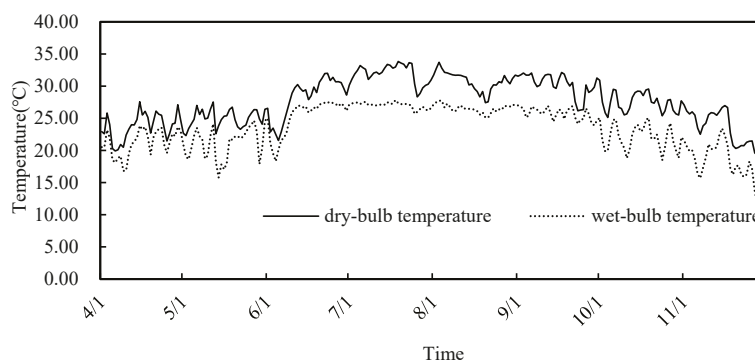


Figure 1. Distribution of outdoor air dry- and wet-bulb temperatures.

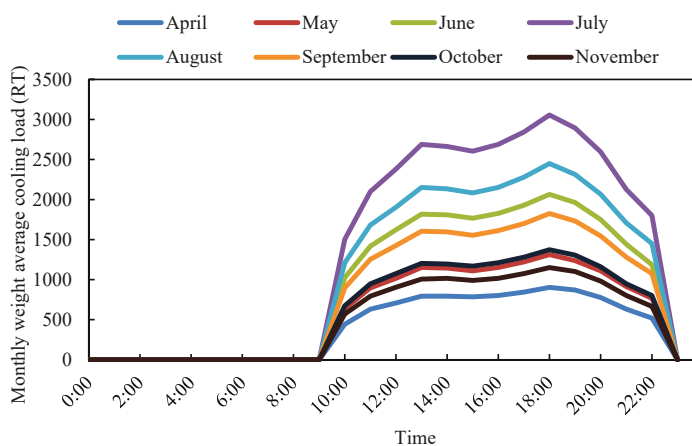


Figure 2. Building monthly weighted average cooling load characteristic.

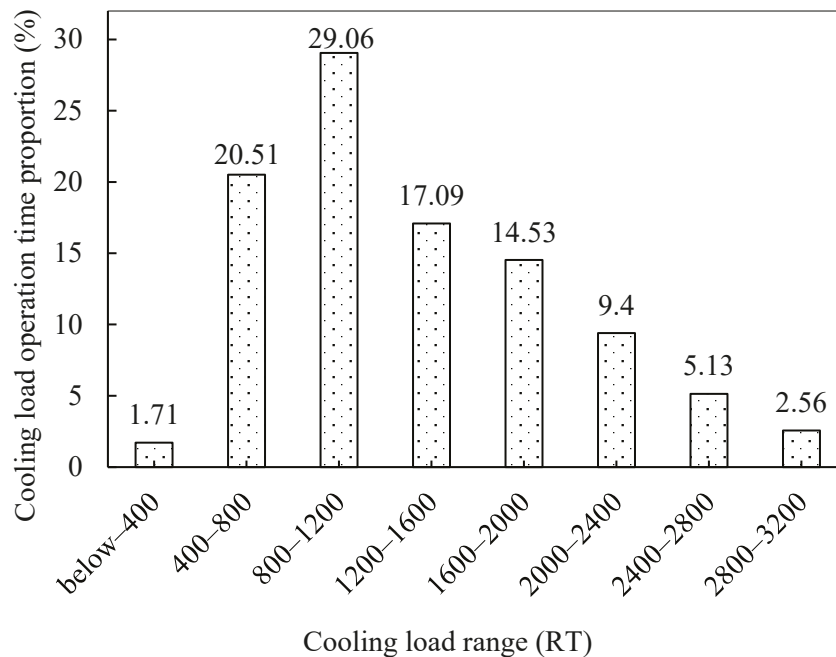


Figure 3. Proportion of cooling load operation time.

Given the extended cooling season in Xiamen—lasting approximately 200 days from mid-to-late April to early November—the potential for energy savings through optimized system design is considerable.

3. System Form and Equipment Selection Optimization

3.1. System Form

The detailed design of the refrigeration plant energy system is illustrated in Figure 4. The system consists of five chillers, seven chilled water pumps, seven cooling water pumps, and seven cooling towers. In terms of control strategy, both the chilled water system and the cooling water system employ a constant temperature difference control method to ensure stable operation and optimize energy efficiency under varying load conditions. Specifically, the temperature difference between the supply and return chilled water is maintained at 6 °C, while that for the cooling water is set at 5 °C. Meanwhile, to enhance cooling efficiency and energy savings, the fan frequency of the cooling towers is regulated using a constant approach temperature control strategy. This approach maintains a fixed temperature difference between the cooling tower outlet water and the wet-bulb temperature, enabling precise adjustment of cooling tower operation and reducing energy consumption.

3.2. Equipment Energy Consumption Models

The total energy consumption of a refrigeration plant system comprises the electrical demand of chillers, water pumps, and cooling towers, as modeled in Equation (1).

$$EC_{total} = EC_{chiller} + EC_{pump} + EC_{tower} \quad (1)$$

where $EC_{chiller}$ is the energy consumption of chillers [kWh].

EC_{pump} is the energy consumption of pumps [kWh].

EC_{tower} is the energy consumption of cooling towers [kWh].

The energy consumption models of chillers, pumps, and cooling towers are described as follows:

(1) Chiller Energy Consumption Model.

By applying the MP model [21] and performing multiple linear regression on the sample parameters, the following energy consumption model for the water-cooled chiller is derived as Equation (2).

$$EC_{chiller} = EC_{chiller,r} \times f(PLR) \times f(T_{eo}, T_{ci})$$

$$= EC_{chiller,r} \times (a_1 + a_2 PLR + a_3 PLR^2) \times (a_4 + a_5 T_{eo} + a_6 T_{eo}^2 + a_7 T_{ci} + a_8 T_{ci}^2 + a_9 T_{eo} T_{ci}) \quad (2)$$

where PLR is the chiller's part load ratio, which is calculated as the ratio of the actual cooling capacity to the rated cooling capacity of the chiller.

T_{eo} is the evaporator outlet temperature, K.

T_{ci} is the condenser inlet temperature, K.

a_1 to a_9 are regression coefficients.

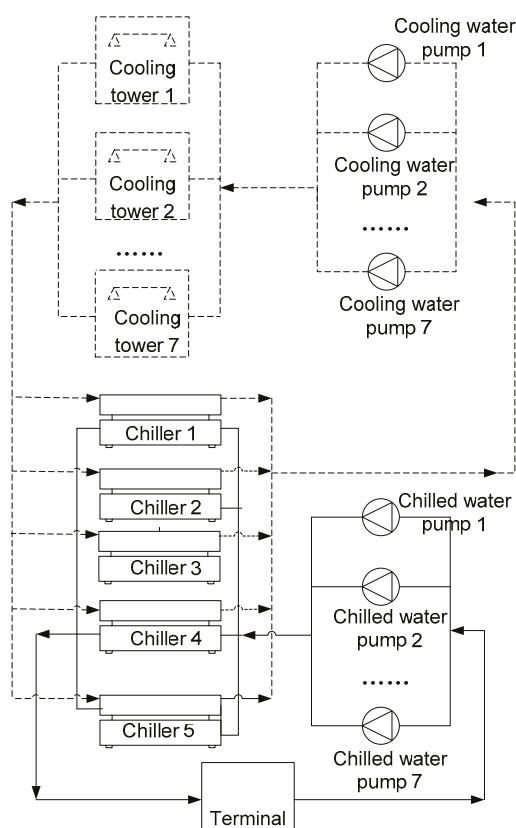


Figure 4. Schematic diagram of the refrigeration plant system.

(2) Pump Energy Consumption Model.

Analysis of the variable-frequency pump sample parameters reveals a quadratic polynomial relationship between its energy consumption and flow rate [22], leading to the following energy consumption model:

$$EC_{pump} = EC_{pump,r} \times f(v) = EC_{pump,r} \times (b_1 + b_2 v + b_3 v^2) \quad (3)$$

where $EC_{pump,r}$ is the rated energy consumption of the pump, [kWh].

v is the water flow, m^3/h .

b_1 to b_3 are regression coefficients obtained by using the multiple linear regression tool based on pump sample parameters.

(3) Cooling Tower Energy Consumption Model.

Cooling tower fan energy consumption is often calculated using the semi-empirical Braun model [23]. While it can predict thermal performance, it fails to capture the coupled relationship between the cooling tower approach temperature, chiller energy use, and overall system efficiency. Consequently, it cannot assess the impact of approach temperature variations on system optimization under identical cooling loads.

To address this limitation, this study utilizes 21,369 validated datasets provided by the Cooling Technology Institute's standardized performance tables. A Gradient Boosting Regressor model was developed with 200 trees, a 0.1 learning rate, and a maximum depth of 4. The model achieves high predictive accuracy: based on our prior studies, the test results show that the coefficient of determination R^2 reaches 0.9977, the root mean square error (RMSE) is 0.1034, the mean absolute error (MAE) is 0.0485, and the mean absolute percentage error (MAPE) is only 4.65% [24]. Crucially, the model accurately quantifies the impact of the approach degree on system efficiency, thereby overcoming a key shortcoming of the traditional Braun model.

Cooling tower energy consumption is calculated as follows [23]:

$$EC_{tower} = C \times EC_{rated} \times \frac{\eta}{\eta_0} \times \frac{Q_{actual}}{Q_{rated}} \quad (4)$$

where C is a dimensionless conversion factor employed to standardize the units between the performance coefficient and power parameters, $C = 0.871$.

EC_{rated} is the rated energy consumption of the cooling tower.

η is the performance coefficient under current operating conditions.

η_0 is the performance coefficient under base operating conditions.

Q_{actual} is the actual water flow of the cooling tower, m^3/h .

Q_{rated} is the rated water flow of the cooling tower, m^3/h .

3.3. Equipment Selection Optimization

(1) Chiller.

The analysis of the building cooling load characteristics (Figure 2) and load operation time distribution (Figure 3) indicates that for 80% of the year, the cooling load falls within the range of 400–2400 RT. To ensure high energy efficiency ratios (EERs) across various load segments, a chiller configuration of four 900 RT units plus one 600 RT unit is selected. Under low-load conditions, the 600 RT unit can operate efficiently. Therefore, the original design, designated as the baseline scheme, consists of four 900 RT constant-speed drive (CSD) chillers and one 600 RT variable-speed drive (VSD) chiller, with a total chiller pressure drop of 8 mH_2O . Based on this configuration, two optimized schemes—optimization scheme 1 and optimization scheme 2—are proposed by separately optimizing the chiller's fixed/variable-frequency configuration and the pressure drop. Relevant parameters are presented in Table 1.

Table 1. Chiller configuration scheme.

Parameters	Baseline Scheme	Optimization Scheme 1	Optimization Scheme 2
chiller configuration	900 RT $\times$ 4 CSD + 600 RT $\times$ 1 VSD	900 RT $\times$ 2 CSD + 900 RT $\times$ 2 VSD + 600 RT $\times$ 1 VSD	900 RT $\times$ 4 CSD + 600 RT $\times$ 1 VSD
chiller pressure drop	8 mH_2O	8 mH_2O	6 mH_2O
chilled water supply and return temperature		6/12 $^{\circ}C$	

Based on the building load characteristics and the energy consumption models of individual equipment, the monthly weighted average energy consumption and annual energy consumption are calculated. Subsequently, by dividing the cooling load by the energy consumption, the monthly weighted average EER and the annual comprehensive EER can be derived. The calculated comparative results of the three schemes are shown in Figures 5 and 6. Compared with the baseline scheme, optimization scheme 1 requires an additional initial investment of 270,000 CNY, with annual cost savings of approximately 42,000 CNY, resulting in a payback period of 6.43 years. In contrast, optimization scheme 2 involves an additional initial investment of 84,000 CNY, achieves annual cost savings of about 35,000 CNY, and has a payback period of only 2.4 years. Although optimization scheme 1 exhibits a slightly higher annual comprehensive energy efficiency ratio (5.17), optimization scheme 2 demonstrates a significant improvement in energy efficiency (5.15) while achieving a substantial reduction in initial investment. Moreover, with a payback period of only 2.4 years, it presents superior techno-economic feasibility.

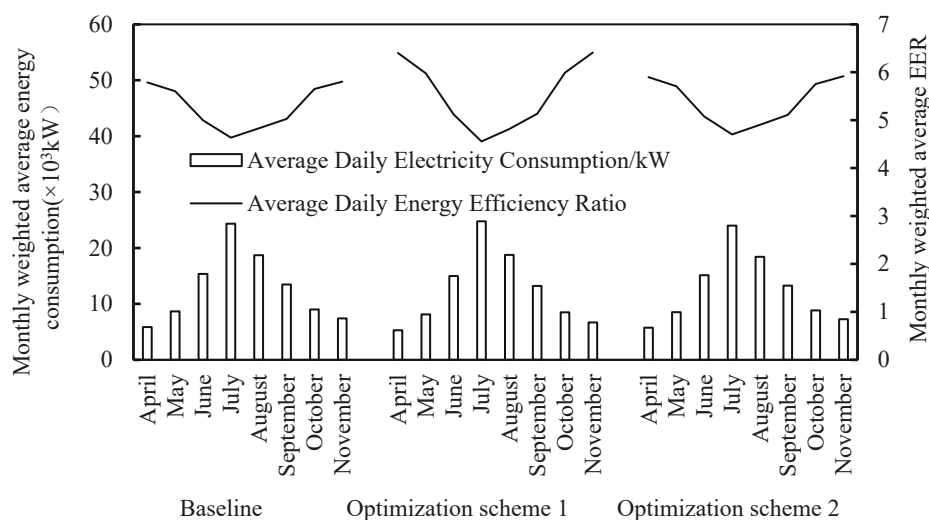


Figure 5. Monthly weighted average energy consumption and EER comparison.

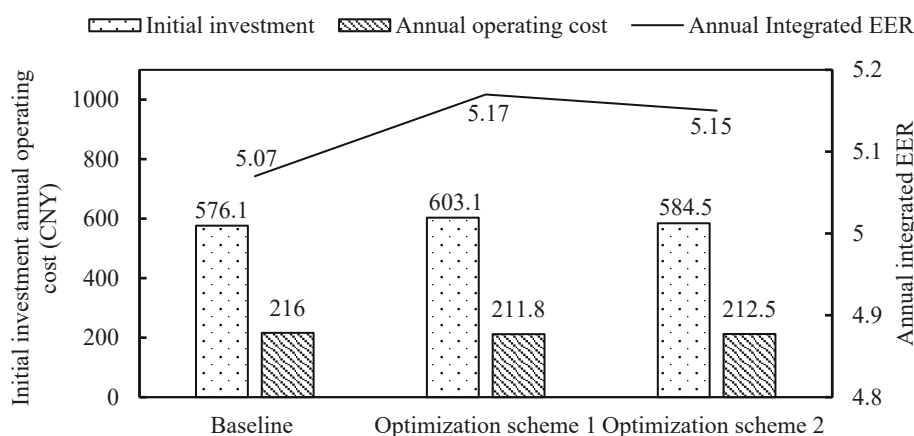


Figure 6. Initial investment, annual operating cost and annual integrated EER comparison.

(2) Pump.

The pump selection primarily adheres to the following principles: the rated flow rate of the chilled/cooling water pump is set 5% higher than the chiller flow rate to provide a safety margin; the pump head is determined based on hydraulic calculations, with an

additional 10% allowance. The total efficiency of the selected pump under rated operating conditions must be $\geq 80\%$ to comply with the energy efficiency evaluation criteria specified in GB 30254-2020. Thus, the original design, i.e., the baseline scheme, employs five variable-frequency chilled water pumps with a capacity of $460 \text{ m}^3/\text{h}$ and two variable-frequency chilled water pumps with a capacity of $310 \text{ m}^3/\text{h}$, along with five fixed-frequency cooling water pumps with a capacity of $650 \text{ m}^3/\text{h}$ and two fixed-frequency cooling water pumps with a capacity of $460 \text{ m}^3/\text{h}$.

To optimize system performance, the cooling water pumps are uniformly upgraded from the original fixed-speed configuration to variable-frequency drive (VFD) control. For the chilled water pumps, a “large + small” combination ($326 \text{ m}^3 \cdot \text{h}^{-1} + 155 \text{ m}^3 \cdot \text{h}^{-1}$) is adopted. This configuration can concurrently serve the 900 RT and 600 RT chillers and enables precise distribution and efficient operation during low-load periods by utilizing the $155 \text{ m}^3 \cdot \text{h}^{-1}$ pump. Based on these principles, the relevant parameters of the proposed optimization scheme are presented in Table 2. This scheme employs four $483 \text{ m}^3/\text{h}$, two $326 \text{ m}^3/\text{h}$, and one $155 \text{ m}^3/\text{h}$ variable-frequency chilled water pumps, as well as five $674 \text{ m}^3/\text{h}$ and two $477 \text{ m}^3/\text{h}$ variable-frequency cooling water pumps. As shown in Table 2, calculations indicate that compared to the baseline scheme, the optimization scheme reduces initial investment by 58.7 thousand CNY. Specifically, the rated power of the $155 \text{ m}^3/\text{h}$ variable-frequency pump in the optimization scheme is only 18.5 kW, significantly lowering the energy consumption of chilled water pumps under extremely low-load conditions. Furthermore, the efficiency of the pumps in the optimization scheme is slightly higher than that in the baseline scheme. Therefore, the optimization scheme is superior.

Table 2. Pump configuration scheme comparison.

Pump	Scheme	Flow Rate (m^3/h)	Head (m)	Power (kw)	Efficiency	Pump Number	Initial Investment (CNY)
Chilled water pump	Baseline scheme	460	41	75	81%	5	332,064
		310	41	55	77%	2	
	Optimization scheme	483	31	55	84%	4	254,520
		326	31	45	79%	2	
		155	31	18.5	82%	1	
Cooling water pump	Baseline scheme	650	35	90	83%	5	291,187
		460	35	75	82%	2	
	Optimization scheme	674	31	75	81%	5	309,992
		477	31	55	84%	2	

(3) Cooling tower.

The comparative analysis of the optimized cooling tower selection schemes is presented in Table 3. The original design, i.e., the baseline scheme, employs eight cooling towers with a flow rate of $360 \text{ m}^3/\text{h}$ each, supplemented by an additional cooling tower with a flow rate of $500 \text{ m}^3/\text{h}$, resulting in a total system flow of $3380 \text{ m}^3/\text{h}$. The design conditions are as follows: a wet-bulb temperature of 28°C and cooling water supply/return temperatures of $32/37^\circ\text{C}$, corresponding to an approach temperature of 4 K. Under this scheme, the total fan power of the cooling towers is 138.5 kW, and the calculated EER of the chiller unit is 4.85.

The optimized scheme utilizes eight cooling towers, each with a flow rate of 420 m³/h, yielding a total flow of 3360 m³/h. Under the same wet-bulb temperature condition (28 °C), the cooling water inlet/outlet temperatures are adjusted to 31/36 °C, reducing the approach temperature to 3 K. The total fan power of the cooling towers in this scheme is 148 kW, and the calculated EER of the chiller unit increases to 5.07. By lowering the cooling water temperatures and improving the overall system energy efficiency, the optimized scheme achieves an annual reduction in operating costs of 98,000 RMB. Although the initial investment increases by 150,000 RMB, the payback period is only 1.53 years, demonstrating favorable economic and energy-saving benefits.

Table 3. Cooling tower configuration scheme comparison.

Scheme	Cooling Water Supply/Return Temperature	Outdoor Air Wet-Bulb Temperature	Flow Rate (m ³ /h)	EER	Operation Cost (×10 ⁴ CNY)	Initial Investment (×10 ⁴ CNY)
Baseline scheme	32/37 °C	28 °C	3380	4.85	225.8	143
Optimization scheme	31/36 °C	28 °C	3360	5.07	216.0	158

This optimized selection scheme ensures the required flow rate while significantly enhancing system efficiency through adjustments in cooling tower configuration and operational parameters. The short payback period further underscores its practical value in engineering applications.

4. Empirical Results and Discussion

Although computational analyses have confirmed the energy-saving benefits of optimized chiller, pump, and cooling tower selection schemes, the long-term operational performance of such optimized systems still lacks empirical validation. Therefore, the following section will analyze and evaluate the actual operational performance of a refrigeration plant implementing chiller optimization scheme 2, pump optimization scheme and cooling tower optimization scheme based on operational data collected between 2024 and 2025.

4.1. Refrigeration Plant Monthly Operational Performance Analysis

The operational monitoring system employs high-precision sensors with certified accuracy grades: electromagnetic flow meters ($\pm 0.5\%$) for chilled/cooling water flow, PT1000 temperature sensors (± 0.1 °C) for supply/return water temperatures, and three-phase multifunction power meters ($\pm 0.5\%$) for equipment power consumption. All selected sensors and instruments adopt mainstream high-precision industrial-grade configurations, meeting the accuracy requirements for building energy consumption measurement and operational monitoring in this industry. According to the error propagation principle of the energy efficiency ratio (EER) mathematical model, the relative uncertainties in the flow rate, temperature difference and electric power are synthesized by the root-sum-square (RSS) method, and the combined relative uncertainty in the EER is calculated to be approximately 2.92%.

The monthly energy consumption, cooling capacity, and EER distribution of the commercial building's refrigeration plant during the 2024 and 2025 cooling seasons (20 April to 10 November) are shown in Figures 7 and 8. The data indicate that both energy consumption and cooling capacity were significantly higher from July to September compared to the other months. This is primarily attributed to the higher outdoor temperatures and the

corresponding greater cooling load during this period. Throughout the cooling season, the monthly EER for both 2024 and 2025 remained above 5.06, with annual comprehensive EER values of 5.44 and 5.28, respectively. According to the efficiency classification limits specified for high-efficiency air-conditioning refrigeration plant systems (with cooling capacity exceeding 1758 kW) in Hot Summer and Warm Winter Zone II (Xiamen) by the standard T/CRAAS 1039-2023 “Energy Efficiency Monitoring and Classification Standard for High-Efficiency Air-Conditioning Refrigeration Plant Systems” [25], the annual energy efficiency level of this commercial building’s refrigeration plant has reached Grade I (the classification efficiency limit is 5.2).

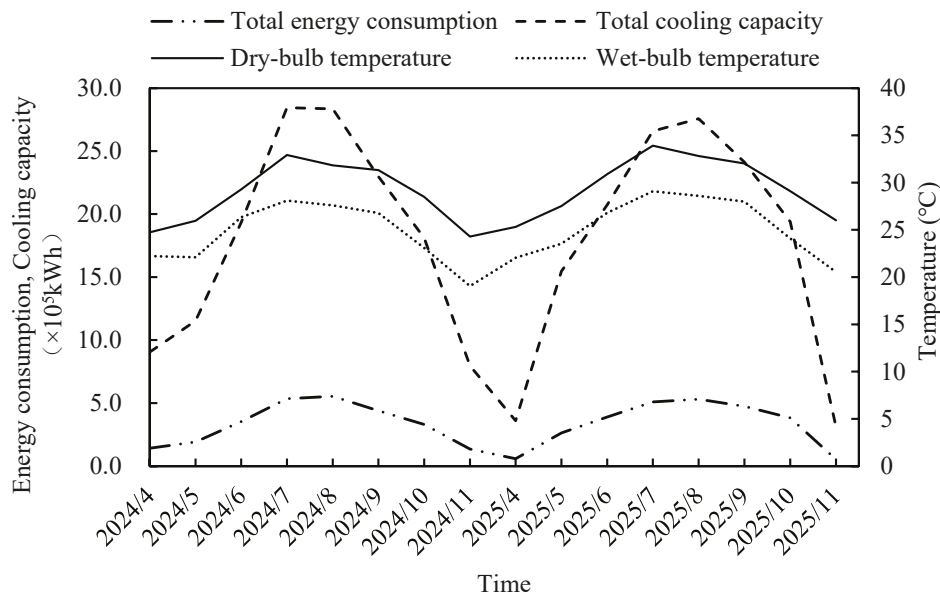


Figure 7. Refrigeration plant monthly energy consumption and cooling capacity.

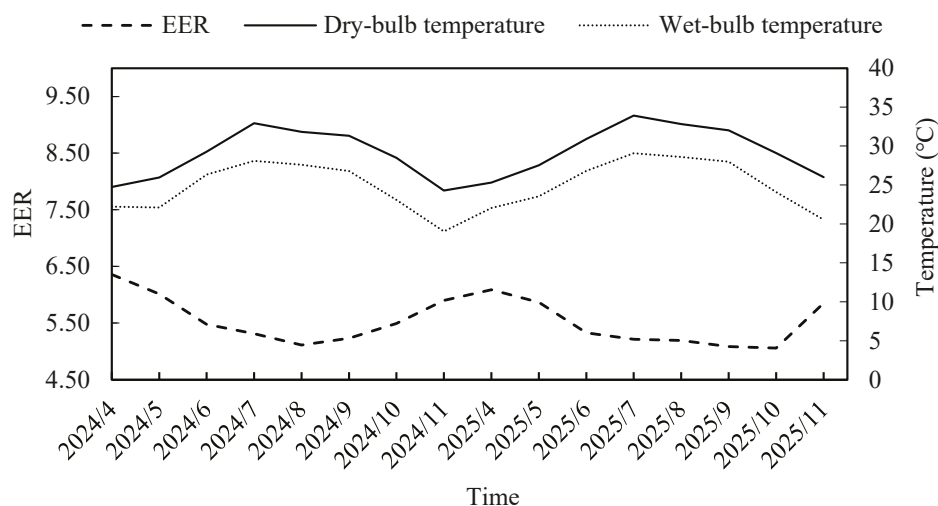


Figure 8. Refrigeration plant monthly EER.

4.2. Refrigeration Plant Operational Performance Analysis During the Hottest Week

An analysis of the hourly energy consumption, cooling capacity, and EER of the refrigeration plant during the hottest week (1–7 July 2025) is presented in Figures 9 and 10. As shown in Figure 9, the peak cooling load of the commercial building in 2025 occurred at 10:00 on 1 July, reaching 8847.46 kW, which is mainly attributed to the higher cooling capacity during the equipment start-up phase.

Figure 10 indicates that during the hottest week, the maximum system EER was 5.56, recorded at 10:00 on 5 July, while the minimum system EER was 4.84, observed at 21:00 on 3 July. The operating statuses of the chillers, pumps, and fans are provided in Table 4. The analysis reveals that at 10:00 on 5 July, three chillers were operating efficiently, and five pumps, along with six cooling towers, were running at high frequencies, resulting in a relatively high overall EER for the refrigeration plant. In contrast, at 21:00 on 3 July, only two chillers were operating at lower efficiency, and four pumps, along with two cooling towers, were running at partial frequencies, leading to a relatively low overall EER for the refrigeration plant.

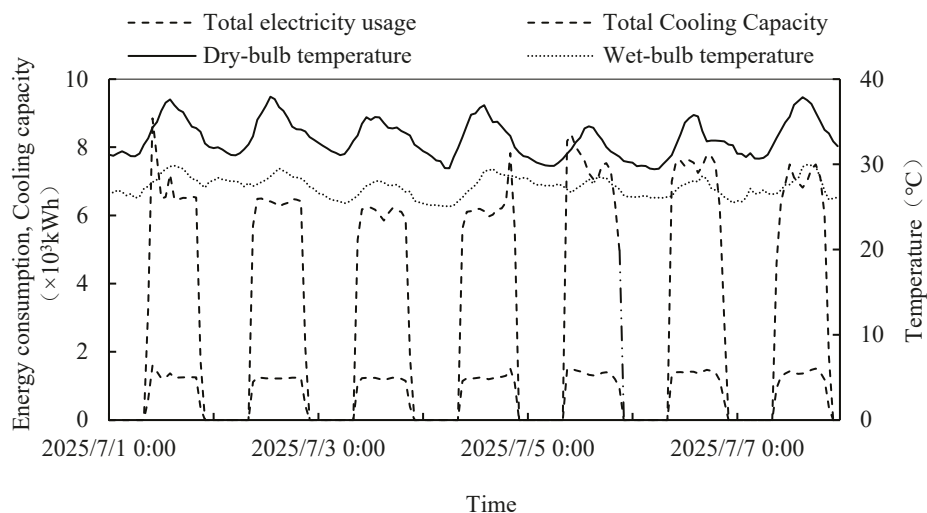


Figure 9. Refrigeration plant energy consumption and cooling capacity during the hottest week.

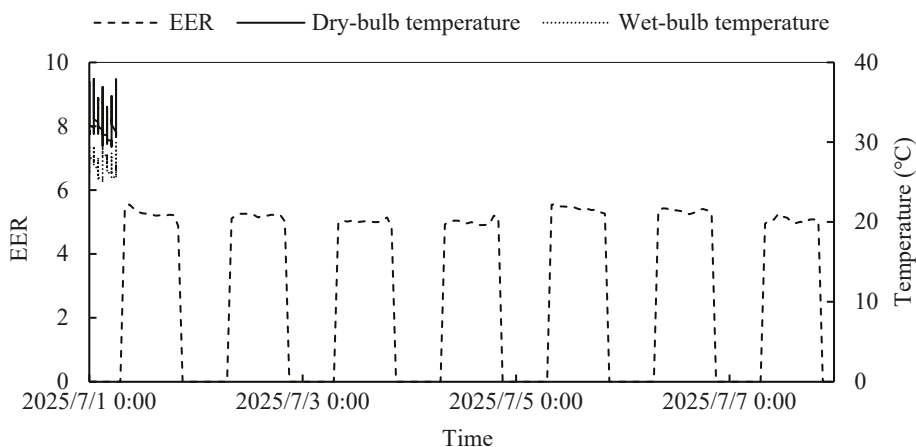


Figure 10. Refrigeration plant EER during the hottest week.

Table 4. Operating status and frequency of chillers, pumps, and cooling towers.

Time	Chiller Operation Status and EER	Pump Operation Status and Frequency	Cooling Tower Operation Status and Frequency
10:00, 5 July	Chillers #2, #4, #5 in operation, EER: 6.69, 6.59, 8.20	Chilled water pumps #2, #5 in operation, Frequency: 40 Hz; Cooling water pumps #1, #4, #5 in operation, Frequency: 40 Hz	Six cooling tower fans in operation; Fan frequency: 50 Hz
21:00, 3 July	Chillers #1, #3 in operation, EER: 5.06, 5.17	Chilled water pumps #1, #3 in operation, Frequency: 32 Hz; Cooling water pumps #1, #3 in operation, Frequency: 15 Hz	Two cooling tower fans in operation; Fan frequency: 18 Hz

4.3. Equipment Operational Performance Analysis

(1) Chiller.

Figures 11 and 12 illustrate a comparison between the actual operating performance and the rated performance of each chiller under different partial load ratios and cooling tower inlet water temperatures. Calculation results show that the energy efficiency ratio of most actual operating condition points is lower than that of the rated condition points, with deviations generally controlled within 5%. A few operating condition points even achieve the rated energy efficiency ratio level. According to the guidelines for efficient operation of chillers provided in ASHRAE Guideline 36-2021 [26], the deviation between the actual energy efficiency ratio and the rated value should typically be maintained within a range of 5% to 15%, with deviations below 10% considered relatively good performance. Thus, it can be concluded that the operational performance of the chiller in this commercial building is superior to that of most similar units, indicating excellent overall operating conditions.

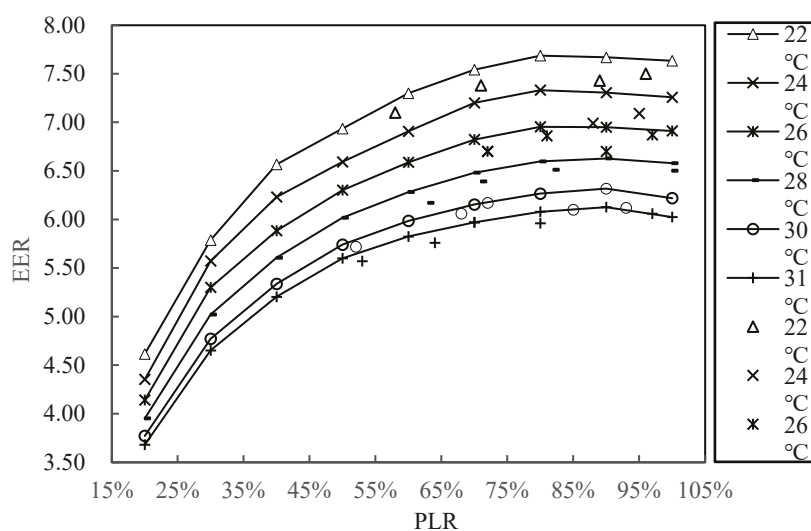


Figure 11. Comparison of actual vs. rated performance for chillers #1–#4.

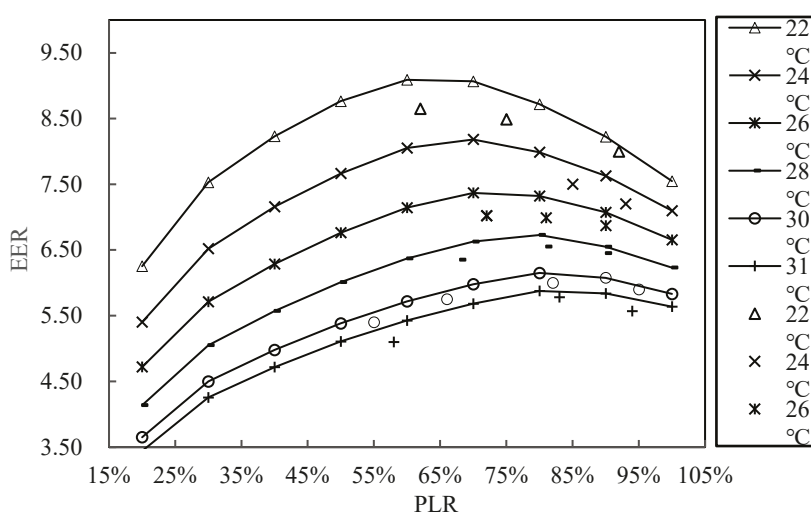


Figure 12. Comparison of actual vs. rated performance for chiller #5.

In addition, Figures 11 and 12 illustrate the impact of cooling water inlet temperature and part-load ratio on the EER. The results indicate that a 1 °C decrease in cooling water

temperature can improve the EER by approximately 3–12%, highlighting the importance of cooling tower performance optimization.

(2) Pump.

The chilled water pump transport factor μ_{ch} and the cooling water pump transport factor μ_c are used to characterize the operational performance of the chilled water system and the cooling water system, respectively. A larger value indicates better performance. They are calculated using Equations (5) and (6), respectively.

$$\mu_{ch} = \frac{Q}{W_{chw p}} \quad (5)$$

$$\mu_c = \frac{Q_c}{W_{cwp}} \quad (6)$$

where Q is the refrigeration capacity of the refrigeration plant, (kW);

$W_{chw p}$ is the chilled water pump power (kW);

Q_c is the heat rejection of the cooling water system (kW);

W_{cwp} is the cooling water pump power (kW).

Operational parameters with high occurrence frequency during the 2024–2025 operational period were selected for the calculation of pump transport coefficients. The calculation results are presented in Table 5. Since both the chilled water pumps and cooling water pumps utilize variable frequency drives, the transport coefficients for the chilled water pumps all exceed 30, and those for the cooling water pumps all exceed 45, indicating excellent operational performance.

Table 5. Water pump transport factor calculation results.

Item	1#	2#	3#	4#	5#
Chilled water pump transport factor	36.7~86.5	37.0~116.8	35.0~105.2	33.8~102.2	33.0~79.7
Cooling water pump transport factor	50.8~143.8	51.2~128.3	50.3~136.8	50.2~137.5	48.0~124.2

(3) Cooling tower.

The cooling tower approach degree and cooling tower efficiency are key indicators for evaluating cooling tower performance and are calculated using Equations (7) and (8), respectively.

$$Approach = T_{out} - T_s \quad (7)$$

$$\eta = \frac{T_{in} - T_{out}}{T_{in} - T_s} \quad (8)$$

where T_{out} is the cooling tower outlet water temperature (°C);

T_s is outdoor wet-bulb temperature (°C);

T_{in} is the cooling tower inlet water temperature (°C).

Based on the annual operational data of the cooling tower, operating conditions close to its rated inlet and outlet water temperatures were selected to compare the actual performance with the rated performance. The results are presented in Table 6. It can be observed that both in terms of approach and cooling tower efficiency, the actual performance of the cooling tower is close to or even better than its rated performance. The reason for this lies in the fact that Xiamen has a maritime climate, with an average annual temperature of

approximately 21 °C and an average relative humidity of 77%. The mild winters, where temperatures rarely drop below 5 °C, eliminate the risk of freezing throughout the year. Additionally, the sea breeze during summer enhances evaporative cooling efficiency, thereby improving the actual operational performance of the cooling tower. This indicates that during equipment selection, fully considering local climatic characteristics and choosing operational parameters—such as a lower approach temperature—that are well-suited to the climate can fully unlock the energy-saving potential of the equipment, thereby achieving synergy between design optimization and operational optimization.

Table 6. Comparison of actual vs. rated performance for cooling towers.

Item	Rated Condition	Operating Time			
		14 June 2024 12:30	29 June 2024 20:00	22 June 2024 11:30	30 June 2024 19:30
Time	/				
Cooling tower outlet water temperature (°C)	31	31.26	31.29	31.33	31.12
Cooling tower inlet water temperature (°C)	36	36.07	35.94	35.98	36.03
Outdoor wet-bulb temperature (°C)	28	28.57	27.77	28.19	28.24
Approach degree (°C)	3	2.69	3.52	3.14	2.88
Cooling tower efficiency	62.50%	64.13%	56.92%	60%	63%

5. Conclusions

This paper studies the equipment selection optimization of a commercial building refrigeration plant and provides measured data for 2024–2025, empirically validating the actual energy efficiency of the optimized design scheme and filling the gap in long-term performance verification data in this field.

- (1) A comprehensive optimization of the refrigeration plant equipment selection for a commercial building in Xiamen was conducted, encompassing chillers, chilled/cooling water pumps, and cooling towers. Theoretical calculations indicated that the optimized schemes (particularly for chillers and cooling towers) would offer significant energy-saving potential and favorable economic returns.
- (2) Empirical analysis based on long-term operational data from 2024 to 2025 confirmed the high efficiency of the optimized refrigeration plant. The system achieved annual comprehensive EER values of 5.44 (2024) and 5.28 (2025), with monthly EER consistently above 5.06. This performance meets the stringent Grade I efficiency classification according to the standard T/CRAAS 1039-2023.
- (3) Equipment-level performance evaluation revealed excellent real-world operation. Chiller efficiency deviations from rated values were generally controlled within 5%. Pump transport factors, benefiting from variable frequency drives, are excellent (≥ 30 for chilled water pumps, ≥ 45 for cooling water pumps). Cooling tower actual performance, favored by Xiamen's maritime climate, was close to or better than its rated performance.

This study integrates optimized equipment selection at the design stage with empirical performance analysis based on actual operation. It provides a practical and data-supported framework for achieving high energy efficiency in commercial building refrigeration plants, contributing valuable findings for engineering practice and future relevant energy efficiency standard development.

The proposed framework—integrating design-stage equipment selection with operational-stage empirical validation—is inherently transferable across climates and building types, although critical parameters (e.g., outdoor dry-bulb and wet-bulb temperatures, load profiles, etc.) require adjustment for local conditions. Future work will extend this methodology to diverse climate zones and building typologies to establish a comprehensive performance benchmark database and quantify the climatic sensitivity of optimal configurations through cross-regional comparative studies.

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Abbreviations

The following abbreviations are used in this manuscript:

EER	Energy efficiency ratio
MPC	Model predictive control
PLR	Part load ratio
CSD	Constant-speed drive
VSD	Variable-speed drive
VFD	Variable-frequency drive
RMSE	Root mean square error
MAE	Mean absolute error
MAPE	Mean absolute percentage error

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Article

Soil Heat Deficit Regulation-Based Cross-Seasonal Heat Storage of Medium-Deep Borehole Heat Exchanger

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Abstract: Medium-deep borehole heat exchanger (MBDHE) is a clean building heating solution that offers improved indoor thermal comfort, but it risks heat extraction capacity deterioration due to insufficient geothermal heat flux recovery. Cross-seasonal heat storage can mitigate heat extraction capacity deterioration. However, previous studies focused on optimizing the heat storage parameters without considering the mutual inhibition between geothermal heat flux recovery and the heat storage. This paper proposes a soil heat deficit regulation-based cross-seasonal heat storage method, which expands the soil heat deficit to enhance the geothermal heat flux recovery and heat storage simultaneously for improved heating performance of the MDBHE. The effectiveness of the proposed method is demonstrated by using an MDBHE model validated against actual engineering data. The results show that the flow rate-based soil heat deficit regulation is more effective than the temperature-based soil heat deficit regulation. Expanding the soil heat deficit by lowering the heat extraction temperature effectively increases the heat extraction capacity of MDBHE by up to 44.82% with a relatively stable heat storage efficiency. Expanding the soil heat deficit by increasing the heat extraction flow rate increases the heat extraction capacity by up to 55.93% and improves the heat storage efficiency by up to 18.71%. Furthermore, the robustness of the soil heat deficit regulation is verified under different heat storage conditions. Therefore, the proposed method can effectively improve the heating performance of MDBHE, enhancing indoor thermal comfort and contributing to the advancement of low-carbon building heating.

Keywords: medium-deep borehole heat exchanger; cross-seasonal heat storage; soil heat deficit regulation; heat extraction capacity; heat storage efficiency

1. Introduction

To achieve carbon neutrality and meet the growing global energy demand, transitioning to sustainable, renewable energy heating solutions is crucial for both environmental sustainability and energy security [1]. Geothermal resources, known for their wide distribution, low carbon emissions, reliability, and ability to enhance indoor thermal comfort, have become a primary source of clean heating in northern China [2]. Medium-deep borehole heat exchanger (MDBHE) technology offers advantages such as higher outlet temperatures, minimal land use, and reduced surface disturbance [3], making it particularly suitable for

large-scale deployment in urban areas with limited land resources. In recent years, the application of MDBHE has rapidly expanded. Since 2013, the integration of MDBHE with heat pump systems has gradually been adopted in northern China, currently serving over 30 million square meters of building area [4].

Extensive research, including field experiments, analytical calculations, and numerical simulation studies, has been conducted to investigate the heating performance of the DBHE system. Wang et al. [5] monitored an MDBHE system for an office building in Harbin, with an average heat exchange rate of 152.6 W/m and a heat pump coefficient of performance (COP) ranging from 4.46 to 6.89. Deng et al. [6] monitored an MDBHE with a depth of 2750 m, reporting an average heat extraction rate of 429.5 kW and a system COP of 3.68. To accurately model the long-term and short-term heat transfer processes of MDBHEs, Luo et al. [7–9] developed a series of analytical models, such as the non-uniform finite line source method [7] and the Response Factor Matrix method [10].

Continuous heat extraction of MDBHE causes a gradual underground soil temperature decrease during long-term operation, which diminishes the heat extraction capacity of MDBHE [11,12]. Chen et al. [13] reported that after ten years of operation, the soil temperature at a radial distance of 1 m from the borehole can fall by up to 7.46%, corresponding to a 9.67% reduction in heat extraction capacity. To mitigate this degradation in heat extraction capability, cross-seasonal heat storage is adopted [14,15]. During the non-heating season, the circulating fluid transfers heat to the underground from high-temperature sources such as solar energy and industrial or commercial waste heat, thereby raising the soil temperature. During the heating season, the stored heat is released to maintain or even enhance MDBHE heating performance.

Currently, cross-seasonal heat storage research mainly focuses on optimizing various heat storage parameters [16–18]. S. Brown et al. [19] conducted a comprehensive sensitivity analysis on an MDBHE, examining the effects of borehole depth, geothermal gradient, and flow rate on heat storage efficiency. Huang et al. [20] demonstrated that for effective heat storage in an MDBHE, fluid inflow from the inner pipe releases more heat into the surrounding rock and soil, increasing heat storage capacity by 10.81 kW compared to inflow from the annular space under the same conditions. Krusemark et al. [21] analyzed how deviations in borehole trajectories significantly influence the amounts of heat stored and recovered, and the overall storage efficiency. Fu et al. [22] examined the influence of borehole layout, the number of boreholes, inter-borehole spacing, and borehole depth on the seasonal heat storage performance of MDBHE arrays. This is primarily achieved by optimizing the heat storage process to mitigate the decline in MDBHE's heat extraction performance and improve heat storage efficiency.

During the heat storage process in the non-heating season, soil temperature recovery involves both geothermal heat flux recovery [23] and heat storage [16]. In the geothermal heat flux recovery process, heat is transferred from the higher-temperature radial and deep soil to the lower-temperature soil surrounding the MDBHE. In contrast, during heat storage, heat is transferred from the circulating fluid inside the MDBHE to the surrounding lower-temperature soil. The directions of heat transfer in geothermal heat flux recovery and heat storage are opposite, creating a mutual inhibition effect that limits the full contribution of both processes to soil temperature recovery. Current research on cross-seasonal thermal storage primarily focuses on optimizing various heat storage parameters to improve heat storage efficiency. However, this approach may inadvertently reduce the effectiveness of geothermal heat flux recovery. Therefore, it is crucial to consider the interaction between geothermal heat flux recovery and heat storage to optimize soil temperature recovery and ensure more efficient overall performance.

This paper proposes a novel soil heat deficit regulation-based cross-seasonal heat storage method for MDBHE. By expanding the soil heat deficit, the geothermal heat flux recovery and heat storage are enhanced simultaneously, thereby improving the heating performance of the MDBHE and enhancing indoor thermal comfort. Section 2.1 explains the working principle of this mechanism, while Section 2.2 discusses the development and validation of the model. Using the developed model, the impact of heat extraction temperature (Section 3.1) and heat extraction flow rate (Section 3.2) on the heating performance of MDBHE is analyzed. Finally, Section 4 demonstrates the robustness of the soil heat deficit regulation across different heat storage conditions. The main contributions of this paper are as follows:

- (1) A novel soil heat deficit regulation-based cross-seasonal heat storage method for MDBHE is proposed, which expands the soil heat deficit to enhance the geothermal heat flux recovery and heat storage simultaneously.
- (2) The effectiveness of the proposed cross-seasonal heat storage method is demonstrated with two soil heat deficit regulation strategies.
- (3) The robustness of the soil heat deficit regulation is verified across different heat storage conditions.

2. Methodology

2.1. Working Principles of Cross-Seasonal Heat Storage Based on Soil Heat Deficit Regulation

To address the decline in heating performance caused by soil temperature degradation during the heating season (Scenario 1 in Figure 1), a conventional approach is to inject heat into the surrounding soil during the non-heating season using a high-temperature circulating fluid, thereby accelerating soil temperature recovery (Scenario 2 in Figure 1). This study proposes a soil heat deficit regulation-based cross-seasonal heat storage method for MDBHE (Scenario 4 in Figure 1). By expanding the soil heat deficit (i.e., lowering the heat extraction temperature or increasing the heat extraction flow rate), the geothermal heat flux recovery and heat storage are simultaneously enhanced, leading to a significant improvement in the heating performance of MDBHE.

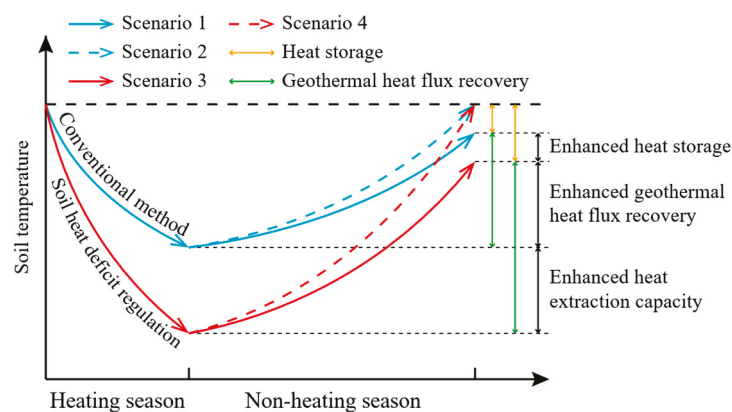


Figure 1. Schematic diagram of the cross-seasonal heat storage of the MDBHE based on the soil heat deficit regulation. Scenario 1—conventional method without heat storage; Scenario 2—conventional method with heat storage; Scenario 3—soil heat deficit regulation without heat storage; Scenario 4—soil heat deficit regulation with heat storage.

During the cross-seasonal heat storage, soil heat recovery involves two primary processes: (i) geothermal heat flux recovery, where geothermal heat flows from the deep and radial soil layers toward the cooled surrounding soil zone around the MDBHE; and

(ii) heat storage, where heat flows from the high-temperature circulating fluid inside the MDBHE toward the cooled surrounding soil zone around the MDBHE. However, the heat transfer direction of these two processes opposes each other, resulting in a phenomenon of mutual inhibition. This opposing heat transfer direction prevents the optimal utilization of geothermal heat flux recovery and reduces the overall efficiency of heat storage, therefore limiting the effectiveness of cross-seasonal heat storage of the MDBHE.

To effectively resolve this issue, this study proposes a soil heat deficit regulation-based cross-seasonal heat storage method for MDBHE. The soil heat deficit is defined as the difference in the amount of heat in the soil before and after the heat extraction, as described in Equation (1). If the soil heat deficit can be replenished during the non-heating season, the heat extraction capacity of MDBHE in the second year can be maintained at the same level as in the first year. By expanding the soil heat deficit, the geothermal heat flux recovery (Scenario 3 in Figure 1) can be enhanced while maintaining sufficient space for cross-seasonal heat storage regulation and improving heat storage efficiency. The effectiveness of this approach will be verified through temperature-based soil heat deficit regulation (Section 3.1) and the flow rate-based soil heat deficit regulation (Section 3.2). The robustness of the soil heat deficit regulation will be demonstrated across different heat storage conditions in Section 4.1.

$$SHD = Q_{be} - Q_{ae} = Q_{ghf} + Q_{hs} \quad (1)$$

where SHD refers to the soil heat deficit; Q_{be} refers to the amount of heat contained in the soil before heat extraction; Q_{ae} refers to the heat contained in the soil after heat extraction; Q_{ghf} refers to the geothermal heat flux recovery; Q_{hs} refers to the heat storage.

2.2. Mathematical Model and Validation

The heat transfer process within the MDBHE is divided into three main parts [24]: (i) heat conduction between the surrounding soil and the outer tube wall; (ii) heat conduction between the outer tube wall and the circulating fluid in the annular space; and (iii) heat conduction between the circulating fluid in the annular space and the circulating fluid inside the inner tube. For simplification, the following assumptions are made: (i) the heat exchange process inside the MDBHE is treated as a quasi-steady-state; (ii) the heat capacity and storage effects of the inner and outer tube walls are neglected, and only heat conduction through the walls is considered; (iii) there are no gaps or thermal contact resistance between the backfill material, tube wall, and borehole wall; and (iv) the thermophysical properties of all materials (including the circulating fluid, backfill, and surrounding soil) are considered as constant.

The numerical model for the MDBHE is developed using the open source finite element software OpenGeoSys 6.3.2 (OGS) [25]. OGS employs a dual-continuum finite element framework to simulate the temporal variations in the inlet and outlet temperatures of the MDBHE, along with the surrounding soil temperature field. In the numerical grid, the MDBHE and its backfill material are represented as one-dimensional line elements, while the surrounding soil is discretized using three-dimensional prismatic elements [3,26]. The governing equations for the circulating fluid inside the inner pipe and annular space are given in Equations (2) and (4), respectively, with Robin-type boundary conditions described in Equations (3) and (5). The heat transfer in the grout surrounding the outer pipe is primarily governed by conduction (assuming that the grout porosity is zero), as expressed in Equation (6), with the corresponding Robin-type boundary condition provided in Equation (7). Finally, the rock temperature T_r is determined by the energy balance

equation considering both heat convection and conduction, as shown in Equation (8), with a Neumann-type boundary condition in Equation (9).

$$\rho_f c_f \frac{\partial T_i}{\partial t} + \rho_f c_f v_i \cdot \nabla T_i - \nabla \cdot (\Lambda_f \cdot T_i) = H_i \quad (2)$$

$$-\Phi_{io}(T_o - T_i) = q_{nT_i} \text{ on } \Gamma_i \quad (3)$$

$$\rho_f c_f \frac{\partial T_o}{\partial t} + \rho_f c_f v_o \cdot \nabla T_o - \nabla \cdot (\Lambda_f \cdot T_o) = H_o \quad (4)$$

$$-\Phi_{io}(T_o - T_i) - \Phi_{og}(T_g - T_o) = q_{nT_o} \text{ on } \Gamma_o \quad (5)$$

$$\rho_g c_g \frac{\partial T_g}{\partial t} - \nabla \cdot (\lambda_g \cdot \nabla T_g) = H_g \quad (6)$$

$$-\Phi_{gr}(T_r - T_g) - \Phi_{og}(T_o - T_g) = q_{nT_g} \text{ on } \Gamma_g \quad (7)$$

$$\frac{\partial}{\partial t} [\phi \rho_w c_w + (1 - \phi) \rho_r c_r] T_r + \nabla \cdot (\rho_w c_w v_w T_r) - \nabla \cdot (\Lambda_w \cdot \nabla T_r) = H_r \quad (8)$$

$$-(\Lambda_w \cdot \nabla T_r) = q_{nT_r} \text{ on } \Gamma_r \quad (9)$$

where ρ_f and c_f refer to the density and specific heat capacity of the circulating fluid, respectively. v_i and v_o denote the inner and outer pipe flow velocities, respectively. Λ_f represents the hydrodynamic thermos-dispersion tensor, which in this case can be simplified as the fluid thermal conductivity. H and Γ are the heat source/sink term and heat transfer boundary, respectively. In the boundary flux terms, Φ_{io} and Φ_{og} represent the heat transfer coefficients between the inner and outer pipes, and between the outer pipe and grout, respectively. The subscript g represents the grout. In the boundary flux term, Φ_{gr} is the thermal resistance between the grout and rock, and T_r is the soil temperature. The heat transfer coefficients Φ_{io} , Φ_{og} and Φ_{gr} are functions of the borehole geometry and thermal resistances. For the detailed calculation method of heat transfer coefficients, readers can refer to [27]. ϕ is the effective porosity, and the subscript w denotes groundwater. The heat flux boundary condition was imposed on the rock/grout interface Γ_r .

A numerical heat transfer model is developed to evaluate and optimize the thermal performance of the MDBHE for cross-seasonal heat storage. To accurately simulate heat transfer, the computational domain is set to $200 \text{ m} \times 200 \text{ m} \times 2300 \text{ m}$, ensuring a sufficient distance from the MDBHE to prevent thermal interference at the boundaries. The boundary conditions are defined as follows: (i) a Dirichlet boundary condition is applied to the top surface, governed by the monthly average ambient temperature; (ii) Neumann boundary conditions are assigned to the lateral boundaries [28]; and (iii) a constant heat flux of 60 mW/m^2 is imposed at the bottom boundary to represent the local geothermal heat flux [29]. The initial soil temperature field is initialized based on local geological survey data, assuming a surface temperature of 15°C and a geothermal gradient of 33.0°C/km [30]. The thermophysical properties of the stratified soil are summarized in Table 1 [31], while the geometric specifications and operational parameters of the MDBHE are summarized in Table 2.

The MDBHE model developed in OGS has been previously validated through analytical solutions, numerical benchmarks, and field experiments [32]. In this study, further validation is carried out by comparing the simulation results with the measured data [33,34] (see Figure 2). In Figure 2a, slight fluctuations in the inlet and outlet temperatures were observed, which can be attributed to variations in outdoor ambient conditions and indoor heating demand. The comparison reveals a maximum relative deviation of 4% between the simulated and measured outlet temperatures. Figure 2b presents a comparison with

720 h of experimental data from another project, where the overall error is found to be less than 6%. Therefore, the developed numerical model is considered reliable for simulating heat transfer between the MDBHE and the surrounding soil, as well as for calculating soil temperatures and the inlet and outlet temperatures.

Table 1. Thermophysical properties of the underground soil with depth.

Depth m	Thermal Conductivity $\text{W} \cdot \text{m}^{-1} \cdot ^\circ\text{C}^{-1}$	Density $\text{kg} \cdot \text{m}^{-3}$	Specific Heat Capacity $\text{J} \cdot \text{kg}^{-1} \cdot ^\circ\text{C}^{-1}$
0–500	1.60	1760	1433
500–740	1.63	1860	1025
740–1440	1.70	1920	978
1440–2500	1.81	2070	948

Table 2. Basic parameters of the MDBHE model.

Parameter	Value/Unit
Borehole depth	2000 m
Borehole diameter	0.2159 m
Outer diameter of inner tube	0.11 m
Wall thickness of the inner tube	0.01 m
Thermal conductivity of the inner tube wall	0.42 $\text{W}/(\text{m} \cdot ^\circ\text{C})$
Outer diameter of outer tube	0.1778 m
Wall thickness of the outer tube	0.0092 m
Thermal conductivity of the outer tube wall	40 $\text{W}/(\text{m} \cdot ^\circ\text{C})$
Grout density	2190 kg/m^3
Grout specific heat capacity	1735 $\text{J}/(\text{kg} \cdot ^\circ\text{C})$
Grout thermal conductivity	0.63 $\text{W}/(\text{m} \cdot ^\circ\text{C})$
Circulation fluid density	998 kg/m^3
Circulation fluid specific heat capacity	4190 $\text{J}/(\text{kg} \cdot ^\circ\text{C})$
Circulation fluid thermal conductivity	0.6 $\text{W}/(\text{m} \cdot ^\circ\text{C})$
Circulation fluid dynamic viscosity	0.000931 $\text{kg}/(\text{m} \cdot \text{s})$

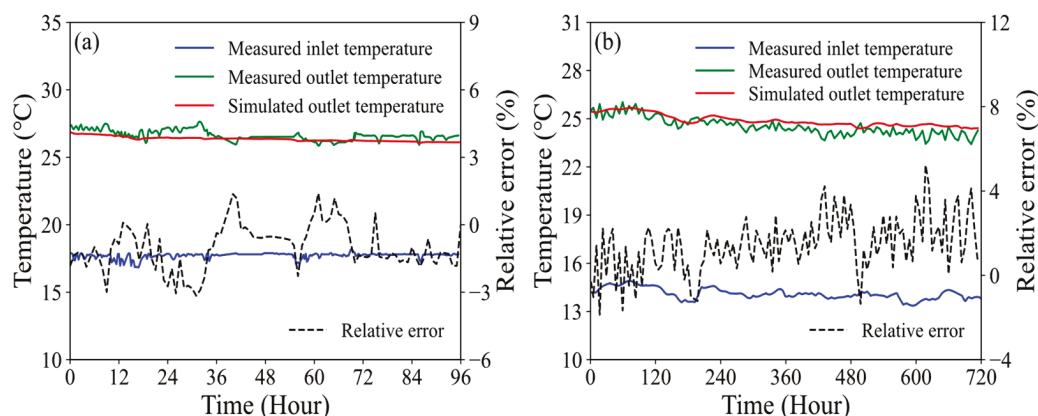


Figure 2. Comparison between measured and simulated outlet temperatures: (a) measured outlet temperature reported in Ref. [33]; (b) measured outlet temperature reported in Ref. [34].

2.3. Evaluation Criteria

To assess the impact of expanding the soil heat deficit on cross-seasonal heat storage performance, two evaluation criteria are employed: heat extraction capacity and heat storage efficiency. The calculation formulas are as follows:

$$q_{\text{extraction}} = c_f m_{\text{extraction}} (t_{\text{out}} - t_{\text{in}}) \quad (10)$$

$$q_{\text{extraction}} = c_f m_{\text{extraction}} (t_{\text{out}} - t_{\text{in}}) \quad (11)$$

$$Q_{\text{extraction}} = \sum_{i=ES}^{EE} c_f m_{\text{extraction}} (t_{\text{out}}(i) - t_{\text{in}}(i)) \Delta \tau \quad (12)$$

$$Q_{\text{storage}} = \sum_{i=SS}^{SE} c_f m_{\text{storage}} (t_{\text{in}}(i) - t_{\text{out}}(i)) \Delta \tau \quad (13)$$

$$\bar{q}_{\text{extraction}} = \frac{Q_{\text{extraction}}}{EE - ES} \quad (14)$$

$$\eta = \frac{\Delta Q_{\text{extraction}}}{Q_{\text{storage}}} = \frac{Q_{\text{extraction}}^{\text{storage}} - Q_{\text{extraction}}^{\text{no storage}}}{Q_{\text{storage}}} \quad (15)$$

where m_{storage} and $m_{\text{extraction}}$ are the mass flow rates of the circulating fluid during the non-heating season and heating season ($\text{kg} \cdot \text{s}^{-1}$); t_{in} and t_{out} are the inlet and outlet temperatures ($^{\circ}\text{C}$); SS , SE , ES , and EE are the start and end times of the non-heating season and heating season (s); $\Delta \tau$ is the simulation time step; $q_{\text{extraction}}$ and q_{storage} are the heat extraction and heat storage capacity (kW); Q_{storage} and $Q_{\text{extraction}}$ are the accumulated extracted and stored heat (J); $\bar{q}_{\text{extraction}}$ is the seasonal average heat extraction capacity; η is the heat storage efficiency (%). The superscript “storage” refers to the heat supplied after cross-seasonal heat storage; “no storage” refers to the heat supplied without storage.

2.4. Simulated Conditions

The soil heat deficit-based regulation for cross-seasonal heat storage of MDBHE involves both temperature-based soil heat deficit regulation and flow rate-based soil heat deficit regulation. The benchmark case in this study is based on the conventional operating conditions commonly used in practical engineering applications. Under the benchmark condition, the heat extraction temperature is set to 21°C , and the heat extraction flow rate is set to $10 \text{ m}^3/\text{h}$. To regulate the soil heat deficit, temperature-based soil heat deficit regulation involves gradually decreasing the heat extraction temperature, with four distinct temperature settings: 18°C , 15°C , 12°C , and 9°C (Section 3.1). Flow rate-based soil heat deficit regulation involves gradually increasing the heat extraction flow rate, with four distinct flow rate settings: 15, 20, 25, and $30 \text{ m}^3/\text{h}$ (Section 3.2). During the non-heating season, the heat storage temperature is consistently set to 50°C , with the heat storage flow rate maintained at $20 \text{ m}^3/\text{h}$. Furthermore, to demonstrate the robustness of the soil heat deficit regulation under varying heat storage conditions (Section 4.1), the heat storage temperatures are set to 50°C , 60°C , and 70°C based on the maximum soil heat deficit (achieved with a heat extraction temperature of 9°C and a heat extraction flow rate of $30 \text{ m}^3/\text{h}$) to compare MDBHE’s performance. These configurations provide a comprehensive evaluation of the impact of soil heat deficit regulation on MDBHE’s cross-seasonal heat storage performance and the robustness of the proposed regulation approach.

3. Results

3.1. Effectiveness of Temperature-Based Soil Heat Deficit Regulation

3.1.1. Soil Heat Deficit Regulation

The temperature-based soil heat deficit regulation is achieved by adjusting the heat extraction temperature. Different heat extraction temperatures affect the heat exchange between the circulating fluid and the surrounding soil, thereby altering the soil heat deficit in the surrounding soil zone. Figure 3 shows the variation in soil temperature at 0.7 m

from the MDBHE under temperature-based soil heat deficit regulation. During the heating season, lowering the heat extraction temperature reduces soil temperature. For example, at the end of the first heating season, when the heat extraction temperature is reduced from the baseline of 21 °C to 18 °C, 15 °C, 12 °C, and 9 °C, the soil temperature at a depth of 500 m decreases by 1.08 °C, 2.17 °C, 3.25 °C, and 4.34 °C, respectively. A further comparison of soil temperature changes at different depths reveals the impact of heat extraction temperature on soil temperature. When the heat extraction temperature is reduced from the baseline of 21 °C to 9 °C, at the end of the first heating season, the soil temperature decreases by 4.34 °C at 500 m, 3.43 °C at 1000 m, 2.75 °C at 1500 m, and 2.17 °C at 2000 m. This phenomenon arises because, as the heat extraction temperature decreases, the temperature difference between the circulating fluid and the shallow soil layers increases, resulting in more heat being absorbed by the circulating fluid from the shallow soil layer. As the circulating fluid moves deeper, its temperature gradually rises, reducing the amount of additional heat absorbed from the deeper soil layers. Therefore, the temperature drop in the shallow soil layers is greater than that in the deeper soil layers. These results indicate that lowering the heat extraction temperature increases the soil heat deficit, with this deficit being primarily concentrated in the shallow soil layers.

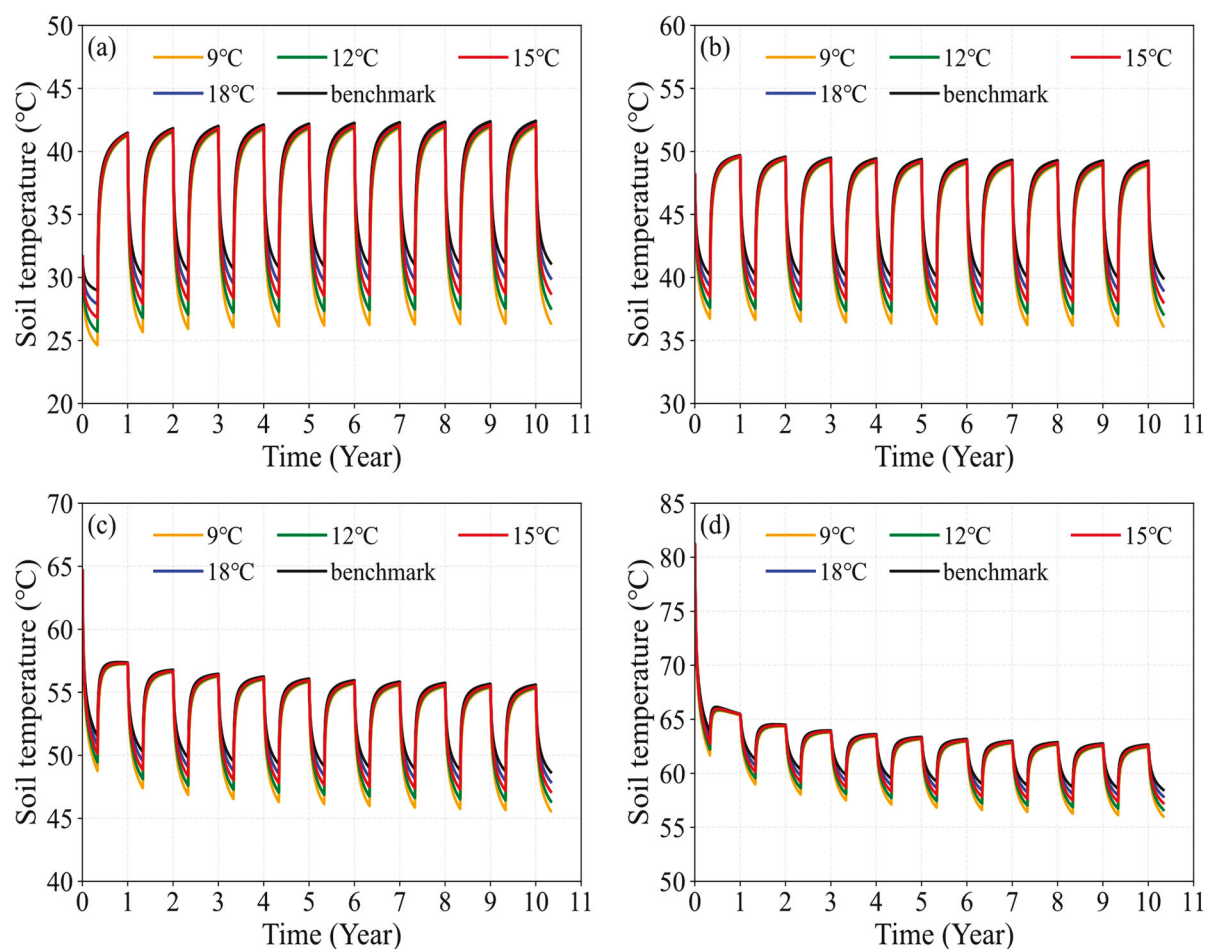


Figure 3. Variation in soil temperature under temperature-based soil heat deficit regulation: (a) 500 m depth; (b) 1000 m depth; (c) 1500 m depth; (d) 2000 m depth.

The heat storage process during the non-heating season accelerates soil temperature recovery. However, different heat extraction temperatures create varying soil temperature fields at the beginning of the non-heating season, affecting heat storage performance. For

example, the soil temperature in the benchmark condition increased by 12.52 °C (43.22%) at a depth of 500 m during the first non-heating season. In contrast, when the heat extraction temperatures are set to 18 °C, 15 °C, 12 °C, and 9 °C, the soil temperature increased by 13.55 °C (48.61%), 14.58 °C (54.43%), 15.61 °C (60.71%), and 16.65 °C (67.61%), respectively. This indicates that as the heat extraction temperature decreases, the increase in soil temperature becomes more pronounced during the non-heating season. Therefore, while the soil temperature difference at the beginning of the non-heating season is more pronounced, this difference diminishes significantly by the end of the non-heating season. For example, at the beginning of the first non-heating season, the maximum soil temperature differences at depths of 500, 1000, 1500, and 2000 are 4.34 °C, 3.43 °C, 2.75 °C, and 2.17 °C, respectively. By the end of the first non-heating season, these differences reduce to 0.22 °C, 0.16 °C, 0.13 °C, and 0.11 °C. This happens because lower heat extraction temperatures lead to a greater soil heat deficit at the beginning of the non-heating season. This, in turn, enhances both geothermal heat flux recovery and heat storage, ultimately accelerating soil temperature recovery. In conclusion, the lower the heat extraction temperature, the stronger the geothermal heat flux recovery and heat storage during the non-heating season, resulting in more significant soil temperature recovery. The soil heat deficit induced by lowering the heat extraction temperature is largely recovered.

3.1.2. Effects on Heat Extraction Capacity

The variation in heat extraction temperature directly influences the heat extraction capacity of the MDBHE, as shown in Figure 4. Compared to the benchmark condition, a lower heat extraction temperature results in a greater increase in the heat extraction capacity of the MDBHE, with the most significant enhancement occurring at the start of the heating season. For example, at the beginning of the first heating season, compared to the benchmark condition, the heat extraction capacities at heat extraction temperatures of 18 °C, 15 °C, 12 °C, and 9 °C increased by 19.70 kW (13.41%), 39.38 kW (26.80%), 59.08 kW (40.21%), and 78.77 kW (53.61%), respectively. By the end of the first heating season, the heat extraction capacities at the same temperatures increased by 10.45 kW (11.12%), 19.69 kW (22.22%), 29.54 kW (33.34%), and 39.38 kW (44.45%), respectively, compared to the benchmark condition. This occurs because, when the heat extraction temperature is lowered, the temperature difference between the circulating fluid and the surrounding soil increases, leading to a significant increase in the heat extraction capacity of MDBHE.

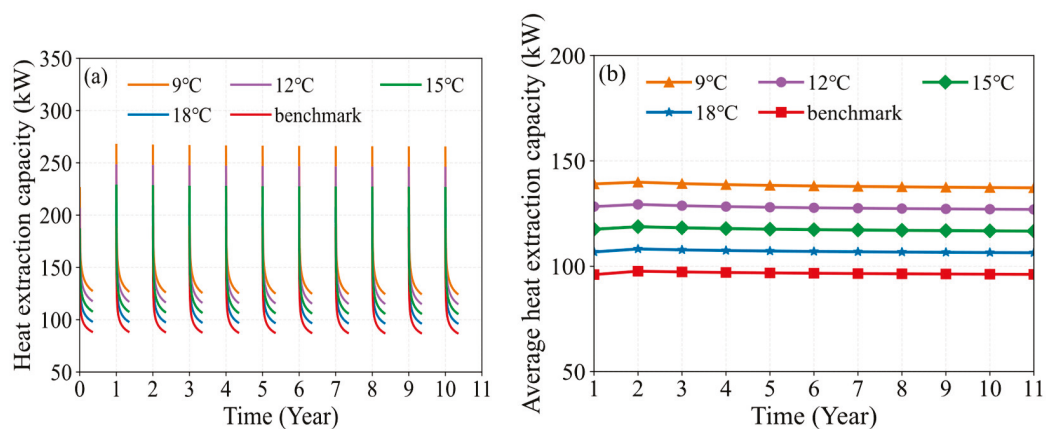


Figure 4. Variation in heat extraction capacity under temperature-based soil heat deficit regulation: (a) hourly variation; (b) seasonal average value.

Figure 4b shows the variation in the seasonal average heat extraction capacity of the MDBHE under temperature-based soil heat deficit regulation. Lowering the heat extraction temperature greatly enhances the seasonal average heat extraction capacity of MDBHE. For example, in the first heating season, reducing the heat extraction temperature to 18 °C, 15 °C, 12 °C, and 9 °C results in increases in the seasonal average heat extraction capacity of 10.76, 21.52, 32.28, and 43.04 kW (11.20%, 22.41%, 33.61%, and 44.82%), respectively, compared to the benchmark condition. Over the entire operational period, temperature-based soil heat deficit regulation produces a maximum increase of 44.82% in the MDBHE's seasonal average heat extraction capacity compared to the benchmark condition. During the entire operational period, the seasonal average heat extraction capacity in the benchmark condition increases slightly from 96.03 kW to 96.14 kW. In contrast, lowering the heat extraction temperature slightly reduces the seasonal average heat extraction capacity. For heat extraction temperatures of 18 °C, 15 °C, 12 °C, and 9 °C, the seasonal average heat extraction capacity decreases by only 0.36%, 0.74%, 1.32%, and 1.06%, respectively. Overall, temperature-based soil heat deficit regulation significantly improves the heat extraction capacity of MDBHE, with only a slight decrease in seasonal average heat extraction capacity over the operational period.

3.1.3. Effects on Heat Storage Efficiency

The variation in heat extraction temperature during the heating season alters the underground soil temperature field, which subsequently affects the heat storage capacity of MDBHE during the non-heating season. Figure 5a shows the changes in MDBHE's heat storage capacity during the non-heating season under temperature-based soil heat deficit regulation. Compared to the benchmark condition, lowering the heat extraction temperature leads to a greater increase in heat storage capacity at the start of the non-heating season. However, the differences in heat storage capacity become minimal by the end of the non-heating season. For example, at the beginning of the first non-heating season, lowering the heat extraction temperature to 18 °C, 15 °C, 12 °C, and 9 °C increases the heat storage capacity of the MDBHE by 16.94, 33.87, 50.81, and 67.74 kW (8.86%, 17.71%, 26.57%, and 35.42%), respectively, compared to the benchmark condition. By the end of the first non-heating season, these increases are reduced to 0.33, 0.67, 1.00, and 1.32 kW (2.13%, 4.41%, 6.53%, and 8.66%), respectively. This occurs because a lower heat extraction temperature creates a larger soil heat deficit at the beginning of the non-heating season, leading to a greater increase in MDBHE's heat storage capacity. However, as heat storage progresses, the soil heat deficit diminishes, which consequently reduces the increase in MDBHE's heat storage capacity.

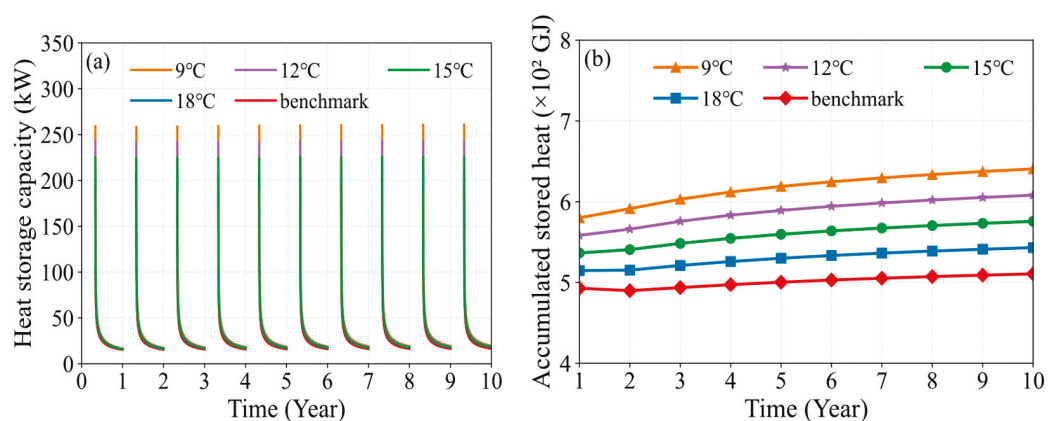


Figure 5. Variation in (a) heat storage capacity and (b) accumulated stored heat of the MDBHE under temperature-based soil heat deficit regulation.

Figure 5b illustrates the variation in the accumulated stored heat of the MDBHE during the non-heating season under temperature-based soil heat deficit regulation. Compared to the benchmark condition, lowering the heat extraction temperature significantly increases the accumulated stored heat, with a more pronounced increase over the operational period. For example, in the first non-heating season, reducing the heat extraction temperature to 18 °C, 15 °C, 12 °C, and 9 °C results in an increase in accumulated stored heat from 492.87 GJ (benchmark condition) to 514.70, 536.54, 558.32, and 580.11 GJ, respectively, corresponding to increases of 4.52%, 9.04%, 13.54%, and 18.05%. Over the entire operational period, the accumulated stored heat under the benchmark condition increases by 3.62%. In contrast, when the heat extraction temperature is lowered to 18 °C, 15 °C, 12 °C, and 9 °C, the accumulated stored heat increases by 5.53%, 7.29%, 8.92%, and 10.43%, respectively. This is due to the larger soil heat deficit induced by lower heat extraction temperatures, which increases over time and enables more heat to be stored. Overall, temperature-based soil heat deficit regulation significantly enhances the accumulated stored heat during the non-heating season.

Figure 6 shows the variation in heat storage efficiency over time under temperature-based soil heat deficit regulation. It can be observed that heat storage efficiency gradually increases over the operational period. However, lowering the heat extraction temperature slightly decreases the MDBHE's heat storage efficiency. For instance, under the benchmark condition, the heat storage efficiency increases from 8.24% to 14.00% over the entire operational period. When the heat extraction temperature is lowered from the benchmark condition to 18 °C, 15 °C, 12 °C, and 9 °C, the heat storage efficiency in the first year decreases from 8.24% to 8.12%, 8.01%, 7.92%, and 7.83%, respectively. Over the entire operational period, the maximum reduction in heat storage efficiency compared to the benchmark condition under temperature-based soil heat deficit regulation is only 0.54%. This is because lowering the heat extraction temperature concentrates the soil heat deficit in the shallow soil layers, where most of the heat is stored during the non-heating season. This increases the thermal influence radius of the shallow soil. Consequently, a larger portion of the heat stored in the shallow soil dissipates during the heating season, leading to a slight reduction in the MDBHE's heat storage efficiency. Overall, temperature-based soil heat deficit regulation significantly enhances geothermal heat flux recovery and heat storage, greatly improving the heat extraction capacity (as shown in Figure 4), while the reduction in heat storage efficiency is minimal (as shown in Figure 6). Therefore, temperature-based soil heat deficit regulation proves to be an effective method for improving the MDBHE's cross-seasonal heat storage performance.

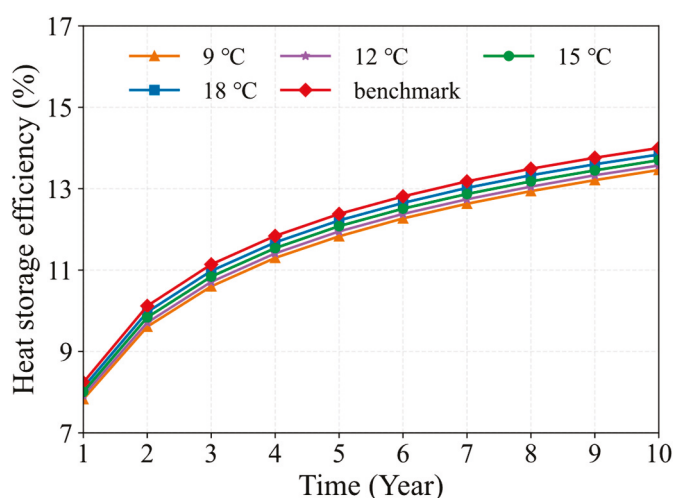


Figure 6. Variation in heat storage efficiency of the MDBHE under temperature-based soil heat deficit regulation.

3.2. Effectiveness of Flow Rate-Based Soil Heat Deficit Regulation

3.2.1. Soil Heat Deficit Regulation

The flow rate-based soil heat deficit regulation is achieved by adjusting the heat extraction flow rate. Variations in the flow rate affect the heat exchange between the circulating fluid and the surrounding soil, which in turn influences the soil heat deficit within the surrounding soil zone. Figure 7 illustrates the variation in soil temperature at a distance of 0.7 m from the MDBHE under different flow rate-based soil heat deficit conditions. During the heating season, increasing the heat extraction flow rate leads to a reduction in soil temperature. For instance, at the end of the first heating season, when the heat extraction flow rate is increased from the baseline of 10 m³/h to 15, 20, 25, and 30 m³/h, the soil temperature at 500 m depth decreases by 1.03 °C, 1.55 °C, 1.83 °C, and 2.00 °C, respectively. When the flow rate increases from the baseline of 10 m³/h to 30 m³/h, the soil temperature at the end of the first heating season decreases by 2.00 °C at 500 m, 4.26 °C at 1000 m, 6.31 °C at 1500 m, and 7.24 °C at 2000 m, compared to the benchmark condition. This occurs because, as the flow rate increases, the heat exchange time between the circulating fluid and the shallow soil layers decreases, resulting in less efficient heat transfer. Consequently, the circulating fluid's temperature during heat exchange with deeper soil layers is lower, allowing it to absorb more heat from the deep soil layers. As a result, the temperature decrease is more significant in the deeper soil layers than in the shallow soil layers. These results suggest that increasing the heat extraction flow rate enhances the soil heat deficit, which is primarily concentrated in the deeper soil layers.

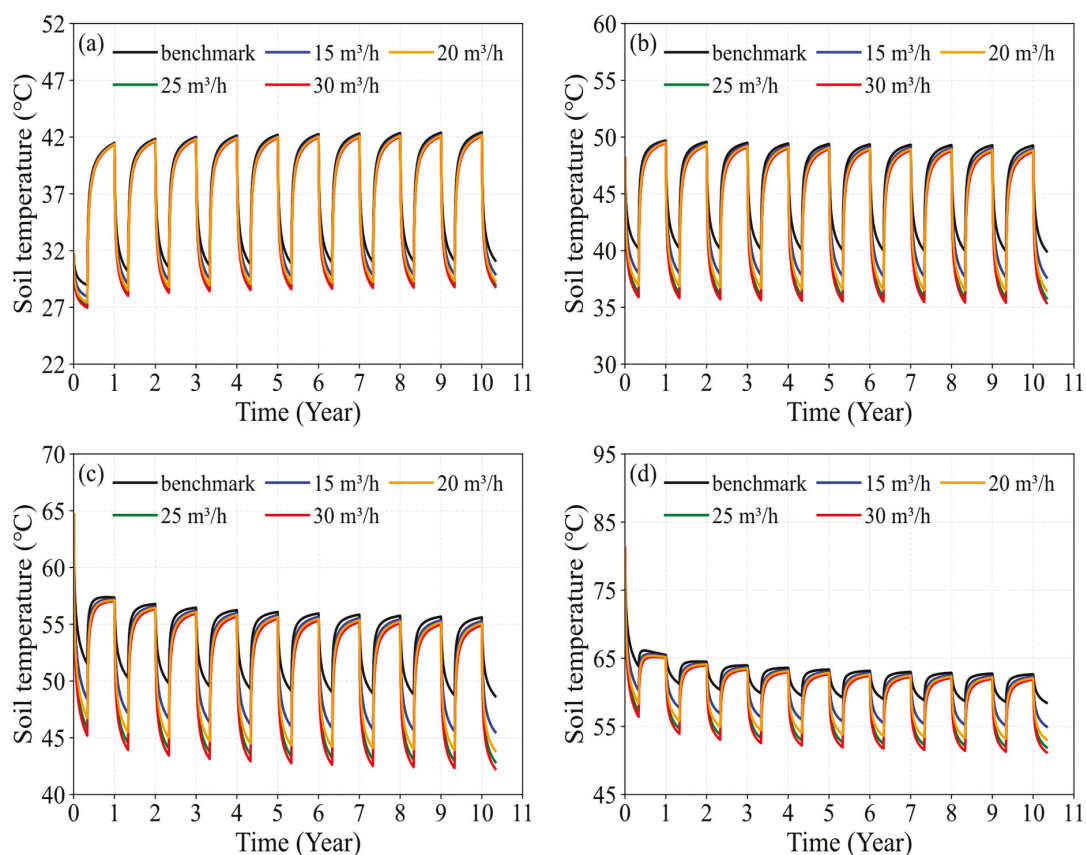


Figure 7. Variation in soil temperature under flow rate-based soil heat deficit regulation: (a) 500 m depth; (b) 1000 m depth; (c) 1500 m depth; (d) 2000 m depth.

The heat storage process during the non-heating season accelerates soil temperature recovery. However, varying heat extraction flow rates create different soil temperature profiles at the beginning of the non-heating season, which subsequently influences heat storage performance. For example, in the first non-heating season, the soil temperature under the benchmark condition increased by 12.52 °C (43.22%) at a depth of 500 m. In comparison, with heat extraction flow rates set to 15, 20, 25, and 30 m³/h, the soil temperature increased by 13.49 °C (48.28%), 13.97 °C (50.94%), 14.23 °C (52.45%), and 14.39 °C (53.37%), respectively. This demonstrates that increasing the heat extraction flow rate results in a more significant increase in soil temperature during the non-heating season. As a result, although the soil temperature difference at the beginning of the non-heating season is more pronounced with higher heat extraction flow rates, this difference decreases considerably by the end of the non-heating season. For instance, at the beginning of the first non-heating season, the maximum temperature differences at 500, 1000, 1500, and 2000 m, compared to the benchmark condition, are 2.00 °C, 4.26 °C, 6.31 °C, and 7.42 °C, respectively. By the end of the first non-heating season, these differences narrowed to 0.13 °C, 0.23 °C, 0.32 °C, and 0.41 °C. This phenomenon occurs because higher heat extraction flow rates lead to a larger soil heat deficit at the beginning of the non-heating season, enhancing both geothermal heat flux recovery and heat storage, thus accelerating soil temperature recovery. The soil heat deficit created by higher heat extraction flow rates is largely restored by the end of the non-heating season.

3.2.2. Effects on Heat Extraction Capacity

As shown in Figure 8, variations in the heat extraction flow rate directly impact the heat extraction capacity of the MDBHE. Compared to the benchmark condition, higher heat extraction flow rates result in greater improvements in the heat extraction capacity of the MDBHE, with more significant increases observed at the beginning of the heating season. For example, at the beginning of the first heating season, when the heat extraction flow rates are set to 15, 20, 25, and 30 m³/h, the heat extraction capacity of the MDBHE increases by 88.81 kW (60.45%), 149.26 kW (101.60%), 186.31 kW (126.81%), and 207.75 kW (141.41%) relative to the benchmark condition. However, by the end of the first heating season, the corresponding improvements in heat extraction capacity are 21.72 kW (24.51%), 32.80 kW (37.02%), 39.08 kW (44.11%), and 42.98 kW (48.50%) relative to the benchmark condition. This phenomenon occurs because, at the beginning of the heating season, the soil temperature is relatively higher. When the heat extraction flow rate is increased, the heat extraction capacity improves more significantly. However, as heat is extracted and the soil temperature decreases, the additional increase in heat extraction capacity with further increases in heat extraction flow rate becomes less pronounced.

Figure 8b illustrates the variation in the seasonal average heat extraction capacity of the MDBHE under flow rate-based soil heat deficit regulation. Increasing the heat extraction flow rate significantly improves the seasonal average heat extraction capacity of the MDBHE. For example, during the first heating season, when the heat extraction flow rate is increased to 15, 20, 25, and 30 m³/h, the seasonal average heat extraction capacity increases by 26.51 kW, 40.55 kW, 48.65 kW, and 53.71 kW, corresponding to improvements of 27.60%, 42.23%, 50.66%, and 55.93%, respectively, compared to the benchmark condition. Over the entire operational period, the seasonal average heat extraction capacity of MDBHE increased by a maximum of 55.93% under flow rate-based soil heat deficit regulation. In contrast, under the benchmark condition, the seasonal average heat extraction capacity slightly increased from 96.03 kW to 96.14 kW. However, increasing the heat extraction flow rate slightly reduces the seasonal average heat extraction capacity. At flow rates of

15, 20, 25, and 30 m³/h, the seasonal average heat extraction capacity decreases by 2.35%, 3.29%, 3.73%, and 3.98%, respectively. Overall, flow rate-based soil heat deficit regulation significantly enhances the heat extraction capacity of MDBHE, with only a slight decrease in the seasonal average heat extraction capacity over the entire operational period.

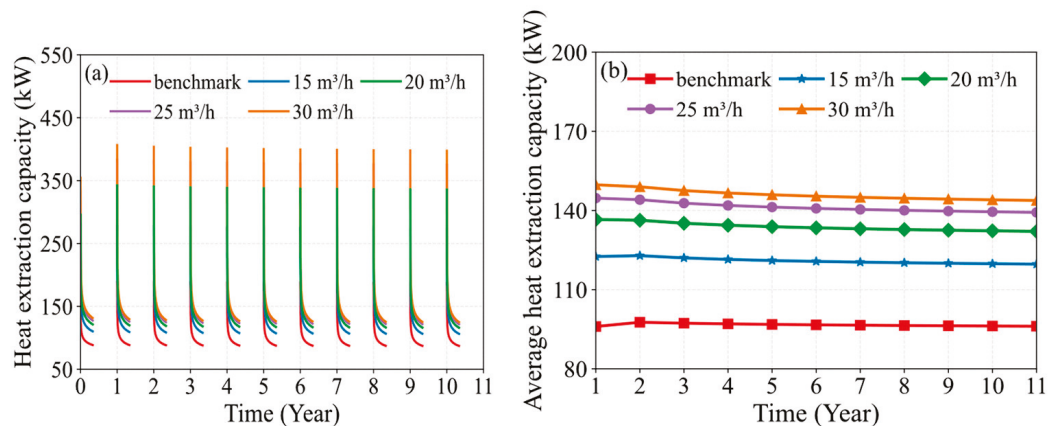


Figure 8. Variation in heat extraction capacity under flow rate-based soil heat deficit regulation: (a) hourly variation; (b) seasonal average value.

3.2.3. Effects on Heat Storage Efficiency

The variation in heat extraction flow rate during the heating season alters the underground soil temperature field, which subsequently affects the MDBHE's heat storage capacity during the non-heating season. Figure 7 illustrates the variation in MDBHE's heat storage capacity during the non-heating season under flow rate-based soil heat deficit regulation. Compared to the benchmark condition, increasing the heat extraction flow rate significantly enhances the heat storage capacity at the beginning of the non-heating season. However, by the end of the non-heating season, the difference in heat storage capacity becomes much smaller. For example, at the beginning of the first non-heating season, when the heat extraction flow rate is increased to 15, 20, 25, and 30 m³/h, the MDBHE's heat storage capacity increases by 33.85, 51.32, 61.26, and 67.46 kW, corresponding to improvements of 17.70%, 26.83%, 32.03%, and 35.28%, respectively, compared to the benchmark condition. By the end of the first non-heating season, the increase in heat storage capacity is reduced to 0.81, 1.23, 1.49, and 1.65 kW, corresponding to improvements of 5.32%, 8.05%, 9.73%, and 10.79%. This occurs because higher heat extraction flow rates lead to a greater soil heat deficit at the beginning of the non-heating season, which leads to a larger increase in MDBHE's heat storage capacity. However, as the heat storage process progresses, the soil heat deficit decreases significantly, which in turn reduces the enhancement in heat storage capacity.

Figure 9b shows the variation in the accumulated stored heat of MDBHE during the non-heating season under flow rate-based soil heat deficit regulation. Compared to the benchmark condition, increasing the heat extraction flow rate significantly boosts the accumulated stored heat, with a more substantial increase observed over the entire operational period. For example, during the first non-heating season, increasing the heat extraction flow rate to 15, 20, 25, and 30 m³/h leads to an increase in accumulated stored heat from 492.87 GJ (benchmark condition) to 541.37 GJ, 566.76 GJ, 581.39 GJ, and 590.58 GJ, corresponding to increases of 9.84%, 14.99%, 17.96%, and 19.82%, respectively. Over the entire operational period, the accumulated stored heat under the benchmark condition increases by 3.62%. In contrast, when the heat extraction flow rate is raised to 15, 20, 25, and 30 m³/h, the accumulated stored heat increases by 8.14%, 8.63%, 9.49%, and 10.02%,

respectively. This occurs because a higher heat extraction flow rate leads to a greater soil heat deficit, and over time, the increasing soil heat deficit allows for more heat to be stored. However, as the flow rate increases, the rate of increase in accumulated stored heat gradually decreases. For instance, during the first non-heating season, for every 5 m³/h increase in heat extraction flow rate, the accumulated stored heat increases by 9.84%, 4.69%, 2.58%, and 1.58%, respectively. In conclusion, increasing the heat extraction flow rate significantly enhances the soil heat deficit, thereby improving MDBHE's heat storage capacity. However, as the heat extraction flow rate increases, the rate of increase in accumulated stored heat gradually diminishes.

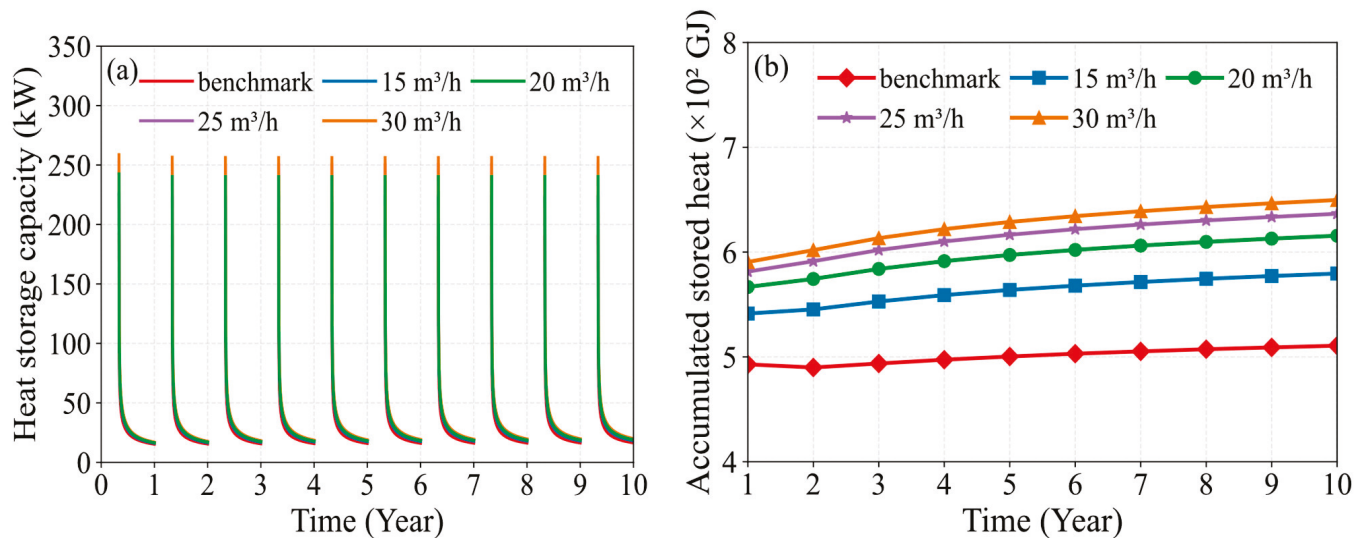


Figure 9. Variation in (a) heat storage capacity and (b) accumulated stored heat of the MDBHE under flow rate-based soil heat deficit regulation.

Figure 10 illustrates the temporal variation in heat storage efficiency under flow rate-based soil heat deficit regulation. It is evident that heat storage efficiency increases gradually over the operational period. Moreover, higher heat extraction flow rates lead to enhanced heat storage efficiency for MDBHE. For example, under the benchmark condition, heat storage efficiency increases from 8.24% to 14.00% over the entire operational period. When the heat extraction flow rate is increased from the baseline of 10 m³/h to 15, 20, 25, and 30 m³/h, the heat storage efficiency in the first year shifts from 8.24% to 8.13%, 8.52%, 8.92%, and 9.23%, while in the last year it increases from 14.00% to 14.56%, 15.41%, 16.10%, and 16.62%, respectively. Over the entire operational period, flow rate-based soil heat deficit regulation results in a maximum improvement of 18.71% (from 14.00% to 16.62%) compared to the benchmark condition. This improvement is attributed to the fact that increasing the heat extraction flow rate focuses the soil heat deficit in deeper soil layers. The lower temperatures in these deeper soils create favorable conditions for the circulating fluid to store more heat, progressively enhancing the heat storage efficiency of the MDBHE. In summary, flow rate-based soil heat deficit regulation significantly expands the soil heat deficit, leading to enhanced geothermal heat flux recovery and heat storage, and consequently improving both heat extraction capacity and heat storage efficiency of MDBHE.

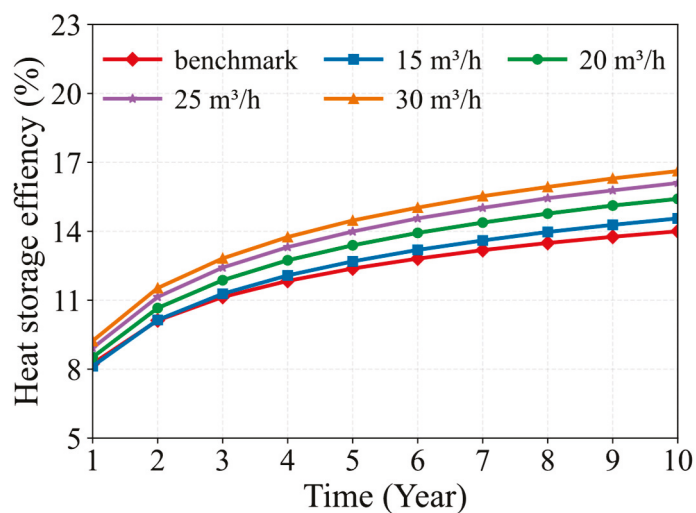


Figure 10. Variation in heat storage efficiency of the MDBHE under flow rate-based soil heat deficit regulation.

4. Discussion

4.1. Robustness of Soil Heat Deficit Regulation

The previous section discussed the effectiveness of the proposed cross-seasonal heat storage regulation under both temperature-based and flow rate-based soil heat deficit conditions. Building on this, the current section examines the robustness of soil heat deficit regulation under different heat storage conditions during the non-heating season. Specifically, while the heat storage temperature in the previous section was set to 50 °C, in this section, the heat storage temperature is adjusted to 50 °C, 60 °C, and 70 °C, all based on the maximum soil heat deficit. The response of heat extraction capacity and heat storage efficiency of MDBHE is then evaluated over multiple years to assess the robustness of the proposed soil heat deficit regulation.

As shown in Figure 11, compared to the benchmark condition, the heat extraction capacity of MDBHE increases significantly with the expansion of the heat deficit, and the higher the heat storage temperature, the greater the heat extraction capacity. However, under different heat storage temperatures, the difference in heat extraction capacity at the beginning of the heating season is more pronounced, while at the end of the heating season, the heat extraction capacity becomes almost identical. For example, when the heat storage temperatures are 50 °C, 60 °C, and 70 °C, compared to the benchmark condition, the heat extraction capacity at the beginning of the first heating season after cross-seasonal heat storage increases by 379.83 kW, 466.8 kW, and 553.85 kW, corresponding to increases of 200.92%, 246.95%, and 292.97%, respectively. However, by the end of the heating season, the increase in heat extraction capacity drops to 95.85 kW, 100.38 kW, and 105.69 kW, with increases of 108.19%, 113.31%, and 118.89%, respectively, and the difference in heat extraction capacity decreases significantly. This is because higher heat storage temperatures raise the surrounding soil temperature (as shown in Figure 12), resulting in a larger difference in heat extraction capacity at the beginning of the heating season. However, as the heating season progresses, the surrounding soil temperature under different heat storage conditions gradually becomes more uniform, narrowing the gap in heat extraction capacity by the end of the heating season.

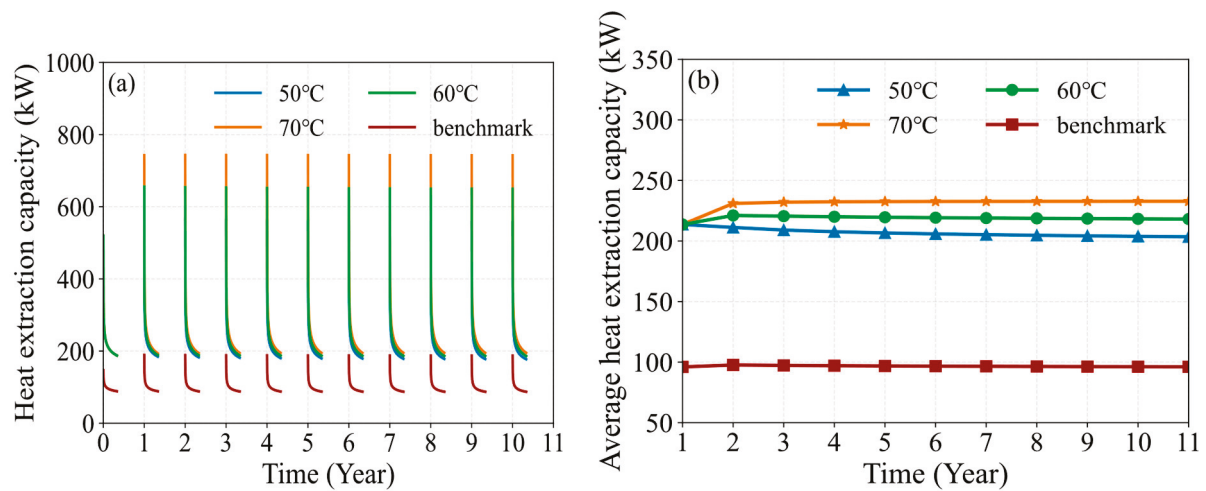


Figure 11. Variation in heat extraction capacity under different heat storage conditions: (a) hourly variation; (b) seasonal average value.

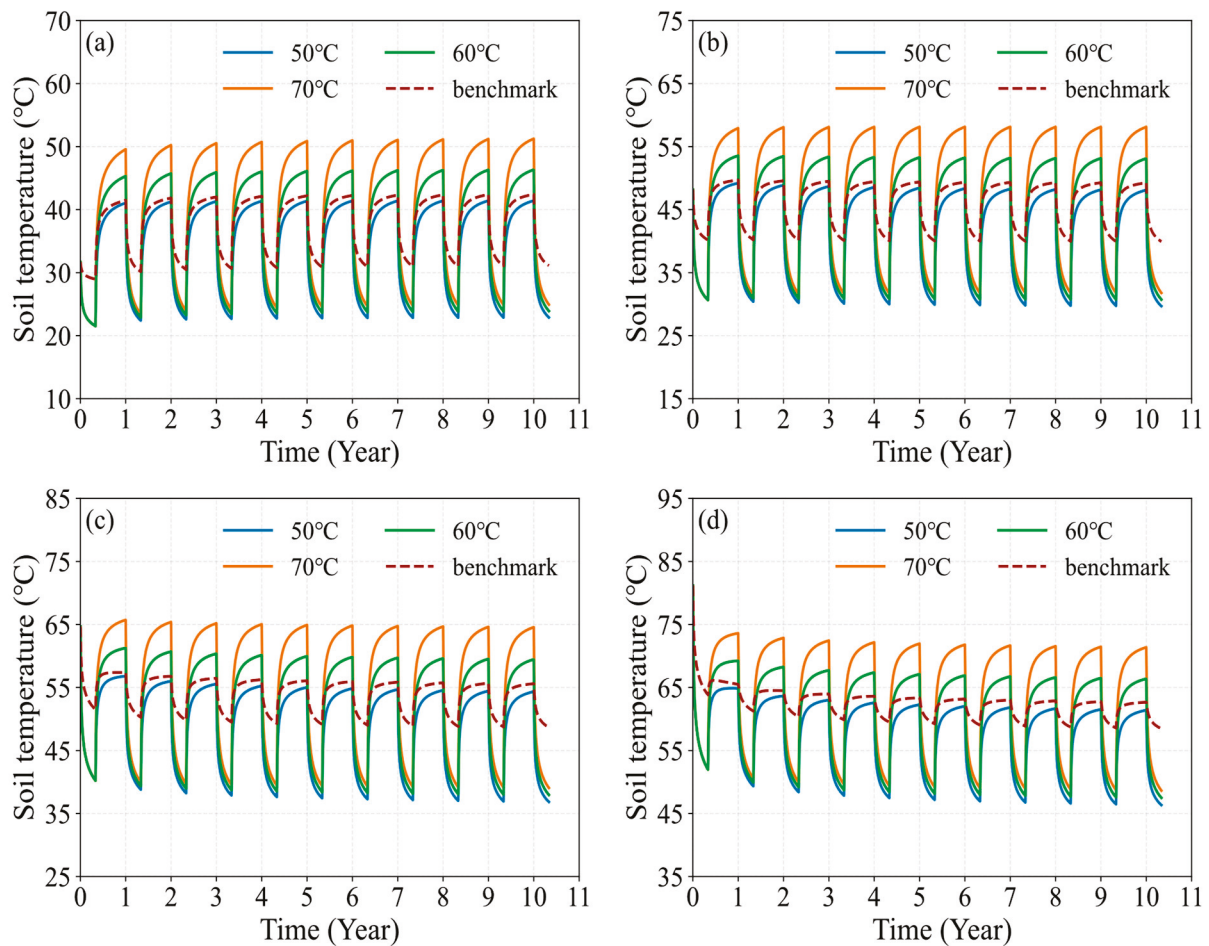


Figure 12. Variation in soil temperature under different heat storage conditions: (a) 500 m depth; (b) 1000 m depth; (c) 1500 m depth; (d) 2000 m depth.

Higher heat storage temperatures and an expanded soil heat deficit result in a stronger seasonal average heat extraction capacity for MDBHE, with a smaller decrease in seasonal average heat extraction capacity over time. In some cases, the seasonal average heat extraction capacity even increases with higher heat storage temperatures and a greater

soil heat deficit. Figure 11b illustrates the variation in MDBHE's seasonal average heat extraction capacity under different heat storage conditions. When the soil heat deficit is maximized, the seasonal average heat extraction capacity is significantly higher at heat storage temperatures of 50 °C, 60 °C, and 70 °C compared to the benchmark condition. Specifically, at heat storage temperatures of 50 °C, 60 °C, and 70 °C, with an expanded soil heat deficit, the seasonal average heat extraction capacity increases by 122.50%, 126.88%, and 142.01%, respectively, compared to the benchmark condition. Under the benchmark condition, MDBHE's seasonal average heat extraction capacity increases slightly from 96.03 kW to 96.14 kW, a 0.11% increase. However, when the heat storage temperature is 50 °C with an expanded soil heat deficit, the seasonal average heat extraction capacity decreases from 213.67 kW to 203.48 kW, a reduction of 4.77%. At a heat storage temperature of 60 °C with expanded soil heat deficit, the seasonal average heat extraction capacity increases from 213.67 kW to 218.12 kW, representing an overall increase of 2.08%. At a heat storage temperature of 70 °C with expanded soil heat deficit, the seasonal average heat extraction capacity increases from 213.67 kW to 232.76 kW, a total increase of 8.93%. This improvement is due to the heat storage during the non-heating season, raising the surrounding soil temperature, thereby increasing the temperature difference between the circulating fluid and the surrounding soil at the beginning of the heating season. The increased temperature difference enhances MDBHE's seasonal average heat extraction capacity. In conclusion, increasing the soil heat deficit and adjusting the heat storage temperature can significantly improve MDBHE's heat extraction capacity.

The heat storage temperature and soil heat deficit directly influence the heat storage capacity of the MDBHE. As shown in Figure 13a, higher heat storage temperatures and increased soil heat deficit significantly improve the heat storage capacity of MDBHE, particularly at the beginning of the non-heating season. For example, at the start of the first non-heating season, compared to the benchmark condition, the heat extraction capacity increases by 161.94 kW (84.68%), 277.31 kW (145.01%), and 392.65 kW (205.32%) when the heat storage temperature is set to 50 °C, 60 °C, and 70 °C, respectively. This improvement is attributed to the higher heat storage temperatures and increased soil heat deficit, which amplify the temperature difference between the circulating fluid and the surrounding soil, thereby boosting the heat transfer driving force and enhancing the heat storage capacity. Furthermore, the accumulated stored heat of MDBHE during the non-heating season also increases significantly, as shown in Figure 13b. For instance, during the first non-heating season, compared to the benchmark condition, the accumulated stored heat increases by 222.63 GJ (45.17%), 1158.83 GJ (235.19%), and 2094.96 GJ (425.05%) when the heat storage temperature is set to 50 °C, 60 °C, and 70 °C, respectively. In contrast, under the benchmark condition, the accumulated stored heat increases slightly over time, with a total increase of 3.62% over the entire operational period. However, under the expanded soil heat deficit, the cumulative heat storage changes by 16.43%, 0.21%, and −4.83% over the entire operational period when the heat storage temperatures are 50 °C, 60 °C, and 70 °C, respectively. This suggests that while increasing the heat storage temperature in combination with an expanded soil heat deficit significantly boosts the heat storage capacity and accumulated stored heat of MDBHE, higher heat storage temperatures lead to a slight decrease in accumulated stored heat.

Figure 14 illustrates the variation in heat storage efficiency of MDBHE under different heat storage temperatures and soil heat deficits during cross-seasonal heat storage. Compared to the benchmark condition, increasing both the heat storage temperature and soil heat deficit significantly enhances the heat storage efficiency of MDBHE. For example, in the first year, the heat storage efficiency under the benchmark condition is 8.24%, while

under increased soil heat deficit, the efficiency at heat storage temperatures of 50 °C, 60 °C, and 70 °C increases to 9.00%, 10.11%, and 10.42%, respectively. Over the entire operational period, increasing both the soil heat deficit and heat storage temperature results in a maximum improvement of 27.79% (from 14.00% to 17.89%) in heat storage efficiency compared to the benchmark condition. However, when the heat storage temperature increases from 60 °C to 70 °C, the improvement in heat storage efficiency is much smaller compared to the increase from 50 °C to 60 °C. For instance, in the first year, when the heat storage temperature increases from 50 °C to 60 °C, the heat storage efficiency increases from 9.00% to 10.11%, a 12.33% increase. In contrast, when the heat storage temperature increases from 60 °C to 70 °C, the efficiency only rises from 10.11% to 10.42%, a 3.07% increase. In conclusion, increasing both the soil heat deficit and heat storage temperature significantly improves the MDBHE's heat extraction capacity and heat storage efficiency, confirming the robustness of the proposed soil heat deficit regulation. However, when the heat storage temperature exceeds a certain level (e.g., above 60 °C), the efficiency gains from further increases in heat storage temperature become less pronounced.

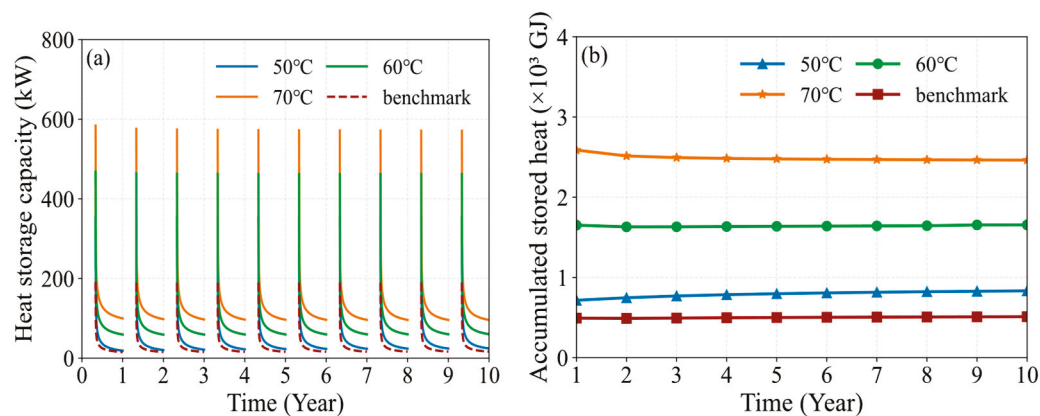


Figure 13. Variation in (a) heat storage capacity and (b) accumulated stored heat of the MDBHE under different heat storage conditions.

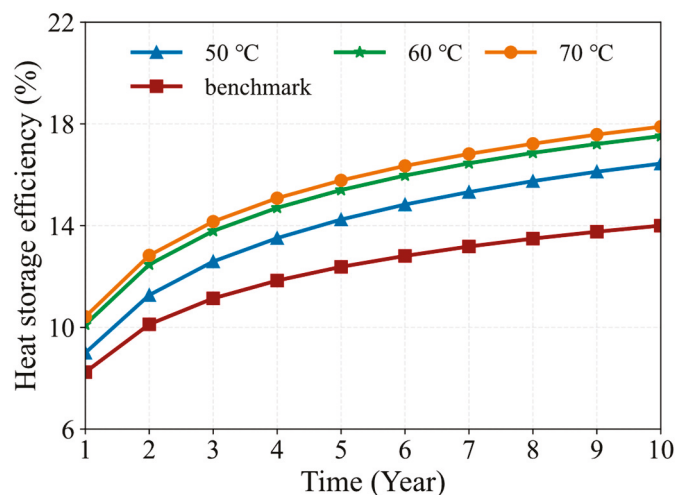


Figure 14. Variation in heat storage efficiency of the MDBHE under different heat storage conditions.

4.2. Limitations and Outlook

This study analyzes the heat extraction capacity and heat storage efficiency of the MDBHE under temperature-based and flow-rate-based soil heat deficit regulation. The results indicate that increasing the soil heat deficit can significantly enhance the heat extrac-

tion capacity of the MDBHE. Furthermore, the robustness of the soil heat deficit regulation is demonstrated across different heat storage conditions. However, it is important to note that while increasing the heat extraction flow rate and raising the heat storage temperature lead to improved performance, they also result in higher operational costs for the MDBHE. As such, careful consideration of economic factors is essential during the design phase. It is worth mentioning that this study focuses on verifying the effectiveness of temperature-based and flow-rate-based soil heat deficit regulation, without providing the optimal combination of heat extraction temperature, heat extraction flow rate, heat storage temperature, and heat storage flow rate. Future research could explore the use of advanced machine learning techniques, such as deep learning models [35], to optimize these parameter combinations, ultimately improving both the heating performance and heat storage efficiency of the MDBHE. Additionally, there is potential to improve heat transfer efficiency to enhance the overall heating performance of MDBHE. Future work will explore enhanced heat transfer technologies, such as enhanced surfaces like spiral tube walls [36], which can boost heat exchange efficiency by increasing surface area and optimizing fluid dynamics.

5. Conclusions

This study proposes a novel cross-seasonal heat storage method based on soil heat deficit regulation for the MDBHE, which expands the soil heat deficit to enhance the geothermal heat flux recovery and heat storage simultaneously, while also improving indoor thermal comfort. The effectiveness of the proposed cross-seasonal heat storage regulation method is analyzed under temperature-based and flow rate-based soil heat deficit regulations. The robustness of the soil heat deficit regulation is demonstrated across different heat storage conditions. The main findings of the study are summarized as follows:

- (1) The effectiveness of the proposed cross-seasonal thermal storage regulation method is analyzed under temperature-based and flow rate-based soil heat deficit regulations. Under the temperature-based soil heat deficit regulation, the soil heat deficit was mainly concentrated in the shallow soil layers, while the flow rate-based soil heat deficit regulation was mainly concentrated in the deep soil layers.
- (2) The flow rate-based soil heat deficit regulation is more effective than the temperature-based soil heat deficit regulation. Expanding the soil heat deficit by lowering the heat extraction temperature increases the heat extraction capacity of MDBHE by up to 44.82% (from 93.03 kW to 139.07 kW), with relatively stable heat storage efficiency. In contrast, expanding the soil heat deficit by increasing the heat extraction flow rate enhances the heat extraction capacity by up to 55.93% (from 93.03 kW to 149.74 kW) and improves heat storage efficiency by up to 18.71%.
- (3) The robustness of the soil heat deficit regulation is verified under various heat storage conditions. Increasing both the heat storage temperature and the soil heat deficit significantly enhances the heat extraction capacity and heat storage efficiency. However, once the heat storage temperature reaches a certain level, further increases in heat storage temperature result in diminishing returns in heat storage efficiency.

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Data Availability Statement: The datasets generated and analyzed during the current study are available from the corresponding author upon reasonable request.

Conflicts of Interest: Authors Jun Liu and Yuping Zhang were employed by the company Shaanxi Coal Geology Group Company Limited. Author Jingyue Zhang was employed by the company Shaanxi Coal Geology Investigation Research Institute Co., Ltd. Author Heng Shi was employed by the company Qingdao Haier Air Conditioner Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Abbreviations

The following abbreviations are used in this manuscript:

MDBHE	Medium-deep borehole heat exchanger
COP	Coefficient of performance
OGS	OpenGeoSys
SHD	soil heat deficit
Q_{be}	Amount of heat contained in the soil before heat extraction
Q_{ae}	Amount of heat contained in the soil after heat extraction
Q_{ghf}	Geothermal heat flux recovery
Q_{hs}	Heat storage
ρ_f	Density of the circulating fluid
c_f	Specific heat capacity of the circulating fluid
v_i	Inner pipe flow velocities
v_o	Outer pipe flow velocities
Λ_f	Hydrodynamic thermos-dispersion tensor
H	Heat source/sink term
Γ	Heat transfer boundary
Φ	Heat transfer coefficients
T_r	Soil temperature
ϕ	Effective porosity
$m_{storage}$	Mass flow rates of the circulating fluid during the non-heating season
$m_{extraction}$	Mass flow rates of the circulating fluid during the heating season
t_{in}	Inlet temperatures
t_{out}	Outlet temperatures
SS	Start time of the non-heating season
SE	End time of the non-heating season
ES	Start time of the heating season
EE	End time of the heating season
$q_{extraction}$	Heat extraction capacity
$q_{storage}$	Heat storage capacity
$Q_{storage}$	Accumulated extracted heat
$Q_{extraction}$	Accumulated stored heat
$\bar{q}_{extraction}$	Seasonal average heat extraction capacity
η	Heat storage efficiency

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Article

Field Study of Relationships Between Indoor Thermal Conditions and Two Major Causes of Allergies—Dust Mites and Mould—In New Zealand Houses

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Abstract: Based on field studies, this study contributes the new physical data of winter indoor thermal conditions of the indoor spaces with four different dust mite allergen levels in New Zealand houses. This study provides a new method to identify the relationships between indoor thermal conditions and indoor dust mite allergen levels. This study found that the indoor mean relative humidity (RH) close to the floor must be controlled below 70%, and there must be less than 30% of time in winter when indoor RH close to the floor is higher than or equal to 75% to maintain indoor dust mite allergens at an undetectable level; and the indoor mean RH close to the floor must be controlled below 75%, and there must be less than 50% of time in winter when indoor RH close to the floor is higher than or equal to 75% to maintain indoor dust mite allergens at a low (acceptable) level. This study also identified the relationship between indoor thermal conditions for dust mites to thrive and for mould spores to germinate. This study provides a strategy or guideline for preventing indoor allergies related to dust mites and mould under the temperate climate zone; the winter is mild and humid.

Keywords: dust mites; New Zealand houses; indoor allergens; indoor thermal conditions; mould

1. Introduction

Auckland, New Zealand, has a temperate climate with warm, dry summers and mild, wet winters [1]. Most of the factors that contribute to poor health, for example, bacteria, viruses, fungi, mites, etc., have increases associated with high indoor RH [2]. Indoor dust mite allergens in New Zealand houses are deemed to be the world's highest levels [3]. Visible mould on indoor surfaces is common in local homes [4]. In New Zealand houses, the main indoor allergens from mould and dust mites are related to the high indoor RH. For optimum indoor air quality, RH should be below 60% [5–7]. Indirect adverse health effects can be minimised by maintaining indoor RH between 40% and 60% [2].

Most Auckland houses with internal insulation and external cladding are designed for temporary heating, and their structures are light-frame construction. There are only a small percentage of local homes that have a permanent heating system [8]. For an Auckland house with recommended insulation in its envelope and a permanent heating system, the minimum indoor temperature must be higher than 20 °C to control indoor mean RH

below 50% for minimising indirect adverse health effects [2,9]. Since 2007, double-glazed windows and the higher insulation (R2.9 for roofs, R1.9 for walls, R1.3 for floors, and R0.26 for glazing) for the local houses have been required by the building codes [10,11]. According to the thermal insulation history of New Zealand houses, there is a large percentage of Auckland houses that do not meet these requirements of the current building codes [10,11]. For those houses without the recommended insulation and double-glazed windows, it is difficult and energy inefficient, and also very expensive for the occupants to maintain the minimum indoor temperature above 20 °C for controlling indoor mean RH below 50%. Based on the field study, this study identified the relationship between indoor thermal conditions and the two major indoor allergens in New Zealand houses—dust mites and mould. This study focuses on the minimum indoor thermal conditions required to maintain the major indoor allergens at acceptable levels.

Previous studies in other countries found dust mites cannot survive when indoor RH is lower than 50% [12,13]. The presence of dust mites and their allergens can be reduced by maintaining indoor RH below 50%, and when indoor RH is maintained within a range of 40–50%, dust mites can be minimised [14–17]. Indoor RH within 60% to 80% are optimal conditions for dust mites' reproduction. Dust mites need indoor RH within 75% to 80% or higher indoor RH to thrive. They prefer temperatures in the range of 18–25 °C [2,14,15,18–20]. Both high ambient temperature and RH could occur during the hot and humid summer of a temperate climate or a tropical climate. The local climate for this study has a high ambient RH and low temperature during the winter and a comfortably warm, dry summer.

The previous study of lab mite cultures showed that the environmental RH for dust mite population at the peak was at about 85% at 28 °C [21]. Except for indoor RH and temperature, indoor ventilation can also impact the dust mite allergen; high indoor ventilation potentials or high indoor air change rates can reduce indoor dust mite allergen levels [22,23]; indoor dust mite allergen levels can also be impacted by occupants' numbers and daily activities [24,25]. Based on the field study data, this study identified four different indoor dust mite allergen levels and their associated indoor thermal conditions. The four different dust mite allergen levels are as follows: undetectable and low (acceptable indoor dust mite allergen levels) and medium and high (unacceptable indoor dust mite allergen levels).

Previous studies in New Zealand have shown that the local homes had generally much higher levels of house dust mite (HDM) allergens than nurseries, kindergartens, and elementary schools [26–28]. Home environments had significantly higher dust mite allergens than indoor spaces of public buildings and were primarily affected by floor coverings [29]. The recent study in another country also showed that dust mite populations in home environments were higher than hotel and school indoor environments [30]. Those studies did not provide indoor thermal conditions. There are limited data and studies on measurements of indoor thermal environments in New Zealand houses; in particular, information on the relationships between indoor thermal environments, indoor air quality and health is sorely limited regarding New Zealand buildings [31,32]. The local previous studies showed that occupants' poor health conditions were closely related to the poor living conditions of the houses with serious dust mite allergy problems [33,34]. This study focused on New Zealand houses and identified indoor thermal conditions related to different indoor dust mite allergen levels. This field study not only tested indoor thermal conditions in sample houses with dust mite allergen problems (unacceptable dust mite allergen levels) but also tested indoor thermal conditions in sample houses without dust mite allergen problems (acceptable dust mite allergen levels). Those thermal conditions of indoor spaces with acceptable dust mite allergen levels can be used as the threshold for keeping indoor allergens at acceptable levels. If the indoor thermal conditions of New

Zealand houses can be maintained below the threshold for keeping indoor allergens at acceptable levels, there will be no indoor dust mite allergen problems.

Previous studies showed that mould cannot survive when indoor RH is lower than 60%. Mould can grow on most building materials when indoor RH exceeds 75–80% [35–37]. For mould spores to germinate, they require both higher indoor RH than mould survival does and the required time to germinate (Table 1) [38]. If indoor thermal conditions can be controlled to prevent mould spores' germination, the mould hardly grows on the indoor surfaces [39,40]. The recent study of mould growths on different building materials in the laboratory environment showed that there is no mould growth on the materials' surfaces when the RH was lower than 75% and the temperature was at 25 °C [41]. Those studies found that there are correlations between surface moisture content of the building materials and mould growth under different RH and temperature [42,43]. The other recent studies found that the short wavelength light can limit the formation and production of mould spores [44]; the optimum indoor ventilation condition in a building with an HVAC system can reduce mould growth [45]. This study focused on the thresholds of indoor thermal conditions for mould germination—not its survival or growth conditions—for preventing mould growth on indoor surfaces. This study also identified the relationships between indoor dust mites and mould under the local climate.

Table 1. Required RH and time for mould spores to germinate [38].

Surfaces	Indoor RH	Time Required (d)
The majority of building internal surface materials	=100%	1
	≥89%	7
	≥80%	30

The healthy minimum indoor temperature recommended by WHO is 18 °C for houses and 20–21 °C for older people and young children's living environments during the winter [46]. There are 1150 heating degree days for the Auckland climate (the base temperature: 18 °C) [47]. Previous studies showed that for minimising infections of the respiratory system during the winter, the required minimum indoor temperature is 16 °C [48,49]. Previous studies have revealed that winter indoor temperatures of local houses are quite lower than the health minimum indoor temperature recommended by WHO [50,51]. Based on data gathered from about 400 sample houses throughout New Zealand, mean winter living room and bedroom temperatures are below 18 °C. Mean living room and bedroom temperatures are 12.6–13.5 °C during the morning, 14.2–15.8 °C during the day, 15.0–17.8 °C during the evening, and 13.6–14.8 °C during the night [11]. As indoor temperature decreases, this is associated with an increase in indoor RH; very low indoor temperature can result in very high indoor RH. As occupants only use temporary heating in their living rooms and bedrooms when they are occupied during the winter or as needed, it is difficult to maintain winter indoor temperature above the healthy minimum indoor temperature recommended by WHO for 24 h a day in winter. This study tried to identify the minimum requirement of indoor temperature for maintaining the two major indoor allergens from dust mites and mould at the acceptable levels in New Zealand housing.

2. Materials and Methods

According to the instructions on how to use the Ventia Rapid Allergen Test, dust samples in the carpets of indoor spaces were collected by a vacuum cleaner with a DUSTREAM® collector (InBio, Charlottesville, VA, USA). In a bedroom, the area of carpet sampled was

beside the bed. In a living room, the areas of carpet sampled were in front of the sofa or in the centre of a living room. The total testing area in a bedroom or a living room is about 1 m². The continued vacuuming time for the sampling area of each room is 2 min. The Ventia Rapid Test cassettes were used for testing dust samples. The four different dust mite allergen levels—(1) undetectable; (2) low (less than 0.2 micrograms of dust mite allergens per gram of dust); (3) medium (0.2–1.0 micrograms of dust mite allergens per gram of dust); (4) high (≥ 1 microgram of dust mite allergens per gram of dust)—can be identified. According to the instructions of the Ventia Rapid Test cassettes, for Levels 1 and 2, no action is needed to reduce indoor levels of dust mite allergens (acceptable levels); for Levels 3 and 4, action should be taken to reduce indoor levels of dust mite allergens to protect occupants' health (unacceptable levels). Undetectable and low dust mite allergens are considered acceptable; medium and high dust mite allergens are considered unacceptable for this study.

In accordance with the instructions of consulting industrial microbiologists Biodet Services Ltd. (Auckland, New Zealand), mould samples were collected by the researchers, using clear, standard Sellotape, from the indoor surface areas with visible mould in the indoor spaces. The mould samples on the Sellotape were then wrapped in the baking paper and put in the plastic bags; the samples were examined both macroscopically and microscopically by a local testing laboratory.

HOBO UX100 Temp/RH Data Loggers (Onset, Bourne, MA, USA) were used for the field studies of indoor thermal conditions (tested indoor thermal environment: -20 to 70 °C; for the temperature range of 0 – 50 °C, the accuracy is ± 0.21 °C; for RH, the range is 15 – 95% , the accuracy is $\pm 3.5\%$; the maximum data memory is 84,650; and the testing intervals can be set from 1 s to 18 h). Air temperatures and RH close to ceilings and floors in the indoor spaces (living rooms or bedrooms) and in the outdoor spaces under the roof eaves on the south sides of sample houses were continuously tested during the winter (15 min intervals and 24 h per day). The indoor measuring points adjacent to the ceilings and the floors were generally located at the room centres. Indoor measuring points adjacent to the floors of the north living room or bedroom could be located close to the south internal walls for avoiding direct sunlight. The outdoor measuring points under the roof eaves were on the south sides of houses, which can minimise the impact of solar gain during the daytime. There are about 635,904 raw data of indoor and outdoor temperatures and RH from the field studies of the 16 sample houses. Based on the field study data, winter indoor mean temperatures and RH, and winter indoor mean temperatures and RH close to the ceilings and the floors of the indoor spaces with different dust mite allergen levels in the new and old sample houses were calculated. Average outdoor temperature and RH for the new sample houses or the old sample were calculated and used for the outdoor reference data. All field study data of indoor and outdoor air temperatures (mean, close to ceiling and close to floor) have been converted into percentages of time in winter when air temperatures are greater than or equal to 16 °C, 17 °C, 18 °C, 20 °C, and 22 °C; all field study data of indoor and outdoor RH have been converted into percentages of time in winter when RH is greater than or equal to 40% , 50% , 60% , 70% , 75% , 80% , and 90% for the purposes of comparing indoor thermal conditions of indoor spaces with different dust mite allergen levels and mould growths in the new and old sample houses.

Indoor thermal environmental data of 17 indoor spaces (living rooms or bedrooms) in 11 old sample houses (floor areas from 51 m² to 106 m²) with unacceptable dust mite allergen levels (medium and high levels) and mould growth present on internal surfaces in the remote sites of Minginui, Te Whaiti, and Murupara areas were used for the study. Dust mite allergen tested results in all the 7 indoor spaces of the five old sample houses were high levels; dust mite allergen tested results in all the 10 indoor spaces of the six old

sample houses were medium levels. Indoor thermal environmental data in 15 indoor spaces (living rooms or bedrooms) of five new houses with acceptable dust mite allergen levels (undetectable and low levels) in Auckland city were used for the study. Dust mite allergen tested results in the 9 indoor spaces of the five new sample houses were undetectable levels; dust mite allergen tested results in the 6 indoor spaces of the four new sample houses were low levels. Dust mite tested results in all the indoor spaces of one new sample house were undetectable levels only; dust mite tested results in the indoor spaces of the other four new sample houses were both undetectable and low levels. Figures 1 and 2 illustrate Auckland and Rotorua climate data. The remote field study sites in Minginui are near Rotorua.

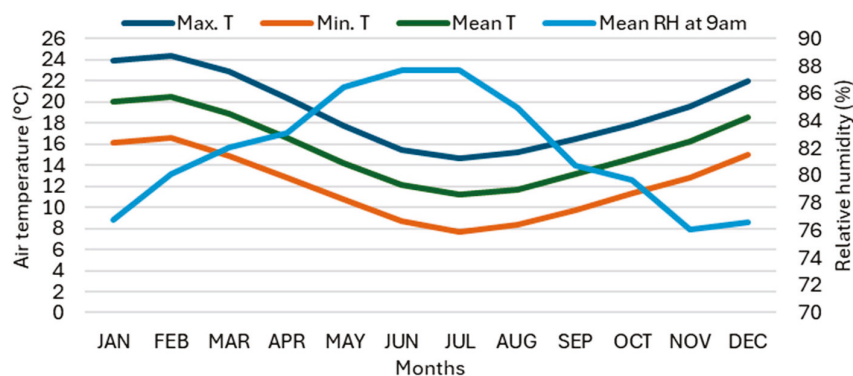


Figure 1. Climate data in Auckland for the 1991–2020 period (Source: National Institute of Water and Atmospheric Research).

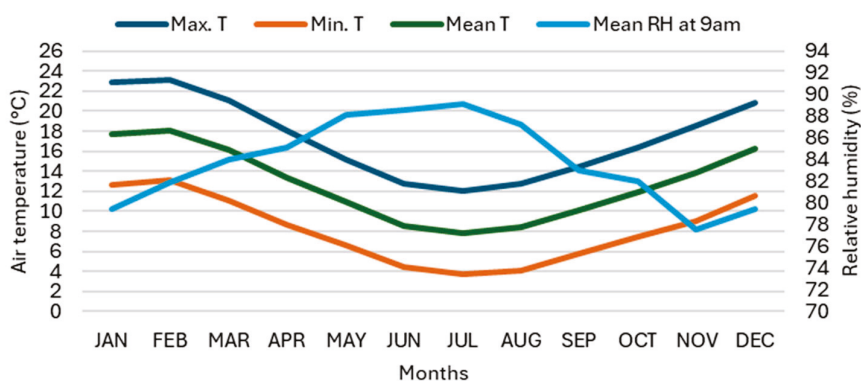


Figure 2. Climate data in Rotorua for the 1991–2020 period (Source: National Institute of Water and Atmospheric Research).

The 11 old sample houses built from the 1920s to the 1970s have light timber structures and single-glazed windows. Roofing materials are old asbestos cement sheets (one house), old, corrugated iron (four houses) and old roof tin shingles (six houses). Wall materials are old brick (one house) and old weatherboard (ten houses). The 11 old sample houses do not have sufficient insulation (eight houses without any insulation and three houses with limited insulation). One house used an electric heater and a fireplace for space heating. Two sample houses used stoves for both cooking and space heating. Eight houses used fireplaces for space heating.

The five new sample houses (floor areas from 210 m² to 256 m²) in Auckland, with a light timber structure, were built in the 2000s (2000–2018). Roofing materials are tile (four houses) and felt (one house). Wall materials are brick veneer and weatherboard (three houses) and polystyrene insulation board (two houses). Three houses have basic insulation (R1.9 for roofs, R1.5 for walls, and R1.3 for floors) [52] and single-glazed windows. Two

houses have sufficient insulation (R2.9 for roofs, R1.9 for walls, R1.3 for floors, and R0.26 for glazing) [8,9] and double-glazed windows. For space heating, three sample houses used portable electric heaters, and two sample houses temporarily used heat pumps. The field studies of measuring indoor thermal conditions and testing for dust mite allergens in the old sample houses and new sample houses were carried out in the winter months (June, July, and August) in 2018 [34] and 2022, respectively, by the authors.

3. Results and Discussion

3.1. Mean Indoor RH and Different Dust Mite Allergen Levels

Tables 2–5 list percentages of time in winter at different ranges of RH and mean indoor RH in the indoor spaces with different dust mite allergen levels. Although there was 99% and 86%, respectively, of time in winter when indoor RH was higher than or equal to 50% and 60%, dust mite survival conditions and ideal growth conditions, respectively [12–17], there were no allergens of dust mites being detected in carpets of the indoor spaces (Table 2). The mean RH of indoor spaces with undetectable dust mite allergens was 67%, with a range of 64–69% (Table 2). The mean RH of indoor spaces with low dust mite allergens was 72%, with a range of 70–75% (Table 3). For maintaining dust mite allergens at acceptable levels, the mean indoor RH does not have to be controlled below dust mite survival conditions ($\geq 50\%$) or ideal growth conditions ($\geq 60\%$). There were clear differences of mean indoor RH between the indoor spaces with acceptable dust mite allergen levels and the indoor spaces with unacceptable dust mite allergen levels (Tables 2–5).

Table 2. RH of the 9 indoor spaces with undetectable dust mite allergens in the five new sample houses in Auckland city.

House Numbers	No. 1	No. 1	No. 2	No. 2	No. 2	No. 3	No. 3	No. 4	No. 5	
Indoor Spaces	N Living	N Bed.	N Bed.	N 2nd Bed.	S Bed.	N Living	S Bed.	N Bed.	N Bed.	Outdoor
Mean RH (%)	69	64	69	69	67	66	65	68	65	84
% Time when RH $\geq 40\%$	100%	100%	100%	100%	100%	100%	100%	100%	100%	100%
% Time when RH $\geq 50\%$	100%	100%	99%	98%	99%	96%	99%	100%	100%	100%
% Time when RH $\geq 60\%$	98%	83%	90%	86%	85%	79%	77%	88%	87%	98%
% Time when RH $\geq 70\%$	39%	8%	50%	54%	35%	35%	25%	34%	17%	89%
% Time when RH $\geq 75\%$	6%	0%	25%	24%	11%	11%	7%	13%	3%	81%
% Time when RH $\geq 80\%$	0%	0%	6%	4%	2%	1%	1%	4%	0%	69%

Table 3. RH of the 6 indoor spaces with low dust mite allergens in the four new sample houses in Auckland city.

House Numbers	No. 1	No. 3	No. 4	No. 4	No. 5	No. 5	
Indoor Spaces	S Bed.	N Bed.	N Living	S Bed.	N Living	S Bed.	Outdoor
Mean RH (%)	72	71	70	71	72	75	84
% Time when RH $\geq 40\%$	100%	100%	100%	100%	100%	100%	100%
% Time when RH $\geq 50\%$	100%	100%	98%	100%	100%	100%	100%
% Time when RH $\geq 60\%$	100%	94%	84%	91%	100%	100%	98%
% Time when RH $\geq 70\%$	77%	57%	53%	60%	66%	89%	89%
% Time when RH $\geq 75\%$	27%	23%	31%	33%	24%	48%	81%
% Time when RH $\geq 80\%$	0%	6%	9%	8%	3%	14%	69%

Table 4. RH of the 10 indoor spaces with medium dust mite allergens in the six old sample houses in the remote sites.

House Numbers	No. 1	No. 2	No. 2	No. 3	No. 4	No. 4	No. 5	No. 5	No. 6	No. 6	
Indoor Spaces	N Living	N Living	S Bed.	N Bed.	S Living	N Bed.	N Living	S Bed.	N Living	S Bed.	Outdoor
Mean RH (%)	83	76	75	81	88	76	74	77	85	92	85
% Time when RH $\geq$ 40%	100%	100%	98%	100%	100%	100%	99%	100%	100%	100%	100%
% Time when RH $\geq$ 50%	100%	98%	96%	100%	99%	99%	96%	99%	100%	100%	100%
% Time when RH $\geq$ 60%	99%	94%	93%	98%	99%	95%	90%	97%	100%	100%	98%
% Time when RH $\geq$ 70%	98%	78%	77%	92%	99%	71%	73%	77%	100%	100%	90%
% Time when RH $\geq$ 75%	95%	62%	59%	84%	97%	55%	57%	57%	99%	100%	82%
% Time when RH $\geq$ 80%	81%	37%	33%	67%	90%	42%	32%	41%	85%	99%	70%
% Time when RH $\geq$ 85%	40%	14%	6%	36%	69%	24%	7%	18%	55%	98%	55%
% Time when RH $\geq$ 90%	2%	0%	0%	3%	37%	3%	0%	1%	13%	83%	35%

Table 5. RH of the 7 indoor spaces with high dust mite allergens in the five old sample houses in the remote sites.

House No.	No. 7	No. 7	No. 8	No. 9	No. 9	No. 10	No. 11	
Indoor Spaces	N Living	S Bed.	S Bed	N Living	S Bed.	S Bed.	S Bed.	Outdoor
Mean RH (%)	77	85	80	80	77	76	75	85
% Time when RH $\geq$ 40%	98%	100%	99%	99%	99%	100%	100%	100%
% Time when RH $\geq$ 50%	95%	100%	99%	99%	99%	98%	98%	100%
% Time when RH $\geq$ 60%	90%	99%	99%	97%	98%	95%	94%	98%
% Time when RH $\geq$ 70%	78%	98%	95%	90%	89%	86%	76%	90%
% Time when RH $\geq$ 75%	64%	92%	79%	80%	67%	68%	55%	82%
% Time when RH $\geq$ 80%	45%	79%	46%	57%	33%	34%	30%	70%
% Time when RH $\geq$ 85%	28%	54%	16%	21%	6%	2%	8%	55%
% Time when RH $\geq$ 90%	9%	15%	1%	2%	1%	0%	0%	35%

There were not many differences in percentages of time in winter when the indoor RH was higher than or equal to 50% and 60% (the conditions for dust mites to survive and the ideal conditions for dust mites to grow, respectively) between the indoor spaces with acceptable dust mite allergen levels (Tables 2 and 3) and the indoor spaces with unacceptable dust mite allergen levels (Tables 4 and 5). There were clear differences in percentages of time in winter between the indoor spaces with acceptable dust mite allergen levels and the indoor spaces with unacceptable dust mite allergen levels when the indoor RH was equal to or higher than 70% (the ideal conditions for dust mites to grow). There were significant differences in percentages of time in winter between the indoor spaces with acceptable dust mite allergen levels and the indoor spaces with unacceptable dust mite allergen levels when the indoor RH was higher than or equal to 75–80% (the conditions for dust mites to thrive [2,14,15,18–20]).

Percentages of time in winter associated with different RH ranges, especially for the threshold for dust mites to thrive (≥ 75 –80%), and the mean indoor RH can be used to identify the different indoor thermal conditions that cause different indoor dust mite allergens. Previous studies have mainly focused on indoor RH related to the threshold for dust mite survival (50%) and maintaining indoor RH below the threshold for dust mite survival (50%) to reduce dust mite populations or eliminate dust mites [12–17]. The study mainly focused on indoor RH related to the conditions for dust mites to thrive (≥ 75 –80%) [2,14,15,18–20]. If indoor thermal environmental conditions can be maintained below the conditions for dust mites to thrive, indoor dust mite allergens can be maintained at the acceptable levels (i.e., low or undetectable).

3.2. Mean Indoor Air Temperatures and Different Dust Mite Allergens

Tables 6–9 list percentages of time related to different ranges of air temperatures and mean indoor air temperatures in the indoor spaces (living rooms or bedrooms) with different dust mite allergens. There were significant differences of mean indoor temperatures between the indoor spaces with acceptable dust mite allergen levels (undetectable and low) and the indoor spaces with unacceptable dust mite allergen levels (medium and high). In the indoor spaces with undetectable dust mite allergens, the mean indoor air temperatures were all higher than 17 °C; the mean indoor temperature was 19 °C with a range of 17–20 °C, which maintained mean indoor RH below 70%. In the indoor spaces with low dust mite allergens (Table 7), the mean indoor temperatures were higher than 15 °C; the mean indoor temperature was 16 °C with a range of 15–19 °C, which maintained mean indoor RH below 75%. Previous studies found that indoor temperatures above 16 °C can limit respiratory infections [48,49]. Sixteen °C can be considered as the minimum indoor temperature required to keep indoor dust mite allergens at the acceptable levels. If the mean indoor temperature in winter can be kept at the higher levels, such as 17–20 °C, this will ensure a better indoor thermal environment for occupants' thermal comfort and health and prevent dust mite allergen problems. It is shown in previous studies that winter indoor temperatures of New Zealand homes are significantly lower than the requirements of thermal comfort or health for occupants [50,51]. A study of about 400 sample houses with different building envelope designs and temporary space-heating methods throughout New Zealand found that mean winter living room and bedroom temperatures were significantly below 18 °C, the health minimum indoor temperature recommended by the WHO [46]. Indoor temperatures of those sample houses were below 16 °C for most of the winter [11,50,51]. Maintaining the indoor air temperature above 16 °C should be the first step for New Zealand houses to phase in the reduction of indoor allergens from dust mites.

Table 6. Temperatures of the 9 indoor spaces with undetectable dust mite allergens in the five new sample houses in Auckland city.

House Numbers	No. 1	No. 1	No. 2	No. 2	No. 2	No. 3	No. 3	No. 4	No. 5	
Indoor Spaces	N Living	N Bed.	N Bed.	N 2nd Bed.	S Bed.	N Living	S Bed.	N Bed.	N Bed.	Outdoor
Mean temperature (°C)	17	19	19	19	19	20	20	18	18	12
% Time when $T \geq 16$ °C	65%	98%	97%	98%	99%	100%	100%	88%	92%	4%
% Time when $T \geq 17$ °C	39%	90%	91%	93%	93%	96%	98%	67%	81%	1%
% Time when $T \geq 18$ °C	18%	76%	70%	72%	74%	87%	92%	41%	60%	0%
% Time when $T \geq 20$ °C	0%	40%	6%	20%	27%	48%	56%	14%	15%	0%
% Time when $T \geq 22$ °C	0%	2%	0%	1%	2%	8%	21%	3%	2%	0%

Table 7. Temperatures of the 6 indoor spaces with low dust mite allergens in the nine new sample houses in Auckland city.

House Numbers	No. 1	No. 3	No. 4	No. 4	No. 5	No. 5	
Indoor Spaces	S Bed.	N Bed.	N Living	S Bed.	N Living	S Bed.	Outdoor
Mean temperature (°C)	15	19	16	17	15	15	12
% Time when $T \geq 16$ °C	17%	100%	52%	64%	26%	11%	4%
% Time when $T \geq 17$ °C	1%	98%	38%	40%	9%	2%	1%
% Time when $T \geq 18$ °C	0%	90%	27%	22%	2%	0%	0%
% Time when $T \geq 20$ °C	0%	25%	8%	3%	0%	0%	0%
% Time when $T \geq 22$ °C	0%	0%	0%	0%	0%	0%	0%

Table 8. Temperatures of the 10 indoor spaces with medium dust mite allergens in the six old sample houses in the remote sites.

House Numbers	No. 1	No. 2	No. 2	No. 3	No. 4	No. 4	No. 5	No. 5	No. 6	No. 6	
Indoor Spaces	N Living	N Living	S Bed.	N Bed.	S Living	N Bed.	N Living	S Bed.	N Living	S Bed.	Outdoor
Mean temperature (° C)	9	10	11	8	9	11	11	11	9	7	9
% Time when $T \geq 16$ °C	1%	4%	7%	1%	0%	8%	7%	1%	0%	0%	1%
% Time when $T \geq 18$ °C	0%	2%	4%	0%	0%	3%	4%	0%	0%	0%	0%
% Time when $T \geq 20$ °C	0%	1%	3%	0%	0%	1%	2%	0%	0%	0%	0%
% Time when $T \geq 22$ °C	0%	0%	3%	0%	0%	0%	1%	0%	0%	0%	0%

Table 9. Temperatures of the 7 indoor spaces with high dust mite allergens in the five old sample houses in the remote sites.

House No.	No. 7	No. 7	No. 8	No. 9	No. 9	No. 10	No. 11	
Indoor Spaces	N Living	S Bed.	S Bed	N Living	S Bed.	S Bed.	S Bed.	Outdoor
Mean temperature (° C)	10	9	12	13	11	13	13	9
% Time when $T \geq 16$ °C	5%	0%	5%	14%	3%	17%	15%	1%
% Time when $T \geq 18$ °C	2%	0%	0%	6%	1%	8%	5%	0%
% Time when $T \geq 20$ °C	1%	0%	0%	3%	0%	4%	1%	0%
% Time when $T \geq 22$ °C	1%	0%	0%	1%	0%	1%	0%	0%

In the indoor spaces with medium or high dust mite allergens, the mean indoor temperatures were all significantly below 16 °C (Tables 8 and 9). In indoor spaces with high dust mite allergens, the mean indoor air temperature was 11 °C, with a range of 9–13 °C. In indoor spaces with medium dust mite allergens, the mean indoor temperature was 10 °C, with a range of 7–11 °C. The mean indoor temperature (11 °C with a range of 9–13 °C) of indoor spaces with high dust mite allergens was slightly higher than the indoor spaces with medium dust mite allergens (10 °C with a range of 7–11 °C). Dust mites prefer warm and humid indoor environments [2,14,15,18–20]. Environmental temperature increase (the range from 10 °C to 25 °C) can cause higher dust mite populations [14,15,18–20]. If indoor temperatures of an indoor space cannot be maintained above 16 °C by temporary space heating, a small increase (a couple of degrees) of indoor temperature can provide a better thermal environment for dust mites to thrive.

3.3. Indoor RH Close to the Floors

Figures 3 and 4 show indoor mean air temperatures and RH in the indoor spaces with different dust mite allergen levels. The mean air temperature is normally higher next to the ceiling than the floor (Figure 3), and the mean RH is higher close to the floor than the ceiling (Figure 4). Dust mite development in carpets is more directly impacted by the RH of the air close to the floor than the mean indoor RH.

Figure 5 illustrates the mean RH close to the floors in indoor spaces with different dust mite allergen levels. The mean RH close to the floors is generally higher than the mean indoor RH. In the indoor spaces with undetectable dust mite allergens, the mean RH close to the floors was 69.4%, with a range of 66.3–70.9%, which was below or close to 70%. In the indoor spaces with low dust mite allergens, the mean RH close to the floors was 73.2%, with a range of 72.3–74.9%, which was higher than 70% and below 75% (the conditions for dust mites to thrive). In the indoor spaces with medium dust mite allergens, the mean RH close to the floors was 84.5%, with a range of 78.6–92.8%, which was significantly higher than the conditions for dust mites to thrive (≥ 75 –80%). In the indoor spaces with high dust mite allergens, the mean RH close to the floors was 82.7%, with a range of 80.0–87.5%,

which was significantly higher than the conditions for dust mites to thrive ($\geq 75\%$). The main difference in mean RH close to the floors between indoor spaces with acceptable and unacceptable dust mite allergen levels was whether the mean indoor RH was higher than 75% (the conditions for dust mites to thrive).

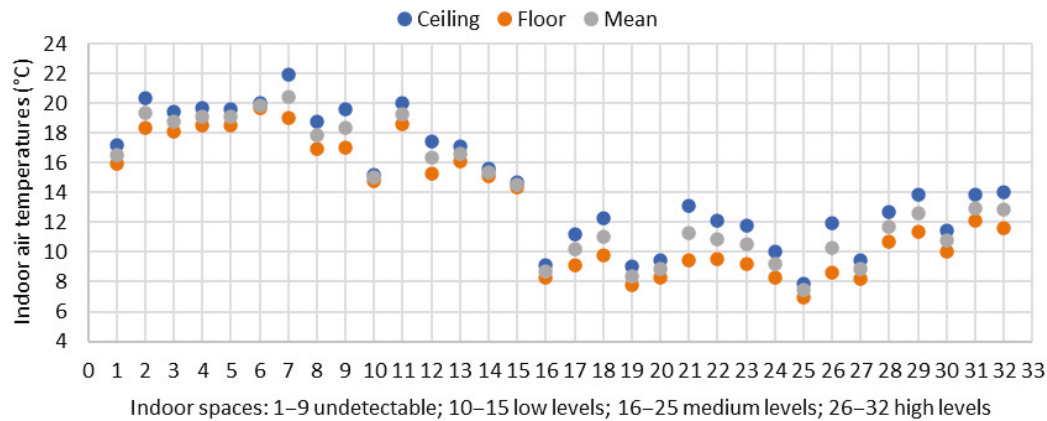


Figure 3. Winter indoor mean temperatures and indoor mean temperatures close to the ceilings and the floors of indoor spaces with different dust mite allergens in the new and old sample houses.

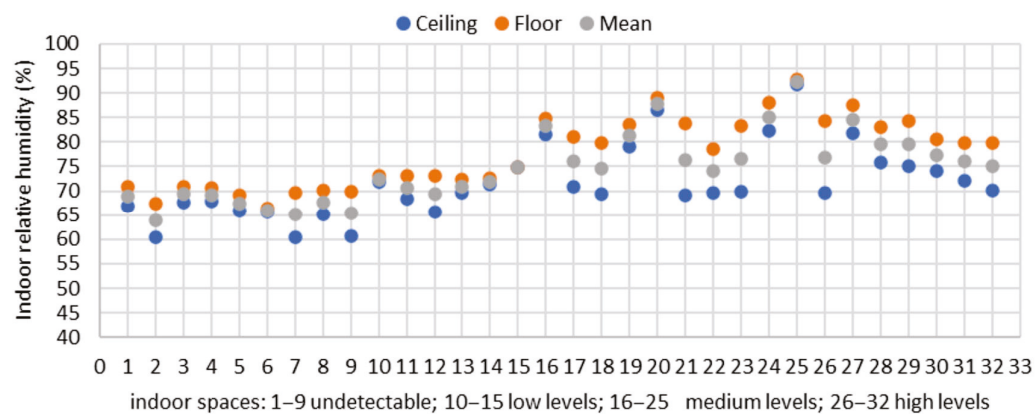


Figure 4. Winter indoor mean RH and indoor mean RH close to the ceilings and the floors of indoor spaces with different dust mite allergens in the new and old sample houses.

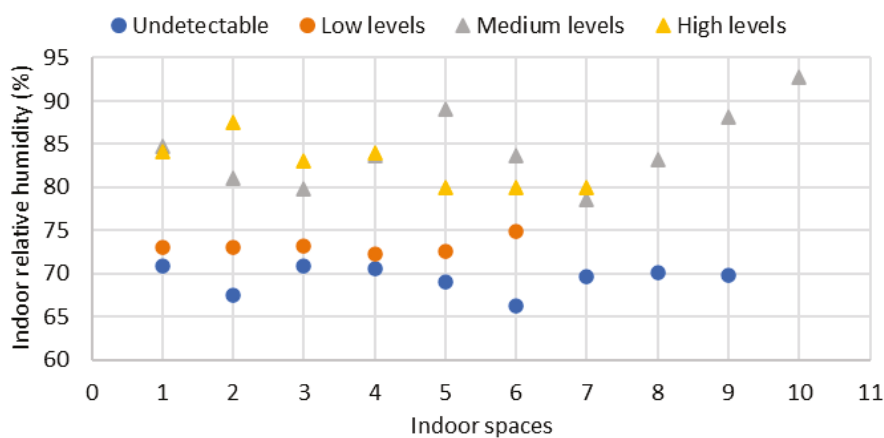


Figure 5. Winter indoor mean RH close to the floors in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

Figure 6 illustrates percentages of time in winter when RH close to the floors was higher than or equal to 75%. In the indoor spaces with undetectable dust mite allergens, the

percentages of time in winter when RH close to the floors was higher than or equal to 75% were all below 30%; the mean percentage was 15.8%, with a range of 3.0–28.7%. In the indoor spaces with low dust mite allergens, the percentages of time in winter when RH close to the floors was higher than or equal to 75% were all below 50%; the mean percentage was 39.9%, with a range of 29.2–48.7%. In the indoor spaces with medium dust mite allergens, the percentages of time in winter when RH close to the floors was higher than or equal to 75% were all higher than 70%; the mean percentage was 92.3%, with a range of 73.3–99.6%. In the indoor spaces with high dust mite allergens, the percentages of time in winter when RH close to the floors was higher than or equal to 75% were all higher than 80%; the mean percentage was 92.5%, with a range of 81.8–98.8%. The main differences between indoor spaces with acceptable and unacceptable dust mite allergen levels were the time in winter when RH close to the floors was higher than or equal to 75%, the conditions for dust mites to thrive.

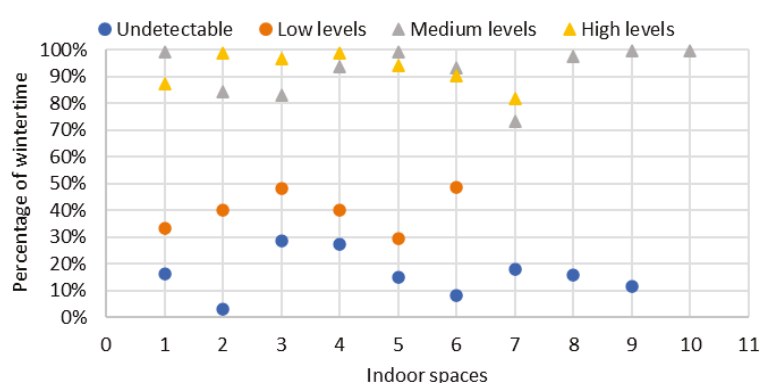


Figure 6. Percentages of time in winter when RH close to the floors was higher than or equal to 75% in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

Figure 7 illustrates average time (h) per day in winter when RH next to the floors was below 75% in the indoor spaces with different dust mite allergen levels. In the indoor spaces with undetectable dust mite allergens, the average time (h) per day in winter when RH close to the floors was below 75% was in all cases more than 17 h; the average time was 20 h, with a range of 17–23 h. In the indoor spaces with low dust mite allergens, the average time (h) per day in winter when RH close to the floors was below 75% was in all cases more than 12 h; the average time was 14 h, with a range of 12–17 h. The study compared and identified the differences of indoor RH in the indoor spaces with different dust mite allergen levels and identified the thresholds of RH to maintain the indoor dust mite allergens at acceptable levels.

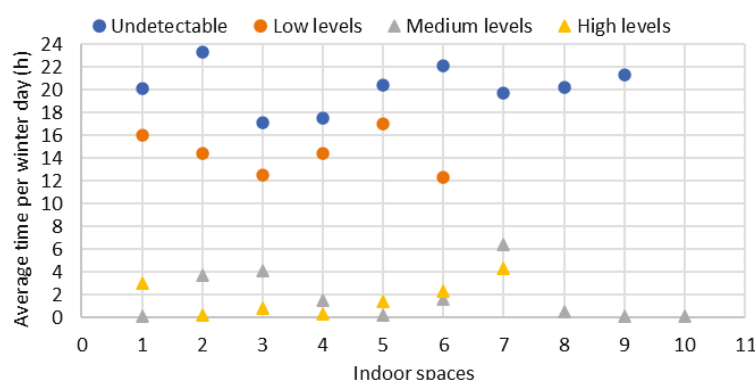


Figure 7. Average time (h) per day in winter when RH close to the floors was below 75% in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

Figure 8 illustrates percentages of time in winter when RH close to the floors was higher than or equal to 80%, the conditions for mould spores to germinate (Table 1) [38]. In the indoor spaces with undetectable dust mite allergens, the percentage of time in winter when RH close to the floors was higher than or equal to 80% was in all cases below 5%; the mean percentage of time was 2%, with a range of 0–5%. In the indoor spaces with low dust mite allergens, the percentage of time in winter when RH close to the floors was higher than or equal to 80% was in all cases below 26%; the mean percentage of time was 10%, with a range of 0–26%. In the indoor spaces with medium dust mite allergens, the percentage of time in winter when RH close to the floors was higher than or equal to 80% was in all cases higher than 47%; the mean percentage of time was 80%, with a range of 47–100%. In the indoor spaces with high dust mite allergens, the percentage of time in winter when RH close to the floors was higher than or equal to 80% was in all cases higher than 50%; the mean percentage of time was 71%, with a range of 53–94%.

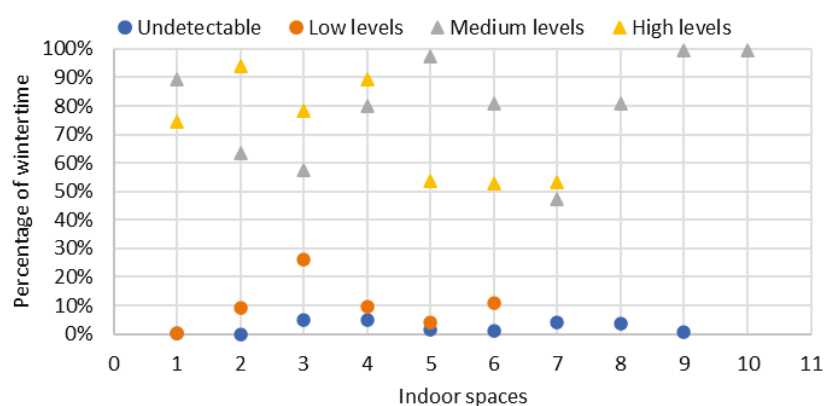


Figure 8. Percentages of time in winter when RH close to the floors was higher than or equal to 80% in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

Figure 9 illustrates the time in winter (days) when RH close to the floors was higher than or equal to 80%, the conditions for mould spores to germinate (Table 1). In the indoor spaces with undetectable dust mite allergens, the time in winter when RH close to the floors was higher than or equal to 80% was in all cases less than 4.7 days. In the indoor spaces with low dust mite allergens, the time in winter when RH close to the floors was higher than or equal to 80% was in all cases less than 24 days. In the indoor spaces with undetectable and low dust mite allergens, the total time in winter when RH close to the floors was higher than or equal to 80% was less than 30 days, the threshold of time required for mould spores to germinate. If indoor mould spore germination can be prevented, there will be no mould allergens in that indoor space [39]. The RH close to the floor is the highest RH in an indoor space. If the RH of the air close to the floor does not meet the conditions for mould spores to germinate, the RH of the air above the floor, close to the walls and ceiling, will never reach the conditions for mould spores to germinate. An indoor space with acceptable dust mite allergen levels is unlikely to have mould allergen problems. In the indoor spaces with medium dust mite allergens, the total time in winter when RH close to the floors was higher than or equal to 80% was in all cases more than 43 days. In the indoor spaces with high dust mite allergens, the time in winter when RH close to the floors was higher than or equal to 80% was in all cases more than 48 days. In the indoor spaces with medium or high dust mite allergens, the time in winter when RH close to the floors was higher than or equal to 80% was in all cases significantly more than 30 days, and indoor RH met the thresholds for mould survival, growth and development conditions for

most of the time in winter (Tables 4 and 5); there were likely to be mould growth problems in those spaces.

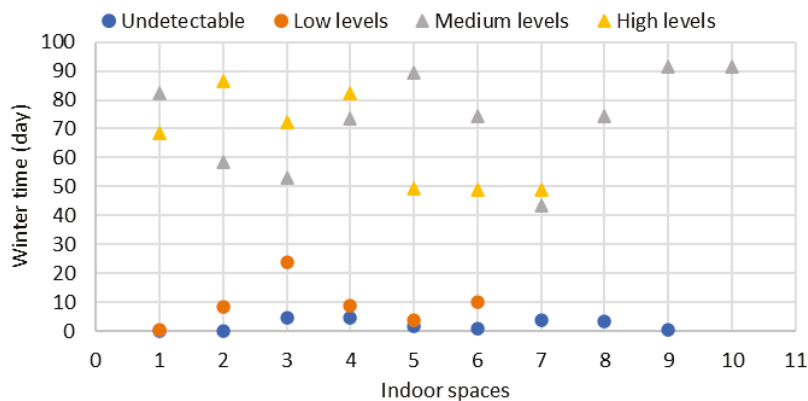


Figure 9. Time in winter (days) when RH close to the floors was higher than or equal to 80% in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

Table 10 lists the indoor spaces with unacceptable dust mite allergen levels and identified mould growths. As mould samples were mainly collected from the limited indoor surfaces with visible mould, mould growth on other surfaces could have been missed, and the accuracy of mould test results could have been influenced. Mould test results also could have been influenced by occupants' daily life and could depend on how often the occupants cleaned the indoor surfaces, especially areas with visible mould. Based on the field study data of the sample houses with unacceptable levels of dust mite allergens, the general relationships between dust mite allergens and mould in test results can be identified. Most of the sample houses with high levels of dust mite allergens also had an abundant or moderate level of *Cladosporium*, and most of the sample houses with a medium level of dust mite allergens also had low to moderate levels of *Cladosporium*. If a house has medium or high levels of dust mite allergens, it is likely to have a mould growth problem, and vice versa.

Table 10. The 17 indoor spaces with unacceptable dust mite allergen levels and identified mould growths in the 11 old sample houses.

Indoor Spaces	Dust Mite Allergens	Mould or Spores Found	<i>Cladosporium</i>	Unidentified Fungus	Miscellaneous or Other Spores
1	Medium	Yes	Moderate	Low	-
2	Medium	Yes	-	Moderate	-
3	Medium	Yes	Low	Low to moderate	-
4	Medium	Yes	Low	Low to moderate	-
5	Medium	Yes	-	Moderate to abundant	Low
6	Medium	Yes	-	Moderate to abundant	Low
7	Medium	Yes	Moderate	Moderate to abundant	-
8	Medium	Yes	Moderate	Moderate to abundant	-
9	Medium	Yes	-	Moderate to abundant	Low
10	Medium	Yes	Low to moderate	Low	-
11	High	Yes	Abundant	-	-
12	High	Yes	Abundant	-	-
13	High	Yes	-	Low	Low
14	High	Yes	Abundant	-	-
15	High	Yes	-	Low to moderate	Low
16	High	Yes	-	Low to moderate	-
17	High	Yes	Moderate	-	-

Figure 10 illustrates the time in winter (days) of indoor spaces with different dust mite allergen levels when RH close to the ceilings was higher than or equal to 80%, the conditions for mould spores to germinate. In some indoor spaces with medium or high dust mite allergens, the total time in winter (days) when the indoor RH close to the ceilings was higher than or equal to 80%, the conditions for mould spore germination, was significantly more than 30 days. If visible mould on the ceilings or walls can be identified, there must be serious dust mite allergens and serious mould growth on hidden indoor surfaces, especially the areas close to the floor. New Zealand asthma prevalence remains at one of the highest rates in the world [53]. About 13% of adults and 14% of children have asthma in New Zealand [54,55]. The previous studies showed that sensitisation to HDM was an important factor for children developing asthma [56]. High indoor dust mite allergens could contribute to the severity of asthma and the significant high rate of asthma prevalence in New Zealand [57]. Young children living in the house with a severe mould allergy could develop asthma and allergic rhinitis [58,59]. Chronic respiratory diseases developed in early years can negatively impact life expectancy [60]. The ability of mould spores to germinate and dust mites to thrive is solely related to high indoor RH; therefore, indoor RH should be controlled to prevent dust mites from thriving and mould from germinating at the same time.

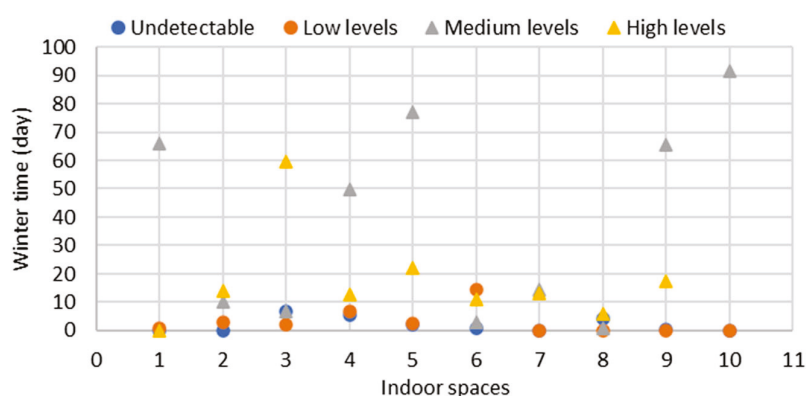


Figure 10. Time in winter (days) when RH close to the ceilings was higher than or equal to 80% in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

3.4. Mean Indoor Air Temperatures Close to the Floors and Different Dust Mite Allergen Levels

Figure 11 illustrates mean indoor temperatures close to the floors of indoor spaces with different dust mite allergen levels. The mean indoor temperatures close to the floors were slightly lower than the mean indoor air temperatures of the spaces. In the indoor spaces with undetectable dust mite allergens, the mean indoor temperatures close to the floors were all higher than or equal to 16.0 °C; the mean indoor temperature close to the floors of these spaces was 18.0 °C, with a range of 16.0–19.7 °C. In the indoor spaces with low dust mite allergens, the mean air temperatures close to the floors were higher than or close to 15.0 °C; the mean air temperature close to the floors of these spaces was 15.7 °C, with a range of 14.4–18.6 °C. In the indoor spaces with medium dust mite allergens, the mean air temperatures close to the floors were all below 10.0 °C; the mean air temperature close to the floors of these spaces was 8.7 °C, with a range of 7.0–9.8 °C. In the indoor spaces with high dust mite allergens, the mean air temperatures close to the floors were all below 13.0 °C; the mean air temperature close to the floors of these spaces was 10.5 °C, with a range of 8.2–12.1 °C. In the indoor spaces with medium and high dust mite allergens, the mean indoor temperatures of those spaces (Tables 8 and 9) and the mean indoor temperatures close to the floors were very low. When winter indoor temperatures

are lower than 12 °C, occupants' blood pressure and blood viscosity can be temporarily increased, which could lead to heart attacks or strokes. When winter indoor temperatures are lower than 9 °C, elderly occupants' core temperature can decrease [61–63]. Those low indoor temperatures can not only increase indoor allergens but also can negatively impact occupants' health [33,34].

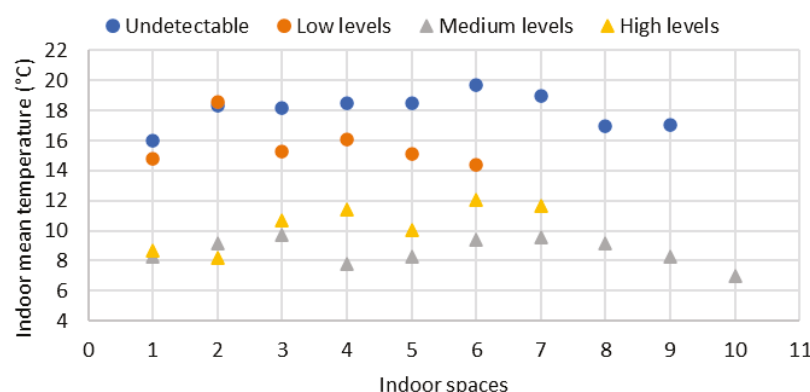


Figure 11. Mean indoor temperatures close to the floors of indoor spaces with different dust mite allergen levels in the indoor spaces with different dust mite allergen levels in the new and old sample houses.

The sample houses in this study having medium or high dust mite allergens in their indoor spaces were built from the 1920s to the 1970s. Before 1977, there was no insulation in the building envelopes of New Zealand houses. The R-value requirements for building elements in Climate Zone 1 (R1.9 for roofs, R1.5 for walls, and R0.9 for floors) were in accordance with the New Zealand standard in 1977 [64,65]. The R-value requirements for building elements in Climate Zone 1 (R1.9 for roofs, R1.5 for walls, and R1.3 for floors) were updated in accordance with the revised standard in 1996 [52,66]. After 1996, the R-value requirements for building elements in Climate Zone 1 were continuously updated. There were no requirements for double glazing until 2004 [67]. The R-value requirements for building elements in Climate Zone 1 (R2.9 for roofs, R1.9 for walls, R1.3 for floors, and R0.26 for glazing) were updated in accordance with the standard in 2009 [8,9]. However, there are still many New Zealand houses without adequate insulation. It is difficult or not energy efficient for those houses without insulation or with limited insulation to be heated up to and maintained at the minimum indoor temperature required for maintaining indoor dust mite allergens at acceptable levels. The low indoor temperature can result in high indoor RH, which can negatively impact indoor air quality. It is widely acknowledged that poor-quality living conditions can negatively impact occupants' health and wellbeing [34,68,69]. The new sample houses with acceptable dust mite allergen levels had sufficient insulation (R2.9 for roofs, R1.9 for walls, R1.3 for floors, and R0.26 for glazing) or basic insulation (R1.9 for roofs, R1.5 for walls, and R1.3 for floors) in their building envelopes. Retrofitting existing houses with adequate insulation in their envelopes, in accordance with the new building standards, should be the first step to improving indoor health conditions. These measures can raise the baseline of winter indoor temperatures, which can make temporary space heating more efficient to maintain the indoor temperatures for controlling the indoor allergens at the acceptable levels.

A limitation of this study is the lack of survey data on occupant behaviour in houses, for example, how occupants used space heating, especially for the indoor spaces with acceptable dust mite allergen levels, and how much space heating energy was needed for maintaining indoor thermal conditions required to control dust mite allergens at acceptable

levels. Further study on indoor dust mite allergens could include the area of beds, sofas, upholstered furniture, etc.

4. Conclusions

Based on the field studies, this study contributes the new physical data of winter indoor thermal conditions of the indoor spaces with four different dust mite allergen levels in New Zealand houses under a temperate climate with mild and wet winters. For maintaining indoor dust mite allergens at acceptable levels, the mean indoor RH does not have to be controlled below the conditions for dust mites to survive (50%) or the ideal conditions for dust mites to grow (60%). There were clear differences of mean indoor RH between the indoor spaces with acceptable and unacceptable dust mite allergen levels. There were significant differences in percentages of time in winter between the indoor spaces with acceptable and unacceptable dust mite allergen levels when the indoor RH was higher than or equal to 75–80% (the conditions for dust mites to thrive).

This study identified thresholds of indoor mean temperatures and percentages of time in winter when indoor RH is higher than or equal to 70% and 75% to maintain indoor dust mite allergens at the acceptable levels (undetectable and low dust mite allergen levels). This study provides a new method to identify the relationships between indoor thermal conditions and indoor dust mite allergen levels. The relationships can serve as guidelines to maintain indoor dust mite allergens at acceptable levels.

If the RH close to the floor can be maintained to keep dust mite allergens in the carpet at the acceptable levels, and the indoor RH never meets the conditions for mould spores to germinate, there will be no mould growth or allergy problems. The main indoor allergens from dust mites and mould can be prevented at the same time in New Zealand housing. Maintaining the indoor air temperature above 16 °C should be the starting point for New Zealand houses to phase in the reduction of indoor allergens from dust mites and mould.

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Article

Subjective Perception and Cooling Effect for Dynamic Ventilation with Fluctuating Air Velocity

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Abstract: Dynamic ventilation has proven effective in enhancing indoor thermal comfort. However, previous studies often expose participants to inconsistent thermal environments, potentially compromising the accuracy of subjective evaluations. To address this limitation, this study implemented dynamic ventilation with fluctuating air velocity in an accurately controlled environmental chamber. Objective measurements of indoor air velocity and air temperature distribution are conducted, and subjective thermal sensation votes are collected under thermally consistent environments among participants. During the experiment, all participants experience similar dynamic thermal environments. The results show that participants experience thermal comfort under dynamic ventilation. Dynamic ventilation enhances convective heat transfer between the human body and the surrounding air and stimulates cutaneous cold receptors. The pronounced cooling effect of dynamic airflow contributes to a reduction in skin temperature on the head, chest, upper arm, forearm, hand, and thigh, with a temperature drop ranging from 1.3% to 2.8%. In addition, dynamic ventilation significantly reduces draft risk, with the proportion of participants reporting a dissatisfied sensation decreasing from 10% to 0%. This study demonstrates the advantages of dynamic ventilation in improving thermal comfort and minimizing draft risk under controlled and uniform environmental conditions for all participants.

Keywords: dynamic ventilation; subjective perception; cooling effect; thermal consistency

1. Introduction

Thermal comfort is essential for the health and well-being of occupants. Although steady thermal environments are widely implemented, they often fall short in meeting the evolving needs of occupants. Studies have shown that a steady thermal environment is not always the optimal solution [1], and it may contribute to sick building syndrome and other potential health risks. Dynamic ventilation creates a non-steady-state thermal environment by introducing variations in air velocity, such as pulsating airflow [2] or simulated natural wind [3]. These airflow fluctuations stimulate the skin's cutaneous cold receptors [4], triggering physiological responses that create a cooling sensation. The cooling effect of dynamic airflow enhances thermal sensation and overall comfort [5,6], while also extending the upper threshold of acceptable thermal comfort temperatures [7]. Dynamic ventilation helps to mitigate the discomfort of draft risk caused by direct airflow under high-temperature conditions. For example, dynamic ventilation with pulsating air velocity

has been shown to reduce the percentage of dissatisfaction from 34% to just 8% [8]. Beyond improving comfort, dynamic ventilation also offers energy-saving benefits. Periodic and stepwise variations in the thermal environment have been reported to lower fan energy consumption by 31–68% and reduce cooling loads by up to 13% [7,9].

To better understand the mechanisms behind dynamic ventilation and its impact on thermal comfort, researchers have investigated the characteristics of indoor airflow fields, with particular attention to their cooling effects. Two key parameters commonly used to describe dynamic airflow are the logarithmic power spectrum slope [10] and turbulence intensity [11,12]. The β value, which represents the negative slope of the airflow's power spectrum curve, quantifies the variability of velocity fluctuations. Studies have shown that mechanical wind typically exhibits lower β values than natural wind, generally falling within the range of 1.2 to 1.6. With the development of the airflow, the air velocity tends to decrease, while the β value of mechanical wind increases. Some studies suggest that the β value can serve as a useful metric for characterizing dynamic airflow and optimizing ventilation strategies for thermal comfort [13]. Turbulence intensity, another crucial indicator, describes the ratio of the standard deviation of the velocity fluctuations to the mean airflow velocity [14]. Higher turbulence intensity is associated with stronger convective cooling effects and improved thermal sensation [12]. For instance, quantitative analysis indicates that at an airflow velocity of 1.0 m/s, a 15% increase in turbulence intensity can enhance the convective heat transfer coefficient by approximately 3 W/(m² K) [5]. Under sedentary conditions in an indoor environment with temperatures ranging from 26.0 °C to 32.0 °C, dynamic airflow can increase the overall convective heat transfer coefficient by approximately 10 W/(m²·K), reduce the average skin temperature by around 0.5 °C, and lower thermal sensation votes by one scale [5]. Complementary findings from Du et al. [15] indicate that the fluctuations in convective heat loss under dynamic airflow can range from 10 W/m² to 50 W/m², further highlighting its physiological cooling potential.

Airflow characteristics and physiological responses form the basis for understanding the cooling mechanisms of dynamic ventilation, while the evaluation of occupants' thermal comfort under dynamic ventilation relies primarily on subjective experiments [16]. For instance, Tian et al. [8] reported that more than 87% of participants experienced thermal comfort under pulsating air supply based on the subjective questionnaire results. Zhou et al. [17] demonstrated that pulsating airflow can effectively reduce overall and local thermal sensation by as much as 0.21 and 0.28 scale units, respectively. However, many existing studies lack environmental consistency. For instance, participants seated at different distances from the supply air vents may experience varying airflow velocities and temperature fluctuations, leading to inconsistencies in the thermal environment. These spatial inconsistencies may have influenced participants' thermal perceptions and subjective evaluations, thereby reducing the comparability and representativeness of the experimental results.

To address this limitation, the study aims to investigate the effects of dynamic ventilation with fluctuating air velocity on thermal comfort in a controllable environmental chamber. The objective is to determine whether dynamic airflow can enhance thermal comfort and reduce draft risk compared to steady ventilation. To achieve this goal, both subjective evaluations and objective measurements are conducted, ensuring that all participants experience a consistent and repeatable thermal environment. Our subjective evaluations effectively eliminate the confounding effects of environmental variability on thermal comfort assessments, thereby enhancing the accuracy of experimental data and the consistency of subjective responses. In order to gain a deeper understanding of the spatial characteristics and attenuation behavior of airflow under dynamic ventilation and steady ventilation, the distribution of air velocity and air temperature is measured and analyzed. These results are presented in Section 3.1,

providing a foundation for understanding how airflow propagates from the supply vent to the occupied zone and how its amplitude diminishes with distance. Section 3.2 summarizes the overall and local thermal sensation votes reported by participants. To validate these subjective findings, the analysis also incorporates skin temperature measurements, allowing for a quantitative assessment of the enhanced cooling sensation induced by dynamic airflow. Finally, participants' satisfaction with air movement is compared under dynamic ventilation and steady ventilation. The results demonstrate that dynamic airflow can effectively reduce draft risk while improving overall acceptance of air movement. The main contributions of this study are summarized as follows.

- (i) A controlled and consistent experimental environment is established for all participants, within which both subjective evaluations and objective measurements are conducted under dynamic ventilation and steady ventilation;
- (ii) A comprehensive comparative analysis is performed between subjective questionnaire responses and physiological indicators, thereby validating the reliability and accuracy of subjective thermal comfort assessments;
- (iii) The implementation of dynamic ventilation demonstrates a significant improvement in participants' thermal comfort satisfaction and a notable reduction in draft risk.

2. Methodology

2.1. Measurement Site

The subjective and objective experiments on dynamic ventilation are conducted in the environmental chamber located at the Innovation Harbor Campus of Xi'an Jiaotong University. The chamber measures 6.80 m in length, 4.60 m in width, and 2.85 m in height, providing a controlled environment for determining whether dynamic airflow can enhance thermal comfort and reduce draft risk compared to steady ventilation. The dynamic ventilation system implemented in this study is developed based on the patented method (ZL202210158090.1). By adjusting the supply voltage applied to the fans to precisely regulate the rotational speeds, thereby enabling dynamic variation in the supply air velocity. The dynamic ventilation system is controlled by a Siemens Programmable Logic Controller (PLC), which provides a high degree of programmability and operational stability. A user-friendly graphical user interface is provided, allowing users to specify parameters including the range, frequency, and mode of variation for supply velocity and temperature. Supported variation modes include sinusoidal, step-wise, or user-defined functions. Once the parameters are configured, the PLC executes real-time control of fan operation based on predefined logic, generating airflow fluctuations that replicate real-world ventilation dynamics. For the experiments, the environmental chamber is arranged in a classroom setting, with a row of seated participants remaining still. Each participant is exposed to identical thermal conditions to ensure consistency in the experimental data. The layout of instrumentation within the chamber is shown in Figure 1.

The technical specifications of the instruments used in the experiment are detailed in Table 1. To monitor the thermal environment, the WWFWZY-1 wireless universal wind speed and temperature recorder is employed, capable of measuring air temperature from $-20\text{ }^{\circ}\text{C}$ to $80\text{ }^{\circ}\text{C}$ and wind speed from 0.05 m/s to 30 m/s , with an accuracy of $\pm 0.5\text{ }^{\circ}\text{C}$ and $\pm 0.05\text{ m/s}$, respectively. In addition, the WWSZY-1 wind temperature and humidity recorder is used to continuously monitor air temperature ($-40\text{ }^{\circ}\text{C}$ to $100\text{ }^{\circ}\text{C}$, accuracy $\pm 0.3\text{ }^{\circ}\text{C}$) and relative humidity ($0\text{--}100\%$, accuracy $\pm 3\%$ RH). The WHQZY-1 global temperature recorder is used to measure radiant temperature, with a range of $-20\text{ }^{\circ}\text{C}$ to $80\text{ }^{\circ}\text{C}$ and an accuracy of $\pm 0.3\text{ }^{\circ}\text{C}$. More details about the recorders can be found at https://www.ybzhan.cn/st25259/product_722351.html (accessed on 1 August

2025). To capture physiological responses, DS1922L-F5# iButton sensors are attached to the participants' skin to record skin temperature, operating within a range of $-40\text{ }^{\circ}\text{C}$ to $85\text{ }^{\circ}\text{C}$ with an accuracy of $\pm 0.5\text{ }^{\circ}\text{C}$. These instruments ensure accurate and reliable data collection of both environmental and physiological thermal parameters during the experiment.

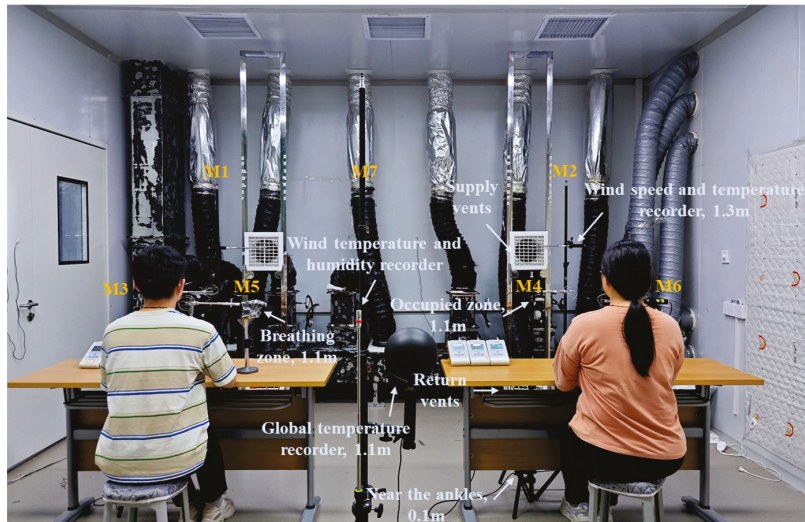


Figure 1. Schematic plan of the environmental chamber interior layout.

Table 1. Detailed information on experimental instrument.

Instruments	Parameter	Location	Height	Range	Accuracy
WWFWZY-1	Air temperature	M1–M6	1.3 m (M1–M2)	$-20\text{--}80\text{ }^{\circ}\text{C}$	$\pm 0.5\text{ }^{\circ}\text{C}$
			0.1 m (M3–M4)		
	Air velocity		1.1 m (M5–M6)		
WWSZY-1	Air temperature	M1–M2	0.2 m (M1–M2)	$-40\text{--}100\text{ }^{\circ}\text{C}$	$\pm 0.3\text{ }^{\circ}\text{C}$
	Relative humidity			0–100%	$\pm 3\%$
WHQZY-1	Global temperature	M7	1.1 m (M7)	$-20\text{--}80\text{ }^{\circ}\text{C}$	$\pm 0.3\text{ }^{\circ}\text{C}$
DS1922L-F5# iButton sensors	Skin temperature	-	-	$-40\text{--}85\text{ }^{\circ}\text{C}$	$\pm 0.5\text{ }^{\circ}\text{C}$

Seven measurement lines, labeled M1 through M7, are strategically arranged within the chamber. Measurement lines M1 and M2 are positioned in front of the supply vents and return vents. Each of these lines is equipped with the WWFWZY-1 wireless universal wind speed and temperature recorder positioned at the supply vents at a height of 1.3 m, and the WWSZY-1 wind temperature and humidity recorder placed at the return vent at a height of 0.2 m. M1 and M2 are used to monitor the air temperature and air velocity at the supply vents, as well as the air temperature and relative humidity at the return vents. Measurement lines M3 and M4 are placed along the front edge of the desks. The WWFWZY-1 wireless universal wind speed and temperature recorders are installed at heights of 0.1 m (near the ankles). Lines M5 and M6 are arranged at the center of the desks, with WWFWZY-1 wireless universal wind speed and temperature recorders placed at a height of 1.1 m (representing the breathing zone). Measurement line M7 is positioned at the center of the chamber, where the WHQZY-1 global temperature recorder is installed at a height of 1.1 m to measure radiant temperature. The recording interval for all the aforementioned instruments is set to 5 s. The DS1922L-F5# iButton sensors used for recording skin temperature are attached directly to the participants' skin using medical adhesive tape. Considering both cost efficiency and accuracy, the recording interval for the DS1922L-F5# iButton sensors is set to 10 s in this study.

To better understand the spatial distribution of airflow, the attenuation of air velocity and temperature from the supply vent to the occupied zone is analyzed. As the supply

air travels through the room, it mixes with the surrounding air, leading to momentum loss and thermal dissipation, which in turn reduce both the air velocity and temperature. Quantifying this attenuation is crucial for evaluating the effectiveness of dynamic ventilation in delivering air and maintaining thermal comfort within the occupied zone. Equations (1) and (2) describe the attenuation of air velocity and temperature from the supply vent to the occupied zone.

$$AR_v = \frac{v_s - v_o}{v_s} \times 100\% \quad (1)$$

$$AR_T = \frac{T_o - T_s}{T_s} \times 100\% \quad (2)$$

where AR_v and AR_T are the attenuation rate for air velocity and air temperature; v_o and v_s denote the air velocity at the occupied zone and the supply vent (m/s); T_o and T_s denote the air temperature at the occupied zone and the supply vent (°C).

2.2. Subjective Questionnaire

A priori power analysis is conducted using G*Power 3.1.9.7 software [18,19] to estimate the minimum required sample size prior to the experiment. For the statistical test, the effect size (f), error probability (α), and statistical power ($1-\beta$) are set at 0.4, 0.05, and 0.95, respectively. The calculation indicates that a minimum of eight participants is sufficient. To enhance the accuracy and robustness of the results, ten participants (five female and five male) are recruited for the subjective survey. All participants are graduate students from Xi'an Jiaotong University who have lived in Xi'an city for over one year. All participants are in good health and have no harmful habits. Each participant is exposed to steady ventilation and dynamic ventilation and completes the questionnaire for both conditions. The basic information of the participants is presented in Table 2.

Table 2. Basic information on participants.

Gender	Age (Years)	Height (cm)	Weight (kg)
Male	24.6 ± 2.5	173.83 ± 2.94	77.65 ± 8.83
Female	24.0 ± 1.0	163.94 ± 7.57	54.37 ± 7.03

To ensure the reliability of the questionnaire data, the entire experimental procedure and relevant precautions are thoroughly explained to all participants prior to the study. The content of the questionnaire is clearly interpreted, and a pre-experiment session is conducted to familiarize participants with the process and ensure the smooth execution of the main experiment. To minimize response bias, participants are informed before the experiment that their responses would be used solely for research purposes and that they should answer truthfully based on their actual experience. In addition, we cross-check subjective responses with physiological measurements (e.g., skin temperature) to ensure consistency. If a participant's responses are found to be inconsistent or invalid, the data are excluded from analysis. Participants are instructed in advance to maintain adequate sleep and a balanced diet on the day before each experimental session. They are also advised to avoid taking any medications within 24 h prior to the experiment and to ensure they are in good physical condition [20]. Additionally, they are required to abstain from caffeine, alcohol, and strenuous physical activity for at least 12 h before testing. Participants are required to wear typical indoor summer clothing during the experiment, including a short-sleeved T-shirt or light blouse and lightweight long trousers, along with short socks and athletic shoes.

As shown in previous studies on dynamic ventilation strategies [8,13,17], experimental durations ranging from 70 to 110 min are commonly used to capture reliable subjective responses. In this study, a total experimental duration of two hours is adopted, which is adequate for a comprehensive assessment of participants' thermal perception over time. The experiment consisted of two phases: the first 30 min served as an acclimatization period, intended to eliminate the influence of thermal history on subjective responses [21]. The remaining 90 min comprised the formal experimental phase, during which subjective and physiological data are continuously collected. Participants are instructed to arrive at the environmental chamber 15 min prior to the experiment. During this time, they are briefed on the experimental guidelines, equipped with iButton skin temperature sensors, and asked to complete a personal information questionnaire (Questionnaire A1: Basic Information of Participants in Appendix A). The detailed procedure of the subjective experiment is illustrated in Figure 2.

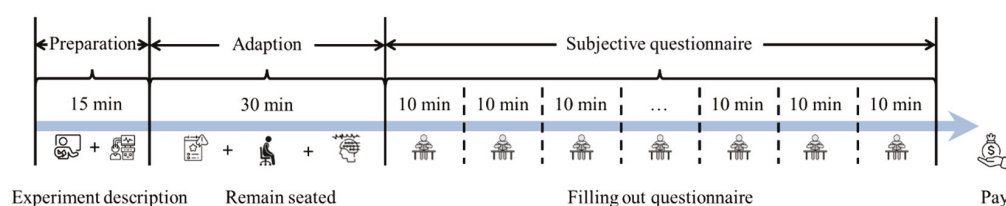


Figure 2. Schematic diagram of the experimental procedure.

The iButton devices automatically log skin temperature data at 10 s intervals and store the skin temperature in a data logger. Participants are encouraged to engage in calm activities such as reading or office work to minimize the influence of emotional arousal. Every 10 min, participants are required to complete a thermal environment evaluation questionnaire (Questionnaire A2: Thermal Comfort Evaluation Questionnaire in Appendix A), resulting in a total of one personal information questionnaire and twelve thermal sensation surveys. At the end of the experiment, a total of 240 subjective thermal environment questionnaires are collected. All participants receive compensation upon completing their participation in the study. The subjective questionnaire administered during the experiment primarily focused on thermal environment-related evaluations. These evaluations included overall thermal sensation votes, local body thermal sensation votes (covering the head, chest, upper arm, forearm, hand, thigh, calf, foot, and back), and satisfaction with air velocity. The thermal sensation and local body thermal sensation are assessed using the ASHRAE 7-point scale according to ANSI/ASHRAE Standard 55-2023 [22], where 0 indicates a neutral feeling, values below 0 indicate a sensation of cold, and values above 0 indicate a sensation of warmth. Air velocity satisfaction is measured using a Likert 7-point scale, with a score of 4 indicating acceptability. Scores below 4 reflect dissatisfaction, while scores above 4 indicate satisfaction.

2.3. Physiological Parameters

The purpose of skin temperature measurement in this study is to obtain both the local and overall mean skin temperatures of the participants during the adaptation phase and the experimental process, in order to support the reliability of the subjective questionnaire results. Considering the physiological characteristics of different body regions and the variation in skin temperature caused by thermoregulatory mechanisms, such as differences in blood perfusion, metabolic activity, and proximity to the body core [23], this study adopts a 9-point mean skin temperature calculation method [24]. The iButton sensors are placed at the following body locations: forehead, left chest, right upper arm, left forearm,

right hand, front thigh, front calf, right foot, and left upper back. The arrangement of these measurement points is illustrated in Figure 3. The overall skin temperature for each participant is calculated using Equation (3) [25]. According to the weighting factors summarized by Liu et al. [26], the proportional contributions of each site in the 9-point mean skin temperature calculation method are as follows: 0.07, 0.18, 0.07, 0.07, 0.05, 0.19, 0.13, 0.06, and 0.18, respectively.

$$T_{sk} = 0.07t_1 + 0.18t_2 + 0.07t_3 + 0.07t_4 + 0.05t_5 + 0.19t_6 + 0.13t_7 + 0.06t_8 + 0.18t_9 \quad (3)$$

where T_{sk} represents the overall skin temperature ($^{\circ}\text{C}$); t_1 to t_9 represent the local skin temperatures of the forehead, left chest, right upper arm, left forearm, right hand, front thigh, front calf, right foot, and left upper back ($^{\circ}\text{C}$).

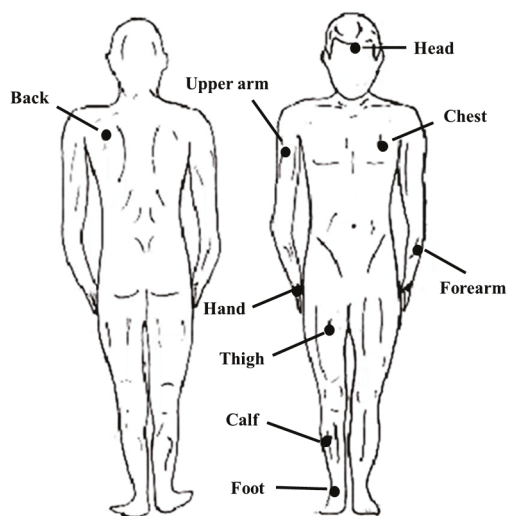


Figure 3. Diagram of skin temperature measurement sites.

3. Results

3.1. Spatial Transmission Characteristics of Airflow

This section examines how airflow parameters are spatially propagated from the supply vent to the occupied zone under both dynamic and steady ventilation conditions. Figure 4a illustrates the air velocity variation at the supply vent under dynamic ventilation and steady ventilation modes, which serve as the initial conditions for spatial transmission toward the occupied zone. In dynamic ventilation mode, the air velocity exhibits a distinct periodic fluctuation, ranging approximately from 1.6 m/s to 2.2 m/s, with a clear pulsating pattern and a cycle of about 180 s. This choice is based on Wigö's findings [27], which indicate that a thermal plume requires roughly 45 s to be disrupted by an air jet and another 45 s to recover after the jet ceases, resulting in a disruption–recovery cycle of 90 s [13]. In this study, two consecutive disruption–recovery periods ($90\text{ s} \times 2$) are applied to ensure a stable oscillatory airflow pattern, yielding a full cycle of 180 s. In contrast, the steady ventilation mode maintains a nearly constant air velocity, set equal to the peak value of the dynamic mode. The airflow remains stable at approximately 2.3 m/s. Figure 4a clearly highlights the differences in airflow characteristics between the two ventilation strategies.

Figure 4b presents box plots of air supply temperature under both dynamic ventilation and steady ventilation modes. As shown, the air temperatures remain within relatively stable ranges under both modes. The dynamic ventilation mode shows an average temperature of $24.8\text{ }^{\circ}\text{C}$ with a standard deviation of $0.3\text{ }^{\circ}\text{C}$. For dynamic ventilation, the minimum

observed temperature is 24.1 °C, with the first quartile (Q1) at 24.6 °C, the median at 24.8 °C, the third quartile (Q3) at 25.0 °C, and the maximum at 25.6 °C. The average temperature is 24.9 °C with a standard deviation of 0.2 °C under steady ventilation. The minimum observed temperature is 24.5 °C, with the first quartile (Q1) at 24.7 °C, the median at 25.0 °C, the third quartile (Q3) at 25.1 °C, and the maximum at 25.2 °C for steady ventilation. The average temperature is 24.9 °C. In both cases, the actual supply air temperatures closely align with the target setpoint of 25 °C, demonstrating a high level of temperature stability.

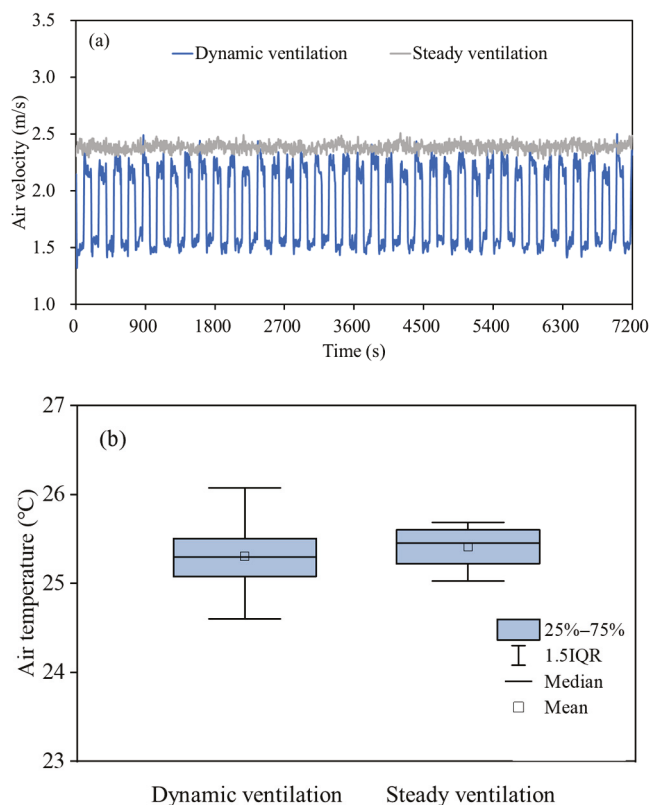


Figure 4. Air velocity and air temperature at the supply vent: (a) air velocity; (b) air temperature.

Previous studies [8,28,29] on subjective evaluations under dynamic ventilation have frequently reported inconsistencies in the thermal environments experienced by different occupants. To objectively investigate whether the dynamic ventilation shown in Figure 4 can provide uniform thermal conditions at multiple occupant locations, the temporal variations in air velocity and air temperature experienced by the two participants (M5 and M6, as shown in Figure 1) during the experiment are presented in Figure 5. The trends of the two curves within the occupied zone are highly consistent. The fluctuation amplitude, frequency, and average values of both air velocity and air temperature are nearly identical. This indicates that the thermal environmental parameters experienced by the two participants are almost the same. Therefore, it can be concluded that M5 and M6 are exposed to equivalent thermal conditions during the experiment, providing a reliable basis for subsequent analysis of subjective thermal perception and human thermal responses.

Figure 6a illustrates the variation in air velocity over time after the airflow reaches the occupied zone (location M5 in Figure 1) under both dynamic ventilation and steady ventilation modes. It can be observed that in dynamic ventilation mode, the air velocity continues to exhibit strong fluctuations, ranging from 0.8 to 1.2 m/s, with an average value of 1.0 m/s with a standard deviation of 0.3 m/s, maintaining its characteristic pulsation. This indicates that the pulsating characteristic of dynamic ventilation is effectively transmitted

to the occupied zone during airflow propagation. In contrast, while some fluctuations are also present under steady ventilation, they are relatively minor, and the overall air velocity remains stable, with an average value of 1.2 m/s and a standard deviation of 0.1 m/s in the occupied zone. These results highlight a clear distinction between the airflow characteristics of the two ventilation strategies within the occupied zone.

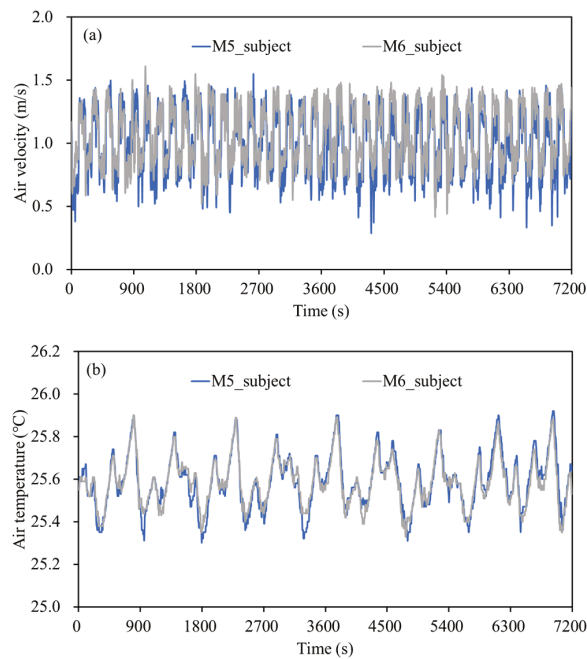


Figure 5. Air velocity and temperature distributions in the occupied zone of M5 and M6: (a) air velocity; (b) air temperature.

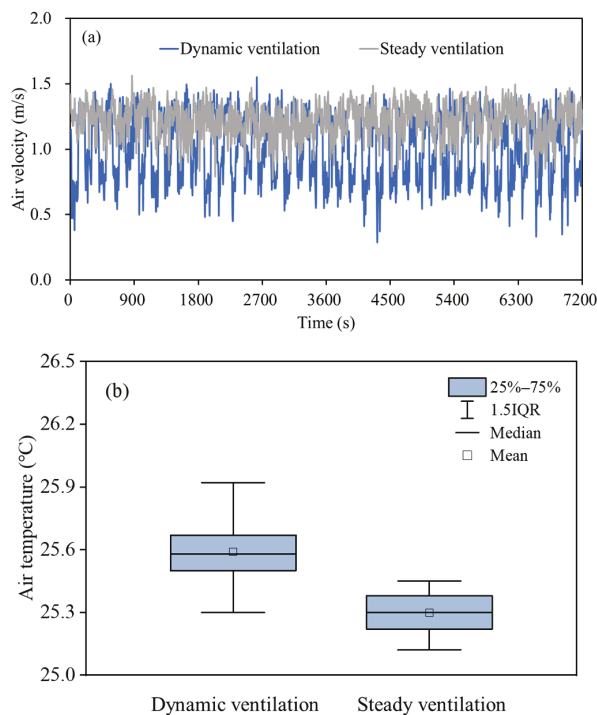


Figure 6. Air velocity and temperature distributions in the occupied zone of M5: (a) air velocity; (b) air temperature.

Under dynamic ventilation, the average air temperature of the occupied zone tends to be slightly higher compared to steady ventilation. The dynamic ventilation mode shows an average temperature of 25.6 °C with a standard deviation of 0.1 °C. The average temperature is 25.3 °C with a standard deviation of 0.1 °C under steady ventilation. This difference can be attributed to the increased turbulence intensity which enhances the mixing of supply air with the surrounding air, thereby leading to greater thermal dissipation. As shown in Figure 6b, the range between the maximum and minimum air temperatures is noticeably larger under dynamic ventilation compared to steady ventilation. This wider range reflects greater variability and fluctuation in the thermal environment caused by dynamic airflow. Specifically, the temperature range under dynamic ventilation spans approximately 0.6 °C, while that under steady ventilation is around 0.3 °C.

To further quantify the spatial transmission effects, Figure 7a,b present the attenuation of air velocity and air temperature as the airflow propagates from the supply vent to the occupied zone under both ventilation modes. As the supply air enters the occupied zone from the supply vent, entrainment and mixing with the surrounding air lead to a decrease in air velocity and an increase in air temperature. Under dynamic ventilation, the high air velocity decreases from 2.2 m/s to 1.2 m/s, corresponding to an attenuation rate of 43.6% (Equation (1)). The low velocity drops from 1.6 m/s to 0.8 m/s, resulting in an attenuation rate of 49.7%. The average supply air velocity under dynamic conditions decreases from 1.9 m/s to 1.0 m/s, with an attenuation rate of 46.5%. In comparison, under steady ventilation, the average air velocity decreases from 2.3 m/s to 1.2 m/s, corresponding to an attenuation rate of 47.9%. For air temperature, the dynamic ventilation condition shows an increase from 24.8 °C to 25.6 °C, yielding a temperature increase rate of 3.1% (Equation (2)). Under steady ventilation, the temperature rises from 24.9 °C to 25.3 °C, with an increase rate of 1.6%.

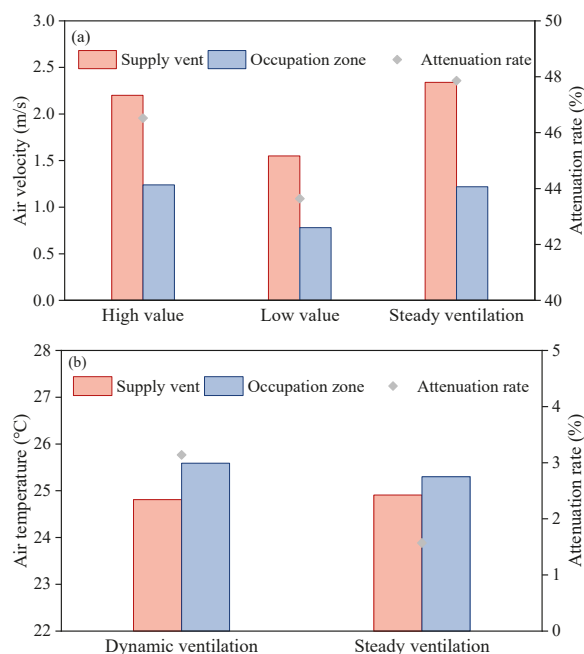


Figure 7. Attenuation of air velocity and air temperature: (a) air velocity; (b) air temperature.

3.2. Analysis of Subjective Surveys

The subjective survey results are analyzed to assess perceived thermal comfort. Figure 8 presents the average thermal sensation votes of all participants under dynamic ventilation and steady ventilation conditions across 12 voting sessions. The bar charts represent the mean thermal sensation votes for each questionnaire under the two venti-

lation modes, while the error bars indicate the corresponding 95% confidence interval, reflecting the variability and statistical reliability of the data. As shown in Figure 8, thermal sensation vote values for both dynamic ventilation and steady ventilation are predominantly concentrated within the range of -0.5 to 0.5 (yellow dotted lines), indicating that participants generally perceived a thermally comfortable indoor environment. Notably, the error bars under dynamic ventilation are generally shorter, suggesting a higher level of consistency in participants' subjective responses, with less variability and a more stable comfort experience. In contrast, the confidence intervals under steady ventilation are wider at certain time points, indicating greater individual differences in thermal perception.

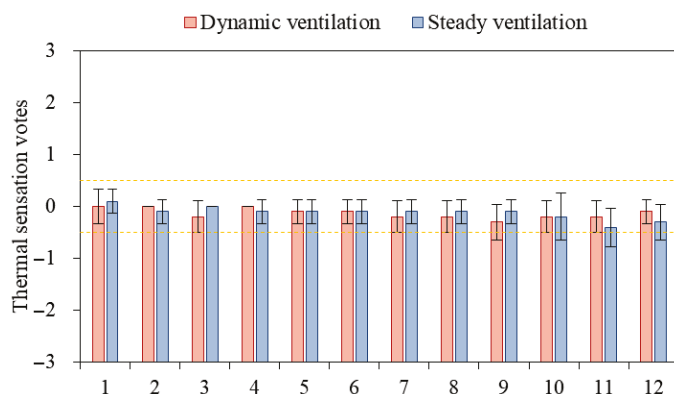


Figure 8. Overall thermal sensation votes under dynamic ventilation and steady ventilation.

To complement these subjective assessments, skin temperature is also monitored as an objective physiological indicator of thermal response under both ventilation modes. Figure 9 presents the overall average skin temperature distributions of two participants (one male participant and one female participant) under dynamic ventilation and steady ventilation modes. For both participants, the overall skin temperature is consistently lower under dynamic ventilation compared to steady ventilation. This can be attributed to the enhanced cooling effect of dynamic airflow. Specifically, dynamic ventilation enhances convective heat transfer between the human body and the surrounding air by increasing turbulence intensity [30]. Furthermore, the low-frequency fluctuations generated by dynamic airflow introduce correspondingly low-frequency variations in convective heat loss, causing the skin temperature to oscillate and more strongly stimulate cutaneous cold receptors [31]. The resulting cooling sensation contributes to an improved perception of thermal comfort [12,32]. During the initial adaptation phase (the first 30 min), both participants show either an increase or a decrease in skin temperature, which is likely related to their short-term prior thermal exposure before the experiment. After this adaptation period, the female participant's skin temperature stabilizes with a variation of only $0.2\text{ }^{\circ}\text{C}$ under steady ventilation, but shows a greater change of $0.5\text{ }^{\circ}\text{C}$ under dynamic ventilation, potentially indicating a greater sensitivity to thermal stimuli. When comparing the overall skin temperatures of the two participants, the male participant consistently exhibited slightly lower temperatures than the female, which can be attributed to individual physiological differences.

Figure 10 presents the local thermal sensation votes of the female participant shown in Figure 9a under dynamic ventilation and steady ventilation. Although the participant reported a neutral overall thermal sensation in both ventilation modes, the local thermal sensation data reveal more nuanced differences, providing further insight into the role of dynamic airflow in enhancing local thermal comfort. Compared to steady ventilation, the local thermal sensation votes under dynamic ventilation are noticeably lower in the upper arm, forearm, and hand, indicating a stronger perception of local coolness. Specifically,

the average thermal sensation votes for the upper arm and hand are -0.1 and -0.3 , respectively, suggesting a slight cool feeling in these areas under dynamic airflow, while both are perceived as thermally neutral under steady ventilation. The forearm exhibited negative thermal sensation votes under both ventilation modes, but the value is more pronounced under dynamic ventilation, with an average vote of -0.4 . This is primarily because the hands and forearms have a high density of cold thermoreceptors [33]. Cold thermoreceptors are sensitive to low-frequency fluctuations in air velocity [34]. As a result, they are more responsive to airflow stimulation and more likely to exhibit changes in thermal sensation, which is characteristic of dynamic ventilation.

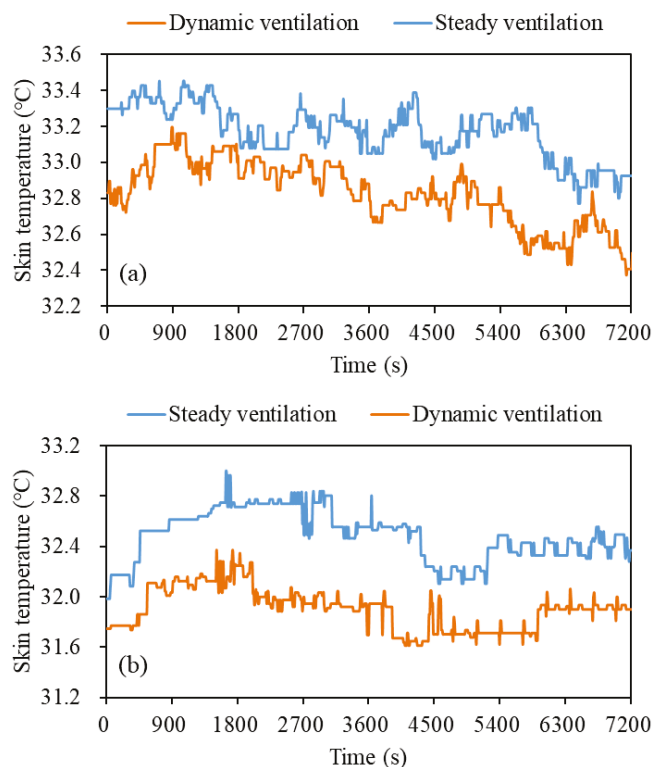


Figure 9. Mean skin temperature: (a) female participant; (b) male participant.

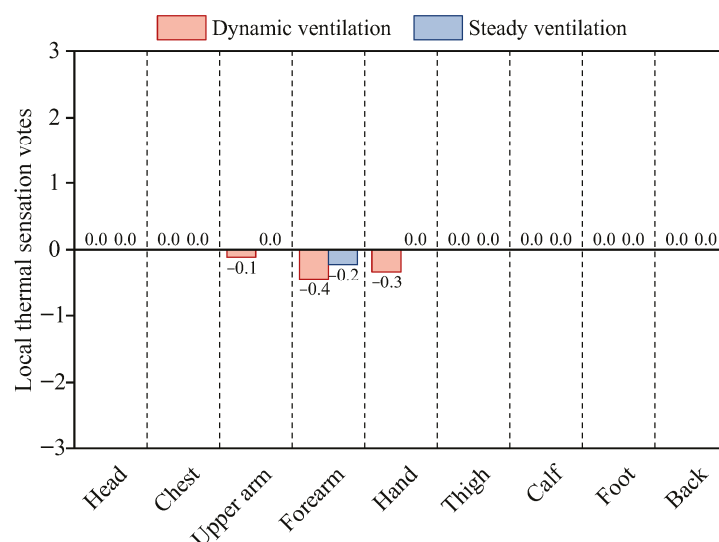


Figure 10. Local thermal sensation votes under dynamic ventilation and steady ventilation.

Local skin temperature has been recognized as a significant physiological indicator of an individual's overall thermal sensation [35]. Individual body parts contribute with different weights to warm and cool sensations, and strong local sensations can dominate the overall thermal perception. When all local sensations are near neutral, the overall sensation tends to approximate the average of all body segments [36]. Figure 11 presents the local skin temperature of the female under dynamic ventilation and steady ventilation. The participant's hands and feet exhibited lower skin temperatures compared to other parts of the body, with the head showing even lower temperatures than the extremities. In contrast, the chest and back display higher skin temperatures. This conclusion is consistent with the findings reported by Song et al. [37]. The results indicate that under dynamic ventilation, the skin temperatures of the head, chest, upper arm, forearm, hand, and thigh are lower than those recorded under steady ventilation. The reduction in skin temperature can be attributed to the intermittent and periodic fluctuations in air velocity introduced by dynamic ventilation. This airflow pattern enhances convective heat transfer between the body and the surrounding air, thereby improving cooling efficiency. This effect is particularly evident in areas with a higher density of cold thermoreceptors [38,39], making these regions more sensitive to airflow variations. For this participant, the skin temperatures of the head, chest, upper arm, forearm, hand, and thigh under dynamic ventilation are 31.2 °C, 34.2 °C, 32.8 °C, 30.2 °C, 29.0 °C, and 31.7 °C, respectively. Under steady ventilation, the corresponding temperatures are 31.7 °C, 35.1 °C, 33.4 °C, 31.1 °C, 29.5 °C, and 32.2 °C, respectively. This corresponds to temperature reductions of 1.3%, 2.7%, 1.9%, 2.8%, 1.6%, and 1.6%, respectively. This further supports the local thermal comfort voting results of the same participant shown in Figure 9. For this participant, the calf temperature shows minimal difference between the two modes, measured at 32.4 °C under dynamic ventilation and 32.4 °C under steady ventilation. For the foot and back, the participant exhibits lower skin temperatures under steady ventilation compared to dynamic ventilation. This localized deviation reflects individual physiological variability, such as differences in peripheral blood flow or local thermal sensitivity. This interpretation is supported by the weighted skin temperature, which for this participant remains lower under dynamic ventilation compared to steady ventilation (Figure 9a).

"How satisfied are you with the overall air velocity in the current environment?" This question is designed to assess participants' subjective perception of satisfaction with air velocity during the experiment. The participants rated the perceived air velocity satisfaction on a Likert 7-point scale: (1) very dissatisfied, (2) dissatisfied, (3) slightly dissatisfied, (4) acceptable, (5) slightly satisfied, (6) satisfied, (7) very satisfied. The orange dotted line in the figure marks a score of 4, representing the "acceptable" threshold. Figure 12 summarizes these responses for both dynamic ventilation and steady ventilation. The bar charts represent the mean satisfaction votes for air velocity under the two ventilation modes, while the error bars indicate the corresponding 95% confidence interval. During the initial adaptation phase (the first three voting sessions), participants report lower satisfaction with air velocity under dynamic ventilation compared to steady ventilation, primarily due to the influence of thermal history. Satisfaction with dynamic ventilation progressively improves over the course of the experiment. From the sixth voting session, it consistently surpasses that of steady ventilation, suggesting that a fluctuating airflow is more appealing to them. The voting results are consistent with the findings of Uğursal et al. [30], indicating that the majority of participants prefer a dynamic environment.

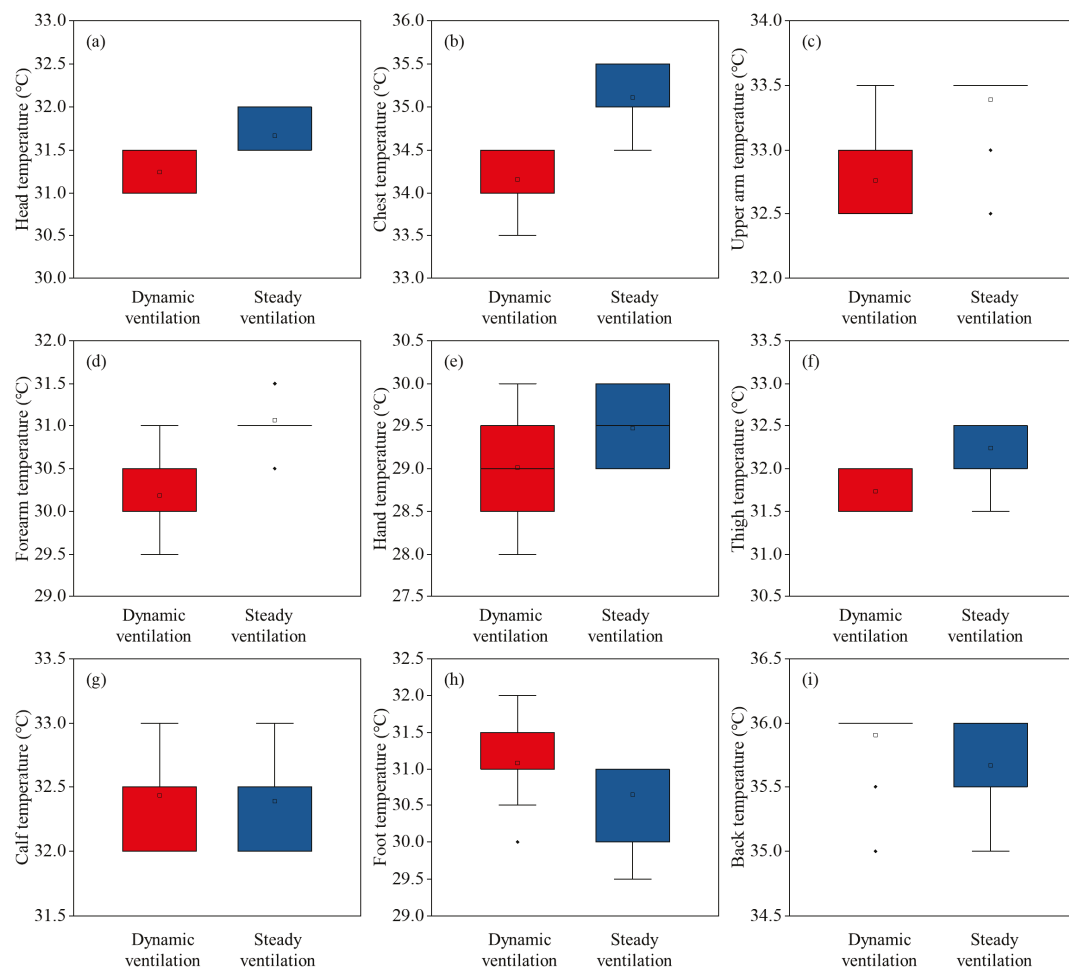


Figure 11. Participants’ local skin temperatures under dynamic ventilation and steady ventilation: (a) head; (b) chest; (c) upper arm; (d) forearm; (e) hand; (f) thigh; (g) calf; (h) foot; (i) back. Note: In the box plot, the five horizontal lines from top to bottom represent the maximum value, the upper quartile (Q3), the median, the lower quartile (Q1), and the minimum value, respectively. The white rectangle indicates the average value, and the black diamond represents an outlier.

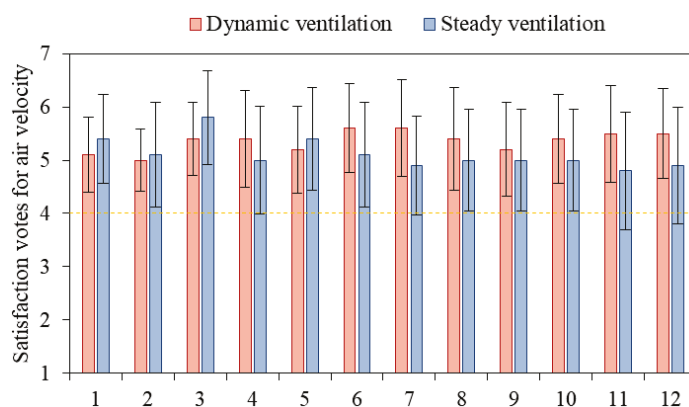


Figure 12. Voting statistics of the participants’ satisfaction with air velocity.

Draft is the unwanted local cooling caused by air movement [40]. A vote of “dissatisfied” or “acceptable” accompanied by a desire to reduce air velocity is considered indicative of draft sensation. Conversely, a vote of “acceptable” with a preference to maintain or increase air velocity or a vote of “satisfied” is interpreted as an absence of draft

sensation. After thermal adaptation in the steady ventilation, the draft rate (defined as the proportion of votes indicating dissatisfaction) with air velocity is 10%. Under dynamic ventilation, this proportion decreases to 0%, as illustrated in Figure 13, showing that a dynamic environment not only improves overall satisfaction but also eliminates draft risk.

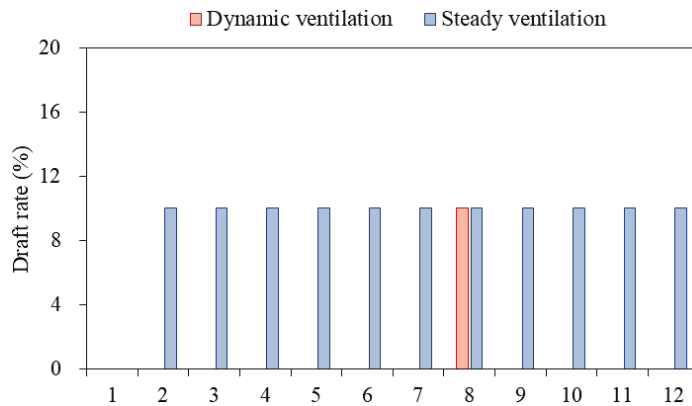


Figure 13. Draft rate under dynamic ventilation and steady ventilation.

4. Discussion

Section 3 primarily presents the results of subjective thermal environment surveys. It further explores subjects' perceptions of air quality and acoustic environment under both dynamic ventilation and steady ventilation modes. Figure 14 shows the voting statistics results of the average perceived air quality of the subjects. It can be seen that the 95% confidence interval results of the dynamic ventilation and steady ventilation are both greater than (with higher values indicating better air quality, and four representing a moderate level as marked by the orange dotted line), which indicates that both ventilation methods maintained good indoor air quality during the experiment. Dynamic ventilation enhances indoor turbulent mixing, enabling fresh air to be delivered more efficiently to the breathing zone while accelerating the dilution and dispersion of pollutants. This process contributes to improved perceived air quality [41]. Moreover, the fluctuating airflow patterns under dynamic ventilation mimic the rhythm of natural wind, which is more likely to be associated—at a psychological level—with sensations of freshness [42]. Such associations may reinforce subjects' perception of the air as cleaner and more pleasant, leading to more favorable subjective evaluations.

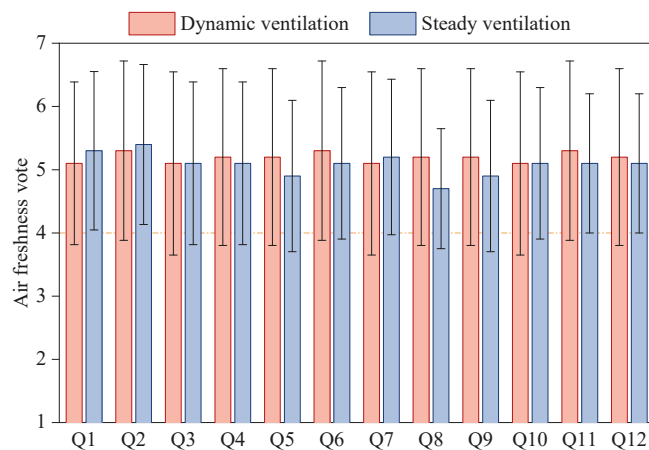


Figure 14. Mean air freshness votes under dynamic ventilation and steady ventilation.

“Do you perceive any noise in the room?” This question is designed to assess subjects’ subjective perception of the acoustic environment during the experiment. Subjects rate the perceived noise level on a Likert five-point scale: (1) no noise, (2) slight noise, (3) noticeable noise, (4) loud noise, (5) unbearable noise. Figure 15 illustrates the acoustic ratings reported under both dynamic ventilation and steady ventilation. The results indicate that the perceived noise levels are generally higher in dynamic ventilation mode than in steady ventilation mode. This is primarily attributed to the intermittent operation of the fan in dynamic ventilation mode, which reduces the continuous exposure to noise generated by the fan. Compared to steady ventilation mode—where the fan operates continuously—the non-continuous operation effectively decreases cumulative noise exposure and mitigates auditory fatigue, thereby lowering the perceived noise level [43].

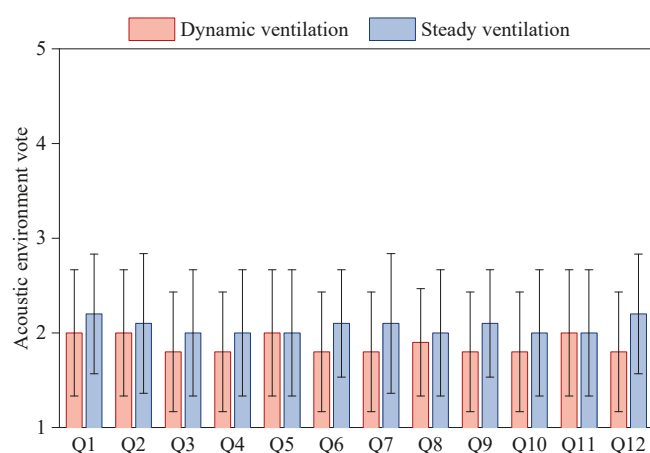


Figure 15. Acoustic environment votes under dynamic ventilation and steady ventilation.

The subjective survey results indicate that dynamic ventilation mode provides a favorable indoor environment across multiple dimensions. Subjects report thermally neutral and comfortable conditions, with less variability in thermal sensation responses compared to steady ventilation. Perceived air quality is also rated higher under dynamic ventilation, likely due to enhanced airflow mixing and psychological associations with natural wind patterns. Importantly, the acoustic environment is perceived more favorably under dynamic ventilation. The intermittent operation of the fan helps to reduce continuous noise exposure and auditory fatigue, resulting in lower perceived noise levels than under steady ventilation. These findings suggest that dynamic ventilation not only maintains thermal comfort but also supports positive perceptions of air quality and acoustic conditions.

5. Future Studies

Dynamic ventilation strategies that rely solely on variations in supply velocity are prone to exceeding the upper limit of acceptable air velocity in the occupied zone. As a result, the allowable range of velocity fluctuations is constrained, which significantly limits the cooling effect of dynamic airflow. Therefore, future research should explore low-frequency dynamic ventilation strategies that combine both supply temperature and velocity, aiming to enhance the cooling effectiveness of dynamic ventilation. Additionally, future studies may extend the application of dynamic ventilation to different indoor environments to assess its effectiveness across various spatial and occupancy scenarios.

6. Conclusions

This study investigates the thermal comfort performance of a dynamic ventilation strategy characterized by a fluctuating supply air velocity (1.6 m/s to 2.2 m/s), a constant supply temperature of 25 °C, and a pulsing period of 180 s. Subjective evaluations and objective measurements are conducted in a controlled and consistent experimental environment to minimize environmental variability between participants. Based on data collected from the recruited participants during the 2 h experimental period, the key findings from both subjective perceptions and physiological measurements can be summarized as follows:

- (1) Dynamic ventilation is capable of maintaining the participants in a thermally neutral environment, which further reduces thermal sensation votes for the upper arm, forearm, and hand to -0.1 , -0.4 , and -0.3 , respectively.
- (2) The intensified convective heat exchange coupled with the rhythmic stimulation of cutaneous cold receptors generates a stronger and more sustained cooling effect, which in turn results in a noticeable reduction in skin temperature under dynamic ventilation.
- (3) Dynamic ventilation completely eliminates draft-related dissatisfaction, with the draft rate decreasing from 10% under steady ventilation to 0% under dynamic ventilation.

These findings confirm that dynamic ventilation offers notable advantages in improving thermal comfort and reducing draft risk. The results provide theoretical and empirical support for incorporating dynamic airflow strategies into the design and operation of energy-efficient ventilation systems.

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Data Availability Statement: The datasets generated and analyzed during the current study are available from the corresponding author on reasonable request.

Conflicts of Interest: Author Chunfeng Lao, Jing Ling, Jing Li, Yan Yan, Yue Yin and Mingliang Gu were employed by the company Qingdao Haier Air Conditioner Co., Ltd.; National Engineering Research Center of Digital Home Networking. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Appendix A

The following includes the pre-experiment demographic questionnaire used to collect participants' personal information (Questionnaire A1), as well as the subjective evaluation questionnaire administered during the experiment (Questionnaire A2).

I. Questionnaire A1: Basic Information of Participants

1. Name: _____
2. Gender: _____
3. Age: _____

4. Height (cm): _____
5. Weight (kg): _____
6. Your current clothing—Upper body: _____
7. Your current clothing—Lower body: _____
8. Your current clothing—Underwear: _____

II. Questionnaire A2: Thermal Comfort Evaluation Questionnaire

1. Name: _____
2. Questionnaire number: _____
3. Your current overall thermal sensation (Single answer only):
- ☐ Cold
- ☐ Cool
- ☐ Slightly cool
- ☐ Neutral
- ☐ Slightly warm
- ☐ Warm
- ☐ Hot
4. Your current local thermal sensation (Single answer only):

Body Part	Cold	Cool	Slightly Cool	Neutral	Slightly Warm	Warm	Hot
Forehead	☐	☐	☐	☐	☐	☐	☐
Chest	☐	☐	☐	☐	☐	☐	☐
Back	☐	☐	☐	☐	☐	☐	☐
Upper arm	☐	☐	☐	☐	☐	☐	☐
Forearm	☐	☐	☐	☐	☐	☐	☐
Hand	☐	☐	☐	☐	☐	☐	☐
Thigh	☐	☐	☐	☐	☐	☐	☐
Calf	☐	☐	☐	☐	☐	☐	☐
Foot	☐	☐	☐	☐	☐	☐	☐

5. Your current overall satisfaction with air movement (Single answer only):
- ☐ Very dissatisfied
- ☐ Dissatisfied
- ☐ Slightly dissatisfied
- ☐ Acceptable
- ☐ Slightly satisfied
- ☐ Satisfied
- ☐ Very satisfied
6. Your current overall air movement acceptability (Single answer only):
- ☐ Very unacceptable
- ☐ Unacceptable
- ☐ Neutral
- ☐ Acceptable
- ☐ Very acceptable
7. What is your air movement preference at the moment (Single answer only)?
- ☐ Stronger
- ☐ No change
- ☐ Weaker

8. How would you rate the current air quality (Single answer only)?

- ☐ Very poor
- ☐ Poor
- ☐ Fairly poor
- ☐ Neutral
- ☐ Fairly good
- ☐ Good
- ☐ Very good

9. Do you think there is noise in the room (Single answer only)?

- ☐ No noise
- ☐ Slight noise
- ☐ Noticeable noise
- ☐ Loud noise
- ☐ Unbearable noise

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Article

A Concise Grid-Based Model Revealing the Temporal Dynamics in Indoor Infection Risk

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Abstract: Determining the transmission routes of pathogens in indoor environments is challenging, with most studies limited to specific case analyses and pilot experiments. When pathogens are instantaneously released by a patient in an indoor environment, the peak infection risk may not occur immediately but may instead appear at a specific moment during the pathogen's spread. We developed a concise model to describe the temporal crest of infection risk. The model incorporates the transmission and degradation characteristics of aerosols and surface particles to predict infection risks via air and surface routes. Only four real-world outbreaks met the criteria for validating this phenomenon. Based on the available data, norovirus is likely to transmit primarily via surface touch (i.e., the fomite route). In contrast, crests of infection risk were not observed in outbreaks of respiratory diseases (e.g., SARS-CoV-2), suggesting a minimal probability of surface transmission in such cases. The new model can serve as a preliminary indicator for identifying different indoor pathogen transmission routes (e.g., food, air, or fomite). Further analyses of pathogens' transmission routes require additional evidence.

Keywords: attack rate; fomite; infectious diseases; modeling; principle of parsimony; spreading events

1. Introduction

Pathogens in the environment can be transmitted from the source patient to susceptible individuals through various routes, including air, fomites, food, and water [1–3]. However, determining the relative contribution of each route to the transmission of a pathogen is often challenging [4].

Most studies investigating pathogen transmission routes have relied on indirect evidence from specific case studies [5,6] or controlled experiments conducted using pathogen surrogates [7,8]. Some studies have used animals in laboratories as the hosts to explore the transmission pathways of live viruses. For example, the influenza A virus was shown to be transmissible between guinea pigs via aerosolized fomites [7]. In field studies, different tracers (e.g., nonmicrobial markers, microbial surrogates, and genetic materials) have been used to observe the spatiotemporal dynamics of pathogens, demonstrating that contaminated surfaces contribute to disease transmission. However, animals and tracers do not perfectly represent humans and live viruses due to differences in behavior and viability. Although target viruses or their RNA materials have been detected on environmental surfaces in some case studies [9–11], they were present in very low levels, and few studies have isolated viable viruses on environmental surfaces [12]. Some studies have combined modeling and numerical simulation to predict or reproduce the infection risks of

pathogens along different transmission routes in real outbreak scenarios [13,14]; however, these models and conclusions may not be generalizable to other scenarios. In most cases, it is difficult to balance a model's complexity with its broad applicability.

Therefore, drawing on the principle of parsimony [15], we aimed to develop a streamlined, widely applicable model containing only the most essential elements to help identify indoor pathogen transmission routes. For a target disease from an instantaneous source, as the pathogen spreads across more surfaces or a larger air volume, more areas become contaminated. Simultaneously, the infection risk per unit area decreases due to pathogen degradation (e.g., death, dilution, or increased surface adhesion [16]). At a certain point during this process, the highest infection risk (quantified using models such as the multi-hit model [17]) is reached.

Inspired by the aforementioned phenomenon, we developed a model to describe dispersion events, in which the total “effect” of an assumed spreading agent is considered a combination of its unit intensity and effective scope. We then used the model to simulate infection risks for both fomite and airborne transmission routes under typical indoor conditions. Data in existing case studies suggest that the new model can provide a straightforward approach to identifying indoor pathogen transmission routes. Moreover, this new model could potentially be extended to characterize the temporal patterns of other diffusion phenomena, such as disease pandemics, product sales, and information dissemination.

2. Materials and Methods

2.1. Model Development

2.1.1. Model Framework

Pathogens spread across surfaces or through the air while simultaneously degrading in the environment. Thus, we developed a concise model to describe the latent rise-and-fall pattern in infection risk during the dispersion of pathogen-laden particles in an indoor environment (see Figure 1a).

The spread and degradation of pathogens was modeled using a two-dimensional (2D) grid. Each column in the model grid represents a surface (or a unit volume of air), and each row represents a certain pathogen-survival rate. Pathogen-laden particles spreading outward and downward from the top-center grid cell simulate a pathogen spreading across surfaces (or air parcels) while degrading. A single dispersion step of the model represents one instance of pathogen exchange between adjacent surfaces (or air volumes) or a one-level reduction in the survival rate. The detailed design of the model framework is introduced as follows.

Grid design: As shown in Figure 1b, we assume a vertical 2D grid with N columns and M rows representing different surfaces (or air volume) and different levels of infectivity, respectively.

Spreading rule: To simulate pathogen spread, a given number of pathogen-laden particles are released from the middle cell in the top row, and their movement is simulated step by step. At each step, each particle randomly moves one cell horizontally (left or right), moves one cell downward, or remains stationary. Based on these rules, in each step, a particle has a $1/4$ probability of moving downward. This defines the degradation rate of the pathogens.

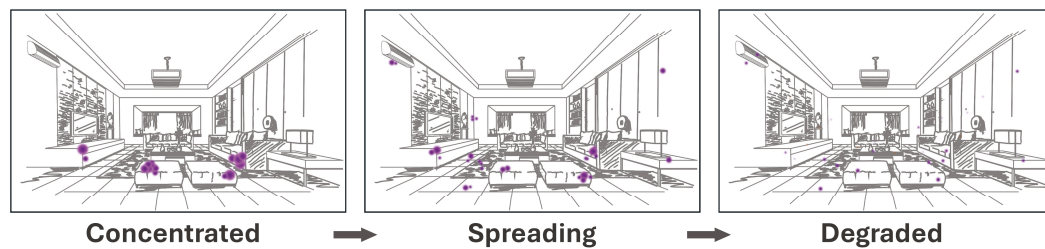
Pathogen survival rate: In the 2D grid model, the steps are assumed to last equal time intervals. During each step, particles are exchanged between adjacent columns, with each particle having a $1/4$ probability of degrading (i.e., moving downward, as mentioned earlier). Statistically, the average degradation time equals a quarter of the total steps

(i.e., $T/4$). The pathogen survival rate S is modeled as an exponential decay under constant indoor conditions and no particle interaction, as shown in Equation (1).

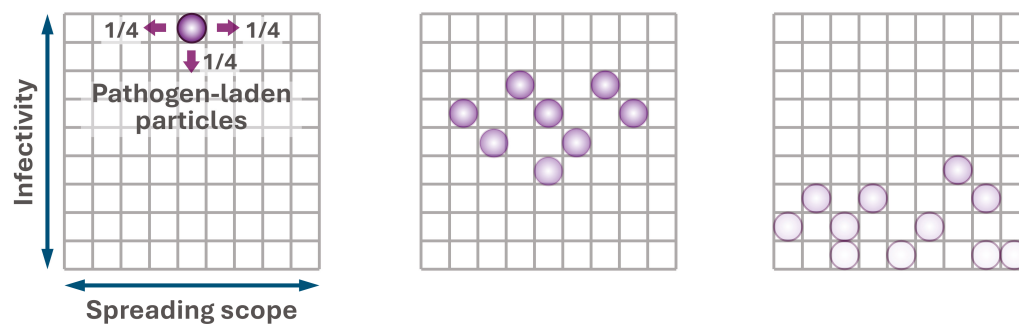
$$S = \exp[-\lambda \times (T/4)] \quad (1)$$

where T represents the number of steps, and $T/4$ is the average number of rows by which a pathogen (particle n) moves downward in the model grid after T steps. According to the exponential decay principle (see Appendix A.1), $\lambda = \ln(2)/\text{half-life of the pathogen}$.

a. Schematic diagram of pathogen spread in a room at three different moments



b. Schematic of a 2D model representing the three moments above



c. Schematic of a 3D model with two dimensions for particle spread

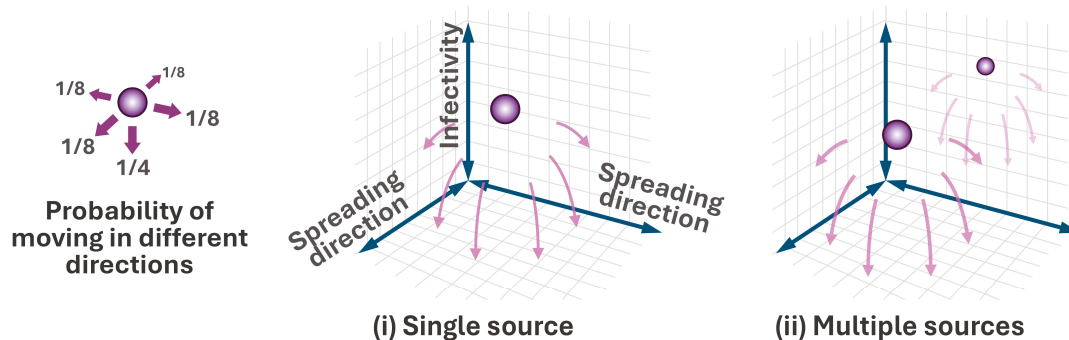


Figure 1. Modeling of pathogen spread. (a). Schematic diagram of pathogen spread in a room at three different moments. (b). Schematic of 2D model representing these three moments. (c). Schematic of a 3D model with two dimensions representing the spread of pathogen-laden particles. Each purple arrow, labeled with its corresponding fraction, is used to indicate the direction of particle movement with the given probability. The level of transparency is used to represent the decay of each pathogen-laden particle.

Model of 3D grid: As shown in Figure 1c, the model can be extended to a 3D grid, where the two horizontal axes represent the spatial spread of particles, and the vertical

axis represents particle degradation. In this framework, we assume that each particle has an equal probability of moving one cell downward ($1/4$), horizontally along the x -axis ($1/8$ left and $1/8$ right), along the y -axis ($1/8$ forward and $1/8$ backward), or remaining stationary ($1/4$).

2.1.2. Infection Risk in an Indoor Environment

Assume that an individual, upon touching a surface or inhaling an air volume (i.e., site n), acquires all the pathogen particles present, and any single particle retaining infectivity can result in the individual's infection. Then, the infection probability of site n (p_n) for this individual can be calculated using Equation (2).

$$p_n = 1 - [(1 - S_{n1})(1 - S_{n2}) \dots] \quad (2)$$

where $S_{n1}, S_{n2}, \dots$ denote the survival rates of particles 1, 2, $\dots$ at site n , as calculated by Equation (1).

Equation (2) reflects a nonlinear relationship between pathogen viability and infection probability. When an individual randomly touches a surface or inhales an air volume (e.g., site n) with a touch or inhalation probability q_n and acquires all the pathogen particles present, the average infection probability in the entire setting can be calculated using Equation (3).

$$P = \sum_{n=1}^N (p_n \times q_n) \quad (3)$$

where P reflects the ideal infection risk per unit occupancy time in an indoor environment at a specific moment of pathogen transmission, and $\sum_{n=1}^N q_n = 1$. Although P should be a function of λ and T , we do not specify this in Equation (3). In practice, individuals can stay in a setting for a period during which they touch multiple surfaces (or inhale multiple air parcels). Thus, the practical infection risk should be calculated as described in Equation (4).

$$P_R(e, \lambda, T_1, T_2) = 1 - \prod_{T=T_1}^{T_2} [1 - e \times P(\lambda, T)] \quad (4)$$

where e denotes the infective efficiency of a pathogen particle once it has been acquired via touch or inhalation, and P_R is the practical infection risk for an individual during time steps T_1 – T_2 . Thus, if a group of people is exposed to pathogens for the same period, the maximum-likelihood value of the attack rate (A)—defined as the ratio of infected people to the total number of people in the group—is equivalent to the practical infection risk for an individual, i.e., $A = P_R$.

Different from the ideal infection risk P in Equation (3), the term A (or P_R in Equation (4)) relates specific situations with a given exposure period (T_1 – T_2), infective efficiency (e), and pathogen viability (λ). Of these parameters, λ determines the timing of the crest, while T_1 – T_2 and e along with the particle numbers are approximately proportional to P_R and have an equivalent effect such that an increase in one can compensate for another.

2.2. Simulation of Pathogen Spread

2.2.1. Particle Spread in Idealized Grids

To examine the effects of model parameters, we simulated particle spread in idealized grids. First, a 2D grid with 21 columns and an infinite number of rows was established. Particles—equal in number to the columns—were released from the central top cell at step 0, spreading outward and downward while degrading with a half-life of 3 rows.

We further tested 3D grids with 11×11 columns and infinite rows, introducing varying particle densities (equal to 1, 4, 16, and 64 times the number of columns) and multiple release points (2, 4, and 8 sources randomly located in the top row). As a comparison, continuous sources were simulated by releasing one particle per step in 2D grids, or six per step in 3D grids, for the first 300 steps, maintaining a consistent particle-to-column ratio between the 2D and 3D grids.

Each scenario was simulated 10 times in parallel. The average infection risk (P in Equation (3)) was recorded at each step to capture its evolution over time.

2.2.2. Pathogen Spread via Surface Touch

Pathogen spread via surface touch is simulated based on typical parameter values collected from real-life scenarios [18,19]. According to prior studies, in a household or office environment [8,20,21], approximately 400–2000 surfaces are touched 10,000–200,000 times by occupants every day (as summarized in Table 1 in Zhao et al. [21]). Therefore, we assume that a source patient releases virus-laden particles at a single moment in an indoor environment. These particles gradually spread across 500 surfaces via surface touch, with each surface being touched an average of 50 times daily. Viruses on fomites have been observed to decay by 1–3 orders of magnitude over a period of two days to six weeks [22]. Accordingly, the pathogen's half-life on these surfaces is set to one day.

Based on the aforementioned parameters, we set the model grid to have 500 columns representing the 500 surfaces. As each surface is estimated to be touched 50 times per day, the pathogen particles randomly move left, right, or downward or stay stationary for 50 steps/day and consequently drop 12.5 rows daily on average. Thus, the value of λ in Equation (1) is calculated to be $\ln(2)/12.5$. We simulated the release of 500 pathogen-laden particles from the top-center grid cell. Step-by-step simulations were performed to track the variation in infection risk (P in Equation (3)).

We also changed the number of surfaces (N) to 250 and the pathogen's half-life to two days ($\lambda = \ln(2)/25$) to observe how the spreading scope and degradation rate affect the infection risk.

2.2.3. Pathogen Diffusion in Indoor Air

Pathogen diffusion in indoor air is simulated for typical settings. For a room with an area of 30–100 m² and a height of 3–4 m [8,20,21], the air volume is approximately 90–400 m³. The typical airspeed for minimal comfort is around 0.15 m/s [23]. Therefore, we assume that pathogen-laden particles are continuously released by a source patient over 2 h in a 100 m³ room, with an average indoor airspeed of 0.1 m/s. The room is divided into 100 air parcels of 1 m³ each, which exchange pathogens every 10 s. The pathogen's half-life in the room is set to 30 min [23] considering the effects of air ventilation (with a typical air-change rate of 0.5 h⁻¹ [24]), particle deposition (with deposition rates of 2–4 h⁻¹ in residential houses [25]), and microbial death (decay by 1–3 orders of magnitude over a period of 2 days to 6 weeks [22]).

Based on the aforementioned parameters, we set the model grid to have 100 columns, representing the 100 air cubes, each with a volume of 1 m³. An airspeed of 0.1 m/s yields a particle exchange interval of 10 s between adjacent air cubes. Therefore, each simulation step corresponds to 10 s, resulting in 360 steps/hour. The decay rate λ is thus calculated as $\ln(2)/45$. We simulated the gradual release of 720 pathogen-laden particles over a two-hour period, with one particle released every 10 s (i.e., one particle per step) from the top-center grid cell. Step-by-step simulations were conducted to track the variation in infection risk (P in Equation (3)).

We also adjusted the release rate to one particle every two steps (20 s) over a four-hour period in order to observe its effect on infection risk. Note that a minimal airspeed of 0.1 m/s was set in the simulation to capture the most prominent potential crest of infection risk. A higher airspeed would lead to faster particle dispersion and an earlier crest that would merge with the source release period and become unobservable.

2.3. Model Validation

To validate the new model, we collected reports of disease outbreaks in confined spaces, which describe situations where multiple groups of visitors, without inter-group contact or overlapping visiting times, were infected after successive visits. Using Google Scholar, we searched for case studies published before 2024 using keywords such as “outbreak”, “attack rate”, “multiple groups”, and “infection.” Only four outbreaks, all caused by norovirus, met the criteria, as shown in Table 1.

Table 1. Reports of indoor disease outbreaks involving multiple groups of visitors without inter-group contact or overlapping visiting times.

No.	Study	Location	Time	Onset/Total Interviewed People
I.	Kimura et al., 2011 [26]	A hotel	2–10 December 2006	345/approx. 2700
II.	Isakbaeva et al., 2005 [27]	A cruise ship	16 November 2002–4 January 2003	587/approx. 20,000
III.	Thornley et al., 2011 [28]	A flight cabin	13–18 October 2009	27/63
IV.	Jones et al., 2007 [29]	Four houseboats	9–29 May 2004	20/27

We use the attack rate A as a measure of infection risk in an indoor environment. Under every setting listed in Table 1, the pathogen attack rate is calculated separately for each of the consecutive visiting groups. Thus, the temporal variation in the infection risk and the potential crest of infection risk can be observed.

We assume that each setting includes 500 surfaces and an equal number of pathogen-laden particles. The maximum-likelihood attack rate via fomite route is also fitted based on Equation (4) with different e and λ values. Reasonable values of e and λ obtained from the simulation can be considered as a validation of fomite transmission.

2.4. Statistical Analysis

Each dataset of attack rate values was normalized so that the sum equaled 1, and its cumulative distribution was then calculated. The Kolmogorov–Smirnov (KS) test was conducted to compare the paired datasets from model prediction and empirical measurements in order to validate the new model. A p -value greater than 0.05 was considered indicative of no significant difference between the two distributions.

3. Results

3.1. Model Framework and Simulation Insights

The simulation results (Figure 2) show that for instantaneous pathogen sources (Figure 2a), the infection risk (P , as defined in Equation (3)) exhibits a crest over time, where time is measured in simulation steps. The timing and magnitude of this peak are independent of the grid dimension and column number. Comparing plots I and II, higher dimensions and finer grids improve prediction accuracy as they reduce variation across simulations. In fact, the half-life of the infectious particles primarily affects the infection risk after the crest, with minimal influence on the timing of the crest. In contrast, higher particle concentrations

increase the average infection risk, and multiple sources accelerate spatial spread, leading to an earlier crest.

When particles are released continuously over time (Figure 2b), the infection risk forms a plateau. The duration of this plateau is determined more by the release behavior than by environmental spread. Once the release stops, the risk rapidly declines.

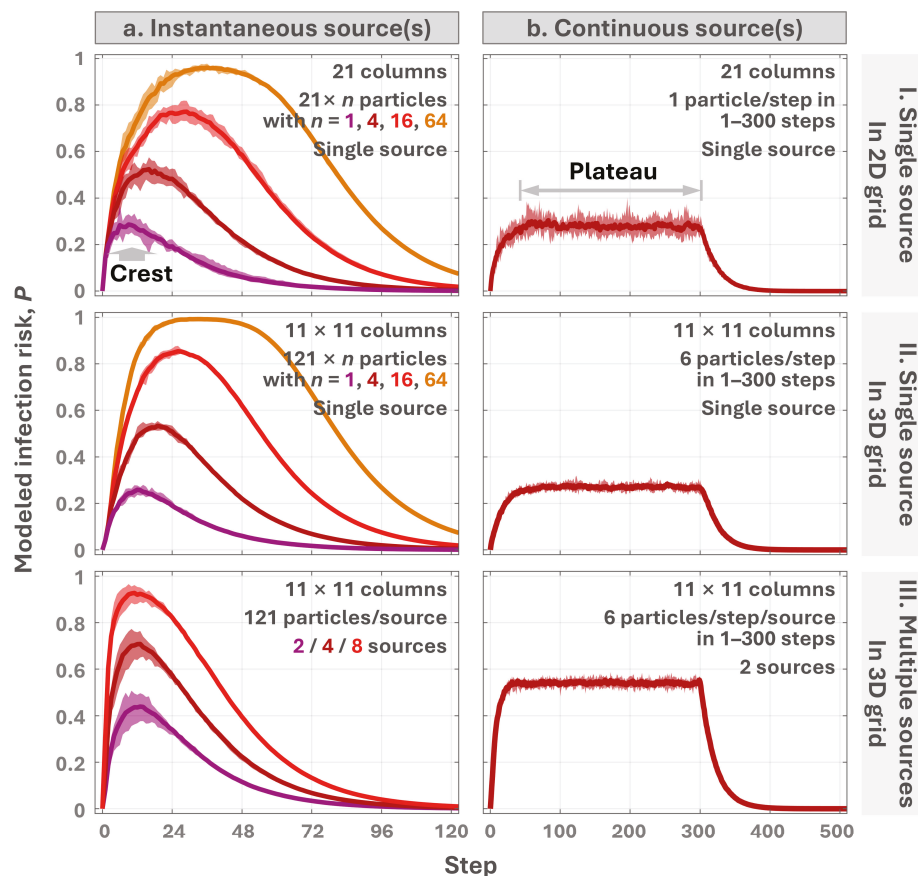


Figure 2. Simulations of particle spread in idealized grids. Particles equal to or 4, 16, or 64 times the number of columns were released from (a) instantaneous source(s) at step 0 or (b) continuous source(s) over steps 0–300 and dispersed within a 2D or 3D grid while degrading with a half-life of 3 rows. Parameter settings are indicated in each plot. Each curve represents the mean of 10 simulations, and the shaded area shows the full range of these simulations.

These two release modes reflect key differences between surface-touch and airborne transmission. In surface-touch scenarios, infection risk is mainly governed by the spread and persistence of particles. In contrast, airborne transmission exhibits infection risk that closely follows the active release period due to rapid air diffusion. Accordingly, we further analyze these two transmission routes within representative indoor environments.

3.2. Temporal Variation in the Infection Risk for Surface and Air Pathogens

The grid-based model was used to simulate pathogen transmission via indoor surfaces and air. In each simulation, the parameters were set to represent typical real-life scenarios. Figure 3 presents the results from 10 simulations for each scenario. The infection risk curve is not smooth, with the crest containing multiple maxima. This variability is due to the random movement of pathogen-laden particles, which can change both the intensity and scope of the viral load.

For pathogen transmission on surfaces (Figure 3a), the highest infection risk may appear on the second day in some cases. A smaller network size (250 vs. 500 surfaces) and slower pathogen degradation (half-life of 2 days vs. 1 day) increase the infection risk, with the slower degradation rate also delaying the crest's occurrence.

In indoor air, pathogens degrade faster due to the nature of the airflow [4]. Consequently, the pathogen release time must be considered. Under such circumstances, the pathogen source is close to a continuous source, and the infection risk reaches a “plateau” (Figure 3b), the width of which is determined by the pathogen release duration. Once the infected individual leaves the room, the infection risk drops rapidly.

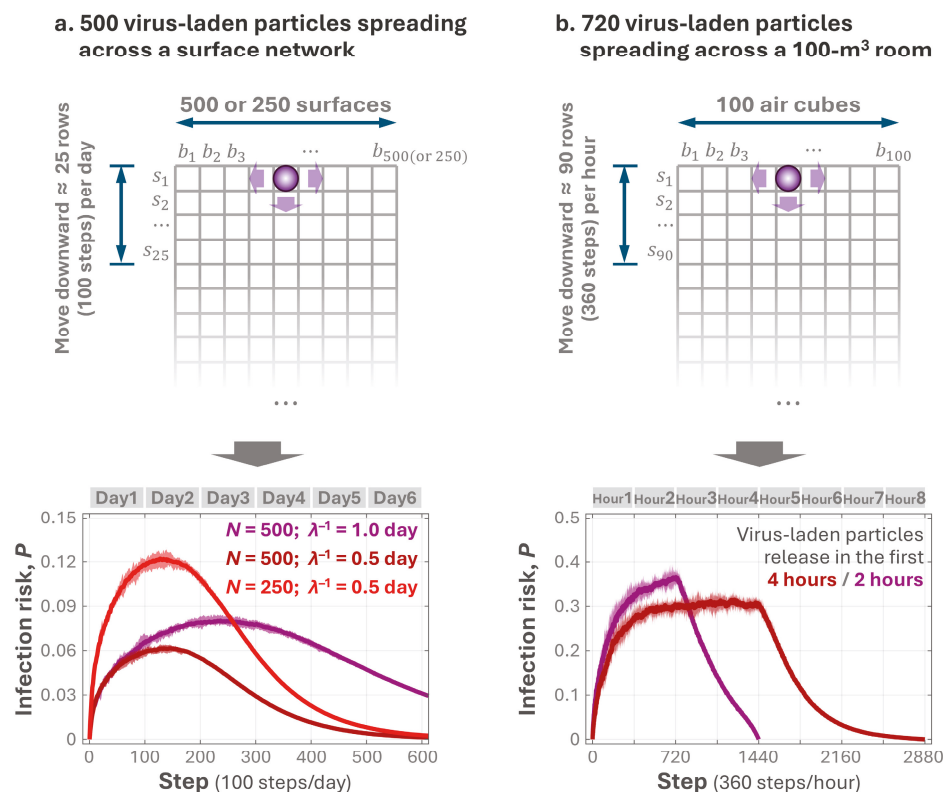


Figure 3. Modeling of spread of pathogen-laden particles via (a) surface touch and (b) indoor air. The simulated infection risk (P in Equation (3)) varied by simulation steps is presented for different scenarios. Each curve represents the mean of 10 simulations, and the shaded area shows the full range of these simulations.

3.3. Validation of the New Model

Data from four studies were used to validate the new model. The attack rates reported in these studies can be used as a measure of infection risk, although these two parameters are not equivalent. The rise-and-fall pattern can be observed from the reported attack rates. As the datapoints in Figure 4 show, cross-infection occurred among different groups of visitors in the indoor spaces, despite no inter-group contact or overlapping visiting times. Furthermore, of the seven curves in Figure 4, the crest appears at the first datapoint on four curves, at the second datapoint on two curves, and at the third datapoint on one curve. After the first several datapoints, the attack rate curves show a decreasing trend over time.

Pathogen transmission was also simulated for each scenario in Figure 4 based on the mechanism governing the new model. Different pairs of e and λ values were applied, and the maximum-likelihood attack rates were plotted, with different colors corresponding to the real data. As shown in Figure 4, of the seven pairs of attack rates tested using the K–S test,

six yielded p -values greater than 0.05, indicating strong agreement between the simulated and empirical datasets. For the seven simulations, the median of λ is 0.251, indicating that the pathogen has a half-life of 2.76 days. This value falls into the ranges commonly reported in the literature [22]. In contrast, if we simulated the pathogen transmission based on the parameters of airborne route, the λ value would deviate substantially from normal range.

In Case IV in Figure 4, the downward trend of the curve is less evident. This could be partially due to the small sample size (Table 1). More importantly, each group was exposed to the established settings for nearly a week, which exceeds the symptom onset time. Thus, more visitors may have become sources of infection before leaving, and some visitors may have been infected by secondarily infected individuals. Such infection chains can cause prolonged outbreaks [30,31], where infections can proliferate over months or cause epidemics. In such circumstances, the pathogen transmission routes are difficult to identify based on the new model.

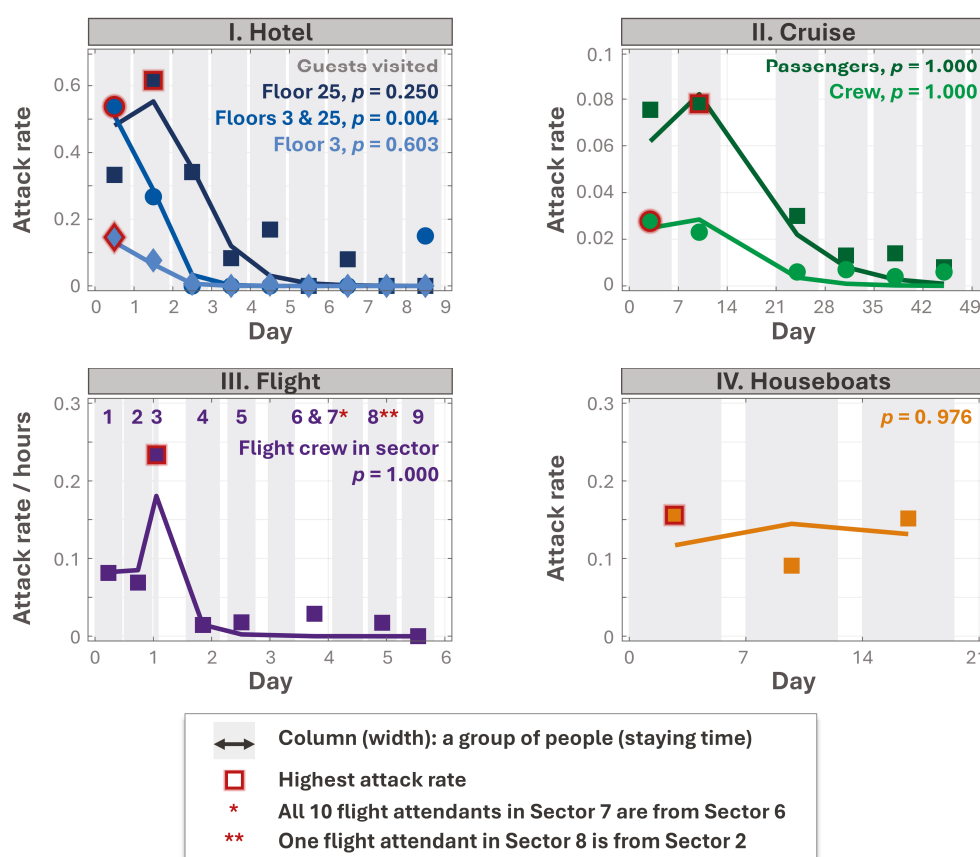


Figure 4. Attack rates per unit time for different groups in four indoor norovirus outbreaks (listed in Table 1). The infections occurred without inter-group contact or overlapping visiting times, except for one instance (Flight Sector 8 in Case III). For each data group (distinguished by colors), the maximum-likelihood attack rates via fomite route were calculated using the new model and are presented by the curves in corresponding colors. In Case III, “attack rate/hour” is used as the parameter for the vertical axis because of the different visiting periods of the different groups. p -values from Kolmogorov–Smirnov tests are presented to evaluate the agreement between simulated and measured attack rates.

4. Discussion

4.1. Mechanism of the Rise-and-Fall Pattern

The rise-and-fall pattern proposed in this study differs from conventional epidemic curves in that it arises from the nonlinear variations in both the intensity and spatial extent (breadth) of the spreading agent, which are typically independent of each other. This independence allows the use of a 2D or higher-dimensional grid-based model for simulation. The nature of such a rise-and-fall pattern is like a firework explosion, during which a crest of illumination occurs at a certain moment after its initial explosion (see Appendix A.2). In contrast, traditional SEIR-based epidemic curves and those shaped by periodic effects in self-organizing systems may follow mechanisms distinct from that of this study.

4.2. Conciseness and Flexibility of the New Model

This study introduces a concise model to capture the temporal variation in infection risk during pathogen transmission. It focuses on the changes in the pathogen survival and spatial diffusion while deliberately omitting other variables. The model is self-organizing and employs minimal mathematical expressions to constrain state transitions. Its transmission rules can be flexibly adjusted to represent various scenarios. Additionally, this model allows customized behavior settings for specific sites (e.g., a surface or an air cube), individuals, and dispersal agents (e.g., pathogen-laden particles), offering greater flexibility than most existing physical models.

Previous studies have employed a range of methods—such as computational fluid dynamics, Monte Carlo simulations, and agent-based models—to explore infection risk. These approaches primarily aim to evaluate key factors, including the role of high-touch surfaces in pathogen transmission [21], the impact of close-contact frequency between individuals [32], and the effect of handwashing frequency [33]. However, when distinguishing between transmission routes—particularly airborne and surface-touch transmission—such complex methods are often unnecessary. The simple mathematical model proposed in this study can clearly elucidate the fundamental differences between these two routes.

4.3. Occurrence of the Crest as an Indicator of Fomite Transmission

The occurrence of the rise-and-fall pattern during pathogen spread is determined by environmental parameters and specific spreading routes. For surface pathogens in indoor environments, the infection risk could remain for days or weeks, and the risk crest could occur in the days following the pathogen's release. In contrast, the crest of infection risk in aerial transmission occurs shortly after the pathogen's release due to the nature of aerial diffusion. Thus, pathogen release duration must be considered when simulating airborne transmission. The numerical simulations in this study are based on previous data related to surface-touch behaviors and air ventilation [14,34]. In contrast, the transmission of respiratory diseases in unconfined spaces or outdoor environments is rare because of the high efficiency of air ventilation [35].

Previous outbreak case studies have primarily focused on three transmission routes: airborne, fomite, and foodborne transmission. In airborne transmission, the infection risk usually exhibits a plateau and rapidly decreases after the source individual's departure [36–38]. For example, in a SARS-CoV-2 outbreak at a restaurant [19], all infected customers had overlapping dining times with the source individual. Although contaminated meals or water can cause infections across groups, the attack rate mainly fluctuates with the amount of food consumed [39–41]. Thus, the infection risks in airborne and foodborne transmissions seldom exhibit a rise-and-fall pattern. Conversely, fomite-based transmission likely causes

continuous infection across consecutive groups of visitors without inter-group contact or overlapping visiting times, with the infection lasting days or weeks; the infection risk (or attack rate) exhibits a declining trend after a probable crest.

All four outbreak cases illustrated in Figure 4 involved norovirus; thus, our analysis suggests that norovirus can be transmitted via fomites. Additionally, norovirus is known to spread through food and air [39,42]. For other pathogens, different facilities and materials have been identified as transmission agents, such as drainage systems for SARS-CoV-2 [6], cooling towers for *Legionella* [43,44], and skin-surface temperature probes in hospitals for *Candida auris* [45]. However, aside from norovirus, we did not identify any outbreaks in confined indoor spaces that exhibited a rise-and-fall pattern. In an outbreak at an aquarium in Australia [44], the number of infected individuals coincidentally exhibited a rise-and-fall pattern in relation to the visiting dates, although the specific attack rates were not reported. However, based on the aforementioned case study [44], the outbreak may have been linked to the operation period of a poorly disinfected cooling tower rather than fomite transmission.

4.4. Limitation and Potential Applications

Although the crest in attack rate described by our new model has been observed in empirical data (see Section 3.3), the number of usable outbreak samples remains very limited. This is because the crest becomes more pronounced only in outbreaks involving multiple groups of visitors without inter-group contact or overlapping visiting times. Nonetheless, the underlying infection risk may still exhibit a rise-and-fall pattern, even if this is not reflected in the reported daily attack rates.

We observed this phenomenon only in norovirus outbreaks, which may serve as preliminary evidence that norovirus is transmitted via surface-touch routes. In contrast, for pathogens of greater public concern—such as SARS, MERS, and SARS-CoV-2—we did not observe a similar crest in their time-varying attack rates. This suggests that surface touch is unlikely to be the dominant environmental transmission route for these respiratory viruses, consistent with the conclusions of many previous studies [1,14,46].

However, the absence of a crest in attack rate should not be interpreted as evidence against surface-touch transmission. It merely indicates that the probability of such transmission is likely low for these pathogens. The presence of a crest may serve as an early indicator of surface-touch transmission; however, more definitive conclusions regarding transmission pathways require further direct evidence.

Different spreading objects can exhibit the rise-and-fall pattern in their “effects” because of the nonlinear variation in the spreading extent and effect intensity over time [47]. For example, as a finite volume of a hazardous material leaks from a source to its surroundings, a larger residential area is polluted, while the concentration decreases with increasing distance from the source. Thus, the number of affected people can exhibit a peak with respect to the distance from the source. Similarly, many other spreading events can involve a crest in their “effects” with respect to both space and time, irrespective of whether the spreading objects are physical substances or information. The significance of this study lies in its use of a simple model to describe the rise-and-fall pattern. In the future, it could be used to characterize various diffusion events and identify their peak “effects”—such as the optimal sales range of a product or the most effective coverage of an advertisement. It may also be applied to suppress unwanted diffusion events, such as the spread of hazardous materials or pathogens.

5. Conclusions

This study identified and validated a distinct pattern of infection risk peaking at specific moments after a pathogen's release in indoor settings. This phenomenon was observed in simulating fomite transmission and corroborated by data on norovirus outbreaks. In typical indoor settings, airborne and foodborne transmission typically do not exhibit such patterns. Thus, this phenomenon may serve as a preliminary indicator for evaluating pathogen transmission routes in confined indoor spaces. However, the analysis of transmission routes for more pathogens requires additional data support. This research can help in understanding and managing indoor pathogen transmission. The new grid-based model is likely to be applicable in analyzing other types of spreading objects as well.

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Appendix A

Appendix A.1 Assumptions in Modeling Indoor Pathogen Spread

One-dimensional spread of pathogens. We applied a grid-based model to simulate pathogen spread, with one or two dimensions representing spatial propagation. In reality, the network structures underlying both surface and airborne transmission are complex. However, the efficiency of pathogen spread is more strongly influenced by the transmission medium than by the structural complexity of the network. Based on typical indoor air change rates (ACH), airborne particles generally decay within a few hours, while surface pathogens can persist for days or even weeks due to relatively low transmission frequencies and slower decay rates. Therefore, focusing on the fundamental characteristics of different transmission routes—while abstracting away from specific environmental structures—enables us to better capture the core dynamics of infection risk.

First-order decay of pathogens. The emergence of the rise-and-fall pattern remains unaffected, regardless of whether we adopt a linear decay, first-order decay (i.e., exponential decay), or second-order decay.

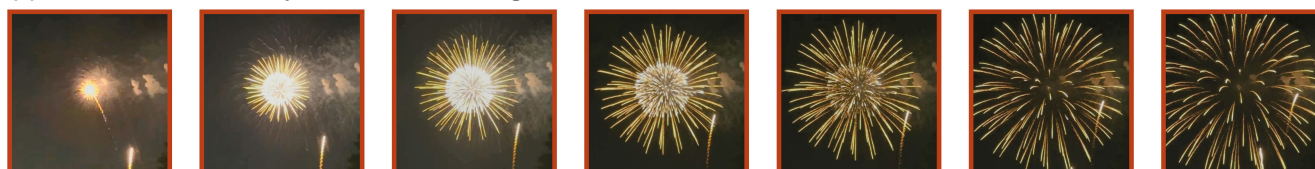
Here we chose first-order decay because it is a canonical form of decay where the relative proportion degraded per unit time remains constant. This type of decay is widely used to describe radioactive decay, the deterioration of consumer goods, and other scenarios. It also best matches our research context—assuming indoor environmental conditions remain constant and pathogen particles do not interact. In this situation, the theoretical degradation of pathogen particles should follow first-order kinetics. In contrast, if we suppose that two pathogen particles interact (e.g., by competing for limited resources) and lead to the elimination of one, then the pathogen degradation would follow second-order decay, similar to the mechanism in the Malthusian population model.

Appendix A.2 Firework Explosion

The crest of infection risk resembles a firework explosion: As a firework explodes in the sky, the illuminated area expands, while the brightness per unit area decreases from the point of the explosion. Due to the nonlinear variations in the firework's intensity per unit area and its expanding scope, the visual impact reaches its peak at a specific moment. Therefore, we can term this rise-and-fall pattern in spreading events the “firework phenomenon.” A comparison between the explosion of a firework and the spread of an indoor pathogen is illustrated in Figure A1.

In addition, the state of a spreading agent, when measured by entropy, also exhibits the rise-and-fall pattern, with the entropy value initially increasing and then decreasing. However, entropy differs slightly from the infection risk defined in this study. Assume that (i) an individual provides a single receptor receiving pathogens via touching surfaces or inhaling air and (ii) the receptor has a 100% chance of capturing a pathogen-laden particle. At this point, an individual's infection risk is independent of the pathogen concentration; hence, we ignore the cumulative effect of multiple pathogen particles on the infection risk as per the assumption in Equation (2). The infection risk in such cases is equivalent to Shannon entropy.

(a) Process of a firework explosion — the total brightness first increases and then decreases.



(b) Process of pathogen spread via surface touch — the infection risk first increases and then decreases.



Figure A1. An analogy between the brightness of a firework and the infection risk of an indoor pathogen.

(a) A series of screenshots in a video capturing a real firework explosion. (b) Three schematic diagrams depicting a spreading event of a pathogen in an indoor environment.

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Article

Rethinking Indoor Environment Demand and Control in Hot Summer from a “Yin Summer-Heat” Perspective in Traditional Chinese Medicine

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Abstract: Air conditioning is the most common and efficient measure against summer heat. However, overcooling issues exist widely in well-conditioned buildings, and the health risks and causes require further exploration. This study aims to rethink the indoor environment control and demand in hot summer from a novel perspective of yin summer-heat in traditional Chinese medicine. The core idea was to reflect health risks embodied in the indoor environment control that was oriented by the average comfort zone in air-conditioned buildings. Three research questions were explored, namely, indoor–outdoor environment features in hot summer, the heterogeneity of demands and behaviors, and relationships between personal attributes and lifestyles. Eleven field tests were conducted in residential buildings, together with experiments in an office building and three questionnaire surveys with 765 responses from 2020 to 2023 in China. Results showed that notable indoor–outdoor environment gaps appeared due to air conditioning. Yin summer-heat symptoms, such as a heavy feeling in the body, were reported by individuals of vulnerable constitutions even in neutral air-conditioned environments. In addition, Chinese medicine theories, including pathogenic factors, constitutions, and health preservation principles, worked well to interpret diverse environment perceptions, demands, and adaptive behaviors. These findings will add to the scientific basis of wellbeing in indoor environments.

Keywords: indoor environment; overcooling; thermal adaptability; yin summer-heat symptoms; constitutions in Chinese medicine

1. Introduction

1.1. The Increasing Usage of Air Conditioners and the Narrowing Comfort Zone

Climate change has permanently changed summer characteristics, with extreme heat events occurring globally from tropical to temperate climates. Almost half of the global population are exposed to extreme heat currently [1]. Hot ambient temperature and associated heat stress can increase mortality and morbidity [2], e.g., causing cardiovascular, respiratory, cerebrovascular, and endocrine diseases, etc., in addition to heat-related illness [1,3]. Moreover, recent studies highlighted the significant impact of heat stress on human mental health, including an increased suicide rate with high temperature [4], anxiety disorders in hot and humid environments (mouse experiment) [5], and less perceived happiness on hot days [6].

Globally, space cooling has been one of the fastest-growing end-uses in recent years against such hot summers [7,8]. However, long-term AC use reduces occupants' thermal

adaptability, leading to a preference for lower temperatures and a relatively stable environment. Alnuaimi et al. [9] carried out a large-scale study based on a global database including over 90 thousand responses from 27 countries in warm climates across thirty years. It was found that 17% of the occupants were overcooled according to their TSVs (Thermal Sensation Votes) and TP (Thermal Preference). The comfort temperature was 22~25 °C and the observed temperature was lower by 0.4~3.0 °C in summer. “Well-conditioned” buildings enable a tight control of indoor climate, where occupants become less tolerant of discomfort, compared with naturally ventilated buildings. A field study in Brazil with 20 university students [10] showed that occupants with air conditioning exposure preferred cooler environment, though their TSV was similar to those without, even at an indoor temperature of 24.5 °C. A field study in China [11] examined the neutral temperature of naturally ventilated (NV) and split air-conditioned (SAC) buildings based on 465 and 345 responses from healthy students. The NV group had a higher neutral temperature of 26.2 °C, compared to 25.5 °C of the SAC group. A large-scale study in Australia [12] analyzed occupant’s thermal sensitivity using the Griffith Constant, i.e., a coefficient of changes in TSV by temperature, based on 11,500 samples in different types of buildings. The G value of occupants in naturally ventilated spaces (0.208/°C) was about half of those in air-conditioned environments (0.440/°C), indicating less sensitivity to indoor thermal environments.

In addition, more exposure to HVAC systems suggests higher thermal expectations. A 15-month field measurement with 41 Australian households [13] found that heavy users (over 500 h annual HVAC system usage) maintained a significantly higher temperature in winter in the bedroom (20.9 °C, 0.6 °C higher) and living room (21.4 °C, 1.4 °C higher) than medium or light users. A field study in Brazil [14] with over 3000 responses in naturally ventilated classrooms (air temperature 26.9~31.6 °C) had similar findings. Students who routinely spend over 12 h per day in an air-conditioned environment reported the most warm or hot feelings (41%) and strongest needs for cooling (78%), compared with less exposed or unexposed students. A climate chamber study [15] investigated the physiological acclimation of 60 healthy young people when the temperature ranged within 20~32 °C and the humidity was suitable. Half of the participants were from central air-conditioned (CAC) buildings and the other half from SAC buildings. The CAC group reported a significantly higher mean skin temperature under the non-neutral conditions than the SAC group. Also, the 90% acceptable standard effective temperature (SET) was 24.8~27.4 °C and 24.4~30.1 °C for the CAC and SCA groups, with the range of the SAC group being wider by 3.1 °C.

As pointed out by de Dear [16], the comfort zone has become progressively narrower over the past seven decades. Short-term and long-term thermal experiences vary from seconds to years, impacting thermal sensation, comfort, acceptability, and expectations in different manners [17]. Individuals would easily and quickly improve their thermal expectations but take more effort to lower them, as found by a study with people with varied long-term thermal experiences in China [18]. Prolonged exposure to a stably cool indoor environment would restrain humans’ adaptive capacity, while a dynamic environment was more beneficial [17]. Thus, a more relaxed thermal comfort zone is called for, as it was in the adaptive model in ASHRAE Standard 55 since 2004 version [16], to encourage more climate-adaptive practices. Therefore, the narrowing thermal comfort zone has led to a preference for prolonged stays in air-conditioned environments, and the associated health risks have increasingly become a focus of research attention.

1.2. Health Risks Associated with Air-Conditioned Environments

Escalating health risks are associated with such air-conditioned environments due to their stable indoor conditions. Despite diseases caused by polluted AC systems

and air contaminants [19,20], studies since the 2000s have found that sickness symptoms still appear in non-problem AC buildings, with a higher prevalence than NV buildings. For example, Muzi et al. [21] compared self-reported irritative symptoms of 198 and 281 employees in AC and NV buildings in Italy and found more ocular, upper airway, and cutaneous symptoms in the AC than NV group, with 29.8% (AC) and 14.9% (NV), 25.3% and 9.6%, and 14.1% and 3.6% for the three symptoms, respectively. Wong and Huang [22] investigated the bedroom environment of three residential buildings and SBS symptoms of 105 occupants in Singapore. Occupants sleeping with operating air conditioners mostly presented SBS symptoms. Also, more symptoms appeared when they turned from NV to AC. Cao et al. [23] examined twelve categories of discomfort related to neural, digestive, and respiratory systems, and the skin and mucous membranes of 2595 responses in China. Significant differences were found in the prevalence of certain symptoms among people exposed to an air-conditioned environment for different hours per day. Unsurprisingly, overcooling and sickness issues exist in buildings at the same time. A field test [24] with eleven office buildings and 599 occupants in Thailand, Indonesia, and Singapore during 2017 to 2018 found that the mean operative temperatures were all lower than 24 °C indoors. Self-reported symptoms, such as headache and fatigue in the body, were most severe and gentle at an operative temperature of 21.0 °C and 24.5 °C.

Interdisciplinary exploration has provided medical evidence on the working mechanism of indoor thermal environment factors on human health. A comprehensive review by Liu et al. [25] showed that temperature was the most important and frequently discussed factor, followed by humidity and air movement, which were associated with SBS, metabolic symptoms, and respiratory diseases. Chamber tests with 20 people [26] found that the NV group presented stronger physiological regulatory capacity for heat shock from 26 to 36 °C, particularly showing a more rapid skin temperature response, a lower “chest–feet ΔT ”, a higher volume of sweat, a smaller rise in HRV (Heart Rate Variability), and a higher level of HSP₇₀ (Heat Stress Protein 70) than the AC group. In addition, experiments with 26 healthy subjects at ambient temperatures of 24 °C and 28 °C and air velocities of 0 m/s and 1 m/s [27] displayed that cold draft would restrain upper respiratory mucosal immunity. In particular, the concentration of secretory immunoglobulin A (S-IgA) remained low after one day in the 24 °C and 1 m/s case, indicating a prolonged adverse impact.

Health risks also exist for temperature steps or sudden temperature changes, which are quite common in indoor–outdoor space switching. A climate chamber study with 16 persistent allergic rhinitis patients and 16 control subjects in Brazil [28] found that the former were more susceptible to the sudden temperature change of 14 → 26 °C. Another laboratory experiment in China [29] with 24 subjects exposed to temperature step changes between 32 and 37 °C, 26 and 37 °C, and 22 and 37 °C reported a notable shrink in perspiration at sudden temperature drops of 12 and 15 °C. A list of biomarkers were found to reflect the impact of sudden temperature changes on human immune, thermal metabolism, respiratory, and cardiovascular systems [30], including interleukin-6 (IL-6), oral and skin temperature, heart rate, and heart rate variability. Also, a significant and strong correlation existed between such physiological parameters and self-reported symptoms.

1.3. “Yin Summer-Heat” Perspective in Traditional Chinese Medicine

Evidently, reliance on or even “addiction” to air conditioning not only increases energy use in buildings but also leads to the sickness of occupants. How, then, should we manage indoor comfort needs and maintain the health of occupants in air-conditioned buildings? It is not easy to answer this question, since a large number of variables are involved, which

are coupled in a complex manner and present varied features in different time scales. The “yin summer-heat” concept in traditional Chinese medicine (TCM) may help with this question.

The idea of cold-induced illness in hot summer, i.e., “yin” symptoms in TCM, were proposed by Jiegu Zhang (Jin Dynasty, 1131~1234), which were common among inactive people who got away from hot weather inside massive buildings, in comparison with heat-induced illness in summer, i.e., “yang” symptoms, for active people walking around outdoors. The expression of “yin summer-heat” was formally proposed by Jingyue Zhang (Ming Dynasty, 1563~1640) as typhoid in hot summer [31], which was common but quite covert.

Modern Chinese medicine interprets yin summer-heat from the perspective of chills in summer [32]. Clinical observations of 120 patients [33] confirmed that yin summer-heat illness was caused by the pathogenic factor of cold excess, usually combined with another pathogenic factor of dampness excess, resulting from overly taking cold drinks, shelter from the sun, etc. An epidemiological study of 203 patients with respiratory tract infections in summer in Beijing, China [32], showed that most patients usually caught a “cold” via air conditioners (152 ppl.), electric fans (11 ppl.), chill natural wind (16 ppl.), rain or shower (3 ppl.), or cold drinks (3 ppl.). The pathogenic factors could invade in two ways, i.e., from an external boundary such as the skin or the internal body such as organs, and induce different symptoms, e.g., fever, headache, anhidrosis, chills, and body aches, or vomiting, diarrhea, and abdominal pain, respectively [34]. Such pathogenic factors and symptoms work well to explain discomfort feelings in air-conditioned environments [35]. Chinese medicine theories and methods, e.g., substantial classic prescriptions targeting summer heat [36], are advantageous to avoid and treat such yin summer-heat diseases. The classification of common yin summer-heat symptoms is elaborated in Table 1, together with a comparison of the concept, symptoms, and pathogens among building-related illness, SBS, and yin summer-heat illness.

Table 1. Comparison of building-related illness, SBS, and yin summer-heat illness.

Concept	Symptoms	Causes
Building-related illness [19]	Physical: sick building syndrome Allergic: asthma, extrinsic allergic alveolitis, humidifier fever Infections: bacterial, viral, fungal	Air contaminants, etc.
Sick building syndrome [20]	Mucous membrane irritation: eye irritation, throat irritation, cough Neurotoxic effects: headaches, fatigue, lack of concentration Respiratory symptoms: shortness of breath, cough, wheeze Skin symptoms: rash, pruritus, dryness Chemosensory changes: enhanced or abnormal odor perception; visual disturbances	Air contaminants, ventilation, work organization, and host factors
Yin summer-heat [33]	Cold excess invading skin: aversion to cold, fever, headache and heavy head, limb tightness, soreness, no or poor perspiration, slight cough, upset, white and thin tongue coating, etc. Cold excess approaching lungs: persistent, clear sputum with thin foam, or white sticky sputum, may be accompanied by aversion to cold, fever, fatigue, weakness, thin or white tongue coating. Cold excess trapped in the body: abdominal distention, pain, nausea, vomiting, chest tightness, belching, poor appetite, loose and frequent stools, slippery tongue coating, may be accompanied by aversion to cold and limb soreness, etc.	The pathogenic factors of cold and dampness excesses

Notes: there are six TCM pathogenic factors, namely, wind, cold, summer heat, dampness, dryness, and fire that would invade the body if one were weak and cause illness [37].

Furthermore, constitutions in Chinese medicine would help to understand individual differences in environmental adaptability, responses to the pathogenic factors, and potential symptoms in air-conditioned environments, when gender or age are not the only reasons for individual disparities in thermal comfort [38]. Constitutions in Chinese medicine comprise a scientific and systematic description of the physiological and mental conditions of an individual, considering physical characteristics, common manifestations, psychological features and symptoms, etc. The national standard (ZYYXH/T157-2009 [39]) stipulates a series of questions about health status to evaluate an individual's constitution type. The questions adopted five-point Likert scales and the score of questions reflects one's tendency toward certain constitutions. The segmentation of constitutions has been calibrated and validated by interdisciplinary epidemical studies with more than 129 thousand people [40]. As illustrated in Table 2, there are nine constitutions, indicating different environment and social adaptabilities. An individual of a certain constitution will be vulnerable to corresponding TCM pathogenic factors and get sick in a given environment.

Table 2. Identification of nine constitutions in Chinese medicine and corresponding environment adaptabilities [39].

Constitution	Abbr.	Environment Adaptability and TCM Pathogenic Factors
Balanced constitution	BC	Strong adaptability
Qi-deficiency constitution	QDC	Vulnerable to wind, cold, summer heat, dampness excesses
Yang-deficiency constitution	YADC	Vulnerable to wind, cold, dampness excesses
Yin-deficiency constitution	YIDC	Vulnerable to summer heat, dryness, fire excesses
Phlegm dampness constitution	PDC	Poor adaptability to damp environment
Dampness–heat constitution	DHC	Poor adaptability to damp and hot environment
Blood stasis constitution	BSC	Vulnerable to cold excess
Qi stagnation constitution	QSC	Poor adaptability to rainy days
Inherited special constitution	ISC	Poor adaptability

1.4. Research Aim

Continuous extreme heat has brought great challenges to human beings living on this planet. Existing studies have evidenced that long-term exposure in air-conditioned environments is associated with a series of health risks, including weakened environmental adaptability and building-related illness, or more specifically, overcooled symptoms. However, there has been limited exploration of the complex links between the sensation, adaptability, and symptoms of the occupants in air-conditioned environments. Also, the so-called comfort environment is not necessarily healthy for modern urbanites who live and will still live in such well-conditioned buildings. TCM concepts, including yin summer-heat illness, which is associated with occupant's strong desire for cool feelings in hot summer (e.g., relying on air conditioners, cold showers, cold drinks, etc.), six pathogenic factors, and constitutions, could work well to explain the discomfort feelings and individual adaptability in air-conditioned environments. Therefore, this study aims to introduce the pathogenic factors and prevention strategies of “yin summer-heat” illness in masterpieces of Chinese medicine to rethink indoor environment control from the following aspects. Two questions are to be answered:

- Question 1: What are the features and health risks of the space where people shelter from heat?
- Question 2: How do environmental preferences and demands differ among diverse populations?

The findings of this study will facilitate human-centric demand-facing personalized indoor environment control strategies in hot summer.

2. Data and Method

To answer the questions above, field tests and questionnaire surveys were carried out from 2020 to 2023, in typical subtropical or temperate climatic cities in China, to explore how buildings and occupants adapt to or get rid of hot summer and potential health risks. In particular, this study did not manage to cover all disparities induced by geographical or demographic heterogeneity, but tried to point out and showcase the challenges faced by the “concrete forest” cooled down by numerous air conditioners in the modern city. The research framework of this study is displayed in Figure 1.

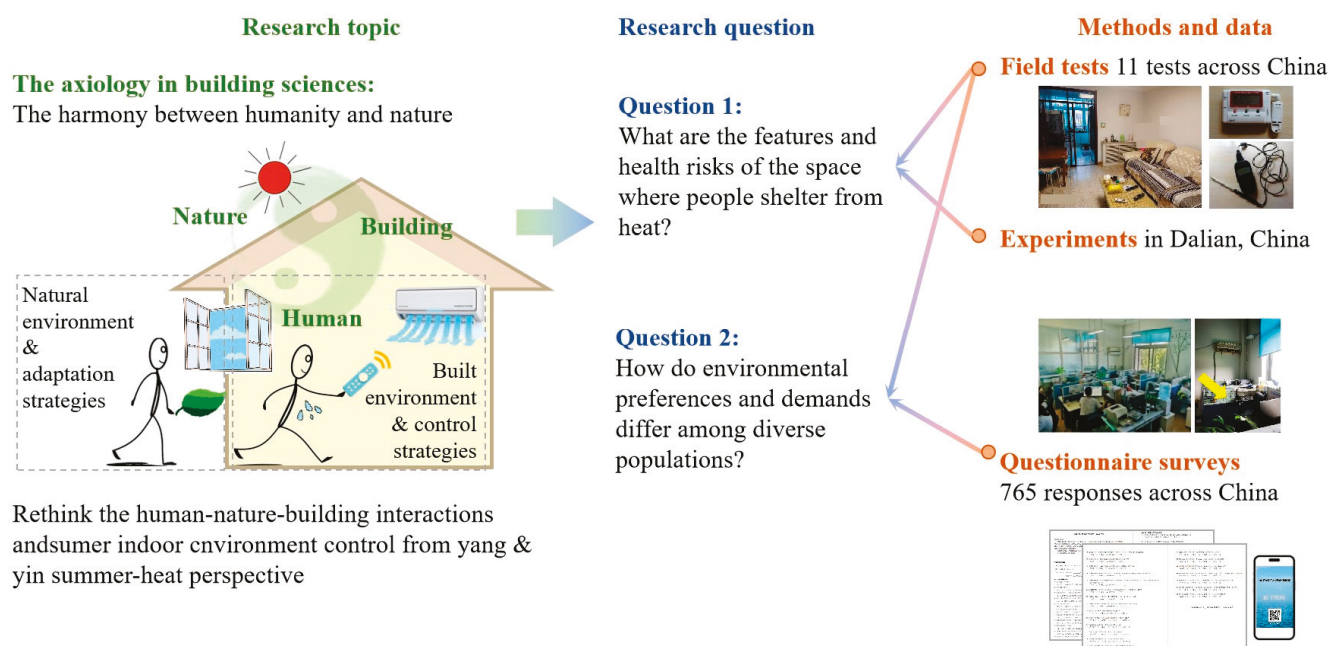


Figure 1. The research questions, methods, and data of this study (illustration by the authors).

2.1. Field Tests and In Situ Experiments

Eleven field tests were conducted in residential buildings to investigate the indoor and outdoor environment features in hot summer, as elaborated in Table 3. Participants of the field tests comprised young, middle-aged, and elderly people, who lived in Dalian, Nanning, Changsha, and Hong Kong in China. The participants maintained their lifestyles, including daily dress habits, during the tests. The monitoring in residential buildings lasted for three to fourteen days. All tests were conducted during extreme hot days of the year, with outdoor temperature and humidity being stably high, as shown in Figure 2. Existing studies showed that large temperature steps, low air humidity, and drafts appearing in the overcooled indoor environment, together with indoor–outdoor switching, would lead to discomfort feelings or illness of occupants [27–30,35]. Therefore, relevant indoor and outdoor environment parameters were obtained in the field tests. Thermal diaries and follow-up interviews were also applied to some participants to record their instant feelings and comments. In addition, the constitution of the participants of the field tests were identified. Typical meteorological year (TMY) data were also collected for targeted cities.

Furthermore, in situ experiments (Test L in Table 3) with young healthy staff in an office building on a university campus were employed to demonstrate how yin summer-heat risks appear in air-conditioned environments. The first experiment was to demonstrate

thermal experience related to the typical routine of “returning to work after lunch break” on hot summer days with typical summer clothing. In the experiment, as shown in Figure 3, two healthy young male participants staying outdoors walking or standing for at least 20 min, with shading from time to time to adapt to the outdoor environment, and then entered the building to work in the air-conditioned offices. The real-time thermal environment changes experienced by the two participants were recorded using portable sensors that they carried with them, together with clothes and activity levels during the experiment. Thermal images of the upper-body skin temperature of the two participants were recorded when just stepping into the office and one hour later. In the office, Participants A and B were seated nearby and away from the outlet of air conditioners. Therefore, Participant A would feel more breezy and cool airflows. In addition, Participant A and B were of DHC and BC constitutions, respectively. There were also follow-up thermal image experiments to investigate the physiological responses of staff of different constitutions when they worked in the air-conditioned office with their daily clothing. All participants had lived in the city for years and adapted to the climate.

Table 3. A summary of field tests and experiments during hot summer in 2020~2023 in China.

No.	City	Region	Building Type	Duration	Operation Mode	Occupants	Key Parameters
A	Dalian	N	Residence	31 July–6 August 2020	NV	2Y	T, RH
B	Dalian	N	Residence	31 July–6 August 2020	NV	1Y	T, RH
C	Nanning	S	Residence	29–30 August 2020	Mixed	2M+1Y	T, RH
D	Changsha	S	Residence	8–11 August 2022	Mixed	2E	T, RH
E	Changsha	S	Hotel	8–11 August 2022	Mixed	1E+1Y	T, RH
F	Hong Kong	S	Residence	19–28 July 2023	Mixed	1Y	T, RH
G	Hong Kong	S	Residence	19–28 July 2023	Mixed	1Y	T, RH
H	Dalian	N	Residence	31 July–6 August 2023	Mixed	2Y	T, RH
I	Dalian	N	Residence	19 July–1 August 2023	NV	1E	T, RH
J	Dalian	N	Residence	19 July–1 August 2023	NV	1E+1Y	T, RH
K	Dalian	N	Residence	19 July–1 August 2023	Mixed	2M	T, RH
L	Dalian	N	Office	20 July 2023	Mixed	Several Y	T, T _g , T _{skin} , RH, v, thermal images

Notes: 1. N and S in the column of “Region” refer to North and South China, which are mainly temperate and subtropical climates, respectively. 2. NV or mixed in the column of “Operation mode” means that the common indoor environment control approach is natural ventilation, or a combination of natural ventilation and air conditioning, respectively. 3. Y, M, and E in the column of “Occupants” refer to young, middle-aged, and elderly people aged below 45, 45~59, and 60 and above years old, respectively. 4. T, T_g, T_{skin}, RH, and v refer to air temperature, black globe temperature, skin temperature, air relative humidity, and air velocity. 5. Test A, B, E, and G included rainy days.

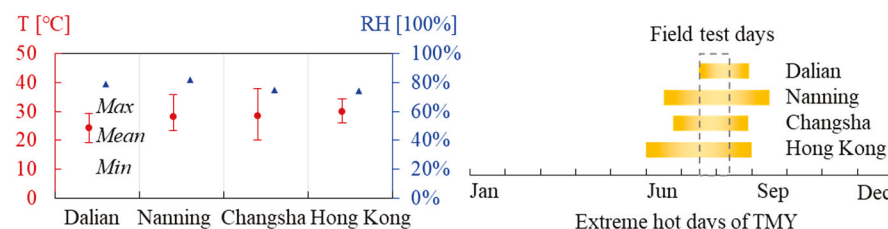


Figure 2. Outdoor environment of extreme hot days in TMY of the targeted cities.

The sensors used in the field tests were produced by reliable global suppliers with acceptable accuracy, and were calibrated regularly, such as temperature and humidity data loggers by T&D (e.g., TR-72Ui, ± 0.1 °C), air velocity meters by Kanomax (e.g., KA-12, ± 0.001 m/s), the thermal comfort meter by LSI-LASTEM (e.g., R-Log 7730, temperature ± 0.1 °C, relative humidity ± 0.1 °C, globe temperature ± 0.5 °C), and the infrared thermal imaging camera by FLIR (e.g., T420, ± 0.01 °C). Data analyses involved the calculation of

operative temperature (T_o), SET, the spectral analysis of air velocity, descriptive statistical analysis and regression analysis, etc.

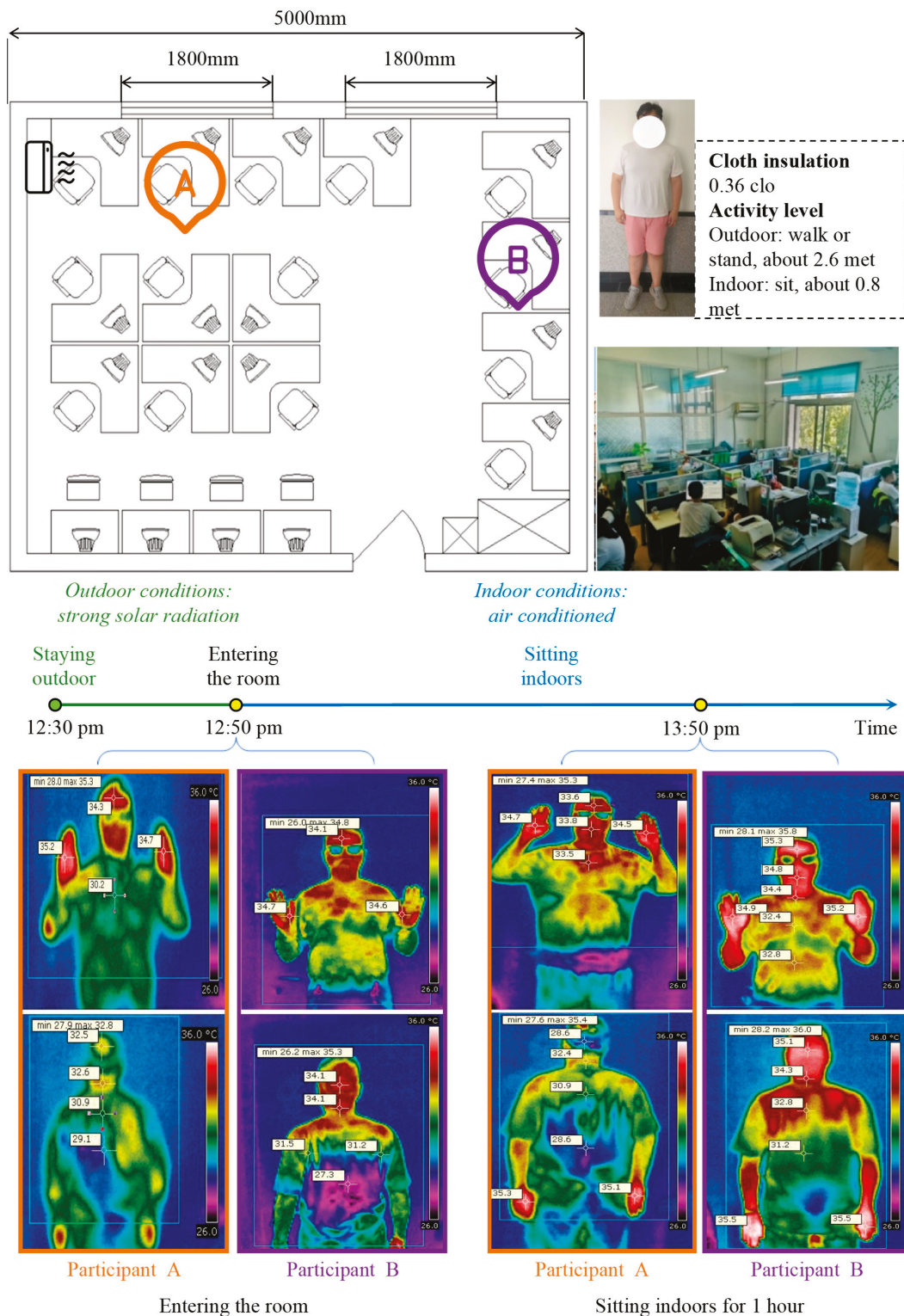


Figure 3. A typical outdoor-to-indoor routine and individuals' responses to environment changes on extreme hot days (Data from Test L).

2.2. Questionnaire Survey

Three questionnaire surveys were carried out during the hottest days in 2020, 2022, and 2023, as shown in Table 4. The 10 min surveys contained about twenty to forty questions in six aspects, namely, a. demographic and physiologic characteristics, b. cooling-related adaptive behaviors, c. wellness habits and TCM knowledge, d. indoor environment sensation, acceptance, and preference, e. illness symptoms indoors, and f. test of constitutions in Chinese medicine (taking another 5 min). The definition of and identification approach to constitutions were mentioned above in Section 1.3. Example questions are shown in Appendix A Table A1. Questionnaires were delivered by well-trained researchers to ensure data quality. The participants were recruited by both online platforms and site visits roughly in a snowball process. The participants were required to understand and answer the questionnaire correctly. Finally, a total of 765 valid responses were obtained, in which 157 responses provided additional constitution identification results. The sample size qualified for descriptive statistical analysis. The participants of the questionnaire surveys lived in different cities across China, from both the subtropical climatic zone with long summers and the temperate climatic zone with four distinct seasons (Figure 4). The map of China used in Figure 4 is sourced from the WPS software, Version 12.0 [41], and the data for Questionnaire I and Questionnaire II are derived from the author's research team [42,43]. There were 54.5% males and 45.5% females. Most participants ranged from eighteen to seventy years old (90.2%), and there were 30 and 45 responses from teenagers and the elderly aged more than seventy, respectively. The geographical and demographic diversity of the sample enabled comparative analyses of the varied demand, perception, and interaction with the built environment of different occupant segments.

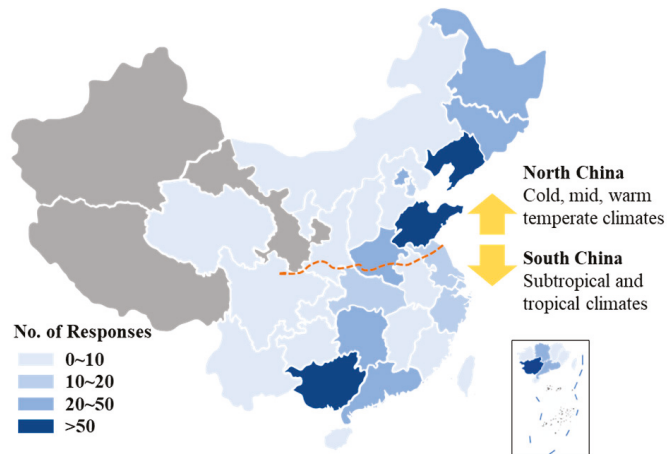


Figure 4. Regional distribution of responses of questionnaire surveys (the map was produced by WPS Office [41]; the data were collected by the authors in previous work [42,43]).

Table 4. A brief overview of questionnaire surveys during hot summer in 2020~2023.

No.	Survey Time	Regional Distribution	Building Type	Sample Size
I	August 2020	All in South China	Office	107
II	August 2022	Well distributed	Not specified	557
III	July 2023	Two thirds in North China	Not specified	101

3. Results and Analysis

Question 1 and 2 are examined in Sections 3.1–3.3, respectively.

3.1. Outdoor and Indoor Environment Features During Extreme Hot Period

Extreme hot days refer to a period when the outdoor temperature and humidity remain relatively high according to the typical meteorological year (TMY) data, as shown in Figure 2. First, the comparison of the TMY data before, during, and after the extreme hot period showcased the temporal changes in outdoor conditions to be adapted to by humans. Dalian was taken as an example, to which other cities showed analogous results. Statistical and analysis results of the TMY data are presented in Figure 5, including air temperature, apparent temperature, relative humidity, and absolute humidity. In particular, apparent temperature (T_s), was defined by Equation (1) in a tourism industrial standard [44] to reflect an individual's perceived outdoor temperature determined by air temperature, velocity, and humidity.

$$T_s = \begin{cases} T + \frac{15}{T_{\max} - T_{\min}} + \frac{RH - 70}{15} - \frac{v - 2}{2}, & T \geq 28^\circ\text{C} \\ T + \frac{RH - 70}{15} - \frac{v - 2}{2}, & 17^\circ\text{C} < T < 28^\circ\text{C} \\ T - \frac{RH - 70}{15} - \frac{v - 2}{2}, & T \leq 17^\circ\text{C} \end{cases} \quad (1)$$

where T refers to real-time air temperature [$^\circ\text{C}$]; $T_{\max}$ and $T_{\min}$ refer to the daily maximum or minimum air temperature [$^\circ\text{C}$], respectively; RH refers to real-time relative humidity [%]; and v refers to real-time air velocity [m/s].

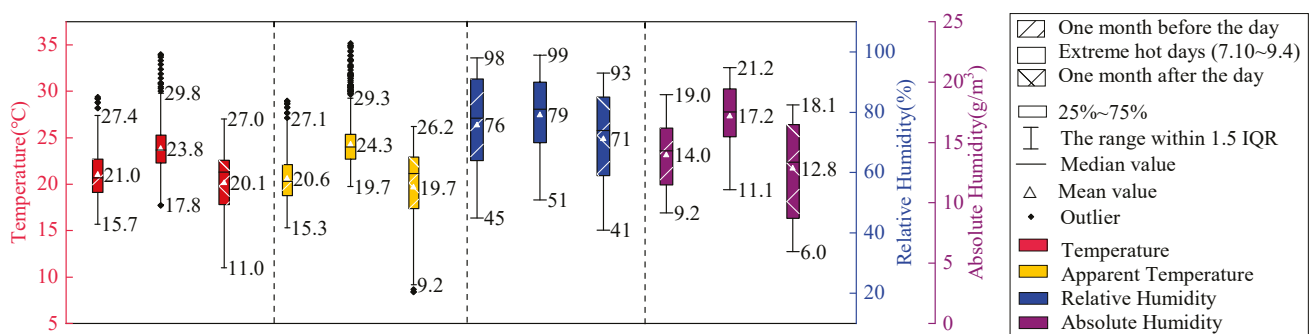


Figure 5. TMY data before, during, and after extreme hot days in the city of Dalian (data from Li et al. [45]).

Generally, the variation in outdoor temperature and humidity narrowed down during extreme hot days, and it would be continuously hot and humid outside with more gentle fluctuations. The increase steps of environment parameters were smaller than the decrease steps, indicating a milder emerging of heat and dampness but a slightly faster appearing of coldness and dryness, before and after extreme hot days, respectively. Specifically, air temperature and apparent temperature during extreme hot days in Dalian became separately 2.8°C and 3.7°C higher than the former month. This supported the necessity to consider the combined impact of climatic indicators on an individual's outdoor thermal feelings. In addition, absolute humidity became 3.2 g/m^3 and 4.4 g/m^3 higher during extreme hot days in Dalian. Meanwhile, relative humidity remained similarly high in the transition and hot days, suggesting the advantages of absolute humidity in describing changes in moisture. This study therefore prioritized absolute humidity in analyses.

Second, outdoor conditions during the extreme hot period were examined in different cities from North to South China, where the field tests were conducted (Figure 6). Though certain discrepancies existed in the TMY data and the field test records, the TMY data helped to understand the fundamental climate characteristics of different cities. The diurnal temperature range (DTR), i.e., the difference between day and night temperatures, was roughly 4.9°C , 8.0°C , 7.3°C , and 4.0°C separately for Dalian, Changsha, Nanning, and

Hong Kong in the TMY data during the extreme hot period. Moreover, the average TMY wind speed was 3.5 m/s, 2.2 m/s, 1.5 m/s, and 3.8 m/s in Dalian, Changsha, Nanning, and Hong Kong, respectively. Wind directions remained mostly the same or were more varied for seaside cities, but became to some extent converged in the inland city of Changsha.

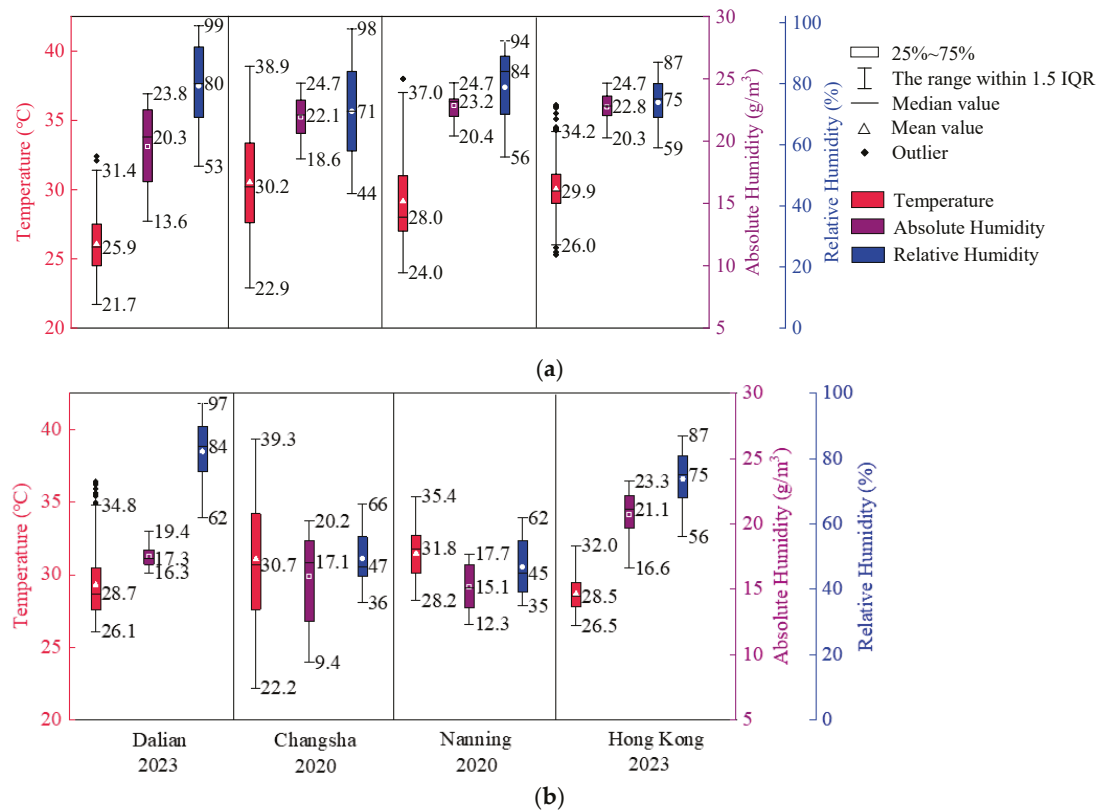


Figure 6. (a) TMY records during extreme hot days; (b) Outdoor climate conditions during field test periods in selected cities (data from Li et al. [45] and field test results. Extreme hot days were defined as in Figure 2).

Third, indoor conditions in Tests A~K on typical hot summer days are presented in Figure 7. As for air temperature, it was hotter outdoors for most of the time and the orientation of certain rooms and rainy days (e.g., in Tests A, B, F, and G) could explain other situations. The maximum real-time indoor–outdoor temperature differences in a day even reached 12.2 °C (Test J), which usually appeared in the daytime. Generally, in the daytime (10:00~16:00), the average real-time temperature difference between the outside and living room or study room was −1.0 °C~10.4 °C, while at nighttime (22:00~6:00) the difference between the outside and bedroom was −4.2 °C~7.2 °C in different tests. In addition, temperature gaps existed among different rooms. For example, a sectional analysis of air temperatures in different rooms in Test C, Nanning, at 17:00 on August 30 in 2020, showed varied indoor air temperature in the master bedroom (28.9 °C), second bedroom (25.6 °C), living room (30.1 °C), kitchen (31.0 °C) and dining room (31.9 °C) when it was 32.5 °C outside. The occupant would experience a sudden temperature drop of 6.9 °C when entering the second bedroom from the outside, and then a temperature rise of 6.3 °C when going to the dining room to drink water. That is, occupants would experience unignorable sudden drops or rises not only with indoor–outdoor space switching but also with movement between different rooms, indicating potential health risks related to temperature steps, as discussed in Section 1.1.

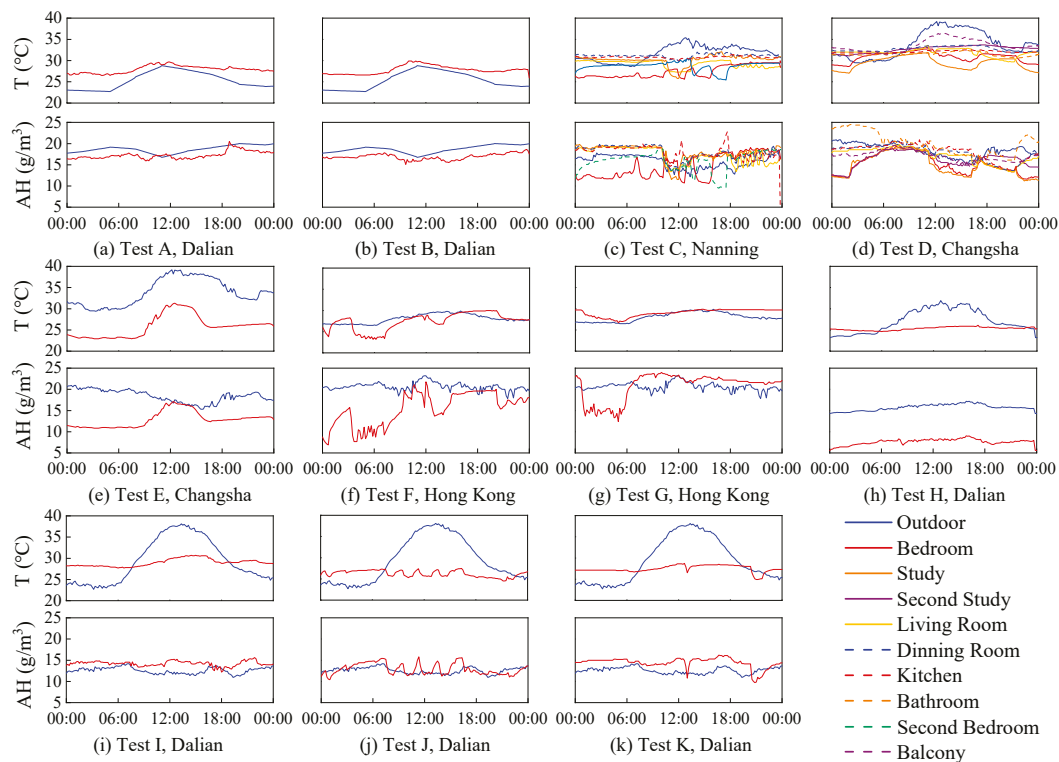


Figure 7. Indoor thermal conditions in monitoring periods. (a) Test A in Dalian; (b) Test B in Dalian; (c) Test C in Nanning; (d) Test D in Changsha; (e) Test E in Changsha; (f) Test F in Hong Kong; (g) Test G in Hong Kong; (h) Test H in Dalian; (i) Test I in Dalian; (j) Test J in Dalian; (k) Test K in Dalian. The houses were mostly occupied on the selected day (both sunny and rainy (Tests A, B, F, and G). Data from Tests A–K). T and AH refer to air temperature and absolute humidity.

As for air humidity, it is more humid indoors for most tests, except for Tests I, J, and K in Dalian. The maximum real-time indoor–outdoor humidity differences in a day reached 13.6 g/m^3 (Test F), excluding the regular high-humidity moments in daily life, such as cooking in the kitchen and showering in the rest room. Similarly, absolute humidity gaps existed not only between indoor and outdoor environments but also among different rooms, and the gaps could be larger in the latter situation. For example, the absolute humidity was 11.4 g/m^3 in the master study room and 11.7 g/m^3 in the bedroom, but 15.9 g/m^3 in the living room, 16.1 g/m^3 in the kitchen, 16.6 g/m^3 in the dining room, and 21.9 g/m^3 in the bathroom when it was 18.3 g/m^3 outdoors, for Test D, Changsha, at 22:30 on 10 August in 2022.

Summarily, extreme hot periods in Chinese cities featured a continuously hot and humid outdoor environment. Obvious indoor–outdoor and indoor–indoor gaps appeared in thermal environments during extreme hot periods in the observed cities, e.g., the maximum real time indoor–outdoor temperature and absolute humidity differences in a day reached $12.2 \text{ }^\circ\text{C}$ and 13.6 g/m^3 (Q1).

3.2. “Yin Summer-Heat” Risks in the Well-Conditioned Buildings

The in-situ experiments elaborated the process and stress of thermal accumulation in air-conditioned environments. Figure 3 displays the skin temperature changes of the two participants entering the air-conditioned office room from the sunny outside. Both of them experienced rapid changes in ambient micro-climates when entering the room, with the sudden drops in air temperature and absolute humidity being $8.3 \text{ }^\circ\text{C}$ ($35.8 \text{ }^\circ\text{C} \rightarrow 27.2 \text{ }^\circ\text{C}$) and 5.4 g/m^3 ($17.5 \text{ g/m}^3 \rightarrow 12.1 \text{ g/m}^3$). After resting for one hour, the forehead and chest

skin temperature of Participant A decreased by $0.7\text{ }^{\circ}\text{C}$ ($34.3\text{ }^{\circ}\text{C} \rightarrow 33.6\text{ }^{\circ}\text{C}$) and increased by $1.9\text{ }^{\circ}\text{C}$ ($30.2\text{ }^{\circ}\text{C} \rightarrow 32.1\text{ }^{\circ}\text{C}$), separately, while those of Participant B increased by $0.8\text{ }^{\circ}\text{C}$ ($34.1\text{ }^{\circ}\text{C} \rightarrow 35.3\text{ }^{\circ}\text{C}$) and decreased by $2.2\text{ }^{\circ}\text{C}$ ($34.6\text{ }^{\circ}\text{C} \rightarrow 32.4\text{ }^{\circ}\text{C}$). In particular, the TCM acupoint of Wind Pool (“Feng Chi” in Chinese) was located nearby the back of neck, which is sensitive to drafts [46].

Further analyses indicated wind characteristics in the air-conditioned experiment office. The wind azimuth angle was investigated using the power spectral density function (PSD), and the negative slope of the curve (β) indicated horizontal directional fluctuations in wind at the dominant frequency, as proposed and elaborated in [47]. Natural wind samples typically exhibit β values exceeding 1.1 [47]. In this experiment, the β value of air conditioning supply airflow measured 2.5 (Figure 8). A maximum air velocity of $0.4\text{--}0.6\text{ m/s}$ was observed, exceeding or reaching the upper limit of the draft rate (DR) values specified in ASHRAE standards [48]. Accordingly, Participant A positioned within the airflow trajectory reported thermal discomfort due to the elevated air velocity. Also, a much lower skin temperature was observed on the back of neck with Participant A than Participant B, being $28.6\text{ }^{\circ}\text{C}$ and $34.3\text{ }^{\circ}\text{C}$, respectively, after one hour of staying indoors. The different skin temperature changes of the two participants could be explained by the fact that Participant A (DHC) naturally preferred cooler feelings than Participant B (BC) and therefore stayed near the air conditioner, which aroused yin summer-heat risks. For example, the SET of Seat A and B was roughly $18.1\text{ }^{\circ}\text{C}$ and $28.1\text{ }^{\circ}\text{C}$.

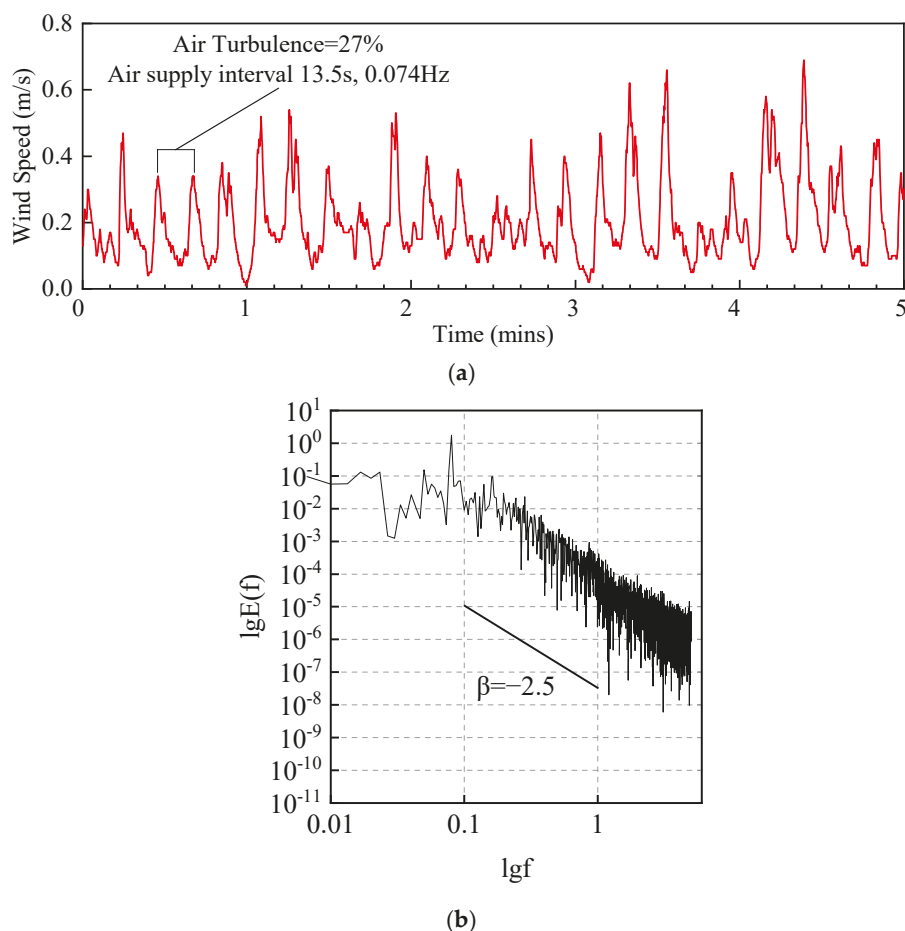


Figure 8. Dynamic features of airflows in the SAC office (data from Test L). (a) Air velocity of the supply air of the split-type air conditioner near the neck of Participant A; (b) logarithmic power spectrum curves of supply air.

That is, the enlarged gaps between indoor and outdoor conditions due to air conditioning would bring about notable regulation stress for occupants, evoke cold and wind excesses, and then cause yin summer-heat symptoms such as muscle soreness or stiffness. There would also be other symptoms, including respiratory discomfort related to low humidity. Also, yin-summer heat risks would appear to different extents due to the exposure to the cool supply air of air conditioners and occupants' constitutions.

Follow-up analyses of four staff sitting near the air conditioners suggested different yin summer-heat risks for vulnerable individuals (Figure 9). Staff 1, 2, 3, and 4 were of BC, DHC, DHC, and QDC constitutions, respectively. The operative temperature (T_o) was clearly lower for the four seats than others distant seats. The thermal and humid sensation votes of the four staff were mostly neutral and occasionally slightly cool or dry. However, they showed somewhat varied skin temperature. For example, Staff 1 (BC) with strong adaptability showed a slightly higher face and palm temperature than Staff 2 and 3 (DHC), who tended to feel hot. Also, they adjusted their clothes differently. Staff 3 wore slippers, and the exposed foot skin showed a relatively low temperature (27.8 °C). Staff 4 with poor environment adaptability wore more clothes (e.g., a coat). Despite such behavioral adaptation, yin summer-heat risks hid in the so-called “neutral” air-conditioned environment. It is essential to figure out how to utilize artificial cooling appropriately without covert health risks in the long run.

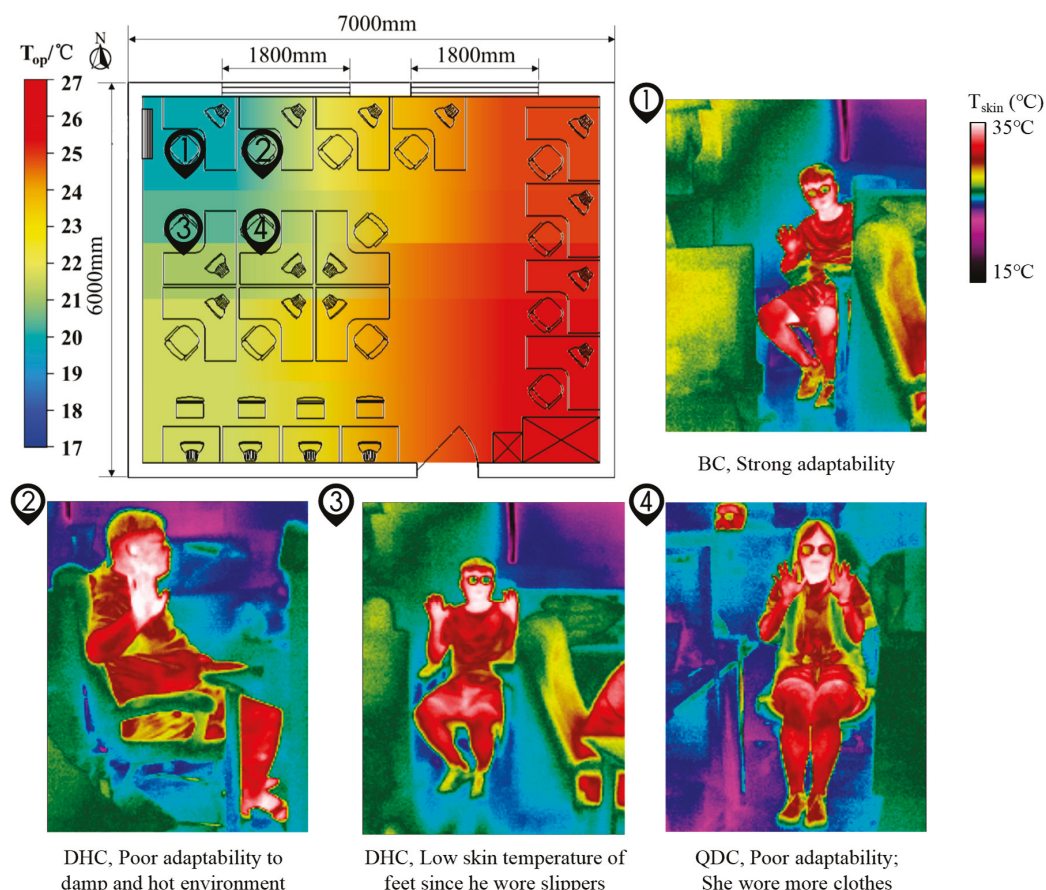


Figure 9. Yin summer-heat risks in the SAC office for young people of different constitutions (data from Test L).

Summarily, the experiment results showed that yin summer-heat risks were quite common with daily exposure to air-conditioned environments, even with thermally neu-

tral cases, which presented quite different air temperature, humidity, and air movement characteristics than a natural environment (Q1).

3.3. The Diverse Lifestyles and Demands of Occupants with Different Constitutions

Field test results in bedrooms for participants of different constitutions indicated that indoor thermal environment control modes would vary according to individuals' constitution features. As shown in Figure 10, the longest air conditioner usage time during non-work time was observed with the young male in Dalian, who was of the dampness–heat constitution (DHC), being less tolerant of damp and hot environments. The room air conditioning kept operating almost all the time when the room was occupied. The room temperature and absolute humidity were kept roughly at 25.3 °C and 7.3 g/m³. Moreover, much less air conditioner usage time was found with a young female and a young male in Hong Kong. The young female was of the yang-deficiency constitution (YADC), being vulnerable to wind, cold, and dampness excesses. She turned on air conditioners only for a while before sleeping or when it was getting hot and humid (28.6 °C and 18.3 g/m³ around 4:00 a.m.). The rapid drops in temperature and humidity, e.g., towards as low as 22.6 °C and 7.2 g/m³ after 4:00 a.m. probably suggested her compensatory expectation of a cool and dry sleeping environment. The young male was of the balanced constitution (BC), with strong environment adaptability. He moderately used air conditioners during sleeping when the room temperature decreased gradually from 30.1 °C to 25.9 °C. As for the elderly female in Changsha, who was of the yin-deficiency constitution (YIDC), being vulnerable to summer heat, dryness, and fire excesses, he could only endure the contradictory dehumidifying and overcooling effects of air conditioning when sleeping.

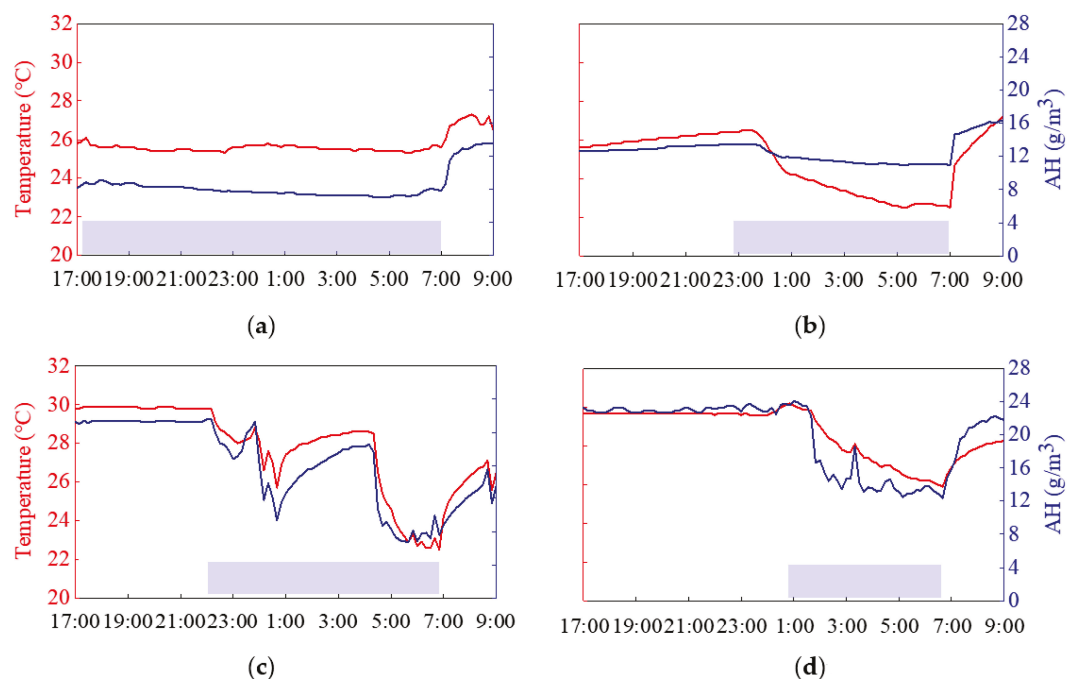


Figure 10. Operation modes of room air conditioners during non-work time (17:00~9:00 (the next day)) for participants of different constitutions. (a) Test H, Dalian, DHC, AC kept open in rest time; (b) Test E, Changsha, YIDC, Low temperature, high AH; (c) Test F, Hongkong, YADC, Greater humidity drop; (d) Test G, Hongkong, BC, Shorter AC time and higher set temperature. Abbreviations such as BC refer to the constitutions listed in Table 2.

Furthermore, significant correlations existed between the participants' constitutions and environment perceptions in the field tests. Generally, the major constitution worked well to interpret an individual's tendency in thermal, humid, and air movement sensation. As illustrated in Figure 11, participants with more YADC features, being vulnerable to wind, cold, and dampness excesses, tended to feel warm and breezy indoors. Participants with more YIDC features, being vulnerable to summer heat, dryness, and fire excesses, would feel dry easily. Participants with more DHC features, being less tolerant of damp and hot environments, were sensitive to heat. However, there seemed to some extent to be contradictory or irrelevant results for BC, QDC, and PDC features. This might be explained by potential pathogenic excesses associated with individuals' minor constitutions, which needs further exploration.

The large sample size of the questionnaire surveys enabled further exploration of population heterogeneity in the physiological and behavioral adaptation to hot summer, in the view of personal properties such as gender, age, and constitutions. There were four typical constitutions, which varied a lot in the tolerance of heat and coldness. Specifically, BC showed good thermal acceptance, with a wide thermal comfort zone, according to the environment adaptability and TCM pathogenic factors elaborated in Table 2; QDC presented a narrow zone easily affected by both heat and coldness; and YADC could not bear coldness and YIDC could not bear heat. Figure 12a illustrates that about 29% of males and 10% of females were of constitutions with strong thermal acceptance (BC). Generally, males were more resistant to cool conditions (BC + YIDC, 80%) and disliked warm conditions, while females were less tolerant of cool conditions (QDC + YADC, 68%) and preferred warm conditions. The constitution features explain the gender differences in thermal acceptance in naturally ventilated (NV) and air-conditioned environments (Figure 12b). That is, AC environments were usually cooler and poorly accepted by individuals vulnerable to cool conditions, such as many females, but well accepted by males.

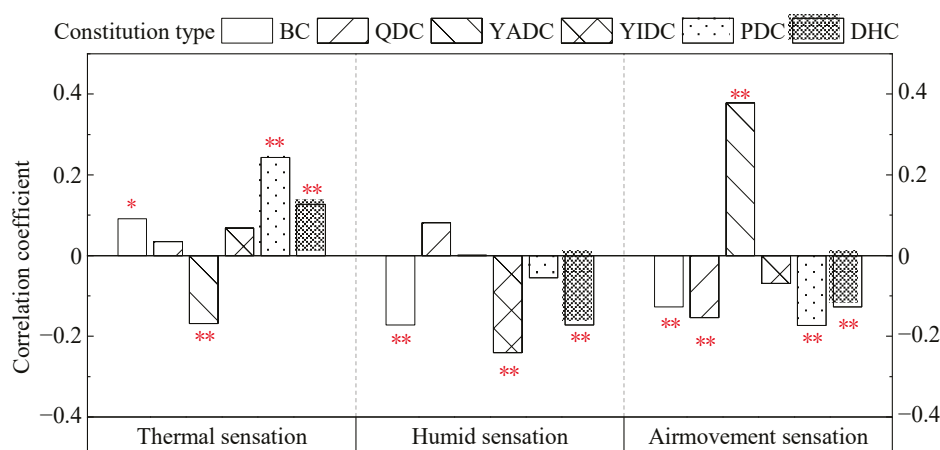


Figure 11. Correlations between constitutions and thermal, humid, and air movement sensation. Thermal, humid, and air movement sensation were measured with 7-point scales, with “−3” and “3” referring to hot and cold, very dry and very humid, and very still and very breezy, respectively. Lack of data for BSC, QSD, and ISC constitutions. * and ** mean significance at $p < 0.01$ and 0.05 levels, respectively.

Adaptive behaviors in hot summer changed with age. As displayed in Figure 13, young people would sometimes find it too hot to sleep and frequently use electric fans or air conditioners. Meanwhile, middle-aged and elderly people would only be annoyed by

sleepless hot nights occasionally. Elderly people preferred natural cooling measures, such as hand fans and opening windows.

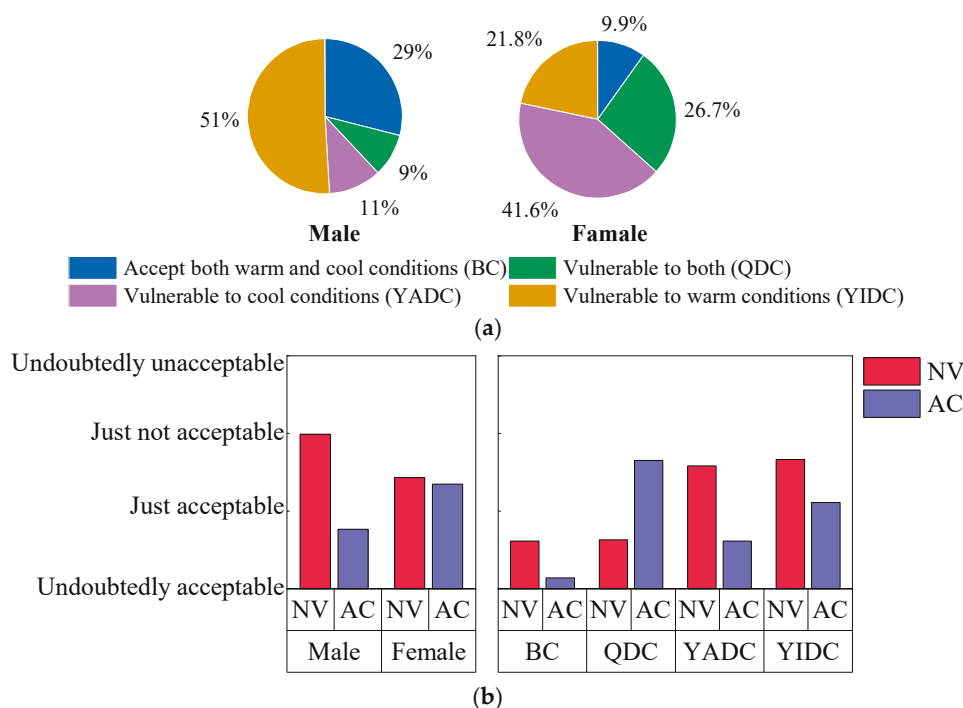


Figure 12. Thermal acceptance with gender and constitutions (data from Questionnaire I). NV and AC refer to naturally ventilated and air-conditioned buildings, respectively. (a) Gender distribution of constitutions; (b) thermal acceptance with gender (left) and constitutions (right).

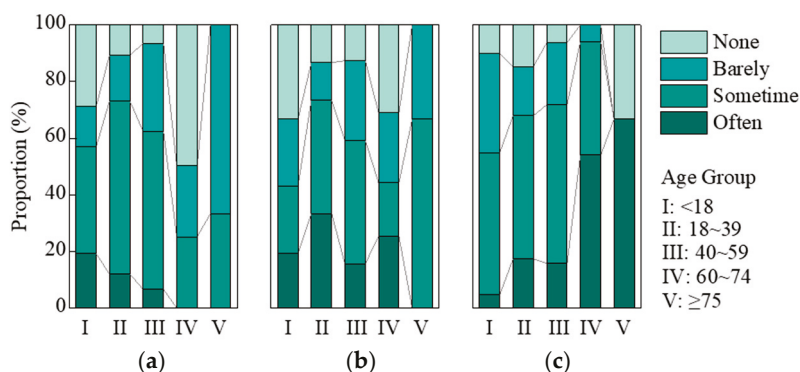


Figure 13. Behavioral adaption in hot summer days of different age groups (data from Questionnaire II). (a) How often will you feel too hot to sleep; (b) How often will you sleep with air-conditioner or electric fan operating; (c) How often will you use natural cooling measures.

Differences in adaptive behaviors could not only be explained by the tendency toward warmer environments when people get older or their heat exposures at earlier ages, mentioned in a large-scale study in Australia [49], but also be associated with lifestyles cultivated silently by the TCM culture. TCM knowledge guides individuals to maintain wellbeing in everyday life, e.g., how to adjust clothes, food, and housing conditions to prevent illness and improve health. The health preservation principles originated from the concept of organic wholeness as the core idea of TCM, e.g., the unity of internal and external environments and the unity of the body and the spirit. As shown in Figure 14, individuals with TCM health preservation knowledge relied less on air conditioning and

preferred natural cooling measures, such as hand fans, compared with those without TCM knowledge. Individuals (82%) with TCM knowledge would take hot tea or drinks at room temperature when they were sweating, while those without TCM knowledge preferred cold drinks (56%). However, such reliance on air conditioners and cold drinks for quick cool feelings in hot summer was accompanied by yin summer-heat risks.

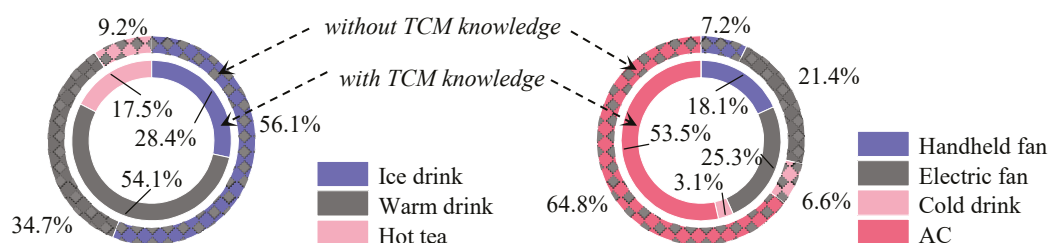


Figure 14. Summer adaptive behaviors reported by participants who were with or without TCM health preservation knowledge (data from Questionnaire II).

As for the usage time of air conditioners (Figure 15), 15%, 33%, 23%, and 30% of responses reported less than three hours, three to eight hours, eight to twelve hours, and more than twelve hours staying in air-conditioned environments, including the workplace and home, in a day. Restrained usage of air conditioners, i.e., less than three hours of exposure to air-conditioned environments, was common for QDC and QSC participants with weak constitutions. Meanwhile, DHC participants who were less tolerant of dampness and heat would stay in air-conditioned environments for even more than twelve hours per day.

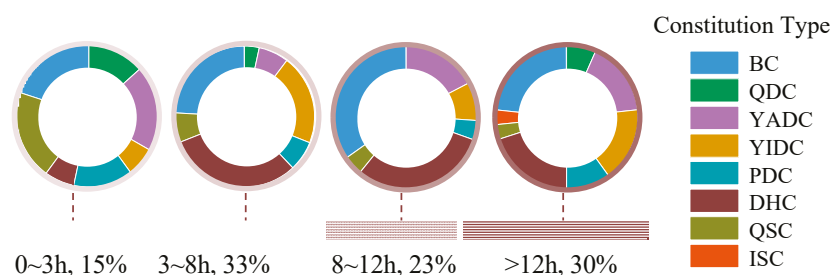


Figure 15. Daily time in air-conditioned environments by constitutions (data from Questionnaire III).

The average number of specific yin summer-heat symptoms appearing in the questionnaires of participants of the nine constitutions was examined, as shown in Figure 16. This study employed the indicator of the average number of all symptoms per capita to show the overall impact. Twenty symptoms were investigated, namely, a heavy feeling in the body and feeling lazy, upset for no reason, headache, less urine, cough, cold with sweating, cold with chills, stuffy nose, no perspiration, eczema, sticky stools, diarrhea, muscle soreness, rheumatism and joint pain, athlete's foot, halitosis, poor appetite, indigestion, chest tightness, and poor breathing. A higher proportion of BC participants stated no symptoms compared to other constitutions. A greater variety and number of discomfort feelings were reported by YADC, PDC, and DHC participants. YADC participants, who used air conditioners rather frequently, topped the nine constitutions in the average number of symptoms. Many non-BC participants reported a heavy feeling in the body, which was a typical yin summer-heat characteristic. Furthermore, extended AC exposure might exacerbate yin summer-heat symptoms. As shown in Figure 17, YADC, YIDC, PDC, and DHC participants, who suffered from cold excess associated with overcooling, showed severe symptoms such as diarrhea, poor appetite, and chest tightness, indicating that cold excess

was trapped in the body, other than just invading the skin or approaching the lungs. Also, more symptoms would emerge when these participants felt cold, even in a comfortable manner, than when it was neutral or not cool.

Summarily, the field tests and questionnaire survey results suggested the large variation in indoor environment demands and adaptive behaviors due to personal attributes, including gender, age, health preservation knowledge, and the Chinese constitution. In particular, constitutions worked well to explain the thermal perceptions, the usage mode of air conditioners, and the yin summer-heat symptoms of different populations (Q2).

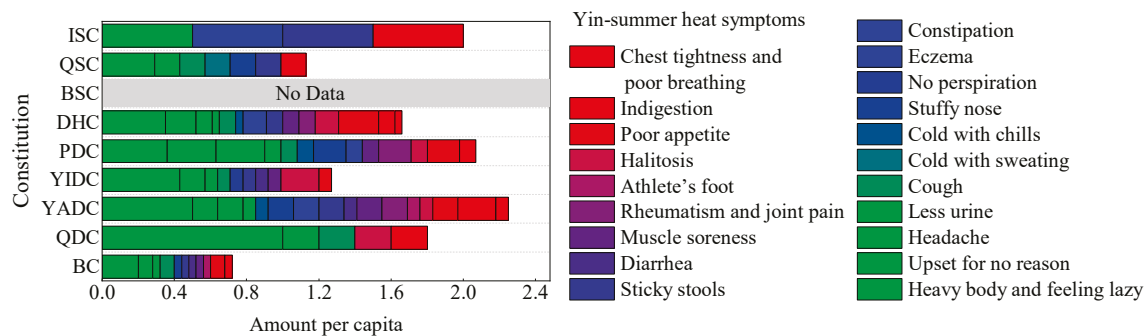


Figure 16. Amount of yin summer-heat symptoms per capita of different constitutions (data from Questionnaire III). Lack of BDC data in the analyzed data.

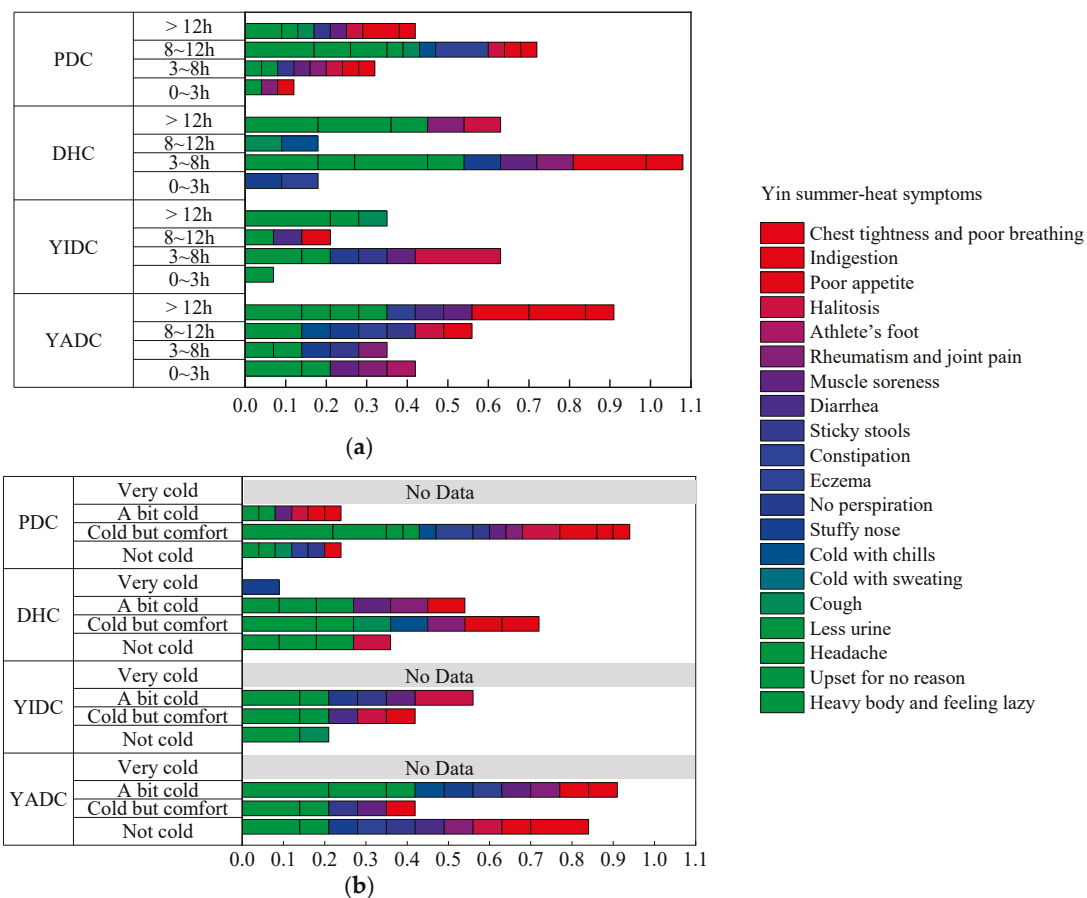


Figure 17. Cross-analysis results of the number of symptoms per capita in air-conditioned environments by constitutions, AC hours, and thermal sensation (data from Questionnaire III; limited data for “very cold” scenario). (a) Amount of symptoms per capita with different AC hours and constitutions; (b) Amount of symptoms per capita with different thermal sensation and constitutions.

4. Discussion

4.1. Health Risks in Well-Conditioned Buildings and Comfortable Environments: Should the Environment Be Isolated or Open?

Occupants in well-conditioned modern buildings are becoming accustomed to efficient and standardized control strategies and losing patience with natural solutions, such as adjusting clothing, using hand fans, and opening windows. As shown in Figure 13, young people were more often bothered by sleepless hot nights than elderly people (e.g., about 60% aged below 60 vs. 30% aged above 60), and tended to rely on air conditioners and electric fans for night cooling. It was also found that, for individuals with a lack of TCM knowledge to strengthen their adaptation to seasonal natural environment changes, air conditioners (64.8%) were their first choice for quick cooling in hot summer (Figure 14). Another study with Chinese urbanites [50] stated similar findings that young people performed fewer wellness habits, set lower indoor temperatures, and operated air conditioners more frequently, indicating the trend of the dependence on and precedence of air conditioning in summer indoor environment control.

However, this study found that staying long-term in “neutral” or “comfortable” air-conditioned environments was accompanied by potential yin summer-heat risks, aroused mainly by cold excess. For example, the number of symptoms per capita in the “cold but comfortable” environment was about 0.9, 0.7, 0.4, and 0.4 for participants with vulnerable constitutions of PDC, DHC, YIDC, and YADC, respectively (Figure 16). The health risks could firstly appear as trivial discomforts, which were easily ignored, for example, occasional coughs, but accumulate to be severe body symptoms in the long run according to TCM theories. Similar worries were proposed in the existing literature that the use of air conditioning for cooling would remove stimuli for physiological and perceptual adaption to the heat [51]. Some researchers used the metaphor of an “antibiotic” [52] to demonstrate the overuse of cooling devices and the necessity of demand management in modern buildings. A climate chamber study with seven healthy young subjects in China [53] found that short-term appropriate cold or hot exposure may help to enhance respiratory mucosal immunity by testing salivary secretory immunoglobulin E (S-IgE) at a temperature of 15–33 °C. This suggests that appropriate short-term cold and heat exposure may enhance human immune competence. However, the mechanisms underlying the beneficial versus detrimental effects of rapid thermal stress exposure require further investigation in future studies, particularly regarding dose–response relationships and individual constitutional variations. A multidisciplinary approach integrating thermo-physiology, immunophenotyping, and constitution assessment would be critical to establish personalized thermal exposure protocols.

4.2. The Ignored Role of Occupants in Indoor Environment Control and Their Diverse Demands: Should the Control Targets Be Fixed or Flexible?

Another issue is the overlooking of the proactive role of occupants in indoor environment control and their diverse demands due to contextual factors in existing thermal models and IEQ standards. Reflections on the narrow comfort zone (22–25 °C) and the standardized thermal comfort models, e.g., the predicted mean vote (PMV) model, led to the development of personal thermal comfort models, such as an adaptive model that extended the comfort zone to 21–29 °C [51] considering outdoor conditions and the thermal adaptation of occupants. It is now widely accepted that occupants should not be treated as “passive receipts of the external environment”, since multifaceted and comprehensive interactions exist between occupants and the external environment [54]. Occupant-centric control (OCC) strategies, compared to traditional centralized control strategies, also

follow the fundamental principle of delivering building services according to occupants' needs [55].

Generally, organic wholeness thoughts shaped the environment values of Chinese people, as well as their sensing of and reaction to the external environment. Such culture factors might explain the greater slope of 0.45 in the adaptive model tailored to China's naturally ventilated buildings [56], indicating a wider comfort zone than the slopes of 0.31 and 0.33 in ASHRAE 55-2020 [57] and EN 15251-2007 [58], respectively. In this present study, diverse demands and adaptive behaviors were found at the individual level, varying with gender, age, and, particularly, constitution. In particular, constitutions in Chinese medicine help to understand the environmental adaptability and adaptive behaviors of Chinese people. For example, DHC individuals who were less tolerant of hot and damp environments reported roughly the most usage of air conditioners in hot summer. Moreover, TCM knowledge on health preservation inspired ways to avoid potential health risks and create a wellbeing-centered indoor environment. For example, BC individuals displayed strong thermal adaptability. Individuals of constitutions such as YADC, YIDC, PDC, and DHC, being vulnerable to cold or wind excesses, showed more and severe yin summer-heat symptoms in an air-conditioned environment despite it being "comfortable". Thus, appropriate indoor micro-climate control targets and strategies should be set correspondingly for occupants of different constitutions. Flexible control in both temporal and spatial manners should be encouraged for shared spaces, e.g., in multi-member families and multi-person offices.

5. Conclusions

This study proposed reflections on indoor environment control in hot summer from a "yin summer-heat" perspective under TCM theories. Eleven field tests were carried out in residential buildings, together with experiments in an office environment and three questionnaire surveys with 765 responses from 2020 to 2023 in China. Overall, it was proved that introducing TCM theories, including pathogenic factors, constitutions, and health preservation principles, would enable a better understanding of diverse occupant demands and facilitate the design of indoor environment control strategies. The conclusions of this study are as follows:

- Question 1: Notable indoor–outdoor temperature, humidity, and air movement gaps existed in modern buildings due to air conditioning. The maximum observed real time indoor–outdoor temperature and absolute humidity differences were 12.2 °C and 13.6 g/m³. Such gaps added to the thermal regulation stress of occupants. Yin summer-heat risks associated with cold and wind excesses existed even in so-called "neutral" air-conditioned environments.
- Question 2: Individual of different constitutions in Chinese medicine usually presented varied adaptive behaviors and environment perceptions. Individuals who were vulnerable to cold or wind excesses, such as DHC, etc., would suffer from yin summer-heat symptoms in air-conditioned environments to different extents. A personalized threshold system based on constitution typology is recommended.

The novelty of this study is the firsthand data to demonstrate the inspiration of TCM theories for indoor environment control. Also, the constitutions in Chinese medicine and associated pathogenic excesses enable a much more systematic interpretation of the heterogeneity of occupant demands and potential health risks related to environmental factors. The findings will facilitate the development of user-tailored IEQ control targets and personalized environment control technologies. Moreover, this study is limited in the sample size. Further exploration of other TCM knowledge, such as the nine-palace–

eight-wind theory and the 24 solar terms, which are associated with environment adaptation, is expected to inspire interesting reflections on indoor environment control in modern buildings.

When referencing certain philosophical ideas and methodologies from Chinese classical literature or traditional Chinese medicine, the quantitative scientific evidence remains insufficient. Extensive scientific validation is required to transition from empirical discourse to scientific substantiation. Key questions for future research include how to effectively utilize the clinical samples accumulated in traditional Chinese medicine and how to translate these into regulatory threshold indicators for engineering applications in the field of built environments. These issues will undoubtedly become critical areas for investigation and validation in the years to come.

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Conflicts of Interest: The authors declare no conflicts of interest.

Appendix A

Table A1. Contents and example questions in questionnaire surveys.

Aspect	Contents or Example Questions	Answers
a. Demographic and physiological characteristics	Age Gender City of residence	/ Male/female /
b. Cooling-related adaptive behaviors	What will you do when you feel hot recently? What will you do when returning from the outside? How will you rank the following cooling measures?	Open windows/adjust clothes/turn on AC/take cold drinks, etc.
c. Wellness habits and TCM knowledge	Do you maintain a regular routine? How often will you work out? Is it true that physical and mental status are related?	Frequency or agreement scales
d. Indoor environment sensation, acceptance and preference	How do you feel about the temperature here and now? How do you feel about the humidity here and now? How do you feel about the air velocity here and now?	Scales in reference to ASHRAE standards, etc.
e. Illness symptoms indoors	Have you been sweating a lot recently? Do you have following symptoms recently? Do you often feel tired in summer?	Yes or no or scales
f. Test of constitutions in Chinese medicine	Do you have cold hands and feet? Do you feel your hands and feet hot? Do you get tired easily?	Frequency scales

Notes: three questions from each aspect are shown as examples in this table.

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Article

Innovative Prefabricated Wall Panel for Solar Utilization and Energy Efficiency: Building-Integrated Heat Pipe-Embedded System for Cooling-Dominant Zones

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Abstract: Buildings are major contributors to carbon emissions, emphasizing the need for energy efficiency. However, existing solar-integrated building façades often face integration and adaptability challenges. The aim of this study is to propose and evaluate an innovative building-integrated heat pipe-embedded (BiHPe) prefabricated wall panel for sustainable building design. By embedding heat pipes into concrete walls, the system transfers solar energy to domestic water. The performance of the system is evaluated using a comprehensive approach that integrates dynamic modeling, experimental validation, parametric analysis, and a case study. A dynamic energy balance model was developed and experimentally validated, identifying key factors affecting system performance, such as heat pipe spacing, absorber material, and heat pipe placement and configuration. Parametric analysis was conducted to assess the impact of these variables. Simulation results from a case study in Hong Kong show that the system reduces wall heat transmission to 76.1%, achieves a water gain efficiency of 16.7%, and saves 162 kWh/m² of electricity annually. Additionally, the system stabilizes indoor temperatures, improving thermal comfort. The BiHPe panel offers a multifunctional solution that combines energy efficiency, thermal comfort, and water heating, demonstrating exceptional adaptability and performance in cooling-dominant zones, making it a promising option for sustainable building design.

Keywords: heat pipe-embedded concrete wall; solar thermal utilization; energy saving; hot water supply

1. Introduction

Zero-carbon or low-carbon buildings have emerged as a crucial strategy to mitigate carbon emissions and reduce reliance on fossil fuels. Achieving zero operational carbon in buildings typically follows two key pathways: (1) reducing energy demand through energy-efficient measures and (2) maximizing the utilization of renewable energy via passive and active designs [1]. Regardless of the method used, efficient heat transfer equipment is indispensable. Among such devices, heat pipes stand out due to their exceptional thermal properties, with heat transfer coefficients between the evaporator and condenser zones ranging from 10^3 to 10^5 W/(m²·K) and thermal resistance as low as 0.01 to 0.03 K/W [2]. Thanks to this characteristic, heat pipes are widely used in a variety of energy efficiency systems and renewable energy application systems.

Most buildings rely on conventional water heaters powered by fuels such as town gas, liquefied petroleum gas, and electricity for year-round hot water supply. However,

this approach often results in inefficient energy conversion, exacerbates air pollution, and contributes to greenhouse gas emissions. Solar water heating systems offer a sustainable alternative, with heat pipe evacuated tube collectors demonstrating superior performance compared to conventional flat plate and single-phase thermosyphon designs [3,4]. Since Bienert's pioneering study on heat pipe solar energy collection in 1973 [5], advancements such as U-shaped, finned, and enlarged heat pipe condenser configurations have further enhanced thermal performance [6]. There are also improvements made to the heat pipe evacuated tube collector, including the addition of parabolic troughs, a tracking mechanism, and a glass cover, which have enhanced the heat collection efficiency [7]. Some improvements also focus on functional diversity, such as placing photovoltaic cells inside the vacuum glass tube, enabling both photovoltaic electricity generation and water heating [8]. Despite their advantages, these systems are typically roof-mounted, limiting energy-saving potential due to constrained roof area availability.

Building façades, which generally have a larger surface area than roofs, play a significant role in building heating and cooling demands. The integration of energy-efficient technologies into building envelopes is critical for achieving zero-carbon buildings. Early studies explored the use of serpentine water pipes embedded in walls for heat transfer [9–11], but these designs faced issues such as corrosion, scaling, freezing, and maintenance challenges. Albanese et al. [12] demonstrated that the heat pipe solar wall showed significant improvement over the concrete wall and water pipe wall in different climates [13,14]. Using heat pipes instead of water pipes can be considered as a good choice by eliminating corrosion and scaling problems as well as water freezing and sanitation problems [15].

The façade-integrated heat pipe system was proposed in the 2000s. Varga et al. investigated the performance of Z-shaped heat pipe panels for space cooling [16]. The heat pipe panels were installed outside the concrete wall with the evaporator section facing the indoor space. Yu et al. [17] proposed a phase change material (PCM) wall combined with a Z-shaped gravity micro-heat pipe and sky radiative cooling plate. When the heat pipe evaporator section faces the outdoor space, a heating effect is provided [18]. Gong et al. [19] proposed an L-shaped flat heat pipe solar house. The evaporator sections are put between the absorber and the exterior wall, and the condenser sections are bent into the interior thermal mass wall [20]. Guan et al. [21] proposed a micro-heat pipe and PCM solar house. Z-shaped gravity micro-heat pipe arrays were put in the block wall with the evaporator section exposed and blackened to directly absorb solar radiation energy. It can be concluded that heat pipes are used to transfer more solar heat to the outdoors or indoors for cooling purposes or heating purposes, respectively. However, solar heat is not utilized fully, especially in the cooling season; hence, the energy saving potential is limited.

If feeding water flows through the heat pipe condenser section, the water can be heated for domestic hot water supply, making the solar energy conservation potential greater. Zhao et al. [22] and Wang et al. [23] reported their investigation on a building-integrated loop heat pipe water heating system. Liu and Chow put heat pipe rings in metallic panels for façade integration [24]. Feed water flowed through the heat pipe condenser section. It had the dual function of domestic hot water supply and solar transmission reduction to the indoor space. He et al. [25] and Zhang et al. [26] investigated the heat pump-assisted façade loop heat pipe water heating system. The thermal efficiency of the loop heat pipe with heat pump assistance is higher than that without the heat pump. Meanwhile, the coefficient of performance (COP) of the loop heat pipe-assisted solar source heat pump system is higher than the air source heat pump system. Many other versatile designs were proposed, such as a combined heat pipe solar façade with heat pump and photovoltaic (PV) panels [27–29] and a heat pipe–PV/thermal wall with solar cells laminated on a base

panel and heat pipes welded underneath [30], which have the triple function of hot water supply, heating/cooling load reduction, and electricity generation.

The several types of heat pipe walls mentioned above are installed outside the solid wall. There are two disadvantages: its influence on the façade appearance and the hidden danger of parts falling on the street. Heavy glazing covers are usually set on the wall panel to obtain high thermal efficiency, which further increases the risk of falling. Putting heat pipes embedded in walls is a solution. Zhang et al. proposed a passive solar wall implanted with heat pipes [31]. The heat pipe evaporator and condenser sections were implanted in the inner and external mortar layer, respectively. During the heating season, the heat pipes in the south exterior wall work; thus, solar heat can be transferred to the indoors through the south exterior wall [32] and this improves the thermal environment, thus reducing the dissatisfaction rate [33]. Indoor heat was dissipated through the north heat pipe wall in summer, but a continual temperature increase occurred because of the limited heat transfer rate [34]. The energy saving potential of this design is very limited.

Based on the literature research above, existing façade-integrated systems, such as serpentine water pipes and externally mounted heat pipe systems, face significant limitations, including maintenance issues, aesthetic concerns, and safety risks. To address these challenges and identify a building façade that can utilize solar energy while reducing building energy demand in cooling-dominant zones, this study proposes a novel building-integrated heat pipe-embedded (BiHPe) system. Unlike conventional designs, the BiHPe system embeds heat pipes directly into reinforced concrete slabs, eliminating the need for external mounting. This approach improves safety, preserves façade aesthetics, and enhances energy efficiency by simultaneously reducing air-conditioning loads and preheating domestic water. It consists of an array of independently working heat pipes embedded vertically in a reinforced concrete slab. The evaporator section of each heat pipe is fitted with a metallic plate to improve heat absorption and ensure better alignment of the heat pipes. The condenser sections at the upper parts are inserted into a water tank or a horizontal water-flow channel. The flow channels of the neighboring wall panels can be inter-connected to form a part of the hot water supply network. The array of heat pipes conveys the wall-absorbed thermal energy to a feed water circuit. Accordingly, the solar heat transmission through the external wall is reduced and hence, the space air-conditioning load is reduced. The water discharge can be utilized as pre-heated feed water to the services hot water system so that the fossil fuel consumption in water heating can be reduced.

The system is primarily designed for prefabricated modular wall panels, offering versatility and ease of installation. This study aims to quantify the energy-saving potential of the BiHPe system, specifically focusing on its impact on energy efficiency, air-conditioning load reduction, and water heating energy savings. To evaluate the system's performance, a dynamic model based on energy balance equations was developed and solved by the "semi-implicit" finite difference method [30] in Section 2. In Section 3, the developed model was validated through experimental testing in a test chamber, where a BiHPe modular wall panel was embedded in the southern wall. The methodology also includes parametric analysis to identify key factors influencing system performance in Section 4, such as heat pipe spacing, absorber material, and pipe placement and configuration. Section 5 presents a case study in Hong Kong which is in a cooling-dominant zone. A discussion is provided in Section 6, and finally, the main conclusions are summarized in Section 7.

2. System Description and Modeling

Figure 1 illustrates the schematic diagram of the BiHPe wall panel, which primarily consists of three components: the concrete wall panel, heat pipes, and water flow channel.

Both the heat pipes and water flow channel are embedded within the concrete wall panel, seamlessly integrating with the wall without affecting its exterior appearance. Each heat pipe consists of an evaporator section, a condenser section, and a short adiabatic section. The evaporator sections absorb heat from the wall and transfer it to the condenser sections through the two-phase flow of the working fluid inside. To enhance heat transfer between the wall and the evaporator sections and to maintain a fixed spacing between the heat pipes, these heat pipes are welded to a metallic absorber plate. The condenser sections are submerged in a water box or flow conduit at the top end. The absorbed heat is then conveyed to the cold feed water in the water box. To prevent heat transfer from the water tank to the indoor environment, thermal insulation was applied to the side of the tank adjacent to the indoor space. In the figure, δ_w represents the thickness of the concrete wall, W is the heat pipe spacing, and L_e denotes the length of the evaporator section. The symbols “W1”–“W6”, “b”, “e”, “c”, and “w” represent the respective heat node labels. The specific meanings of these heat node labels are described in detail in the following paragraph.

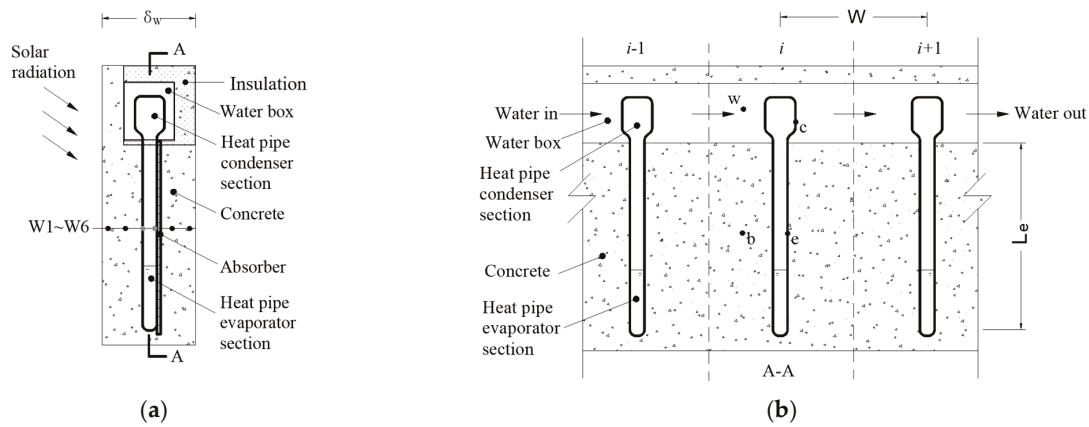


Figure 1. BiHPE system: (a) side cut; (b) front cut.

Assuming the number of heat pipes embedded in each panel is N_{hp} , water flows around each heat pipe condenser section in sequence, and is heated gradually. The BiHPE panel is divided into N_{hp} equal segments in order to simulate the temperature distribution more accurately. Each segment consists of ten nodes, as shown in Figure 1. These nodes are designated as follows: three wall nodes (“W1”, “W2”, and “W3”) located between the wall’s exterior surface and the position where the heat pipes are embedded; the absorber node “b”; the heat pipe evaporator section node “e”; the heat pipe condenser section node “c”; the flowing water in the water box node “w”; and another three wall nodes (“W4”, “W5”, and “W6”) between the location where the heat pipes are embedded and the wall’s interior surface. For the wall nodes “W1”, “W2”, and “W3”, in the i th ($i = 1, 2, \dots, N_{hp}$) segment, the energy balance equations are as follows:

$$c_W M_{W1} \frac{dT_{W1,i}}{dt} = h_{a1} A_{a1} (T_a - T_{W1,i}) + h_{sky1,i} A_{sky1} (T_{sky} - T_{W1,i}) + h_{12} A_{12} (T_{W2,i} - T_{W1,i}) + \alpha_W G_W A_W / N_{hp} \quad (1)$$

$$c_W M_{W2} \frac{dT_{W2,i}}{dt} = h_{12} A_{12} (T_{W1,i} - T_{W2,i}) + h_{23} A_{23} (T_{W3,i} - T_{W2,i}) \quad (2)$$

$$c_W M_{W3} \frac{dT_{W3,i}}{dt} = h_{23} A_{23} (T_{W2,i} - T_{W3,i}) + h_{3b} A_{3b} (T_{b,i} - T_{W3,i}) \quad (3)$$

where the parameter subscript containing the symbol “ i ” indicates that the value of the parameter corresponds to the i th segment, and that the parameter value varies between each segment. If the subscript does not contain the symbol “ i ”, it means that the parameter

value is the same for all segments. The same applies to the later sections of the text. c_W is the heat capacity of the concrete wall. M_{W1} , M_{W2} , and M_{W3} represent the masses of wall nodes “W1”, “W2”, and “W3” in the i th segment, respectively. $T_{W1,i}$, $T_{W2,i}$, and $T_{W3,i}$ represent the temperatures of wall nodes “W1”, “W2”, and “W3” in the i th segment, respectively. A_{a1} and A_{sky1} represent the heat transfer area between the wall node “W1” and the ambient air and the wall node “W1” and the sky environment, respectively. h_{a1} represents the convective heat transfer coefficient between the wall node “W1” and the ambient air [35]. $h_{sky1,i}$ represents the radiative heat transfer coefficient between the wall node “W1” in the i th segment and the sky environment [35].

$$h_{sky1,i} = \varepsilon_W \sigma (T_{sky}^2 + T_{W1,i}^2) (T_{sky} + T_{W1,i}) \quad (4)$$

A_{12} , A_{23} , and A_{3b} and h_{12} , h_{23} , and h_{3b} represent the heat transfer areas and heat transfer coefficients between wall nodes “W1” and “W2”, “W2” and “W3”, and “W3” and the absorber, respectively, which are equal in each segment.

$$A_{a1} = A_{sky1} = A_{12} = A_{23} = A_{3b} = A_{45} = A_{56} = A_{WR} = A_W / N_{hp} \quad (5)$$

where A_W is the total wall area where the absorber is embedded. The ambient temperature is denoted by T_a . G_W refers to the total solar radiation incident on the wall, while α_W indicates the absorptivity of the wall’s exterior surface. The heat transfer coefficients between adjacent wall nodes h_{12} and h_{23} in Equations (1)–(3) and h_{45} and h_{56} in Equations (23)–(25) are given by Equations (6) and (7), and h_{3b} is given by Equation (8). h_{mn} represents the heat transfer coefficient between wall nodes “W_m” and “W_n”, and δ_{mn} equals to the central distance between wall nodes “W_m” and “W_n”. Here, $m, n = 1, 2, \dots, 6$, and $m \neq n$.

$$h_{mn} = \frac{k_W}{\delta_{mn}} \quad (6)$$

$$\delta_{mn} = (\delta_{Wm} + \delta_{Wn}) / 2 \quad (7)$$

$$\frac{1}{h_{3b}} = \frac{\delta_{W3}}{2k_W} + \frac{\delta_b}{2k_b} \quad (8)$$

where k_W is the thermal conductivity of the concrete wall, and δ_{Wm} represents the thickness of wall node “W_m”.

For the nodes absorber “b”, heat pipe evaporator section “e”, and condenser section “c” in the i th segment, the energy balance equations are as follows:

$$c_b M_b \frac{dT_{b,i}}{dt} = h_{3b} A_{3b} (T_{W3,i} - T_{b,i}) + h_{be} A_{be} (T_{e,i} - T_{b,i}) + h_{b4} A_{b4} (T_{W4,i} - T_{b,i}) \quad (9)$$

$$c_e M_e \frac{dT_{e,i}}{dt} = h_{be} A_{be} (T_{b,i} - T_{e,i}) + h_{e4} A_{e4} (T_{W4,i} - T_{e,i}) + \frac{T_{c,i} - T_{e,i}}{R_{hp,i}} \quad (10)$$

$$c_c M_c \frac{dT_{c,i}}{dt} = \frac{T_{e,i} - T_{c,i}}{R_{hp,i}} + h_{cw,i} A_{cw} (T_{w,i} - T_{c,i}) \quad (11)$$

where A_{be} and h_{be} are the heat transfer area and coefficient between the absorber node and evaporator node, respectively, which are equal in each segment.

$$A_{be} = \delta_b H \quad (12)$$

$$h_{be} = \frac{2k_b}{x}, x = \frac{W - D_{e,o}}{4} \quad (13)$$

where δ_b and H are the thickness and the length of the absorber plate, respectively. k_b is the thermal conductivity of the absorber. W is the heat pipe spacing. $D_{e,o}$ is the outer diameter of the heat pipe evaporator section. A_{b4} and h_{b4} are the heat transfer area and coefficient between the absorber node and the wall node “W4”, which are also equal in each segment.

$$A_{b4} = WH \left(\frac{L_W - D_{e,o}}{L_W} \right) \quad (14)$$

$$\frac{1}{h_{b4}} = \frac{\delta_b}{2k_b} + \frac{\delta_{W4}}{2k_W} \quad (15)$$

where L_W is the width of each absorber node. δ_{W4} is the thickness of the wall node “W4”. The heat transfer area and coefficient between the evaporator node “e” and the wall node “W4” can be given by

$$A_{e4} = \left(\frac{\pi}{2} + 1 \right) D_{e,o} H \quad (16)$$

$$h_{e4} = 2k_W / \delta_{W4} \quad (17)$$

$R_{hp,i}$ is the thermal resistance between the heat pipe evaporator section and the condenser section in the i th segment, relating to temperatures and heat fluxes of the heat pipe evaporator and condenser nodes; for details, refer to chapters 2 and 3 of Ref. [36]. A_{cw} is the heat transfer area between the heat pipe condenser node and the water,

$$A_{cw} = \pi D_{c,o} L_c \quad (18)$$

where $D_{c,o}$ is the outer diameter of the heat pipe condenser section; L_c is the length of the condenser. $h_{cw,i}$ is the heat transfer coefficient between the heat pipe condenser node and the water node in the i th segment, distinguishing whether water is flowing or not [37].

For the node water “w” in the i th segment, the energy balance equation is:

$$c_w M_w \frac{dT_{w,i}}{dt} = h_{cw,i} A_{cw} (T_{c,i} - T_{w,i}) + h_{wR} A_{wR} (T_R - T_{w,i}) + c_w \dot{m} (T_{w,in,i} - T_{w,out,i}) \quad (19)$$

where A_{wR} and h_{wR} represent the heat transfer area and coefficient between the water and room air.

$$h_{wR} = k_{in} / \delta_{in} \quad (20)$$

where k_{in} and δ_{in} are the thermal conductivity and thickness of the insulation. $T_{w,in,i}$ and $T_{w,out,i}$ represent the inlet and outlet water temperatures in the i th segment, respectively, with the average water temperature calculated as

$$T_{w,i} = (T_{w,in,i} + T_{w,out,i}) / 2 \quad (21)$$

and

$$T_{w,out,i} = T_{w,in,i+1} \quad (22)$$

The mass flow rate of water is symbolized as $\dot{m}$.

For the wall nodes “W4”, “W5”, and “W6” in the i th segment, the energy balance equations are as follows:

$$c_W M_{W4} \frac{dT_{W4,i}}{dt} = h_{b4} A_{b4} (T_{b,i} - T_{W4,i}) + h_{e4} A_{e4} (T_{e,i} - T_{W4,i}) + h_{45} A_{45} (T_{W5,i} - T_{W4,i}) \quad (23)$$

$$c_W M_{W5} \frac{dT_{W5,i}}{dt} = h_{45} A_{45} (T_{W4,i} - T_{W5,i}) + h_{56} A_{56} (T_{W6,i} - T_{W5,i}) \quad (24)$$

$$c_W M_{W6} \frac{dT_{W6,i}}{dt} = h_{56} A_{56} (T_{W5,i} - T_{W6,i}) + h_{6R} A_{6R} (T_R - T_{W6,i}) \quad (25)$$

where T_R is the room air temperature. A_{45} , A_{56} , and A_{6R} represent the heat transfer areas between the wall nodes “W4” and “W5”, “W5” and “W6”, and “W6” and the room environment, respectively.

To solve the energy balance equations, the “semi-implicit” finite difference method (FDM) is utilized [30], where linear terms are treated implicitly and nonlinear terms explicitly. The discretized equations were solved using matrix-based solutions and custom-developed MATLAB programs.

3. Model Validation by Experimentation

3.1. Experimental Rig

To verify the accuracy of the developed BiHPe system model, an experimental rig was set up on the flat roof of a multi-story laboratory building located in Changzhou, Jiangsu Province, China (119.95° E, 31.79° N). Figure 2 shows the experimental setup used in this study. The test room, designed as a light-weight isolated hut with thermal insulation, measures 3 m (length) × 3 m (width) × 2.8 m (height). On its south face, there are two concrete walls placed side-by-side, each measuring 1.5 m (height) × 0.5 m (width) × 0.15 m (thickness). The concrete wall shown on the left in Figure 2, as viewed from outside, is the heat pipe-embedded wall, while the other serves as the reference wall. Apart from the heat pipe embedded and the copper absorber attached to the heat pipe-embedded wall, its construction and the thermal properties of the concrete material are identical to that of the reference wall. Furthermore, to minimize heat flow at the side edges, the edges of both walls are insulated with thermal insulation layers.

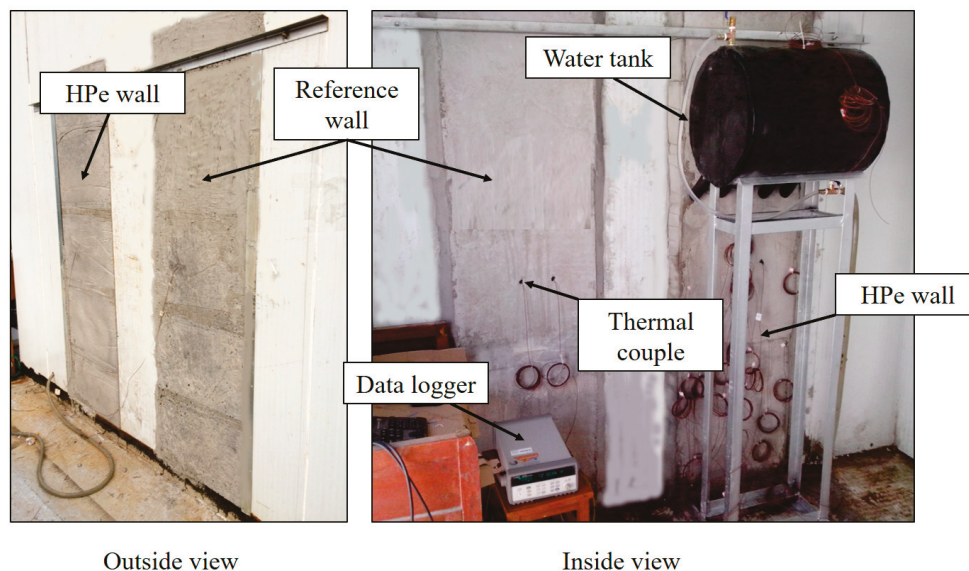


Figure 2. BiHPe system experiment test chamber.

Figure 3 illustrates the arrangement of the BiHPe system. Embedded within the wall is a copper absorber plate measuring 1 m (height) × 0.5 m (width) × 1 mm (thickness). Five heat pipes are securely tin soldered to the copper plate, ensuring excellent thermal contact, and the plate acts as an extended heat transfer area. The pitch between the heat pipes is 0.1 m. The distance between the exterior wall surface and the copper plate is set to 1/8 of the wall’s thickness. The total length of the heat pipe is 1.6 m, with the evaporator section measuring 1.0 m, the adiabatic section 0.425 m, and the condenser section 0.175 m. The inside

and outside diameters are 8 mm and 10 mm, respectively. These parameter settings were determined through preliminary parameter analysis using the theoretical model. To simplify the experimental setup, a large water tank was placed indoors, with the condenser section bent into the tank. The exposed portion inside the room is the adiabatic section, which was thermally insulated. In practical applications, water circulates between the water flow channel embedded in the wall and the water tank. In this experimental validation, the water flow mass is zero, and the temperature of the water node is the average water temperature of the tank. This experimental setup does not introduce differences compared to the theoretical model established for practical applications. To prevent freezing, water is deemed unsuitable as the working fluid due to the potential for the ambient temperatures in the Jiangsu province of China to drop below 0 °C during winter. Instead, the chosen working fluid for the heat pipe is R134a. Moreover, preliminary parameter analysis using the theoretical model showed that R134a provided better system performance than water. The filling ratio, which represents the ratio of the volume of the working fluid in the tube to the inside volume of the evaporator section, is set at 30%, equivalent to a liquid volume of 16 mL.

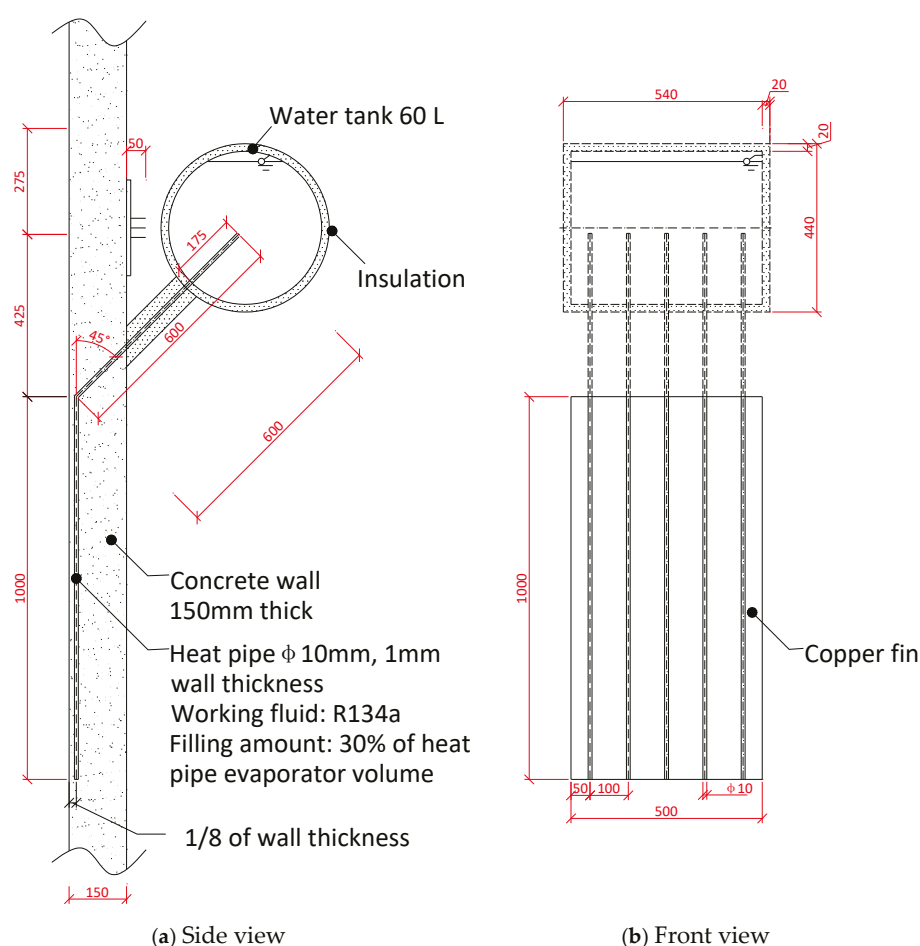


Figure 3. BiHPe system with heat pipe condensers bended into a water tank in the test room.

The water tank has dimensions of $\Phi 0.4 \text{ m} \times 0.5 \text{ m}$ (diameter $\times$ length), providing a water storage capacity of 60 L. Positioned within the test room, the water tank is fixed in place. To accommodate this setup, the adiabatic and condenser sections of the heat pipes are bent at a 45° angle towards the room. Despite this configuration, the vapor condensates can still flow back to the evaporator section through gravity force. To control the room air

temperature, a split-type air conditioner with both cooling and heating modes is installed. This air conditioner offers the flexibility to regulate the temperature as needed.

To collect data, pyranometers are used to measure the global solar radiation on both the vertical wall and horizontal roof surfaces. Additionally, temperatures are recorded using T-type thermocouples. The data are subsequently captured and logged using an automated data acquisition system (Agilent 34970A, a legacy model originally produced by Keysight Technologies, a U.S.-based company formerly known as Agilent Technologies, Santa Clara, CA, USA). To obtain comprehensive temperature measurements, a total of 42 thermocouples are utilized. These thermocouples measure temperatures on the outside surfaces of the heat pipes and the copper plate, as well as the interior and exterior surfaces of the heat pipe-embedded wall and the reference wall. Furthermore, water temperature in the water tank and indoor and outdoor air temperatures are also monitored using the thermocouples.

3.2. Model Validation

To simplify the experiment, the water tank was filled and a continuous 3-day operation experiment was conducted starting from 12 November. The weather was sunny on the 12th and 13th, and cloudy on the 14th. During this period, no water was added or drained from the tank. This approach ensured a stable and controlled environment, allowing us to focus on evaluating the thermal performance of the heat pipe-embedded system without introducing variability from water flow. The test data were recorded at 30 s intervals for model validation. After a complete water change in the tank, two additional sets of 3-day continuous experiments were conducted starting from 16 November, with all six days being sunny. Similarly, during each experiment set, no water was added or drained from the tank, and the tank was completely refilled after each set. The comparison between experimental results and simulation results can be made using data from any single day or from two or three consecutive days. The root-mean-square errors (RMSEs) between the simulated and experimental temperatures during the winter period from 12 to 21 November are given in Table 1. It can be seen that the RMSEs for the water temperature range from 0.1 °C to 0.2 °C. Most of the other RMSEs range from 0.3 °C to 0.8 °C, and the maximum value is 1.0 °C. It can be concluded that the developed model can predict the performance of the BiHPe system with good accuracy.

Table 1. RMSEs between the experimental and simulated temperatures in the BiHPe system.

RMSE Period	Outside Surface T_{so} (°C)	Inside Surface T_{si} (°C)	In the Middle of the Wall T_W (°C)	Copper Plate T_b (°C)	Evaporator Section T_e (°C)	Condenser Section T_c (°C)	Water in Tank T_w (°C)
12–13 November	0.6	0.5	0.4	0.6	0.5	0.6	0.1
17 November	0.7	0.3	0.4	0.8	0.7	0.8	0.2
18 November	0.9	0.4	0.6	1.0	0.9	0.8	0.1
16–18 November	0.5	0.6	0.4	0.7	0.7	0.5	0.2
19–21 November	0.7	0.3	0.5	0.7	0.6	0.7	0.2

Figure 4 presents the temperature distributions in the wall, heat pipes, and water during the period from 16 to 18 November. The experimental and simulated temperatures of the heat pipe evaporator are denoted by T_{e_exp} and T_{e_sim} , respectively. Similarly, T_{c_exp} and T_{c_sim} represent the experimental and simulated temperatures of the heat pipe condenser. For the water tank, T_{w_exp} and T_{w_sim} indicate the experimental and simulated mean water temperatures. Lastly, T_{si_exp} and T_{si_sim} correspond to the experimental and simulated inside

surface temperatures of the heat pipe-embedded wall. As shown in Figure 4, the simulated and experimental values of these four sets of temperatures align closely, demonstrating the high accuracy of the theoretical model. Moreover, over three days and two nights, the temperature of the evaporator section begins to decrease after 2 p.m. each day due to a drop in the ambient temperature, which lowers the wall temperature. This decline continues until 7 a.m. the following morning. In contrast, the temperature of the condenser section decreases only slightly from 5 p.m. to 7 a.m. the following morning, primarily due to heat dissipation from the water tank and the thermal conductivity of the heat pipe. This phenomenon demonstrates the thermal diode effect of unidirectional heat flow.

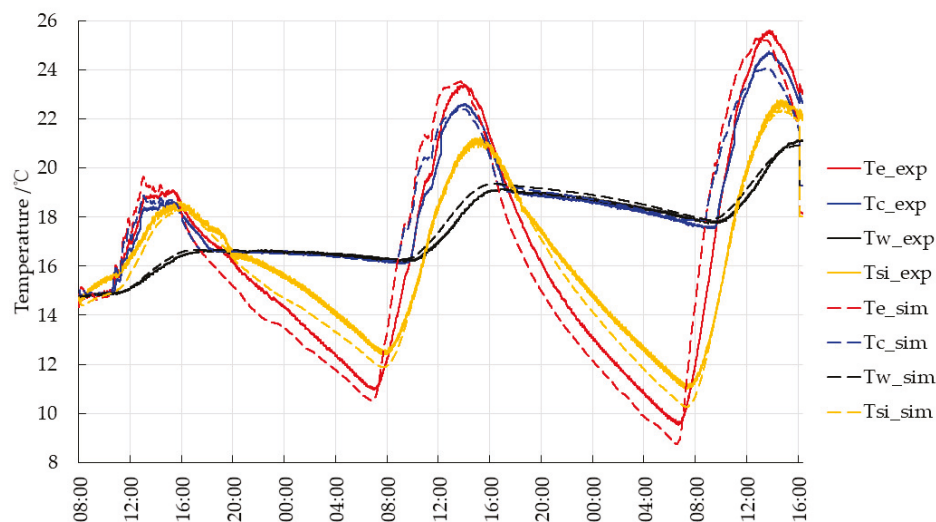


Figure 4. Experimental and predicted temperature distributions in the BiHPe system during the period from 16 to 18 November.

4. Parametric Analysis

The energy performance of the BiHPe system is intricately linked to both climatic conditions, such as solar radiation, ambient temperature, and wind speed, as well as its operating parameters, including wall orientation, incoming water flow rate, and water temperature. Furthermore, the system's physical parameters, such as heat pipe dimensions, tube spacing, their specific location within the wall, and the absorber materials used, directly influence its performance. In this section, we thoroughly investigate the impact of heat pipe spacing, absorber material, and heat pipe placement and configuration using the validated model and computational methods mentioned earlier. Unless otherwise specified, the wall dimensions, thermal properties, and operating conditions of the heat pipe-embedded system are provided in Table 2. To calculate the electricity savings, assuming a perfect heat conversion efficiency of 1.0 for the electrical water heater and a COP of 3.0 for the air conditioner in cooling mode, the savings are derived as the sum of the water heat gain divided by 1.0 and the reduction in indoor heat gain (compared with the reference wall) divided by 3.0.

Table 2. Thermal properties of wall and other parameters of the heat pipe-embedded wall.

Wall dimensions	1 m (width) × 1 m (height) × 0.15 m (depth)
Wall thermal properties	$k_W = 1.74 \text{ W/(m} \cdot \text{K)}$, $\alpha_W = 0.48$, $\varepsilon_W = 0.48$
Operating conditions	$G_W = 400 \text{ W/m}^2$, $T_a = 33 \text{ }^\circ\text{C}$, $T_R = 25 \text{ }^\circ\text{C}$, $u_{wind} = 1 \text{ m/s}$

Table 2. Cont.

Heat pipe dimensions	$L_e = 1 \text{ m}$, $L_c = 0.11 \text{ m}$, $D_{e,o} = 0.001 \text{ m}$, $D_{e,i} = 0.0086 \text{ m}$, $D_{c,o} = 0.024 \text{ m}$, $D_{c,i} = 0.022 \text{ m}$, axial groove wick
Working fluid and fill ratio	Distilled water, $V' = \frac{V_i}{V_e} = 0.3$
Initial water temperature	25 °C

4.1. Impact of Heat Pipe Spacing

Figure 5 illustrates the energy performance of the BiHPe system at different heat pipe spacings. The solar heat transmitted into the indoor space through the reference wall (no heat pipe embedded) is simulated to be 92.2 W/m^2 . When a single heat pipe is embedded in the middle of the wall, the heat transmission through the heat pipe-embedded wall is simulated to be 86.2 W/m^2 , with a water heat gain of 5.1 W/m^2 . By increasing the number of heat pipes embedded, a further decrease in the indoor heat gain can be achieved, while the water heat gain increases, leading to greater electricity savings. The energy performance of the BiHPe system is significantly enhanced. For instance, with ten heat pipes embedded in the wall, only 23.4 W/m^2 heat is transmitted to the indoor environment. The water heat gain is 140.6 W/m^2 and the electricity saving is 163.5 W/m^2 . This is attributed to the increased total heat transfer areas at the heat pipe evaporators and condensers, which reduce thermal resistance. With a decrease in tube spacing, the water gain efficiency (water heat gain divided by solar radiation) approaches 45%. However, the increased number of heat pipes raises the production cost. Over 12 heat pipes, i.e., tube spacing less than 0.08 m , the performance improvement becomes marginal, with less than 1% efficiency gain achieved for each additional heat pipe.

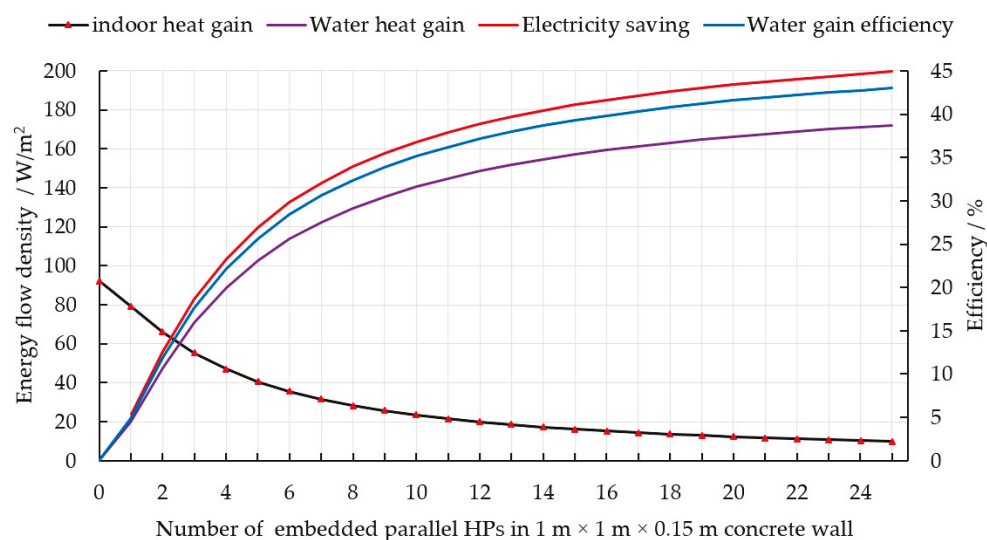


Figure 5. Energy performance against embedded wicked heat pipe density.

For installations with 8 to 12 heat pipes, the thermal efficiency can exceed 30%. Specifically, a spacing of 0.13 m corresponds to 8 heat pipes, while 0.08 m corresponds to 12 heat pipes, representing the recommended range for a practical optimal choice. Based on the above analysis, this range was selected to balance thermal efficiency and material costs. A smaller spacing increases heat pipe density, improving heat transfer but significantly raising material and installation costs. Conversely, a larger spacing lowers both costs and thermal performance. The chosen range represents a practical compromise, ensur-

ing effective heat transfer while remaining economically feasible. Even when the solar power incident is 800 W/m^2 , this recommendation remains valid. Moreover, when using Freon R134a instead of distilled water in the heat pipe, the performance of the heat pipe-embedded wall shows similar characteristics, and the suggested range for tube spacing, between 0.08 m and 0.13 m, remains valid.

4.2. Impact of Absorber Material

The evaporator sections of the heat pipes are affixed to a metallic absorber plate, designed to act as a fin and enhance heat transfer from the wall to the heat pipes. To assess the impact of the absorber material, four scenarios are selected: no absorber, carbon steel absorber, aluminum absorber, and copper absorber. The system performance with the recommended tube spacing of 0.1 m is applied for these four scenarios, and the results are summarized in Table 3.

Table 3. The performance of the BiHPe system with different absorbers attached (heat pipe spacing: 0.1 m).

Absorber Material	Thermal Conductivity [W/(m·K)]	Indoor Heat Gain Through Wall Transmission (W/m^2)	Water Heat Gain (W/m^2)	Water Heat Gain Efficiency (%)	Electricity Saving (W/m^2)	Water Temperature Rise ($^{\circ}\text{C}$)
	/	33.5	117.9	29.5	137.5	0.72
Carbon steel	43	32.6	120.0	30.0	139.9	0.73
Aluminum	238	23.4	140.6	35.2	163.5	0.86
Copper	390	22.5	142.7	35.7	165.9	0.87

In the absence of an absorber, the water heat gain and indoor heat gain through wall transmission are 117.9 W and 33.5 W, respectively, with a water gain efficiency of 29.5% and electricity saving of 137.5 W. For the other three scenarios with carbon steel, aluminum, and copper absorber materials, the indoor heat gains decrease by 2.7%, 30.1%, and 32.8%, respectively. Meanwhile, the water heat gains increase by 1.8%, 19.3%, and 21.0%, and the electricity savings increase by 1.7%, 18.9%, and 20.7%, respectively. It is evident that the performance improvement is observed in the order of absorber materials from carbon steel to aluminum, and finally to copper, due to their progressively increasing thermal conductivity. However, the improvement when using a carbon steel absorber is found to be very small. On the other hand, aluminum and copper absorbers significantly enhance the performance of the BiHPe system. Although the thermal conductivity of the copper absorber is approximately 64% higher than that of the aluminum absorber, the water heat gain efficiency increases by only 1.4%, and the reduction in heat transferred indoors is 3.8%, indicating a negligible performance improvement. From a performance optimization perspective, copper is the ideal material for the absorber; however, from an economic standpoint, aluminum is the more cost-effective choice.

4.3. Impact of Heat Pipe Location

To investigate the impact of the heat pipe location within the wall, the performance of the BiHPe system is analyzed under both summer and winter conditions, assuming a tube spacing of 0.1 m, which is considered to be the optimal spacing. Copper is chosen as the absorber material, and other working conditions remain consistent with previous cases, except for the variable of heat pipe location in the wall.

Tables 4 and 5 present the performances of the BiHPe system in summer ($T_a = 33\text{ }^{\circ}\text{C}$) and winter ($T_a = 10\text{ }^{\circ}\text{C}$) conditions, respectively, when the heat pipes are embedded at different locations within the wall with an overall thickness of $\delta_W = 150\text{ mm}$. The distances between the heat pipe and the wall exterior surface are $3\delta_W/4$, $\delta_W/2$, $\delta_W/4$, and $\delta_W/8$. Notably, heating demand during winter in Hong Kong is minimal, and as such, the electricity saving in winter can be considered equal to the water heat gain. The results indicate that as the distance between the heat pipes and the wall exterior surface reduces, the water heat gain, water temperature rise and electricity saving all increase, both in summer and winter conditions. This enhancement is due to the reduced thermal resistance between the heat pipes and the outer wall layer. Consequently, the BiHPe system with the heat pipes embedded at $\delta_W/8$ from the wall exterior surface exhibits the best performance in both summer and winter conditions.

Table 4. The summer performance of the BiHPe system with different distances of heat pipes from the wall exterior surface.

Distance Between Center of Heat Pipes and Wall Outside Surface (mm)	Indoor Heat Gain Through Wall Transmission (W/m^2)	Water Heat Gain (W/m^2)	Water Heat Gain Efficiency (%)	Electricity Saving (W/m^2)	Water Temperature Rise ($^{\circ}\text{C}$)
112.5 ($3\delta_W/4$)	22.8	130.3	32.6	153.4	0.8
75 ($\delta_W/2$)	22.5	142.7	35.7	166.2	0.9
37.5 ($\delta_W/4$)	23.1	160.5	40.1	183.8	1.0
18.75 ($\delta_W/8$)	23.6	171.1	42.8	194.2	1.1

Table 5. The winter performance of BiHPe system with different distances of heat pipes from the wall exterior surface.

Distance Between Heat Pipes and Outside Surface (mm)	Indoor Heat Gain Through Wall Transmission (W/m^2)	Water Heat Gain (W/m^2)	Water Gain Efficiency (%)	Electricity Saving (W/m^2)	Water Temperature Rise ($^{\circ}\text{C}$)
112.5 ($3\delta_W/4$)	3.6	112.2	28.1	112.2	0.7
75 ($\delta_W/2$)	3.3	119.6	29.9	119.6	0.7
37.5 ($\delta_W/4$)	2.7	131.8	33.0	131.8	0.8
18.75 ($\delta_W/8$)	2.3	139.0	34.8	139.0	0.9

4.4. Impact of Heat Pipe Configuration

The wick inside a heat pipe provides a capillary force that aids in the flow of the condensed working fluid back to the evaporator section. In vertically installed heat pipes, with the evaporator section located below the condenser section, the condensed liquid can flow back to the evaporator section due to gravity force. Hence, wickless heat pipes can be suitable for use in vertical walls or walls with a close-to-vertical orientation. For the investigation, assuming a heat pipe spacing of 0.1 m , no absorber attached, and the heat pipes located in the middle of the wall, the thermal performance of both an axial groove wick heat pipe-embedded system and a wickless heat pipe-embedded system are analyzed. The results are presented in Table 6. Interestingly, the presence of the axial groove wick reduces the system performance. This is because the wick's existence increases the thermal resistance of the heat pipe, resulting in a lower amount of heat being transmitted to the water. Therefore, the thermal performance of the wickless heat pipe-embedded

wall is slightly better than that of the wicked heat pipe-embedded wall. Furthermore, the production cost of wicked heat pipes is higher than that of wickless ones.

Table 6. The performance of the BiHPe system with different configurations of heat pipes.

Wick Form	$D_{e,o}$ (mm)	L_e (m)	Indoor Heat Gain (W/m^2)	Water Heat Gain (W/m^2)	Water Gain Efficiency (%)	Electricity Saving (W/m^2)	Water Temperature Rise ($^{\circ}C$)
Axial groove	8	1	33.5	117.9	29.5	137.5	0.7
No wick	8	1	33.3	118.3	29.6	137.9	0.7
No wick	8	1.5	38.2	107.1	26.8	125.1	1.0
No wick	8	2	42.3	97.9	24.5	114.5	1.2
No wick	10	1	30.9	123.7	30.9	144.1	0.76
No wick	15	1	27.4	131.6	32.9	153.2	0.8
No wick	20	1	25.5	135.9	34.0	158.1	0.8
No wick	25	1	24.3	138.6	34.7	161.2	0.9

As depicted in Table 6, it is evident that both water gain efficiency and electricity savings decrease, while indoor heat gain increases with an increase in the length of the heat pipe evaporator section. Specifically, for each 0.5 m increase in heat pipe length, the water gain efficiency drops by approximately 2.3–2.8%. It can be explained that by increasing the length of the heat pipe evaporator section, although the heat obtained by each heat pipe increases, the heat gain per unit wall area decreases. When the outer diameter of the heat pipe evaporator section increases from 8 mm to 10 mm, 15 mm, 20 mm, and 25 mm, there is a corresponding increase in the water heat gain and the water temperature rise. Additionally, electricity savings increase by 4.5%, 11.1%, 14.6%, and 16.9%, respectively. This improvement can be attributed to the increased heat transfer area between the absorber and the heat pipes, resulting in more energy being transferred to the water. However, it should be noted that this also leads to a higher initial cost due to the larger and more expensive heat pipes.

5. Case Studies

Due to its high demand for hot water and large exterior wall area, a gymnasium in Hong Kong was selected as the case study to assess the system's effectiveness. However, this approach can also be applied to other building types with similar characteristics. A validated dynamic model is used to analyze the annual thermal performance of the BiHPe system applied to the hypothetical gymnasium room in Hong Kong (22.304° N, 114.17° E). In this case study, heat pipes are embedded in the front wall of the hypothetical gymnasium room. The front wall is facing the southwest direction. An isometric view of the rectangular room is shown in Figure 6. The floor area is 200 m² (20 m × 10 m). The dimensions of the front wall are 20 m (W) × 3 m (H). A total number of 200 heat pipes are soldered on copper plates with a pitch of 100 mm and embedded in the front wall. The dimensions of the copper absorber plates are 20 m (W) × 2.3 m (H). The distance between the copper plates and the wall exterior surface is set to 1/8 of the wall thickness. A water tank with a total volume of 1.25 m³ is incorporated at the upper-inner side of the heat pipe-embedded wall. The adiabatic and condenser sections of these heat pipes bend at 45° towards the indoor space. Some design data and thermal properties of the BiHPe systems in use are listed in Tables 7–9. According to the building energy code in Hong Kong, the recommended maximum U-value for opaque walls should not exceed 3.3 W/m²·K [38]. It

is important to note that conventional wall constructions in Hong Kong are typically built without any insulation layer.

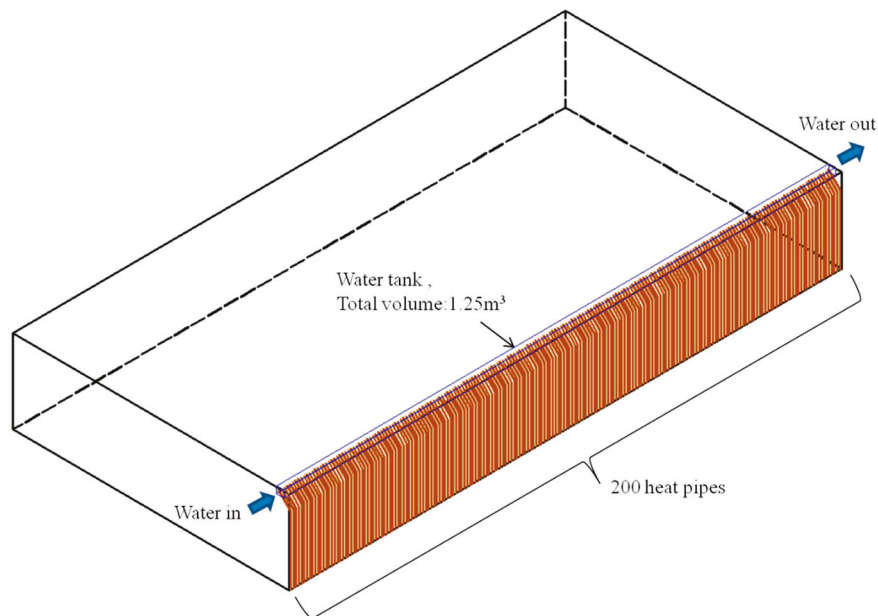


Figure 6. Arrangement of the BiHPe system.

Table 7. Thermophysical properties of the front wall.

Material	Thickness δ (mm)	Thermal Conductivity k [W/(m·K)]	Thermal Resistance R (m ² ·K/W)	Density ρ (kg/m ³)	Thermal Capacity c_p [J/(kg·K)]
Concrete	150	1.51	0.099	2300	920
Absorptivity of exterior surface				0.7	
Emissivity of exterior surface				0.35	
Heat transfer coefficient $U = 1/(0.044 + 0.299 + R)$			2.26 W/(m ² ·K), not exceeding 3.3 W/(m ² ·K) [38]		

Table 8. Design data and thermal properties of the thermal absorber and water tank.

Parameters		Data Values
Thermal absorber (copper plate)	Total length	20 m
	Width	2.3 m
	Thickness	1 mm
	Thermal capacity	385 J/(kg·K)
	Density	8933 kg/m ³
	Thermal conductivity	390 W/(m·K)
Water tank	Cross section	0.25 m × 0.25 m
	Internal length	20 m
	Volume	1.25 m ³
	Tank surface heat loss coefficient	1.2 W/(m ² ·K)

The annual performance of the BiHPe system for application in Hong Kong was predicted by means of self-developed MATLAB computer programs based on the hourly TMY weather data [39]. The operating schedule of the gymnasium was obtained by referring to the Guidelines on Performance-based Building Energy Code [38], as shown in Table 10. According to the guidelines, the designed occupant density was set at 3 m²/person and the

hot water consumption, mainly for showers, was 20 L/person. The water was discharged according to the hot water demand for each hour shown in Table 10. For example, on Sunday at 15:00, the maximum number of occupants was 67 (i.e., $200/3 = 67$ persons) and the maximum hot water demand is 1340 L ($20 \times 67 = 1340$). The hot water demand was 1072 L ($1340 \times 0.8 = 1072$). The discharge water temperature was set equal to the water temperature in the water tank, and the charging water temperature was set to be the same as the mean ambient temperature of the day. The indoor temperature was maintained at 24 °C during the seven summer months (April to October) and 21 °C for the remaining five months. Each day, the initial water temperature in the tank was set equal to the day's average ambient temperature. The air-conditioner operated daily from 6:00 to 22:00. Thermal performance metrics, including solar heat transmission through the heat pipe-embedded wall, water heat gain, and the interior surface temperature of the wall, were analyzed and compared to the reference wall case.

Table 9. Parameters of the heat pipe (wickless, R134a-copper, filling ratio: 30%).

Parameters	Value
Total Length	3.05 m
Length of evaporator section	2.45 m
Length of adiabatic section	0.425 m
Length of condenser section	0.175 m
Outside diameter of evaporator and adiabatic section	10 mm
Inside diameter of evaporator and adiabatic section	8.6 mm
Outside diameter of condenser section	24 mm
Inside diameter of condenser section	22 mm
Thermal capacity of container wall	385 J/(kg·K)
Density of container wall	8933 kg/m ³
Thermal conductivity of container wall	390 W/(m·K)

Table 10. Operating schedule of the gymnasium for the three day types.

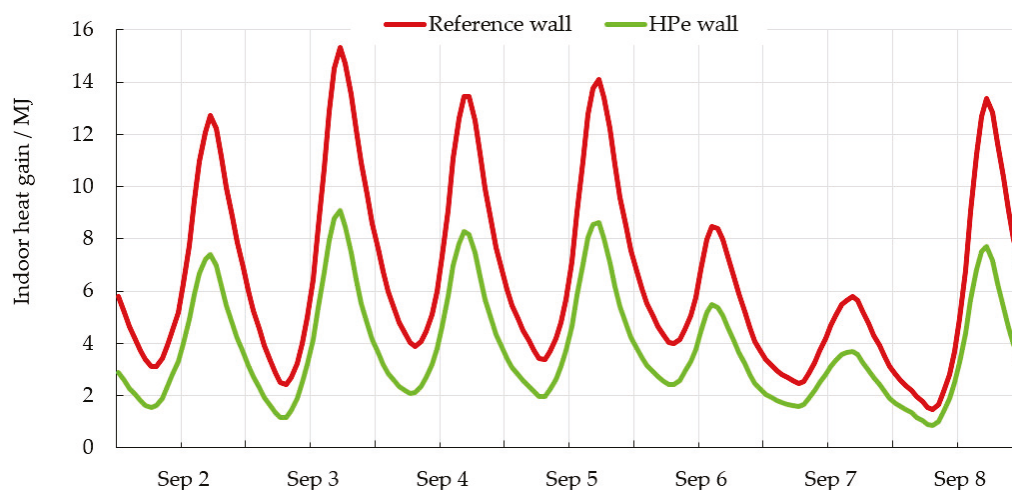
Time	AC	Percentage of Hot Water Consumption			Hot Water Consumption (L)		
		(Monday–Friday)	(Sat)	(Sun)	(Monday–Friday)	(Sat)	(Sun)
01:00~06:00	off	0.05	0.05	0.05	67	67	67
07:00	on	0.2	0.3	0.3	267	400	400
08:00	on	0.3	0.5	0.5	400	667	667
09:00	on	0.4	0.6	0.6	533	800	800
10:00~11:00	on	0.5	0.7	0.8	667	933	1067
12:00~14:00	on	0.3	0.5	0.5	400	667	667
15:00~17:00	on	0.5	0.8	0.8	667	1067	1067
18:00	on	0.6	0.8	0.8	800	1067	1067
19:00~21:00	on	0.8	0.8	0.8	1067	1067	1067
22:00	on	0.5	0.5	0.5	667	667	667
23:00~00:00	off	0.05	0.05	0.05	67	67	67

Table 11 shows the heat transmissions through the heat pipe-embedded wall and the reference wall. In Hong Kong, the cooling demand is dominant. By incorporating the heat pipe-embedded wall, the annual transmission heat gain, which contributes to the cooling demand, is reduced to 72.7% compared to the reference case. The effect on transmission heat loss is minimal, and the total heat transfer through the wall (including both heat gain and loss) decreases to 76.1% of the reference case.

Table 11. Annual heat transfer through the heat pipe-embedded wall and the reference wall.

Location	Wall	Transmission Heat Gain/MJ	Transmission Heat Loss/MJ	Overall Heat Transmission/MJ
Hong Kong	Reference wall	27,157.1 (100%)	3601.8 (100%)	30,758.9 (100%)
	Heat pipe-embedded wall	19,734.9 (72.7%)	3681.5 (102.2%)	23,416.4 (76.1%)

Figure 7 illustrates the hourly indoor heat gains through wall transmission during a typical summer week. It can be observed that the indoor heat gain for the heat pipe-embedded wall is consistently lower than that of the reference wall throughout the week, with a more noticeable difference occurring in the afternoon. The heat pipes embedded in the wall effectively transfer heat away, converting it into hot water, which reduces the indoor heat gain. Specifically, the peak values of indoor heat gains during the summer week are reduced to just 59.3% of those in the reference case. This significant reduction demonstrates the heat pipe-embedded wall's superior performance in mitigating indoor heat gain, especially during periods of strong solar radiation. As a result, the heat pipe-embedded wall reduces the cooling load required to counter excessive heat, particularly in the afternoon. The data in Figure 7 clearly highlight the effectiveness of the heat pipe-embedded wall in improving thermal management and energy efficiency under typical summer conditions.

**Figure 7.** Hourly transmission heat gain during a typical summer week.

The annual total solar radiation on the vertical wall facing the southwest direction in Hong Kong is calculated as 3182.1 MJ/m² using the Perez model [40]. Throughout the year, the water heat gain of the BiHPE system is simulated to be 530.7 MJ/m², resulting in an average water gain efficiency (total water heat gain percentage of total solar radiation incident) of 16.7%. Comparing with existing research results, for a heat pipe solar water heating system with front glass cover and insulation backside, which is placed outside the wall, the water heat gain efficiency was over 50% [22]. In contrast, the BiHPE system in this study, lacking the greenhouse effect of the front glass cover and the insulation effect of the backside, obtains an annual average efficiency as low as 16.7%. This paper aims to propose a new system with stronger applicability that does not require a glass cover, has no impact on the building's appearance, and eliminates the risk of the collector panel dropping down, although the heat collection efficiency is inevitably lower. There is still research space for efficiency improvement in this system, such as enhancing the absorptivity of the wall

exterior surface to increase solar heat absorption, reducing the wall surface emissivity to minimize heat loss, using aerated concrete materials to reduce heat transfer from the wall to the indoor space, and decreasing the heat transfer resistance between the wall and the heat pipe evaporator sections. Future research will focus on improving the efficiency of this system. Assuming an air-conditioning system COP of 3.0 and 100% thermal efficiency for an electric water heater, the corresponding annual electricity savings for air-conditioning is 2447.5 MJ (overall heat transmission reduction in the heat pipe-embedded wall compared to the reference wall/COP), and the annual electricity saving for water heating is 24,412.2 MJ. Thus, the annual total electricity saving is 26,859.7 MJ (7461 kWh), that is 162 kWh/m².

The temperature of a wall's interior surface significantly influences occupant thermal comfort. Figure 8 compares the interior surface temperatures of the heat pipe-embedded wall ($T_{si,HPE}$) and the reference wall ($T_{si,ref}$) during a typical summer week, with T_a representing the ambient temperature. The interior surface temperatures exhibit a trend similar to the ambient temperature but with a slight time lag, and the variations are smoother, without noticeable fluctuations. The results demonstrate that the heat pipe-embedded wall effectively lowers the interior surface temperature, achieving maximum reductions of up to 2.5 °C. Furthermore, the heat pipe-embedded wall shows smaller temperature fluctuations compared to the reference wall. Therefore, it can be concluded that the heat pipe-embedded wall contributes to maintaining a more stable and comfortable interior surface temperature, improving indoor thermal comfort.

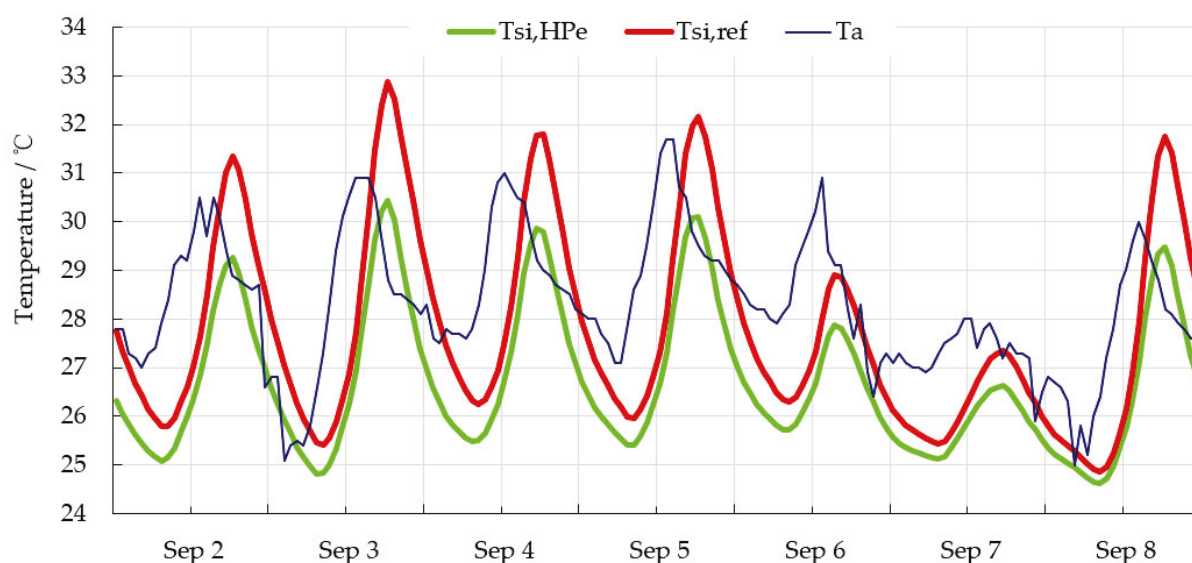


Figure 8. Temperature profiles of the interior surfaces for the heat pipe-embedded wall and the reference wall during a typical summer week.

In summary, implementing the BiHPE system in Hong Kong demonstrates three key benefits. First, total heat transfer through the wall decreases to 76.1%, significantly lowering air-conditioning energy consumption. Second, the system achieves an annual water heat gain of 530.7 MJ/m² with a thermal efficiency of 16.7%. This results in an annual electricity savings of 162 kWh/m² for the air-conditioning system and water heating. Lastly, indoor thermal comfort is enhanced as the heat pipe-embedded wall reduces interior surface temperatures by up to 2.5 °C and minimizes fluctuations.

6. Discussion

The originality of this paper lies in the introduction of a novel BiHPe system, which represents a significant advancement over existing façade-integrated systems. This innovative system distinguishes itself by integrating heat pipes directly into the wall structure. The BiHPe system offers several key advantages: (1) no adverse impact on façade aesthetics; (2) simplicity in design, manufacturing, and installation; (3) elimination of external-mounted weights, reducing safety risks; (4) low maintenance requirements due to the absence of moving parts and corrosion issues; (5) prevention of reverse flow circulation, minimizing water heat loss; and (6) enhanced energy savings and thermal comfort.

This study demonstrates the effectiveness of the innovative BiHPe system in addressing solar energy utilization challenges in urban environments. By integrating heat pipes within concrete walls, the system successfully reduces indoor heat gain while providing pre-heated water and improving indoor thermal comfort, offering three key benefits for energy-efficient building design. Its practical integration makes it particularly suitable for prefabricated and zero-carbon buildings, enhancing their sustainability.

The BiHPe system's adaptability and cost-effectiveness make it especially advantageous for space-constrained, cooling-dominant urban regions. However, further research is required to evaluate the system's performance in other climate conditions, as varying solar radiation levels and temperature profiles may affect its efficiency. Furthermore, for climate zones dominated by heating demands, the focus is on how to deliver more solar energy to the indoor environment, and the preheated water of this BiHPe system can be utilized for radiant heating. Life-cycle cost analysis, including initial installation, maintenance requirements, and long-term performance, is essential to validate the system's broader applicability and economic feasibility. Additionally, improving water gain efficiency remains an area for optimization. Future studies could focus on increasing solar absorption rates and minimizing heat dissipation from the exterior wall surface to further enhance the system's performance.

In summary, the BiHPe system offers a promising and efficient solution for sustainable building design, effectively reducing thermal loads and supporting renewable energy utilization in urban settings. Continued research into performance improvement, climate adaptability, maintenance considerations, and long-term economic benefits will be critical to fully realizing its potential in diverse environments.

7. Conclusions

This study proposed the BiHPe system, developed a theoretical model, conducted experimental validation, and performed a parameter analysis. A year-round performance evaluation was further conducted for a gymnasium in Hong Kong as the case study. The key conclusions are as follows:

(1) The BiHPe system integrates heat pipes within the exterior wall, with the evaporator section fixed on a metallic absorber plate and the condenser section inserted into a water duct. This configuration utilizes solar energy absorbed by the wall for water heating while reducing the heat transferred indoors, thereby lowering air-conditioning cooling loads. Additionally, the interior wall surface temperature is significantly reduced with less fluctuation, improving thermal comfort. The heat pipe-embedded panel can be modularly prefabricated and tightly assembled between panels without altering the building's appearance, providing three functions: a reduction in energy consumption for both air conditioning and water heating, and an improvement in indoor thermal comfort, applicable for cooling-dominant climate zones.

(2) A theoretical model of the system was established based on energy balance equations for each thermal node. An experimental setup was constructed to validate the model, with results showing good agreement between simulations and measurements, confirming the model's accuracy.

(3) A parametric analysis was conducted, considering factors such as heat pipe spacing, absorber material, and heat pipe placement and configuration. The results suggest that a heat pipe spacing of 0.08–0.13 m represents a balance between system efficiency and cost-effectiveness. From a performance perspective, copper was identified as the preferred absorber material, while aluminum is recommended for cost efficiency. For heat pipe placement, closer positioning to the exterior surface of the wall was found to be more effective. Furthermore, wickless heat pipes demonstrated superior performance compared to wicked heat pipes.

(4) Year-round simulation results for the Hong Kong case showed that the BiHPe system could reduce wall heat transmission to 76.1% of the reference case and achieve an annual average water gain efficiency of 16.7%. This results in an estimated electricity saving of 162 kWh per square meter of absorber area. Additionally, the interior wall surface temperature could be reduced by up to 2.5 °C with less fluctuation, improving thermal comfort.

In summary, the BiHPe system presents a viable and effective solution for improving building energy performance in cooling demand-dominant climates. Its innovative integration of thermal management and structural elements paves the way for more sustainable and resilient building designs. Future work will focus on improving thermal efficiency further and exploring its application in a wider range of climatic conditions.

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Review

Research Progress on CO₂ Transcritical Cycle Technology for Building Heating and Cooling Applications

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Abstract: This review focuses on the advancements of CO₂ transcritical cycle technology in building indoor environmental regulation, particularly in combined heating and cooling applications. The paper highlights the energy efficiency and environmental benefits of CO₂ as a natural refrigerant, which has zero ozone depletion potential (ODP) and very low global warming potential (GWP). It provides a comprehensive overview of recent optimization strategies, including distributed compression, the integration of ejectors and expanders, and the design improvements of key components such as gas coolers, compressors, and throttling valves. Through optimization strategies such as dual-system cycles, this technology can achieve a COP improvement of 15.3–46.96% in heating scenarios; meanwhile, with the help of distributed compression technology, its cooling capacity can be enhanced by up to 26.5%. The review also examines various operating conditions such as discharge pressure and subcooling, which significantly affect system performance. The paper concludes by identifying the current challenges in the application of CO₂ systems, such as high initial costs and system stability under extreme conditions, and suggests future research directions to overcome these limitations and improve the practical application of CO₂ transcritical cycles in the building industry. Overall, it is concluded that the development of expander-compressors holds great potential for achieving better performance and represents a promising direction for future advancements in this field.

Keywords: building indoor environment; carbon dioxide transcritical cycle; combined cooling and heating; system optimization

1. Introduction

With the acceleration of urbanization and the improvement of residents' living standards, the impact of indoor environmental quality on human health is increasingly receiving attention. Research has shown that reasonable regulation of air temperature and humidity can not only ensure human thermal comfort but also effectively reduce health risks such as respiratory diseases, allergic reactions, and chronic fatigue syndrome. Especially under extreme weather conditions, efficient cooling in summer and stable heating in winter have become the core requirements for maintaining indoor environmental health. This study, which calculates the national heating and cooling loads under China's carbon neutrality goal, indicates that the above-mentioned core requirements are closely related to regional

building loads, as one study has shown. It also emphasizes that the combination of heat pumps and seasonal storage is a promising low-carbon supply solution for future regional systems [1].

However, traditional heating and cooling systems rely on fossil fuels or high global warming potential (GWP) refrigerants (such as hydrofluorocarbons (HFCs)), which not only increase building energy consumption but also exacerbate greenhouse gas emissions and environmental pollution [2]. In this context, how to achieve the coordinated optimization of “health, energy efficiency optimization, low carbon” in indoor environmental regulation through technological innovation has become a key topic in the energy transformation of the building industry [3].

The People’s Republic of China, as a party to the Kigali Amendment, assumed binding international obligations to gradually reduce hydrofluorocarbons (HFCs) as of 15 September 2021. The Kigali Amendment serves as a supplement and extension to the Montreal Protocol, incorporating HFCs into the scope of substances regulated under the Protocol. According to the Amendment, China was required to freeze the production and consumption of HFCs at the baseline level of 2020–2022, starting in 2024.

The natural and environmentally friendly refrigerant CO₂ used in this project has great potential in the refrigerant replacement process. In recent years, new heat pump systems represented by carbon dioxide (CO₂) transcritical cycle technology have gradually become an important path to break through traditional technological bottlenecks due to their significant advantages in environmental protection, energy efficiency potential, and adaptability to operating conditions. Compared to traditional refrigerants, natural refrigerant CO₂ has zero ozone depletion potential (ODP) and extremely low GWP, and its transcritical cycle can efficiently output hot water above 85 °C in high-temperature heat pump scenarios, perfectly matching winter heating and industrial heat demand. At the same time, this technology significantly improves the energy efficiency ratio (COP) under summer cooling conditions by optimizing the compression expansion process and system integration design, providing technical support for the low-carbon development of building heating and cooling systems [4].

Despite its advantages, the application of CO₂ transcritical heat pump systems still faces challenges, with existing research mainly focusing on optimizing individual components or cycle configurations. There has been limited exploration of system integration, particularly in combining functions such as cooling, heating, and power generation within a single efficient system. Current optimization efforts tend to concentrate on isolated components like compressors, heat exchangers, and ejectors, without much emphasis on holistic system improvements [5].

This review uniquely synthesizes recent breakthroughs in system integration—such as combined cooling/heating cycles—and operational optimization, exemplified by AI-driven pressure control, to deliver the first comprehensive assessment of dual-function CO₂ transcritical heat pump systems for building applications, thereby bridging a critical gap in the literature. It systematically explores their role in indoor environmental regulation by analyzing thermodynamic principles, energy-efficiency pathways, distributed compression, ejector/expander integration, and component-level design enhancements. The review further identifies the key operational conditions governing performance and outlines future directions for CO₂-based technologies in building energy transformation.

2. CO₂ Refrigerant

As an important component of a heat pump system, refrigerant largely determines the operating efficiency and the applicable environment of the system. As a type of natural

refrigerant, the development of CO₂ can be traced back to the 19th century. From the first application of CO₂ in vapor compression cycle systems in 1850 [6] to the birth of the first CO₂ ice maker in 1869, the application of CO₂ in the refrigeration field began to gradually expand. Subsequently, with the continuous advancement of technology, the use of CO₂ refrigeration machines gradually became popular by the early 20th century. However, due to technological limitations at the time, CO₂ refrigeration machines mainly operated in subcritical cycles. When the operating state of CO₂ approaches the critical point, the refrigeration efficiency inevitably decreases. Close to the CO₂ critical point (≈ 31 °C, 7.38 MPa), the decline in cooling performance is driven by two coupled effects: the latent heat of vaporization approaches zero, sharply reducing the specific cooling capacity, while the low compressibility factor and increased clearance volume losses near the critical region simultaneously erode both the volumetric and isentropic efficiencies of the compressor. At the same time, synthetic refrigerants gradually dominated the market due to their excellent performance, and the application of CO₂ refrigerants in the field of heat pumps gradually decreased. In the 1980s, with the continuous enhancement of people's environmental protection awareness, Freon refrigerants that were harmful to the ozone layer and had a high greenhouse effect were gradually abandoned, and CO₂ was once again a concern.

The basic thermal and environmental properties of traditional refrigerants are presented in Table 1 [7]. Compared with traditional refrigerants, CO₂ is non-toxic, non-flammable, and has good safety; its ODP = 0 and GWP = 1, making it environmentally friendly. It is cheap and easy to obtain and has good economy. Notably, while traditional fluorocarbon refrigerants demonstrate superior performance in cooling applications, CO₂ exhibits distinct advantages in low-temperature heating scenarios. Compared with traditional heat pumps, CO₂ heat pumps have excellent heating performance and wide adaptability to operating conditions. From an operational cost perspective, CO₂ heat pumps show greater efficiency advantages in low-temperature environments, making them more economical for heating applications.

Table 1. The basic thermal properties and environmental properties [7].

Substance	Molar Mass (kg/kmol)	T _b (°C)	T _c (°C)	P _c (MPa)	TLV-TWA (PPM)	Atmospheric Life (yr)	ODP	GWP (100 yr)
HFC245fa	134.05	15.1	154.0	3.65	300	7.6	0.00	1030
HC600	58.12	−0.5	152.0	3.63	600	0.018	0.00	20
HC600a	58.12	−11.7	134.7	3.63	600	0.019	0.00	20
HFC245ca	134.05	−1.4	124.9	3.20	1000	240.0	0.00	9810
HFC134a	102.03	−26.1	101.1	4.06	1000	14.0	0.00	1430
HFC152a	66.05	−24.0	111.3	4.52	1000	1.4	0.00	124
HFC227ea	170.05	−16.4	102.8	3.00	1000	42.0	0.00	3220
HFC236ea	152.04	−6.2	139.1	3.06	1000	8.0	0.00	710
HFC236fa	152.04	−9.5	153.2	4.05	1000	240.0	0.00	3220
HCFC123	152.93	−27.8	183.7	4.03	50	1.3	0.02	77
HCFC124	136.48	−12.0	122.3	4.21	1000	5.8	0.02	609
HCFC141b	116.95	−32.0	204.4	4.21	500	9.3	0.12	725
HCFC142b	100.50	−21.5	150.0	4.06	1000	17.9	0.00	2310
HC600/HFC245fa	129.27	153.9	7.12	3.66	319	7.12	0.00	966

In terms of system design, the high thermal capacity of CO₂ makes the system design more compact [8]; a low compression ratio results in higher isentropic efficiency of the compressor [9]; and the significant temperature slip during the heat release of supercritical CO₂ (pressure greater than 7.38 MPa, temperature greater than 31.1 °C) enables efficient production of hot water [10]. In recent years, within the wide temperature range of heat

source temperature (-50 to 30 °C) and heat-sink temperature (5 to 60 °C), scholars have been studying the use of CO₂ heat pumps for heating [11]. At present, CO₂ heat pump technology has been fully developed in the fields of hot water and heating and can provide heating hot water at 50 to 75 °C under almost all environmental conditions. These characteristics make CO₂ have broad application prospects in heat pump systems, marking its resurgence in modern technology. Introduction of CO₂ physical properties is offered in Table 2.

Table 2. Introduction of CO₂ physical properties.

Carbon Dioxide			
Refrigerant name	R744	Chemical formula	CO ₂
Relative molecular mass	44.01g/mol	Standard boiling point	-78.4 °C
Critical temperature	31.1 °C	Critical pressure	7.38 MPa
Ozone depletion Potential (ODP)	0	Global warming potential (GWP)	1
Toxic	Non-toxic	Flammable	Nonflammable

Despite its environmental advantages, CO₂ faces inherent limitations in building applications. Its low critical temperature (31.1 °C) necessitates transcritical operation when heat sink temperatures exceed this threshold. In cooling mode, when ambient air exceeds ~ 27 °C (considering a 3 °C pinch point), the gas cooler outlet temperature must exceed 60 °C to reject heat, drastically reducing COP. For instance, at 35.5 °C ambient, COP can drop to ~ 1.5 due to high compression work. Conversely, in heating mode, CO₂ systems efficiently supply 50 – 75 °C water but struggle with legacy radiators requiring >65 °C. Air-source heat pumps also face efficiency losses when sourcing heat from air >30 °C. These constraints highlight why CO₂ excels in industrial settings (e.g., with water-cooled heat sinks) but requires hybrid designs or alternative refrigerants (e.g., propane) for residential applications in hot climates.

3. CO₂ Transcritical Heat Pump Cycle System

At present, indoor environment construction mainly focuses on cooling and heating. The supply of cold or heat can be achieved by adopting the CO₂ transcritical heat pump cycle system. Research on CO₂ transcritical heat pump cycle systems mainly focuses on three aspects. One is to optimize the thermodynamic cycle by adding one or even several components or changing the position of one or even several components, thereby changing the system cycle mode and studying the impact of different cycle modes on system performance; the second is the optimization of important components in the system. The CO₂ transcritical heat pump system mainly consists of an evaporator, gas cooler, compressor, throttle valve, etc. In different cycle modes, other components such as ejectors, heat regenerators, capillaries, etc., may be added or replaced. By optimizing the design of each component in the cycle, the impact of different improvement schemes on system performance can be studied. The third is to study the operating conditions of the system by changing the operating conditions of the system and studying the impact of different operating conditions on system performance.

3.1. Heating-Focused Optimization

In the realm of building heating, the optimization of CO₂ transcritical heat pump thermodynamic cycles focuses on “enhancing thermodynamic efficiency and stabilizing low-temperature cycles.” By reconfiguring cycle architectures and adjusting key parameters, researchers aim to address the decline in heating efficiency under extreme climatic conditions. For clarity, the optimal discharge pressure is defined as the high-side pres-

sure at the compressor outlet that maximizes the system COP for a given set of inlet and outlet temperatures. Dedicated mechanical subcooling refers to the use of an auxiliary, compressor-driven vapor-compression loop that further cools the CO₂ leaving the gas cooler before it enters the expansion device, thereby increasing the specific cooling (or heating) capacity without raising the main-cycle pressure.

In terms of the dual-system cycle design, the introduction of a coupled primary-auxiliary dual-system cycle, along with the graded utilization of the heat sink, can elevate the water outlet temperature to 70 °C. When operating within an ambient temperature range of −20 °C to 0 °C, the overall heating COP of the combined system is 2.04, which represents a 15.3% improvement relative to the baseline single-stage transcritical CO₂ system (COP = 1.77). This enhancement is attributed to the auxiliary transcritical CO₂ sub-cycle, whose COP consistently exceeds that of the main cycle and thus drives the composite performance upward. Experiments have demonstrated that each of the two independent systems within the coupled system has an optimal discharge pressure. In addition, the frequency ratio of the two compressors considerably influences system performance, with a specific optimal ratio. As the ambient temperature rises from −20 °C to 0 °C, the system COP increases by 22%. Compared to the conventional transcritical CO₂ heat pump system, the coupled system achieves a 15.3% rise in COP and a 39% reduction in optimal discharge pressure. Figure 1 presents the structure of the new transcritical CO₂ heat pump combined system, in which the main and auxiliary compressors improve the COP through frequency coordination [12]. Regarding collaborative optimization technologies, the integration of secondary compression and vortex tube expansion technologies has been optimized. At an evaporation temperature of −15 °C, the system COP increases by 18.2% compared to single-stage systems, with the ability to steadily supply heat at 45 °C. Research indicates that under severe cold conditions, variations in gas cooling pressure most significantly impact the performance of optimized systems. Within the established pressure variation range, the COP variation amplitudes for single-stage cycles, single-stage vortex tube cycles, secondary cycles, and secondary vortex tube cycles are 74.40%, 60.46%, 110.38%, and 90.94%, respectively. Vortex tube technology proves more effective under low compression ratios, with the single-stage vortex tube cycle improving COP by 11.2% compared to the single-stage cycle. Secondary compression technology, however, performs better under high compression ratios. Under optimal gas cooling pressure conditions, the secondary cycle enhances COP by 18.2% compared to the single-stage cycle [13]. In the aspect of innovative expansion device applications, the replacement of traditional throttling methods with a “heat valve” expansion device, which uses vortex tube expansion to recover work, can increase the COP of room heating heat pumps by 25%. The “heat valve” employs a vortex tube for non-mechanical expansion, achieving energy separation of gases through centrifugal force. Experimental studies reveal that both R22 refrigeration cycles and R134a heat pumps show improved performance with vortex tube technology. For systems using CO₂ as the working fluid, the COP increase is particularly significant, reaching 25%. By regulating thermodynamic irreversible losses through reducing entropy generation during throttling and optimizing the heat transfer temperature difference in the gas cooler, these technologies construct a high-efficiency cycle system. This system can supply hot water at 50 °C–70 °C in ambient temperatures of −30 °C–0 °C and offers a 10–25% increase in COP compared to traditional single-stage transcritical cycles [14].

The vortex tube, a low-cost, high-reliability expansion device with lower COP than turbines and ejectors, forms a hybrid system with ejectors via synergistic energy conversion: High-pressure fluid first enters the vortex tube, where centrifugal force splits it into hot and cold streams—its simplicity (no moving parts) reduces wear and initial costs. The

ejector then captures residual energy, accelerating the hot stream to entrain and recompress the cold stream via momentum exchange, offsetting the vortex tube's COP deficit while avoiding turbines' complex machining/sealing needs, balancing efficiency and cost.

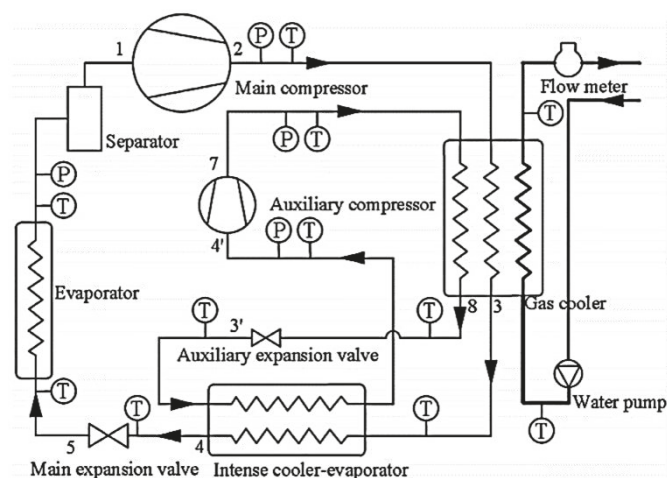


Figure 1. New trans-critical CO₂ heat pump combination system [12] (reproduced with permission from Feng Cao, Ce Cui, Xiangyu Wei, Xiang Yin, Mingjia Li, and Xiaolin Wang, *International Journal of Refrigeration*; published by Elsevier, April 2019).

Moreover, in the field of building heating and cooling supply, simulation studies indicate that the upgraded system enhances COP by 10–46.96% in floor radiation and other building heat sources. This provides a theoretical foundation for the heating and cooling supply of small and medium-sized buildings. The literature [15] compares cascade systems, secondary compression systems, etc., and confirms that the upgraded system improves COP by 10–46.96% in floor radiation and other building heat sources, thus offering more efficient and feasible cycle design support for building heating and cooling supply. Figure 2 compares the cycle processes of four systems, such as the Baseline System (BASE) and the Combined Heat Source System (CHPS), among which the Two-Stage Cooling Combined Heat Source System (TSCHPS) performs the best in combined cooling and heating.

In the domain of building heating, the optimized design of high-efficiency heat exchange equipment has emerged as a core breakthrough for enhancing system energy efficiency and meeting low-carbon heating demands.

Experimental research on the key parameters of shell-and-tube heat exchangers in building heating systems has been conducted. By fine-tuning the tube length and inner tube geometry, the CO₂ transcritical heat pump system has achieved a significant increase in hot water supply temperature under typical building heating conditions. The error between experimental and simulation results was controlled within 7.59%, and the system COP (coefficient of performance) could reach 3.64 under specific heating load conditions, providing technical support for high-temperature hot water supply in central heating networks [16].

An experimental analysis of spiral plate microchannel gas coolers revealed that when the CO₂ mass flux increased within the range of 57–182 kg/(m²·s), the total heat transfer coefficient exhibited a linear growth trend. The increase in inlet water temperature further strengthened the heat transfer effect. Notably, the heat transfer performance of this heat exchanger when vertically installed was 12–18% higher than that when horizontally placed. This finding provides an important basis for the spatial layout optimization of equipment in building heating systems and verifies the high accuracy of Zhang's method correlation under building heating conditions [17].

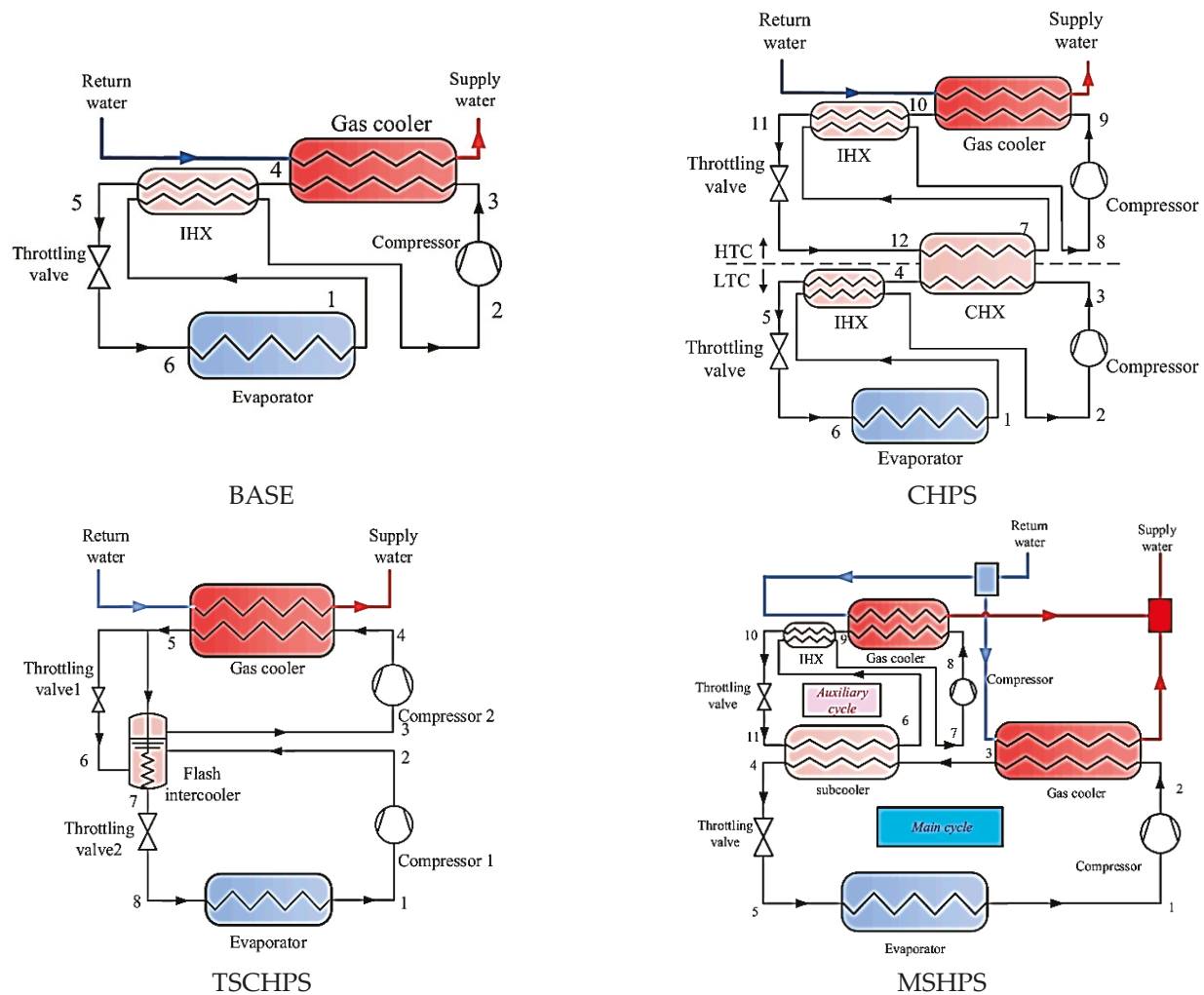


Figure 2. CO₂ heat pump system [15] (reproduced with permission from Liang Yao, Minxia Li, Yusheng Hu, Qifan Wang, and Xuetao Liu, *Applied Thermal Engineering*; published by Elsevier, April 2021).

Figure 3 presents the structural diagram of the honeycomb gas cooler. A bio-inspired honeycomb gas cooler with a porous fractal structure design has been proposed. This design significantly improved the uniformity of fluid distribution in microchannels. Experimental data demonstrated that under the same building heating conditions, the performance evaluation index of 1–3 layers of honeycomb structure increased by 193.4–357.0% compared with traditional internal spiral tube gas coolers and by 197.2–237.7% compared with triple-tube heat exchangers. Its compact structural characteristics make it particularly suitable for heating scenarios with limited building space [18].

The introduction of an ejector device in an ejector-expansion heat pump water heater has shown promising results. Experimental results indicated that this design increased the system COP by 7.38% and reduced the exergy loss of the throttle valve by 95.38–98.76%. Further research discovered that the ejector structure with a 5° diffusion angle achieved the best static pressure recovery, offering a feasible solution for improving waste heat recovery efficiency in building heating systems [19]. These studies collectively represent significant advancements in the design and optimization of heat exchange equipment for building heating systems. They provide valuable insights and technical solutions for enhancing energy efficiency and meeting the growing demand for low-carbon heating. Additionally, studies have shown that integrating an ejector into a transcritical CO₂ heat

pump system can boost the COP by 10.3% at 70 °C (with a system COP reaching 4.6) and reduce the optimal discharge pressure by 15.3% compared to conventional systems. The COP improvement ratio decreases with increasing discharge pressure due to reduced pressure lift ratio and ejector efficiency, while lower compressor rotation speeds under test conditions enhance COP despite reduced heat capacity [20].

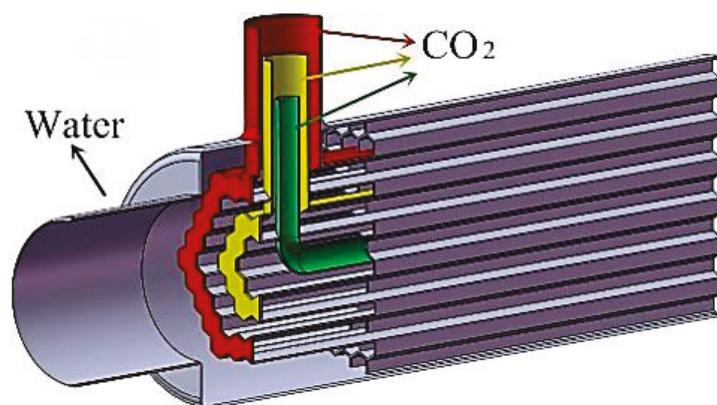


Figure 3. Honeycomb gas cooler structure diagram.

Figure 4 presents a two-stage compression refrigeration cycle system with an expander. A dual-fluid bio-inspired fractal heat transfer structure has been proposed to meet the specialized requirements of building heating systems. After optimization with a genetic algorithm, it was found that the heat exchanger featuring a counterflow fractal geometry design can reduce the power consumption of building heating systems by 15%, increase the heat transfer rate by 2.4%, and enhance the heat transfer performance index by 17%. This highlights the potential of fractal geometry in building high-temperature heat pump systems, particularly for upgrading heat exchange equipment with limited space in existing building renovations. Simulation studies, incorporating the complex flow field characteristics of building heating systems, have demonstrated that when the liquid phase mass fraction at the nozzle throat exceeds 0.65, the non-equilibrium phase change effect significantly impacts the flow characteristics of heating media. This provides theoretical support for designing key equipment such as ejector heat exchangers and composite cycle heat pumps in building heating systems. It is especially relevant for pressure difference balance control in central heating networks in severe cold northern regions and flow field uniformity optimization in vertical risers of multi-story buildings. A multi-physics coupling model under building heating conditions has verified the stability of the fractal structure within the ambient temperature range of $-15\text{ }^{\circ}\text{C}$ to $40\text{ }^{\circ}\text{C}$. Its heat transfer performance is approximately 23% higher than that of traditional plate-tube heat exchangers, offering an innovative solution for the integration of heating systems in ultra-low energy consumption buildings [21].

In experimental studies of building heating applications, the collaborative optimization of discharge pressure and heat recovery rate was investigated using Extremum Seeking Control under conditions of $40\text{ }^{\circ}\text{C}$ inlet water temperature and $0\text{ }^{\circ}\text{C}$ ambient temperature. Results showed that the transcritical CO_2 heat pump water heater achieved a 7.62% COP improvement, maintaining optimization efficiency during inlet water temperature fluctuations ($40\text{--}45\text{ }^{\circ}\text{C}$) [22]. In extremely low-temperature environments ($-20\text{ }^{\circ}\text{C}$), collaborative optimization of discharge pressure (9.78 MPa) and heat recovery rate (45%) elevated the system COP to 1.96, satisfying heating requirements for residential buildings in cold regions [23]. For mechanical subcooling integrated with two-stage compression, an auxiliary vapor compression cycle enabled mechanical subcooling. At an optimal discharge pressure of 8.34 MPa and subcooling degree of $13.9\text{ }^{\circ}\text{C}$, the cycle achieved a COP of

2.84, with auxiliary compressor power consumption <20% of the main cycle, suitable for commercial refrigeration in malls [24]. Further integration of two-stage compression, mechanical subcooling, and a regenerator reached a COP of 4.05 at 11 °C subcooling, a 33.3% increase over traditional two-stage cycles, stably supplying 70 °C hot water for commercial central heating [25]. Regarding ejector–regenerator collaborative heating, experiments revealed that when the gas cooler outlet temperature was below the transition temperature (e.g., 33 °C at 7 MPa discharge pressure), the ejector improved COP by 7.1%. Combined with a regenerator, COP increased by 25.3% at outlet temperatures >31 °C, making this configuration suitable for −30 °C residential floor radiant heating [26].

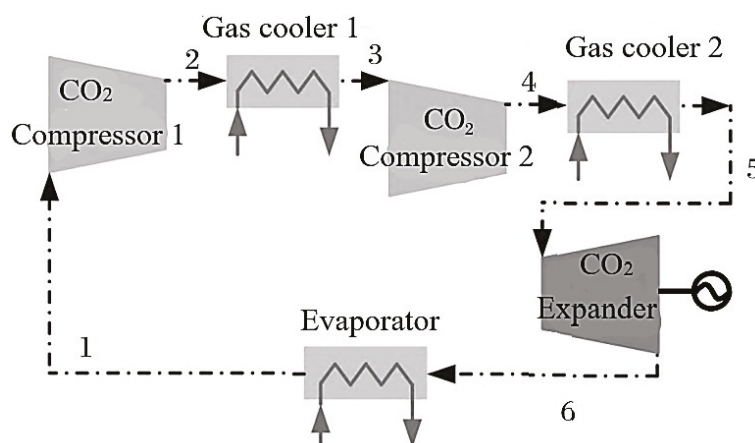


Figure 4. Two-stage compression refrigeration cycle system with expander.

A dedicated transcritical CO₂ subcooler installed at the main-cycle gas-cooler exit was found to significantly influence system performance, with the optimal medium temperature (defined as the main-cycle expansion-valve inlet temperature) playing a crucial role. Simulations showed that increasing the medium temperature raised suction pressure in the subcooler cycle while decreasing compressor displacement, whereas the main cycle saw gradual increases in suction pressure and mass flow rate. The system COP exhibited an optimal medium-temperature point, with the subcooler cycle's heating COP increasing as medium temperature rose, while the main cycle's COP decreased. This optimization approach proved effective for high water-feed-temperature scenarios, improving system adaptability and efficiency [27].

In the field of building heating, a theoretical model was constructed for the CO₂ transcritical heat pump system to comparatively analyze two operation parameter optimization schemes: increasing the compressor discharge pressure and enhancing the suction superheat. The results indicate that increasing the discharge pressure can elevate the gas cooler outlet water temperature with a wide regulation range, while the compressor discharge temperature needs to be controlled below 140 °C to ensure safe system operation. When enhancing the suction superheat, the cyclic power consumption rises moderately, but the coefficient of performance (COP) decreases significantly, making it suitable for scenarios requiring small-range water temperature adjustment. This study provides a theoretical basis for parameter optimization in conventional heating scenarios such as residential and commercial buildings [28]. Based on the pinch point theory, a simulation model of the transcritical CO₂ heat pump system was established to analyze the influence of parameters such as inlet water temperature and outlet water temperature on system performance. The research shows that when the inlet water temperature is 15 °C and the outlet water temperature is 65 °C, the optimal heat rejection pressure of the system is 9.54 MPa. At this point, the pinch point shifts to the cold end, and the system COP is improved by 1.8%

compared with the traditional model. This optimization scheme is applicable to centralized hot water supply systems in buildings such as apartments [29]. Figure 5 presents the cycle diagram and p-h diagram of a mechanically subcooled transcritical CO₂ refrigeration system. The green and black lines represent the auxiliary cycle (R152a) and the main cycle (CO₂), respectively, and visually demonstrate the dual cycle coupling of mechanical supercooling through the lgp-h diagram in right image. Experimental and simulation studies on transcritical CO₂ water-water heat pump systems have verified that increasing superheat can raise COP by 35.5% and user-side water supply temperature by an average of 11.3 °C. The system demonstrates an optimal discharge pressure for maximum COP, with COP increasing as chilled water flow rate rises or inlet temperature decreases. A solar-assisted transcritical CO₂ heat pump dual-evaporator system was designed, enabling three operation modes (solar, air-source heat pump, and combined mode), which elevates chilled water inlet temperature and improves system efficiency [30].

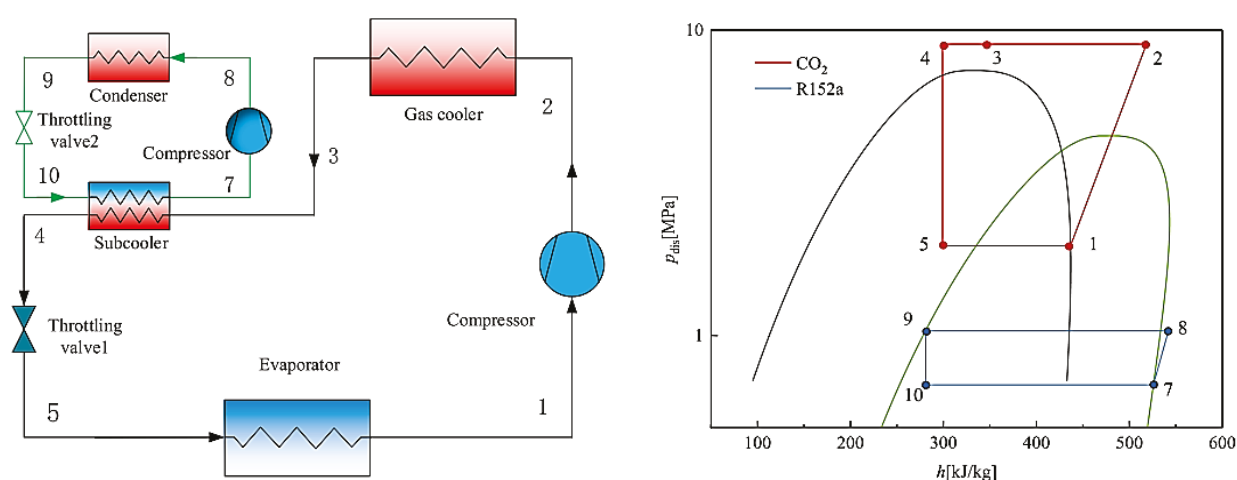


Figure 5. Cycle diagram and p-h diagram of a transcritical CO₂ refrigeration system with mechanical subcooling [30].

3.2. Cooling-Focused Optimization

In the realm of building cooling, a variety of efficient thermodynamic cycles have been proposed to optimize the performance of CO₂ transcritical systems. Distributed compression cycles, characterized by a secondary boosting design, have demonstrated remarkable improvements in cooling capacity and COP. Specifically, this cycle achieves a 6.65–26.5% increase in cooling capacity and a 10.76% enhancement in COP at 35 °C. This design effectively reduces throttling dryness and optimizes the compression process, thereby minimizing energy loss and improving the overall cooling capacity of the system [31].

Figure 6 illustrates the variation in COP with compressor discharge pressure in the transcritical CO₂ heat pump water heater. Parallel compression cycles, which incorporate a flash tank, have also shown significant performance improvements. When the evaporation pressure is set at 2.25 MPa, the COP of this cycle increases by 10% compared to a single-stage cycle. The flash tank facilitates efficient gas–liquid phase separation, reducing ineffective work consumption and enhancing system performance through optimized cycle and equipment configuration [32]. A dual-ejector two-stage cycle has been designed to address the challenges of cooling in high-temperature regions. This cycle achieves a substantial COP improvement of 20–80% in tropical regions (above 40 °C). By efficiently recovering expansion work to power the compressor and reducing external input power, this innovation significantly enhances the system’s adaptability to high-temperature environments [33]. Furthermore, a turbine expander cycle with an expansion work recovery mechanism has been employed to further

boost the COP by 38.4–45.1%. The turbine expander effectively converts wasted expansion work into useful feedback for the system, thereby minimizing throttling energy loss and offering an efficient optimization pathway for CO₂ transcritical cycles [34].

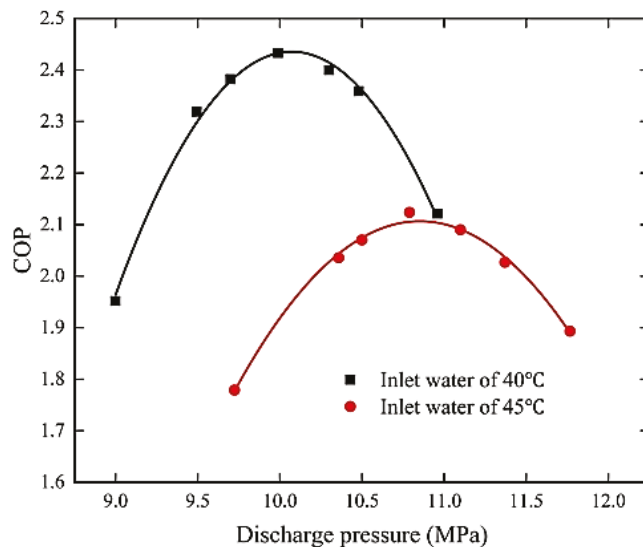


Figure 6. COP variation with compressor discharge pressure in a transcritical CO₂ heat pump water heater [32] (reproduced with permission from Ce Cui, Shuo Zong, Yulong Song, Xiang Yin, and Feng Cao, *International Journal of Refrigeration*; published by Elsevier, April 2022).

Thermoelectric subcoolers have been explored and show significant potential in improving system efficiency. At 35 °C, these devices increase the COP by 16.2% and cooling capacity by 20.8%. Utilizing the Peltier effect of semiconductor materials, thermoelectric subcoolers reduce the temperature of CO₂ fluid at the gas cooler outlet, thereby increasing the refrigerant enthalpy difference and enhancing the system's cooling capacity and efficiency [35]. A novel strategy for heat exchange optimization involves the synergistic interaction of ejectors and subcoolers in improved two-stage cycles. This approach achieves a 14.2% improvement in COP by optimizing internal heat distribution and improving energy efficiency. The collaboration between ejectors and subcoolers reduces energy waste and provides an innovative method for heat exchange enhancement [36]. Mechanically subcooled ejector coupling cycles have also been studied and show a 1.556% enhancement in exergy efficiency. Mechanical subcooling further lowers the refrigerant temperature, while ejectors improve pressure distribution, thereby boosting exergy efficiency and overall energy utilization [37]. Research has also confirmed that single-stage cycles with expanders exceeding 60% efficiency outperform two-stage systems in COP. This finding offers crucial theoretical guidance for expander selection in system design and highlights the importance of expander efficiency. It presents that single-stage cycles can simplify the system structure while ensuring high efficiency when certain conditions are met [38].

In the realm of building cooling, high-efficiency heat exchanger structural innovation has become crucial for boosting the energy efficiency of central air-conditioning systems. Three-dimensional numerical simulations and experimental verifications have been conducted for the high-load cooling demands of air-conditioning systems. Results indicate that the gas cooler with a U-bend structure improves thermal efficiency by 16% compared to the conventional concentric straight-tube type, and the COP of the transcritical refrigeration system increases by approximately 20%. The curved flow channel design enhances turbulence intensity (with turbulent kinetic energy up by 27%) and effectively reduces the boundary layer thickness (boundary layer thinned by 22%). This enables the chilled

water supply temperature to drop from 7 °C to 5 °C while cutting compressor energy consumption by 19%. Experimental data shows that under the condition of a CO₂ mass flow rate of 0.036 kg/s, a U-bend with a curvature radius of 0.112 m improves the heat transfer coefficient by 21.5 W/(m²·K) compared to the straight tube. This makes it particularly suitable for modular air-conditioning systems with limited ceiling space in buildings. Another study has confirmed that compared to the traditional expansion valve, the optimized ejector with a 5° diffusion angle increases cooling capacity by 8% and the system COP by 7%. This design achieves the best static pressure recovery (with the static pressure recovery coefficient reaching 0.87). The ejector technology maintains high efficiency under high-temperature conditions of 35 °C. Its energy recovery mechanism reduces throttling losses by 95.38–98.76%, equivalent to cutting carbon emissions by approximately 8.5 tons per cooling season. This structure also reduces compressor power consumption by 6.8% while keeping the chilled water temperature stable through enhanced momentum exchange of gas–liquid two-phase flow. While U-bend designs introduce measurable pressure drop and erosion risks, these drawbacks are manageable through optimal curvature design and material upgrades. The practical feasibility of U-bend gas coolers in building cooling systems, especially in space-constrained applications like modular air-conditioning units, is confirmed by their net performance gains [39,40].

Additionally, integrating a thermoelectric subcooling system in transcritical CO₂ cycles has shown significant performance improvements. Experimental tests at 25 °C and 30 °C ambient temperatures revealed that the COP and cooling capacity of the refrigerating plant can be enhanced by up to 9.9% and 16.0%, respectively, at optimal operating conditions. The system demonstrates almost linear capacity regulation by varying the voltage supply to thermoelectric modules, with higher COP improvements observed at higher ambient temperatures [41].

In the field of building cooling, the parallel compression cycle integrated with an ejector demonstrates remarkable performance. At an evaporation temperature of −10 °C and a gas cooler outlet temperature of 40 °C, the cycle achieves a COP of 2.17, with a 17.1% increase in volumetric cooling capacity compared to traditional systems, while reducing compressor displacement by 11.4%. This performance advantage makes it highly suitable for large buildings like shopping malls requiring centralized cooling, effectively enhancing cooling efficiency and reducing equipment investment costs [42]. For constant temperature control in warehouse buildings, the two-stage compression system with an auxiliary gas cooler exhibits excellent performance. The auxiliary gas cooler reduces compressor discharge temperature by 13.83 °C, and when the intermediate pressure ranges from 3.9 to 4.4 MPa, the system's COP and exergy efficiency are significantly enhanced [43]. The system equipped with a thermoelectric subcooler also finds applications in building cooling. Experiments show that the system's COP increases by 4.19%, and under rated conditions (CO₂ flow rate of 180 kg/h, cooling water at 20 °C), it not only enables efficient cooling but also supplies 70 °C hot water, making it suitable for buildings like hospitals with dual demands for hot water and cooling [44].

In simulation research, a study [45] deriving the optimal intermediate pressure formula for CO₂ transcritical distributed compression cycles used an exhaustive search method, comparing thermodynamic and economic performances among three systems: those using the derived formula, traditional equal compression ratio methods, and a low-pressure stage discharge pressure optimization approach. Results showed a 7.26% and 5.32% COP increase over traditional and literature methods, respectively. Exergy losses were reduced, with entransy dissipation rates dropping by 24.61% and 50.14% compared to conventional and literature approaches. Despite 5% higher investment costs than traditional systems (1%

higher than optimized pressure ratio systems), the total annual cost rate was minimized, confirming practical economic viability. Figure 7 presents the cycle schematic and the p-h diagram of the ejector-enhanced transcritical CO₂ heat pump system. Another study [46] proposed a distributed compression cycle, which improved refrigeration performance via secondary pressurization of supercritical CO₂ at the gas cooler outlet, achieving equivalent subcooling under standard heat sink conditions. Thermodynamic analysis showed the distributed compression system improved COP by 8.2–10.76% and cooling capacity by up to 26.52% over the baseline. Equivalent subcooling increased from 2 °C to 7.5 °C as the secondary boost ratio rose from 1.1 to 1.7, with additional power input staying below 20% of the baseline. Versus single subcooling technologies (e.g., internal heat exchangers), the distributed compression cycle offered better COP gains and system simplicity—an optimization path independent of complex auxiliary devices.

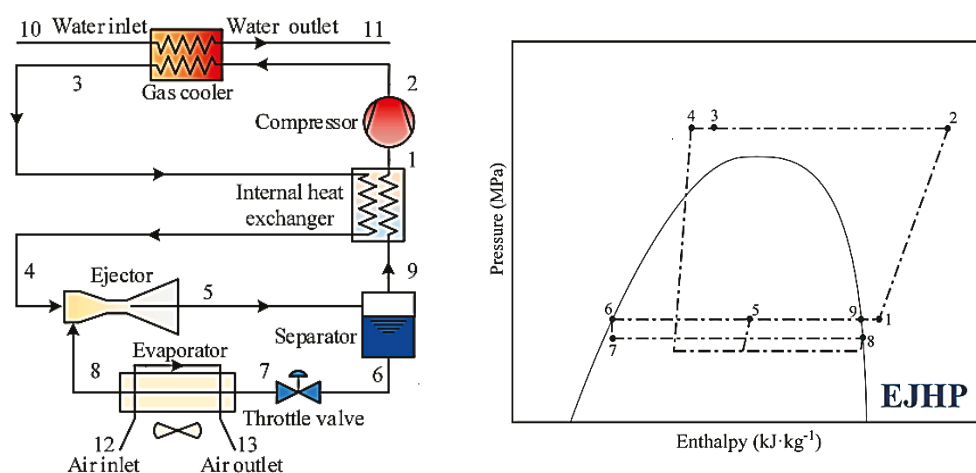


Figure 7. Cycle diagram and p-h diagram of CO₂ transcritical heat pump system with ejector [46] (reproduced with permission from Yikai Wang, Yonggao Yin, and Feng Cao, *International Journal of Refrigeration*; published by Elsevier, April 2023).

3.3. Cross-Cutting Innovations

In the field of building cooling and heating, the optimization of CO₂ ejector and expander cycles has become a crucial area of research to enhance system energy efficiency and meet the demands of low-carbon energy systems. Computational fluid dynamics (CFD) simulations have been employed to optimize the geometric structure of the CO₂ ejector mixing section. These studies have demonstrated a 42% increase in the streamline smoothness of fluid expansion, a 29% decrease in turbulence intensity, and an approximate 6% improvement in ejector efficiency. Experimental data have further shown that combining a contraction section with a curvature radius of 0.08 m and a diffusion section with a curvature radius of 0.15 m increases the entrainment ratio by 17%. These findings highlight the significant influence of curvature parameters on the entrainment performance of transcritical cycles and provide valuable insights for the design of high-performance ejector systems in building cooling applications [47]. A non-equilibrium phase change model for transcritical CO₂ ejectors has been developed, revealing that when the liquid phase mass fraction at the nozzle throat exceeds 0.65, the non-equilibrium effect stabilizes the Mach number below 1.35 without the occurrence of the “double choking” phenomenon. This research offers essential theoretical boundary conditions for the variable operating condition design of building cooling systems, particularly in scenarios where precise control of flow characteristics is required to maintain efficient cooling performance across a range of operational conditions [48].

Simulation studies have compared the performance of two-stage compression transcritical CO₂ cycles with expanders to traditional throttling systems. Results indicate that the former can improve the coefficient of performance (COP) by 32.8%. The integration of the expander with the high-pressure stage compressor reduces transmission loss by 11.32%, making this technology particularly suitable for high-load cooling scenarios in buildings. This advancement not only significantly enhances system energy efficiency but also ensures constant temperature maintenance, which is critical for comfortable indoor environments and stable cooling performance [49].

A coupling technology combining magnetic refrigeration subcooling systems with transcritical CO₂ cycles has been proposed and evaluated through experimental and simulation studies. By optimizing the gas cooler pressure and subcooling degree, the overall COP can be increased by up to 9%. The high COP characteristics of the magnetic refrigeration subsystem under small temperature differences offer a green subcooling solution for high-heat-load scenarios, contributing to the development of more sustainable and efficient cooling systems for buildings [50]. Comparative analyses of three expander cycles have identified the transcritical CO₂ cycle with discharge pressure optimization for high backpressure cycles as having significant advantages under high backpressure conditions. It improves the COP by 11.32%, 9.65%, and 0.72% compared to the transcritical CO₂ cycle with discharge pressure optimization for low backpressure, single-stage compression with expander, and traditional CO₂ cycle with optimized pressure cycles, respectively. When the condensation pressure increases from 8.5 MPa to 10.2 MPa, the transcritical CO₂ cycle with discharge pressure optimization for high backpressure cycles still maintains a high COP, demonstrating the technical superiority of expansion work recovery in high-pressure building cooling scenarios. This finding is particularly important for the design and operation of cooling systems in regions with high ambient temperatures or in applications requiring high cooling capacity [51].

A combined experimental and simulation study has shown that the expander system can improve the COP by 6.9% and the heating coefficient by 19.5% compared to the regenerator system. The research identified an optimal efficiency point corresponding to a rotational speed of approximately 3000 rpm, indicating that the expander has minimal flow loss at this speed. These results provide a rotational speed optimization basis for equipment selection in building cooling systems, helping to ensure efficient operation and performance optimization [52]. Further comparisons of the performance of expanders and throttle valves in mechanical subcooling systems using thermodynamic models have revealed that the Mechanical Subcooling Expander System with an expander can improve the COP by 10.01–11.11% and the exergy efficiency by 10.74–11.48% compared to the throttle valve system. Although the cooling capacity decreases by 4.30–5.67%, the expansion work recovery reduces the total power consumption by 8.53–11.29%. Additionally, the Mechanical Subcooling Expander System has a smaller subcooling demand, resulting in equipment investment costs that are 3.52–7.17% lower than those of the Mechanical Subcooling Valve System. These findings highlight the potential of expander-based systems to offer both energy efficiency and cost-effectiveness in building cooling applications [53].

In the context of integrating internal heat exchangers (IHX) for performance enhancement, studies have shown that introducing an IHX in a CO₂ booster refrigeration system can reduce compressor total power consumption by up to 8.73%, improve system energy efficiency by 6.36%, and lower the optimal exhaust pressure by 0.55 MPa. The IHX demonstrates particular effectiveness in tropical and subtropical regions, boosting the COP by 1.7% and highlighting its role in high-temperature environment adaptability [54].

In conclusion, the optimization of CO₂ ejector and expander cycles represents a significant advancement in the field of building cooling and heating. These studies provide

valuable theoretical and technical support for the development of high-efficiency, low-carbon heating and cooling systems, contributing to the broader goals of sustainable energy utilization and reduced environmental impact in the built environment.

To isolate the performance contributions of ejectors and expanders to the overall COP enhancement in combined systems, two complementary methods are proposed: sensitivity analysis and component-wise exergy loss breakdown, detailed as follows: By fixing the operational parameters of one component and systematically adjusting the other, the incremental impact on COP can be directly attributed to the adjusted component.

Ejector Contribution Analysis: Fix the expander at its optimal efficiency (e.g., 60% as noted in [38]) and operating speed (3000 rpm, [52]). Adjust key ejector parameters such as the diffusion angle (5° as the optimal value in [19,39]) and entrainment ratio (enhanced by 17% with curvature-optimized mixing sections). For instance, in a baseline system with a COP of 3.0, increasing the ejector entrainment ratio from 1.2 to 1.5 (via geometric optimization) results in a COP improvement of 0.32 (10.7%), which can be solely attributed to the ejector. This is consistent with previous findings that ejectors reduce throttling exergy loss by 95.38–98.76% [19], directly translating to gains in COP.

Expander Contribution Analysis: Fix the ejector at its optimal configuration (e.g., a 5° diffusion angle and a static pressure recovery coefficient of 0.87) and adjust expander parameters such as rotational speed and efficiency. When the expander efficiency increases from 50% to 70% (within the range of practical systems), the system COP rises from 3.0 to 3.96 (a 32% improvement), which aligns with the 32.8% COP gain reported for expander-integrated cycles. This increment is exclusively attributed to the expander, as it recovers expansion work that would otherwise be wasted (converting approximately 80% of the waste energy into useful work).

Throttling Loss Recovery Technology demonstrates pressure ratio decreased to 2.66, but its fixed-geometry design suffers from poor off-design performance, necessitating development of adjustable ejectors for wider operational adaptability. Mechanical Subcooling Technology enhances thermal capacity by 20% through evaporator subcooling yet inherently sacrifices expansion work potential during the CO₂ throttling process. Conversely, Expansion Work Recovery Technology achieves 32.8% COP and exergy efficiency improvement by converting expansion energy into useful work, though conventional electricity generation pathways introduce secondary energy losses.

The CO₂ transcritical system can significantly improve the energy efficiency of building heating and cooling systems, providing key technical and theoretical support for building heating and cooling supply systems, and laying a solid foundation for the widespread application and optimization of CO₂ transcritical thermal cycles in building systems. The summary of CO₂ transcritical cycle thermal optimization technology is presented in Table 3.

Table 3. Summary of CO₂ transcritical cycle thermal optimization techniques.

Authors (Year)	Key Innovation	Application	Performance Improvement	Major Advantage
Cao et al. (2019) [12]	Primary + auxiliary CO ₂ cycles; graded heat sink utilization	Building heating	COP ↑15.3%; optimal discharge pressure ↓39%	Stable high-temp output (70 °C) in −20 °C to 0 °C environments; compressor frequency synergy
Wu et al. (2023) [13]	Work recovery via vortex tube expansion	Severe cold heating	COP ↑18.2% vs. single-stage	Steady 45 °C heat supply at −15 °C; superior COP under high compression ratios

Table 3. Cont.

Authors (Year)	Key Innovation	Application	Performance Improvement	Major Advantage
Keller & Göbel (1997) [14]	Vortex tube replaces throttling valve	Room heating	COP ↑25%	Reduces throttling losses; optimizes heat transfer irreversibility
Yao et al. (2021) [15]	Cascade/secondary compression integration	Floor radiation heating	COP ↑10–46.96%	Enhanced efficiency for small/medium buildings
Lü et al. (2024) [24]	Secondary supercritical CO ₂ pressurization	Commercial cooling	Cooling capacity ↑26.5%; COP ↑10.76% at 35 °C	Minimizes throttling dryness; compression process optimization
Chesi et al. (2014) [25]	Parallel compression cycle (with flash tank configuration)	R744 refrigeration cycle	Best COP ↑10%, cooling capacity increased by over 25%	Significantly enhanced system performance by reducing throttling losses and optimizing flash tank separation efficiency
Manjili & Cheraghi (2019) [26]	Expansion work recovery for compressor power reduction	Tropical cooling (>40 °C)	COP ↑20–80%	Adaptability to high-ambient temps; reduces external power input
Zhang et al. (2022) [27]	Converts expansion work into useful feedback	General cooling	COP ↑38.4–45.1%	Minimizes throttling energy loss; high exergy efficiency
Casi et al. (2022) [28]	Peltier-effect subcooling at gas cooler outlet	Cooling at 35 °C	COP ↑16.2%; cooling capacity ↑20.8%	Enhances refrigerant enthalpy difference; no moving parts
Li et al. (2024) [29]	Two-stage cycle with internal heat distribution	Combined cooling/heating	COP ↑14.2%	Optimizes energy efficiency; reduces waste in heat exchange
Sun et al. (2016) [42]	Expander efficiency > 60%	Cooling	Outperforms two-stage cycles in COP	Simplifies system structure while maintaining efficiency

4. Research Status of CO₂ Transcritical Cycle System for Combined Heating and Cooling Supply

The CO₂ combined heating and cooling system is a device that achieves simultaneous heating and cooling in the CO₂ transcritical heat pump cycle system. It can meet the heating and cooling demands of different indoor environments at the same time, improving energy utilization efficiency.

The CO₂ combined heating and cooling system integrating ejector and mechanical subcooling technology constructs an efficient heating and cooling source supply system through the collaborative operation of the main cycle and auxiliary cycle. In the heating mode, the gas cooler serves as the heat source to output heat at a supply water temperature of 60 °C, while in the cooling mode, the evaporator acts as the cold source to provide cooling at a supply water temperature of 7 °C. The system realizes mode switching via a four-way reversing valve. This system demonstrates remarkable advantages in heating and cooling performance: compared with the conventional mechanical subcooling system and traditional ejector system, the COP is increased by 10.90% and 5.58%, respectively, in the heating mode, and by 8.99% and 18.12%, respectively, in the cooling mode. Notably, the performance improvement becomes more significant under lower ambient temperatures (heating mode) or higher ambient temperatures (cooling mode). In practical applications,

the system can meet the annual heating and cooling demands of buildings in different climate zones [55]. In transcritical CO₂ combined heating and cooling systems, the working fluid diversion strategy significantly enhances cold and heat source supply efficiency. Diversion between the gas cooler and subcooler or between the subcooler and throttle valve maximizes the comprehensive cycle COP by 17.62%. This optimization improves heat transfer in gas coolers (60 °C heat source in heating mode) and evaporators (7 °C cold source in cooling mode). By reducing throttling losses and increasing compressor suction pressure, the strategy ensures stable heating and cooling outputs. The optimal discharge pressure, determined by gas cooler outlet temperature, enhances energy efficiency across climates. Unlike diversion between the evaporator and throttle valve (no performance gain), these strategies synergize heating and cooling performance for efficient integrated energy supply [56]. The air-air transcritical CO₂ heat pump system [57] achieves simultaneous cold and heat supply through a parallel compression configuration, with its air handling unit enabling synchronous output of 210 kW cooling capacity and 110 kW heating capacity. Leveraging transcritical cycle thermodynamics, the system uses a gas cooler as the heat source in heating mode and an evaporator as the cold source in cooling mode, dynamically allocating refrigerant via a three-way valve to match demand. The system directly processes air through the air handling unit without water loops, enabling simultaneous heating for three theaters/foyers and cooling for the projector room. Its finned-tube heat exchanger uses ambient air for heat rejection/absorption, demonstrating consistent cold-heat co-supply efficiency in real-world applications. Guo et al. [58] studied the variation in various performance parameters of the CO₂ transcritical cycle cold and hot combined supply system with control parameters such as compressor discharge pressure, gas cooler outlet working fluid temperature, and evaporation temperature under the constraint of narrow point temperature difference. The system leverages the gas cooler as the heat source and the evaporator as the cold source: heating temperature rises with increasing discharge pressure and gas cooler outlet temperature but decreases with higher evaporation temperature, while cooling temperature is solely determined by the evaporation temperature. The study also reveals that the COP of heating and cooling decreases with the increase in gas cooler outlet temperature but increases with the rise in evaporation temperature. When the gas cooler outlet temperature is 45 °C, the comprehensive COP first increases and then decreases with the increase in compressor discharge pressure, which reflects the optimal balance between heat rejection capacity of the gas cooler and compression work consumption. This research highlights the system's ability to dynamically adjust the output of cold and heat sources, providing a theoretical foundation for the efficient application of integrated heating and cooling systems.

In terms of system coupling design, the CO₂ transcritical cycle combined heating and cooling unit [59] realizes synergistic supply of cold and heat sources through a water-water heat pump configuration. The system couples the gas cooler and evaporator via a refrigerant flow regulation mechanism: When the compressor frequency increases from 80 Hz to 120 Hz under rated conditions, the optimal discharge pressure rises from 8 MPa to 8.5 MPa, while the maximum heating COP_h decreases from 3.9 to 3.3. By adjusting the inlet temperature of the ethylene glycol solution in the evaporator and cooling water in the gas cooler, the system achieves dynamic matching of heating and cooling outputs—for every 5 °C increase in cooling water temperature, the heating capacity decreases by 6.7% and the cooling capacity by 7.2%, whereas a 10 °C rise in ethylene glycol temperature boosts heating and cooling capacities by 12.5% and 16.1%, respectively. This coupling design enables simultaneous heat recovery in the gas cooler and cold production in, with the comprehensive energy efficiency COP reaching up to 6.8 when the ethylene glycol inlet

temperature is 10 °C and the cooling water temperature is 20 °C. The system's optimal performance under variable conditions highlights its adaptability for integrated cold-heat supply in industries like slaughtering and brewing, where concurrent thermal and cooling demands exist. For a 5000-square-meter building, considering summer cooling, 670 kWh of electricity is saved per day when the COP improves from 3 to 6.8.

For buildings with simultaneous hot water and cooling demands, the ejector + sub-cooler + internal heat exchanger (IHX) integrated system (Two-Stage Cooling Combined Heat Source System, TSCHPS) emerges as the most promising configuration. Its superiority stems from synergistic energy recovery, enhanced heat transfer, and flexible mode switching, as detailed below: COP Enhancement: Compared to baseline systems, TSCHPS achieves a 14.2% higher COP in combined cooling and heating mode. The ejector recovers expansion work (reducing throttling losses by 95.38–98.76% [27]), the subcooler increases specific cooling capacity (by 20.8% at 35 °C [20]), and the IHX optimizes heat exchange between high-pressure and low-pressure streams (reducing compressor power consumption by 8.73% [40]). Dual-Temperature Output: In heating mode, the gas cooler delivers 60–70 °C hot water, while in cooling mode, the evaporator supplies 7 °C chilled water. Mode switching via a four-way valve ensures seamless transition between demands.

In the transcritical CO₂ heat pump cycle studied in [60], the optimal discharge pressure is determined to enhance heating and cooling performances across diverse ambient temperatures (−15 to 30 °C) and gas-cooler exit temperatures (25 to 45 °C). The study reveals that gas-cooler temperature significantly influences ODP.

To guide practical selection of CO₂ transcritical cycle configurations for buildings with simultaneous heating and cooling demands, Table 4 consolidates COP data of key configurations across ambient temperatures (−20 °C to 40 °C). It synthesizes findings from experimental and simulation studies to clarify optimal choices under specific climatic conditions.

Table 4. COP of key cycle configurations for combined heating and cooling across ambient temperatures.

Ambient Temperature Range	Baseline Single-Stage Cycle (COP)	Ejector-Enhanced Cycle (COP)	Expander-Integrated Cycle (COP)	Subcooler-Equipped Cycle (COP)	Distributed Compression Cycle (COP)	Dual-Ejector Two-Stage Cycle (COP)	Optimal Configuration
−20 °C to −10 °C	1.5–1.8 [12,13]	1.6–1.9 [27]	2.0–2.3 [13,35]	1.7–2.0 [20]	N/A (not optimized for low temps)	N/A	Expander-Integrated
−10 °C to 0 °C	1.8–2.2 [12]	1.9–2.4 [28]	2.3–2.6 [35,38]	2.0–2.3 [43]	N/A	N/A	Expander-Integrated
0 °C to 10 °C	2.2–2.5 [15]	2.4–2.7 [27]	2.5–2.8 [35]	2.3–2.6 [20]	2.4–2.7 [54]	N/A	Expander/Ejector (tie)
10 °C to 30 °C	2.0–2.3 [21]	2.3–2.6 [28]	2.4–2.7 [35]	2.5–2.8 [20,43]	2.5–2.9 [16,54]	N/A	Subcooler-Equipped
30 °C to 40 °C	1.7–2.0 [18,21]	2.0–2.3 [30]	1.9–2.2 [35]	2.1–2.4 [20]	2.2–2.5 [16,54]	2.0–2.6 [18]	Distributed Compression
>40 °C	1.5–1.7 [18]	1.8–2.0 [30]	1.7–1.9 [35]	1.9–2.1 [20]	2.0–2.2 [54]	2.3–3.1 [18]	Dual-Ejector Two-Stage

5. Challenges, Limitations, and Future Works

The narrow optimal operating range of CO₂ systems remains a primary bottleneck. Performance degrades significantly when ambient temperatures or loads deviate from design conditions, causing efficiency losses and a sharp drop in COP. Additionally, the required high operating pressures impose demanding durability requirements on heat exchangers, piping, and seals. Long-term risks like fatigue cracking and material embrittlement require further resolution. Real-time coordination of dynamically shifting

heating/cooling loads also challenges control systems—traditional methods often cause instability during rapid load transitions, leading to excessive compressor cycling and reduced system lifespan.

A critical barrier to global adoption is the lack of comprehensive international standards for CO₂ combined heating and cooling systems. Current regulations primarily address conventional refrigerants or single-function CO₂ equipment, failing to provide specific guidance for the following: integrated thermal load management; high-pressure safety protocols in combined-cycle architectures; performance testing for dual-function operation across variable climates; and unified environmental compliance frameworks.

While regional standards are evolving, harmonized global standards are essential for safety, interoperability, and scalability.

Future research should focus on the following directions:

Co-design of variable-geometry nozzles and adaptive diffusers; development of high-fatigue-limit aluminum–graphene composite piping and self-healing coatings to enhance high-pressure reliability; implementation of AI-driven control using deep reinforcement learning-based multivariable predictive control; and integration with phase-change thermal storage and PV-direct-driven DC bus systems to enable peak shaving, valley filling, and reduced initial investment.

6. Conclusions

The carbon dioxide transcritical heat pump system, as a cutting-edge technology in the field of building environment regulation, provides innovative solutions to the core contradiction of “health, energy conservation, low-carbon” collaborative optimization. It effectively avoids the environmental risks of traditional refrigerants based on the cyclic characteristics of natural working fluids and significantly improves the climate adaptability of building energy supply systems through the high-temperature heating and high-efficiency cooling capabilities of transcritical cycles. Further research has shown that through compression process optimization, expansion work recovery, and system integration design, CO₂ heat pumps can maintain high-efficiency operation under extreme conditions, providing key technical support for reducing building energy intensity and carbon emissions. However, its large-scale promotion still needs to break through bottlenecks such as high initial investment costs and insufficient performance stability under high-temperature conditions.

Looking forward, carbon neutrality and net-zero emissions are the global trends, with HFCs facing a phased elimination. CO₂-based systems show great potential in this regard, but the expansion stage remains crucial for improving the performance of CO₂ transcritical cycles.

Overall, the development of expander-compressors holds great promise and is a promising direction for future advancements in CO₂ system technologies. It is necessary to combine material innovation and intelligent control technology to build the most efficient technology path throughout the entire life cycle in order to accelerate its application process in green buildings and low-carbon cities.

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Abbreviations

COP	Coefficient of Performance
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
HFCs	Hydrofluorocarbons
CFD	Computational Fluid Dynamics
PID	Proportional-Integral-Derivative

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